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A Four Degrees of Freedom, Cockpit Motion Machine for Flight Simulation

By L. H. WARTON
Aero/Flight Dept., R.A.E., Bedford

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Aero/Flight Dept., R.A.E., Farnborough

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Summary

This Report gives a detailed account of the development of a machine for reproducing aircraft motions on a flight simulator cockpit.

The four freedoms of the machine are actuated by hydraulic servos.

The equipment is part of the piloted flight simulator facility used for research and development studies in the Aerodynamics Department of the Royal Aircraft Establishment at Bedford.

The information presented concerns the engineering design and servo drive considerations involved.

A description is given of the control and safety procedures used when operating the machine within the simulator facility.

The limitations of the performance achieved are discussed both with a view to finding ways to improve the existing machine, and to point out those particular aspects of design which need careful attention and uncompromising implementation on any future project.

* Replaces R.A.E. Technical Report 72075—A.R.C. 34 056.

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1. Introduction

Flight simulation began in Aero Flight Division at R.A.E. Bedford in 1956 with the acquisition of the aiming flight simulator (AFSim) equipment developed by Armament Department at Farnborough.

Up to that time simulation work within R.A.E. had been carried out on static cockpits and the only 'flight simulator' available then, presenting the pilot with physical motion cues, was the Link Trainer. On this machine, however, the aircraft response characteristics were only approximated in the crudest way.

Within the Department, during the years 1956–1960, experience in tackling research simulation problems associated with aircraft stability, control and handling characteristics grew. In almost every case the simulator was substantially more difficult to control than the real aircraft modelled by it. This was thought to be mainly due to inadequate visual cues and the total lack of motion cues conveyed to the pilot. Over-controlling, poor sensitivity and general lack of realism contributed to an inability to reproduce representative flight behaviour and at times even caused reversed control responses and nausea in the pilot.

In 1960 an extensive modernisation programme was decided upon with the object of increasing the simulation capability and fidelity. A significant part of this programme was the provision of a motion machine capable of driving a single seat cockpit in heave (identified as 'pitch' at that time) and roll.

This machine was designed and constructed by Short Bros. & Harland Ltd. at Belfast and installed in the simulator at Bedford during 1961. It was used continuously for aircraft handling studies throughout the following eight years.

During this time a further major improvement in the scope of the aero flight simulator facility was made by the acquisition of a closed circuit television visual system and the addition of a three-axis motion machine driving a large-aircraft two-seat cockpit, in pitch, roll and heave. A description of the flight simulator currently in use in Aero Flight Division, Bedford, is given in Ref. 1 and an account of initial experience gained with the facility is given in Ref. 2.

As a result of experience gained from using the simulator to study the handling problems of a wide variety of aircraft ranging from vertical take-off to supersonic transport types, it was decided that amongst other improvements, additional motion freedoms were needed to enhance the validity of simulations to be carried out by Aerodynamics Department in the future.

The extension of the capability of the single-seat cockpit motion machine, by the addition of pitch and yaw freedom, was seen as a convenient way to investigate motion fidelity requirements for adequate simulation both in general and specific (VTOL) tasks. The possibility existed both of adding to the number of freedoms on the three-axis motion system and eventually of specifying a more advanced machine of up to five freedoms on the basis of the experience gained in this way.

The object of this Report is to describe in detail the development of the Short's cockpit motion machine, to summarise and assess its performance as a servo mechanism and, as a result of experience gained with it, list the features which need careful consideration from a servo engineering aspect when specifying and designing such machines. The Report does not consider programming techniques or recommend minimum motion capabilities for adequate simulation of specific flying tasks.

The Report is divided into seven main sections. A discussion of the principles of motion generation is followed by a general description of the machine, with detailed accounts of the specification, design considerations and performance of the four freedoms. Heave and roll are treated in one section as they comprise the basic Short's design. The original development work on heave and roll carried out at Belfast is described in detail by a Short Bros. Technical Report.[7]

Section 7 discusses general problems of control and safety involved in the use of a motion system in a flight simulator and describes the manner in which these problems were resolved for the Short's machine.

The final section summarises the experience gained in the development and use of the machine so far. Ways are suggested to improve its performance. Features requiring special attention when specifying and designing a motion machine are mentioned. The pitch and yaw motion mechanisms were designed and installed over the same period of time. Neither modification was able to benefit from experience gained on the other. This point is worth mentioning only in that it explains the overlap of similar problems discussed in the following text.

The text is supported by a number of illustrations and figures covering overall power requirements, system layout and performance. A general summary of the current performance and loading of the machine is given in Table 1.

2. Principles of Cockpit-Motion Generation for Flight Simulators

The object of imparting motion to the cockpit of a ground-based flight simulator is to provide the pilot with motion cues equivalent to those which he would receive in real flight. In full scale flight the aircraft responds in six degrees of freedom over an almost unlimited range of linear and angular motion.

Owing to the physical and economic limitations necessarily imposed when building a machine for this purpose, it will be more or less restricted in its degrees of freedom, in frequency response, and in the maximum accelerations, rates, and displacements it can achieve. The machine will only be able to fulfil its objective within these limitations.

In order to make the most effective use of the limited motion capability available from a particular machine it is necessary to have a thorough understanding of the significance of motion cues in the pilot's control task, and experience in the drive techniques found to be most successful within the machine's limitations.

It has become increasingly evident in simulator studies of aircraft handling problems that the physical sensations experienced by the pilot as a result of aircraft motion are of considerable importance in defining his control task accurately. However, very little work has been done to investigate this aspect of control.

When the engineer is required to provide systems for the automatic, or assisted control of aircraft, such as auto-stabilisers, flight directors and autopilots, he immediately finds it necessary to provide phase-advanced information obtained from sensitive rate gyros and accelerometers to the control servos.

This experience should serve to illustrate, in a general way, the significance of motion cues in the control of the aircraft. In manual flight the pilot is acting both as the control servo drive and as the 'sensing' instrumentation.

For simulation purposes a real pilot is available, but motion, although real, is only available within restricted channels and is subject to degradation both in response and sensitivity.

Some insight into the quality of motion generation required for simulation purposes may be obtained from these considerations.

The limited range and performance of the simulator motion machinery places a restriction on the level and duration of the accelerations which can be applied to a simulator cockpit. This leads directly to the adoption of a drive technique in which 'initial' accelerations are applied, usually at a reduced scaling, and the resultant rate and position of the cockpit is 'washed out', i.e. the cockpit tends to return to its neutral position at rate and acceleration levels low enough to be unnoticed or ignored by the pilot. The technique makes it possible to generate high acceleration cues without frequently driving to the motion limits of the machine.

With this type of drive the resultant cockpit motion does not in general, correspond to that of the simulated aircraft with respect to the outside world. This implies that visual information systems (outside world displays) are best carried integrally with the cockpit on the motion machine. These systems may then be driven by computed signals defining aircraft position relative to the outside world. Visual displays which cannot be carried on the cockpit would need to have appropriate correction terms added to their drive signals. Such a technique requires the synchronisation of a variety of servos, each one having its own response characteristics.

At the time when the Short's machine was being specified, it was realised that its primary task was the application of accelerations to the pilot. However, position servos, accurate enough to provide smooth position-following over a specified frequency range, were regarded as adequate for the motion drives. The accelerations associated with the position-following response would produce the 'motion cues' to which the pilot would react.

As a result of this simple concept, no great emphasis was laid on what we now define as desired 'quality' or smoothness of acceleration response, when the original specification for the machine was written.

During the subsequent use and development of the machine it was realised that the ability to reproduce accurately controlled acceleration and rate of change of acceleration is one of the most important aspects of cockpit motion for aircraft simulation, if the pilot is to be able to register the required motion cues with sufficient sensitivity.

Pilots will only accept simulated aircraft motion if it gives a really credible imitation of the essentially smooth sensations so characteristic of flight. It is, however, occasionally possible to do some useful work with a machine lacking this smoothness, if the simulation is restricted to rough air flying. If gusts of sufficient severity are programmed into the computation they may swamp the inadvertent jerks and hesitations associated with poor motion drive mechanisms, and allow some usable motion cues to be sensed in addition to providing a simulated rough air environment.

The use of 'rate of change of acceleration' as a criterion on which to base and assess motion system performance is supported by the realisation that pilots react to sensed changing acceleration rather than to absolute acceleration levels.

A full appreciation of the problem of analysing human response to both rotational and linear motion as experienced in flight involves a knowledge of the functions of the vestibular organs of the ear.[3,4] It is reasonable to assume, however, that pilots sense and utilise rates of change of acceleration in performing the flight control task.

Experience has shown that a simple position servo-mechanism, built to relatively high quality general engineering standards, is not likely to achieve the quality of cockpit motion needed for adequate flight simulation of aircraft handling problems.

Attempts have been made,[5] with some success, to improve the acceleration-following response of motion systems by the addition of rate and acceleration feedback loops, together with appropriate addition of computed rate and acceleration terms to the drive signal.

Considerable work remains to be done to extend our understanding of the role of motion cues in flight handling and the technique of simulator motion programming, and to perfect servo drives capable of the required motion quality. Ref. 6 gives a useful review of the position.

3. General Description of the Four Degrees of Freedom Motion Machine

The motion machine described in this Report was originally designed and manufactured by Short Bros. to an Aero Flight specification in 1961. In its original form it provided roll and heave freedom only.

The machine now carries a general purpose single-seat cockpit, together with television projection, television monitor and shadowgraph display equipment.

A general view of the assembly is given in photographs in Figs. 1 and 2. The principal dimensions of the machine and the arrangement of the four movement axes are illustrated by Figs. 3 and 4 respectively.

The yaw motion system was designed more recently by Cranfield Institute of Technology to an R.A.E. specification. Manufacture and installation was carried out by R.A.E. workshops. Design, manufacture and installation of the pitch motion drive was carried out completely within R.A.E.

As shown in Fig. 4 the cockpit pitches about an axis in the heave beam aft of the pilot's shoulders and parallel to the cockpit floor. The pitch drive jack (Fig. 5) is carried centrally under the heave beam and moves the cockpit through an attachment bearing mounted on the centre line of the cockpit back structure at a torque arm of 203 mm. No pitch balance ram is provided, the dead weight being taken entirely by the servo jack.

The heave beam pivots about an axis in a cruciform casting, the axis being parallel to and 1·38 m aft of the pitch axis. The heave drive jack (Fig. 6) is carried along the top centre line of the cruciform casting, and drives through a crosshead and connecting rod attached to a bearing lug rivetted to the top of the heave beam. The torque arm of the heave jack is 165 mm. Dead weight of the cockpit and heave beam is taken by a vertical balance ram acting between gymbal bearings on the main support structure and on the heave beam.

Suitably programmed signals to the pitch and heave freedoms can provide pitch rotation and heave translation cues in aircraft body axes to the pilot.

The cruciform casting is free to roll about a fore and aft axis parallel to the laboratory floor and carried in bearings in the main pyramid structure of the machine. The roll drive jack (Fig. 7) is mounted vertically on the left hand side of the support frame and drives through a crosshead and connecting rod attached to a bearing lug built into the cruciform casting at a torque arm of 165 mm.

The precise nature of the cues provided by the roll freedom is affected by the position of the heave and pitch drives. Consequently it does not always generate pure roll sensations as compared with the heave and pitch system, which provide relatively pure heave and pitch motion cues. In practice, however, this feature of the roll motion has not been an embarrassment. It is neutralised to some extent by the use of 'washed out' drive to the heave servo, which tends to minimise the average displacement of the cockpit from the roll axis.

The main supporting structure is carried on a 1·53 m diameter ball race sandwiched between a steel floor plate and a steel base plate. The axis of the bearing lies on the centre line of the machine 241 mm forward of the heave axis and approximately under the centre of mass of the whole machine.

The yaw drive jack (Fig. 8) is anchored to the floor plate and drives tangentially on to the base plate. The yawing arm between the pilot seat centre and the yaw axis is 2 m, with heave and pitch at zero. Yawing motion sensed by the pilot contains sideways translation and other elements depending on the position of roll, pitch and heave.

The hydraulic circuit arrangements for the complete machine are given in Fig. 9.

The source of hydraulic power for pitch, roll and heave is a 20·7 MN/m^2 (3000 lb/in^2), 23 litre/min (5 gal/min) aircraft pump of the two-stage piston self-regulating type backed up by a 4·5 litre (1 gal) accumulator and driven by an electric motor.

The source of hydraulic power for the yaw motion is a 43 litre/min (9·5 gal/min) piston pump driven by an 18·7 kW motor and regulated by a pilot-operated pressure relief valve to give a load cycle of 17·25–15.9 MN/m^2 (2300–2500 lb/in^2).

Solenoid operated spool valves controlled by pressure switches monitoring the pressure in a parallel pair of 18 litre (4 gal) back-up accumulators maintain the load cycle sequence for the yaw hydraulic supply.

Operation of the machine is protected by a comprehensive electro-hydraulic safety circuit described in Section 7 and illustrated by Fig. 10.

4. Heave and Roll Freedom

4.1. Requirement

The original intent in specifying a motion system for use in the Aero Flight simulator was to provide the primary longitudinal and lateral cues of heave, pitch rotation and roll rotation.

The layout of the machine was defined by treating the longitudinal cues as predominant and therefore referred to body axes in a way typical of conventional aircraft where the pilot is normally at some distance forward of the pitch axis.

The roll freedom, added on a 'layer' principle was effectively referred to a 'wind' or 'flight-path' axis.

The geometry of the longitudinal freedom gave a fixed combination of 'normal' and rotational cues with the possibility of separating these at a later date by the addition of a separate pitch drive, located at the end of the heave arm.

4.2. Design Specification

The motion system was intended to drive a cockpit with a mass of 275 kg whose moments of inertia in pitch and roll about its CG were 154 kg m^2 and 103 kg m^2 respectively. The CG of the cockpit was to be 2·18 m forward and 0·4 m below the motion heave axis in its neutral or level position.

The dynamic performance was to be as follows:

	Heave freedom		Roll freedom
	Pitch rotation	Vertical translation	
Operational range	$+20°$ $-10°$	0·69 m 0·35 m	$+15°$ $-15°$
Maximum rate	$\pm 40°/s$	$\pm 1{\cdot}39$ m/s	$\pm 30°/s$
Maximum acceleration	$\pm 100°/s^2$	$\pm 3{\cdot}54$ m/s^2	$\pm 100°/s^2$

This specification intended the following performance to be achieved:

Heave: step inputs of 'normal' acceleration up to $\frac{1}{3}$ g and sinusoidal inputs of 'normal' acceleration up to $\frac{1}{3}$ g, limited by frequency and amplitude as shown in Fig. 11.

Constant 'normal' accelerations of between $\frac{1}{3}$ g and $\frac{1}{6}$ g over excursions of up to 10 degrees as shown in Fig. 11.

Heave and roll: sinusoidal rotations as follows:

15 degrees amplitude up to 0·4 Hz (heave) 0·3 Hz (roll)
10 degrees amplitude up to 0·5 Hz
5 degrees amplitude up to 0·7 Hz.
2 degrees amplitude up to 1·0 Hz

The phase lag in response to sinusoidal inputs of position demand was not to exceed 15 degrees at 1 Hz for 2 degrees amplitude in both heave and roll (*see* Appendix I).

Motion quality was to be such that slowly varying demands could be followed without perceptible roughness, 'stiction' or vibration.

The final performance of the heave and roll freedom did not meet the design specification on a number of counts as follows:

(1) Heave maximum rate was limited at 23 deg/s thus limiting the sinusoidal normal acceleration and position response.

(2) The phase lags in heave and roll were 33 and 29 degrees at 1 Hz respectively.
(3) The motion quality was poor, particularly in roll.

The reasons for this are discussed in Section 4.4.

4.3. Servo Drives

The basic designs of the heave and roll freedom servos were similar, using symmetrical area short stroke jacks driving the load through crosshead slides and linkages at a torque arm of 165 mm in each case.

The substantial dead weight of the heave freedom was taken by a balance ram, leaving the servo jack to drive against inertia, friction and damping forces only.

The relatively small symmetrical out of balance loads in the roll motion were taken by the roll servo jack itself.

The power requirements to drive inertia loads to the design specification are small and the use of short stroke jacks and high system pressure 20·7 MN/m^2 (3000 lb/in^2) enabled flow rating of the servo valves to be kept down to a level covered by single stage spool valve designs. 'Power margin' curves in terms of simultaneous rate and acceleration performance for heave and roll servo drives are shown in Figs. 12 and 13. The specified design performance in each case is also plotted. The heave drive could have been well matched to the load if a slightly bigger valve had been used and had it not been necessary to throttle the balance ram as discussed later. In the case of roll, the drive is substantially overpowered and would possibly benefit from a reduction in system pressure.

It was not appreciated at the design stage how high the stiffness of structure and hydraulic drive would need to be to meet the frequency response specification of 15 degrees phase lag at 1 Hz, i.e. sufficient to give a natural frequency of 10 Hz.

The design, based mainly on strength rather than stiffness, turned out to have natural frequencies of approximately 3 Hz and 5 Hz in heave and roll respectively, with the resilience shared equally between the hydraulics and the structure.

The heave and roll jacks are controlled by an R.A.E. design of single stage hydraulic servo valve, fitted with shaped spools to cancel the effects of flow forces in the valves which would otherwise restrict their maximum flow performance. These valves are electrically driven in push–pull by differential valve amplifiers utilising HT voltages of ± 300 volts. Rotary position feedback potentiometers were originally provided, mounted somewhat remotely from the servo jacks. In the case of heave a rotary torque 'motor', mounted remotely from the heave jack, was used to give rate feedback. In roll an inductive, linear, rate pick-up mounted directly across the roll jack was used to give roll rate feedback. Fig. 14 illustrates the circuit arrangements for heave and roll, which were similar in each case.

4.4. Development and Performance

The low natural frequencies of the design made it difficult to achieve the phase lag specification without introducing closed loop instability. When the gains were reduced to avoid instability the phase lags increased, and the dead zones or motion thresholds were too high.

Some attempts were made to stiffen the structure but any significant increase would have meant a complete re-design.

It was decided during the development trials therefore to settle for a reduced frequency response and attempt to improve the motion quality.

To this end, the servo damping was increased by introducing bleed orifices across the jacks which also tended to dissipate shock pressures and made for smoother motion reversals.

In heave however, it was considered necessary to apply substantial additional damping by throttling the flow of oil to and from the balance ram. The effect of this was to produce substantial damping forces, increasing the load which the servo drive had to overcome with the result that the maximum achievable rate was reduced to 23 deg/s.

The motion machine was finally accepted for use in the Aero Flight simulator with only 50 per cent of the maximum specified heave rate available and with a poor frequency response, improved only at the expense of motion quality by increasing the servo loop gains.

Motion quality was poor at both low and higher gain settings. At high gains the feel tended to be 'hard' with an undercurrent of high frequency oscillation or vibration and with motion-reversal thumps sharply defined. At low gains the general feel of the motion was smooth but the reversal thumps were still obtrusive, less sharp but accentuated by additional dwell caused by the increased threshold level.

During the initial period of use at R.A.E. some improvement in motion quality was achieved by mounting better quality position (double track) and velocity transducers directly across the servo jacks rather than at relatively remote positions on the movement structure. This enabled the higher gains to be used more effectively. Low friction PTFE seals were introduced into both jacks and the balance ram. The results of tests carried out to measure the seal friction under pressure of the pitch jack with nitrile rubber and then PTFE seals, are given in Fig. 15.

Reversal thumps were still excessive particularly in roll. Main bearings and jack rod end bearings were inspected for slack but only normal running fits were evident.

Attempts to eliminate valve threshold by superimposing restricted a.c. inputs of between 10–400 Hz made no significant difference until the amplitude of the a.c. was large enough actually to cause visible jack movement. The discontinuity in the acceleration at reversal then largely disappeared but was replaced by a continuous resonance in the structure at the exciting frequency which was quite intolerable.

Measurements of starting friction in the heave and roll freedoms, inclusive of jack seal friction, gave figures of 271 N m (200 lb ft) and 244 N m (180 lb ft) respectively. Roll friction with the jack detached measured 136 N m (100 lb ft). If these levels of friction were to appear as break out forces they would generate acceleration jerks of 7 deg/s^2 in heave and 51·6 deg/s^2 in roll, i.e., 7 per cent of the full heave acceleration capacity and 50 per cent of the full roll acceleration capacity could be generated as spurious acceleration at reversal or initiation of motion. These figures demonstrate the accentuated sensitivity of the roll freedom to acceleration 'noise' due to the low inertia in this freedom. Recordings illustrating typical motion quality are shown in Figs. 16, 17 and 18. The heave acceleration trace shows lack of symmetry due to the action of the balance ram. Full compensation of the dead weight of the cockpit is usually initially set at the zero heave position, but tends to drift with gas/oil temperature during operation. Compensation is reduced as the ram extends (heave up), and increases as the ram contracts (heave down). If the 100 per cent compensation position happens to coincide with the lower motion reversal, then the cockpit will momentarily come to rest in balance, while the drive servo crosses friction, backlash, and valve dead space thresholds. The result is that the acceleration drops to zero for this interval. The upper motion reversal is assisted by a fraction of unbalanced dead weight and the acceleration tends to peak at this point.

Attempts so far to improve the frequency response or the motion quality by the application of acceleration feedbacks have failed. Acceleration feedback round the jack leaves backlash, bearing friction and structural resonance outside the loop. Acceleration feedback from the cockpit creates instabilities from cross coupling between the various freedoms of motion and structural resonances other than the fundamental one and is upset by discontinuity resulting from normal engineering clearances in the moving parts.

The present performance of the heave and roll drive is shown in Figs. 19 and 20, illustrating the limits of undistorted output in response to simple harmonic demands and by Figs. 21 and 22 giving the frequency response as position-following servos in terms of amplitude ratio and phase lag for heave and roll respectively. Relevant information is also given in Table 1 'Current Performance and Loadings'.

A detailed account of the initial development of the machine by Short Bros. is given in Ref. 7.

4.5. Modification to Heave Balance Ram

The addition of television display equipment and the pitch motion modification requiring increased cockpit structural stiffness, increased the loads in the heave and roll drives by 50 per cent. This necessitated the replacement of the balance ram by one of increased capacity. The increased loading had negligible effect on the frequency response characteristics but increased the stress on the machine in general, particularly in runaway cases. It is largely because of the latter effect that no attempt has been made to increase the maximum rate of heave to the design specification figure. The performance of the new ram is defined by the figures given in Table 1.

5. Pitch Freedom

5.1. Requirement

Although the original design specification of the Short's cockpit foresaw the requirement for pitch freedom distinct from 'heave' movement, it was not added until 1969.

In anticipation of the need to perform simulations for vertical take-off aircraft a suitable shadowgraph display was proposed to be used in conjunction with the Short's motion system and cockpit. This display machine, too large to be carried on the motion system, would in fact prohibit the use of the heave or yaw freedoms. Since it was thought essential that some longitudinal motion cues be available for VTOL simulations, this strengthened the case for the development of a pitch motion drive with a negligible heave component.

The use of pitch in conjunction with heave and the other freedoms was regarded as a secondary benefit, if it could be made to work in spite of the known lack of structural stiffness of the machine. If it could not be used successfully, then it was to be locked or replaced by a static strut.

Since its installation, pitch, in combination with roll and heave, has been continuously used in general aircraft simulations.

The use of pitch prohibits the use of the available television projector and display screen because it introduces relative movement between the two. A television monitor display is used instead, mounted on the cockpit nose in front of the pilot and above the flight instrument panel (*see* photographs Figs. 2 and 3).

5.2. Pitch Design Specification

The following performance was thought to be adequate for the pitch drive:

	operational range	$\pm 15°$,
	overrun	$\pm 5°$,
	total mechanical travel	$\pm 20°$,
	maximum rate	$\pm 50°/s$,
and	maximum acceleration	$\pm 100°/s^2$.

This specification is designed to ensure unlimited response to sinusoidal demands of amplitude,

2·5° up to 1 Hz,
5° up to 0·7 Hz,
and 15° up to 0·4 Hz.

The frequency response was to be such as to give not more than 15 degrees phase lag at 1 Hz (*see* Appendix I). Smooth motion response was called for but no limit of signal to noise ratio was specified.

5.3. Pitch Freedom—Servo Drive

The power requirements to drive the cockpit with the specified response are relatively low and it was decided to use a small 20·7 MN/m^2 (3000 lb/in^2) hydraulic servo ram which could be run from the hydraulic system supplying heave and roll. The cockpit hinge pivot attachment points had to be redesigned to allow the required ± 20 degrees of total pitch freedom and to give sufficient strength to tolerate the anticipated yaw drive loads. The heave beam and the back of the cockpit had to be stiffened locally to provide attachment points for the drive jack. To avoid running the cockpit into the laboratory floor at full mechanical travel in heave, pitch and roll, it was necessary to reduce the overrun of the heave drive from the original values of 10 degrees down to 3 degrees down. This was done by restricting the heave jack stroke by increasing the length of an internal sleeve in the end of the cylinder bore. The operational travel of heave down was not altered.

Because of the small size of the jack required it was possible to provide sufficient thrust in it to take the dead weight loading as well as the dynamic loading. The nearest standard size commercially available jack was selected. The estimated hydraulic stiffness in conjunction with the cockpit pitch inertia of 440 kg m^2 defined a natural frequency of 7–10 Hz for the drive. A single spring-centred two-stage servo valve (MOOG21) of 330 ml/s flow capacity at 7·0 MN/m^2 (1000lb/in^2) was chosen to drive the system. The 'power margin' of the system is illustrated in terms of combined rate and acceleration capacity in Fig. 23.

The curves for estimated valve/jack performance are based on a $Q \propto \sqrt{P_v}$ relationship between valve flow and pressure drop across the valve. This is only true for the valves used up to a specific value of pressure drop, after which the flow through the valve saturates. This saturation level is shown in Fig. 23. The extreme asymmetry of the drive between 'pitch up' and 'pitch down' performance is evident in spite of the use of a differential area jack. The effect of using a balance ram to fully compensate for the dead weight is also shown. Asymmetry is still present but in a reversed sense. 83 per cent balancing of the dead weight would make the pitch up and pitch down curves coincident. A substantial power margin over that required to drive the load would still exist and since the stiffness and therefore the jack size must not be reduced, a better power match could be achieved by reducing the system pressure, probably by as much as 67 per cent, i.e. to 6·9 MN/m^2 (1000 lb/in^2).

An integrated circuit d.c. operational amplifier working through a single power output stage, driving single ended into the servo valve is used (Fig. 24) to drive the servo-jack. D.C. position (double track) and velocity transducers are mounted directly across the jack, serving as the feedback elements in the control loop. Initially these transducers were driven from a bracket on the back of the cockpit structure several inches away from the jack attachment bearing. During initial testing a transverse structural resonance about the jack attachment destabilised the servo loop. Repositioning the transducers avoided the problem.

5.4. Pitch Freedom—Development and Performance

The pitch motion when driven as a simple position loop had a substantial resonance at about 3·7–3·8 Hz. The addition of simple negative velocity feedback was ineffective in damping the drive at this frequency.

Provision of 'acceleration' feedback up to 4 Hz by differentiating the output of the velocity transducer and cutting off the higher frequencies with a filter, substantially improved the motion quality to the extent that it became acceptable for use in simulation exercises.

The structural natural frequency of the cockpit in general was so low (4·2 Hz) that further attempts to remove the residual resonances failed and only a low loop gain could be used.

A block diagram of the final control loop is given in Fig. 24.

Motion quality, with and without acceleration feedback, may be compared in Figs. 25 and 26. As with the heave and roll motion, a discontinuity is evident at the peaks of the acceleration trace, particularly pronounced in the reversal from 'pitch up'. This is no doubt due to the effect of dead weight which demands the valve to be slightly open in the 'pitch up' sense whenever the cockpit is stationary. 'Pitch up' motion and reversal from 'pitch down' tends to be smoother therefore because it is less likely to encroach upon or cross the 'valve closed' condition. Pitch down, particularly at relatively slow rate, constantly requires the servo valve to cross its centre. This behaviour repeatedly excites the poorly damped resonant frequency of the drive as illustrated by the acceleration trace of Fig. 25.

Tests of the motion system with both heave and pitch freedom energised demonstrated some significant reactions from one servo to the other, due to the low stiffnesses in both drives. These were not large enough to preclude the use of both freedoms together, but coordination of the pitch and heave drives to produce pure 'normal' acceleration had to be restricted to open loop programming of the two input signals and could not utilise position feedback compensation from one drive to the other. Use of the latter system almost certainly would have produced violent and divergent oscillations involving the two freedoms, as well as introducing additional phase lag.

The present performance of the pitch freedom is defined by Fig. 27, illustrating the limits of undistorted output in response to simple harmonic inputs and by Fig. 28, giving the frequency response as a position-following servo in terms of amplitude ratio and phase lag.

The maximum rates achieved by the pitch drive are substantially lower than those estimated and in the case of 'pitch up', 25 per cent lower than the specified requirement.

This is due to a combination of pressure drop down the servo line, low initial system pressure and low usable servo gain.

Acceleration limits have not been experimentally measured because of the hazard to the cockpit assembly involved.

Additional performance and loading information is given in Table 1.

6. Yaw Freedom

6.1. Requirement

The lack of yaw or transverse acceleration cues has always been a source of criticism in simulation exercises performed with the Aero-Flight Simulator motion systems. Its lack was felt to be detrimental to the faithful reproduction of coordinated turns, sideslip manoeuvres and particularly the study of asymmetric engine failures.

It was also hoped to use the yaw motion to apply lateral vibrations equivalent to those being increasingly experienced in flight as a result of the structural modes associated with long-nosed aircraft configurations.

The suggestion was made that 'transverse', rather than 'yaw', acceleration was the more useful motion cue. However, the overall design and installation of the Short's motion system and cockpit were more suited to the addition of a yaw freedom, the layout of which could provide a combination of translational and rotational cues.

6.2. Yaw Design Specification

A survey of the range of yaw accelerations and rates allowed for in the computation of past simulation exercises gave the following figures:

Aircraft	Acceleration	Rate
Avro 707A	25 °/s^2	10 °/s
First SST and Trident	5 °/s^2	10 °/s
VTOL type and helicopters	33 °/s^2	40 °/s
Jet flap type	33 °/s^2	—
Comet	12·5 °/s^2	Rate 1 turn 3 °/s
Ames SST. Engine failure on cruise	8 °/s^2	Rate 4 turn 12 °/s
Comet. Engine failure at take-off	3 °/s^2	—
BAC 221 and Airbus	—	10 °/s

These are modest acceleration figures compared with those required to represent a structural mode, typically 2 cm sideways translation of the pilot at 3 Hz say. On an arm of 2 m this is equivalent to amplitudes of approximately 1/2 degree in position, 180 deg/s^2 acceleration and 10 deg/s rate.

An overall aircraft yawing oscillation of up to ± 5 degrees amplitude at 1 Hz was thought to be a generous maximum excursion which the motion system and cockpit could withstand, equivalent to 30 deg/s rate and 190 deg/s^2 acceleration.

Based on these figures it was proposed to design a yaw servo capable of 40 deg/s maximum rate and 200 deg/s^2 maximum acceleration.

A satisfactory frequency response criterion was specified as not more than 15 degrees phase lag at 1 Hz, regarded as adequate for motion cues in aircraft simulation (*see* Appendix I).

Motion quality was to be based on experimental figures for acceleration perception thresholds stated to be between 4 and 20 cm/s^2. Acceleration 'noise' should not exceed 5 cm/s at the pilot's station, i.e. $<0{\cdot}007$ g.

Physical limitations of the motion system and cockpit located within its 9 m diameter display dome allowed an operational travel of ± 15 degrees with an additional ± 5 degrees for overrun.

6.3. Yaw Freedom—Servo Drive

The most practical scheme for the addition of yaw to the motion system was to lift the complete machine from its floor fixing and mount it on a large turntable placed with its axis directly under the CG of the machine. This caused the least disturbance to the existing servo drives and structure and was economic both in time and expense. The resulting axis of rotation is some 1·5 m behind the pilot and therefore generates a combination of yaw and sideways translation in the cockpit.

Since in the majority of conventional aircraft the pilot also sits foward of the yaw axis to a greater or lesser extent, this feature of the engineering solution was regarded as acceptable.

The installation adopted, minimised the design requirements for the servo drive and main yaw bearing. The yaw moment of inertia is at a minimum and the direct static loads on the bearing are evenly distributed. Turning moment loads imposed by the motion of heave and roll during normal operation are relatively light and substantially offset by dead weight. The bearing has to be of suitable design and strength to sustain the maximum loads imposed by deceleration at the end stops in combined servo runaway failure cases.

It was decided to mount the motion system squarely on a solid steel plate 51 mm thick in order to retain the load bearing continuity of the frame, and to sandwich a high quality, precision, low friction, large diameter slewing ring between this plate and a second similar 51 mm plate bolted and grouted to the floor. This led to a relatively inexpensive and simple installation with a minimum increase in height of the machine. This latter point has particular significance in allowing the continued effective use of the display dome in which the machine is housed.

The direct solution of a rotary servo motor for a rotary motion freedom was thought to have little to offer in terms of increased stiffness and performance over that of a linear actuator, in view of the difficulty and expense in supplying a power gear box and circular rack free of backlash and friction.

The non-linearity introduced over ± 15 degrees by the use of a linear tangential drive was not regarded as significant in this application.

The low mechanical-hydraulic stiffness of the heave and roll drives on this motion system were thought to limit the frequency response of those freedoms and to be in part due to the small radius (165 mm) at which the servo actuators operate. The turntable configuration of the yaw installation called for a long stroke actuator acting at a substantial radius, the actuator attachment point being outside the radius of the slewing ring. The servo should therefore be less sensitive to backlash and friction in the slewing ring, although more sensitive to jack friction and allow high stiffness to be achieved in the servo drive.

It was decided to specify the actuator size on a basis of adequate hydraulic stiffness to meet the frequency response specification and then check that acceleration and rate performance would be adequate in combination with system pressure and selected valve gear.

Local mechanical stiffness of the yaw bearing sandwich and jack attachments was high enough to present no problem. Stiffness of the rest of the motion machine and cockpit structure was very low with no practical remedy possible.

In view of the multiple freedoms at the cockpit it is difficult to monitor the yaw motion in the neighbourhood of the pilot's seat for control purposes. The servo loop, or loops, were therefore to be contained around the jack and yaw bearing leaving the main flexible high inertia load outside the loop. This limits the range of servo techniques applicable and probably limits the performance achieved.

To achieve a natural frequency of at least 10 Hz the required jack area was found to be 18·5 cm^2. The nearest commercially available standard jack was of 5·08 cm bore with thrust areas of 20·2 cm^2 and 10·7 cm^2.

The very substantial difference in jack piston areas was imposed by the need for a heavy duty rod design to give adequate 'strut' strength. A single ended design was chosen to give a compact installation in view of the long stroke required.

To achieve a maximum rate of 40 deg/s required a piston velocity of 63·5 cm/s and equivalent mean oil flow of 984 ml/s approximately. This was well within the capacity of a pair of MOOG22 servo valves.

The estimated inertia in yaw was 4700 kg m^2 about the centre of rotation.

The required maximum acceleration of 200 deg/s^2 or 3·5 rad/s^2 is equivalent to 5050 N m (3730 lb ft) acting on a 0·91 m radius arm requiring a pressure of 15·5 MN/m^2 (2250 lb/in^2) to be available at the jack. The oil supply requirements of the proposed yaw design were well outside the capability of the existing 20·7 MN/m^2 (3000 lb/in^2) system driving heave and roll.

It was decided therefore, to share the services of a 15·9–17·3 MN/m^2 (2300–2500 lb/in^2), 43 litre/min (9·5 gal/min) power rig to be installed for other hydraulic equipment. This power rig included two 'back up' hydraulic accumulators of 18 litres (4 gal) each.

The demanded 'power' for a sinusoidal motion of peak rate and acceleration of 40 deg/s and 200 deg/s respectively, is illustrated by the load curve of Fig. 29 and the load capability of the selected valve-jack combination is shown to be substantially in excess of this for an extending jack and just matched for a retracting jack.

The maximum allowable friction torque to ensure smooth motion through reversals of direction was assessed at 136 N m (100 lb ft), equivalent to acceleration noise levels of 0·005 g.

The expected friction torque from the bearing was 136 N m (100 lb ft) and from the servo-jack seal friction, 1 per cent of stall thrust torque, equivalent to 272 N m (200 lb ft).

It was thought that this assessment of 408 N m (300 lb ft) total friction torque was pessimistic and that the effects of excessive friction could be overcome by the tightness of the servo loop.

An integrated circuit, d.c., operational amplifier working through a single power output stage, driving single ended into two parallel wired servo valves was used to control the yaw servo (Fig. 30). A twin track d.c. position transducer was mounted in parallel with the jack between the floor pedestal and the yaw base frame to serve as the feedback element for the position control loop. A linear d.c. velocity transducer as used in the other servo drives was not available with sufficient stroke to be used directly on the yaw jack.

Instrumentation to sense rate and acceleration was not initially provided, since the retrospective fitting of a rate gyro or an accelerometer would present no difficulty.

6.4. Yaw Freedom—Development and Performance

Initial tests of the yaw drive as a simple position loop displayed a poorly damped oscillation with a frequency of about 7 Hz. This frequency superimposed on the demanded motion was evident at quite low gains and became progressively more prominent as the gain was increased and finally limited the positional stiffness that could be used. In addition to this resonance, very substantial 'reversal thumps' and breakout acceleration levels were evident.

Typical 'motion quality' recordings of acceleration and rate in the basic position control configuration are illustrated by Fig. 31.

Attempts to improve the waveform by deriving an acceleration feedback damping term from a short-stroke linear-rate pick-up mounted temporarily across the jack were reasonably successful, as illustrated by Figs. 32 and 33. At the highest loop gains used there was a strong tendency for high frequency (20–50 Hz) oscillation to occur, stalling the valve and upsetting the motion quality. Some of this trouble was due to insufficient stiffness in the linkage driving the rate pick-up itself.

In spite of the substantially improved general waveform achieved, the 'reversal thumps' were not significantly reduced, as illustrated by Fig. 34. The 'dwell' at zero rate followed by oscillating breakout is particularly noticeable as the frequency of motion is reduced. Although from the recordings the discontinuity may not appear very large and a record of position may look perfectly smooth, by actually riding on the motion it was apparent that the level of acceleration 'noise' at reversal of motion direction was not acceptable. At low frequencies, less than 1/4 Hz, say, the required acceleration cues are small and the 'jerk' produced by the motion system swamps the cue level and destroys the illusion of aircraft motion for the pilot as well as negating any possibility of the motion cue being an aid to controlling the simulated aircraft.

Further work on the yaw motion was concentrated on trying to eradicate the reversal thump rather than to improve the general servo performance. Attempts to use pressure feedback and feedback from an accelerometer only succeeded in destabilising the system. Hydraulic bleeds across the jack were ineffective as were attempts to remove valve threshold by backing off one valve against the other at zero input.

Backlash in jack attachment bearings was taken up as much as possible which did make an improvement of 20–30 per cent in the reversal thump.

Alternating current voltage inputs were superimposed on the demanded motion input at a range of frequencies and amplitudes. This technique was only successful when the level of a.c. applied was sufficient to cause visible jack movement at a frequency of 14·3 Hz which appeared to coincide with a jack resonance of some kind. The effect is illustrated in Fig. 35.

The audible noise and the vibration levels set up in the machine swamped any improvement in waveform and were quite unacceptable.

To look more closely into the cause of the acceleration 'noise' the servo jack was detached from the cockpit motion machine so that tests could be carried out on the servo drive and the cockpit yaw freedom independently.

Acceleration responses of the unloaded servo jack under open loop and closed loop, low and high gain position feedback conditions are illustrated in Fig. 36. The waveforms exhibit similar characteristics to those of the fully loaded servo but the dominant 'noise' frequency is in the region of 12 Hz. As frequency and amplitude are increased the reversal discontinuity increases proportionally, the period being about 0·1 second but the general smoothness of the waveform improves.

In Fig. 37 are shown the results of tests in which the yaw movement was activated in a smooth sinusoidal manner by pushing manually at the cockpit nose. This was done initially with the weight of the cockpit and heave beam supported on the static legs and secondly with the dead weight supported by the balance ram. The latter is a less stiff configuration and more representative of the conditions under which yaw would be used. Accelerations were measured on the floor frame in the first test and at the cockpit seat in the second test.

In this case also the general waveform has similar characteristics to those of the fully loaded servo drive. The floor frame acceleration shows a resonant frequency of 4–5 Hz and the cockpit structure acceleration shows resonant frequencies of about 10 Hz. The reversal discontinuity is still apparent.

It was felt that the problem with yaw motion quality stemmed from main support bearing friction, servo jack friction, valve threshold and jack attachment backlash. It has not been possible so far to isolate any one of these factors as being the most significant.

It was noticeable, however, that at the higher frequencies and amplitudes, above 1/2 Hz and ± 3 degrees for example, the reversal jerks were less apparent. This may be due to the momentum of the load being sufficient to overcome bearing friction and tighten up the backlash through the retardation and subsequent acceleration of a motion reversal.

The structural resonances in the overall machine limit the loop gains that can be used and make it unlikely that any reduction in the acceleration noise level will be achieved by normal servo techniques.

If it were possible to reduce the yaw bearing friction to practically zero by the use of an air bearing for example, the system would perhaps then tolerate the normal engineering 'running fits' in the jack attachments. There would be no dwell while servo pressure builds up to overcome main load friction and the servo jack would 'see' the load inertia directly. The load inertia would perhaps be sufficiently large to smooth out the inherent acceleration noise in the servo jack and the resultant motion quality might then be acceptable.

This possible solution to the yaw motion problem needs proving by appropriate laboratory tests before serious action is taken to change the present yaw bearing.

The limits of undistorted output in response to a simple harmonic input are given by Fig. 38 (Appendix III). Current performance and loadings are given in Table 1. The frequency response given in Fig. 39 is for a simple low gain position-follower configuration only. The corresponding circuit arrangement is illustrated in Fig. 30.

The maximum rates achieved by the servo drive are approximately 30 per cent down on those estimated. This brings the 'yaw left' rate below the specified requirement. This is probably due to a combination of pressure loss in the servo line, low servo gain and some friction resistance.

Acceleration limits have not been experimentally measured because of the hazards to the cockpit assembly involved.

The yaw motion quality attainable at present is not good enough for general use in simulation exercises. No permanent rate or acceleration stabilising loop has been installed. However it has been found possible to employ the yaw motion in simulations of aircraft with a lively lateral response in the presence of heavy turbulence. The inherent roughness of the motion is swamped by the mean level of turbulence and some useful yaw cues get through to the pilot. This experience substantiates the value of a yaw freedom and encourages further efforts to improve the existing drive.

7. Control and Safety

7.1. General Principles

Two general control and safety problems have to be resolved when using a motion machine of this type where human operators are in close proximity to it, or actually riding in it.

The first one is to be sure that the following procedures, 'start up', 'shut down' and 'computer-motion engage' and 'disengage' are at all times safe and under full control. The other is to be sure that all likely sources of control failure are 'fail safe' and result in the machine being brought to rest without damage to itself or personnel associated with it.

At the design stage a choice has to be made between automatic and manual control for the 'start up' and 'shut down' procedures. Automatic control sequences permit remote control but they must be 'fail safe' in operation and they inevitably increase the complexity of the machine, leading to greater risk of unserviceability.

Manual control may involve additional hazard to the operating staff, if it brings them into closer proximity to the machine. However, each stage of the manual sequence can be monitored and can be reversed if necessary. Individual freedoms can be energised without necessarily energising the others.

7.2. Short's Motion Machine Controls

In the original Short's motion machine, since it was a simple two degrees of freedom device, the start up and shut down controls were entirely manual. It was decided not to interfere with the original manual controls of roll and heave and to retain manual controls throughout. Since pitch works from the original hydraulic supply, pitch control was made an extension of the existing control panel on the motion support pedestal. The yaw controls were mounted on a remote hydraulic accumulator rack from which the yaw servo is energised.

A safety barrier isolates the motion machine from the rest of the laboratory in which the accumulator rack and yaw controls stand. A gate in this barrier is interlocked with the yaw motion so that access to the motion machine cannot be obtained while the yaw motion is energised. Effectively then the yaw motion must be the last freedom (or the only one) to be energised, and the first freedom to be shut down in the control sequence.

As illustrated in the hydraulic circuit (Fig. 9), manual control of the hydraulic systems and servos involves flow bypass valves for each of the two hydraulic supplies, and individual shut off valves for each of the four freedoms. In addition, each hydraulic system has a solenoid operated 'emergency stop' valve mounted in the pressure line downstream of 'back-up' hydraulic accumulators and solenoid operated 'dump' valves between the accumulators and the return or tank lines. These shut-off valves together with the solenoid-operated emergency stop valves must lie between the hydraulic back-up accumulators and the servo jacks, otherwise, after valve closure, full drive motion would still be possible until the accumulators are exhausted.

The solenoid-operated emergency stop and dump valves are wired to a common interlocked safety circuit (Fig. 10), together with the start/stop circuits for the hydraulic pump drives and computer-motion engage control circuits. The circuit incorporates a number of emergency stop switches (pilot, controller, motion control panel etc.) motion travel overrun limit micro switches for all four freedoms, and electric power supply monitoring for servo amplifier units.

The safety circuit has to be primed by a manual 'reset' button as part of the start up sequence and will only self-lock providing all interlocks and switches are in the 'operational' condition. It is designed to a 'fail safe' principle in that it functions by relaxing relays rather than by energising them.

In the event of failure of any of the monitored supplies, overrun of the motion limits or operation of the emergency stop switches, the safety circuit operates, disengaging the motion from the computer, isolating the servo drives from system pressures, stopping hydraulic pumps and dumping hydraulic accumulator pressures.

The system is then completely shut down and it will not be possible to start again until the safety circuit is reset. The motions subside immediately with no violent reactions.

7.3. Computer—Motion Coupling

In normal use when the motion system is energised it is necessary to be able to engage and disengage the motion machine and computer without incurring violent reaction. A secondary control unit is incorporated in the system to do this. Effectively the motion machine can only be engaged with the computer in the static 'reset' condition and with motion signal input lines slugged (*see* Figs. 14, 17 and 30). 'Initial condition' motion offsets are taken up gently. At the initiation of dynamic simulation the slugs are removed from the motion inputs and full servo response prevails. Each motion input is driven by a low impedance computing amplifier driving into a small reservoir capacitor. When computer 'reset' or 'motion disengage' is selected or in the event of a 'fail safe' shut down, the computer is switched out upstream of the reservoir capacitor. The reservoir capacitor 'holds up' the remaining signal level, allowing it to leak away slowly through the motion input impedance and thereby allowing the motion to disengage gently and return to its datum position.

The drive signals from the computer are diode limited at the input circuit so that the cockpit movements cannot normally be driven beyond their maximum operational ranges.

7.4. Servo Runaway Failures

Double-track position feedback potentiometers are used on each servo to avoid loss of control in the event of pot-track failure through wear.

In the event of some kind of servo control failure causing motion runaway into the operational limits, the limit micro switches operate the safety circuit described in Section 7.2.

In roll, pitch and heave the safety circuit relaxes relays to isolate the servo valves from the servo amplifiers, providing additional deceleration by creating partial hydraulic lock in the servo jacks. This feature was originally included in the yaw drive also but the resulting deceleration proved to be too severe owing to the larger mechanical advantage at which the yaw jack acts.

In roll and heave, deceleration results from the loss of drive pressure and the development of back pressure in the jacks. An additional protection in pitch is provided by the inclusion of hydraulic cushions within the jack body at each end of the stroke. The levels of deceleration in roll, pitch and heave are ultimately limited by the flexibility of the structure, a feature which is coincidental rather than intentional.

In yaw the deceleration is brought about by loss of drive pressure, development of back pressure in the jack from flow throttled through the servo valve, hydraulic cushions built into the overrun at each end of the jack, and finally, reaction from a large helical spring stop, compressed throughout the overrun distance.

Deceleration in roll, pitch and heave controlled in this way are relatively mild. In the case of yaw, deceleration of a runaway is still somewhat severe. Although the method used to stop runaways is simple and avoids complication of the servo jack assemblies it provides no means of adjustment. Progressive isolation of the servo drives from system pressure by mechanical cam action would be better, but this means individual throttling valves and cam drives for each freedom, resulting in complicated and bulky hydraulic servo packs.

Roll, pitch and yaw servo jacks are each provided with pressure relief valve protection. Ostensibly aimed at dissipating peak pressure developed in the jacks as a result of sudden closure of the valve during high rates of motion, their action during deceleration of a runaway can be embarrassing in the presence of dead weight or drive pressure. If the pressure relief valve lifts at this time it allows jack oil to escape via a relatively low restriction compared to the servo valve and in the presence of 'driving' dead weight or system pressure, the pressure relief valve may be held open allowing the load to accelerate into the jack limit of travel.

Two such instances have occurred in the motion machines used in the Aero Flight simulator—once, as a result of dead weight acting in the pitch freedom of the Short's machine and on the other occasion, on a machine in which throttling of the jack outflow against the progressive build up of system pressure as the movement decelerates was the designed system.

If it is essential to give some protection to the jack by pressure relief valves, then the relief valves should have throttled outlets at least as restrictive as the servo valves used on the jack. The use of balance rams to sustain dead weight is also preferable in this case.

7.5. General Safety Precautions

Other general features relating to safety in operation of the motion machine involve 'low pressure' and 'low oil level' switches on the hydraulic supplies which shut the system down in the event of sudden loss of pressure or gradual unobserved leakage of oil. Illuminated warning notices give indication that the motion system is 'live' which is not always obvious when the machine is static.

Smoking is prohibited within the motion 'operations' room and unauthorised personnel are forbidden access to it.

Pilots only transfer to and from the cockpit when the motion machine is fully shut down and energising is only commenced when the pilot is strapped in his seat.

8. Discussion and Recommendations

The cockpit motion machine described in this Report has given ten years of reliable service and been in constant use for simulation exercises throughout that time.

The design specifications which in some measure it did not meet, were, and still are, very demanding for this type of machine. Nevertheless the performance atained was sufficient for the machine to become a very effective vehicle on which to mount simulation exercises.

A comparison of the figures illustrating motion quality for the four freedoms does not obviously indicate why the performance of one axis rather than another should be more acceptable to the pilot. The reasons for this are several, some fundamental and some related to the particular motion system and cockpit design.

The motion cues derived from the pitch and roll movements appear to be more significant to the pilot than those derived from the other axes. This may be partly due to the fact that it is these movements which destroy the pilot's sense of the vertical referred to the simulator laboratory floor. This is a major factor in giving the simulator pilot a real sense of being airborne. In pitch and roll, attitudes together with accelerations are frequently used as composite cues, thus adding to the effectiveness of these motions. This increased effectiveness of the motion cues increases the tolerance the pilot has of some of the imperfections in the drive.

Another factor influencing pilot sensitivity to acceleration 'noise' is the transient response of the drive to the initial disturbance. A short time constant, inherent in a light stiff drive, may result in more objectionable sensations to the pilot than those produced by a heavier more flexible drive with a longer time constant. In the latter case the movement sensed by the pilot may be interpreted as realistic, if erroneous, aircraft motion, and therefore more tolerable than the harder 'knocks' and 'vibrations' of the former case.

The root cause of acceleration 'noise' in the motion drives is undoubtedly backlash, bearing and jack seal frictions and servo valve thresholds. However, the resultant level of noise to which the pilot is subjected also depends on the inertia, damping, stiffness and resonances in the motion system and cockpit structure. These vary from one axis to another as determined by the engineering design.

For direct comparison of the motion qualities to which the pilot is subjected by a number of servo driven movements it is necessary to record the motion on instrumentation mounted in close proximity to the pilot's seat.

The order of susceptibility to noise problems for the present motion system is as follows: yaw, roll, pitch and heave. The severity of the yaw motion problem is masked to some extent in the figures showing yaw rate and yaw acceleration because in most cases the rate pick-up and accelerometer was attached to the motion system frame rather than to the cockpit structure close to the pilot's seat.

It is still hoped to improve further the 'quality' of the motion cues provided by this machine, particularly in yaw which is a much needed addition to the system's capability.

In the event of a replacement motion system being required to fulfil a similar role, namely to provide fast smooth motion cues to a light single-seat cockpit, several lessons may be learnt from present experience and several points still need to be resolved.

There are numerous ways in which motion can be provided in a simulator cockpit. The manner and order in which it is done is a fundamental consideration involving not only the kinematics of the motion cues but also engineering design problems.

The motion system must be designed not simply to be strong enough to hold up against applied loadings but rather to be stiff enough to facilitate adequate servo control over the full input frequency range. When assessing the stiffness required one must take into account the basic motion structure, the servo drive and also the cockpit structure and any substantial additional loads such as display equipment.

The motion geometry used may have a substantial effect upon the stiffness which can be economically achieved. The stiffness requirement applied to the hydraulic servo drive may involve the use of large jack diameters and correspondingly large valve flow ratings, in which case torque outputs can be kept reasonably low by the reduction of system pressure. This in turn will help to reduce jack seal friction and pressure shock waves and relieve installation problems associated with high pressure systems.

The degrading effects of friction and backlash in the system are considerable. Every effort must be made to reduce these features to a minimum, to the extent of using special close tolerance fittings or backlash-free design and frictionless jacks and bearings. Normal engineering clearances on jack rod attachment bearings are not adequate, especially in the presence of load friction.

Main bearing design must be such as to reduce the effects of friction to a minimum even possibly by using air or fluid bearings. Small radius bearings may have some advantage over larger ones as the friction torque is reduced for a given loading.

The advantage gained in drive symmetry by the use of equal area jacks may outweigh the increased complexity of jack design and installation. Wherever substantial dead weights are to be sustained it is best to do so by the use of balance rams. By doing so drive symmetry is retained, physical damping forces can be applied by throttling the ram inlet port, and runaway deceleration loads are minimised.

Adequate provision for arresting runaway movements must be provided. In this respect the simple built-in hydraulic cushion often provided in hydraulic jacks was found to be unpredictable and ineffective in restraining substantial inertia loads, although desirable to give some protection to the jack itself. In the case of runaway due to hydraulic valves which have jammed in an open position, any buffer mechanism designed to restrain the load must work against the build up of drive pressure, ultimately developing stall thrust in the jack as the load slows down, in addition to dissipating kinetic energy. Cutting the supply pressure to the servo valve was found to be remarkably effective, even too severe, in stopping runaways. Progressive throttling of the supply pressure line to the valve or the driving valve line to the jack should provide more gentle deceleration and avoids building up excessive pressures in the jack. Throttling of outlet lines from the jack can lead to over-stressing the jack, or failure to restrain the load should the jack be protected by pressure relief valves.

The criterion by which a motion machine should be judged is on a basis of acceleration and rate of change of acceleration. Motion fidelity in relation to demanded acceleration is required down to very low levels, and acceleration 'noise' must therefore be kept to a minimum.

'Noise' levels in the motion cues should at least be less than those defined by the thresholds of perception for human beings. Approximate figures for these levels are 0·005 g to 0·01 g for 'normal' acceleration, 0·01 g to 0·02 g for lateral acceleration and 2 to 5 deg/s^2 for angular accelerations.

Linear acceleration 'noise' levels of the order of 0·05 g are experienced on the Aero Flight machine.

Acceleration 'noise' is the most persistent problem met with in motion machines for simulators, particularly where hydraulic drives are employed. The hydraulic jack or motor controlled by flow valves, tends to become locked by closure of the servo valve whenever the resultant servo drive signal (error signal) is zero. The inherent compliance of the oil locked in the cylinder means effectively that the load is driven via an undamped spring. These two factors appear to be fundamentally different from say an electric drive where it is possible for the load to 'slip', even at zero servo drive signal, and reactions within the motor are damping forces rather than spring forces. This suggestion seems to be borne out by the experience with some American electrically driven motion simulators which are reputed to be less susceptible to acceleration 'noise' problems. However, the facility with which hydraulics can be used in this application makes hydraulic drive the obvious choice if an acceptable compromise can be found.

Carefully filtered acceleration feedback can be used to stabilise the resonant modes of the hydraulic servos and load, where cross coupling from one axis to another can be avoided. The increased damping resulting from this added stabilisation degrades the frequency response of the drive. However, in the computer programmes for flight simulation, analog signals of acceleration and rate are normally available, and can be used as compensating lead commands to the basic position servos and in conjunction with acceleration and rate feedback can be used to improve motion fidelity and extend the available frequency response.[6]

Where adequate acceleration and rate feedback for each motion drive is not easily obtained, a simple way to improve the frequency response is to factor the command signal from the computer by a network equivalent to, or, a reasonable approximation to, the inverse transfer function of the servo.

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APPENDIX I

Frequency Response for Motion Drives

It is generally understood that a human controller—such as the pilot of an aircraft—is subject to an effective response lag of the order of 0·25 seconds. In order not to invalidate flight control activities in which pilot reaction limits are important, it is essential during simulation not to present the pilot with cues which themselves are lagging behind the true motion by a significant amount, i.e. significant in relation to the 0·25 seconds pilot's delay. From such considerations a time constant of no more than 0·04 seconds is thought to be an appropriate design target for a mechanism to be used to generate acceleration sensations in piloted flight simulators. The time constant of 0·04 seconds is equivalent to a phase lag of 15 deg at 1 Hz for a single lag system. This is the origin of the frequency response specification for Aero Flight simulator motion equipment.

Hadekel[8] shows that the transfer function of a hydraulic jack-type position-servo driving an inertia load may be written

$$\frac{\theta_{\text{out}}}{\theta_{\text{in}}} = \frac{K}{K + S(S^2(M/\lambda) + S(MK/\mu) + 1)},$$

where K is the servo loop gain—the steady state velocity per unit error under 'no load' conditions,
M is the inertia load referred to the jack output,
λ is the rigidity of the mechanism resulting from the combined effect of structural elasticity and oil compressibility.
and μ is the static output stiffness, power on, i.e. the steady load which must be applied in order to deflect the servo through unit position error.

It is also shown that for adequate damping an optimum ratio of $\mu/\lambda = \frac{1}{3}$ occurs, defining the maximum usable gain factor

$$K = \frac{2}{3\sqrt{3}}\sqrt{\frac{\lambda}{M}} = \frac{2\omega_0}{3\sqrt{3}},$$

where ω_0 is the undamped natural frequency of the inertia load under the flexible restraint of the hydraulic jack and supporting structure with the servo drive unenergised.

With the values of damping and gain factors given above, the phase lag of the mechanism depends only on F, the ratio of forcing frequency to the undamped natural frequency

$$\text{phase lag} = \frac{\tan^{-1} F(F^2 - 1)}{2/\sqrt{3}(\frac{1}{3} - F^2)},$$

where $F = \omega/\omega_0$.

For small values of F this is approximately equivalent to:

$$\text{phase lag (in degrees)} = 150\,\text{F}.$$

It follows, that, in order to achieve a phase lag of 15 deg at 1 Hz specified for the Aero Flight motion drives, the natural frequency of the equipment would have to be at least 10 Hz.

APPENDIX II

Natural Frequency of the Motion Drive

It is shown in Appendix I that the phase lag in the motion response is directly related to the natural frequency of the mechanism. The natural frequency is determined by the ratio of elastic to inertia forces. The inertia forces are largely determined by the mass distribution of the cockpit. The elastic forces arise from structural deflection and compressibility in the oil.

The stiffness associated with oil compressibility, measured at the jack, is given by:

$$\lambda_{\text{oil}} = \frac{4NA}{L},$$

where N = oil bulk modulus,
A = jack piston area
and L = jack stroke.

When referred to the load this rigidity is reduced as the square of the movement ratio r between jack and load, so that

$$\lambda_{\substack{\text{oil}\\ \text{at the load}}} = \frac{4NA}{Lr^2}.$$

Since, however, the load movement S is specified and is related to the jack stroke by $S = Lr$, then

$$\lambda_{\substack{\text{oil}\\ \text{at the load}}} = \frac{4NA}{Sr}.$$

Thus for a given load movement S and constant jack area A the greatest stiffness is achieved with the smallest movement ratio r, i.e. ideally the jack should act directly on the load.

The natural frequency F_n of the drive and load combination is defined by the combined structural stiffness and oil stiffness in the relationship

$$F_n = \frac{1}{2\pi}\sqrt{\frac{\lambda_{\text{oil}} + \lambda_{\text{structure}}}{M}}.$$

The choice of jack to load gearing for maximum stiffness however is compromised in practice by other considerations, such as the availability of adequate valve gear, limitations on jack stroke and overall compatibility of the motion geometry.

APPENDIX III

Maximum Undistorted Sinusoidal Output Characteristics

The overall performance of the motion drives has been illustrated in Figs. 18, 19, 25 and 36, by envelopes defining the limits of sinusoidal positional response over the full input frequency range.

The limitations are imposed by the available positional range, maximum steady rate defined by friction, viscous resistance and valve flow rating, and maximum acceleration at zero rate defined by inertia and jack stall thrust.

The amplitudes of the input signal required to drive the motions up to the margin defined by this envelope are determined by the attenuation curve of the frequency response characteristics.

TABLE 1

Four Freedom Motion Machine—Current Performance and Loadings

	Range			Measured maximum rate	Estimated maximum acceleration	Servo ram				Valve flow at 7 MN/m^2 (1000 lb/in^2)	Approximate loadings
	Working	Overrun	Total			Area	Stroke	Torque arm	System pressure		
PITCH Up	15°	5°	20°	37°/s	184°/s^2	11·4 cm^2 (1·77 in^2)	14 cm (5·5 in.)	20·3 cm (8·0 in.)	20·7 MN/m	330 ml/s	440 kg m^2 (300 sl ft^2) about pitch axis. 4 kN (900 lb wt) Dead wt.
PITCH Down	15°	5°	20°	72°/s	936°/s^2	7·5 cm^2 (1·16 in^2)			(3000 lb/in^2)		
HEAVE Up	20°	15°	35°	23°/s	122°/s^2	14·8 cm^2 (2·3 in^2)	14·5 cm (5·7 in.)	16·5 cm (6·5 in.)	20·7 MN/m^2	200 ml/s	2360 kg m^2 (1600 sl ft)^2 about heave axis. 5·12 kN (1150 lb wt) Dead wt.
HEAVE Down	10°	3°	13°	23°/s	122°/s^2	14·8 cm^2 (2·3 in^2)			(3000 lb/in^2)		
ROLL Left	15°	15°	30°	57°/s	537°/s^2	8·06 cm^2 (1·25 in^2)	16·2 cm (6·4 in.)	16·5 cm (6·5 in.)	20·7 MN/m	115 ml/s	294 kg m^2 (200 sl·ft^2) about roll axis
ROLL Right	15°	15°	30°	57°/s	537°/s^2	8·06 cm^2 (1·25 in^2)			(3000 lb/in^2)		
YAW Left	15°	5°	20°	30°/s	234°/s^2	10·6 cm^2 (1·65 in^2)	67 cm (26·4 in.)	96·5 cm (38 in.)	17·3 MN/m^2	660 + 660 ml/s	4700 kg m^2 (3200 sl ft^2) about yaw axis.
YAW Right	15°	5°	20°	43°/s	445°/s^2	20·3 cm^2 (3·14 in^2)			(2500 lb/in^2)		
	Heave balance ram					20·2 cm^2 (3·14 in^2)	38 cm (15 in.)	42 cm (16·5 in.)	Up to 20·7 MN/m^2 (3000 lb/in^2) available	Torque arm referred to heave axis	

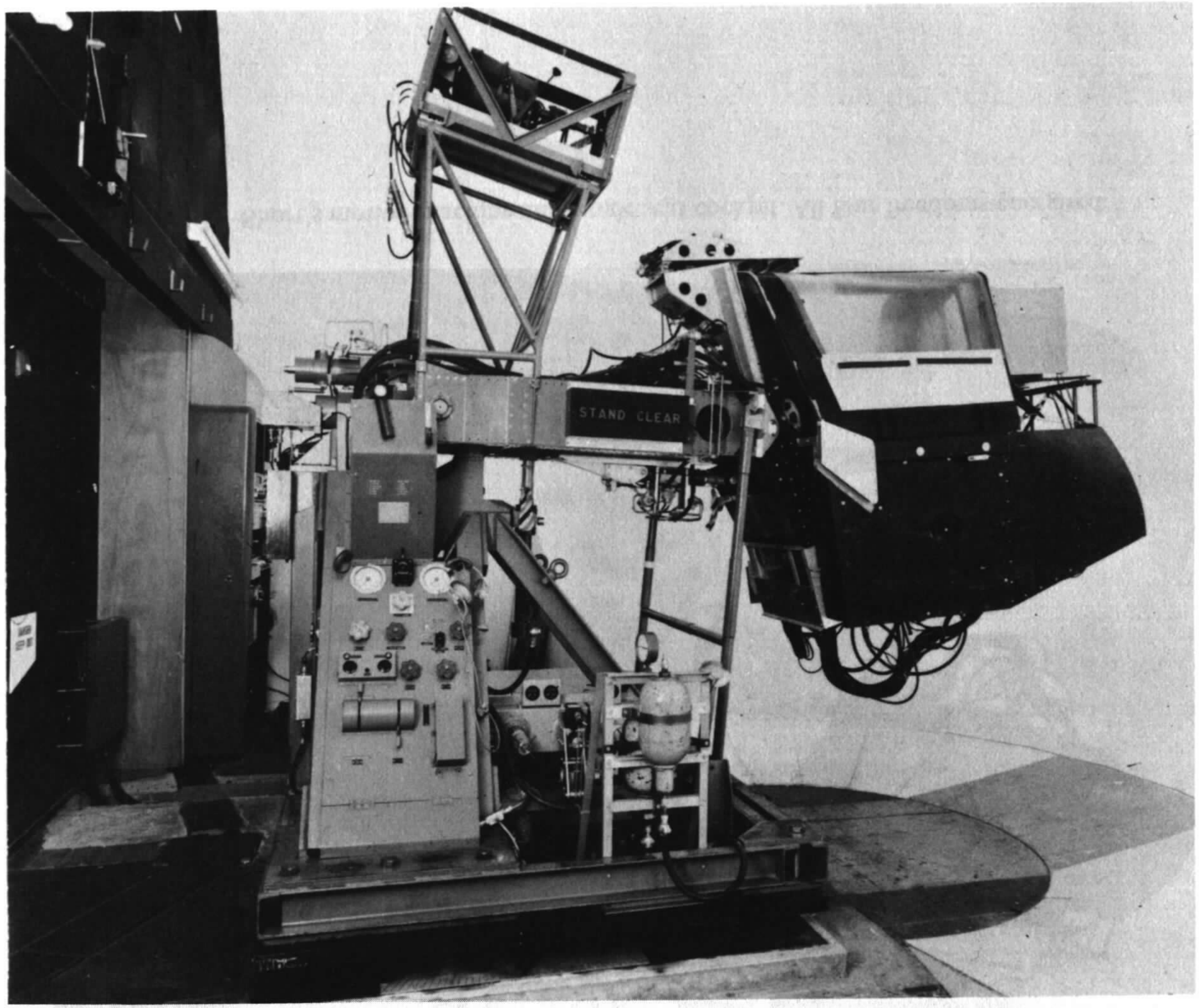

FIG. 1. Short's motion machine and single seat cockpit. All four freedoms locked.

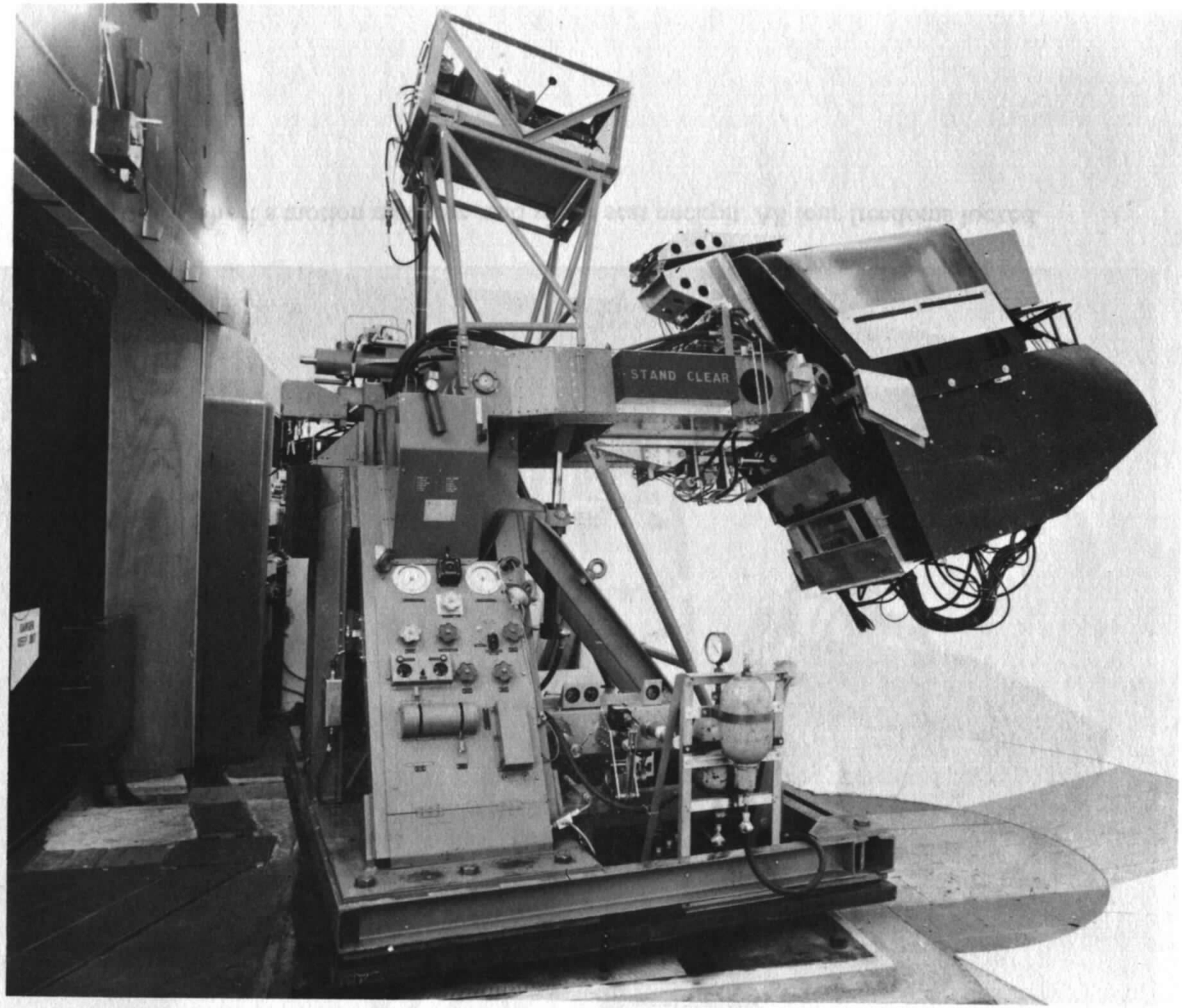

FIG. 2. Short's motion machine and single seat cockpit. All four freedoms energised.

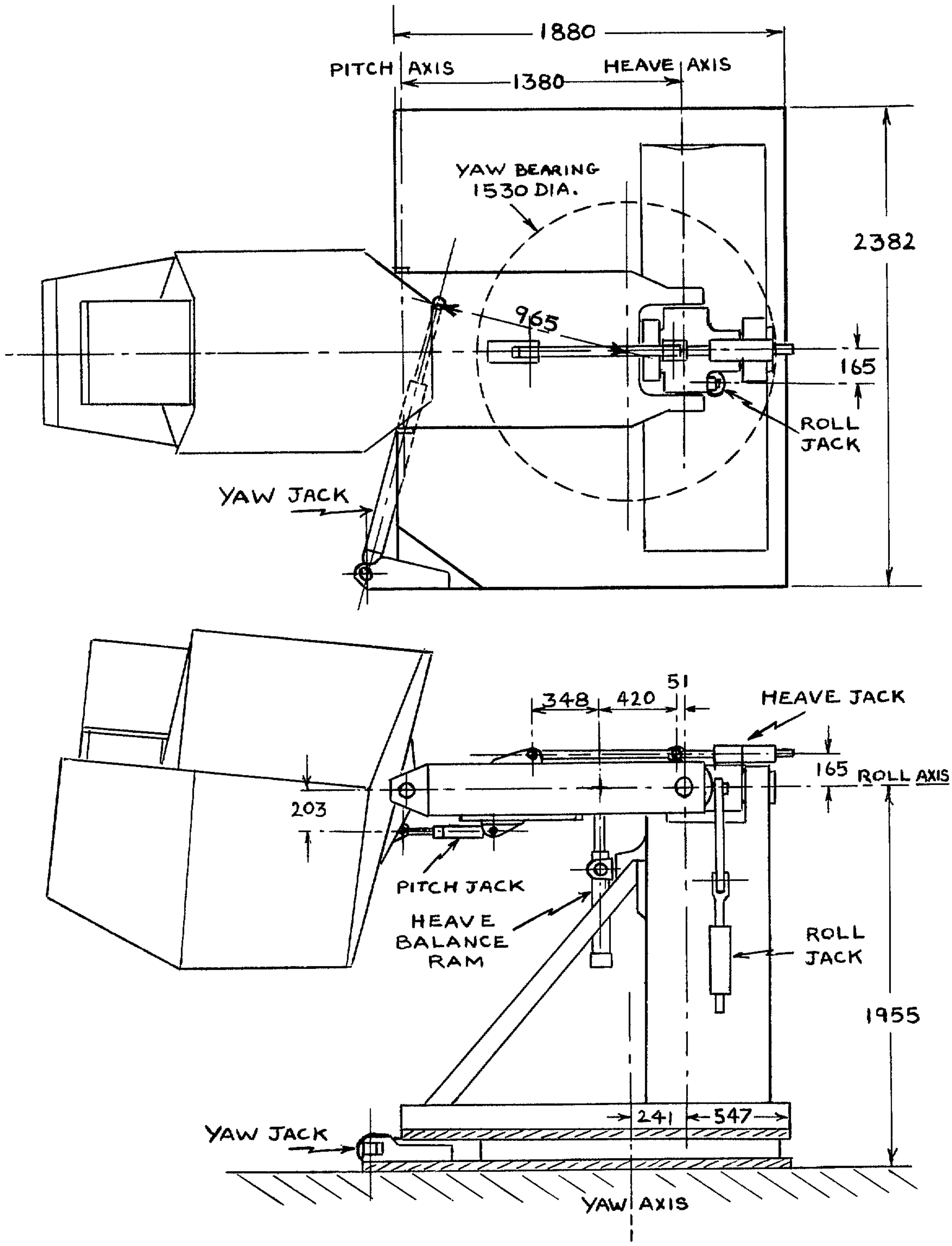

FIG. 3. Four freedom cockpit motion machine. G.A. (dimensions in mm).

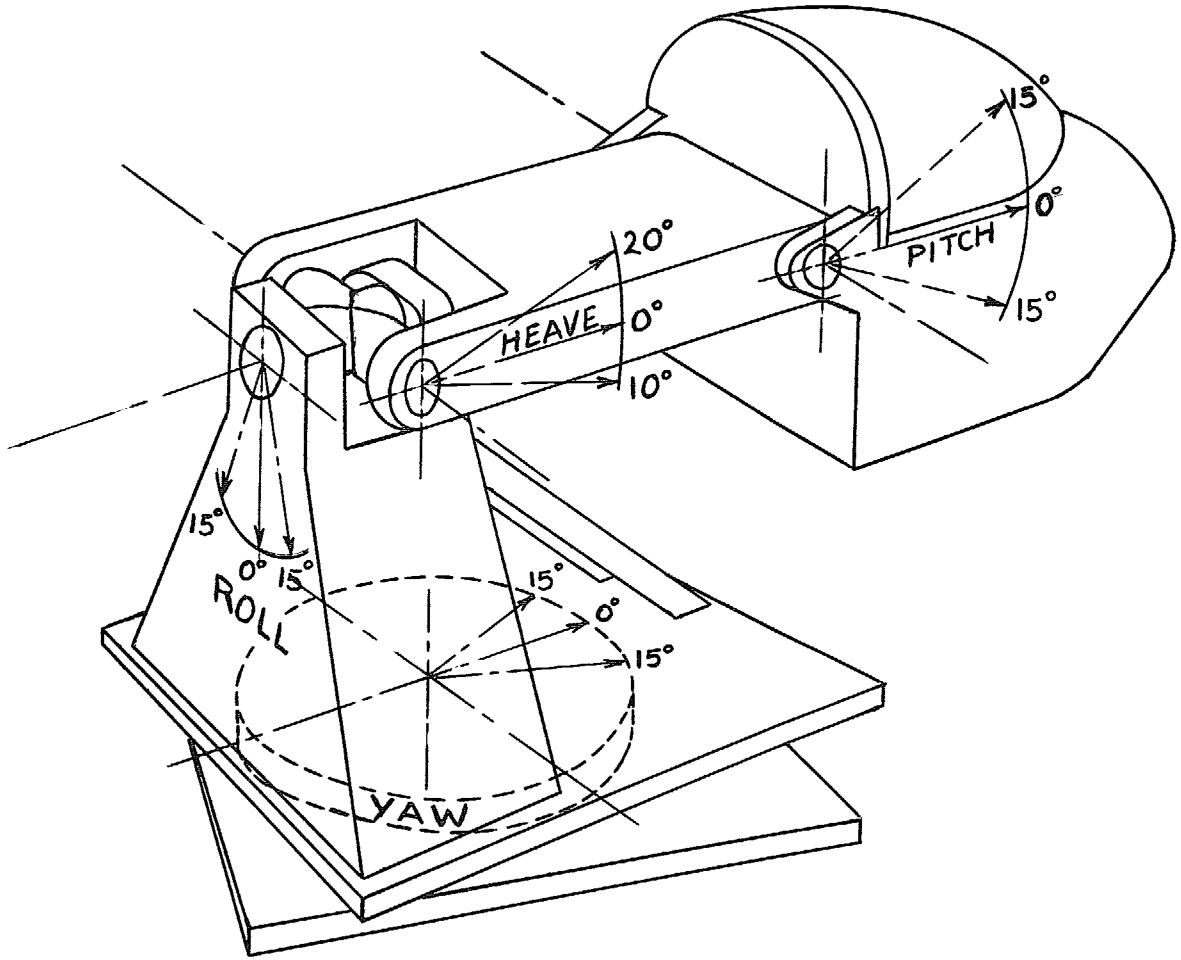

FIG. 4. Four freedom cockpit motion machine. Kinematics and operational ranges.

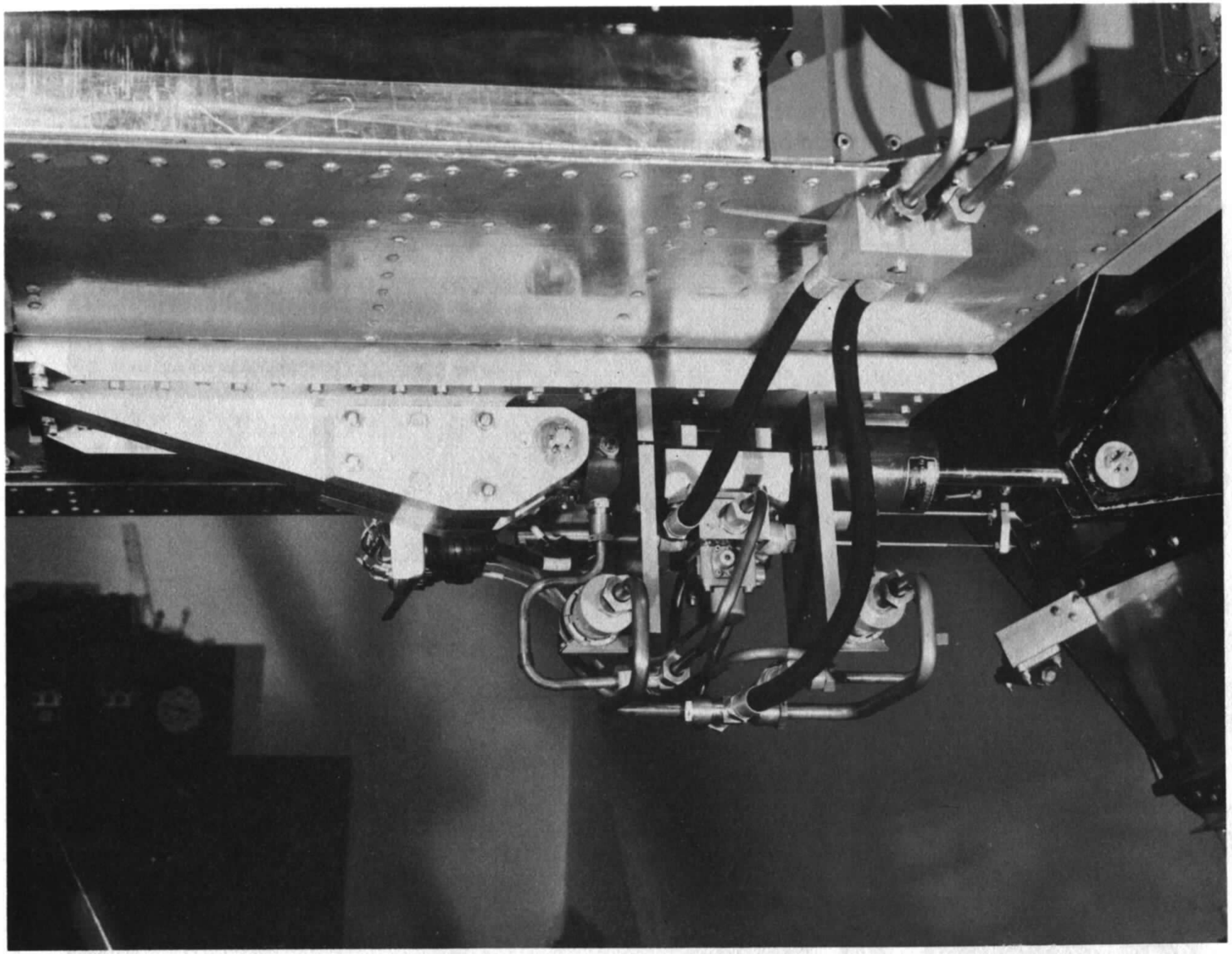

FIG. 5. Pitch freedom hydraulic servo drive assembly.

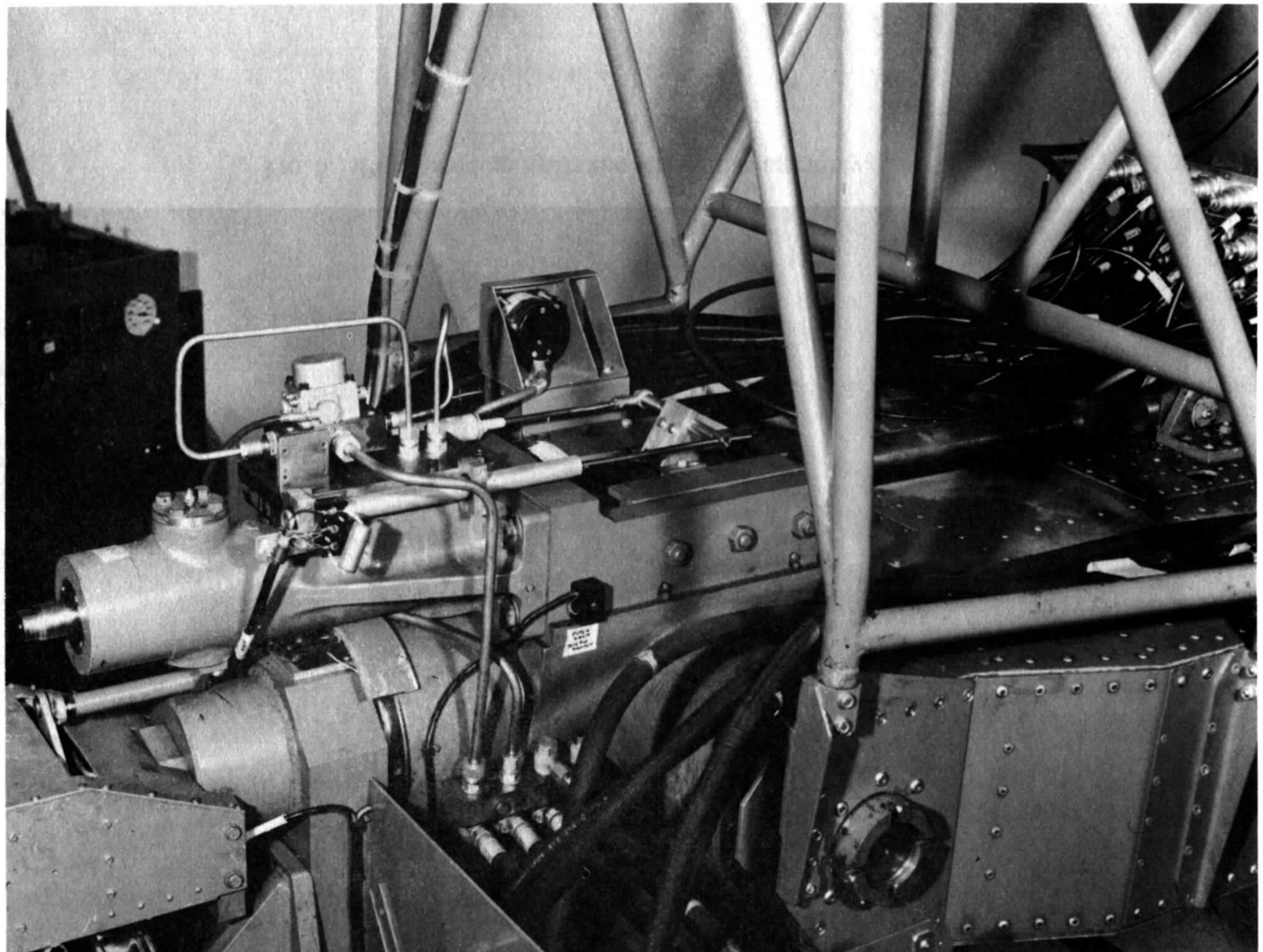

FIG. 6. Heave freedom hydraulic servo drive assembly.

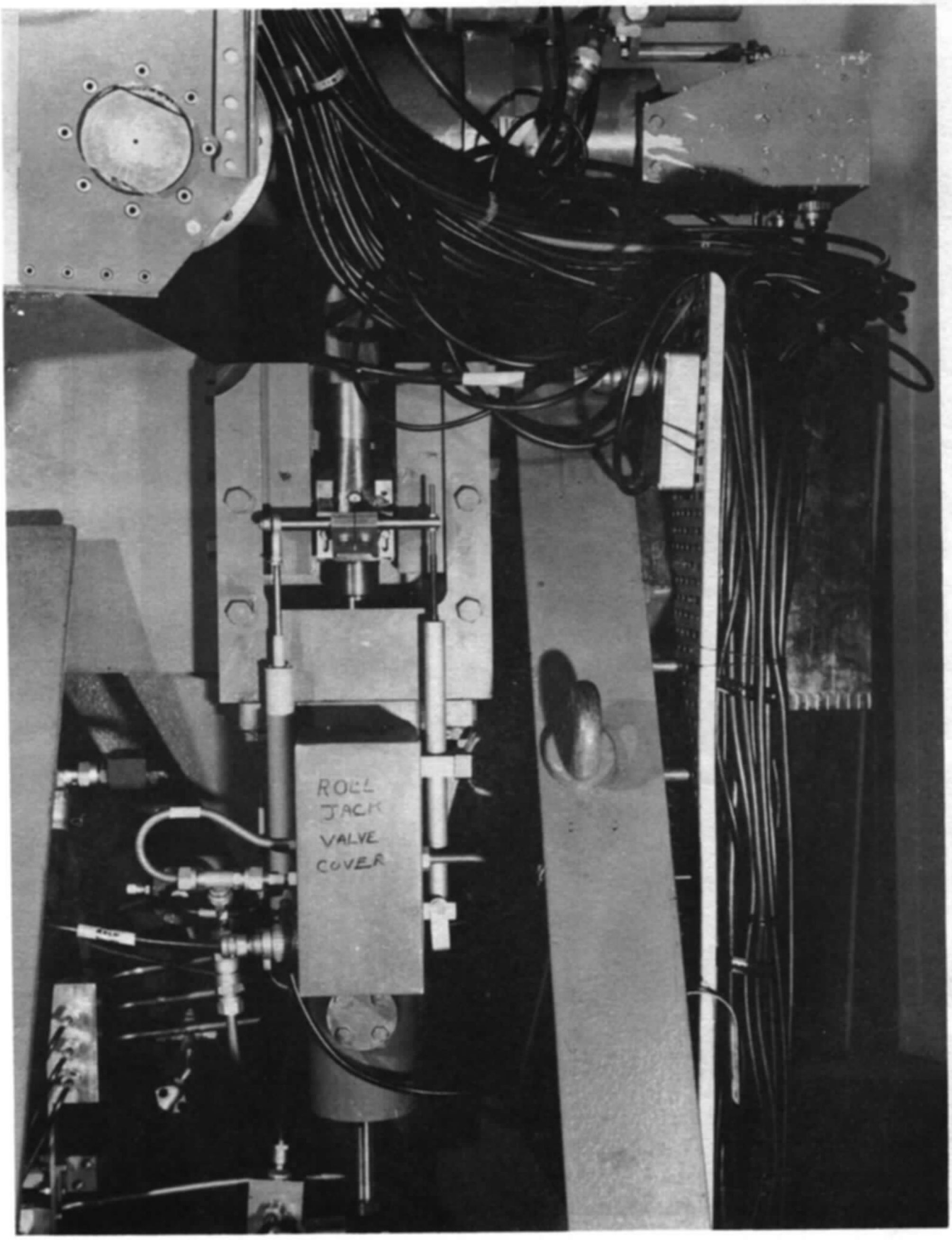

FIG. 7. Roll freedom hydraulic servo drive assembly.

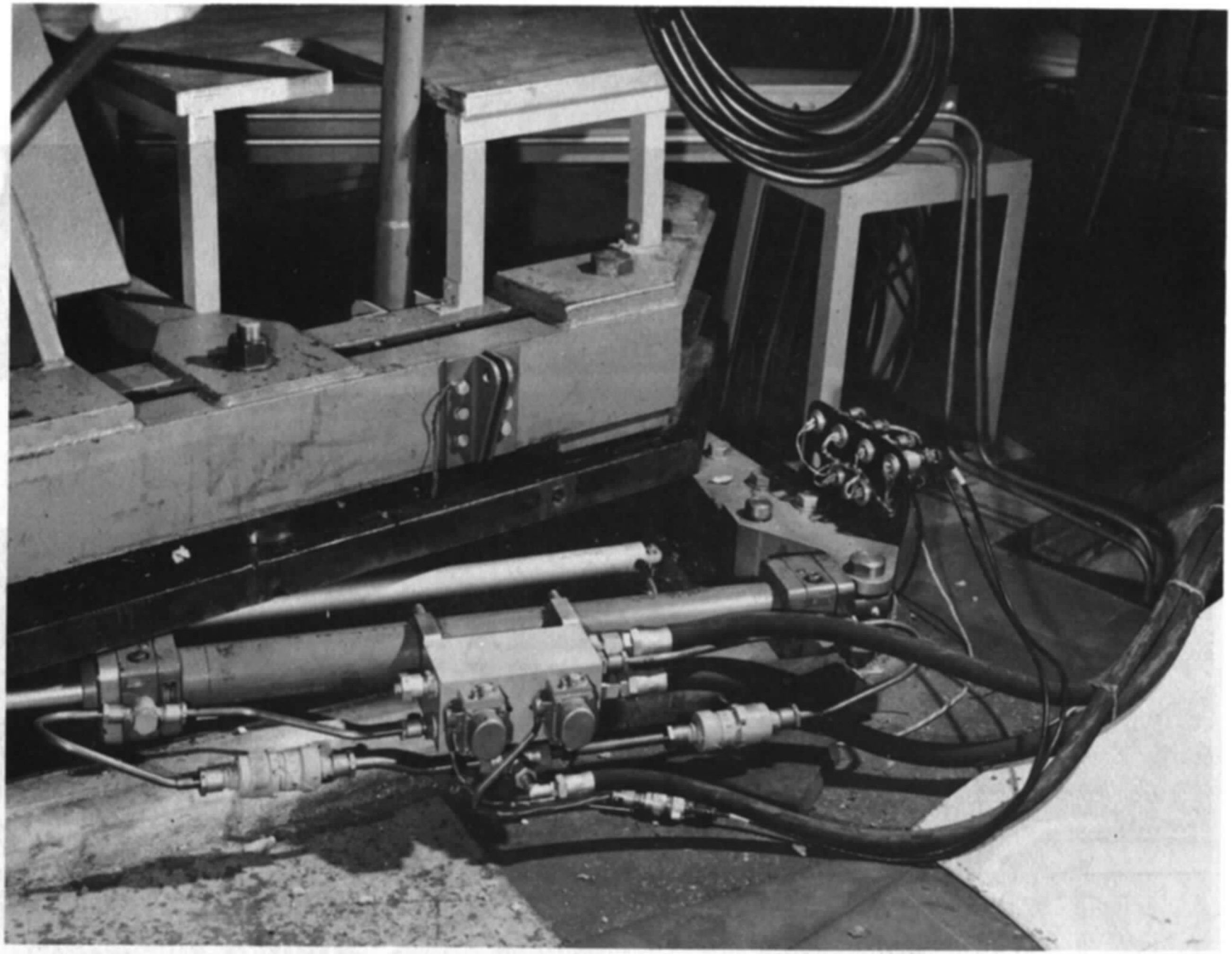

FIG. 8. Yaw freedom hydraulic servo drive assembly.

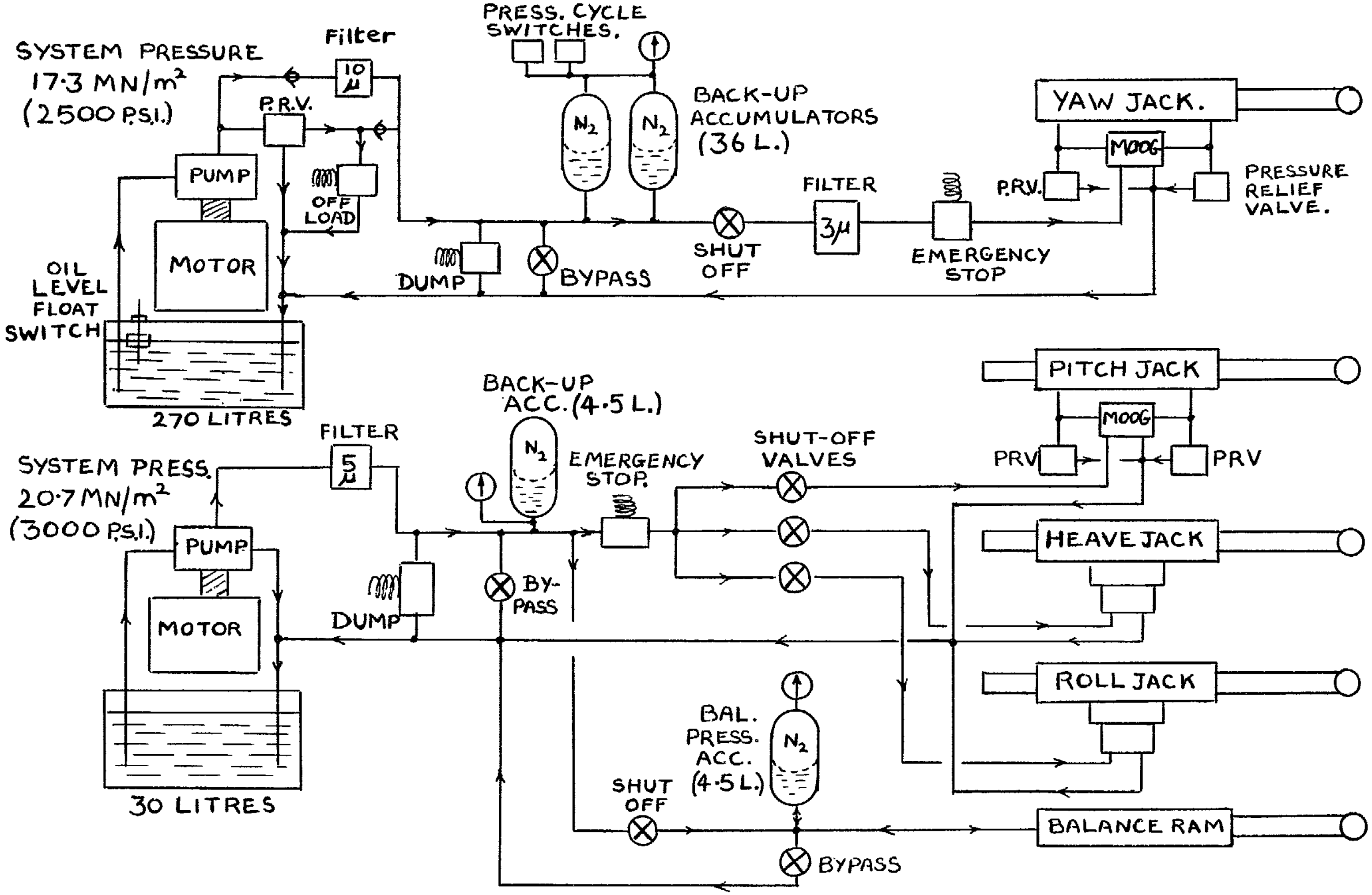

FIG. 9. Four freedom cockpit motion machine. Hydraulic system.

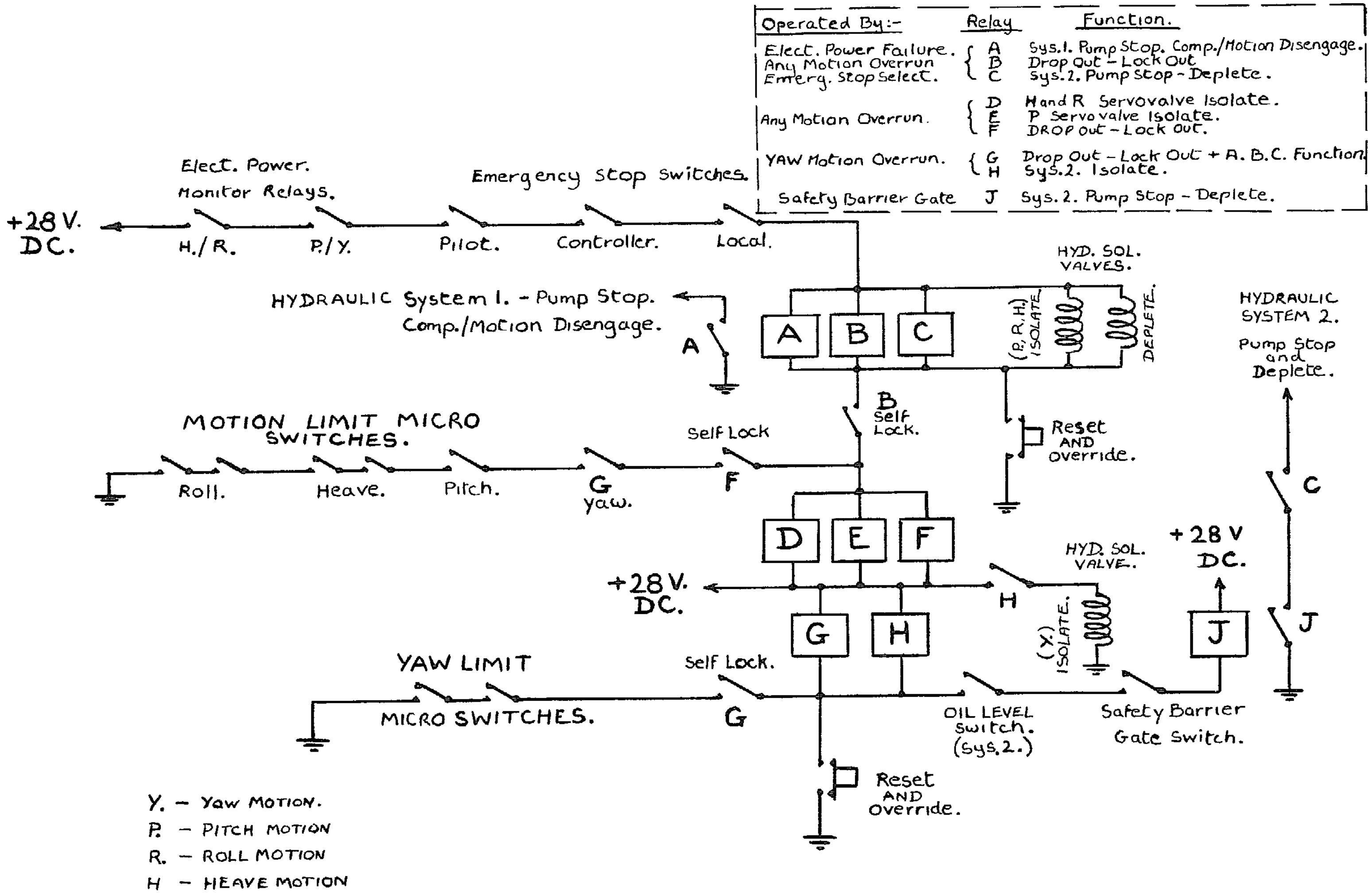

FIG. 10. Four freedom cockpit motion machine. Electrohydraulic safety circuit.

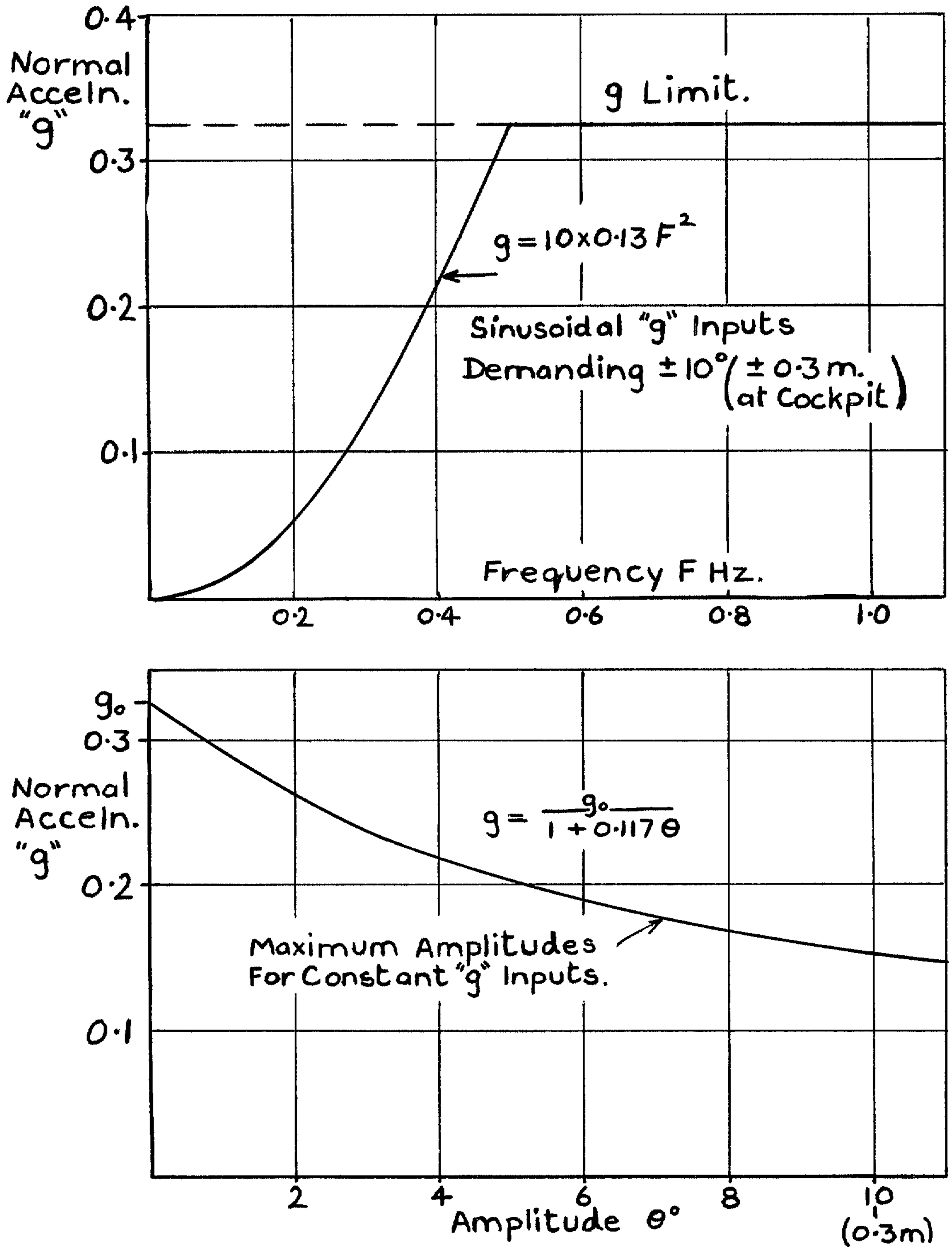

FIG. 11. Heave freedom: Limits of 'normal' acceleration available within the specified performance.

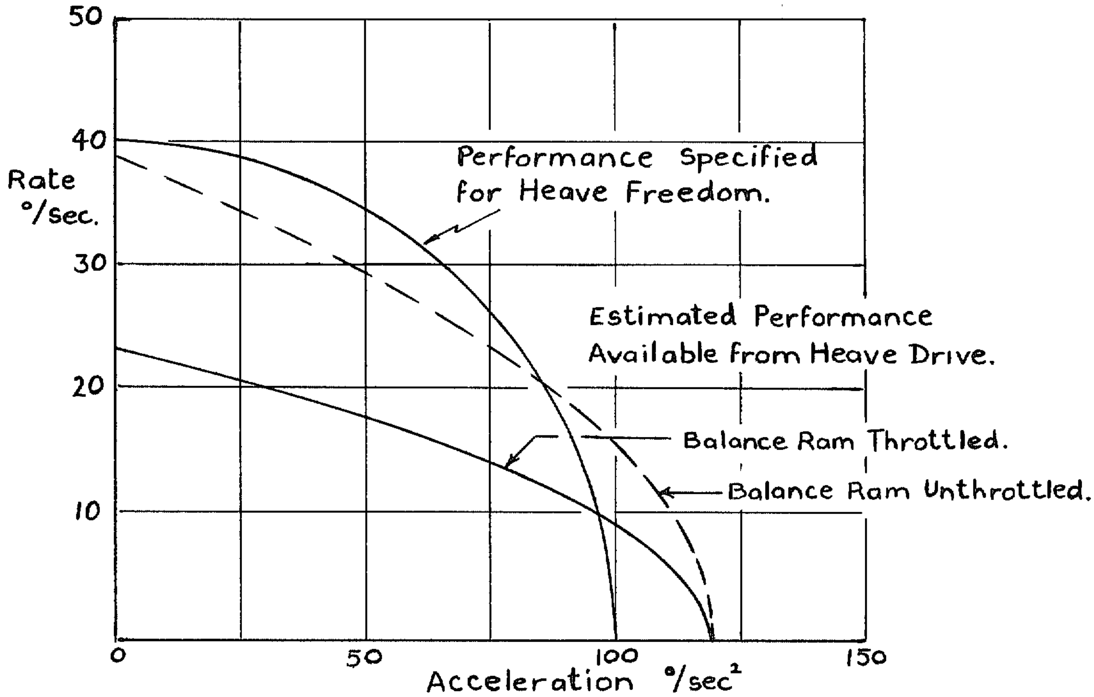

FIG. 12. Heave freedom: Simultaneous rate/acceleration (power) available from the hydraulic drive (Design estimate).

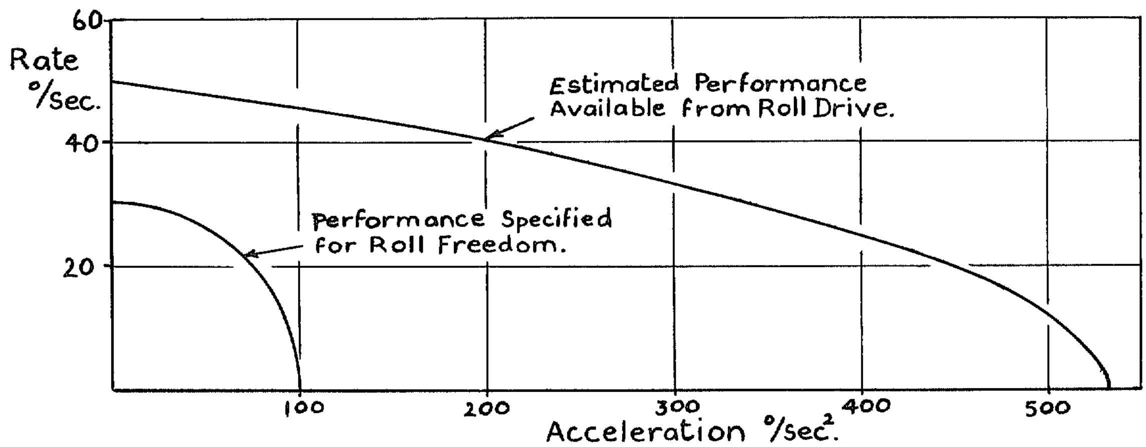

FIG. 13. Roll freedom: Simultaneous rate/acceleration (power) available from the hydraulic drive (Design estimate).

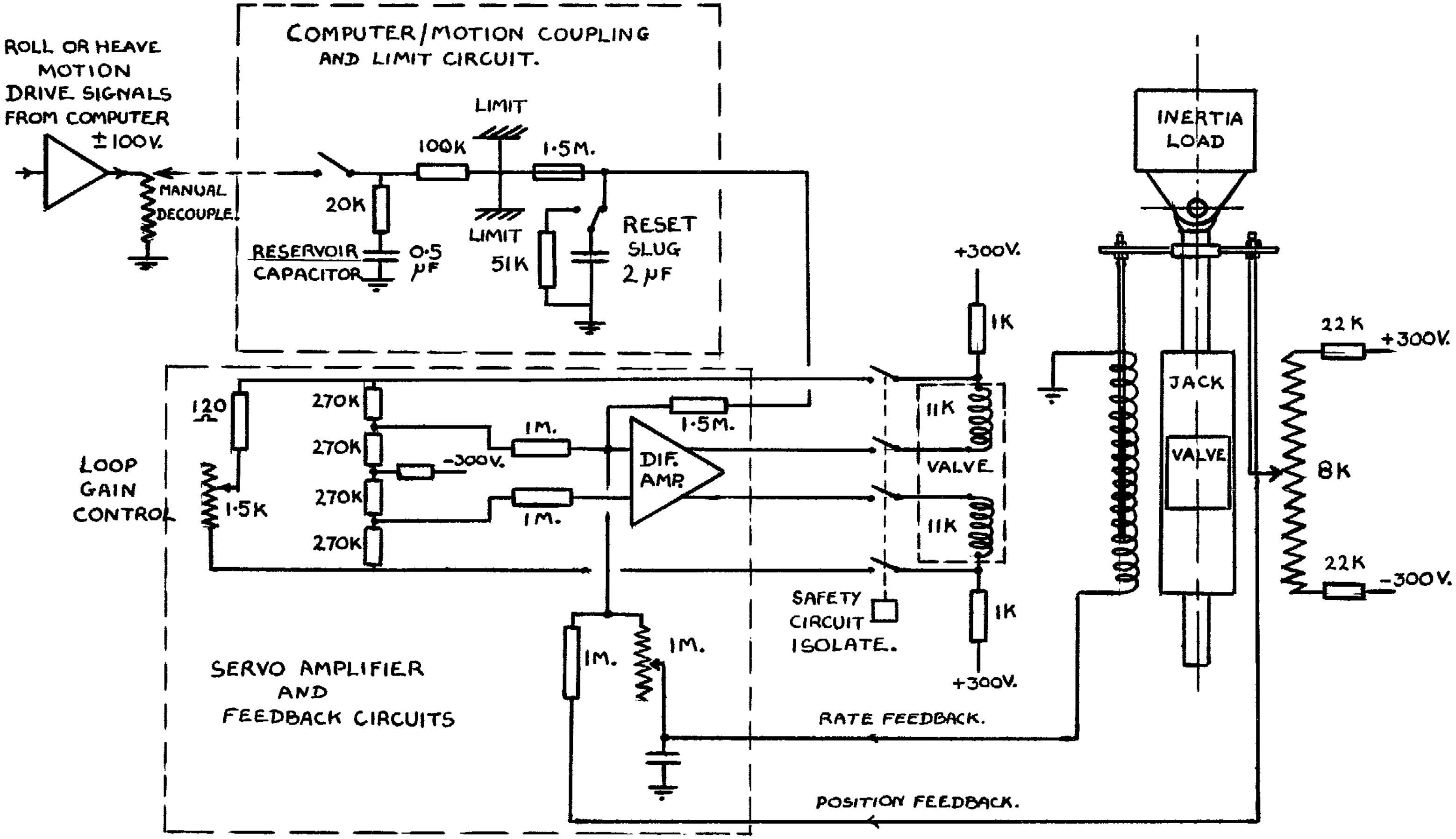

FIG. 14. Heave and roll freedom servos: Circuit arrangement.

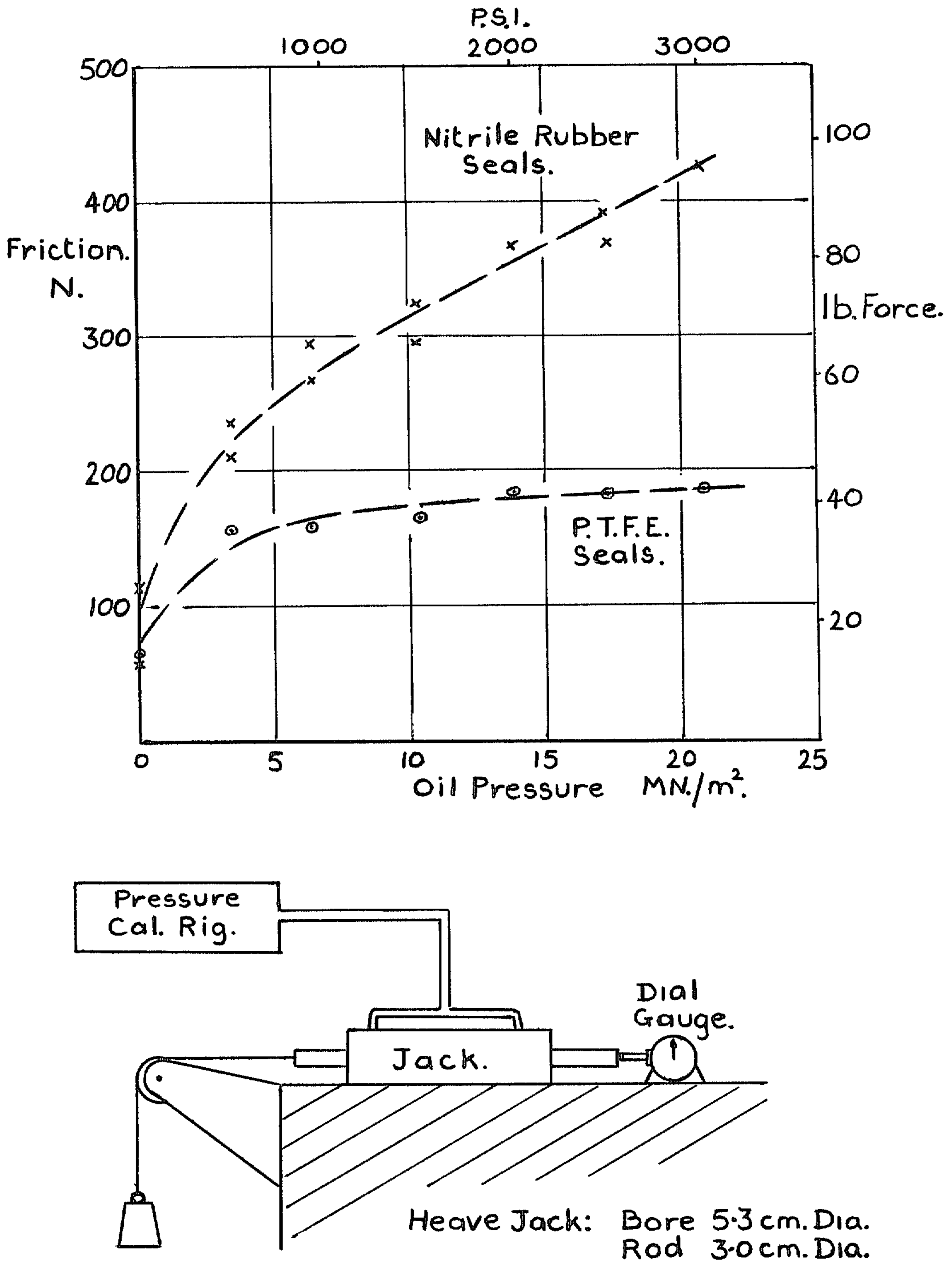

FIG. 15. Measurement of jack seal friction throughout the operation pressure range.

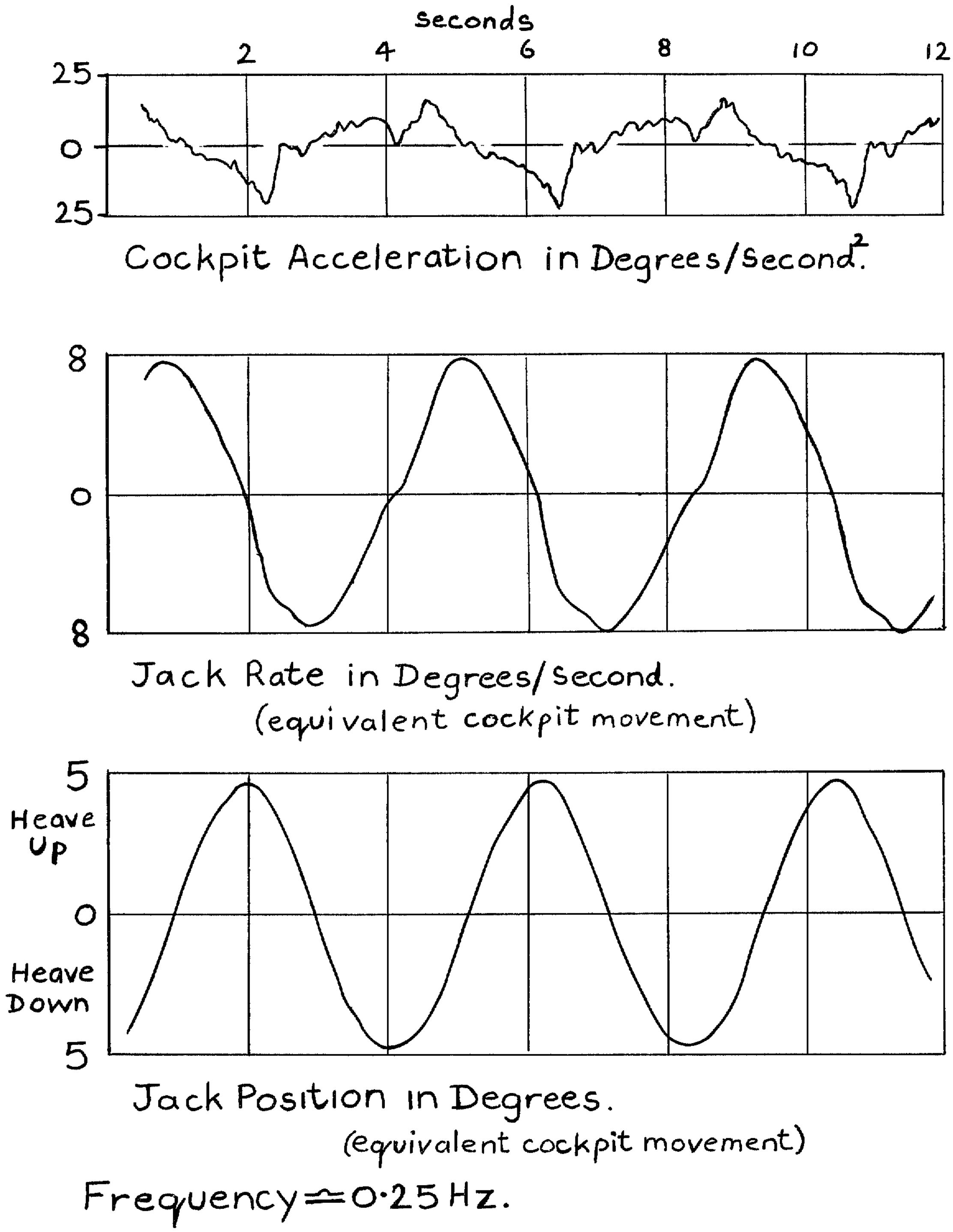

FIG. 16. Heave motion quality: Response to sinusoidal input signal.

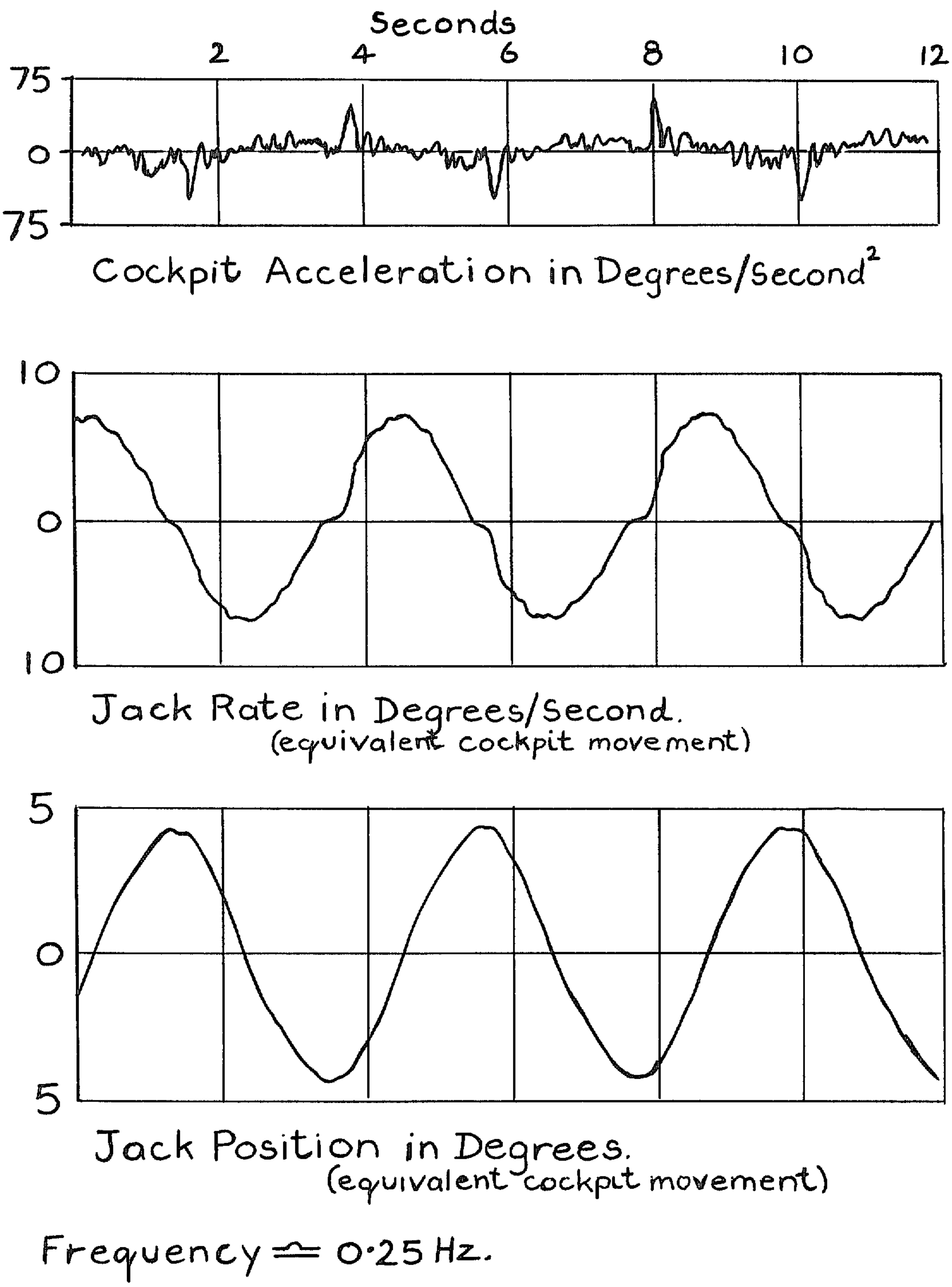

FIG. 17. Roll motion quality: Response to sinusoidal input signal.

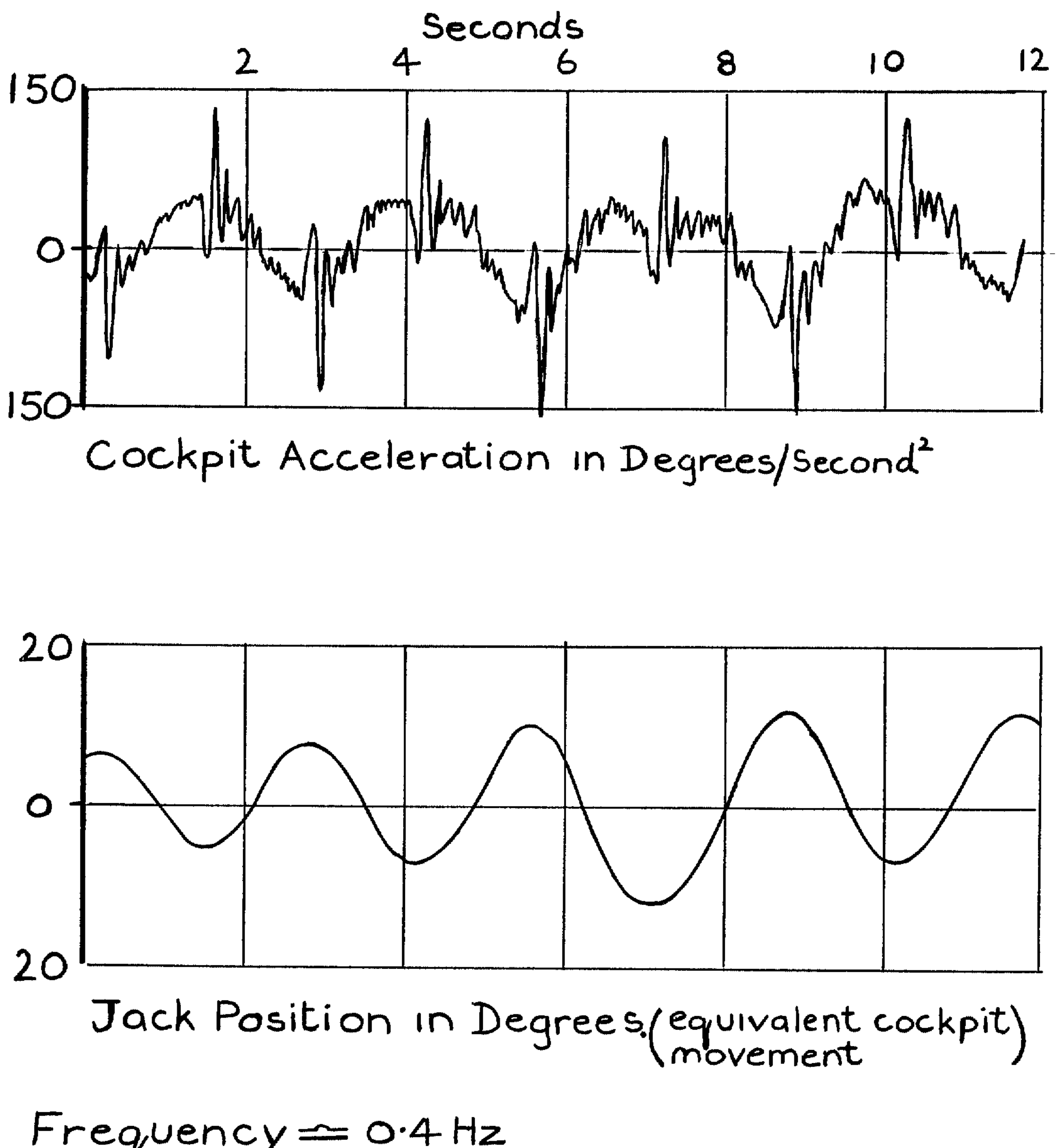

FIG. 18. Roll motion quality during aircraft (H.P. 115) simulation.

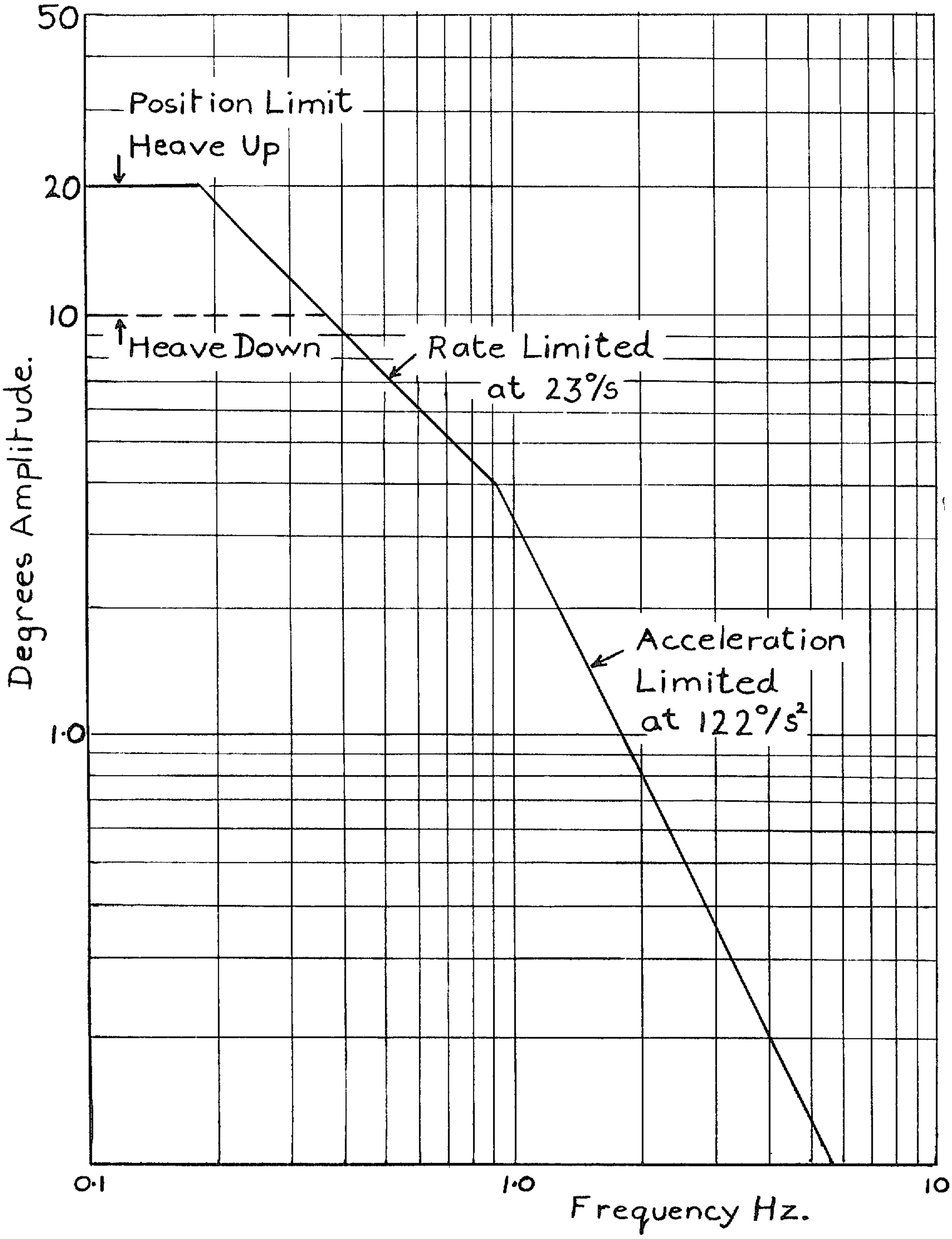

FIG. 19. Cockpit motion: Heave freedom servo—Maximum undistorted sinusoidal output.

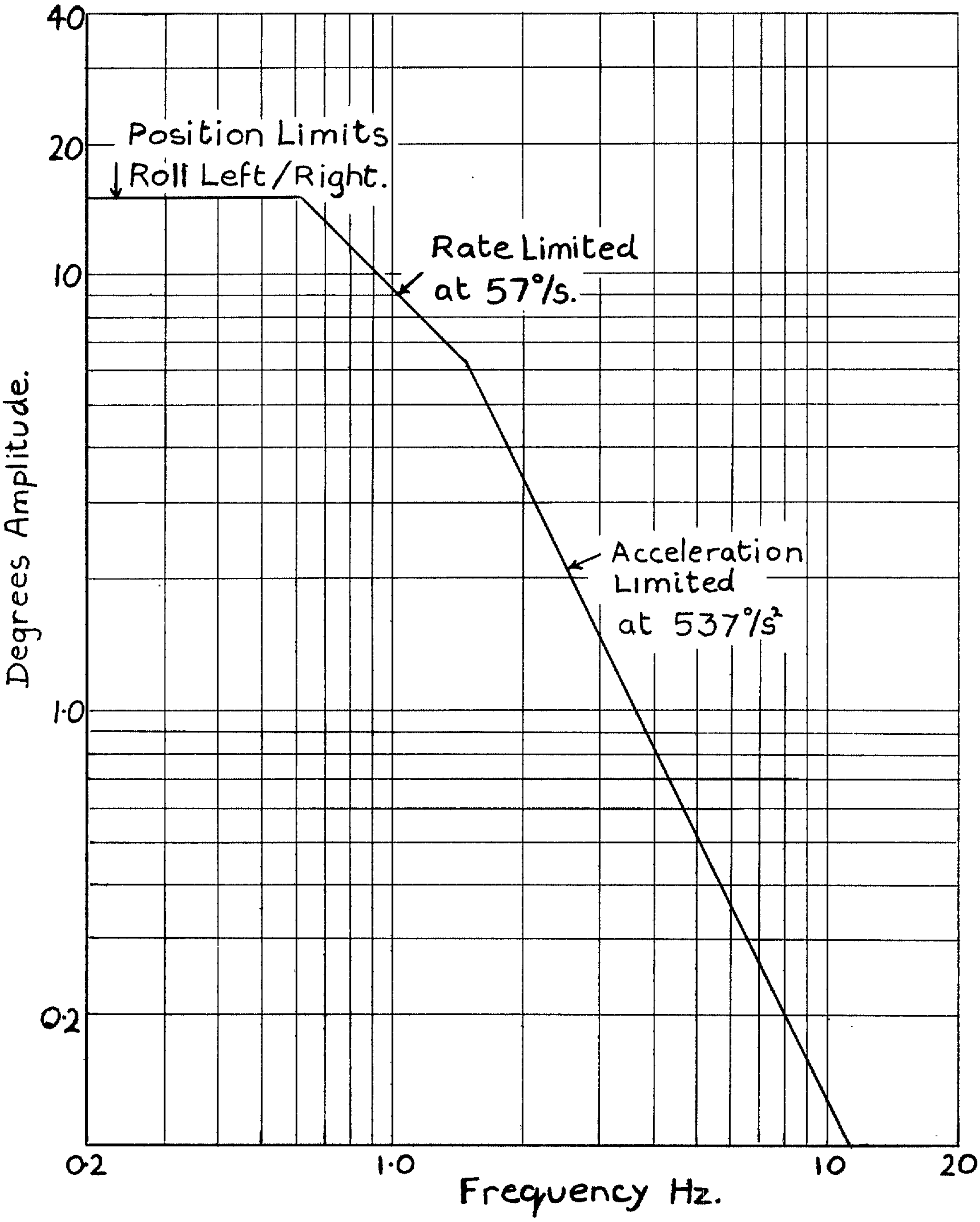

FIG. 20. Cockpit motion—Roll freedom servo. Maximum undistorted sinusoidal output.

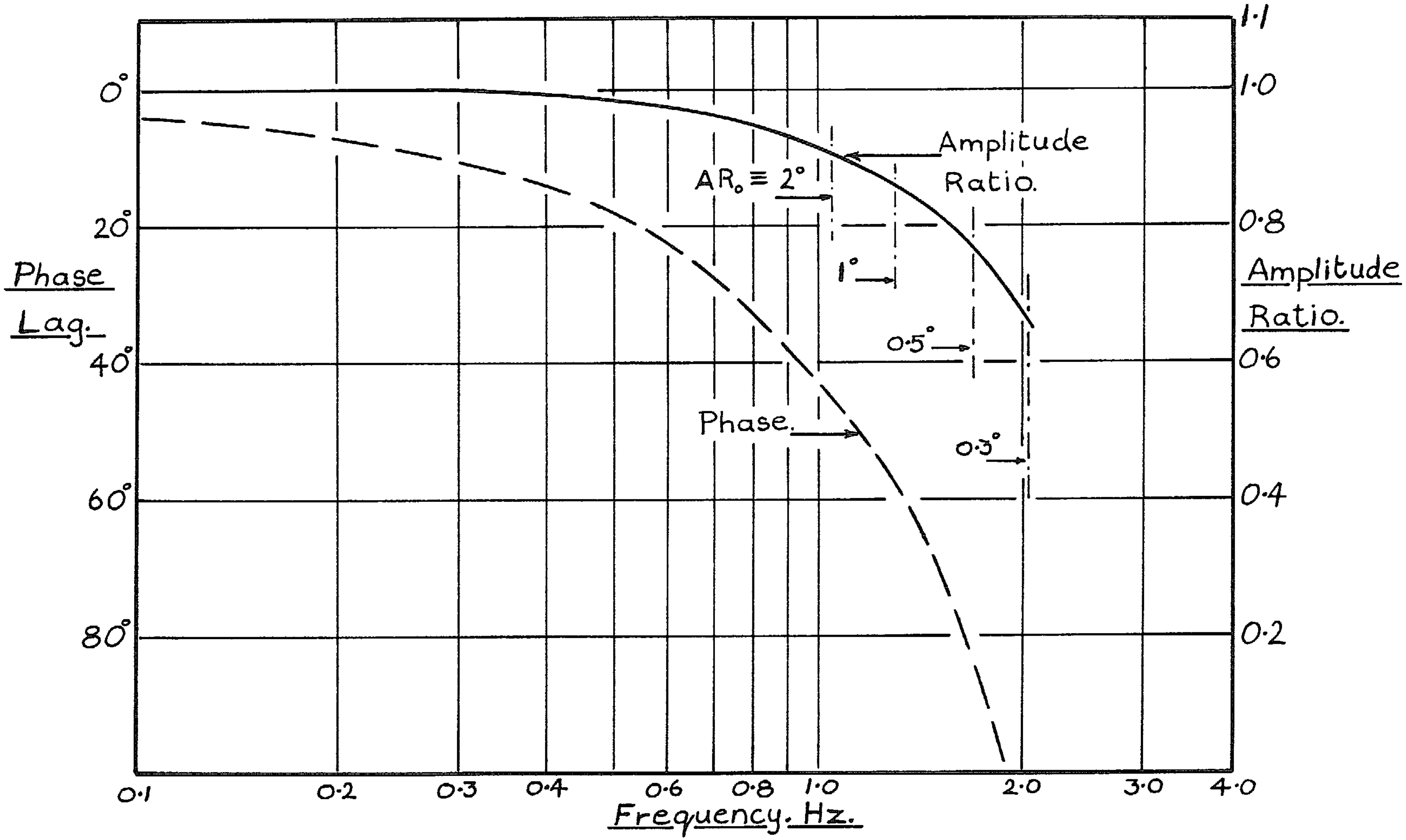

FIG. 21. Heave freedom servo: Frequency response as a position follower with damping increased by rate feedback.

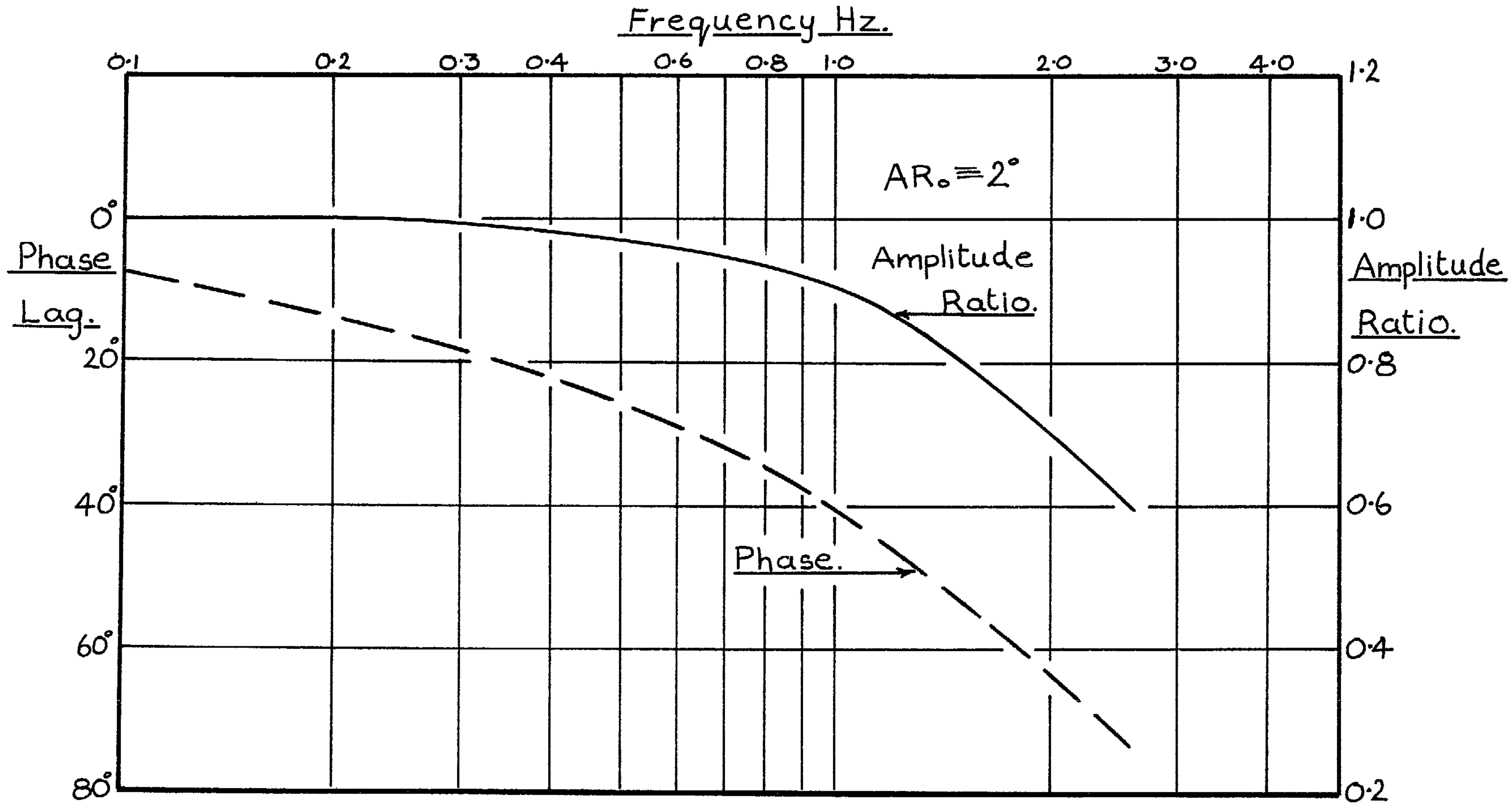

FIG. 22. Roll freedom servo: Frequency response as a position follower with damping increased by rate feedback.

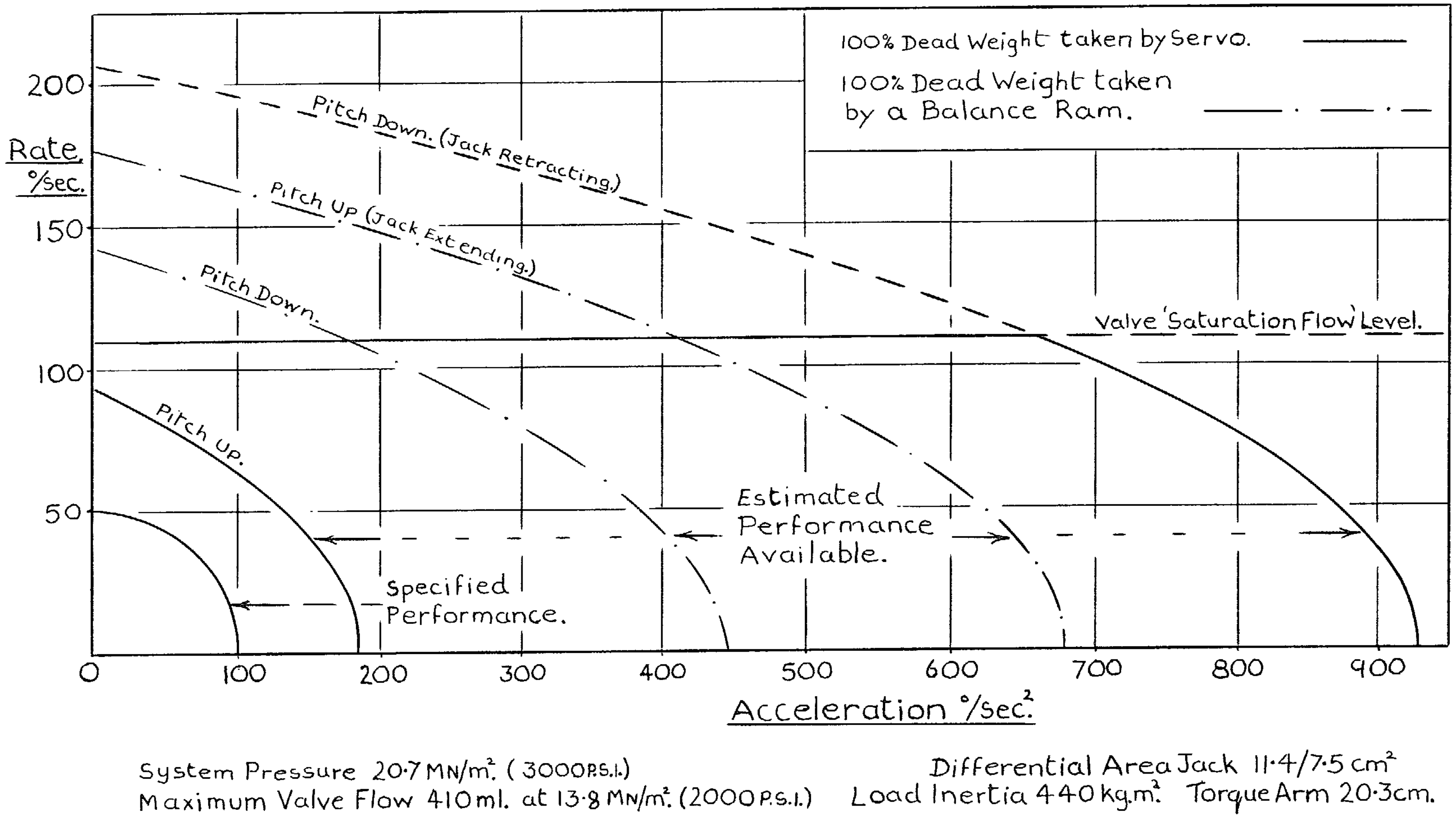

FIG. 23. Pitch freedom: Simultaneous rate/acceleration (power) available from the hydraulic drive (Design estimate)

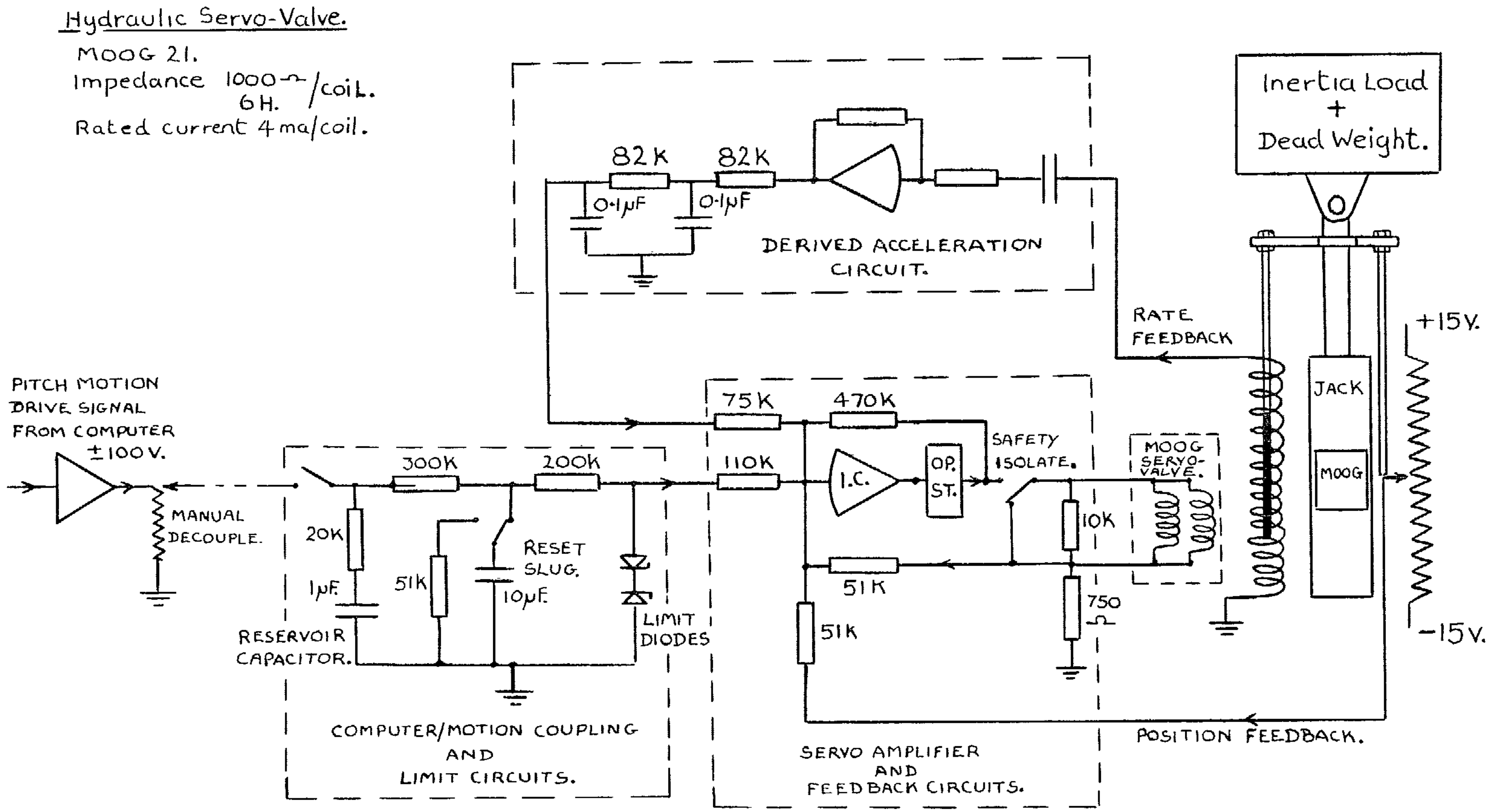

FIG. 24. Pitch freedom servo: Circuit arrangement.

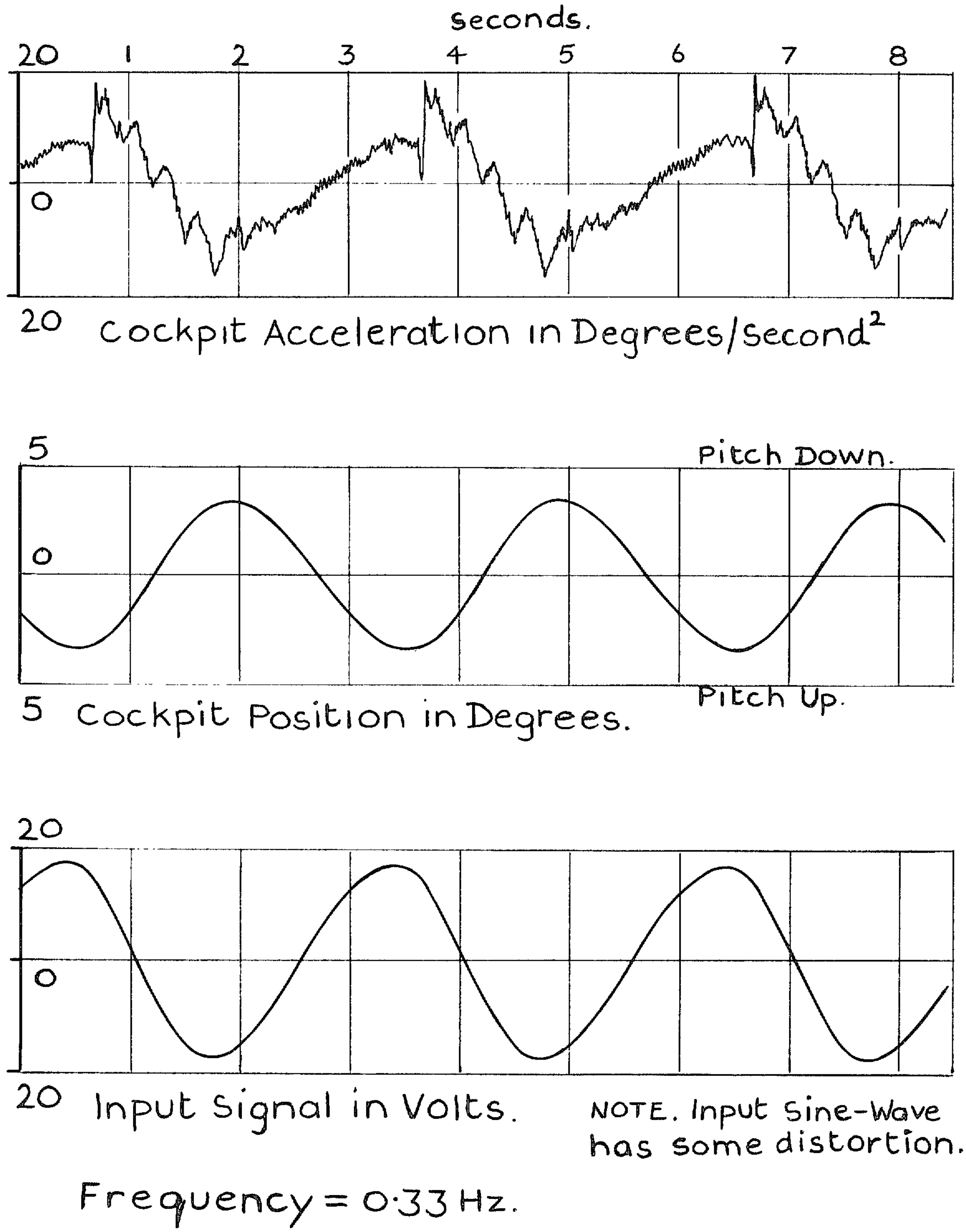

FIG. 25. Pitch motion quality: Response to sinusoidal input signal. Servo feedback loops—position only.

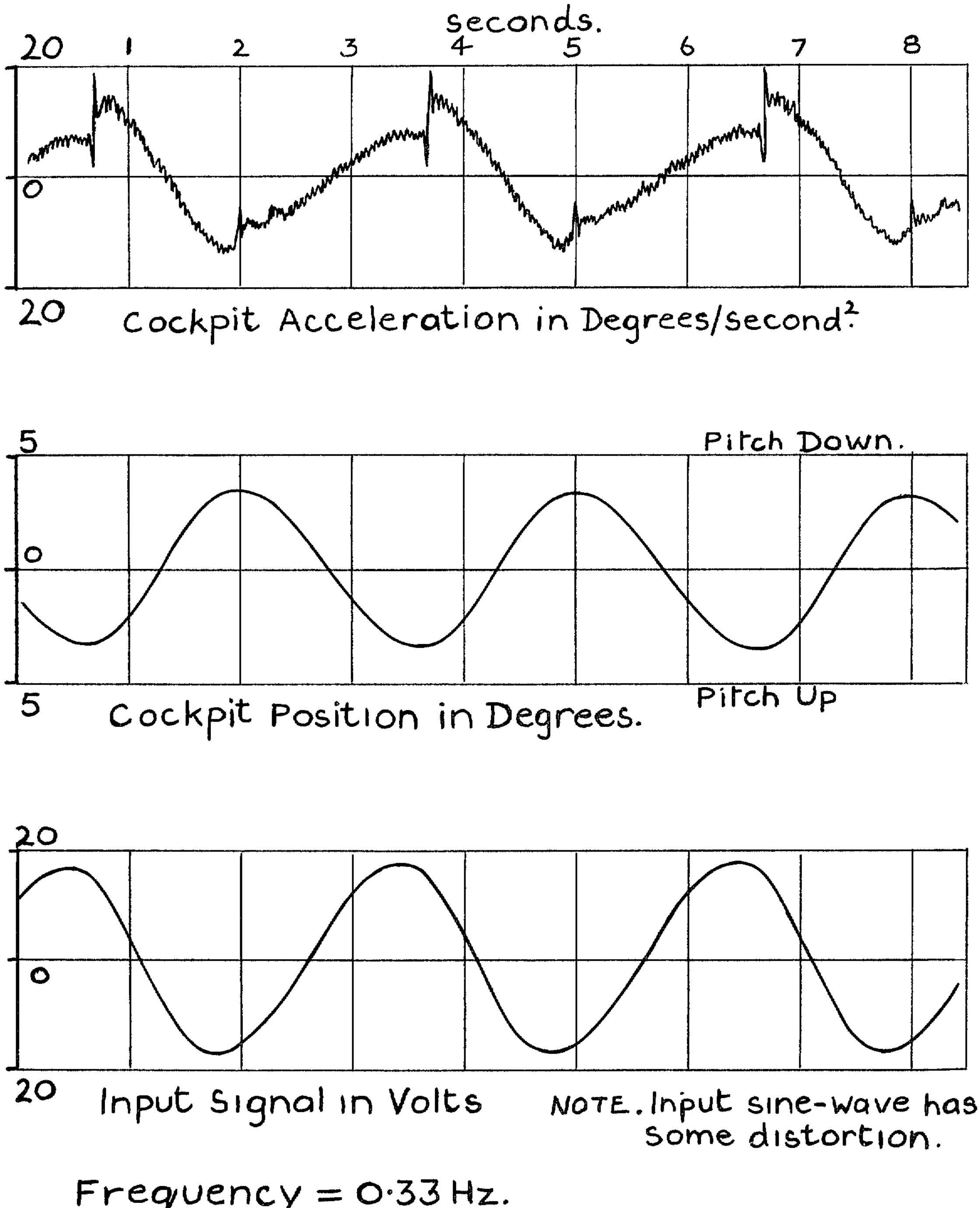

FIG. 26. Pitch motion quality: Response to sinusoidal input signal. Servo feedback loops—position + acceleration.

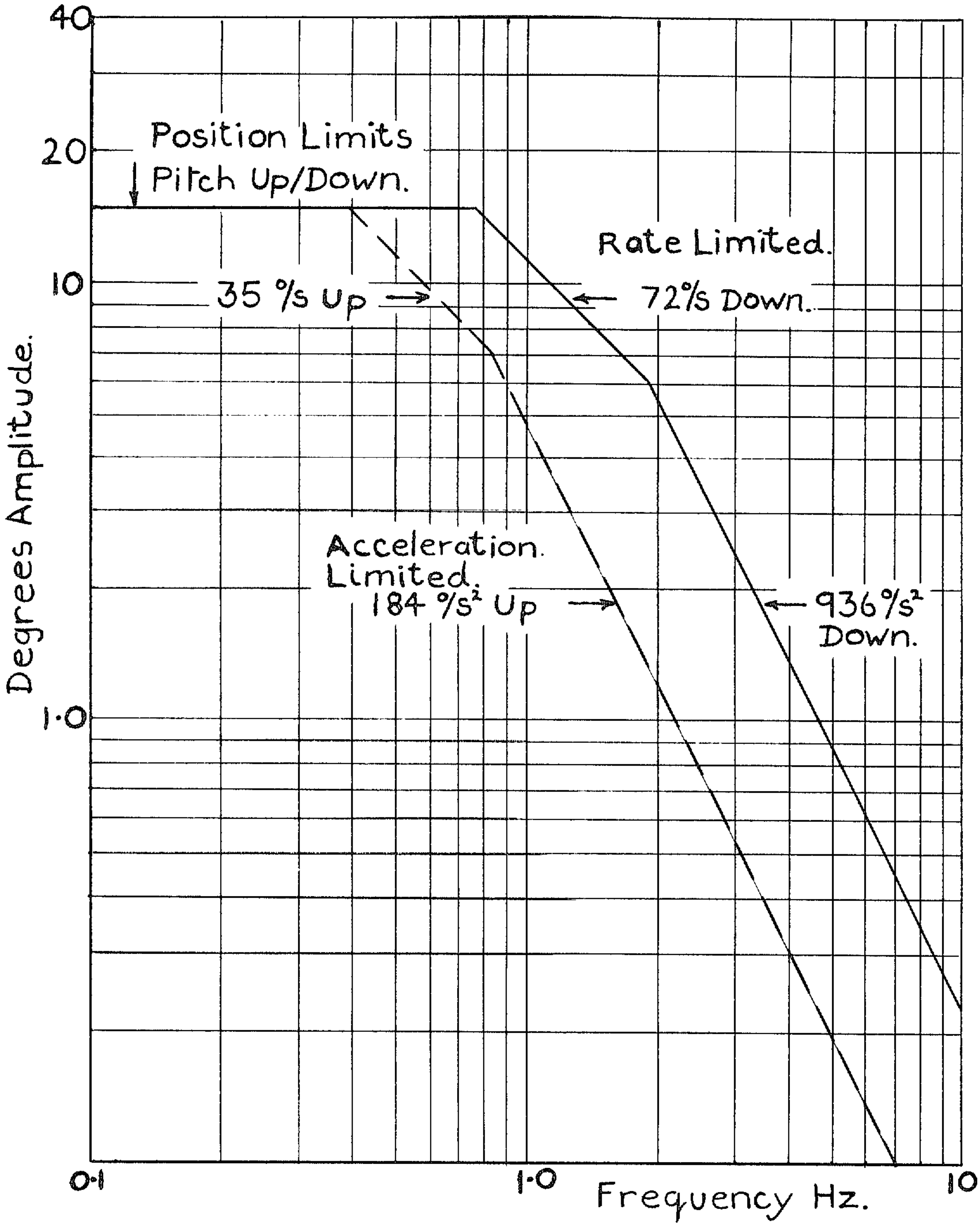

FIG. 27. Cockpit motion: Pitch freedom servo. Maximum undistorted sinusoidal output.

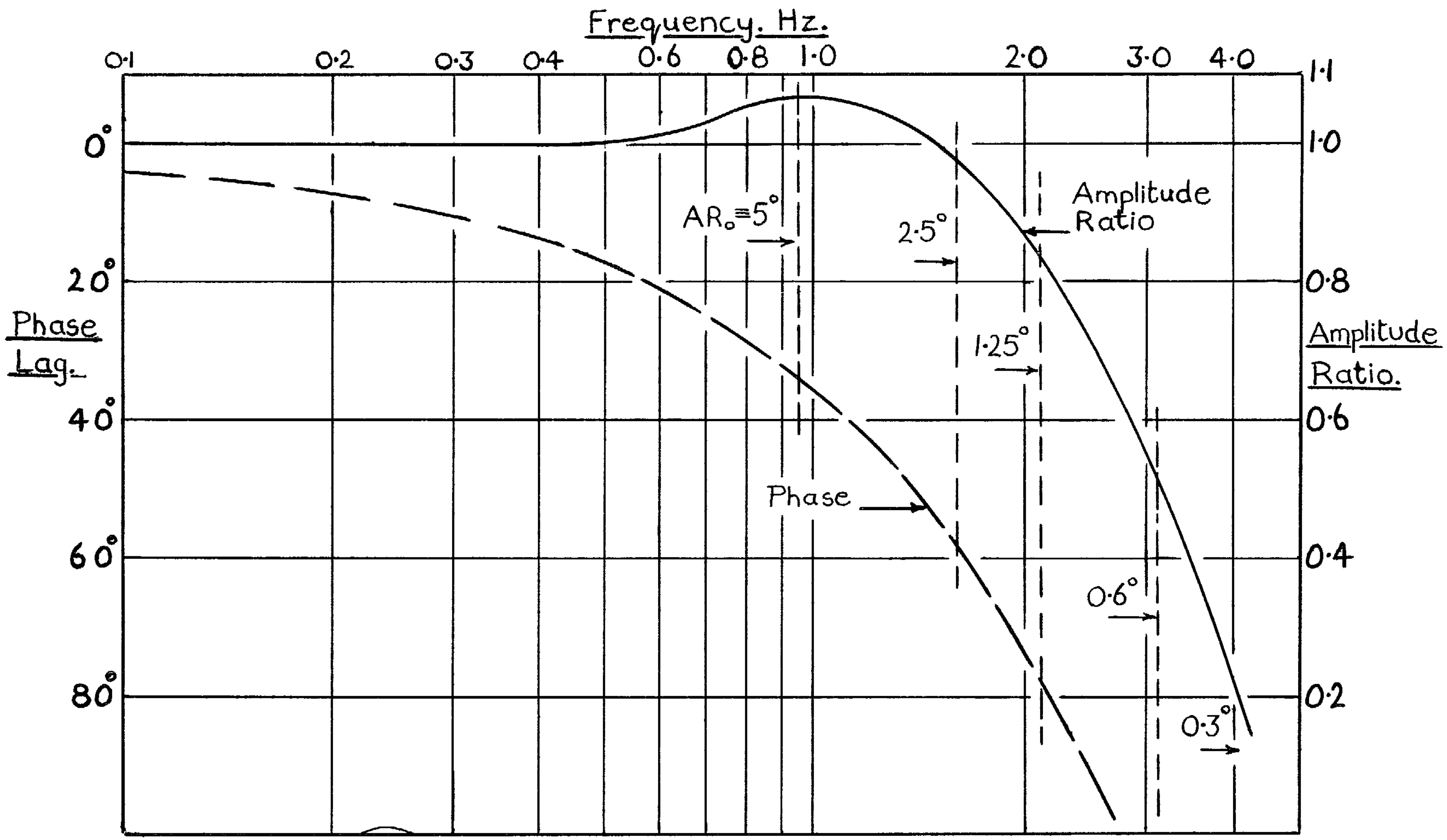

FIG. 28. Pitch freedom servo: Frequency response as a position follower with position + acceleration feedback loops.

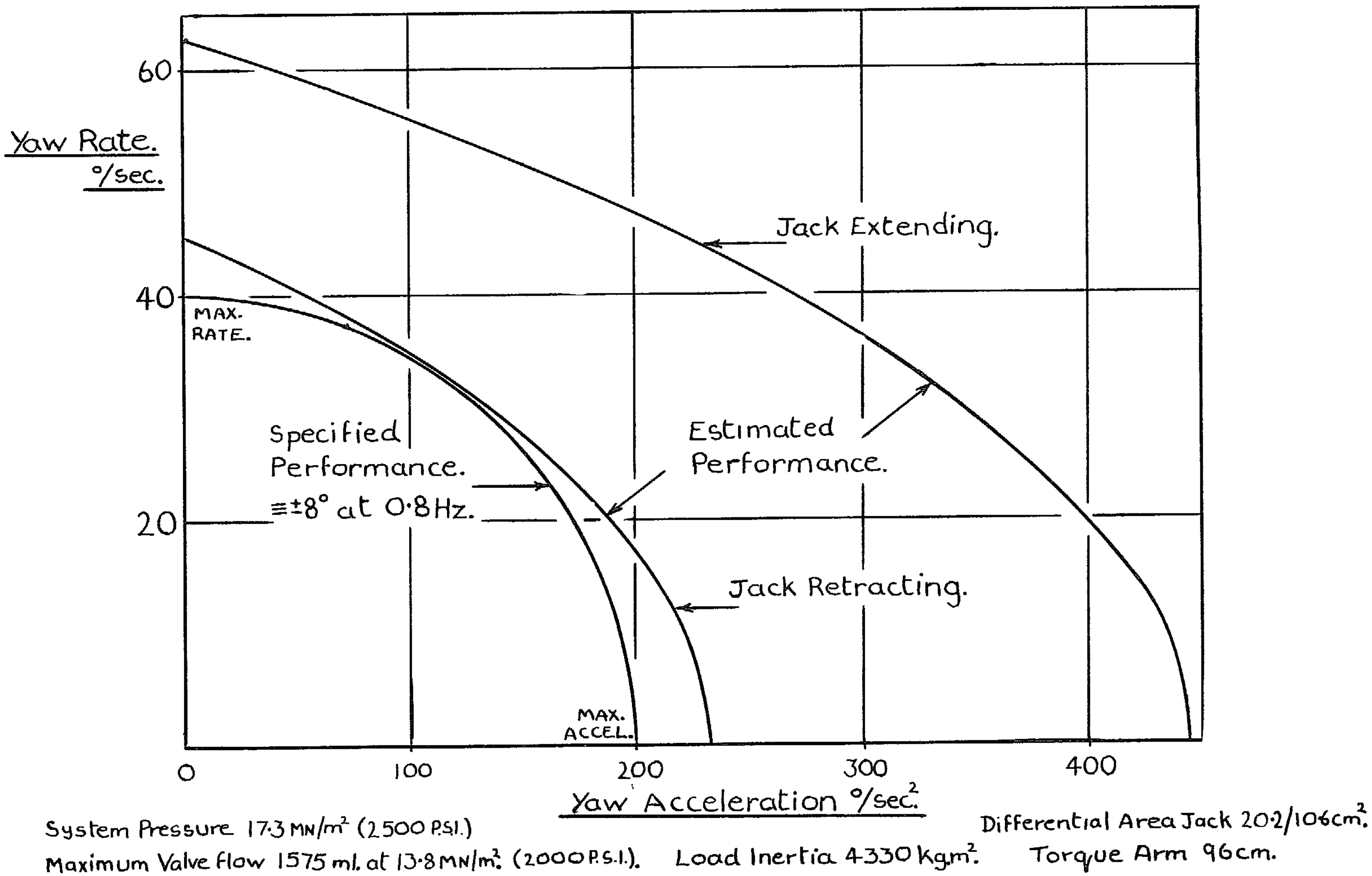

FIG. 29. Yaw freedom: Simultaneous rate/acceleration (power) available from the hydraulic drive (Design estimate).

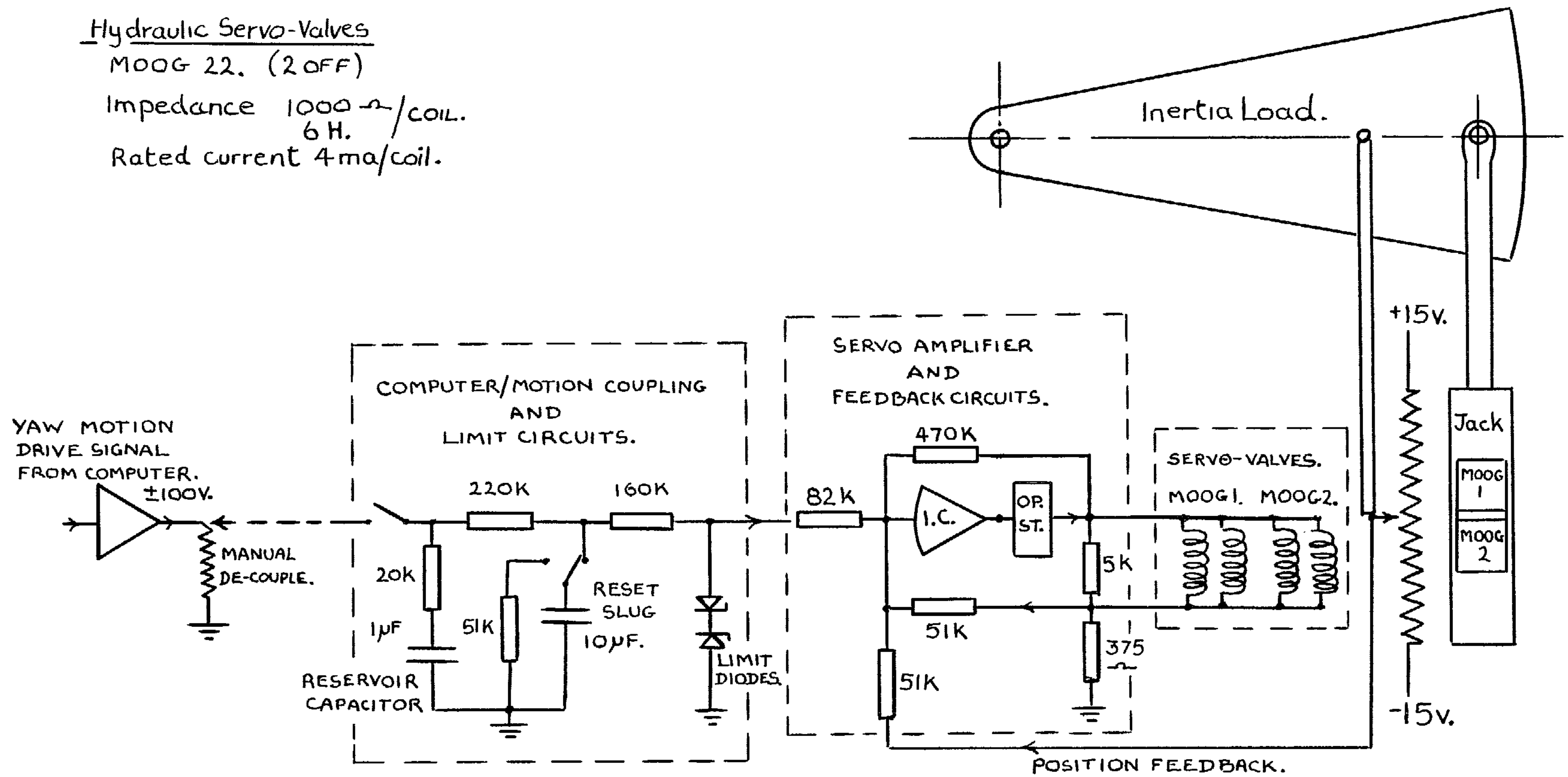

FIG. 30. Yaw freedom servo: Circuit arrangement.

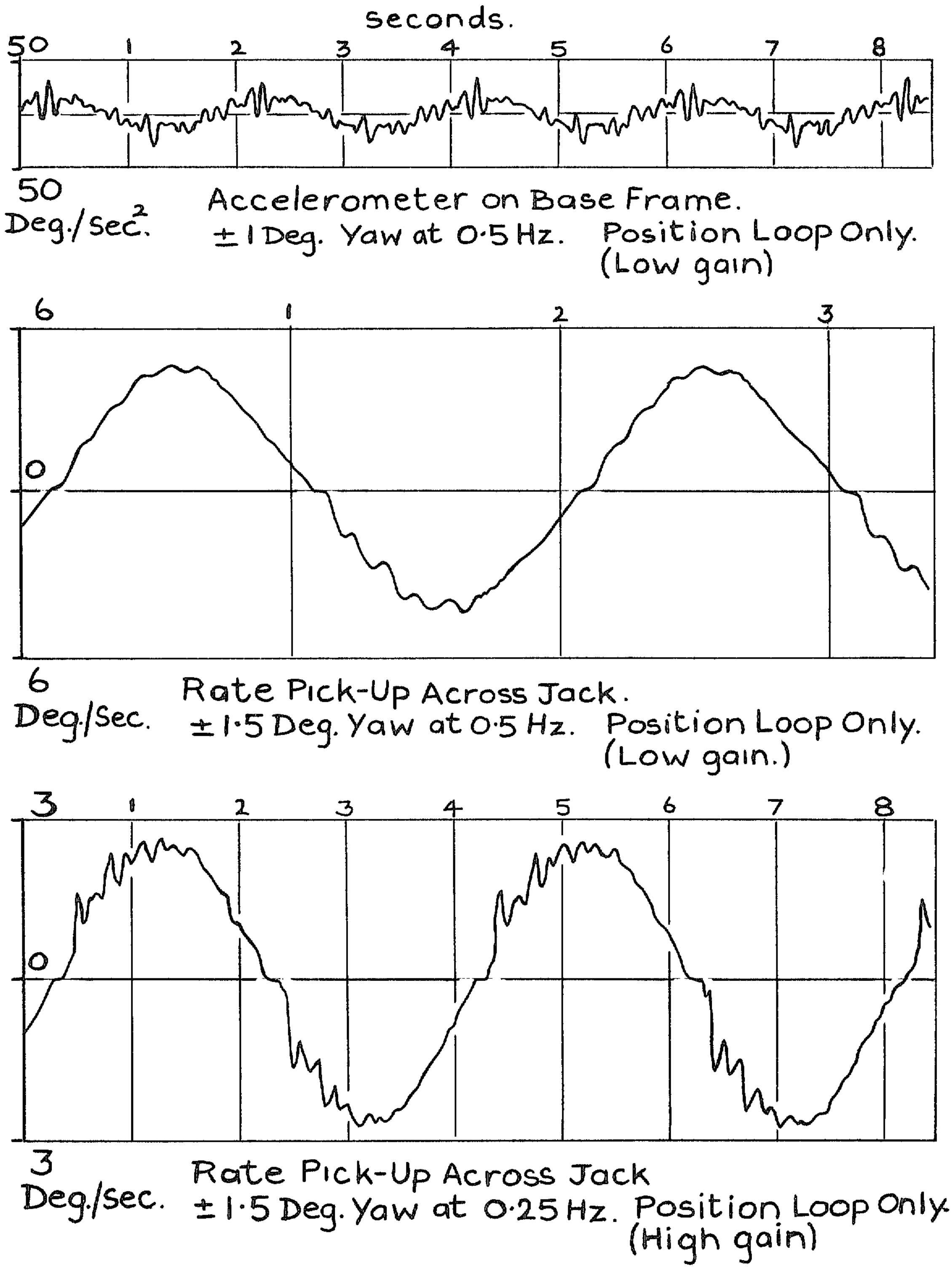

FIG. 31. Yaw motion quality: Response to sinusoidal input signal. Servo feedback loops—position only.

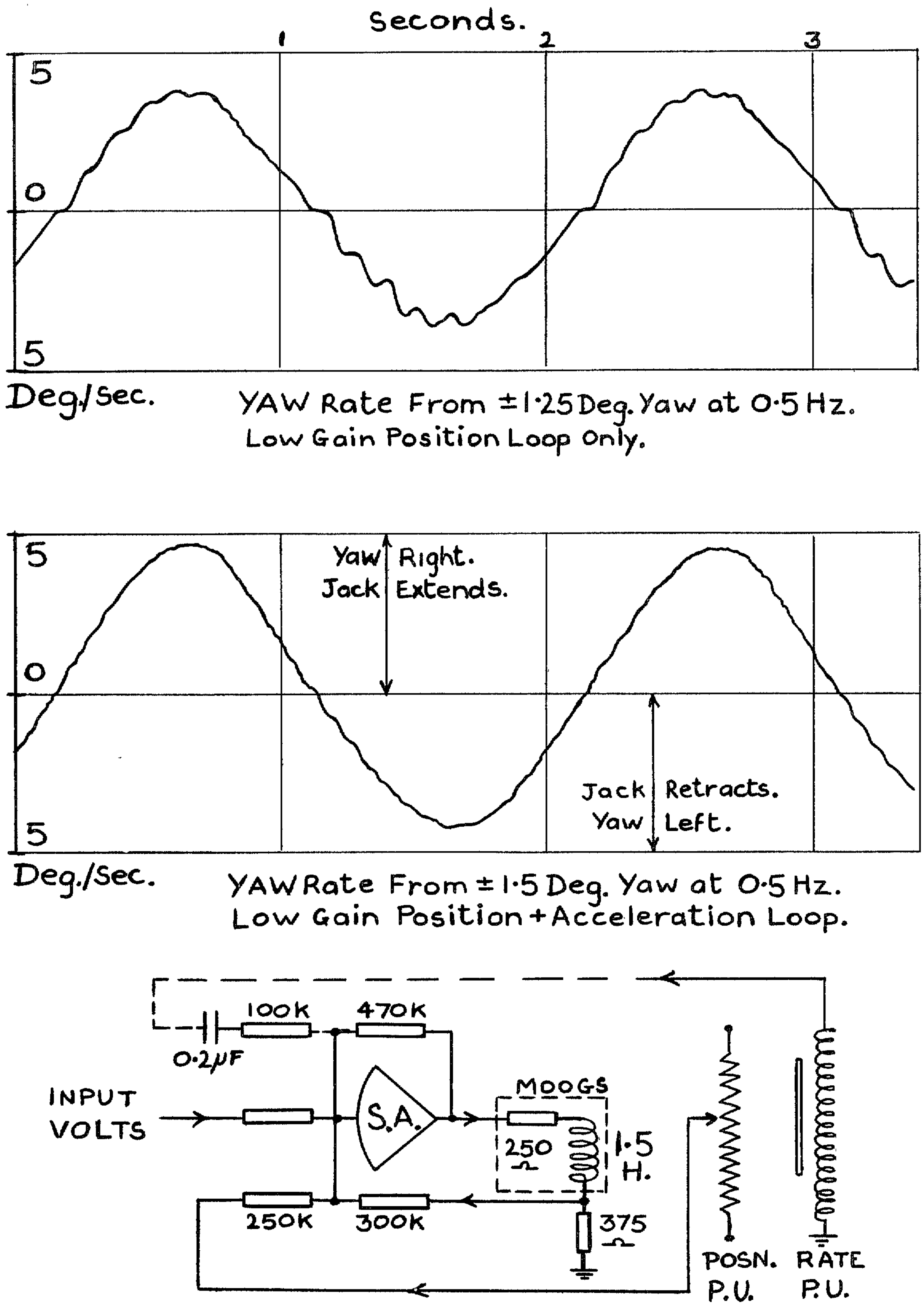

FIG. 32. Yaw motion quality: Improved waveform resulting from the use of position + acceleration feedback.

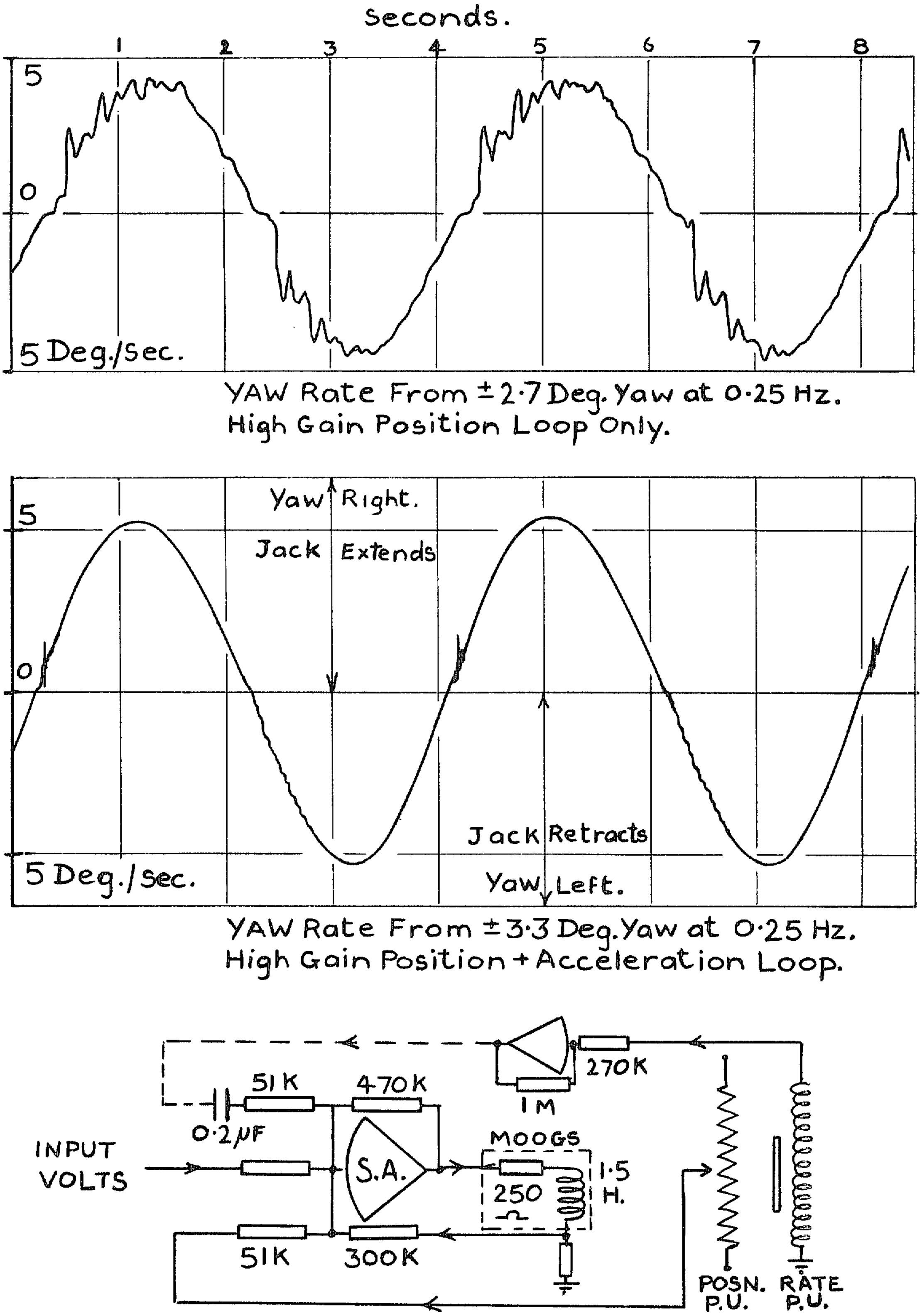

FIG. 33. Yaw motion quality: Improved waveform resulting from the use of position + acceleration feedback.

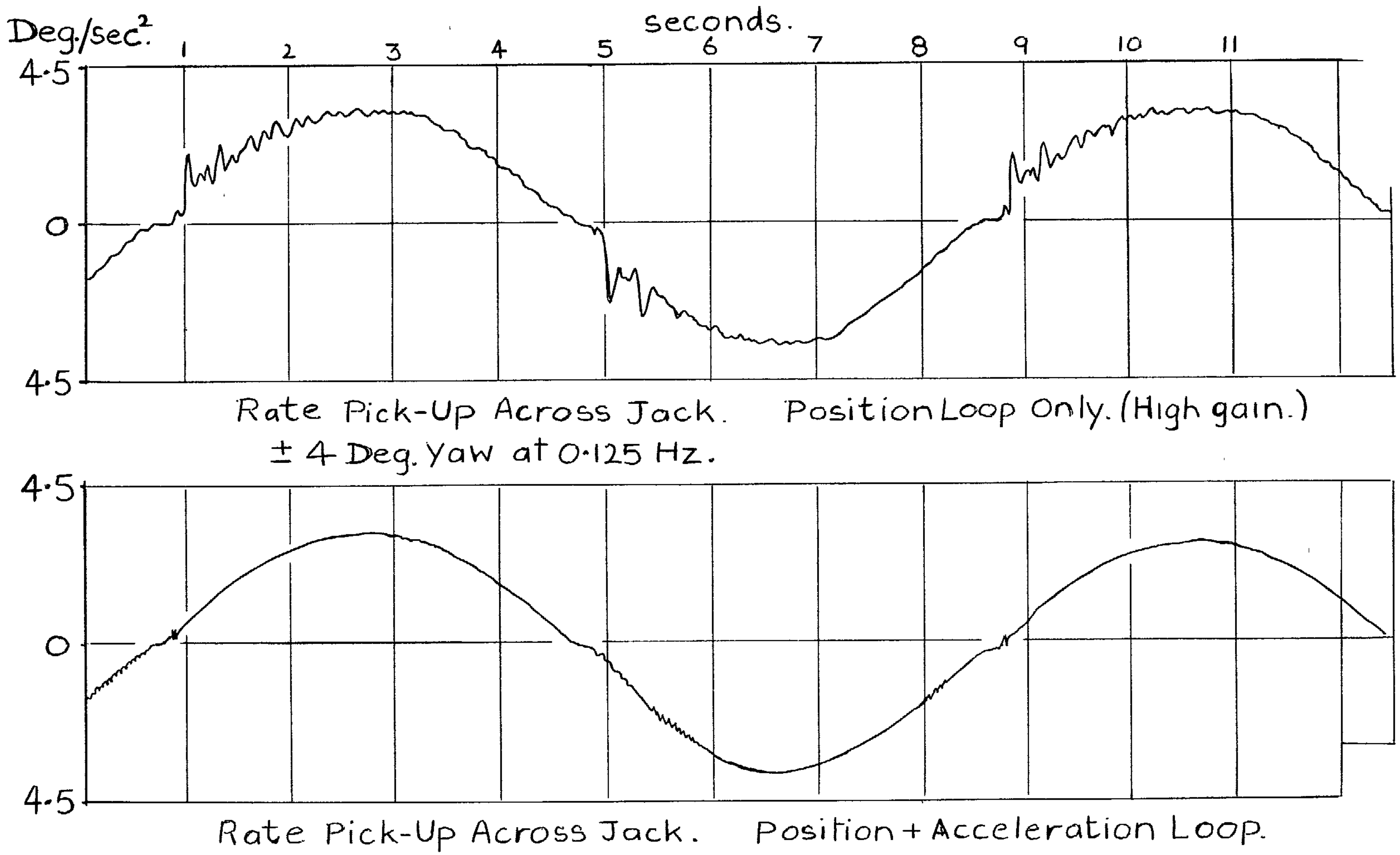

FIG. 34. Yaw motion quality: Improved waveform resulting from the use of high gain position + acceleration feedback. Response at low frequency.

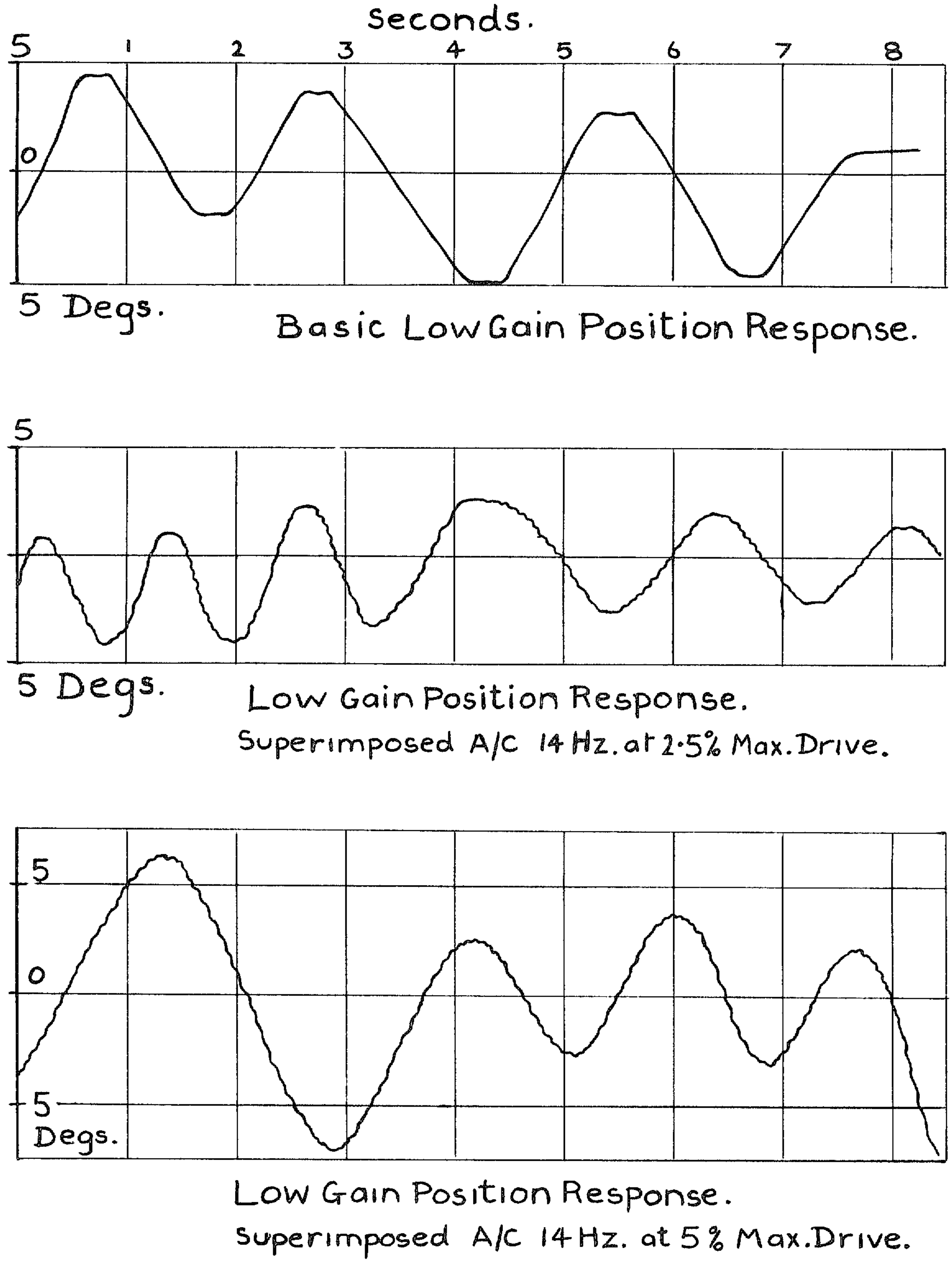

FIG. 35. Yaw motion response to smooth input demands: Low gain position loop with and without superimposed a.c. voltage.

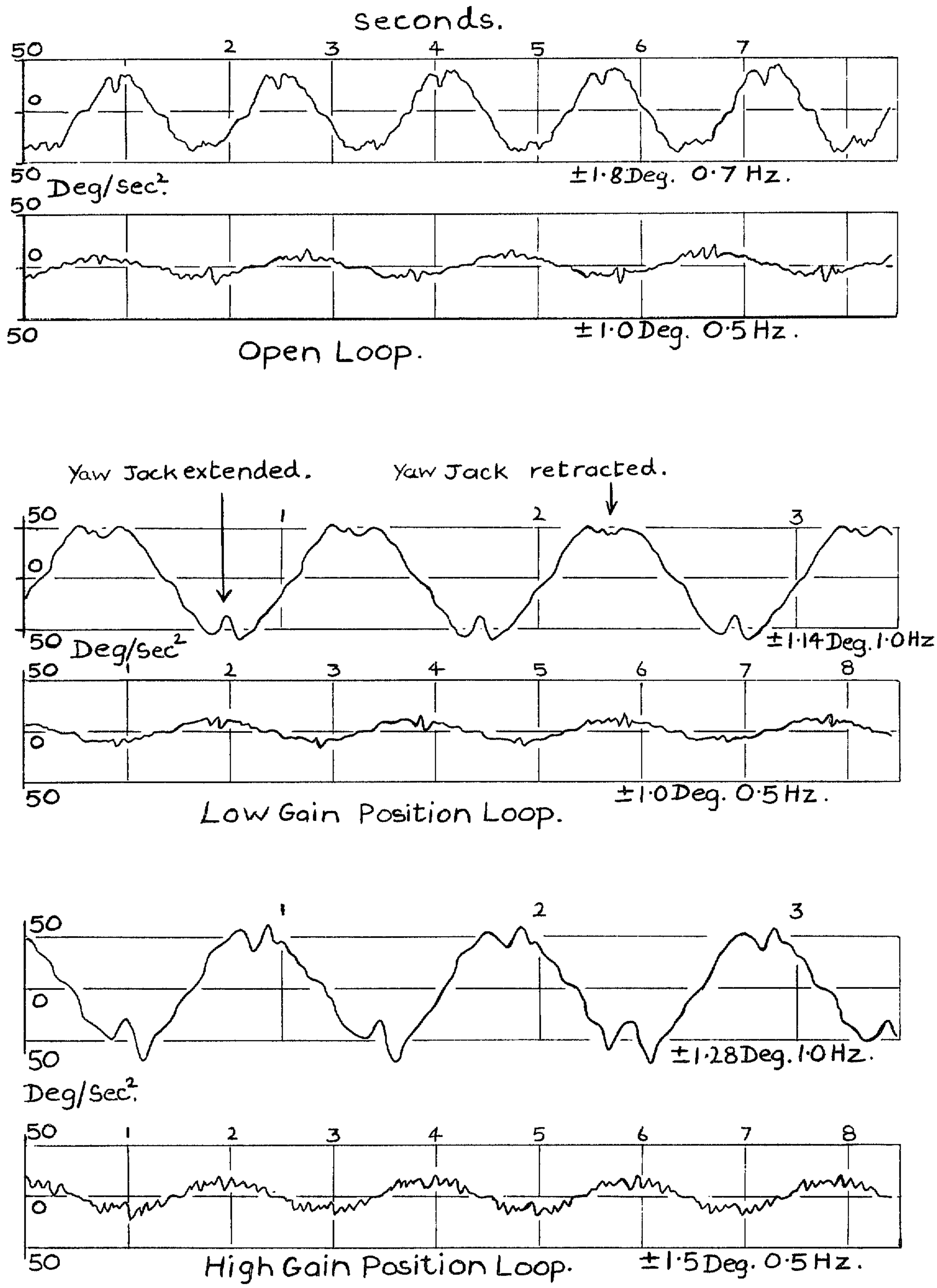

FIG. 36. Response of yaw servo jack in terms of equivalent cockpit acceleration. Jack servo detached from motion machine.

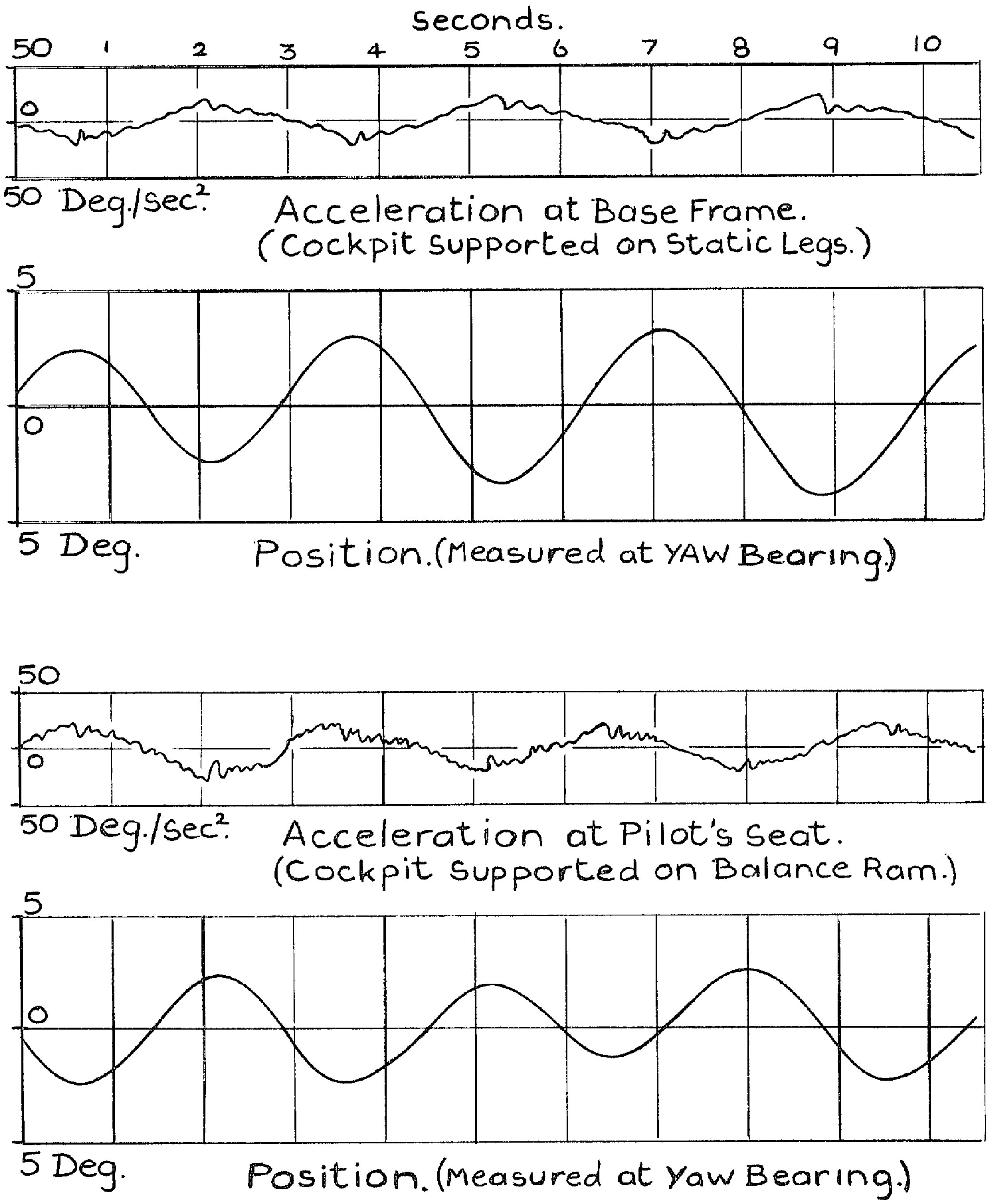

FIG. 37. Yaw motion quality: Response to smooth manual drive. Servo jack detached from the machine.

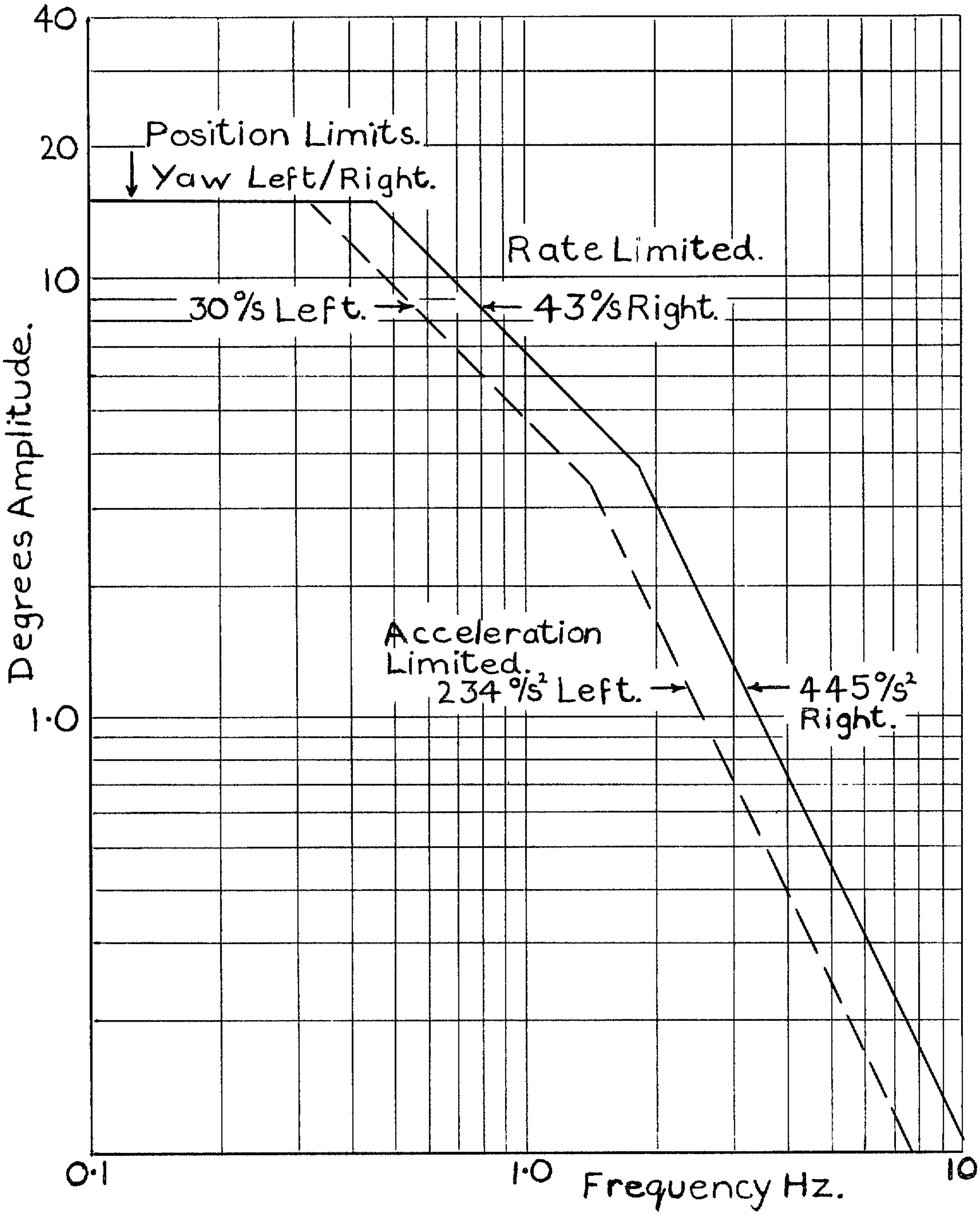

FIG. 38. Cockpit motion: Yaw freedom servo. Maximum undistorted sinusoidal output.

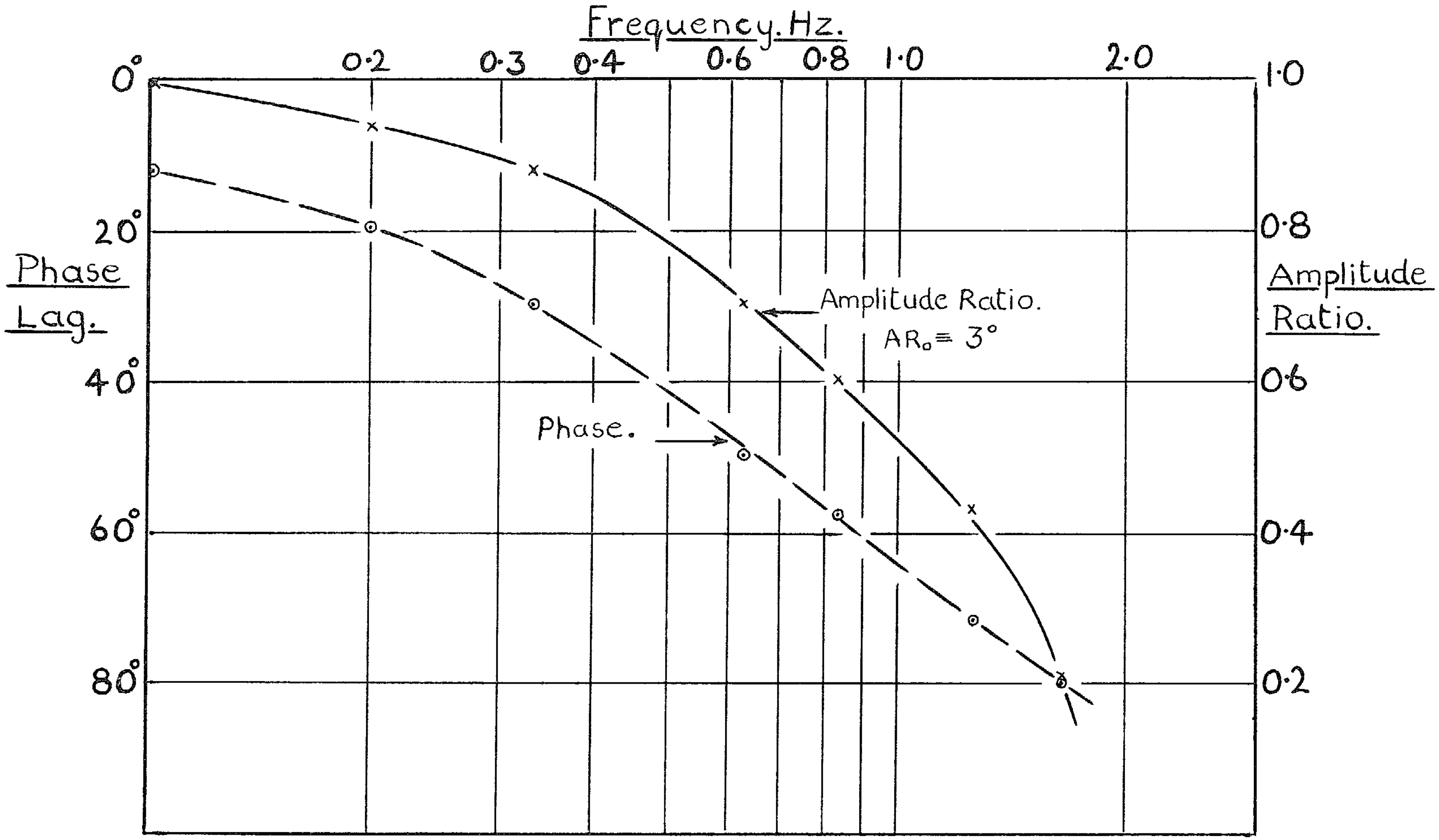

FIG. 39. Yaw freedom servo: Frequency response as a position follower with low gain position feedback loop only.

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