PS/FES/TN/349
7.12.1972

The Electro-Hydraulic Servo Actuator and Hydraulic System for the Serpukhov Ejection

S. Milner

This report describes the functions, principles and designs of the actuator, and the hydrostatic guiding mechanism for the intermediate mobile septum magnet of the Serpukhov ejection. Guide-lines for operation and maintenance are also given.

Contents :

		Page
1.	Introduction	1
2.	General	1
3.	Performance	3
4.	Stroke Definitions	4
5.	Actuator	5
6.	Accumulators	7
7.	Mechanical Feedback	8
8.	Velocity Stop	9
9.	Locking Device	9
10.	Servo-Valve	10
11.	Servo-System	11
12.	Construction	13
13.	Pump Group	15
14.	Fluid Power Transmission Lines	17
15.	Control and Interlock	17
16.	Start and Stop	19
17.	Magnet Guiding	26
18.	Check and Maintenance	28
19.	Acknowledgements	32
20.	References	33

1. Introduction

As part of the CERN-Serpukhov collaboration, a fast-ejection system[1] for the Serpukhov 70 GeV accelerator has been designed and manufactured at CERN.

This system consists of three ejection magnets with their pertaining vacuum tanks, high voltage pulse generators, electronic, beam diagnostic and various accessories.

The magnets are the following : one kicker magnet, one intermediate moving septum magnet and one stationary septum magnet.

The actuator and the hydraulic system displace in a programmed way, the intermediate septum magnet from its rest position or so-called "Base line position" into its "ejection position" at approximately 1 mm from the circulating beam. In this position, the beam, when kicked by the kicker magnet in straight section 16, jumps over the septum into the aperture of the intermediate mobile septum magnet in straight section 24. Its field provides for further deflection towards the stationary septum magnet in straight section 26. The latter magnet produces the final deflection and ejection of the beam into the beam transport line (see Fig. 1). After an ejection cycle, the magnet is withdrawn to its "base line" from where the next displacement takes place.

2. General

The actuator is situated in straight section 24 (see Fig. 2) and the hydraulic pump group supplying the hydraulic power is placed in the basement of the ejection building in the so-called pump station (PST). The (PST) is approximately 60 m from the local control room (LCR) and 200 m from the actuator in ss24. The hydraulic power is led to and from the actuator by means of steel tubes situated in a tube trench up to straight section 16, and then along the basement of the ring tunnel. The central control and interlocks are in the (PST); Remote control can be performed from the LCR.

The main parameters of the system are given in Table 1.

The displacements are controlled by an electronic programme generator[2] supplying an analogue program voltage to the input of the servo-amplifier which controls the input current to the first stage electrohydraulic servo-valve of the actuator (see Fig. 3). The programme is set on the "hydraulic programme selector" on which the strokes and the required times for the displacement are chosen. The start and withdrawal times are triggered by pre- and post B + T pulses from the general ejection timing system. The programme can be set to perform 3 full strokes with different timings within one acceleration cycle. Fig. 4 shows some possible movements. Fig. 4(a) shows the movement for a typical single shot operation where 250 ms are needed to move the magnet to its ejection position, 100 ms for keeping it in the ejection position and 250 ms to bring it back to the base line again. Fig. 4(b) shows a possible movement for multi-shot operation. In this type of operation it is in general not necessary to withdraw the magnet completely between the shots. However, as the beam position may be slightly different from shot to shot, the working position of the magnet is changed accordingly. For particular operations, it may be convenient to withdraw the magnet partially between the shots, thus allowing e.g. a short target burst in the intervals (Fig. 4(c)).

The septum magnet itself is clamped to the end of a hollow supporting shaft that is connected to the moving part (carriage) of the guiding system (see Fig. 5). The carriage is located outside the vacuum tank and is composed of two parallel accurately aligned guiding shafts fixed on a transversal on which the magnet shaft is also supported. Both guiding shafts slide in two hydrostatic bearings. The high voltage coaxial pulse connections and the water cooling tubes for the magnet, are fed through the hollow magnet shaft. Flexible striplines connect the rigid pulse transmission system and the coaxial pulse feed-through (see Fig. 6).

The actuator itself is fixed rigidly on a steel frame which also supports the hydrostatic guiding system. The frame is fixed on a supporting girder and can be horizontally aligned by means of a three-point screw adjustment. The supporting girder is rigidly anchored in a heavy concrete block cast into the underlying rock. The block is separated from the foundations of the main magnets so that shocks are not transmitted to the magnets and vacuum system.

3. Performance

A prototype actuator was thoroughly tested for its transfer functions, accuracy and long-term performance (200,000 cycles), concerning mechanical wear and position drift. The test showed that the actual behaviour was very near to the calculated behaviour and that the small derivations observed could be explained by the approximations made[5]. After these tests, the prototype was installed in the CERN PS where it actuates the "bare kicker" magnet in ss13; it has been in continuous operation there since 1971.

The final actuator was tested[6], less extensively, but its performance was compared with the one of the prototype. As the pressure variations in the supply pressure were smaller than those for the prototype, and the absolute supply pressure was 200 bar instead of the 150 bar, the measured accuracy of positioning was better. The error from cycle to cycle was 0,01 mm with a 500 kg dummy load and the long term drift, order of days, was less than $\pm$ 0,1 mm. Fig. 7 shows the step response of the actuator at different feed-back tunings without load and Fig. 8 shows the step response with 500 kg dummy load at different feedback tuning and step sizes. Fig. 9 shows the actual programmed movement with magnet and hydrostatic guiding systems connected.

Fig. 10 shows the damping effect of the accleration feedback H4. The trial with this feedback was made in order to determine the best possible performance of the actuator. However, this feedback was not installed in the final system, as the system would then be more complex and less easy to adjust. Also the extreme response time and accuracy obtainable with this feedback was not necessary for the operation of the actuator as the relative soft input programme (a sine function) did not excite the higher order oscillations.

4. Stroke Definitions

The main definitions concerning the stroke of the actuator are given in Fig. 11. The theoretical beam position is the centre of the accelerator aperture, but as the beam position in the moment of ejection is dependent on the requirement of the accelerator operation, the beam will be within a certain range from the theoretical central orbit. This range ± 30 mm from the central orbit, is called the "ejection range" and is one of the basic parameters for the design of the actuator. The tolerances on the actuator design permits the range of be extended to ± 40 mm.

The "base line" is the starting and return position of the front of the septum, before and after an ejection cycle. This position 110 mm from the central orbit is well outside the interference range of the beam at injection, so as to minimize the influence on the beam from the weak remenant field of the magnet, and to keep the actuator dimensions within reasonable limits. The program range of 135 mm is situated between the baseline position and the extreme ejection position. By manual control, the max. stroke of 160 mm is limited at two chosen points "servo-stops", 20 mm from the dash pots, where a mechanical feedback overrides the electrical feedbacks independent of a possible excessive electrical input signal. The locking of the actuator takes place in the servo-stop (out) position. The physical stroke of the piston extends 20 mm beyond both servo-stop positions. These parts of the stroke can only be reached under fault conditions. The last 15 mm between the servo-stops and the really physical stops on the bearings are called "dash pots" owing the their damping action. The indication and interlock for the locking position is activated by the locking and servo-stop switch 2 mm before reaching the servo-stop. The interlock action of this switch in manual operation, where the actuator is brought back to the servo-stop position (out) for the locking operation, is bypassed by a discriminator when the input signal corresponds to an absolute stroke of 25 mm from the end of the dash pot (out).

5. Actuator

The choice of an electrohydraulic actuator was determined by the requirements for high acceleration of the large moving mass, fast movement and accurate positioning. The system combines the inherent flexibility of electrical components together with the high power gain and reliability of hydraulic power amplifiers. It is the most compact device for its power output, as compared with electrical, pneumatic or mechanical systems.

The principle of the actuator is shown in Fig. 12 and an engineering cut is given in Fig. 13.

The actuator is a three-stage hydraulic amplifier preceded by an electric torque motor. The two first stages of the hydraulic amplifications are provided by the servo-valve and the third stage is in the distributor valve. The feedbacks for the spool and ram displacements provide for stability and proportionality between input and output of the actuator.

The servo-amplifier obtains its input from the program generator and supplies a d.c. current to the servo-valve proportional to the program voltage (zero current corresponds to the middle position of the servo-valve spool). The servo valve supplies an oil flow Q to the spool piston proportional to the amplifier current and causes the distributor spool to move. The controlling orifices of the spool, representing variable restricted passages in the two oil circuits between supply and the return ports, then open and change the flow resistance proportionally to the spool displacement (simplified approximation - see Ref. 8 for more complete relationship). The flow passing through each orifice depends on the pressure drop across them as well as on their size and the acceleration of the ram and compressibility of the oil columns in the ram cylinders. Actually, there can be a small flow into the compartment without movement, for example, during peak acceleration, when the oil compresses until the required acceleration pressure is reached. For simplification here, it is assumed that the ram movement is only governed by the amount of oil Q_1, Q_2, Q_3 and Q_4 flowing into and out of the cylinder compartment, the compressibility thus only being of importance for the calculation of the stability.

In the stationary state, (zero velocity of the ram) oil flow in the two circuits passing through the supply and return ports, are equal, i.e. $Q_1 = Q_2 = Q_3 = Q_4$, and the distributor is said to be in the zero position.

By forward movement of the ram (y positive), the distributor spool is displaced in the opposite direction (x positive). Herewith the orifices (2) and (4) become larger and (1) and (3) smaller, Q_2 and Q_4 will increase, and Q_1 and Q_3 decrease. As a consequence, the oil volume in compartment (I) will increase and the volume in compartment (II) will decrease. This has the result that the ram is pushed in a forward direction (y positive). In the reverse case, the orifices (1) and (3) become larger and (2) and (4) smaller, so that the ram is pushed backwards (negative direction).

The spool and ram move at uniform velocities at step inputs and the velocity is proportional to the magnitude of the step, respectively : the electrical input or servo-valve spool position for the velocity of the distributor spool, and the distributor spool position for the ram velocity. To provide proportional action, the spool and ram displacement are measured by feedback transducers (19) and (15) and fed back via attenuation potentiometers (H2) and (H3) to the summing point. If any difference exists between input and feedback signals, the resulting difference or error signal is amplified and the output current is fed to the servo-valve. The output current is determined by the magnitude and polarity of the error signal.

Assuming stationary conditions for the actuator, i.e. the sum of the feedback signals is equal to the input signal, but with the opposite polarity, and the spool of the servo-valve and the distributor spool are in their zero position, an instant applied input signal would create an error signal equal to the change of the input signal. The servo-valve spool would follow the error signal in a proportional way until saturation, with a small time lag, and would supply a constant flow rate for displacement of the distributor spool. However, as the distributor spool moves, the feedback of the spool tends to decrease the difference between input and the feedback signals, thus decreasing the error signal and flow rate of the servo-valve. As the distributor spool is displaced from its zero position, a certain flow rate is supplied to the ram cylinder and moves the ram until the ram position feedback corresponds to the input signal. The ram is then in its desired position with its velocity zero and the distributor spool in the zero position with corresponding zero feedback signal. The above behaviour is only valid for slow feedback tunings where the transients for the spool and ram are aperiodic. For fast feedback tunings where non-linearity and overshoots are inherent to the system during the transient time, the behaviour is more complex (see Fig. 8 and Refs. 4, 5).

6. Accumulators

The operation of the actuator is characterized by large demands of energy over short time intervals, requiring corresponding large oil flow surges at constant supply and return pressure. Therefore, in order to avoid shocks in the long transmission lines, due to the rapid change of oil velocity and corresponding high oil column acceleration, two accumulators (21) and (22), large enough to supply and absorb the peak power, have been installed directly on the housing of the actuator. This provides for short and large section oil connection, assuring small pressure drops and small masses of the accelerating oil columns. The supply accumulator (21) has an energy reserve large enough, when the piston is half its maximum stroke, to supply the peak demand. It is recharged during the longer intervals between signals when the power demand is below average.

The return accumulator (22) stores the exhausted oil for a short time and lets it flow smoothly over the orifice and pressure regulator back to the return line. The pressure regulator maintains a certain maximum back pressure that prevents the accumulator piston from hitting the upper seat.

Two larger accumulators have been installed at each end of the long high pressure transmission line for damping of the remaining oscillations due to the pulsating operation of the actuator. In the return line they are not needed as the flow surges are smoothed sufficiently by the return orifice and pressure regulator.

A magnetic position switch for interlock indication of the piston position is installed at the top of each accumulator. This switch comes into action when the piston comes up too high, due to high oil pressure or leakage of nitrogen.

Pressure setting of the back pressure regulator is given in Table IV, and the nitrogen pressures are given in Table III.

7. Mechanical Feedback

A mechanical feedback linkage (see Fig. 12) protects the actuator, magnet and supporting structure against uncontrolled shocks due to wrong manipulations or an oversteered amplifier, as may result from a bad contact in a feedback cable.

In the normal working range, this feedback linkage is out of function, but it becomes a direct mechanical feedback when the ram comes to the servo-stop (14) position, defined to be 20 mm from the end of dash pot (see Fig. 11). It then overrides the electrical feedbacks and brings the magnet to a gradual deceleration and limitation of the ram stroke, defined by the servo-stop position, before the ram piston reaches the rigid mechanical stop in the dash pot.

When the spool is steered to the left from its zero position, the ram moves to the right. Here the top of the lever (28) is drawn to the right by a pivot (11) and (28) pivots around (17), and the roller (13) moves to the left. When (13) touches the so-called servo-stop (14), it stops its lefwards motion and the pivot point (17) is drawn to the right by the ram. Then, the spool is drawn to the right through the connection linkage (29) to its initial position and closes the distributor ports in a gradual way, well defined by the linkage ratio and the stroke of the ram.

8. Velocity Stop

As the velocity of the actuating ram is proportional to the spool stroke, adjustment of the maximum velocity can be done by limitation of the latter. This is done by means of the two stops (12), which can be adjusted to give between 0 and 10 mm spool stroke. This corresponds to an unloaded ram velocity between 0 and 10 m/s. The maximum velocity range is used during test periods to allow large step inputs corresponding to the full spool stroke. Narrower limits can be set for a standard mode of operation, so as to reduce shocks in case of malfunctions.

9. Locking Device

When the hydraulic system is stopped and without pressure, the magnet would be drawn into the accelerator aperture by the forces created by the pressure difference over the vacuum sealing bellows. So, in case of an interlock stop, the actuator withdraws automatically to its servo-stop position and automatically locks by the action of the servo-stop switch, with the magnet well outside the injection beam envelope. The system is locked manually after a normal stop. The locking device consists of two pneumatic cylinders, one with a fork engaging in slots on the traversal of the magnet carriage of the magnet, the second as a safety latch on the first (see Fig. 18).

10. Servo-Valve

The servo-valve (which is Moog valve) (see Fig. 12) consists of a polarized electrical torque motor and two stages of hydraulic power amplifications. The polarizing magnetic flux is generated by two alnico permanent magnets (2), arranged in parallel, between upper and lower polepieces. The motor armature (1) extends into the air gaps of the magnetic flux circuit and is supported in this position by a flexure tube member (6). The flexure acts also as a seal between the electromagnetic and hydraulic section of the valve. Two motor coils surround the armature, one located on each side of the flexure tube. The flapper of the first stage hydraulic amplifier is rigidly attached to the midpoint of the armature. The flapper extends through the flexure tube and passes between two nozzles (7) creating two variable orifices between the nozzle tips and the flapper. Filtered fluid from the pressure source is supplied to the two variable orifices through two fixed upstream orifices (3). The pressure developed in the intermediate chambers between the fixed and the variable orifices is applied to the end of the second stage spool. The servo-valve spool is, like the distributor spool, a conventional four-way sliding spool , in which the output flow from the valve is proportional to spool displacement from the zero position, at a fixed valve pressure drop. A cantilever feedback spring is fixed to the armature and extends through the flapper to engage a slot at the centre of the spiral. Displacement of the spool deflects the feedback spring which creates a torque on the armature-flapper assembly.

As the input signal is applied to the motor coils, a torque is developed on the armature, causing it to pivot around the flexure tube support. The resulting motion of the flapper increases the size of one nozzle and decreases the size of the other. This unbalance between the variable orifices produces a differential pressure which causes spool displacement. As the spool moves, a torque, proportional to spool displacement, is applied to the armature by the feedback spring. This torque opposes that developed by the motor and a condition of torque equilibrium will exist when the feedback spring torque equals the electrical motor torque. The position of the spool and herewith the size of the orifices are then proportional to the input signal. With constant pressure drops over the orifices without load and a constant input, the flow rate from the servo-valve will be constant and hence spool velocity constant. The flow rate will increase proportionally with the input size.

11. Servo-System

The block diagram of the servo-system is shown in Fig. 15. Detailed calculations of the dynamic behaviour and transfer functions have been given elsewhere (see Refs. 3, 4 and 8). The results are summarized here.

The electrohydraulic servo-valve behaves in good approximation as a second order system. The parameters are adjusted and published by the manufacturer (Table II). The displacement of the actuator with respect to the distributor flow is governed by a third order transfer function. For slow movements, the ram velocity is proportional to the spool position, yielding a pure integration. However, for higher frequencies, the moving mass together with the fluid compressibility and cylinder wall flexibility introduces an additional second order function. The same holds true for the spool with respect to the servo-valve flow, but with higher proper frequencies. In the linear approximation, i.e. for small stroke amplitudes, one obtains the closed loop transfer function.

$$G = \frac{y}{x} = \frac{K.G_1G_2G_3}{1 + K.G_1G_2(H_2 + H_3G_3)}$$

$$\text{width} = G_1 = \frac{a_1}{1 + a_2.s + a_3.s^2} \quad ;$$

$$G_2 = \frac{a_4}{a_5.s + a_6.s^2 + a_7.s^3}$$

$$G_3 = \frac{a_8}{a_9.s + a_{10}.s^2 + a_{11}.s^3}$$

where the constants H_2,H_3, K and a_1 through a_{11} are given in (Table II).

The measured frequency response at different strokes is shown in Fig. 14. The feedback tuning was low in order to reach the characteristic parts of the curves with a not too high frequency which may otherwise overstress the actuator supports and foundations.

For longer strokes, two types of non-linearities set limits to above approximation. One is saturation, i.e. the maximum stroke of the distributor spool, the second is the pressure drop over the ram piston necessary for strong acceleration, which upsets the proportionality between ram velocity and spool position. These two non-linearities are inherent to the system.

Fig. 7 shows the influence of ram position feedback (H3) and spool position feedback (H2) for step input signals and output amplitudes of about 15 mm of the unloaded ram. The influence of the response time of (H3) and the damping influence of (H2) are clearly disceribles.

Fig. 8 gives step responses of the actuator loaded with 500 kg mass for a number of step amplitudes and feedback tunings. Here the non-linear behaviour is already obvious from the different response shapes and slower responses for greater amplitudes. The same is obvious from the amplitude and flow responses to sinusoidal input signals (Fig. 14).

The commercial solid state d.c. amplifier (Mcog) (Fig.20) is built into a chassis with feedback attenuation potentiometers and test point connectors built in on the front panel for easy tuning and check out. Three additional input channels with attenuation potentiometers are provided for test purposes, e.g. for introducing step inputs, sine waves or d.c. signals.

For the 400 Hz feedback signals, there is an a.c. amplifier and demodulator, since the summing point works at d.c. To reduce static friction, a generator superimposes a 600 Hz square wave (dither) of 50 mV peak to peak on the output current.

The feedback transducer (H2) for the distributor spool displacement is a conventional linear differential transducer and the one (H3) for the ram stroke is a rotary resolver, whose output amplitude varies with the size of the input angle.

Both transducers produce a 400 Hz amplitude modulated signal with the zero point in the centre position. The linear relationship between the ram stroke and its feedback signal is restored by a transmission link (16 in Fig. 12) making the ram position a sine function of the resolver angle.

12. Construction

The basic principle of the actuator design is outlined in Fig. 13, Figs. 16 and 17 give an exploded view and a side view, and Ref. 7 gives detailed information on the construction. The actuator housing, (a sphero-casting), is provided with a rectangular flange on the front side for fixation to the support of the hydrostatic guiding system of the septum magnet. Two bronze bearings are inserted for guiding of the actuating ram. They are provided with labyrinth seals on the high pressure side and sealed off from the outside by low pressure standard lip seals. The leaks are vented through the housing to the main return tube. The leak pressure is kept at about 15 bar by means of the back pressure regulator. The light oil film attached to the ram surface is scraped off by a soft double lip scraper and collected in the drop oil collectors in the bearing flanges. These small leaks are vented through the flanges and collected in a separate reservoir in order not to contaminate the main circuit with the dirty drop oil (see also Fig. 18 for the oil circuits). The seals are interchangeable without dismantling the ram and the main bearings.

The cylinder sleeve is inserted in the housing and is easily interchangeable in case of wear or dammage. The sleeve is kept in place between a recess in the housing and the rear bearing land. This allows for different thermal expansions of the sleeve and housing without deforming the sleeve. The sleeve has axial-directed grooves on the outside, in order to provide oil channels between the distributor parts and the piston compartments.

The ram is of a high grade steel, surface hardened and ground and chrome plated. The ram piston is provided with two berrylium copper piston rings which provide the sealing effect during the acceleration and deceleration periods. The piston diameter is a few tenths of a millimeter smaller than the cylinder diameter, so as to provide a well-defined guiding of the ram. The ram has on the two ends grooves for fixations, one for the ball connection to the hydraulic guiding and one for the pivot block of the mechanical feedback lever.

The distributor valve is mounted in the housing and kept in place, between a recess and the bearing flange. Precise information on tolerances, materials and heat treatments are given on drawing FES.304.231.0. The spool and sleeve are two very accurately machined parts since flow characteristics and leakage are highly dependent on port dimensions, axial and radial tolerances.

The spool is made of a high-quality alloy steel which is nitrated and ground. In order to decrease friction forces on the spool, soft seals on the high pressure sides of the spool piston are avoided. Labyrinth seals are used on each side of the spool piston. This type of seal also decreases the hydraulic lock effect. A small disadvantage is the relatively large leak across the labyrinth. However, the advantages of decreasing the friction between the sleeve and the spool provide for better resolution and stability around the 0-position of the spool. As for the ram piston, two berrylium bronze piston rings provide for sealing of the spool piston. At the rear end there is a connection slot for the linkage of the mechanical feedback and the velocity stops. At the other end there is a small flange connection for the probe of the feedback transducer. The spool chambers on the return side are shaped somewhat like a turbine blade in order to give a sort of flow force compensation.

The sleeve is made of a stabilized quality steel without special surface treatment. It is provided with milled ports around the circumferences. The port dimensions are kept very accurate and are finally adjusted together with the spool in order to obtain a symmetric flow characteristic of the distributor valve. The sleeve is sealed with O-rings on the outside. The tolerances are chosen so as to secure guiding only at the ends and leave a few hundred millimetres of play in the other guidings, to allow for slight deformation of the housing without deforming the sleeve. The principle of the soft seals on the low pressure side is the same as for the ram seals.

13. Pump Group

The hydraulic power for the actuator is supplied by the pump group, situated in the pump station (PST). Figs. 19, 19(a) and 19(b) show the installed pump group and Fig. 20 the central control and interlock racks.

The pump group is a compact unit with two separately controlled, parallel connected pumps, with common oil reservoir, filters and cooling system.

The pump works at 200 bar and its 90 dm^2/min delivery per pump, can supply up to three full movements per acceleration cycle at a repetition rate of 10 cycles/min. The second pump is reserve and may be used for higher repetition rates. Fig. 21 shows a simplified diagram of the pump group and Fig. 18 the hydraulic system. There are two variable displacement swash plate pumps with pressure controlled delivery. This type of pump makes it possible to maintain a preset pressure of 200 bar, delivering exactly the demand of the actuator and the small losses in pump and valves.

The function of the pump is explained in Fig. 18. The spring in the swash plate actuator acts on the larger piston side (A1) and brings the swash plate to a maximum swash angle position. The system pressure acts directly on the annulus of the adjusting piston (A2) as well as through the throttle (220) on the larger piston surface (A1) whereby the spring force is reinforced. The system pressure acts also on the measuring surface of the pressure reducing valve (224) which maintains a constant pressure on (A1).

Should the pressure rise above the set pressure of the pressure reducing valve (224), the connection between system pressure and (A1) would be cut off, thus connecting (A1) with the exhaust. The flow through the orifice between (A1) and (A2) creates a pressure drop over the piston and moves it against the spring to a smaller swash angle resulting in a smaller output flow and pressure. When the system pressure drops, (A1) will again be connected via (224) to the system pressure and be pushed in the direction of a larger swash plate angle, and a corresponding larger output flow. A solenoid directional control valve or by-pass valve (227) is placed between valve (224) and (A1), whereby the cylinder can be connected with the exhaust or with the valve (224). When the solenoid is de-energized, the cylinder is connected to the exhaust so that swash angle moves towards zero at a system pressure of approximately 20 bar. The motor then must only start up with a minute torque. The switching to full pressure takes place after commutation of the motor switches from star to triangle. The solenoid is then energized and the valve (222) connects the cylinder to the set pressure of the valve (224).

The oil is then passing through the acoustic damper (206) and the check valve (313) into the first accumulator (001) of the high pressure transmission tube. For double safety a fast acting relief valve (307) is installed close to pump output in addition to the common relief valve (316). Furthermore, a solenoid-operated discharge valve (321) allows automatic fast unloading of the accumulators into the return tube, when the pumps are stopped. The discharge time however is adequate for withdrawing and locking of the actuator in case of emergency stops. The return tube goes through a cooler (808) which stabilizes the oil temperature at $30^{\circ}C$ by controlling the water flow. The main filter 20 μm is placed after the cooler and is rated at 90 dm^3/min. The by-pass relief valve opens when the flow rate is greater or when the filter is dirty. The check valve (110) prevents backflow from the tank when the filter is opened for cleaning. There is another full flow filter in the oil leak return tube.

The pump group forms a compact unit (Fig. 19). A heavy base frame bears the two motor pump groups, the common reservoir, filters, valves and cooling system. The oil tank is placed above the pump, thus giving a slight overpressure at its inlet and reducing the risk of cavitation. The oil level is 1,5 m under the actuator axis. The tank is therefore pressurized with dry air (0,2 bar) so as to avoid air bleeding of the system after long stops. The manometers and all pressure and temperature switches for the control and interlock system are easily accessible on one side of the pump group.

The swash angle is picked up by a potentiometer (227 Fig. 18) placed on the side of the pump housing and visualised on the central and remote control panel for indication of the pump flow (see Figs. 20 and 22).

14. Fluid Power Transmission

From the pump group the oil is transmitted to the actuator and back through stainless cold drawn steel tubes of interal diameter 38 mm for the supply and 36 mm for the return tube. Their diameter limits the pressure drop to about 3% for the full flow rate of 90 dm^3/min and oil viscosity of 8^oE.

The high pressure transmission tube is provided with two accumulators for reasons as indicated in Section 6.

The leak tube was originally foreseen for the bearing leak of the actuator also, but as this leak had no serious contamination it could be fed back through the return tube and there was no case for this tube any more.

15. Control and interlock

The central controls, monitoring and interlock racks are located in the pump station (PST)(Fig. 20). The controls and main supply relevant to operation of the ejection system are repeated on the "remote control panel" in the LCR. Also, the electronic program generator for the input signal is located here (see Fig. 22).

The start-up of the complete system initial check ups and selection of mode of operation is carried out from the central control racks (PST). Thereafter the control is given over to the remote control LCR. Figs. 23 (a), (b), (c), (d) and (e) give a clearer picture of the control panels in the (PST).

The high supply pressure and inflammability of the hydraulic oil, as well as the strong thrust and high speed of the actuator ram, make this system inherently dangerous. An intricate chain of interlocks, therefore, protects personnel and equipment against consequences of faulty conditions (see Fig. 20(f))for parameters with interlock functions. The numbers on the lamps correspond to those on Fig. 18. A fail safe philosophy has been followed as much as possible. The block diagram is shown on Fig. 23.

The interlock circuits impose a particular sequence of manual and automatic actions for start-up and shut-down of the hydraulic system (see Fig. 24). The automatic sequence withdraws the actuator and locks it at its "servo-stop out" position, after an interlock or manual emergency stop. A faulty control action has normally no influence, only the correct action gives the required result. The majority of fault conditions stop the whole system and lock the actuator. In addition, there are a number of warning signals. Action is requested by pressing the relevant button, which then starts flashing. If the order is executed correctly, the signal will flash continuously. Persistent flashing indicates failure of the element in question.

Figs. 20(a) shows how the control and interlock units are grouped with servo-amplifier and actuator monitoring system into two racks in the PST. The control and interlock system operating from a 24 d.c. supply has for each interlock transducer a repeater relay, acting immediately on the signal lamps. Additional contacts of the repeater relays are continued to various interlocking functions inside the control unit, which contain the necessary logic circuitry, except for two dual time delays, located in an additional chassis with the $\pm$ 15 V, 24 V and the 25A flashing supplies. The delays by-pass some interlocks during start-up and prevent starting both pumps together within 60 seconds.

Two interlocks are not directly indicated on the interlock panel. They are :

(a) 220 Volt program generator "not on"

(b) 220 Volt servo-amplifier "not on"

(a) prevents the pump to be started, as the interlock action is in the chain of the emergency stop. When the program generator is "not on" it results in flashing emergency lamp. Program generator "on" is indicated on the program generator itself.

(b) prevents the actuator from being moved by manual or program input. 220 Volt on is indicated on the amplifier itself.

16. Start and Stop

The sequence of starting operations are listed below.

1. Switch on the main distribution box for :

   (a) 380 Volt pump 1

   (b) 380 Volt pump 2

   (c) 220 Volt for the central and interlock racks

   (d) 220 Volt to the distribution box N1 after intercommand television

2. Switch on the 220 Volt on rack.

3. Switch on the Lambda 24V power supply rack.

4. Press (ON SUPPLY 24 Volt) on the central unit rack.

5. Press (RESET).

6. Press for selection (PUMP 1) or (PUMP 2).

Effect :

(a) The suction valve for the pump selected opens.

(b) The pump unit for the hydrostatic bearings starts.

(c) The interlock chain of the pump selected comes into operation.

7. Wait 1 minute until the pressure and the flow of the hydrostatic bearings have reached their nominal values.

8. Then press (RESET) again.

Effect :

(a) All interlocks stop flashing, except :

(SERVO STOP EXT)
(OIL SUPPLY PRESSURE 409) Low
(HYDRAULIC SYSTEM READY)

9. Press (PUMP STATION CONTROL).

Conditions :

(a) Ram on (SERVO STOP EXT).

(b) Ram (LOCKED)

(c) (STROKE CONTROL) Zero

(d) (MANUAL)

Effect :

(a) Start and stop operation can now only take place· from the control point selected. However, the (EMERGENCY SWITCH), (PUMP 1 OFF), (PUMP 2 OFF) and (RESET) are always operationl from both control points.

10. Press (ON) for selected pump.

Conditions :

(a) (PUMP 1) or (PUMP 2) selected.

(b) Ram on (SERVO STOP EXT).

(c) Ram (LOCKED).

(d) (STROKE CONTROL) zero.

(e) MANUAL.

(f) (EMERGENCY) switch OK.

(g) Hydraulic interlock chain OK.

(h) Actuator interlock chain OK.

(i) Interlock chain of the selected pump OK.

(j) Program generator LCR on.

Effect :

(a) Start of selected pump.

(b) Water shut-off valve 807 opens.

(c) Discharge valve 321 closes.

(d) After 15 sec. the motor starter switches from star to delta and closes at the same time as the by-pass valve (221) or (222) of the pump selected.

11. Wait a minute, until the system pressure has reached its normal value and the lamps (OIL SUPPLY PRESSURE 409) low and (HYDRAULIC SYSTEM READY) stop flashing. After stops longer than 1-2 hours, leave the pressure on for 15-20 minutes before going to the next point. This is in order to avoid vibration of the actuator when unlocked.

12. Press (UNLOCK).

Conditions :

(a) Ram on (SERVO STOP EXT).

(b) Ram (LOCKED).

(c) (STROKE CONTROL) zero.

(d) (HYDRAULIC SYSTEM READY).

Effect :

(a) Locking device opens.

(b) The input of the servo-amplifier switches from emergency input to manual input.

13. Press (PROGRAM).

Conditions :

(a) Ram on (SERVO STOP EXT).

(b) Ram (UNLOCKED).

(c) (STROKE CONTROL) zero.

(d) (MANUAL).

(e) (HYDRAULIC SYSTEM READY).

Effect :

(a) The input of the servo amplifier switches from manual zero (ram on SERVO STOP EXT) to program zero (ram in base line position).

14. Watch the previous point (13) and the following operation on the oscilloscope.

15. Turn the (STROKE CONTROL) to full open dial "10".

    Effect :

    (a) Magnitude of input program corresponds to that set on the program selector in LCR.

    (b) Actuator follows the input program.

16. A general check of ram movements, spool stroke and and input program on the monitoring must now be undertaken, together with a check of the system pressure.

17. Now if LCR control is desired then turn (STROKE CONTROL) to zero.

18. Press (MANUAL).

19. Press (LOCK).

20. Select (LCR CONTROL).

    Effect :

    (a) Transfer of control to LCR for : START, STOP, UNLOCK, LOCK, MANUAL, PROGRAM and STROKE CONTROL.

    (b) EMERGENCY STOP, PUMP OFF and the RESET is still available from the pump station.

    (c) Lamp (LCR CONTROL) lights up.

    LCR OPERATION :

21. Press (UNLOCK).

22. Press (PROGRAM).

23. Turn (STROKE CONTROL) to "10".

24. Check actuator movement on the slow monitoring oscilloscope.

25. For the set-up of actuator programming, see instructions.

STOP PROCEDURE

The hydraulic system must be stopped from the control point selected.

Normal stop :

1. Turn (STROKE CONTROL) to zero.

2. Press (MANUAL).

Effect :

(a) Actuator goes from base line to the servo-stop position.

3. Press (LOCK).

Conditions :

(a) Ram on (SERVO STOP EXT).

(b) (STROKE CONTROL) zero.

(c) (MANUAL).

Effect :

(a) Locking device keeps the actuator in the locking position.

4. Press (PUMP 1 OFF) and (PUMP 2 OFF).

5. If the control has been executed from LCR, go to pump station and continue the operation there.

6. Press (PUMP STATION CONTROL).

7. Press (SUCTION VALVES CLOSED).

8. Press (OFF, SUPPLY 24 VOLT).

   Conditions :

   (a) Suction valves closed.

9. Switch off the main distribution boxes.

10. Check the mamometers on the pump group and fill in the check list.

STOP CAUSED BY INTERLOCK

Effect :

(a) Pump will stop.

(b) Discharge valve 321 opens.

(c) By-pass valve 221 or 222 opens.

(d) Actuator is automatically drawn back to the servo-stop position and locked.

Before starting again have a look on the panel HYDRAULIC SYSTEM SIGNALISATION, eliminate the fault, press the (RESET) and start operation again as from point 10.

EMERGENCY STOP

Emergency stop can be made from the pump station on LCR on the straight section 24, independent of the selected control point.

Effect :

(a) Pump will stop.

(b) Discharge valve 321 opens.

(c) By-pass valve 221 or 222 opens.

(d) Actuator is automatically drawn back to the servo-stop position and locked in this position.

The effect is the same as for interlock stops.

17. Magnet Guiding

The principle of the guiding system has been given in Figs. 3 and 5, and the hydrostatic functioning is given in Fig. 25. The carriage consists of two parallel cylindrical guiding shafts connected by a traversal. The magnet supporting shaft is connected to the centre of this traversal. Each guiding shaft slides in two hydrostatic bearings, which are fixed on a rigid steel base. This arrangement provides a well-defined and accurate guiding of the septum magnet as the play in the bearings when pressurized is zero. The derivations in the horizontal plane are given by elastic deformations of the guiding shafts and amounts to a few hundredths of a millimeter measured on the magnet. The angular deflection of the magnet is dependent on the alignment of the bearings and the shafts, but as they have been aligned to within 5 μm, it limits the angular deflection to an unimportant parameter in comparison with the beam tolerances and keeps the tension of the vacuum bellow to a strict minimum, which is essential for the proper functioning of this element.

The construction and way of functionning of the bearing are illustrated in Fig. 26 and 27. It consists of the bearing shell proper, which is contained in a housing closed by two end flanges with sealings and scrapers. The shell is provided with two 8 recesses, uniformly distributed around the circumferences, each pair of recesses separated by a groove for the return oil. Each recess is supplied with oil through an orifice. There is a common high pressure distribution channel for all recesses, a common collector duct for the return oil and another separate one for the leak oil. Special self-sealing connectors on top of the bearing may be used for air bleeding and pressure measurement.

If unloaded, the guiding shaft is centred on the bearing shell and the pressure in the recesses is equally and symmetrically distributed around the shaft. The total flow resistance between the recesses and the exit of the bearing is equal to the resistance in the orifice and therefore the pressure build-up in each recess is half the supply pressure.

When the shaft is loaded, it takes an excentric position with respect to the bearing shell. The changes in clearance causes the flow resistance and the pressure to increase in recesses of the lower half circumference and decrease in the recesses of the upper half. The change of pressure, multiplied by the active area on the shaft, gives an upward resultant force which balances the load.

Fig. 27 gives a more detailed explanation. In Fig. 27 (a) the guiding shaft is sealed by the external force (F) on the lower half of the bearing, at zero supply pressure ($P_s = 0$). In Fig. 27 (b) when the supply pressure builds up, it builds up also in the bearing recesses, mostly in the lower recess where it equals the supply pressure, and less in the top recess where the resistance to flow is smaller due to the bearing clearance. In Fig. 27 (c) the recess pressure (P_2) is built up to a value where the (P_2) multiplied by the lower recess area minus (P_1) multiplied by the upper recess area is just enough to counter-balance the external force and lift the shaft from the bearing. In Fig. 27 (d) as the shaft separates from the bearing, the pressure (P_2) drops to a lower value and (P_1) increases to a higher value, due to the decreased resp. increased flow resistance between the recesses and the exit of the bearing. However, as the surface on which the pressure acts on the lower half is larger than in Fig. 27 (c), the external force is still balanced by the pressure force. In Fig. 27 (e) if the external force is increased by (4F), the bearing clearance will decrease at the lower half and increase at the upper half and accordingly increase and decrease the flow resistance at the resp. places. So, the recess pressure will rise in the lower and drop in the upper half, until the pressure force in the lower recess minus the pressure force in the upper recess is large enough to carry the new load.

The load can be increased theoretically until the recess pressure in the lower half is equal to the supply pressure. In Fig. 27 (f) if there is no load at all, the pressure in the upper and lower recesses will equalize as well as the flow and bearing clearances and the shaft will be in the ideal centred position.

The bearings are supplied with oil of 50 bar, so that without load, the pressure in the recesses is about 25 bar. When applying an external force of 5000 N the pressure increased to 36 bar in the two lower recesses and decreased to 14 bar in the two upper ones. With this load, the shaft was still not touching the bearing shell and could, in the absence of scrapers, be moved just by blowing on it with an air jet. The radial clearance between shell and shaft is 0,025 mm and the diameter of the orifice 0,20 mm. For an oil viscosity of $8^{o}E$ (at $25^{o}C$), the flow rate is 1.1 dm^3/mm for each bearing.

The oil is supplied to the bearings by an autonomous small pump unit (Fig. 28) giving a pressure of 50 bar and a maximum flow of 12 dm^3/min. It includes water cooling, 5 μm filtering in the pressure line and 100 μm in the return line, and an accumulator. The high pressure oil stored in the latter permits withdrawing and locking of the actuator in case of power failure, and Fig. 18 shows the hydraulic circuit.

18. Check and Preventive Maintenance

Trouble free operation for long periods of time can be attained by maintaining the following conditions :

(a) Pressures and temperatures within ranges as indicated in Tables III and IV. Numbers on components relative to these are found on drawing 404.516.0.

(b) Working fluid of oil blends of highly refined petroleum base oils with incorporated anti-oxydant, anti-rust, anti-foam and anti-wear additives. This oil must also be suitable for continuous working pressure of 200 bar.

The oil used at CERN was Shell Tellus 27 and the oil corresponding to that can be recommended. Complete change of the oil must be undertaken each 2000 working hours.

(c) New oil must be filtrated to 10 μm before it is filled into the system.

(d) Water contamination of the oil must be strictly avoided by not having oil drums stored outside in an upright position. Collected water on top of the drums can leak through an untight plug.

(e) Tight nitrogen and oil tubes.

(f) Clean filters.

(g) Pump group and actuator shall always be kept in clean and proper condition, in order to be able to trace minute oil and water leaks before more serious leaks occur.

Furthermore, periodical checks and inspections of the entire equipment must be made for timely elimination of detected faults or troubles :

1. Pressure, temperature, oil level and mode of operation for the pump group as indicated on check lists 1 and 2 - compare with table IV for the right values. The check list must be filled out the first time 1-2 hours before a run starts and then after each two hours until the run is over. If a nitrogen pressure is too low, recharge the corresponding accumulator to the maximum pressure level indicated in table IV, column "Before Start".

2. Pressure, temperature and oil level for the pump unit of the hydrostatic bearings as indicated on check list 2, compare with table IV. The check list chall be filled out before the pump unit is started and some 15 minutes later.

3. Check on the actuator monitoring the movement of the ram, spool and the input program immediately after start and compare the movement with the master photo. Check these movements each two hours together with the pressure checks.

4. Check visually before each run the movement of the actuator and membrane with programmed movements, and see that it moves softly and smoothly. Pay attention that the membrane is not touching the plexiglass cover.

5. Check the shafts of the hydrostatic guiding for grooves or other damage.

6. Check for water and oil leaks around the actuator and tighten if there is any leak, and dry out the oil film on the actuator components before leaving the ring.

7. Check conditions of the tubing in the cellar and tube trench in each shut down.

Observation of the above mentioned points and elimination of detected fault conditions prolong or eliminate completely the intervals between serious damage and repair, but does not exclude the necessity of effecting preventive maintenance.

Maintenance

The operation of the high pressure power supply and the fast moving actuator is unavoidable accompanied (due to thermal and mechanical stresses) by wear of certain parts such as high and low pressure sealings, pistons and cylinders, ball bearings of the pump motors, and so on.

In order to obtain trouble free operation over long periods of time, it is therefore a necessity to execute some preventive maintenance according to a fixed schedule as follows :

Each Week

Complete cleaning of the pump group for oil, water and dust deposit. This is for easy localisation of new leak points.

Each Shut-down

1. Checking and filling of all accumulators according to table IV.

2. Complete cleaning of the actuator and guiding system.

3. Tighten leak points.

Every Two Months

1. Cleaning of all filters, 801, 803, 611 and 815. If a large amount of swarf is found in the filters, the wear points should be traced. Wash all filter elements and housings with petrol and dry with air jet.

2. Take 200 cm^3 sample of oil in the pump group and the pump unit of the HBS and send it for analysis to determine the contents of acid and water together with an indication of viscosity change. Keep a record of results.

Every Six months or 2000 working hours (whichever comes first)

1. Complete change of oil in the circuit of the pump group and pump unit of the HBS. Check whether the oil tanks are free from sediment and clean with light, clean oil.

   Fill again with oil corresponding to Shell Tellus 27.

2. Check proper functioning of the pressure switches by lowering the accumulator pressure to the required set point. For the temperature switches raise or lower the temperature (see table III).

3. Check the bolts on the support of the actuator and the hydrostatic guiding.

Each Year or 8000 working hours

1. Complete overhaul of the system in a long shut down.

2. Check of the high pressure pump having been most in use.

3. Check the pump of the HBS.

4. Check the piston sealings of all the accumulators.

5. Check seal of the actuator.

6. Check seals of the hydrostatic bearings.

7. Check the actuator parts for excess wear.

8. Check the hydrostatic bearings for excess wear and measure the flow for each bearing.

9. Grease the electromotor bearings.

10. Control the tightness of all Ermeto flanges and fittings on the transmission lines.

11. Clean all interlock and feedback cables and connections with alcohol.

12. Change the Moog Servo Valve and send it back to the factory for an overhaul.

19. Acknowledgements

I wish to thank Messrs B. Bouchet, P. Griessen and A. Ijspeert for their excellent contributions to the design, manufacture, assembly, computation and testing of the system. Also, I wish to thank Mr. B. Kuiper for his support and helful comments through the stages of the work.

20. References

1. B. Kuiper et al., Design study on the fast ejection system of channel A, Serpukhov 70 GeV proton synchrotron, CERN/PS/FES/TN-49, February 1969.

2. R. Bossart, Function generator of hydraulic movement, PS/FES/TN-P4, 15.3.1972.

3. A.J. Ijspeert, The parameters of the transfer function of the hydraulic actuator (Serpukhov prototype), CERN/PS/FES/TN-88, July 1969.

4. A.J. Ijspeert, Calculation of the dynamic behaviour of the hydraulic actuator (serpukhov prototype), CERN/FES/TN-108, 29.10.1969.

5. A.J. Ijspeert, Test results of the hydraulic actuator (Serpukhov prototype), CERN/PS/FES/TN-253, December 1970.

6. A.J. Ijspeert, Test results of the final Serpukhov actuator, CERN/PS/FES/TN-350, December 1972.

7. B. Bouchet, Construction de l'actuator electrohydraulique et du système de guidage hydrostatique pour l'éjection de Serpukhov, PS/FES/TN-341, June 1972.

8. S. Milner, Derivations of the transfer functions for the Serpukhov actuator, CERN/PS/FES/TN-275, March 1971

TABLE 1

MAIN CHARACTERISTICS FOR THE SERVO-ACTUATOR

	Symbol	Unit	Programmed Operation 300 ms	Operation with step inp.
Moving mass, total	m_1	Kg	500	500
Ram stroke maximum	y_{max}	m	0,170	0,2
Ram stroke nominal	y	m	0,105	0,145
Ram velocity	$\dot{y}$	m/s	0,45	2
Ram acceleration	$\ddot{y}$	m/s^2	16	88
Supply pressure	p_1	bar	200	200
Return pressure	p_2	bar	15	15
Piston area, ram	$A_{1,2}$	m^2	$22 \cdot 10^{-4}$	$22 \cdot 10^{-4}$
Acceleration force ram piston	F	N	8000	44000
Piston area, spool	$A_{3,4}$	m^2	$12.4 \cdot 10^{-4}$	$12.4 \cdot 10^{-4}$
Mass of spool	m^2	Kg	8.9	8.9
Spool stroke	X	m	$0,57 \cdot 10^{-3}$	$8 \cdot 10^{-3}$
Spool velocity	$\dot{X}$	m/s	$0,005 \cdot 10^{-3}$	$0,933 \cdot 10^{-3}$
Setting time, ram	t_s	ms	300	150
Static error, drift in 2 hours, ram	-	m	$\pm 0,1 \cdot 10^{-3}$	$\pm 0,1 \cdot 10^{-3}$
Dynamic error, cycle to cycle, ram	-	m	$\pm 0,05 \cdot 10^{-3}$	$\pm 0,02 \cdot 10^{-3}$
Internal leak of distributor	q_1	dm^3/min	< 10 stat.	-
Internal leak of the bearings	q_2	dm^3/min	<3,3 stat.	-
Internal leak of the servo valve	q_3	dm^3/min	< 3 stat.	-
Oil consumption per cycle	V	m^3	$0,462 \cdot 10^{-3}$	$0,88 \cdot 10^{-3}$
Total oil consumption with 10 cpm +leak	Q	dm^3/min	< 21	< 25
Power consumption with 10 cpm + leak	P	KW	< 7	< 8,2

TABLE II

CONSTANTS AND FEEDBACKS

FOR

THE TRANSFER FUNCTIONS

Symbol	a_1	a_2	a_3	a_4	a_5	a_6	a_7	a_8
Unit	$\frac{m^3}{A.sec}$	sec.	sec.	m^{-2}	./.	sec.	sec.^{2}	sec.$^{-1}$
Value	0,224	$1,36.10^{-3}$	$1,29.10^{-6}$	806	1	$1,56.10^{-5}$	$4,82.10^{-8}$	1100

Symbol	a_9	a_{10}	a_{11}	K	H_2	H_3	H_4	
Unit	./.	sec.	sec.^{2}	mA/V	V/m	V/m	$\frac{V\text{-}sec.^2}{m}$	
Value	1	$2,21\text{-}10^{-5}$	$4,44.10^{-4}$	2,03	508	35,5	$0,99.10^{-3}$	

TABLE III

SET POINTS AND ADJUSTABLE RANGES

FOR THE

PRESSURE AND TEMPERATURE SWITCHES

COMP. Nr.	PRESSURE SWITCH CONTACT OPEN WHEN:	SET POINTS	ADJUSTABLE RANGE
301	OIL PRESSURE PUMP 1	≥210 bar	1_400 bar
302	OIL PRESSURE PUMP 2	≥210 "	1_400 "
409	OIL PRESSURE SUPPLY LOW	≥160 "	1_400 "
409	OIL PRESSURE SUPPLY HIGH	≥210 "	1_400 "
002	NITROGEN PRESSURE FIRST ACCU.	≤120 "	1_200 "
408	OIL RETURN PRESSURE	≥ 3 "	0,1 _10 "
410	OIL LEAK PRESSURE	-	0,1 _10 "
509	OIL PRESSURE SECOND ACCU.	≥210 "	1_200 "
511	NITROGEN PRESSURE SECOND ACCU.	≤ 120 "	1_200 "
634	OIL SUPPLY PRESSURE ACT.	≥ 210 "	1_400 "
631	NITROGEN PRESSURE THIRD ACCU.	≤120 "	1_200 "
633	OIL RETURN PRESSURE ACT.	≥25 "	1_200 "
632	NITROGEN PRESSURE RETURN ACCU.	≤ 12 "	1_200 "
635	OIL LEAK PRESSURE ACT.	-	1_200 "
719 HB	NITROGEN PRESS. HYDR. BEARINGS	≤ 30 "	1_200 "
820	WATER PRESSURE PUMP GROUP	≤ 2 "	0,1 _10 "
101	AIR PRESSURE OIL TANK	≤ 0,15 "	0,02 _1,25 "
709 HB	OIL PRESSURE HYDR. BEARINGS	≤ 45 "	
718 HB	WATER PRESSURE HYDR. BEARINGS	≤ 2 "	
703 L	AIR PRESSURE LOCKING ACT.	≤ 5 "	
115	OIL TEMPERATURE	< 20° > 35°C	0- 100°C
703 HB	OIL TEMPERATURE H.S.B.	> 32°C	20-50°C

TABLE IV

PRESSURE RANGES FOR THE

MANOMETERS

COMP. Nr.	PRESSURE	BEFORE START bar	NORMAL OPER. bar	MIN. bar	MAX. bar
301	OIL PRESSURE PUMP 1	—	190–195	180	210
302	OIL PRESSURE PUMP 2	—	195–200	180	210
409	OIL PRESSURE SUPPLY	—	190–200	180	210
002	NITROGEN PRESSURE FIRST ACCU.	125–130	190–200	180	210
408	OIL RETURN PRESSURE	—	1,1	0,5	3
410	OIL LEAK PRESSURE	—	—	—	—
509	OIL PRESSURE SECOND ACCU.	—	190–200	180	210
511	NITROGEN PRES. SECOND ACCU.	125–130	190–200	180	210
634	OIL SUPPLY PRESSURE ACT.	—	190–200	180	210
631	NITROGEN PRESURE THIRD ACCU.	125–130	190–200	180	210
633	OIL RETURN PRESSURE ACT.	—	16–20	13	25
632	NITROGEN PRES. RETURN ACCU.	12–13	16–20	13	25
635	OIL LEAK PRESSURE ACT.	—	—	—	—
719 HB	NITROGEN PRES. HYDR. BEARINGS	30–40	45–50	45	50
820	WATER PRESSURE PUMP GROUP	3	3	2	5
101	AIR PRESSURE OIL TANK	0,2	0,2	0,15	0,22
710 HB	OIL PRESSURE HYDR. BEARINGS	—	48	45	50
722 HB	NITROGEN PRES. ACCU. HYDR. BEARINGS	30-40	45-50	45	50
723 HB	WATER PRESSURE HYDR. BEARINGS	3	2,5	2	5
	AIR PRES. ACT. LOCKING VAC.	6	6	5	7

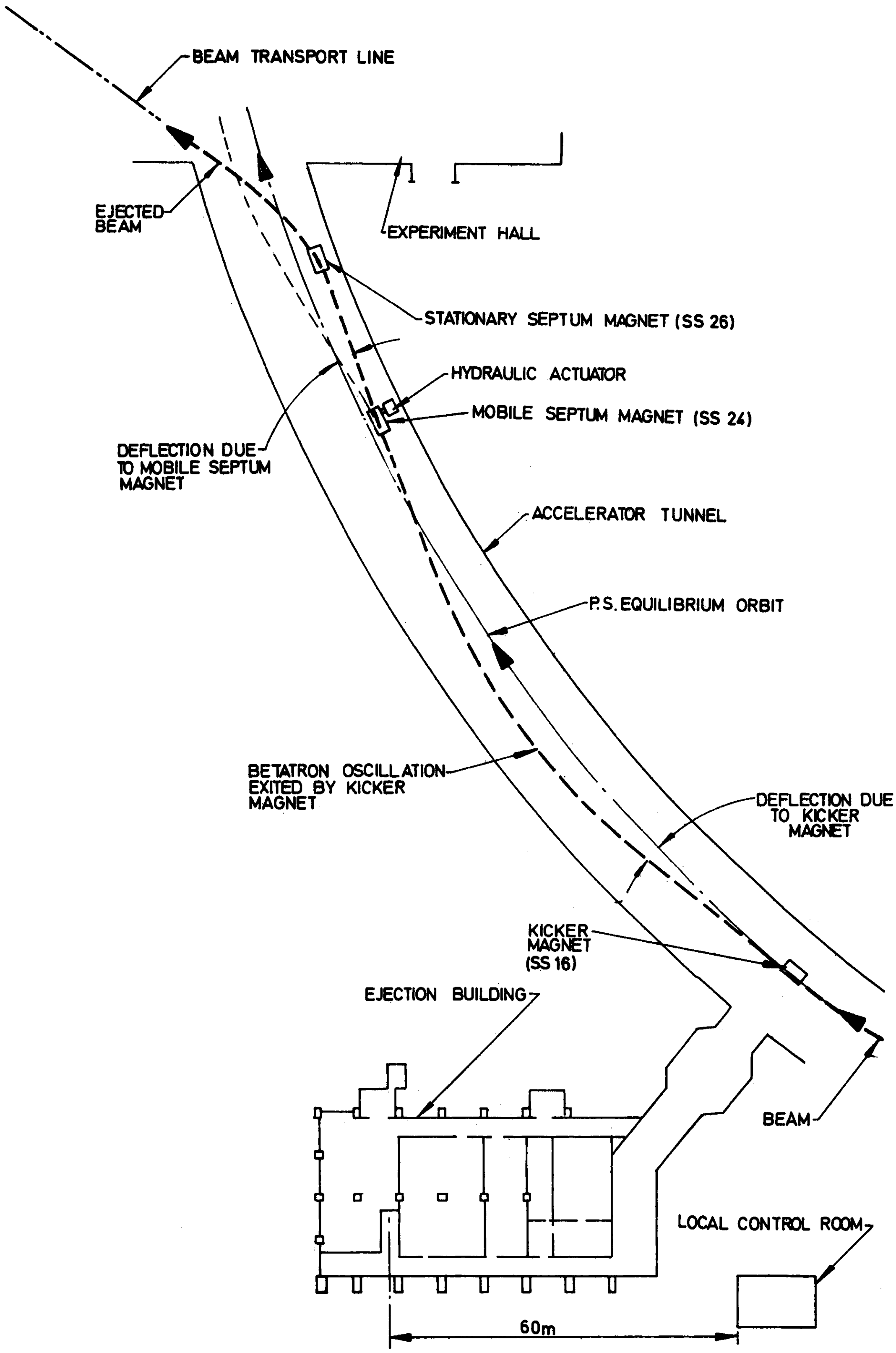


Fig1 PRINCIPLE OF THE FAST EJECTION SYSTEM

SIS/R/17089

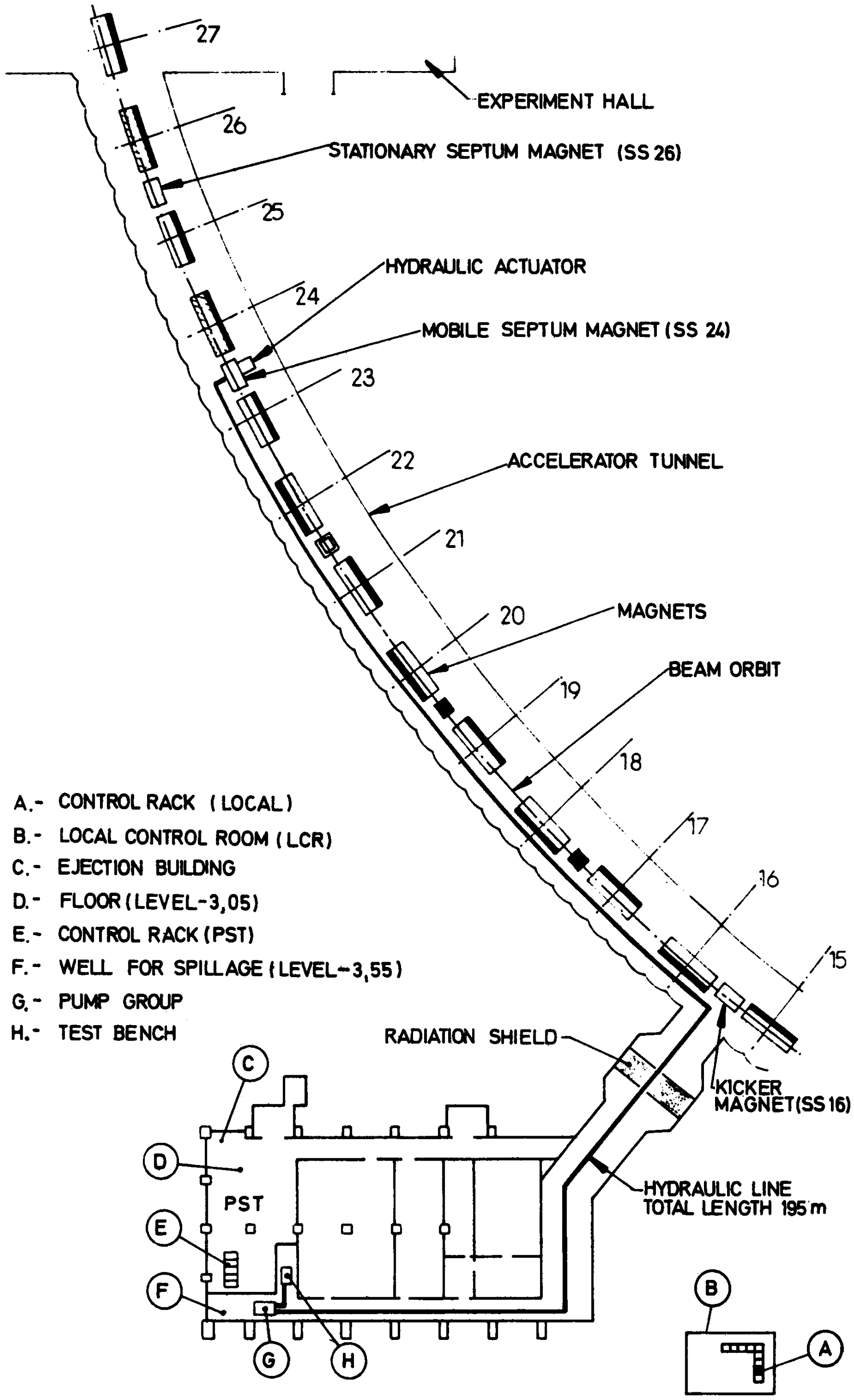


Fig 2 SITUATION PLAN

OF THE HYDRAULICS

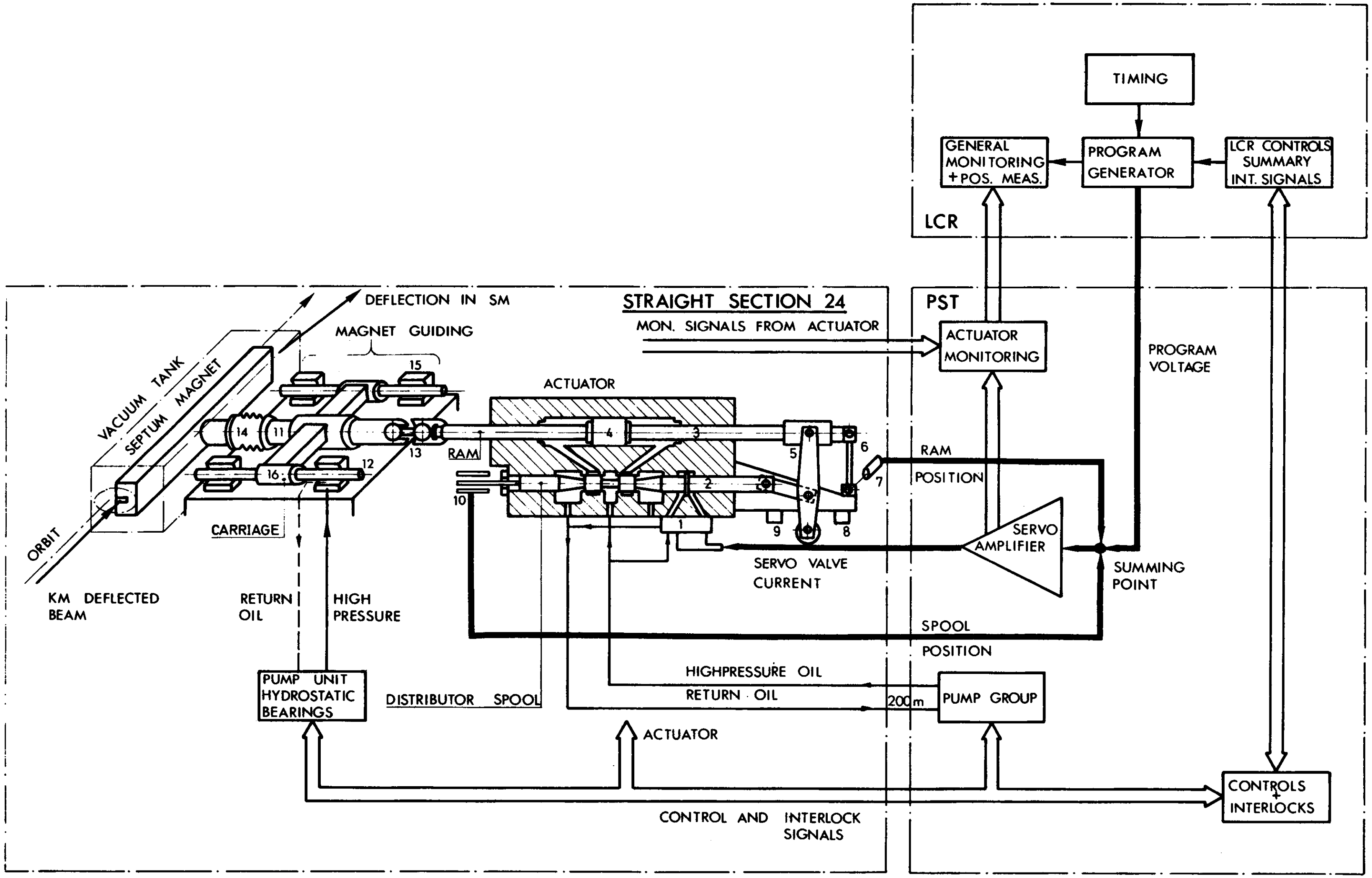


Fig. 3 BLOCK DIAGRAM FOR THE ELECTRO HYDRAULIC SERVO ACTUATOR

(1) servo-valve, (2) distributor spool, (3) actuating ram, (4) ram piston, (5) mechanical feedback lever, (6) lever for ram position feedback, (7) feedback resolver, (8) mechanical servo-stop (in), (9) mechanical servo-stop (out), (10) spool position feedback, (11) magnet connecting shaft, (12) guiding shaft, (13) ball connection, (14) membrane, (15) hydrostatic bearings, (16) supporting traversal.

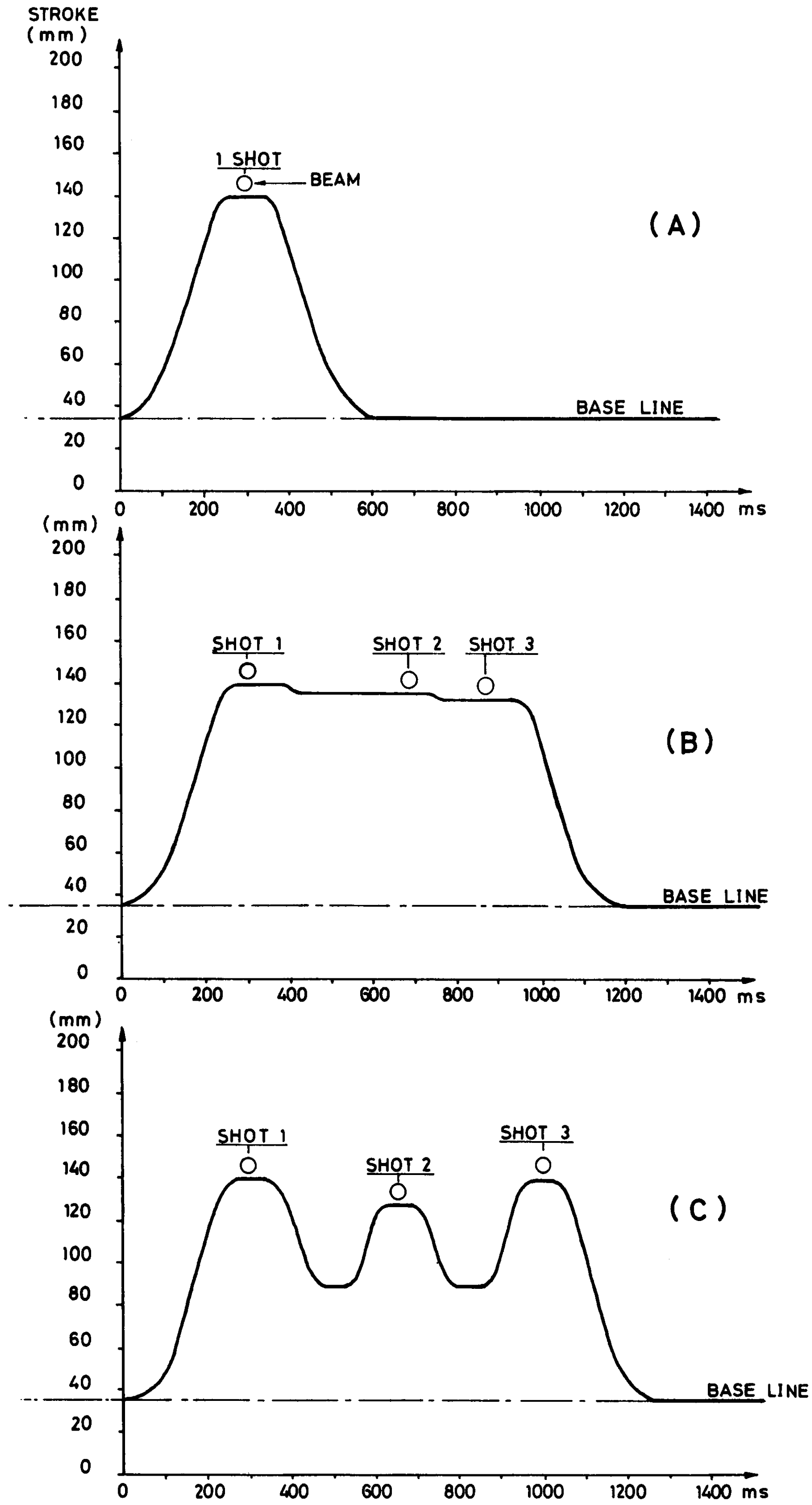


FIG. 4 SOME POSSIBLE MOVEMENTS OF THE ACTUATOR

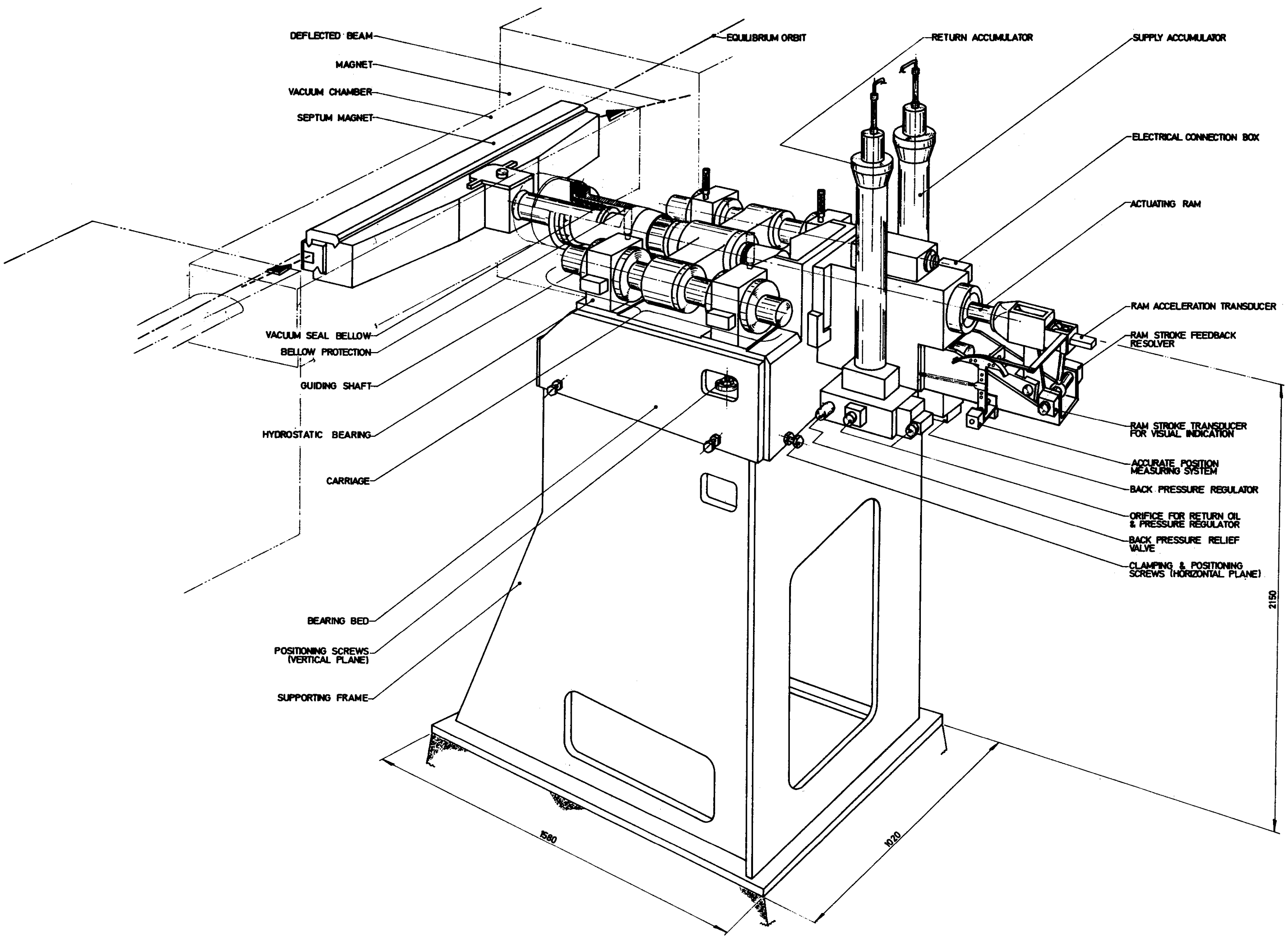


Fig.5 PERSPECTIVE VIEW OF ACTUATOR & HYDROSTATIC GUIDING SYSTEM

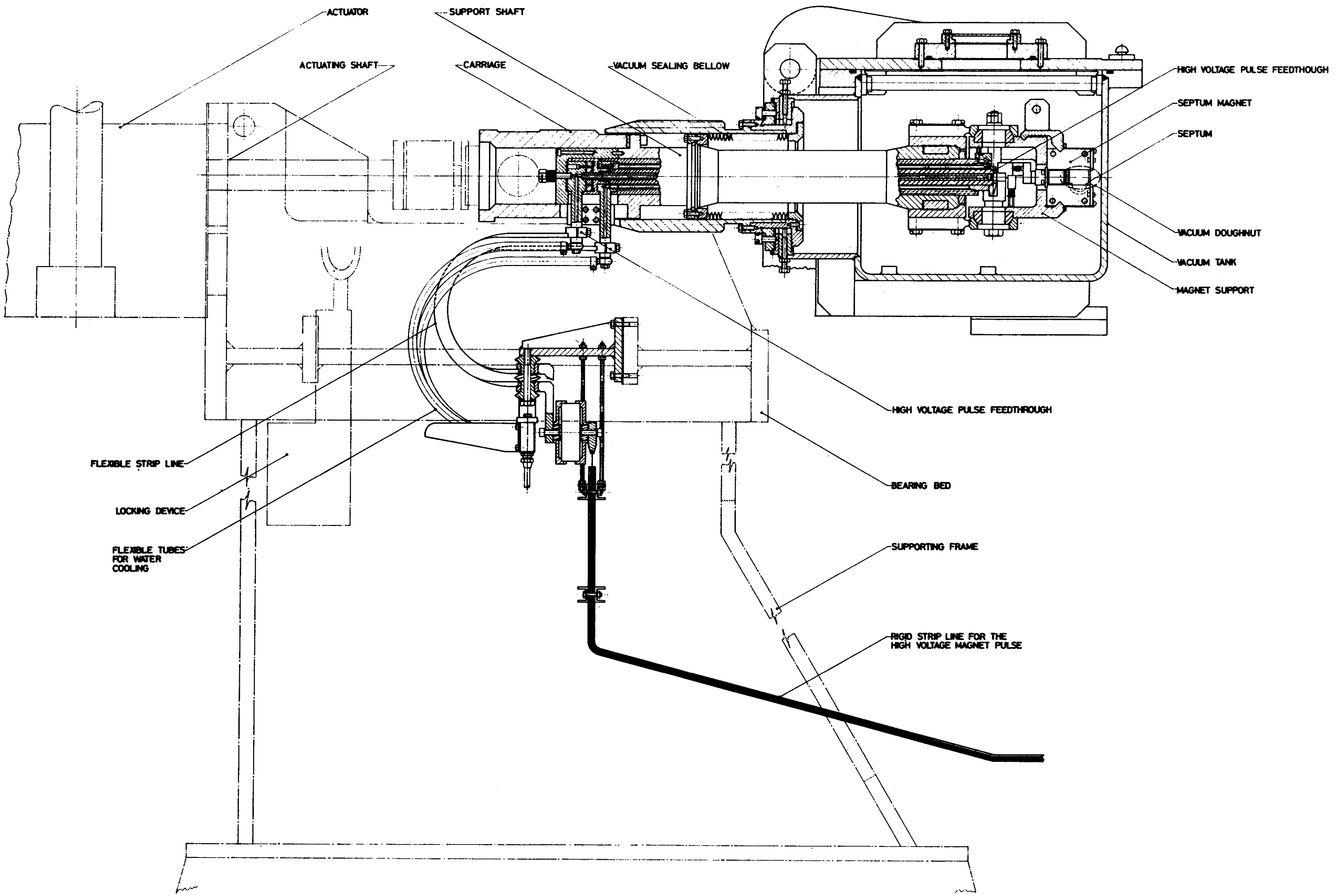


Fig.6 ACUATOR AND CONNECTED SEPTUM MAGNET

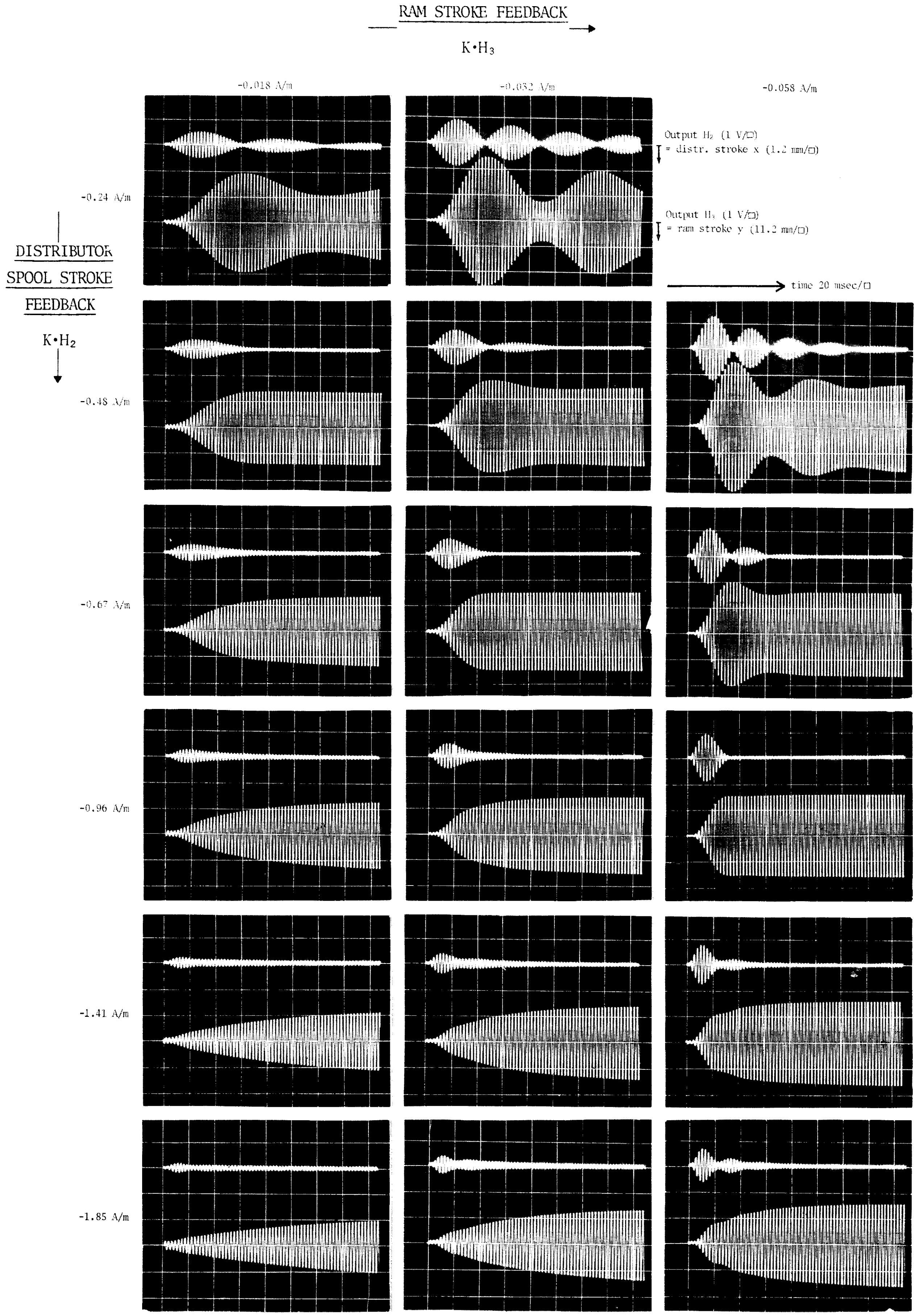


Fig. 7 STEP RESPONSE OF THE ACTUATOR

Without dummy load.

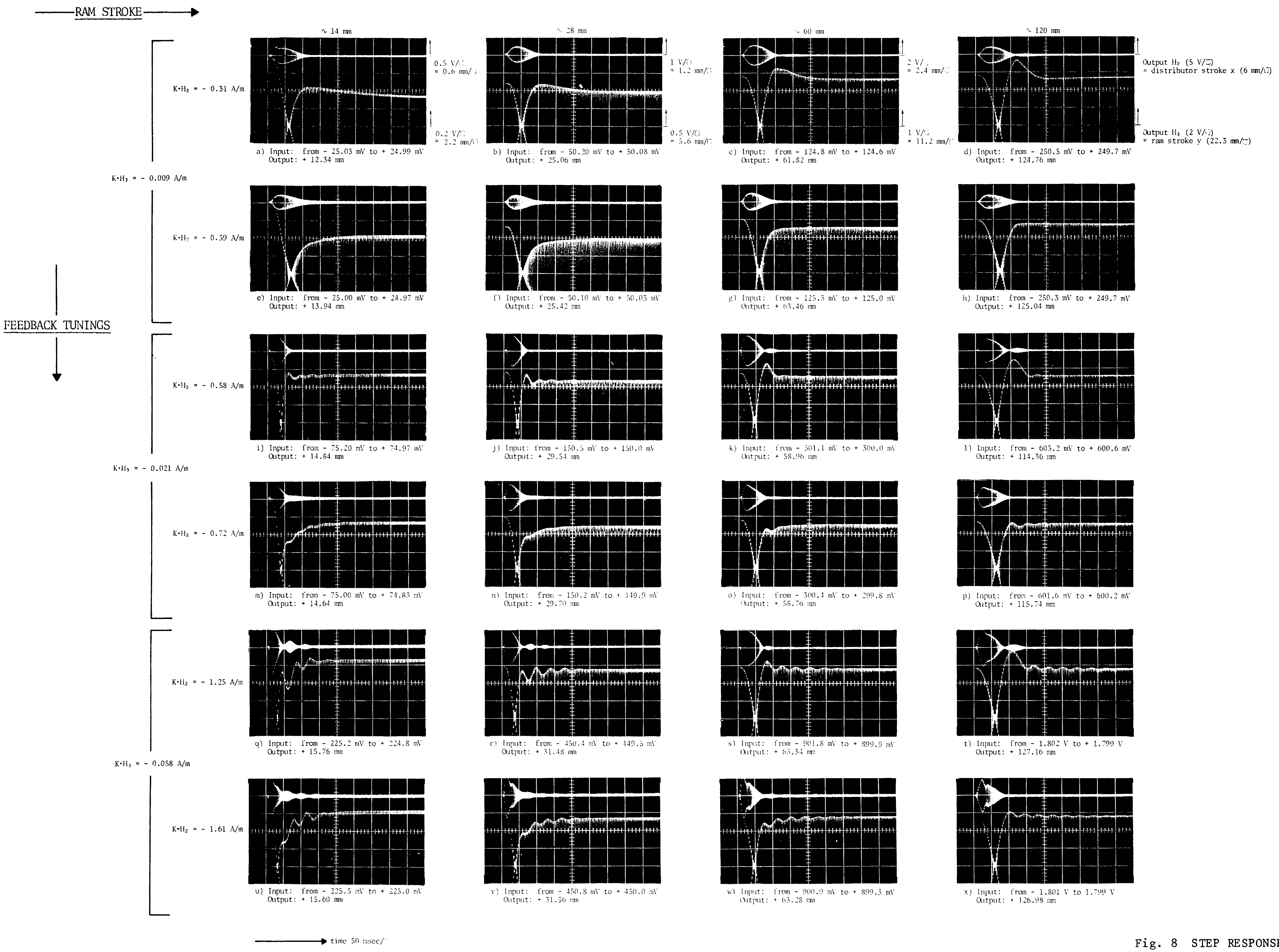


Fig. 8 STEP RESPONSE OF THE ACTUATOR

With different feedback tunings and a dummy load of 500 kg.

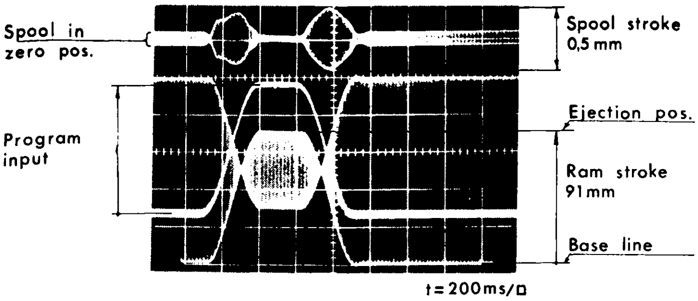
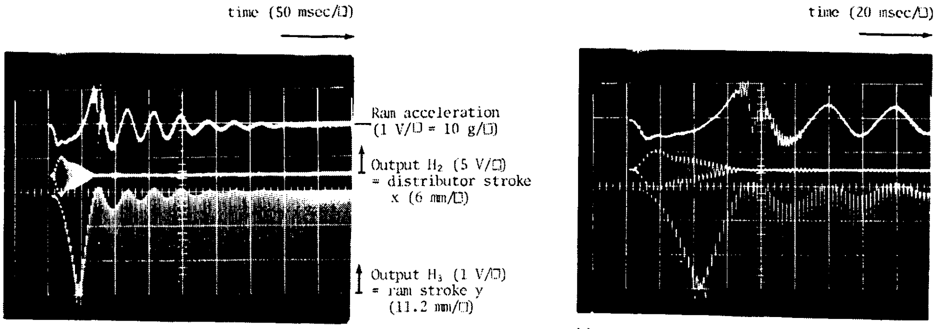


FIG.9 PROGRAMMED ACTUATOR MOVEMENT

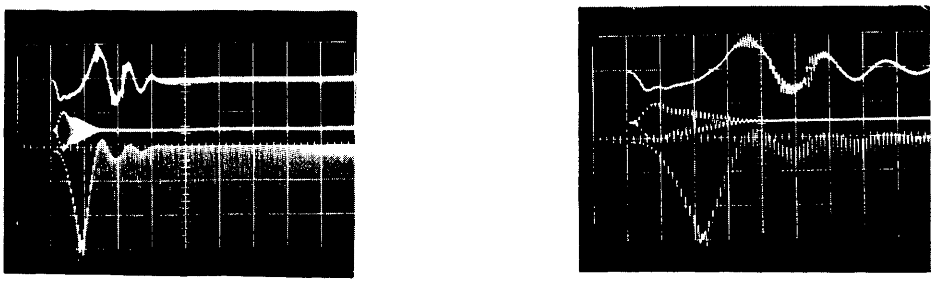
Negative acceleration H_4 feedback: zero



a) Input: from + 954.6 mV
to - 956.2 mV
Output: jump of - 66.98 mm
Pressure influence: - 0.006 mm/bar
Reproducibility: ±0.02 mm
Resolution: 2 mV
Resolution at zero: 7 mV

b)

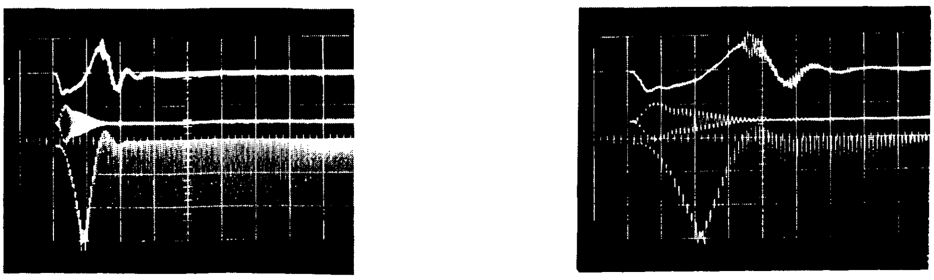
Negative acceleration H_4 feedback: -3×10^{-6} A sec²/m



c) Input: from + 957.6 mV
to - 959.3 mV
Output: jump of - 67.06 mm
Pressure influence: - 0.004 mm/bar
Reproducibility: ±0.02 mm
Resolution: 2 mV
Resolution at zero: 7 mV

d)

Negative acceleration H_4 feedback: -5×10^{-6} A sec²/m



e) Input: from + 956.2 mV
to - 957.9 mV
Output: jump of - 67.00 mm
Pressure influence: - 0.004 mm/bar
Reproducibility: ±0.02
Resolution: 1 mV
Resolution at zero: 8 mV

f)

Fig. 10 STEP RESPONSE OF THE ACTUATOR WITH ACCELERATION FEEDBACK

Dummy load 500 kg
$K \cdot H_2 = -1.34$ A/m for the distributor spool
$K \cdot H_3 = -0.058$ A/m for the ram stroke

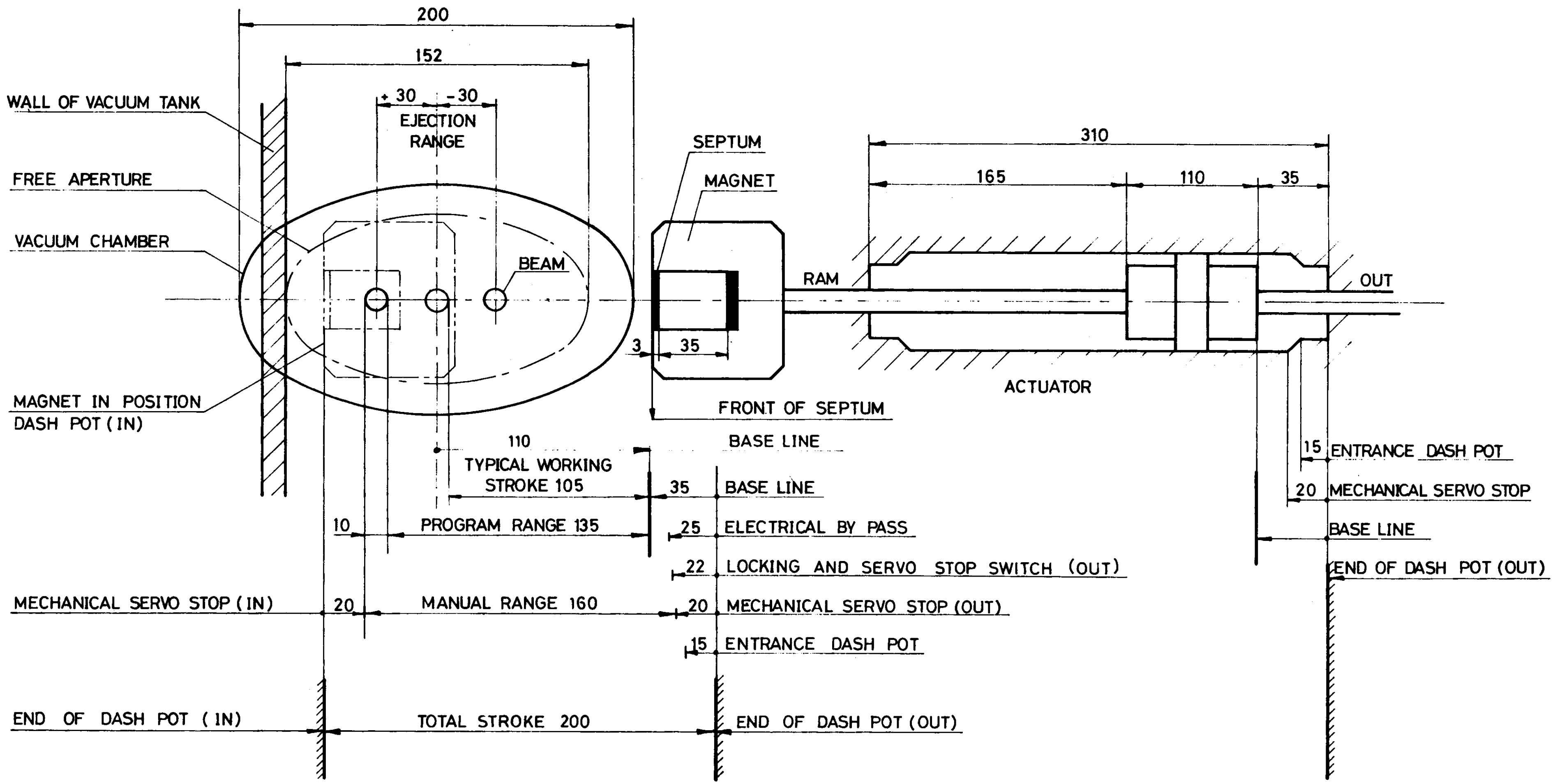


Fig. 11 MAIN DEFINITIONS FOR THE STROKE OF THE ACTUATOR AND THE SEPTUM MAGNET

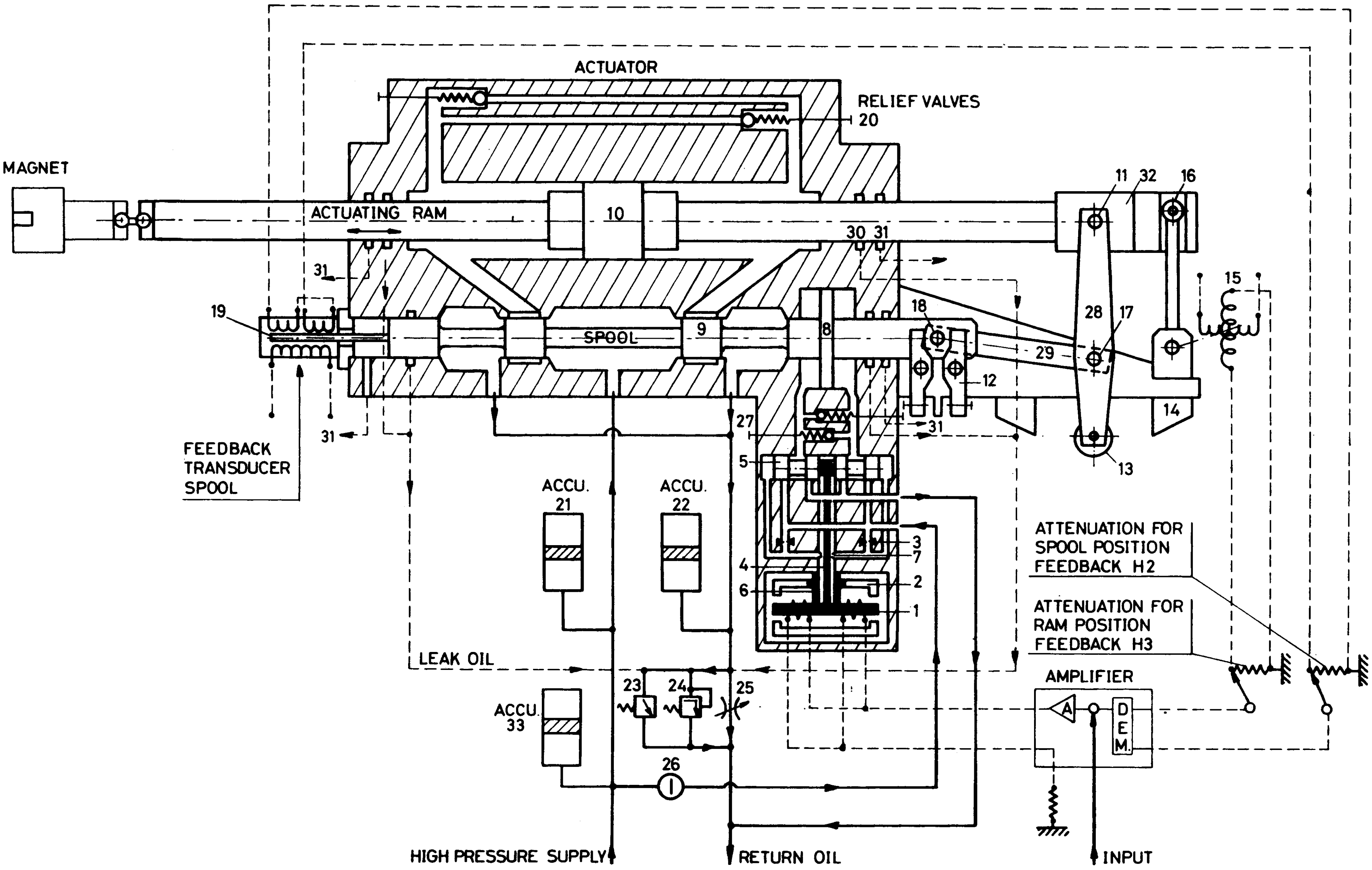


FIG.12 PRINCIPLE OF THE ELECTRO HYDRAULIC SERVO ACTUATOR

(1)_TORQUE MOTOR ARMATURE, (2)_POLEPIECE, (3)_INLET ORIFICE, (4)_FEEDBACK SPRING, (5)_SPOOL, (6)_FLEXURE TUBE, (7)_NOZZLE, (8)_SPOOL PISTON, (9)_SPOOL, (10)_RAM PISTON, (11)_PIVOT POINT, (12)_ADJUSTABLE STOPS FOR SPOOL STROKE, (13)_ROLLER, (14)_STOP FOR THE MECHANICAL FEEDBACK LEVER, (15)_RAM POSITION FEEDBACK RESOLVER, (16)_ROLLER GUIDING FOR ROTATING FEEDBACK LEVER, (17)_PIVOT POINT FOR THE MECHANICAL FEEDBACK LEVER, (18)_SPOOL KNOCKER, (19)_DIFFERENTIAL TRANSDUCER, (20)_RELIEF VALVES, (21)_HIGH PRESSURE ACCU., (22)_RETURN ACCU., (23)_RELIEF VALVE, (24)_PRESSURE REGULATOR, (25)_ADJUSTABLE ORIFICE, (26)_FILTER, (27)_RELIEF VALVES, (28)_MECHANICAL FEEDBACK LEVER, (29)_CONNECTING LINKAGE, (30)_LOW PRESSURE LEAK CHAMBER, (31)_DRIP OIL COLLECTORS, (32)_PIVOT BLOCK. (33)_SECOND TRANSMISSION LINE ACCU

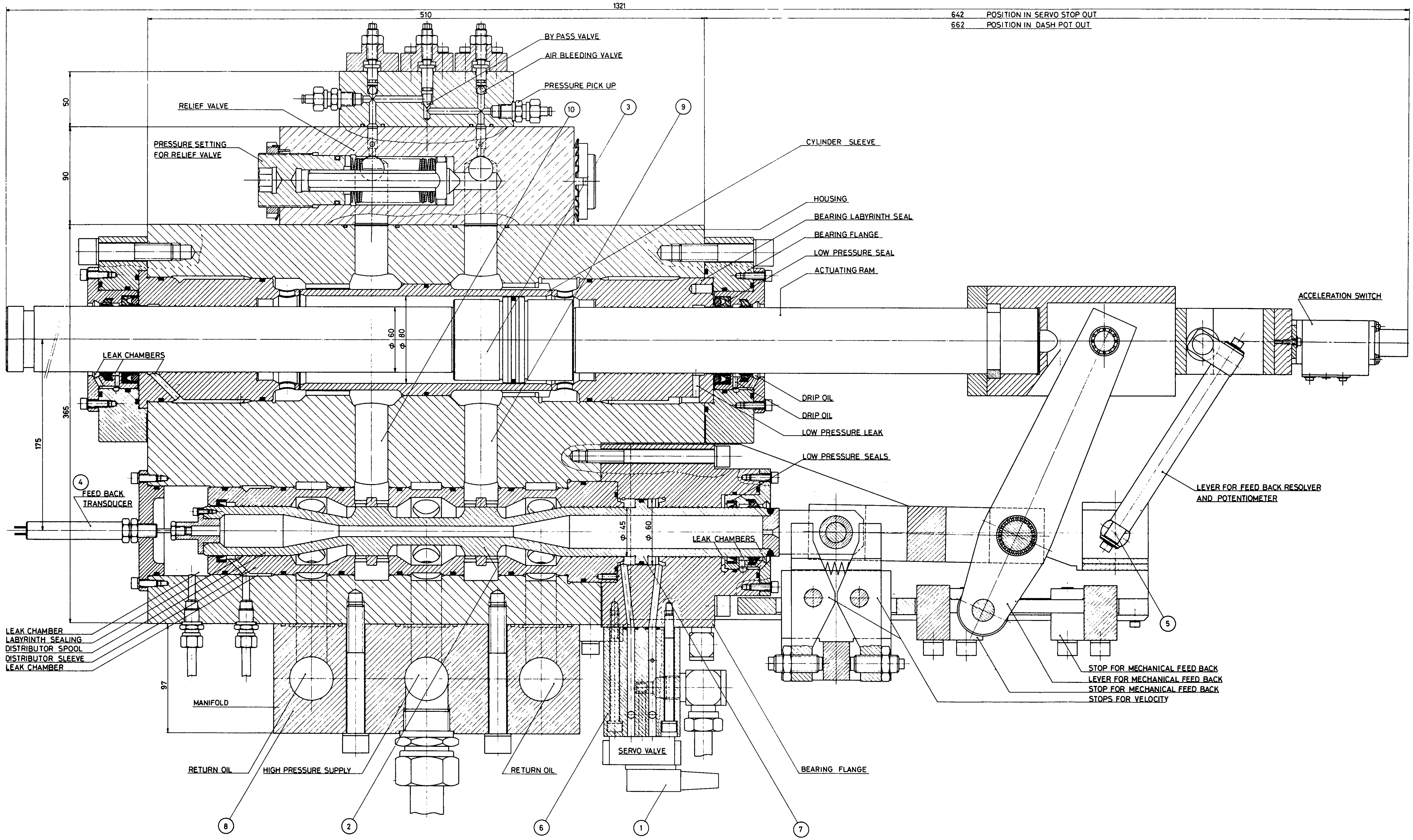


Fig 13 ACTUATOR

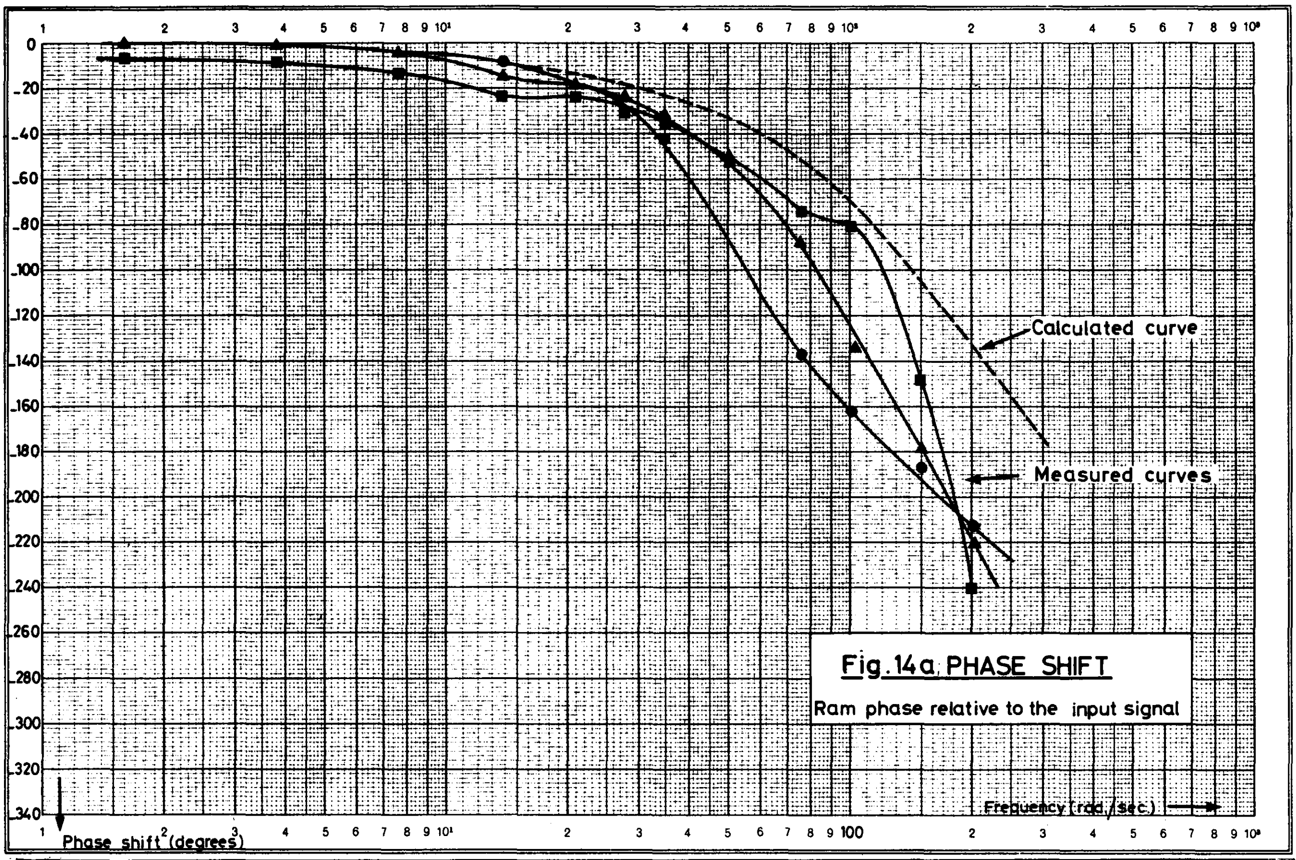


Fig. 14a PHASE SHIFT

Ram phase relative to the input signal

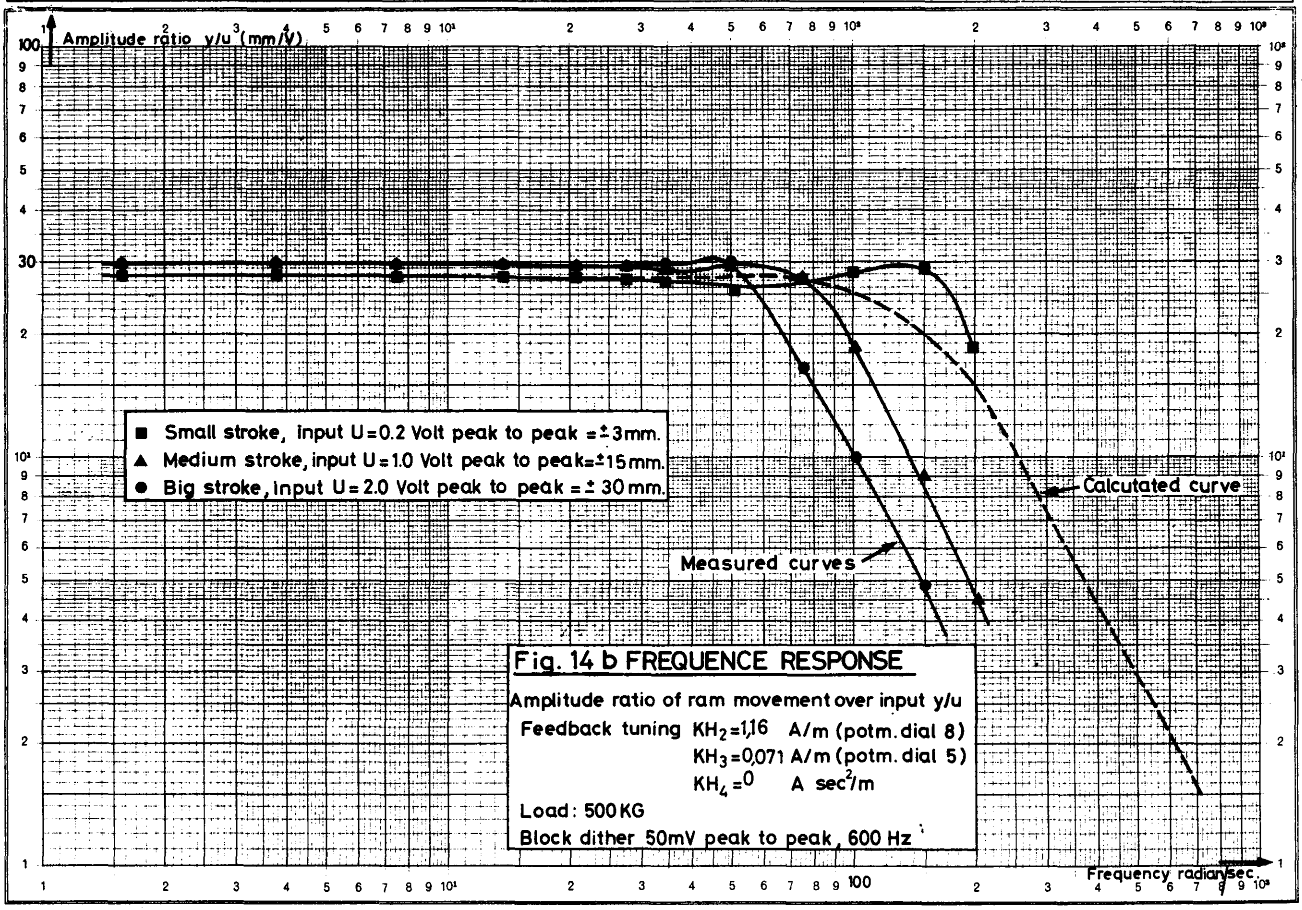


Fig. 14 b FREQUENCE RESPONSE

Amplitude ratio of ram movement over input y/u

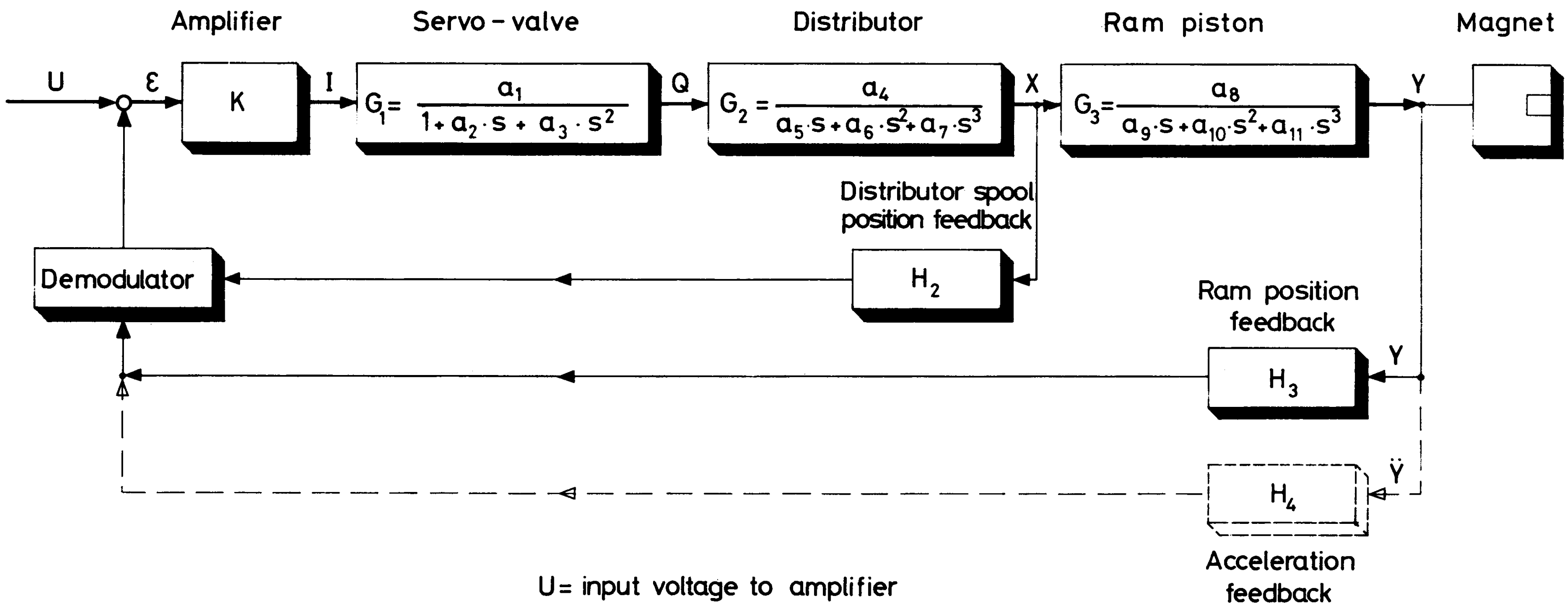
Feedback tuning KH_2=1,16 A/m (potm. dial 8)

KH_3=0,071 A/m (potm. dial 5)

KH_4=0 A sec^2/m

Load: 500 KG

Block dither 50mV peak to peak, 600 Hz



U= input voltage to amplifier
I = output current from amplifier
Q= output oil flow from servo-valve
X = distributor spool position
Y = ram position

Fig. 15 Block diagram of the Servo system

H_4, Acceleration feedback optional

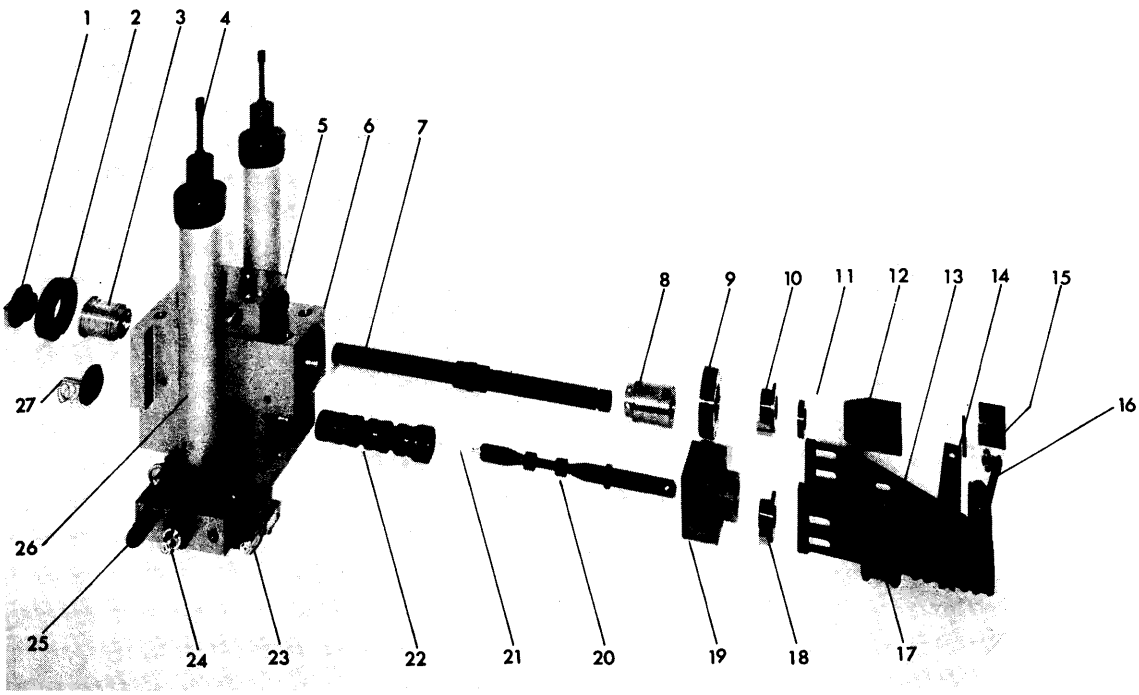


Fig. 16 EXPLODED VIEW OF THE ACTUATOR

(1) seal flange (front side), (2) main flange (front side), (3) front bearing, (4) prolongation tube for the magnet of the magnetic switch, (5) manifold for relief and by pass control, (6) actuator housing, (7) actuating ram, (8) rear bearing, (9) main flange (rear side), (10) seal flange (rear side), (11) flange for the ram head, (12) ram head, (13) support for velocity, mechanical and electrical feedbacks, (17) adjustment for velocity limitation, (18) seal flange (spool), (19) spool piston cylinder, (20) spool, (21) probe for spool position transducer, (22) distributor sleeve, (23) pressure regulator for the back pressure, (24) orifice for return oil, (25) relief valve for the back pressure, (26) low pressure accumulator (return), (27) spool position transducer feedback.

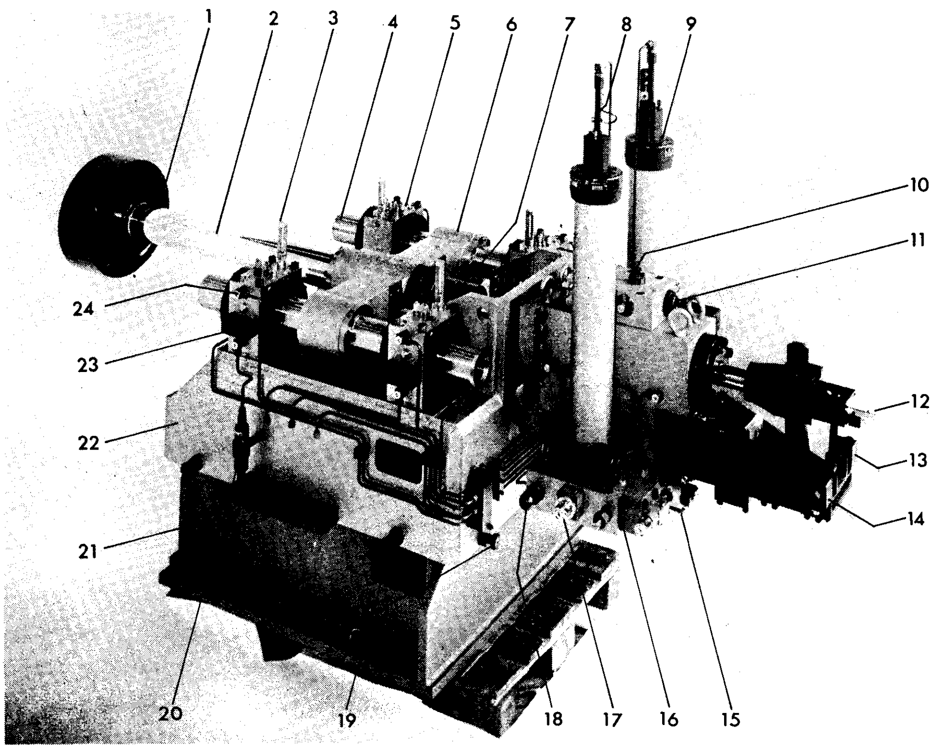


Fig. 17 ACTUATOR AND HYDROSTATIC GUIDING SYSTEM (side view)

(1) dummy load 400 kg, (2) shaft, (3) thermometer, (4) shaft of the hydrostatic guiding system, (5) hydrostatic bearing, (6) supporting traversal of the hydrostatic guiding system, (7) ball connections, (8) magnetic switch for max. position of the accumulator piston, (9) high pressure accumulator, (10) air bleeding valve, (11) pressure setting screw of the relief valve, (12) ram acceleration transducer, (13) ram stroke feedback resolver, (14) ramstroke transducer for visual indication, (15) electro-hydraulic servo valve, (16) pressure regulator for back pressure, (17) orifice for return oil, (18) relief valve for back pressure, (19) adjustment screws for the supporting frame, (20) flow meter, (21) support for the test, (22) supporting frame, (23) temperature switch, (24) supply pressure for the hydrostatic bearings.

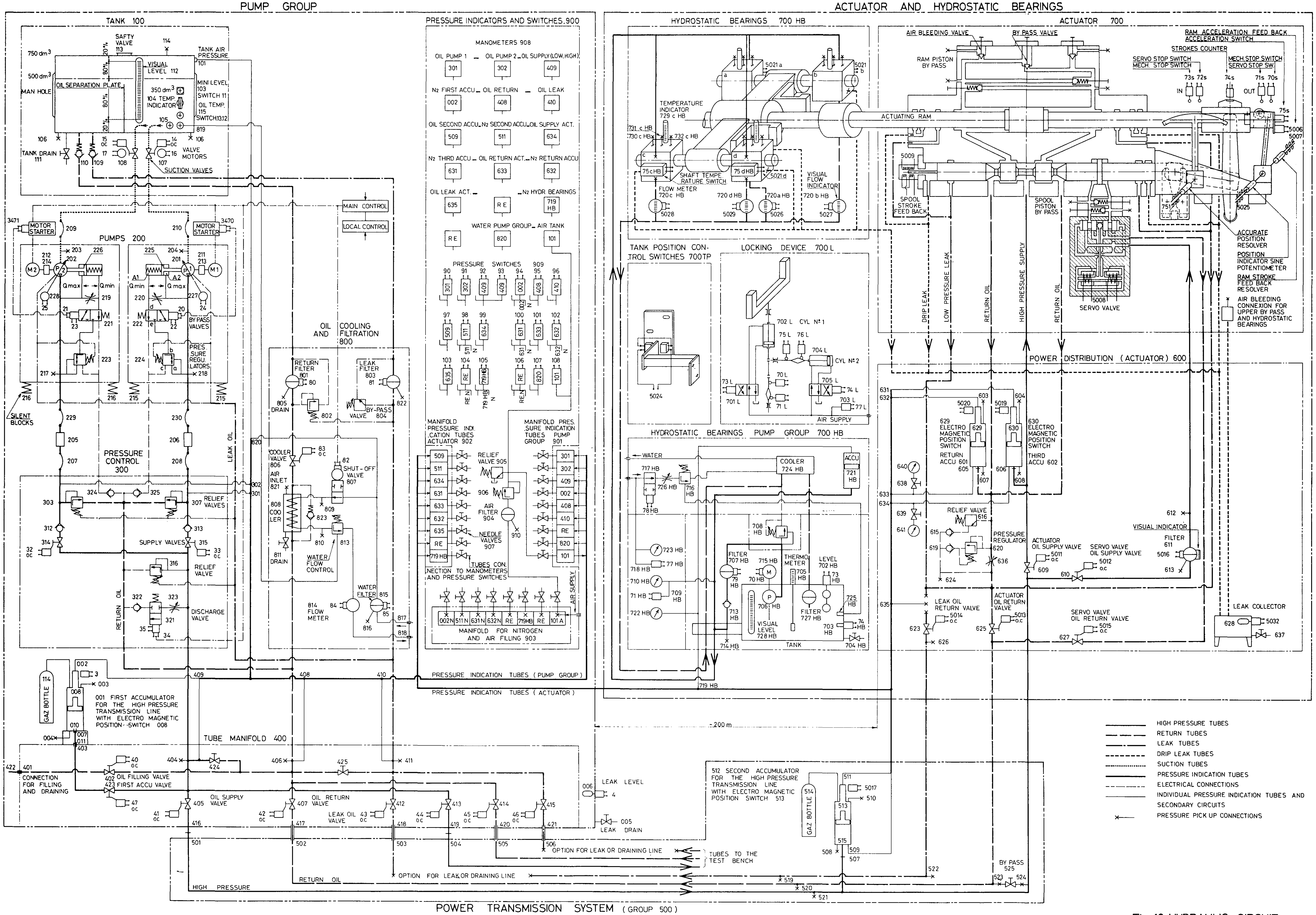


Fig. 18 HYDRAULIC CIRCUIT

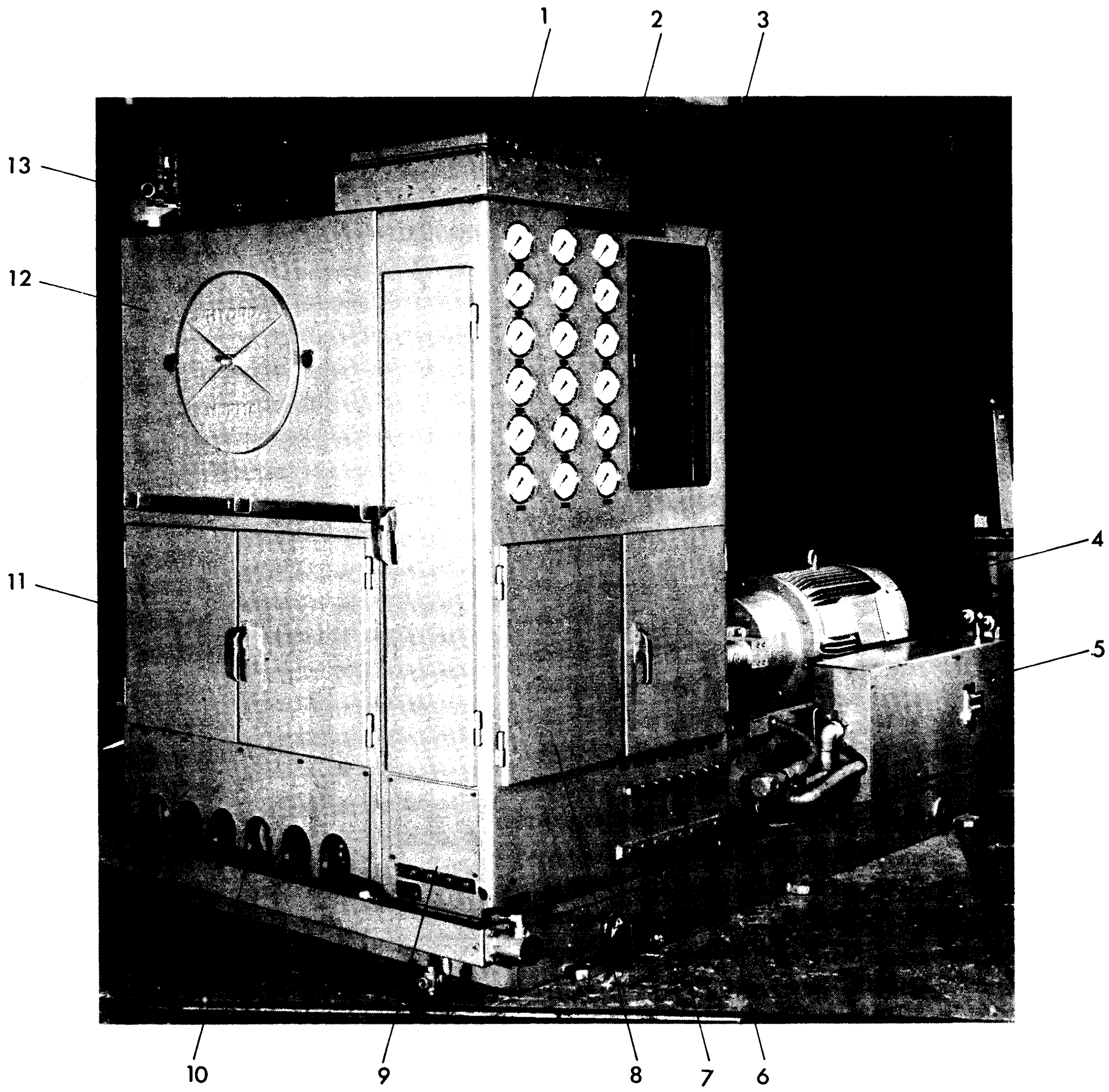


Fig. 19 PUMP GROUP (2 × 90 dm^3/min, 200 bar) (general view)

(1) electrical connection box for the interlocks, (2) manometers for the main pressures, (3) compartment for temperature and level indicators, (4) pump motor 40 kW, (5) compartment for the cooling water filtration and temperature control system, (6) manifold for nitrogen and air filling system (7) shut off valves for the nitrogen and air filling system, (8) compartment for the pressure switches of the interlock system, (9) connections for the pressure indications tubes, (10) connections for the high pressure and return oil power transmission system, (11) compartment for the general pressure controls and main manifold with shut off valves for the power transmission system, (12) oil tank, (13) high pressure accumulator for the supply pressure.

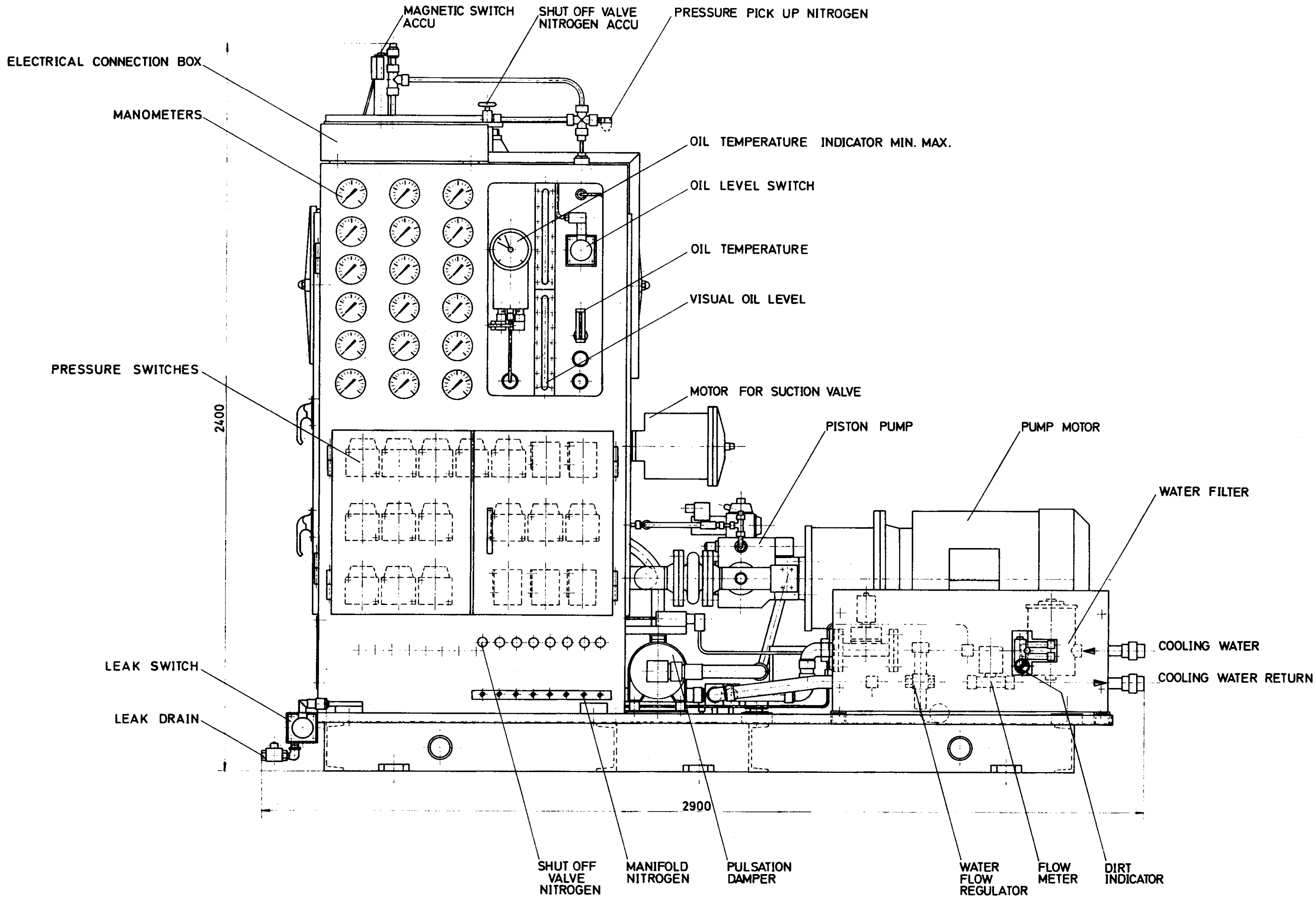


Fig. 19a PUMP GROUP
(side view)

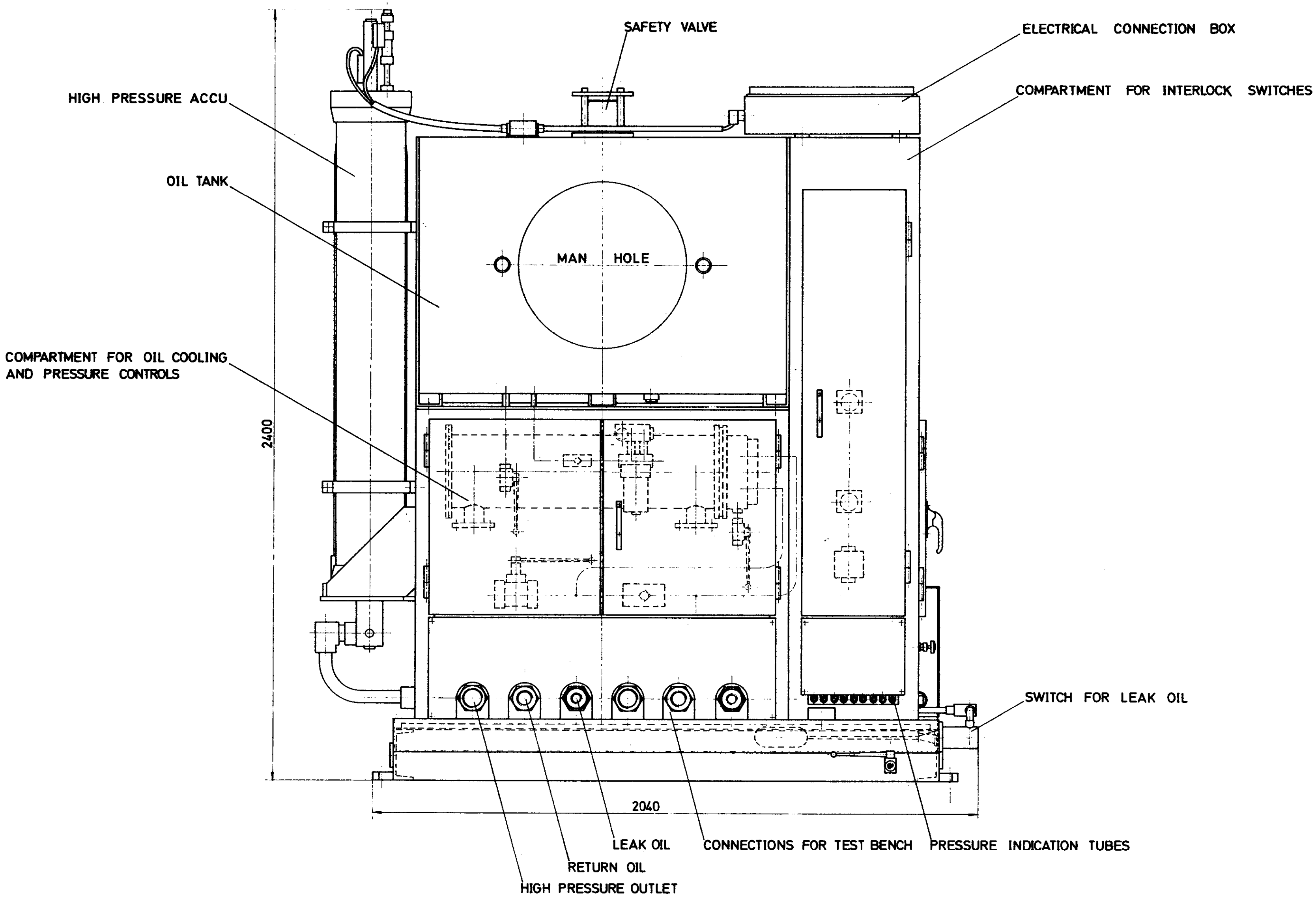
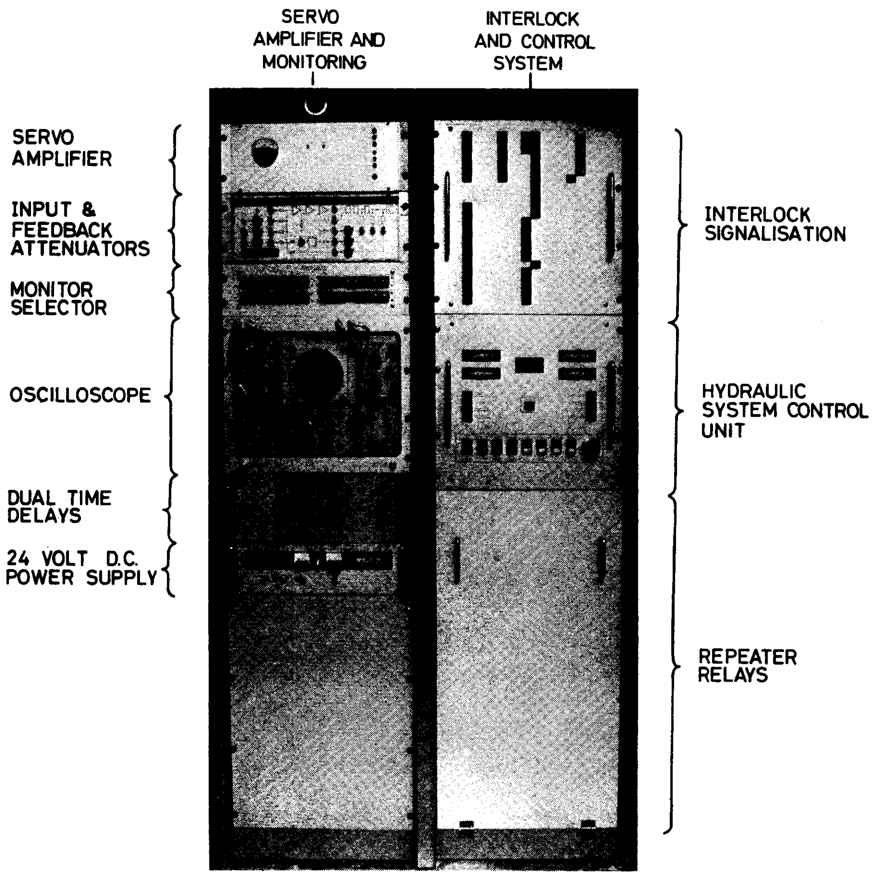


Fig 19b PUMP GROUP
(front view)



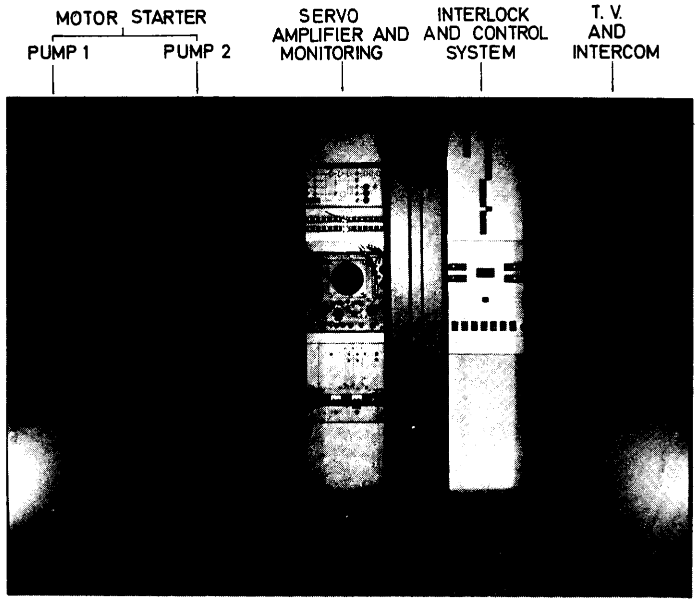


Fig 20 CENTRAL CONTOL AND INTERLOCK RACKS IN PUMP STATION

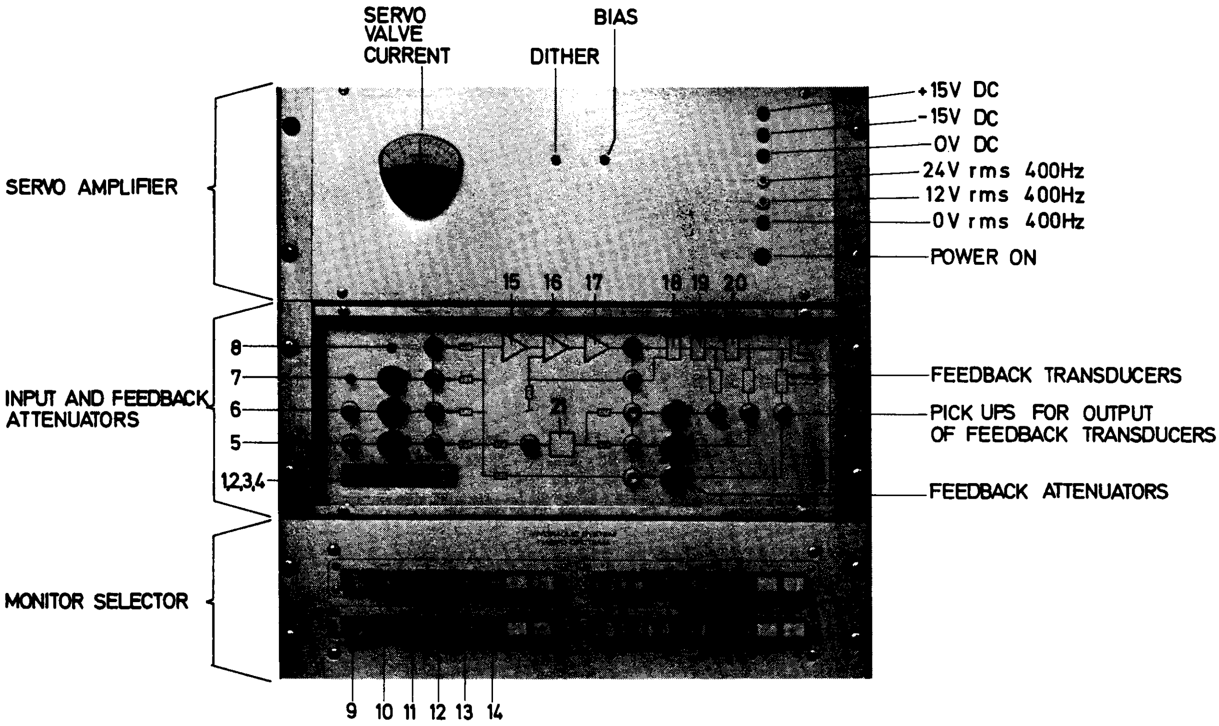


Fig. 20a SERVO AMPLIFIER AND MONITOR SELECTOR

(1) electronic only, (2) pumps only, (3) test bench, (4) cancel, (5 and 6) test inputs, (7) step input ±5 V, (8) program attenuation, (9) program, (10) servo-valve current, (11) spool stroke, (12) ram stroke, (13) ram acceleration (14) demodulator output, (15) D.C. amplifier, (16) differ. amplifier, (17) power amplifier, (18) servo valve, (19) distributor valve, (20) ram, (21) demodulator.

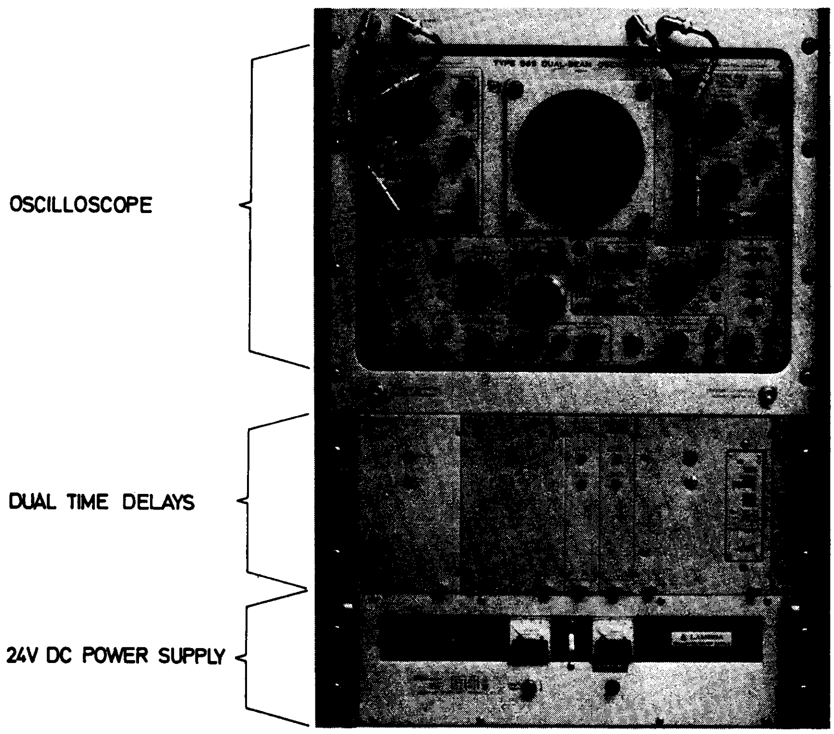


Fig. 20b FOUR CHANNEL OSCILLOSCOPE FOR MONITORING

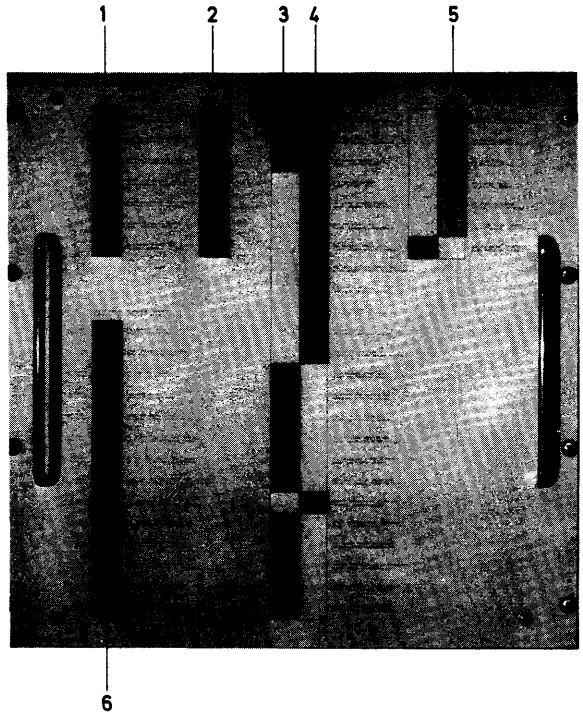


Fig. 20c HYDRAULIC SYSTEM SIGNALISATION (PST)

(1 and 2) interlock lamps for pump 1 and 2, (3 and 4) system pressures too low or high, (5) signalisation lamps only, (6) interlock lamps for the shut-off valves in straight section 24.

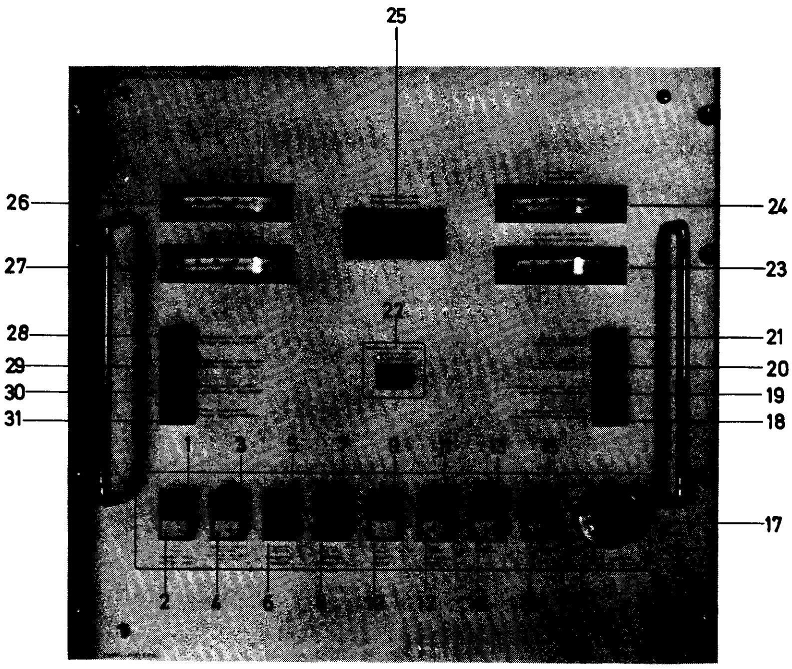


Fig. 20d HYDRAULIC SYSTEM CONTROL UNIT (PST)

(1) 24 V on, (2) 24 V off, (3) reset, (4) suction valve close, (5) selection pump 1, (6) selection pump 2, (7) pump station control, (8) local control (L.C.R), (9) pump 1 on, (10) pump 1 off, (11) pump 2 on, (12) pump 2 off, (13) unlock, (14) lock, (15) program, (16) manual, (17) stroke control, (18) stroke control zero, (19) hydraulic system ready, (20) servo stop (INT), (21) servo stop (EXT), (22) emergency switch, (23) ram position, (24) servo input, (25) stroke counter, (26) flow of pump 1, (27) flow of pump 2, (28) mechanical stop (EXT), (29) mechanical stop (INT), (30) acceleration high, (31) tank alignment.

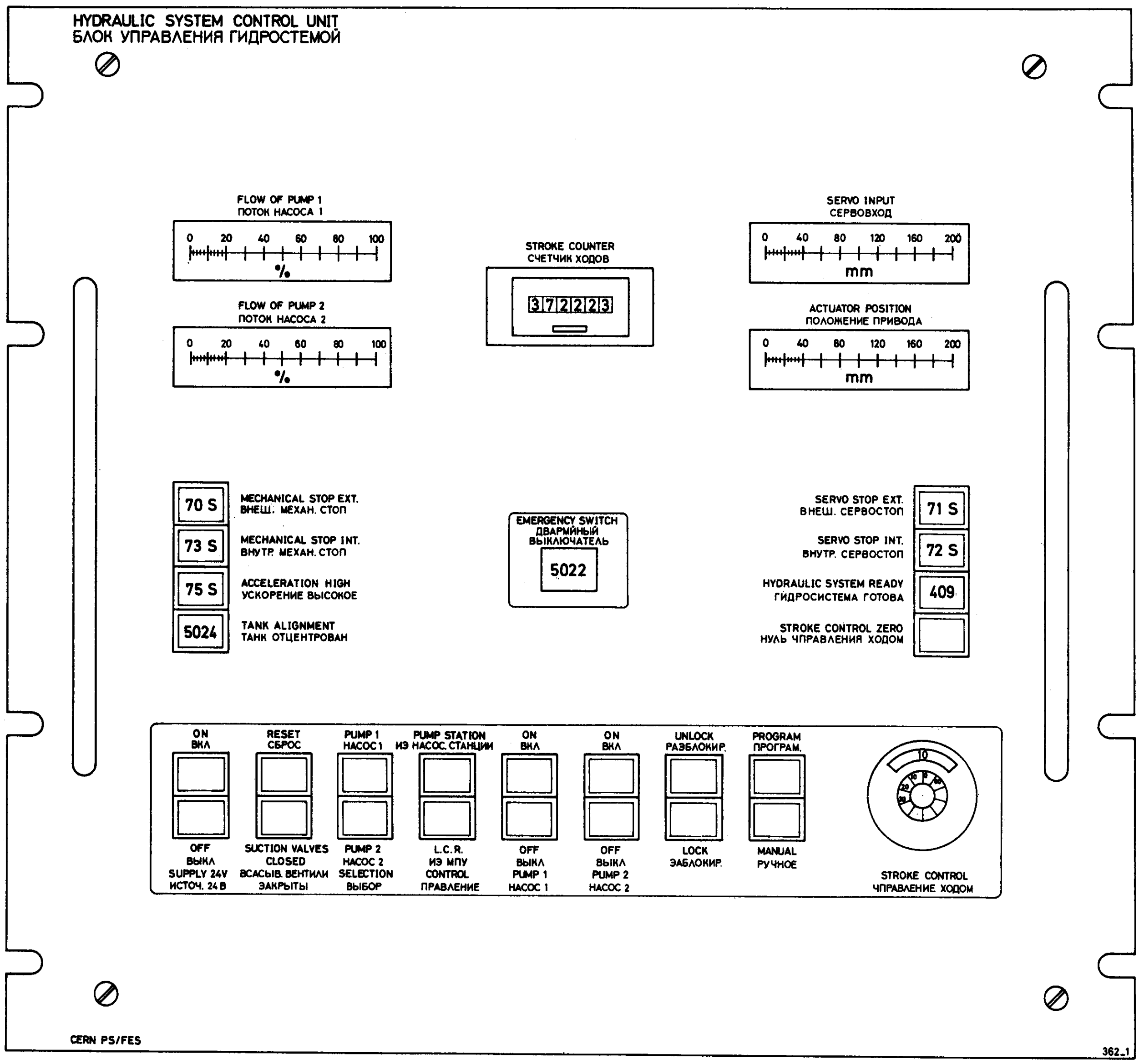


Fig. 20e CENTRAL CONTROL (PST)

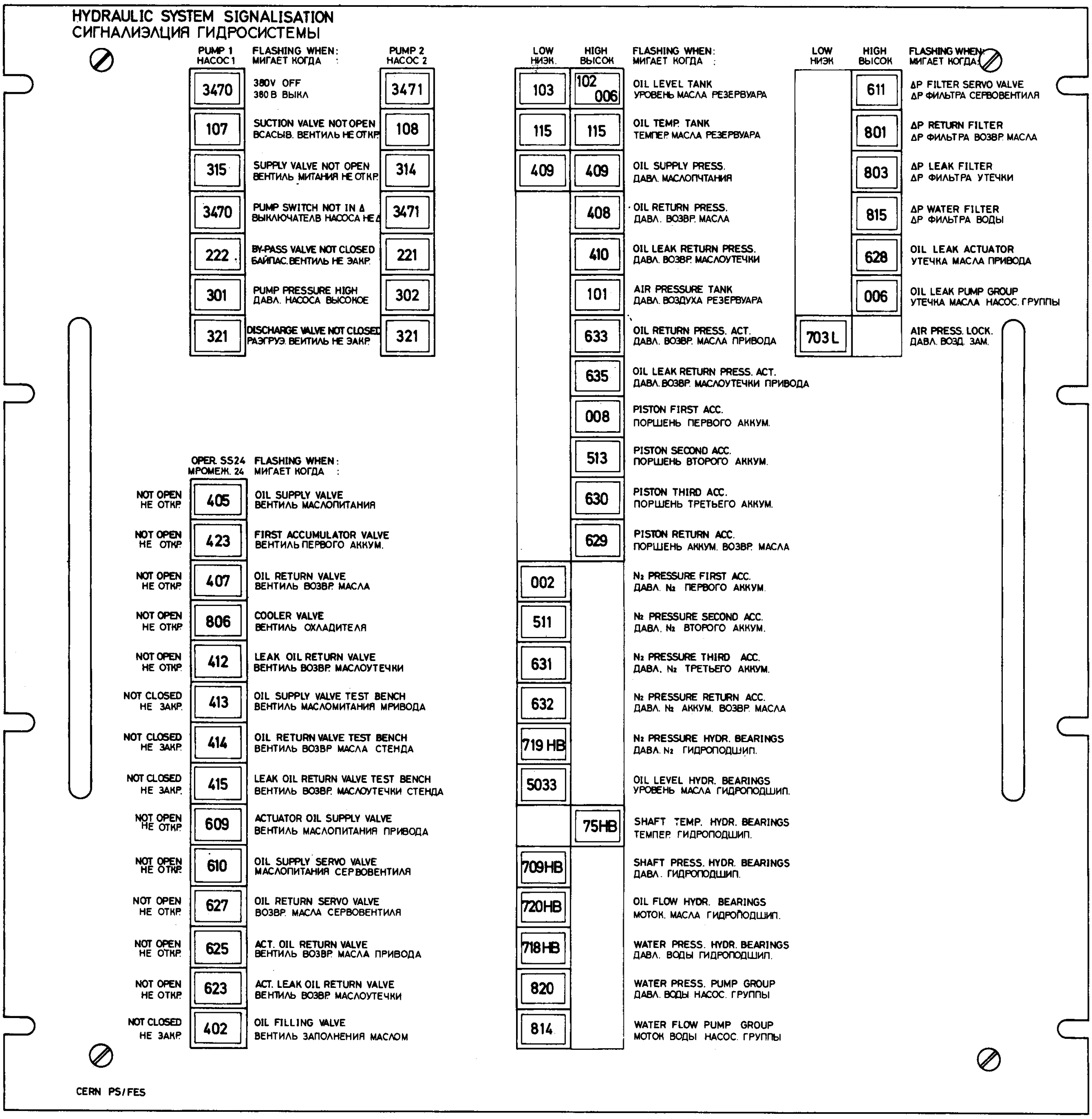


Fig.20f HYDRAULIC INTERLOCK SIGNALISATION (PST)

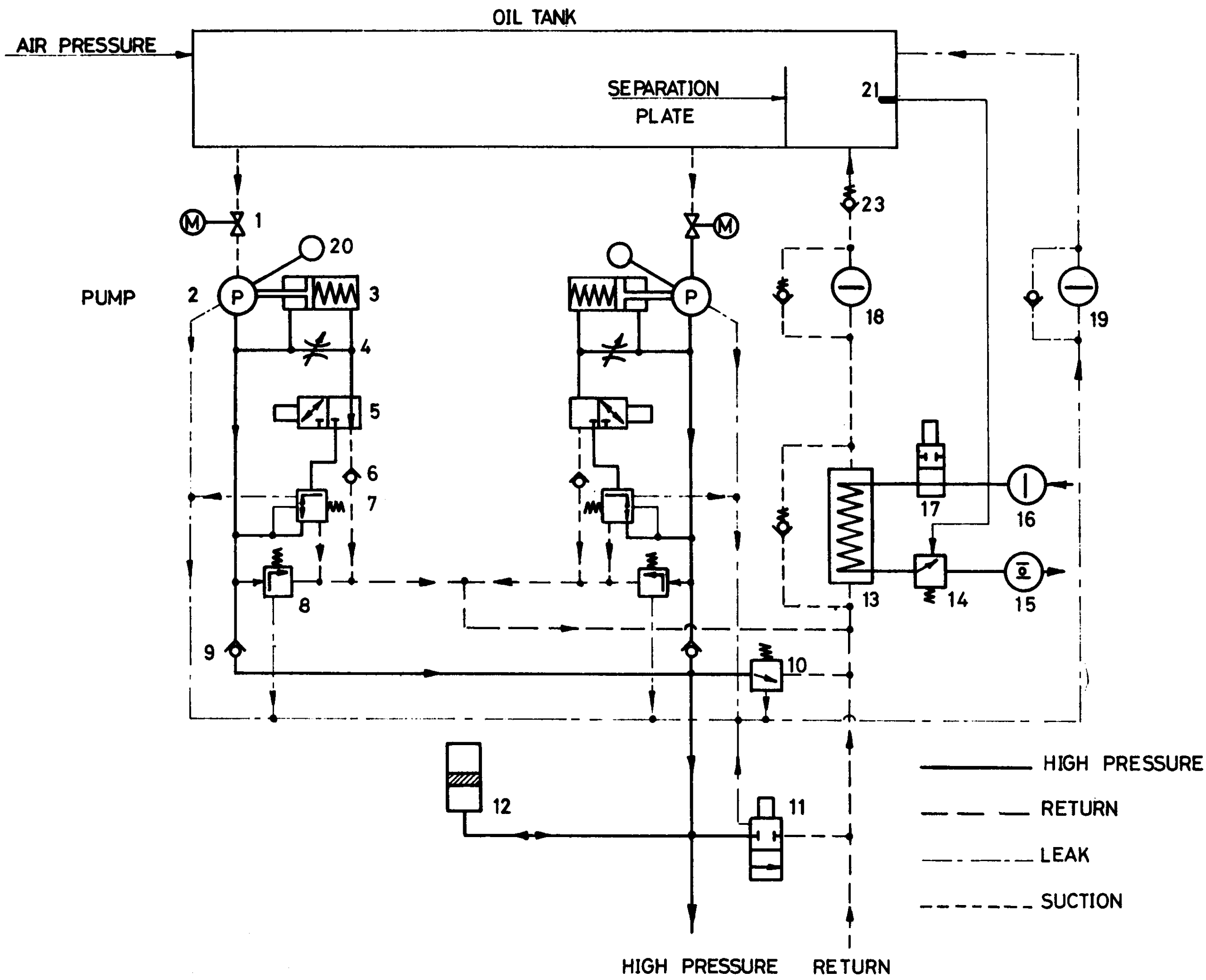


Fig 21. SIMPLIFIED DIAGRAM OF THE HYDRAULIC PUMP GROUP

(1)-SUCTION VALVE WITH MOTOR, (2)-VARIABLE DISPLACEMENT PUMP, (3)-SWASH PLATE ACTUATOR, (4)-THROTTLE, (5)- 4 WAY BY PASS VALVE, (6)- CHECK VALVE, (7)-PRESSURE REGULATOR, (8)-RELIEF VALVE, (9)-CHECK VALVE, (10)-RELIEF VALVE, (11)-DISCHARGE VALVE, (12)-ACCUMULATOR, (13)-COOLER, (14)- WATER FLOW CONTROLER, (15)- FLOW METER, (16)-WATER FILTER, (17)- SHUT OFF VALVE, (18)-RETURN FILTER, (19)- LEAK FILTER, (20)-SWASH ANGLE TRANSDUCER, (21)-THERMOMETER, (22)-AIR PRESSURE INLET (23)-CHECK VALVE

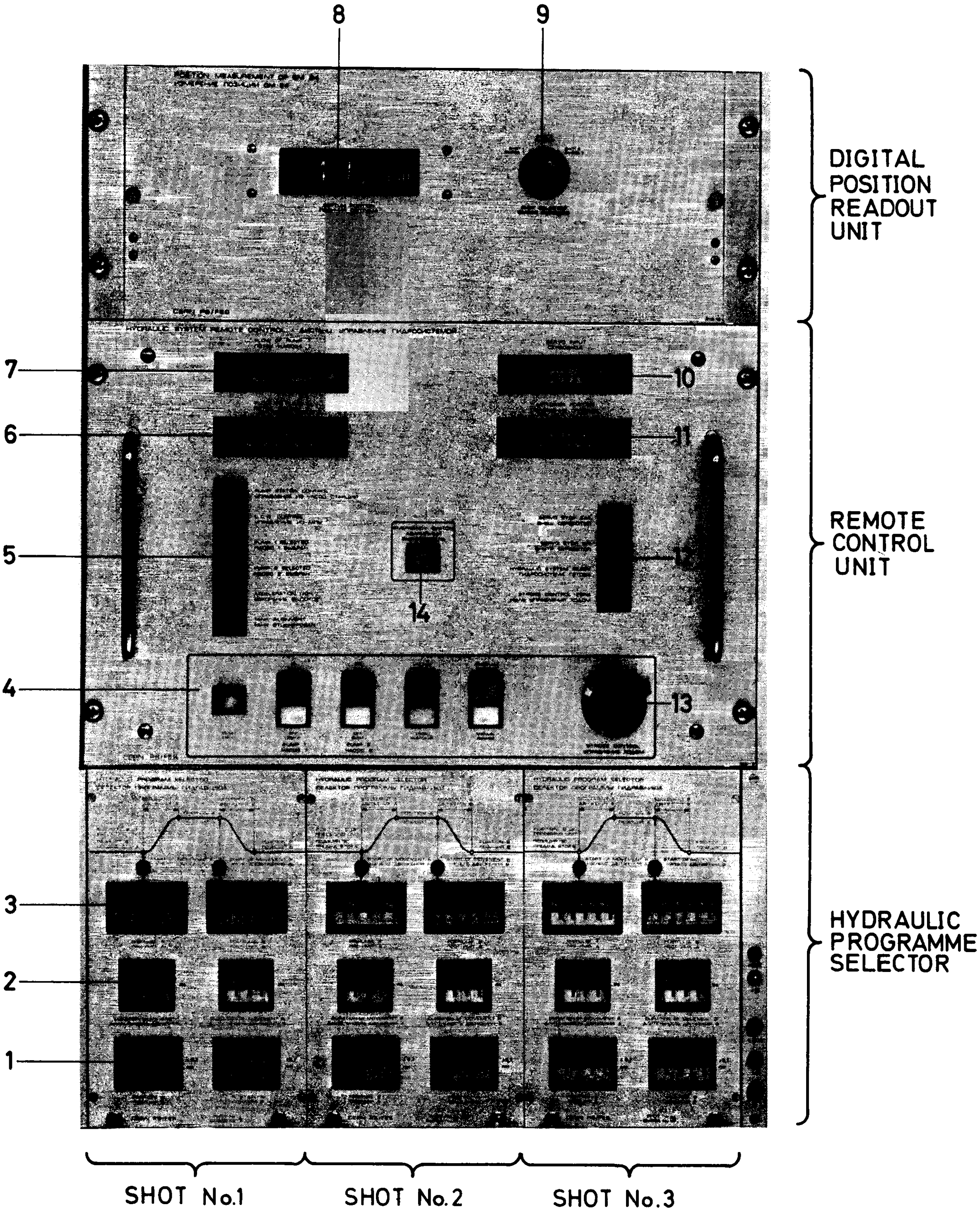


Fig. 22 REMOTE CONTROL STATION IN THE LOCAL CONTROL ROOM

(1) ejection and withdrawal positions, (2) duration time in ejection position, (3) timing of inwards and return movements, (4) buttons for: reset, lock-unlock, program-manual, (5) summary interlocks and mode of operation, (6) flow of pump 2, (7) flow of pump 1, (8) ram position read out, (9) shot selector, (10) servo input, (11) ram position, (12) servo stop indications and hydraulic system ready indications, (13) stroke control, (14) emergency stop.

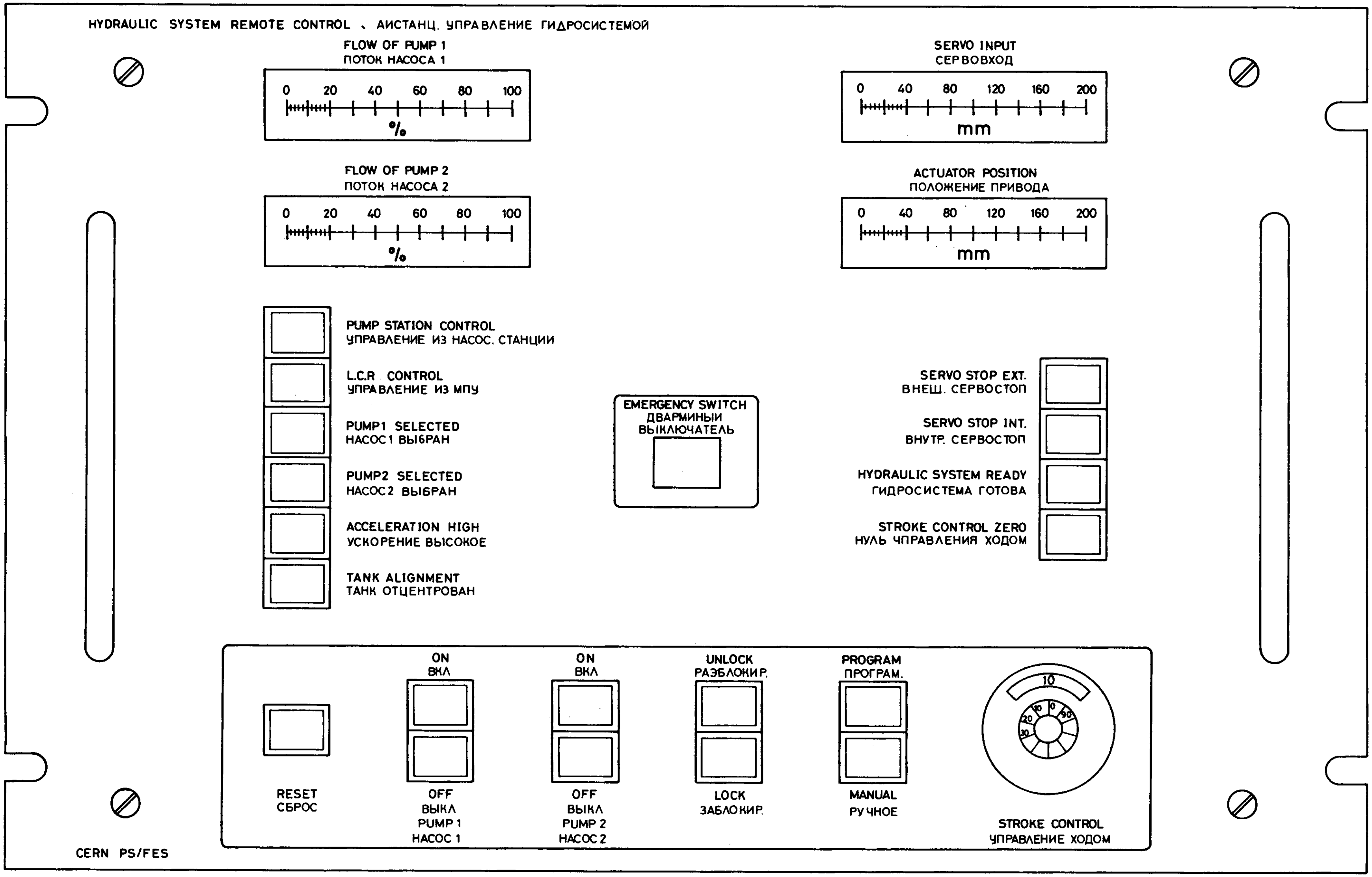


Fig. 22a REMOTE CONTROL (LCR)

1. SUFFIXES GIVEN TO INTERLOCK NAMES ie (HIGH / LOW) etc. ARE 'FAULT CONDITIONS'

2. CONTACTS ARE SHOWN CLOSED ( ) SIGNIFYING 'NO FAULT' EXCEPT IN THE CASE OF BY-PASSED INTERLOCKS WHICH ARE SHOWN OPEN ( ) WITH THE BY-PASS CLOSED

3. IF A WRITTEN CONDITION IS FULFILLED THE SIGNAL IS 'YES'

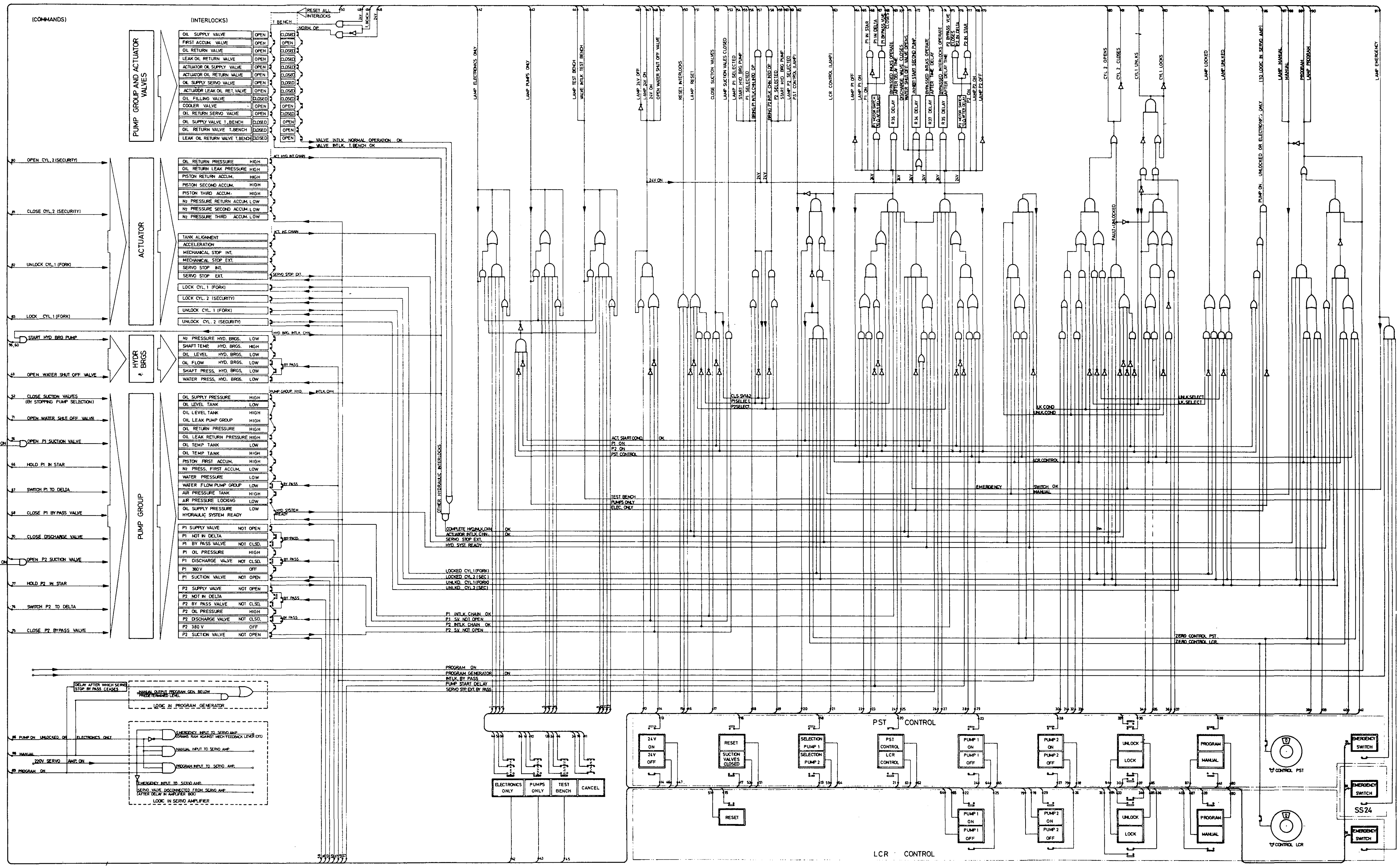
4.a. THE ACTIONS OF THE EMERGENCY SWITCH, INTERLOCK FAULT OR 220V BREAK DOWN OF THE PROGRAM GENERATOR ARE THUS:-
STOP PUMPS
EMERGENCY INPUT TO SERVO AMPLIFIER
AUTOMATIC LOCKING

b. A 220V BREAK DOWN OF THE SERVO AMPLIFIER GIVES THE SAME ACTIONS TOGETHER WITH:-
THE START OF A TIME DELAY AFTERWHICH THE SERVO VALVE IS DISCONNECTED FROM THE SERVO AMPLIFIER (THIS AVOIDS BANGING OF THE RAM THE MOMENT THE CAPACITORS OF THE AMPLIFIER POWER SUPPLY BECOME EXHAUSTED)

5. PUSH BUTTONS ARE SHOWN IN THE 'UNPUSHED' POSITION

6. AND OR INVERTER

FIG 23. BRIEF EXPLANATORY NOTES FOR THE BLOCK DIAGRAM OF THE HYDRAULIC INTERLOCK SYSTEM



PES DRG NO. 362 - 119 - 0

FIG. 23 BLOCK DIAGRAM OF THE HYDRAULIC INTERLOCK SYSTEM

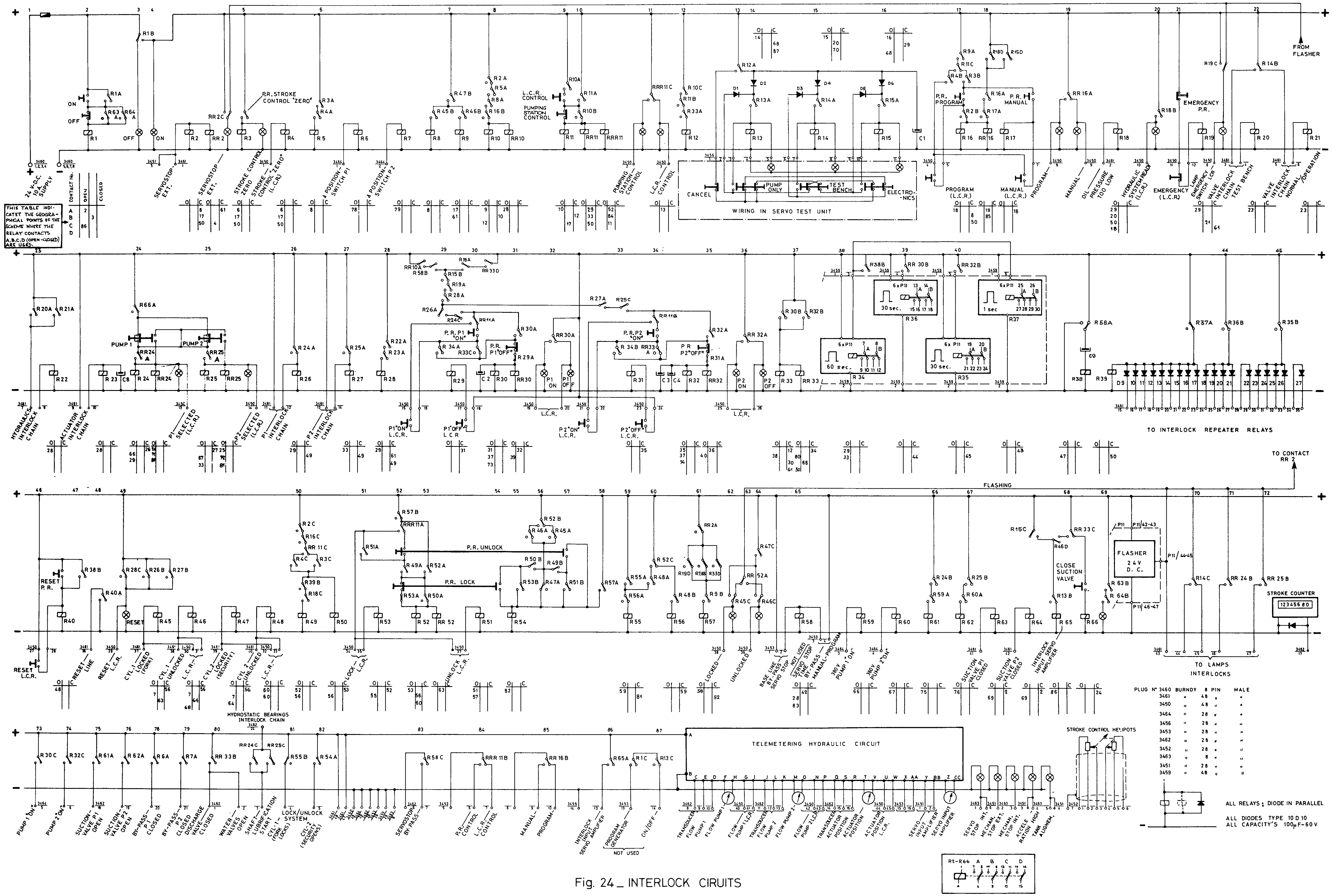


Fig. 24_ INTERLOCK CIRUITS

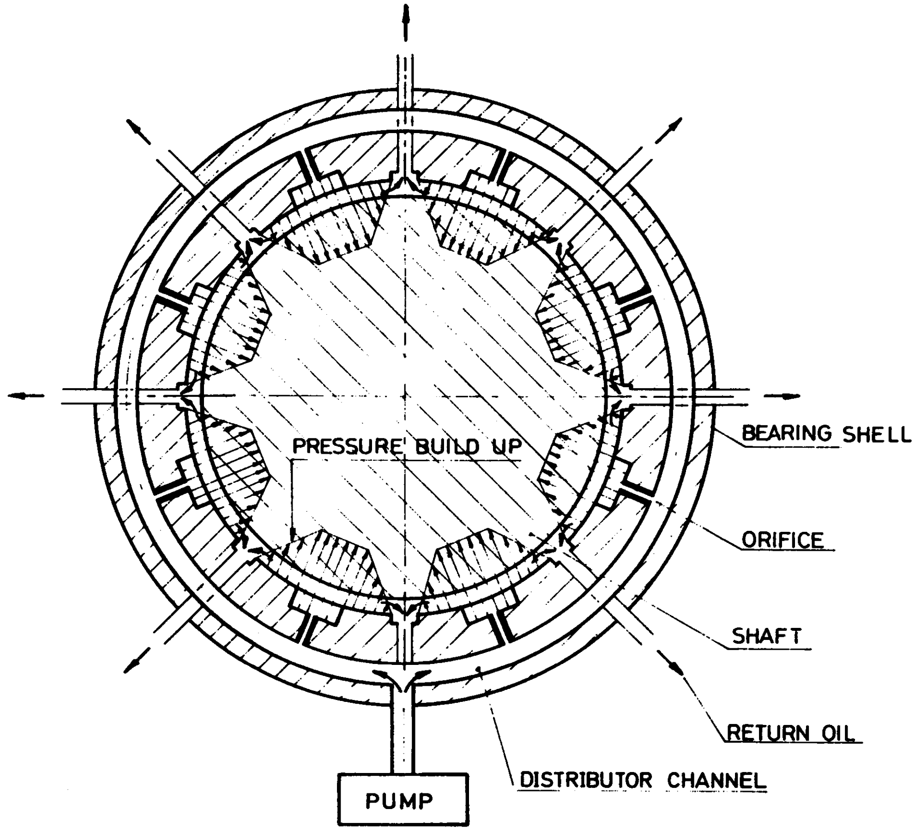


Fig.25a.HYDROSTATIC BEARING
UNLOADED
SYMMETRICAL PRESSURE BUILD UP

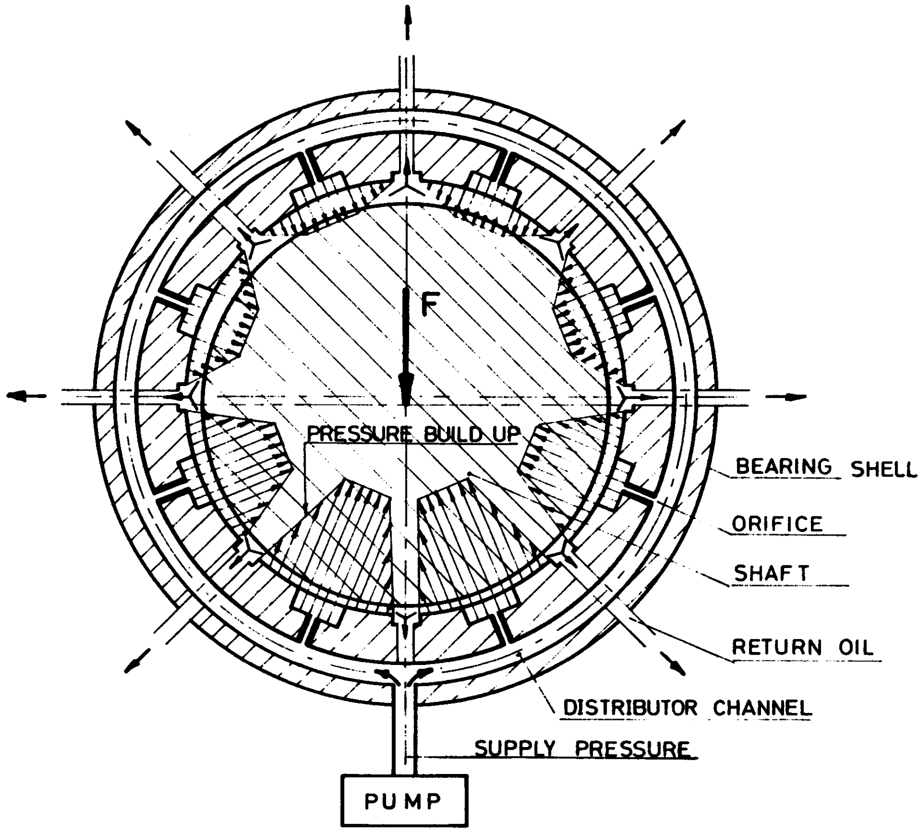


Fig.25 b.HYDROSTATIC BEARING
LOADED
UNSYMMETRICAL PRESSURE BUILD UP

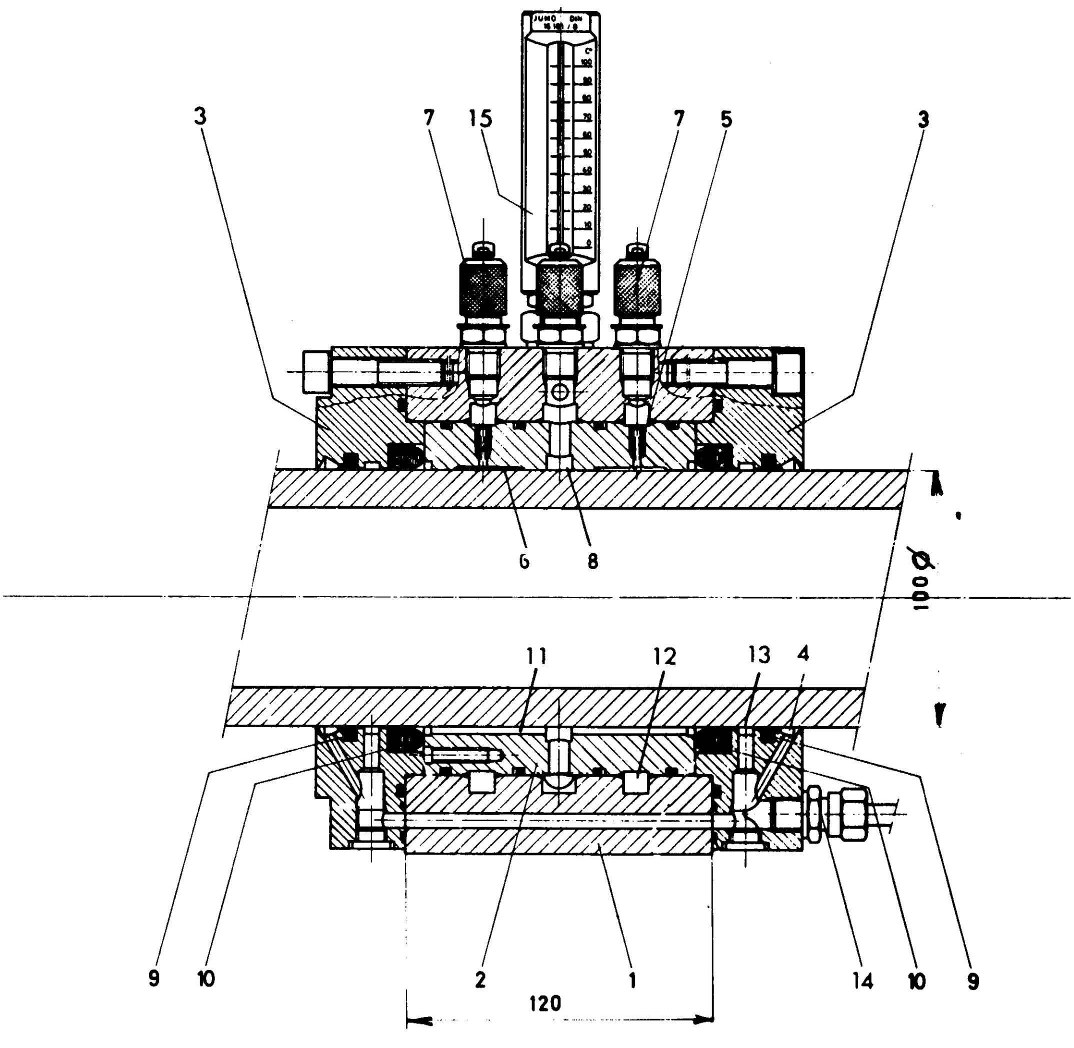


Fig. 26 HYDROSTATIC BEARING

(1) housing, (2) bearing shell, (3) flange, (4) drip collector, (5) orifice, (6) recesses, (7) self sealing pressure connector, (8) collector duct, (9) scraper, (10) seal, (11) groove for return oil, (12) high pressure distribution channel, (13) leak oil, (14) leak and drip oil collector, (15) thermometer.

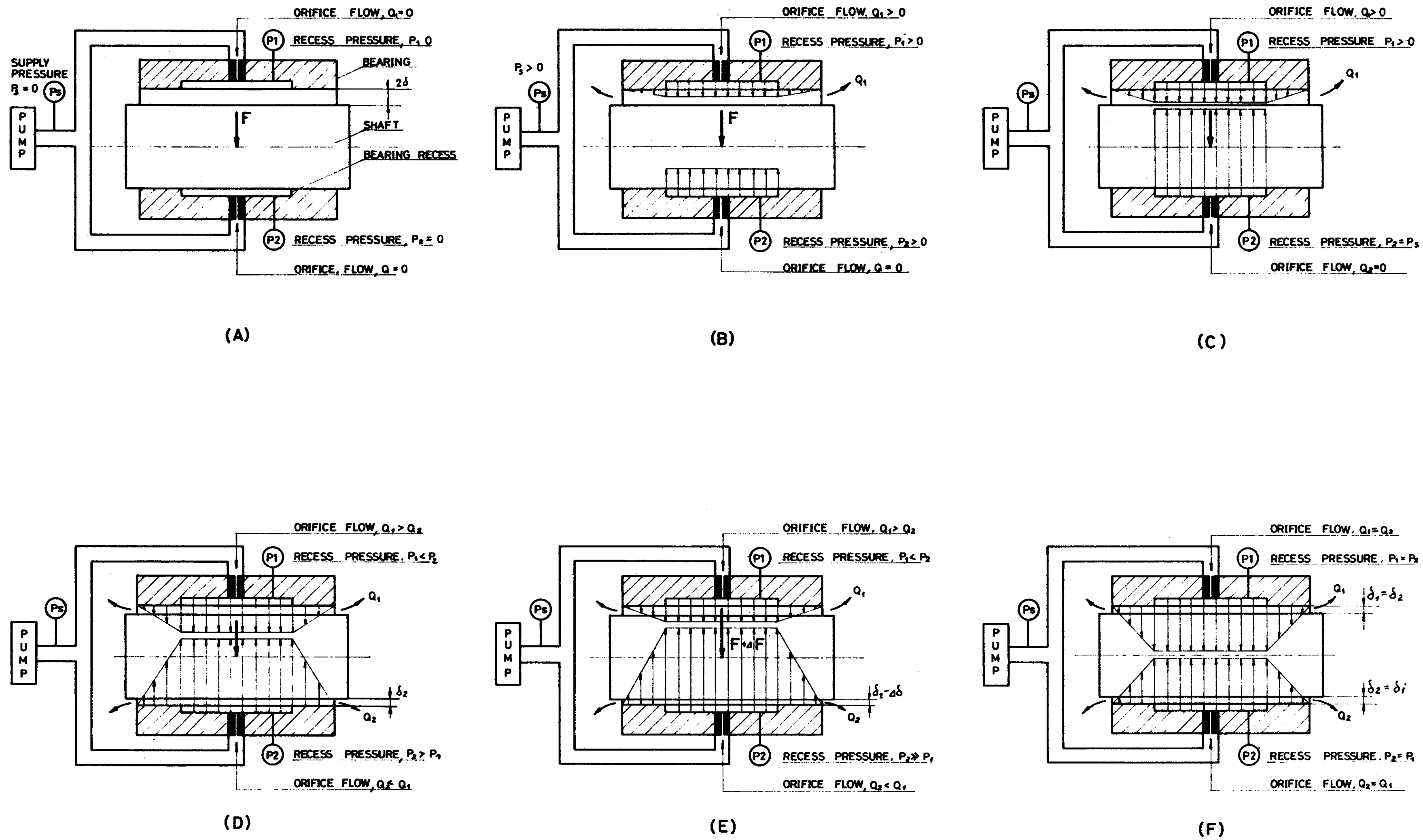


FIG. 27 PRINCIPLE OF THE HYDROSTATIC BEARINGS

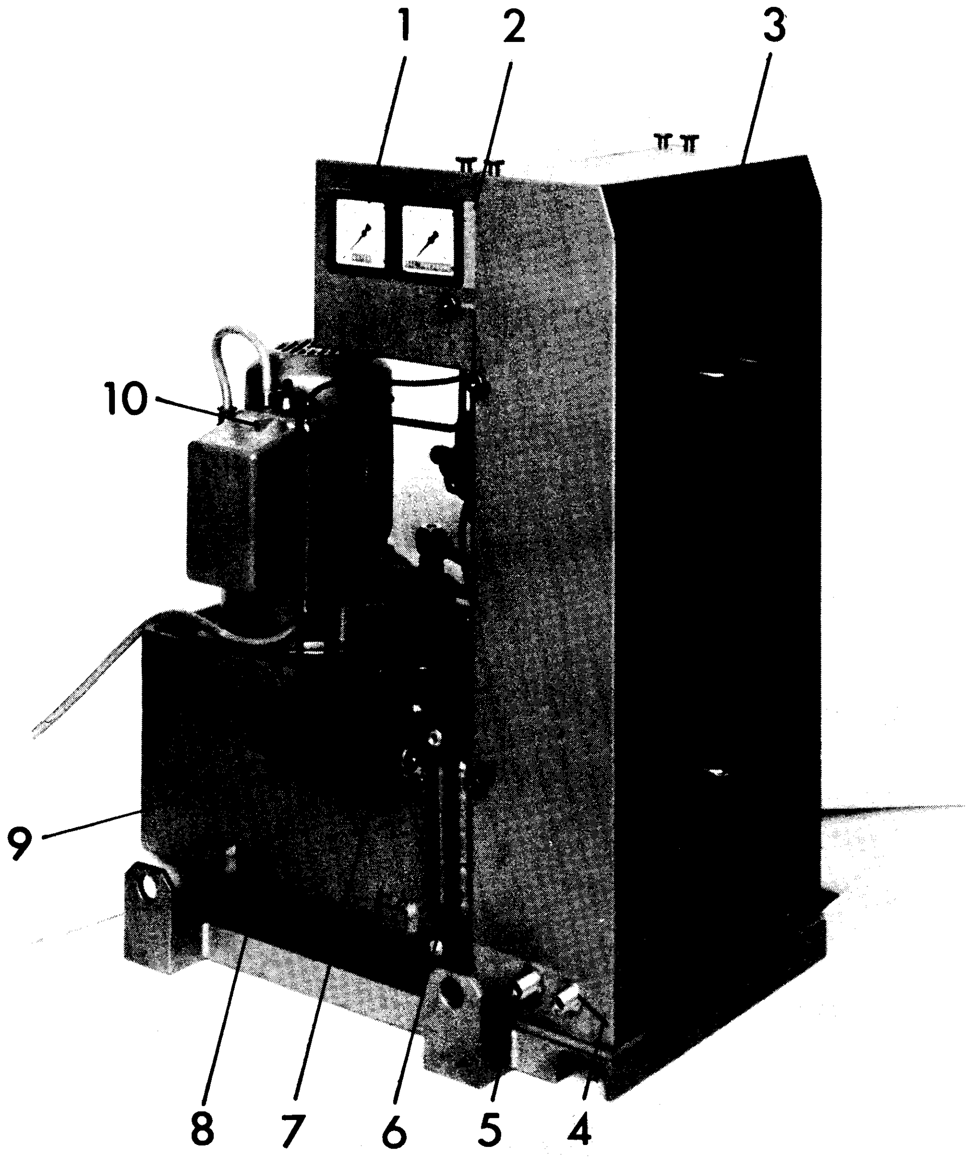


Fig. 28 PUMP UNIT FOR THE HYDROSTATIC BEARINGS (general view)

(1) manometer for pressure of cooling water, (2) manometer for pressure of supply pressure, (3) compartment for the water cooling system, (4) cooling water inlet, (5) cooling water return, (6) visual oil level, (7) supply pressure outlet, (8) return oil, (9) connection for nitrogen indication and filling system, (10) manual start button.