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TECHNICAL PROPOSAL

For The

ESSCO

SYNERGISED ANTENNA SUBSYSTEM

and

ACCESSORIES

VOLUME I

1971

ELECTRONIC SPACE SYSTEMS CORPORATION
Old Powder Mill Road
Concord, Mass. 01742

3.3 ELEVATION OVER AZIMUTH PEDESTAL

3.3.1 Environmental

The review of all applicable environmental influences and forces effecting the pedestal design, described in the previous section, have been distilled to those that clearly represent the conditions which the pedestal must withstand and operate within. Since the pedestal is to operate within the protected environment of an ESSCO Metal Space Frame Radome, the pertinent specifications are reduced to the following:

a) Ambient temperature ranges from -20F to +140°F.

b) Barometric pressure from sea level up to 10,000 ft. altitude.

c) Salt atmosphere as encountered in coastal regions, or during ocean transport and use.

d) Ozone as encountered in Arctic regions and in the vicinity of heavy electrical equipment.

e) Sand and dust environments.

The methods of design, fabrication and material choice, to comply with the requirements of efficient operation and reliability, while exhibiting the lowest possible service life maintenance, must be completely compatible with the foregoing list. The materials and finishes to be utilized represent ESSCO's accumulated experience gathered over extended periods of performance time.

3.3.2 Design, Construction and Physical Characteristics

In achieving a balanced design, ESSCO has employed a thorough analysis to insure the efficient utilization of the basic materials for the pedestal. The discussion that follows of ESSCO's analytical approach and methods includes a detailed analysis of the drive train as well as other components, such as brakes, stow pins, mounting interface and mechanical stops. In addition, because of its importance to system performance, the natural frequencies of the pedestal have also been analyzed in detail.

Following the discussion of the analytical design methods, the details of the pedestal assembly are described.

3.3.2.1 Mechanical and Structural Design

A. Drive Train Analysis

The drive train in the pedestal has been analyzed to determine its stiffness, strength and life expectancy.

The stifness of the drive train is a particularly critical system parameter because it is generally the lowest of all the stiffnesses of elements that make up an overall antenna installation. Therefore, it is the controlling factor in determining the lowest natural frequency of vibration of the total installation and thus has a considerable impact on the stability of the servo system which controls the motion of the antenna.

The stiffness is calculated by adding the compliance of all the deflecting elements in the drive train wherein the compliance is defined as the rotation of the output axis resulting from a unit torque applied to the output axis. The stiffness is then calculated from the following equation:

$$\frac{1}{K_G} = \sum \left(\frac{1}{K}\right)_i$$

K_G = Stiffness of drive train

$(1/K)_i$ = Compliance of i^{th} element in drive train

The elements in the drive train which contribute to the compliance are bending and torsional displacement of the gear cluster shafts, approach and methods includes a detailed analysis of the drive train as well as other components, such as brakes, stow pins, mounting interface and mechanical stops. In addition, because of its importance to system performance, the natural frequencies of the pedestal have also been analyzed in detail.

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$(1/K)_i$ = Compliance of i[th] element in drive train

The elements in the drive train which contribute to the compliance are bending and torsional displacement of the gear cluster shafts, radial and axial displacement of the bearings and local deflections of the gearbox around the bearing housings.

At the same time that the compliance of the drive train elements are estimated, the compliance of the overall pedestal structure is also determined. The stiffness of the entire pedestal is then calculated from the following equation:

$$\frac{1}{K_p} = \frac{1}{n K_c} + \frac{1}{K_s}$$

K_p = Stiffness of pedestal

$1/K_s$ = Compliance of overall pedestal structure

n = Total number of drive trains transmitting torque to output

To determine the strength and life expectancy of the drive train, the loads on each cluster are calculated:

$$\sum \tau = I \alpha$$

$\sum \tau$ = Resultant torque on cluster

I = Mass moment of inertia

α = Angular acceleration of cluster

In order to account for the effects of friction on the drive train, an iterative procedure is used applying the above equation first to the overall pedestal to determine the angular acceleration, and then to each cluster in succession to determine the loads acting on the cluster gears. Equilibrium equations are then applied to determine the reactions at the cluster bearings.

Knowing the loads on all the gears and bearings, the friction torque developed by each is then estimated. At this time, the friction torque due to the bearing seals is also estimated. Next, the torque equation is again applied to the overall pedestal, this time adding algebraically the total friction torque determined in the first iteration to the original value of ΣT and a new value of α computed. This procedure is repeated until the changes become insignificant. At this point, all the gear tooth loads and the bearing loads are known for the particular loading condition analyzed. All probable loading conditions have been analyzed, including appropriate combinations of torque generated by the motor, brake, mechanical stops, etc. Tooth friction torque is estimated using the following expression for power loss in the mesh. *(Ref. No. 20)

$$P_L = \frac{50 f \cos^2 \psi}{\cos \phi_n} \left(\frac{H_s^2 + H_t^2}{H_s + H_t} \right)$$

$$H_t = \frac{m_G + 1}{m_G} \left[\sqrt{\left(\frac{r_o}{r}\right)^2 - \cos^2 \phi_n} - \sin \phi_n \right]$$

$$H_t = (m_G + 1) \left[\sqrt{\left(\frac{R_o}{R}\right)^2 - \cos^2 \phi_n} - \sin \phi_n \right]$$

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*Ref. No. 20 - "Gear Handbook", Darle W. Dudley

P_L = Power loss in mesh (%)

f = Coefficient of friction

ψ = Helix angle

ϕ_n = Normal pressure angle

m_G = Mesh reduction ratio

r_o = Outside radius of pinion

r = Pitch radius of pinion

R_o = Outside radius of gear

R = Pitch radius of gear

The coefficient of friction depends on the pitch line velocity, the lubrication and the tooth loading and is selected for each mesh using data presented in Ref. No. 20.

The friction torque from each bearing is computed from

$$B_F = \frac{fRD}{2}$$

B_F = Bearing friction torque

R = Resultant load on bearing

D = Bearing bore diameter

The resultant load, R, is determined from the iterative analysis and includes the bearing preloads. The friction coefficient f and the associated bore diameter are obtained from manufacturer's recommendations.

The bearing seal friction torque is determined from:

$$S_F = \frac{\pi f w D_s^2}{2}$$

S_F = Seal friction torque

w = Seal radial load per unit length

D_s = Shaft diameter

The friction coefficient, f , and the radial load per unit length, w , are obtained from manufacturer's recommendations.

Using the critical gear tooth loads obtained from the iterative analysis described, the gears have been checked for surface durability and bending strength using the following equations from Ref. 20.

$$S_c = C_p \sqrt{\frac{W_t C_o C_s C_M C_F}{C_v F d I}} \leq S_{ac} \frac{C_L C_H}{C_T C_R}$$

$$S_t = \frac{W_t K_o P_d K_m K_s}{K_v F J} \leq S_{at} \frac{K_L}{K_T K_R}$$

S_c = Calculated control stress (psi)

W_t = Tangential tooth load (lb.)

d = Pinion pitch diameter (in.)

F = Face width (in.)

S_{ac} = Allowable contact stress (psi)

C_p = Elastic properties coefficient

C_o = Overload factor

C_v = Dynamic factor

C_s = Size factor

C_M = Load distribution factor

C_F = Surface condition factor

I = Geometry factor

C_L = Life factor

C_H = Hardness ratio factor

C_T = Temperature factor

C_R = Safety factor

S_t = Calculated bending stress (psi)

W_t = Tangential tooth load (lb.)

P_d = Transverse diametral pitch

F = Face width (in.)

K_o = Overload factor

K_v = Dynamic factor

K_s = Size factor

J = Geometry factor

S_{at} = Allowable bending stress (psi)

K_L = Life factor

K_T = Temperature factor

K_R = Safety factor

K_M = Load distribution factor

The various factors are selected appropriately from the data presented in Ref. No. 20 in predicting the life expectancy of the gears under normal operating conditions and their capability to survive extreme environmental loading conditions.

The bearings have been similarly checked using analytical methods and allowable loads supplied by the manufacturers.

The drive train analyses just described have been programmed by ESSCO for routine analysis on an in-house time sharing high speed computer terminal. These programs compute the inertias of the cluster in the drive train, the friction torque in the bearings, seals, and gear meshes, the loads on the gear teeth and the bearings, and the surface contact and bending stresses on the gear teeth and their life expectancy.

B. Special Components

(1) Brake - An electro-mechanical brake is used for stopping and for holding the pedestal in a stationary position. It may be included anywhere in the drive train, on the output axis or on the motor. Its required characteristics will vary considerably with its position in the drive train. These characteristics are determined by including the brake in the drive train analysis discussed previously. The inertia and the torque capability of the brake are parameters considered in the analysis. The analytical evaluations performed assure that the brake will provide the stopping and holding torques required without overloading the drive train components or burdening the drive train with inertia that might unnecessarily lower the natural frequency of the antenna installation.

(2) Mechanical Stop - The mechanical stop is positioned so it will intercept the moving load and dissipate its kinetic energy in a controlled manner if the load accidentally passes its limit of travel in elevation. The performance of the mechanical stop is evaluated by inserting it into the drive train analysis described previously. The capacity of the stop is selected to be able to bring the load to rest within the required stopping distance when the load engages it at maximum speed with the motor on and the brake off. The position of the stop, the force and the stroke are all considered in the analysis and selected to provide the required energy dissipation without overloading the drive train components. The force determined from the analysis is used to check the strength of

the mounting of the stop on the pedestal and also the structure that it engages on the reflector.

(3) Stow Pins - The stow pins hold the reflector in the stow position when it is not being operated. The holding capacity of the stow pins must be great enough to hold the reflector against survival wind torque. The pins engage between a fitting on the pedestal structure and a hole in the bull gear. The strength of the pedestal structure, the fitting, the pins and the bull gear are all analyzed for the survival condition and as many pins as are required to develop the full wind torque are designed into the pedestal.

(4) Mounting Interface - The pedestal mounting interfaces occur where the pedestal mounts to the tower and where the reflector mounts to the pedestal. These interfaces are analyzed to assure that load paths of sufficient strength and stiffness are provided to transmit shears, torques and direct loads across the interfaces. Careful attention is given to the size and location of shear pins and bolts and to stiffening gussets to make sure that localized deflections at these interfaces will not contribute unnecessarily to the overall compliance of the pedestal.

(5) Pedestal Structure - Most of the critically loaded areas of the pedestal structure, as well as the previously discussed special components, are analyzed for strength. There are additional areas, however, where high loads are transmitted across welded connections, such as the housings for the bull gear bearings. Such areas are analyzed individually as required to assure overall soundness of the pedestal structure.

C. Natural Frequency

Natural frequencies of the pedestal are calculated considering the stiffnesses and reflected inertias of the drive trains, the stiffness and

and mass of the pedestal structure, the inertia and stiffness of the load on the pedestal and also the mass and stiffness of the tower foundation.

To obtain the lowest natural frequency for the entire antenna installation, the frequencies are separately calculated for the pedestal with the load on it, then for the tower with the mass of the pedestal and load on it, and finally for the foundation with the mass of the entire installation. The lowest natural frequency for the entire installation is then computed using Dunkerley's equation:

$$\frac{1}{\omega_A^2} = \frac{1}{\omega_P^2} + \frac{1}{\omega_T^2} + \frac{1}{\omega_F^2}$$

ω_A = Lowest natural frequency of the antenna installation

ω_P = Lowest natural frequency of the pedestal with the load on it

ω_T = Lowest natural frequency of the tower with the mass of the pedestal and load on it

ω_F = Lowest natural frequency of the foundation with the entire mass of the installation on it

Because of the low stiffness of the pedestal compared to those of the tower and the foundation, this approximation gives an accurate prediction of the lowest natural frequency of the total installation.

Several modes of vibration are of interest and mathematical models for each have been formulated and analyzed.

A symmetrical mode of vibration about the elevation axis is shown in Figure 3-34(a) and the model for this mode which treats the pedestal and the load on it is shown in Figure 3-34(b), consisting of rotary inertias and torsional springs. The moving load on the pedestal is repre-

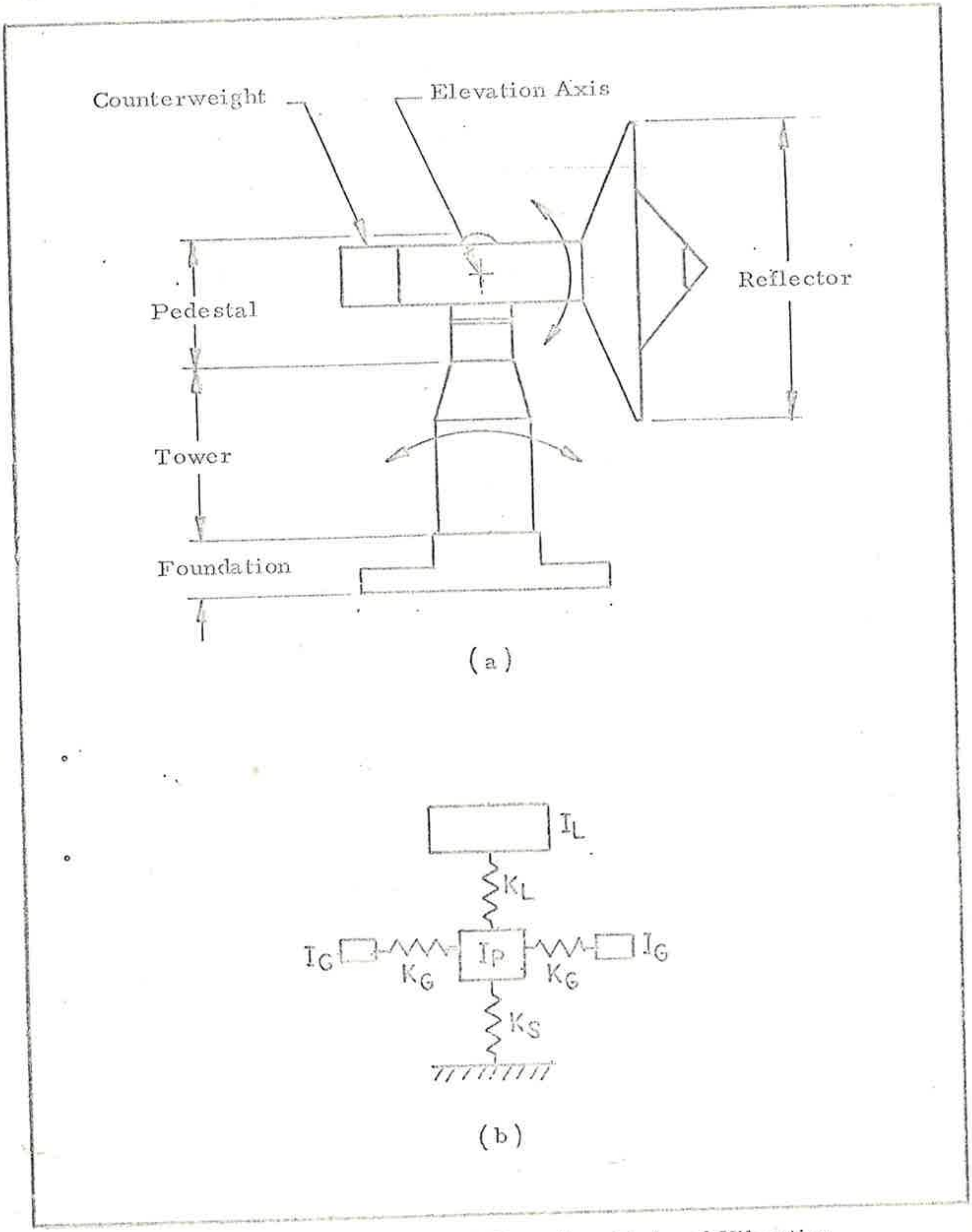

Figure 3-34 Symmetrical Elevation Mode of Vibration

sented by I_L and the stiffness of the load structure by the spring K_L. I_G is the inertia of the drive train reflected to the output axis and K_G is the stiffness of the drive train, I_P and K_S the inertia and stiffness of the pedestal structure. The values K_S and K_G are those calculated in the equations employed in the drive train and pedestal stiffness analysis.

The mode of vibration about the azimuth axis is modeled in the same way with values of inertias and stiffnesses appropriately selected for this mode of vibration.

An antisymmetrical mode of vibration about the elevation axis is also possible as shown in Figure 3-35(a) and this mode is analyzed using the model shown in Figure 3-35(b). I_R is the inertia of the reflector about its focal axis and K_R is the associated stiffness. I_C is the inertia of the counterweights and supporting structure about the elevation axis. K_L is the torsional stiffness of the structural tie across the supporting arms which may be a torsion box included specifically to provide this stiffness or the torsional stiffness of the reflector backstructure.

The vibration modes just discussed are called free-free modes because the motor armatures are assumed to be capable of displacement relative to the pedestal structure.

Also needed for servo analysis are locked rotor modes of vibration in which it is assumed that the motor armature is fixed to the pedestal structure. The locked rotor natural frequencies in elevation and in azimuth are evaluated using the model shown in Figure 3-36. It is the same as the free-free model except that the drive train inertias are eliminated and the stiffness of the pedestal is represented by K_P, the value computed previously in the pedestal stiffness analysis. The antisymmetrical mode model is shown in Figure 3-37. In this model the inertia of the drive train is eliminated and half the stiffness of the pedestal acts on each of the supporting arms as shown in the figure.

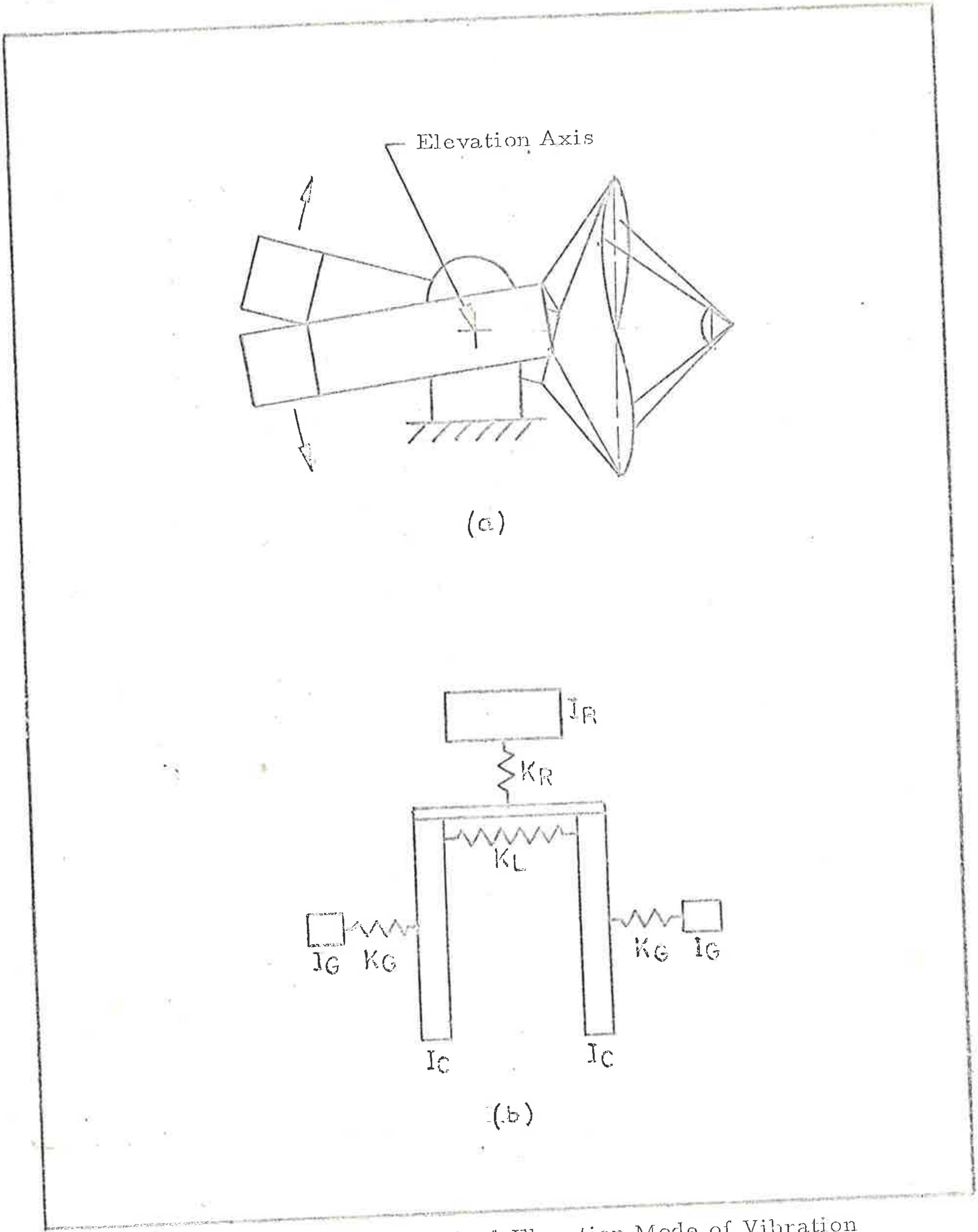

Figure 3-35 Antisymmetrical Elevation Mode of Vibration

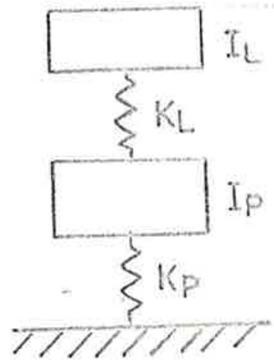

Figure 3-36 Locked Rotor Symmetrical Vibration

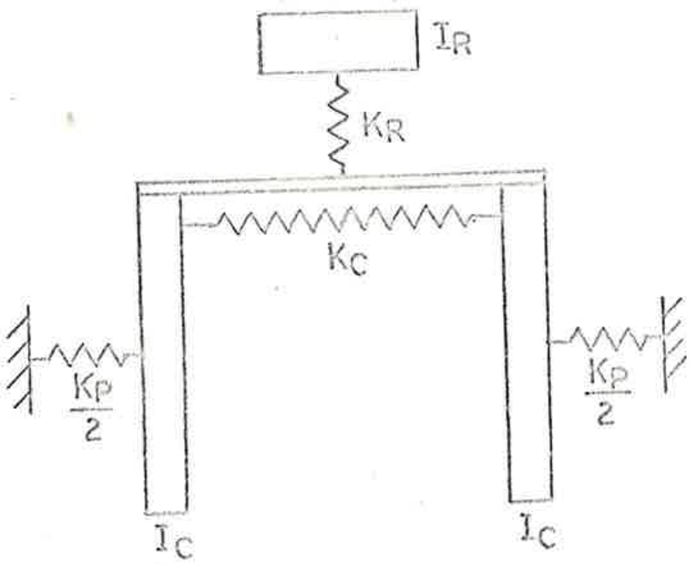

Figure 3-37 Locked Rotor Antisymmetrical Vibration

The natural frequencies for each model are determined by writing expressions for the total kinetic energy and potential energy of the system and using La Granges equation for a conservative system.

$$\frac{d}{dt}\left(\frac{dL}{d\theta_i}\right) - \frac{dL}{d\theta_i} = 0$$

$L = T - U$

θ_i = Angular displacement of i^{th} inertia

T = Kinetic energy of system

U = Potential energy of system

Performing the differentiation with respect to each displacement, θ_i, in the model yields n differential equations of motion where n is the number of degrees of freedom in the model. Assuming harmonic motion the solution of the n equations may be written in terms of ω^2 where ω is the circular frequency of the harmonic motion. If n is not too large, the n equations may be conveniently reduced by standard algebraic methods to a polynomial in ω^2. The roots of the polynomical are readily determined on ESSCO's time sharing computer facility.

3.3.2.2 Physical Characteristics

The preceding section presented the methods of mechanical and structural analysis that were used in the design of the ESSCO elevation over azimuth pedestals employed in the synergised antenna subsystem. In the sections which follow, a descriptive summary is presented of the major physical and mechanical features of this pedestal line.

Basically, the pedestal consists of a yoke type elevation axis assembly mounted on a king post azimuth axis assembly. A yoke-king post configuration was chosen because of its inherent rigidity and because it allows easy access to the electronics of the feed system located in the area behind the vertex of the reflector. Each of these assemblies is made up of rugged structural frame into which low friction bearings have been installed along with independently controlled drive systems. In addition, each assembly is equipped with a data package, stow lock, brake, handwheels, and limit packages.

The basic pedestal assembly for a 45 foot antenna is shown in Figure 3-38.

3.3.2.2.1 Elevation Axis Assembly

(1) Structure and Mechanical Arrangement - The elevation portion of the pedestal is a yoke type configuration as illustrated in Figure 3-39.

The yoke arm is a rigid box, fabricated of ASTM A-26 steel plate. A large diameter tube is welded to the center section of the cross-arm to capture the thrust and radial bearings. Those surfaces of the yoke arm, which are used to mount the gear boxes and pillow blocks of the elevation assembly, are machined to insure an accurate mounting arrangement. The complete yoke arm structure is stress relieved prior to machining. The elevation axis employs two pillow block bearing assemblies which are mounted on the yoke arm as shown in Figure 3-39. Each

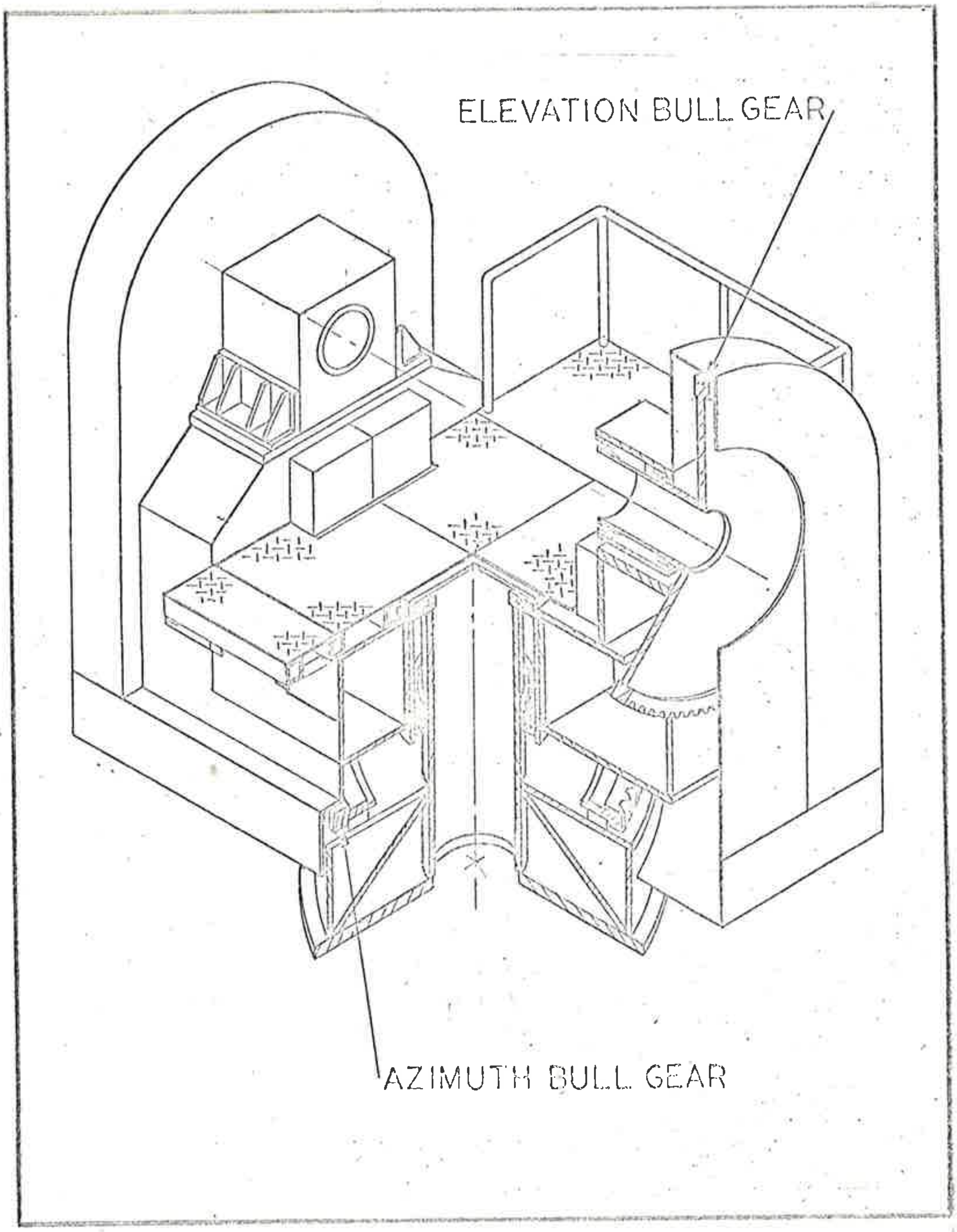

Figure 3-38 Elevation Over Azimuth Pedestal Assembly

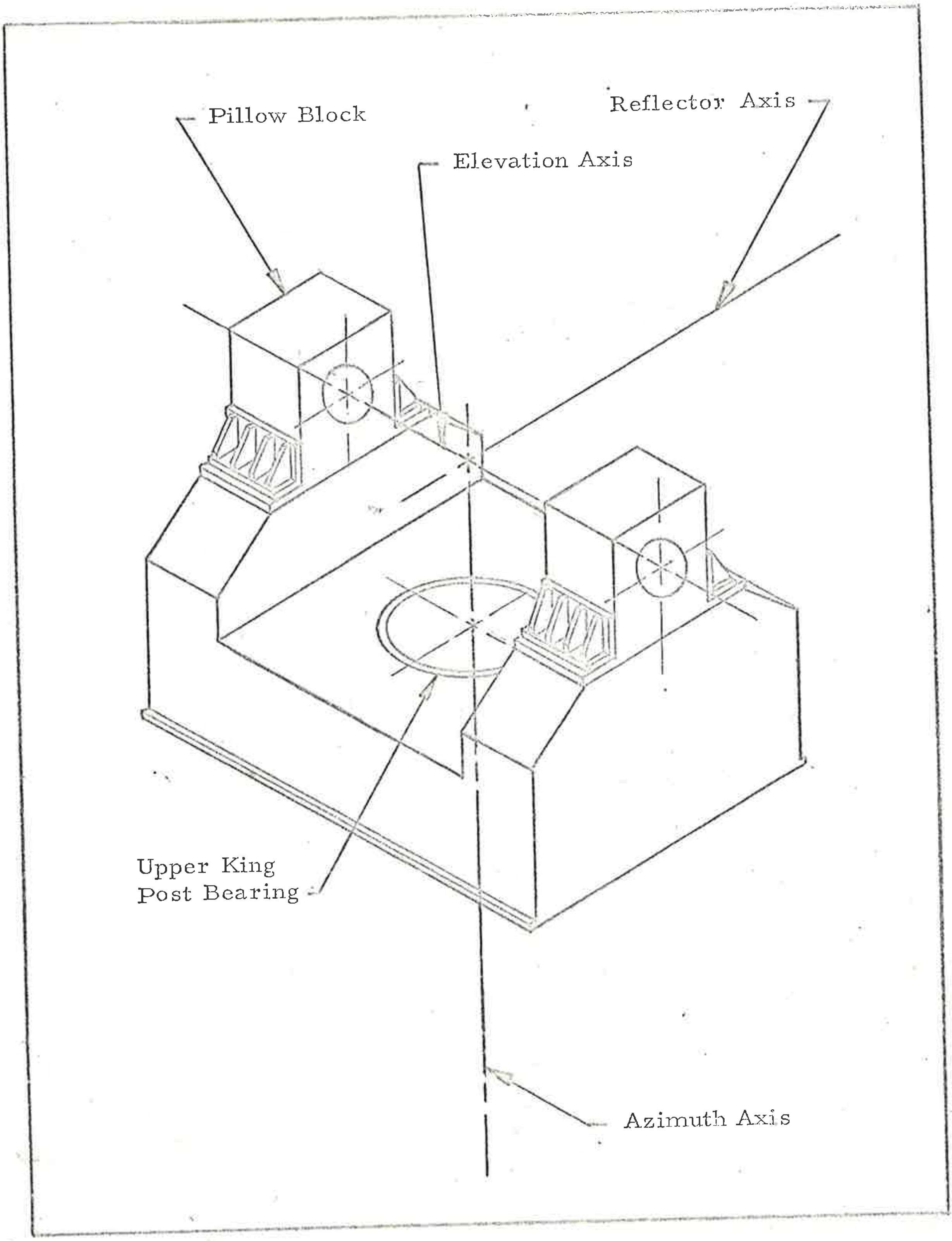

Figure 3-39 Elevation Yoke Structure

pillow block assembly consists of two tapered roller ball bearings which are pre-loaded into the pillow block housing. It will be noted that there is a substantial separation of the two tapered roller ball bearings in the pillow block housing which provides a very rigid arrangement. Each of the pillow block assemblies has a load carrying capacity well in excess of the combined loading it will experience under operating conditions. The elevation axis shaft passes through the pillow block bearings and extends to support a bull gear on each side of the elevation axis. The reflector support structure, including the counterweights, is attached directly to the outer side of the elevation bull gears. The elevation axis employs one or two bull gears as dictated by the accuracy, stiffness and economy requirements. In either case, two drive assemblies are used per bull gear. Each drive assembly is composed of a low speed section, a high speed section and a brake.

The elevation and azimuth axes employ the same drive and gear box assemblies providing commonality in the components.

One side of the yoke assembly is used to locate and support the gear driven data package. The data package contains the synchro transmitters, limit switches, sine-cosine potentiometer, and linear potentiometer. It should be noted that the yoke design provides the additional advantages of allowing personnel convenient access to the feed, and associated electronics mounted on the reflector directly behind the vertex.

In addition, all drive components and data equipment including wiring and interconnecting cabling for the elevation axis, are easily accessible from the yoke platforms. Removable guard rails are provided on the platforms.

As shown in Figure 3-38, ladders and walkways are provided on the elevation yoke assembly. These are removable assemblies which are in accordance with MIL-STD-1472. Access to the walkways is from a ladder with a removable safety cage which is permanently attached to the elevation yoke. The removable ladders and walkways remain in the "service use" position for all operational modes of the antenna system with the exception of the "plunge" condition.

Electrical interlock switches have been provided on the removable parts of the ladders and walkways. These interlock switches prevent the initiation of the "plunge" condition in elevation from the antenna control console until the removable sections have been dismounted.

The elevation motors, tachometer and brakes are all easily accessible from the walkways. The walkways also make more convenient the natural access provided by the yoke design to the area behind the reflector vertex where the feed and other electronic components are conveniently located.

(2) Elevation Drive Train - The drive train for the elevation axis employs either one or two bull gears as dictated by accuracy, stiffness and economy requirements. In either case, two drive assemblies are used with each bull gear. Each drive assembly, in turn, consists in general of a low speed section and a high speed section. A typical drive train of a pedestal for a 45 foot diameter antenna system is shown in Figure 3-40. In this case, the high speed section of the drive assembly has two stages and the electro-mechanical brake is mounted between stage 1 and stage 2.

The front mesh of the low speed section of the drive train consists of a large bull gear and pinion to give the largest possible reduction consistent with overall size of the pedestal and its load capacity. The design of this first mesh is very important in achieving the maximum stiffness and torque capacity of the pedestal. The pinion is integrally cast with the main gear in the next mesh of the low speed section of the drive.

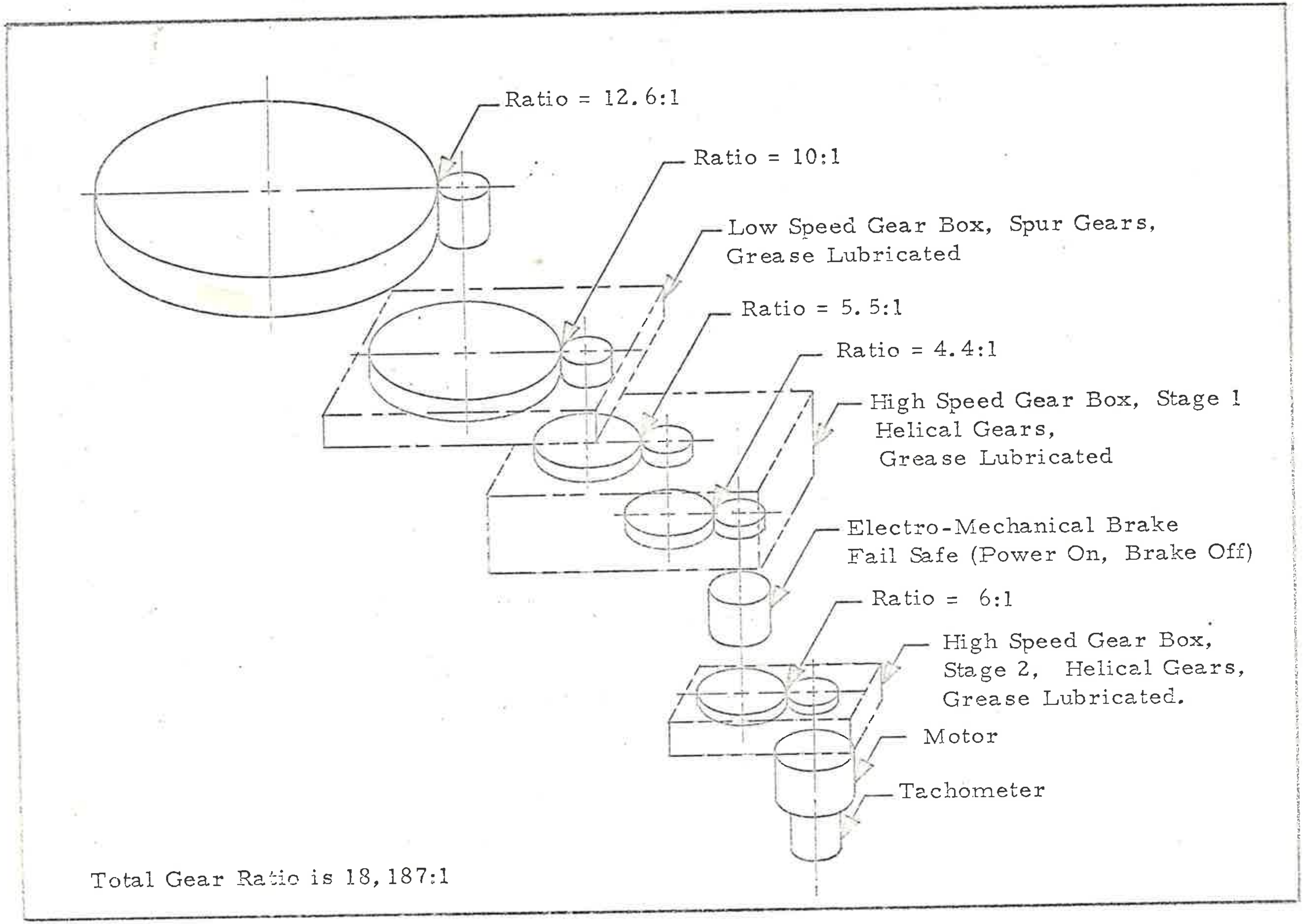

Figure 3-40 Typical Synergized El-Az Drive Train

In addition, the distance between the support bearing of the pinion and the bull gear is kept small to provide a minimum pinion overhang. This design approach is used to reduce to the lowest possible level the separating deflection between the pinion and the bull gear. The first two meshes comprising the low speed section of the drive train are standard for each of ESSCO's pedestal sizes. The low speed section of the pedestal drive is illustrated in Figure 3-41. Variations in the speed and motor requirements are made by an appropriate selection of the high speed section of the drive train. As shown in the typical example of Figure 3-40, two stages are used to meet the requirements for a 1 degree per second slew speed.

It will be noted that both stages of the high speed gear section utilize helical gears with grease lubrication. The gears are very hard steel with an excellent finish to reduce wear to a minimum. The use of grease lubrication in the gear boxes allows the elimination of oil seals which, in turn, results in a much lower drive train friction than is possible with the usual oil lubricated gear boxes.

During the assembly, the backlash of the gear trains is controlled so that the maximum backlash of the trains does not exceed ± 0.05 degrees. The gear boxes are pinned to the structure to insure that this backlash level will not change during subsequent operation. This modest backlash is allowed to insure that the friction remains low during operation. In order to provide a zero backlash system, the two drives per bull gear are biased so that the two drives oppose one another during operation.

(3) Data Package and Limit Switches - The elevation data package is driven by dual precision, Class I anti-backlash pinions from a 48 pitch precision data bull gear with a pitch diameter of 18 inches mounted on the elevation axis. Precision Class I anti-backlash gearing is used within the data package to drive 1:1 and 72:1 synchro torque transmitters (TX).

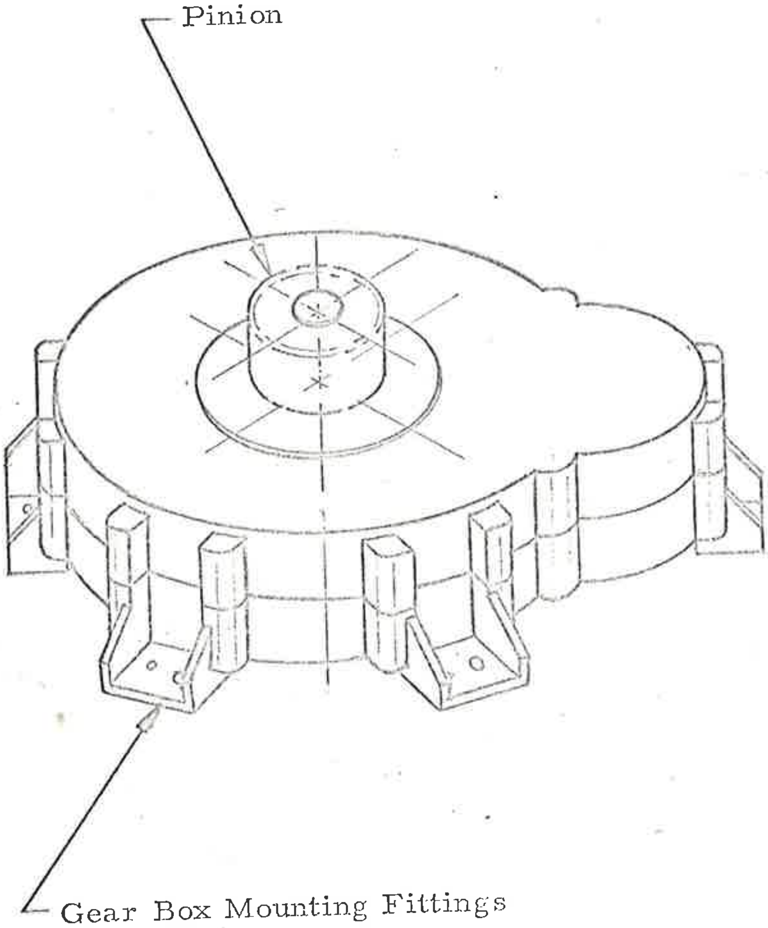

Figure 3-41 Low Speed Gear Box Elevation and Azimuth Drives

The synchros are size 23, 60 Hz and are connected to the data shafts with zero backlash couplings.

The data shaft position accuracy, including backlash repeatability, is within 0.5° for the 1:1 shafts and 0.03° for the 72:1 shafts (referred to the output axis).

Five electrical limit switches are included with the data package and are of the rotary type mounted on a 1:1 axis of the data gear box. Each switch is capable of handling a current of 100 ma and indicates a travel limit by an open circuit. The limits are adjusted and set in accordance with the prescribed limits.

The limit package has a prelimit position and a final limit position at each extremity of travel and an additional switch to indicate the quadrant of operation. The prelimit positions reduce the rotational speed of the elevation axis to approximately 10% of its rated value. The final limit switch dynamically brakes the motors.

(4) Brake - The optimum location of the brake in the elevation drive train is a function of capacity/cost of the brake, the inertia of the brake reflected to the output (1:1 point of the train) and the details of the mechanical mounting. For the typical pedestal drive system shown in Figure 3-40, the most desirable location of the brake turned out to be the first stage of the high speed section of the drive train. The brakes are designed to be fail safe devices in that they are applied in the event of a power failure.

(5) Mechanical Stops - Energy absorbing mechanical stops are provided at the final mechanical limit of the elevation axis. The two stops that are provided for each direction of rotation are hydro-mechanical type and are designed to intercept a specially provided member attached

to the reflector support structure in the event that the electrical limits fail. These stops are capable of absorbing the kinetic energy associated with the maximum rotational speed of the antenna and associated drive train inertias in addition to the maximum torque of the motors.

(6) Stow Lock - A stow lock has been provided for stowing the antenna in the zenith position. The capacity of the stow lock is given in Table 1-4. The stow pin is actuated manually by means of a lever. To insure safety, two limit switches are provided on the stow pin assembly. One limit switch is located at the fully locked position (stow limit) and the other at an intermediate position ("unstow" limit) prior to any mechanical interference. These limit switches can be used as follows: When the "stow" limit switch is activated an indicator marked "Stow" is lighted on the control panel; when the "Unstow" limit switch is activated the indicator marked "Unstow" is turned off and the motors and the servo system is de-energized; when the stow pin is completely retracted the indicator marked "Unstow" on the control panel is lighted and the motors and servo system are energized.

(7) Handwheel - A handwheel has been supplied on the elevation axis. The handwheel is adjacent to the motor mounting and is easily accessible. The handwheel is used to position the elevation axis manually in the event of a power failure. A slip clutch is provided in the handwheel to limit the torque that may be applied. In addition, the handhweel is spring loaded so that an axial hand thrust load is required to engage the handwheel gearing. When the handwheel is engaged, a safety interlock is activated, which de-energizes the power to the pedestal. The handwheel automatically disengages when the hand is taken from the handwheel.

3.3.2.2.2 Azimuth Axis Assembly

(1) Structure and Mechanical Arrangement - As previously

noted the azimuth is of the king post design. Basically, the king post is a stationary steel cylinder supported by a base structure assembly as shown in Figure 3-42. The upper and lower king post bearings are spaced sufficiently apart and the wall of the king post cylinder is sufficiently thick so that it contributes less than 15% of the total pedestal compliance.

The upper bearing is an X-roller type which can support radial, axial, as well as overturning loads. The lower bearing is a radial roller type bearing which can support only radial loads. Thus, the upper bearing reacts all of the thrust loading and the radial and overturning loads are reacted by the two bearings acting as a unit through the king post. Both of these bearings are grease lubricated and are covered to prevent the entry of dust and dirt.

The azimuth bull gear is fixed, being attached to the base structure assembly as shown in Figure 3-42, and its dimensions are identical to the elevation bull gear. This arrangement requires that the azimuth drive train rotate with respect to the fixed base.

(2) Azimuth Drive Train - As noted above, the azimuth drive train assemblies are identical to the elevation drive train assemblies thereby providing interchangeability. Either two or four complete drive trains employed in an anti-backlash mode are used for the azimuth axis depending upon accuracy, stiffness, and economy requirements. The load and stiffness capability of the azimuth axis is equal to or greater than that of the elevation axis.

(3) Data Package and Limit Switches - The azimuth data package is driven from the azimuth bull gear with a 10 inch diameter - pitch, anti-backlash pickoff gear. Precision Class I anti-backlash gearing is used within the data package to drive 1:1 and 72:1 synchro torque transmitters (TX). The synchros are size 23, 60 Hz and are connected to the data shafts with zero backlash couplings.

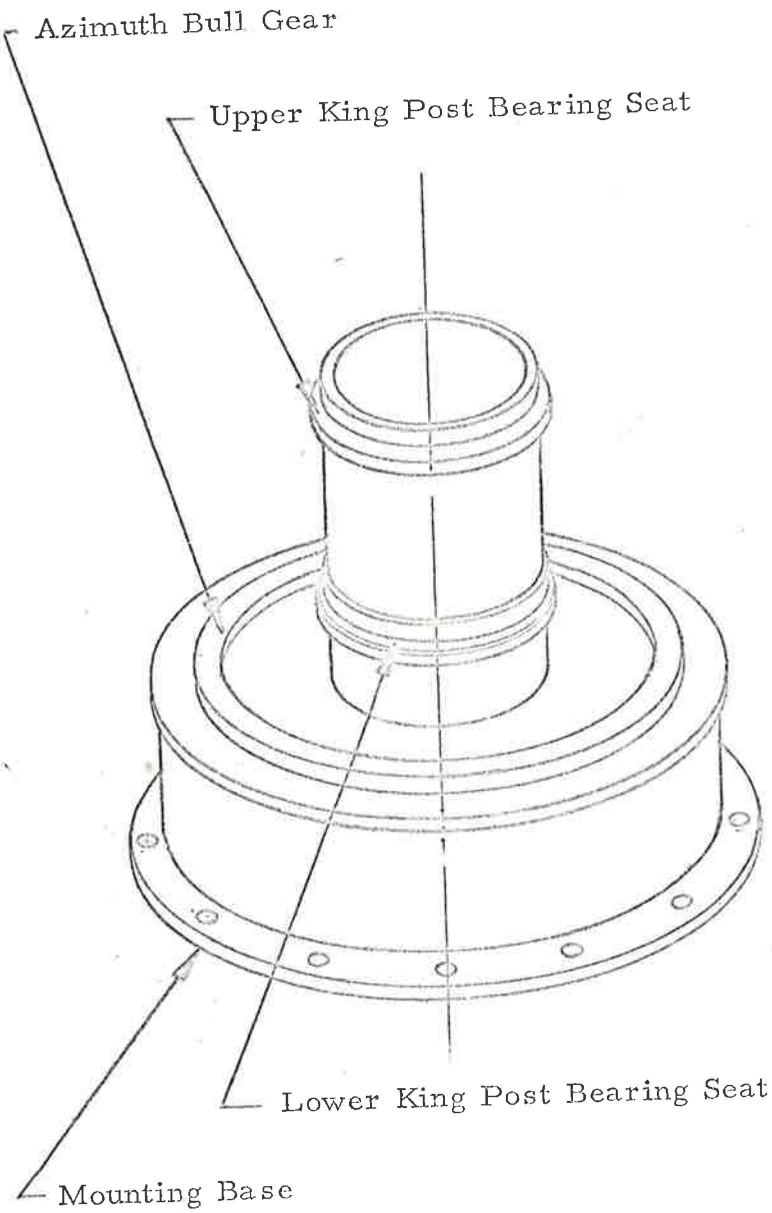

Figure 3-42 Azimuth King Post Structure

The data shaft position accuracy, including backlash repeatability is within 0.5° for the 1:1 shafts and 0.03° for the 72:1 shafts (referred to the output axis).

The azimuth limit package consists of 4 rotary switches mounted on a 1:3 axis of the data gear box. Each switch is capable of handling a current of 100 ma and indicates a travel limit by an open circuit. The limits are adjusted and set in accordance with the prescribed limits.

The limit package has a pre-limit position and a final limit position at each extremity of travel. The pre-limit positions are used to limit the rotational speed of the azimuth axis to approximately 10% of its rated value. The final limit switch dynamically brakes the motors.

(4) Brake - The brake for the azimuth axis is identical to the one described for the elevation axis.

(5) Stow Lock - The stow lock provided for the azimuth axis is the same as that described for the elevation axis. The actuation lever for the azimuth stow lock is located on the yoke cross arm between the pillow block supports.

(6) Handwheel - The handwheel supplied for the elevation axis is identical to that described for the elevation axis.

(7) Cable Wrap - The cable wrap for the azimuth axis consists of two plates, each of which has 12 holes approximately 2" in diameter, through which the cables are threaded. The upper plate is attached to the elevation yoke platform and the lower plate is attached to the base of the azimuth king post. Thus, the upper plate moves while the lower plate remains fixed. The cables are threaded through the holes in the upper plate, down through the king post, and out the holes in the lower plate. A clamp is attached to the cables on the upper plate fixing them in position. Rollers are provided on the lower plate to allow the cables to move freely without chaff-

ing as the elevation yoke is rotated. Upon rotation of the elevation yoke, the cables are drawn up through the lower plate and wrapped around a support tube within the azimuth king post.

The cables are provided with a large loop below the base of the pedestal to allow free motion of the cables during azimuth rotation.