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CR - 152193

TECHNICAL REPORT

Final Report No. C4952-1

FEASIBILITY AND CONCEPT STUDY
TO CONVERT THE NASA/AMES VERTICAL MOTION SIMULATOR
TO A HELICOPTER SIMULATOR

By

C.A. Belsterling
R.C. Chou
E.G. Davies
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Franklin Research Center
(formerly Franklin Institute Research Laboratories)

Prepared under Contract No. NAS2-9884

September, 1978

for

National Aeronautics and Space Administration



(NASA-CR-152193) FEASIBILITY AND CONCEPT
STUDY TO CONVERT THE NASA/AMES VERTICAL
MOTION SIMULATOR TO A HELICOPTER SIMULATOR
Final Report (Franklin Inst. Research Labs.)
150 p HC A07/MF A01

N80-16070

Unclas
44854

CSCL 14B G3/09



Franklin Research Center

A Division of The Franklin Institute

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1. INTRODUCTION

1.1 Background

The National Aeronautics and Space Administration (NASA) has previously designed and is currently completing the construction of an advanced Vertical Motion Simulator (VMS) at Ames Research Center. It has been fitted with a six-degree-of-freedom (6 DOF) platform to provide the longitudinal and angular motions required in the simulation of fixed-wing aircraft. The objective of the work described in this report is to develop the conceptual design for converting the Vertical Motion Simulator from an aircraft to a dual-purpose aircraft and helicopter simulator. This objective has been met most effectively through the creative design of a unique, high-performance, 4 DOF motion system to replace the present 6 DOF platform.

1.2 Description of the Vertical Motion Simulator (VMS)

The VMS now nearing completion at NASA/Ames is illustrated in Figure 1. It consists of a 6 DOF synergistic motion system mounted on a structure with a lateral displacement capability of 40 feet, in turn mounted on a vertical-motion platform capable of 60 feet of vertical displacement. The VMS is housed in a tower addition to Building N-243, especially designed to be compatible with the original VMS. Vertical and lateral motions are powered with DC motor drive systems and the 6 DOF platform is powered with electrohydraulic-actuators supplied through two symmetrical catenaries attached to the tower walls.

1.3 Design Constraints

In the conceptual design of the modification of the VMS to accommodate the requirements for helicopter simulation at least four important constraints must be considered as follows:

- Building envelope

The existing building was constructed with certain special features to integrate with the fixed-wing VMS requirements. For helicopter simulation longitudinal and angular motions are greater, and the cockpit must be enclosed in a 6.25 m (20.5 ft.) OD sphere attached to the moving platform. Therefore, the building envelope imposes a severe limitation on the new motion system kinematics.

- Weight

The vertical and lateral drives of the VMS were originally designed to provide the required performance with a certain calculated gross weight on the lateral carriage. Although there was provision for adding drive power, further addition of gross weight could compromise vertical and lateral performance. Therefore weight of the new 4 DOF motion system is a second severe design constraint.

- Interchangeability

Although interchangeability between the 6 DOF platform for fixed-wing simulation and the new 4 DOF system for helicopter simulation is allowed in the NASA plan, it would add a large element to operating cost. It would include not only the manpower cost in making the conversion but also reflected costs of unavailable VMS simulation time. Therefore the only acceptable concept for VMS modification must accommodate both fixed-wing and helicopter simulation requirements.

- Cost

Finally there is a limitation to the amount of funds available to complete the conversion of the VMS to a helicopter simulator. The use of exotic materials for construction and for building special drive machinery might alleviate the effects of some other constraints. However, funding would not be adequate to support applied R&D or the use of unproved techniques and components.

1.4 Design Study Approach

To satisfy the objectives of the required conceptual design study within the constraints described above, we have made a systematic approach to the definition of the most suitable conceptual design. We first consider the motion and visual display requirements for both fixed-wing and helicopter simulations including the effective loads. A total of 5 kinematic arrangements are initially considered. Three are eliminated at an early stage due to the severe envelope, weight or cost constraints. Two configurations are explored in more detail, considering tradeoffs in envelope, weight and actuator construction. The result is the selection of the most promising approach.

A tradeoff study on hydraulic versus electric drives is carried out, but it is limited at an early stage because of the severe weight constraint.

The remainder of the work is concentrated on confirming, through preliminary designs, that the selected concept, including all associated subsystems, is capable of meeting all objectives of the program most effectively.

The results are in the form of specifications for the VMS modification based on the selected conceptual design, preliminary designs of all subsystems and estimates of costs and time schedules for detail design.

2. DESIGN GOALS

2.1 Objectives

The objectives of the work to be covered by this Report are to develop the concept for the "best" 4 DOF motion system which

- will integrate with the two VMS translational motions
- can easily be converted from helicopter to fixed-wing simulation
- interfaces with an advanced visual display system
- makes maximum use of proven techniques, convenient materials and off-the-shelf components
- limits stresses on materials and components to well below their design limits
- provides continuous monitoring of critical components and easy access for maintenance and repair

2.2 Fixed-Wing Simulation

To satisfy the requirements for fixed-wing simulation, a 4 DOF motion system mounted on the VMS lateral carriage must meet the following requirements:

2.2.1 Motions (not simultaneous)

Mode	Displacement	Velocity	Acceleration
• Longitudinal	$\pm 0.76\text{m}$ (2.5 ft)	$\pm 0.61\text{ m/sec}$ (2 ft/sec)	$\pm 4.88\text{ m/sec}^2$ (16 ft/sec ²)
• Roll	$\pm 0.38\text{ rad}$ (22°)	$\pm 0.26\text{ rad/sec}$ (15°/sec)	$\pm 0.87\text{ rad/sec}^2$ (50°/sec ²)
• Pitch	+ 0.45 rad (26°) - 0.42 rad (24°)	$\pm 0.26\text{ rad/sec}$ (15°/sec)	$\pm 0.87\text{ rad/sec}^2$ (50°/sec ²)
• Yaw	$\pm 0.50\text{ rad}$ (29°)	$\pm 0.26\text{ rad/sec}$ (15°/sec)	$\pm 0.87\text{ rad/sec}^2$ (50°/sec ²)

2.2.2 Loads

- Gross Weight 3,632 kg (8,000 lb)
- Mom. of Inertia I_{xx} 2,939 n m sec² (26,000 lb in sec²)
- Mom. of Inertia I_{yy} 3,504 n m sec² (31,000 lb in sec²)
- Mom. of Inertia I_{zz} 3,504 n m sec² (31,000 lb in sec²)

2.2.3 Frequency Response

Based on the projected characteristics of the existing VMS, all motions should exhibit less than 0.52 rad (30°) phase lag at 9.42 rad/sec (1.5 hertz).

2.3 Helicopter Simulation

To satisfy the requirements for helicopter simulation, a 4 DOF motion system mounted on the VMS lateral carriage must meet the following requirements:

2.3.1 Motions (simultaneous)

Mode	<u>Simultaneous</u> Displacement	<u>Worst Combination of Two</u>	
		Velocity	Acceleration
• Longitudinal	$\pm 1.22\text{ m}$ (4 ft)	$\pm 1.22\text{ m/sec}$ (4 ft/sec)	$\pm 3.05\text{ m/sec}^2$ (10 ft/sec ²)
• Roll	$\pm 0.31\text{ rad}$ (18°)	$\pm 0.52\text{ rad/sec}$ (30°/sec)	$\pm 1.04\text{ rad/sec}^2$ (60°/sec ²)
• Pitch	$\pm 0.31\text{ rad}$ (18°)	$\pm 0.52\text{ rad/sec}$ (30°/sec)	$\pm 1.04\text{ rad/sec}^2$ (60°/sec ²)
• Yaw	$\pm 0.42\text{ rad}$ (24°)	$\pm 0.52\text{ rad/sec}$ (30°/sec)	$\pm 1.04\text{ rad/sec}^2$ (60°/sec ²)

2.3.2 Load

- Configuration 6.25m (20.5 ft) dia sphere (section)
- Gross Weight 5,448 kg (12,000 lbs)
- Mom. of Inertia I_{xx} , I_{yy} , I_{zz} Up to 40,718 nm sec² (30,000 lb ft sec²)

2.3.3 Frequency Response

- Less than 0.52 rad (30°) phase lag at 9.42 rad/sec (1.5 hertz)
- Characteristics of a second order system with natural frequency of 18.8 rad/sec (3 hertz) and damping factor of 0.7 up to 12.6 rad/sec (2 hertz)
- Tolerances on matching this response shall be ± 2 db in amplitude and $\pm 20\%$ in phase.

2.4 Cab/Visual System Envelope

The payload on the moving 4 DOF system is essentially a section of a 6.25m (20.5 ft) OD sphere enclosing the cab and visual systems. The preliminary layout as applied in the conceptual design studies reported here is shown in Figure 2. The estimated gross weight is 5,448 kg (12,000 lb) with an inertia up to 40,718 nm sec² (30,000 lb ft sec²).

2.5 Building Envelope

The VMS is housed in a specially-designed building originally intended to accommodate fixed-wing simulations. The building and the major structural components are shown in Figure 3 excluding the 6 DOF platform on the lateral carriage. Note that the allowable operating envelope is limited to less than 9.46m (31 ft) in the longitudinal direction due to the projections of the loading ramp and the east wall vertical track. With a 6.25m (20.5 ft) spherical envelope around the cockpit and a required longitudinal displacement of ± 1.22 m (4 ft), this limitation imposes one of the most severe constraints on the new motion system design.

3. KINEMATIC CONCEPT DEVELOPMENT

During the course of the design studies covered in this program, five different kinematic concepts were evaluated. The first three to be described here required only a brief investigation before being discarded as unsuitable for the application. The remaining two were considered in further detail before final selection.

3.1 Six DOF Synergistic Platform

Clearly the most flexible and proven concept is the 6 DOF synergistic platform illustrated in Figure 4. It requires the movement of the minimum of dead weight, therefore is capable of the highest performance with minimum power. Using the mathematical model previously developed when the Franklin Research Center (FRC) invented this motion system, the length of stroke of the actuators necessary to provide the simultaneous displacements required for helicopter simulation (Para. 2.3.1) were calculated. The result was 3.66m (12 ft). This in itself represents an unrealistic design condition. Further calculations and design layouts indicated that, in certain failure modes, the spherical visual system envelope would contact the building walls. Therefore the 6 DOF synergistic platform was eliminated as a candidate concept.

3.2 Air Bearing Sphere

Considering the major constraints of operating envelope and minimum weight, the concept of supporting the spherical visual system on air bearings at its outer diameter as illustrated in Figure 5 is a most appealing concept. It eliminates the need for cascaded gimbals and minimizes longitudinal overtravel. The technology is an advanced one but FRC has been at the leading edge for many years. Based on this experience, calculations of the size and layout of the air pads were made which determined how the loads must be transmitted through the spherical shell. The results indicated the need for increasing the weight far beyond the estimated amount (Para. 2.4). The location of this additional mass raised the estimated inertia to an unrealistic $51,575 \text{ nm sec}^2$ ($38,000 \text{ lb-ft-sec}^2$). The excessively high drive power required to achieve the required performance, plus the anticipated difficulties in machining and maintaining such large spherical surfaces, led to the early elimination of this concept from further consideration.

3.3 Gimballed Four DOF

The mechanically-gimballed 3 DOF cockpit on a separate translating carriage as shown in Figure 6 is a familiar concept that also minimizes longitudinal overtravel. It has been proven in many years of service in the FSAA Simulator at NASA/Ames. Applied to helicopter simulation on the VMS, it requires some exceptionally-large structures to accommodate the 6.25m (20.5 ft) diameter spherical envelope. The weight of the gimballed system was estimated at an excessive 12,712 kg (28,000 lb) for the structure, not including the weight of the drive systems, and the location of the additional mass raised the estimated inertia to $47,504 \text{ nm sec}^2$ ($35,000 \text{ lb ft sec}^2$), compromising drive system performance and power requirements. Therefore the mechanically-gimballed system concept was eliminated because of the weight constraint.

3.4 Synergistic Four DOF Platform

A new kinematic arrangement was conceived to maintain the minimum weight benefit of the 6 DOF platform and to control the overtravel displacement. The concept is illustrated in Figure 7. The moving platform is restrained from any lateral motion by the hinged center post. It is limited to a small vertical displacement by the fixed height of the center post. It is free to move in the remaining four degrees of freedom as driven by the four actuators. Overtravel can be limited by means of mechanical stops on center post hinge and platform pitch angles.

The mathematical model was developed for the new 4 DOF synergistic system and programmed for computer simulations. Calculations showed that the concept can be the most efficient kinematic arrangement for producing 4 DOF motions. However, because all motions are produced by the 4 actuators from a common reference (as they are in the 6 DOF synergistic system), the actuators must have exceptionally long strokes. To accommodate both fixed-wing and helicopter simulations on the VMS, these strokes are approximately 4.58M (15 ft). This results in an impractical cylinder design.

3.5 Three DOF Platform on a Longitudinal Carriage

To overcome the handicap of exceptionally-long actuators, a second new kinematic arrangement was conceived as illustrated in Figure 8. Here the longitudinal motion is provided by a translating carriage supporting a 3 DOF platform. The platform is restrained from any translations by a fixed center post. Only angular motions are driven by the three canted actuators thereby limiting the required strokes of the actuators. Overtravel is conveniently limited by mechanical stops and there is no kinematic coupling between translatory and angular motions.

The mathematical model for the 3 DOF platform was developed and programmed for computer simulations. Calculations showed that the maximum actuator stroke to accommodate both fixed-wing and helicopter simulations is 1.53m (5 ft), a reasonable length for practical design. Preliminary estimates showed that the concept can be constructed within the weight limitation of 11,804 kg (26,000 lbs). Preliminary layouts showed that the operating envelope remains safely within the building envelope without modifications.

3.6 Selection of Concept for Design Study

Having rejected the first three concepts on gross violations of design constraints, only the last two need be considered in greater detail. Both the 4 DOF synergistic platform and the 3 DOF platform on a carriage can be designed to fit within all of the constraints. The tradeoffs are found in weight, ease of construction and complexity of control. Since the weight restriction can be met with the more easily controlled 3 DOF concept and the construction of the long actuators for the 4 DOF concept

would be very difficult, the former is clearly the most suitable concept for the VMS modification.

In summary, the selected concept using a 3 DOF platform on a longitudinal carriage, as illustrated in Figure 8, has the following outstanding features:

- operates within the existing building envelope
- can be built within the specified weight limitations
- provides both fixed-wing and helicopter simulations without mechanical changes
- capable of outstanding dynamic response with reasonable power
- minimizes coupling between motions
- uses proven operating principles and readily-available components
- simple and reliable kinematic design
- can be built for relatively low cost

4. DRIVE SYSTEM EVALUATIONS

In the course of the early design studies, various types of power drive systems were considered as follows.

4.1 Electric Motor With Rack & Pinion

The rack and pinion driven with an electric motor was considered for the longitudinal drive primarily because of its similarity to the lateral and vertical VMS drives. Power requirements were estimated at 60 horsepower. To provide smooth motion at very low velocities, it is necessary to provide a separate speed reducer. The preliminary estimates on weight of this system are:

• 60 hp motor	345 kg (760 lbs)
• speed reducer	118 kg (260 lbs)
• rack & pinion	<u>68 kg (150 lbs)</u>
Total relative weight	531 kg (1170 lbs)

Also considered were the mechanical problems of maintaining the proper mesh between rack and pinion and the acoustical problems inherent to running gears.

4.2 Electric Motor With Ball-Screw Actuator

The electrically-driven ball-screw actuator is a drive concept that is also similar to proven equipment in service at NASA/Ames. Preliminary calculations indicated the need for linear velocities that exceed the accepted limits on ball velocity and screw pitch. Power requirements for platform drives were estimated at 75 hp. Again a speed reducer would be required for smooth motion at low velocity. Weight estimates amount to

• 75 hp motor	341 kg (750 lb)
• actuator	272 kg (600 lb)
• speed reducer	<u>136 kg (300 lb)</u>
Total relative weight	749 kg (1650 lb)

4.3 Linear Induction Motor

Linear Induction Motors (LIM) as a longitudinal drive has a number of attractive features. It eliminates the need for a rack and pinion, a speed reducer or any other mechanical coupling except bearings. The thrust can be generated in the structural material in the moving base and be distributed over a wide area to minimize stresses. Based on LIM designs previously used by FRC in vehicle propulsion, the estimated weight of the motors is 2,361 kg (5,200 lb).

4.4 Hydraulic Motor Drives

Rotary hydraulic motors were considered with both rack-and-pinion and ball-screw actuation. The motors themselves are smaller and lighter than equivalent electric motors but the actuation is subject to the same critical comments as before.

4.5 Electrohydraulic Actuators

Electrohydraulic actuators are most widely used in the design of high-performance motion systems and have proved to be very reliable and maintenance-free. FRC has over 100 man-years of experience in designing and building these systems. They eliminate the need for rack and pinions or ball-screws and provide a direct-acting force for a minimum of weight. For the longitudinal motion, the estimated electrohydraulic drive weights were

• cylinder	279 kg (615 lb)
• servovalve	45 kg (100 lb)
• accumulators	<u>91 kg (200 lb)</u>
Total relative weight	415 kg (915 lb)

4.6 Selection of Drive System

Tradeoffs to be considered in the selection of the drive system for the new 4 DOF conceptual design are primarily mechanical complexity and weight. All those evaluated, with the possible exception of the LIM drive, have been well-proven and components are readily available. Also considered was the availability of maintenance and operating skills available to the VMS facility.

In review of the foregoing evaluation of the various drive systems, it is clear that the electrohydraulic actuator is the simplest, lightest and most maintenance-free. The existing 6 DOF platform on the VMS is hydraulically powered and the necessary operating and maintenance skills are readily available. Therefore electrohydraulic drives have been selected for all motions in the new conceptual design. Guided by the exceptional experience of FRC personnel, special cylinders will be designed with equal area pistons to minimize servo-valve requirements and provide for smoother control. Servo valve third stages will also be designed with proprietary porting arrangements which will prevent the occurrence of the "hydraulic bump" phenomena.

5. LONGITUDINAL SYSTEM DESIGN

5.1 Calculation of Loads

All preliminary calculations indicate that the gross weight constraint for the new 4 DOF conceptual design can be met. The calculation of loads on the longitudinal system is based on the total amount of 11,804 kg (26,000 lbs). This provides an initial "cushion" due to the weight of the parts to be fixed to the lateral carriage.

5.2 Mechanical Layout

The mechanical design of the longitudinal system is based on a slide mechanism driven with a hydraulic cylinder as illustrated in Figure 9. Ball bearings have been chosen to provide the smoothest motion and lowest noise. The base structure is to be attached directly to the box section of the existing VMS lateral carriage to minimize weight and match platform height to the existing access ramp. The hydraulic cylinder is mounted beneath the moving carriage but is accessible for service or removal when the carriage is extended. Preloaded clevis pins are used to eliminate backlash. The servovalve is mounted directly on the cylinder for close coupling and maximum hydraulic stiffness. Accumulators located close to the actuators satisfy the peak demands of the motion systems, minimize pump capacity and provide emergency system power.

5.3 Mechanical Details

Figure 10 shows the details of the bearing section of the longitudinal system.

Two rows of 0.032m (1 1/4") dia. steel balls spaced by plastic cages are captured between hardened steel upper and lower races. The unique feature of this design is that the bearings are preloaded internally by controlling the tolerances. The preload is contained within the bearing unit and will not reflect to the structures. Therefore the preload will be kept at the controlled value regardless of structure deflections, which are relatively large for the bearing clearance. Also, because of this arrangement, the bearings can be fabricated and bench tested as a single unit before installation.

5.4 Hydraulic Cylinder Design

The hydraulic cylinder is specially-designed to provide equal piston areas in both directions. This insures balanced peak forces and hydraulic gains, and eliminates the need for excessive flow through the servovalve. A layout of the cylinder is shown in Figure 11 (Dwg No 2824).

The total length of conventional equal piston area cylinders requires two and half times the total stroke plus the dead length of the connecting eyes, glands and piston. The excessive length presents problems in space requirements and large lateral deflections.

The presented design "folds" the pressure area for extension stroke inside of the piston rod and saves about 50% of the total length. The only disadvantage of this design is that a rather large amount of fluid is trapped inside the cylinder which tends to decrease the hydraulic stiffness and the natural frequency of the total system.

The hydraulic stiffness of this design is estimated at $10.5 \times 10^6 \text{ n/m}$ ($6 \times 10^5 \text{ lb/in}$) which is adequate for specified performance requirements.

5.5 Cylinder and Servovalve Sizing

With reference to Para. 2.2.1, the maximum longitudinal acceleration is 4.88 m/sec^2 (16 ft/sec^2) for the fixed wing case. However, the worst-case loading is due to the pitch motion with a maximum acceleration of 1.04 rad/sec^2 ($60^\circ/\text{sec}^2$) and maximum velocity of 0.52 rad/sec ($30^\circ/\text{sec}$) coupled with 3.05 m/sec (10 ft/sec) longitudinal acceleration for the helicopter case.

Assuming the center of mass of the rotational load [weight 6,810 kg (15,000 lb) and I_{xx} of $45,212 \text{ nm sec}^2$ ($400,000 \text{ lb in sec}^2$)] is located 1.097m (43.2 in) above the platform pivot point (see Figures 2 and 8), plus dead weight of longitudinal carriage of 10,000 lb, the longitudinal

acceleration due to pitch motion coupled with a longitudinal acceleration of 3.05 m/sec^2 (10 ft/sec^2) requires a longitudinal cylinder force of $10,442 \text{ kg}$ ($23,000 \text{ lb}$). With a $2.07 \times 10^7 \text{ n/m}^2$ ($3,000 \text{ psi}$) supply pressure and an assumed $5.31 \times 10^6 \text{ n/m}^2$ (770 psi) drop across the servovalve, the piston area should be approximately $6.65 \times 10^{-3} \text{ m}^2$ (10.31 in^2).

The worst-case maximum velocity is the 1.22 m/sec (4 ft/sec) longitudinal motion. The flow rate is 1.22 m/sec (4 ft/sec) $\times 6.65 \times 10^{-3} \text{ m}^2$ (10.31 in^2) = $8.11 \times 10^{-3} \text{ m}^3/\text{sec}$ ($495 \text{ in}^3/\text{sec}$) or 128 gpm .

5.6 Control System Design

With the definition of longitudinal system loads and component sizing, the preliminary design of the control system can proceed. Transfer functions were derived for the actuator/load, the two-stage commercial servovalve and the specially-designed third stage spool. Following previous experience, a position loop is closed around the three-stage servovalve to extend its bandwidth to an acceptable frequency. The remainder of the complete closed-loop control system is then filled-out as shown in Figure 12.

The position loop is closed to provide a stable reference and to respond to position commands as required. The velocity loop is closed to provide effective damping and to allow for direct response to velocity commands. The acceleration loop is closed to insure smooth, low velocity tracking and to compensate for structural resonances, if necessary. A feedforward path is also provided to enhance the dynamic response of the closed-loop system. The numbers in the transfer functions have been calculated from the drive requirements, the characteristics of the components and the computer studies of the frequency response of the longitudinal system.

5.7 Predicted Frequency Response

The mathematical model of the longitudinal control system as represented in Figure 12 was programmed for simulation on FRC's PACER 100 hybrid computer facility. Parameter studies were run, comparing the predicted frequency response with the requirements of Para. 2.3.2. The results compared with those requirements are shown in Figure 13.

The calculated closed-loop phase shift at 9.42 rad/sec (1.5 Hz) is -0.43 rad (25°), which is 17% less than the 0.52 rad (30°) specified. The calculated amplitude is within the $\pm 2 \text{ db}$ allowed by the specification, up to 12.6 rad/sec (2 Hz). The phase margin and the gain margin for system stability of the position servo are 1.08 rad (62°) and 12 db , respectively, which represents a safety factor of better than 4. This is more than adequate to insure stability in cases of variation of load and minor malfunctions.

The results of the simulation studies confirm that the frequency response requirements for the longitudinal control system can be met and exceeded with the proposed conceptual design.

6. ROTATIONAL 3 DOF SYSTEM DESIGN

6.1 Calculation of Loads

Because of the coupling between cylinders driving the 3 DOF platform, it is not a simple matter to calculate loads on the actuators and structural stresses. As in all our synergistic system designs, the mathematical model was developed and programmed for computer analysis to aid in this design. The model simulates the behavior of the 3 DOF platform under all possible combinations of conditions and computes structural stresses and actuator loads as well as cylinder accelerations, velocities, and displacements. The program searches for and prints out maximum values, as appropriate.

The dominate factor of the structural design for dynamic machinery such as flight simulators is often stiffness rather than stress. In the case of the center post design (Figure 20) which works as a cantilever beam to support all the lateral forces of the payload, it is designed to have a natural frequency above 87.9 rad/sec (14 Hz) and the stress is at a rather low level.

In the case of structural design, worst-case modes of interactive operation were explored to find maximum stresses. These are beyond the simultaneous motion requirements of Para. 2.3.1. In the structural design, a safety factor of 3 has been applied on all members and components which will be retained in the final design as long as the weight constraint continues to be met.

6.2 Mechanical Layout of 3 DOF Platform

The kinematics of the 3 DOF are best illustrated with the photographs in Figures 14 through 18. A general layout is shown in Figure 19.

The hydraulic cylinders are attached to the base and to the moving platform with universal joints having preloaded clevis pins to eliminate lost motion. The platform is mounted on the center post with a ball bearing on top of a universal joint, decoupling yaw motion from pitch and roll. The servovalve is mounted directly on the cylinder for close coupling and maximum hydraulic stiffness. Accumulators located close to the actuators satisfy the peak demands of the actuators, minimize pump capacity and provide emergency shutdown power.

6.3 Mechanical Details of the 3 DOF Platform

Figure 20 shows further detail on the design of the center post. It is a simple conical section with a 3 DOF joint at the top and an attachment flange at the bottom. It will be fabricated of aluminum welded plate stock and bolted to the longitudinal platform to avoid welding distortions and alignment problems.

A special universal joint is designed to connect the center post and rotational platform; it allows the platform to rotate in all required roll, pitch and yaw motions but supports the loads and provides restraint in all the translational motions; longitudinal, lateral and heave.

The rotational platform is designed to support the universal cab (Figure 2) and with provision for mounting the 6.25m (20.5 ft) dia. visual display screen. It will be fabricated of welded aluminum box sections (two channels welded face to face) to provide stiffness and strength.

6.4 Calculation of Cylinder Force and Velocity

As in the case of load calculations for structural design, computer simulation of the mathematical model of the 3 DOF platform has been a valuable aid in determining normal cylinder forces and velocities. Because of the nonlinearities in the platform kinematics and the many possible combinations of motions, the determination of maximums required a comprehensive series of sensitivity studies. For example, Figure 21 shows a plot of maximum cylinder velocity as a function of yaw commands at several operating points in roll.

Enough of these manual searches have been completed to establish preliminary design values for maximum cylinder force, velocity and displacement. We have

- displacement 1.52m (60 inches)
- velocity 1.07 m/sec (42 in/sec)
- force 83,660n (18,800 lb)

6.5 Cylinder and Valve Sizing

The maximum cylinder force required for platform motion is calculated to be 83,660 n (18,800 lb). A piston area of $5.35 \times 10^{-3} \text{ m}^2$ (8.30 in^2) is chosen to provide the necessary force from the common supply pressure of $2.07 \times 10^7 \text{ n/m}^2$ (3,000 psi) less an assumed $5.07 \times 10^6 \text{ n/m}^2$ (735 psi) drop across the servovalve.

The worst-case velocity is defined as 12.8 m/sec (42 in/sec). With the $5.35 \times 10^{-3} \text{ m}^2$ (8.30 in^2) piston area, the maximum flow rate is $5.72 \times 10^{-3} \text{ m}^3/\text{sec}$ ($349 \text{ in}^3/\text{sec}$) or 90 gpm.

6.6 Control System Design

With the definition of the loads, cylinders and servovalves, the design of the control system for the 3 DOF platform drives proceeded. Transfer functions were defined for the actuator/load and servovalves, and a position loop closed around the third stage spool to extend its bandwidth. The remainder of the control system components are defined in Figure 22.

The position loop is closed to provide a stable reference and to respond to position commands as required. The velocity loop is closed to provide effective damping and to allow for direct response to velocity commands. The acceleration loop is closed to insure smooth, low-velocity tracking and to compensate for structural resonances, if necessary. A feedforward network is also provided to enhance the dynamic response of the closed-loop systems. The numbers in the transfer functions of Figure 22 are preliminary design values determined from component characteristics and parameter studies of the response of the system as simulated on our hybrid computer.

Note that the design of the control system for the 3 DOF actuators is far more complex than for the longitudinal actuator. This is due to the wide variation (156:1) in loads reflected onto a given actuator as the geometry changes.

To account for the wide variation of effective load mass on the actuator, acceleration is measured and fed back to the input to form a minor feedback loop as shown in Figure 22. With the proper acceleration feedback gain, the closed-loop characteristics of the minor loop will not be a function of actuator load mass and the overall closed-loop of the position servo will be insensitive to the variations in actuator loading. As a result (and an important one), all of the rotational servoactuators will have the same speed of response regardless of differences in load sharing and the resultant platform motion will be smooth and uniform.

6.7 Predicted Frequency Response

The mathematical models for two worst-case loading conditions were programmed for computer simulation and parameter studies were run comparing the predicted frequency response with the requirements of Para. 2.3.2. The results compared with those requirements are shown in Figure 23.

First note that the calculated closed-loop frequency responses exhibit no significant differences for cases when the actuator load mass is $2,234 \text{ kg sec}^2/\text{m}$ (maximum) or $14.3 \text{ kg sec}^2/\text{in}$ ($0.8 \text{ lb sec}^2/\text{in}$) (minimum).

Second, note that the calculated closed-loop phase shift at 9.42 rad/sec (1.5 Hz) is -0.47 rad (27°) which is 10% less than the 0.52 rad (30°) specified. The calculated amplitude response is also within $\pm 20 \text{ db}$ of the specified second order system up to 12.6 rad/sec (2 Hz). The phase margin and gain margin for system stability of the position servo are 1.0 rad (58°) and 8 db respectively. This represents a safety factor of better than 2.5 for fluctuations in the loop gain.

The results of the computer simulations confirm that, with proper choice of compensating techniques and networks, the frequency response requirements for helicopter and fixed-wing simulation can be met with the proposed conceptual design.

7. HYDRAULIC POWER SUPPLY

7.1 Longitudinal Drive System

The peak hydraulic power requirements for the longitudinal actuator have been defined in Para. 5.5 as $2.07 \times 10^7 \text{ n/m}^2$ (3000 psi) and 128 gpm. Assuming an average based on sinusoidal motion and close-coupled accumulators capable of providing the difference, the pump must supply $0.637 \times 128 = 82 \text{ gpm}$ (average).

7.2 Rotational Drive Systems

The peak hydraulic power requirements for each of the three actuators on the 3 DOF platform were defined in Para. 6.5 as $2.07 \times 10^7 \text{ n/m}^2$ (3,000 psi) and 90 gpm. A search of the computer printouts from simulated rotational maneuvers indicates that pure yaw near the limits requires the peak flow to all three actuators simultaneously for a total peak flow of 150 gpm. Assuming sinusoidal motion and close-coupled accumulators capable of providing the difference, the pump must supply the average of $0.637 \times 150 = 96 \text{ gpm}$.

7.3 Power Supply Design

With simultaneous longitudinal and yaw motions, the hydraulic power supply must deliver $82 + 96 = 178 \text{ gpm}$ at a pressure of $2.07 \times 10^7 \text{ n/m}^2$ (3,000 psi) at the longitudinal carriage on the VMS. Allowing for a $3.45 \times 10^6 \text{ n/m}^2$ (500 psi) drop in the long catenary

lines, the hydraulic power supply must be rated to deliver 180 gpm at 2.41×10^7 n/m² (3,500 psi). This is equivalent to 367 horsepower and assuming a 90% conversion efficiency, a 400 horsepower electric motor is required to drive the pump.

To conserve electric energy and minimize the cooling requirements, a variable-delivery type of pump should be used. Other details of the hydraulic power supply are illustrated in Figure 24.

Note that the pressure sensor for the hydraulic pump is remotely located on the longitudinal carriage and is connected electrically to a servo-regulator to control pump delivery. This is to force the pump to compensate for the pressure drop through the long catenary supply lines and insure that the maximum force developed by an individual actuator cannot exceed safe limits under any conditions.

8. COMPUTING REQUIREMENTS FOR COORDINATE CONVERSION

Since the entire VMS simulation is controlled from the central computer, it is assumed that the command signals to the individual pure motions (longitudinal, pitch, roll, yaw) are compensated for differences in centers of rotation for various vehicles. Then the computing requirements to be considered here are limited to the 3 DOF rotational system alone.

8.1 Math Model of 3 DOF Motion System

The mathematical model developed for design purposes includes not only the actuator displacements and velocities in response to angular commands but also actuator loads and structural stress. For the purposes of real-time calculation in VMS simulation, the latter parameters are eliminated to simplify and minimize computer requirements. The simplified equations are of the form

$$(\text{linear displacement})_{1,2,3} = f_1 (\text{pitch, roll, yaw angles})$$

$$(\text{linear velocity})_{1,2,3} = f_1 (\text{pitch, roll, yaw rates})$$

Note that these are continuous functions and in real time so the computer must be adequate to handle enough increments at a fast enough rate to insure smooth motion at the cockpit.

The solution of the equations involves the calculation of all sines and cosines, multiplication of the three sine-cosine matrices, multiplication times the cylinder vector and addition of the vector components. We estimate that the equivalent of 200 digital multiplications must be done in each iteration period.

In the past there has been continuing controversy regarding the relative merits of analog versus digital computation in dynamic systems in real time. Analog computers have practically infinite resolution and can be built as a unit dedicated to the coordinate conversion for the 3 DOF platform at a reasonable cost. On the other hand, most simulation facilities have centralized digital computers with capacity available to handle the coordinate conversion at little additional cost. Both situations were investigated in these studies.

8.2 Digital Computation

With reference to Para. 2.3.1 an acceptable level of spurious linear acceleration is 0.03g. Applying this value to the linear actuators of the 3 DOF platform and assuming fundamentally-sinusoidal steps, one can calculate the acceptable computing increment as follows:

Acceleration threshold, $a = 0.03g$

System bandwidth, $f = 12.6$ rad/sec (2 hertz)

$$\therefore \text{Maximum increment, } d = \frac{ag}{(4\pi)^2} = 1.9 \times 10^{-3} \text{ m (0.073 in)}$$

At the maximum velocity this increment should be recalculated at the following rate:

Actuator velocity, $v_1 = 1.07$ m/sec (42 in/sec)

Maximum increment, $d = 1.9 \times 10^{-3}$ m (0.073 in)

$$\therefore \text{Iteration rate} = \frac{v}{d} = 575/\text{sec}$$

With extrapolation = 50/sec

At the calculated v/d rate there is not enough time (0.0017 sec) to make a complete set of calculations, therefore extrapolation is used to calculate intermediate points (about 10). For the large ratio of iteration rate to bandwidth (50/2), this will present no problems. During periods of high acceleration, the extrapolation will be less accurate but the threshold is much higher in these cases.

To establish the necessary digital word length for position commands, we divide the total stroke of the actuator by the previously established increment

- Actuator stroke, $l = 1.52\text{m}$ (60 in)
- Maximum increment, $d = 1.9 \times 10^{-3}\text{m}$ (0.073 in)
- Resolution, $r = \frac{l}{d} = 822/1$
- Word length = 12 bits minimum

However, greater resolution is required for accuracy in the computation of sines and cosines, so

- Use word length = 16 bits

8.3 Analog Computation

With reference to the equations in Para. 8.1, it has been estimated that the coordinate conversion can be performed in real time with practically infinite resolution (no spurious accelerations) with the following complement of dedicated analog hardware

- 0 integrators
- 3 sine function generators
- 3 cosine function generators
- 27 multipliers
- 3 dividers
- 24 summers
- 6 operational amplifiers
- 9 potentiometers
- 1 power supply

9. OPERATING CONTROLS AND DISPLAYS

Operating controls and displays are required for a variety of purposes related to the operation of the new 4 DOF system with the VMS. They are:

- Normal startup and shutdown including subsystem supervisory functions and interlocks
- Emergency shutdown including sensors and controls to implement both safety systems and catastrophic failure
- Diagnostic and service including sensors, manual controls, computing equipment and displays to minimize downtime.

9.1 Normal Startup and Shutdown

In normal startup, the operator must be informed in the status of various subsystems and interlocks and subsystems such as electronic power and hydraulic pressure must be brought on in proper sequence to avoid loss of control. Figure 25 illustrates an operators' control panel that,

coupled with supervisory computer capacity, provides these functions. Before starting the hydraulic pumps, the "ready" light must be on summarizing the status of all interlocks. If the ready light does not come on, individual small lamps indicate which interlock is not satisfied so the operator can take appropriate action.

The operator is also presented with the status of certain subsystems that allow time for the exercise of human judgment. For example, if the oil temperature exceeds a preset limit but the VMS is in the process of a costly simulation run, the operator can choose to complete the run if it can be done safely.

In normal startup, power is applied in the proper sequence and all subsystems assume their nominal status with the 4 DOF system in a position just opposite the loading ramp. The operator can then transfer control (through built-in "bumpless" circuits) to computer control or to the service panel.

The operator also has on his panel an emergency shutdown button. This initiates a level 2 shutdown. For example, it removes control of the 4 DOF system from the central VMS Operators' Console and returns the platform to a "rest" position.

9.2 Safety Systems

9.2.1 Level 1 - Controlled Safety Systems

Safety systems are built into all control systems to detect and control:

- acceleration
- velocity
- displacement

Except in those cases where the actuator loadings are quite variable, acceleration limits are built-in, being largely determined by the available supply pressure. On the other hand, the deceleration involved in stopping under emergency conditions must be limited so as not to overstress man or machine. A simple mechanization uses relief valves arranged to limit pressure differential across the hydraulic cylinder.

Velocity limits are vitaly important because displacements are limited by the cylinder stroke. With both displacement and deceleration limited, the velocity must be automatically limited so the servo actuator can be stopped before it makes a mechanical contact (bottoms). A simple velocity limit would be set at fixed level for the worst case and the stopping sequence tripped with a limit switch a fixed distance from the end of stroke.

However, this limits the useable simulation stroke to the distance between the two fixed limit switches. If the servo actuator is following a low-velocity command signal under normal conditions, a simulation run may be needlessly interrupted by this scheme. We believe that the VTOL Motion Control System would be significantly compromised unless it was fitted with "smart" velocity limits, sensitive to distance from the end-of-stroke of the hydraulic cylinders, such as illustrated in Figure 26.

The circuit of Figure 26 can be used to limit maximum acceleration, velocity and position commands using position command as an input. The circuit configuration shown represents a second order system in which the first and second derivatives then represent velocity and acceleration, respectively. If limiting circuits are incorporated, maximum values for each of the parameters can be controlled. Judicious selection of circuit constants, limiting levels and limitation of the input position command, X_1 , will not only yield the desired maximums, but will allow programming the output functions, X_0 , into its limits at a predetermined acceleration.

9.2.2 Level 2 - Emergency Systems

An emergency system for hydraulic actuator driven simulators to prevent loss of control of the mechanical equipment has been developed to handle such emergency conditions as:

- excessive acceleration
- excessive velocity
- displacement beyond limits
- excessive servo loop errors
- subsystem malfunctions
- interlock trips
- loss of hydraulic pressure
- loss of electric power
- operator's options

Emergency system operation can be initiated automatically by any desired predetermined parameters, or it can be initiated manually at the discretion of the operator.

Under emergency operation the motion of the platform must be arrested with controlled deceleration and returned by controlled power to its "at rest" position. Powered return of the platform is always possible using the hydraulic fluid which is stored under pressure in accumulators connected to the hydraulic lines. The system is interlocked in such a manner that the moving platform cannot be operated unless accumulator pressure is available.

In normal system operation, the valve solenoids are energized, allowing pressure to be exhausted from the emergency actuators, so that emergency actuator pistons are withdrawn from main servo valve chamber, thus allowing the servo valve to control the system.

Initiation of emergency system operation shuts off electric power from all electrohydraulic components including the main pump motor and the solenoid valves. The valves now admit accumulator pressure to the "down" chamber of the main servo valve and drains pressure from the "up" chamber of main servo valve. Accumulator pressure is also admitted to the emergency actuator, thus forcing it to move the main valve spool to a predetermined position in the "down" direction. The slight opening of the valve orifice thus provided causes a drop in pressure on the bottom piston area of the actuators, and the moving platform is driven to its "at rest" position.

The speed at which the moving platform is driven to the "at rest" position is controlled by the amount of opening in the second stage servo valve and is adjustable. The speed of valve spool movement to its predetermined "down" position is also adjustable so as to control initial acceleration.

When platform has reached its "at rest" position and all accumulator pressure has drained off the main pilot operated check valves close thus hydraulically locking the platform in place.

A digital computer program is available for verifying the action of the moving platform during emergency return to "at rest" position. The program determines the platform coordinates from the actuator length by solving simultaneous nonlinear equations, calculates the linear and angular displacements of the center of gravity of the payload, and derives the pressure in both the upper and lower chambers of the actuators. The actuators are then allowed to retract a short distance as determined by the pressures and the given orifice opening, after which the computation cycle is repeated until the platform arrives at the "at rest" position.

The accuracy of this program has been proved by checking the results obtained from it against actual retraction speeds of the simulator. Not only was close correspondence between calculated and actual speeds observed, but the amount of orifice opening of the second spool as determined by the setting of the emergency actuator was corrected on the basis of computed results so as to achieve uniform retraction speed. Uniformity of retraction speeds cannot be taken for granted because the displacement of the servo valve spool under emergency conditions is of the same order of magnitude as the manufacturing tolerances.

Figure 27 shows a typical time history of platform motion during an emergency shutdown initiated during a large displacement in three degrees of freedom.

9.3 Service and Diagnostic Controls

For routine maintenance and setup a convenient control panel such as illustrated in Figure 28 is provided. During normal operation the observer can monitor servo loop errors to anticipate impending problems. When operation is transferred to him, he has the manual controls and other displays, coupled with computer diagnostic algorithms, to aid in isolating malfunctions.

Further controls, computation and displays can be used on remote panels to pinpoint a malfunction and to aid the service person in correcting it.

10. SUMMARY OF SPECIFICATIONS

10.1 Technical Specification Package

The conceptual design studies covered by this report were intended to produce technical specifications and preliminary design drawings to support a detail design effort. The total package is to be made up from the following four categories:

1. The functional requirements and restraints as originally presented in the Statement of Work and summarized in Para. 2 of this report.
2. Additional functional requirements as developed during the conceptual design studies and presented in Para. 10.2 to follow.
3. Preliminary design requirements as developed in this report and presented in general, block diagrams and circuit schematics.
4. Preliminary mechanical design drawings including overall layouts, kinematic arrangements and details of critical mechanical couplings and structures.

The Specification Package is assembled and delivered as a separate item including the following documents.

- Technical Specification
- Circuits and Block Diagrams
- Mechanical Drawings

10.2 Additional Design Requirements

10.2.1 Safety Factor

The motion system shall be designed with a minimum safety factor of 2 on yield stress of materials using the following motions as the basis for calculating the loads:

<u>Mode</u>	<u>Acceleration</u>
Vertical	14.6m/sec ² (48 ft/sec ²)
Lateral	7.32 m/sec ² (24 ft/sec ²)
Longitudinal	3.05 m/sec ² (10 ft/sec ²)
Roll, Pitch and Yaw	1.04 rad/sec ² (60°/sec ²)

10.2.2 Natural Frequencies

The lowest natural frequency of the motion system structure shall be greater than 50.2 rad/sec (8.0 hertz). Design provisions shall also be made to avoid excitation of any natural frequency greater than 50.2 rad/sec (8.0 hertz).

10.2.3 Noise Threshold

Noise thresholds (responded uncorrelated with commands) shall be determined by monitoring the appropriate variable while exercising each degree of freedom of motion with a 0.63 rad/sec (0.10 hertz) sine function having an amplitude of 10% of maximum displacement.

<u>Mode</u>	<u>Threshold (peak)</u>
Linear	0.03 g
Angular	0.004 rad/sec ² , 0.04 rad/sec

10.2.4 Smoothness

Throughout the performance envelope, motion noise levels shall be no greater than the noise threshold levels stated in Para. 10.2.2.

10.2.5 Static Accuracy

Static error between actual and commanded position should be less than 1% of maximum displacement.

10.2.6 Noise Control

Goal for the audible noise level of the new 4 DOF motion system as measured at the cab/visual interface shall be below 75 db over the entire audio frequency range.

10.2.7 Position-Rate-Acceleration Envelopes

Operating envelopes for the longitudinal and rotational motions of the new 4 DOF motion system shall be within 5% of those illustrated in Figures 29 and 30, respectively.

10.2.8 Safety Systems

Safety systems shall be included in the design of the new 4 DOF to automatically limit maximum accelerations and velocities to safe levels under all normal and emergency situations.

10.2.9 Actuator Replacement

Design of the new 4 DOF motion system shall allow for the replacement of a complete electrohydraulic actuator assembly within 2 hours.

10.2.10 Hydraulic Fluid

Hydraulic fluid shall be MIL-H-5606 or equivalent.

11. ESTIMATED COSTS

We estimate that the cost of construction, installation and checkout of the new 4 DOF Motion System is under \$2 million.

12. ESTIMATED TIME SCHEDULES (Figure 31)

● Detail Design	9
● Construction and In-Plant Tests	12
● Installation and On-Site Tests	6
	<u>27</u> months

13. CONCLUSIONS AND RECOMMENDATIONS

The objective of the design studies covered in this Report was to develop the conceptual design for converting the Vertical Motion Simulator to a multi-purpose aircraft and helicopter simulator. This objective has been met most effectively through the creative design of a unique, high-performance 4 DOF motion system to permanently replace the present 6 DOF synergistic system. The new 4 DOF system has the following outstanding features:

- will integrate with the two large VMS translational modes and their associated subsystems
- can be converted from helicopter to fixed-wing aircraft simulation through software changes only
- interfaces with an advanced cab/visual display system of large dimensions
- makes maximum use of proven techniques, convenient materials and off-the-shelf components
- will operate within the existing building envelope without modifications
- can be built within the specified weight limit and avoid compromising VMS performance
- provides maximum performance with a minimum of power consumption
- simple design minimizes coupling between motions and maximizes reliability
- can be built within existing budgetary figures.

In conclusion, the proposed conceptual design for the new 4 DOF meets every goal of the NASA simulation program. It satisfies the major objective of the design studies with a concept that is based on proven equipment and fits within all design constraints. Therefore it meets the objective most effectively and for the lowest cost.

We believe that the selection of the conceptual 4 DOF motion system described in this Report for implementation with the VMS will be in the best interest of the NASA. We recommend that selection.

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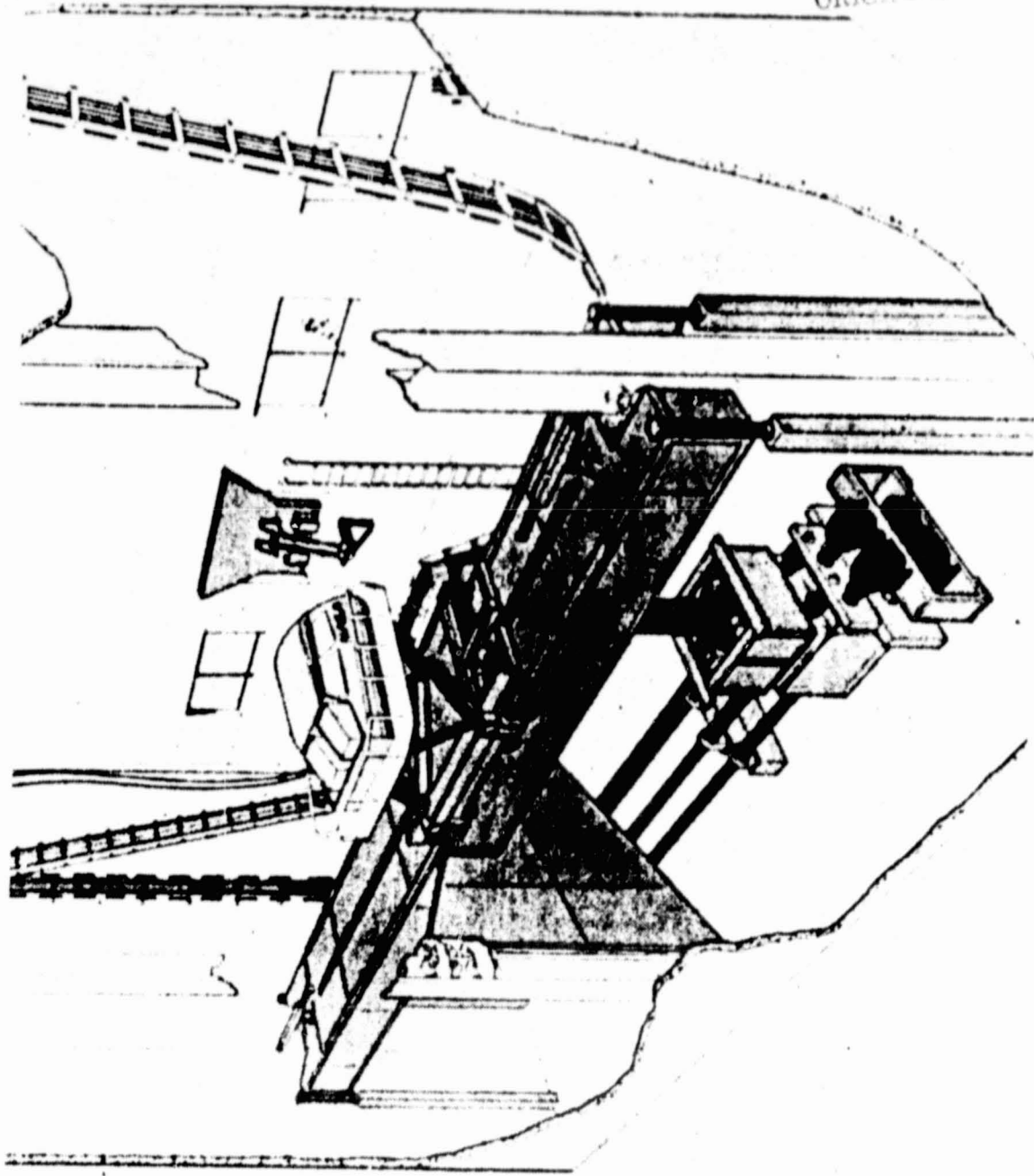


Figure 1. Illustration of VMS Now Under Construction

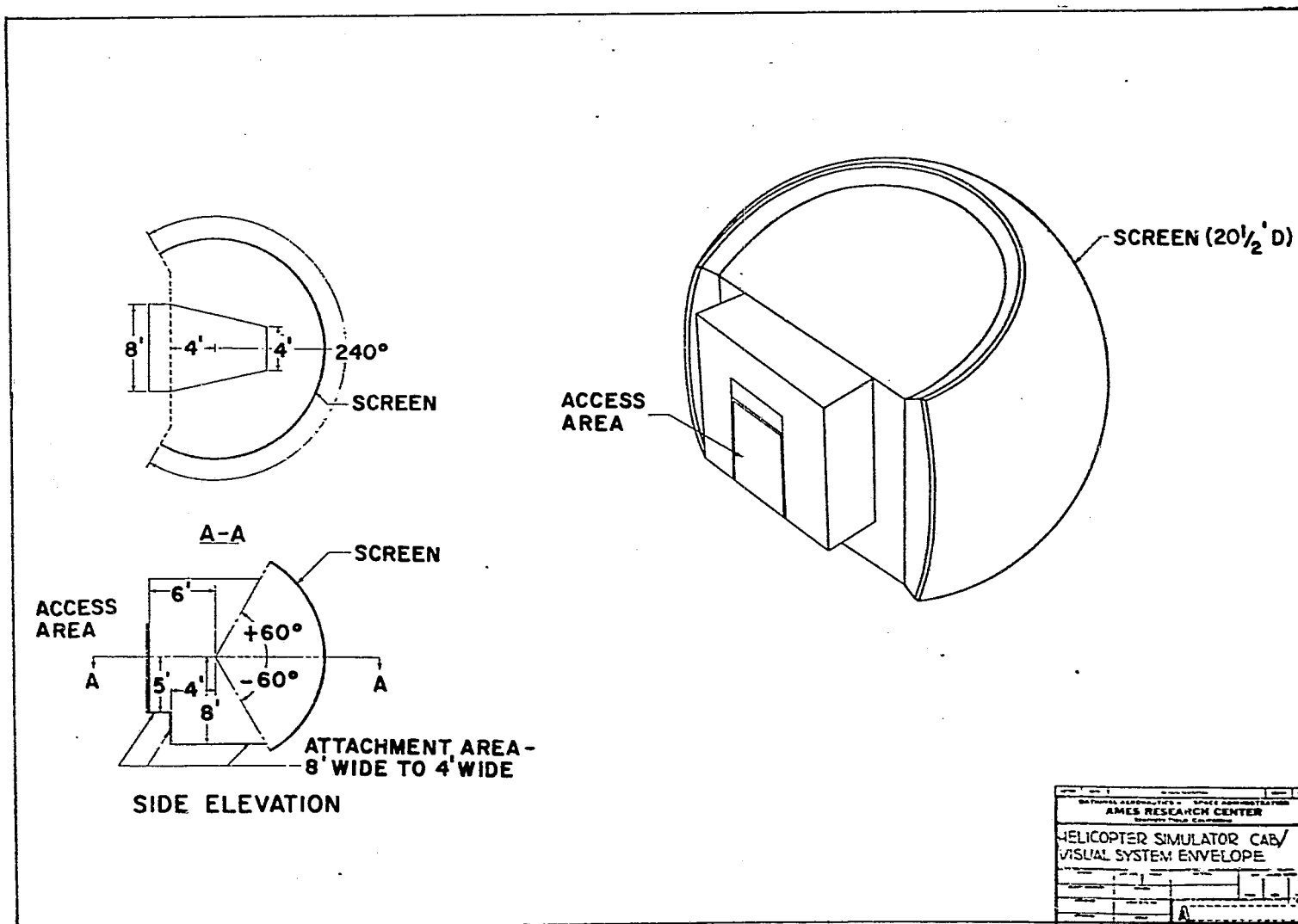


Figure 2. CAB/Visual System Envelope (Preliminary)

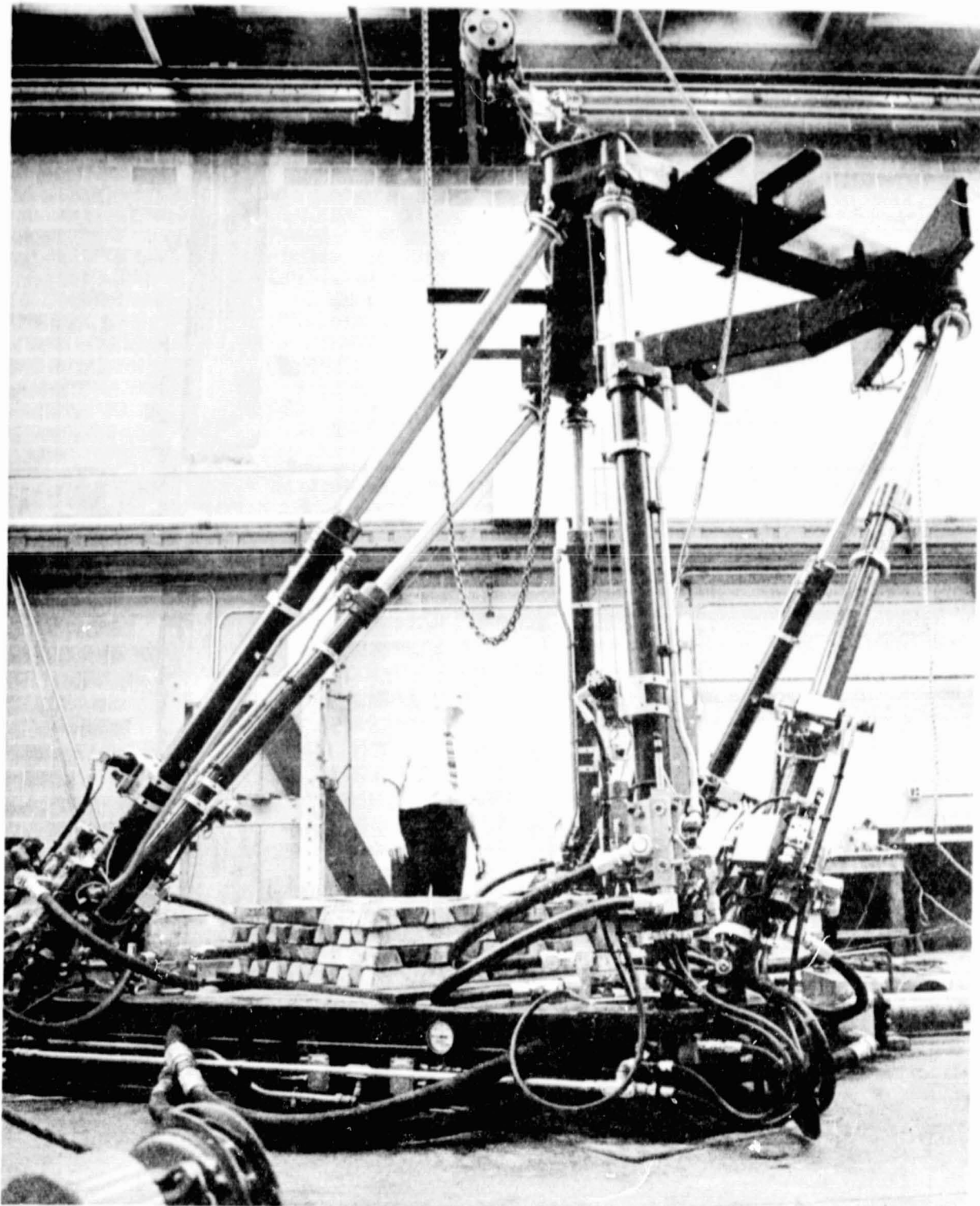


Figure 4. Six DOF Synergistic Platform

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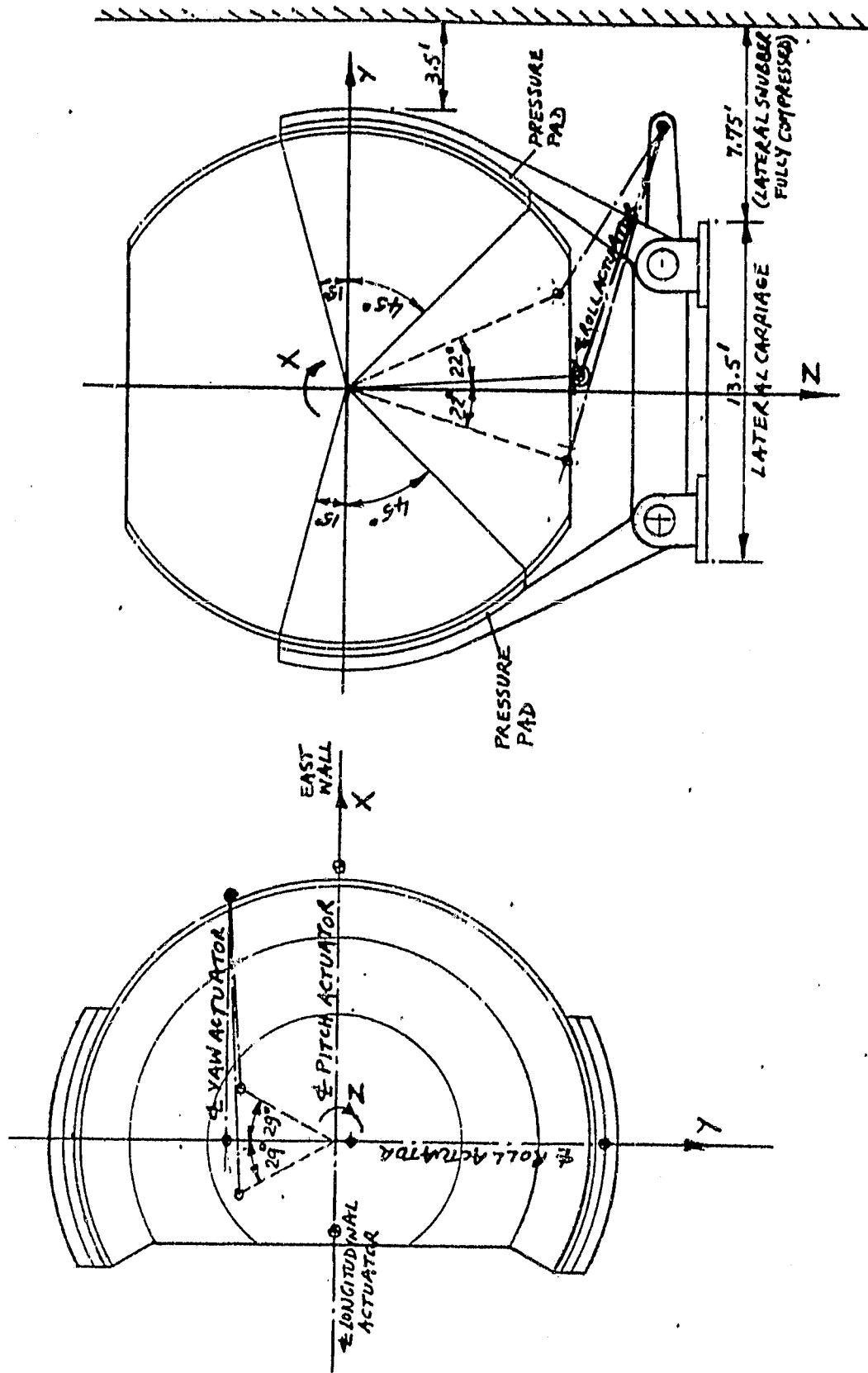
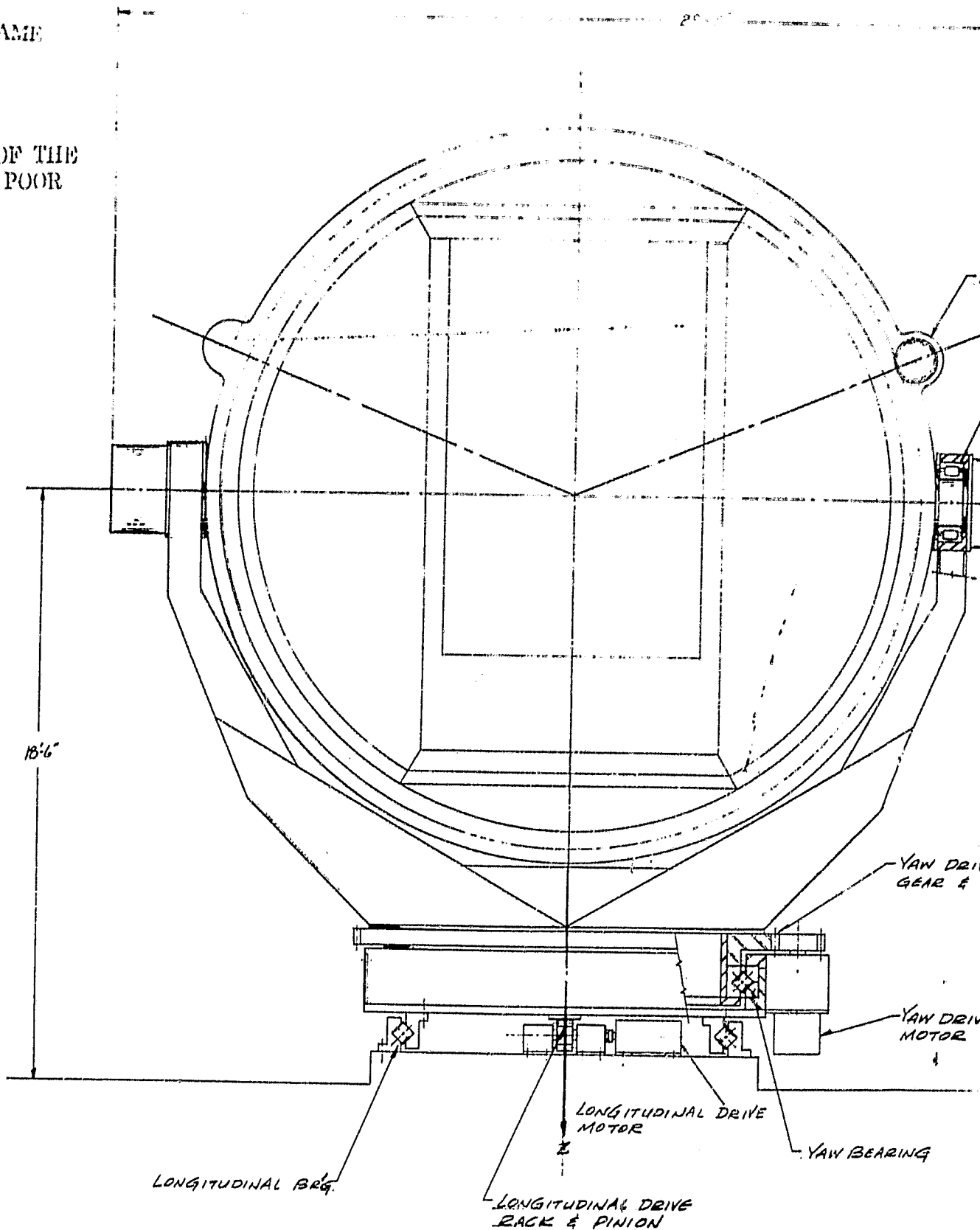


Figure 5. Air Bearing Sphere on Longitudinal Carriage

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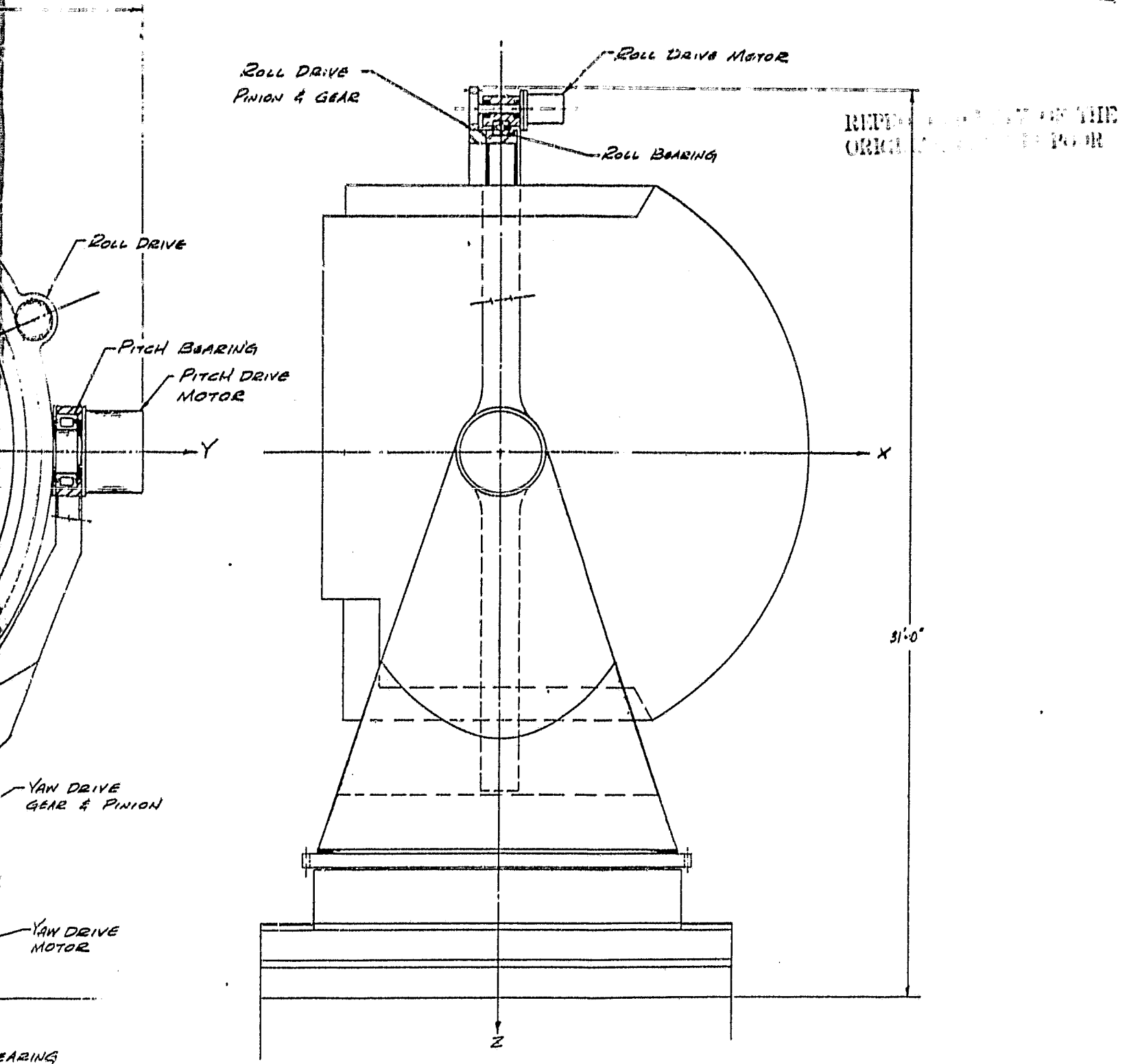


Figure 6. Gimbaled Three DOF on Longitudinal Carriage

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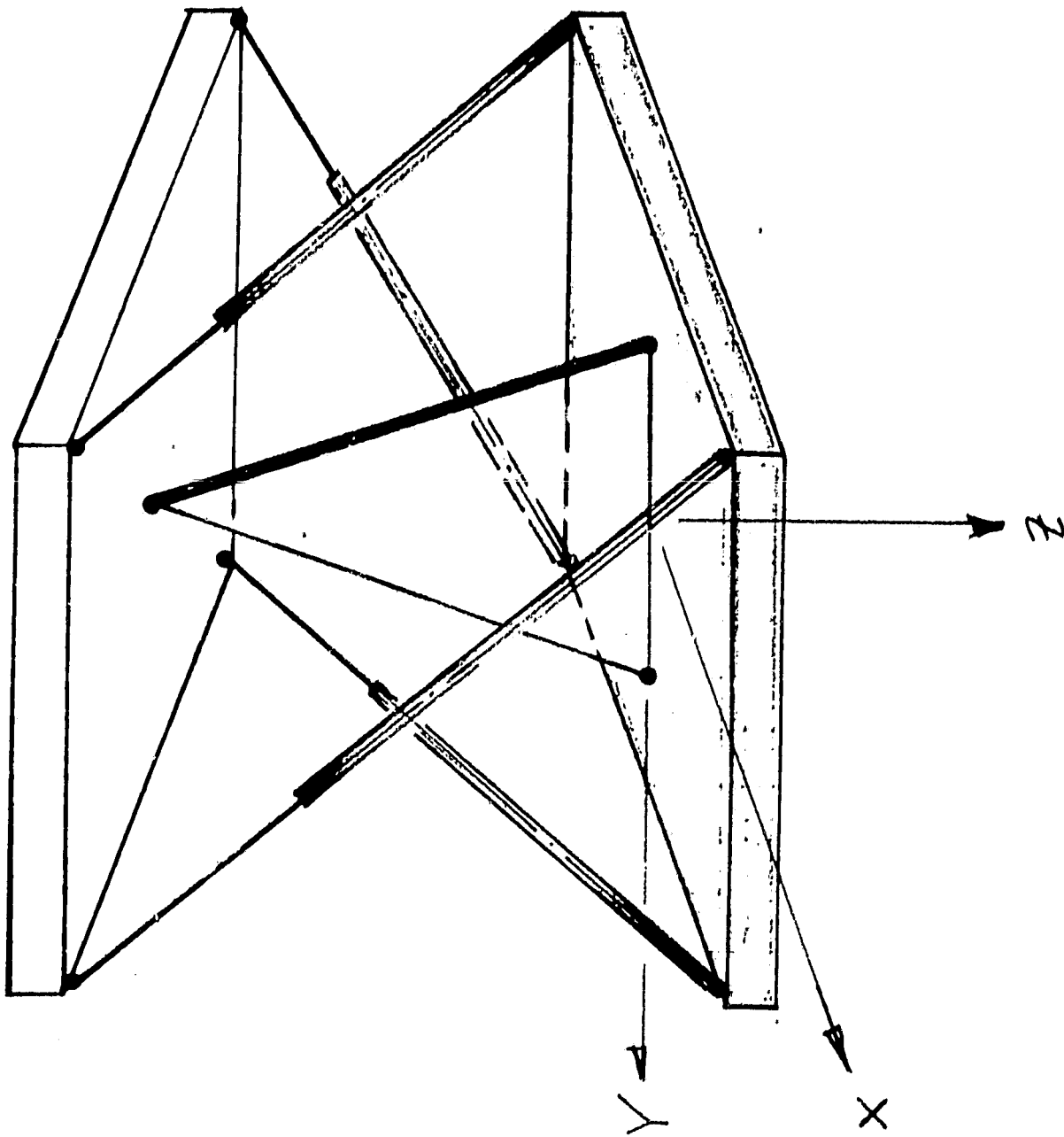


Figure 7. Four DOF Synergistic Platform

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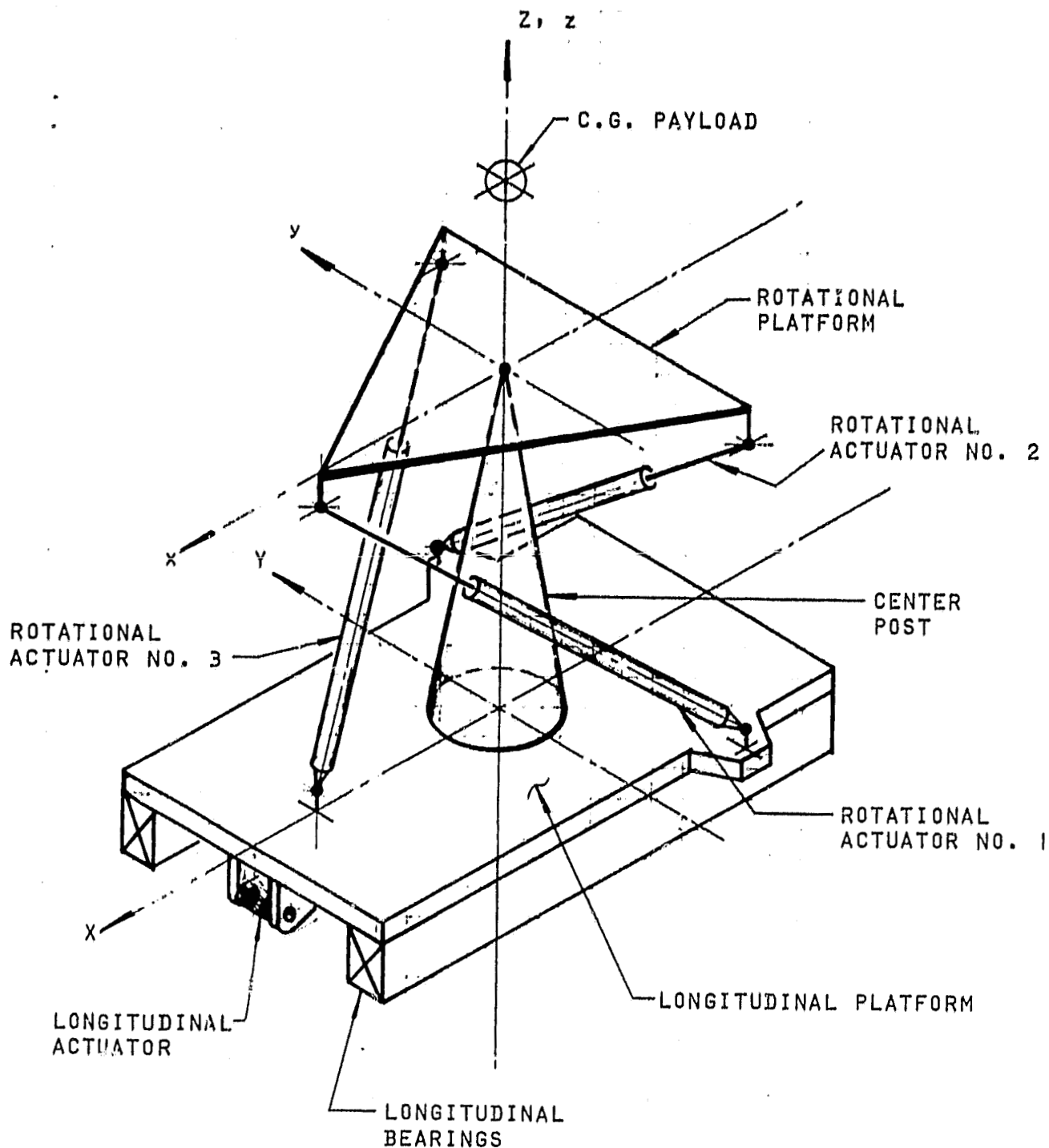
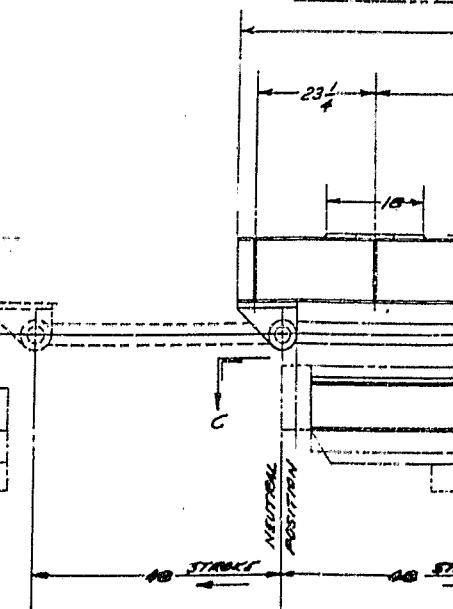
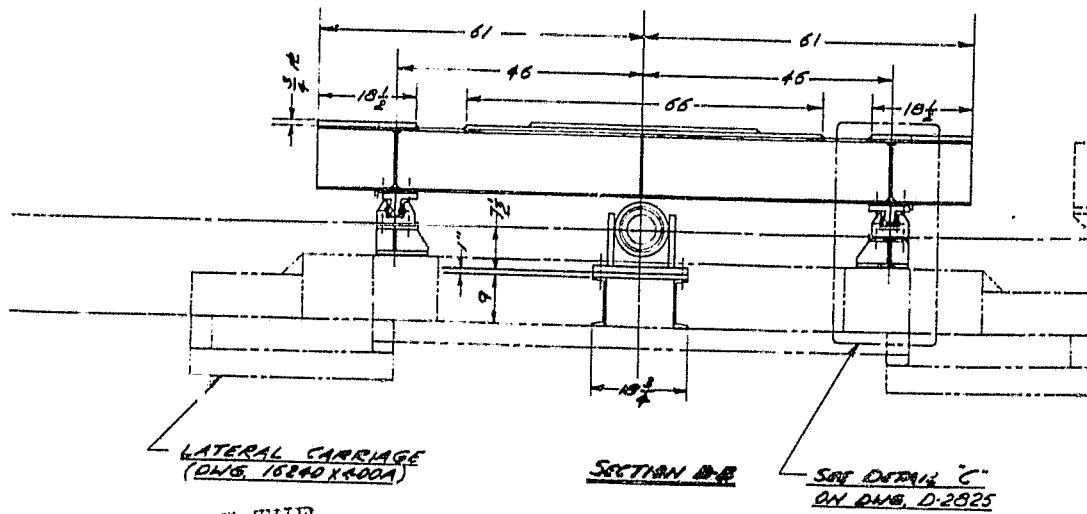
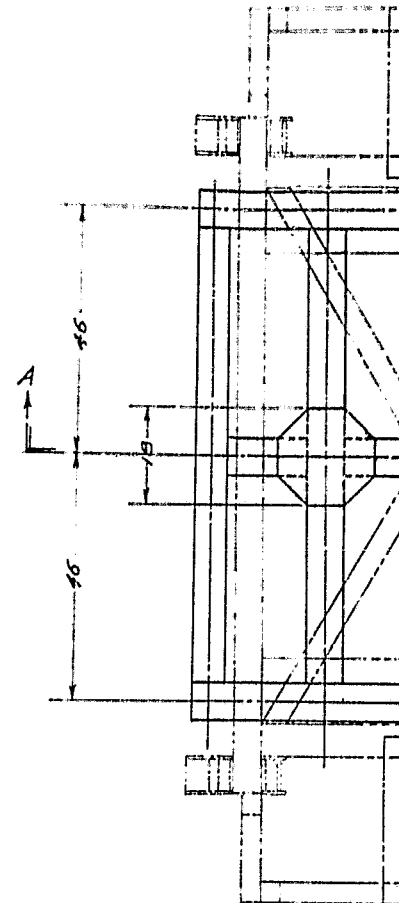
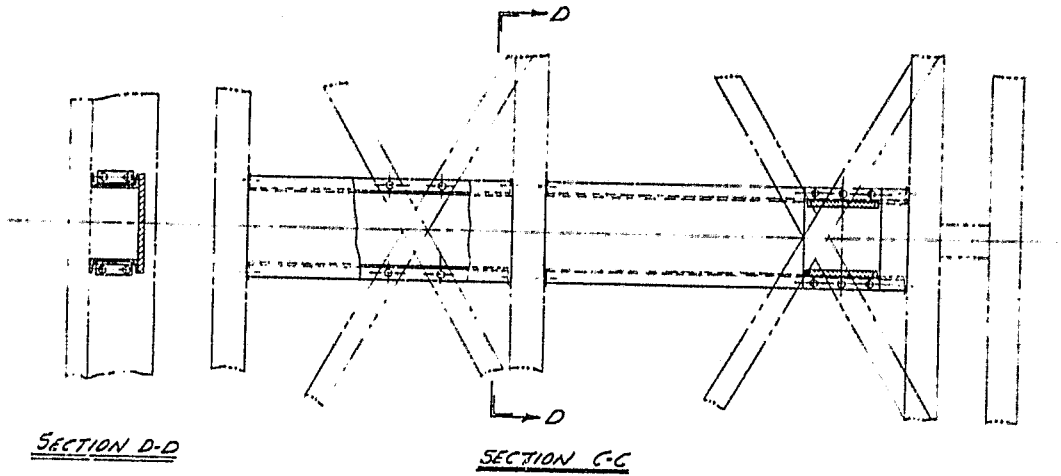


Figure 8. Three DOF on Longitudinal Carriage

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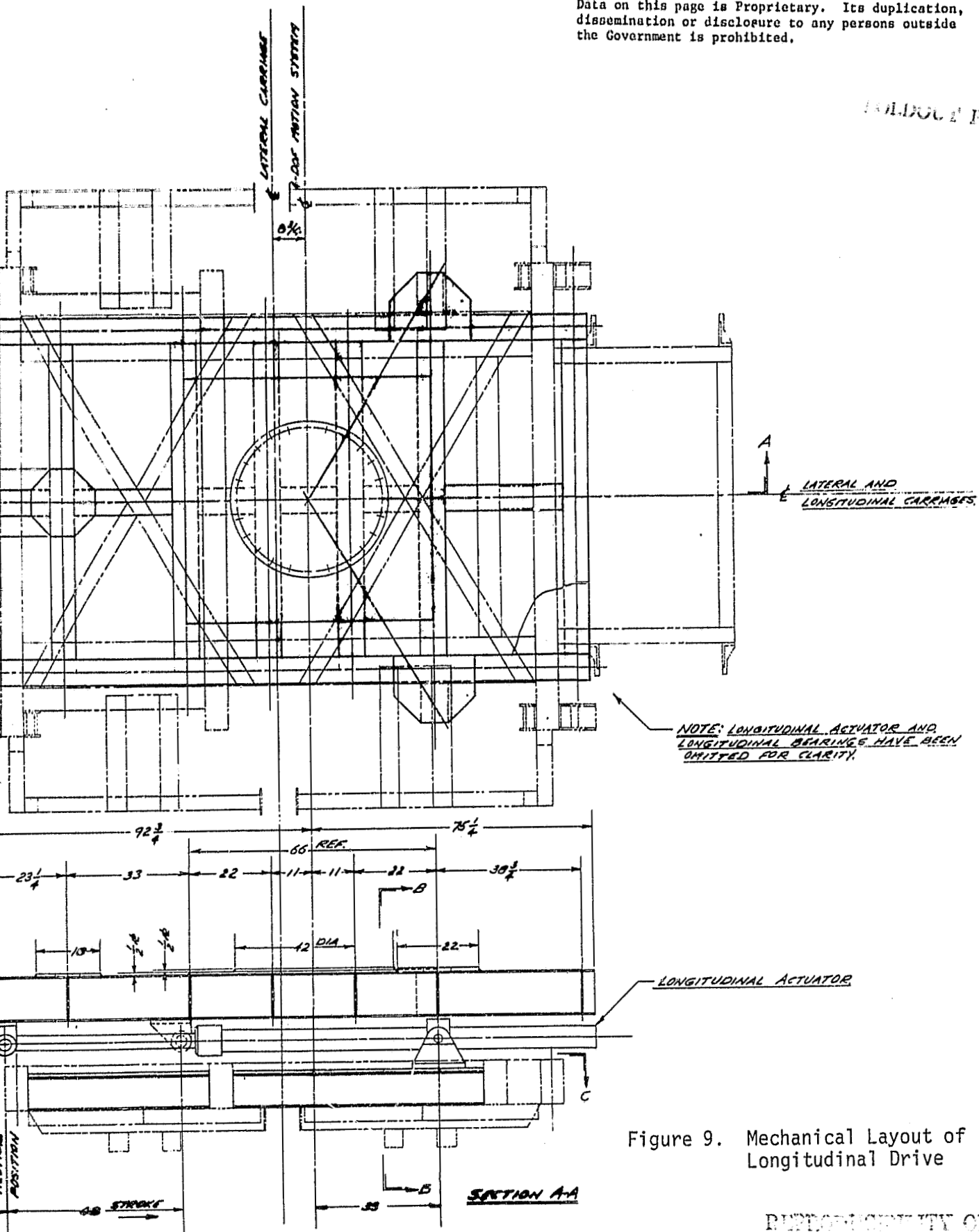
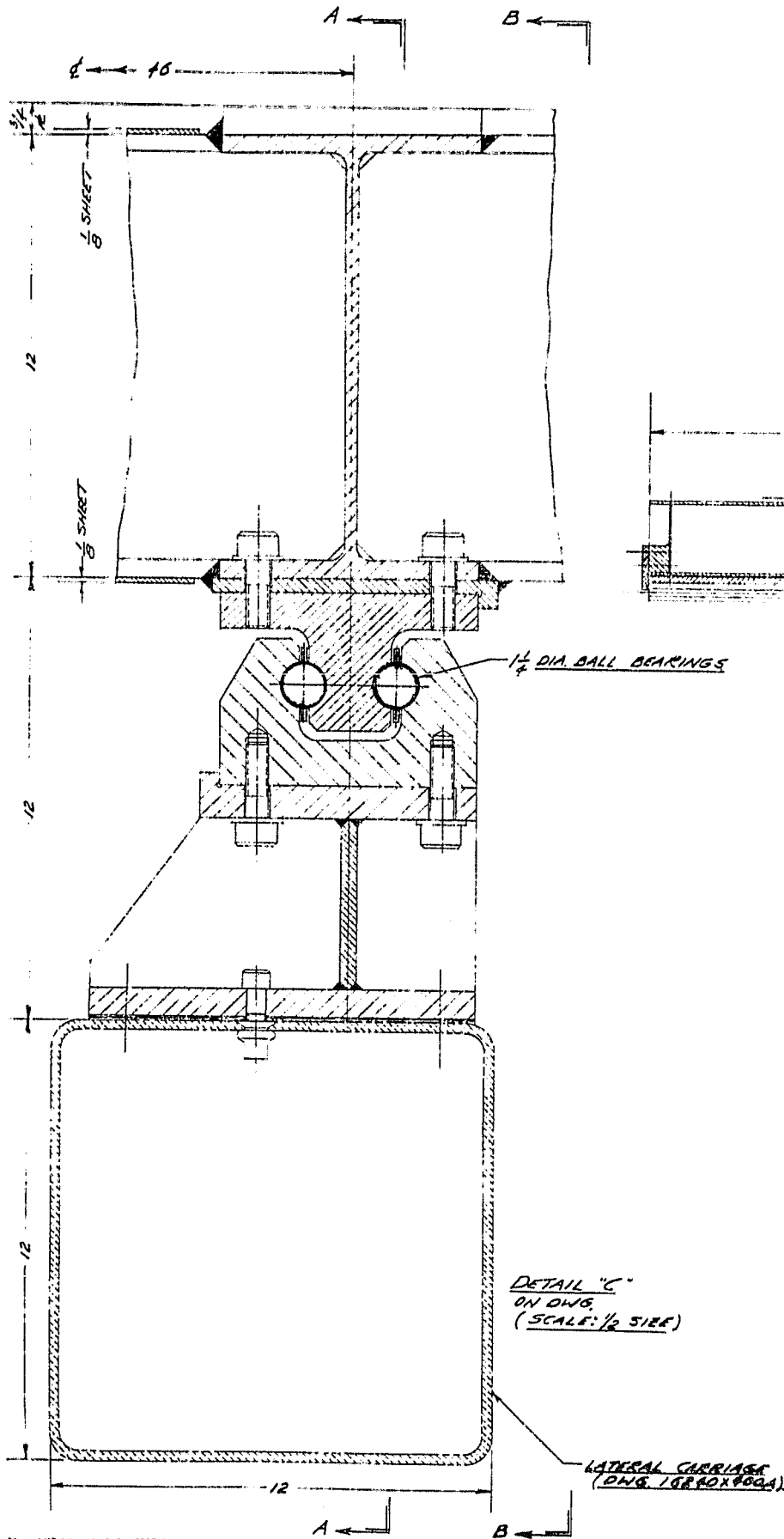


Figure 9. Mechanical Layout of Longitudinal Drive

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DETAIL "C"
ON DWG.
(SCALE: 1/2 SIZE)

LATERAL CARRIAGE
(DWG. 10660X804)

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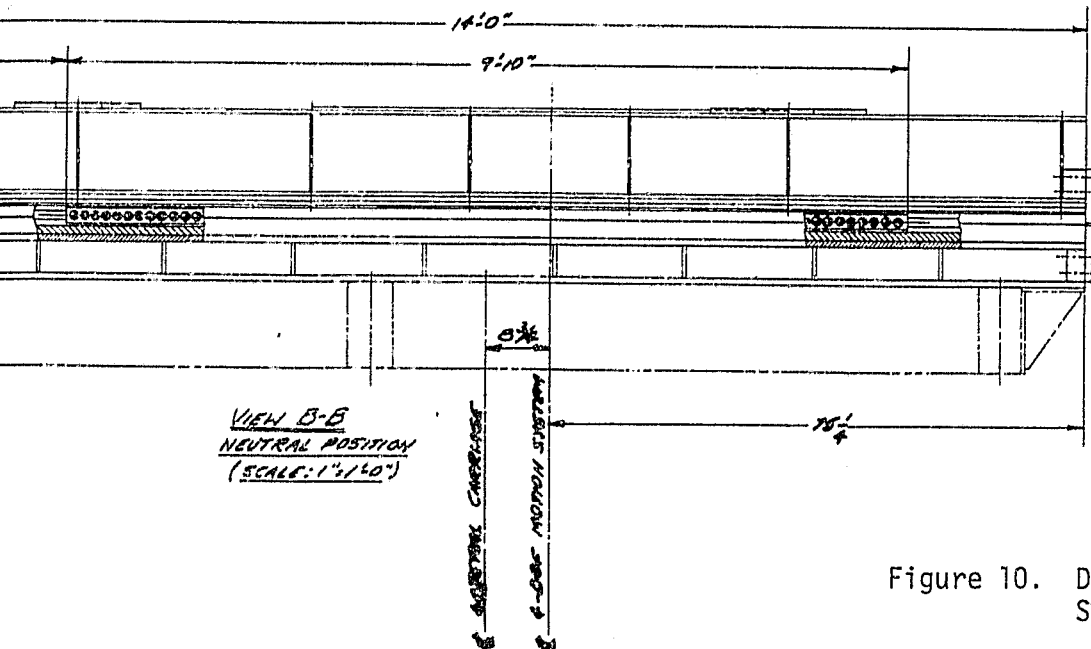
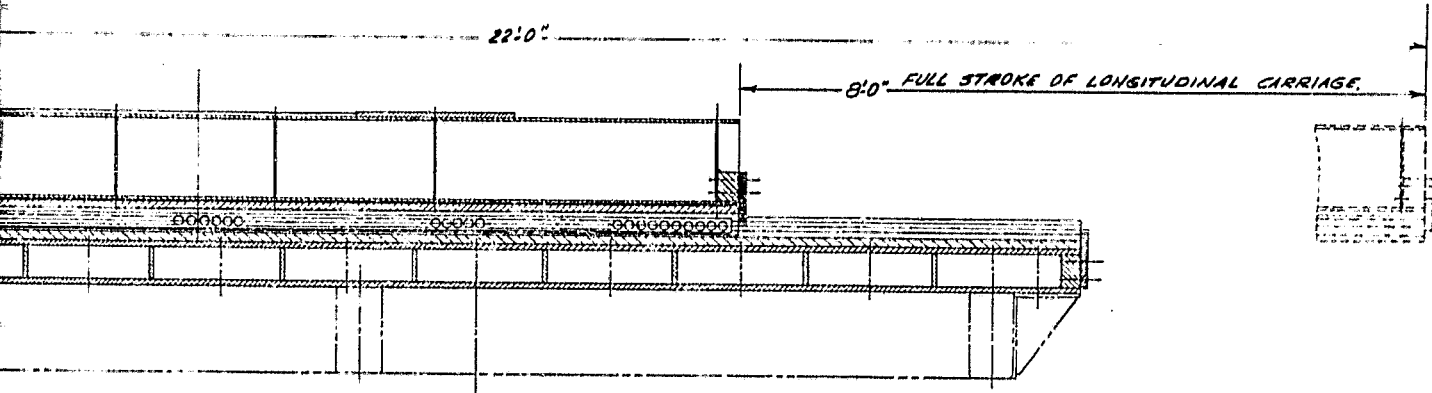
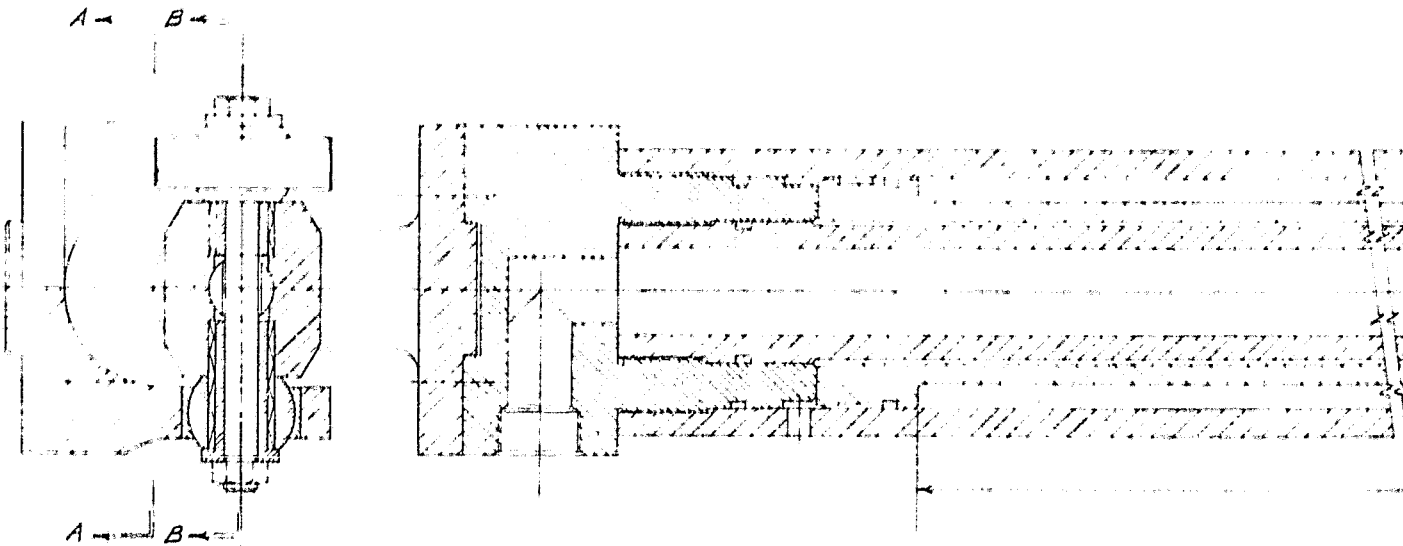


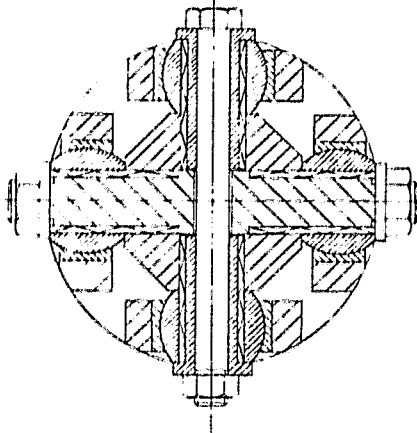
Figure 10. Details of the Bearing Section

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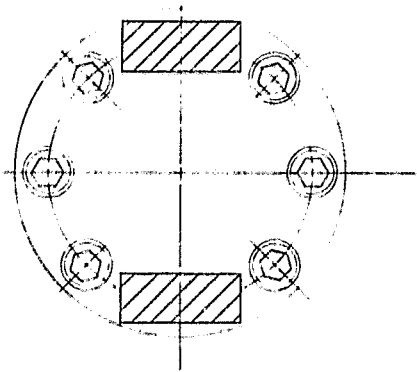
FOLDOUT FRAME



Z — 146 $\frac{3}{4}$ NE



SECTION B-B
(TYPICAL BOTH ENDS)



SECTION A-A

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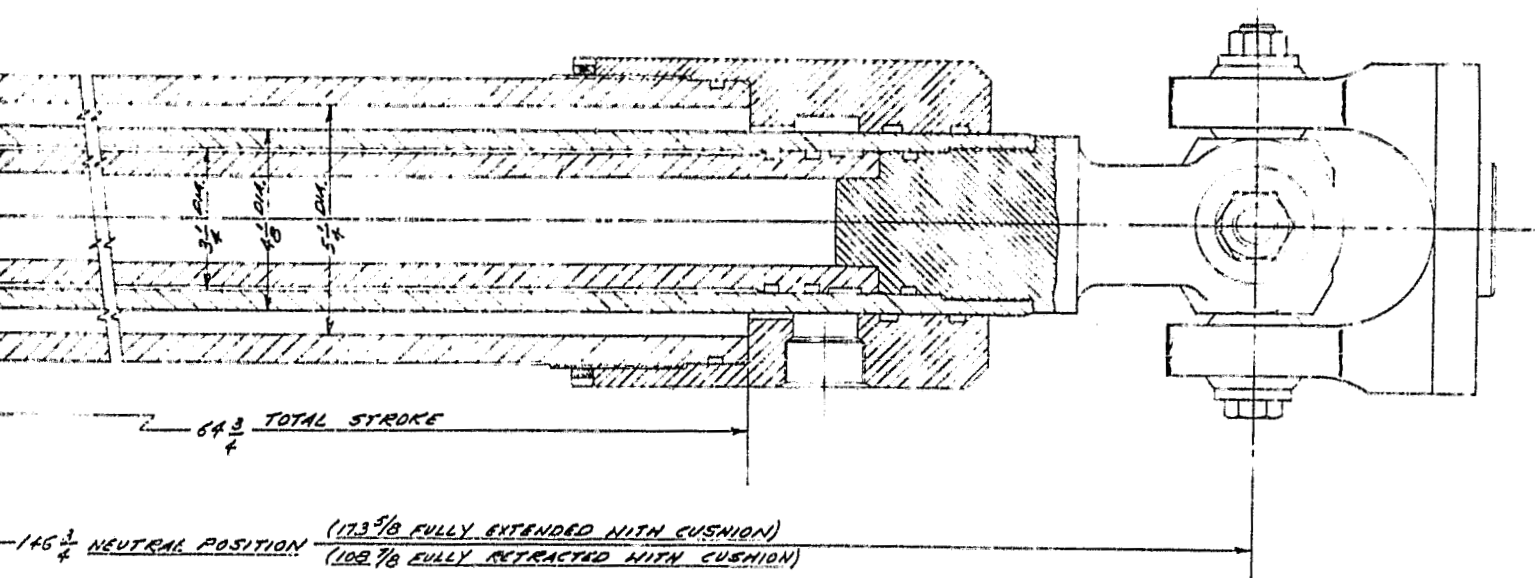


Figure 11. Layout of Equal-Area Hydraulic Cylinder

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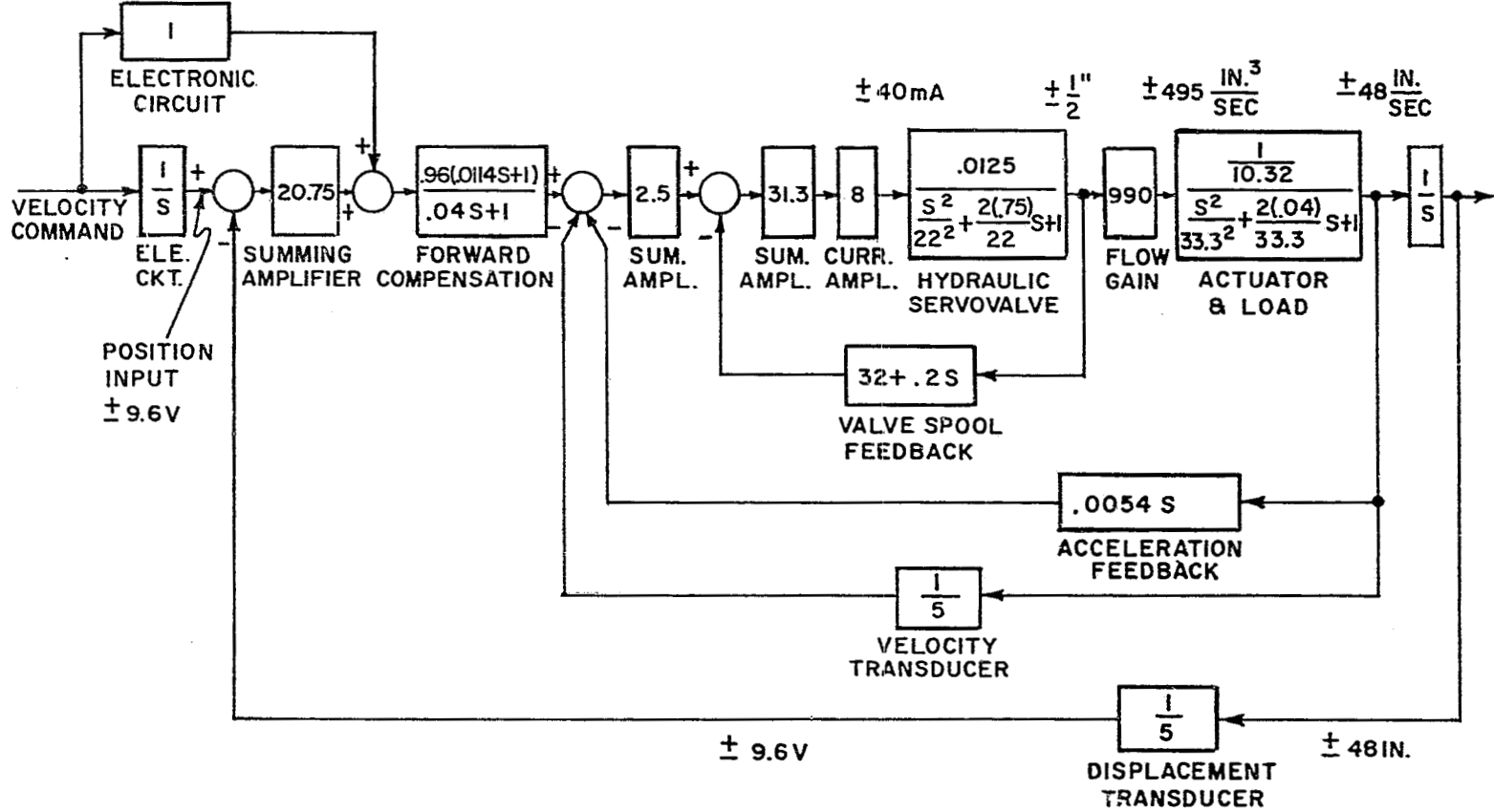


Figure 12. Control Block Diagram for Longitudinal System

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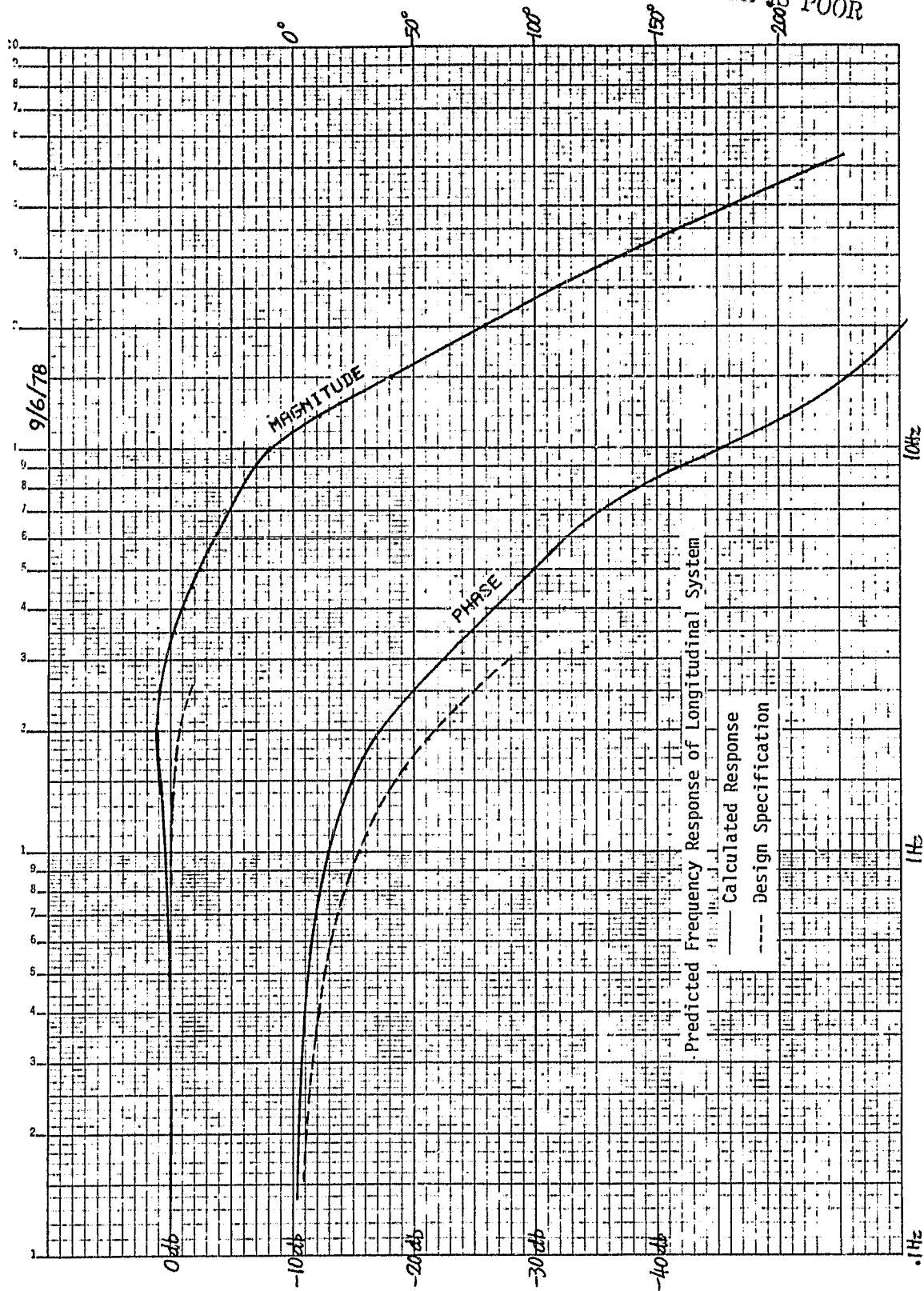


Figure 13. Predicted Frequency Response of Longitudinal System

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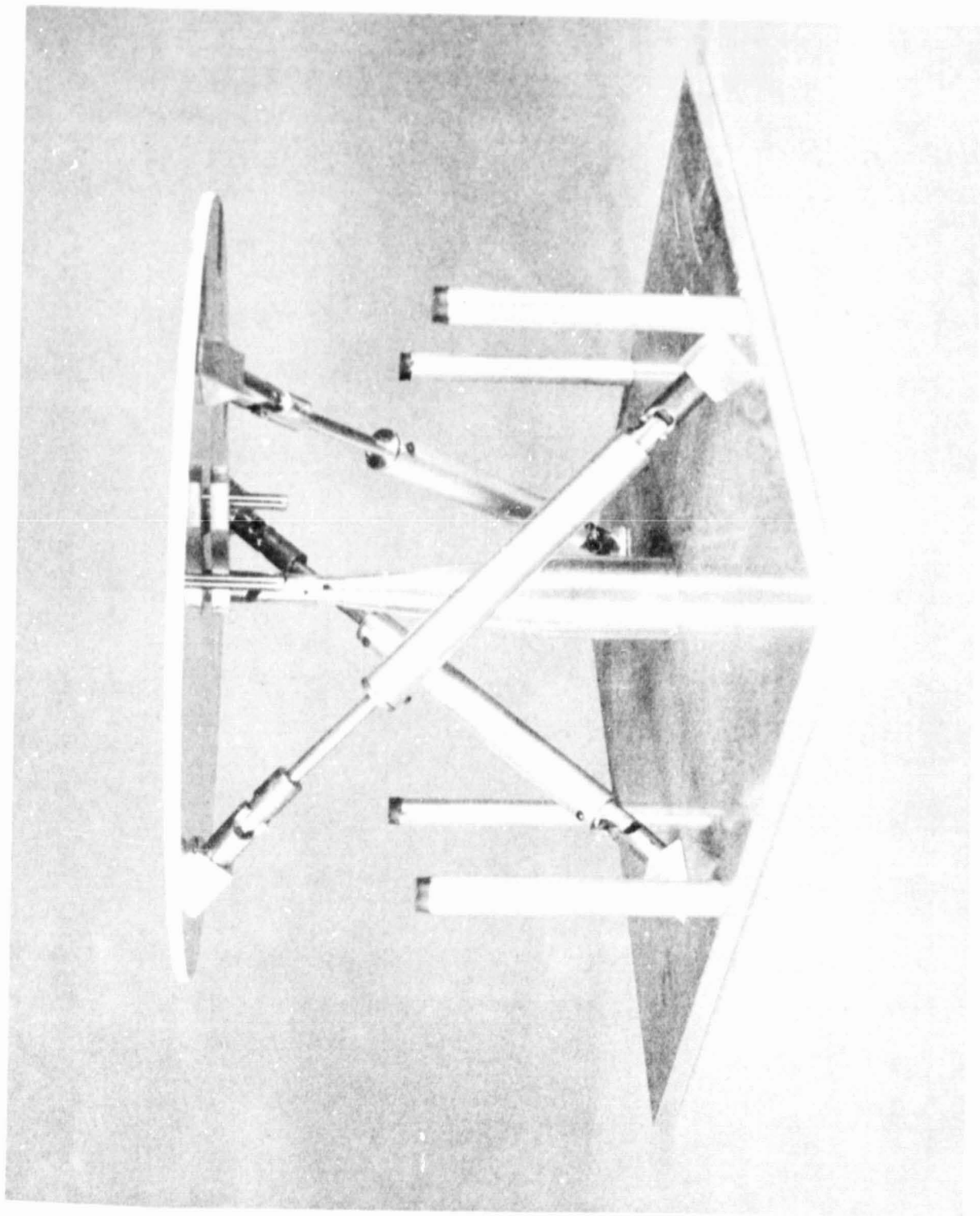


Figure 14. Kinematics of the 3 DOF Platform (Neutral)

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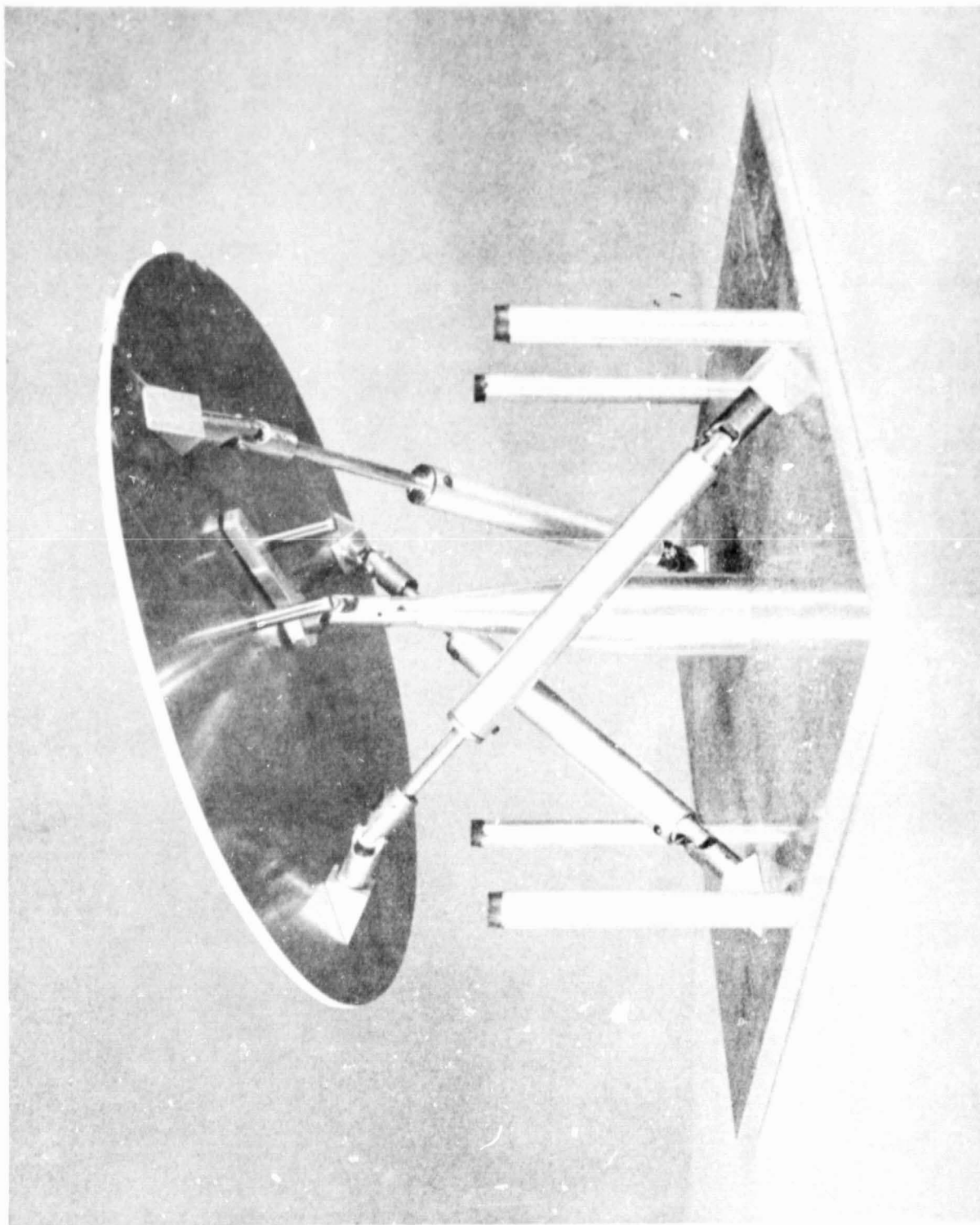


Figure 15. Kinematics of the 3 DOF Platform (Cont.) Pitch Up

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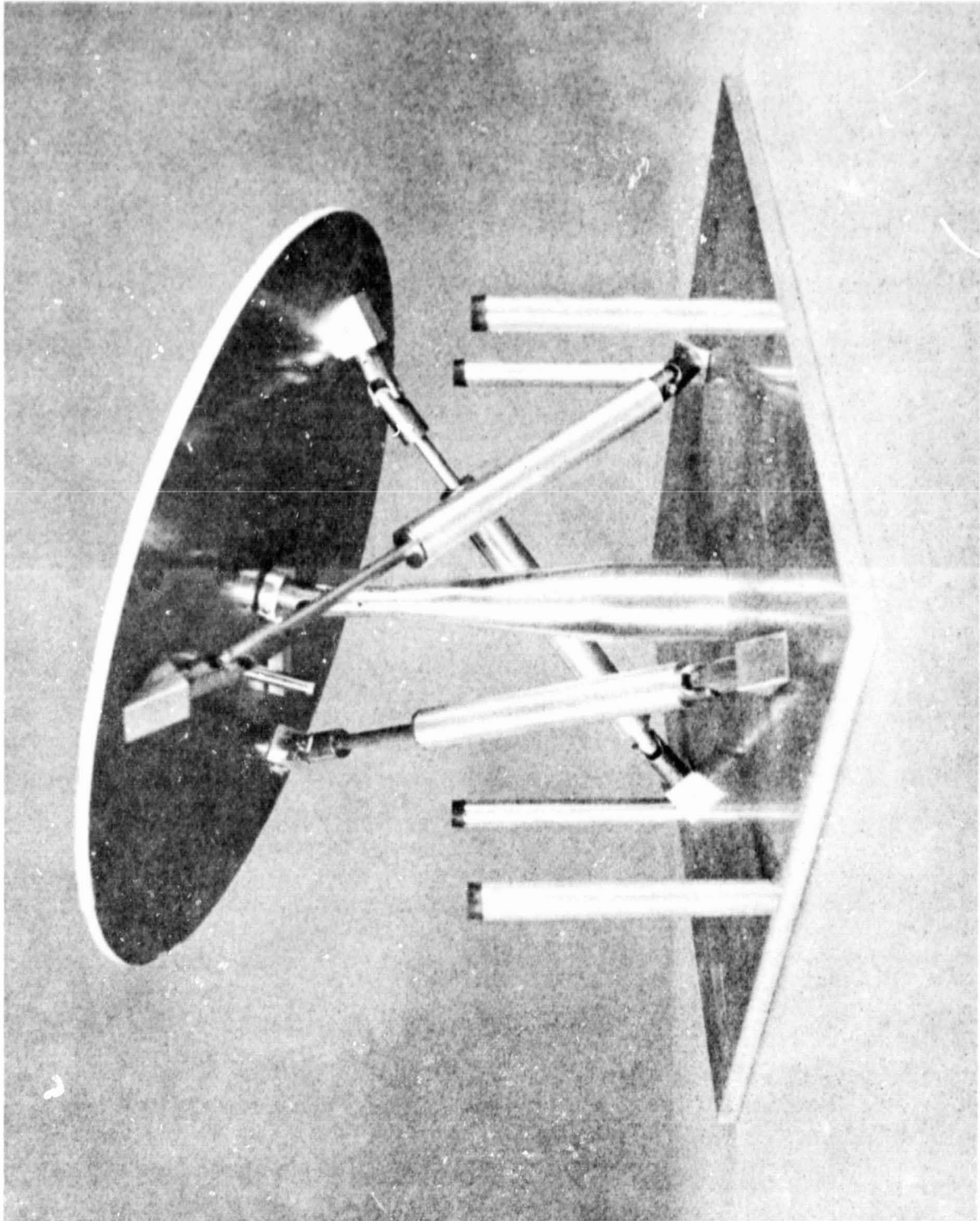


Figure 16. Kinematics of the 3 DOF Platform (Cont.) Roll Right, Yaw Right

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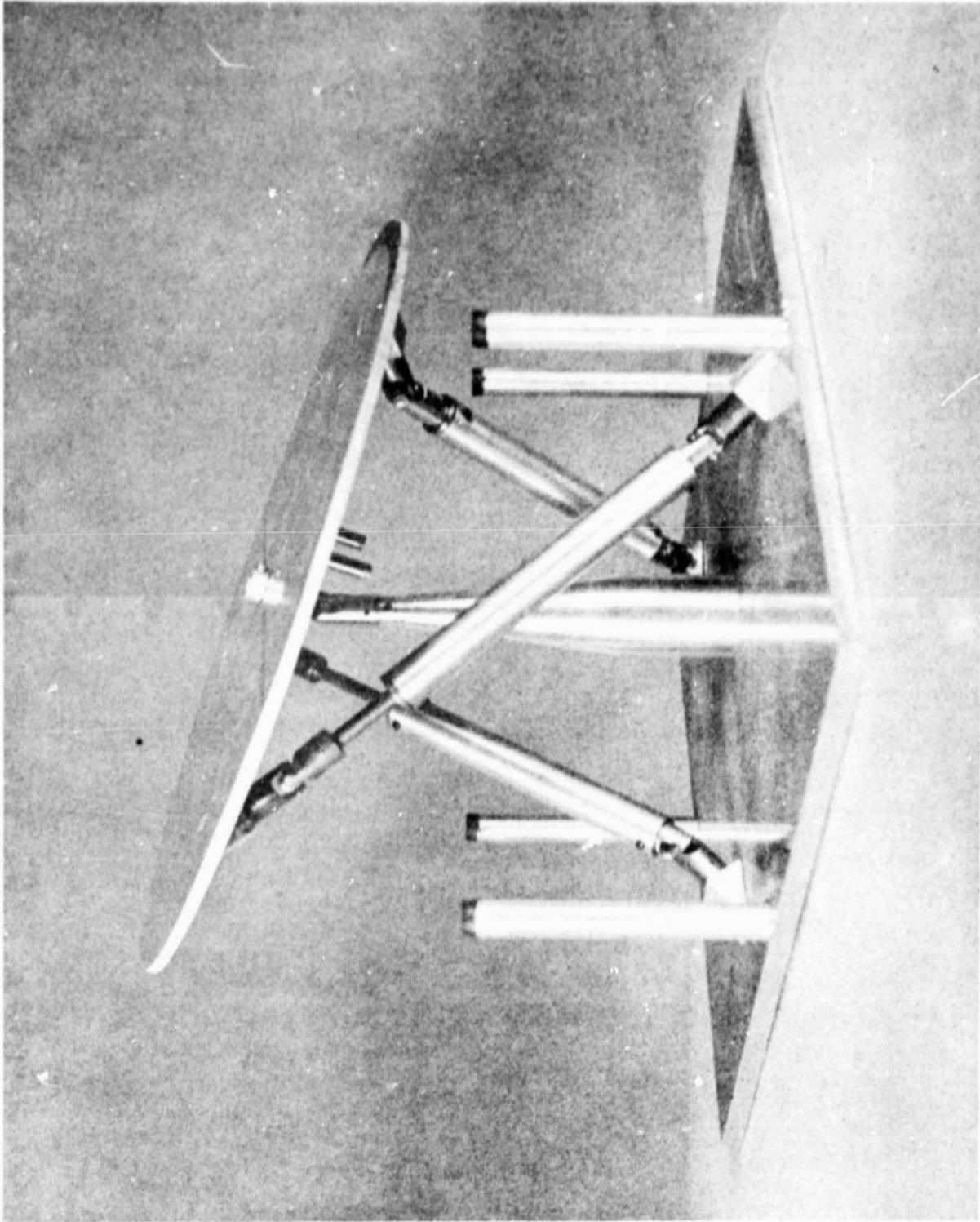


Figure 17. Kinematics of the 3 DOF Platform (Cont.) Pitch Down, Yaw Left

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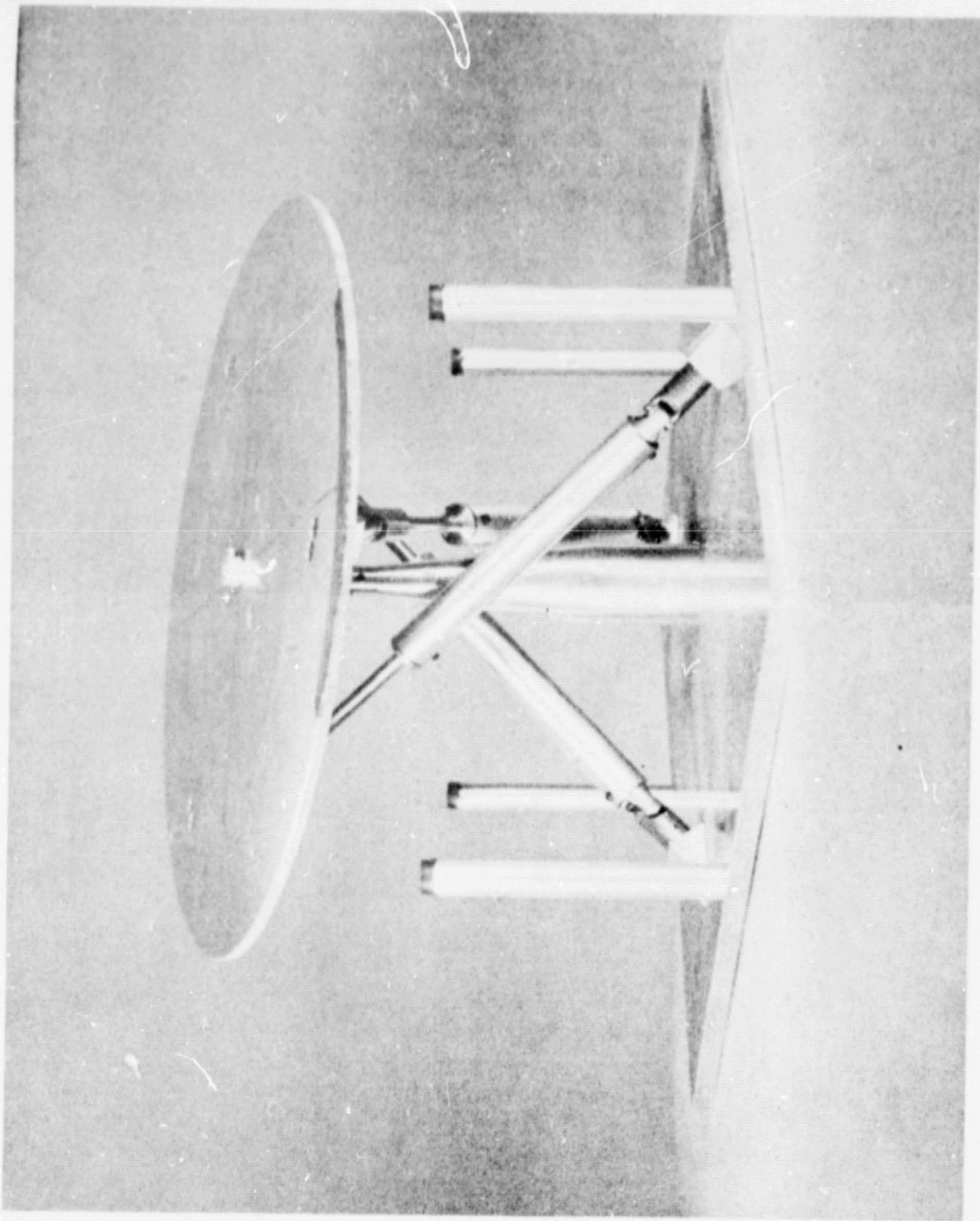


Figure 18. Kinematics of the 3 DOF Platform (Cont.) Pitch Down, Yaw Right

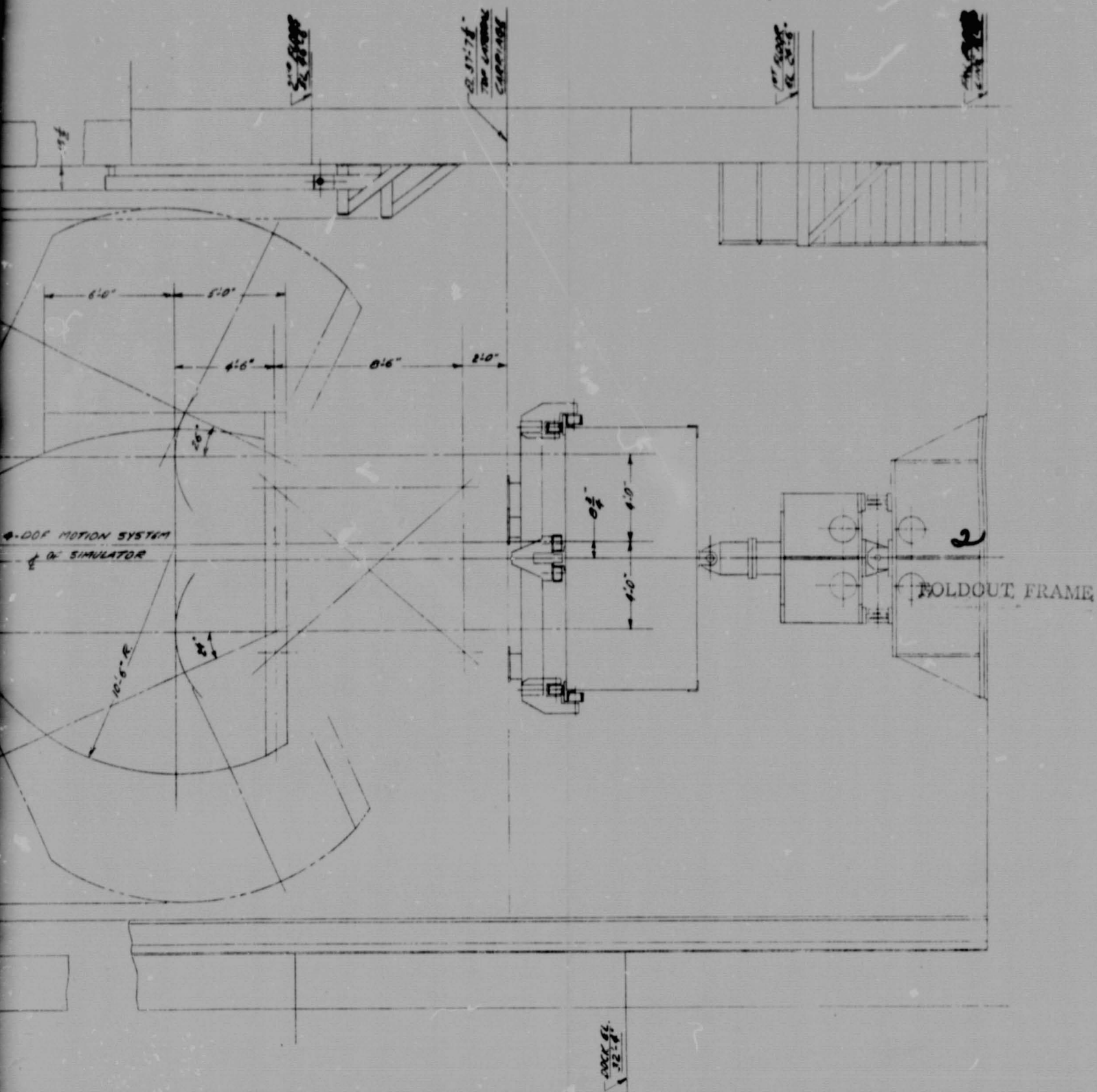
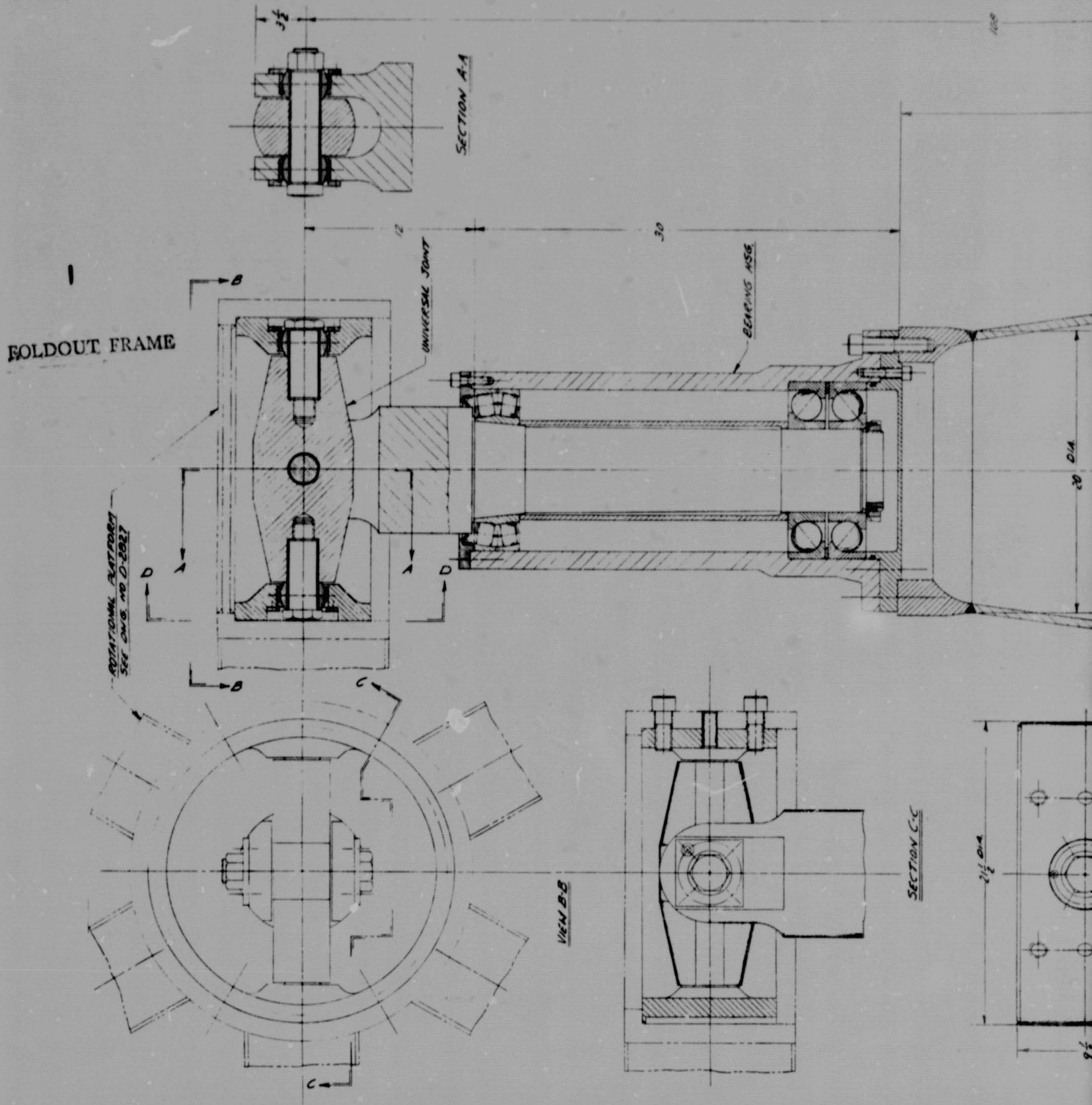


Figure 19. Layout of 3 DOF Platform

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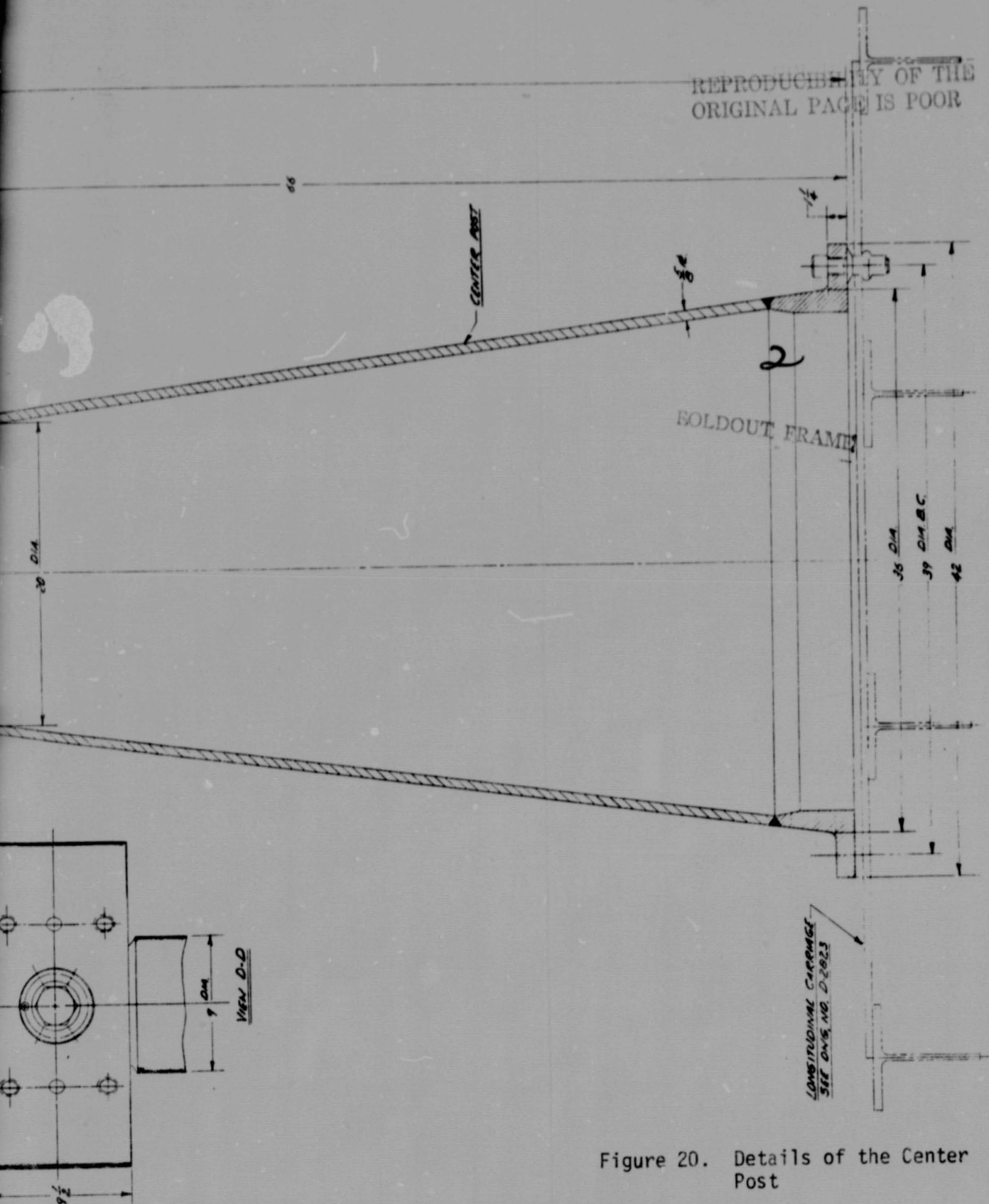


Figure 20. Details of the Center Post

86478

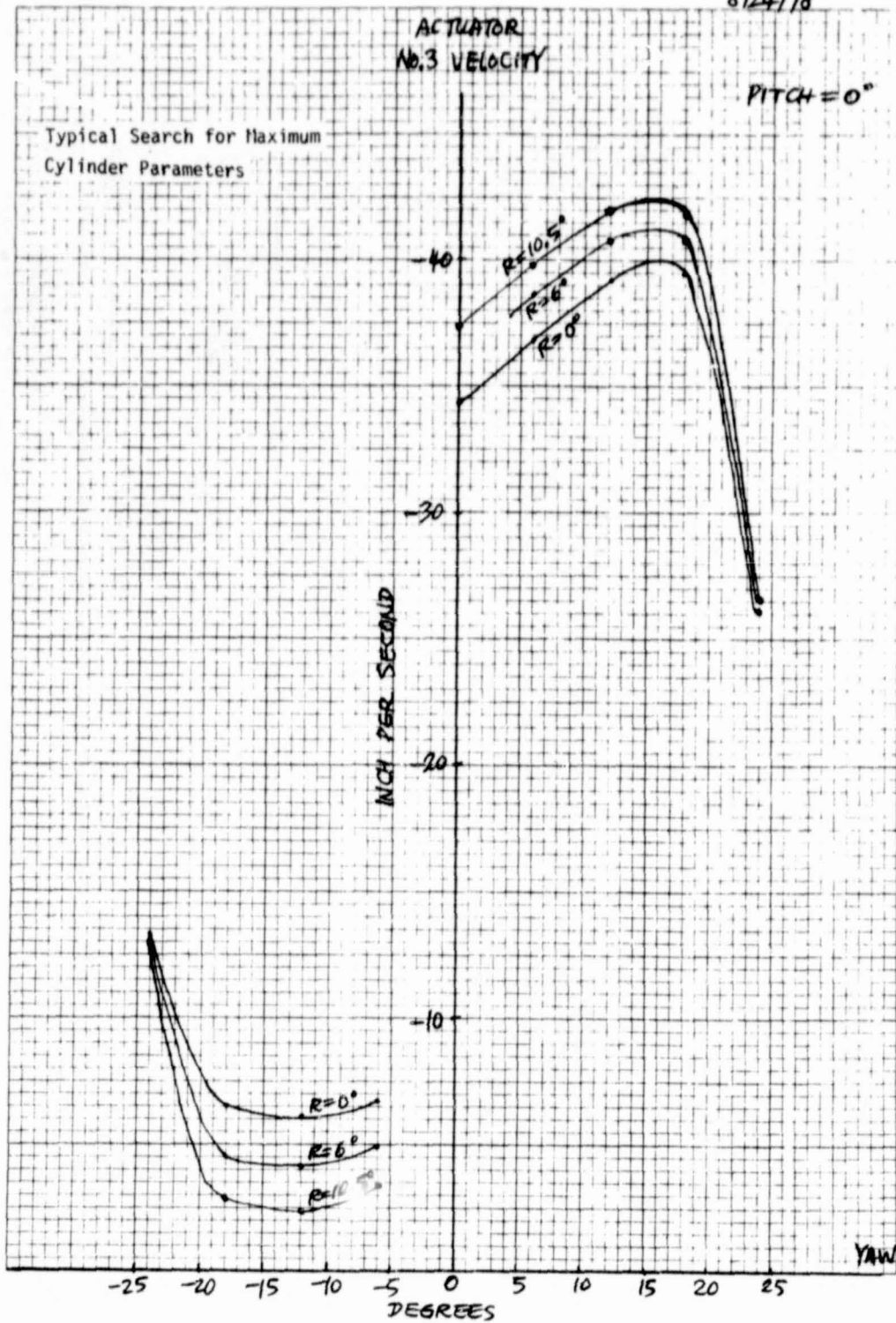


Figure 21. Typical Search for Maximum Cylinder Parameters

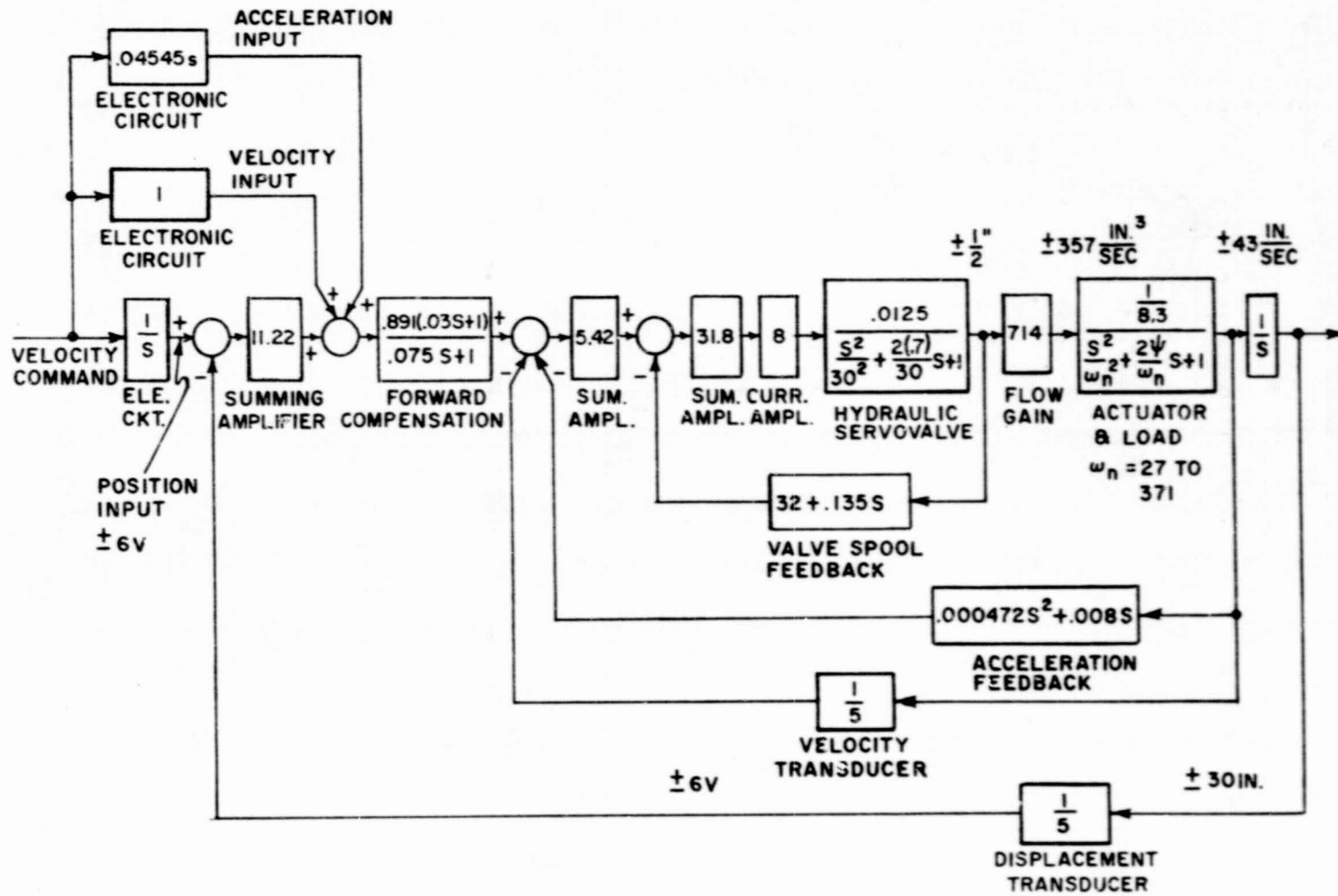


Figure 22. Control Block Diagram for 3 DOF Actuator

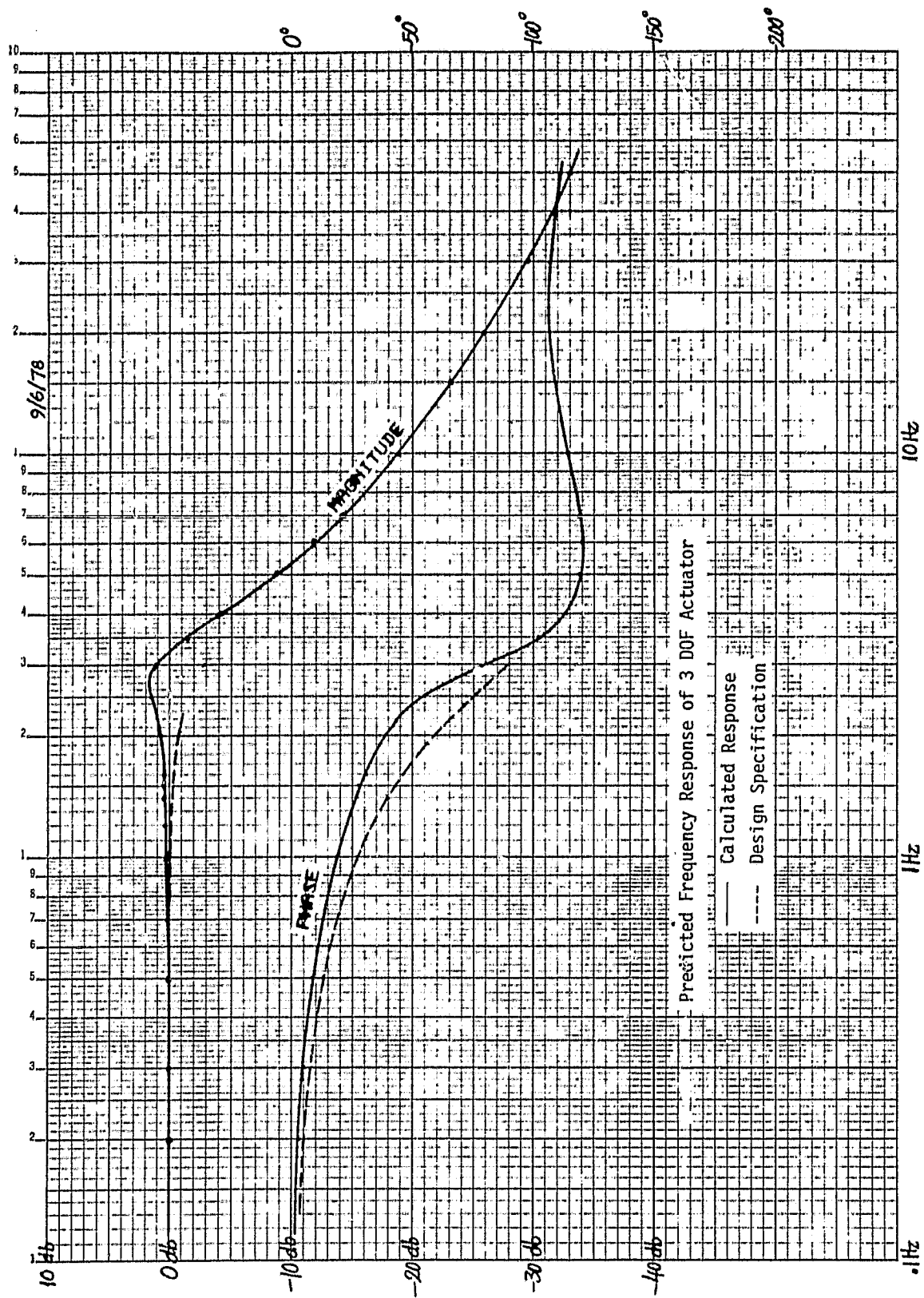
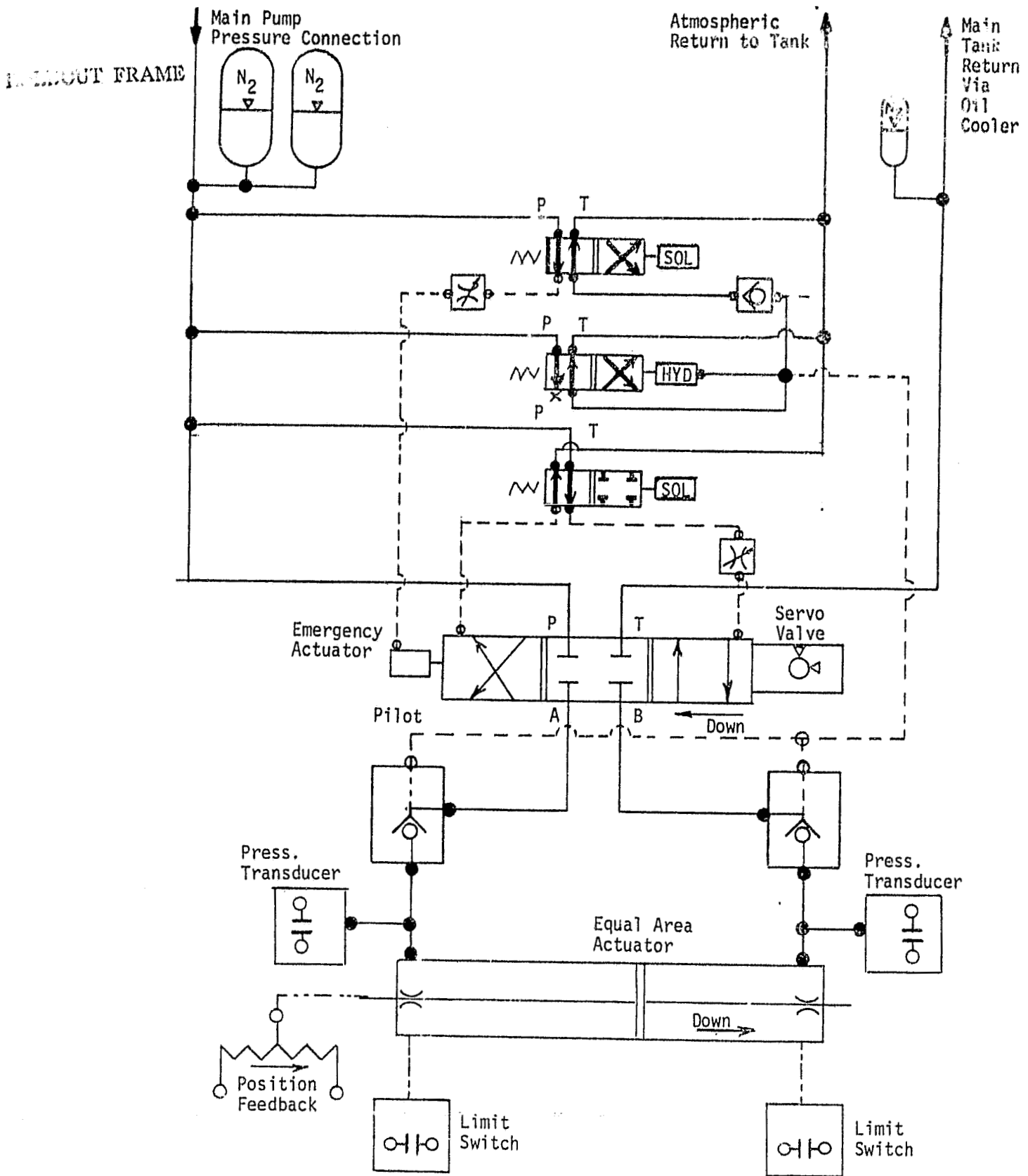
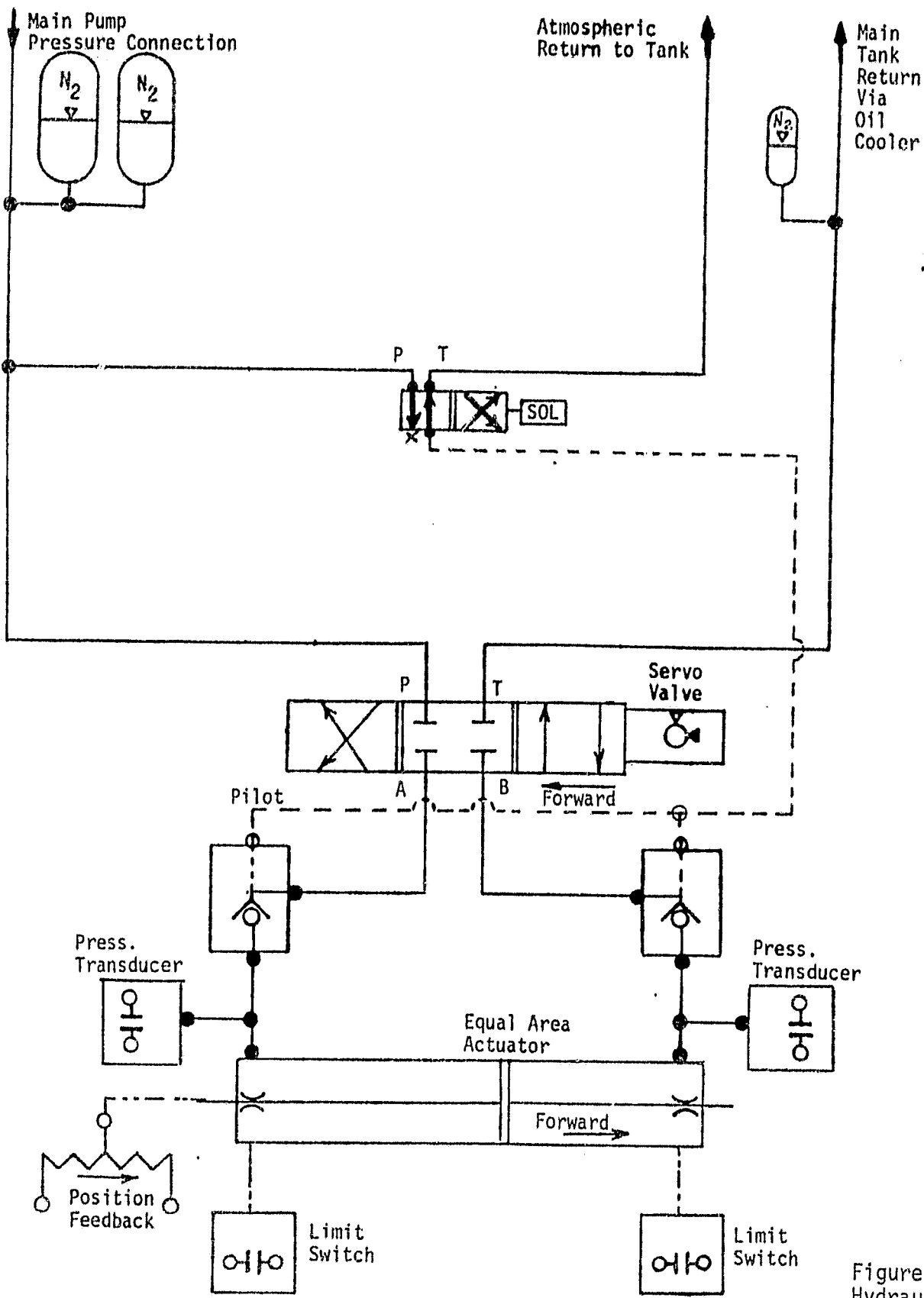


Figure 23. Predicted Frequency Response of 3 DOF Actuator



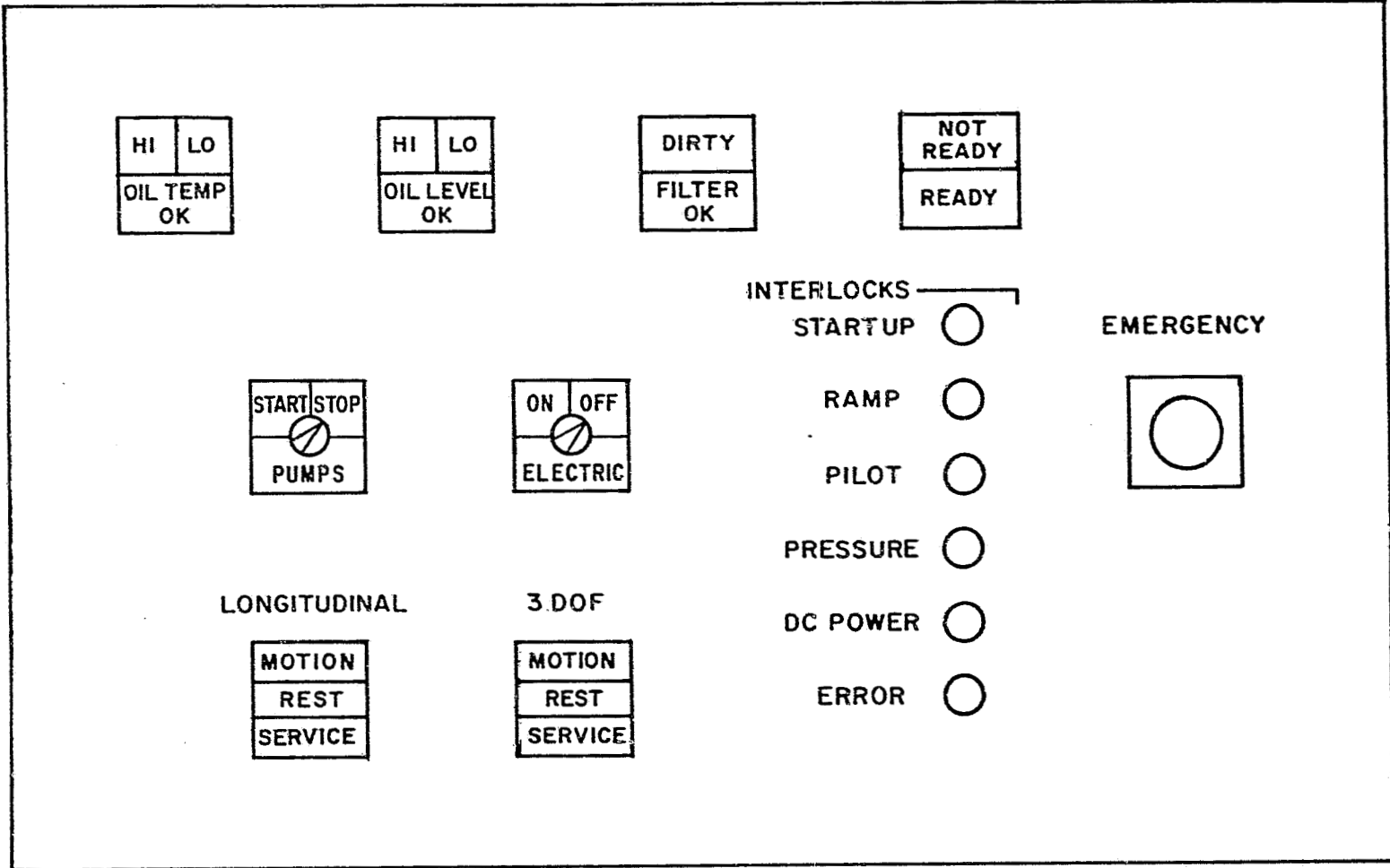
Hydraulic Schematic for Rotational Displacement Actuators
 (Typical for each of three actuators)

2



Hydraulic Schematic for Longitudinal Displacement Actuator

Figure 24. Hydraulic Power Supply Schematics



OPERATING PANEL

Figure 25. Preliminary Operators' Control Panel

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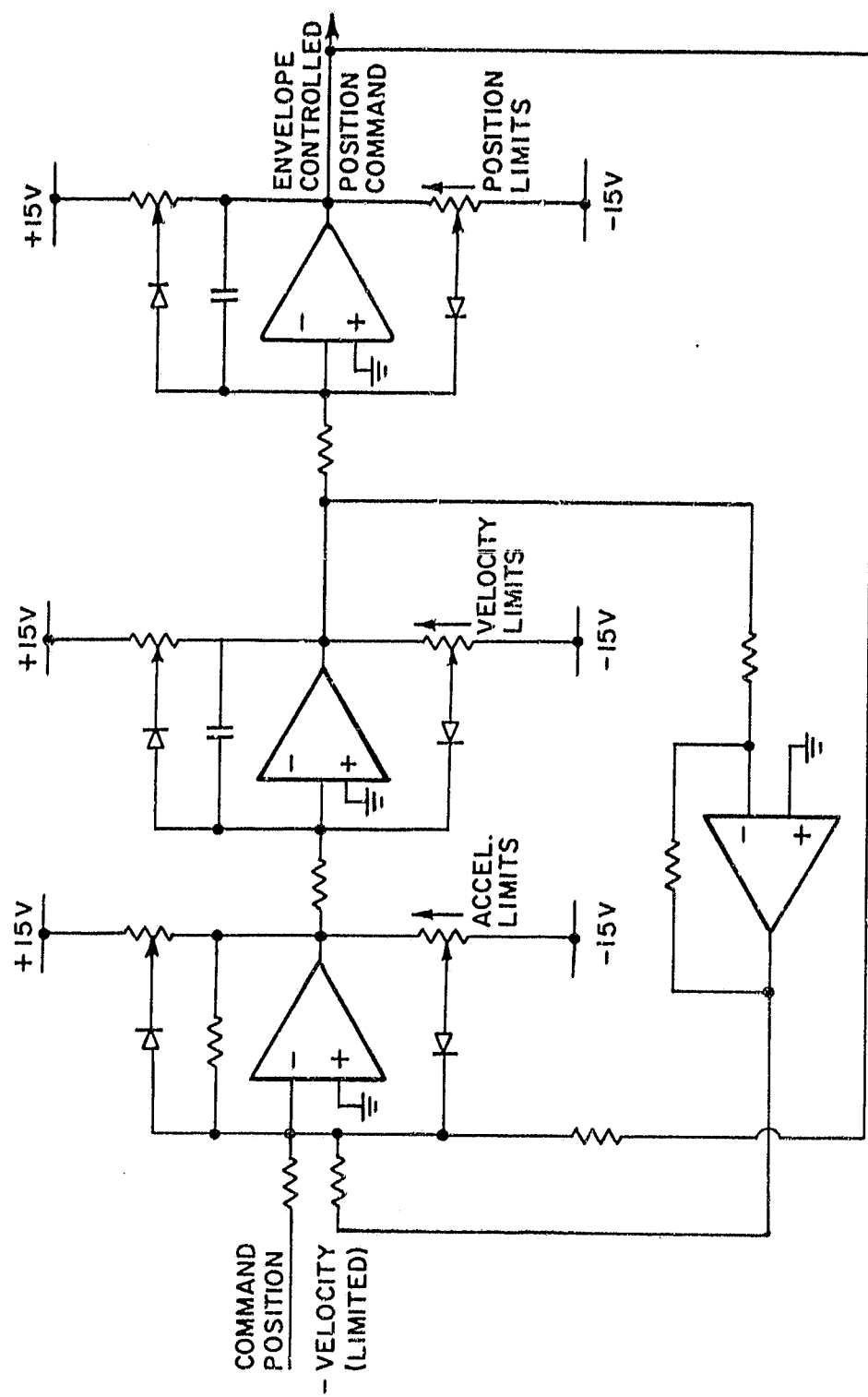


Figure 26. Performance Envelope Limiting Circuits

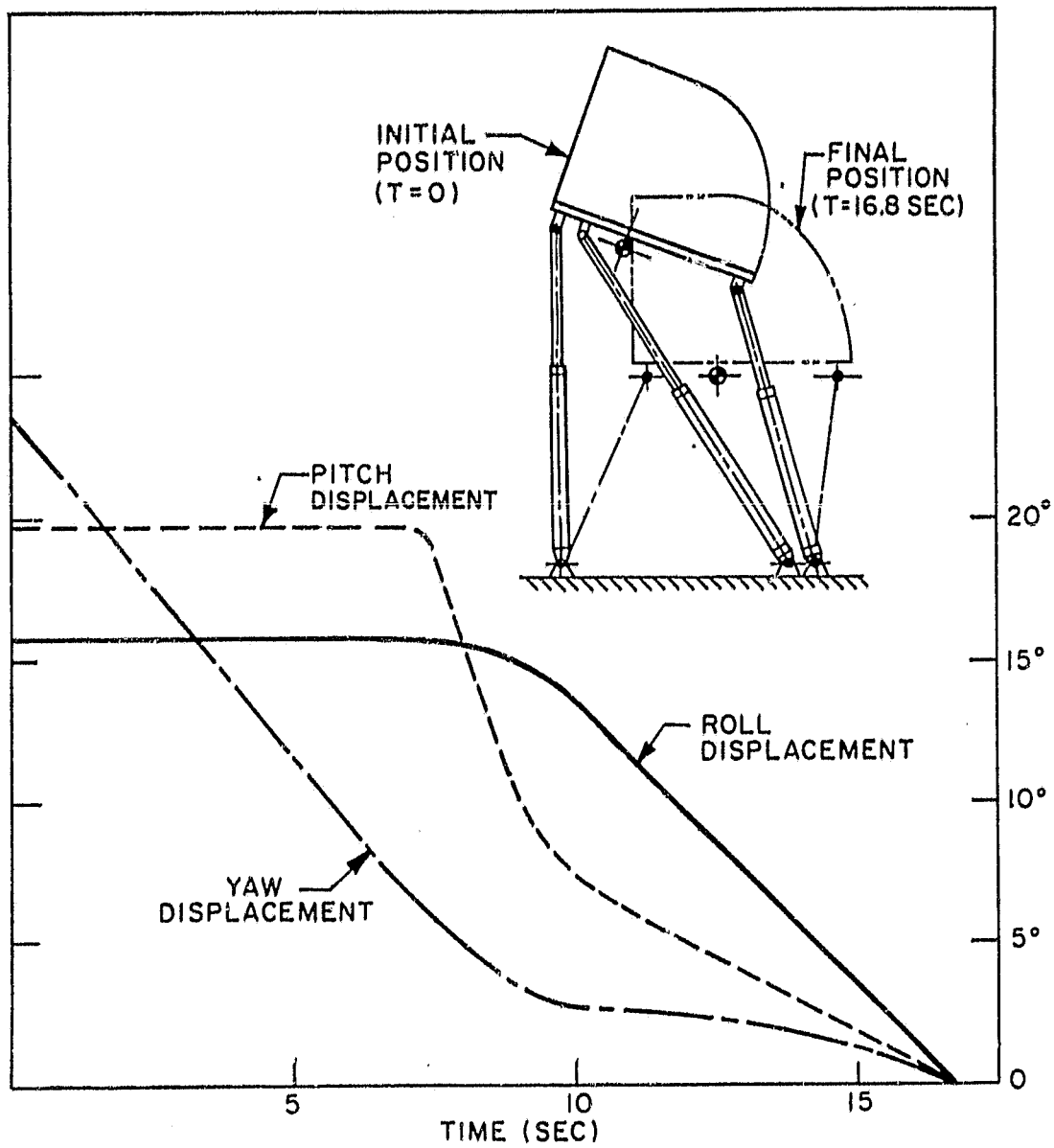


Figure 27. Time History of Emergency Shutdown

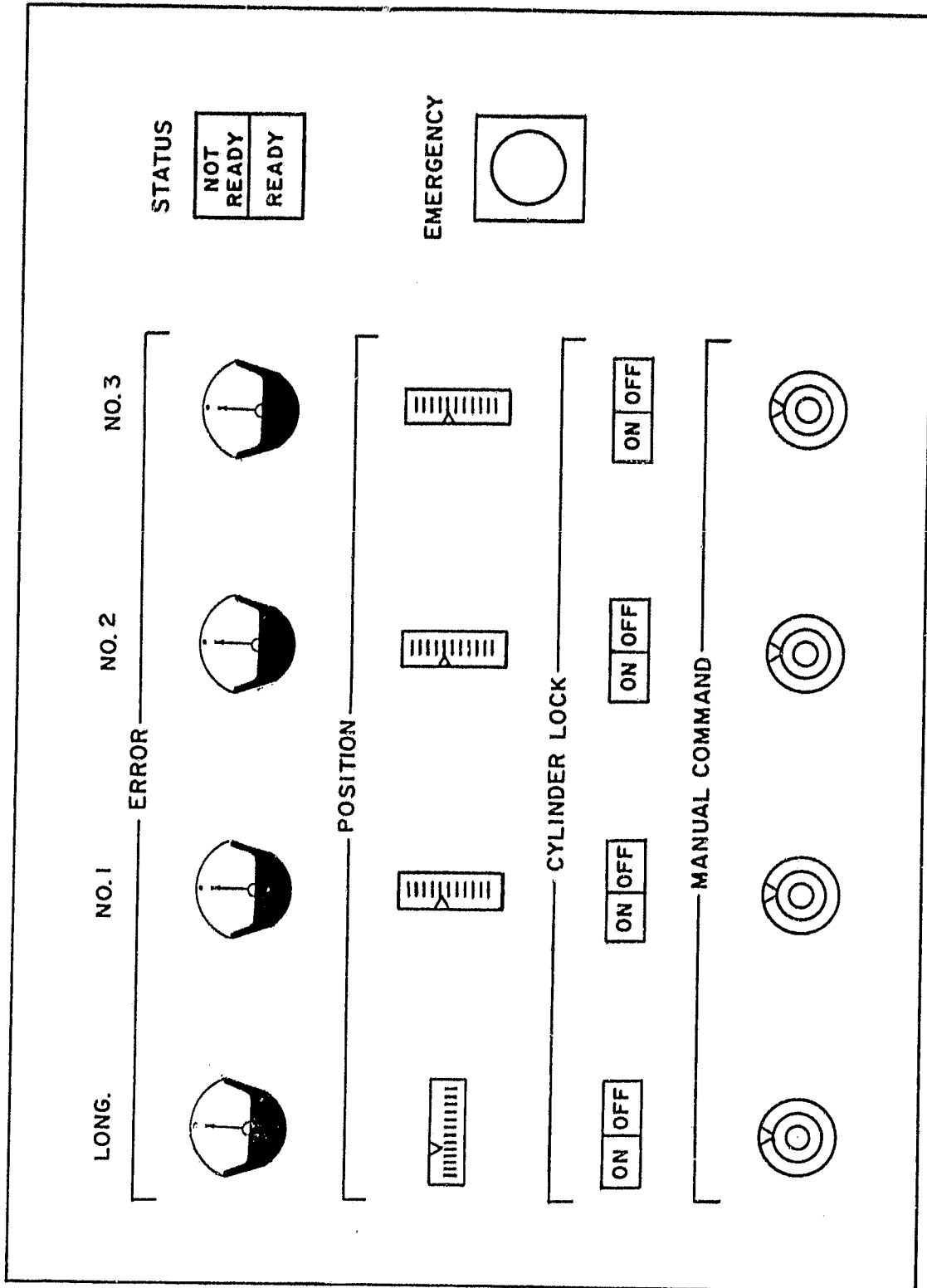


Figure 28. Preliminary Operators' Service Panel

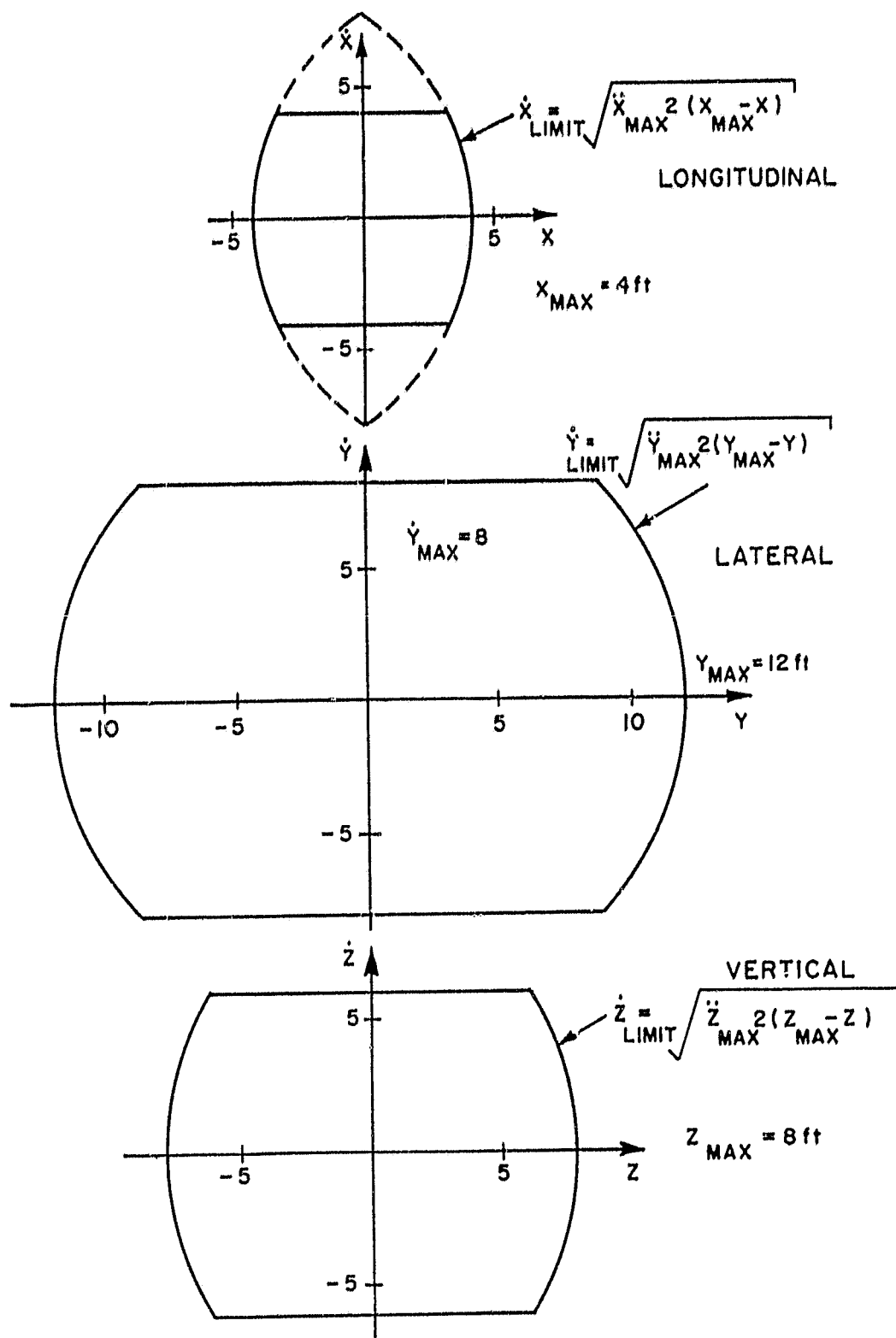


Figure 29. Longitudinal Position-Rate-Acceleration Envelopes

MAXIMUM ROTATIONAL PERFORMANCE ACCELERATION,
DECELERATION = 1.0 RAD/SEC² = α

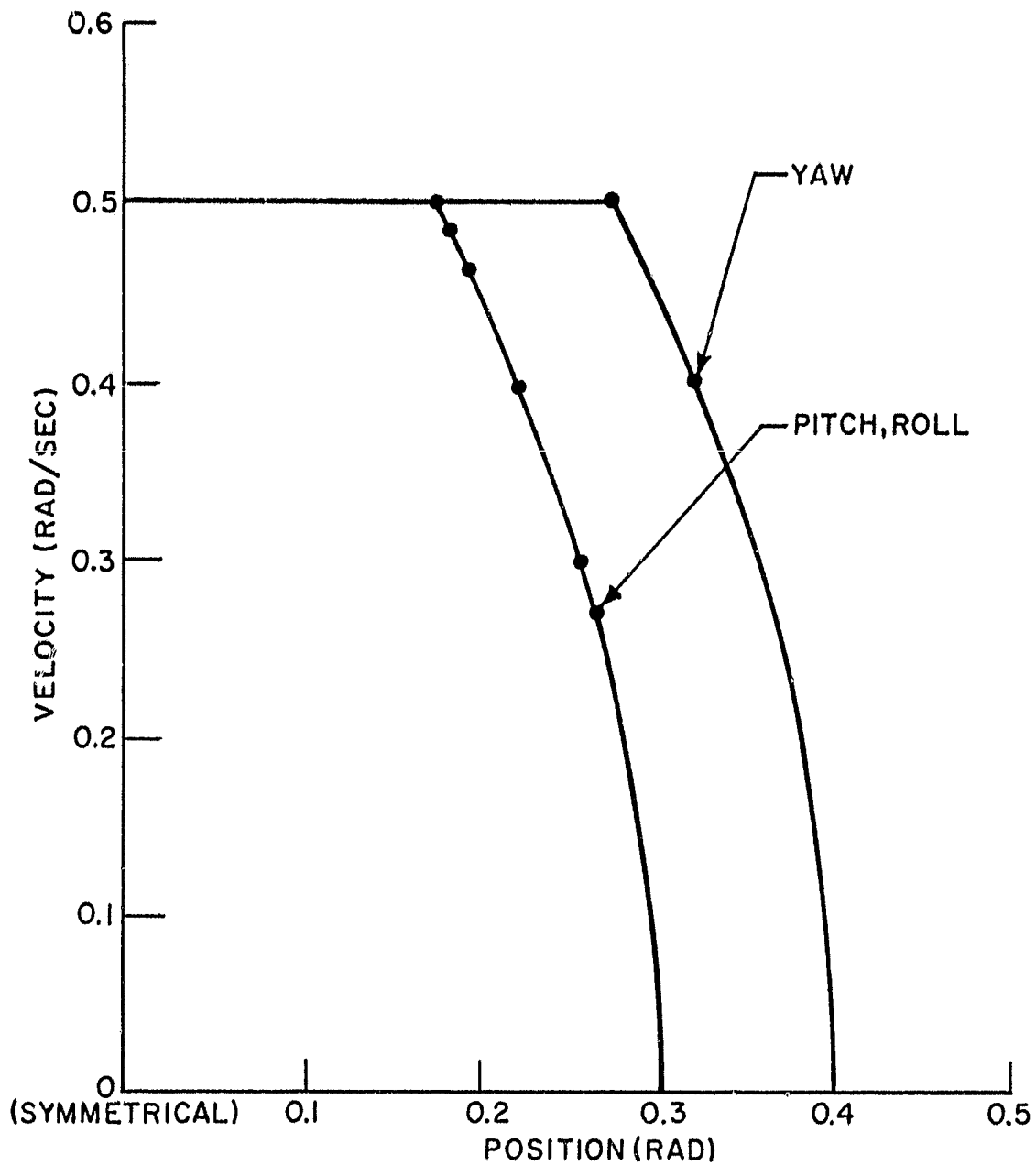


Figure 30. Rotational Position-Rate-Acceleration Envelopes

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TASK	MONTHS AFTER DATE OF CONTRACT																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Layout, Calculations	█																							
Tech. Report (Progress)				▲																				
Tech. Report (Final)																								
Acceptance Test Plan																								
Tech. Data Package																								
Final Dwg. & Assoc. Lists.																								
Instruction Manual																								
4 DOF Simulator																								
Final Design																								
Order Parts																								
Fabricate																								
Shop Tests																								
Ship																								
Field Installation																								
Acceptance Testing																								
Instruction																								

Figure 31. Time Schedule - Detail Design Construction and Test