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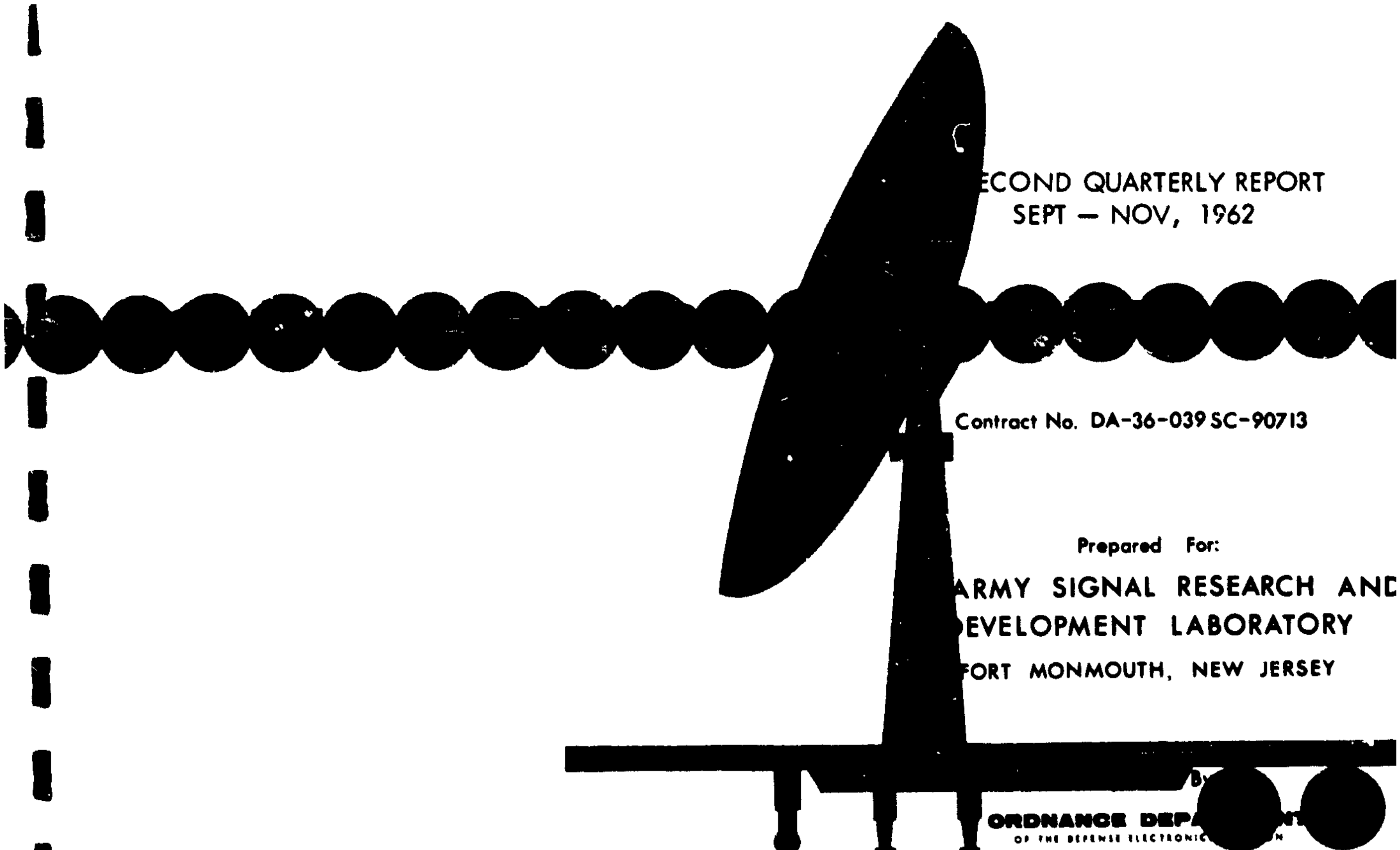

ECOND QUARTERLY REPORT
SEPT — NOV, 1962

Contract No. DA-36-039 SC-90713

Prepared For:

ARMY SIGNAL RESEARCH AND
EVELOPMENT LABORATORY
ORT MONMOUTH, NEW JERSEY

By

ORDNANCE DEPA
OF THE DEFENSE ELECTRONIC
GENERAL ELECTRIC
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REPORT NO. 2 - U.S. ARMY PROJECT NO. 00069-PM-62-91-91 (81

DESIGN STUDY OF A TRANSPORTABLE ANTENNA SYSTEM
(Second Quaterly Report, 1 September through 30 November, 1962)

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Report No. 2

Contract DA-36-039SC-90713

Signal Corps Technical Requirements No. SCL-2101N
dated 14 July 1961

Project No. 00069-PM-62-91-91 (8101)
Prepared For
U.S. Army Signal Research and Development Laboratory
Fort Monmouth, New Jersey

Objective: To conduct an applied research study of design factors for a transportable automatic tracking antenna most suitable for use as a ground terminal(s) for 1.0-10kMc satellite communication systems, and to review state-of-the-art techniques with emphasis on reliability and maximum tracking precision. Specific areas to be studied include parabolic and/or spherical collimators, pedestal and base design, radomes (thin-wall inflatable and thin-skin space frame), drive systems, accessory equipment and maintenance techniques.

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PURPOSE

PURPOSE

The purpose of the program as defined in Army Contract No. DA36-039SC-90713 is (1) to conduct an applied research study of design factors for a transportable automatic tracking antenna most suitable for use as a ground terminal(s) for 1.0-10kMc satellite communication systems, and (2) to review state-of-the-art techniques with emphasis on reliability and maximum tracking precision. Specific areas to be studied include parabolic and/or spherical collimators, pedestal and base design, radomes (thin-wall inflatable and thin-skin space frame), drive systems, accessory equipment and maintenance techniques.

After careful review of the basic objectives of the program, the twelve month study program was divided into the following three phases:

Phase 1 - Exploratory phase of three months
Phase 2 - Optimization phase of six months
Phase 3 - Final phase of three months

The purpose of each phase is defined in the following paragraphs:

1-1 EXPLORATORY PHASE

The exploratory phase of the program was originally divided into four basic parts, with the objective of establishing broad parameters and defining those specific areas requiring special study. These parts included error study, Design A and Design B studies (mechanical design and power drive design for each), and radome study.

This dual design approach (Designs A and B) to the problem was taken in the exploratory phase because the error allowance and erection time requirements appear to be mutually incompatible. This apparent incompatibility made it desirable to inquire into two designs, one design primarily endeavoring to meet this erection time requirement, the other the accuracy requirement. The two approaches taken were designated Design A and Design B, and were defined as follows:

Design A. The objective was to synthesize a minimum cost design that would meet the erection time requirement and optimize accuracy at minimum weight. An adaptation of an existing Marine Corps Search Antenna design was contemplated to serve as a springboard and to save time and money. The original pallet-loaded design was retained in preference to adaptation to trailer loading. An absolute accuracy goal of 0.3 milliradians circular error probable (CEP) or better was established as a target. This design will establish the optimal accuracy level that may be obtained within the erection time limit of 12 hours, at minimal cost and weight.

Design B. The objective was to synthesize a design that would meet or approach the specification accuracy requirement and optimize the erection time and other parameters. An adaptation of the existing Atlas Missile Tracker design was contemplated to serve as a platform to effect study time and money economy. An accuracy goal of 0.1 milliradians CEP was established as a design target. This design would establish the optimal erection time for greatest possible accuracy.

As a result of the exploratory phase of the effort as reported in the First Quarterly report and stemming also from Conference 1, the objective of the study has now crystalized into the development of a design manual for the initial design synthesis of transportable satellite trackers.

Four designs (instead of the original two designs) are now to be generated which will serve different applications. This modified philosophy is developed in the subsequent text of this report.

Pertinent design parameters were originally set forth as follows:

Reflector: 30-40 foot diameter paraboloid, horn fed and Cassegrain
Radio Frequency: 0.1 to 10 kilomegacycle (kMc)
RF Power: 20 Kilowatt (peak) transmit
Acquisition: Velocity 10 degrees/sec., and acceleration 8 degrees/sec.^{2}
Tracking: Velocity 1 degree/sec. and acceleration 0.5 degrees/sec.^{2}
Transportation: Via C-124, C-133, truck and rail
Erection Time: 12 hours to 48 hours
Accuracy: 0.05 to 1.0 milliradian (CEP) absolute pointing error

The objective of the study may now be further stated as follows: To generate a design manual that will permit, as a matter of the state of the art, an initial design synthesis of a radar tracker within the stated parameter ranges that will approach the optimal for a specific application. It will also provide a basis for initial estimates of parameters such as cost, weight, accuracy, erection, etc. The scope of the study is to be kept broad, but a comprehensive bibliography is included for use for detailed examination of any subject matter.

In the light of the objective of the study as it has just been presented, the current optimization phase of the study consists of a large number of special studies, reportable only in part at this time. It is with the current status of these studies that this Second Quarterly Report is concerned.

The work continues in the focus of the overall plan as follows:

Phase 1: (First Quarter) Exploratory Phase. Reported in First Quarterly Report
Phase 2: (2nd and 3rd Quarters) Optimization Phase, currently in process.
Phase 3: (4th Quarter) Concluding Phase.

The contemplated four designs are being worked out to a further degree and may be described as follows:

Design A - A design which is transportable and capable of erection in twelve hours. Accuracy and cost are to be optimized and the accuracy of the order of 0.3 milliradian (CEP) or less. The balance of considerations lead to a pallet-loaded design (transported in M-35 trucks) with geared electric drive and thin skin radome. Weight, 15,000 lbs. Prototype cost, $400,000.

Design B - A design which is transportable and of a maximum absolute accuracy of the order of 0.1 milliradian (CEP) or less, the erection time and cost to be optimized. The balance of considerations lead to a trailerized design incorporating direct electric drive and a randomized space frame radome. Weight, 100,000 lbs. Prototype cost, $2,000,000.

Design C - A design which is transportable and capable of erection in twelve hours with an accuracy of one milliradian (CEP) or more. The design process leads to a trailerized design without radome utilizing a geared hydraulic drive. No data yet on weight or prototype cost.

Design D - A design which is transportable and capable of erection in 48 hours with an accuracy of 0.75 milliradian (CEP). The requirements result in a trailerized design, with dual wall radome and direct electric drive. No data yet on weight or prototype cost.

ABSTRACT

ABSTRACT

This is the Second Quarterly Report on the Design Study of a Transportable Antenna System covering the first half of the optimization phase discussed under PURPOSE. As a result of the exploratory phase effort, the First Quarterly Report, and Conference 1, the objective of the study has been crystalized into the generation of a design manual for transportable satellite trackers with large, paraboloid reflectors which can be readily erected. The current optimization phase effort consists of a large number of special studies now in process and therefore not reportable in full at this time.

Covered in part in this report are preliminary findings in that area of the studies covered to date in the following categories:

Section I: Introduction
Section II: Error Analysis
Section III: Mechanical Design Considerations
Section IV: Mechanical Design Alternatives
Section V: Transportation
Section VI: Power Drive Design
Section VII: Data Unit Survey
Section VIII: Accuracy State of the Art

Included also in this report is a part of the bibliographic material; subsequent reports will contain the balance of this important material.

Basic parameters are given for the four antenna designs which are to be further developed during the third and fourth phases of the program. Conclusions are drawn and plans for both the Third and Fourth (final) Quarterly Reports are outlined. Tentatively, the third report will serve as an Introduction to and as a basic format for the final document. This document will constitute a thorough and comprehensive refinement of the significant findings of the study; mainly, considerations for System Design, Electro-Mechanical Design, Ancillary Design, and Integrated Design comprising the parameters for Design A, Design B, Design C, and Design D.

This report is presented in essentially the same format that will be followed by the Fourth (and final) Quarterly Report, which, as stated under Purpose, will constitute a design manual for the initial design synthesis of transportable satellite trackers.

In the light of this objective and considering the evolutionary character of the studies in process, it should be borne in mind that wording and substance will be expanded, modified and refined in future reports as the work progresses, and that in certain areas of this Second Quarterly Report, only the unintegrated results of special studies as conducted to date are given. Bibliographic material is provided in some areas, and will be expanded as the studies proceed, as previously noted.

Contained in the PROGRAM FOR NEXT INTERVAL is a detailed though tentative outline of the projected format for the Fourth (and final) Quarterly Report, as well as information on the projected studies for the third and fourth quarters.

PUBLICATIONS, LECTURES, REPORTS, AND CONFERENCES

PUBLICATIONS, REPORTS AND CONFERENCES

PUBLICATIONS

First Quarterly Report, 1 June through August 30, 1962.

Seven reports have been prepared by the Ordnance Department, General Electric Company, and delivered to the Signal Corps. These include:

1. First Monthly Letter Report, submitted 29 June, 1962
2. Second Monthly Letter Report, submitted 31 July, 1962
3. First Financial Quarterly Report, (DD Form 1097) submitted 9 September, 1962
4. Third Monthly Letter Report, submitted 9 October, 1962
5. Fourth Monthly Letter Report, submitted 7 November, 1962
6. Second Financial Quarterly Report, (DD Form 1097) submitted 10 December, 1962
7. Fifth Monthly Letter Report, submitted 31 December, 1962

CONFERENCES

Four conference. took place between General Electric Ordnance Department personnel and Army Signal Corps personnel. These were:

1. Pre-Award Conference, held at the Signal Corps Research and Development Laboratory at Fort Monmouth, New Jersey on 24 January, 1962. The subject of discussion was the overall problem of designing and producing an accurate and reliable transportable antenna system.

2. Post-Award Conference, held at the General Electric Ordnance Department, Pittsfield, Massachusetts 20 June, 1962. The conference objective was to reach agreement on - and obtain Signal Corps approval of - a specific outline of study as proposed by Ordnance Department.

3. First Quarter Study Report Review Conference, held at Fort Monmouth, New Jersey, 2 November, 1962, to review and discuss the draft copy of the First Quarterly Report, Design Study of Transportable Antenna System.

4. First Quarter Study Report Review Conference held at Forth Monmouth, N.J., 2 November 1962, to review and discuss the draft copy of the First Quarterly Report.

FACTUAL DATA

FACTUAL DATA

Section I
INTRODUCTION

1-1 GENERAL

This is a report on a Design Study for a Transportable Antenna System. The study is conducted in accordance with Specification SCL 4345, Contract DA-36-309-SC-90713, General Electric Technical Proposal CDE 41005 and various conferences between the General Electric Company and United States Army Signal Research Development Laboratory. Specifically, the purpose of the study was stated in Specification SCL4345:

"(1) to conduct an applied research study of design factors for a transportable automatic-tracking antenna most suitable for use as ground terminal (s) for 1.0 to 10 KMC satellite communications system and (2) to review state-of-the-art techniques with emphasis on reliability and maximum tracking precision."

1-2 APPROACH

There were two possible fundamental approaches to this study problem:

1. Examine all existing trackers, their parameters and evaluations; catalog this data in some logical manner and then explore integration of the concepts and design data into appropriate designs for the specific applications involved in the study. This approach necessarily would be based on assumption of a deep level of knowledgability on the part of the reader.

2. Begin with fundamentals and generate a manual for the design synthesis of trackers for specific applications. Such an approach will involve the assumption of only general engineering knowledgability on the part of the reader.

The latter approach was selected for a number of reasons, the main one being that such an approach would prove to be of greater value, in that it would result in a fundamental "text book" for radar tracker synthesis which does not exist in the art. At the same time, the study would be conducted in such depth as to meet basic objectives of the study. It will be of particular value to the reader who is knowledgable in one aspect of radar tracker design but not familiar with the entire spectrum of design which is involved.

1-3 STUDY ORIENTATION

In particular, this study is oriented to the electro-mechanical design of trackers and does not become involved in microwave considerations except as background. Where possible, in each area of study, a catalog of feasible alternatives if given, together with considerations for and against the use of each alternative, plus analytical information which permits the selection of the optimal alternative for a specific application. Specifically, the study provides:

1. Initial, nearly optimal design synthesis of a specific application.

2. An extensive listing of bibliographic references for detailed study.

3. Design curves as a basis for management decision-making, showing the relationship of cost, accuracy and erection time for radar trackers within the general definitions contemplated by this study.

4. Coverage within the limits of the overall concept of the study plus bibliographic references which permit added detailed investigation of any one specific area.

5. A base upon which further study may be organized for comprehensive treatment of the topic.

The study is divided into five basic sections:

1. System design section

2. Mechanical design section

3. Power drive section

4. Ancillary design section

5. Integrated design section

The system design section dwells on the design synthesis process and upon error analysis. The error analysis is emphasized first because error is considered in all subsequent decision making. Mechanical design deals with fundamental mechanical design of reflector assembly mount and radome including consideration of erection and transport problems. The power drive section deals with the synthesis of an optional power drive for specific applications. Ancillary design treats of the many associated design problems in a radar tracker from bore-sight scope and data unit selection to collimation, alignment and leveling instrumentation techniques, and error.

The integrated design section utilizes the suggested design synthesis techniques to build up four designs for four different applications. In the process, a design curve relating accuracy, cost and erection time is generated.

Section II
ERROR ANALYSIS

2-1 INTRODUCTION

A radar system or a radar tracker specification will generally contain an accuracy requirement. Thereafter, every step in the design synthesis involves decisions relating to angular error. In fact, one of the first steps in the synthesis is the preparation of an error budget. For these reasons, the subject of angular error is treated first in this study to give the reader broad insight into the topic antecedent to the detailed discussions that follow later in the text. Work in this section is nearly in its final phase and requires only a review for accuracy check and final additions. This section on error analysis is divided into six subsections as follows:

- 2-1 Introduction
- 2-2 Radar Tracker Error Classification
- 2-3 Microwave Error
- 2-4 Mechanical Error
- 2-5 Servo Error
- 2-6 Error Analysis Procedure

In passing, necessary elements of system and microwave antenna design are introduced as background. However, this study is focused on the electromechanical design of radar trackers; hence, mechanical and servo errors are treated in greater depth than microwave errors. Only noise-excited error, of all the microwave error sources, is considered because of its strong influence on the electromechanical design.

2-2 RADAR TRACKER ERROR CLASSIFICATION

One approach to the classification of the types of error is to examine an evaluation of the accuracy of an existing radar tracker. There are several techniques for such evaluation but the most effective is by comparison of the radar tracker with a much more accurate instrument, such as a ballistic camera or optical tracker, tracking the same target simultaneously. The error of the radar tracker will be found to be of the idealized form shown in figure 2-1 (a), Total Error. For convenience, this total error may be resolved into three components: bias, cyclic error and noise. The character of these error components is indicated in figure 2-1 (a), (b), (c) and (d) which indicates the form of each of the three components.

Table 2-1 indicates some of the alternative nomenclature that is used in the technology to identify these three total error components. For the purpose of this discussion, the terms, noise error, cyclic error and bias error will be used. The term total error or accuracy will be used for the combined sum of noise, cyclic and bias errors. Occasionally the term precision will be used to indicate noise and/or cyclic error (the context deciding which or both).

Referring to figure 2-1, the terms noise error, cyclic error and bias may be defined as follows:

Noise Error - Noise error is that error which is random and totally unpredictable except in statistical terms as a random variable. The noise inducing the error may be electrical or mechanical in character. An example of electric noise is thermal noise in electronic devices. An example of mechanical noise is bearing roughness.

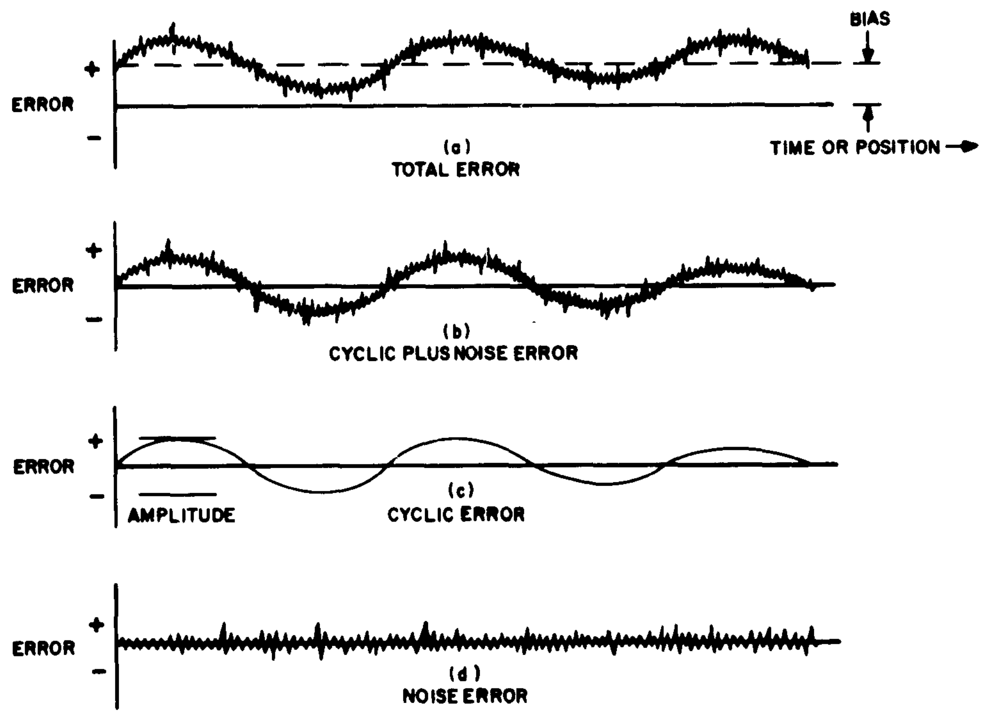


Figure 2-1. Radar Tracker Error.

Cyclic Error - Cyclic error is that error which is a function of time, tracker position, tracker velocity or tracker acceleration or other continuous function. Cyclic error is variable, but not necessarily periodic, and may be predictable and calibrated out if a correlation with various tracker parameters may be made.

This entire study is predicted upon not using a computer in real time or using subsequent data processing to partially eliminate this cyclic error. Hence, cyclic error is treated statistically as a random error herein. A limited discussion of the use of a computer to eliminate this source of error is discussed in Section VIII, Accuracy State of the Art, Error Suppression and Compensation Schemes. In this total connection, Armed Services Technical Information Agency (ASTIA) Report AD 270800 is an interesting report on accuracy determinations of the range instrumentation at the Atlantic Missile Range.

Bias Error - Bias error is that error which is a time and position invariant. When established as such, the error may be removed by calibration. However, "constant" bias will change from day to day and test to test. Hence, even bias must be treated statistically as a random variation around a mean.

TABLE 2-1. ERROR TERMINOLOGY

Classical Terminology	Statistical Terminology	Communications Terminology	Mixed Terminology		This Report Term	Examples
Random	Random Variable	High Freq. >0.5 cps	Random	Uncorrelated Random	Noise	(1) Thermal noise (2) Mechanical Noise
Systematic		Low Freq. <0.5 cps	Systematic	Cyclic or Correlated Random	Cyclic Error	(1) Dynamic Lags (2) Out of Level
	Mean	Nearly Zero Frequency	Constant Bias	Constant Bias	Bias	(1) Servo Dead Band, (2) Circuit Drift
Total	Total	Total	Total	Total	Total	Total

All individual errors are treated statistically in analysis and their variances summed to obtain the variance of the total error. In other words, the standard deviation of the individual errors are added root sum square to extract the standard deviation of the total error. These are different expressions of the same thing in that the variance is the square of the standard deviation. The variance or standard deviation so obtained is that of a Gaussian (normal) distribution. This procedure is justified by the Central Limit Theorem which demonstrates that the sum of several errors of any statistical distribution is a Gaussian or normal distribution. This means that the resulting system error will be less than the standard deviation on 68 percent of all occasions and that the peak error is plus or minus 3.29 σ where σ (sigma) is the symbol for standard deviation. Table 2-2 indicates other relationships between the standard deviation and probability of the occurrence of various levels of error.

The standard deviations of the error will be generated in the azimuth and elevation axes. In practice, it is convenient to have a single figure of merit for accuracy and the CEP (Circular Error Probable) is used. The CEP is a radial value of error in space that will not be exceeded on 50 percent of all occasions. The conversion may be made by the following approximate expression:

$$CEP = 0.59 \quad (\sigma_A + \sigma_E)$$

The bibliography lists several basic texts on mathematical statistics as well as several ASTIA reports that are of value in this connection.

The foregoing is a classification of error into three types (noise, cyclic and bias), as a matter of mathematical statistics to facilitate analysis. Every error, regardless of the source, will fall into one or more of these categories. Sources of error in a radar tracker may be classified as shown in figure 2-2. The text following is logically organized into discussions of the indicated three basic sources of error: microwave, mechanical, and servomechanical error.

TABLE 2-2. ERROR PROBABILITY

Probability of Occurrence Error	Standard Deviations
0.50	±0.67
0.68	±1.00
0.90	±1.65
0.95	±1.96
1.000	±3.29

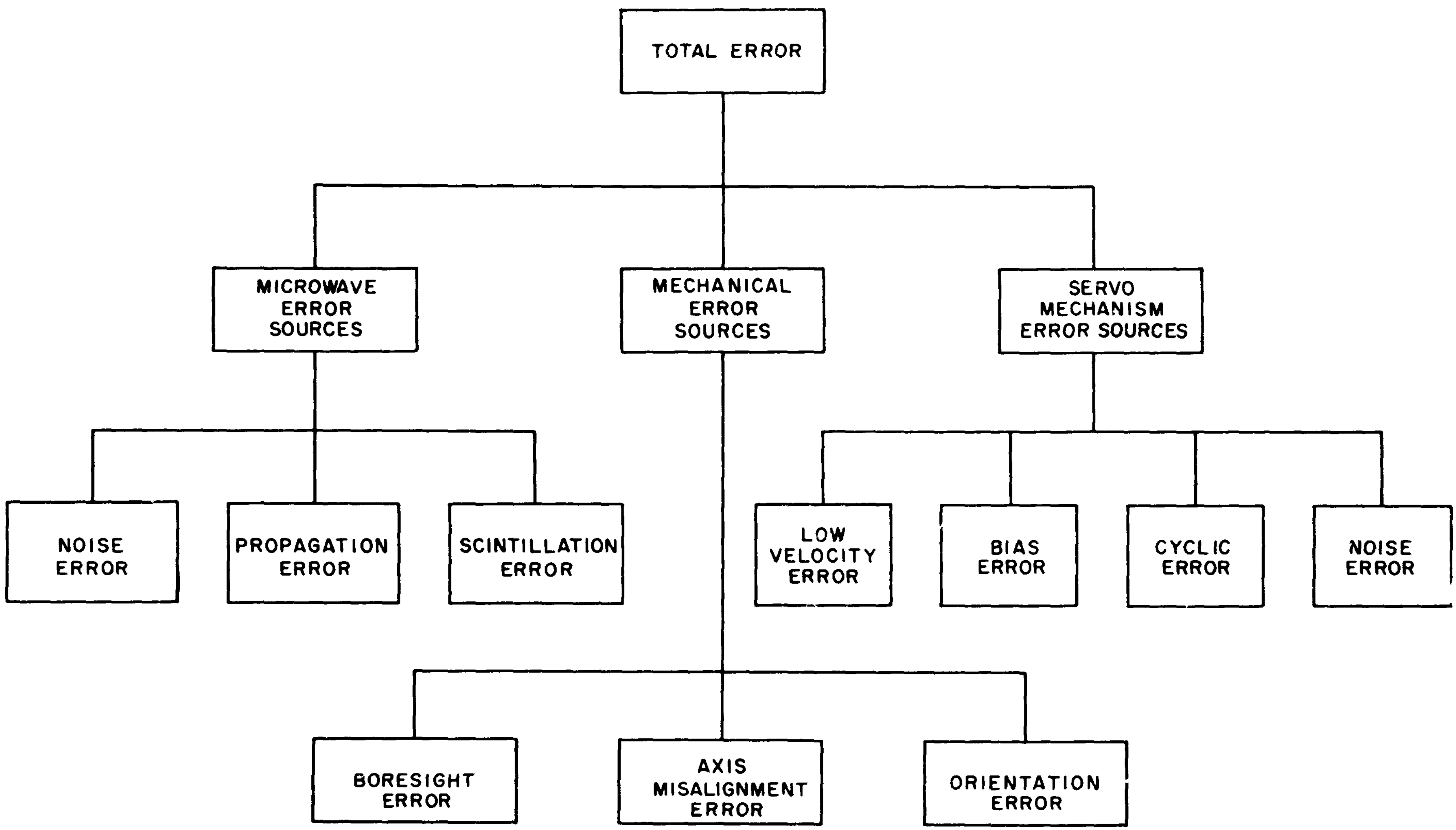


Figure 2-2. Sources of Error.

2-3 MICROWAVE ERROR

2-3.1 INTRODUCTION. This study is directed at the electromechanical design of radar trackers on the explicit premise that all system and antenna microwave requirements have been reduced to mechanical terms. However, an understanding of certain elements in the microwave area is beneficial to the electromechanical designer. These subjects are introduced through the consideration of the angular error introduced by microwave aspects. Table 2-3 is a classification of microwave error into three sources; noise, propagation and scintillation errors. Only the angular error due to noise will be considered in detail herein because of its influence on the electromechanical design.

2-3.2 ANGULAR ERROR DUE TO NOISE. One approach to consideration of the angular error due to noise is to consider a system parameter, beam sensitivity, and then manipulate it into an expression for angular error.

Beam sensitivity may be defined as:

$$K_s = \frac{\Delta}{E} \qquad \text{Eq 2-1}$$

where K_s = beam sensitivity

Δ = radar error signal

E = angular error

For a number of reasons it is desirable to make this into a dimensionless ratio. The radar error voltage is normalized by the signal strength (S) and the angular error placed on a per beam width basis. Hence,

$$K_s = \frac{\Delta/S}{E/\theta} \quad \text{(a numeric)} \qquad \text{Eq 2-2}$$

Where: S = Radar signal strength

θ = Half-power radar beamwidth

Consider now equation 2-2 in the presence of noise

$$K_s = \frac{(\Delta + N)/(S + N)}{E/\theta} \qquad \text{Eq 2-3}$$

where N = noise strength

TABLE 2-3. MICROWAVE ERROR CLASSIFICATION

I Noise Errors	II Propogation Errors	III Scintillation Errors
1. Thermal Noise 2. Man-made Noise 3. Cosmic Noise 4. Atmospheric Noise	1. Refraction 2. Diffraction 3. Reflection 4. Rain Clutter 5. Ground Clutter	1. Wander 2. Fading

But $S >> N$ and the equation reduces to:

$$K_s = \frac{(\Delta + N)/S}{E/\theta} \qquad \text{Eq 2-4}$$

In the case where the radar error voltage is zero ($\Delta = 0$), the equation reduces to:

$$K_s = \frac{N/S}{E/\theta} \qquad \text{Eq 2-5}$$

The error signal, $(\Delta + N)$ or N, is used in a servomechanism to drive the antenna to reduce the angular error towards zero. The servo acts as a low pass filter on the noise and the servo filter factor K_f is introduced to account for this reduction in noise.

Hence, rearranging terms:

$$\frac{E_n}{\theta} = \frac{1}{K_s \ K_f} \quad \frac{N}{S} \qquad \text{Eq 2-6}$$

where: K_f = servo filter factor (a numeric)

E_n = angular error due to noise

Consider now the well known radar range equation for beacon tracking:

$$\frac{(S)^2}{(N)} = \frac{P_b \, G_b \, G_r \, \lambda^2 L}{(4\pi)^2 R^2 N_f K T B} \qquad \text{Eq 2-7}$$

where p_b = Power of beacon (watts)

G_b = Antenna gain of beacon (a numeric)

G_r = Antenna gain of radar tracker (numeric)

λ = Wavelength (meters)

L = Losses

R = Range (meters)

N_f = Noise figure of receiver

KT = Boltzman's constant times absolute temperature (watts per cycle)

B = Receiver IF bandwidth (cycles per second)

Combining equations 2-6 and 2-7 to eliminate S/N yields:

$$\frac{E_n}{\theta} = \frac{1}{K_s \ K_f} \sqrt{\frac{(4\pi)^2 \ R^2 \ N_f \ \ KTB}{P_b \ \ G_b \ \ G_r \ \lambda^2 \ \ L}} \qquad \text{Eq 2-8}$$

Reducing this equation to the factors under the control of the antenna designer results in the following simplified expression:

$$\frac{E_n}{\theta} = K \ \frac{1}{K_s \ K_f} \sqrt{\frac{1}{G_r}} \qquad \text{Eq 2-9}$$

where: K = a system constant

Each of these factors, K_s, K_f, G_r and θ is discussed in detail in the text that follows. For crude analysis, the following values may be used:

$$K_s = 1.5 \qquad \text{Eq 2-10}$$

$$K_f = \sqrt[4]{\frac{f_p}{\pi f_c}} \qquad \text{Eq 2-11}$$

where: F_p = pulse repetition rate

f_c = servo open loop cutoff frequency

$$\theta = 70 \frac{\lambda}{D} \quad \text{(degrees)} \qquad \text{Eq 2-12}$$

$$G_r = \frac{27,000}{\theta^2} \quad \text{(numeric with } \theta \text{ in degrees)} \qquad \text{Eq 2-13}$$

2-3.3 AUTOMATIC TRACKING AND BEAM SENSITIVITY. A target may be tracked by an operator observing the echo on a presentation tube. As the echo tends to disappear, the operator may move antenna controls to place the antenna on target by maximizing the echo. One difficulty is that as the echo tends to disappear, the operator does not know the direction in which to move the antenna. This problem may be solved by having the operator oscillate the controls and thereby establish the direction of the error. This may also be accomplished automatically by two techniques; sequential lobing and monopulse radar. Sequential lobing may be effected by conical scanning wherein a beam is rotated in a cone to effect an error signal to drive the antenna servo. Monopulse accomplishes the same purpose by splitting the RF power into two beams, simultaneously, with a squint angle between them (the squint angle is α in figure 2-3). In each instance the basic

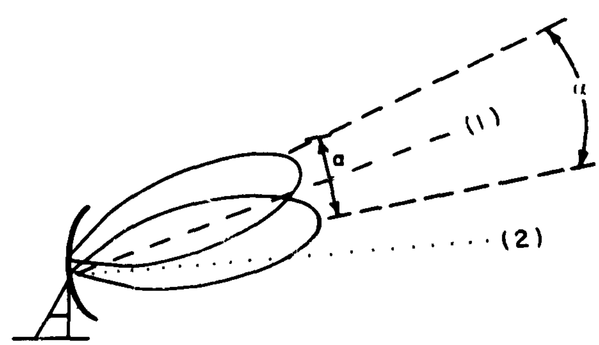


Figure 2-3. Lobes in Space.

principle is the same as indicated in figure 2-3 which displays the two lobes in space. If the target is on the dashed line (1), then the strength of the returns from the two lobes are equal and no error signal results. If the target is on the dotted line (2), then there is a difference in the strength of the two return echos. This difference can then be utilized as an error signal for the servos. The direction is known, i.e., it opposes the direction of the lobe with the larger return. Thereby, automatic tracking is effected.

There are three types of monopulse:

(1) Amplitude comparison monopulse
(2) Phase comparison monopulse
(3) Combined (amplitude and phase)

The beam patterns of an amplitude monopulse will have the form shown in figure 2-4. for reasons that will emerge, the sum and difference patterns are of greater value. The form of these patterns is shown in figure 2-5.

Earlier, in equation 2-2, beam sensitivity was defined as: $K_s = \frac{\Delta / S}{E / \theta}$

where K_s = beam sensitivity

Δ = radar error signal

S = signal strength

E = angular error

θ = beamwidth

The beam sensitivity may be established from the sum and difference patterns of figure 2-5 as follows:

$$K_s = \frac{\Delta / S}{E / \theta} = \frac{\Delta / \Sigma}{E / \theta} \qquad \text{Eq 2-14}$$

where: Δ = A-B

Σ = A+B

A = Strength of beam A

B = Strength of beam B

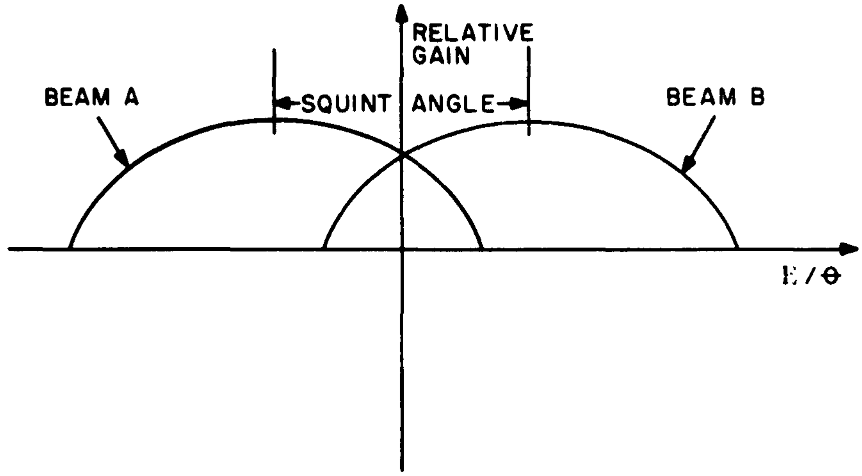


Figure 2-4. Amplitude Monopulse Beam Patterns.

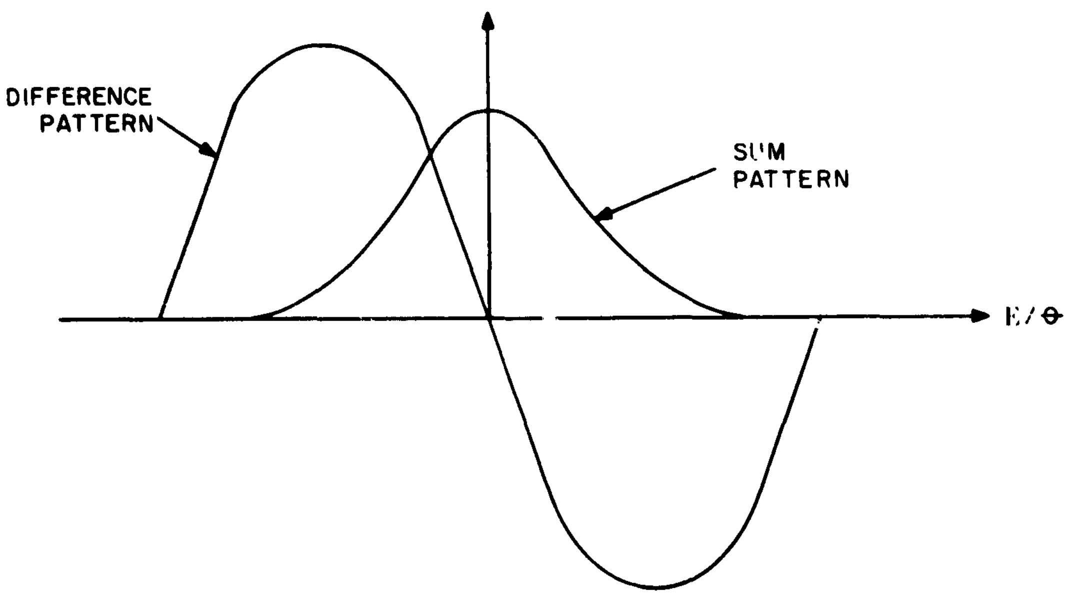


Figure 2-5. Beam Pattern Forms.

The value of $\Delta/E/_\theta$ may be established from the difference pattern and is divided by the sum (Σ) to obtain the beam sensitivity. While it is desirable to obtain a high value for the beam sensitivity by making the squint angle large, penalty, in loss of gain, must be paid as the total gain is the sum of the gain in the two lobes. While considerations of linearity may be important, in general the optimal trade-off exists wherein K_s G = maximum. Practical values of the beam sensitivity will vary from 1 to 3.

2-3.4 SERVO FILTER FACTOR. The error signal generated by the radar and utilized in the tracker servo to effect automatic tracking has noise superimposed upon it from a number of sources including thermal noise in the receiver. Fortunately, the servo acts as a low pass filter to reduce the noise power. While a derivation and the assumptions will not be given here, it can be demonstrated that the following expressions define the servo filter factor (K_f).

$$K_f = \sqrt{\frac{f_p}{\pi f_c}} \qquad \text{(for monopulse)} \qquad \text{Eq 2-15}$$

$$K_f = \sqrt{\frac{f_p}{2\pi f_c}} \qquad \text{(for sequential lobing)} \qquad \text{Eq 2-16}$$

where: f_p = pulse repetition rate

f_c = servo open loop bandwidth

2-3.5 GAIN AND BEAMWIDTH.

2-3.5.1 Introduction. Consider a plane aperture upon which uniform and equiphased illumination falls and from which a beam forms. It can be demonstrated that the gain of the beam is:

$$G = \frac{4 \pi A}{\lambda 2}$$

where G = gain (a numeric)

A = aperture area

λ = wavelength of radiation

Gain is defined as the ratio of the radiated power per unit solid angle in a given direction in the beam that is formed to the same power when radiating in all directions equally. The radiation from the apeture can be shown to be of the form $G(a) = \left(\frac{\sin u}{u}\right)^2$

where u is a function of directional angle (α). This function indicates that sidelobes are formed, in addition to the main lobe, due to diffraction effects. For the same reason, a tapered lobe is formed instead of a cylindrical beam.

Figure 2-6, curve A, indicates qualitatively the beam and sidelobes which result from uniform, equiphased illumination upon the aperture. Curve B indicates qualitatively the results when the aperture is illuminated by a non-uniform illumination which is peaked in the center and tapers towards the edge. Note that the non-uniform illumination broadens the beam and thereby lowers the gain but that the sidelobe level has decreased relative to the main lobe. Low sidelobe levels are a desirable attribute in most applications.

Consider now a perfect parabolic surface which has the following useful geometric properties:

(1) Rays from the focal point to the surface of the paraboloid will emerge parallel to each other.

(2) The aperture plan (projected surface of the paraboloid) will be equiphased surface because the length of ray from focal point to the reflector surface to the aperture plane surface is the same on all cases.

These properties permit a parabolic reflector to be analyzed as if it were the plane aperture discussed earlier.

A perfect paraboloid with a circular aperture and illuminated with uniform and equiphased radiation will prodice the following gain and beamwidth:

$$G = \frac{4\pi A}{\lambda^2} = \left(\frac{\pi D}{\lambda}\right)^2 = \text{Gain} \qquad \text{Eq 2-17}$$

$$\theta = 50.5 \frac{\lambda}{D} = \text{Half power beamwidth} \qquad \text{Eq 2-18}$$

First sidelobe will be 17.5 decibel (db) down from the main lobe

There are two influences which will cause departures from these ideal results:

(1) Non-uniform illumination of the parabolic surface

(2) Phase errors, i.e., departures from an equiphased front in the plane of the aperture.

These two influences will cause the gain to be less than the theoretical maximum or:

$$G = f \frac{\pi D}{\lambda}^2 \qquad \text{Eq 2-19}$$

where f = gain factor which is less than 1

In addition the full power from the feed horn will not be intercepted by the reflector and, together with other considerations, cause an inefficiency. Hence, the need for an illumination efficiency factor and modifying Eq 2-19

$$G = n_f \frac{\pi D}{\lambda}^{2} \qquad \text{Eq 2-20}$$

where n = efficiency factor which is less than 1

Hence, f and n for a given design must be established from experience, experiment or analysis.

Non-uniform primary illumination is deliberately introduced into the design to achieve the desirable lower sidelobe levels at the penalty of loss of gain and broadening of the beam. Phase errors arise accidentally from three sources and are to be suppressed as a deleterious influence on performance. They are:

(1) Departure of the reflector surface from a true parabolic contour
(2) Defocusing (feed not at focal point)
(3) Horn phase center errors

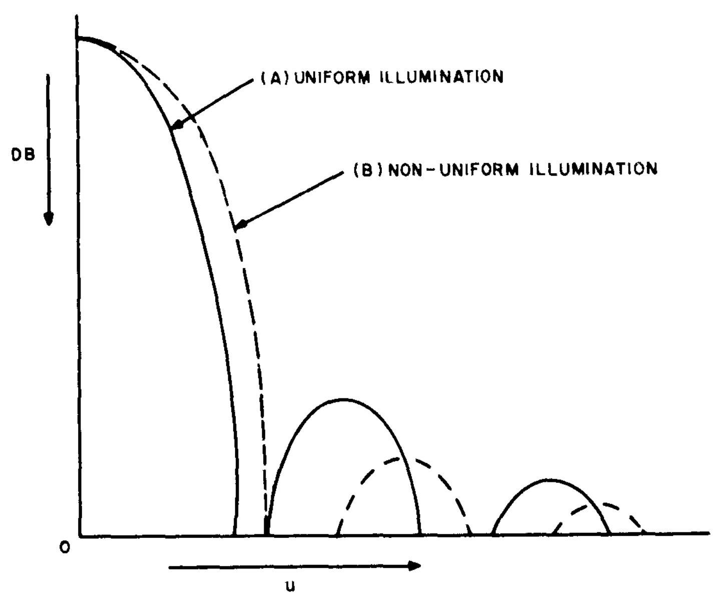


Figure 2-6. Beam and Sidelobe Patterns.

Blockage of the aperture by horn and horn supports may also be considered as phase errors. The remainder of this section will be devoted to the relationships between the factors discussed.

2-3.5.2 Primary Illumination. Table 2-4 indicates the antenna parameters with a cosine shape to the primary illumination with various power levels at reflector edge (no phase errors):

TABLE 2-4

Edge Attenuation db down	Gain Factor (f)	Beamwidth Constant (K) (θ in degrees)	First side lobe db down
10	0.81	67	21
12	0.79	69	22
15	0.77	71	23.5
20	0.75	73	24.5

For comparison, recall that uniform illumination provides a beamwidth constant of 50.5 with sidelobe level of 17.5 db. Note that as the gain decrease the beam broadens, and sidelobes are reduced as taper is increased to a limit of 20 db edge attenuation where further beneficial results cannot be obtained. The results of $\cos^2$ and $\cos^3$ primary illumination will increasingly display the same trend with greater loss of gain and lesser side levels. A Gaussian primary illumination would eliminate sidelobes entirely at great loss of gain. Figure 2-7 indicates the range of taper versus first side lobe level that may be achieved. The sole conclusion to be drawn from the foregoing is that there is a trade-off between gain and sidelobe levels to be made in every design. Figure 2-8 is a representation of these concepts relating side lobe level to beamwidth constant (K) where percent blocking of the aperture is a parameter. From the relation $\theta = \frac{\lambda}{D}$ of figure 2-9, the beamwidth may be established. The beamwidth may be translated into gain by figure 2-10. Thus, the relations between beamwidth, gain and sidelobe levels, together with reflector diameter, may be established if phase errors are assumed to be zero.

The subject of the desirability of low sidelobe levels will not be pursued herein except to state, as a generality, that the lower the sidelobe levels, the lower will be the level of noise pick up.

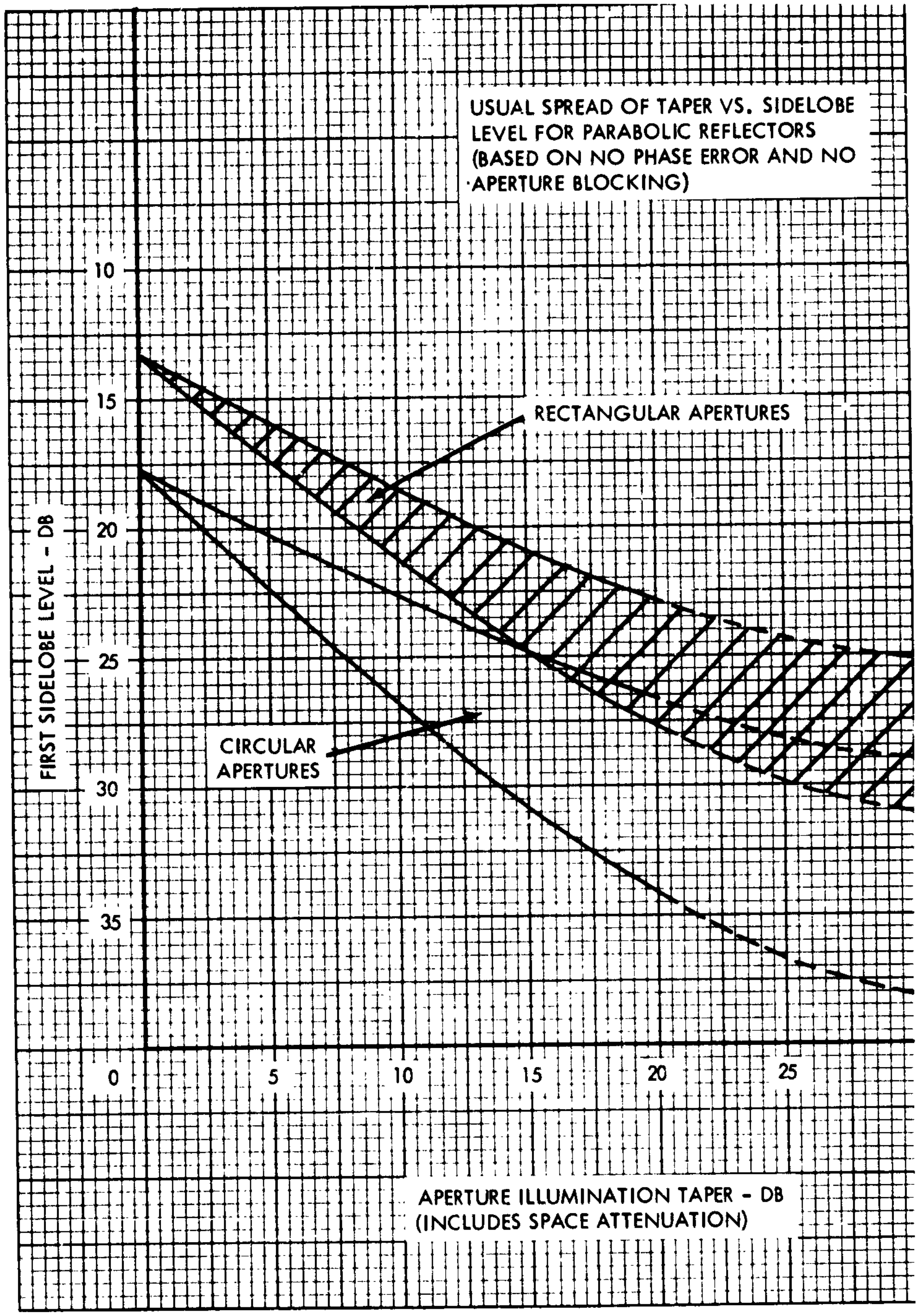


Figure 2-7. Taper Vs. Sidelobe Level.

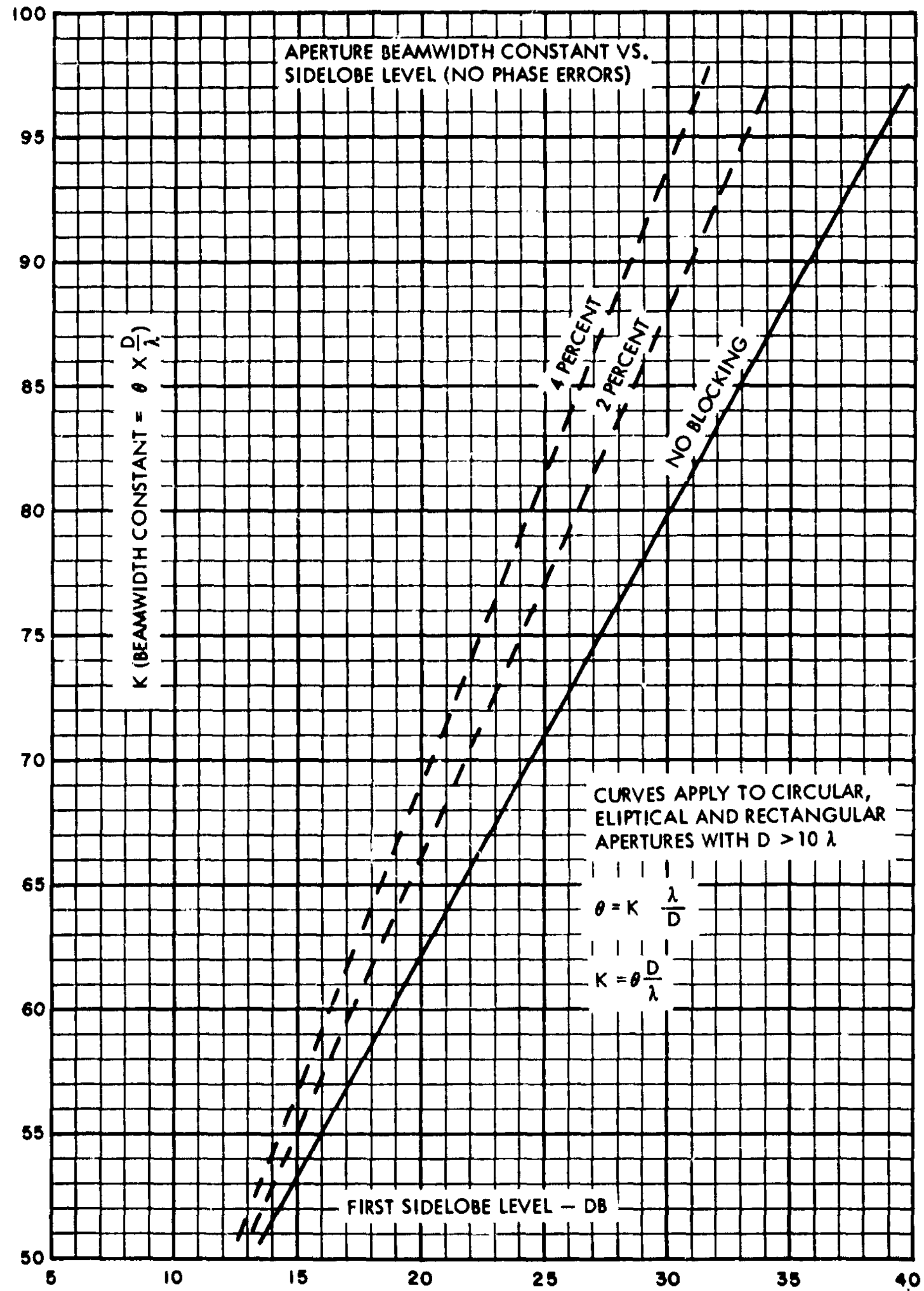


Figure 2-8. Aperture Beamwidth Constant Vs. Sidelobe Level.

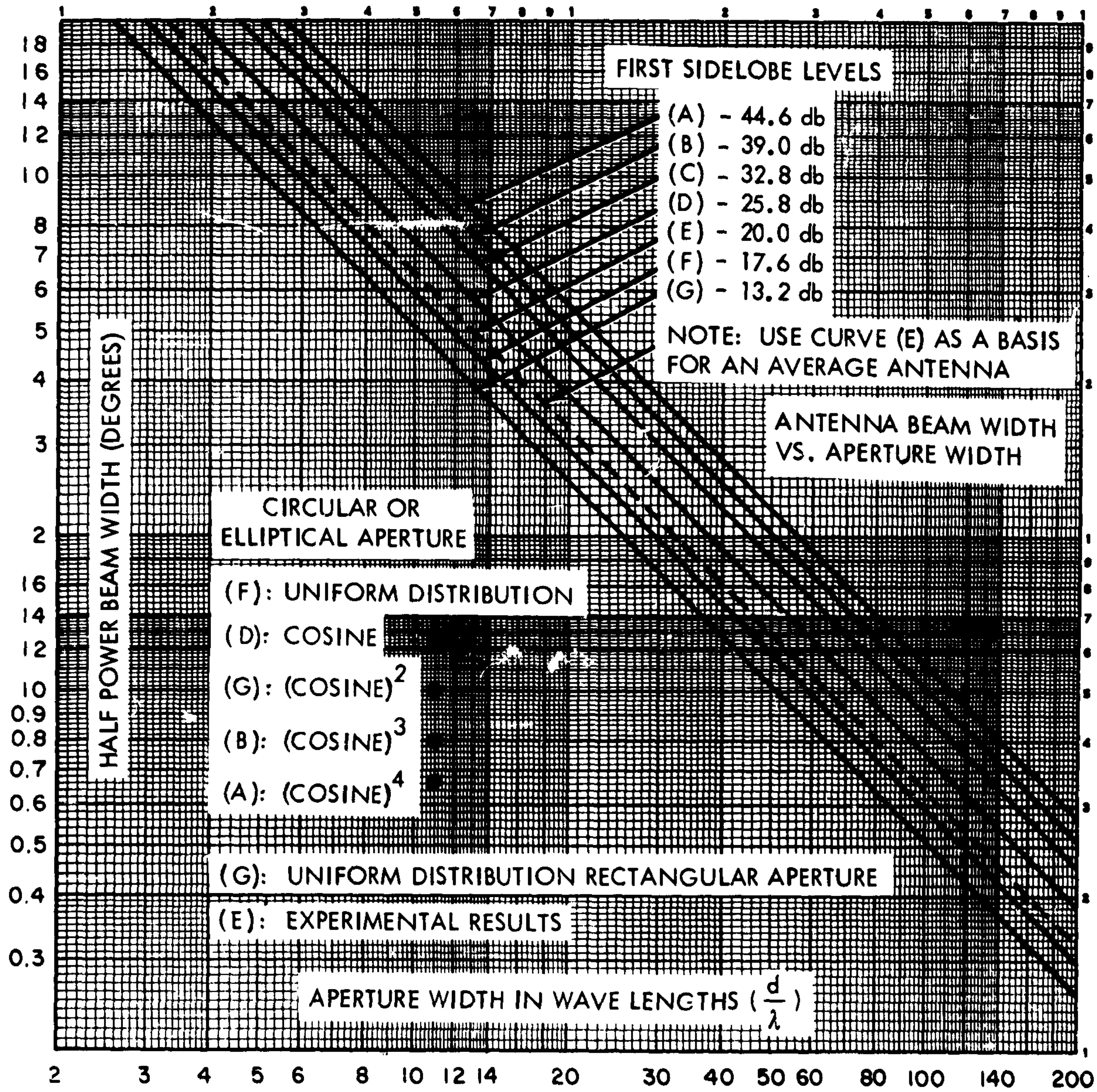


Figure 2-9. Antenna Beamwidth vs. Aperture Width.

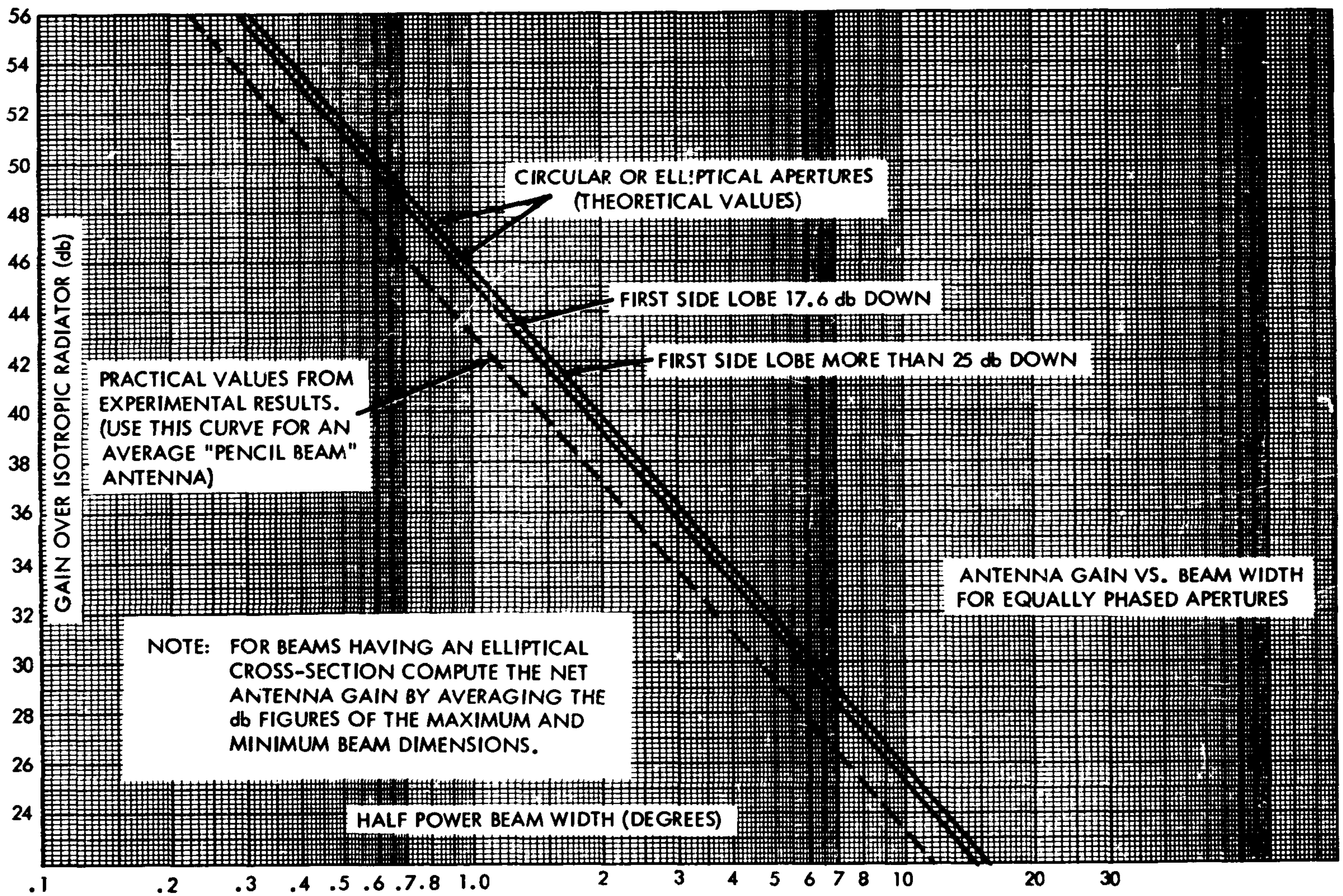


Figure 2-10. Antenna Gain Vs. Beamwidth.

2-3.5.3 Phase Errors. As previously indicated, phase errors are caused by departures from an equiphased front in the plane of the aperture and arise from three sources:

(1) Imperfect reflecting surface contour
(2) Defocusing
(3) Feed phase center errors

Only the first source, imperfect reflecting surface contours will be disucssed but the theory is the same in all three cases. These phase errors will cause modifications to the relationships between beamwidth, gain, sidelobe levels and reflector diameters established in section 2-3.5.2 wherein no phase errors were assumed. Phase errors generally cause loss of gain and an increase in sidelobe levels. As such, they are a deleterious influence. Figure 2-11 reflects the loss of gain due to phase errors in the plane of the aperture. Figure 2-12 indicates the increase in sidelobe level due to reflector contour errors. The correlation interval refers to the distance, or average, for which random contour errors become independent. These curves may be used to establish the influence on the radar beam due to reflecting surface imperfections including tolerances permitted.

2-3.6 CONCLUSION. The reader is cautioned to use the material in this Section 2-3, Microwave Error, with great care. The purpose has been to give the electromechanical designer broad insight into the topic. Rigor and comprehensiveness have been sacrificed in order to encapsulate the topic into a readily understood survey. The references in the bibliography should be studied for a full understanding.

2-4 MECHANICAL ERROR

2-4.1 INTRODUCTION. This section on mechanical error does not purport to be a comprehensive treatment of the subject. It is written to organize the topic in an orderly fashion together with such definitions as will provide an adequate basis for detailed discussions that follow later in this text. Figure 2-13 indicates a classification of mechanical error categorized as follows:

(1) Excitation sources of mechanical errors
(2) Geometric error classification
(3) Mathematical classification

In brief, the geometric errors which may be caused by any of the itemized excitations cause spatial errors which translate into errors in the azimuth and elevation axes of the tracker. The errors may be considered as one of three statistical types; noise, cyclic or bias error and treated accordingly in analysis.

2-4.2 EXCITATION SOURCES. Geometric errors may be excited from a multiplicity of causes which are classified and enumerated in figure 2-13. Each of these major classifications will be discussed below.

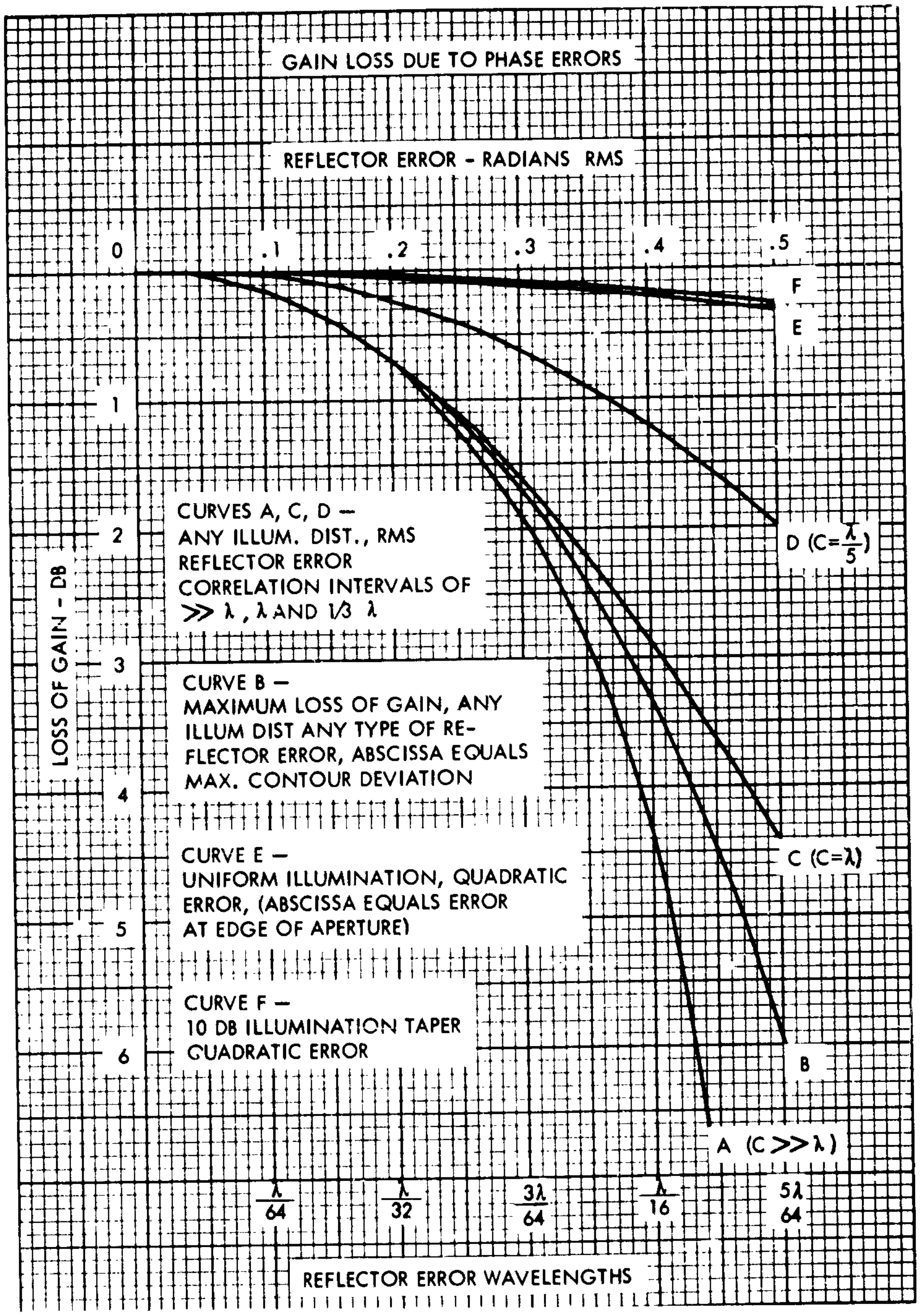


Figure 2-11. Gain Loss Due to Phase Errors.

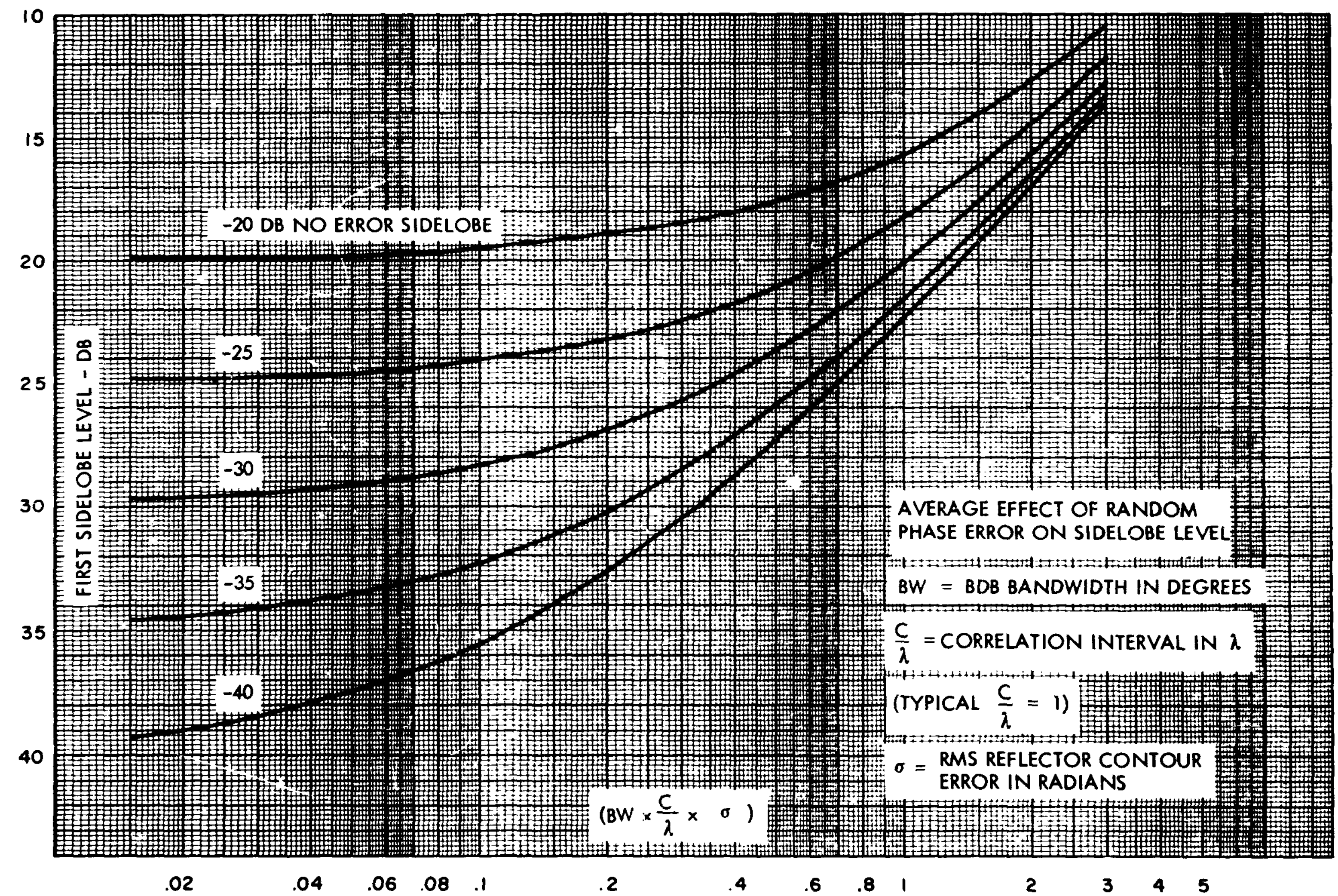
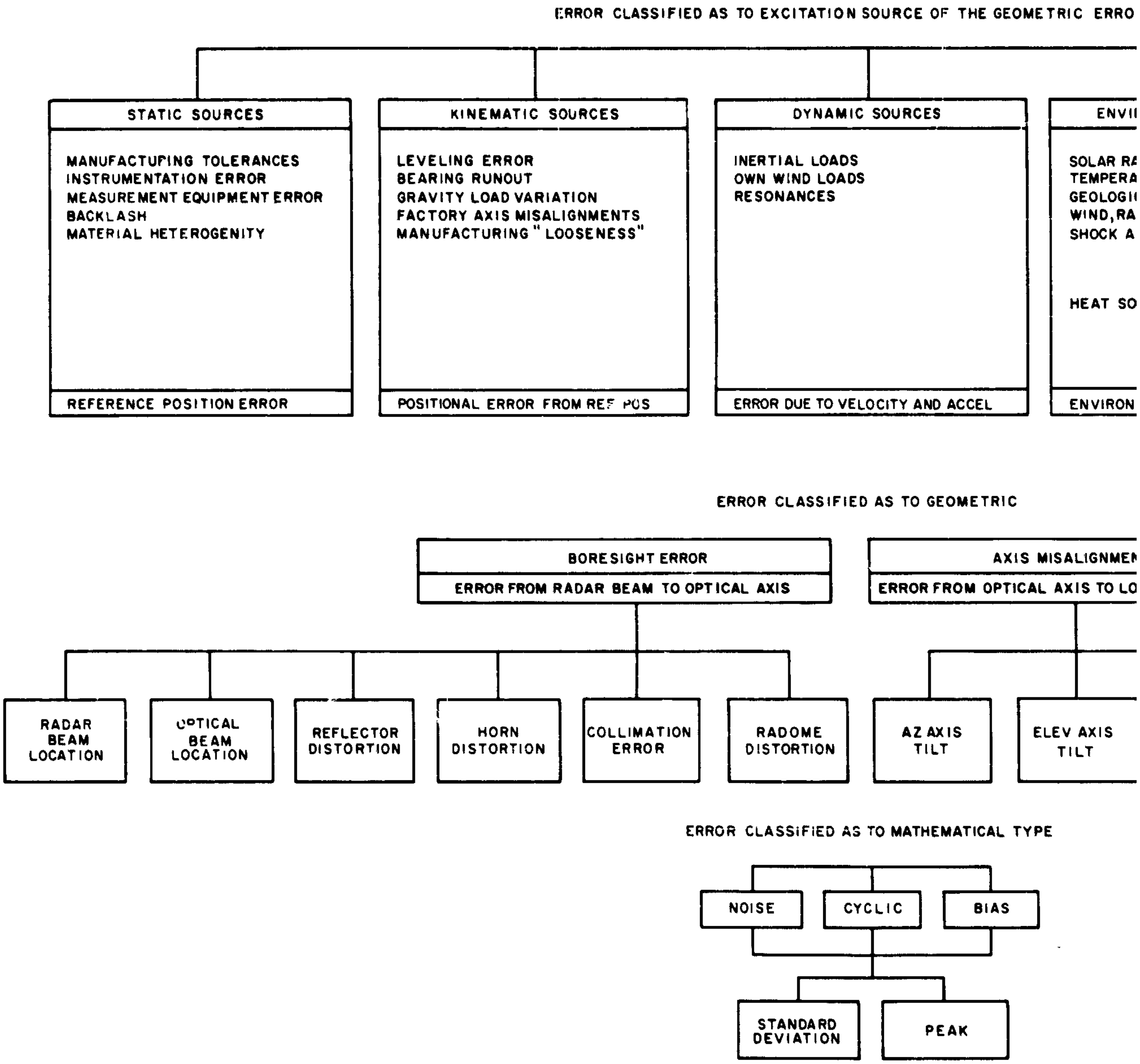


Figure 2-12. Average Effect of Random Phase Error on Sidelobe Level.


ERROR CLASSIFIED AS TO EXCITATION SOURCE OF THE GEOMETRIC ERRO
STATIC SOURCES
MANUFACTURING TOLERANCES
INSTRUMENTATION ERROR
MEASUREMENT EQUIPMENT ERROR
BACKLASH
MATERIAL HETEROGENITY
REFERENCE POSITION ERROR
KINEMATIC SOURCES
LEVELING ERROR
BEARING RUNOUT
GRAVITY LOAD VARIATION
FACTORY AXIS MISALIGNMENTS
MANUFACTURING "LOOSENESS"
POSITIONAL ERROR FROM REF POS
DYNAMIC SOURCES
INERTIAL LOADS
OWN WIND LOADS
RESONANCES
ERROR DUE TO VELOCITY AND ACCEL
ENVI
SOLAR RA
TEMPERA
GEOLOGI
WIND, RA
SHOCK A
HEAT SO
ENVIRON
ERROR CLASSIFIED AS TO GEOMETRIC
BORESIGHT ERROR
ERROR FROM RADAR BEAM TO OPTICAL AXIS
AXIS MISALIGNME
ERROR FROM OPTICAL AXIS TO LO
RADAR BEAM LOCATION
OPTICAL BEAM LOCATION
REFLECTOR DISTORTION
HORN DISTORTION
COLLIMATION ERROR
RADOME DISTORTION
AZ AXIS TILT
ELEV AXIS TILT
ERROR CLASSIFIED AS TO MATHEMATICAL TYPE
NOISE
CYCLIC
BIAS
STANDARD DEVIATION
PEAK

1

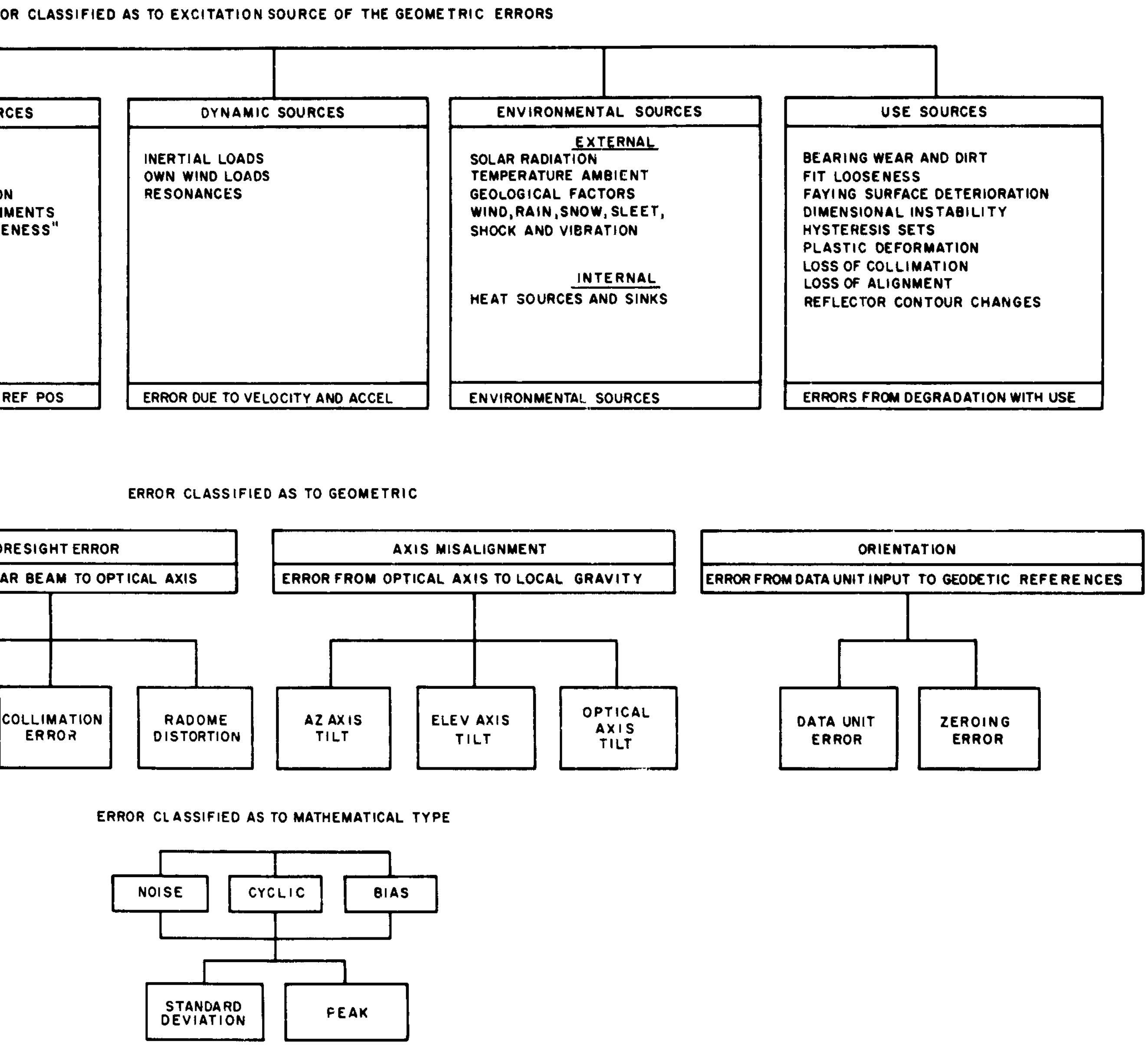


Figure 2-13. Error Classified as to Excitation Source of Geometric Errors.

2-4.2.1 Static Excitations. The radar tracker will be leveled, have its axes aligned and be collimated at some elevation and azimuth angle at the factory. At this static position, a number of errors will exist; for instance, the mount will be imperfectly leveled; the alignment of the elevation axis perpendicular to the azimuth axis will be subject to error; the boresight scope will not be truly perpendicular to the elevation axis and the parallelism between the boresight scope optical axis and radar beam axis may be imperfect. There are other sources of such error; measuring instrument error, for example. Since these errors exist at a given static or reference position, the errors may be termed static errors.

2-4.2.2 Kinematic Excitations. As the tracker is moved from the reference position to some new fixed position in azimuth and elevation, additional errors are introduced. For example, the level and bearing runout error may be made close to zero at the reference position but at some new position these errors become sizable and must be considered as error. Bearing runout and out-of-level will cause the elevation and azimuth axis to nutate with negligible error at some angle increasing to a peak error at other angles. In addition, unbalance of gravity load may cause different deflections than at the reference position and introduce error. Gearing errors, relative to the "perfect" reference position may be introduced. Such errors may be termed kinematic error.

2-4.2.3 Dynamic Excitation. In the event the tracker is moving at some velocity, own wind loads will be generated, cause deflection and hence introduce geometric error. If moving at some acceleration, inertial loads will be generated, cause deflections and hence generate error. These errors may be categorized as dynamic errors.

2-4.2.4 Environmental Excitation. Error may be introduced by a change in environmental conditions from those existing at the time the tracker is leveled, aligned and collimated. For instance wind, solar radiation, a change in temperature ambients, etc., will cause deflections and hence introduce error. The errors may be said to be environmentally induced and termed environmental errors.

2-4.2.5 Use Excitation. As a result of use, the physical condition of the radar tracker will change from what it was at the factory. Bearings will wear and introduce further runout and/or will become dirty and introduce noise. Shock and vibration may cause dimensional change. Surfaces may become dirty or nicked and cause misalignments. Fasteners may be loosened. Such errors are induced by use and may be categorized as use errors.

2-4.2.6 Conclusion. In a given application all sources of excitation must be considered and their influence suppressed into negligibility or the resulting error entered into the error analysis.

The foregoing information is not comprehensive but it does sum up excitation sources into a rational classification which will be of significance later in this report.

2-4.3 GEOMETRIC ERROR.

2-4.3.1 Introduction. At some point and under certain conditions the tracker may be regarded as having zero error. Dimensional changes or other inaccuracies may be induced by any of the excitation sources heretofore discussed, thus introducing spatial errors. These dimensional changes or inaccuracies, at various physical points in the tracker, must be translated into terms of their influence on error in the angle readouts in azimuth and elevation. An understanding of this aspect must be founded on a clear comprehension of the geometry of a two-axis (azimuth and elevation) tracker. In addition, it is convenient to divide the error from a line to the target in space to the gravity vector into two parts; boresight error and axis misalignment. In addition, there is a third category of error because the azimuth and elevation angles of the tracker must be established in relation to references such as gravity (elevation) and true north (azimuth). This latter category may be termed orientation error and is discussed in paragraph 2-4.3.5. A subsequent report will consider boresight error, misalignment error and orientation error more in detail.

2-4.3.2 Two-Axis Azimuth and Elevation Geometry. The three-dimensional aspects of the two-axis tracker and the relationship between the various axes is a complicated matter. The essence of the problem lies in the fact that errors in space are in the traverse plane and need to be reduced to the azimuth plane of the tracker. Further, errors in one axis introduce errors which may not be negligible in other axes. Consideration of all significant ramifications of the problem indicates that consideration must be given to the azimuth axis tilt, the elevation axis tilt, and the boresight axis tilt as they relate to a fourth axis, gravity.

1. Azimuth Axis Tilt. The azimuth axis of the tracker may depart from parallelism with the gravity vector for a number of reasons, including out of level, azimuth bearing face runout, and mechanical noise in the bearing. The general effect is that the azimuth axis nutates or wobbles as the tracker is rotated 360 degrees in azimuth (the azimuth axis is perpendicular to the azimuth plane established by the bearing face). Hence, at a fixed elevation angle, the line of sight "nods" as it rotates. This introduces elevation error which enters directly without need for mathematical manipulation or coordinate transformation.

2. Elevation Axis Tilt. The elevation axis is rigidly fixed and perpendicular to the azimuth axis, though not aligned perfectly at right angles to the azimuth axis. This is meaningless at zero elevation, but as the tracker rotates in elevevation, the elevation plane is not perpendicular to the azimuth plane but is slanted to the true vertical. This introduces an azimuth error which varies with the elevation angle; it also introduces an elevation error which is negligible. The previously discussed azimuth axis tilt causes the elevation axis to tilt and produce the same error as the absence of perpendicularity between the elevation and azimuth axes. The result is that azimuth axis tilt and absence of perpendicularity of the elevation and azimuth axis produce error in accordance with the following relation:

Eq (A) Azimuth error = product of tilt and tangent of the elevation angle.

3. Boresight Axis Tilt. The boresight optical axis is not perfectly perpendicular to the elevation axis at zero elevation. The boresight scope axis at zero elevation may possess error in elevation and azimuth due, perhaps, to scope resolution. The azimuth error at zero elevation must be manipulated as a function of the elevation angle as follows:

Eq (B) Azimuth error = product of the azimuth error at zero elevation and the secant of the elevation angle.

In conclusion, equations (A) and (B), though slightly simplified, represent the only coordinate transformation necessary in a simplified error analysis of geometric errors. A subsequent report will deal more in detail with these considerations.

2-4.3.3 Boresight Error. For the purpose of the discussion and assuming a boresight scope with a perfect optical axis, those considerations which introduce a lack of parallelism between the "perfect" optical axis and the radar beam axis may be termed boresight error. This error may be introduced from the following sources:

1. Collimation error
2. Mislocation of radar beam axis
3. Reflector distortion
4. Horn distortion
5. Mislocation of optical axis
6. Radome imperfections

Collimation error is introduced in the process of making the radome axis and the optical axis parallel under a given set of conditions. All microwave sources of error present in the system as a whole are present in the collimation process, plus others. Collimation is conducted under a given set of conditions; any departure from these conditions may introduce errors. Items (2) to (6) inclusive, just previously stated, constitute possible error sources. (The subject of collimation will be discussed more in detail in a subsequent report.)

The radar beam may be displaced from its mechanical position during collimation by a number of factors, including relative deflections between the horn and the reflector (from any or all load sources including thermal) even though the reflector surface may remain stable. This will introduce lack of parallelism between the reference optical axis and the radar axis and thereby introduce error.

The reflector surface may be changed from its manufactured state by the influence of wind and inertial loads, thermal distortion and other factors. Among other results, the radar axis will move relative to the perfect "optical axis" and thereby introduce boresight error.

Wind and inertial loads, thermal deflections and other excitations may cause the feed-horn to move from its collimation position, relative to the reflector. This will cause a shift of the radar axis in space relative to the "perfect" optical line to the target and thereby introduce boresight shift and the accompanying angular error.

While we have considered the boresight scope or optical axis as "perfect", the resolution of the scope and optical refraction offer sources of error, and these will be considered later in a subsequent report.

Radome imperfections may be essentially eliminated at one aperture through the radome, e.g., at the collimation position. However, as the radar beam "looks" through different portions of the radome, or as the physical form of the radome is distributed by environmental and use factors, the beam through the radome is "bent" differently and radome boresight shift errors are introduced. This topic also will be discussed fully in detail in a later report.

2-4.3.4 Axis Misalignment Errors. Consideration of the two-axis geometry discussed in paragraph 2-4.3.2 together with the discussion of error excitation in paragraph 2-4.2 should provide the reader some insight into the nature of errors due to misalignment. The subject will be further discussed in a subsequent report.

2-4.3.5 Orientation Error. In azimuth, references such as true north and local gravity must be selected. The determination of the reference location will be imperfect and thus introduce error. The process of "zeroing" data units to the reference will also be imperfect and introduce error. Data units will be characterized by inherent error; these three sources are combined and termed orientation error. The orientation error is measured from the input to the data unit to the geodetic references selected. Occasionally, error antecedent to the data input and not accounted for elsewhere must be considered.

Data unit error is discussed in detail in section VII later on in this report. Orientation error will be more fully discussed in a subsequent report.

2-5 SERVO ERROR

If the microwave and mechanical errors previously discussed were equal to zero, an error angle between the order signal (the target position) and the radar axis would remain. This error is the summation of a large number of error components which are entirely attributable to the power drive servos. In order to facilitate the discussion of each of these individual components of servo error, it is desirable to classify these errors in some manner as was done previously for microwave and mechanical errors.

2-5.1 CLASSIFYING ERRORS. Several methods of classifying errors are possible, each being oriented towards a different aspect of the consideration of errors. For instance, errors can be classified as to their point of origin, whether from within the various parts of the equipment or from without. Errors can also be classified as to the

physical nature of the error, such as electrical, magnetic, optical, etc. A third method of classification is as a function of the mathematical description of the error such as random (noise), bias and cyclic errors. It should be recognized from the outset that servo errors are not readily susceptible to classification since any particular error originating from a particular source may have random, bias and cyclic components and be the result of electrical, mechanical and magnetic actions at one and the same time. In addition an error classified as cyclic in one context of mission and performance requirements, may be treated by statistical methods in another context.

Since the major purpose of classifying errors in this study is to enable and facilitate a mathematical treatment of the errors in an error analysis, the mathematical classifications will be used, i.e., noise, bias and cyclic. Each individual error, or group of errors, all of which operate the same, will be discussed as to its nature, source and effect on the servo, its content of noise, bias and cyclic components, and its relative and/or absolute magnitude. The definitions given in paragraph 2-4 apply.

2-5.2 TABULATION. A tabulation of the errors and their classification is given in table 2-6. Beside each error an X is placed in appropriate columns to indicate whether that particular error can be classified as either noise, bias or cyclic. When an X appears in more than one column, either one of two situations is thereby indicated: (1) the error is either noise, bias or cyclic, but only one, depending on the situation, or (2) the error has more than one component. In practice, when making an error analysis, it is necessary to establish for each equipment and for each mission how to treat each error. Because of this, the errors are discussed below separately from their categorization, the emphasis being on the supply of sufficient information of the nature of the error that the reader can decide how to treat the error when the context is known. (Text to appear in final report).

2-6 ERROR ANALYSIS PROCEDURE

2-6.1 INTRODUCTION. This section on procedure will dwell on the preparation of error budgets antecedent to the initiation of design; the development of precise definitions in terms of the errors to be included in the error budget; their mathematical manipulation and environmental conditions, and the procedure utilized to generate error estimates in this study.

2-6.2 ERROR BUDGETS. Initial error budgets for the radar tracker may be predicted on the following rule:

"No one error is to be permitted to exceed one-fifth of the total allowable error and errors less than one-twentieth of the total allowable error may be considered negligible".

This rule is founded on the method of summing errors as the square root of the sum of the squares in accordance with the Central Limit Theorem to obtain the standard deviation of a normal distribution of the total error. For example, consider the case where no one

error exceeds one-fifth of the total. The allowable individual error in accordance with the rule is: Total Error/5 = E. But individual errors are added RSS style or, Total Error = $\sqrt{25E^2}$. Hence, 25 individual errors of size $\frac{(\text{Total Error})}{5}$ are permissible. If a total of 16 major errors were contemplated, then each error should not exceed $\sqrt{\frac{1}{16}} = 1/4$ of the total error. Errors less than one-twentieth of the total have negligible weight when added RSS style and may be ignored.

In a relatively inaccurate equipment, error analysis is a simple task as only a few large error sources need be considered. As accuracy requirements become more severe, many more sources of error must be suppressed and/or factored into error analyses.

In any event, the first task in a design synthesis is to prepare an error budget in accordance with the above technique modified, as appropriate, by experience. As design data is obtained, the error budget should be adjusted on a continuing basis over the life of the design.

2-6.3 ACCURACY DEFINITION. It is essential to obtain a clear definition of the accuracy involved in a given application from the system designers. A clear understanding of the mathematical definition of the errors must be obtained (see section 3-2), and the errors to be included in the total tracker errors must be specified. Also the environmental conditions and the portion of the target course in which full accuracy is required must be established. The period of time over which the accuracy is desired must be defined. To demonstrate some of these ideas, the specification requirement of "0.05 beamwidth at 8 kmc is referred to ground" is considered. It was established that this was a peak value (never to be exceeded) which was radial in space. The expansion of this brief definition into a full definition follows including the large number of assumptions that were made.

A restatement of the accuracy requirement follows: The tracker, employing a 30-40 foot diameter parabolic reflector, shall have a peak radial pointing error of 0.05 beamwidth of 8.0 kmc as referred to local gravity and an arbitrary, perfect north reference. This error allowance assumes operation over a modest range of environmental conditions without benefit of a prepared foundation. The stated error does not include orientation errors (such as geodetic error and local gravity error) or microwave error (such as thermal noise, propagation error or scintillation errors). It does include radar collimation error and error from boresight shifts originating in the radome, reflector, horn and mount which are excited by thermal, inertial, gravity, and wind loads or brought about by other factors such as use. It does include data unit error. The 0.05 beamwidth translates into more convenient values as follows:

1. A 40-foot reflector will have a 0.21 degree beam (55 percent efficiency).

2. The peak pointing error is then -

$$0.05 \times 0.21 \text{ degrees} = 0.011 \text{ degree} = 40.0 \text{ seconds} = 0.2 \text{ milliradian}$$

3. The standard deviations in the azimuth and elevation axes are more convenient design values. Hence, using a Rayleigh probability density function and assuming that azimuth error is approximately twice the elevation error, as found in practice, results in $\sigma_A = 0.05$ milliradian and $\sigma_E = 0.03$ milliradian. These values are standard deviations (σ = sigma) in azimuth and elevation, respectively, of a normal probability density function.

4. The CEP pointing error is most easily generated from the approximation CEP = 0.59 $(\sigma_A + \sigma_E)$ or 0.59 (0.05 + 0.03) = 0.05 milliradian.

Table 2-5 summarizes the specification error allowances:

2-6.4 ASSUMPTIONS. The following assumptions were made in connection with the allowable error definition:

1. Mission preparation. It is assumed that ample time prior to a mission will be available to tune the system, including on-site collimation if required, leveling, orientation, temperature stabilization, amplifier warm-up, etc.

2. Mission time. The full accuracy requirement is assumed to apply to a twenty to thirty minute mission only. Degraded performance may result from longer mission periods.

3. Environmental conditions. Error evaluation is based on consideration of moderate conditions of environment and not on the possible extremes.

4. Elevation track sector. Tracking from 0 degrees to 60.0 degrees in elevation is assumed for estimates of error. The tracker may track to 80.0 degrees or more with reduced accuracy.

TABLE 2-5. ERROR ALLOWANCES

Error Definition	Error Allowance
Radial Pointing Error (Peak)	----- 0.2 milliradian
Radial Pointing Error (CEP)	----- 0.05 milliradian
Azimuth Error (Std. Dev.)	----- 0.05 milliradian
Elevation Error (Std. Dev.)	----- 0.03 milliradian

5. Orientation. Orientation errors, such as errors in geodetic location and variations of local gravity from the vertical, are not part of the allowable error. Data unit error is included.

6. Microwave Error. Microwave errors, such as a propagation error, noise, and scintillations, are not considered as part of the allowable error. Collimation error and radome boresight shift are considered part of the error allowance.

7. Real Time Solution. Angular position is desired in real time without use of a computer or use of data reduction techniques.

8. Absolute Position Accuracy. The tracker must determine the satellite absolute angular position and not merely provide minimum off-RF boresight tracking error in space.

9. Low Velocity Errors. Errors unique to extremely low angular velocity rates such as stiction and backlash "breathing" are not included in the allowable error.

10. Foundation. Prepared concrete foundations shall not be used.

2-6.5 ANALYTICAL PROCEDURE. All errors in the designs in section 8 in this report have been suppressed into negligibility except those indicated in table 2-6.

The analytical procedure may be sequenced as follows:

1. All errors that have not been suppressed into negligibility must be tabulated and a defined, quantitative value of error assigned to each.

2. This assigned value may be a standard deviation or a peak to peak error. Peaks must be reduced to standard deviation as follows:

(a) Sinusoid error distribution: standard deviation = 0.7 peak
(b) Uniform error distribution: standard deviation = 0.6 peak
(c) Normal error distribution: standard deviation = 0.34 peak

3. Noise, cyclic error and biases will be summed in root sum square (RSS) style in accordance with standard practice to obtain the standard deviation of the total error.

4. Azimuth error is a function of elevation; it is desirable to express these azimuth errors in one figure rather than as a function of elevation. Hence, the average of the error from 0 degrees to 0.60 degrees is developed and used. The error at 0.30 degrees and 0.60 degrees is generated for reference.

5. The CEP is used as a single figure of merit instead of stating errors in one azimuth and elevation axis. Three such values are developed:

TABLE 2-6. RESIDUAL ERROR SOURCES

		Geometric Error	
Boresight Error	1	Collimation Error	Static Excitation
	2	Radome boresight shift	Kinematic Excitation
	3	Reflector-horn distortion from gravity load	Kinematic Excitation
	4	Environmental induced boresight shift	Environmental Excitation
	5	Use induced boresight shift	Use Excitation
Axis Misalignment Error	6	Azimuth axis tilt from leveling error	Kinematic Excitation
	7	Azimuth axis tilt from bearing runout	Kinematic Excitation
	8	Elevation axis tilt from factory misalignment	Static Excitation
	9	Elevation axis tilt from bearing runout	Kinematic Excitation
	10	Boresight scope misalignment	Static Excitation
	11	Environmentally induced axis misalignment	Environmental Excitation
	12	Use induced axis misalignment	Use Excitation
Orientation Error	13	Data Unit Error	Static and Kinematic Excitation
		Servo Error	
	14	Bias (Dead band) Error	
	15	Cyclic (Dynamic lags) Error	
	16	Servo noise error	

(a) CEP under factory conditions without the use of environmental error being considered.

(b) CEP under modest environmental conditions.

(c) CEP under modest environmental and use errors.

2-6.6 CONCLUSIONS. Experience has indicated that these methods result in an estimated error that correlates well with the actualities that develop from evaluations and on a conservative basis. The one exception are those errors that result from environmental factors, especially thermally induced deflections and use factors. There is a paucity of data in these areas and analysis is not of too much value. The values for these errors that are used in the error anclyses herein have been developed from experience tempered with judgment. More advanced information will be found in Section 7, Accuracy State of the Art, in this report. Added detailed coverage of Two-Axis Geometry as the result of studies currently in process will be found in a subsequent report, as will detailed bibliographic material.

Section III
MECHANICAL DESIGN CONSIDERATIONS

3-1 LOADS

Loads applied to the radar mount may be considered from the standpoint of structural integrity, or from the standpoint of their effects on the accuracy of the equipment performance. For the former, the interest is in the maximum or critical loads, and the strength of the structure, and in its anchorage; a degree of permanent set may be tolerated if it occurs only occasionally and if it does not jeopardize actual operation.

3-1.1 LOAD TYPES. To insure structural integrity, the loads must be evaluated from every phase of the equipment life; test, storage, transportation, erection and dismantling, operation, and survival when stowed or non-operational. Generally, the designer tries to determine a logical combination of loads that will produce the maximum stresses that may occur at critical or typical sections. For certain parts such as gears and field connections for transportable mounts, wear may result in ultimate failure. Where loads and stresses are repetitive fatigue is a danger. Vibration and shock, particularly during transportation, may provide the limiting loads.

If the radar mount is provided with an air-supported radome, it is usual to design the mount to withstand survival wind loads after failure of the radome deteriorates both by aging and possible accidental damage; it has a relatively short life. The assumption is also made that when the radome fails it falls free; that it does not wrap up on the mount so as to provide additional wind loads on the primary structure.

3-1.2 STRENGTH-STRESS. If a rigid, structural framed type radome is provided with a safety factor adequate for both the radome and the anchorage, there does not appear to be any justification for requiring the mount to be designed to withstand survival wind loads by itself; a radome should be fully as reliable as the mount. If may be speculated that if a rigid radome should fail there is less chance that it would fall entirely free; it appears more likely that it would entangle or cling to the radar mount, damage it, and in all probability cause it to overturn or be carried away.

Regular examination of any anchorage system should be scheduled where the character of the soil or the effect of weather could produce either loosening of the tie-downs, or a loss of the holding anchorage power. The reliability of the anchorage may be lower than that of the radome or of the mount.

3-1.3 DEFLECTION-ACCURACY. If, instead of strength performance is examined, a number of ways in which the loads enter into the design of the mount will be found. The stresses are usually limited well within the elastic range. In the case of transportable mounts permanent deflections or settling may be expected in the soil at the mount support areas.

Periodic checks will be required to establish possible need for re-orientation of the mount. The use of a radome should materially reduce the rate at which settling will occur.

The design of members for the imposed loads will frequently be based on their allowable deflections during operation, so that the stresses are extremely low. Deflections under operating loads will cause distortions of the reflector or of the feed support structure, or introduce a misalignment of rotational axes causing direct errors in data readout or transmission. For loads that occur in a periodic manner, the natural frequency and damping capacity of the structure may become important in controlling the maximum deflections.

A simplified model of the inertias and spring constants of the radar mount (as they apply to both the azimuth and elevation drives) will be needed for the design of the servo system. The sizing of the members may be determined by servo requirements rather than by ultimate strength or maximum allowable deflections. If the antenna is unprotected by a radome, the operational wind loads will usually be the source of the major power requirement.

The wind torques and associated loads for azimuth and elevation positions of the antenna can be determined best by the test of an accurate scale model in a wind tunnel. An examination of wind tunnel data for similar configurations may then be made to predict the power requirements. Corrections will be required for perforated reflectors as against solid ones, for types of backing structure, and for the offset of the reflector from the rotational axes. After these effects have been provided for, the estimated torques will be in proportion to the relative areas of the reflectors in the test data and the particular application. In a similar manner the power requirements of existing radar mounts can be scaled to a new project.

Occasionally, the nature of the new configuration is so different that past tests or experience cannot be applied. In these circumstances, resort to a simplified analysis is indicated for preliminary estimates of torque and power. A semigraphical method, applying the wind loading to the projected area or shadow from the wind direction, will supply a rough estimate of torque for each azimuth and elevation orientation.

The maximum torques will usually occur during slewing; if the tip velocity of the reflector is more than a few percent of the operating wind speed, the effect of antenna rotation on wind torque must be considered. For applications where continuous rotation of the reflector is required, the effect of relative wind across the reflector may result in significant increased average power and maximum torque.

A radar mount enclosed in a radome, unaffected by the external wind, has but little wind load from its own motion. A slight fanning effect would be present for high speed continuous rotation, but this load can be ignored for most tracking rates. The wind load torque for the maximum slewing rate can be estimated by assuming that the air is held still, and using the relative velocity of the reflector as wind velocity across its face. Since the fanning effect decreases to zero at zero velocity, it will not contribute to the system error which results from very slow motions.

There are other loads which must be taken into account in order to estimate power requirements or deflections. These include:

1. Friction of bearings, seals, slip rings, and rotating joints.

2. Friction from gearing.

3. Cable drapes.

4. Dead loads, static unbalance.

5. Inertia loads, acceleration.

Of these, the main azimuth bearing and seal friction may have the greatest influence on performance. Unless there is merely shearing action of a fluid, as in a hydrostatic bearing, there will be a sharp rise in friction at the start of motion. This is sometimes labeled as "stiction" and may be twice as great as the running friction. Where there is only shearing action, the friction load is proportional to speed, and smoother low speed motion can be anticipated.

Slip rings and rotating joints usually have small normal loads and although their coefficient of friction may be high the torque will be much less from these sources than from bearings and seals. Cable drapes may be considered as presenting loads proportional to both speed and displacement. Static unbalance will vary with the elevation angle, but the variation is smooth unless counterweights or their equivalents are free to shift suddenly. The friction of gearing can be significant, especially for power drives. Where anti-backlash devices are used the friction-stiction load is fairly independent of the applied torque and hence, affects low speed motion. The friction of the gearing on higher speed gears can also be damaging to performance; although the normal loads of the gear teeth are corresponding smaller, the friction affect is restored by the speed ratio or distance through which the friction load acts.

Inertia loads produced by acceleration are multiplied by the square of their speed ratio to the reflector. Where a motor is used with a gear reduction unit the inertia affect of the motor itself may well exceed the inertia of the one-speed structure.

3-2 THERMAL CONSIDERATIONS

A high degree of antenna pointing accuracy requires a well-defined radar beam and precise knowledge of the location of the beam with reference to specified coordinates. Distortion of the antenna structure will change the shape of the beam or change its relation to the reference coordinates or it may change both. The distortions can be caused by the application of forces such as wind or inertia loads to the structure, or by changes in the temperature of the structure.

3-2.1 COEFFICIENT OF EXPANSION. If every part of the antenna had the same thermal coefficient of expansion and if every part of the antenna were always at the same temperature, the structural changes caused by a change in temperature would produce a negligible pointing error. An increase in temperature would cause the antenna to "grow" in size with no change in proportions. If the wavelength of the radiated energy is considered as fixed, the increase in focal length of the dish would tend to cause defocusing of the beam. In an aluminum antenna having a 16-foot focal length, a change in temperature of 25.0 degrees F would increase the focal length by about 1/16 of an inch or by one twentieth of a wavelength at 8.0 kMc.

It is not practical to build an antenna in which all parts have the same coefficient of expansion. However, the coefficients of expansion of the materials are known quantities. The location of each particular material within the structure is also known. Therefore, the designer can circumvent or minimize most problems arising from differences in coefficients of expansion.

3-2.2 ANTENNA TEMPERATURES. The problem of temperature variations in the antenna is less manageable. These variations are only indirectly under the control of the designer. He can control their magnitudes, but the occurrence of a particular pattern of distortion is largely unpredictable. All three heat transfer mechanisms (conduction, convection and radiation) act upon the antenna. Heat is exchanged by conduction and convection between the antenna and ambient air, by conduction through the leveling jacks between the antenna and earth and by radiation between the antenna and its surroundings. The surroundings include the ground and the radome wall, or, in the absence of a radome, the ground out to the horizon and the sun and sky.

Conduction through the jacks need not be a major problem. The temperature of the ground will fluctuate very slowly with the seasons. Since the outrigger legs will respond to more rapid inputs such as ambient air temperature, this may give rise to substantial temperature gradients across the jacks. However, the jack affords a relatively small area for heat flow between the antenna and ground. The flow can be reduced to a negligible value by selection of a material, such as stainless steel, which has a relatively low conductivity.

The heat exchange between the antenna and the ambient air can be evaluated by considering the ratio $\frac{k}{hL}$. Here k is the thermal conductivity in BTU/hr/ft^2/°F/ft; h is the surface heat transfer coefficient (or surface film conductance) in BTU/hr/ft^2/°F, and L is the length in feet along the path of heat flow from the surface to the center of the body. For a tubular backing truss, L would be the wall thickness of the tube. If the value of the fraction k/hL (called the internal-external conductance ratio) is greater than 5.0, heat can flow through the body more easily than it can enter the body through the surface. As a result, the body temperature is substantially uniform. When a body having $\frac{k}{hL} > 5$ is thrust into an airbath, its temperature approaches the temperature of the bath exponentially.

The time necessary for the body temperature change to reach 63.2 percent of the final body temperature change is called the time constant. Its value in hours is equal to $\frac{CW}{hA}$ where C is the specific heat in BTU/lb/°F, W is the body weight in pounds, A is the surface area in ft.^{2}, and h is the surface film conductance as before.

The value of h depends on the nature of the surface and the wind velocity. For a painted metal surface, it will vary from 2.0 in still air to 8.0 at 30 mph. In an antenna exposed to a wind, therefore, the response of every portion will be different, depending on the effects of the wind on the surface film.

3-2.3 AMBIENT TEMPERATURE. It appears from the foregoing discussion that the ambient air temperature in a radome is not a critical factor as far as structural distortion is concerned. It is important for other reasons, however. The viscosity of lubricants (and hence the friction force in bearings) is a function of temperature. For this reason, and for others, such as personnel comfort, some degree of ambient air temperature control is desirable in a high-performance antenna.

3-2.4 CONCLUSIONS. In summary, the expedients available to the designer for minimizing thermal distortion due to convection and conduction to ambient air are these:

1. Provide a radome so that values of h will be fixed in magnitude and uniform over the antenna. This will provide uniformity of response to changes in ambient air temperature.

2. Select materials and structural geometry so that $\frac{k}{hL}$ for structural elements will be 5 or larger. This will minimize thermal gradients within the structure.

3. Select materials and structural geometry so that $\frac{CW}{hA}$ is as nearly as possible the same number for all structural components. This will allow the components to respond at nearly the same rate to changes in ambient air temperature. As a result, there will be little tendency for heat to flow from one component to another within the antenna.

4. Stir the air inside the radome; for instance, by exercising the antenna, so that the air is all at the same temperature.

5. Eliminate closed pockets in the structure which will keep the air from mixing uniformly.

6. Insulate sensitive portions of the structure in order to slow down the heat exchange between the structure and ambient air.

7. Ventilate the radome to minimize vertical temperature gradients in the enclosed air.

8. Select radome materials and finishes to reduce the effects of solar radiation on the temperature of the enclosed air.

9. Surround the structure with piping carrying a coolant, such as water, to distribute thermal energy within the structure or to remove it from the structure.

3-3 THERMAL RADIATION

Thermal radiation travels in straight lines at the speed of light. The solar radiation striking the antenna, therefore, can change abruptly with time and location on the antenna in response to motions of the sun, of the antenna, or of cloud shadows. Because of the directional property of the radiation, the antenna will be heated unsymmetrically. The ability to produce rapidly changing, unsymmetrical, thermal inputs makes radiation potentially more damaging than conduction and convection to antenna pointing accuracy.

Even in the absence of solar radiation, thermal gradients due to radiant heat exchange may be great enough to cause pointing error. This is illustrated by an experience with the Atlas tracker. The tracking antenna operates in a radome which is air conditioned and which is insulated except for an aperture which is covered by a Hypalon-Dacron window. On a cloudy, winter afternoon, the antenna was elevated from 0 degrees to 90.0 degrees within 30 minutes, there was a boresight shift in elevation which reached a magnitude of three seconds. Elevating the antenna had changed the attitude of the reflector with respect to its surroundings and upset the thermal radiation balance. The resulting temperature changes had warped the structure. When the antenna was depressed back to zero elevation, the boresight error returned slowly to zero, as the initial equilibrium was reestablished.

3-3.1 LIMITING FACTORS. The antenna designer must attempt to limit the effects of thermal radiation on the structure. Air conditioning is not directly effective for this purpose. Thermal radiation does not affect the temperature of the air through which it passes. An air conditioner could remove heat from the air after the air had been heated by contact with the antenna, which had been heated, in turn, by absorbing radiant thermal energy. However, the conditioned air would come in contact with all parts of the antenna, regardless of temperature.

A more direct approach to the problem is to install radiation shields on sensitive portions of the structure. The shields are an effective solution, wherever their use is feasible. Unfortunately, they cannot be used on the reflecting surface of the reflector where they are most needed. A deterrent to extensive use of shields is the weight and bulk or erection time that they add.

A reasonably effective expedient is to select finishes for the antenna which minimize radiant heating of the structure. The properties of interest in this problem are the emissivities and absorptivities of the finishes. For a given body at a given temperature, emissivity and absorptivity are equal. Emissivity is a pure number, the ratio of the total emissive power of the body to the total emissive power of a black body at the same temperature. Absorptivity

is the fraction of incident radiation absorbed by a body. For an opaque body, the properties of the surface are those which apply in steady state radiation problems. Therefore, the behavior of a body subjected to radiant heating can be controlled to some extent by the application of a proper finish.

3-3.1.1 Equilibrium Temperature. For a body exposed to solar radiation only and to no other thermal effects, the steady state equilibrium temperature depends on the relative magnitudes of the absorptivity of the body at solar temperature and its emittance at its own temperature. For a body in surroundings at a temperature close to its own, the steady state equilibrium temperature depends upon only the emissivity of the body. These two cases represent idealized models of an antenna with and without solar radiation.

In the first case, it appears desirable to have a low value of solar absorptivity and a high value of emissivity at ambient temperature. In the second case, a low value of emissivity appears desirable. It is not possible to optimize both these cases.

3-4 MATERIAL SELECTION

In a highly accurate antenna of the general type under study, the overall structural design will be determined by stiffness rather than by strength. If the antenna has a sufficiently high natural frequency and if deflections are kept to the low level required, the antenna will usually have adequate strength to survive environmental loadings. This is particularly true if the antenna is protected by a radome. As a result, the modulus of elasticity of the material rather than the strength of the material may be the parameter of most interest to the designer. This is not to say that the designer can ignore stress levels. Actually, in some portion of the structure the level of stress may be critical.

3-4.1 STIFFNESS. No signal parameter will determine the selection of a material. The objective is to get a good combination of desirable properties.

The specific stiffness (the ratio of modulus of elasticity to density) is a significant parameter in a material to be used in an accurate antenna. Fortunately for the designer, this ratio is nearly the same for the commonest structural materials, steel, aluminum and magnesium. Values of specific stiffness for some structural metals are given in table 3-1.

3-4.1.1 Beryllium. Beryllium is being used increasingly as a structural material and has a more favorable weight-stiffness ratio than any of the common metals. However, it is not presently a practical material for antenna construction because of its cost. This situation may change in a few years as the current development work on beryllium begins to pay dividends.

3-4.1.2 Invar. Invar is another structural material of interest because of its low coefficient of expansion. It is an iron-nickel alloy containing about 36 percent nickel and less than one percent total of carbon, manganese and silicon. By varying the nickel content in tenths of a percent and by proper combinations of heat treatment and cold work, the thermal

TABLE 3-1

Material	Modulus of Elasticity in Tension $E \times 10^{-6}$ lb^2/in	Weight (P_3) lb/in	Specific Stiffness E max/p inches	Specific Heat BTU/lb/°F	Thermal Conductivity BTU/hr/ft^2/°F/ft	Coefficient of Thermal Expansion $10\,in^{-6}$/in/°F	Cost Dollars/lb
Wrought Aluminum Alloy	10 - 10.6	.093-.107	105	.22 - .23	67.4 - 135	11.7 - 13.7	$ 0.26
Carbon Steel	29 - 30	2.82	106	.10 - .11	27	8.1 - 8.4	.04
Magnesium Alloy	6.4 - 6.5	.060-.0635	102	.245	24 - 80	14 - 16	.36
Beryllium	44	.0667	657	.45	87	6.4	62.00
Invar (Nickel Alloy)	24	.292-.295	81	.12	7.8 - 10.3	---	1.50
Titamium Alloy	14.5 - 19	.160-.170	111	.12 - .13	4.3 - 9.8	4.9 - 7.1	
Ampcoloy (Cast) (Aluminum Bronze)	15 - 18	.264-.272	66		22 - 33	9 - 9.5	
Wrought Alloy Steel	29 - 30	.280	106	.10 - .11	21.7 - 38.5	6.3 - 8.6	.04
Wrought Stainless Steel	24 - 29.4	.269-.289	104	.11 - .14	11.7 - 21.2	9 - 10.2	.30

coefficient of linear expansion can be reduced from small positive values to zero, and even to negative values. In applications where thermal distortions are large relative to elastic distortions, the use of Invar may be desirable.

3-4.2 CHOICE OF MATERIALS. In order to make a choice among the possible materials, the designer should consider cost, availability, thermal coefficient of expansion, thermal conductivity, long term dimensional stability, ease of fabrication, corrosion resistance, and specific heat.

On the basis of specific stiffness, cost, availability, ease of fabrication and corrosion resistance, aluminum and steel are undoubtedly superior to the other materials. An antenna mechanical designer will probably select aluminum or a combination of aluminum and steel for the antenna structure.

Certain important components such as gears, bearings and shafts will usually be made of steel. In order to match coefficients of thermal expansion, the designer may choose to make a steel supporting structure for these steel components. In some cases, gears are made of an alloy of aluminum and bronze, such as Ampcoloy. This is a material which has a coefficient of thermal expansion close to that of aluminum. If Ampcoloy gears are used in an aluminum gear box, the backlash in the gear train will not be affected by changes in temperature.

Most reflectors being made today in the 40-foot diameter range are aluminum. This is because the minimum material thickness needed for stiffness in steel are impracticably small. Once the designer selects aluminum for his reflector, he may decide to continue to use aluminum for the balance of the primary structure, thus avoiding some of the problems that arise from differences in thermal coefficients of expansion and in electrical potential.

3-5 GEOLOGICAL CONSIDERATIONS

Every structure, whether a building, bridge or a radar unit must be founded on soil, and many of the problems encountered in the foundations of these structures are due to the nature of soil. If a choice of soil as a foundation bearing material were generally possible, many foundation problems could be alleviated. However, more often than not, the soil must be used as it exists at a given site. Where the character of the soil is unsatisfactory, it may be possible to improve it by one of the many soil stabilization methods.

3-5.1 TYPES OF SOILS. It is impossible to use some soils as foundation material. Peats and organic silts are compressible and should be avoided where possible. The ideal foundation materials are sands, gravel, stiff clays, cemented soils and rock. Many combinations of the above soils in nature make the problem of representative soil properties difficult. Specimens of soil taken at points a few feet apart may prove to possess different properties.

Whereas the application of the principles of mechanics to materials such as steel is relatively straightforward and the relationships between stress and strain, Poisson's ratio, and

strength in homogeneous, elastic materials like steel are relatively simple, these relationships are not constant in soils and can vary greatly even with similar types of soil. Heterogeneity characteristics and the stratification of soils also tend to complicate the problems in soil prediction techniques.

Problems of soil mechanics as applied to a transportable radar tracker are quite complex. Erection on any type soil must be possible without the unit being subjected to any detrimental effects on the operational requirements of the tracker. Main problems encountered in erecting transportable units due to soil conditions are as follows:

1. Long-term settlement of the soil occurring under the tracker footings. Settlement in itself is not a problem unless it be differential settling, thus introducing beam pointing errors.

2. The effect of the soil under dynamic conditions. The elasticity of the soil must be considered in the dynamic response of the structure and resulting effects on the servo response must be evaluated.

Types of soils found in nature can vary from peat or quicksand having very little bearing capacity, to rock having an extremely high capacity. Even within this classification, a soil property can vary from season to season, depending on drainage conditions and location of the water table. Soils seemingly having the same physical properties in the laboratory can prove to have entirely different properties when tested in situ. A major reason for this situation would be variations in the nature and composition of the underlying stratum. A layer of sand over a layer of soft clay will be characterized by different elastic properties and different settlement rates than one over rock. The same holds true for similar soil which may be located over water tables at different depths.

To more completely understand the many problems inherent in soil mechanics, some information on factors affecting the strength of soils is necessary. The following by no means constitutes a complete treatise in the field of soil mechanics but represents only an attempt to illustrate the magnitude of the problem.

3-5.2 SUPPORTING POWER. The supporting power of a soil in general depends upon the density of the soil. Supporting power may be defined as the ability of a soil to resist penetration or lateral flow when loaded. This, in turn, depends upon the ability of the soil to resist shear. The shearing resistance of a soil depends upon the internal friction and cohesion properties of a soil. Internal friction is the resistance of soil grains to sliding over each other; cohesion is the resistance of soil grains to pulling apart. Internal friction is greatest in sand, less in silts and least in clays; cohesion is greatest in clays, less in silts and least in sands.

3-5.2.1 Shear Strength of Soils. The shear strength of soils can be expressed by the use of Coulomb's law:

Eq 3-1

$S = C + \sigma \tan \theta$, where; S = shear strength, C = cohesion stress, σ = normal external stress, and θ = angle of internal friction. The shear strength of a frictional soil is porportional to the normal pressure on the plane of shear. The strength of any points, therefore, varies with the direction of the plane on which the shear stress acts. In cohesive materials, the shearing resistance is more like that of a solid material, being the same in all planes and independent of the normal pressure.

In the above formula, for frictional soils C = 0

$$S = \sigma \text{ Tan } \emptyset$$

For cohesive soils $\emptyset = 0$

$$S = C$$

3-5.2.2 Bearing Strengths of Soils. Shear failure or rupture and lateral displacement of a soil beneath a footing occurs when the contact pressure reaches the ultimate bearing value of the soil. Security against rupture is obtained by estimating the ultimate bearing capacity of a given soil and limiting the contact pressure to some value less than the ultimate.

One procedure for determining the ultimate bearing capacity is the application of semi-empirical equations proposed by Terzaghi.

For round footings:

$$q = 1.3\, C\, N_c + .6\, \gamma\, ,\, B \frac{N\gamma}{2} + \gamma_2\, D\, Nq \qquad \text{Eq 3-2}$$

For square footings:

$$q = 1.3\, C\, N_c + .8\, \gamma\, ,\, B \frac{N\gamma}{2} + \gamma_2\, D\, Wq \qquad \text{Eq 3-3}$$

where; C = cohesion stress

γ_1 = density of soil below footings

B = footing size

γ_2 = density of surcharge

D = depth of surcharge

The values of N_c, $\frac{N\gamma}{2}$ and Nq which depend only on the friction angle may be obtained from figure 3-1.

The equations are intended only as expressions which are approximate and conservative. They give estimates which are of practical value but must be tempered with considerable judgment.

3-5.3 SOIL SETTLEMENT. For cohesionless soils the settlement of a footing depends upon the stress-deformation characteristics of the material. The rigidity of sand increases markedly with an increase in relative density and is approximately proportional to the confining pressure. Most of the settlement, about 95 percent, will take place during the first few minutes after the load is applied to the footing. Additional settlement will occur because of the compaction effect on the soil of vibratory loads. This is particularly significant in footings located on loose sand.

For cohesive soils, the rate of settlement is much smaller than that of cohesionless soils. Clays are so impervious that the water is almost trapped in the pores. As a load is applied to the soil, a pressure develops in the pore water. This pressure tends to make the water flow out. The flow is rapid at first, but as the pressure decreases the flow decreases. As the water is forced out of the loaded areas the soil particles move closer together; thus the surface settles.

There are empirical equations available in any text book on soil mechanics for the estimation of footing settlements. This will not be presented here because of the nature of our problem, which will be discussed in the summary.

3-5.4 SOIL DYNAMICS. The science of soil mechanics has not yet found a good solution to the problem of predicting the natural frequency of vibrating foundations; the primary reason is that soils do not behave in a truly elastic, semi-infinite isotropic manner and exhibit many non-theoretical characteristics.

3-5.4.1 Vibrating Structures. A vibrating structure will become the source of periodic impulses which are transmitted into the subgrade, similar to sound waves. These impulses set the soil particles directly below the foundation into motion in phase with the foundation. At a greater distance, the soil will vibrate opposite to the foundation motion or 180.0 degrees out of phase. There are alternating areas of in-phase and out-of-phase zones with the spacing of the zones depending on the velocity of propagation of the elastic waves in the soil. If the soil is stratified, the problem is complicated by the reflection and refraction of the elastic waves that occur at the interfaces. Oscillator tests indicated that the amplitude of vibration was magnified many times when located on layered subgrade as compared to a semi-infinite medium. This magnification, however, did not exist when the depth of the first layer of soil was approximately three times the diameter of the foundation.

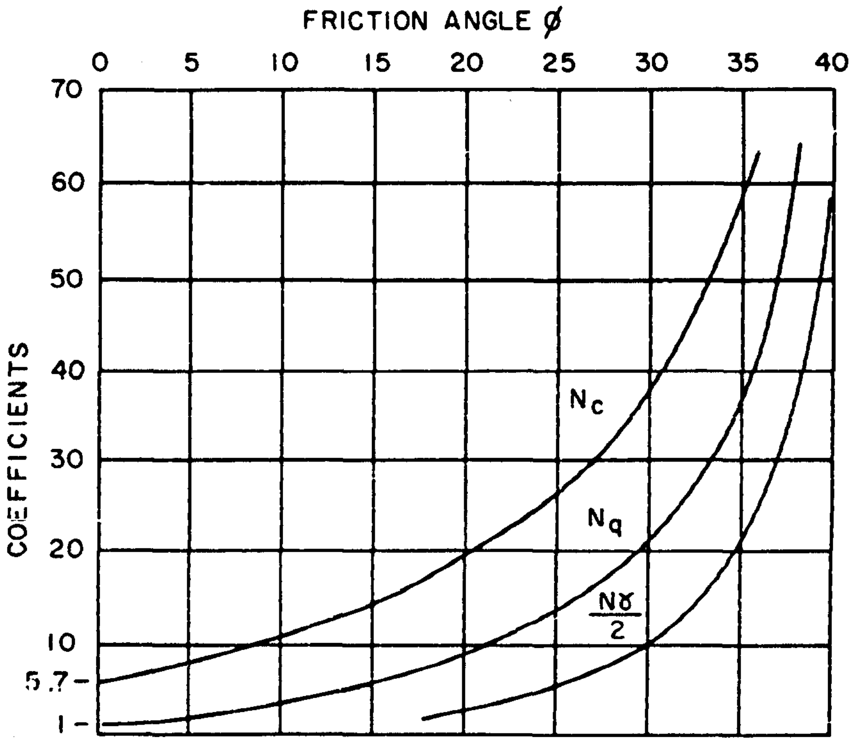


Figure 3-1. Bearing Strength Values.

3-5.4.2 Elastic Properties. The soil dynamics problem, as applied to a transportable radar unit, consists mainly of the determination of the elastic properties of the soil. These include the soil stiffness, effective soil vibrating mass, and soil damping. The above data will be included as a mass, spring and dash pot combination in the lumped-parameter system analyzed to determine the natural frequencies and response of the mechanical-servo structure.

3-5.4.3 Shear Modulus. One method of determining the soil stiffness and effective vibrating mass has been presented by Richards*. As a result of field experiments, he developed a relationship between the shear modulus of elasticity of the soil and the spring constant of the soil in the vertical direction.

$$K = \frac{4Gr_o}{1 - \upsilon} \qquad \text{Eq 3-4}$$

where G = shear modulus

r_o = radius of footing

υ = Poisson ratio

The effective vibrating soil mass is approximated by the equation

$$m_e = \rho r_o^3 \qquad \text{Eq 3-5}$$

where ρ = mass density of soil

The shear modulus for soils depends on the confining pressure as well as on the water content, which may have seasonal variations near the surface. One method of estimating the shear modulus of soils is to determine the velocities of the elastic waves through the soil. The velocity of propagation of the elastic waves varies with the type of soil and the water content. The velocity of propagation of the shear wave is given by the expression:

$$V_s = \frac{G}{\rho} \qquad \text{Eq 3-6}$$

where G = shear modulus

ρ = mass density of soil

* This reference will be keyed to the bibliography in a subsequent report.

From this equation, it can be seen that as the water content of the soil increases (ρ decreases) the shear modulus decreases.

Another method to obtain the shear modulus is by evaluating the dynamic behavior of a small oscillator at the site. Data obtained from oscillator tests will be fairly reliable when borings show the subgrade to be reasonably uniform. When the subgrade is stratified, the information is less reliable.

For a transportable unit, the structure as previously mentioned must be capable of operating on any type soil, therefore, the value of shear modulus will not be known before actual field erection. However, a range of values from 1000 to 6000 psi can be expected to be encountered in the field. In the dynamic analysis of the system, the two extreme soil conditions can be used in calculations and their effect on the servo response computed. If the dynamic analysis indicates that the lower value of shear modulus has a detrimental effect on servo response the soil stiffness may be increased by use of various stabilizing techniques.

3-5.4.3.1 Damping. The dynamic response of a system also depends upon the amount of damping in the structure. The damping inherent in the soil could be an important factor. Unfortunately, very little information is available on the damping of various soils. Here again the percent damping could vary from values of about five to 20.0 percent.

3-5.5 FOUNDATIONS. It may be necessary to increase the bearing capacity or the stiffness of the soil at a site. Therefore, it is important to find methods of modifying the properties of soils in place. These methods vary from providing proper drainage at a site to injecting materials into the soil to change its chemical structure. A few methods will be discussed here and comments made on its practicality for transportable radar units.

3-5.5.1 Soil - Cement Mixtures. In order to provide an adequate strength increase an intimate mixture of soil and cement is required. To be economical, the soil-cement mixture must be made in place, so this limits the use almost entirely to surface treatment. The mixtures can be made with almost any soil; the result is equivalent to a low grade concrete. The cement content generally varies from 8.0 to 12.0 percent by volume.

In the field the soil is pulverized to the desired depth, cement is added, thoroughly mixed and rolled in place. This method has proven quite satisfactory for surface stabilization and can be used with a minimum of extra equipment.

3-5.5.2 Cement Grouting. Cement grouting is accomplished by pumping into the ground a slurry consisting of a water suspension of cement. Small amounts of chemicals are often added to modify the characteristics of the grout. This type of grouting is ineffective in medium to fine sands and in silts. For these soils chemical grouts may be used.

The process consists of the consecutive injections of two solutions into the natural soil through a pipe driven to the required depth. A chemical, such as a modified silicate of soda solution, which will penetrate into the soil wherever water will pass, is injected into the soil. A second chemical, the reagent (calcium chloride solution) is then injected into the ground. Upon contact of the two solutions in the ground a hard calcium silicate gel is formed. The result is a soil having very high bearing capacity and high stiffness.

The problem with both the cement and chemical grouting is the necessity of having special equipment available for pumping the materials into the soil under high pressure. This requirement would preclude the use of these methods for a transportable radar unit.

The first method, soil-cement mixture, seems to be the most practical for transportable units. If the soil at a site has very poor bearing capacity or has low stiffness, the soil directly under the ground pads may be removed and pulverized, mixed with cement and properly replaced. This could be accomplished quite simply by two or three men equipped only with pick and shovel.

3-5.5.3 Conclusions. The soil mechanics problem facing the designers of a field transportable tracker are two fold. The effects on system accuracy due to the settlement of the soil beneath the footings must be considered. Settlement will occur when the soil is loaded by the dead weight of the structure and by the various live loads, including wind and self induced vibratory loads. The other parameter to consider is the effect on the dynamic response of the system due to the elastic properties of the soil.

As previously stated, uniform settlement is not detrimental, but differential settlement of the footings will cause beam pointing errors. This type of settlement will occur if the footings are unequally loaded or if the soil beneath each of the footings have different subgrade stratification. This difficulty can be alleviated by designing the antenna so that all footings are equally loaded and by choosing a site that has a uniform subgrade. Since the area involved in a site is relatively small there is a good possibility that the subgrade will be uniform. However, only borings taken at the site can substantiate this condition.

If differential settlement does occur, beam pointing errors will be introduced into the system. These errors can be minimized by periodically checking and correcting the level of the antenna mount. There are methods in which any "out-of-level" condition can be sensed and corrected for automatically. These methods are discussed in another part of this report.

The soil dynamics problem is somewhat more complicated. As previously explained, the soil parameters at a site cannot be predicted before field erection. It would then seem reasonable to design the system for the worst possible soil condition that could exist in the field. This is a conservative approach, however, one that will not significantly penalize the transportability of the tracker.

As seen from equation (6), the soil stiffness increases with the size of the footing. Therefore, by designing a footing that is as large as can practically be transported, the effective soil stiffness will be relatively high when compared to a footing designed based on allowable soil bearing capacities. A side benefit derived from this approach is the low value of soil bearing pressure under the footing. By keeping the bearing pressure low, the settlement of the footings will also be minimized.

If the soil conditions are such that additional stiffness is required, then the cement soil stabilization method may be advantageously utilized.

Section IV
MECHANICAL DESIGN ALTERNATIVES

4-1 REFLECTING SURFACES

An excellent discussion of reflecting surfaces can be found in volume 26 of the Radiation Laboratory Series entitled, "Radar Scanners and Radomes". Since the publication of this work, in 1948, additional types of surfaces have come into use. The sandwich construction originally developed for aircraft structural applications is now being widely used for reflector panels. The combination of high stiffness in relation to weight which is characteristic of sandwich construction is desirable in antennas as well as in aircraft. In addition, the characteristically smooth, uniformly supported skin of a sandwich panel makes an excellent reflecting surface. Sandwich panels can be built to close contour tolerances and can retain their contour accuracy under service conditions.

The sandwich skin materials most frequently used on reflectors are aluminum and fiberglass. When fiberglass is used, the skin which acts as the reflecting surface must be made electrically conducting. This can be done in a number of ways. Metal foil or metal screening can be bonded to the surface or the surface can be metal sprayed.

If the panel has a doubly curved surface, the skins must be preformed before bonded into the sandwich. In the case of a fiberglass skin, the sheet is laid up on an accurately contoured male mold and then cured. The skin can be built up to any desired thickness by adding layers of cloth. Aluminum skins are usually spun, stretch formed, or Androformed. The size of available rotating machine tools restricts the use of spinning to skins for reflectors smaller than twenty feet in diameter. The other two processes are in common use for larger sized reflectors. In stretch forming, the aluminum sheet is pulled down against an accurately formed male mold of the reflecting surface until the yield point of the material is exceeded. When the sheet is released, it retains very closely the shape of the mold. No mold is required for Androforming; the sheet is pulled through an adjustable, curved die which stretches the material beyond its yield point. The machine can be adjusted to produce a wide range of double curvatures. Ordinarily, the curvature of the reflecting surface is gradual enough so that skins of the same contour can be used on both sides of a panel.

A type of aluminum skin which does not require forming resembles the "Lap-strake" construction of small boats. The panel skin consists of many overlapping flat strips which are individually so narrow that they closely approximate the reflecting surface. The bonding operation which joins the skins to the core also joins the skin segments to one another. This construction is somewhat less expensive than those using preformed skins. One drawback is that each overlap makes a step in the reflecting surface, which limits the contour accuracy obtainable.

The usual lower limit for aluminum skin thickness is about 0.030 inches, where the skin is to be exposed to possible hail damage or rough handling. If durability is not critical, the skin can be thinner. Honeycomb panels have been made using aluminum skins 0.005 inches thick. The honeycomb core pattern tends to show through very thin skins. This could be a disadvantage if extreme contour accuracy were required.

Paper, aluminum and fiberglass honeycomb are commonly used for core material. Since honeycomb is the expensive element in the sandwich, many lower cost core materials have been tried. The low cost core most frequently used is foamed plastic.

Honeycomb tends to form a saddle-shaped surface even when curved in one direction. The pressure applied as part of the bonding process is often enough to force the core into uniform contact with the skins. If the core is thick or if the curvature is sharp, the core may have to be preformed.

The severe environments encountered by missiles and high-performance aircraft have led to expensive refinements in sandwich construction. Stainless steel skins and stainless steel honeycomb cores are brazed in vacuum to form components of supersonic aircraft. The environments encountered by antenna reflectors are not severe enough to warrant the use of exotic and therefore expensive materials.

The ribbon thickness and cell size of the honeycomb core determine the shear strength of the core. The thermal conductivity of the panel is largely determined by the same core parameters, since heat flows most easily from skin to skin through the core ribbon rather than through the air in the cell.

The core ribbon may or may not be perforated. If the adhesive system used is one which generates gas during curing, perforated core will be used to prevent entrapment of the gas. If water enters the panel later on, as a result of damage, the perforated core is less effective than the unperforated in restricting its dispersion through the panel. The presence of water in the panel is undesirable since it may cause enough internal corrosion to damage the panel. If it accumulates in sufficient quantities to fill a complete cell and then freezes, it can rupture the core. An important consideration in panel design is the elimination of points of water entry. Leakage can occur at panel edging, at inserts, at rivet holes, through the cores of blind rivets or anywhere else the skin is interrupted.

The skin and core materials may be used in various combinations in sandwich panels. Aluminum honeycomb core, for instance, may be used with fiberglass skins. At assembly, the skins are coated with adhesive and, with the core between them, are laid up on a male mold. Heat and pressure are applied to cure the adhesive. When curing is complete, the panel is an integral unit. The contour accuracy of the mold is critical because it controls the final shape of the panel. Considerable manufacturing know-how is involved in the curing process. Unless the proper amounts of heat are applied to the various glue lines, they will cure and contract at different rates and, at release, the panel will warp away

from the mold. An accurate mold is a necessary but not a sufficient condition for the production of accurately contoured panels. The bonding process also must be controlled in such a way as to exactly reproduce the mold contour in the panel.

The mold is an expensive item of tooling. The expense must usually be written off against a particular antenna. Hence, special effort must be made to reduce the cost of the mold. The same mold can be used for both stretch-forming and for bonding if this use matches the production schedule. Symmetrical reflectors can be made with a minimum of mold cost since they permit the use of small repetitive portions of the contour as a mold shape.

The bonding system used is closely tailored to the performance desired from the sandwich. High temperature, high strength systems are relatively expensive but, fortunately are not needed in reflector construction. The thermal conductivity of the adhesive has an important effect on the conductivity of the panel. A film of adhesive lies between the skin and the edge of the core ribbon. In flowing from skin to skin, heat has to pass through the adhesive film. If the film has a high conductivity, the panel conductivity will also tend to be high. This will reduce thermal gradients across the panel and hence, the warping of the panel.

Because of its structural efficiency, the sandwich type of construction is used not only for reflector panels, but for reflector primary structure as well. One reflector fifteen feet in diameter consisted of a single, contoured slab of alternate layers of fiberglass and aluminum honeycomb. After the slab was bonded, the rear surface was machined to taper the structure toward the edges and thereby reduce weight.

Another type of construction uses fiberglass-foam sandwich for radial and circumferential beams. The reflecting surface is metallized fiberglass and the back is fiberglass sheet. The entire structure is assembled by bonding.

4-2 REFLECTOR BACKING STRUCTURE

The function of the reflector backing structure is to hold the panels in the proper alignment with respect to one another and to the reference axes of the antenna. Since it is an elastic structure, it can maintain these alignments only within a certain tolerance under design loads.

4-2.1 DESIGN. The magnitude of the tolerance is indirectly defined by the requirement for antenna pointing accuracy; that is, the designer must interpret reflector deflection in terms of beam shift and beam shape. Then he must decide whether this error, in combination with all the other errors, produces an acceptable pointing error. The deflection requirements, therefore, provides one critical design parameters. A second, equally critical, design parameter is the natural frequency requirement. The stiffness of the reflector must be high enough to permit the attainment of certain minimum structural natural frequencies

which are needed for proper servo performance. A third important design parameter is the strength requirement. The structure must survive maximum loads without being damaged. Usually, if the reflector meets the two stiffness requirements, it will be strong enough.

To meet the requirements of this study, the reflector must be designed for compact storage and rapid erection. Therefore, the designer will attempt to build the reflector from a few basic elements which stack or nest compactly and which require relatively few interconnections to form a stable structure. Probably the reflector panels will not contribute greatly to the strength of the structure. The number of interpanel connections necessary to utilize the panel as part of the structure is prohibitive from the standpoint of erection time.

The simplest backing structure would consist of an array of identical radial ribs as shown in figure 4-1(a). This arrangement does not permit installation of a feed at the reflector vertex. If the ribs are cut away at the center of the reflector to make room for a feed, a moment-carrying member must be provided at the inner ends. Figure 4-1(b) shows the structure modified for a Cassegrain feed.

In structures such as those in figures 4-1(a) and 4-1(b), a single rib can deflect without transferring load to adjacent ribs. If one or more concentric rings are added, as in figure 4-1(c), each rib can be supported by adjacent ribs. Under symmetrical loadings, the rings carry hoop stresses. The structure forces more members to resist an applied load than do the structures in figures 4-1(a) and 4-1(b).

If diagonals are added, as in figure 4-1(d) the stiffness of the structure is further increased. The diagonals are especially important in resisting torsional deflections about the focal axis. Vibrations about this axis tend to be excited when the antenna moves in azimuth while pointing at a target near the zenith.

An actual structure for a 40-foot diameter reflector will be more complicated than figure 4-1(d). There will be more radial members since the maximum span between them will be limited by the largest practical panel size. There will probably be more circumferential members also. It will not be necessary to have diagonals between each pair of rings.

The final complexity of the structure depends upon the relative weights the designer assigns to the conflicting requirements he faces. As the structure develops from figure 4-1(a) through figure 4-1(d), it becomes heavier and more expensive, takes longer to erect and requires more shipping space. It also becomes stiffer. The handicaps have to be accepted in some degree in order to achieve the stiffness which is all-important.

The ribs and rings of figure 4-1 should be visualized as being members of considerable depth. They may be either trusses or built-up beams. A secondary function of the rings is to stabilize the ribs against twisting.

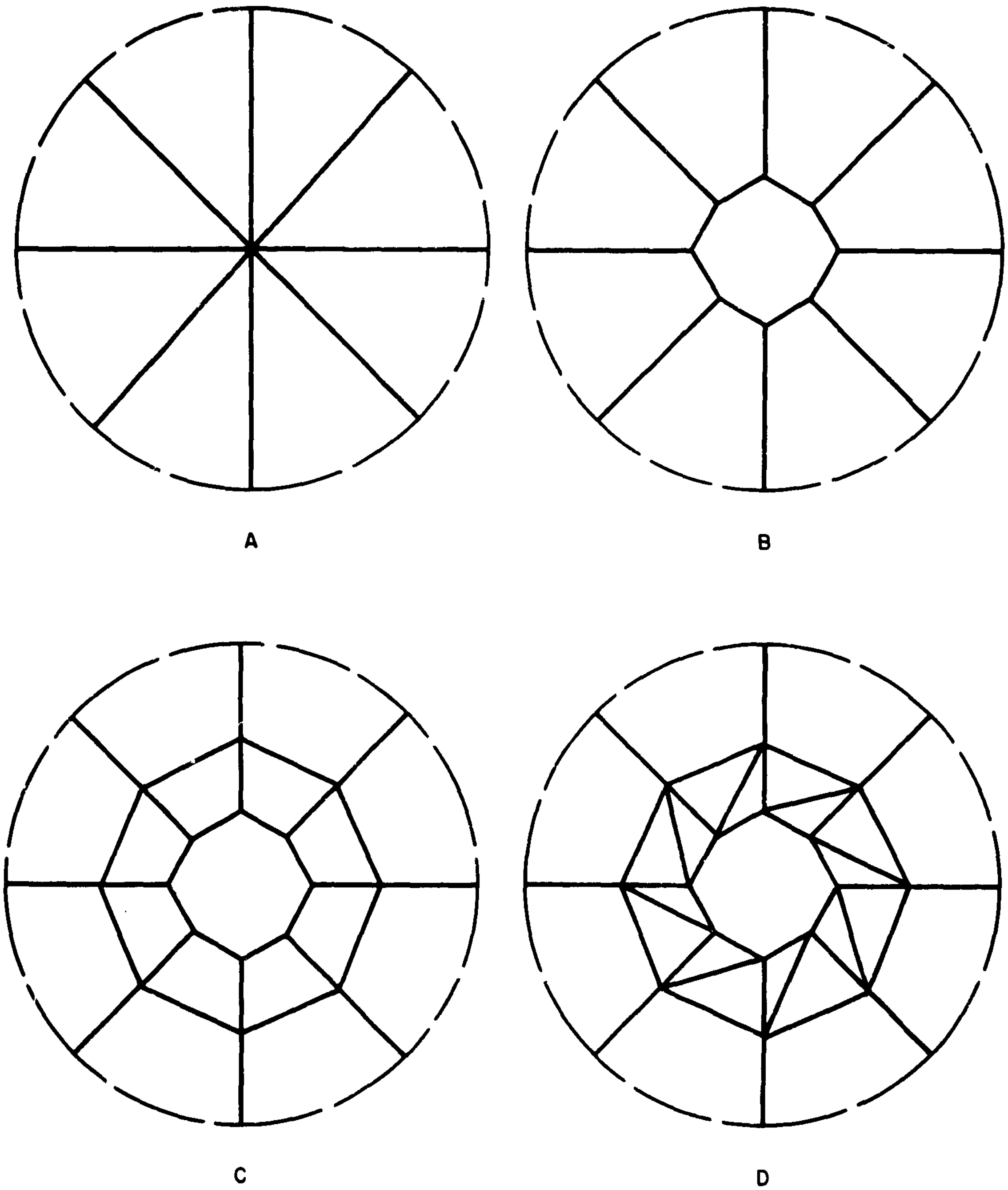


Figure 4-1. Backing Structure Arrangements.

For a given structural pattern such as any of these in figure 4-1, the stiffness can be increased by making the ribs deeper. This expedient has a definite limit, however. As the reflector structure is made deeper, the center of gravity of the structure is moved away from the elevation axis. The distance of the center of the gravity from the elevation axis should be restricted in order to minimize unbalance and moment of inertia with respect to that axis.

4-2.2 CONCLUSIONS. The reflector should not be imagined as resisting only axial symmetrical loads. It may be loaded to some level of intensity in any direction. A good rule to observe is: "A space structure is stable under any system of loads if its surface is divided into triangles". The whole reflector structure does not need to be configured in this way. However, there should be a central core structure which is universally stable.

The reflector structure may require local reinforcement wherever large, concentrated loads are to be applied. The attach points for horn supports and the support points for the reflector proper are typical examples.

An adapter will be required to make a transition from six or eight points on the reflector structure to the elevation shaft.

4-3 AZIMUTH PLATFORM

The azimuth platform is that portion of the antenna structure which lies between the azimuth and elevation bearings. The azimuth platform rotates in azimuth and carries on it the structure which rotates in elevation. For an antenna using a single azimuth bearing, the dimensions of the platform and its position with respect to the structure are roughly established by requirements which are described elsewhere in this study.

4-3.1 AZIMUTH AND ELEVATION BEARING LOCATION. The need for a high overall natural frequency for the antenna has been covered elsewhere in this report. Bearing stiffness and critical bearing load parameters are also discussed elsewhere. In the light of the material in these two sections, the azimuth bearing should be located vertically as close to the elevation axis as possible. For antenna configurations, like Designs A and B, such a location has the following advantages:

1. In an antenna which is subjected to wind loads, the center of wind pressure at maximum wind load is near the reflector vertex. Therefore, overturning moments on the azimuth bearing will be smallest if the bearing is close to the vertex. The wind overturning moment will probably determine the maximum load per rolling element or per pocket in a bearing.

2. The higher the azimuth bearing is mounted the less will be the inertia of the structure mounted on it. For a given azimuth bearing, then, the higher the location on the structure, the higher the natural frequency in rocking modes for the portion of the structure

which rotates in azimuth. The motor torque required to accelerate the antenna about the azimuth axis will be lower. This latter is a minor consideration, however, since the reflector inertia is very much larger than that of the platform.

3. The higher the azimuth bearing is mounted, the less will be the antenna dead weight on the bearing.

Some of the same consideration apply to location of the elevation bearings which form one bound of the azimuth platform. The elevation axis should be located on the focal axis as close to the vertex of the reflector as possible. This location minimizes the moment of inertia of the structure which rotates in elevation. It also, on the average, minimizes wind torques about the elevation axis. Both these effects are favorable to the elevation power drive.

The position of the bearings along the elevation axis is not so clearly indicated. From the standpoint of low inertia, they should be located close together, since considerable structural and drive weight is associated with them. However, the reflector structure requires fairly widely spaced supports and, for rigidity, the bearings should be close to those supports. There is also a requirement for access to the vertex of the reflector in a Cassegrain configuration. The final location is a compromise between these conflicting requirements.

The placement of both elevation and azimuth bearings close to the reflector vertex results in a relatively compact Azimuth platform. Figure 4-2 shows a number of possible configurations which can conform to the general requirements outlined above. The selection of a configuration should, if design time permits, be made on the basis of relative weights, inertias, stiffness and torque requirements.

4-4 BEARING SELECTION

By definition, a bearing is a means of positioning one part with respect to another in such a way that a relative motion is possible.

4-4.1 BEARING TYPES. For the purposes of this discussion, we shall exclude the trivial classes of bearings used for very limited angular motion such as bonded elastromer bushings "crossed-springs", elastic suspensions, etc., and consider only the broad general classes of slider-element bearings and roller-element bearings. The former of these two classes is typified by the classical journal bearing, wherein the moving elements are separated by a thin film of lubricating fluid; the latter by the well-known modern ball or roller bearing, where the moving elements are separated by a number of identical spherical or cylindrical rolling elements.

In further consideration of the slider-element class we recognize two sub-classes which depend on the method of establishing the supporting fluid film.

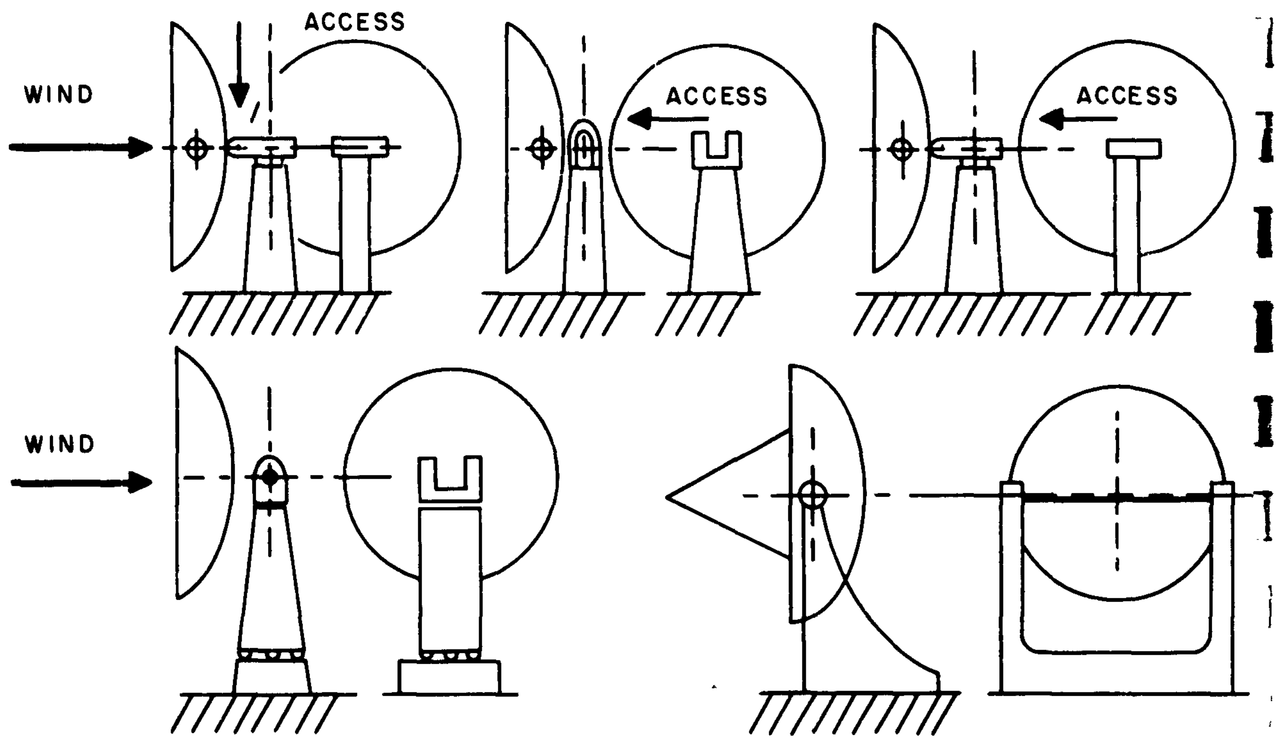


Figure 4-2. Elevation and Azimuth Bearing Location Configurations.

The hydro-dynamic fluid bearing depends entirely upon relative motion between the parts to establish and sustain the fluid film. The hydrostatic fluid bearing depends upon an external pressure system to establish and sustain the supporting film.

Because of the nature of the requirement for satellite trackers for frequent starts, stops, reversals and generally slow, uniform motions, the use of pure hydro-dynamic bearings for the main power drives or axis support is universally shunned as a poor choice so we shall exclude them from further discussion.

The remaining sub-class of hydrostatic bearings and the second general class of so-called "anti-friction" or rolling-element bearings are the important ones to be considered in the following discussion. The reader is referred to reference (1)* for a comprehensive discussion of bearing types and factors affecting selection and application. References (2) and (3) contain interesting chapters on hydrostatic bearing design. Reference (3) is primarily mathematical in treatment.

A further restriction will be made to the scope of this discussion by limiting an examination of bearing selection and application primarily to the major axes support. The problems of bearing selection for prime movers (motors) power drive gear boxes, control gear trains, etc., not peculiar to satellite trackers is well covered in the literature (see references (1) and (9) so we shall exclude these applications and confine our attention to the unique problems of axis bearing choice.

4-4.2 FACTORS AFFECTING BEARING SELECTION. Below are listed some of the important factors affecting bearing selection for tracker use, not necessarily in order of importance, nor independent.

1. Geometrical configuration, axis arrangement
2. Applied Loads, duty cycle, preload
3. Stiffness
4. Friction
5. Smoothness, of motion
6. Cost, Weight
7. Reliability
8. Environment, sealing
9. Lubrication
10. Life
11. Maintenance
12. Support structure

* Numbered references as they appear in this section apply to the Bibliography for section IV in the back of this report.

4-4.2.1 Geometric Configuration. In considering geometrical configuration, we can conclude that the actual number of axes, whether one, two, or three is relatively unimportant since they are truly independent of each other. However, the arrangement may have an effect; i.e., whether an azimuth-elevation or an equatorial arrangement is necessary. Factor (1) is also strongly influenced by factors (2) and (8). For instance, if the tracker is enclosed in a radome providing protection from wind, rain, sunlight, etc., no overturning moments need be resisted; sealing is minimal, and differential expansion of bearings and mount may be minimized. Thus, the azimuth axis in an altazimuth mount in a radome may be supported on a single thrust bearing whose primary load is the deadweight of the structure, with perhaps a small radial bearing for centering only. See figure 4-3. In the same radome, an equatorial mount will require a combined radial and thrust load on one bearing and at least a radial load in the other. See figure 4-4. Alternatively, a single bearing which can carry combined radial, thrust and overturning moment loads, such as a four-point contact (Gothic Arch) ball bearing or X-type roller bearing could be specified for either. See figures 4-5 and 4-6.

The presence or absence of slip rings on the azimuth axis also may affect the selection of axes bearings. If slip rings are required, the most compact arrangement with the best access to the slip rings, dictates a single bearing scheme, rather than two widely separated bearings.

4-4.2.2 Applied Loads. For factor (2), the load and duty cycle has little relative impact on the type of bearing per se, since any type can be designed for any requirements but these factors do affect overall weight, cost, reliability, etc.

For the selection of "standard" ball or roller bearings or specifying the requirements to a bearing vendor for a special application, usually the designer must specify not only the applied loads but the duration or the number of load applications as well.

As a minimum requirement, the bearing designer generally needs to know at least three ranges or magnitudes of load and their relative durations. Thus the highest load the equipment may ever experience under non-operating conditions (so called survival conditions) will set the static brinelling capacity of the bearing. The duration or the number of times of application of this kind of load (usually very small) also needs to be estimated. Secondly, a maximum operating or peak load with an estimate of the number of cycles or revolutions of the bearing required at this load is of interest. The third category is a load which represents a fair average for the major portion of the expected life of the bearing.

Since the major source of these loads is wind impinging on the reflector dish and other structures, the magnitudes may be calculated with considerable accuracy, but the durations and number of cycles experienced during the life of the equipment, can only be guessed at, with some help from statistical wind or weather data. The load calculations must include not only thrust and radial components for two-bearing systems but also "overturning" or rotation torques in addition, applicable to single-bearing Gothic-Arch" ball or X-type roller bearings.

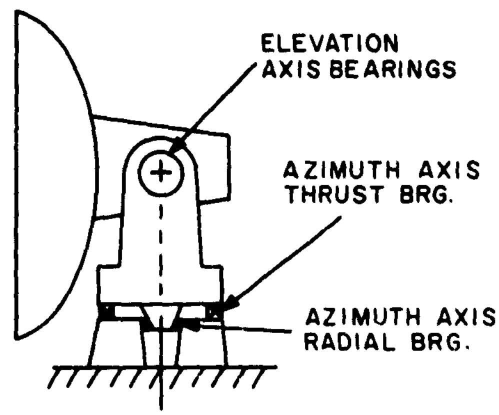


Figure 4-3. Alt-azimuth Mount.

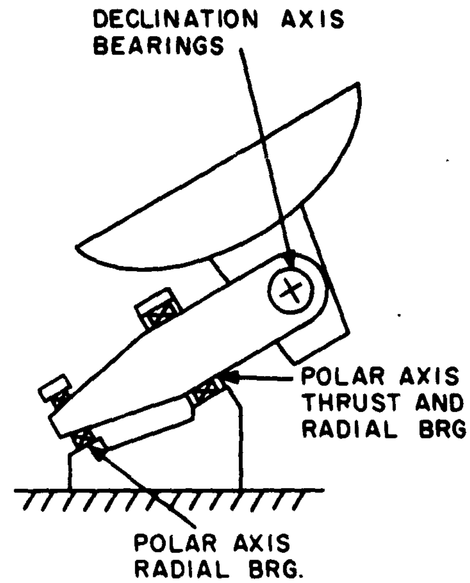


Figure 4-4. Equatorial Mount.

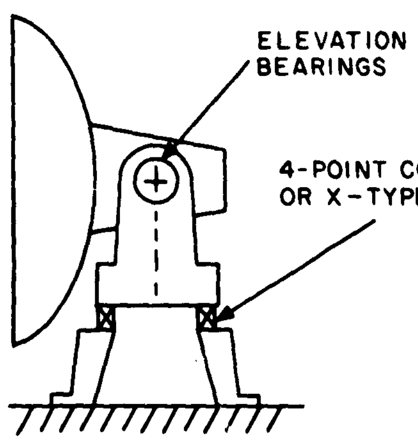


Figure 4-5. Single Bearing Mount.

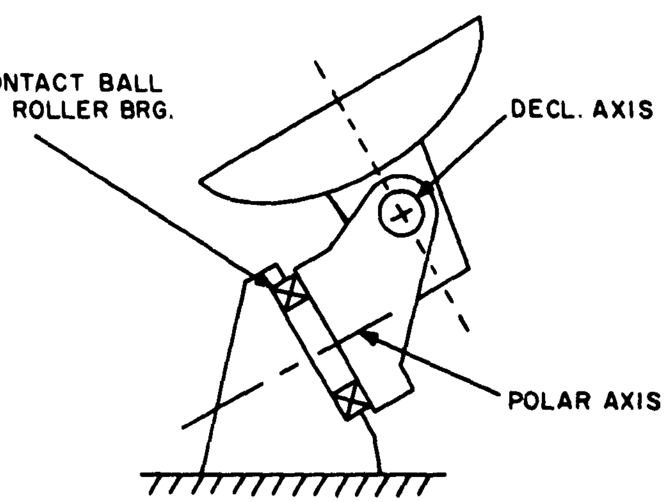


Figure 4-6. Single Bearing Mount.

For simplicity of estimation, the life of the bearing is frequently broken up into three fractions such as an average load for 90 percent of its life, a peak load for 9.9 percent of its life and a survival load for 0.1 percent. Another bearing vendor may favor 90 percent, nine percent and one percent breakdown of applied load duration or some other arbitrary scheme.

Preload is here defined as an internal or "built-in" load for the purpose of eliminating "shake" or looseness in a bearing, and for increasing its stiffness. Since deflection of a ball or roller in a race is not a linear function of load, but increases much slower with load as initial deformation develops, a considerable increase in stiffness can be attained by permitting small deformations to occur by deliberately loading the bearing elements at assembly.

Thus, a typical axial deflection curve for an angular contact ball bearing looks like figure 4-7.

By utilizing the upper portions of the curve away from the origin, a substantial reduction in deflection per unit load or an increase in unit stiffness is attained. Preload prevents operation near the origin where the spring rate is highest.

Preload is attained in double row bearings and duplex angular contact bearings by slight axial displacement of the internal race with respect to the external race. Four point contact bearings, which are in effect, two angular contact bearings have common balls and races, can be conveniently preloaded by a similar technique.

Since the preload, ordinarily, is only a small part of the working load deformation it does not increase the friction significantly. If, however, the preload is appreciable or comparable in magnitude to working loads, it can be the preponderant source of the bearing friction.

Preload in a hydrostatic system occurs inherently when a system for resisting axial load in either direction is specified, since the fluid film pressure on one side of a thrust plate is bucking or opposed to the load applied through the film pressure on the other side. Such systems need to be designed with care, since instability can result if common pumps or pressure systems are used.

4-4.2.3 Stiffness and Friction. Stiffness and friction need to be discussed jointly since they appear to be related in any type bearing and some sort of compromise has to be reached in specifying either.

Reference (7) succinctly sums up the dilemma by concluding that stiffness in a ball bearing is maximum with a large number of small balls and friction is minimum for a small number of large balls.

The four point ball bearing studied in reference (6) with a calculated overturning stiffness of 1.7×10^9 ft, lb/rad. has a measured coefficient of friction of 0.0035 under 35,000 lb. axial load, including shields and grease at room temperature. It should be understood that this friction was measured under pure thrust conditions. Under combined loads of thrust, radial loads and overturning moments, the friction would certainly increase.

By comparison the hydrostatic bearings can enjoy several orders of magnitude less friction. Since the only friction is that due to viscous shearing of the supporting film this can approach zero for zero relative motion but will, of course, increase with increasing speed. Since the axial stiffness of a hydrostatic thrust bearing with constant fluid flow varies inversely as the fourth power of film thickness (see reference 4) it can be appreciated that only a small reduction in film thickness is required for a large increase in stiffness.

Balancing this is the increase in cost due to the finer machining, more careful filtering, etc., required for the thinner film, and of course, an increase in friction.

4-42.4 <u>Theoretical Values.</u> Reference (8) shows a complete theoretical study of a hydrostatic bearing for possible application on the Atlas Tracker. Following are some pertinent numbers taken from this report, for the azimuth thrust bearing.

Mean radius	24 in.
No. pockets	20
Pocket depth	0.003 in.
Supply pressure	200 psi
Pocket pressure	48 psi
Capillary	0.0035 in x 1 in. long
Film thickness	0.0003 in.
Fluid flow	0.00025 GPM
Friction torque at 1 RPM	53 ft. lb.
Friction coefficient (1 rpm)	0.0013
Axial stiffness	1.2×10^8 lb./in.
Angular stiffness (overturning)	1.1×10^{10} in. lb./radian
Flatness tolerance for runner and thrust plate	plus or minus 0.00005

It is interesting to note that the coefficient of friction at 1 RPM is well within the range of what can be expected with a ball thrust bearing, but whereas a ball bearing will show considerable variation of friction due to the unpredictable variation of cage friction and ball position, the hydrostatic bearing friction is very uniform with a smooth reduction to zero at zero velocity.

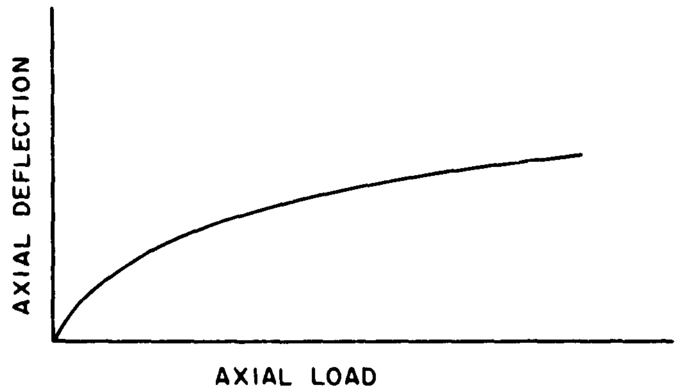


Figure 4-7. Axial Deflection Curve.

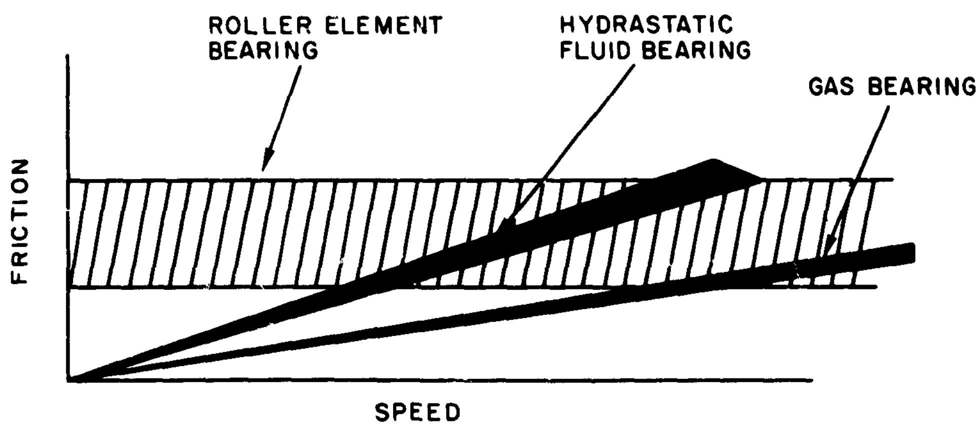


Figure 4-8. Speed and Friction Vs. Types of Bearings.

4-4.2.5 Smoothness of Motion. It is well to point out here that the smoothness and low friction of a fluid bearing system can be completely swamped by the friction and roughness of gear drives, particularly the type using opposed drives for the elimination of backlash in the gearing (see Section 6, Power Drive Design).

It is also only fair to point out that hydrostatic bearings may not save on overall power consumption due to low friction, since the external pressurization system may require considerable power. This is an important consideration in transportable tracker systems, since the power drain, bulk and weight of the pressurization system with its pump, motor, sump, filter, valves, piping, heat exchanger, etc., must inevitably be contended with.

A hydrostatic bearing using air for the fluid (so called gas bearing) can be considered since it offers the advantages of weight saving over oil systems and very much less friction at usable speeds than the oil bearing. It has the outstanding disadvantage of being much less stiff for the same film thickness. Conversely stated, a gas bearing with comparable stiffness would require a much thinner film, and therefore correspondingly finer machining and flatness limits (or higher cost).

Under factor (5), smoothness of motion, the hydrostatic bearings are far superior to any rolling-element bearing by an order of magnitude at least.

4-42.6 Cost and Weight. With regard to cost and weight, the roller-element bearings would seem to have the advantage of lower cost, principally because of its relative simplicity and secondly because the technology design and manufacture is better known at the present time.

Weight comparison is more difficult to assess since the amount of material attributable to each type for proper support and stiffness is somewhat difficult to define. If it can rather arbitrarily be assumed that back-up structure for the hydrostatic bearing is comparable in weight to the ball or roller paths and support structure of the roller-element bearing, then we are left with the weight of roller-elements and cage (or retainers) versus the weight of one or more motor-pump and auxiliaries for the hydrostatic system. Except for very large bearings for extremely heavy loads (20 feet in diameter and up) it would seem that the weight advantage is clearly with the conventional ball or roller bearing. To put some relevant numbers on this, the 62-inch diameter bearings studied in reference (6) has an estimated weight of 780 lbs. of which only 80 lbs. was attributable to ball and cage elements. Under the assumption made above, the equivalent hydrostatic bearing would have to limit the motors, pumps, heat exchangers, valves, sump, filters, piping, etc., to a total weight of 80 lbs., which is obviously difficult to do.

4-4.2.7 Reliability and Environment. In discussing relative reliability of the two types of bearing, we inevitably need to consider also the required life and required maintenance (factors (10) and (11)). A ball or roller bearing may have a definite limited life, based on applied load and number of cycles or revolutions, resulting in failure of the race

or rolling elements by "fatigue" or other failure mechanism. This may be arbitrarily set by design to be either comparable with the expected or useful life of the rest of the equipment (economical design) or may be extended out to a safe margin beyond any foreseeable use to be very conservative. The hydrostatic bearing, on the other hand, has no such life limit, since the only "wear" occurring is shear of an oil or gas film, which if properly cooled need not cause any destruction action. Of course, the life of the other components of a hydrostatic system may be limited and will need periodic replacement to keep the bearing system operating.

Based on the number of contributing elements which may fail, the roller-element bearing wins hands down on a reliability comparison. This is not to state that reliable hydrostatic bearings cannot be made, but the cost of maintenance will certainly be higher. There is a tremendous amount of statistical data available for life prediction of roller-element bearings as an entity, but very little data available for reliability experience on hydrostatic bearing systems. A ball or roller bearing can tolerate certain abuses or lack of maintenance for long periods, while failure of the oil supply may quickly ruin a hydrostatic type.

The factors of environment, sealing and lubrication need to be discussed together.

A ball or roller-element bearing requires a sealing element to retain necessary lubricant and exclude environmental contaminants such as rain water, dust, dirt, etc. It may be an integral part of the bearing race or may be an entirely separate element elsewhere in the configuration (see Paragraph 4-5., Seal Selection).

A hydrostatic bearing, by contrast, requires no "sealing" as such because oil or air under pressure is flowing outward from the bearing at all times. It is almost impossible to introduce contaminants, into the bearing system at this point, against the flow. However, the return flow (possibly by gravity) path, the sump or reservoir are fruitful sources for the entrance of contaminants and must be carefully sealed or alternatively rely heavily on a filter to remove ambient contaminants.

4-42.8 Lubrication. In considering lubrication, it scarcely needs to be pointed out that it is the heart of a hydrostatic system, and of perhaps only secondary importance in a roller-element bearing. It should be emphasized that lubrication of a ball or roller bearing is important to ease the inevitable sliding that must occur because of compressive deformation of the rolling elements, and the sliding that must occur between cage or retainer and the rolling elements. The lubrication systems of large roller-element bearings depends greatly on end use and environment. For heavy duty continuous rotation search antennas operating in an all-weather environment, the use of pressure fed oil systems, with provisions for heating and cooling the oil to suit ambient temperature conditions is common. On the other hand, a tracking radar with an intermittent duty cycle operating in a radome with considerably smaller temperature excursions can suffice with grease lubricated bearings with only occasional applications of grease gun for maintenance. While those extreme conditions are easy to prescribe for, the usual application will be intermediate and the

recommendations of the bearing manufacturers should be sought. It should be pointed out here that manufacturers of large roller-element bearings invariably expect their customers to provide the lubrication "hardware" whether it be continuous pressure fed oil systems or simple grease fittings.

4-4.2.9 Support Structure. Support structure considerations involve the "foundations" of the bearing. It goes without saying that the best design and execution of an antenna axis bearing can go for naught if the bearing has insufficient support. Perhaps most important is the stiffness of the support, which involves not only mere massiveness but also the degree of flatness for continuous support of thrust bearing races; degree of roundness for continuous backup of inner and outer races of ball or roller bearings. It is pertinent here to mention that all stiffness data provided by the bearing manufacturer assumes infinitely stiff foundation support both axially and radially.

A very common configuration for a four point contact ball bearing used for the azimuth axis, involves a cylindrical structure or "nest" for the bearing immediately over a tripod configuration of outrigger legs. Particular attention must be paid to the transition between tripod and cylindrical sections, to avoid deflection between the three "hard" points, which will not only decrease the apparent stiffness of the ball-bearing, but may also upset the bearing designers load and life calculations by concentrating the entire load on three small areas over the "hard" points.

A similar problem exists in hydrostatic bearings utilizing a multiplicity of pads on an annular runner. Both the runner support and transitions from pads to adjacent structure must be stiff enough to utilize the full inherent stiffness of a thin fluid film.

4-5 SEAL SELECTION

Protecting the bearing from moisture and contaminants, and retention of lubricants is the prime duty of bearing seals. Seals may be classified into at least three categories; (a) rubbing seals, (b) labyrinth seals, (c) trapped fluid seals. The first is exemplified by the "shaft packing" or stuffing box type, the molded elastomer lip type seal, and the high pressure face type seal. The second category includes all re-entrant angle baffles and close-clearance path devices. The third, which is practical for vertical shafts only, utilizes an annular U-shaped container partially filled with an inert fluid in which rotates a cylindrical member attached to the vertical shaft. This latter type can withstand differential pressures proportional to the depth of the annulus and the density of the fluid, with extremely low friction due to viscous shearing of the fluid film, and complete exclusion of dirt, moisture, etc.

4-5.1 RUBBING AND LABYRINTH SEALS. The use of rubbing seals in large diameters, which are really effective, exacts a high penalty in friction so the usual design compromise has to be made in limiting the rubbing pressure and thus the sealing efficacy. Low friction materials, such as Teflon will materially aid this situation. The second class of seals, using

a multiple plate labyrinth is commonly used. While this does not provide a positive seal against wind-blown dirt, it does act as a filter to screen out heavier particles, since each direction reversal in the labyrinth tends to drop more entrained dirt. The labyrinth is commonly used with a very light pressure seal or shield at the ball bearing; the labyrinth to exclude direct rain impingement, and most of wind blown sand or dirt, and the felt, or elastomer seal at the bearing to aid in lube retention and final dirt exclusion.

4-5.2 TRAPPED FLUID SEAL. The trapped fluid seal is particularly suitable for a large azimuth bearing which is for other reasons continuously lubricated with oil. Here the oil may provide the inert fluid seal, and surface contaminants filtered out. It should be born in mind that the provision for differential fluid level must accommodate the full dynamic head of incident winds, otherwise the fluid may be blown out of the "trap".

Reference (10) is a comprehensive discussion of modern seal practice.

4-6 MECHANICAL FASTENERS

Selection of the proper mechanical fasteners is especially important in the design of an antenna which must be both mobile and accurate. The mobility requirements implies rapid erection and compact storage. Rapid erection is best achieved if the number of component parts, and hence of connections, is small. However, compactness in storage is best achieved if the antenna disassembles into a large number of parts.

Connections can be made more rapidly if the fits of component parts are loose. However, loose fits reduce the accuracy of the assembly.

In selecting fasteners and designing joints, the designer must balance these conflicting demands. Industry provides a wide variety of types of fasteners for every conceivable purpose. A comprehensive listing of fastener manufacturers and their products is presented in "Hitchcock's Assembly and Fastener Directory" and in "The Fastener's Book", issue of Machine Design (21 March 1963).

4-6.1 FACTORS IN SELECTION. In a given application, the suitability of the fastener should be checked against the following requirements:

1. Strength. Obviously, the fastener must be able to carry the design load; most fasteners carry axial load only. Keys or pins may be required to carry loads which put the fastener in shear. A connection may be made more quickly by using a few bolts than by using a larger number of so-called quick-fasteners with less load carrying capacity.

2. Stiffness. The fastener must not deflect excessively under operating loads. Many quick-fasteners apply axial load to the joint through a spring the stiffness of which determines the stiffness of the fastener. Others use a cam which goes past dead center in order to lock, leaving a small amount of backlash. The presence of backlash or of inadequate stiffness in the fastener may permit undesirable distortion of the structure.

3. Fit. In general, connections can be made more quickly if fits are loose. Loose fits are less susceptible to wear through repeated use than tight ones. They are also less likely to jam when dirty. However, the accuracy of the assembly can be lost if the fits are too loose. Some quick-fasteners will not work at all without considerable clearance.

4. Captivation. All removable hardware should be captivated to make sure that the fastener is available at the right place when needed. The captivating device should not interfere with operation of the fastener; for instance, a cantilever spring clip captivates a quick-fastener, but in so doing, restricts the alignment of the fastener with its hole.

5. Grip Length. Each quick-fastener is applicable only over a short range of grip lengths. Because of this quick-fasteners are used mostly on sheet metal assemblies. In addition, some fasteners are sensitive to variations from the nominal grip length. If this variation is exceeded, the fasteners clamp loosely or not at all. Such fasteners are obviously not compatible with certain types of construction or with certain manufacturing processes.

6. Preload. Any fastener should be preloaded to a value which is approximately equal to the peak service load. This practice prevents separation of the joint under load and the loss in accuracy associated with separation. It also prevents fatigue failure of the fastener. In some patented fasteners, the axial force developed is a function only of the grip length, which is affected by variations in material thickness and by manufacturing tolerances. Therefore, preload can be only loosely controlled.

Bolts are excellent fasteners for preloading. However, accurate measurement of the tension in bolts is not feasible during field erection under a time limit. The use of oversize bolts and wrenches of proper length will insure that bolts will get the required preload as a minimum.

The contact stresses under the bolt head and nut should be designed to prevent creep under prolonged load. If the flange material flows, or if the hardware embeds in the flange, the result will be a loss of preload. Long bolts will lose a relatively smaller amount of preload through a given amount of creep than will short bolts under equal axial load.

7. Assembly time. The most likely reason for using fasteners other than bolts is to speed up assembly. In estimating the relative merit of fasteners with regard to assembly time, the comparison should be made on the basis of total time to make a connection rather than on the basis of individual fastener time. The time saving should be weighed against total system erection time for accurate evaluation.

4-6.2 BOLTS. The overall advantages of bolts as fasteners seem to warrant the payment of some time penalty in order to use them. In fact, if proper tools, (ratchet wrenches, for instance,) are provided the workmen, bolted connections can probably be made as rapidly as any other. The principal advantages of bolts as fasteners lie in the areas of cost, strength, stiffness, the capacity to preload, and effectiveness over a wide grip range.

4-6.3 RIVETS. Rivets, of course, are permanent fasteners and hence are not in the same category as the field connectors. They may be used in a reflector structure. Each rivet application should be examined to determine whether the joint will slip under load and, if so, whether the resulting deflection is tolerable.

In a long, riveted, reflector spar, the deflection due to rivet slip may be an appreciable fraction of the contour tolerance.

4-6.4 LOCATING DEVICES. At most joints, the fastener alone will probably not provide adequate shear strength or sufficiently accurate alignment. The holding power of the primary fastener will have to be supplemented by other means, such as pins, rabbets, or keys. To permit rapid erection, even these devices will have to have some looseness. Since any looseness will permit some misalignment of the structure and result in a loss of pointing accuracy, the amount of clearance provided will have to be a compromise between that required to maintain accuracy and that required to permit rapid assembly.

4-6.5 HINGES. Antenna erection can be simplified by the use of hinges to join component parts. The hinge restrains and aligns the parts relative to one another throughout the assembly process.

The utility of hinges in a transportable antenna is limited by the fact that the stowed position of a component is fixed by the hinge pin. This stowed position may not be compatible with the most compact storage arrangement.

The hinge should not be considered a rigid connector. Hinge pins should be clamped after erection of the antenna to eliminate looseness at the joint.

4-6.6 CLEARANCES. The need for clearance has been mentioned in the discussion of some types of fasteners. Looseness of fit reduces wear during assembly and thereby helps to prevent a degradation of antenna accuracy through use. However, if not controlled, it can limit antenna accuracy. In order to determine whether or not clearance is undesirable, each joint should be studied individually. Factors which should be considered include the geometry of the structure, the position of the joint in the structure, the magnitude of the load at the joint, and the direction of the load with respect to the clearance and the effects of changes in direction of load. Wherever it can be shown that clearance will not permit excessive misalignment of the structure, it should be permitted.

Section V
TRANSPORTATION

5-1 AVAILABLE VEHICLES

The selection of a military vehicle to transport a particular piece of military equipment, be it radar or armament, is based on two broad considerations. The first is the equipment itself, with thorough regard for its proposed usage factors. The second, naturally, is the selection of one of several different types of land vehicles in use by the modern military organization.

In order to establish a background of future reference when considering the peculiarities of satellite tracking antennas, it is well to first review some of the more or less obvious, but important properties of different type vehicles.

5-1.1 CONVENTIONAL VEHICLES. Conventional wheeled vehicles such as pneumatic tired cargo trucks, trailers and tractor-semi trailer combinations are used principally by the military for ground transport mainly because of the following favorable characteristics; typical, for example, of the M-35 truck:

1. Speed. The M-35 cargo truck (2-1/2 ton capacity) is governed to a maximum speed of 58 MPH, which enables it to take advantage of modern express highways for long distance transport.

2. Cruising range. The M-35 cargo truck has a rated fuel consumption (loaded) of 6 MPG, which is consistant with commercial truck performance. The 50-gallon tank gives it a theoretical cruising range (between refuelings) of 300 miles.

3. Ride characteristics. Although hardly comparable to the soft ride of the modern automobile, the M-35 cargo truck does produce a relatively vibration-free ride on smooth roads, and is provided with some degree of shock isolation on rough roads so that long distance over-the-road hauling is within the realm of human endurance. The same ride characteristics are also true of wheeled cargo trailers with leaf-spring suspension. Suspension trade-offs can be made to improve riding qualities at the expense of the limited cross-country abilities. Replacement of the standard military lug tires with commercial highway tires will eliminate a high frequency vibration during transport over paved roads, but will sacrifice the improved traction of lug tires in mud and snow. Trailers, being towed, would not need lug tires. Modern suspension refinements such as torsion bars and air springs soften the ride to an even greater degree.

4. Cross-country capabilities. Capabilities of a truck-trailer or semi-trailer combination are severely restricted by the lack of power on the trailer wheels. This shortcoming excludes the use of trailers on soft ground and on steep slopes.

5-1.2 THE "GOER". A recent development in the field of unconventional wheeled vehicles is the GOER (XM437, 16-ton cargo truck). The GOER is a four-wheeled vehicle composed of a two-wheel semi-trailer attached (more or less permanently) to a two-wheeled tractor. In addition to this unorthodox arrangement, the GOER is an outsized vehicle (123 inches wide x 491 inches long) with five-foot diameter wheels. The GOER is still in the development stage, but the test vehicles have shown a marked cross-country mobility differential over conventional wheeled vehicles.

1. Speed. The GOER is engine governed at 31 MPH. Although this is significantly slower than conventional wheeled vehicles, it is important to note that this speed can be maintained over moderate cross-country terrain while carrying a full load.

2. Cruising range. Approximately 300 miles.

3. Ride characteristics. Ride characteristics on cross-country terrain are superior to conventional wheeled vehicles due to the large diameter, relatively soft tires. However, due to the absence of a spring-shock absorber suspension, the vehicle has a characteristic, low frequency resonant bounce at higher speeds. Over rough ground, the maximum vertical acceleration is approximately 4G. The GOER is able to swim inland waters, climb 60 percent slopes and negotiate mud, snow and sand due in large measure to the fact that the trailer wheels are powered at low transmission ratios.

The GOER is an illegal vehicle out of convoy on a state road, since it is over two feet wider than the usual limit of eight feet and exceeds the usual 18,000-pound axle load limits (front axle = 25,670 pounds with no cargo). It would be virtually impossible to thread a GOER through the narrow-sharp cornered streets of many European small towns, although this would not be a requirement in the case of GOER transported satellite tracking antennas (for reasons to be discussed later).

4. Air transportability. The GOER will fit into the C-133 cargo aircraft; however, the high front axle load (25,000 pounds empty) exceeds single axle load limits (20,000 pounds) as specified in aircraft loading instructions. The C-133 will operate at reduced range for loads over 42,000 pounds, up to a maximum of 100,000 pounds, so that a fully loaded GOER of 70,000 pounds is still air transportable at some reduced range. The C-133 has an approximate 12-foot overhead clearance for loads of GOER width. Unless the load overhang is excessive, it is conceivable that a 12-foot high limit can be approached. Since GOER has a 3-foot load platform, a load of almost nine feet high may be air transported.

5-1.3 TRACKED VEHICLES. Tracked vehicles offer a maximum degree of mobility over severe cross-country terrain, although the following performance characteristics indicate the prices to be paid for this mobility:

1. Speed. Tracked vehicles are generally limited to a speed of 30 MPH, which can be maintained over moderate cross-country conditions.

2. Cruising range. Approximately 150 miles.

3. Ride characteristics. In addition to the radom shock and vibration incurred by movement over rough and uneven ground, there are additional steady state vibrations which are peculiar to and quite significant on tracked vehicles. Most significant is that vibration caused by the movement of the track segments over the wheels of the tank. The frequency is obviously a function of road speed and number of segments in the track with the result that the designer of tracked vehicle mounted electronic equipment must provide wide range vibration isolation for such equipment.

4. Load factor. Since the left and right hand sprung members (track bogies)of a tracked vehicle suspension are not interconnected, the space between them (which is usually used for axle clearance) can be utilized as useful cargo space. This generally results in lower silhouette than for wheeled vehicles. However, where the cargo cannot be accommodated in the space between the treads, the load platform will be about the same height as that of a cargo truck. Tracked vehicles are generally short vehciles and therefore not suitable for carrying long loads, since over-handing loads reduce the large angles of attack and departure so essential in traversing extreme cross-country terrain.

5-2 VEHICLE SELECTION

Before an attempt can be made to select a specific vehicle for transportation of an antenna, it is important that the application of the antenna be explored, to determine the following fundamental parameters:

1. Characteristics of antenna site

2. Type of support equipment

3. Relocation distance

4. Domestic or world wide use

Items (1) and (2) determine the degree of mobility of the transporting vehicle. Obviously, the more time that is available to traverse cross-country terrain and the frequency with which it has to be done, will determine the degree of route preparation that will be practical.

In the case of a fire control radar system, tactical mobility must be maintained. In an extremely fluid engagement, this would mean relocation on short notice, with not time or facility for route preparation. Such equipment is usually mounted on tracked vehicles which not only provide a high degree of mobility over undeveloped adverse terrain, but also afford protection of operating personnel from small arms fire.

In the case of a long range, high powered, omni-directional communication antenna array, site relocation is infrequent and is usually preceded by the establishment of a network

of roadways leading to the antenna site. Consequently, ordinary cargo trucks would suffice for antenna transport in this case.

A satellite tracking communication antenna presents the following unique requirements:

5-2.1 SITE REQUIREMENTS.

1. Probably the most important site requirement is the need to be located on high ground in order to view a maximum horizon angle. This does not necessarily mean location on the very peak of a mountain, but it does mean getting up out of the easily traveled valleys and on to somewhat inaccessible hillsides.

2. A second site requirement for a satellite tracking antenna is the need for relatively stable and level ground. This need can be understood when it is realized that a satellite tracking antenna is a precision antenna, and must remain so for longer periods of time than, for example, a tactical fire control radar antenna.

3. The site must also be relatively free of man-made electrical interference such as generated in towns and cities or in proximity to hostile territory be deliberate electronic jamming. In addition, it should be realized that the satellite communication system is the big link in a communication chain and as such is a rear echelon facility that should not be subjected to the hazards of small arms fire, and therefore not requiring the protection of an armored vehicle.

In addition to the aforementioned site parameters, the following communication system considerations are of interest in determining mobility requirements for the antenna.

4. In addition to the antenna, the satellite communication system consists of several support functions, including power generating equipment, control equipment with associated shelters for operating personnel; secondary communication relay systems (land lines, lower power radio links) and other secondary support functions. These support functions are presently transported as vans on wheeled trucks or trailers. It is consistent, then, that the antenna also be transported as a wheeled vehicle.

5. Being a long distance communication link, the satellite communication system when reloacted, will be transported long distances. A wheeled vehicle, being more efficient than a tracked vehicle due to the higher ratio of dead weight to live load, is therefore more desirable for long distance transport. This factor is particularly important when air transport is contemplated.

5-2.2 MOBILITY. Extreme mobility is not an essential requirement for the transportation of a satellite communication antenna, since it is essentially a rear echelon equipment, and as such a part of a more or less fixed facility. When it is relocated, it will be a long distance move made over routes previously developed and traveled. Accordingly, the

transporting vehicles should have a long cruising range and capable of high speed if good roads are available. Extreme cross-country capability is not required, so that tracked vehicles are immediately ruled out. This leaves some fore of wheeled vehicle as the transporting medium. The exact choice depends on the form that the antenna takes when broken down for transport. If the antenna can be broken down into sections no more than 147 inches long, nor more than 88 inches wide, or 93 inches high, it can be loaded into the standard 2 1/2 ton M-35 cargo truck, which in turn can be air-transported in the loaded condition. This can be done for antenna design A so that all of the antenna is moved by two trucks with all the previously mentioned advantages of using the M-35 rather than a more exotic or specialized vehicle.

Design B cannot be broken down sufficiently, requiring the retracted pedestal to be carried some other way. Flat bed, wheeled trailers offer a maximum accommodation for large bulky cargo, while at the same time maintaining many of the advantages of truck transport. A special, low platform, flat bed semi-trailer is suggested for design B. This trailer serves a dual purpose in being utilized also as an erection base for the antenna. Trailers in general have limited cross-country capabilities, due to the lack of driving power at their wheels. A possible remedy for this situation would be the use of powered wheels (wheels equipped with individual, self-contained electric drives) which may be energized for low speed high traction situations. Power is easily transferred through flexible electric cables from a generator on the prime mover to the motors in the trailer wheels. Such a scheme is used on GOER, to provide a 4 x 4 wheeled vehicle, without the problems of a mechanical power transmission through the fifth wheel trailer connection. It is obvious that low bed trailers are susceptible to "hanging up" on high points of uneven ground. This problem has been alleviated on missile trailer design without sacrificing useful load height limitations by using a variable height suspension (trailing arms with air springs). Thus, in the field, the trailer bed is elevated for maximum ground clearance, while for situations of minimum overhead clearance the bed can be depressed.

A 30-foot diameter antenna with a five-foot ground clearance and six-foot diameter reflector mounting pad will require approximately 23 feet of vehicle load length to transport the pedestal intact in a prone position. A vehicle such as the M-172, 25-ton low bed semi-trailer would be suitable for this application, although its 115-inch width makes it an illegal vehicle. Similar sized commercial trailers are available in the usual legal width of 96 inches.

Section VI
POWER DRIVE DESIGN
SPEED REDUCERS FOR ANTENNA APPLICATION

6-1 INTRODUCTION

The fundamental purpose of a "speed reducer" is to match the speed and torque characteristic of a prime mover to a load. Second in importance for consideration are factors such as whether the speed reducer is a part of a servo loop, where backlash, stiffness and reflected inertia are very important; or whether, for example, it is a straight power drive. Thirdly, there are general factors concerned with geometrical configuration, cost, weight, efficiency, life, ease of maintenance, availability, etc., where the order of importance varies markedly with the application.

In considering various classes or generic types of speed reducers, belt and friction drives of all kinds and chain drives will immediately be excluded from further consideration. Friction devices of all types are inappropriate for precision servo drives. Chain drives, with their non-uniform motion transmission, and high compliance (low stiffness) are unsuitable except for very special circumstances of application near the high speed or motor end when the general configuration requires remote (relatively) mounting of the motor.

The remaining class of positive drive gear reducers encompasses a wide range of involute spur gear reducers, in several sub-classes of arrangements, such as straight series or cascaded spur gear trains, planetary spur gear trains and worm gear reducers with sub-classes of single enveloping or double enveloping ("Cone-Drive") worms. Also to be included are gearing variants such as spiral, hypoid, "harmonic" drives, etc., or any combination of these.

The most fundamental attribute of the speed reducer is the gear ratio. This is wholly determined by the ratio of speeds of the prime mover and the drive load. Thus, a relatively high speed prime mover such as a 400-cycle electric motor will require a large speed ratio to couple the motor to a low speed load, such as a satellite tracker. In this particular instance for example, a 10,000 rpm motor might be coupled to an antenna with a top speed of five rpm for an overall gear ratio of 2000.

At the other end of the scale, a hydraulic motor with a peak speed of 500 rpm might be selected to drive an antenna with a ten rpm maximum slewing speed, thus requiring a speed ratio of fifty.

These two extreme cases, in general, might require totally different approaches for the selection of an optimum speed reducer.

With the necessary gear ratio determined, the problem of optimizing the gear set necessary to accomplish the objective is addressed.

Intuitively, it would seem that the fewer the gears required, the better. Indeed, the number of meshes or mated pairs of gears should be minimized for lowest weight, highest reliability, lowest cost, etc.

But this simple objective is difficult to realize with high ratios. Involute spur gears require at least ten to twenty teeth on the pinion for optimum combination of tooth load, wear, tool cost, etc., and the mating spur gear becomes large and relatively unwieldy in the area over 200 to 300 teeth. In fact, a spur ratio above ten requires special design considerations which makes ratios above ten relatively non-standard.

Worm gearing fits nicely into the picture for ratios from ten to 100 teeth, but suffers in overall efficiency compared with a single spur mesh. Typically, a well-lubricated spur meshes with smooth tooth surfaces at moderate speeds to enjoy efficiencies of 98 percent and above. Worm gearing under like conditions may reach 90 percent, but only at reasonably high worm speeds where full hydro dynamic lubrication conditions apply.

Worm gear ratios either above or below this approximate rough range suffer a marked drop in efficiency; to below 50 percent for high ratios and slow speed.

Thus, there are sound reasons for specifying multiple mesh gear systems. To cite an extreme case, consider a set of reduction gears for a turbo-prop aircraft engine, where turbines at 15,000 rpm are driving a propeller at 1000 rpm. Here the utmost in reliability is required and the ten percent loss associated with worm gearing would be intolerable. Furthermore, the 90-degree relation of propeller axis and turbine would be very awkward. A planetary set of spur gears as a reducer between the turbine and propeller is the choice here because of the co-axial coupling, high efficiency and relatively light weight. In this case, the multiple meshes contribute as follows: the three planetary pinions share the driving load and therefore can be made lighter than a single pinion and the six meshes included will permit an overall efficiency of approximately 0.99^6 (Note: 0.99^6 = 0.99 to the sixth power) X 100 = 95 percent to be attained. Also, the favorable disc configuration and co-axial coupling optimize the geometry of turbine and propeller.

6-2 GEOMETRICAL CONSIDERATIONS

The designer is frequently required to shape the gear train to suit some external condition (such as clearance) to other components, maximum accessibility, etc.

Starting with a cascaded spur train, for instance, with all gear shafts parallel, it may become necessary to locate the motor with its axis perpendicular to the output shaft. Such a case arises frequently in an azimuth drive for an antenna where it is desirable to avoid a vertical position of a motor to minimize an oil sealing problem in the motor bearings. Here, a pair of bevel gears, either straight or spiral may be used. These will yield the

highest efficiency and permit the motor axis to intersect the mating gear shaft axis. On the other hand, if a moderately high ratio reduction is to be accomplished in this mesh, hypoid or non-intersecting axis spiroid gears should be considered. Lastly, a worm gear mesh may prove attractive if a sufficiently high ratio can be taken and the non-intersecting shaft configuration is useful.

The bearing arrangement of the output shaft also contributes to the geometrical aspects of reducer gear train selection. The currently popular single large ball or roller bearing used on a vertical azimuth axis is entirely suitable for employment of a spur bull-gear cut as an integral part of the bearing race. A worm gear configuration here would almost certainly be of too high a ratio, with very low efficiency and irreversibility, both of which are undesirable in a servo drive application.

The older type of two-bearing supported train axis which permits smaller bearings and a smaller worm gear might be considered if this configuration could be justified for some other supporting reason.

6-3 SERVO CONSIDERATIONS

For single-ended systems where backlash is an important consideration, it is desirable to employ as large a ratio as possible at the final mesh between the gear driving the load (bull-gear) and its pinion to place the attendant backlash at the maximum radius possible, to minimize the backlash angle. Here, a spur gear mesh has the advantage at the expense of heavier weight. The reason is that for a given ratio, a worm gear pair has a smaller center distance, hence lighter overall weight and higher backlash angle. The backlash angle for the worm gear is also increased because the backlash angle is a function of the product of pressure angle times helix angle, whereas for the straight spur gear, the backlash angle involves only the pressure angle.

It will be apparent that the bull-gear mesh is the most important in respect to backlash when it is realized that subsequent meshes up to the motor contribute to the overall backlash only to the extent of the mesh lash divided by the relative speed of the pinion of that mesh. Thus, if a pinion on a motor shaft has a speed relative to the final drive of 200, the backlash of X thousandths actually adds only $\frac{1X}{200}$ or one half of one percent of X to the total backlash.

Likewise, in multiple mesh configurations, the twist, or wind-up of shafts is usually critical only in the first pair, since wind-up of subsequent shafts is reduced by the speed ratio of the respective shafts.

The gear ratio has an even more powerful effect on the derived properties of gear train spring constants and gear or motor inertias referred to the slow output, or one-speed (relative) shaft.

Consider the calculation of the overall torsional spring constant of the output shaft defined as torque per unit angle of twist, with the high speed or motor end held fast.

Thus, let K_L be defined as the desired spring constant measured at the low speed shaft which is measured by applying a torque T_L and measuring the resultant total angle θ_L with the high speed or motor end held fast. If this same torque is measured at any other point in the gear train, back at the motor for example; the torque is reduced by the gear ratio to that point, or $\frac{T_L}{R} = T_M$. Likewise, the angular motion θ_L at the output, is increased by the gear ratio at the high speed end, or, $\theta_L n = \theta_M$. If the spring constant is $K_L = \frac{T_L}{\theta_L}$ at the output, then the same spring constant referred to the high speed or motor end is $K_M = \frac{T_L}{\theta_L n^2}$.

Usually, the practical problem is to calculate the overall stiffness of a gear train referred to the output, or slow shaft. Here, the individual spring constants of each shaft, gear tooth, etc., are calculated, multiplied by the square of the gear ratio at that point in the train and reciprocally summed to obtain the overall stiffness. For most conventional spur gear trains at least one half of the total windup, or compliance, occurs in the first, or bull-gear mesh.

In a similar fashion, the square of the gear ratio appears in the computation of rotational inertias of elements in the gear train. If it is desired to compute the inertia of the high speed motor rotor and all the elements of the gear train at their several rotational velocities, in terms of the output shaft, the products of each inertia multiplied by its speed ratio squared is added. Because of this big multiplier, the reflected inertia of a small high speed motor can assume a magnitude comparable with the external load inertia, as "seen" from the output end.

Since the gear train is one of the most important links between the servo-motor and its load, the gear train stiffness has an important effect on the overall servo performance. In examining sources of deflection with the objective of increasing the designed stiffness of the gear train, the designer must consider the following:

1. Tooth bending
2. Tooth shear deflection
3. Tooth compressive deformation
4. Shaft bending
5. Shaft torsion
6. Bearing deflection
7. Housing or case deflections

Actually a dimensional analysis shows that in items (1) and (2) the deflection is independent of tooth size or tooth pitch. Item (3) varies as the logarithm of the square root of tooth radius, which means that it changes only very slowly as tooth size or gear diameter

is changed. As a matter of interest, a hundred-fold change in tooth radius will change the tooth deformation by less than 20 percent.

In any case, with regard to tooth size, the designer can only reduce the deflection by reducing the intensity of loading, either by enlarging the gear pair or increasing the face width, both of which will increase weight and cost.

Items (4) and (5) are important components of deflection, particularly in the first or low speed pair. The avoidance of overhung pinions should be stressed since a beam simply supported at both ends has one-half the deflection of a cantilever beam of equal section and one-half the length of the former.

The use of preloaded bearings in the first pair of shafts is indicated to maximize the stiffness by these components, and attention must be paid to the housing, (case supporting the gear train) to provide a stiffness commensurate with the gears and shafts.

In general, it can be said that multiple mesh gear reducers of any combination can be designed with comparable stiffness and comparable weight. An outstanding exception to this is the so-called Harmonic Drive, a proprietary product of United Shoe Machine Corporation. This device uses a thin-section ring which must be appreciably deflected, and while it can transmit sizable loads in a very compact package it is by its nature compliant and cannot be made as stiff as a conventional gear reducer without sacrificing either life or weight advantage, or both.

Certain spur train combinations are of particular interest in servo applications for the reduction of the effects of backlash or for increased stiffness which may be realized. Those are the so-called "locked" gear trains and "bucking" or opposed dual-drives.

Figure 6-1 shows a simple "locked" train where an internal load on the teeth can be created by relative rotation of one drive pinion with respect to the other. By suitable adjustment of the coupling, any desirable degree of twist or torque can be applied. This has the effect of eliminating all backlash with any desired degree of preload with a consequent increase in stiffness. This device is commonly used where it is desired to "run-in" a set of gears under load. In servo applications it is common to preload the system to ten percent of full output torque, which is a constant frictional load "seen" by the motor in addition to any varying external load. Upon reversal of the motor deviation, however, there is no backlash deadband and considerable reduction in shaft windup.

Figure 6-2 shows a simple opposed dual or bucking drive which accomplishes essentially the same objective as shown in figure 6-1, in a somewhat more flexible manner. Here the motor torques are biased in opposing directions so that in effect one motor drives the external load plus a small fraction of this load applied in an opposing direction from the other motor. The flexibility arises from the ability to shift the bias electrically from a control console, to suit the required performance condition.

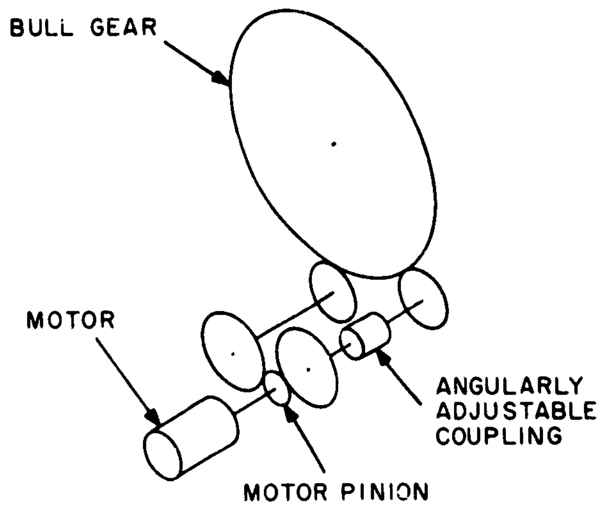


Figure 6-1. Simple Locked Gear Train.

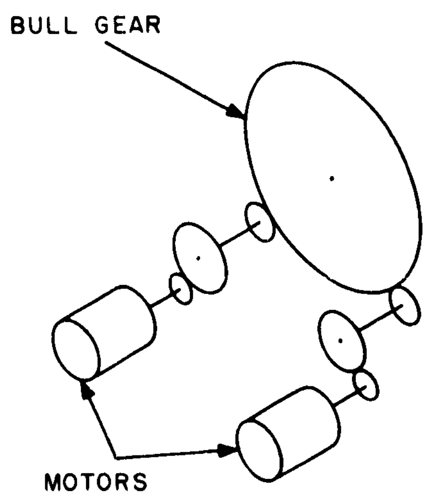


Figure 6-2. Bucking Drive Gear Train.

Both of these devices, effectively increase the apparent stiffness of an equivalent single-ended system, at a price of increased cost and weight, with the locked train being somewhat lighter and cheaper than the bucking drive.

They both, however, have the disadvantage of constant appreciable friction load which may result in difficulties with stiction or "slip-stick" phenomena at very slow speeds. This disadvantage may lead to "ratchety" action at very slow speeds due to higher friction.

6-4 SHAFT COUPLINGS

Important (but often overlooked) in the chain of components between a servo-motor and its external load are the shaft couplings. There is usually at least one in a gear train, between the drive motor and high speed pinion. Unless the external bull-gear is welded to its driving structure (unusual) there also has to be a coupling element joining the bull-gear to its load.

Usually in a precision servo application, the reduction of backlash and increase in stiffness are overriding requirements. These considerations are particularly demanding at the low speed end of the gear train. It is because of these requirements that the use of single keys for fastening a hub to a shaft is not good practice. It is almost impossible to manufacture and assemble a key tight enough to prevent loosening in service under reversing load conditions. A multiple-tooth integrally cut spline is the choice for preventing relative rotation of a shaft and hub. The involute spline is the easiest shape to produce accurately and is coming into wide use. The American Standards Association (ASA) has worked out elaborate and detailed tables showing all pertinent dimensions for a wide range of sizes and pitches. Of the three basic types of fits, major diameter, minor diameter and side bearing, the side bearing is recommended for better control of backlash. For details see ASA B5.15 published by the American Society of Mechanical Engineers (ASME) 29 West 39th Street, New York 18, New York.

A rigid splined sleeve is sometimes used for direct coupling of the shaft of a flange-mounted motor to a gear box shaft. This is possible if the concentricities of the motor and gear shaft and squareness of the mounting flanges can be held to very close limits.

More commonly, the motor is foot-mounted and some sort of flexible coupling must be provided to permit some non-concentric and non-parallel error to exist. There are a great number of basic designs suitable for this application of which two will be mentioned. The so-called gear type coupling in which two external spur gears are splined to the respective shafts and driven by an internal geared sleeve is ideal for ruggedness where some backlash can be tolerated. Due to a slight spherical shape or "crown" of each external tooth, considerable non-parallel non-concentric misalignment can be tolerated.

The second type of coupling is ideal where no backlash can be accepted. This is the so-called disc-type or Thomas coupling in which a stack of flexible discs are pinned to a fork splined to each shaft. The planes of the two forks are mutually perpendicular so that misalignment is accomodated by bending of the thin discs. Actually, the Thomas coupling

is a variant of the classical Hooke's, or universal joint, with the pin-joints replaced by elastic elements to eliminate shake or backlash. This coupling can be designed to be extremely stiff since the principal deformation is that due to torsion in the plane of the discs. Bending of the discs due to shaft misalignment, of course, distorts the plane of the discs and gives rise to secondary deflection, and thus the amount of misalignment is limited by stiffness considerations.

Unfortunately, splines are expensive to manufacture accurately and gears splined to shafts are greatly increased in cost because the work-holding fixtures used in making the gears need the complementary spline element to properly locate the gear, particularly for the recommended side bearing fit.

Wherever possible, gears in a train should be cut as integral pairs to avoid the need for splines to couple them together.

Bull gears are commonly attached to their load-structure by body-bound bolts and/or dowel pins placed at as large a radius as possible to reduce stress and effects of possible loosening. Centering, of course, must be provided by a close shaft fit or rabbeting into a turned fit on the structure.

TABLE 6-1. AMERICAN GEAR MANUFACTURERS ASSOCIATION
QUALITY NUMBERS (EXCERPTS FROM AGMA GEAR CLASSIFICATION MANUAL 390.01 MAY 1961)

AGMA Qual. No.	Diametral Pitch	Pitch Line Runout Tolerance "tenths" Pitch Diameter					Pitch Tolerance "tenths" Pitch Diameter					Profile Tolerance "tenths" Pitch Diameter					Lead Tolerance "tenths" Face Width			
		6	12	25	50	100	6	12	25	50	100	6	12	25	50	100	1	2	3	5
	1		95	115	140	180		16	18	19	22		70	75	78	81				
8	2	58	68	82	100	125	14	14	15	17	19	46	98	50	52	55	7	12	19	24
	3	41	47	56	67	83	11	12	15	15	16	32	34	35	37	39				
	1		50	58	70	87		8.25	8.75	9.75	11		36	38	40	42				
10	2	30	34	40	50	63	7	7.25	8	8.75	10	23	24	25	26	28	5	8	11	16
	4	18	21	24	28	34	5.75	6.25	7	7.5	8.5	16	17	17.5	18	19				
	4	11	12	14	17	21	3.0	3.25	3.5	3.75	4.5	8.25	8.5	9	9.5	10				
12	8	8	9	11	13	15	2.75	3	3.25	3.5	3.75	6	6.25	6.5	6.75	7	3	5	7	10
	16	7	8	8	11	11	2.25	2.5	2.5	2.75	3.25	4.5	4.5	5	5.25	5.5				
	4	5.5	6.25	7.5	9	13	1.5	1.5	1.75	2.0	2.25	4.25	4.25	9.5	9.75	5				
14	8	5	5	5.5	6.5	7.75	1.5	1.5	1.5	1.75	2.0	3	3.25	3.25	3.5	3.75	2	3	3.5	5
	16	5	5	5	5.75	6.75	1.5	1.5	1.5	1.5	1.75	3	3	3	3	3				

Table 6-1 is appended to show something of the state-of-the-art in precision gears. The AGMA has established quality number from 3 to 15 inclusive to cover commercial ranges from very inexpensive, low accuracy gears to the very highest accuracy attainable today. We have chosen to show some of this data for selected numbers near the top of the range which might be of interest to the antenna drive designer. It must be understood that gears of this class are not available off-the-shelf and understandings must be reached with the gear vendor concerning materials, heat treatments, methods of inspection for the required accuracy, etc. It should also be understood that the cost rises with quality numbers, and that some of the standards may not be available in some pitch diameters anywhere at present time. Thus a hardened gear of 100" pitch dia. may not be ground to a quality 12 or 14 grade if no machines are abailable with this capacity.

Section VII
DATA UNIT SURVEY

7-1 THE BASIC PROBLEM IN SELECTING A DATA UNIT

The successful operation of any antenna tracking system depends, among other factors, upon the accurate measurement of the azimuth and elevation angles of the tracker. This measurement must provide electrical signals which can be used in conjunction with computers and other equipment to control the "track" of the tracker.

In the type of applications being considered here, the selection of the data unit is limited by required accuracy of the overall system. To design a tracker having overall accuracy in the order of ±0.05 milliradians requires measurement in the same or more precise order of magnitude. An error of 0.05 milliradians is approximately equal to ten seconds of arc; therefore, the prime concern is to investigate techniques for measurement to an accuracy of ten seconds or less.

Until recently antenna designs incorporated potentiometer and synchro devices for the measurement of angles. However, the use of potentiometers and synchros on trackers has three great disadvantages in so far as space position determination is concerned. First, such analog devices are limited in accuracy and resolution. Second, analog data is difficult to transmit or record without degradation of accuracy. Finally, analog data is not of a form compatible with digital computation requirements. Analog to digital converters are now available which convert analog data such as that provided by potentiometers and synchros to digital computers. However, such conversion devices do not alleviate the inherent accuracy limitations of analog take-offs and provide an additional data processing link with resulting equipment complexity.

These inherent limitations of analog take-off devices have led to the development and subsequent incorporation in tracking equipments of digital take-off devices which permit direct angle to digital conversion.*

The use of analog systems in trackers results in accuracies in the order of minutes. Present day applications require measurement in the order of seconds with perhaps some application aiming for measurement accuracy as low as one second of arc. Such accuracies can be achieved, or at least approached, with the application of digital angle take-off devices and digital converter data processing techniques.

The digital angle take-off devices, commonly referred to as "shaft angle encoders" or just "encoders" are available in a wide number of styles and operate on several different

*Numbered references apply to numbered references for section VII as listed in the general bibliography at the end of the report.

principles. Unfortunately, many of the available encoders do not have the accuracies required for precision work. Those encoders which do have the required precision require relatively large amounts of accessory electronics to assure their performance. Proper application of an encoder requires an understanding of the characteristics and limitations of each particular device.

7-2 ENCODERS - GENERAL PRINCIPALS

7-2.1 ENCODER CLASSIFICATION. In order to evaluate the features of various encoders and related systems an understanding of the general features and principals is required. Many of these features are common to different encoders. Therefore, it will be of value to review some of the basic principals.

Basically the physical design of encoders are divided into two groups of four basic types:

1. Those having a code disk attached directly to the shaft so that full data is available in only one turn of the shaft.

2. Those having geared mechanisms, such that total data is available only after many turns of the shaft.

3. Those having sensing means which do not come in contact with the code disk. The major type of read-out of these devices is:

   a. Magnetic pick-up
   b. Capacitive pick-up
   c. Photo-electric pick-up

4. Those devices having commutator or brush type sensing elements.

A second classification of encoders can be made according to the methods used for obtaining the data as follows:

1. Space encoding-data obtained from a commutator or code disk such that each position of the disk has a discrete set of outputs.

2. Time encoding-data obtained by the measurement of time or phase between a reference and an output wave form.

3. Amplitude encoding-data obtained from voltage measurement and comparison.

The majority of encoders on the market today employ space encoding with either contacting or non-contacting type pick-off. These are used in many commercial applications and find application in tracker applications also. Perhaps the most accurate and useful type in this classification is the photo-electric type with resolution of 2^{18}.

As accuracies in excess of 2^{18} are required, other methods employing time encoding appear to be the most practical approach to the measurement problem.

Before proceeding it may be of interest to review the angular accuracy associated with the number of binary digits in an encoder readout (these figures will be referred to often in the material which follows).

If M = number of binary digits, the resolution = 2^M

M Digits	Resolution (Seconds of Arc)
14	79.10
15	39.55
16	19.78
17	9.89
18	4.94
19	2.47
20	1.24

7-2.2 THE FUNCTION OF AN ENCODER. A shaft position to digital encoder (often called analog to digital converter or shaft angle transducer) is a device used to translate analog infomation into binary digital infomation. The analog input to the encoder is the angular position of the input shaft relative to a zero reference position. The digital output corresponding to the input shaft angle is transmitted on a series of wires. Each of the output wires represents one binary digit, or bit, and is therefore capable of transmitting either a zero or a one to the device receiving the information. The one and zero states are usually represented by a voltage or a no-voltage condition on the output wires. The encoder design is such that for each discrete position of the input shaft there is only one possible combination of ones and zeros on the output wires. Therefore, assuming that the code inherent in the design of a particular encoder were known, it would be possible to determine the position of the input shaft simply by reading the infomation contained in binary form on the output wires.

Although there are many factors that must be considered in designing an encoder to perform a specific task, the general principle on which all encoders is based is quite simple. Given a shaft to carry input infomation and a series of wires to carry output information, the problem reduces to devising a mechanism capable of placing either voltage or no voltage on each of the output wires, the exact combination depending on the position of the input shaft. Thus, an encoder is essentially a switching device.[2]

7-2.3 CODING. Although not all encoders operate on the space encoding principle, an understanding of the coding system on these devices will be of value.

The familiar decimal system of counting based on the number ten, though practical for the visual expression of any quantity, is one which does not readily lend itself to a maximum simplicity and efficiency in designing computing mechanisms, for the digital succession zero to nine represents a system requiring a set of ten different conditions, or "states", which must exist in any device performing purely decimal computations. Since the mark-position procedure in a decimal system is cumbersome from the standpoint of computer and code converter design, a far simpler method requiring only two states which may be "on-off," "conducting-nonconducting," or "yes-no," etc., is more desirable. This system can accurately represent any decimal quantity. In such a two-state system, a combination of only two marks, or symbols, need be employed to express a number. Any pair of symbols representing these two marks could be arbitrarily selected, so long as they differ from each other and are not considered to represent specific numerical values in themselves.

The most commonly used code for encoders (which incidentally is also the most usable code in computer circuits) is the binary code which expresses a number as an ordered array of 0's and 1's. Two binary codes are developed in table 7-1. The weighted binary coding sequence is the simplest form of binary expression and is also known as straight or natural binary. Although the weighted binary system is probably the easiest by which arithmetic functions may be automatically processed, it has serious disadvantages when used in encoders.

Considering the symbol 1 as the encoder conductive bit, it soon becomes evident that more than one bit must change in the true binary system when transferring from one decimal equivalent to another; that is, by one digit higher or lower. For example, in changing from the decimal number 7, (binary 0111) to 8 (binary 1000), a change occurs in all four bits. Since it is impossible to cause all these digits to change state at exactly the same time instant, an ambiguity may exist while the output is changing state. The output word during this transition period may be wildly in error.

To overcome this difficulty, a variation of the weighted binary system is often used, and is termed "cyclic binary" or "Gray" code. This code is the one most often utilized in the design of encoders. An example of cyclic binary code is shown in colume 3 of the table 7-1.

The use of the Gray code has one major disadvantage; that is, systems using Gray code encoders usually require conversion to the conventional code. This means extra electronics in the computer portion of the system. However, this extra complexity is usually justified in precision equipment because of the following advantages of the Gray code.

1. The encoder ambiguity problems are eliminated because the state of only one bit changes at a time (see column (3) of the table).

TABLE 7-1. DEVELOPMENT OF BINARY CODES

(1) Decimal	(2) Weighted Binary	(3) Cyclic Binary (Gray)
0	0000	0000
1	0001	0001
2	0010	0011
3	0011	0010
4	0100	0110
5	0101	0111
6	0110	0101
7	0111	0100
8	1000	1100
9	1001	1101
10	1010	1111
11	1011	1110
12	1100	1010
13	1101	1011
14	1110	1001
15	1111	1000
16	10000	11000
17	10001	11001

2. Only one half the number of segments are required on the rings for the lesser significant bits.

3. Since only one bit changes at a time, a greater amount of sensor misalignment can be tolerated.

4. The added complexity of dual brush or "V scan" mechanisms often required to eliminate ambiguity is eliminated.*

7-2.4 BINARY CODED DECIMAL CODE. In applications where the output of the encoder is fed into a visual display the natural binary code is not always practical. Since few people can read a natural binary number and quickly convert it into its equivalent decimal number, it is sometimes convenient to use a code which is capable of giving a readout in decimal rather than binary form. The Binary Coded Decimal (BCD) code permits such a readout. In the BCD code, as the name suggests, each of the decimal digits in a number to be coded is expressed in binary form. For example, the number 863 in binary coded decimal is 1000-0110-0011. Reference to table 7-1 will indicate that the three 4-digit binary numbers shown above correspond to the three decimal digit values in the number 863. It should be obvious that the BCD code requires four binary digits for each decimal digit since there are ten numbers (0-9) that must be coded. Use of only three binary digits would permit only the decimal numbers from 0 to 7 to be coded. Where a visual decimal display (lamps for example) of the output is required, it is necessary to add some external logic circuitry between the encoder and the lamps to convert the binary output for each decade into its equivalent decimal number.[2]

Although encoders can be obtained with BCD code for special applications, they are not often used in trackers since in general the output of the encoder is to be used in a digital converter in pure binary form.

7-3 SUMMARY OF DIRECT READOUT ENCODING TECHNIQUES

Direct readout encoders can be usually identified by the coded patterns that are placed on the disk or drum. The angular position is determined directly by interrogating the patterns and the coded signal output determines the angular position of the shaft. The angular position is usually available in parallel form; i.e., all bits are available. The interrogation of the patterns may be performed via optical, magnetic, capacitive, brush, or other means. These encoders usually require a number of tracks equal to the number of bits of resolution desired; i.e., a 17-bit system would have 17 binary or Gray coded patterns or tracks.

For the direct readout type of encoder it is usually necessary to divide the least significant circular track into increments equal to the least significant bit desired. Thus, a

*For discussion of dual brush and "V scan" techniques see reference 2 and 3 in the bibliography for this section (VII) at the end of the report.

17-bit disk using conventional binary code needs to be divided into 131,072 equal increments. If the Gray code is used, the pitch of the code is doubled and only 65,536 equal increments are required. However, the accuracy of the tracks must be equivalent to that required for a conventional code.

There are a large number of encoding devices on the market, most of which operate on the space encoding principle. With very few exceptions, most of these devices are limited in their application to systems requiring relatively low resolution. There are, however, a few tracker applications where these types can be utilized. The following material will briefly describe the principles and features of common encoders:

7-3.1 BRUSH TYPE ENCODERS. The simplest encoders are those using brushes as the sensing element. These encoders consist of a drum or disk which has rings of alternate conducting and nonconducting segments. The coded drum or disk is coupled to the input shaft so that it rotates with the shaft. A brush assembly is provided with one brush for each ring of segments. For any angular position of the shaft there is a corresponding set of on-off conditions for the brushes. Individual wires connect the brushes to the external circuits. Brush encoders are limited to low resolution applications in the order of 2^8 or 2^9. They can be obtained to meet requirements of MIL-E-5272A and MIL-E-5400.

Resolution of commutator encoders is limited by the contact area of its brushes. Encoder life is affected by brush wear. A brush about 0.005-inch wide is the minimum average over the life of a unit, while commutator segment (bit) widths are approximately 0.020 inch, minimum. Life of commutator encoders can be extended beyond 1000 hours by making brushes retractable at considerable expense, however. Advantages are:

1. Simple construction
2. Inexpensive
3. Small size

Disadvantages are:

1. Low resolution
2. Mechanical wear of brushes

Discussion of brush type encoders can be found in references (4) and (5).

7-3.2 OPTICAL ENCODERS. Perhaps the most widely used encoder in the medium resolution range is the photoelectric optical encoder.

A command signal to the encoder in the form of a voltage pulse is used to flash the strobe lamp. The lamp fires for a short period of time. The light illuminates a radius of the code disk which is coded radially in a configuration of opaque and transparent elements unique for each angular position.

Bundles of light transmitted through the code disk are collimated by an optical slit system. These light beams then fall upon a photo-sensitive detector assembly which provides an electrical signal representing the presence or absence of light.

The output of each detector element is connected to a transistor amplifier which provides voltage amplification and low output impedance.

Many advances have been made in the art of building optic encoders. The elements in optic encoders are the lamp, the code disk, the optical system and the detectors. The primary advantage of optical encoders is related to ability to place a great number of segments of very narrow width on the code disk. Slit optic techniques and the use of multiple photo cell detectors complete the data transmission path. Direct reading encoders using the photoelectric principal are available in reasonable sized packages with resolution up to 2^{18}.

Optical encoders can provide extremely high resolution as they are limited only by the wavelength of light. As a result of improved optical ruling engines, information-bit width can approach 0.0005 inch. The principal applications for optical encoders are in high-resolution systems where their relatively delicate construction will not be subjected to excessive jarring or impact shock, and where space is available for preamplifiers.

For high accuracy applications to practical systems, the photoelectric types without gears have progressed farthest. Some of the outstanding features of photoelectric units compared to the other types are:

1. Optical wavelengths are small, relative to the dimensions required for the code patterns. This permits not only fine lines, but sharp transitions from "0" to "1" within the pattern.

2. Optical images can be projected accurately over the very short distances required, with sharply defined shadows.

3. Optical readout means are relatively free from edge effects, in comparison to other proximity types of readout.

4. No contact is required between the code medium and the reading means, reducing the electrical noise, mechanical and electrical wear, and friction. Also vibration problems as might be experienced with contacting media are eliminated.

5. The high resolution which can be provided makes gearing and related inaccuracies unnecessary in a majority of applications.[6]

Numerous articles have been written on optical encoders. See references 1, 5, 6 and 7.

7-3.3 MAGNETIC ENCODERS. Magnetic encoders are produced by several manufacturers and in some applications provide improvements over the brush or optical type encoders. The magnetic encoder has the advantages associated with most magnetic devices. They are simple, rugged, inexpensive, little effected by temperature and have practically unlimited life.

The main disadvantage of the magnetic encoder is its limitations in resolution because of the physical size of the code patterns and pick-up coils.

The magnetic encoder consists of a disk on which is deposited a pattern of magnetic material (usually a ferrite). The disk is then processed to magnetize the bits of the pattern.

The pick-off device (one for each bit of the code disk) consist of a two-winding coil or toroid. An alternating current or pulse is applied to one winding of the pick-off. The magnitude of the output pulse depends on the reluctance of the path around the coil. This reluctance is a function of the presence or absence of a magnetic segment on the code disk.

The excitation power for the pick-off is usually in the frequency range of 20 kc. to 200 kc. Output pulses are fed into the computer associated with the encoder.

Presently, available magnetic encoders seem to be limited to resolutions of 2^8; however, encoders of higher order of resolution may become available in the future. Some manufacturers claim higher resolution, but it is achieved by using a multi-turn type encoder. See Reference 8.

7-3.4 CAPACITY ENCODERS. Very little information was found in the search for space encoding device with capacity pick-off.

One such unit is in development by Kearfott. It consists of metallic patterns on a dielectric material. Each bit requires two concentric rings of patterns on the stator and one set of patterns on the rotor. The capacitance between the rotor and each stator ring varies cyclically with rotor displacement. The "state" of any particular bit is determined by the relative capacitance between the pattern rings. The associated electrical equipment requires transformers and flip-flops to convert the encoder data into binary outputs which can be used by the computer.

The manufacturer claims to have built a 2^7 encoder and is working on a 2^{13} development model.[9]

Perhaps future work will make this type device useful for precision trackers, although it does not appear to be a possibility at present.

7-3.5 MULTI-TURN ENCODERS. The encoders being considered in this section of the report are all available in single-turn and multi-turn versions. An understanding of multi-turn encoders is essential to properly interpret their accuracy.

A multi-turn encoder simply consist of a series of two or more code disks assembled with gear trains. Each disk is relatively low resolution type, perhaps 2^8 or less. When the total count on the least significant bit is reached after one complete revolution, the gearing advances the next code disk one bit and the counting process is repeated.

Some models of small size claim to provide resolution as high as 2^{20}. Many such devices require 4000 turns or more to reach the full count.

The accuracy of multi-turn encoders is affected directly by the accuracies of the gear trains required. The resolution claimed by these devices is only an indication of "total count" and cannot be used as an indication of the accuracy of measurement. For all practical purposes the multi-turn encoder cannot be used on precision trackers.

More general information on multi-turn encoders can be found in references 2, 4 and 5.

7-4 SPECIAL ENCODERS FOR HIGH RESOLUTION

The direct readout encoder in general is limited by physical consideration to resolutions of 2^{18} and below. Even in this range of resolution only the photoelectric type shows promise for precision tracker application. Paragraph 7-6.3 of this section will outline some of the reasons why direct reading encoders are limited to approximately 2^{18} resolution.

Methods have been developed by which higher precision can be realized in the measurement of angles. This section of our discussion will briefly present the theory of operation of these methods and devices. Paragraph 7-6 will discuss them from the application point-of-view.

7-4.1 PHASE ENCODING. The useful range of resolution in encoding devices can be extended by the application of phase encoding. Essentially, this method requires a device which will produce a voltage waveform which is shifted in phase with respect to a reference voltage. The phase shift must be proportional to the angular position of the input shaft. Using the output of such a device, high speed timing or counting techniques are used to measure the phase shift and hence the angle of the input shaft.

The principal of the phase measurement can be explained as follows:

1. The two voltages (signal and reference) are fed into detector circuits which determine when the voltage waveform crosses zero.

2. The zero point of the reference voltage is used to start a high speed binary counter. The zero point of the signal voltage is used to stop the counter.

3. While the counter is "ON" high frequency pulses generated in a reference oscillator are counted. The phase angle and hence the shaft position are proportional to the number of counts registered in the "ON" interval of the counter.

Although a large amount of electronics is required to produce a binary representation of the angle using phase measurement, the cost may be justified where extreme accuracy is needed. One compensating factor is that the need of conversions from Gray code to conventional code is eliminated, thus offsetting some of the electronics cost. One great disadvantage is that certain problems associated with the electronics are very difficult to solve.

7-4.2 AVAILABLE SYSTEMS AND METHODS. Present state-of-the-art methods and devices for measurement of angles to an accuracy greater than 2^{18} are relatively few in number. It is, therefore, within the scope of this report to review all the important methods.

The available methods for accurate measurement in binary form are:

1. Phasolver
2. Microgon
3. Inductosyn
4. Industosyn used in conjunction with an instrument servo and photoelectric encoders

7-4.3 THE PHASOLVER (MANUFACTURED BY TELECOMPUTING CORPORATION[10]). The Phasolver is a high accuracy analog measuring device employing a two-disk capacitive type transducer. Binary representation of the input shaft angle is obtained from phase measurement of the output waveforms.

The Phasolver is claimed to have a resolution of 0.32 second of arc when properly applied with the associated electronics.

The Phasolver transducer is essentially a high accuracy electrostatic shifter. It consists of a pair of dimensional low voltage stable non-conducting disks, one of which rotates. Each disk carries a pattern of conducting metal film. The stationary disk carries the input and output patterns and all electrical connections. The rotating coupler pattern couples energy electrostatically from the driver pattern and produces the output signal which is transferred to the output coupling ring in the driver pattern.

The pattern configuration provides an output signal with constant amplitude and with a phase angle which is proportional to the rotary position of the coupler disc. This is accomplished in the following manner: the driver pattern consists of two sets of conjugate, sinusoidal conductive areas which are phase displaced by ninety degrees. On either side and between the sets are the output coupler rings which are electrically connected to furnish the output signal to the load. This pattern is powered by two pairs of push-pull AC signals which have a quadrature time relationship. The coupler pattern resembles two rows of alternate bars and spaces. The rows are separated and bordered by rings formed by the distribution of the spaces. The widths of the bars and spaces are the same and equal to half wave lengths of the sinusoidal patterns on the driver disc.

As the two discs move relative to each other, the first pair of conjugate sinusoidal areas on the driver disc couples a voltage into the corresponding rectangular area on the coupler disc which varies as a sine. At the same time, the second sinusoidal pattern on the driver disc, which is phase displaced by ninety degrees from the first pattern, couples a voltage into the corresponding rectangular area on the coupler disc. Because of the quadrature of the signal voltages for the two sinusoidal pattern pairs, the output of the first rectangular area is $E \cos \theta$ and the output of the second rectangular area is $j E \sin \theta$. When these two are combined the result is a signal of constant amplitude and rotating phase. This phase shift increases continuously from 0 to 360.0 as the two discs move, relative to each other, a distance equal to one sinusoidal pattern.

The foregoing principle is applied, using both coarse and fine pattern pairs on the same disc, to provide a highly accurate, unambiguous readout of angular position. Obviously, if a single sinusoidal pattern is used in the coarse pattern, an unambiguous output position will be established. The resolution produced by the system is then a function of the associated electronics and the number of pattern pairs in the fine pattern, which determines how many times the electrical phase difference will change through 360.0 for one mechanical rotation or portion of a rotation. Because of the symmetry in the arrangement of the pattern pairs and the coupler bars, an average output of all pattern pairs is obtained. This averaging effect results in negligible errors being introduced because of non-linear pattern pair spacing or eccentricities of the rotation centers of the Phasolver elements.[10]

The output voltages from both the coarse and fine patterns are fed into electronic circuitry so that the phase shift can be accurately measured and converted to digital form.

Some desirable features of this transducer are:

a. A two-speed transducer is obtained with only one set of disks.
b. The pattern etching is not as complex as that used on high resolution photoelectric types.
c. Etching errors are averaged.
d. There is no wiring to the rotating disk.

Some undesirable features are:

a. Stable, quadrature signals are required at the input.
b. A high impedance signal is derived from the output.
c. Very stable detectors and zero crossing gates are required.

7-4.4 THE MICROGON (MANUFACTURED BY NORDEN DIVISION OF UNITED AIRCRAFT CORPORATION[11]). The Microgon system can achieve extremely high resolution by a method of generating, detecting and quantizing the phase angle between two signals.

The Microgon is available in models capable of measuring to a resolution of 2^{21} or the equivalent of 0.62 seconds of arc.

The signal generator consists of a grounded metal rotor and two insulated metal stators. The periphery of the rotor is engraved with two rows of perfectly aligned teeth. Each stator also has a similar array of teeth on its inner face, surrounding a corresponding row of rotor teeth.

As the rotor, driven by a constant speed motor, turns within the stators, the capacitance between each of the insulated stators and the grounded rotor varies cyclically as the teeth go into and out of alignment. With a DC voltage applied across each rotor-stator air gap, the resulting alternating component of charging current produces a signal with a frequency equal to the number of rotor teeth which pass a given stator reference point each second.

One of the stators is firmly secured to the body of the transducer, and its output is called the "reference" phase. The other stator is secured to and turns with the input shaft. Its output is called the "variable" phase.

The signals of the reference and variable phases will be in phase whenever the teeth of the two stators are exactly in line. If the variable stator is turned through an angle equal to the tooth pitch, the variable phase signals will shift progressively through 360 electrical degrees with respect to the reference phase signals.

While the variable stator is turning, the instantaneous frequency of its output will differ from that of the reference phase by the product of the number of teeth and the revolutions per second of the variable stator. The change in "phase angle" between these outputs is thus independent of the variations in rotor speed.

It will be noted that the signals are generated by the overall change in capacitance between the entire number of teeth. The averaging effect of this action reduces errors caused by minute variations in tooth spacing, or by eccentricity or runout, to a negligible value.[11]

The two output voltages are analyzed in the electronics package. The heart of the electronics contains a high frequency oscillator, a frequency divider and a phase detector.

The oscillator output goes to the divider, and the divider output is sent to the phase detector together with the amplified signal from one of the transducer phases. The detector compares these two signals and produces a voltage proportional to the "phase difference" between the pulse output of the divider and the sine wave output of the transducer. This output is applied so as to lock the oscillator frequency to a fixed multiple of the transducer frequency.

The output of both oscillators is fed to the "updown separator", which notes the changing phase difference between the two high frequencies caused by the turning of the variable stator (input shaft). The variable phase oscillator gains or loses one full cycle, with respect to the reference phase, when the variable stator (input shaft) turns by one quantum unit (one transducer tooth pitch divided by the primary divider ratio). The up-down separator determines the direction of this increment of stator movement and emits a pulse for each quantum via an "up" or a "down" bus to a reversible "up-down counter".

A "missing tooth" technique is used to establish a fixed zero reference. One tooth is removed from each stator and from the same relative position on each row of rotor teeth. Whenever a missing tooth space on the stator is aligned with a space on the rotor, the tone signal undergoes an exaggerated cycle. This affects the DC output of the phase detector, which is fed to a "missing tooth detector" (MTD), producing a shaped pulse. The "missing tooth" pulse gates the next carry pulse from the primary divider, establishing the instant when the missing tooth spaces are precisely aligned. At this instant, the registers in the primary divider are all transferring to zero.

In the reference phase, the primary divider output is applied to a secondary divider, the output of which is at the same rate as the RPS of the transducer rotor. By using the gated pulse described above to check and, if necessary, by resetting the registers of the secondary divider, a reference zero displacement is precisely defined. Thereafter, the reference phase registers contain data on the transducer rotor displacement from this zero.

A similarly derived gating pulse in the variable phase defines the precise alignment of missing tooth spaces of that phase. This pulse compares the reference phase register data with the up-down counter data, correcting the latter if necessary.

Data in the up-down counter is verified once per revolution of the transducer rotor and shows at all times the encoded angular displacement of the input shaft from the reference zero.[11] For further information see References 11 and 12.

Some desirable features of the Microgon are:

a. A two-speed system is obtained with a single transducer.
b. Errors in machining of teeth are averaged.
c. A correct readout is available during constant slewing operations.

Some undesirable features of the Microgon are:

a. The transducer uses a motor, bearings and brushes.
b. A high output impedance signal is obtained.
c. A jitter is present in the least significant bit. Thus a time averaged form of readout is required to mimimize the jitter effect. This jitter may be due to variations in motor speed.

7-4.5 THE ROTARY INDUCTOSYN. The rotary Inductosyn is essentially a multipole resolver which produces an inductive coupling between the rotor and stator windings. The Inductosyn windings consist of metal patterns which are etched on one side of a glass disk. In assembly, the rotor and stator are in close proximity so that the flux of one pattern is coupled to the other. The pattern behaves like a series of radial bars with end connections. A current through the pattern flows radially inward and outward to produce flux between successive bars that alternates in relative polarity. Such a pattern behaves like a multipole winding with pole spans confined by the arc spans between successive bars. Common Inductosyns are built with 256 and 360 poles. Development work is being done to provide Inductosyns with greater numbers of poles.

Accuracy of a direct mounted 12-inch diameter 360-pole inductosyn is better than ±1.0 second of arc. Special correction techniques can be used to reduce this to ±0.4 seconds of arc.

The coefficient of coupling in an Inductosyn is very low; for this reason Inductosyns are normally excited with a frequency of 10.0 KC.

The Inductosyn is utilized for direct phase measurement of angles in the following manner.

The Inductosyn in a resolver is mounted on the shaft to be measured. Coarse data is obtained from the resolver, fine data from the Inductosyn. If a particular system uses a 360-pole Inductosyn, its output voltage varies through 360 degrees for every two degrees motion of the input shaft (higher resolution can be obtained using more poles on the Inductosyn). The error output from both resolver and Inductosyn is a sine wave having constant amplitude but whose phase shift varies as a function of shaft angle.

Dual electronic channels are used, one for each device. Positive going null crossings of the excitation signal are used to open a gate. This allows pulses from a high frequency generator to be fed to a binary counter. The gate is closed by the positive going null crossings of the error signal thus blocking clock pulses to the counter.

The readout of the Inductosyn takes place within one cycle of the excitation frequency. By proper choice of the ratio "R" between excitation and pulse frequencies, the readout can be obtained to 1/R part of the pitch of the Inductosyn poles, within one count. For example, if a 360-pole Inductosyn is excited at 10.0 KC the accuracy of readout will be ±7.2 seconds of arc using a 1.0 MC counter frequency, or ±1.0 second of arc using a 7.2 MC counter frequency. The accuracy can also be increased by increasing the number of poles on the Inductosyn. Literature from Farrand Controls, Inc., 99 Wall St., Valhalla, N.Y., indicates that Inductosyns of up to 2048 poles are possible.

The circuits associated with the phase measurement of the resolver and Inductosyn outputs are essentially the same as those required in any phase measurement system. The electronics would include:

1. High frequency oscillation.
2. Frequency divider to obtain the excitation frequency
3. Phase shifter to provide quadrature voltages to encite the Inductosyn and resolver.
4. Null crossing detectors and gates.
5. Binary counter circuits.

A general description of Inductosyns can be found in references 13 and 25. Reference 14 describes a typical system for using phase measurement in a two-speed resolver, Induction combination. An advanced theoretical analysis of the Inductosyn along with practical application data can be found in reference 15.

7-4.5.1 Inductosyn Used With An Instrument Servo. Highly accurate systems with resolutions in the range of 2^{20} or higher can be obtained by using Inductosyns and resolvers in an instrument servo to drive two lower resolution photo electric encoders. There are several possible circuit arrangements for such a system all producing the same end result.

A resolver and an Inductosyn are both mounted on the shaft to be measured. The error signals from each are used as the input signals to the instrument servo. Threshhold detectors are used to determine which signal controls the servo. The output shaft of the servo drives one lower order encoder directly and one encoder through a gear train. If the system is designed to have the high speed encoder directly driven, and the low speed encoder driven through gearing, the effects of gear train errors on overall accuracy can be greatly reduced.

An alternate method might utilize a low resolution encoder mounted directly on the shaft to be measured and an Inductosyn driving an instrument servo and a second encoder for the high speed output.

The use of instrument servo methods will vary from one application to another at the discretion of the designer. However a few of the advantages of this method can be summarized as follows:

1. Resolutions of 2^{20} and higher are possible.

2. The Digital Signal can be obtained from two photo electric or other type encoders. These encoders can be low resolution types such as 2.0 seconds to 2^{13} which are readily available at reasonable cost.

3. High frequency generators and high speed counters associated with phase measurement techniques can be eliminated.

4. The accuracies and techniques for the instrument servos required are easily within the present state of the art of servo technology.

7-5 MISCELLANEOUS NOTES ON ENCODER PROBLEMS, ERRORS AND ACCURACY

The resolution of the shaft position encoder is limited by the number of discrete code elements on the encoder. The accuracy is dependent upon a number of factors including quantizing error, disk and slit alignment, tolerance of bearings, mechanical thresholds in the electronics, etc.

Designs usually aim to hold the total accumulation of errors to within one least significant bit of the encoding device.

Most encoders are rated at a maximum speed of rotation for full accuracy of readout. This speed is a function of flash speed of lamp (on photoelectric encoders), time constants of pick-offs and amplifiers and threshold levels of amplifiers.

7-5.1 ENCODER ERRORS.

1. Quantization Error

The correct angle for a given readout is taken to be the mid-point of the angular excursion of the encoder over which that readout remains unchanged. Thus for a given readout, the encoder shaft may be rotated by ±0.5 of a quantum before the output word is changed. Since a readout may be required at any particular position within a quantum, the probability distribution function for quantization error is simply a constant. There is zero probability that the error will exceed ±0.5 bits. These facts yield an RMS error of 0.289 bits due to quantization. For example, in a 17-digit encoder which has a quantum size of approximately 10.0 seconds, the Standard Deviation for quantizing is 2.9 seconds. For a 16-digit encoder, where the quantum size is approximately 20.0 seconds, the Standard Deviation for quantizing is 5.8 seconds. For a 13-digit encoder, where the quantum size is approximately 158.0 seconds, the Standard Deviation for quantizing is 46.0 seconds.

2. Instrument Error

Instrument error is the combination of all errors in the encoder other than quantizing error. This error can be measured by the use of an autocollimator and a calibrated polygon mirror. Instrument error alone is determined by making measurements at the mid-points of the quanta, thus eliminating the effect of quantization. Generally the standard deviation of instrument error is of more significance than the absolute error at any point. In applications where the encoder is used over a limited portion of its total range, the instrument error should be computed over that range only.

3. Encoder Coupling Procedure

To minimize the effects of instrument error it is important that an encoder be correctly oriented when coupled to the driving member. It is incorrect to couple the

encoder at an arbitrary angle such as zero degrees since this may result in a coupling bias error equal to the encoder instrument error at that angle. The correct procedure is to rotate both the drive member and the encoder to an angle where the instrument error is zero and to couple at that point.

In an application which uses only a limited portion of the encoder range, an extra step is required in the coupling procedure. The instrument error plot should be examined over the pertinent range to determine the mid-point between the maximum and minimum errors for this range. This mid-point should be used as the zero error reference for the limited range in following the coupling procedure outlined above.

4. Instrument Errors

Considering photoelectric encoders (although the same general considerations apply to all encoders), periodic variations in accuracy are introduced if there is a lack of concentricity of rotation due to either the bearings or disk centering on the bearings. Included in the latter is the basic concentricity of the disk pattern and centering the disk on the shaft.

Generally, the three main contributors, pattern concentricity, disk centering and bearing run-put, can be held to 3.0 to 5.0 micro-inch each by using great care. Cyclic errors in disk patterns are in the order of 1.5 seconds of arc. It is believed that this can be improved with time as manufacturers make improvements in their circle dividing machines and assembly techniques.

Photoelectric encoders have a fundamental limitation in viewing slit width (in the order of 50.0 x 10^{-6} inches) because of the wavelength of the light source. This can be somewhat improved by using ultra-violet light sources.

7-5.2 EXCITATION VOLTAGE REQUIREMENTS FOR PHASE ENCODING SYSTEMS. One of the limitations of the phase encoding systems such as with the Inductosyn or Phasolver is the specification required for the two-phase excitation voltage. The following data is taken from an analysis of a Phasolver and will apply to the power supply for any phase measuring system:

For a 200-pole pair Phasolver to ensure that periodic error in the output (due to phase errors in the applied voltages) does not exceed one second of arc, it is required that the phase error in the two-phase excitation be no greater than 3.33 minutes of arc.

Unequal amplitudes in the applied voltages of X percent result in a phase error in the output signal of maximum value $\frac{X}{2} \frac{(1)}{(100)}$ radians. For a 200-pole pair Phasolver, the applied voltages must be equal to 0.19 percent to achieve one second of arc accuracy.

For one second of arc accuracy, the second harmonic distortion in the input must not exceed 0.097 percent.

7-5.3 INSTALLATION. The encoder is a precision instrument and must be handled with great care, avoiding shocks to the encoder and its shaft. Care should be exercised to prevent burring of mounting surfaces; burrs may prevent correct installation.

Mounting should take place only after the equipment is installed in its final location.

No alterations of any kind such as drilling or machining should be made except by manufacturer. Such operations may cause serious damage to code disc, readout optics and bearings and other internal mechanisms.

No repairs of any type should be attempted, except replacement of plug-in modules.

Concentricity of coupling shaft and stress loading of the shaft must be held to the manufacturers specifications to assure accuracy.

Additional notes on errors are included in references 1, 4 and 5.

7-6 APPLICATION FACTORS OF AVAILABLE ENCODERS AND SYSTEMS

7-6.1 SUMMARY. The selection of the proper encoding device for a precision tracking system cannot be done by rule of thumb. Many factors must be considered which include the basic type of computers and control circuitry to be used in the overall system.

There are on the market today a multitude of encoding devices which are relatively inexpensive and find wide use in both commercial and military applications requiring resolution in the order 2^8 to 2.0 seconds. Also available are many devices which claim extremely high resolution by the use of multi-turn techniques. Neither of these types are useful in precision tracker applications because of their low resolution and gearing errors.

The next range of encoder types include single turn devices with resolutions in the order of 2.0 seconds to 2^{15}. A 2^{15} encoder has a basic resolution of 39.55 seconds. This resolution is too low for direct application on most precision trackers. These devices do however, find useful application in systems which employ a two speed readout using an instrument servo in conjunction with the Inductosyn.

Of all the direct reading encoder types, the photoelectric type having resolutions of 2^{16} to 2^{18} appear to be the most desirable for tracker application in regards to accuracy and ease of application, although even these devices may be limited in resolution for some installations. Literature from vendors tends to indicate that it is feasible to extend the state of the art or produce direct reading encoders having resolutions of 2^{19} or 2^{20}.

However these devices are not in current production. If such encoders were to be made in the near future they would probably be in the order of 16.0 to 20.0 inches in diameter.

When resolution in excess of 2^{18} is required there are at present only two methods which appear possible.

1. Phase measurement methods using the Phasolver, Microgon or Inductosyn. These devices have the inherent ability of producing resolutions in the order of 2^{20} to 2^{21}; however, the problems associated with the required electronics are not simple to solve, and may be the limiting item in the overall system.

2. Instrument servo measurement using an Inductosyn and two low resolution encoders. This method, while requiring the expense for design and construction of a high precision servo, has been successfully used and can be designed within the present state of the art. Its primary advantage is that it utilizes Inductosyns and low order resolution encoders, which are readily available and have been perfected.

Investigation into available encoding systems tends to leave one with the opinion that no present encoding system is ideal when resolutions of 2^{18} (4.94 seconds of arc) or greater are required. It is necessary to weigh the merits and disadvantages of each system against the requirements of the tracker to determine which system would be most practical for a particular tracker. It must be pointed out, however, that the field of high precision encoder design and manufacture is relatively new. The next few years should bring significant advances in extending the resolution, accuracy and techniques of encoding devices.

7-6.2 LOW RESOLUTION ENCODERS. The lower resolution and multi-turn encoders are not considered adequate for precision tracker application; however, for the purpose of making our survey complete a few words might be said about them.

These encoders may be either brush, photoelectric, magnetic or capacity types. They usually provide data in the Gray code and hence require electronic conversion to conventional binary code in computer section of the system. The type of power required depends upon the particular encoder. In general, all these devices require low voltage DC for brushes or amplifiers. The photoelectric encoders require lamp power and the magnetic or capacitive types require a high frequency excitation.

The following data is typical for this class of encoders:

Length:	1 1/2 to four inches
Diameter:	One to three inches
Weight:	Three to seven ounces
Life:	500 to 1000 hours at 400 rpm
	(10^6 feet of travel on circumference of code disk)
Environment:	Mil-E-5272A
Operating Temp.	0.0 degrees F to 160.0 degrees F (some available to -50°F)

See the list of Encoder manufacturers at the end of this section.

7-6.3 SINGLE TURN ENCODERS. Single turn encoders with resolution of 2^8 (633 seconds of arc) to 2^{15} (39.5 seconds of arc) are fairly common and can be obtained from several vendors. Encoders of 2^{15} and below are not useful for direct readout on precision trackers, but they can be used as elements in a complete two speed system using the instrument servo technique.

7-6.4 HIGH RESOLUTION DIRECT READING ENCODERS. In applications which permit limitation of resolution to 2^{16}, 2^{17}, or 2^{18}, the direct reading photoelectric encoder is undoubtedly the most desirable from the standpoint of accuracy, resolution, simplicity of system and cost. Furthermore, these devices are available and are in a constant state of being perfected.

There is some indication from the literature of various manufacturer that higher resolutions may be available from these devices in the future. For the present, however, the accuracy and resolution of direct reading encoders is limited to 2^{18}.

The survey of available literature shows only four manufacturers of 2^{16} to 2^{18} direct reading encoders. These are:

1. Dychro Corporation, Reference 17.

   a. Model DV-16A -16 BIT (incremental)

2. Computer Controls Company, References 18 and 19.

   a. Model DV-16B -16 BIT (incremental)

3. A. R. & T. Electronics, Inc., (Baldwin Piano Company), References 20 and 21.

   a. Model 1000 -16 BIT
   b. " 1010 -17 BIT
   c. " 1020 -18 BIT
   d. " A21SP18 -18 BIT

4. Wayne-George Corporation, References 22, 23 and 24.

   a. Model RD-16 -16 BIT
   b. Model RD-17 -17 BIT
   c. Model RD-18 -18 BIT

These encoders all have a favorable basis for comparison because they are all direct reading photoelectric encoders with built in amplifiers for each detector. They each

produce their output in Gray code so that in any particular system the electronics required in the computers and control equipment are essentially the same. Minor differences in the input voltages required are relatively unimportant because special power supplies are usually required for the encoders.

7-6.5 DEVICES FOR PHASE MEASUREMENT SYSTEMS. The Phasolver and Inductosyn systems of phase measurement provide in theory an extremely high degree of resolution. However, the restrictions imposed on the electronics provide a serious limitation to the actual degree of accuracy which can be realized. Both devices require excitation by very precise two-phase voltages. See section 7-3.4. The electronic problems associated with the generation of the excitation are on the border line of being beyond the present state-of-the-art if high accuracy is required. The extremely accurate zero detectors used to gate the clock counters are also difficult to design and produce.

In time, advances in electronics may make precision measurement practical with these devices, but at present such methods do not appear desirable for installations requiring high accuracy and high reliability. More information on these two systems including physical descriptions will be given in a subsequent report.

The Microgon

The Norden Microgon system is presently available in two models, one with 2^{19} resolution and one with 2^{21} resolution. Only one or two of the 2^{21} resolution units have been built to date so that one might consider it as being somewhat of a development device. Not much data is available at this time concerning the 2^{21} resolution device. A few of the major features are listed below:

Size:	About 19.0 inches in diameter with 7-inch hole in shaft.
Environment and Reliability:	No definite data available.
Cost:	Believed to be in the range of $60,000 to $80,000.

The 2^{19} resolution Microgon is a much more completely developed system and has been utilized in numerous applications. Its important features are as follows:

Size:	5.3 inches long; 3.375 inches diameter. No hole in shaft.
Mounting:	Synchro type mounting.
Environment:	

Temperature:	-30 degrees to 55.0 degrees C
Humidity:	90 percent
Shock:	5.0 g
Vibration:	0-13 cps at 0.06 double amplitude

Reliability (hours without failure):

Transducer:	5000 hours
Electronics:	500 hours

Pre-Amp: A small pre-amplifier must be located within ten feet of the transducer.
Cost: $20,400, including electronics.

The Norden Microgon system has some features which make it more desirable than other phase measurement systems. Mainly, the Microgon generates its own reference signal, thus eliminating the need for precise two phase voltages. The requirement for precision detectors and high frequency oscillators is the same as in other phase measurement systems.

The missing tooth technique used to provide course data, while providing some advantage in simplifying the transducer, does require added electronic circuitry to provide the course data readout. Although this circuitry is within the present state-of-the-art from the design point of view, the added complexity of the electronics would tend to lower the over-all reliability of the system. The requirement for a continually running motor also may be a problem from the reliability point of view.

7-6.6 INSTRUMENT SERVO SYSTEM. Information on the instrument servo system will appear in a subsequent report.

7-7 LIST OF ENCODER MANUFACTURERS

The following list includes those manufacturers surveyed in this study. It is believed to be very nearly complete as to the number of manufacturers which produce shaft angle encoders which might be used for military applications.

The following code is used in the tabulation:

Type

A	Photoelectric
B	Magnetic
C	Capacity
D	Brush
E	Inductosyn

Readout

1	Single turn direct readout
2	Single turn phase readout
3	Multiple turn readout
4	Incremental readout

	Type	Readout	Range
A. R. & T. Electronics (Baldwin) 1101 McAlmont Street Little Rock, Arkansas	A	1	2" to 2^{18}
Ascop Division of Electro-Mechanical Research Princeton, N.J.	B B	1 3	2^{7} 2^{19}
Coleman Electronics Inc. 133 East 162nd St., Gardena, Calif.	D	3	2^{20}
Computer Controls Company 983 Concord St., Framingham, Mass.	A	1	2" to 2^{16}
Data Tech 238 Main Street Cambridge, Mass.	B B	3 4	Up to 2^{20} 2^{10}
Datex Corporation 1307 S. Myrtle Avenue Monrovia, Calif.	D D	1 3	2" ?
Dayton Instruments Inc. 404 Winston Avenue Dayton, Ohio	D	3	?
Del Electronics Corporation 521 Homestead Avenue Mount Vernon, N. Y.	E	2	20^{20}
Dynamics Research Corporation Stoneham, Mass.	A	4	2^{20}
Dychro Corporation 49 Walnut Street Wellesley, Mass.	A	1	2" to 2^{16}
Electro-Mec 47 33rd Street Long Island City	D	1	2^{8}

	Type	Readout	Range
Farrand Controls Inc.	E	2	2^{20}
99 Wall Street			
Valhalla, N. Y.			
Guidance Controls Corporation			
110 Duffy Avenue	D	1	2^{9}
Hicksville, N. Y.	D	3	2^{19}
W. & L. E. Gurley			
Troy, New York	A	1	2^{13}
Instrument Development Laboratories			
67 Mechanic Street	D	3	to 2^{17}
Attleboro, Mass.			
Kearfott			
Little Falls, N. J.	D	3	2^{15}
	C	1	2^{7}
	B	1	?
Librascope Division	D	3	2^{7} to 2^{19}
General Precision, Inc.	B	3	2^{8} to 2^{19}
Glendale, California			
Litton Systems, Inc.			
5500 Canoga Avenue	D	3	2^{8} to 2^{13}
Woodland Hills, California			
Machine Tool Automation	B	1	2^{10}
Southport, Connecticut	B	3	2^{13}
	B	4	2^{10}
Norden			
Division of United Aircraft Corporation	C	2	2^{21}
Milford, Conn.	B	1	2^{8}
	D	3	2^{7} - 2^{19}
Theta Instruments Corporation	D	3	2^{12}
Saddle Brook, N. J.			
Wayne George Corporation	A	1	2" to 2^{18}
588 Commonwealth Avenue	A	4	2" to 2^{15}
Boston 15, Mass.			

Section VIII
ACCURACY STATE-OF-THE-ART

8-1 INTRODUCTION-ERROR SUPPRESSION AND COMPENSATION

Another approach to this entire study could have been to use existing pulse radar trackers as the point of departure and determine the parameters of each tracker including accuracy, erection time, weight, prototype costs together with problem areas. After such a survey, the study could have been directed at integrating this information into a design or designs for the instant applications. However, it was thought wiser to start with fundamentals, as has been done, and develop the fundamental into such depth and breadth as the study time and money permitted. For a comprehensive study of the topics included, sufficient bibliographic references are given. This approach was taken to meet the defined objective of the study. Hence, this study has been directed at the achievement of minimum absolute errors through simple, straightforward design without complications that, while reducing error, cause a price to be paid in terms of cost, complexity, erection time, reliability, etc.

8-2 SYSTEM ACCURACIES

Preceding the discussion, it is well to understand that this study relates to the study of trackers associated with pulse radar systems. There are a multiplicity of missile and satellite position (and position derivitives) measuring systems, most of which are inherently more accurate than pulse radar. Optical systems such as fixed, tracking and ballistic camera systems, weather limited, are used. There are CW Interferometer Systems, including Doppler, which are more accurate than pulse radar but which require a number of trackers and accurately determined baselines. Table 8-1 indicates the accuracies of some of these systems. The quantitative values given should be taken as suggestive only because of the complex premises, assumptions and definitions (perhaps slightly different for each) used in generating the values. In general, these are absolute accuracies which do not contain propagation errors, scintillation errors or servo dynamic lags, and from which systematic error has been removed (to the extent possible) by calibration.

8-3 PULSE TRACKER ACCURACIES

Table 8-1 is an itemization of the accuracies of various radar trackers. In the same sense the accuracies in table 8-1 are suggestive only; the tracker accuracies should be considered as indicative of the accuracy only because of the varying assumptions, definitions and mathematical techniques involved in establishment of the quantitative values. The quantitative values are average, standard deviations under optimal operating conditions. Dynamic lags are not included, which accounts for the difference between the instant values and other values herein for Designs A, B, C and D. The purpose of table 8-2 is to give the reader a "feel" for the range of accuracies involved antecedant to a discussion of error suppression and compensation.

TABLE 8-1.

System	Precision Random Error Standard Deviation	Bias Systematic Error in Azimuth and Elevation Axes	Accuracy Total Error (MR)
<u>Radar</u>			
SCR584 Mod II	1.5	1.5	2.2
AN/MPS-26	1.0	1.0	1.4
AN/FPS-16 (MPS-25)	0.2	0.1	0.22
Atlas Guidance System	0.1	0.1	0.14
<u>Other</u>			
Ballistic Camera	0.015 (in X and Y)	0.008 (in X and Y)	0.017 (in X and Y)
Fixed Camera	0.115 (in X and Y)	0.500 (in X and Y)	0.51 (in X and Y)
Tracking Camera	0.043	0.100	0.11

Note 1: Unsmoothed data without boresight correction

Note 2: Add dynamic lags: Coefficients: V of 0.0065 mil/mils/sec^2
A of 0.065 mil/mils/sec^2

Note 3: Add dynamic lags: Coefficients: V = 0.004 mil/mils/sec
A = 0.016 mil/mils/sec^2

TABLE 8-2. TYPICAL TRACKING SYSTEM ACCURACIES

Tracker	Total Accuracy (Milliradians) $A^2 + E^2$
Atlas Guidance Tracker Mod III	less than 0.1
Design B	less than 0.1
FPS-16 (MPS-25)	0.15
Atlas Guidance Tracker Mod I	0.3
Design A	0.5
Advent Tracker	0.8
MPS-26	1.0
Design C	1.3
SCR 584-Mod II	1.6
Design D	1.6
SynCom	?

In a given application the proper path to follow in reducing error is to take an existing design prototype, evaluate its total error and "breakdown" the total error into components related to specific sources of error in the tracker. The next step is to consider each component error and to generate techniques by which the component error may be reduced to decrease the total error of the tracker. Instead of using an existing prototype of some design and its evaluation for this purpose, let us use the a priori analysis of Design B herein. The errors of design B are estimated in table 8-3:

TABLE 8-3.

Error Source	Error
Boresight Shift	
Collimation Error	0.04 milliradians (RMS)
Radome Boresight Error	0.02
Reflector-Horn Distortion	0.02
Axis Misalignment	
Azimuth Axis Out-of-Level	0.01
Elevation Axis Misalignment	0.01
Elevation Axis Bearing Runout	0.01
Boresight Scope Alignment	0.01
Servo Error	0.03

After manipulation, these values translate into 0.1 milliradians total error $\left(\sqrt{E^2_A + E_E^2}\right)$ or less in the elevation and azimuth axes, if environmental and use factors are not considered. The problem then is to determine what can be done to reduce these errors: (1) the reflector-horn distortion can be reduced by adding appropriate stiffness and paying the penalty in weight and inertia involved; (2) the elevation bearing runout can be reduced by larger diameter bearings and paying the penalty in size and associated weight and inertia; (3) the elevation axis misalignment and the data unit error are negligible on the premise that all possible errors are to be reduced to less than two seconds of arc as an objective. This reduces us (4) to a consideration of collimation error which probably cannot be reduced much beyond 0.03 to 0.04 milliradian (by calibration). Radome boresight shift may be reduced to 0.01 milliradian by techniques to be discussed. Servo error will remain as a large source of error and will be discussed. The foregoing has been predicated on very mild environmental conditions. All the environmental conditions (thermal deflections) will cause error to grow significantly. The thermal deflections may arise from a number of sources but solar radiation is most troublesome.

This all leads to the conclusion that the state-of-the art limit on tracker accuracy, as defined herein, is grossly as shown in table 8-4:

TABLE 8-4

Collimation Error	0.03 milliradian (RMS)
Radar Boresight Shift	0.01
Out-of-level	0.01
Elevation Axis Misalignment	0.01
Boresight Scope Misalignment	0.01
Servo Error	0.03
Data Unit	0.01
	0.05 milliradian (RMS) or 10 sec./of arc. Standard deviation in the elevation and azimuth axes which translates into 0.07 milliradian total error (RMS).

This estimate of the state-of-the art limit is a gross one, and further inquiry is necessary to have complete assurance of its validity. The figures do not include microwave errors of any character. It does not include servo dynamic lags and the servo error of 0.03 milliradian is the self-induced servo electrical noise and mechanical noise such as found in bearings and deadband. The accuracy problem can then be reduced to the following key areas of considerations which control tracker accuracy.

1. Reduction of collimation error to minimum
2. Reduction of radome boresight shift to minimum
3. Reduction of thermal deflection to minimum
4. Reduction of servo error (including dynamic lags) to a minimum

8-4 ERROR SUPPRESSION AND COMPENSATION

There is a distinction to be made between error suppression and compensation. Error suppression involves a device or technique by which error may be suppressed prior to reaching the data unit. An example might be the use of servo leveling to reduce error brought about by differential settlement of the tracker feet, out-of-level, or thermal deflections.

Compensation, on the other hand, permits the error to occur but compensates for it. Typical of compensation of error might be the comparison of the tracker with a much more accurate instrument, such as a ballistic camera, to establish bias that may be calibrated out by offsetting of data unit zero points. Various uses of computer and data reduction may be viewed as compensation schemes. Data "smoothing" is another example of compensation.

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CONCLUSIONS

CONCLUSIONS

This Second Quarterly Report documents the tentative results of many special studies which will be concluded in the final report. The slight shift in the objective of the study to a design manual for design synthesis of transportable trackers is a desirable one, but does result, because of time and money limitations, in a qualitative rather than quantitative treatment of many important topics. However, the chosen course appears to be of much greater value than merely localizing the study to two or three specific designs.

PROGRAM NEXT INTERVAL

PROGRAM FOR NEXT INTERVAL
Third and Fourth Quarters

The current plan is to have the Third Quarterly Report represent a basic draft of the final Fourth Quarterly Report, serving as an introduction to the final document. In this manner the concluding phase of a study of three months duration can be efficiently utilized to expand and refine the draft of the final report into a document that will be mutually acceptable to the Signal Corps and to the General Electric Company.

THIRD QUARTERLY REPORT

The objectives of the program as a whole as outlined under PURPOSE in the first portion of this report will be further implemented and developed in the Third Quarterly Report which is to be oriented to Electro-Mechanical Design. The report as tentatively planned will not dwell on radar system or microwave antenna design (except as a broad background) or on the assumption that all system and microwave requirements have been reduced to mechanical terms. It will be assumed that the reader will have a fundamental knowledge of electro-mechanical design. A comprehension by the reader of microwave, servo and elasticity theory, as well as system dynamic analysis and statistical mathematics will be helpful but not necessary.

The report will be divided into four parts as follows:

1. System Design
2. Electro-Mechanical Design
3. Ancillary Design
4. Integrated Design

The System Design material will cover germane theory and design in those areas which are fundamental background to the electro-mechanical design and which include microwave antenna design, error analysis, loads and environmental constructions. These topics will be treated more or less lightly but will be supported by a wealth of bibliographic material.

Discussion of Electro-Mechanical Design will dwell principally upon the fundamental areas of structural, power drive and radome design together with transport and erection schemes. In each instance, a catalog of feasible alternatives will be provided plus discussion of the considerations for and against the use of each in a specific application.

Emphasis on Ancillary Design will be given over to areas which in general relate to accuracy. A catalog of possible techniques and associated instrumentation will be set forth together with the accuracy achievable with each. This admits of a selection which will be commensurate with the needs of the application at minimum cost.

The discussion on Integrated Design will tie together all of the previous pieces that have been discussed into an integrated total design of optimal balance between competing considerations. Four designs of varying parameter balance will be synthesized in accordance with methods suggested in the report. These designs will permit the development of a family of curves which will relate accuracy to cost with erection time as a parameter.

The four designs will be so defined as to range over a band of accuracies and a band of erection times while logically employing a geared electric drive, a direct electric drive and a hydraulic drive. A space frame radome, a dual wall radome and a skin thin radome will also be appropriately employed.

FOURTH QUARTERLY REPORT

In the light of stated plans for the Third Quarterly Report and of program objectives, the final or Fourth Quarterly Report has been tentatively organized as follows. It should be brought out here that the following deviates in some respects from plans for the final report as originally set forth under PROGRAM NEXT INTERVAL in the First Quarterly Report. These changes, of course, come as the logical result of the special studies as they have been pursued up to this point and are reflected in tentative plans for organization of the Fourth (and final) Quarterly Report, outlined basically as follows:

TITLE PAGE
CONTENTS
PURPOSE
ABSTRACT
PUBLICATIONS

BOOK I
SYSTEM DESIGN

1-0 INTRODUCTION

2-0 DESIGN SYNTHESIS

- 2-1 Introduction
- 2-2 General Considerations
- 2-3 Concepts
- 2-4 Initial Design Synthesis
- 2-5 Conclusion

3-0 SYSTEM ANALYSIS

- 3-1 Introduction
- 3-2 Error Classification
- 3-3 Microwave Error
  - 3-3.1 Introduction
  - 3-3.2 Noise Angular Error
  - 3-3.3 Automatic Tracking
  - 3-3.4 Servo Filter Factor
  - 3-3.5 Gain, Beamwidth

BOOK II
MECHANICAL DESIGN

BOOK III
POWER DRIVE DESIGN

BOOK IV
INTEGRATED DESIGN

KEY TECHNICAL PERSONNEL

IDENTIFICATION OF PERSONNEL

SECOND QUARTER MANHOUR BREAKDOWN

Total breakdown of hours expended by key personnel will be covered in detail in a subsequent quarterly report.

PERSONNEL RESUMES

Name: Blasangame, P.A., Project Engineer

Education: B.S. in Mechanical Engineering, 1948 George Washington University. Two years of graduate work.

Experience: Mr. Blasangame joined the Ordnance Department of the General Electric Company in 1951 as Project Engineer on the Navy's Gunfire Control System Mk 56.

He has held a series of increasingly important positions, particularly in System and Project Management.

He has complete engineering responsibility for the Navy's Torpedo Workship Equipment Standardization Program.

Upon successful completion of the Program, Mr. Blasangame was made Project Engineer on the AN/FPS-7 radar antenna. In this capacity, he was responsible for the complete engineering.

In December 1955, Mr. Blasangame was appointed Project Engineer of the Radar Tracker for the Atlas Missile Guidance System. In this capacity, Mr. Blasangame had over-all responsibility for production engineering, manufacturing, quality control, cost, schedules, contract administration, and the construction of a two-million dollar facility for testing of the antenna.

Following this Mr. Blasangame was assigned the responsibility of organizing engineering effort on the Navy's 350-ton Guided Missile Launching System Mk 12.

In May 1958, he joined the Heavy Military Electronics Department to supervise the engineering of the organ pipe scanner for the BMEWS. Upon completion of this assignment in October 1961, Mr. Blasangame returned to the Ordnance Department for continuation of his work in Air Force and Naval Ordnance.

Name: Garcia, R., (Design A)

Education: B.S. in Civil Engineering, 1951 City College of New York, M.S. degree, Rensselaer Polytechnic Institute. Special summer courses at M.I.T., and several General Electric Company courses.

Experience: Mr. Garcia has been with the General Electric Company since 1955 where he has worked on various contracts which involved structural analysis and design. He has worked on design of components of the AS 807 and the acquisition radar in the Plato system, and on the design of the shock-mitigating system of the Polaris fire control optical cart. Mr. Garcia has also worked on the structural analysis of the radome for a ballistic missile tracker and has done stress analysis work on the SPS-30. He has also done the structural and vibration analysis for the Triax Tracker.

Prior to joining General Electric, Mr. Garcia was a consulting engineer on the design of large structures.

Mr. Garcia is a registered professional engineer in Massachusetts. He is a member of Chi Epsilon, Sigma Xi, and S.E.S.A.

Name: Germanowski, F., (Radomes)

Education: BME degree, 1955 Clarkson College of Technology.

Experience: In his capacity as a production engineer with General Electric Ordnance Department Electro-Mechanical Equipment Engineering, Mr. Germanowski has been responsible for extensive design and engineering work on such projects as the Navy SPS30 Height Finder, the 024 Atlas Tracker and the MPQ4 mortar locator.

Name: Rote, W.H., (Design B)

Education: BME degree, 1951 Renssalaer Polytechnic Institute; MS degree in Mechanics (RPI) 1959. Graduate draftsman, 1942, General Electric Apprentice Course.

Experience: Mr. Rote is responsible for developing, designing and evaluating such mechanical equipment as radar antennas, gun directors and missile directors. In addition to his functions as design leader, he prepares and evaluates technical proposals for Ordnance equipment.

He has been responsible for mechanical and structural design and analysis on the SPS-30 antenna and acted as Quality Maintenance Engineer on the SPS-8 Antenna. He also conducted structural and experimental stress analysis on the FPS-7 antenna. He has done mechanical and structural design on receiving antenna for the Plato missile detection system and for the AS-207 mobile search antenna. He also conducted studies of thermal stability in connection with the Atlas radio guidance antenna and participated in the development of improved radar windows for the Atlas radio guidance antenna.

Mr. Rote has conducted experimental investigation of dynamic behavior of the SPS-17 antenna and carried out modifications to the structure as a result of the investigation. He designed the structure for the MPQ-4 mortar locator trailer; he completely redesigned a Torpedo Course Plotter to conform to MIL and JAN specs.

Name: Wales, C.C., (Servo Design)

Education: Graduate, U.S. Naval Academy, 1944. Completed two 26-week General Electric Courses, the Servomechanisms Course and the Automatic Control Systems Course.

Experience: Mr. Wales was recently empolyed as a control equipment analysis engineer. He was recently active in the engineering of the Aiming Reference System of the Sugar Grove 600' Radio Telescope, and was responsible for the design of the Inductosyn servos which were to provide angular output signals in both digital and analog form.

Since March 1958, Mr. Wales has been employed as a control equipment design engineer. As such, he was responsible for preparation or revision of purchase specifications for servo components for the ATLAS Radio Guidance Tracking Antenna, and was in charge of electrical engineering on this program during manufacture of the last six antenna systems.

He joined the General Electric Company in 1956 as a technical publications engineer. In that capacity he authored manuals on the theory of operation and maintenance of the Gun Fire Control System, Mk 75 and the Director, Gun and Guided Missile, Mk 73.

Name: B. Wilbur

Name: E. Mackey

Name: T. Black

Name: J. Russell

Background information on the above listed engineering personnel, also involved in the study, will be supplied in a subsequent report.

ABSTRACT CARDS

General Electric Company, Ordnance Department, Pittsfield, Mass., DESIGN STUDY OF A TRANSPORTABLE ANTENNA SYSTEM by P.A. Blasangame et al, Report No. 2 Second Quarterly Progress Report for Sept. 62 - Nov. 62; 158 incl. illus. (Contract DA-36-039SC-90713) DA Project No. 0069-PM-62-91-91 (8101)

Unclassified Report

This report presents the optimization phase of a study to evaluate the feasibility of requirements for a transportable antenna system having an accuracy of 0.05 milliradian CEP and an erection time of 12 hours. The objective of the study has been crystalized into generation of a design manual for transportable satellite trackers. Basic parameters are developed for four basic antenna designs to meet original cost, accuracy and erection time limit requirements. Special studies are expanded to provide a basic format for the final report.

1. Antennas - Error Analysis of antenna systems expanded
2. Antennas - Mechanical Design and Transportation considerations
3. Antenna - Power Drive design; Data Unit and Accuracy survey

I. Design Study of A Transportable Antenna System
II. Blasangame, P.A. et al
III. U.S. Army Signal R & D Lab, Fort Monmouth, N.J.
IV. Contract DA-36-039SC-90713

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