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AFAPL-TR-64-129

DESIGN, FABRICATION, AND INSTALLATION OF SIX-DEGREE-OF-FREEDOM SPACE MAINTENANCE SIMULATOR

TECHNICAL REPORT AFAPL-TR-64-129

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THE MARTIN COMPANY

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Foreword

This is the final report prepared by Advanced Logistics Department of the Baltimore Division Martin Company, Baltimore, Maryland on Air Force Contract AF33(615)-1250, Project No. 8170, Task No. 817008.

This work was administered under the direction of the Air Force Aero Propulsion Laboratory, APFT, Wright-Patterson AFB, Dayton, Ohio, Mr. Chester B. May, Project Engineer.

This report covers work conducted from 26 January 1964 to 28 July 1964. Design, Fabrication and Test were conducted under the auspices of Mr. M. B. Goldman, Manager of Logistic Support, Baltimore Division, Martin Co.

Special acknowledgement is made to A. Hamilton, E. Wojcik, C. Jenkins, J. Messer and W. MacCrehan for their invaluable contributions as a team to the successful completion of this program.

Abstract

A 6-degree-of-freedom space maintenance simulator was designed, fabricated and installed for USAF Aero Propulsion Lab, WPAFB. The simulator supports a 180 lb subject and 110 lb back pack with unlimited freedom in pitch, roll, and yaw; horizontal translation on frictionless air pads over a 20 x 30 ft floor; and vertical translation on air bearings $\pm$ 18 in. from a nominal position. Also, included was a servo controlled work panel capable of horizontal translation simulating a 3K to 7K lb object in orbit. The work panel is suspended from a 20 ft span bridge crane with both axes controlled by servo amplifiers housed in a single rack. A 140 SCF air tank provides a low rate air spring for vertical translation.

This technical documentary report has been reviewed and is approved.

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I. Program Objective

The objective of this program was to design, fabricate and install a six-degree-of-freedom simulator for experimentation at AFAPL, Wright-Patterson AFB, Dayton, Ohio. The simulator would support a 180 lb man and a 110 lb self-maneuvering unit capable of unlimited rotation in pitch, yaw and roll, translation of 3 ft vertically; and horizontal translation on a 20 x 30 ft floor. Less than 1 lb. or 1 ft -lb. of torque applied for 2 sec. would be required to cause motion. In addition, a work panel simulating a 4 x 6 ft section of a 10 ft dia spacecraft was required to respond to a force input of 10 lb for 2 sec, moving horizontally to simulate the velocity of a 5,000 lb mass in orbit.

II. Design

A. General Description

1. Basic Approach

The design approach was based upon the proposal submitted in response to RFP No. 33-(657)-63-5574, Martin Engineering Report 13074P. There are two major functional units, the simulator which supports the subject with six degrees of freedom and the bridge crane work panel which simulates the work object in space with two degrees of freedom.

Figure 1 shows the complete installation at Wright-Patterson AFB. The simulator is at the left, the bridge crane work panel at the right and the control rack, air dryer and simulator air tank are in the rear.

2. Simulator

The simulator (Fig. 2) is designed to support a spacesuited subject and a self-maneuvering unit in a structure which permits six degrees of freedom, unlimited rotation about the pitch, roll and yaw axis and translation along the horizontal axis within the limits of a 20 x 30 ft floor and vertically $\pm$ 18 in. from a nominal position with the CG 69 in. from the floor. The simulator is 92.6 inches high, 102.5 inches long and 72 inches wide overall, and weighs approximately 325 lbs.

Manuscript (Martin Report ER 13604) released by the author August 1964 for publication as an RTD Technical Documentary Report.

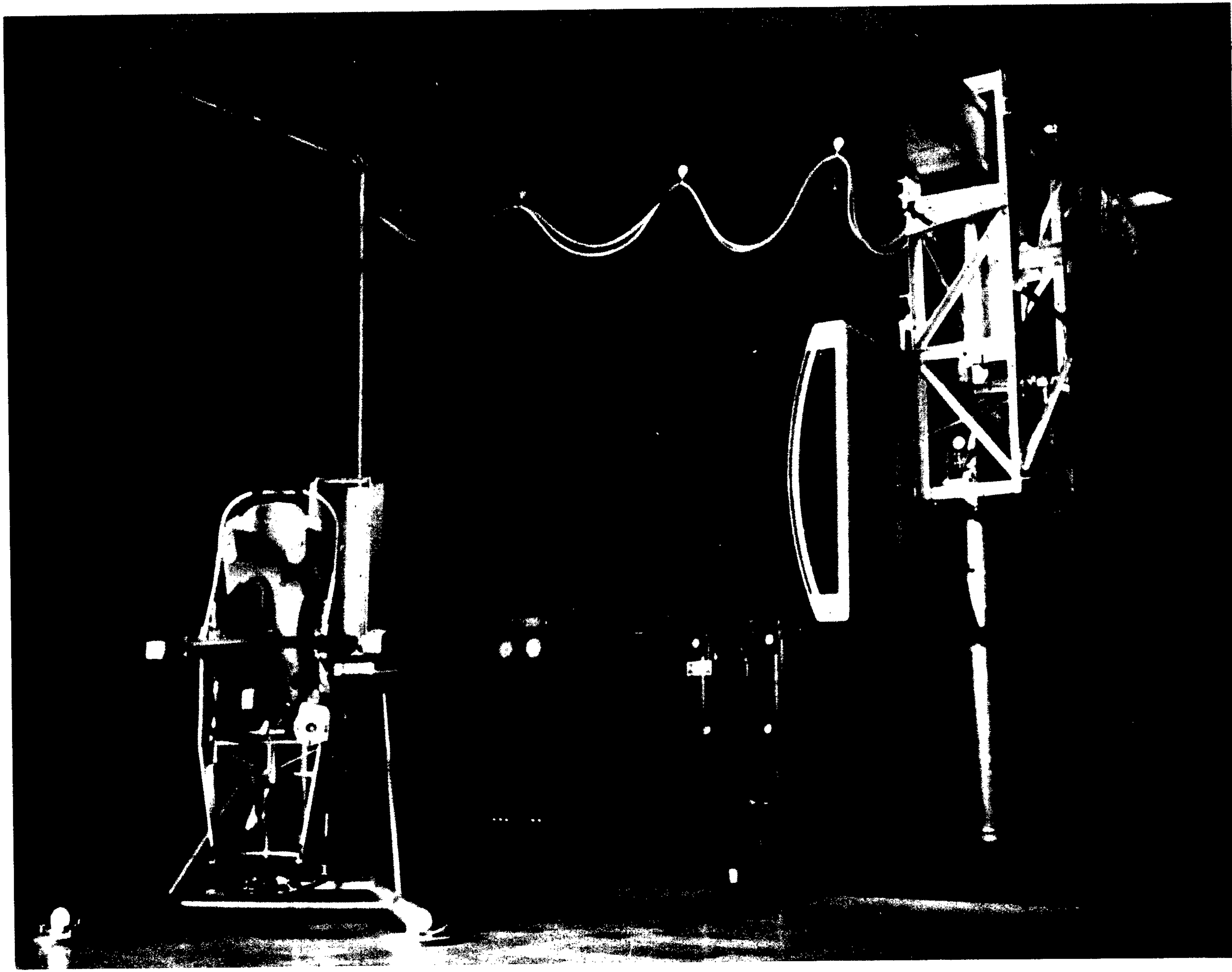


Figure 1. Simulator Installation

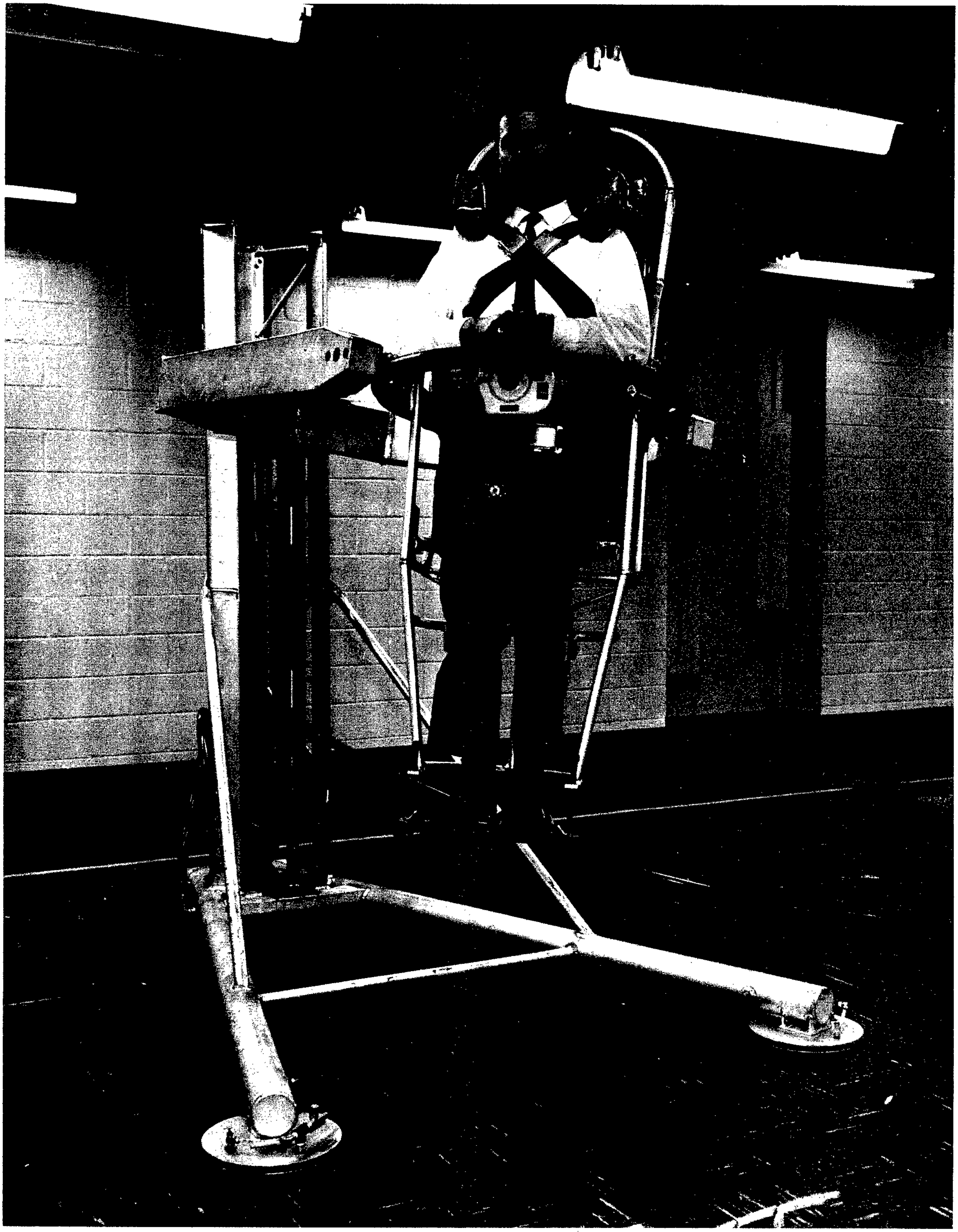


Figure 2. Simulator With Subject

The criteria specified in the contract called for the simulator to support a 180 lb subject wearing a 110 lb back pack. The sensitivity required was 1 lb or 1 ft.-lb applied for 2 seconds to cause motion in any degree of freedom. The 6 degrees of freedom are provided by gimbals rotating on ball bearings for rotational movement and air bearings for translation. From the subject outward the simulator consists of:

a. The body support cradle which holds the subject and back pack rigid, allowing only motion of the subject's arms and head.

b. The yaw ring which holds the body support cradle on two cradle carriers. Each carrier has ball bearings which run on tracks on the yaw ring allowing rotation around an axis thru the center of the ring normal to the plane of the ring. Three tracks are provided, top, bottom, and inside to allow rotation with any orientation of the ring.

c. The roll yoke supports the yaw ring at trunion points which establish the pitch axis. The pitch bearings are bolted to the outer ends of the U shaped yoke which is deep enough to allow unlimited rotation of the subject in pitch. The roll yoke is supported at the rear in the horizontal plane by the roll bearings which provide unlimited rotation in roll with any pitch or yaw attitude.

d. The elevating mechanism holds the roll bearing housing and provides vertical translation. The couple created by the cantilevered roll yoke is reacted by air bearings which allow frictionless vertical translation. The lifting force is provided by a piston and cylinder which are directly coupled by a 3/4 in. ID flex hose to a 140 SCF air tank to form an air spring with a very low spring constant. The system pressure is maintained constant to provide a lifting force in the cylinder exactly equal to the total weight to be lifted. Since the ratio of cylinder volume to total system volume is small, pressure changes caused by movement of the piston from one end of the cylinder to the other will be small, therefore the restoring force when the piston is displaced from nominal will be low and concomitantly the force required for the subject to cause displacement will be low.

   The cylinder is rigidly attached to the roll bearing housing and is used to transmit the couple, produced by the cantilevered load, to the vertical air pads.

   One air pad is provided to the rear of the bottom of the cylinder to take the backward component and two airpads equally spaced from the roll axis and canted 30 degrees back from the transverse axis react the forward component. The piston is supported by a spherical bearing on the top of the piston rod; the bottom of the piston rod is supported in another spherical bearing so that alignment of the cylinder with respect to the vertical air pads is not critical and no horizontal loads are imposed on the piston.

e. The vertical support structure provides the tracks for the vertical air pads and the load path to the base. The structure is essentially two trusses forming a blunt 60^{o} V with the opening of the V toward the front. The lower air pad rides in a track attached to a channel forming the point of the V,and the two upper air pads ride on tracks attached to Z members at the top of the V. The Z members and channel are bent up sheet metal for minimum weight. The open part of the V is braced at the top, bottom and middle of the structure for rigidity; and diagonal braces are provided to each leg of the base.

f. The base of tubular aluminum is V shaped to provide a tripod support for the simulator. The vertical support structure is welded to the point of the V which also provides the support for the elevating mechanism piston rod. Three HOVAIR pads are mounted on the base to provide frictionless movement across the floor. The base is used as a plenum chamber for the air supply to the HOVAIR pads and to the vertical air pads. The air from the base to the vertical pads is fed thru a flexible hose.

g. Separate air supplies are provided for the air pads and the cylinder because of differences in working pressures and flows. The air supply to the air bearings is unregulated shop air at $>$80 psig fed thru the facility Lazy Arm into the base of the simulator. The air for the elevating mechanism is fed thru a control valve, filters and dryers into the 140 cu ft surge tank. From the tank it is fed thru 3/4 ID flexible hose which is hung on the lazy arm and is connected to the bottom of the piston rod. A valve is provided on the simulator for shut off without waiting for the tank pressure to drop. The pressure in the elevating mechanism under full load is approximately 59 psig. Filtering and drying is required because the piston which,is air lubricated by the flow thru the tank, has only .0015 in clearance.

3. The Bridge Crane Work Panel

The bridge crane work panel (see fig. 1) is a 4 x 6 ft section of a 10 ft dia cylinder supported on a 20 ft span bridge crane. The center point of the work panel is capable of translating horizontally over an 11 x 19 ft floor area. The work panel must simulate a 5,000 lb mass and respond to a 10 lb force applied for 2 seconds.

The bridge crane work panel is servo driven in both axis and consists of:

a. The work panel, of typical aircraft or missile structure, is a segment of a 10 ft dia smooth cylinder with its axis horizontal and 69 in. from the floor. The structure is designed so that access panels and "internal" structures can be added to simulate specific space maintenance situations.

b. The work panel vertical support holds the work panel on a parallelogram which restrains the work panel motion to translation in the horizontal plane relative to the support. The force sensing system of hydraulic pistons and pressure transducers, mounted on the vertical support, is coupled to the work panel thru a mechanical linkage which transmits the force with little relative motion. Also mounted on the vertical support are the transverse motor drive and bumper. The motor drive is a servo controlled electric motor driving a rubber wheel thru a two step chain and sprocket drive. The vertical support rides on casters along the bridge crane box beam. Although limit switches are provided to electrically limit the travel, spring bumpers are provided to absorb the shock in the event of servo malfunction and in normal high velocity travel into the limit.

c. The box beam holds the vertical support structure, providing the path for transverse motion of the work panel. The beam is a 12.5 in. square box 20 ft long built up of aluminum angles and skin. Forming a "T" at each end are 43 in. long channels which hold the crane carrier wheels. Along the top of the box, the drive shaft is carried in pillow blocks with rubber drive wheels on each end. At the left and on a small platform above the beam is mounted the longitudinal motor drive which is similar to the transverse drive. Bumpers are provided at both ends of the box beam to avoid twisting during impact.

d. The crane rails are standard crane rails built up of angles and plates to provide an I beam with a narrow lower flange on which the crane carrier wheels ride. The longitudinal drive wheels are held up against the bottom of the rails under a fixed load. End plates limit the travel and absorb the bumper impact.

e. The bridge crane servo system has two channels, longitudinal (A) and transverse (B), which are capable of independent control. Both channels sense the forces applied to the work panel, integrate the impulse and drive the work panel horizontally at a velocity equivalent to the detected impulse acting upon a mass in orbit. Controls are provided for adjusting the simulated weight from 3K to 7K lbs., for positioning the work panel and holding it in place while the experiment is being prepared. A remote jack is provided for operation either from the console or from alongside the simulator. All electronic components except the necessary motors, limit switches and sensors are housed in a standard single rack with access for maintenance from the front and rear. The top of the rack is occupied by the controls for the elevating mechanism air system.

B. Simulator Design

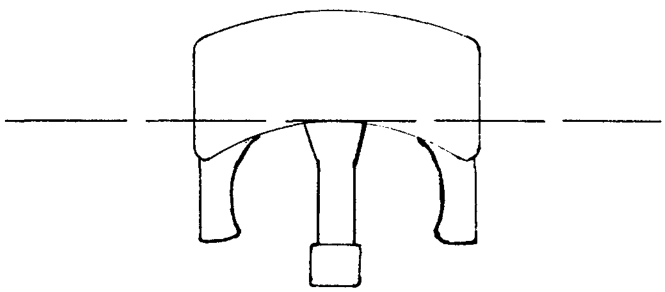
1. Body Support Cradle

The body support cradle is required to hold the subject and back pack rigidly in place allowing only arm and head movement. Unwanted rotation will occur when very small movements of the subject cause unbalance torques greater than 1 ft lb. To minimize this, only necessary movement can be permitted. Previous experience with development and use of the Martin in-house 5^{o} simulator and the NASA-MSC 5^{o} simulator has shown that an efficient restraint could be accomplished with a moulded fiberglass corset and straps supporting the subject in a welded tubular cradle. Additionally, the cradle could be designed to support the back pack separately so that the subject is not required to sustain the full weight during the time he is installed.

The envelope and CG of the self-maneuvering unit were obtained from the Air Force project office and used as the criteria for cradle design. The location of the CG and dimensions of the subject were obtained from references 1 and 2. It was necessary to correct the available data for the position of the subject.

The available data was based on a position with the subject erect with hands at the sides. It was felt that a more realistic position because of the balance criticality was with the hands outstretched to reach the work panel. For simplicity the location of the CG was calculated from data on body segments in reference 3 with the upper arm at 45^{o} from the vertical and the forearm horizontal. See Appendix A for detailed calculations. A subject height and body weight of 186.2 lbs and 72.3 inches was used because this represented the mean $+1\sigma$ for subject size variations and it was close to the 180 lb contract requirement. The new CG position was 5.0 in. from the back plane and 40.4 in. from the floor or 1.3 in. further forward and .6 in. higher than the normal erect position.

The relative locations of the subject and back pack were determined from the relationships of the back plane of the subject and the back pack and the points of contact between the shoulders of the subject and the over the shoulder portions of the pack. The back pack back plane was taken as the vertical plane tangent at the pack centerline to the inner curve of the horizontal cross section. The subject and back pack



back planes were assumed to be parallel with the subject plane 0.5 in. forward to allow for the corset structure. The back pack, when in place, touches the subjects shoulder at the high point of the cushion on the shoulder piece. It was estimated that the contact point was 0.5 in. above the acromion (shoulder point) so 0.5 in. was added to the acromial height from reference 1.

The back pack CG location was referenced to the bottom and rear surfaces. Envelope drawings were scaled to locate the CG relative to the shoulder contact point and a combined CG was calculated at 1.5 in. forward from the subject back plane and 44.5 in. from the floor for the man and back pack.

Layouts were made utilizing this CG location and the back pack envelope to obtain the optimum positions for the cradle members required to support the subject and back pack within the smallest possible envelope. This requirement was imposed because of the problem of increased weight and inertia in the other simulator assemblies which would be required to support a larger cradle assembly.

The basic cradle configuration is a basket completely enclosing the back pack, holding it rigidly in place. The subject is then held separately in the proper relative position. Figure 3 shows the overall cradle structure. In addition, there are two horizontal tubes running between the subject and the back pack to close the structural load path between the sides and to support the corset assembly. The upper tube runs in the space between the back pack and the small of the subject's back. Measurements of anterior-posterior dimensions from reference 1 were used to determine the 1.7 in. depth of available space for this tube. The lower tube ran about halfway between the hips and knees. The subject stands on an adjustable foot plate with his feet and knees strapped in place (Fig. 4).

With the basic cradle configuration established detailed layouts of the cradle were made and a diameter of 30.0 inches established for the yaw-ring.

It was necessary to allow for adjustment of the subject and back pack to obtain a balance with a range of subjects and positions varying from the nominal. Based upon an analysis of the expected tasks the subjects would perform and our previous experience, it was decided that a range of adjustment of ± 1.5 in. vertically and fore and aft and .5 in. transversely would be adequate. The reduced transverse adjustment was based upon the probable performance of tasks immediately in front of the subject without extension of the arms to the sides.

Transverse adjustment is accomplished by moving the subject and corset on the two horizontal support tubes. Figure 5 shows the clamps holding the corset on the lower support. Similar clamps, accessible from the front hold the corset to the upper support tube (Fig. 6).

Fore and aft adjustments are made by sliding the entire cradle in the carriers in the slots provided (Fig. 6).

Vertical adjustments were made by adjusting the back pack, the corset and the foot plate. The corset adjustment is made in 1 in. increments. The back pack is adjusted by screwing the support rods (Fig. 5 shows the lower support) up or down to position the pack and clamp it into place. The foot plate is adjusted by screwing the support rods up or down over a range of 6 in. With the shoulders fixed relative to the back pack,a 6 in. foot movement corresponds to a variation of 3 in. in subject CG locations.

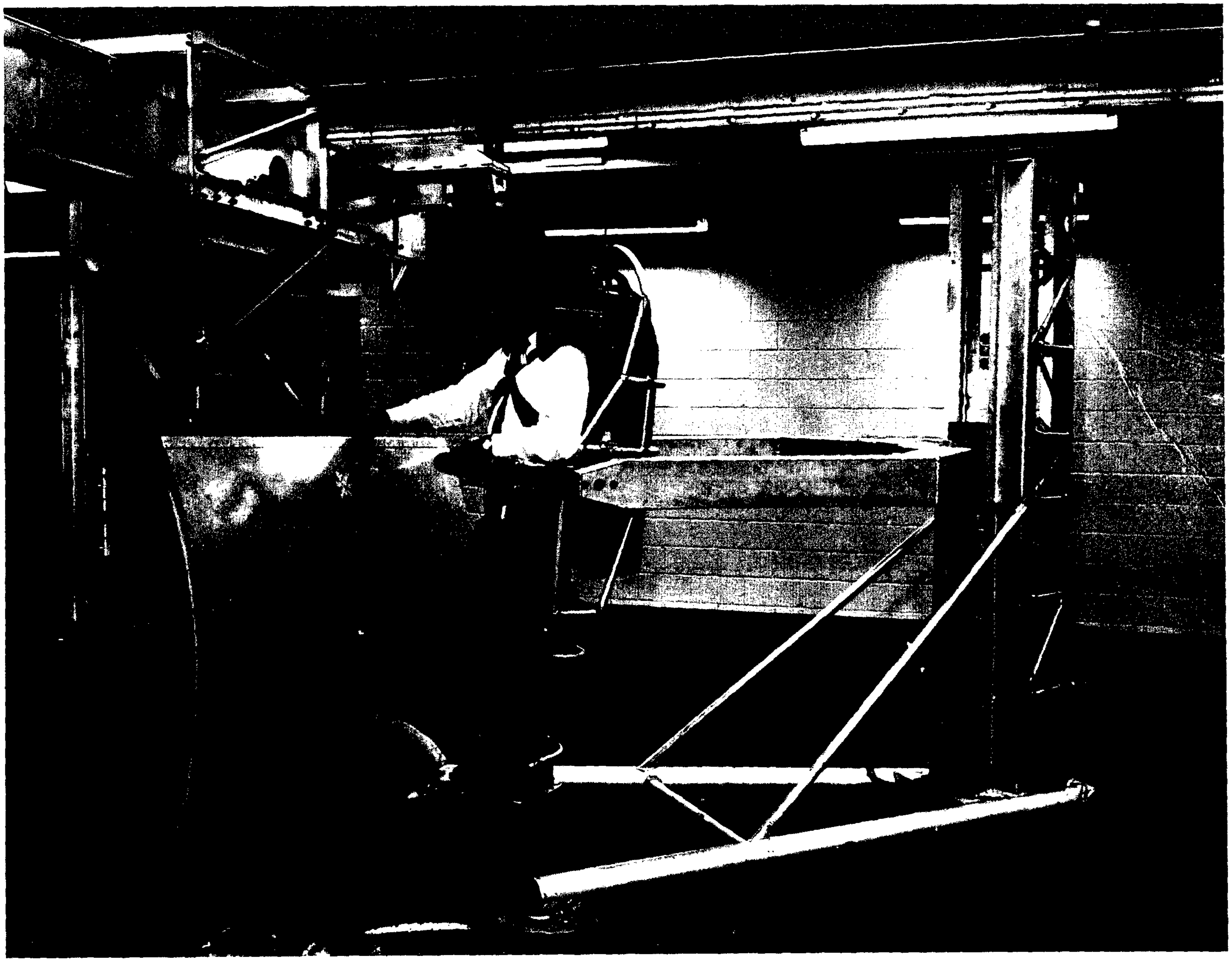


Figure 3. Simulator and Work Panel (Baltimore Installation)

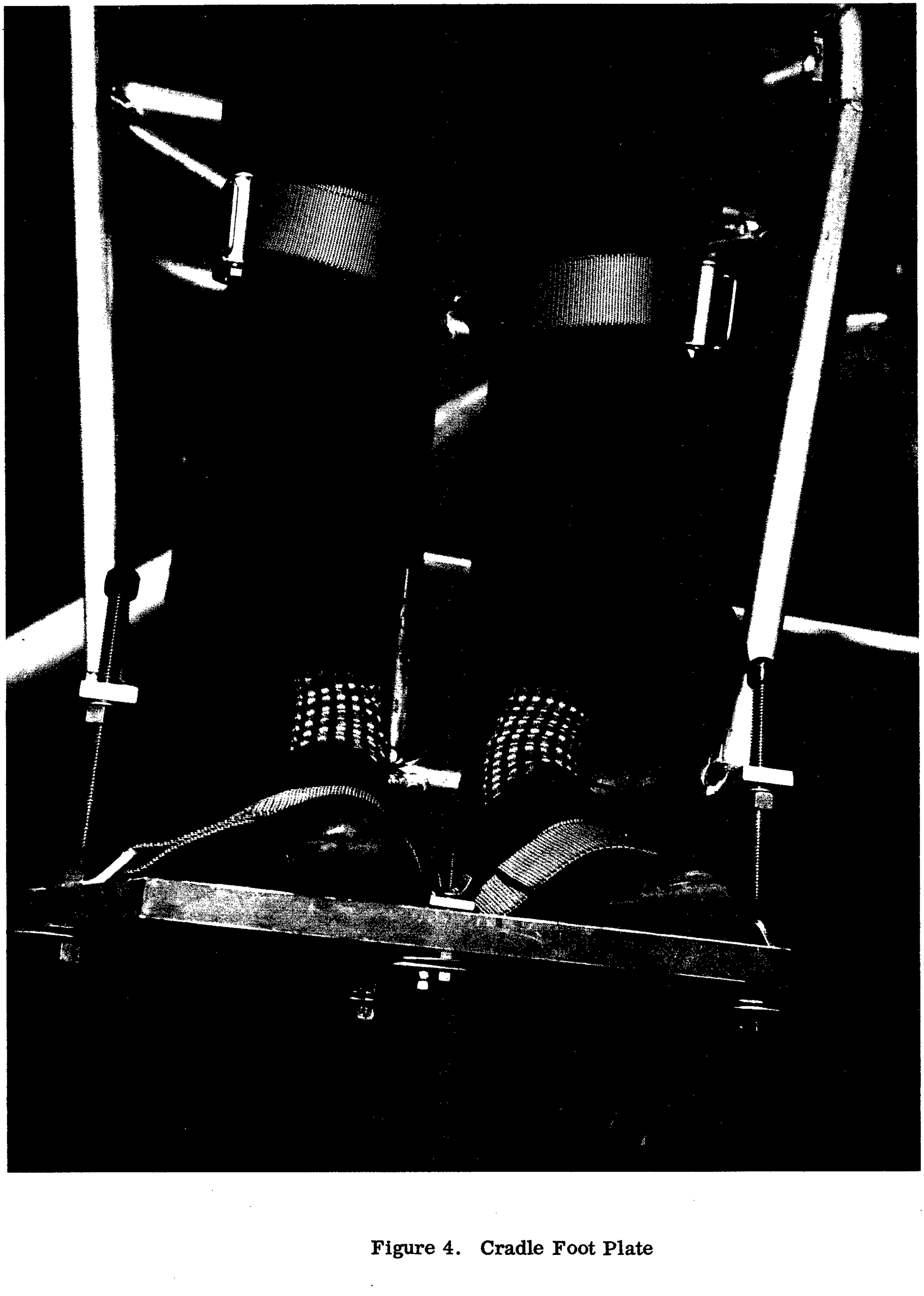


Figure 4.   Cradle Foot Plate

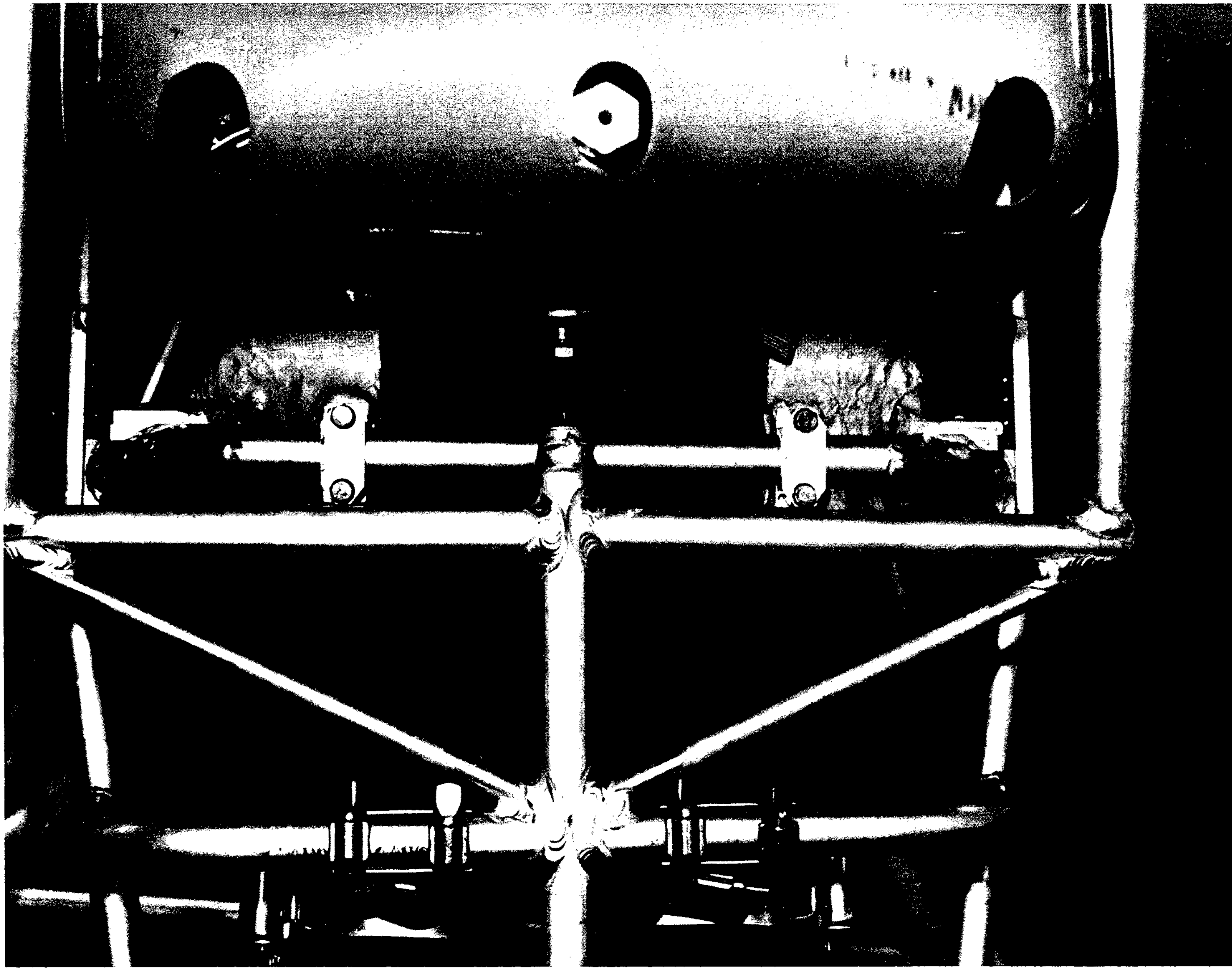


Figure 5. Cradle Corset Lower Support (Rear View)

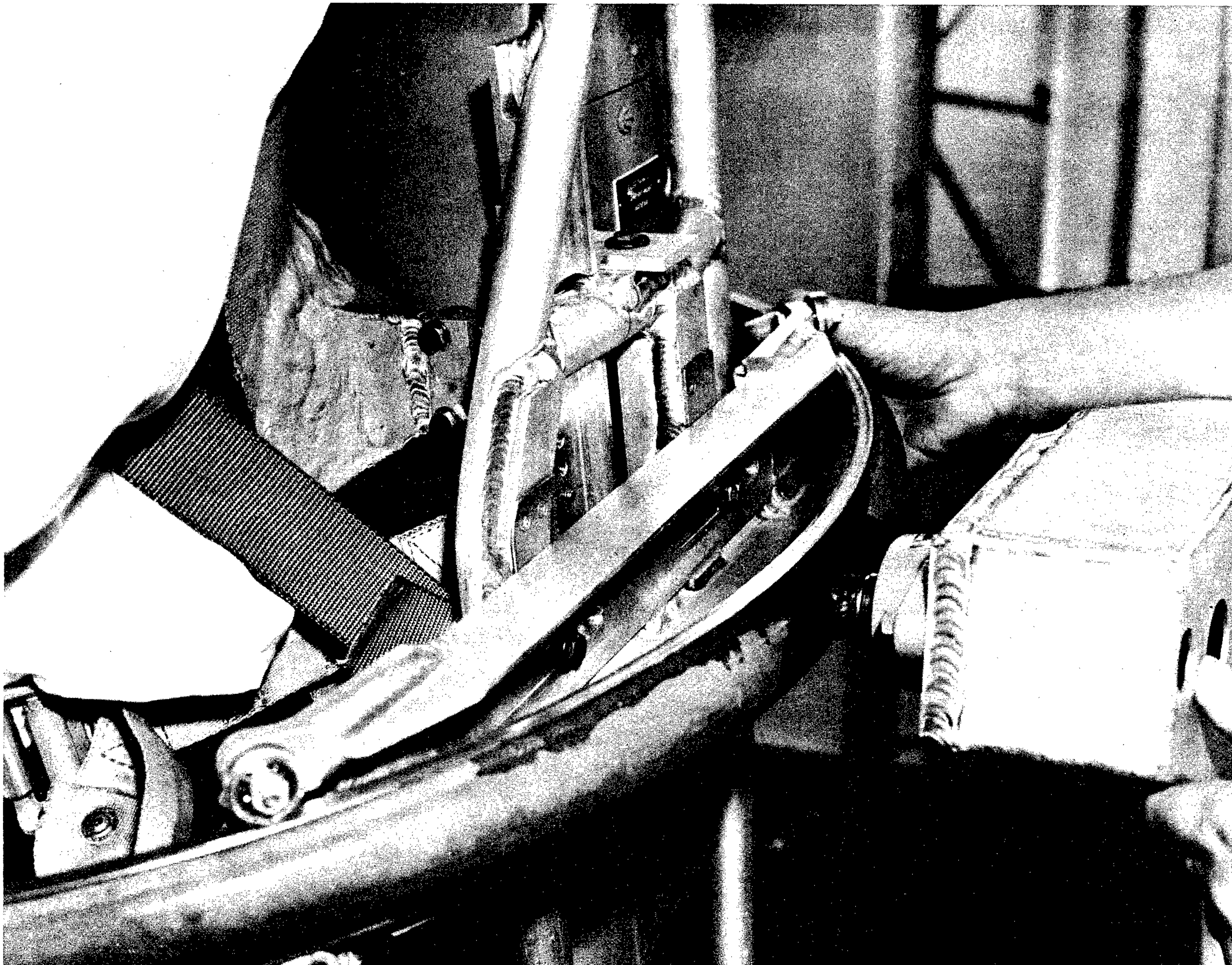


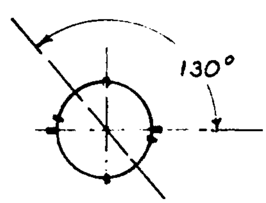
Figure 6. Cradle Carrier and Adjustments

Because the fits between back pack, corset and subject are loose, it is not necessary to make all adjustments simultaneously. "Fine tuning" of up and down balance can be accomplished with the foot plate adjustment alone.

2. Yaw-Ring

The yaw ring was sized from the minimum practical envelope of the cradle and subject allowing for adjustments of the cradle fore and aft and keeping the common CG at the center of the ring. Figure 7 shows the yaw ring viewed along the yaw axis. The subject's hands are resting on the back pack control unit. Two additional sets of bearings are hidden by the back pack. In order to achieve continuous rotation in yaw with minimum structure and deflection, a closed ring was selected. Rotation is required with any orientation of the ring in the roll yoke, therefore the bearings may be loaded from any direction in the plane normal to the ring cross section. A configuration of four sets of three bearings spaced symmetrically at 40° from the transverse axis was analyzed for maximum loads under three different orientations: 1) yaw ring horizontal, 2) yaw ring vertical, cradle rolled 40° and 3) yaw ring at 45°, cradle rolled 40°. A 340 lb load was applied at the center of the yaw ring. Consultations with Fafnir Bearing Company were held, bearings were recommended and a curve of bearing starting torque under the varying loads was developed by Fafnir. From the curve, calculations (Appendix B) were made of the maximum starting torque. In the worst case with the yaw ring at 45° and the cradle rolled 40° to put one set of bearings at the low point, .886 ft lbs should be required to overcome the bearing starting friction.

In addition to bearing friction, deflection of the yaw ring contributes to the force required to yaw. The ring deflects normal to its plane, making it necessary for the cradle to run uphill. The worst condition occurs when the yaw ring is horizontal and the cradle is yawed 130° to put one bearing set at the most forward point and the other on the uphill side in the direction of travel.



All bearings must then run uphill for the next 10° of rotation. In this orientation a maximum of .40 ft lb is available to move the cradle, the rest of the allowable 1 ft lb being required for overcoming bearing friction. Assuming for simplicity that the deflection

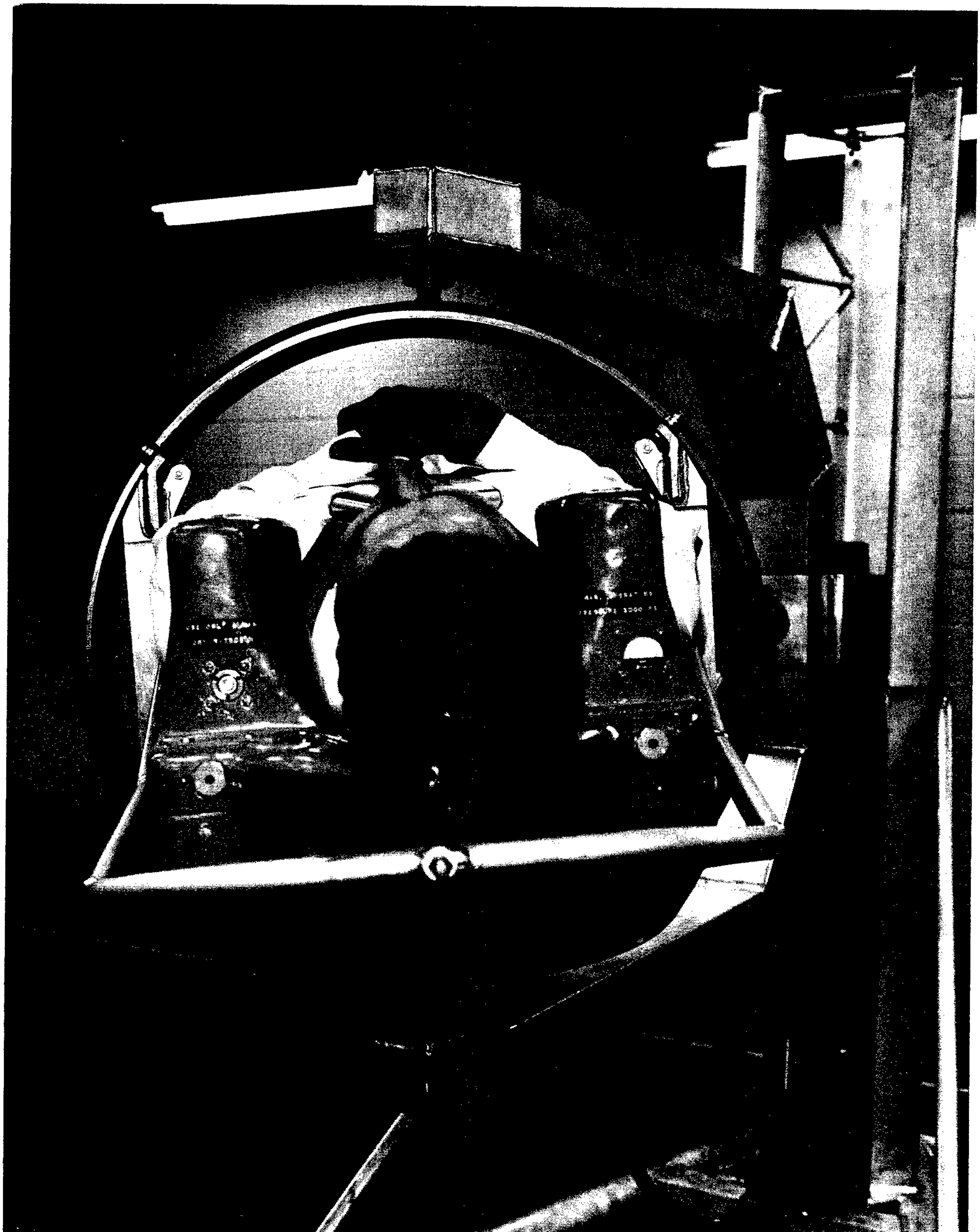
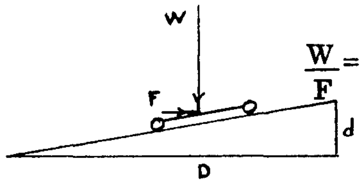


Figure 7. Yaw Ring

produces a uniform helix in the 90^{o} segment of the ring, the maximum allowable deflection is .044 in.



$$\frac{W}{F} = \frac{D}{d}$$

where W = 170 lbs.

F = .40 ft lb at 15 in. = 0.32 lbs

$D = \frac{\pi\ 30}{4}$ in. = 23.5 in.

d = max allowable deflection

$$d = \frac{23.5 \times .32}{170} = .044$$

Actually, since the tube will not bend sharply under the bearing, the track profile should represent an S curve, the initial slope should be less, and therefore the starting torque should be less than the calculated. As the cradle rotates, the S should move with the bearings.

Theoretical   Calculated

The yaw ring design was based upon a maximum .044 in. deflection under the condition of maximum bending load when the cradle is yawed 90^{o}. A conservative load of 400 lbs. was used. With a 1.5 in. dia .065 wall tube of 4130 steel bent in a 30 in. dia circle with bearing tracks .37 x .18 welded to the top, bottom and inside, a theoretical maximum deflection of .040 was calculated.

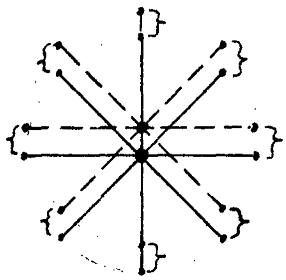
Adjustments are provided by eccentric mountings for all bearings so that minimum shift of CG will occur when the load shifts between bearings during changes in attitude of the yaw ring.

3. Roll Yoke

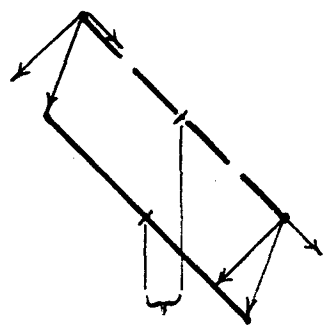
Design of the roll yoke was concerned mainly with two problems: 1) designing the lightest possible structure of adequeate strength and 2) minimizing variations in deflection at different roll attitudes.

The deflection problem arises from the structure of the roll yoke. If a uniform cross section is used throughout the yoke, the deflection of the arms will vary with the direction of the load applied to the ends of the yoke in the roll plane because of the difference between simple bending and twisting deflections in the members at the bottom of the U. With the yoke horizontal, a load applied vertically will create torque in the rear member. The same load applied with the yoke vertical will create pure bending.

The difference in deflection at different attitudes becomes important when the simulator is perfectly balanced, i.e., when the CG is exactly aligned with the axis of rotation. If the deflection of the yoke is constant under in-plane and out-of-plane loads, the outer end of the yoke will appear to rotate around a point directly below the roll axis. However, if the in-plane and out-of-plane



deflections are different the center of rotation of the ends of the yoke will move from side to side as it rotates. This will create



torque around the roll axis making the simulator unstable. With 9.5 in. lbs required to start moving, a 400 lb load will need a horizontal displacment of only .024 in. This corresponds to a difference of .034 in. in the in-plane and out-of-plane deflections at 45^{o}.

For the roll yoke design a built-up box, 6 in. square with .081 walls on the top and bottom, forms the base of the U. The cross section tapers to a vertical rectangle 3 in. x 6 in. at the outer end. Calculations indicated that the roll yoke deflection would be $.208 \pm .010$ in. thru 90^{o} rotation.

4. Elevating Mechanism

a. Basic System

The elevating mechanism is required to support the subject, back pack, cradle, yaw ring and roll yoke at any point in a 3 ft vertical range with less than 1 lb of applied force required for displacement. The basic approach is to maintain a constant pressure in a piston and cylinder such that the total of frictional and pressure changing forces is less than 1 lb. The pressure would be adjusted to produce a lifting force in the cylinder exactly equal to the total weight supported.

There are basically two approaches which could be taken. One is to provide a precision regulator to maintain a constant pressure in the cylinder as the volume changes; the other is to couple the cylinder to a total system volume large enough so that the change in system volume created by displacement of the piston to the extremities will be small and the resultant pressure change will be acceptable.

Allowing for a normal shop air supply of 80 to 100 psig with drops thru filters and dryers, a cylinder of 3.25 in. diameter was chosen with an effective piston area of 7.85 in. With a load range of 400 to 500 lbs , this required 50.9 to 63.4 psig operating pressures with allowable pressure changes from 0.156 psia to 0.164 psia.

For a regulator system this would require a regulator with the following characteristics:

Inlet pressure - 80 to 100 psig
Outlet pressure - adjustable 30 to 65 psig
Flow range - 5 scfm to 50 scfm (max. flow during upward acceleration)
Pressure Regulation 0.1 to psi over full flow range

Investigation of the market showed that a regulator of this type would be a development item with the normal high costs and long schedules.

Opposed to this approach is the surge tank system which was selected. The surge tank volume was calculated at the minimum to be:

$P_1 V_1 = P_2 V_2$ for max load

$P_1 = 78.1$ psia

$P_2 = 78.1 + 0.156 = 78.256$

$$V_2 = V_1 - \frac{\pi D^2 h}{4} = V_1 - \frac{\pi\, 3.25^2 \times 18}{4} = V_1 - 149.3$$

$$78.1\, V_1 = 78.256\,(V_1 - 149.3)$$

$$V_1 = \frac{78.256 \times 149.3}{78.256 - 78.1} = 74,808 \text{ in.}^3 \text{ or 324 gallons minimum}$$

P_1 = nominal pressure

V_1 = nominal volume (tank + 1/2 cylinder)

P_2 = pressure at down position

V_2 = volume at down position

A tank of 1070 gallons was chosen to reduce the maximum restoring force created at the limits of travel to 0.39 lb.

The surge tank is connected to the cylinder on the simulator thru a lazy arm air supply. The basic schematic is shown below.

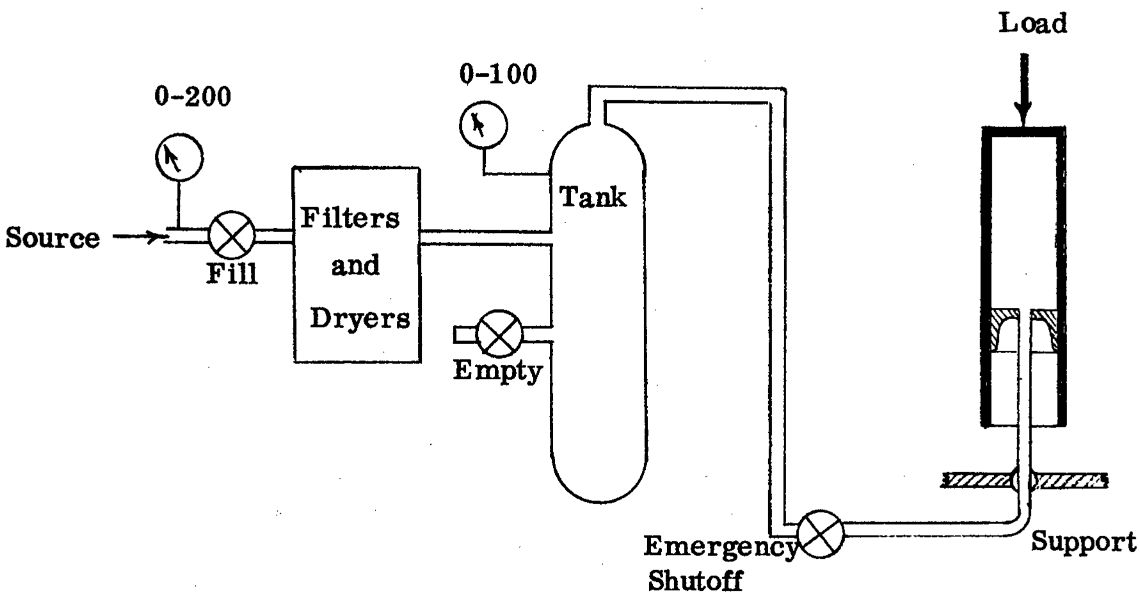


Figure 8. Pneumatic System Schematic

Originally, it was proposed that the piston and cylinder be sealed by a flexible rolling diaphragm. It was discovered that the ratio of stroke to bore was outside the normal limit required for reliable operation so an alternate approach was taken. Instead of a completely closed system, the piston was made air lubricated and a constant flow of lubricating air controlled by the fill valve is fed thru the system. This has an advantage that the frictional forces impeding movement are essentially non-existent. The piston was sized for lubricating film thickness of .0015 to .0023 in. with a flow of 15.6 to 27.8 scfm. Calculations were based upon reference (4).

The close tolerance of the piston and cylinder makes clean dry air a requisite to prevent build up of dirt and oil deposits. A CM Kemp Manufacturing Company two tower air dryer with separator, prefilter and after filter is used upstream of the surge tank. The air dryer has an active cycle of 4 hours at full flow. At the end of 4 hours manual changeover is made to the reactivated tower and the used tower is put thru a 3.5 hr reactivation cycle consisting of electrical heating and purge air flow of .85 cfm dry air. The active agent is silica gel. Details of operation and maintenance are containted in the Operation and Maintenance booklet.

b. Vertical Bearings

Because of the gimbal arrangement the load on the piston is unbalanced and a torque is created as well as the vertical load. It was necessary to design the elevating mechanism so that the torque would be reacted without restricting the vertical motion of the assembly. To accomplish this a vertical track was designed with air bearings riding on the track reacting the couple.

The vertical couple required reaction forces acting in the roll plane. It was necessary to split the upper support to allow movement of the roll housing. Two air bearings 11.5 in. apart canted 30 degrees are used to react the upper forward component and one bearing 43 in. below the upper bearing reacts the rearward component. The bearing loads were calculated with a total of 500 lbs acting on a cantilevered arm of 59.25 in. The two upper bearings see a 397 lb load and the lower bearing see 689 lbs. The bearings were conservatively sized because of uncertainties in air supply and loads. With a source of 75 psig the 4 in. x 6 in. upper bearings should support 576 lbs at .003 in. from the track. With lighter loads, the clearance would be proportionately greater. The 4 in. x 8 in. lower bearing should support 768 lbs. Calculations were straightforward application of reference (4). The air supply required is 50 scfm for the lower bearing and 37.5 scfm for each upper bearing.

The upper air bearings structurally are supported on the roll bearing housing with a two axis coupling which permits movement around axis parallel to the track (Fig. 9). This allows the bearings to adjust for slight misalignments and deflection of the tracks. The lower bearing is mounted on an arm extending rearward from the bottom of the cylinder and was pivoted around a horizontal axis parallel to the rear track (Fig.10). The figure also shows the air supply hose and the piston and piston rod extending from the bottom of the cylinder in the top limit of travel.

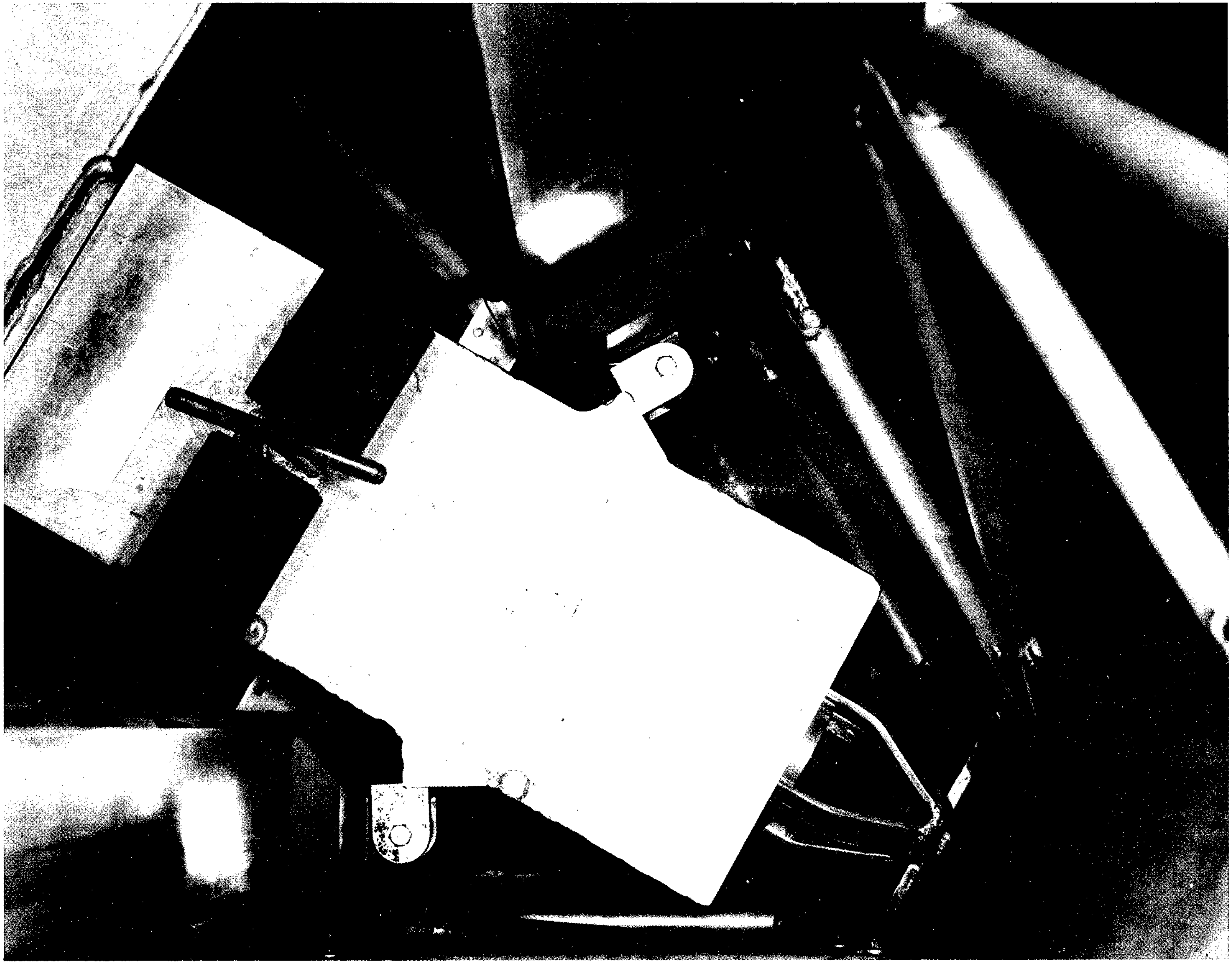


Figure 9. Upper Air Bearings (Top View)

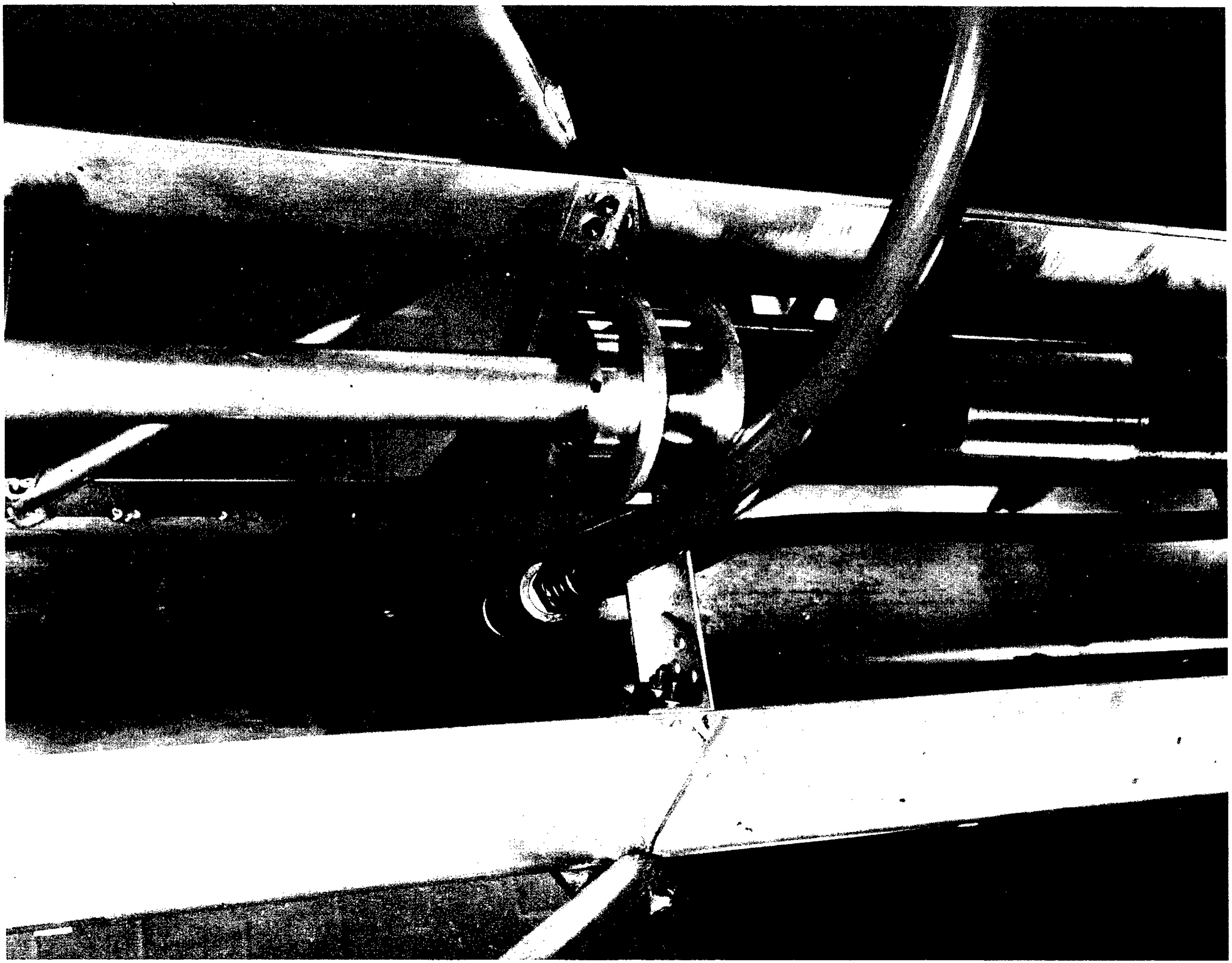


Figure 10. Bottom Rear Bearing

The tracks are 1/2 aluminum plates welded to the vertical support structure and machined flat to .0005 in. A critical aspect of the tracks is the flatness over the length of the pad.

Burrs or embedded particles that project above the surface of the track will cause drag,and if they occur they should be polished off with a medium grade abrasive.

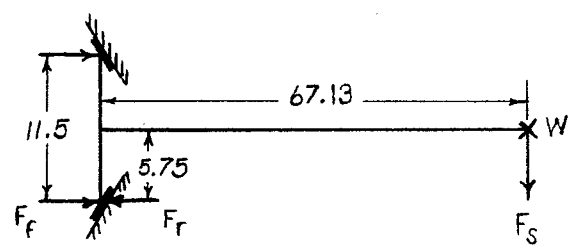
Waviness or slight misalignment is compensated in the pad mountings and structural characteristics.

The original design assumed balanced loads in the roll and yaw planes by virtue of the geometry of the gimbals which concentrated the loads on the pitch and roll axis. Any off-axis load applied by the subject would be resolved by the gimbal system into independent rotational and translational components. Analysis showed that side loads, forces tending to translate the subject to his right or left, would be reacted by the V configuration of the two upper tracks. With the cantilevered yoke applying a 345 lb forward load to each upper bearing, the side load necessary to lift the bearing off the track would be 59 lbs.

$W = 500$ lbs

$F_f = \dfrac{59.25\ W}{43 \times 2} = 345$ lb

$F_r = \dfrac{67.13\ F_s}{11.5}$



When $F_f = F_r$

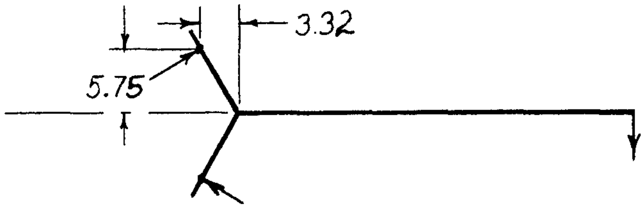
$F_s = \dfrac{345 \times 11.5}{67.13} = 59$

Since this load would be transmitted to the vertical structure and base which were floating on air bearings on the floor, this side load would accelerate the base rapidly with the result that the air pad would never actually lift off the track.

During the test phase additional significant factors became apparent necessitating more detailed analysis and modification.

In order for side loads on the subject to be transmitted to the base, reaction forces on the pads must be normal to the tracks. If the forces are not normal the parallel components will cause the pads to move sideways without restraint until they reach the edge.

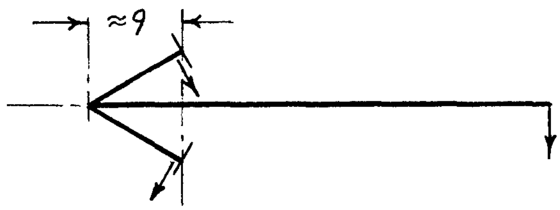
To apply loads normal to the two upper pads, a center of rotation located 3.32 in. forward of the pad pivots is required.



With the two pads moving frictionlessly parallel to the tracks, the apparent center of rotation is a point approximately 9 in. behind the pad pivots.

Figure 11 is a plot locating the apparent center of rotation of a rigid bar 11.5 in. long constrained to move with the ends in contact with two tracks at an included angle of 120°.

If a torque is applied around this center of rotation, it will be seen as almost a pure side thrust on the pad and only a small fraction of the force will be applied to the track structure.



Therefore, the pads will move sideways until they strike the side of the track before they can accelerate the base in yaw or translation.

If the side load applied has a vertical component, the friction between the pad and the side of the track will damp the vertical motion.

Since the center of rotation of the upper bearings is approximately 9 in. behind the bearing pivots and the center of rotation of the lower bearing is approximately 10 in. behind the upper bearing pivots, the elevating mechanism hangs somewhat like a gate on a gatepost that is not plumb. It will swing toward a stable point in the center but the force required for displacement is not large.

In addition to the upper pad side forces, it was discovered that significant side force is seen by the lower pad causing it to drag on the side of the track. This force consists of side thrust generated by slight vertical misalignment of the piston and cylinder.

NOTE: Lines indicate loci of randomly chosen points in the horizontal plane containing the pivot points for the upper bearings, as the pivot points move along the tracks. .

O - Bearings at left extremity
X - Bearings at mid-point
Δ - Bearings at right extremity

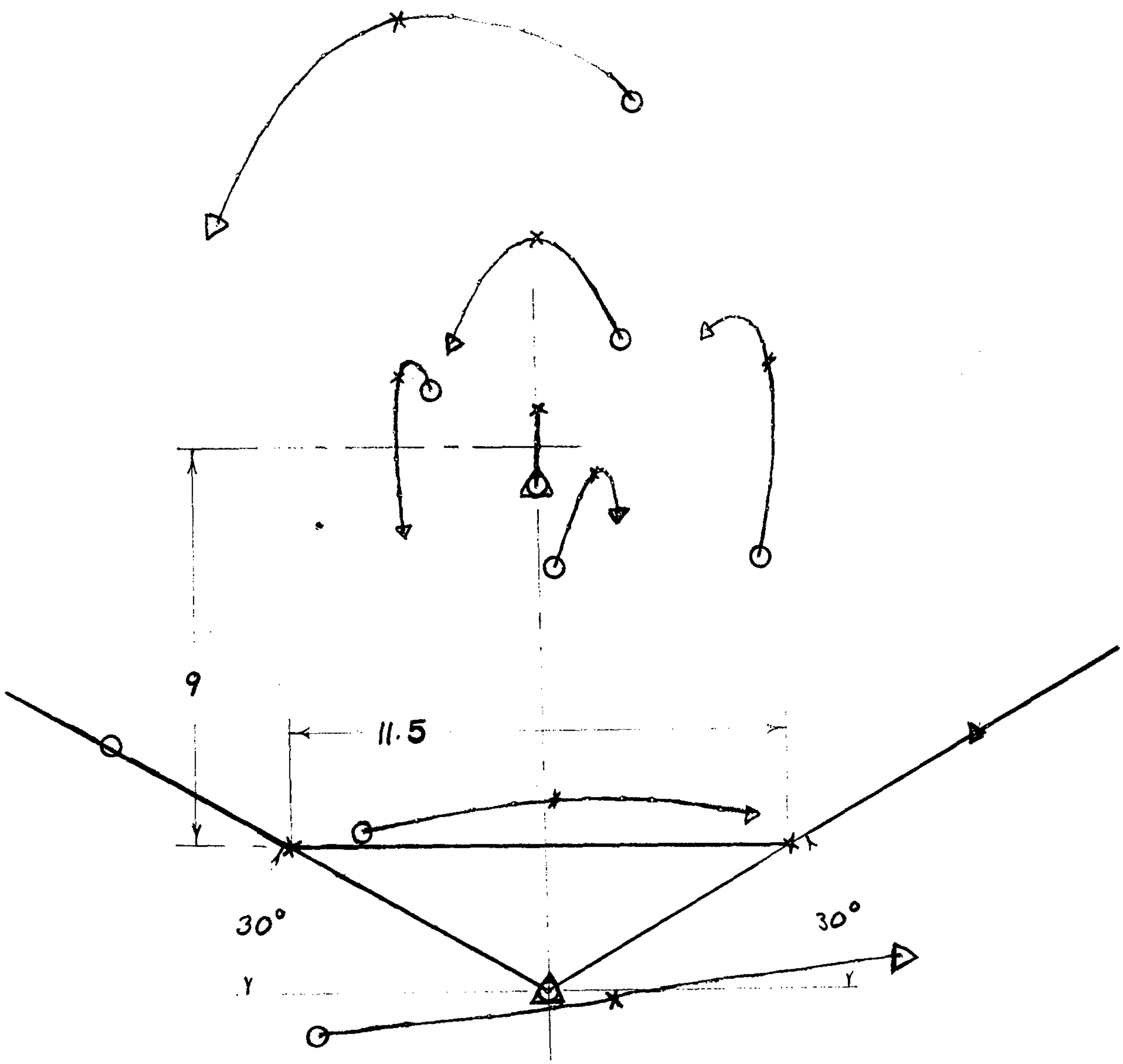
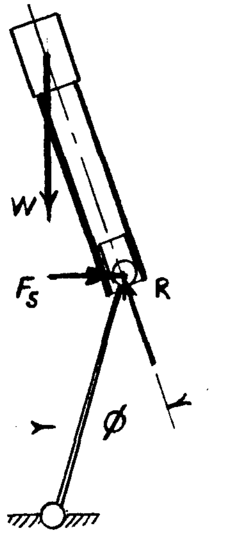


Figure 11. Location of Upper Air Bearing Center of Rotation

The vertical load is applied on the roll axis and is reacted by the piston thru the center of the piston support bearing. Since the piston maintains its position in the center of the cylinder, tilting of the cylinder from the vertical will cause misalignment of these two forces with a resultant which causes the piston and cylinder to move sideways. This is an unstable condition.



$F_s = W \tan \phi$ for $W = 500$ lb.
$F_s = 5$ lb.

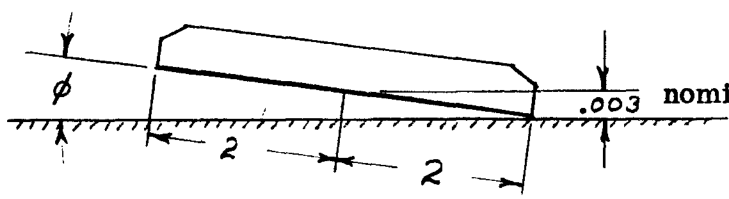
$$\tan \phi = \frac{F_s}{W} = \frac{5}{500} = .01$$

$$\phi = .57^{\circ}$$

With the elevating mechanism at the nominal point, the roll axis is 26.37 in. above the piston bearing axis. A tilt of $.57^{\circ}$ will occur if the piston moves .26 in. relative to the roll axis, thus creating a 5 lb. side thrust at the piston and a 3 lb. side thrust on the rear air pad. At the nominal point the cylinder can tilt a maximum of 2.78° unless it is restrained by the rear pad. The rear pad must be aligned so that the centerline of the support arm is parallel to the roll axis or tilting of the cylinder will occur.

Horizontal misalignment of 1° will result in a vertical misalignment of $.97^{\circ}$ in the cylinder.

Unless the rear pad is allowed to align itself with the track, a tilt of more than 0.086° around its vertical axis would cause one edge to drag on the track.



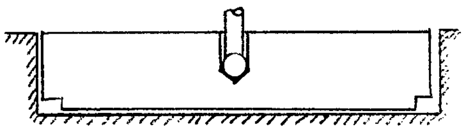
$$\sin \phi = \frac{.003}{2} = .0015$$

$$\phi = 0.086^{\circ}$$

With no freedom around the vertical axis, this would require holding the roll axis to within 0.086^{o} alignment with the centerline of the base. This corresponds to a lateral motion in the upper pads of .014 in. As a result of this further analysis, modifications were made in the design.

To provide a more suitable load path for transmitting side motion of the subject to the base structure, a set of ball bearings was provided between the vertical tracks. The bearings (Fig. XII) ride on the edges of the tracks and take the load before the air bearing can hit the side of the track.

The rear bearing and track were redesigned to provide a rear pad with three bearing surfaces.



The main bearing surface reacts the rearward component of the couple created by the cantilevered load and the side surfaces react the side loads. A ball joint permits the pad complete freedom to align itself in the track. Because the side surfaces required additional air capacity, the pad size was increased from 4 x 8 in. to 4 x 10 to reduce the required operating pressure. The side surfaces are 1 x 10 in.

An internal manifold and needle valves to adjust the flow to each surface were incorporated. The adjustments can be seen in the top of the pad in figure 13. The outer two are adjusted to give the minimum flow necessary to maintain the .0015 clearance. The inner two are adjusted for maximum flow to the main surface without starving the sides. The four sets of double holes at the edges of the pad vent the air from the bearing surfaces to eliminate oscillation caused by cross coupling of air flows.

The track for the new bearing is a built up channel bolted on the face of the old track (Fig. 12). The bearing support arm was shortened to compensate for the change in track thickness.

c. Flexible Hose Selection and Configuration

The air supply requires a flexible hose between the base and the elevating mechanism. It was necessary that the hose be capable of flexing under pressure with a force of less than 1 lb. Consultation was held with Aeroquip and U.S. Rubber and samples of hose obtained for evaluation. It was quite apparent that the hose manufacturers had not been exposed to this type of requirement and no data

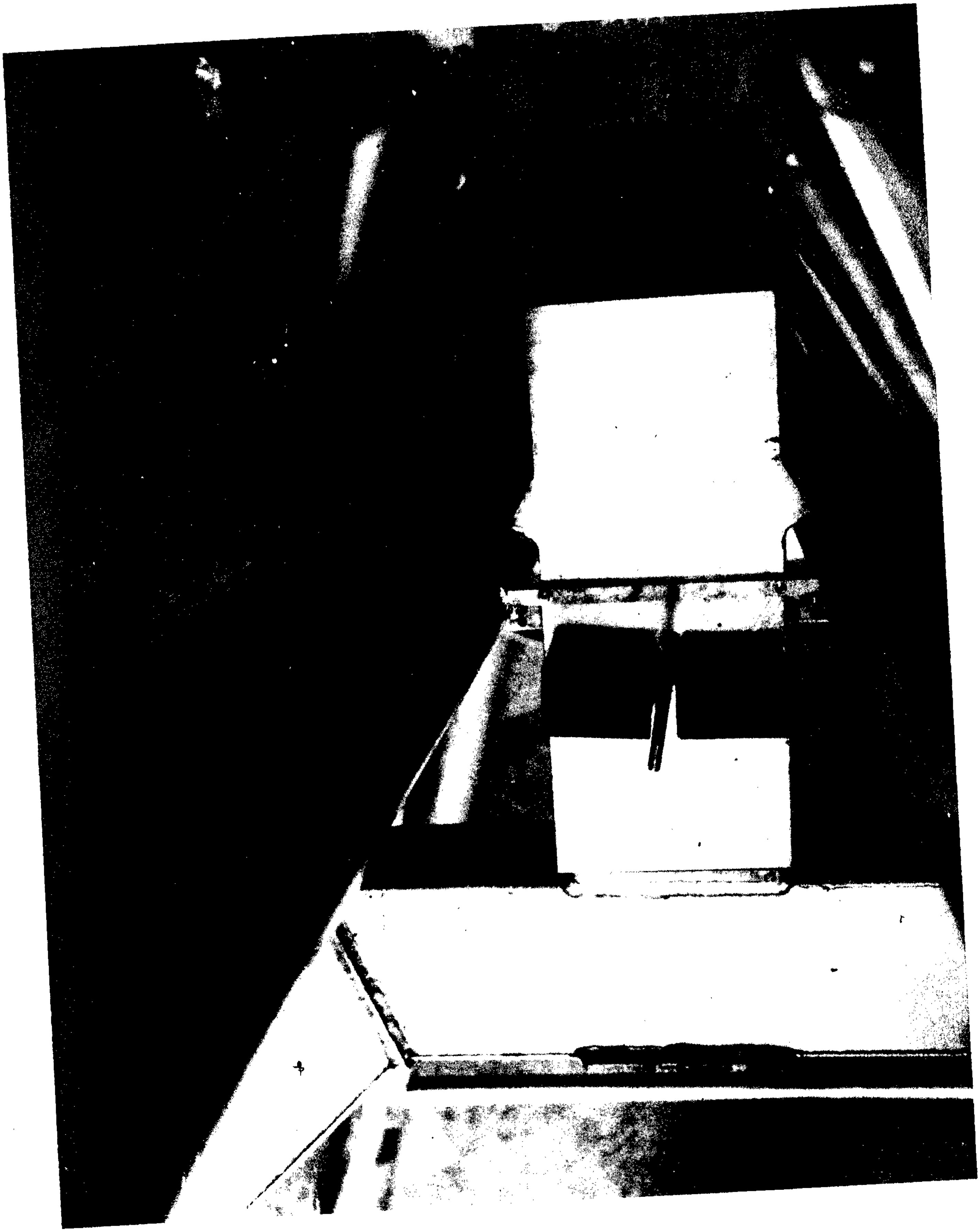


Figure 12. Elevating Mechanism Ball Bearings

Figure 13. Bottom Rear Bearing (Modified)

on the relationships of bend radius, bending force and hose pressure in the required ranges was available.

Although the actual configuration of hose planned for the simulator involved both bending and twisting, initial screening was done by simple bending. Subjective evaluation of bending indicated that most high pressure industrial hose was far too stiff and hose flexible enough, like vacuum cleaner hose, would not take the pressure. The samples found closest to being useable were U.S. Rubber gas pump hose and vinyl garden hose. Two samples of 3/4 in. ID and 1 1/4 in. ID were obtained for evaluation along with a 3/4 in. ID vinyl garden hose.

Since there was a question of the suitability of the vinyl hose under pressure, it was burst tested in our Quality Laboratory. A three ft. section with hose fittings on each end was bent in a 12 in. dia. semi-circle and slowly pressurized. Swelling of about 1/4 in. was observed when pressure was initially applied but this stabilized above 50 psi. The pressure was incrementally increased in 50 lb. steps and held for 1 minute at each pressure. The hose burst after 30 seconds at 200 psig. An additional specimen was maintained at 100 psig for 24 hours with no indication of creep swelling.

The vinyl hose and the rubber gas pump hoses were evaluated for bending by measuring the force necessary to bend a 3 ft. U shaped section of hose in the horizontal plane. The hose was restrained only at the ends. The initial configuration was with the ends 18 in. apart. The hose was bent to 9 in. The results are tabulated below:

	Hose	Force Required to Hold U Shape	
		With 18 in. Separation	With 9 in. Separation
1.	3/4 ID Vinyl	3 oz.	8 oz.
2.	3/4 ID Rubber	10 oz.	18 oz.
3.	1 1/4 ID Rubber	80 oz.	16 oz.

1. 3/4 ID vinyl - 1/8 wall garden hose
2. 3/4 ID Rubber - 3/32 wall wire reinforced gas pump hose
3. 1 1/4 ID Rubber - 1/4 wall rayon reinforced gas pump hose

Sample 3 was included for comparison purposes.

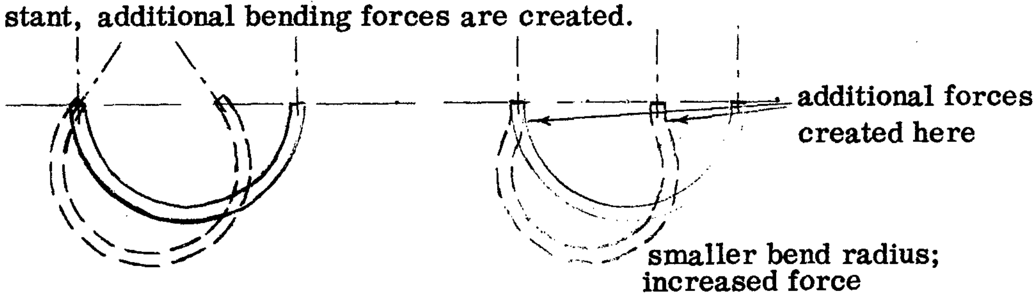
An evaluation of the geometry of the hose configuration indicated that the bending in the installed configuration would be more complex than the simple test configuration so additional tests were made to determine the optimum installation configuration. Factors considered were:

1. End fitting alignment

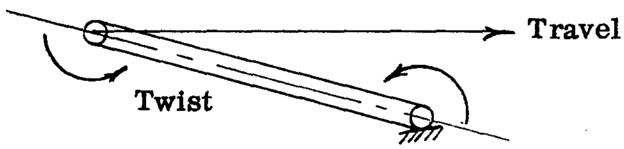
2. Direction and range of travel

3. Weight of hose supported on moving structure

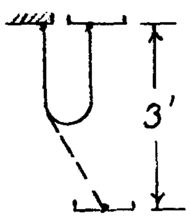
The design of the elevating mechanism restricted the movement of the end of the hose to a straight vertical line. Simple bending of the hose with the end moving in a straight line requires a change in the angle of the hose axis relative to the line of travel. If the hose axis alignment is held constant, additional bending forces are created.



In addition, if the direction of travel is not in the plane of the hose, twisting will occur as the plane of the hose rotates.



As the hose end moves vertically, the weight distribution varies. At the top there will be a minimum of 1.5 feet more hose to be supported than at the bottom.



Development of the final configuration was made experimentally. A vertical track and carriage were set up with a fitting for the hose coupling. The other end of the hose was clamped in place and the carriage was counterweighted to be stable at the nominal position. The clamp was positioned at a point corresponding to the outside of the vertical track and measurements were made of the forces necessary to move the carriage up and down 18 in. from the nominal. Adjustments were made in the vertical position of the clamp, length of hose being flexed and preset twist. The preset twist was introduced to compensate for the weight of the hose. The optimum was foundto be a 4 ft. hose forming a horizontal U with one end just behind the cylinder and the other 18 in. above the base on the right 2 member and with a twist in the hose that increased as the elevating mechanism moved downward. This twist compensated for the weight of hose to give a relatively uniform displacement force which was almost 0 at the nominal point. A force of 11 oz. was required to move the carriage to the extremes of travel. The hose was then pressurized but without flow. The force requirement increased to 15 oz.

A manifold of "T" connectors and flexible 1/2 in. hose is used to distribute air from the 314 ID vinyl hose to the three pads.

5. Vertical Structure and Base

The vertical structure and base support the elevating mechanism and provide a tripod support for the simulator with three HOVAIR air pads which float the simulator on a thin film of air.

The vertical structure is essentially two welded trusses with the longitudinal (vertical) members consisting of a bent up channel holding the lower rear track and two bent up Z members holding the upper tracks. The Z members were designed to bolt to the gussets of the truss so that the three tracks could be machined after welding but prior to assembly. Trusses are aligned in a blunt V of 60^{o} with the wide end forward and pacers are attached to the top and center. The top spacer also serves as an upper stop to limit vertical travel.

The vertical structure is welded to a base made from 4 in. dia. 1/4 wall 6061-T6 aluminum tubing. Diagonal braces are provided to reduce the stresses in the area of attachment of the vertical structure.

Trade-offs involving weight, strength, material availability, corrosion resistance and ease of manufacture were made between aluminum and magnesium square and round tubing and the 4 in. dia. round aluminum tube was selected because the weight/strength/size combination was equivalent to magnesium and the aluminum offered significant advantages in corrosion resistance and ease of manufacture. The corrosion resistance was considered important because the base would be used as a plenum chamber at 80 to 100 psi. Corrosion during the life of the simulator could weaken the tube walls below a safe limit. Protective coatings would be difficult to apply internally with assurance of complete coverage.

The base is supported on three General Motors 12 in. dia. HOVAIR pads (Fig. 14). These air pads provide a frictionless air film flotation with the added feature of a flexible plenum which conforms to the floor surface to permit operation on standard office or lab quality tile floors. Because of the flexible plenum, the HOVAIR pads have an effective clearance of approximately 1/4 in. and they require an operating pressure of 2.5 psig. The input is regulated by a flow orfice which is adjusted to give maximum lift without oscillation.

The air pads are rigidly mounted to the tubular base to form a horizontal triangle with a 60 in. base and a height of 86 in. The subject center of rotation is located 5 in. behind the base of the triangle. Because the CG of the loaded simulator is to the rear of the point of application of translation forces ions, the base will tend to trail the subject and sideways velocities would be minimized, therefore the transverse dimension of the base need not be as great as the longitudinal. With the subject at the top of his travel, impacting sideways at approximately 13 fps would be required to tip it over. This would require a 40 lb. thrust applied for 6.1 seconds over a distance of 39 ft.

At the rear of the base is the connection for the air bearing air supply. Clamps are provided on the vertical structure to attach the lazy arm tubing securely. The flexible hose for the elevating air is secured to the vertical structure parallel to the lazy arm tube. Standard hose and pipe fittings were used.

6. Miscellaneous Notes

Capability of restricting motion in each axis was provided for ease of loading and unloading. The pitch and yaw axis are locked by a bar (Fig. 15) which clamps the yaw ring and roll yoke to limit pitch motion and a pin which holds the cradle attached to the bar to limit yaw. The roll axis is locked by a T pin inserted thru the roll axle and housing (Fig. 9).

Horizontal translation can be limited by shutting off air flow at the source or at the pads. Because impulse forces on the vertical bearings increase when the base is not floating, it is recommended that the base be floating when the vertical system is used.

Vertical motion can be limited by partially closing the emergency shutoff valve to restrict the flow between the cylinder and the tank. This has the effect of stiffening the spring constant of the system. Care should be taken to have the system flow adjusted properly prior to closing the valve.

The straps in the restraining system are standard aircraft seat belt straps with quick release buckles.

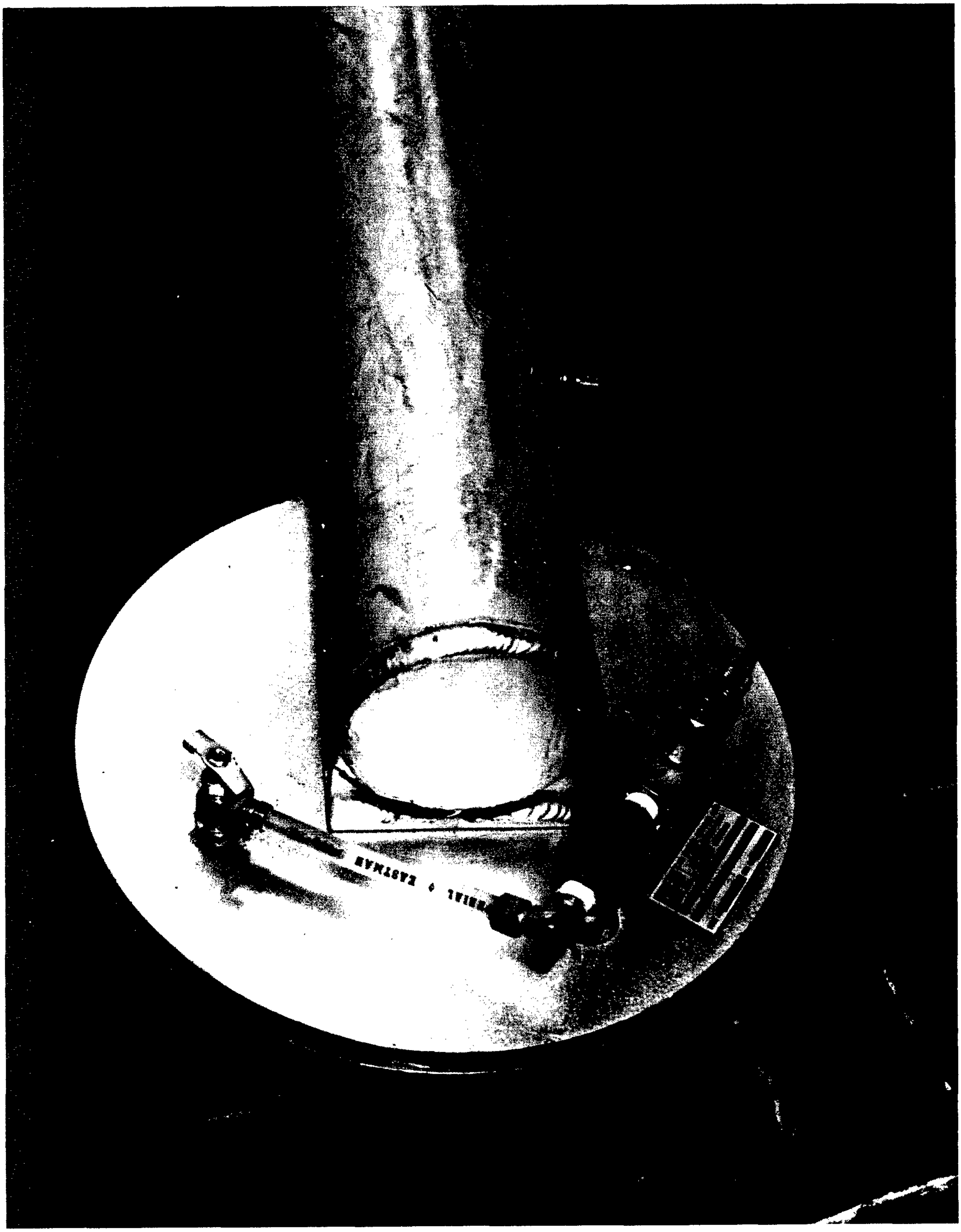


Figure 14. HOVAIR Pads

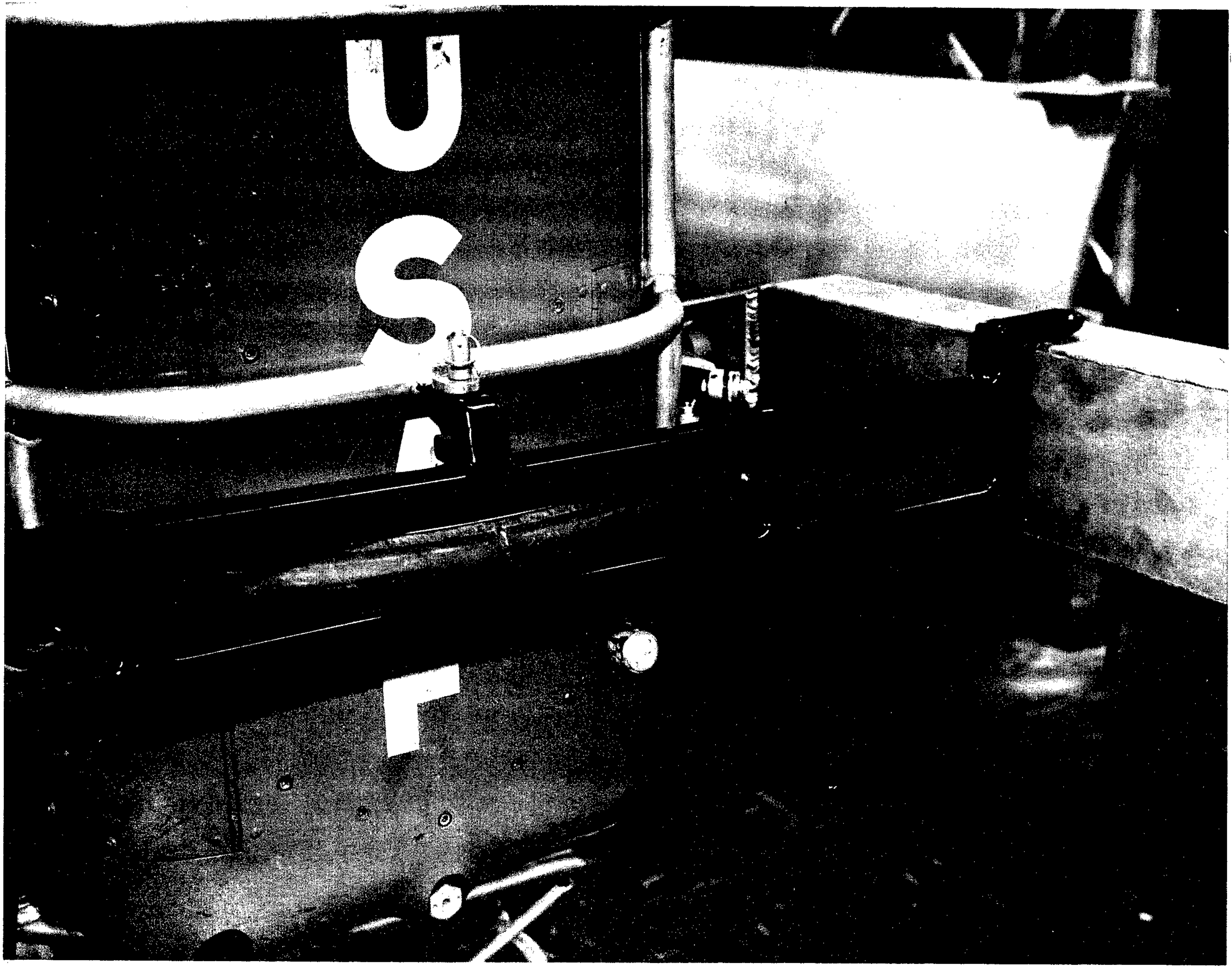


Figure 15. Pitch and Yaw Locking Bar

The air tank is a standard commercial tank 4 ft. in diameter, 12 ft. high with hemispherical domes capable of withstanding 150 psi normal operating pressure. A 40 in. diameter flange supports the tank on end. A drain valve at the bottom and a manhole are provided.

All bearings are lubricated with a small quantity of light oil. Excessive oil will increase the bearing friction thereby reducing simulator sensitivity.

C. Bridge Crane Work Panel Design

1. Work Panel

The work panel consists of 4 x 6 ft. aluminum skin spot welded to a typical aircraft structure of ribs and stringers with a supporting framework for attachment to the vertical support in the rear. The ribs are curved I beams with a 60 in. radius which were available from the Titan Program. Seven stringers of bent up aluminum sheet run horizontally on 6.5 in. centers. The two main ribs are equally spaced 14 in. from the vertical centerline. The linkage to the force sensing system is attached in line with the main ribs.

The linkage with the force sensing system is a parallelepiped with a triangular horizontal section and parallelogram vertical section. Spherical bearings on the vertical members allow limited movement in the horizontal plane. The 3 vertical arms are at the corners of triangle with a 30 in. base and 18.25 in. altitude with the base next to the work panel. The corners of the lower triangle carrying the spherical bearings are attached to the work panel 18 in. above the centerline. The rear bearing is held by an outrigger of tubing extending 18.25 in. behind the rear plane of the panel. The upper triangle is attached to the vertical support. Figure 11 shows the work panel construction and the linkage attaching it to the vertical support structure. On the centerline of the panel, at the bottom of the support structure is the force sensing system. The original design directly connected the panel to the system. Because of changes in the system, a mechanical advantage was introduced by adding the lever directly below the supporting linkage. The lever has an 6:1 advantage and is attached to a fulcrum on the inside of the bottom structural angle (Fig. 16). The lever on the far side can be seen directly in line with it. Forces are sensed in any horizontal direction and are resolved into transverse and longitudinal components which are the inputs to the two servo channels.

2. Work Panel Vertical Support

a. Structure

The vertical support is a vertical trusswork box which suspends the work panel from the crane and provides transverse

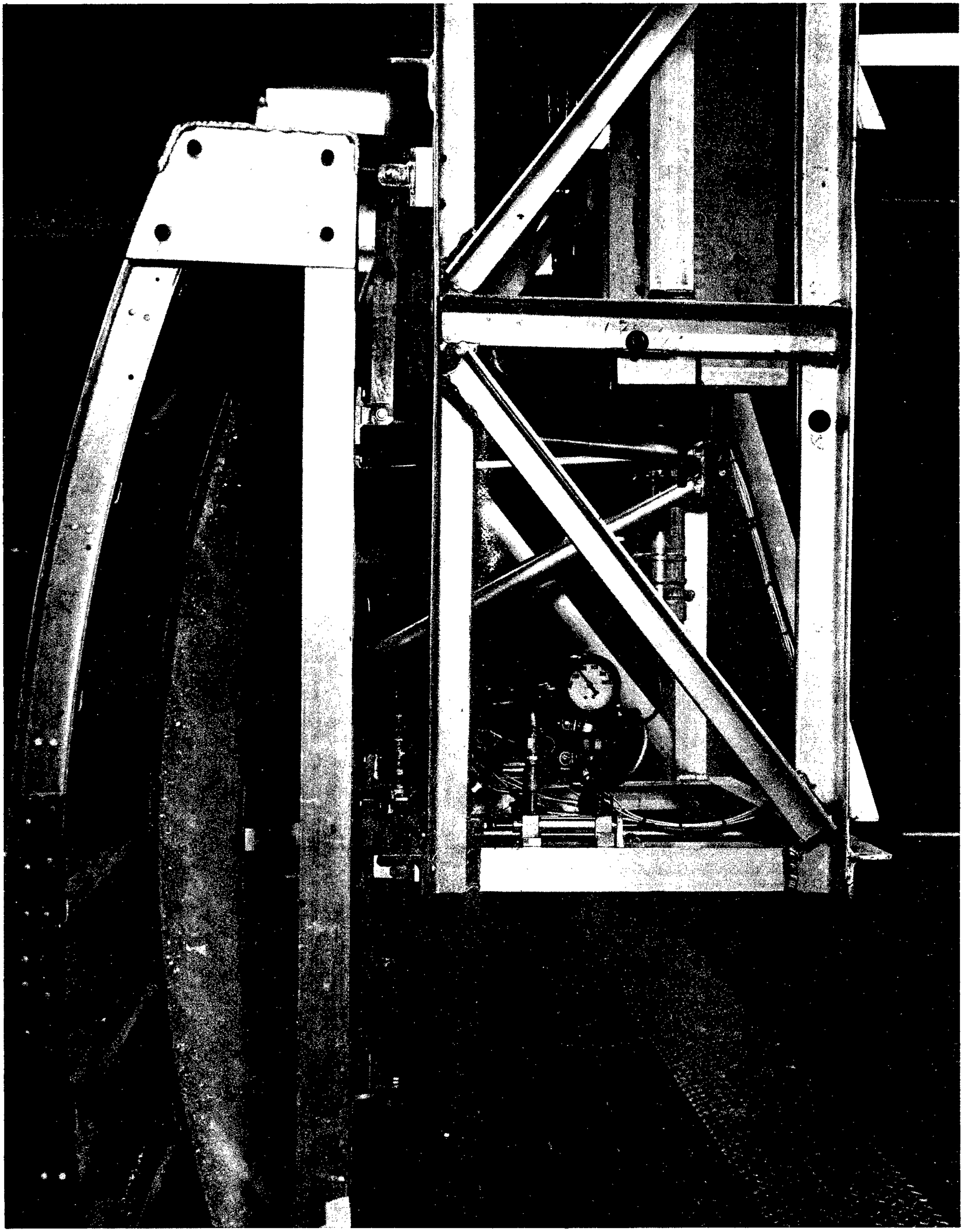
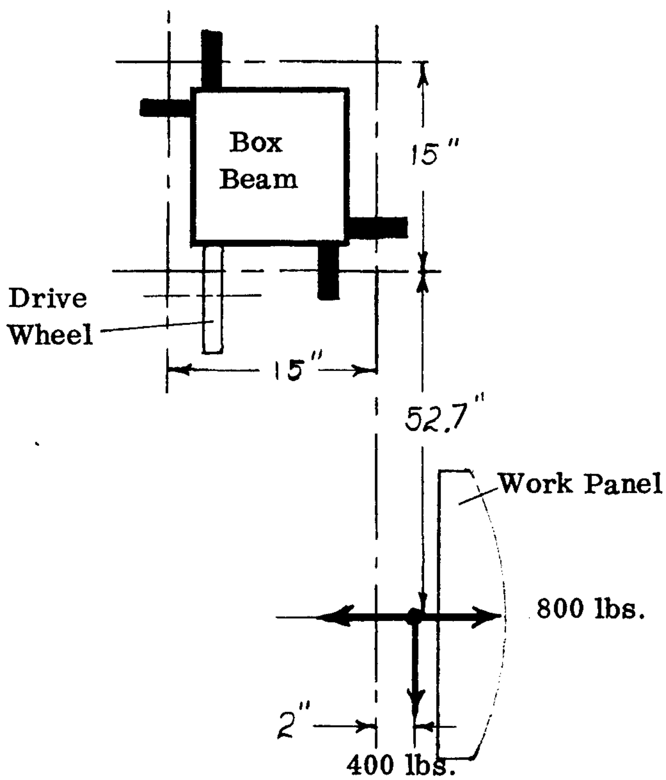


Figure 16. Work Panel and Vertical Support

movement. The box is approximately 71 in. high, 30 in. wide and 18 in. deep. The centerline of the work panel is approximately at the bottom plane of the box, 76.5 in. from the floor and 7.5 in. above the CG of the subject in the nominal position.

The top 1/4 of the box encloses the box beam with casters which support the assembly.

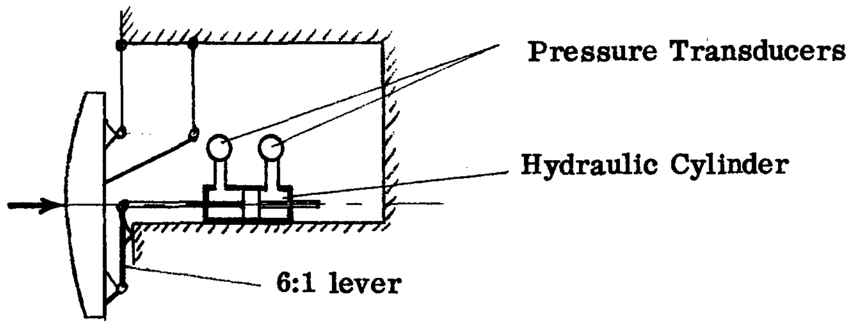
The forces anticipated on the assembly were in the range of ±150 lbs. applied horizontally (ref. 5). However, because the simulator is intended for evaluation of propulsion packs which could malfunction, the structure was sized for impact loads against the limits of travel of 800 lbs. The point of application of force is at the bottom of the vertical structure forward of the centerline so the caster arrangement shown below was selected to provide the minimum number of casters necessary while holding the structure securely on the beam.



Two casters on each face of the beam carry the loads except on the top. Four casters are used to carry the additional load of the drive wheel pressure and the structure weight. Each caster is capable of carrying 200 lbs. normal load.

b. Force Sensing System

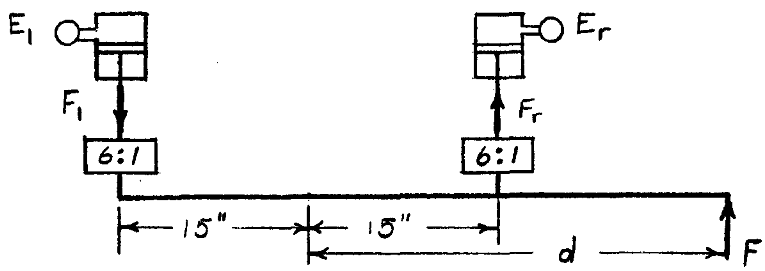
At the bottom of the vertical support is the force sensing system. The basic layout is shown below. Figure 16 shows the actual structure.



The work panel is constrained to move only in horizontal translation by the upper parallelogram linkage. The horizontal component of force seen by the panel is multiplied by the lever and applied to the cylinder. The cylinder is pressurized with each segment closed off. In the absence of a force the pressure is equal and the pressure transducers have equal outputs. When a force is applied, one cylinder chamber sees an increase and the other a decrease in pressure which results in a differential electrical signal out of the pressure transducers.

The mechanical linkage of 6:1 is acting on a double ended hydraulic cylinder of .375 ID with a .25 dia. shaft.

Although the pressure change in the cylinder is dependent upon where the force is applied to the panel, the resultant change in electrical output is independent of where the force is applied because the outputs from each side are electrically summed.



$$F_r = \frac{F\ (d-15)}{30} \times 6 \qquad \Delta P_r = \frac{F_r}{.393} \qquad E_r \approx +\Delta P_r$$

$$F_l = \frac{F\ (d+15)}{30} \times 6 \qquad \Delta P_l = \frac{F_l}{.393} \qquad E_l \approx -\Delta P_l$$

$$E_r + E_l \approx \Delta P_r - \Delta P_l = \frac{F\ (d-15)}{1.965} - \frac{F(d+15)}{1.965} = -\frac{F\ 15}{1.965}$$

At the worst condition d = 36 in. and $\triangle P$ = 389 psi for F = 150 lb. A nominal pressure of 400 psi was selected for the hydraulic system with pressure transducers operating from 0-800 psi. The selected transducers were not available so a change in operating range to 0-1700 psi was made. This change affected only the electronic portions.

The transverse force sensing is accomplished in the same manner by a cylinder located midway between the two levers with the piston rod attached to both. Side forces on the panel act thru both levers to develop the pressure changes in the cylinder.

The hydraulic system is filled and pressurized by a 50 cubic inch hydraulic accumulator and manifold with shutoff valves for isolating each pressure chamber. Any standard hydraulic oil can be used in the system. The accumulator is pressurized to 400 psig by compressed air or nitrogen applied to the accumulator. System pressure is read from a gage in the manifold. Pressures are equalized in the cylinders by opening all valves and closing them prior to operation. Since there is no actual flow in the system, the accumulator volume will compensate for fluid loss around the cylinder seals, and only occasional refilling is required.

c. Transverse Motor Drive

Beside the force sensing system, the vertical support structure carries the transverse motor drive. The drive motor is a Bendix (Exlipse) Aviation type 1235 Generator-Motor controlled by the electronic system. The drive motor drives a two step chain and sprocket 18:1 reduction to turn the 5 in. dia. rubber drive wheel (Fig. 17).

The first step of the drive is a 12-tooth sprocket driving a 72 tooth for a 6:1 reduction. This is further reduced 3:1 by a 20 tooth sprocket driving the output 60-tooth sprocket. Originally it was intended to use a 6:1 V belt drive but the total frictional load was too high and the system was redesigned.

The motor, drive reduction, and drive wheel assembly are mounted on a supporting bed bolted to the vertical support structure. Elongated slots in the bed permit adjustment of the contact pressure on the drive wheel. A belt dressing compound was applied to the drive wheel track to improve the drive friction and reduce the drive pressure required, thereby reducing the load on the motor.

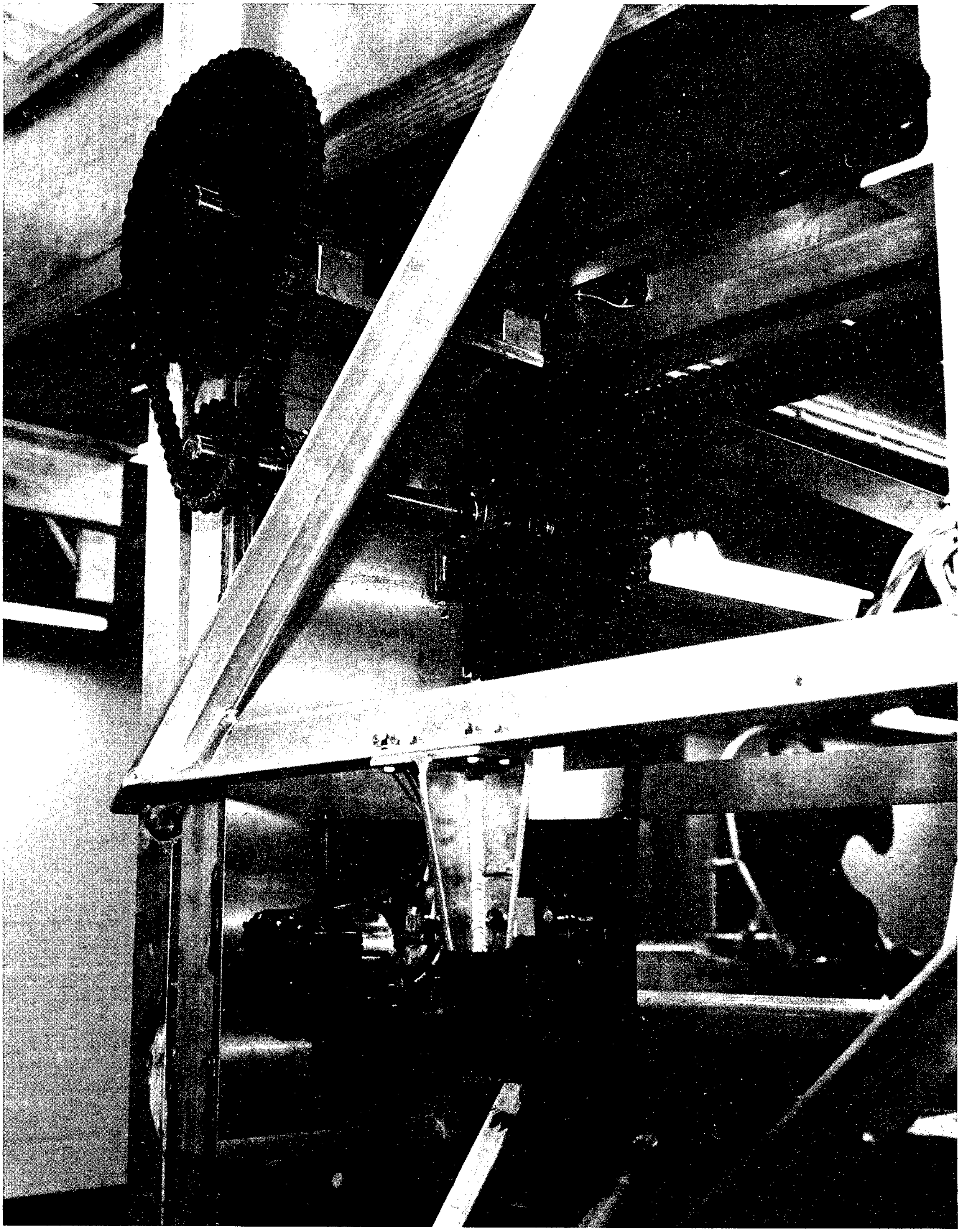
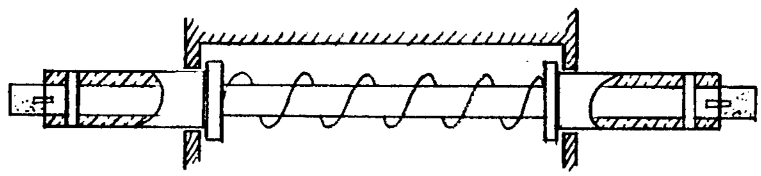


Figure 17. Transverse Drive Mechanism

d. Bumper

A spring bumper was included in the area just under the box beam (Fig. 3). The bumper is intended to absorb the impact of high velocity travel into the end of the beam. The bumper is double acting and consists of a shaft riding in a concentric spring.



At each end of the spring are loose washers which transmit the spring pressure to either the structure or to the shaft. The shaft, on the outer ends, is pinned to a concentric tube which transmits the load. Attached to the shaft ends to absorb the initial impact are commercial "dead blow" hammer faces. The spring is sized to absorb the impact of 400 lbs. traveling at 6 fps in a distance of 12 in.

3. Box Beam

The box beam is a 12.5 in. square beam 20 ft. long constructed of sheet metal riveted to square frames with the edges formed by 2x2x3/8 aluminum angle intermittently welded to the skin. The edges were machined flat to .125 in. over the 20 ft. span and parallel to .015 in. The edges form the tracks for the vertical support casters.

At the ends of the box beam, the carriers for the crane wheels are attached. These are 43 inch long channels built up out of 4x1-5/8x3/16 6061-T6 channels forming the legs and 10 1/4 wide x 1/2 inch 6061-T6 plate forming the web.

The channels are bolted to the top surface of the box beam on centers 229.7 in. apart. In the center of the channel, 20 inches fore and aft of the box beam center, are the wheel trolly assemblies. These are standard American Monorail Co. assemblies which mate with the crane rail. Each assembly consists of a U member with two 4 in. dia. flanged wheels mounted on the inside of the U with a clearance of 1 1/4 in. between them. At the bottom of the U the hanging bolt attaches the load, the trolly assemblies are allowed to swivel to adjust to track irregularities and the hanging bolt is used to adjust the tension on the longitudinal drive wheels.

Across the top of the box beam, the 1 in. drive shaft is mounted in 5 equally spaced pillow blocks. At each end of the shaft adapters reduce the diameter to 3/4 in. for mounting the 5 in. dia. rubber drive wheels.

On the left end, a small platform is mounted 13 in. above the upper surface 16 in. from the end to hold the motor drive (Fig. 18). The motor drive consists of a Bendix aviation generator-motor driving an 18:1 reduction to the output shaft. The first step of the reduction is a 12 tooth sprocket driving a 112 tooth sprocket for a 9:1 reduction. The second step is a 4 in. V belt pulley driving an 8 in. for a 2:1 reduction. The V belt was retained to facilitate the modification from 6:1 to 18:1 during final testing.

At the right of figure 18 the longitudinal limit switches can be seen. The upper switch is engaging the actuating angle mounted on the crane rail.

Terminal boards for the transverse and longitudinal drives are mounted under the motor mounts.

Just below the channels for the trollys are the longitudinal bumpers which are similar to the transverse bumpers but sized for 1600 lb. impact loads.

Across the back of the box beam a 3/32 in. wire rope and pulley assembly carries the cables for the transverse drive and force sensors.

4. Crane Rails

The crane rails are standard American Monorail Co. 9" deep steel girder rails, 25 ft. long. The rails are modified with detachable plates on each end which limit the crane travel.

The rails are hung on 230 in. centers with the top flange 154 in. from the floor. Standard hangars are attached to the facility structure at three hanging points for each rail. Each hanging point carries approximately 1000 lbs. max.

5. Servo System

a. General Considerations

The servo system consists of two channels, one controlling transverse motion of the work panel and the other longitudinal. Except for the fact that the longitudinal forces are sensed by two hydraulic cylinders while the transverse forces are sensed by

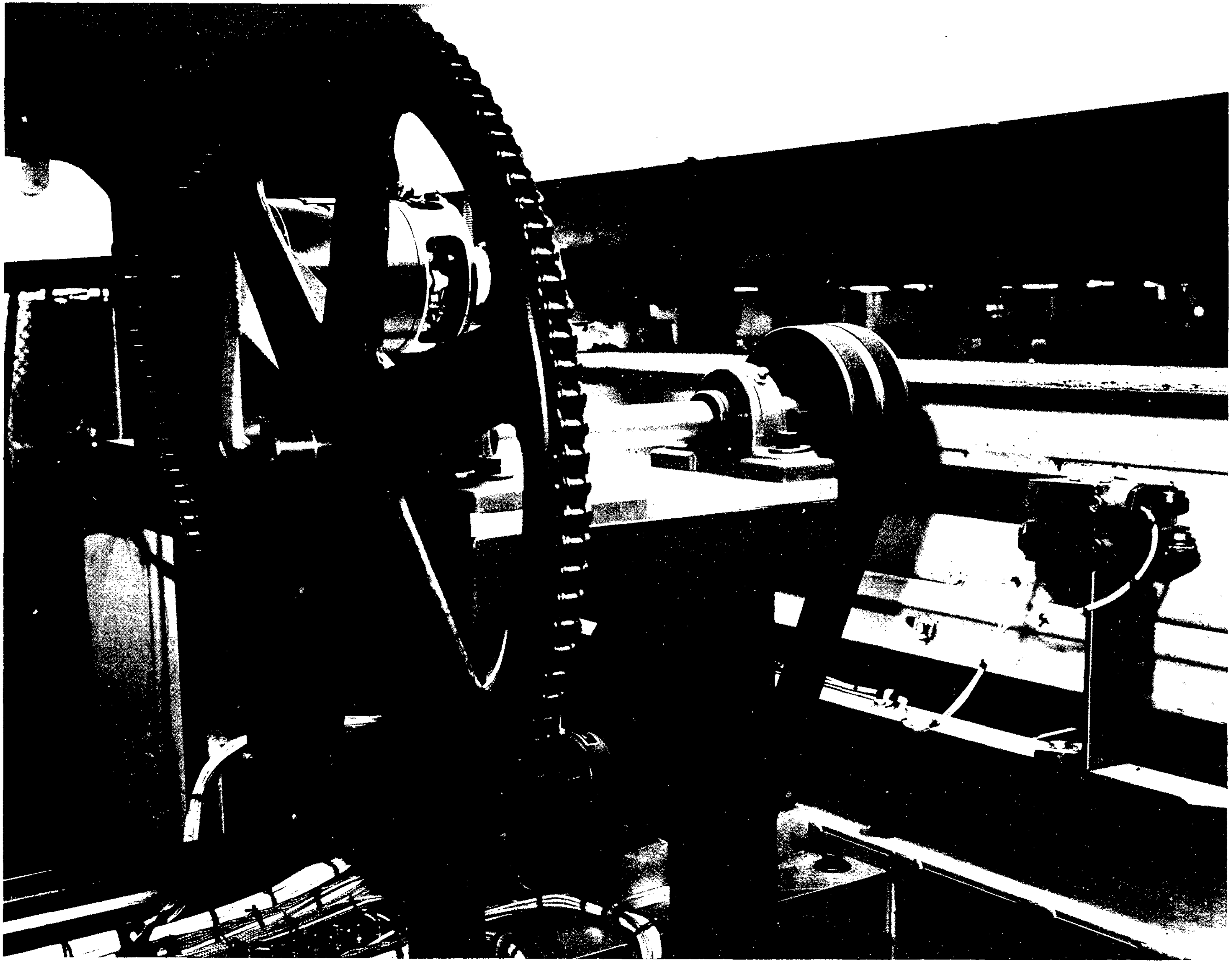


Figure 18. Longitudinal Drive Mechanism

one, the two channels are identical and components are interchangeable.

All discussion will apply to either channel with the force input assumed parallel to the sensitive axis of the channel.

Each channel consists of a first integrator and an SCR power servo. The first integrator drives a potentiometer at a rate proportional to the amplitude of the force input which varies $\pm$ 150 lbs. max. Generally, the forces will be of short duration, much lower amplitudes and of highly irregular force/time profiles as a result of performing tasks in the simulator.

The output of the first integrator provides the control signal to the SCR power servo to drive the work panel at a constant velocity.

It is necessary to provide means for positioning the work panel prior to an experiment, for holding the work panel in place while the experimental set up is being adjusted, and shutting off the motor drive at the limits of travel. In addition, it is desirable to have the capability for adjusting the servo to simulate various masses in orbit and to be able to control the panel from a point near the subject in the simulator.

These items were provided although they were not specifically required under the contract.

b. Description of Operation

1) General

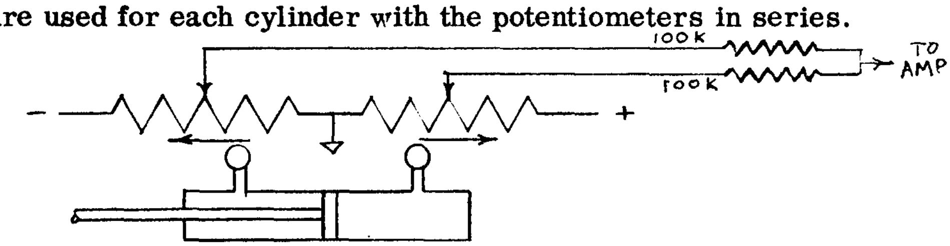
Servo inputs are DC voltages proportional to differential pressures sensed in hydraulic cylinders mounted on the work panel supports. Differential pressures in each cylinder are proportional to forces applied to the cylinder. This input signal is summed with the signal from a DC permanent magnet tachometer. The resulting error voltage is chopped at 60 cps and fed to an AC servoamplifier. The amplified signal then drives an AC servomotor which has the DC tachometer on an integral shaft. Use of a direct current tachometer allows drift free operation. The gain of the AC amplifier is high enough to permit the motor to receive its starting voltage with a force input of 10 pounds.

Since gain of the first integrator is the product of tachometer scale factor $\left(\frac{\text{rad/sec}}{\text{volts}}\right)$, gear ratio and potentiometer "gain" (volts per radian), scaling in the tachometer feedback essentially determines the gain of the servo. In order to adjust the apparent weight of the work panel, a "system scaling control" potentiometer is inserted in series with the tachometer output. Acceleration can be read directly from the differential output of the pressure transducers. In order to prevent component loading of the acceleration readout to the measurement jacks, high impedances are inserted before the readout summing junction.

The SCR power servo is a closed loop around the work panel drive motor. Like the first integrator, DC summing is used. The threshold of this servo is set at the volocity proportional to a 10 pound force applied for two seconds to the "heaviest" apparent mass. The input to this integrator is the output shaft position of the first integrator read from a potentiometer. This is summed with the tachometer output voltage (negative). The resulting error voltage is amplified by an operational amplifier and fed to a power amplifier which drives a DC motor. Automatic switching between the 28 volt DC power supply and the motor field (controlled by the power amplifier) permits the motor to operate in either direction. These amplifiers and the motor are the same as those used in the LEM simulator built by Martin. Power requirements are 115 vac, 60 cps, 15A and 28 vdc, 5A.

2) Force Sensors

The hydraulic cylinder pressures are detected by 6 Bourns Laboratories Model 304 pressure transducers. Two transducers are used for each cylinder with the potentiometers in series.



As can be seen, a differential pressure in the cylinder causes both wipers to move in the same direction. However, variations in system pressure merely change the null positions without changing the electrical output.

All sensors are summed across 100K resistors at the input to the amplifier and in a separate network of 1 meg resistors at the channel readout jack.

3) First Integrator

The first integrator is operated in two modes, run and hold. The run mode is used when an experiment is in progress and the work panel must respond to input forces. The hold mode is used when the panel is being prepared and it should not respond.

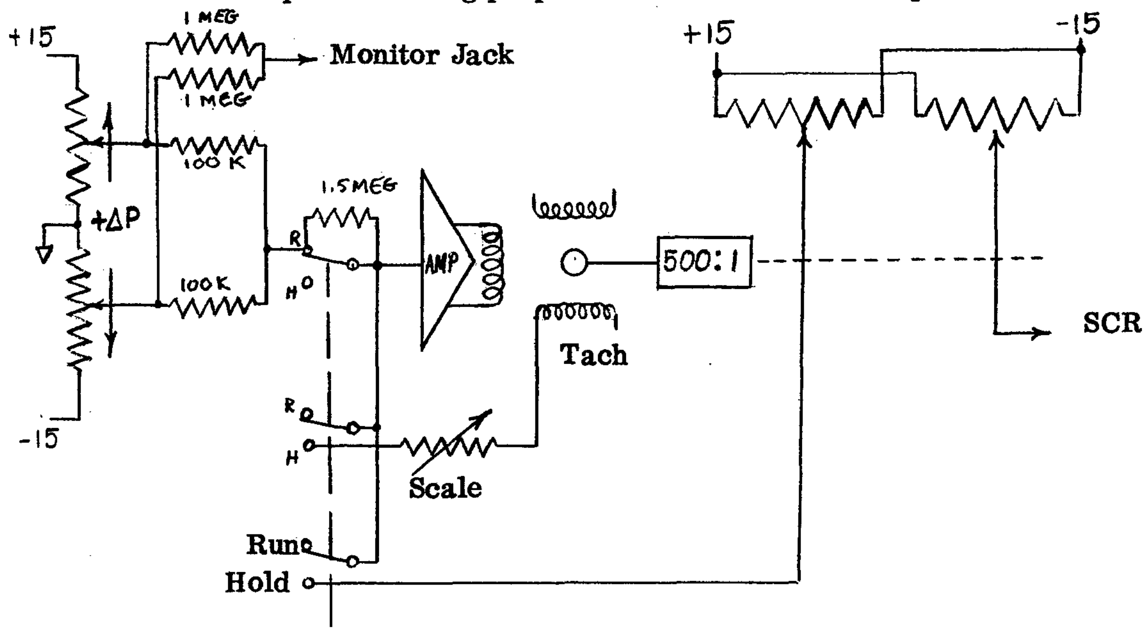


Figure 19. First Integrator Simplified Schematic

In the run mode, the input signal is mixed with a signal from the DC tachometer and fed thru a standard AC operational amplifier to drive the motor. The motor drives the tachometer directly and a dual potentiometer thru a 500:1 gear reduction.

The voltage on the potentiometer is proportional to the desired velocity of the work panel. This follows from the fact that the angle thru which the motor turns the pot is proportional to the force amplitude and time it is applied, FT=MV. The gain of the integrator determines the apparent mass. This is controlled by a potentiometer in series with the tachometer output. The potentiometer was sized to give a mass range from 3k to 7k lbs.

When the integrator is switched to the hold mode, a 1.5 meg resistor is inserted in the signal input to reduce the signals and a position pot signal is inserted into the input. The output pot and the position pot are driven in tandem by the motor. The servo will then seek a null position on the position pot. The mechanical alignment of the output pot is adjusted for the same electrical null on both pots.

4) SCR Power Servo

The SCR power servo controls the drive motors to maintain the velocity indicated by the output of the first integrator. In addition, circuits are provided for slewing the work panel to a desired position and stopping the motor when the limits of travel are reached.

The input signal from the first integrator is fed thru the slewing and limit relays to the summing network where it is mixed with the tachometer signal. Mounted on the pre-amp chassis are 5 circuit boards, 3 standard transistor amplifiers, 1 polarity sensitive relay and 1 SCR gate driver.

The signal is fed thru the first amplifier to the second and third. The second amplifier drives the polarity sensitive relay which controls the motor field switching. The third amplifier feeds the gate driver to control the silicone controlled rectifier amplifier.

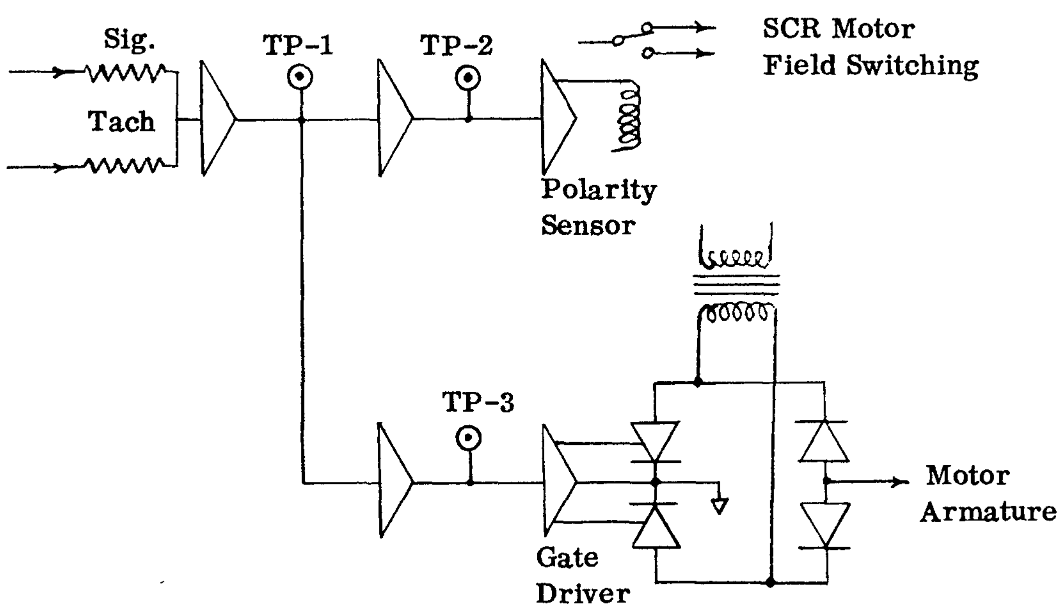


Figure 20. SCR Power Servo Simplified Schematic

Two 2N-683 silicon controlled recitifiers are driven by the output of the gate driver which is adjusted for a full 180° conduction angle at max. signal in. The rectifiers are protected by 25A quick blow fuses in series with the Stancor RT2012 transformers.

Separate power switches are provided for each channel as well as an emergency stop switch which removes power from the SCR's.

5) Physical Layout

The servo systems are housed in a Martin Co. standard rack with access from front and back for maintenance. The top of the rack is devoted to the valves and gages for the elevating mechanism. At normal operating level, the servo controls are mounted on a panel which swings down for maintenance. Figure 21 shows the front panel controls. Since the photograph was made, scaling pot dial indications were added for setting the system to 3, 4, 5, 6, or 7 K lbs. The connector in the lower right is for the remote cable which will parallel the slew and hold controls.

Immediately behind the panel (Fig. 22) are the first integrators. Between the channel amplifiers are stacked the motors, tachometers and, not visible in the picture, the output pots. The sensor mixing nets are on the circuit card at the right just behind the front panel.

Figure 23 shows the area just below the first integrators. Behind the blank panel, now removed, are the pre-amp chassis. The bottom panel contains the power control switches. Power is controlled separately to the SCR's for ease of maintenance.

Behind the pre-amp chassis (Fig. 24) mounted on the side of the rack, is the power supply. The relays above the pre-amp chassis are the slewing relays. Mounted in the bottom of the rack is the SCR chassis (Fig. 25).

All chassis terminal boards and connectors and major components have been marked to correspond to the designations in the schematic booklet delivered with the simulator.

D. Design Review

Prior to final release of drawings to the shop for fabrication a design review meeting was held on 16 March 1964 between the Air Force Project Officer, Mr. Chester May and the Martin Company Technical Director, Mr. Allen Holmes for the purpose of approving the design for fabrication.

A set of drawings and design notes were forwarded prior to the meeting and used as the basis for discussion.

In addition to the review of the simulator design, discussion of the interface between the simulator/bridge crane and the

Figure 21. Servo Operating Controls

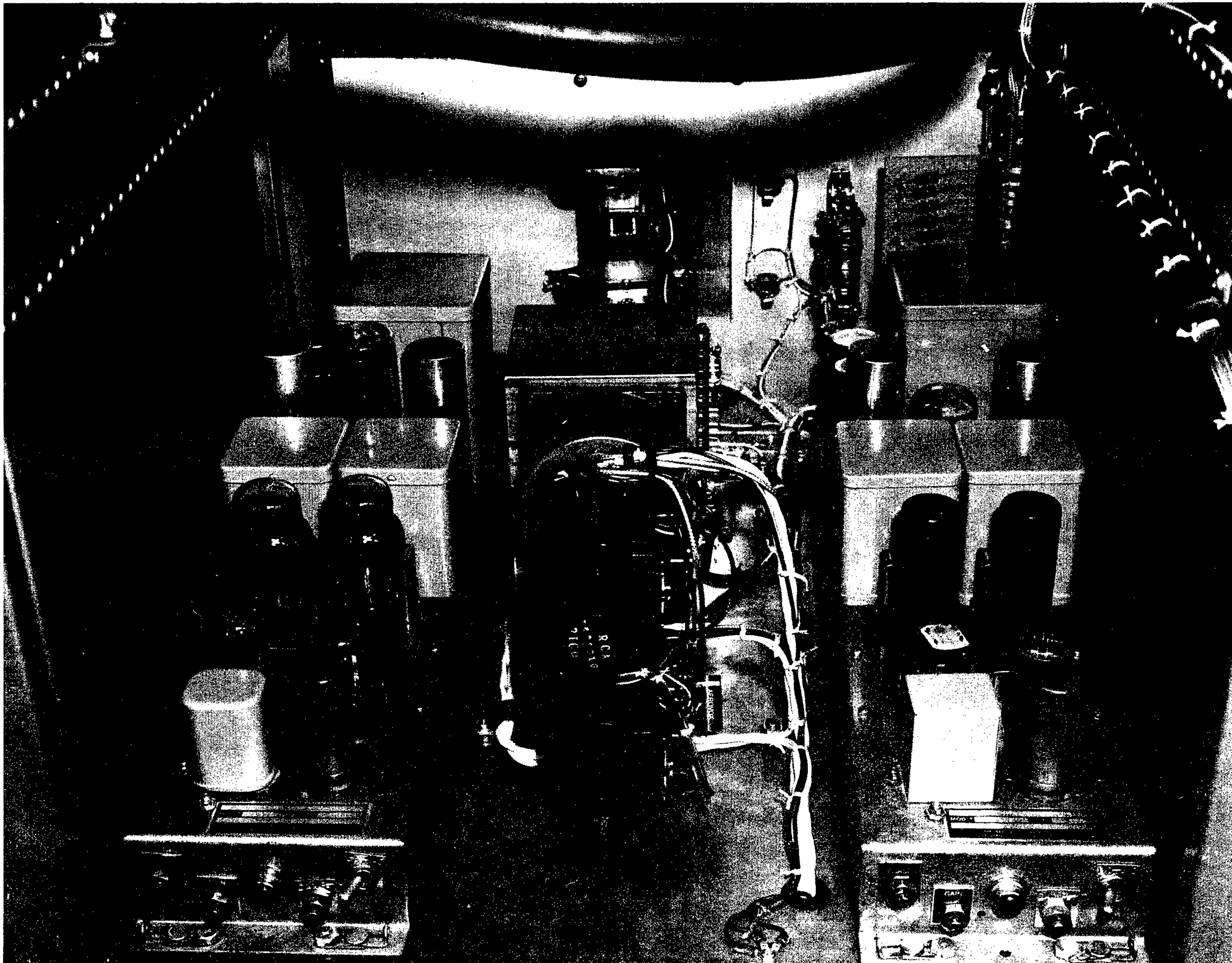


Figure 22. First Integrators

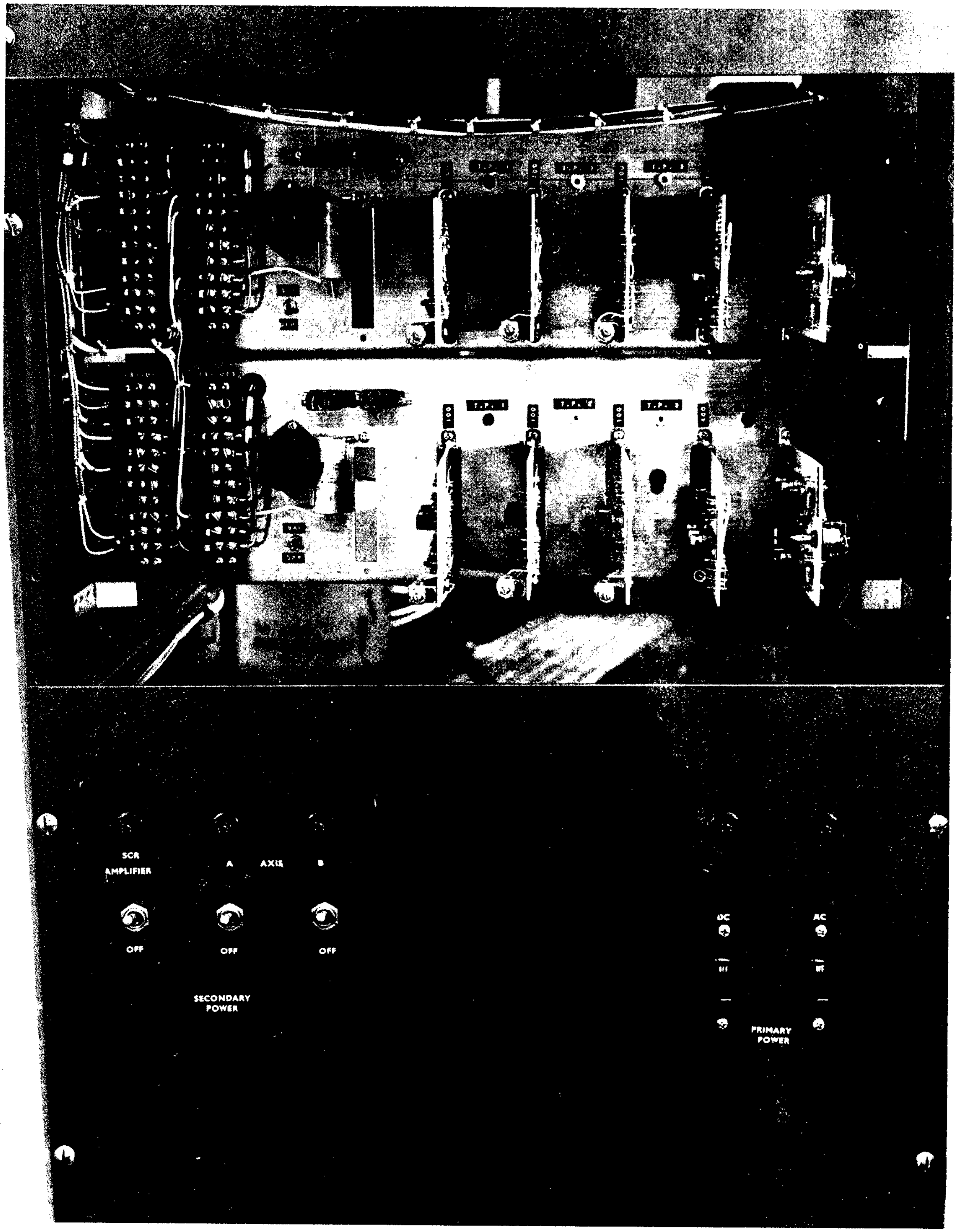


Figure 23. Pre-Amp and Power Controls

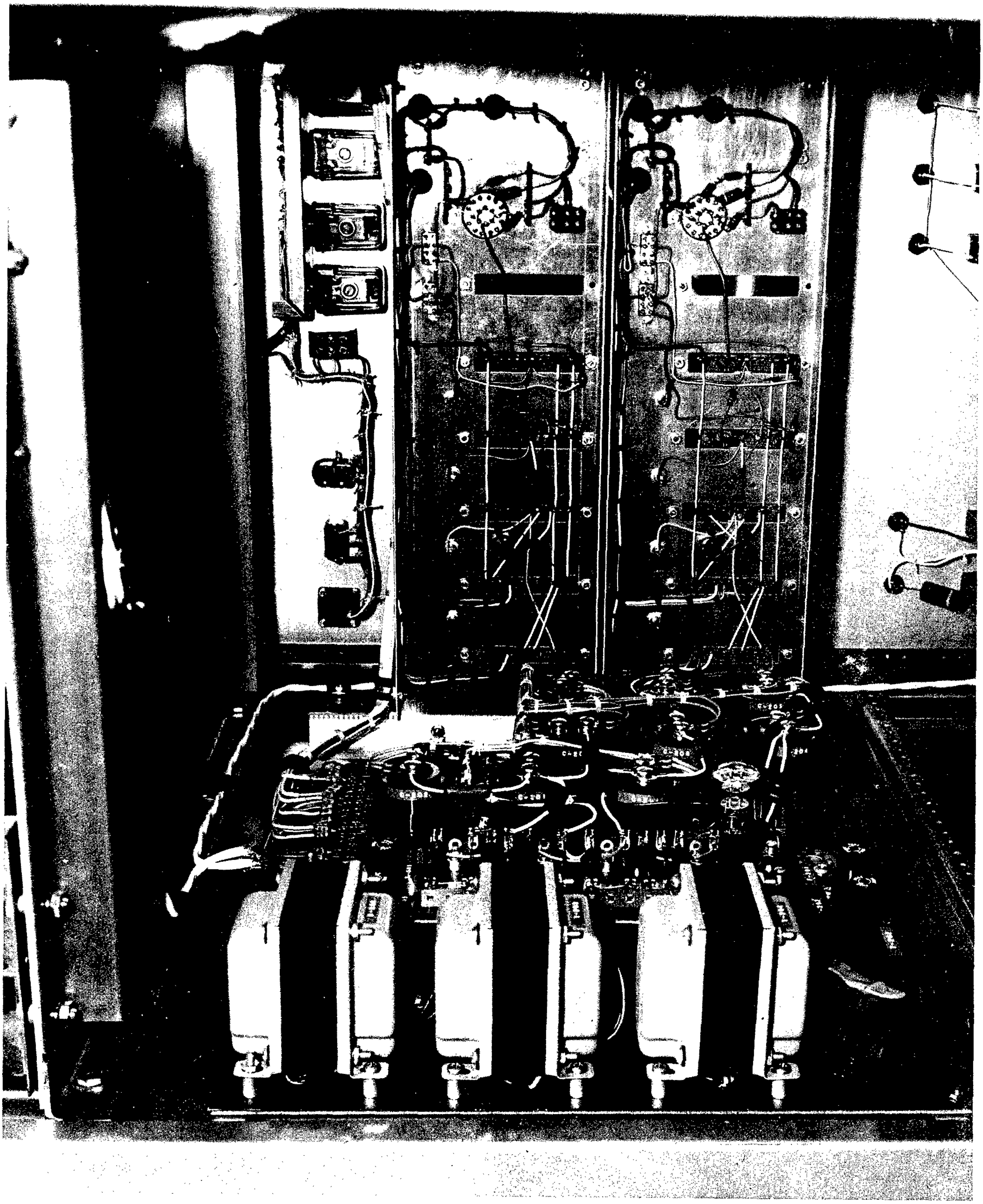


Figure 24. Power Supply

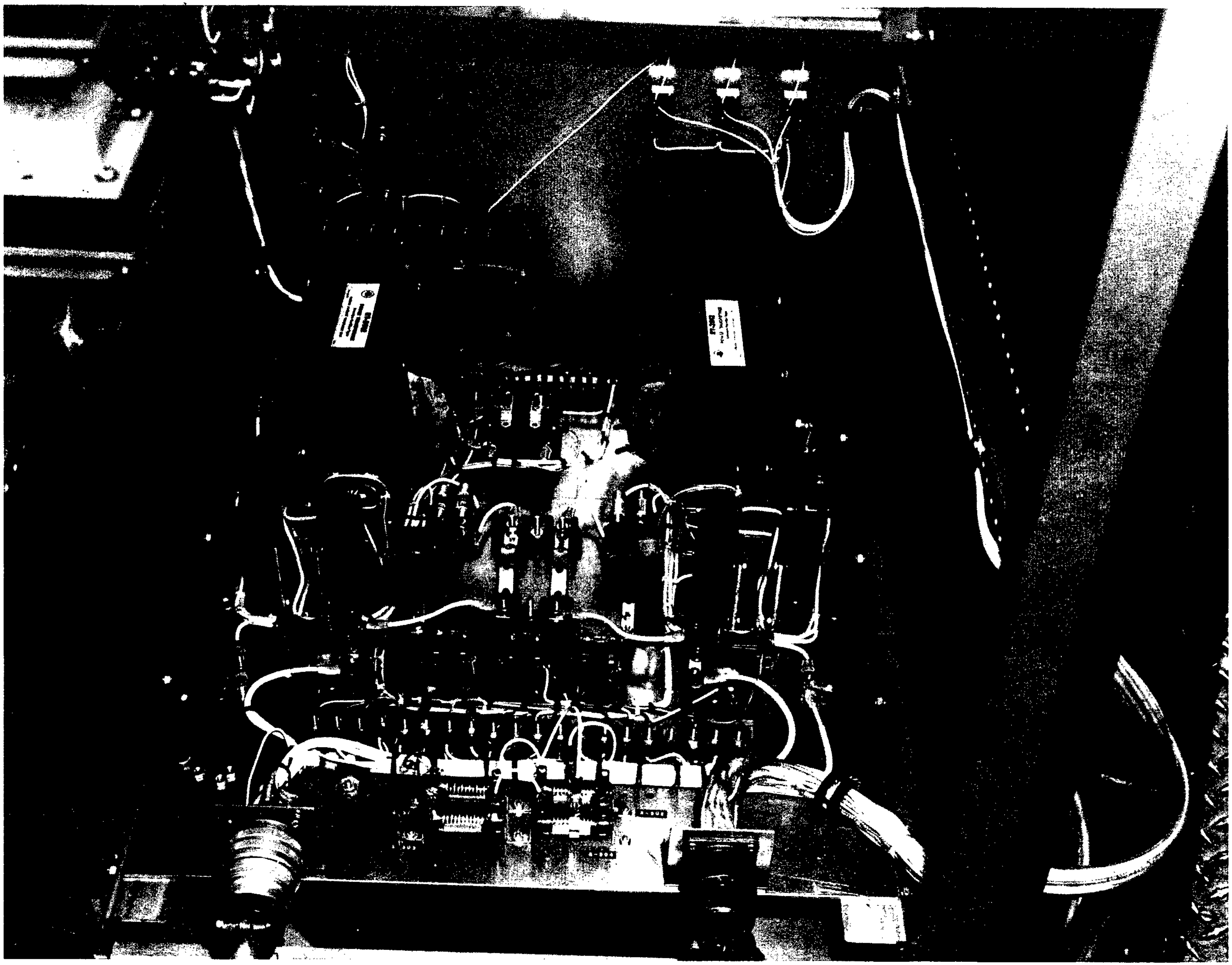


Figure 25. SCR Chassis

facility was held and the interface points initially defined. The interface points were the attachment of the crane rails to the facility hanging structure, the electrical power connections, the facility air inputs to the simulator and the lazy arm connections to the simulator.

Engineering liaison was maintained throughout the fabrication and test phases to resolve problems as they arose.

E. Study of Servo Balancing

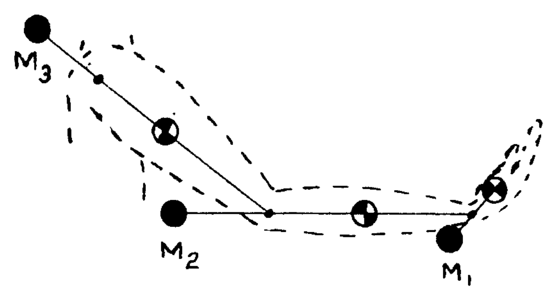
As a part of this program, the advantages and disadvantages of servo controlled balance mechanisms were weighed. It is our conclusion that the next major step forward in the development of this type of simulator must come thru the devising of a simple reliable means of detecting unbalance forces on the order of magnitude of the frictional breakout forces and automatically rebalancing before significant motion has occurred. No attempt has been made to numerically define the criteria for a servo balance system because much of the necessary basic data must be developed and this program's scope did not allow for extensive investigation along these lines. Data is required on such areas as acceptable balance accuracies, low reaction force detectable response thresholds, rates of CG displacement versus limb motion, and range of CG travel for the task spectrum.

Changes in center of gravity location have been estimated to be within a box 1.7x1.8x5.0 in. for an astronaut with pressure suit and self-maneuvering unit (reference 3). For a 200 lb. man without spacesuit, the range increases to a box 12.3x9.2x7.9 in. This range is related to the extreme limits of motion of a human. The normal task performance for probable maintenance tasks will not require such a range of motion, and the range of CG locations will therefore be less. In addition, spacesuit mobility restrictions will also reduce the range of motion. However, state-of-the-art advances in spacesuit design may increase the range of mobility to the point where the spacesuit is no longer a restricting factor.

The simulator is balanced by placing the center of gravity at the intersection of the pitch, roll, and yaw axes. Therefore, any shifts in the range of the center of gravity are significant to the balance problem. If the subject moves, his CG position shifts and is no longer aligned with the axis of rotation. In all probability the motion will have a horizontal component which produces a torque thru the action of the subject's weight. For a 180 lb. subject with a suit and back pack with total gross weight of 350 lbs., a horizontal shift of .0028 in. will develop a torque of 1 ft. lb. which is the required threshold of the simulator. Lower thresholds require even smaller displacements to produce unwanted rotation of the subject as he moves. This and previous similators have shown sensitivities in pitch to the clenching and unclenching of the subject's fists.

There are two possible approaches in a simulator of this type which relies on rotational balance to produce the simulated orbital motions; static balance in which the subject is counterbalanced so that his CG position never shifts when he moves, and dynamic balance in which the shift of CG is detected and the subject is repositioned in the simulator to restore the balance. This latter approach further subdivides into systems which measure CG location and systems which derive CG shift from measured movement of the subject.

The counterbalance approach is felt to be undesirable because of the complexity and added mass. Assuming that only the head and arms are free to move and the rest of the body is restrained as one mass, there are eight interacting masses to be balanced. The counterbalances must, of necessity, be placed in positions where they will not be restricted in their motion by simulator structure or back pack structure. This will be very difficult to achieve, especially for the head counterbalance. The mass increase will be large. For one arm alone the weight required is 51.2 lbs.



Weight in lbs.		Counterbalance in lbs.
Hand	1.24	$M_1 = 1.24$
Lower arm	3.44	$M_2 = \dfrac{3.44 \times 4.38 + 2.48 \times 10.19}{4.38} = 9.21$
Upper arm	6.05	$M_3 = \dfrac{6.05 \times 5.95 + (9.21 + 3.44 + 2.48)\ 13.65}{5.95} = 40.75$

$$\text{Total} \quad M_1 + M_2 + M_3 = 51.20$$

This assumes the counterbalance is placed at a moment arm equal to the distance of the segment CG from the hinge and the weight of supporting structure is neglected.

The dynamic approach appears more feasible. If the system of measuring limb motion and computing CG shift is utilized, a major problem will be the accurate measurement of motion of all significant body sections, preferrably including the legs and feet and the computation in three dimensions of the resultant CG shift. This implies measurement of the following minimum movements:

Hand	2	x2	4
Lower arm	3	x2	6
Upper arm	2	x2	4
Shoulder	2	x2	4
Head	2		2
Foot	1	x2	2
Lower leg	1	x2	2
Upper leg	3	x2	6
		Total	30

Also, factors must be introduced to account for tool weights.

This means that at least 37 channels interconnecting the subject and computer would be required, 30 inputs, 3 outputs and 3 feedbacks (P, R, Y), and 1 power. Compromises could be made at the sacrifice of accuracy. With accuracy of CG location required on the order of .003 in., this may not be feasible. Thirty-seven sets of slip rings per axis or a telemetry encoding system would be required. At the expense of limiting rotation to one or two revolutions, a continuous wire system could be installed, but higher rotating friction would be encountered as the wire is flexed.

The computer required would depend upon the ultimate degree of accuracy desired and feasible and the response time required for the calculation. A response of approximately 1 second would be required to rebalance before appreciable motion of the subject would occur.

In the system where the CG location is measured directly, a major problem is discriminating between torques caused by CG shifts and torques caused by reaction forces. If the subject pushes horizontally with a force of 10 lbs. at a point 6 in. above his CG, he produces the same torque as if his CG shifted .014 in. There is no way of determining whether the instantaneous value of torque causing rotation is due to pushing or CG shift.

However, if the time history of the force is considered, it appears possible that the distinction could be made. An unbalance force is a function of the horizontal displacement from the axis and the rate of rotation. A decaying sine wave will be the force/time wave shape with the period equal to period of the pendulum oscillation of the subject. During the time when a balancing response would be required, the force would be essentially constant.

A reaction force on the other hand, will be of highly irregular shape and of short duration because the subject will either rotate or translate out of arms reach quickly unless very small forces are applied. Low pass filter circuits which react to only the unbalance motions could be designed so that the CG would be shifted in a manner which cancels low rates of rotational acceleration, but which does not react to high rates. Two possible mechanizations of this approach would be: 1) an electrical filter which measures angular rates thru pickoff on each axis and controls the axis balance servo loop and 2) a gyro referenced system which maintains subject attitude around the reference gyro to null out detected low acceleration rates but which will reorient the reference attitude under high rates. A problem with both systems will be determining the threshold rates so that rotational accelerations caused by small reaction forces will not be seen as a slight unbalance. Also, the problem of simultaneous reaction acceleration and CG shift would require circuits capable of detecting the acceleration of CG displacement superimposed upon constant velocities resulting from push away accelerations. Experimentation is required to determine the acceptable threshold and response rates for such systems before design analysis would be worthwhile.

An additional approach to the problem would be to utilize the principle that a free body tends to rotate around its CG. If the subject were caused by a rotating mass to oscillate as a conical pendulum, he would tend to oscillate around his CG, but the gimbal structure would restrain him. As a result stress would be set up in the cradle supports proportional to the displacement of the CG from the support point. By measuring forces in two axis on each support and comparing the phasing with the driving pendulum or comparing relative amplitudes, the location of the CG relative to the rotational axis can be fairly simply calculated. Actually a simple null seeking servo of sufficient sensitivity is all that would be required in addition to a comparison circuit which derives the x, y and z components of the unbalance. This system has the advantage of being sensitive to holding tools and parts which are seen as a portion of the total mass.

Before design of an automatic balancing mechanism can be fruitfully started, much investigation into the parameters of the problem will be required. At this point all that can be said is that some of the concepts discussed above appear fruitful. The advisability of proceeding to investigate the problem is not really arguable when one considers that improvements in spacesuit mobility will make development of "acrobatic" techniques for maneuvering in space desirable as a back up if not as the primary mode. The use of simulators capable of studying this area as it applies to maintenance, assembly and handling will be mandatory for several years to come because of the problem of limited availability of orbital facilities and subjects.

F. Design Data

A complete set of Martin Co. drawings 861-00068 sheets 1 thru 7 and 861-00070 sheets 1 thru 11 were submitted along with a schematic booklet for the servo system.

In addition, an informal Operating Instructions was prepared and delivered with the equipment. Air dryer operation and maintenance instructions from the dryer manufacturer were also delivered.

III. Fabrication

Fabrication of the simulator was accomplished in the Martin Advanced Manufacturing Technology Laboratory by the same team that fabricated the two 5° simulators built previously for Martin in-house use and for NASA-MSC.

Most of the fabrication was a straightforward application of standard manufacturing techniques with emphasis on close engineering and procurement liaison to assure quick reaction to problems as they arose.

Insofar as possible, the simulator was designed and built with the maximum fitting at assembly to minimize tolerance problems in detail manufacture. Quality control was exercised over all phases of fabrication with basic quality guidelines being laid down in design reviews which coordinated engineering tolerance requirements with manufacturing capability and ease of fabrication. Those areas of fabrication worthy of special note are the yaw ring and the body support corset.

The yaw ring was fabricated from two tubes of 4130 steel, 1 7/8 in. in dia. bent in a semi-circle. Since the tube bending dies which were available did not cover the 1 7/8 tube for a 30 in. dia., a simple but ingenious approach was taken. A die of the next largest diameter, with a 15 inch bend radius was used and a tube of the proper diameter was obtained. The 4130 tube was slipped inside the larger tube and centered with three short wires of appropriate diameter at each end. Then the whole tube was filled with "Serabend" a lead-like material with a very low melting point and the complete assembly was bent in the tube bender, after which the Serabend was melted out and the 4130 tube removed.

Both 180° sections of tube were then trimmed and butt welded to form a continuous ring. The cradle bearing tracks were formed and welded to the tube with short intermittent welds spaced widely around the ring to equalize heating and minimize distortions. Welds were added until a complete bead was formed on all tracks. Approximately 1/8 inch had been allowed for warpage during welding but only .035 in. distortion was measured on a surface table. After welding the ring was hardened and the track faces machined.

The body support corset was fabricated from a mold of a 186 lb., 6 ft. 0 in. subject. The subject was wrapped in a polyethelene sheet and laid on the foundry table. Sand was packed around him from the neck to the knees (Fig. 26). The sand which was treated with silica gel was hardened by forcing a tube into the pile and blowing carbon dioxide thru it. After the mold was cleaned up, a plaster positive was cast (Fig. 27). On the positive mold, fiberglass and "Renite" epoxy was laid up 1/4 inch thick. After the renite had hardened, the shell was removed, split and trimmed and the mounting brackets were attached.

IV. Installation and Test

A. General

The simulator was designed to be installed and operated at Wright-Patterson AFB, Dayton, Ohio. Early in the design phase interfaces were defined with the facility, and problems of installation which affected design were studied and resolved. Prior to shipment to WPAFB, the simulator was installed in the manufacturing area for final testing and debugging.

The installation at Martin Co. differed from the final installation only in areas where the effect upon test results were negligible. Because the simulator and bridge crane were essentially two separate pieces of equipment with only the control rack in common, it was possible to set up and test them without reproducing all features of the final installation.

The bridge crane was hung so the work panel was only 30 inches from the floor at the centerline in order to make it easier to work on the equipment. See figure 1 and 28 for a comparison of installations.

The relationships between control rack, air dryer, air tank and simulator were also different, but since flexible hose was used for interconnecting components no problem was encountered. The simulator was not used with a lazy arm. Tests of horizontal displacement forces were made over short distances and hose drag accounted for. The air tank was not installed in its vertical position because of limited ceiling clearance.

B. Test Equipment

1. Impulse Tool

The basic test equipment was utilized in both places. An impulse tool was designed for use with bridge crane and the simulator (Fig. 29). It consists of a horizontal H beam with the web vertical. A small cart rides on the beam on ball bearings. Extending forward from the cart is a 3/4 in. dowel which transmits the impulse to the work panel. To apply the impulse a vertical beam is attached to the front of the H beam. At the top and bottom pulleys carry a small flexible wire rope

Figure 26. Subject in Mold

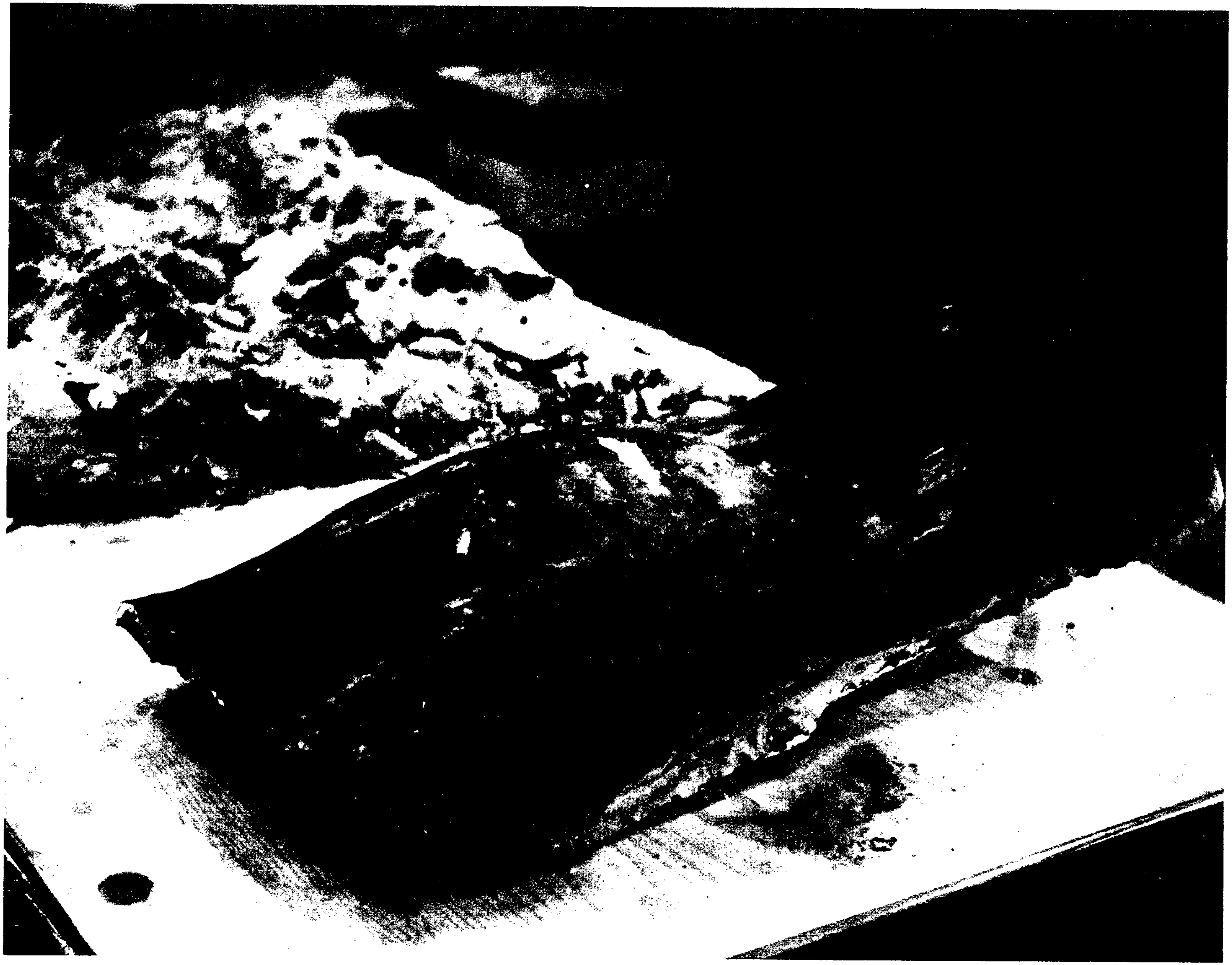


Figure 27. Body Cast

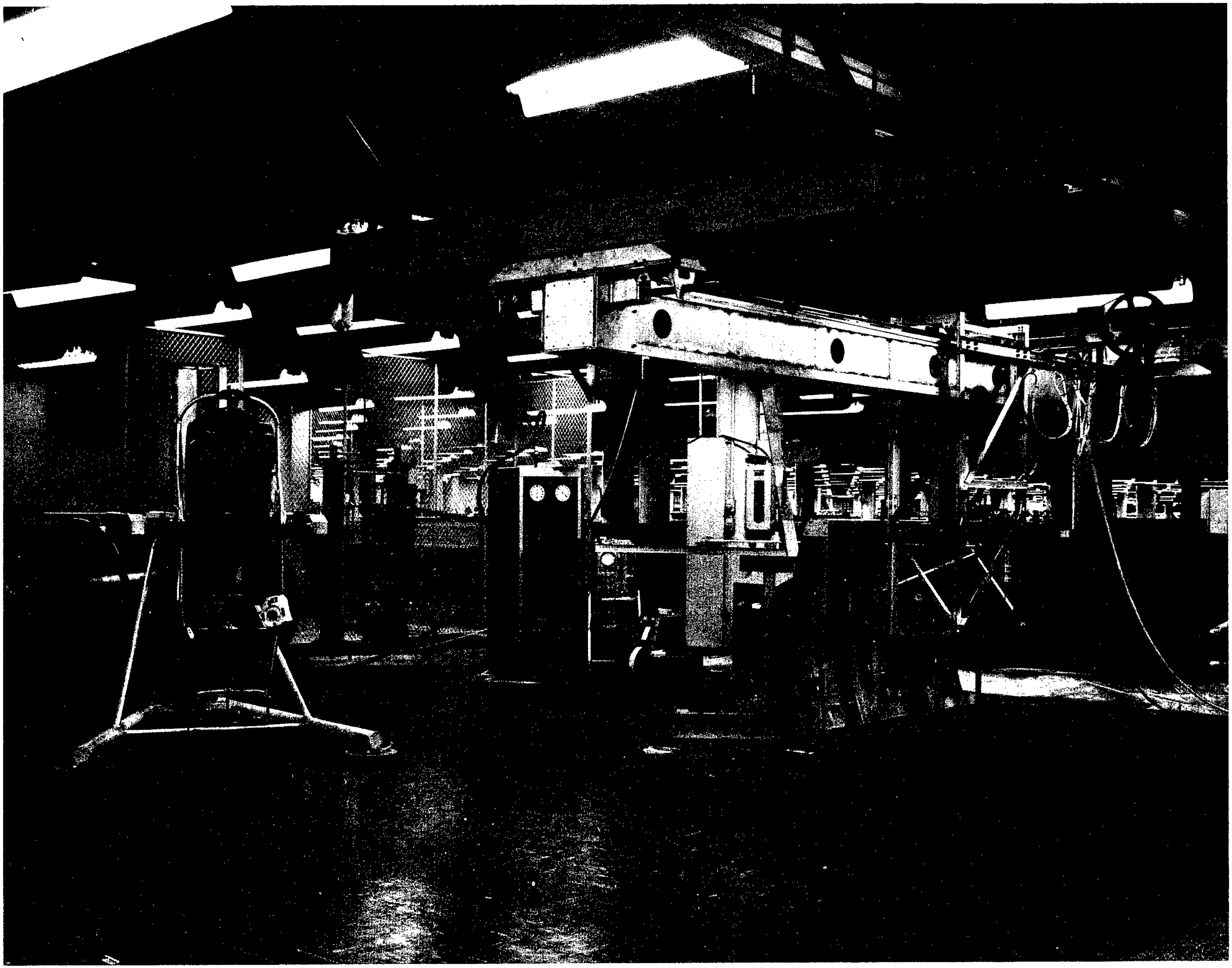


Figure 28. Test Set-Up (Baltimore)

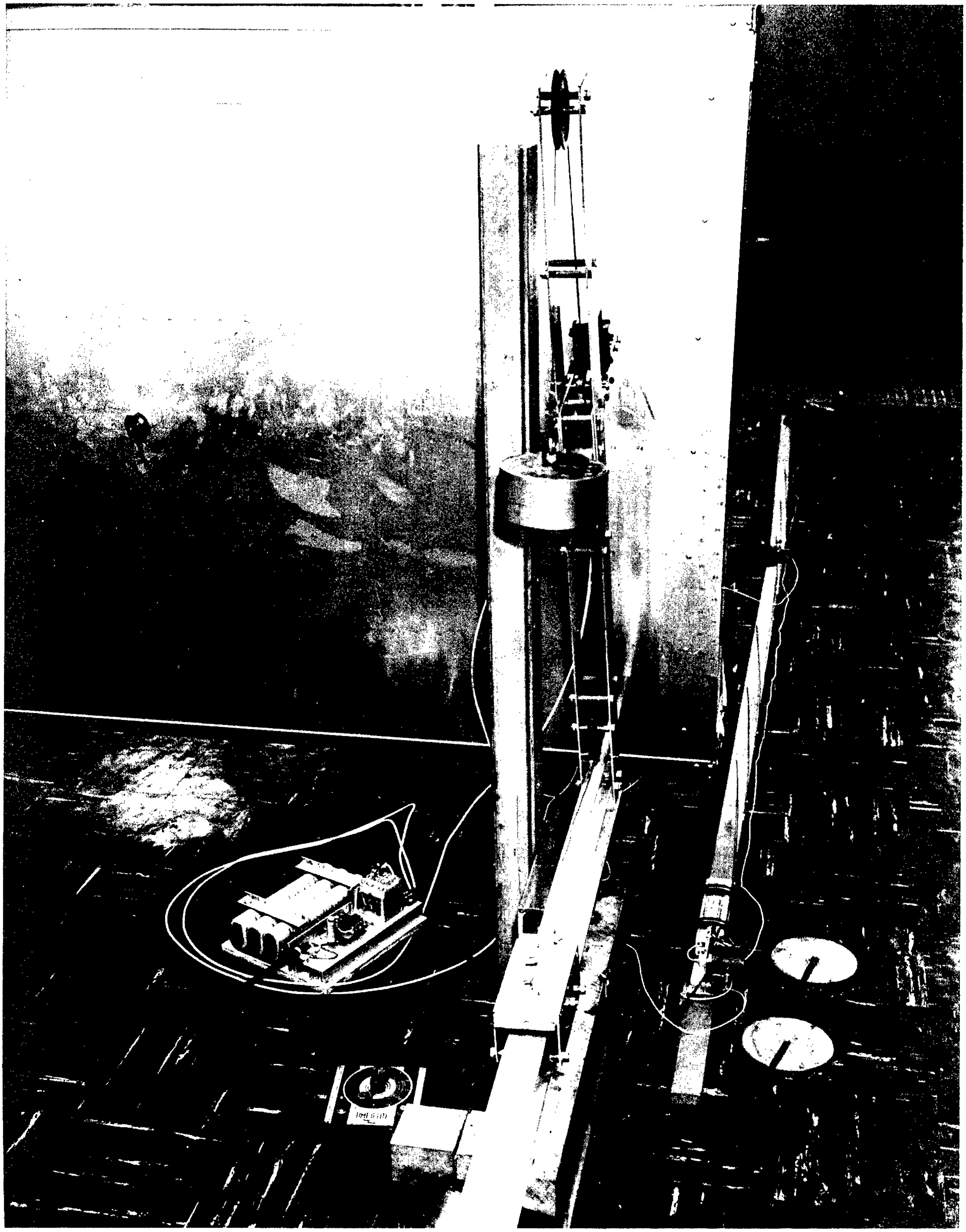


Figure 29. Impulse Tool Set-Up

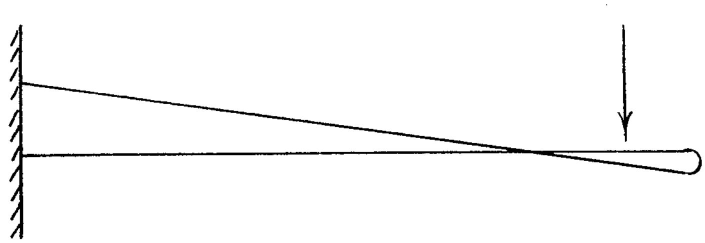
attached at one end to the cart and suspending a weight from the other. Various weights can be used to apply a constant force over a distance of up to 24 inches. The duration of the impulse is controlled by a clutch on the vertical beam which holds the force of the weight off of the horizontal cast until it is electrically released. A spring re-engages it when the circuit is broken. The clutch solenoid is controlled by a standard photo timer. Since the main purpose of the tool was to initiate motion in the direction of the impulse a tail off in force was accepted for design simplicity. At only the very low velocities is the time it takes for the clutch to take up the weight and remove the impulse force significant. The tool is supported on an angle bolted to a heavy metal base. It is adjustable to three positions corresponding to the top, center and bottom of the work panel in the test installation. For tests at WPAFB the tool was set on a standard Air Force work platform which was adjusted to the proper height.

In figure 29 the impulse tool is shown set up to apply a 20 lb. impulse to the bottom right corner of the work panel. The photo timer is at the left close to the base. Between the work panel and the photo timer is the DC battery supply which was used because the solenoid required a DC input. At the right are 5 and 10 lb. lead weights which permitted testing in increments of 5 lbs. up to 35 lbs.

2. Distance Markers

At the right of the work panel extending rearward is the distance marker tool consisting of piano wire markers set at 1 ft. intervals for recording the travel of the work panel. A battery supplies power to the circuit which is closed by a horizontal finger, clamped to panel, contacting the vertical marker. Leads are run to the recorder which provides the time base for deriving velocity from the distance markers.

The markers are shaped as shown below to limit their oscillations to a plane normal to the direction of travel for accuracy of measurement. The markers were set to the nearest 1/16 in., a tolerance of 0.5 percent.

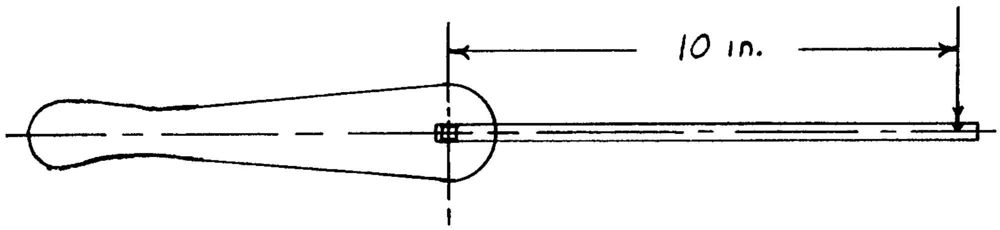


3. Recorder

A two channel Brush Mark II recorder with event markers was used for all recordings. Generally, simultaneous recording was made of force input, integrator output signal and distance markers with paper speeds of 5 mm per second.

4. Force Measurements

Standard Chatillon Co. spring balances (fish scales) were used for measuring forces required to cause simulator motion. A 0-2 lb. and 0-50 lb. scale were used. In addition, a Snap-on model TQ 3-FU -30 in. lb. torque meter was used for measuring rotational friction and was also used with adapter for applying linear forces up to 3 lb.



To apply a linear force, a 10 in. adapter was attached and the wrench was held securely perpendicular to the line of the applied force. With the force applied at a 10 in. radius, the dial reads directly in 10ths of a lb. With a 16 in. adapter the dial will read directly in oz. Care must be exercised to maintain the perpendicularity of the adapter and the line of the force.

C. Bridge Crane/Work Panel Installation and Test

Because the manufacturing schedule completed the bridge crane first, it was tested first. The rails were hung and leveled and adjusted for parallelism to within 1/8 inch. The end plates were removed and the box beam was lifted by two fork lifts and slid onto the ends of the rails. After the box beam was hung the vertical support was positioned under the beam, the upper casters and channel were removed and the vertical support was lifted up against the beam. While it was held in place the top casters were replaced. The work panel was then hung on the support linkage and connected to the force sensing system.

At Wright-Patterson AFB the same sequence of installation was followed. However, prior to installing the crane rails the air tank was lifted upon end and carried into place by fork lift. The facility supports at Wright-Patterson had been terminated just above the crane hangars. Angles to support the crane rails were located and welded in place to position the rails so that the top flange was 13 ft. from the floor.

After all structural elements were in place the final wiring was done. Cables had been made up prior to installation but the cable to the vertical support had not been terminated. Terminal boards were installed on the box beam and the cables run to the vertical support, cut to length and terminated.

In the final installation a 1/4 in. dia. wire rope was stretched parallel to the crane rails 1 ft. above the longitudinal motor mount. The cables between the rack and the crane were hung on pulleys, free to move along the wire rope with the crane. The cables between the control rack and the bridge crane were connected and power was applied. Since the system had been bench checked prior to final assembly no hook up problems were encountered.

When power was applied it was discovered that the SCR power servo fuses would blow in both channels when slewing. The cause was finally traced to the drag in the system overloading the motor. Tests were made of the forces required to cause translation and they were found to be about 50% higher than expected. Corrective action for the transverse channel was taken in two areas. The motor drive system friction was reduced by switching from V belt to chain drive and by cleaning the grease from the bearings. A light oil was used because the anticipated speeds were quite low. In addition, the casters were contributing significant friction so they were modified. Ball bearing casters which would be a direct substitute could not be found, so the casters were disassembled, the wheels bored out and a roller bearing pressed into each wheel. The drive wheel pressure was eased and a standard belt dressing was sprayed on the drive wheel track to improve the traction and reduce the pressure required. After modifications were completed the force required to move transversely was measured at 25 lbs. and the SCR's operated satisfactorily. On the longitudinal axis the trolley wheels were already ball bearing types so the only corrective action to be taken was reduction of drive wheel pressures by backing of the large nut on the trolleys and change from V belt to chain drive. Since the change of the second step of the drive mechanism would have involved removal of the crane from the rails, it was decided to make the change only on the first step of the drive and test it to see if it would work before changing the second step (refer to fig. 18). change was sufficient.

After the SCR operation was resolved, attention was turned to calibration of the system scaling controls. The work panel and test equipment were set up for longitudinal movement forward, i.e., toward the back of the panel. The force sensing system was pressurized to 400 psig and impulses applied to the center of the panel. Originally we intended to record force and distance on each run but when the velocities obtained from a standard 10 lb. 2 sec. impulse were found to vary greatly it was decided to record integrator output as well. Repeated runs showed that the integrator voltage was varying both in level achieved by the impulse and during the travel of the work panel over 10 ft.

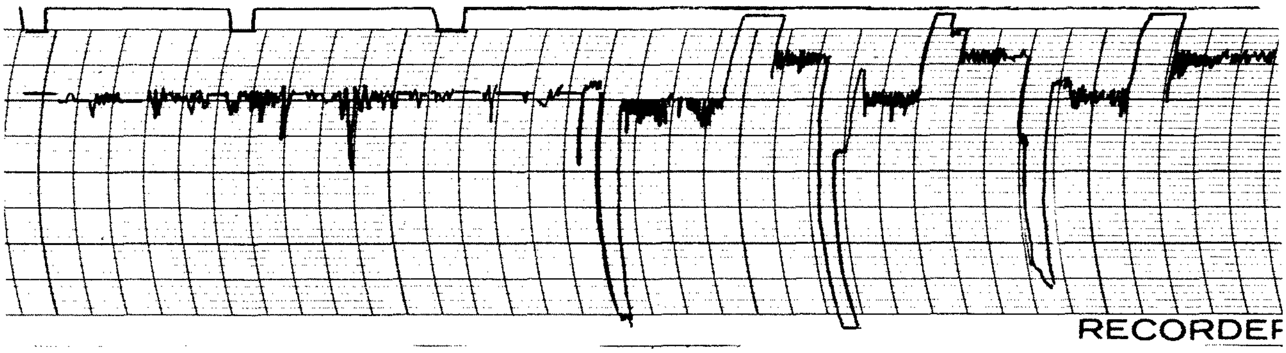
The integrator had been calibrated on the bench with simulated force signals. It was sensitive to 100 MV input. The force sensing system was creating the variations in integrator voltage at a level not discernible in the recorder force trace but at a level to which the integrator would respond. The trouble was found to be in the friction of the O ring seals in the hydraulic cylinders. They require a breakout force of about 8 lbs., and depending upon the random preloading introduced by panel vibration, wide variations in signal would occur when the same force was applied. In effect, the signal to noise ratio in the hydraulic system was too low. The variations in force signal caused by piston friction were above the threshold of the integrator amplifier so that even with no force applied a differential pressure existed great enough to generate a voltage above 100 MV which caused the integrator to run.

The amplifier gain was reduced to increase the required threshold but this resulted in a decreased sensitivity and an inadequate range of scale pot settings. Because the problem was the ratio of desired force signal input at 10 lbs. to the signal resulting from the failure of the piston to center itself, it was decided to increase the force signal seen by the piston by a factor of 6:1 so that the amplifier gain could be reduced but the same equivalent output voltage would be seen when a force was applied.

Both channels were modified and retested to determine the sensitivity. Final calibration was delayed until the final installation at WPAFB, but tests were run to confirm that the response to 10 lb. for 2 sec. would be variable over a range of simulated weights from 2.5K to 7.5K.

Figure 30 shows recordings taken at WPAFB confirming the nulls for both channels. The recordings were made in the run mode. Manual impulses were applied and released and the panel allowed to return to a null. Recording speed was 1 mm/sec. It was discovered that a mechanical bias was required to attain a good electrical null. To set the bias, the hydraulic system valves are opened and the panel pulled forward so that the lever makes a small angle from the vertical. The valves are then closed and the null checked by applying alternate pushes and pulls and monitoring the force signal with no force on the panel. If it is below 100 MV in both directions, the null is satisfactory. If not, repeat the procedure. The direction of adjustment, i.e., increase or decrease of bias angle, is determined by the average value of the nulls.

After the nulls were adjusted, calibration runs were made on each channel but because it was felt that the installation at WPAFB could possibly introduce slightly different loads and, therefore, change the response slightly, final calibration was delayed until the final installation. Fig. 31 shows typical calibration recordings. The right hand trace is the integrator voltage, the left is the force signal and the left markers are the distance indications.

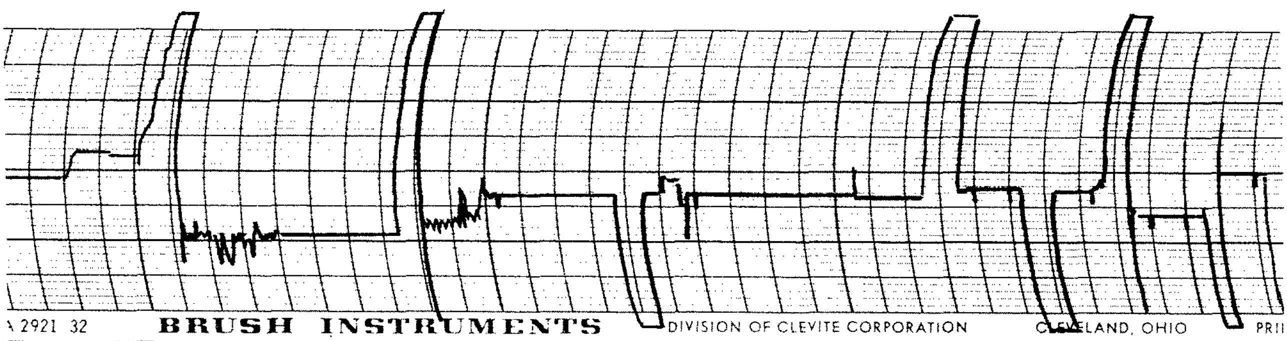


Channel A Null Check (7/24/64)

Null after:

pull -60 mv
push 0
pull -60 mv
push +1 mv
pull -60 mv
push +15 mv

Null spread: .075v peak-to-peak



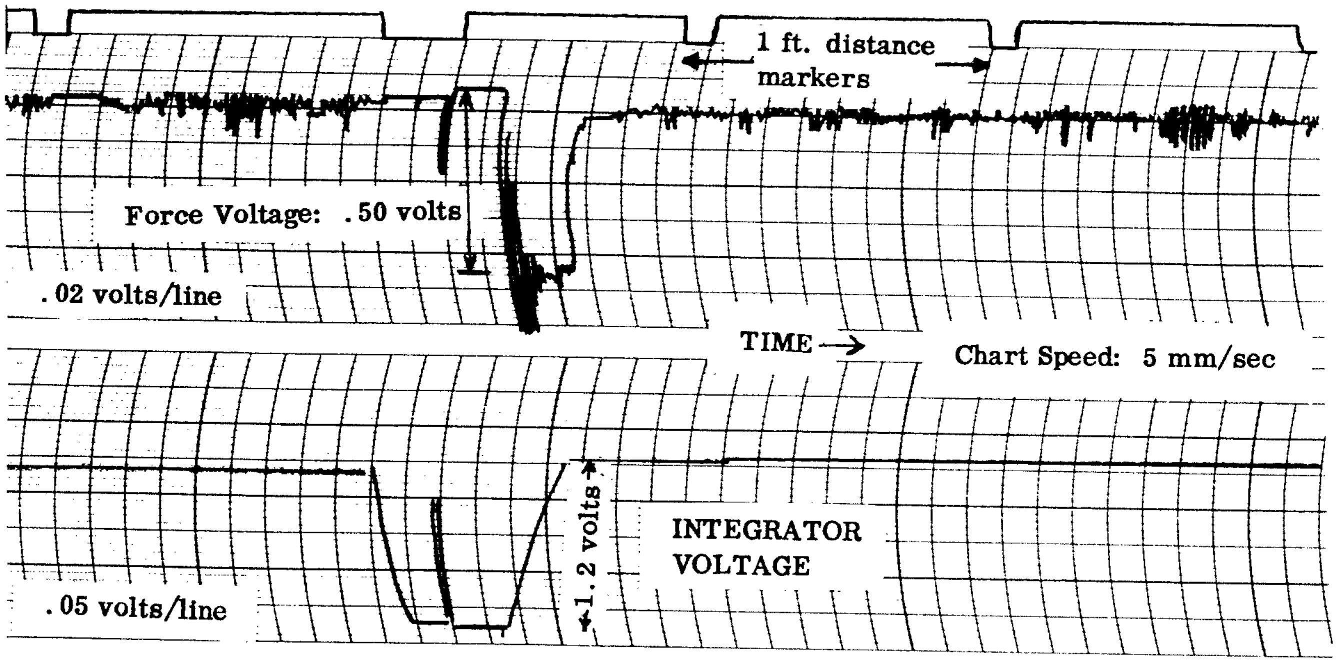
Channel B Null Check (7/24/64)

Null after
push to:

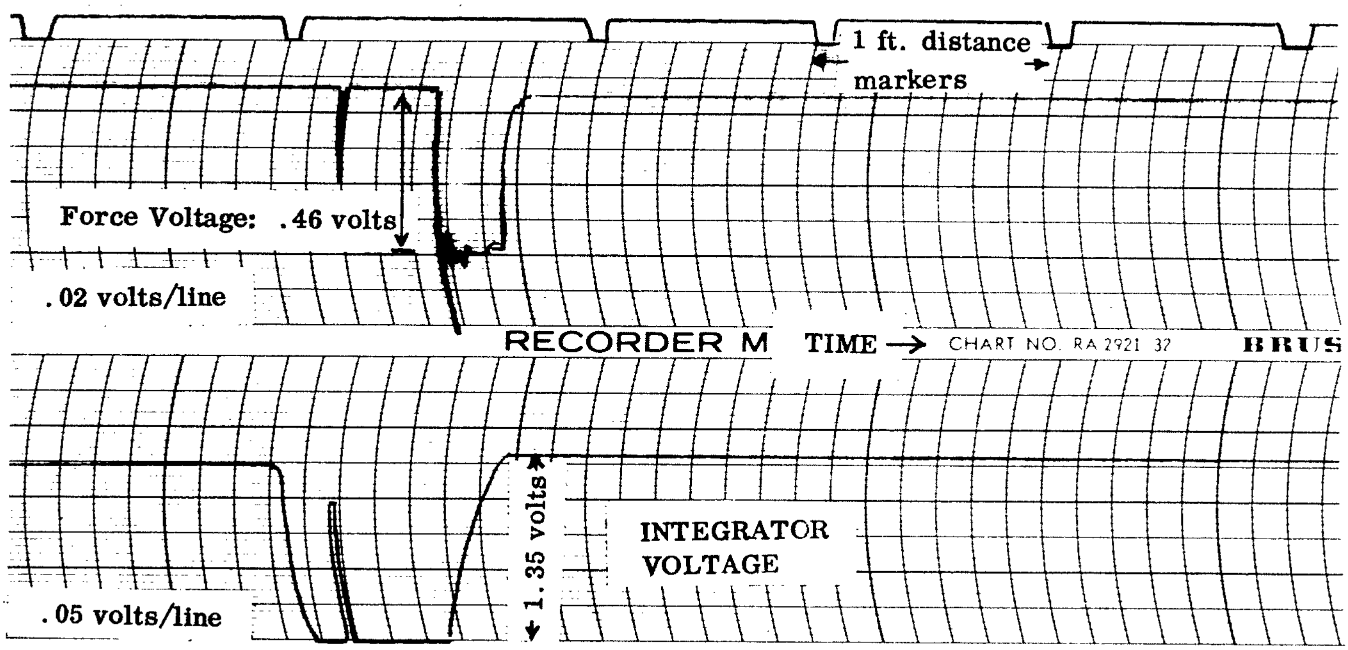
R +60
L +30
R +25
L +30
R +70
L +90

Null spread: .090v peak-to-peak

Figure 30. Null Voltages



Channel A - Scale pot at 55°
Force = 10 lb. for 2 sec.
Velocity = .113 ft./sec.
Simulated mass = 7667 lb.



Channel B - Scale pot at 70°
Force = 10 lb. for 2 sec.
Velocity = .151 ft./sec.
Simulated mass = 4250 lb.

Figure 31. Typical Calibration Runs

Although the crane has a travel of 19 ft. longitudinally and 11 ft. transversely, only 10 ft. of travel was utilized. The impulse tool was set to apply an impulse prior to contacting the first marker. The panel was allowed to travel 10 ft. and then it was stopped and returned. Runs were made at increments of dial rotation and a curve plotted for each channel (Figs. 32 and 33). Figure 34 was used to convert the velocity in seconds per foot to simulated mass with an impulse of 10 lb. for 2 sec. The dial calibration marks were derived from figures 32 and 33 and are noted on the figures.

In figure 31 the comparison between channel A and B shows the noise level from vibration of the structure. Channel A has a noise level which was significantly higher at WPAFB than it was at Baltimore. This was caused by a slight misalignment of the supporting structure which caused an inward force on the right crane rail. This made the flange of the trolley wheels rub on the track. It was not possible to adjust the trolley and rail to compensate completely so the gain of the integrator was adjusted for a threshold just above the noise level and the system was calibrated at that gain setting. After the calibration of each channel the channels were set for a simulated 5000 lb. mass and further tests made. Figure 35 shows the typical response of the system to varying forces. The maximum possible velocity was measured by manually driving the integrator to maximum with the SCR power servo turned off. After the integrator was at the maximum 15 volts, the SCR's were switched on and the travel recorded at 25 mm/sec. (Fig. 36). The maximum velocity attained in channel A was 1.92 ft./sec., in channel B it was 2.08 ft./sec. Note that in both channels the work panel was accelerating thru the first recorded foot of travel as indicated by the longer time it took to travel 1 foot.

D. Simulator Installation and Test

1. General

The simulator installation and test were conducted after the bridge crane tests in Baltimore and at WPAFB. The air tank, air dryers, control rack and simulator were connected to the facility air in accordance with the schematic shown in Figure 8. At Baltimore the lazy arm was not used and a flexible hose was directly connected. Although the facility air pressure at WPAFB was only 75-80 psi as compared to 90-100 in Baltimore, the feed to the simulator was of larger diameter with the result that in Baltimore the vertical air pad inlet pressure was 45-50 while at WPAFB it was 60 psi.

2. Rotational Tests

The back pack was installed in the simulator along with a 176 lb., 5 ft. 10 in. subject. The simulator was balanced on all three axes.

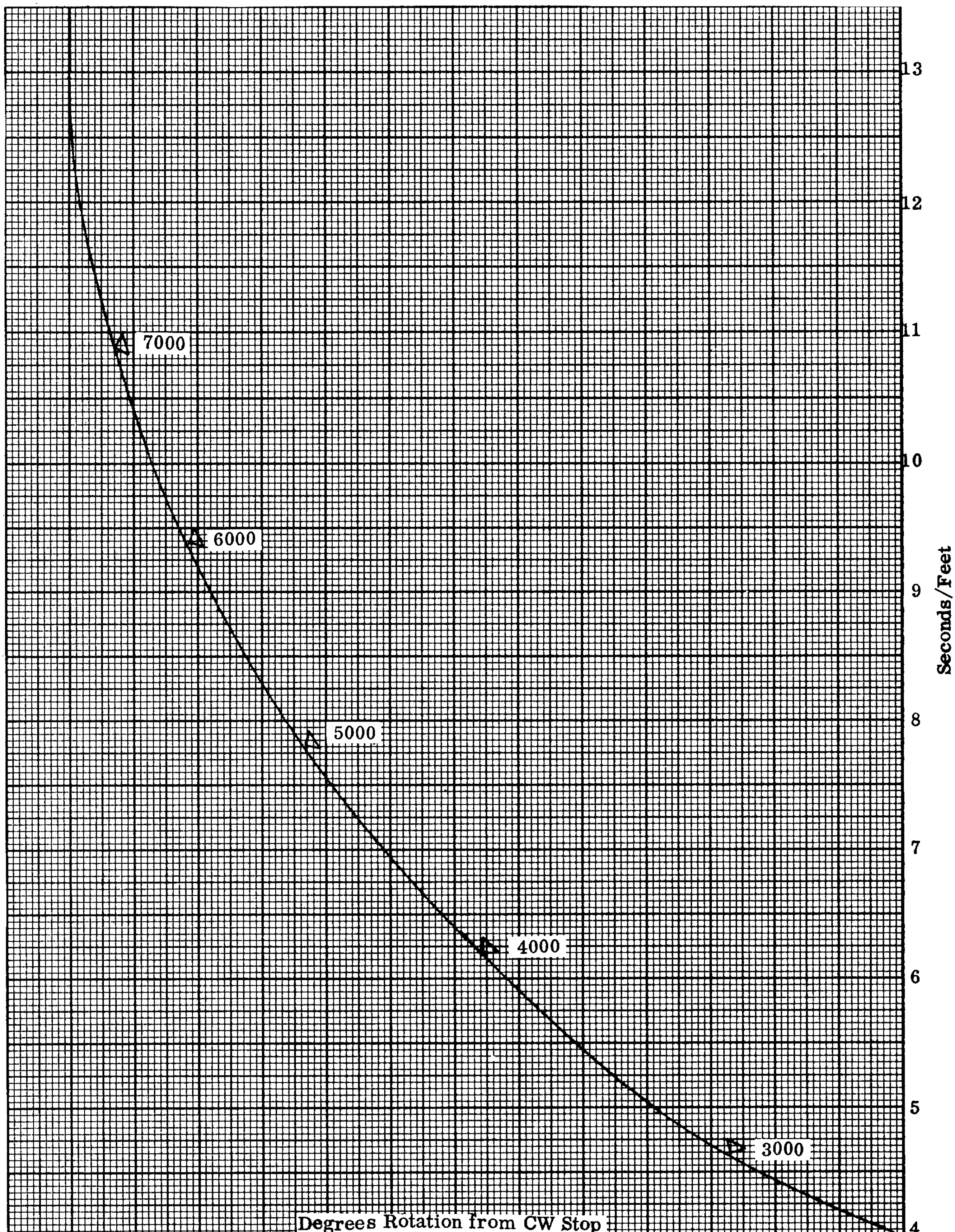


Figure 32. Channel A Scale Pot Calibration

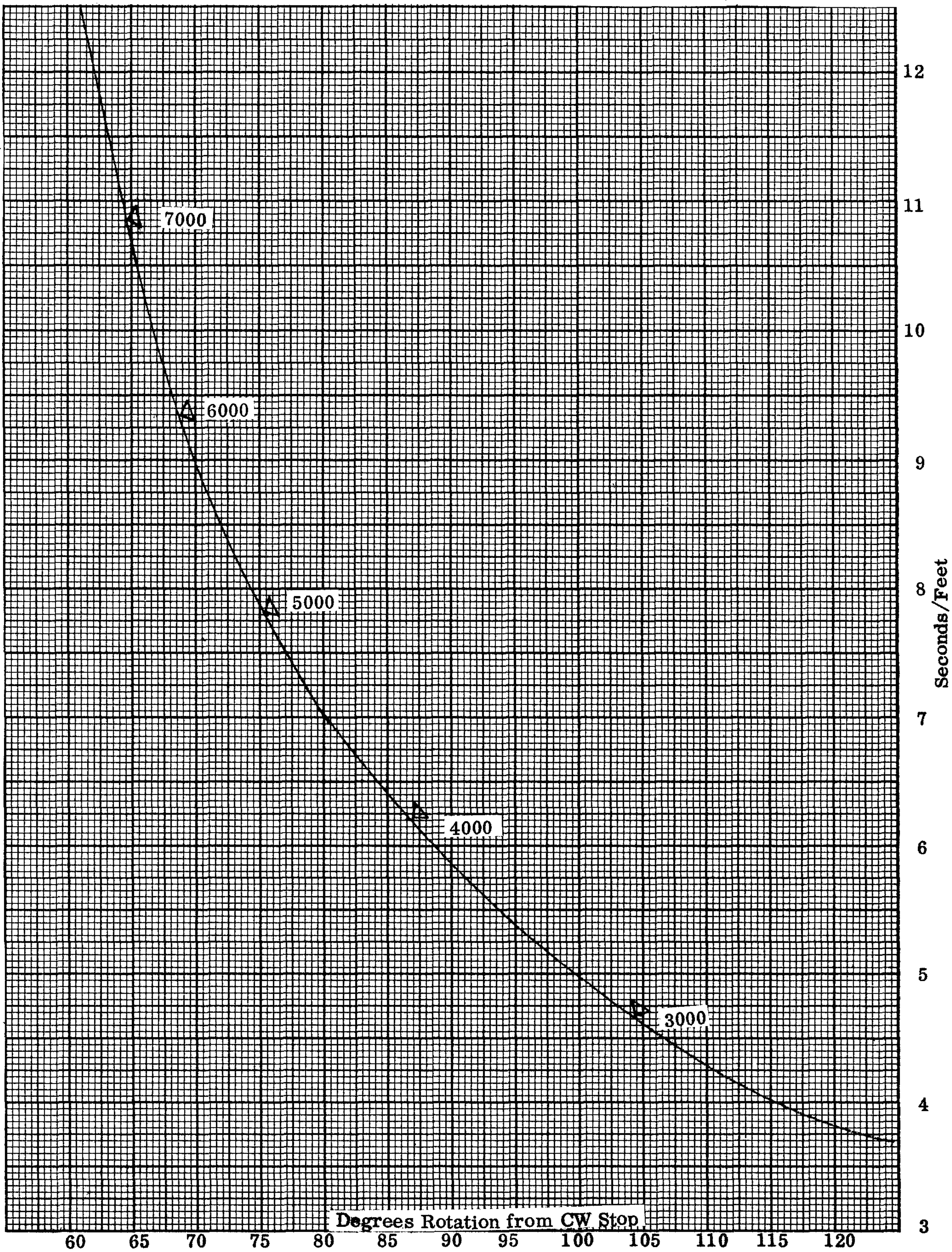
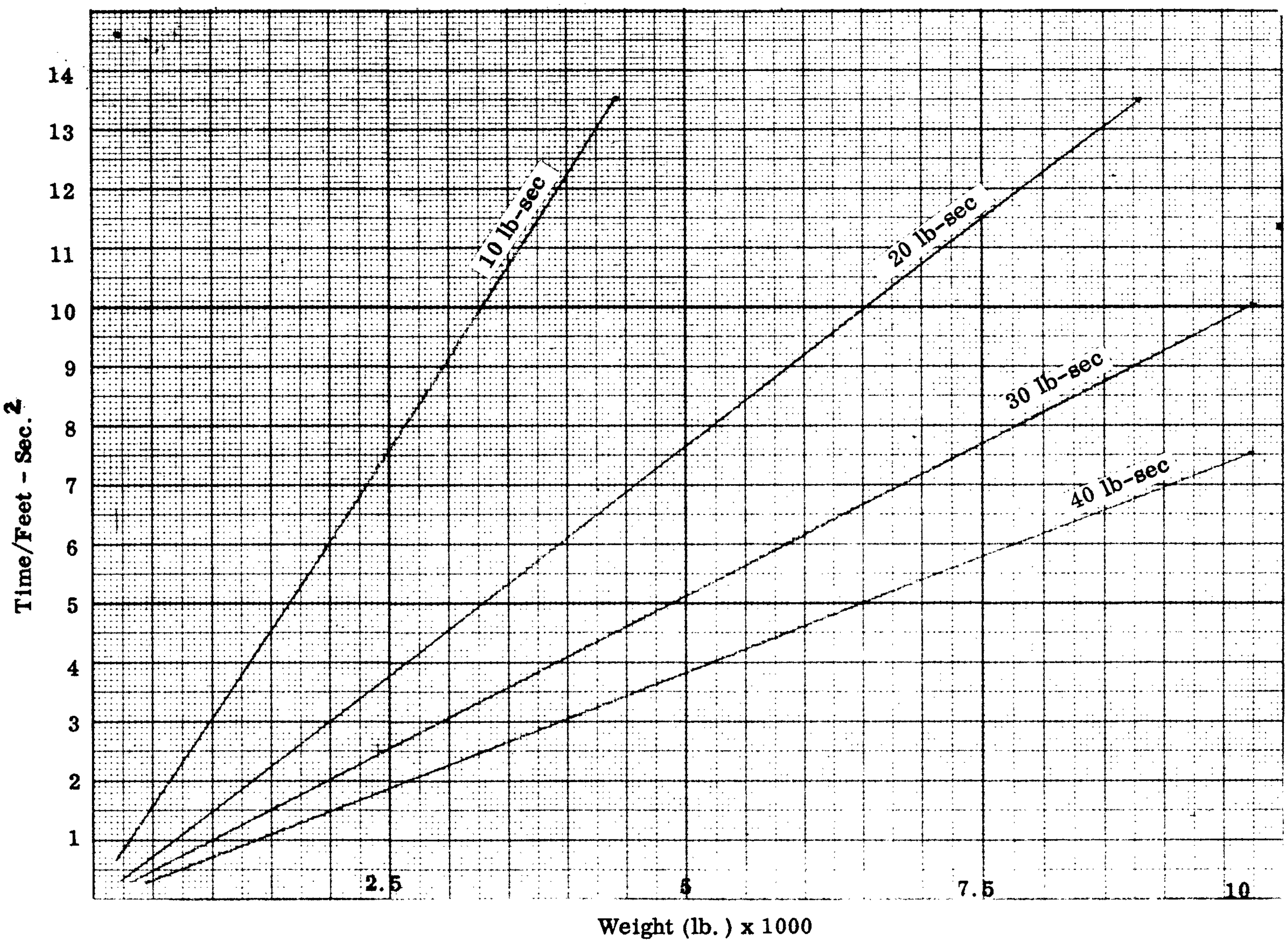


Figure 33. Channel B Scale Pot Calibration



$$FT = mv = \frac{w}{g} \times \frac{d}{t}$$

$$FT = \frac{W}{32.2t}$$

$$W = 32.2\ FT\ t$$

F = Force in.
T = F duration
W = Weight
d = 1 foot
t = time/ft.

Figure 34. Conversion: Time/ft vs Weight

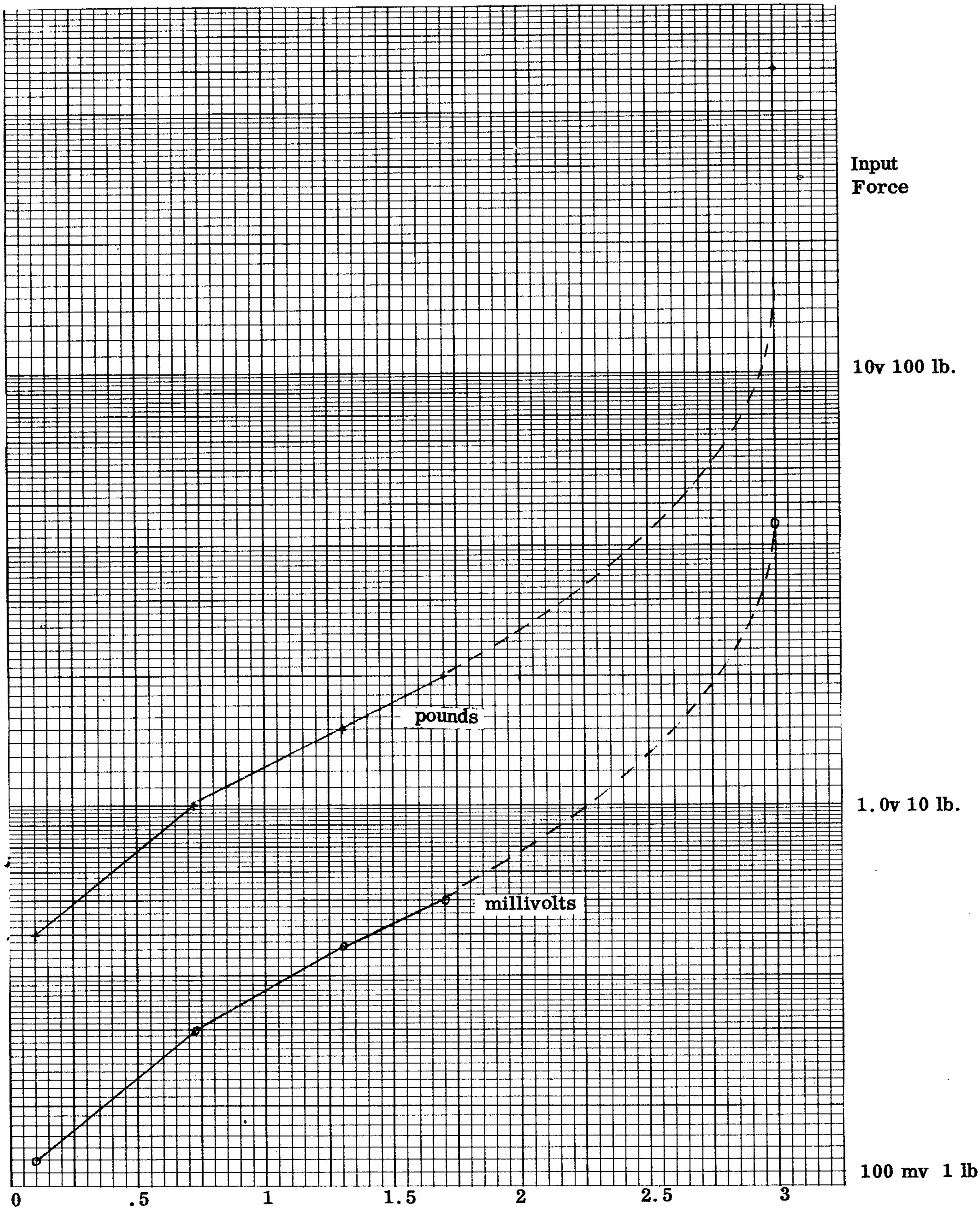


Figure 35. Integrated Rate vs Force Input

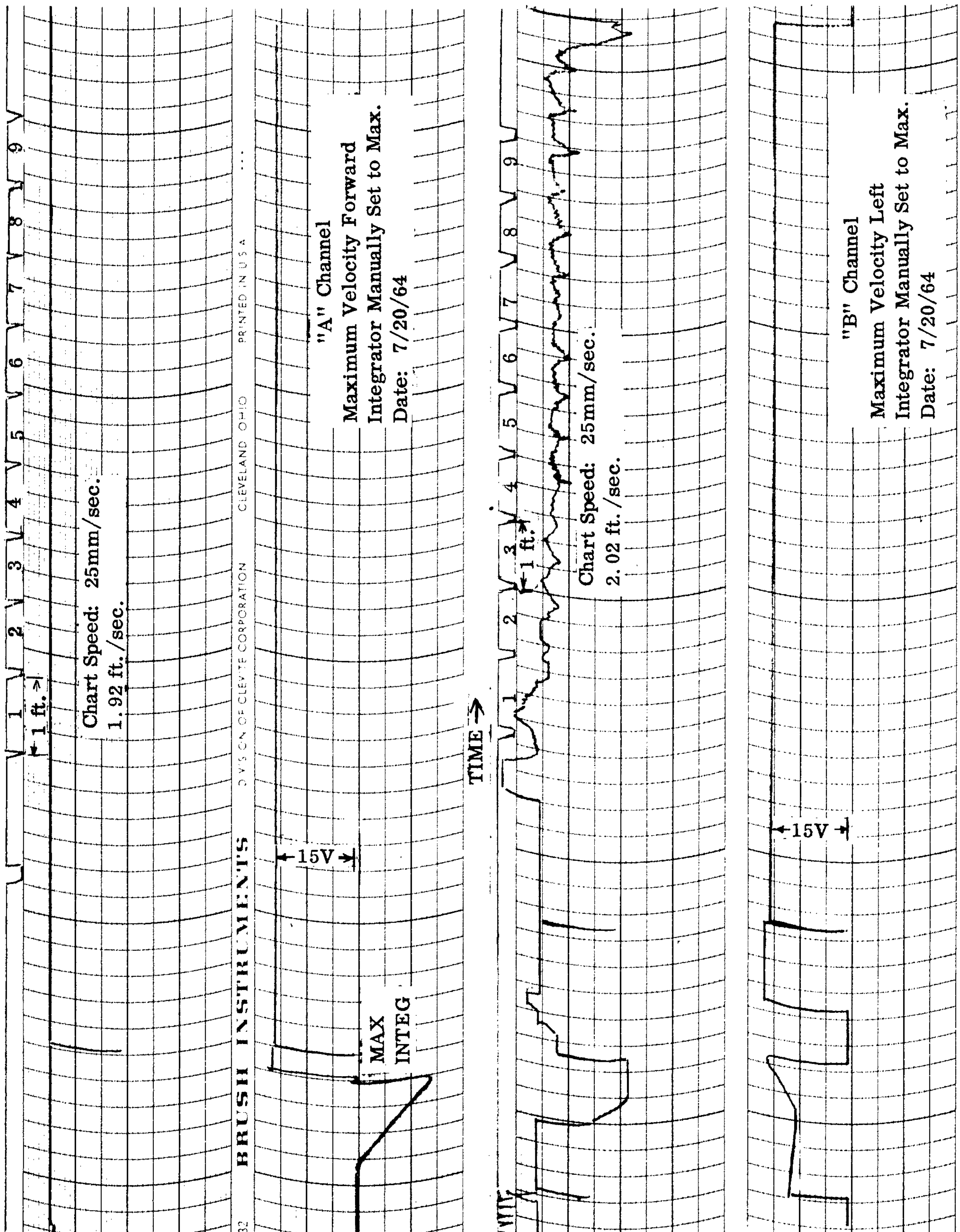
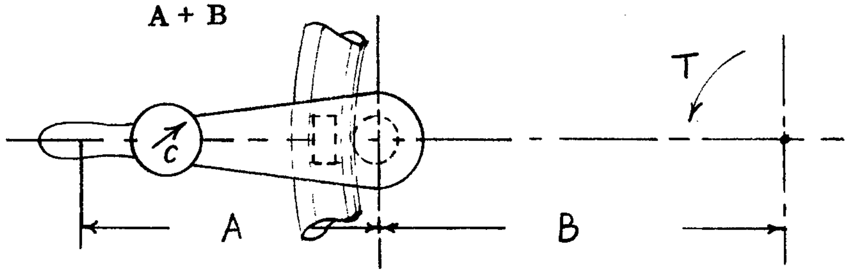


Figure 36. Maximum Velocities

Minor difficulties were encountered in making the fore and aft adjustment of the cradle and rework to enlarge the slotted adjusting holes (ref. Fig. 6) and lubricate the sliding parts was accomplished.

Tests of the forces required to cause rotation were made and average values of 5 in. lb. in pitch, 7 in. lbs. in roll and 20 in. lbs. on the inner bearing axis of the forward right hand set of yaw bearings were measured. The yaw measurement was made at a radius of 13.3 in. from the center. Using a standard torque wrench conversion formula, $C = \frac{AT}{A + B}$ the 20 in. lbs. is equivalent to 4.84 ft.-lbs.



around the yaw axis. Investigation into the problem was carried on in two directions. Discussions were held with Fafnir Bearing Co. on possible causes of increased friction and possible bearing substitutions and checks of quality records and measurement of deflections were made.

Fafnir had no bearings which could be substituted with any assurance of success. They did suggest that alignment of the bearings must be held to within 1.5 degrees or excessive loads will occur to increase the friction.

Quality records showed that the alignment of the bearing axis had been held to within the normal manufacturing tolerance of .5 degrees. Measurements with a dial indicator of the vertical deflection of the bearing axis under load were made. With a 1/2 in. thick machined plate resting firmly on the yaw ring track as a reference surface, the deflection of two points, one on each side of the bearing, was measured under no load and under 186 lb. load in the cradle. Only .001 in deflection was noted over a distance of 1.5 in. A 1 degree deflection would have given .025 in. At the same time yaw ring deflection was measured at a point 90 degrees from the pitch axis at the front. A load of approximately 85 lbs. was applied and a deflection of .017 in. noted. The reference surface was the same 1/2 thick plate. All bearings were removed and washed out and one drop of light oil was used as lubricant. The surface of the tracks was polished clean and checked against standards for an RMS 32 finish as recommended.

During final acceptance testing at WPAFB a 175 lb. subject with the back pack was tested in both clockwise and counterclockwise directions and an average of 24 measurements gave 3.9 foot lbs. to cause yaw movement. This was judged acceptable in that back pack forces will cause yaw motion and forces applied to the work panel will be at an 18 to 24 in. radius which means that the yaw axis would be sensitive to a 2 to 3 lb. side force applied to the work.

In all measurements of pitch, roll and yaw forces, the balance of the simulator with a live subject was highly critical. It was impossible for any subject to hold his position thru significant angular displacements and, consequently, the unbalance forces masked the frictional forces after a rotation of 5-10 degrees. The subjects also tend to actively control their attitude by slight changes in body position.

While measuring responses to impulses, the subject, even though requested to freeze, shifted position enough to stop rotation after about 10 degrees. It was possible, however, to demonstrate that the simulator will move under a 1 ft. lb. torque applied to the pitch and roll axis for 2 seconds. The impulse tool was used with the point of application on the lower front member of the cradle. The thrust axis of the tool was aligned perpendicular to the axis under test. The other axis was left free and the impulse was applied to a point 1 ft. below the centerline of the pitch bearings. A rotation of ab out 10 degrees was detected.

As an illustration of the balance criticality, the subject was requested to balance himself in an erect position by moving his arms. When he was erect, a torque wrench was placed on the pitch trunion and the subject moved one hand 3 inches up and 4 inches forward from its original position just touching the yaw ring. This motion created a 30 in. lb. pitch torque.

3. Translational Tests

With the simulator loaded, air was turned on to the base and the valves on the Hovair pads opened. The simulator was operated on a standard shop tile floor. Obstacles up to 1/16 in. were placed on the floor and the simulator moved over them. There was no detectable drag.

The impulse tool was set up to apply an impulse to the rear of the loaded simulator. A 1 lb. force applied for 2 sec. caused motion of approximately 6 inches. At that point the weight and stiffness of the temporary air hose stopped the simulator. The Hovair pads were purchased from General Motors Defense Research Division. Although these particular pads have not been extensively tested, the basic approach has been highly developed and design information is available at the source. Because of the developmental work done by General Motors no tests other than those described above were made by us. Appendix C shows the results of tests made by General Motors on the pads.

The vertical translation was tested initially with only the back pack in the simulator. A lifting force of approximately 35 psig was required. The initial tests of vertical friction showed extremely high values and extensive tests and analyses were made to determine the causes. As discussed in the design section, the force systems involved were more complex than originally anticipated and each item corrected a portion of the problem until the friction was reduced to the minimum of 1.5 lbs. shared between the flexible hose and the ball bearings.

Initially the misalignment of the air bearings was severe enough to measure with feeler gages under the corners and edges of the pads. As the friction was reduced this system of measurement was no longer subtle enough and it was necessary to utilize a continuity check between the air pad and the track to determine if the pads were riding free. Since there were 4 air bearing surfaces in operation (3 pads plus the piston) and all were suspected of contributing to the friction, this method was not highly fruitful. Greatest reliance was placed upon close visual inspection of the tracks for rubbing as an indication of the area where the frictional forces were being generated and visual inspection with a light shining under the pad and the line of sight parallel to the long axis of the track.

During testing in Baltimore, a severe high pitched squeal developed in the upper pads. The problem was assessed as an effective spring rate that was too low. The pads were removed and the faces machined to reduce the pad plenum depth from .009 to .003 in., thereby, reducing the volume of air serving as a spring under the pad. This was effective in removing the squeal. However, another squeal developed during the last stages of testing. The piston was vibrating at a low frequency about 2/3 of the way up in its travel. The remedy was discovered at WPAFB while attempting to damp out the squeal by weighting the piston rod to change its natural frequency. The origin of the squeal was a misalignment of the piston and cylinder from the vertical.

The measurement of the vertical translation friction was complicated by the sensitivity of the system to changes in flow. After the initial lifting force is developed the flow must be adjusted to establish a precise balance. With a pressure change of approximately .2 psi sufficient to raise or lower the elevating mechanism, the flow must be carefully controlled to avoid excessive drift. During operation of the simulator it was apparent that small adjustments with reasonable waits between adjustments will most quickly bring the system into a good balance. Initial fill rates are dependent upon source pressure but, in general, 5-10 minutes is required to build up full lifting pressure and about 5 minutes is required to bleed off the air to a point where the subject can be removed safely.

V. Summary of Simulator Characteristics

Simulator		
Length Overall		8 ft 6.5 in.
Width Overall		6 ft 0 in.
Height Overall		7 ft 8.6 in.
Nominal CG Height		5 ft 9 in.
Weight		325 lbs approx
Motion	Amount	Threshold force
Pitch	Unlimited	.42 ft lbs
Roll	Unlimited	.58 ft lbs
Yaw	Unlimited	3.9 ft lbs
Horizontal translation	20 x 30 ft.	$<$1.0 lbs
Vertical translation	36 in.	1.5 lbs
Subject		
Weight		186 lbs $\pm$ 20 lbs
Height		72.3 in. $\pm$ 3 in.
Operating air pressure		
Inlet		75-100 psig
Elevating mechanism		21-65 psig
Tank Capacity		140 SCF (1070 gallons)

Bridge Crane Work Panel	
Simulation Range 3K lbs. to 7K lbs. both channels	
Travel	
Longitudinal	19 ft
Transverse	11 ft
Threshold force	
Channel A longitudinal	2.5 lbs
Channel B transverse	2.0 lbs
Maximum velocity	
Channel A	1.92 ft /sec
Channel B	2.08 ft /sec
Crane size	
Width	20 ft
Length of rails	25 ft
Height of rails (top)	12 ft 10 in.

Work Panel	
Height	4 ft
Width	6 ft
Height of centerline from floor	6 ft 3.5 in.
Radius	60 in.
Power requirements	
110 VAC 60 CPS 15A	
28 VDC 5A	

VI. Conclusions

The simulator will perform the tasks for which it was intended in an adequate manner. Subject performance in the simulator during the test phases demonstrated the sensitivity of the simulator to the reaction forces produced in simulating maintenance tasks and to the thrust levels of the prototype self-maneuvering unit. Additional features, not originally requested, were included to improve operating ease and versatility.

VII. Recommendations

Development of automatic balance mechanisms has been demonstrated, by this as well as other programs, to be a necessary step to continuing development of simulators capable of being used to investigate man's capabilities to perform in a simulated orbital gravity. Basic study of the parameters affecting true weightless simulation in the areas of motor performance learning transfer from simulators to actual orbital situations should be studied to identify and define the critical and essential simulator characteristics.

Methods and techniques of achieving simulation with 6 degrees of freedom with less simulator mass should be developed to allow more accurate evaluation of orbital problems by the simulator.

Work panel simulation should be extended to 6 degrees from the present 2 for a closer evaluation of the interacting forces between subject and work object.

REFERENCES

1. Alexander, M., & McConville, S. T., Anthropometric Data in Three-Dimensional Form: Development and Fabrication of USAF Height-Weight Manikins, AMRL-TDR-63-55, Aerospace Medical Research Laboratories, Wright-Patterson AFB, Ohio, June 1963.

2. Santschi, W. R., DuBois, J. & Ornato, J., Moments of Inertia and Centers of Gravity of the Living Human Body, AMRL-TDR-63-36, Aerospace Medical Research Laboratories, Wright-Patterson AFB, Ohio, May 1963.

3. Griffin, J. B., Feasibility of a Self-Maneuvering Unit for Orbital Maintenance Workers, ASD-TDR-62-278, Aeronautical Systems Division, Wright-Patterson AFB, Ohio, August 1962.

4. Gross, W. A., Gas Film Lubrication, John Wiley & Sons.

5. Baker, W. E., McClernan, L. M., Thayer, W. S., & Fineman, H. E., Measurements of Human Abilities to Apply Forces While Tractionless, Institute of Environmental Sciences - 1963, Proceedings of, April 1963.

6. Holmes, A. E., Frictionless Simulator Final Report for NASA-MSC, Martin Company, Unpublished, December 1963.

Appendix A

Center of Gravity Calculations

95% man of 200.8 lb (ref 3)

Head	12.4
Torso	14.3
Upper Leg	14.3
Lower Leg	18.8
Ankles & Feet	3.1
Total Height	73.1

Subject: Norm $+1\sigma$ (ref 2)
Weight: $166.4 + 19.8 = 186.2$
Height: $69.4 + 2.9 = 72.3$

Body Segments	Length	CG Distance From Upper Hinge	Weight (ref 3)
Correction factor	$\frac{72.3}{73.1} = .989$		$\frac{186.2}{200.8} = .927$
Upper Arm	13.8	6.02	6.6
Corrected	13.65	5.95	6.05
Lower Arm	10.3	4.43	3.75
Corrected	10.19	4.38	3.44
Hand	7.0	3.99	1.35
Corrected	6.92	3.95	1.24
		Total	10.73

CG location Standing with arms down — Subject: Norm $+ 1\sigma$

$X_1 = 3.5 + .2 = 3.7$ From back plane

$Z_1 = 31.0 + 1.45 = 32.45$ From top of head

$$186.2X = (186.2-10.73)\,3.7+2(6.05)(8.65)+2(3.44)(20.6)+2(1.24)(30.45)$$

$$X = \frac{931.5}{186.2} = 5.00 \text{ in. from back plane}$$

$$186.2Z = \underbrace{(186.3X32.45)-\overbrace{2(6.05)(19.3)-2(3.44)(31.2)-2(1.24)(41.0)}^{\text{arms down}}+2(6.05)(18.2)+2(3.44)(23.6)+2(1.24)(23.6)}_{\text{arms up}}$$

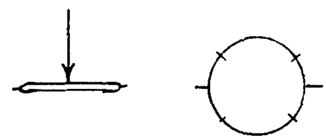
$$Z = \frac{5933.4}{186.2} = 31.87 \text{ in from top of head}$$

Appendix B

Yaw Ring Bearing Friction

Load 340 lbs applied in center of ring.

Condition I   Ring horizontal



4 bearing carrying 340 lb $\frac{340}{4}$ = 85 lb/bearing

friction/bearing at 85 lb load = .06 in.lb*, bearing radius = .5 in.

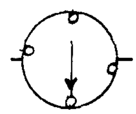
force/bearing $\frac{.06}{.5}$ = .12 lb

total force .12 x 4 = .48 lb

torque @ 15 in. R $\frac{.48 \times 15}{12}$ = .60 ft lb

. . . .

Condition II   Ring Vertical, cradle at 40^{o}

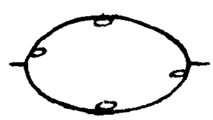


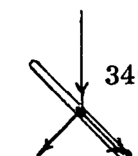
friction @ 340 lbs .25 in. lb *
force per bearing .25 = .5 lbs total .5 x 1 = .5

torque @ 13.7 in radius $\frac{.5 \times 13.7}{12}$ = .57 ft lb

. . . .

Condition III   Yaw Ring at 45^{o}, cradle at 40^{o}





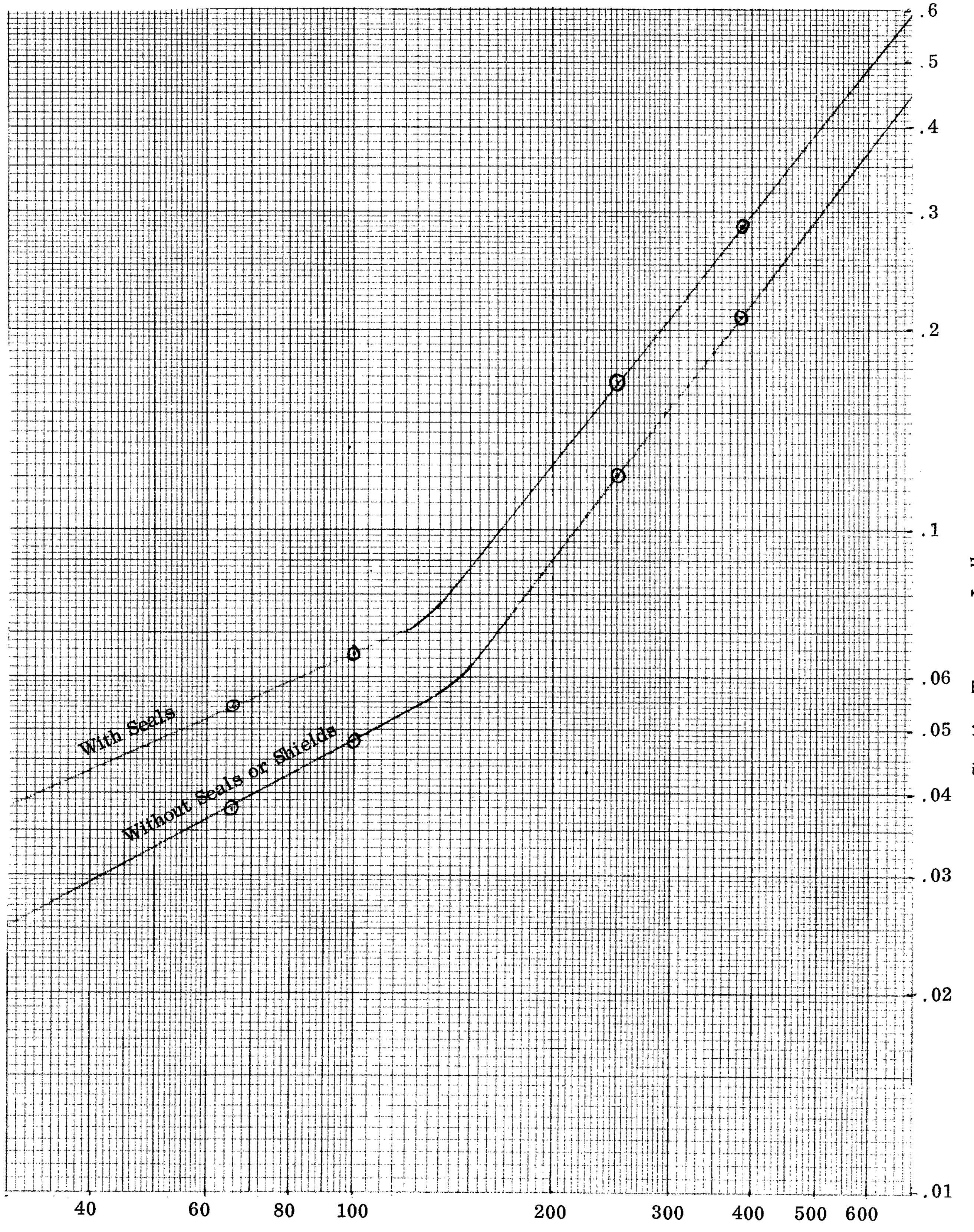
240 lbs normal to ring on 4 bearings
240 lbs parallel to ring on 1 bearing

<u>parallel</u>
friction at 240 lb = .16 in. lb *
force $\frac{.16}{.5}$ = .32 lb
torque @ 13.7R $\frac{.32 \times 13.7}{12}$ = .367

<u>normal</u>
friction at $\frac{240}{4}$ = 60 = .052*
force $\frac{.052}{.5}$ = .104
total force .104 x 4 = .416
torque @ 15R = $\frac{.416 \times 15}{12}$ = .519

* From chart of Average Starting Torque

total torque .367 + .519 = .886



Average Starting Torque - KP4R16-2
(Based on 7 sets of 16 bearings each)

APPENDIX C
Hovair Pad

SK06416 TEST REPORT

Test Set-Up No. 1

1. Surfaces: Granite surface table
   Shop floor (smooth sealed concrete)
   Aluminum sheet

2. Air manifold pressure control: 0-100 PSI regulator

3. Force measurement: 0-6 ounce-inch torque arm with 1 inch moment arm

4. Load on Hovair pad ass'ys. = 620#

5. Operating pressure in the Hovair pad ass'ys approximately 2.5 PSIG

6. Damping valve setting at 4 1/2 turns from closed position

Supply Regulator (PSIG)	Control Valve Setting (First Color Band)	Force to Sustain Motion in Two Opposing Directions (Ounces)
20	4.0	Under 3
35	3.0	" 3
50	2.5	" 3
70	2.0	" 3
85	1.25	" 3

Resulting coefficient of friction = .0003

Results were the same on all three surfaces described.

Control valve settings above those stated resulted in vertical oscillations when excited by a vertical force.

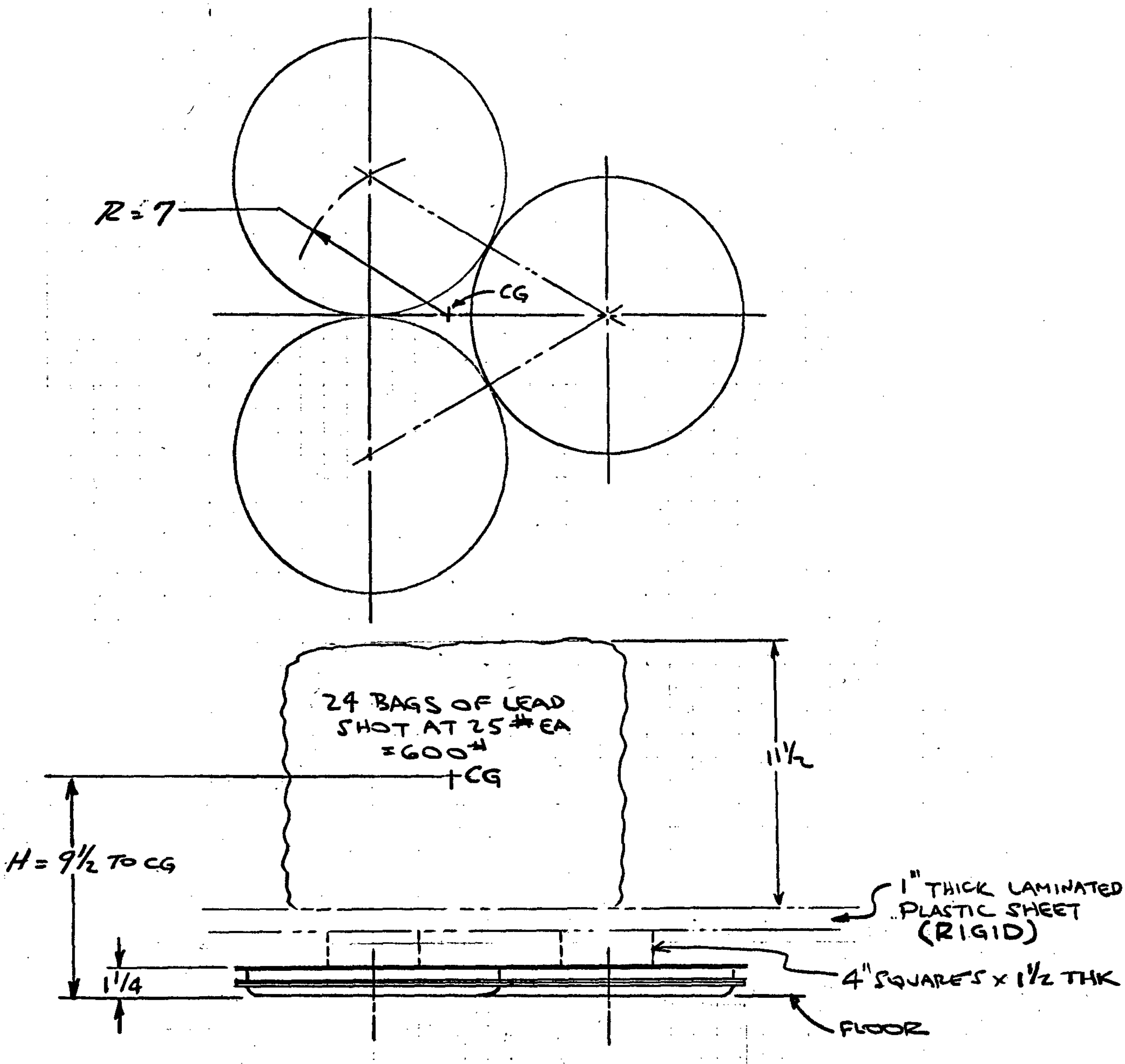
Test Set-Up No. 2

Surface: Shop floor (smooth sealed concrete)

Load and instrumentation identical to No. 1

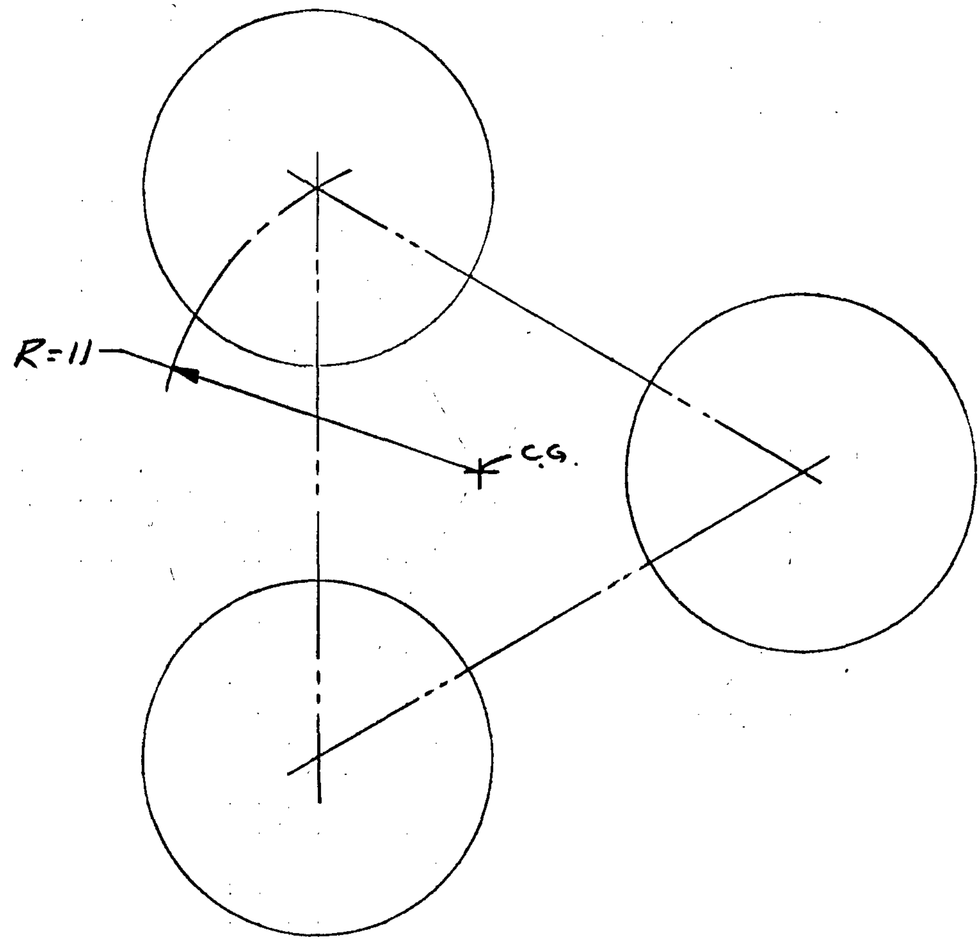
Color valve settings could be opened up to approximately the same values but on the second color band indicating lower H/R and non-rigid load mounting result in less sensitivity to vertical oscillations.

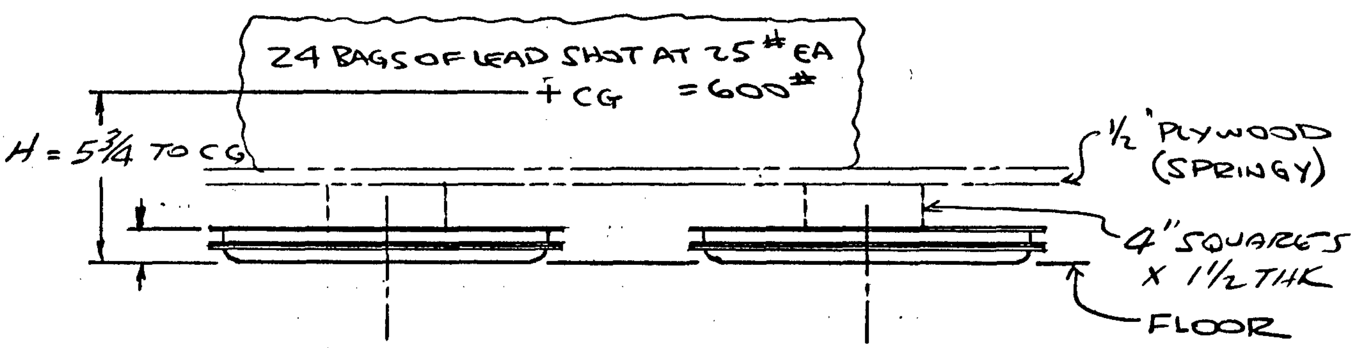
FRICTION COEFFICIENT TEST – TEST SET-UP NO. 1



$$H/R = \frac{9\frac{1}{2}}{7} = 1.36$$

FRICTION COEFFICIENT TEST - TEST SET-UP NO. 2





$$H/R = \frac{5\frac{3}{4}}{11} = .52$$

SK06416 OPERATING INSTRUCTIONS

1. Install SK06416 Hovair pad ass'ys to space simulator.

2. Connect flow control valve to regulated pressure manifold of space simulator with 3/8 inch polyflow tubing (or equivalent).

3. Close all flow control valves and damping chamber valves.

4. Apply operating load.

5. Set space simulator manifold to desired operating pressure.

6. Open all three flow control valves in increments, such that settings are equal, until either frictionless operation is obtained or vertical oscillations can be induced that do not dampen out.

7. If latter occurs first, open damping chamber valves until vertical oscillations are dampened out.

8. Repeat steps 6 and 7 until frictionless stable operation is obtained.

Note: Valve settings cannot be preset at GM Defense Research Laboratories since they are dependent on the nature and geometry of the load.