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The Design of a Hydraulic Fast Stub Driver for ITER

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CONTENTS

1. INTRODUCTION

1.1 The Ion Cyclotron Heating system for ITER is foreseen to include automatic tuning systems utilising fast capacitive stubs the design of which is the subject of an additional report prepared by ERM/KMS . The system proposed by ERM/KMS would be similar to that presently used on the TEXTOR machine with the additional requirement that it must operate under steady state conditions over a prolonged period of time. The TEXTOR capacitors are driven by a servo electric motor and ball screw combination.

1.2. This report considers the design and performance capability of a servo hydraulic fast stub driver as an alternative to the servo electric system.

1.3. The maximum velocity and duty cycle of existing capacitor designs are limited by water cooled bellows on the RF current path. Pressure rises and fluid movements within the bellows restrict the maximum velocity and provide heavy damping, whilst resistive heating of the bellows prevents steady state operation. Capacitors of this type operate on TEXTOR and are limited to .25m/s and 50m/s2 at a duty cycle of 60 to 1.

1.4. ERM-KMS [Durodie F, 1994] have proposed a capacitor that eliminates the requirement for a current carrying bellows. This design will have no effective velocity limitation, an acceleration limit based on its mechanical strength and negligible internal damping. This report considers a capacitor of this type.

1.5. This report encompasses the design, selection, performance simulation and costing of the hydraulic actuator, servo control valve, transducer and associated hydraulic equipment. The design and manufacture of control electronics are not considered although these will be within the scope of standard electronic control systems.

2. OPERATING PARAMETERS

2.1. The following operating parameters were agreed with ERM/KMS as being representative of the proposed ITER capacitative stub design which does not incorporate current carrying bellows:

Spring rate: The spring rate is due to the capacitor internal bellows and could be + or - from the mid point null position. This was foreseen to be the same as TEXTOR Comet capacitor type CV7W330FSC at 1400N/m.

Vacuum Load: The vacuum load is proportional to the capacitor bellows effective area. The bellows are assumed to be diameter 70mm which is the same as TEXTOR capacitor type Comet CV7W330FSC giving a vacuum load of 350N.

Damping: Negligible, since no water cooling.

Moving Mass of capacitor: Assumed to be twice as large as on TEXTOR and this to be 7Kg.

Total moving mass: Considered to be the capacitor mass plus 8Kg for a attachment structure (equivalent to a diameter 28mm x 1.5metre long bar) giving a total moving mass of 15 Kg.

Maximum Travel: The total required capacitor stroke was assumed to be the same as at present and to be 70mm. This would gives a capacitance range of 20 to 300pf.

Maximum Acceleration: This is limited by the strength of the capacitor and driver assemblies. This is assumed to be the same as the limit on the present TEXTOR capacitors, 50 m/s^2.

Working Stroke: This is the distance over which the capacitor is required to move dynamically within the specified time constant. This is assumed to be + or - 30mm i.e. + or - 120 pf.

Velocity: Virtually unlimited since there is no pressure build up within the cooled capacitor bellows. In practice limited by the maximum acceleration, available stroke and valve flow rating. For valve sizing a 50m/s^2 acceleration over 30mm was assumed giving the maximum velocity 170 cm/s. For dynamic performance simulation, 50m/s^2 over 15mm stroke acceleration followed by 50m/s^2 over 15mm stroke deceleration was assumed, giving a peak velocity over this period of 122 cm/s and an average velocity of 61.2 cm/s.

Resolution: Positional accuracy required within the specified response time 0.5mm.

Response time: Defined as the time taken to achieve the requested position within the specified resolution following a step input. This was specified as 50msec within the specification for task D14.

Maximum steady state duty: This is used to define the size of the hydraulic plant. Taken to be 30mm travel at maximum acceleration with a 0.5Hz square wave cycle time. This was considered as being suitable for RF load variations during "sweeping" of the divertor plates for the full 1000 second pulse duration.

Space envelope for actuator assembly: In order to minimise backlash and moving mass, the actuator and its transducers should be situated as close as possible to the capacitor. The capacitor is situated at the base of a length of stub line of length 1.5 metres and inside diameter .235m . The actuator and valve assembly is envisaged to fit close to the capacitor within this space envelope (Figure 1).

Hydraulic Fluid bulk modulas: 13793 bar. This is a typical figure for an oil based hydraulic fluid.

Volume between servovalve and cylinder: 0.2 litres. Equivalent to a diameter 20mm pipe in two lengths each 300mm long.

Neutron Loading: Assumed to be negligible

3. SYSTEM OVERVIEW

3.1.1 All linear servo hydraulic position control systems operate by controlling flow to one or both sides of a hydraulic cylinder (Figure 2). The force that can be applied is a function of system pressure and cylinder area. The maximum velocity of the cylinder is limited by the maximum system pressure, the cylinder volume and the throttling effect of the valve, pipe work and cylinder ports. It is normally considered that this throttling effect within the valve obeys laws relating to flow through square edged orifices. The dynamic response of the system is limited by the valve, moving mass and the elasticity of both the structure and the fluid between the valve and the actuator. System stiffness can be improved by increasing the cylinder area and valve response can generally be improved by increasing system pressure.

3.1.2. A schematic diagram of a hydraulic position control system as would be applied to each of the ITER tuning stubs is shown in figure 3. The system comprises of a hydraulic cylinder to drive the capacitor, a transducer to give the position, a servovalve to control the cylinder position and associated plant. The diagram shows a power pack incorporating, a pump, low pressure oil filters and a recirculating conditioning filter. A start up by pass valve is also shown to reduce motor starting torque. To enable a lower pump rating and improve response at peak flows an accumulator is shown close to each servo valve. In line high pressure filters are incorporated immediately before each servo valve to maintain the very low level of fluid contamination essential for reliable operation. An adjustable cross port pressure relief valve is shown immediately after each servovalve, this is used to provide a fixed limit on the maximum force and hence acceleration that can be applied. The servo actuator items shown within the chain dotted line would be a single assembly in order to minimise fluid volumes. Selection of these items will be discussed later within this report.

3.2 Servovalves.

3.2.1. It can be seen that the valve in figure 2 is in the form of a spool. If the spool moves to the left fluid is free to flow from the pressure port P1 to the cylinder port C1 and from the cylinder port C2 to the return line R2 causing the piston to move to the right. If the spool moves to the right the opposite flows occur and the piston moves to the left. With the spool In its central position both the pressure and return ports are covered and the piston does not move.

3.2.2. In most servo valves an electrically driven primary stage is used via a "fluid amplifier" to actuate the main spool secondary stage (figure 4). The fluid amplifier operates by an electric current in the torque motor armature causing the flapper to move to the left or the right with a force proportional to current. This causes one of the two opposing nozzles to close causing pressure to increase on that end of the spool. The spool then moves to open port P to one cylinder port and the return port R to the other cylinder port. In this example a mechanical feed back spring provides a restoring force to the torque motor which is proportional to spool movement, other valves may use a separate electrical feedback transducer connected directly to the spool.

3.2.3. Servo valves have near linear no load flow gain characteristics as shown in figure 5. The frequency response of a Moog Controls Ltd series 760 high response valve are given for example in figure 6. The higher the frequency response generally the lower the valve rated flow (figure 7). A standard type valve would have a flow rate of up to 80 litres per minute at 70 bar and a typical response of 90 Hz for a 90 degree phase lag at 210 bar (typical cost 1300 ECU per valve). Higher response valves would be rated at typically up to 35 litres per minute at 70 bar and would have a 210 bar 90 degree phase lag frequency response of 140 to 280 Hz. High response high flow rate valves (1000 ltr min (@70bar with 50Hz 90 degree phase lag) are available at relatively high cost (typically ECU 12000 per valve).

3.3 Cylinders

3.3.1. The servo cylinder must have sufficient area to provide the required maximum actuator force at the system pressure. Also a larger diameter cylinder has a better dynamic response due to its higher fluid stiffness. However a very large cylinder will require more expensive valves and pump to accommodate the high flow rates. The efficiency of a system with a large diameter cylinder is reduced because a smaller proportion of the system pressure is used to drive the load.

3.3.2. The cylinder can either be single or double ended. Double ended cylinders have the advantage that loading on the cylinder is symmetrical regardless of travel direction. Single ended cylinders are cheaper and more compact.

3.3.3 Both rod wiper, bearing and piston seals are required. These should be of low friction to minimise the effect on the dynamic performance due to the "stick slip" characteristics of the seal. To minimise this effect seals are typically made from spring energised PTFE, for example those manufactured by Busak+Shamban Ltd. Typical seal friction utilising double rod seals on a diameter 40 mm cylinder have been quoted as 670N. Typical seal life at a velocity of 0.2m/s has been quoted as 5000 linear Km, seal life may reduce at higher velocities and for small amplitude movements. Very low friction and unlimited wear may be achieved using hydrostatic rod bearings however seal leakage is increased. A small amount of hydraulic fluid leakage is usually unimportant in most hydraulic system environments. However following experience at JET and in the conditions expected in the environment of the ITER RF transmission lines elimination of external fluid leakage is essential. This can be achieved by:

- Using double rod seals with an interspace drain between the seals. This would have higher friction, more wear on outer rod seal which has poor lubrication.

- Enclosing the actuator assembly within a sealed enclosure fitted with a pumped drain using a rubber bellows to cover and seal the piston rod. This has low friction and would tend to be a more bulky assembly. Seal failure would not be important as excess fluid is drained. Degradation of the seal normally takes the form of progressively increasing leakage.

3.3.4 Cylinders often incorporate hydraulic dampers or "cushions" in order prevent damage and to limit forces in the event of a cylinder over run. To eliminate backlash, position transducers are often incorporated and sealed within one end of a hollow piston rod on a double ended cylinder. To minimise fluid volumes servovalve and crossport relief valves may be incorporated directly on a manifold that is integral to the cylinder. End mountings can either be screw, ball joint or clevis fittings as standard. Facility to bleed air from the servo cylinder volume can be provided.

3.3.5. Cylinders incorporating the above features are available as standard items. Cylinders that have geometrical or other constraints may require special design.

3.4 Position Transducer.

3.4.1. To maintain optimum performance the transducer must have a frequency response several times larger than the system. Transducers are readily available with frequency responses in excess of 500Hz. The transducer must have linearity and resolution compatible with the 0.5mm accuracy required. The transducer must fit within the space envelope of the hydraulic cylinder rod. The transducer must have a long service life. LVDT type position transducers satisfy all the above requirements and are frequently used in servo hydraulic applications.

3.5 Accumulators

3.5.1. Accumulators provide a compressible volume which is used to store hydraulic pressure energy. An accumulator should be fitted before each servovalve to accommodate rapidly changing fluid flows and allow peak flows much higher than the pump capacity. All accumulators consist of a volume which contains both a precharge of compressed gas and hydraulic fluid. The fluid and gas are separated by a piston, membrane or rubber bladder. The accumulator will be full at the maximum operating pressure and will discharge with an adiabatic expansion as the pressure is reduced. A typical large accumulator size is 50 litres. Precharges can be as high 90% of the system pressure.

3.6 Pumps

3.6.1 Pumps provide the hydraulic system pressure. Pumps can be of fixed or of variable displacement. Pumps for high pressure systems are either gear, vane or piston pumps. Higher pressure and flow pumps are usually piston pumps. Pistons can be either axial or radial. The displacement of an axial piston pump is controlled by an angled swash plate. Variation of the angle controls the piston displacement. This can be controlled so as to provide a constant system pressure over a wide flow range. Pressures up to 400 bar and displacements 2000 cc per rev are possible with axial piston pumps. Variable displacement pumps are the most efficient power sources. A typical running speed is 1500 rpm.

3.7 Filters

3.7.1. Servo valves incorporate very close tolerances and small openings within the pilot stage. Experience at JET has shown that a very high degree of fluid cleanliness is essential for reliable operation . Prior to operation all components and fluids must be thoroughly cleaned. The system must be flushed and all welds must be pickled to remove loose debris. To maintain this cleanliness a high degree of properly maintained filtration is required. A 10 to 15 micron high pressure filter is recommended before the servo valve. A 3 to 5 micron low pressure filter is recommended on the return line. Additionally new fluid should be filled by means of a transfer pump including a 3 micron filter. A recirculating pump can continuously "polish" the oil through a 3 to 5 micron filter. Tanks must be properly designed, cleaned, covered and fitted with a filtered breather cap. Dirt alarm pressure switches should monitor the pressure drop across the filter and indicate when filters need to be changed. It is recommended that when a filter element is changed the tank volume is flushed through the filter at least 100 times. Flushing fluid through the servovalve can be avoided if flushing blocks are fitted in place of the valve. Regular monitoring of fluid quality to a standard agreed by the valve manufacturer is recommended. Fluid the fluid should at least be clean to ISO DIS 4406 Code 14/11 .

3.8 Pipework

3.8.1. All pipe work must be clean and free from debris. Where possible seamless precision steel metal rigid pipework to DIN2391/C shall be used. Flexible lines should be thermoplastic, Teflon or nylon lined tube rather than rubber. Use O ring fittings rather than tapered threads. Contamination from fragments of PTFE sealing tape can cause problems. If tapered threads must be used, a liquid type sealant rather than a tape should be employed. Use pipe cutter not hacksaws for cutting pipe. Cold bend pipe if possible. Descale pipes after hot bending or welding.

3.9 Fluid

Preferred fluid is hydraulic oil to DIN 51524 ,Viscosity 60 to 450 SUS at 30° C.

4. SYSTEM DESIGN

4.1 Force Requirements

Ft = Total maximum force on actuator = Fl + Fa + Fe + Fs.

Where:

Fl = force due to load = Static vacuum load + maximum spring loading from bellows (at a 35mm deflection).

Fl = (35 x 9.81)+(1400 x 5e-3)=392N.

Fa = Force due to acceleration of load = Maximum acceleration x Mass.

Fa = 50 x 15=750N.

Fe = Any external force acting on actuator.

Fe = Nominally zero, however to allow a margin say =100N.

Fs = Friction loading. This is dependant on the type and number of fluid and wiper seals used on the cylinder. Cylinder manufacturers Parker Hannifin Ltd have quoted 670N for a diameter 40mm cylinder with double rod seals (see following section).

Ft = 392+750+100+670=1.912 KN. say 2KN.

4.2 Cylinder Size, Servovalve and System Pressure

4.2.1 The cylinder must be sufficiently large to provide the maximum required force of 2KN at the operating pressure (Moog Controls Ltd recommend that an additional margin of 30% is allowed for this force) . It must also have sufficient mechanical stiffness such that its elasticity has a small effect on the overall system.

4.2.2. For optimum efficiency and energy transfer to the load, it is recommended that the system pressure should be such that the load pressure drop across the valve is 33% of the main system pressure. This also ensures that valves utilise the full spool travel and as result increases the system accuracy. Moog Controls recommend 100 bar as a minimum operating pressure. However servo valve performance improves at higher pilot pressures (ref. figure 8). Higher system pressures increase the forces available from the pilot stage for spool movement and reduce the amount of spool movement required for any given flow. This produces higher pressure gains and as a result faster system response times at the expense of some decrease in efficiency, resolution and system accuracy.

4.2.3 210 bar is the highest system pressure with a wide range of available components, and has been chosen as the preferred pressure for this system. Computer simulations show that a lower pressure 100bar system sized using the 33% ratio of load pressure drop to system pressure will not achieve the required 50msec response time.

4.2.4. An alternative that would improve system efficiency without the reduction in system performance associated with low pressure operation would be to use a servo valve incorporating a separate 350 bar supply "5th" port for the pilot valve and a 100 bar supply to drive the load.

4.2.5 At 210bar a cylinder of diameter 13mm with a 10mm rod diameter would be required to provide the operating force. A cylinder of this size would not have sufficient mechanical or hydraulic rigidity for the required dynamic and positional performance. A standard sized 40mm diameter cylinder and 28mm diameter rod has therefore been chosen as the smallest cylinder with sufficient hydraulic stiffness.

4.2.6. Note: Hydraulic energy is proportional to the pressure drop multiplied by the flow rate. The energy transferred to the load equals the pressure drop across the load times the flow rate . The energy lost at the valve equals the system pressure minus the pressure drop across the load multiplied by the flow rate. The efficiency is the ratio of these two pressure drops. The worst efficiency will occur for low velocity movements where the force required to move the load is small and hence almost the full system pressure is lost across the valve. However at low velocities the flow rates are small and so the quantity of energy lost at the valve is correspondingly small.

4.3 Servovalve Dynamic Response

Load Dynamics

Diameter 40mm actuator, 28mm rod, double acting 4-way connection, supply pressure set to 210 bar.

Problem: calculate the resonant frequency of the mechanics and hydrualics utilising moog hydrualics method.

Hydraulic Load Resonant Frequency (assuming an equal area cylinder)

K0	=	hydraulic stiffness (N/m)
β	=	fluid bulk modulus (N/m^2) = 1.3793 10^9 N/m^2
Mass	=	moving mass (Kg) = 15 kg
Area	=	working area of double ended piston (m^2)
Xu	=	piston stroke used (m) = 0.03m
Xt	=	total piston stroke (m) = 0.07m
Volume	=	total volume of fluid between servovalve ports and the piston (m^3)
η	=	volumetric coefficient
Ft	=	applied max force to move load = 2 x 10^3 N
bore	=	bore of cylinder (m) = 0.04m
rod	=	rod diameter of cylinder = 0.028
Fn	=	natural frequency (Hz)
QL	=	maximum loaded flow through cylinder per valve (m^3/s)
amax	=	maximum acceleration (m/s^2) = 50 m/s^2
VL	=	maximum loaded velocity
PL	=	load pressure drop (m/s) across valve (N/m^2)
QnL	=	no load flow m^3/s
Qr	=	rated flow at 70 bar with a 10% safety margin (m^3/s)

$$\text{area} = \text{bore}^2 \frac{\pi}{4} - \text{rod}^2 \frac{\pi}{4} = 6.409 \times 10^{-4} \times m^2$$

Making an allowance of 0.0002 m^3 for the pipe volume

$$\text{Volume} = (\text{Xt} \times \text{area}) + 0.0002\ m^3 = 2.449 \times 10^{-4}\ m^3$$

$$\eta = \text{area} \times \frac{\text{Xu}}{\text{Volume}} = 0.079$$

$$K0 \quad = 4.\beta 0 \text{ area } \frac{\eta}{Xt} \quad = 3.966 \times 10^6 \text{ N/m}$$

$$Fn = \frac{1}{2.\pi}\sqrt{\frac{K0}{Mass}} \quad = \quad \underline{81.84 \text{ Hz}}$$

Maximum loaded velocity

$$VL = \sqrt{2.a\max.Xu} \quad = \quad 1.732 \text{ m/s}$$

QL = area x Vl = 1.11 x 10^{-3} m^3/s

$$PL = \frac{Ft}{area}.1.1 \quad = 3.433 \times 10^6 \text{ N/m}^2 \text{ (with a 10\% margin)}$$

Suggested supply pressure = Ps = 3 x load pressure drop across valve.

Ps = 3 x PL Ps = 1.03 x 10^7 N/m^2 (103 bar)

However set supply pressure (Ps) at 210 bar (210 x 10^5 N/m^2)

For square edged orifices

$$QnL \quad = QL\sqrt{\frac{Ps}{Ps - Pl}} = 1.214 \times 10^{-3} \text{ m}^3/\text{s}$$

$$Qr \quad = 1.1 \times QnL \times \sqrt{\frac{70x10^5}{Ps}} = 7.937x10^{-4}\,m^3 / s$$

Qr x 60.1000 = <u>47.622</u> x litres/min.

4.3.1 From the above calculation it can be seen that for maximum acceleration over a 70mm stroke the servovalve needs to have a rated flow at a 70bar valve pressure drop of more than 46 litres/min. For optimum performance it is recommended that the 90 degree phase point exceeds the load resonant frequency of 81.8Hz previously calculated by a factor of three or more, i.e. a 90 degree phase lag of 243 Hz. No single valve is available with this combination of frequency response (which is significantly above the 50msec required response time) and flow rate. However this flow rate at a 90 degree phase lag of 80 Hz is possible with a Moog 760-406 valve (reference appendix 1). Higher flow rates can be achieved by using two or more lower flow rate valves in parallel in order to achieve the required flow rate without sacrificing frequency response. With multiple valves it is possible to have a potential recirculating leak path through the valves which would increase null flow rates. The performance of:

- Single M760-406 standard response valve,
- Single and twin Moog 760-233 high response valve (90 degree phase lag 140 Hz),
- Three Moog 760-912 super high response valves (90 degree phase lag 280 Hz),

have been be modelled using the Hydrosoft package. Full details of these valves are given in Appendix 1.

5. PERFORMANCE SIMULATION

5.1.1 The Hydrosoft hydraulic modelling and simulation package (Reference Flowtron Ltd) was used to model the performance of the systems selected; data files and transient response plots are given in Appendix 2. Valve data was extracted from the Moog controls information given in appendix 1. The resonant frequency response (400 Hz -3db and 90 degree phase lag) of an LVDT type displacement transducer was also included in later runs of the higher performance valves. For all cases Hydrosoft was used to optimise system gain with phase and gain margins of 45 degrees or 7db.

5.1.2. It can be seen from the step response characteristics (Appendix II figure II 1.1) that a single standard M760-406 valve does not provide ± 0.5mm accuracy within the required 50msec response time (time constant was 67msec).The single and dual Moog 760-233 high response valves and the triple 760-912 super high response valves reacted to a 3cm step input with time constants of 34msec and 25msec respectively. Natural frequencies were of 50 and 63 Hz respectively (45 Hz with LVDT transducer).

5.1.3 The acceleration limit of 50 m/s2 will limit the response from a 30 mm step input to an average velocity of 61.2 cm/s. This is shown for the three high response systems in appendix 2. A 5g acceleration and deceleration over a 30mm displacement for the twin 760-233 valves has been approximated with a 10hz sinusoidal input. The time constants of the three systems with a 61.2 cm/s ramped input were within 50msec .

5.1.4. The steady state velocity error of the system is directly proportional to velocity and system gain. System gain can be increased by increasing the hydraulic stiffness, by minimising the volume of compressible fluid between the servovalve and the actuator and by maximising the cylinder cross sectional area within the flow constraints of the servo valve. With the fluid pipe volume of 0.2 litres assumed velocity errors for the 760-233 valve with basic closed loop position feed back control are as follows:

Velocity m/s	Error mm
0.10	0.5
0.25	0.7
0.612	1.4
0.86	2.2

5.1.5. Maximum actuator velocities are limited by the saturation flow of the servovalve. These and a summary of other performance data are given in the table below (at 210 bar and for a 40mm diameter actuator):

System	1	2	3	4	5
Valve	1 off 760-406	1 off 760 - 233	2 off 760 - 233	3 off 760 - 912	3 off 760 - 912 Maximum fluid system stiffness
System natural frequency Hz	26	46 *	50 44 *	63	92 *
70 bar flow rating l/min	65	38.5	77	57.6	57.6
210 bar flow rating l/min	100	60	120	90	90
Maximum velocity, diameter 40mm actuator	260	154	308	230	(diameter 70mm cylinder) 62
Optimised loop gain (7db or 45 degree phase lag) sec^{-1}	145	179*	205 180*	232	442*
Velocity error (62cm/s) mm	--	1.8mm*	1.4	1.4	0.8mm*
Velocity error (170cm/s)mm	11	(150cm/s limit 8.35*	8.26 8.68*	7.3	--
Response time in msec to a 3cm step ±0.5mm	67	30*	34 34 approx	25	20*

* Includes LVDT position transducer response.

Note: System 5 utilises three super high performance 760-912 valves, a diameter 70mm cylinder and a pipe fluid volume of only 0.05 litres. The dynamic performance will be close to the achievable maximum, limited by the valve response, transducer response and the fluid compressibility.

5.1.6. Following these simulations it is proposed that a single 760-233 valve combined with a 40mm diameter cylinder (28mm diameter rod) provides the simplest and most cost effective solution to the performance specification.

6. ASSOCIATED PLANT

6.1 Pump Selection

Cylinder bore diameter 40 mm rod diameter 28mm.
Area = 6.41e-4 m2
Stroke at steady state 30mm at 0.5 Hz.
Maximum average velocity over 30mm stroke = 0.612 m/s.

6.1.1 Steady state flows

Volume per 30mm stroke = 19.23 e-6 m3 = 19.23 cc
Steady state .5Hz consumption = 19.23 x 2 = 38.46 cc/s = 2307cc/min.
Total for 72 stubs = 166147 cc/min = 166 litres/min steady state.

Valve leakage flow is less than 2.3 litres/min at 210 bar total for 72 stubs=165.6 litres/min

<u>Maximum steady state flow rate for 72 stubs =166+165=331 litres/min</u>

6.1.2 Peak flows

Peak flow at 0.612m/s stroke over say a 10 second period
Swept volume over 10s =6.41e - 4 x.612 x 10 = 3.92e - 3m^3 per stub.
Peak volume flow rate per stub=3.92e - 4m 3m^3/s = 392cc/s = .392 litres/s

Peak flow for 72 stubs = 1693 litres/min.

6.1.3. This peak flow over a 10s period can be accommodated by a 50 litre accumulator with a 100 litre gas back up bottle mounted close to each servovalve so as to minimise pipe losses.

6.1.4. Due to the wide range of required flow rates at constant pressure a variable displacement axial piston pump with constant pressure feed back will provide maximum efficiency.

6.1.5. Two pumps will enable continued operation at a reduced duty cycle in the event of a pump failure. The pumps must be rated each at a minimum of 166 litres per minute at 210 bar. Two Rexroth Hydraulics series A10VS size 140 variable displacement axial piston pumps with a constant pressure controller will provide the required steady state flow rate at 210 bar. The power requirement at 1500 rpm 210 bar (reference figure. 9 attached) is 80 kW per pump. A large proportion of this power will be used to heat the hydraulic fluid, this is particularly the case when the system is not operating at full duty. Providing pipes are unlagged then a large amount of the heat will be lost from the pipework. However an additional oil cooler may be necessary.

6.2 Cylinder

6.2.1. The cylinder, servo valve, and cross port pressure relief valve shall comprise a single modular assembly incorporating the following features:

- Fit within a space envelope of length 1.5 metres and diameter .235m
- Double ended cylinder.
- Low friction PTFE seals.
- Transducer incorporated within cylinder rod.
- Integral servovalve manifold.
- Cylinder cushioning.
- Cylinder air bleed.
- 210 bar rating.
- Fitted within an additional enclosure incorporating a pumped drain and a piston rod sealing bellows.
- Piston diameter 40mm, 28mm rod diameter rod.

Note: Hydrostatic seals and bearings may be considered if the lifetime and maintenance requirements of low friction PTFE seals are considered to be unacceptable.

6.3 Pipe work

6.3.1. Pipe to and from the pump shall be a ring main of 50mm bore seamless precision steel metal pipework to DIN2391/C. If pessimistic assumptions are made over flow rates this will produce pressure drops of less than 1 bar over a 50 meter length. Joints shall be welded where possible. All welds shall be pickled and the system flushed prior to operation.

6.3.2. Each servovalve assembly shall be linked to the ring main by means of 12.5 mm steel bore pipes. Each link shall be fitted with one way valves to prevent flow reversal. Isolation valves shall be fitted to enable maintenance. Links between the 12.5mm pipe and the servo valve shall be by flexible thermoplastic hoses. Connections to the servo actuator assembly shall be by means of quick release self sealing couplings. Accumulators shall be connected as closely as possible to the servovalve.

6.4 Mechanics and Support Structure

6.4.1. Free play on connection from the actuator to the load should be 3 to 10 times less than required position accuracy ie 0.166mm to .05mm. Stiffness should be 3 to 10 times higher than hydraulic stiffness (hydraulic stiffness from Hydrosoft programme $= 2 \times 10^3 \div 0.04367 = 45.8$KN/mm) ie stiffer than 137 KN/mm. The servo actuator assembly should be mounted within the inner conductor of the stub within a fluid tight enclosure with a separate pumped drain. Connection of the servo actuator should enable easy removal for maintenance.

6.4.2. The capacitor bellows should be sized such that its resonant frequency is higher than the operating frequency of the system.

6.5 Control

6.5.1. The systems modelled within this report have used a proportional type controller. Further improvements in performance may be possible by utilising more advanced control techniques such as:

- Multiple feed back incorporating velocity control loops.
- Phase advance passive networks.
- Adaptive control
- PID type control.
- Dither. A small amplitude oscillation can reduce hysteresis errors.

Standard plug in hydraulic control units are available for servovalve systems.

If a digital control system is utilised at least 12 bit DAC's should be used to avoid degrading servovalve resolution. A low pass analogue filter will smooth signal and extend valve life.

7. COSTING

Pump set 2 off .

Pump:165 Ltr/min, pressure 210 bar	5K ECU
Motor:80KW 1500 rpm.	3K ECU
Filter on tank 3 micron 165 ltr/min .	4K ECU
Return Filter. 3 micron 165 ltr/min. .	4K ECU
Recirculating pump .	8K ECU.
Recirculating filter 3 micron. .	4K ECU
Pressure relief valve 210 bar 165ltr/min .	2K ECU
Pressure switch 210 bar .	3K ECU
Motor control .	6K ECU
Tank capacity approx 1000 ltr	3K ECU
Tank Breather .	1K ECU
Tank lid + seal .	1K ECU
Level indicator .	1K ECU
Level Switch .	1K ECU
Pressure Transducer.	3K ECU
Pressure gauge .	3K ECU
Cylinder case drain line pump .	5K ECU
Tank breather. .	2K ECU
Fluid 1000 ltr .	4K ECU
Pump Control system	6K ECU

Total for each pump 22.5 KECU Total for two pumps = 45 K ECU

Pipe work and line fittings

144 Quick release self sealing 1/2" couplings		= 13	K ECU
Tube steel: 100 meters 50mm bore, 300 meters 12.5mm bore. Say 200 ECU/meter		= 80	K ECU
Bleed valves 72.		= 9	K ECU
144 One wave valve		= 20	K ECU
In line high pressure filters 15 micron x 72 = 0.32KECU ea	Total	= 23.04	K ECU
Isolation Valves x 144. 0.1 KECU each.	Total	= 14.4	K ECU
Accumulator. 50 ltr with gas back up * 72.62KECU ea,	Total	= 446.4	K ECU
	Total for fittings	= 606	K ECU

Actuator assemblies each of 72 tuning elements (2 per each of 36 lines)

Servovalve type Moog 760 type or equivalent		= 2.5	K ECU
Transducer type .		= 4	K ECU
Cylinder case drain line and enclosure. .		= 6	K ECU
Mechanical mounting system		= 1	K ECU
Flushing Valves.		= 0.15	K ECU
Cross Port Pressure relief valve		= 0.75	K ECU
Cylinder 210 bar 28mm bore. 70 mm stroke			
Manifold			
Servo actuator assembly	Total	= 4.5	K ECU

Cost per actuator assembly = 9.9 KECU. Total for 72 assemblies = 712 K ECU

Total overall Cost = (Pump set + pipework and fittings + servo actuators
= 712 + 606 + 45 = 1363 K ECU

Say total = 1.4 MioECU (excluding: vacuum capacitor, stub, control electronics, and commissioning costs).

8. CONCLUSIONS

8.1. A 50 msec, 3cm ± 0.5mm step response time is possible utilising commercial hydraulic servovalve technology.

8.2. A system operating under closed loop position feed back control at 210 bar and utilising a single Moog controls Ltd high response servovalve type 760-233 or equivalent to a cylinder of 40mm bore and 28mm rod diameter will provide the required response. Performance can be expected to be as follows:

- Natural frequency 46 Hz.
- 3cm ±0.5mm step response time (no acceleration limit) 30msec.
- Optimised loop gain 179 sec^{-1}.
- Velocity error at 62 cm/s =1.8mm.
- Velocity limit for saturated valve flow at 210 bar =150 cm/s.

8.3. The best achievable response will be close to a system utilising 3 off 760-912 super high performance valves operating in parallel to drive a 70mm diameter cylinder with a pipe volume between the valve and cylinder of .05 litres. Performance can be expected to be as follows:

- Natural frequency 92 Hz.
- 3cm ± 0.5mm step response time (no acceleration limit) 20msec.
- Optimised loop gain 442 sec^{-1}.
- Velocity error at 62 cm/s =0.8mm.
- Velocity limit for saturated valve flow at 210 bar =62 cm/s.

8.4 The efficiency of this system will be low due to the very small pressure drop across the load and high flow rates. The flow rate is more than three times the flow rate in the system recommended in paragraph 8.2 above. Accumulators, pumps and pipework would have to be sized to accommodate this flow, many will be outside the range of standard equipment. It would be difficult to fit a system such as this within the space envelope of the stub. If the system had to be mounted some distance from the capacitor many of the performance advantages would be lost.

8.5. The efficiency could be improved by reducing the system pressure from 210 bar to 103 bar and utilising a servovalve with a separate high pressure (210 or 350bar) connection for the pilot stage. System response will be better than a completely 103 bar system but worse than a 210 bar system due to the reduced valve pressure differential. Also a separate low flow high pressure supply line will be required to drive each pilot stage.

8.6. Some improvement in position and velocity error could be achieved by utilising more sophisticated control techniques such as incorporating an additional velocity feedback loop, "dither", utilising "phase advance" and adaptive control techniques.

8.7. Servo hydraulic system are capable of higher powers, faster linear velocities and accelerations at higher frequencies than servo electric systems. Due to the large linear forces available the advantages of servo hydraulics increase with larger accelerations and inertia's. This is particularly the case at peak powers in excess of 15KW per servovalve.

8.8. The performance of the modelled systems would be close to that shown for the step response if the maximum 5g acceleration limit was removed.

9. REFERENCES

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Busak+Shamban, 5 Saint James Terrace, Ewart House, Nottingham, NG1 6FW, England. Sealing systems.

G.L.Rexroth Ltd, Cromwell Road, St Neots, Cambs, PE19 2ES.England. Pumps, servovalves and general hydraulic equipment.

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Walters. R.B. Hydraulic and Electro-hydraulic control systems. Elsevier Applied Science series. ISBN 1-85166-556-0

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Durodie F, 1994 ERM/KMS, Laboratoire de Physique des plasmas, - Association EURATOM-Etat Belgé, Ecole Royal Militaire, 30 Av. de la Renaissance, B1040 Bruxelles, Belgique.

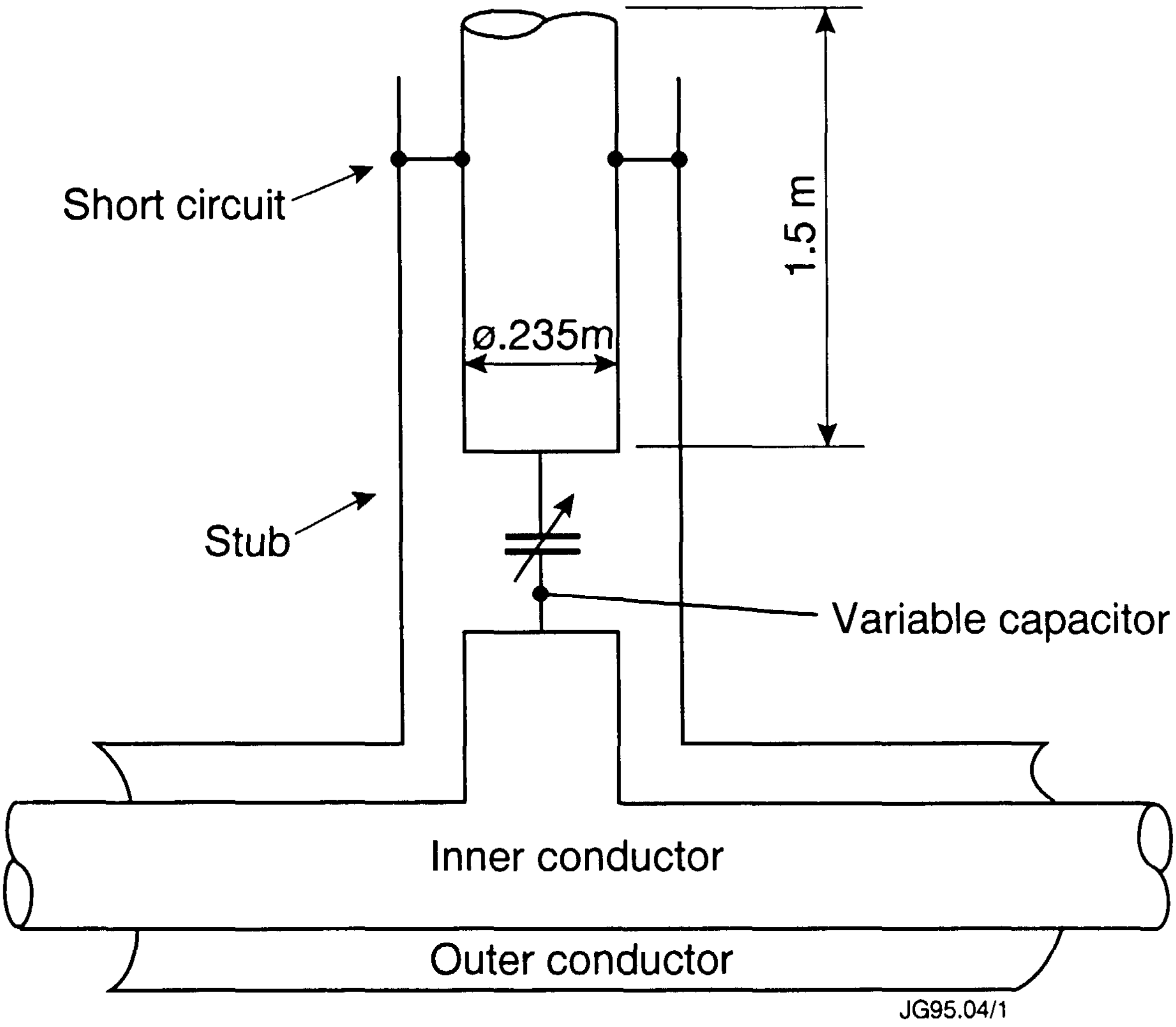


Figure 1 Space envelope for capacitor actuator

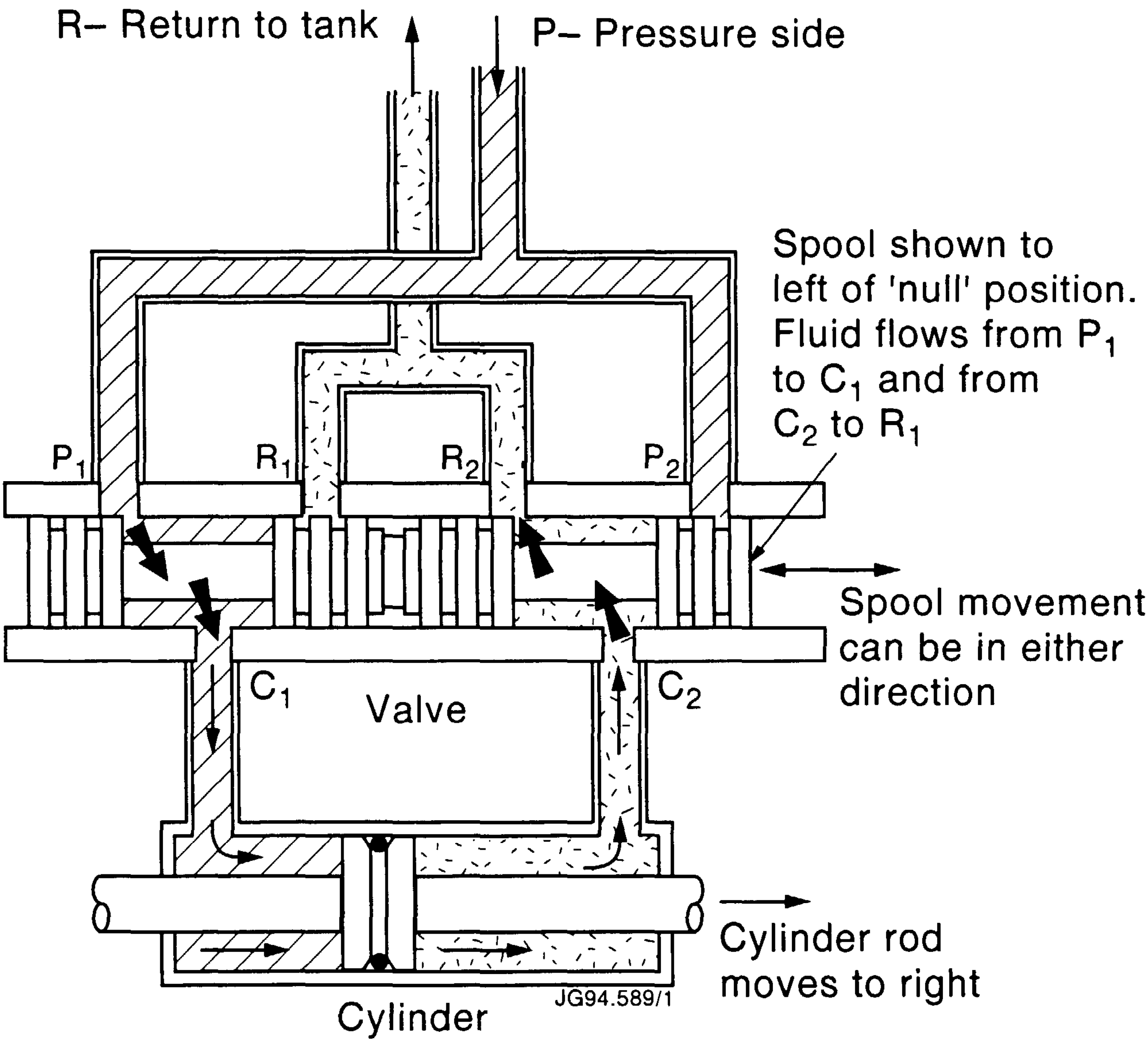


Figure 2 Cylinder position control with an open loop single stage valve

Component list

1. Motor
2. Pump
3. Low pressure filter
4. Recirculating filter pump
5. Oil cooler
6. Pressure gauge
7. Pressure switch
8. Start up by pass valve
9. Accumulator
10. High pressure filter
11. Servovalve
12. Cross port pressure relief valve
13. Cylinder
14. Position Transducer
15. Capacitor

One way valve

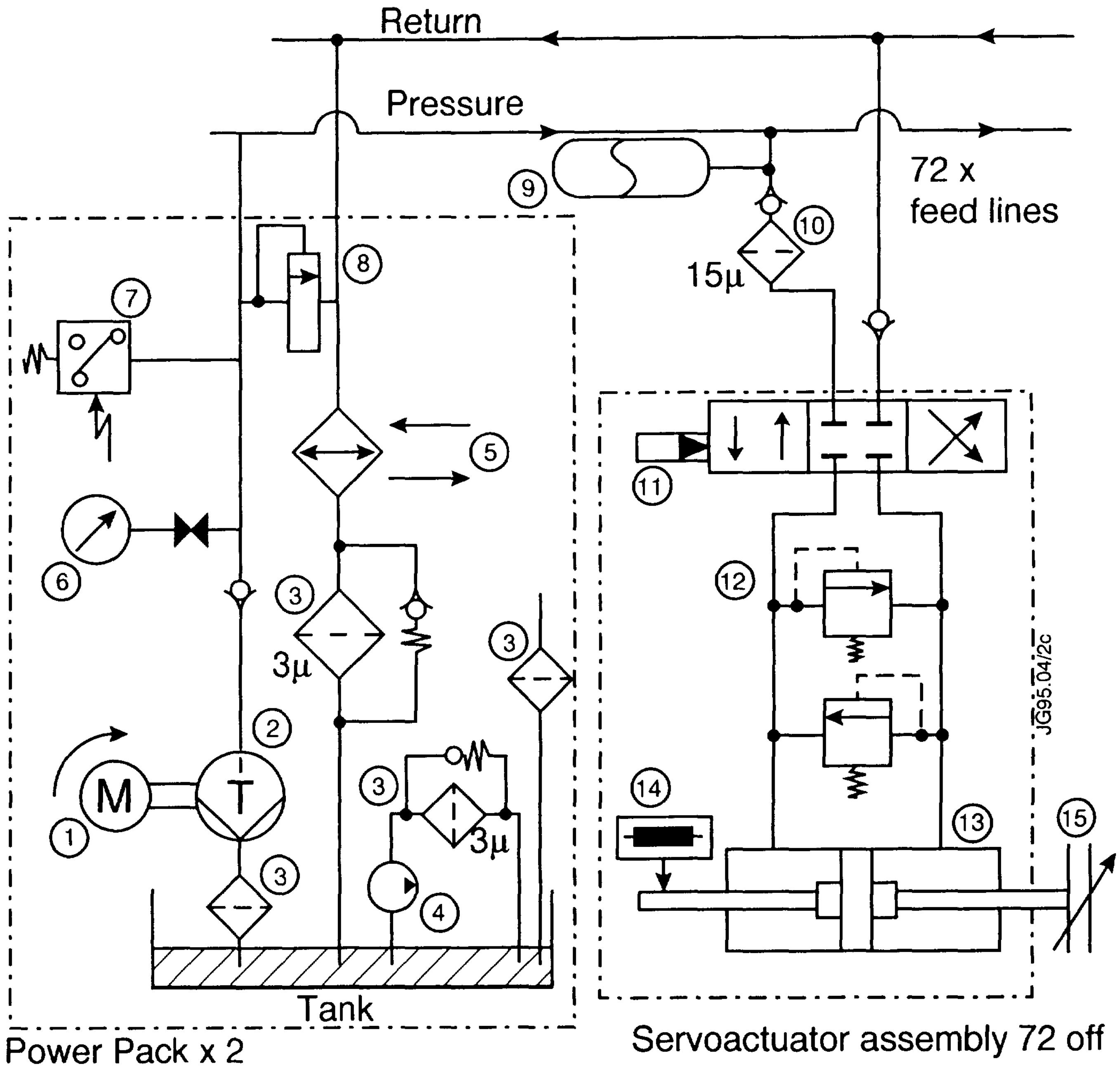


Fig. 3 Schematic hydraulic circuit for ITER fast stub

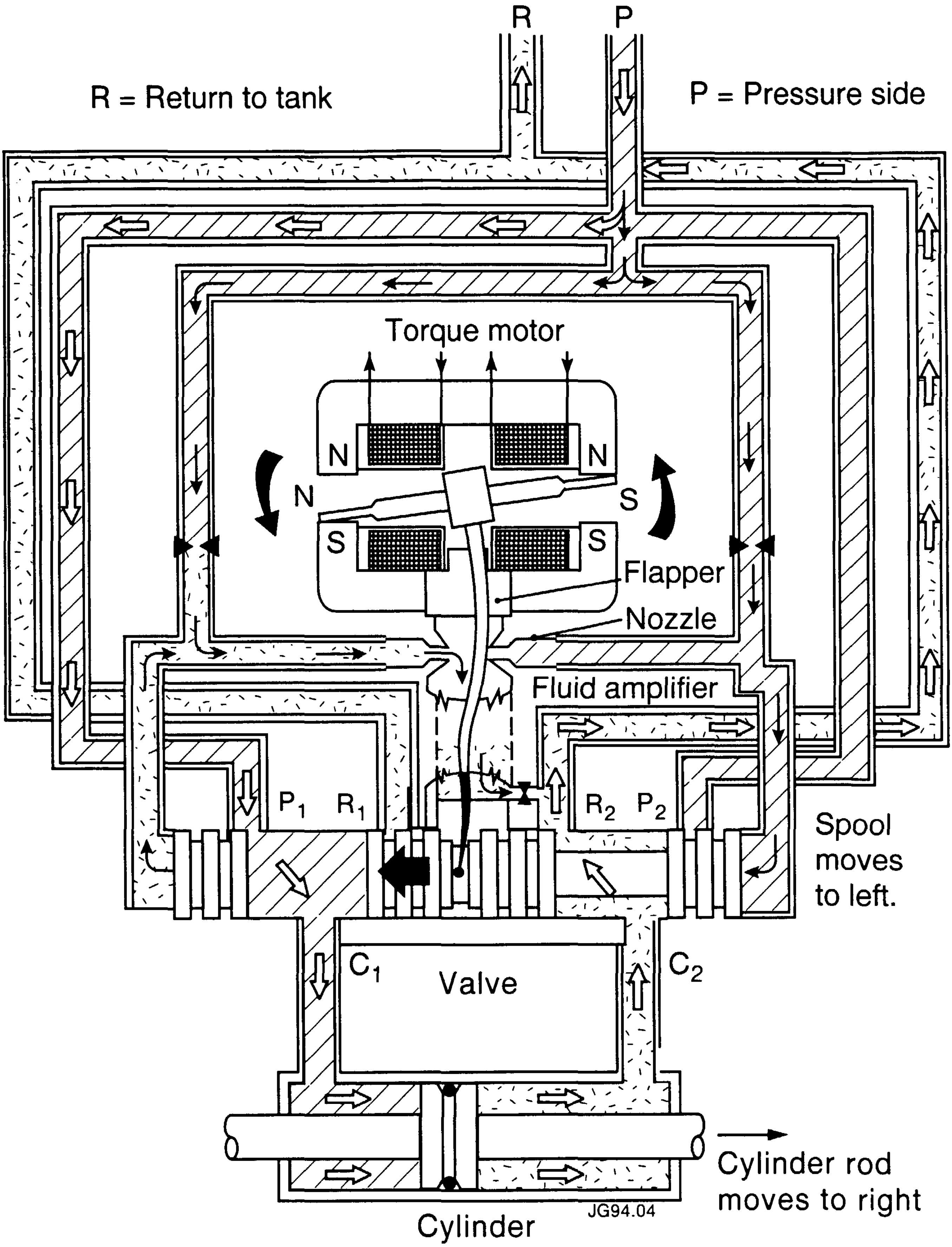


Figure 4. A two stage Servo valve with mechanical feed-back.

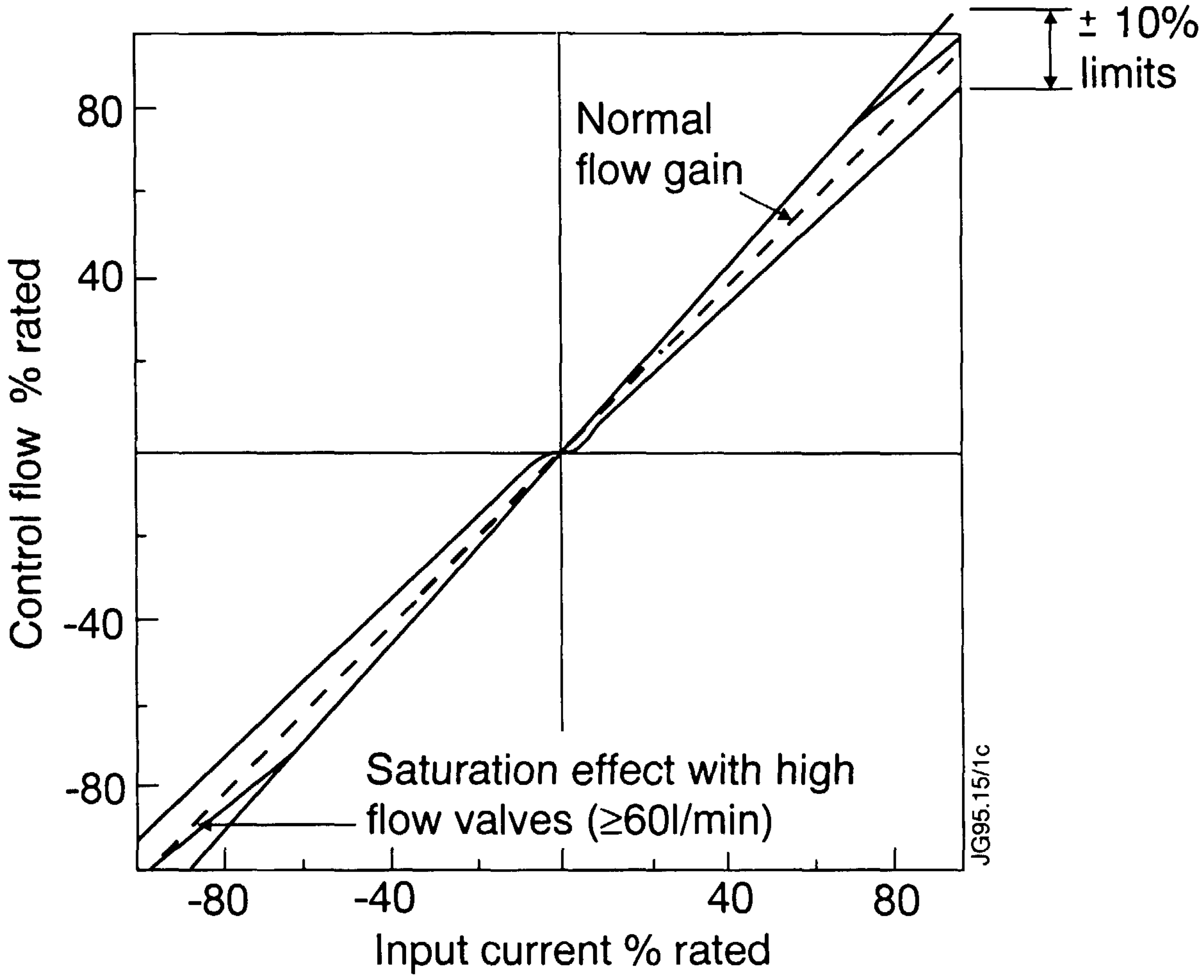
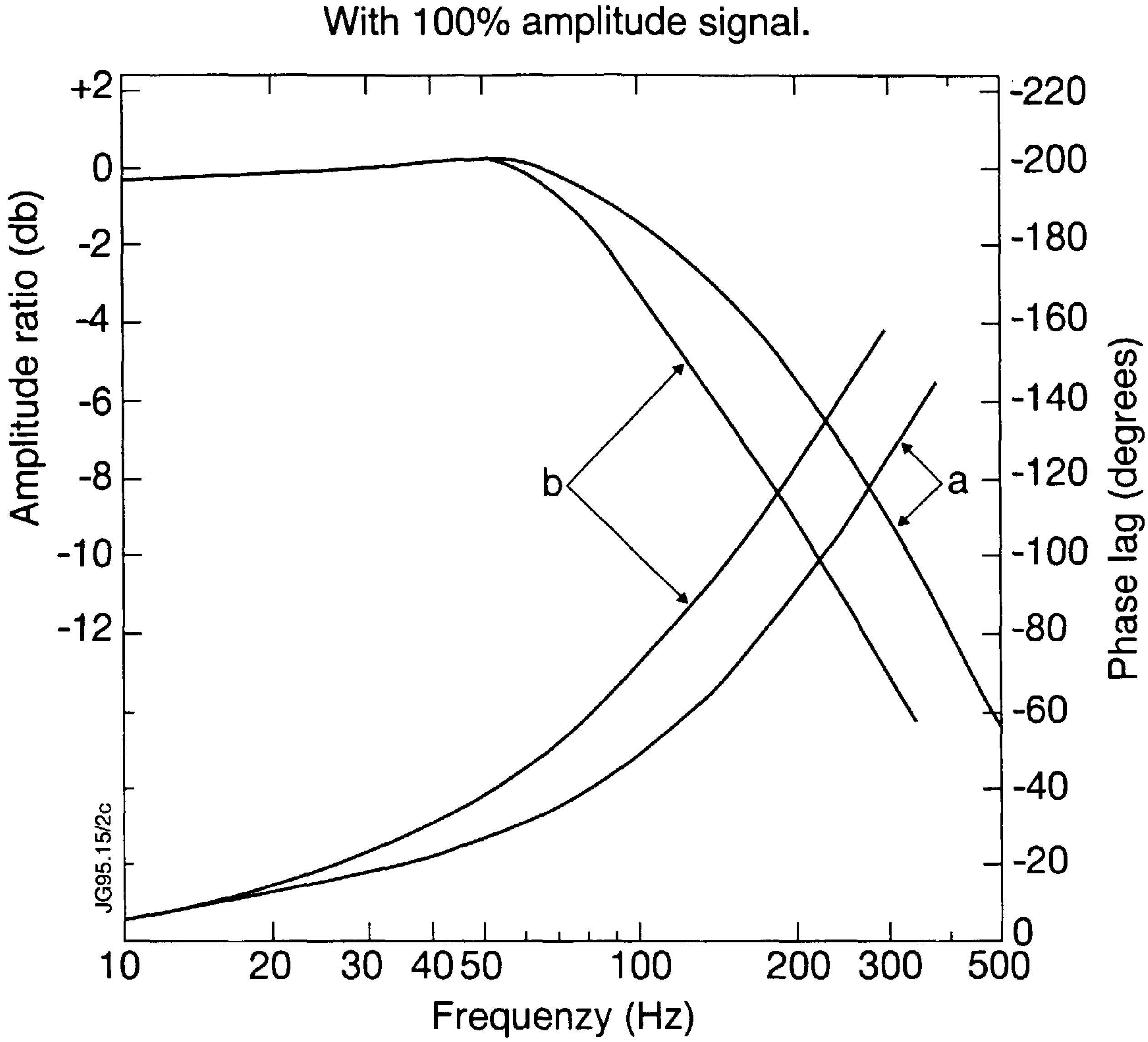


Figure 5 Typical no load flow gain tolerances
(Ref. Moog Controls Ltd.)



a = Model E760 – 230, 231, 232 b = Model E760 – 233

Figure 6 Frequency responce of Moog 760 high responce valves

(Ref Moog Controls Ltd.)

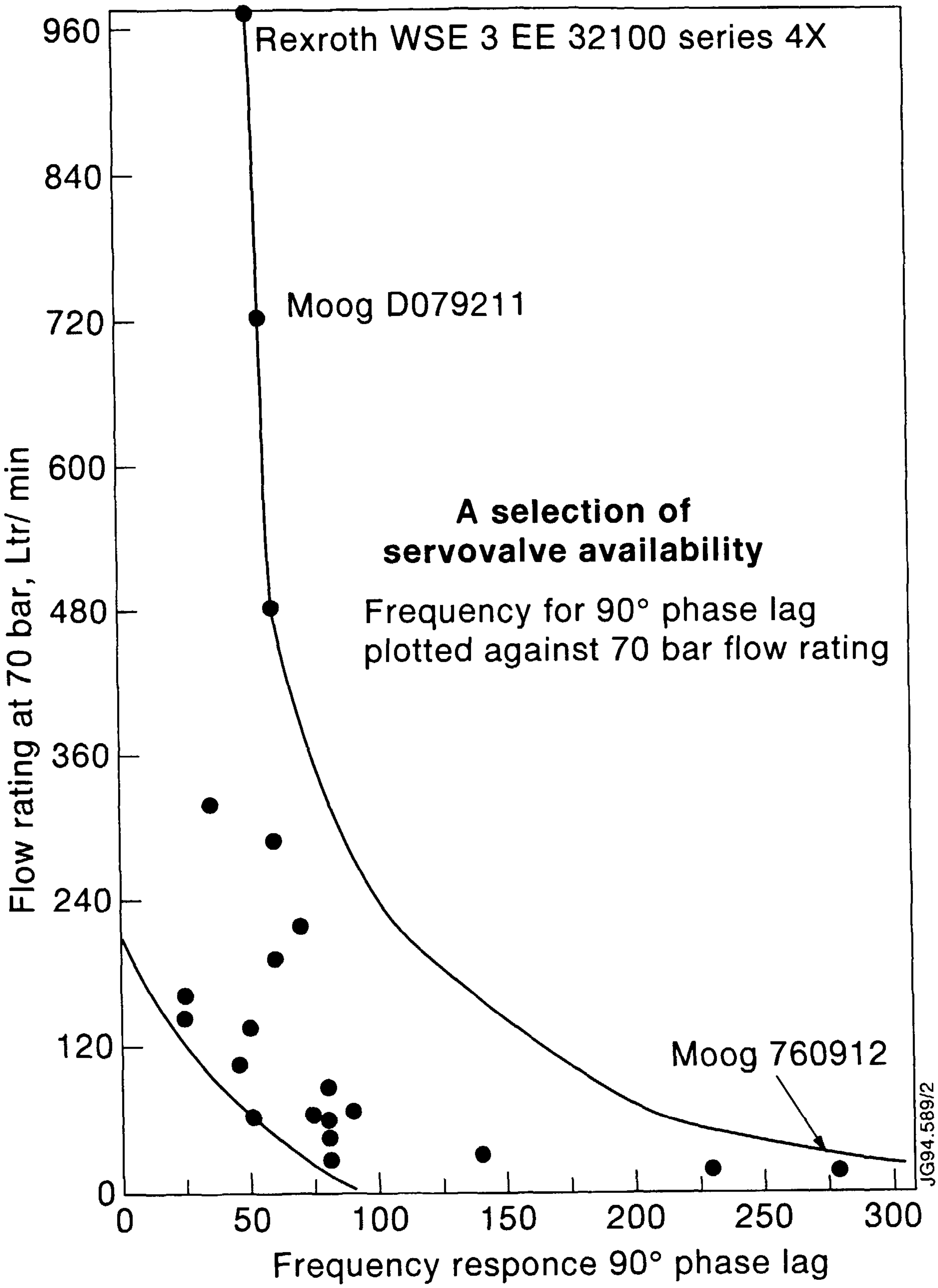
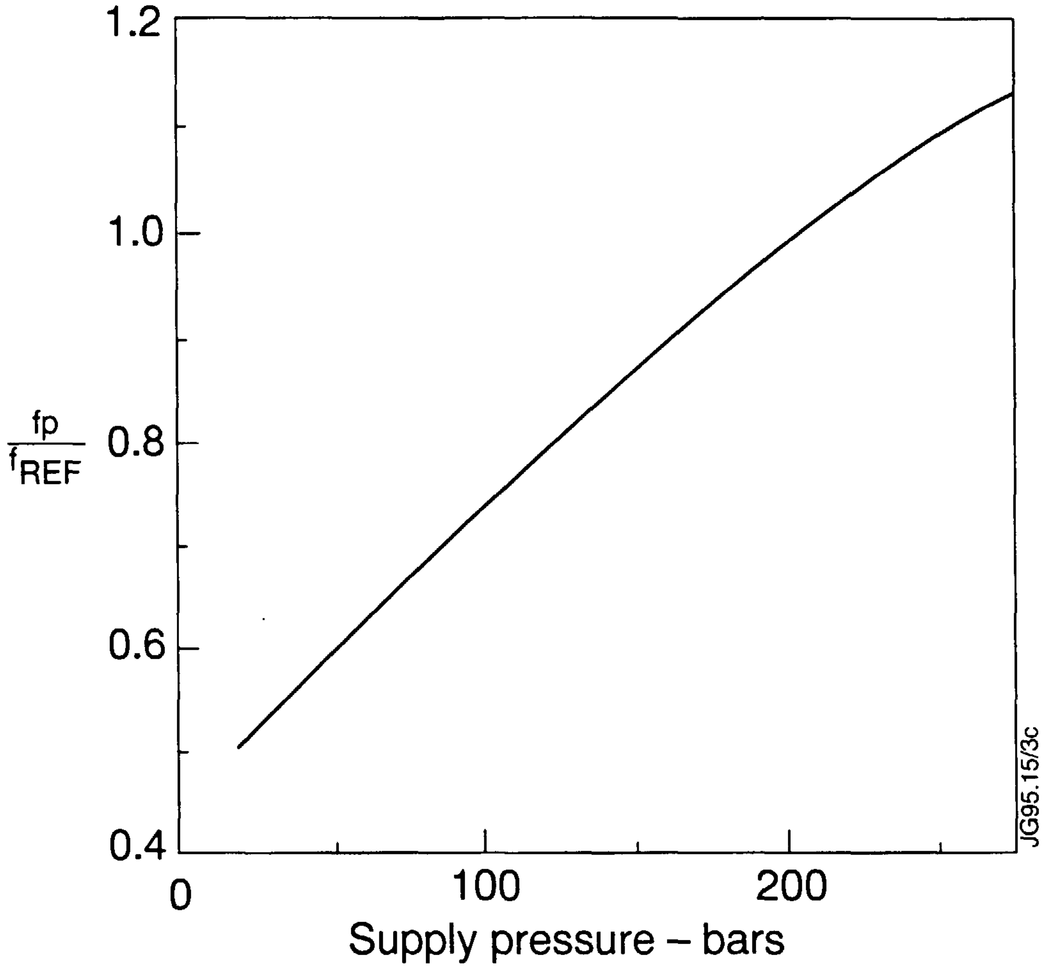


Figure 7 Performance of available servo valves



$$\frac{fp}{f_{REF}} = \frac{\text{Natural frequency at other pressure}}{\text{Natural frequency at 210 bars}}$$

Figure 8 Typical variation in frequency response with pilot stage pressure
(Ref Moog Controls Ltd)

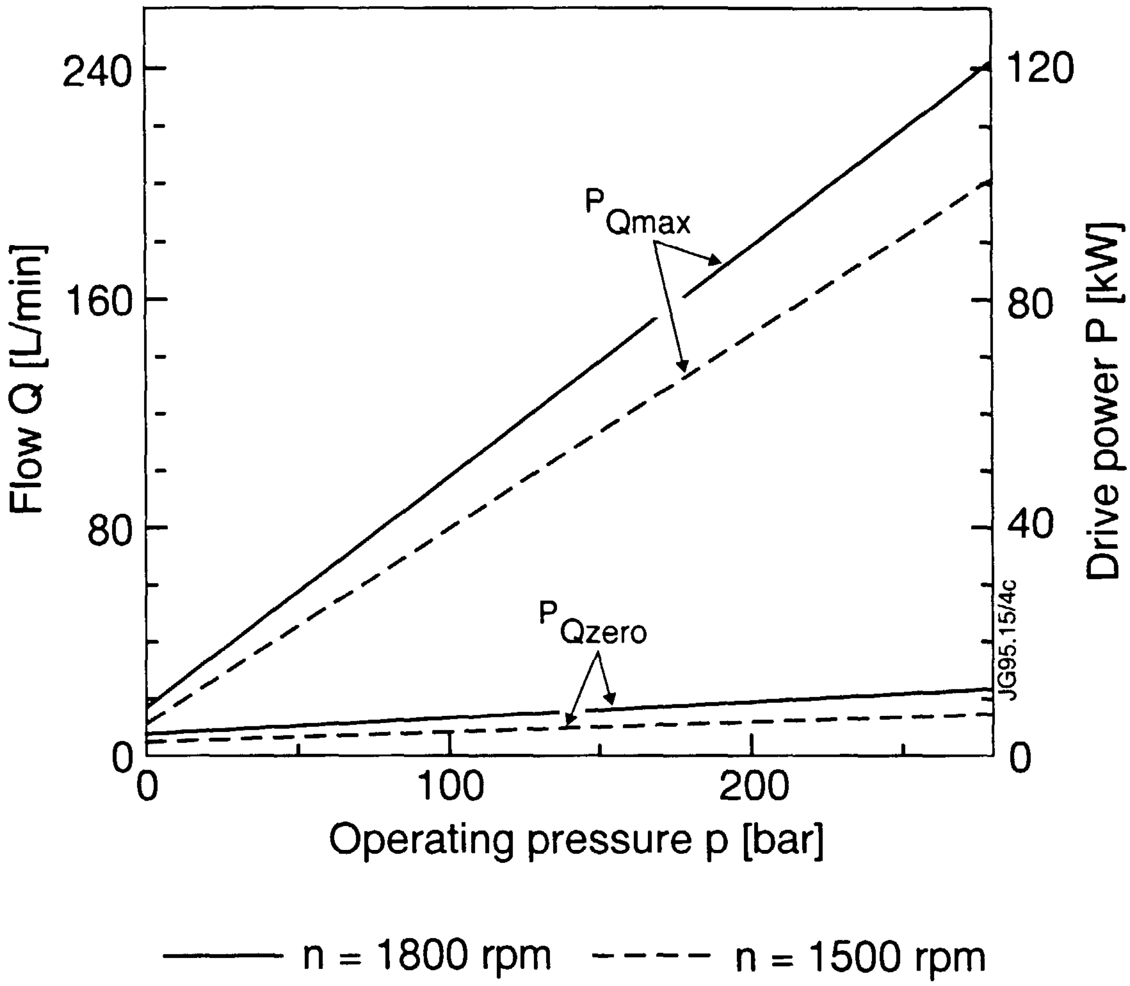


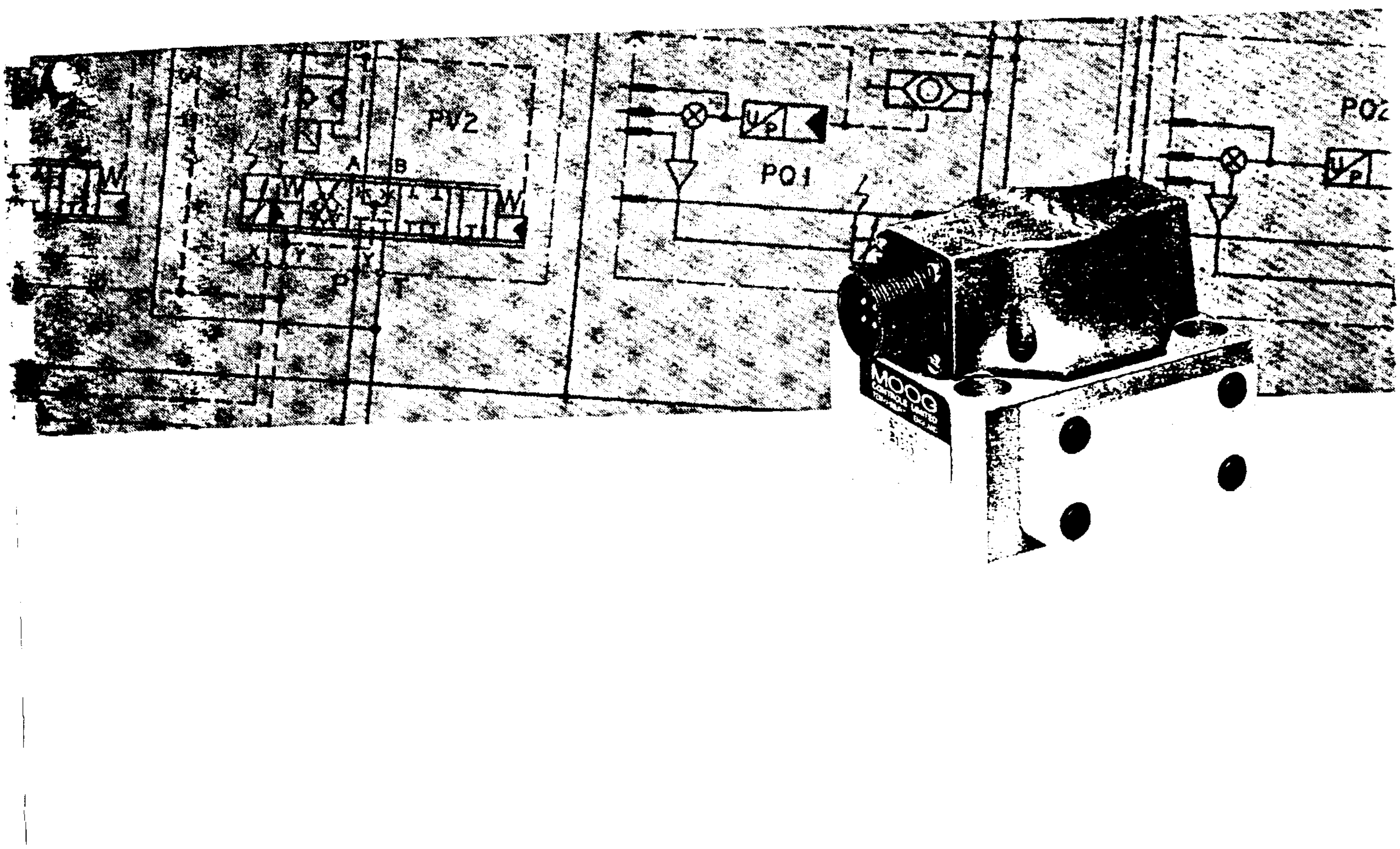
Figure 9 Flow power characteristics of Rexroth Ltd. variable displacement axial piston pump with constant pressure controller type A10VS 140

APPENDIX 1

Performance Data on Moog 760 series valves.

MOOG

Servovalves series E760



Features of the E760

Design

The two-stage design incorporates a friction-less pilot stage driving a spool and bushing power stage.

Pilot

The magnetic components of the pilot stage are sealed from the fluid in a dry compartment, preventing contamination from ferrous particle pick-up.

Feedback

The feedback from power to pilot stage is transmitted mechanically through a simple cantilever spring.

Response

Optimum response is achieved by restricting the oil volume between pilot and power stage to a minimum.

Null shift

Temperature and pressure null shift are minimised by using a symmetrical construction in the pilot stage.

Stability

Stability is improved by the incorporation of a new design of bushing and spool.

Null bias

The series E760 has a convenient mechanical null bias adjustment.

Fifth port

The optional fifth port facility provides flexibility for operating the main spool stage at low system pressures, whilst still achieving maximum performance from the first stage. This is particularly useful when a 'soft start' facility is required.

Construction

All features described are incorporated in a design featuring rugged construction and high performance. The electrical coils are protected from extremes of thermal and vibration stresses by resilient potting.

Intrinsic Safety

Approved to Harmonised European standards EN50 014 & 020 (BS 5501 Parts 1 & 7) up to EEx ia IIc T6 dependent upon coil configuration.

Also approved by Factory Mutual Corporation up to class I, II, III, Div. I, app. groups A, B, C, D, E, & G dependent upon coil configuration.

Refer to Moog Technical Bulletins for futher information on intrinsic safety

Hydraulic characteristics

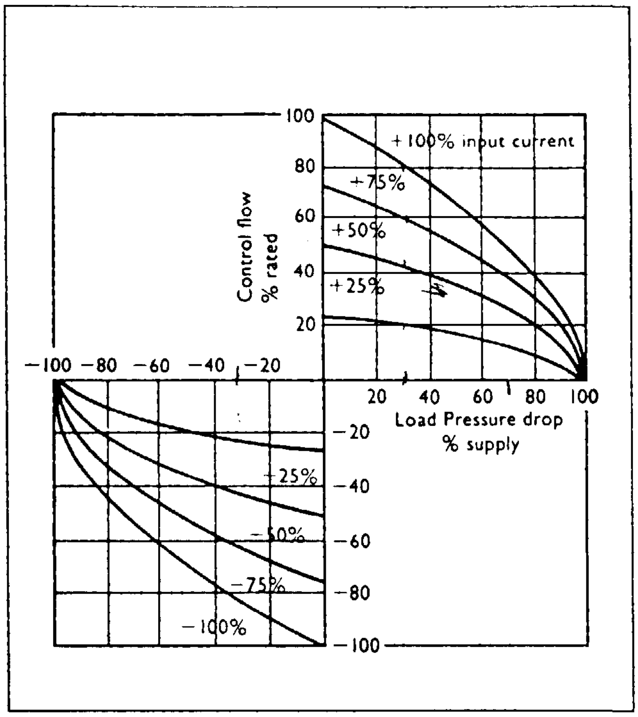
Flow-load characteristics
Control flow to the load will change with load pressure drop and electrical input as shown below. These characteristics follow closely the theoretical square-root relationship for sharp-edged orifices, which is $Q_L = Ki\sqrt{P_V}$

Q_L = control flow
K = valve sizing constant
i = input current
P_v = valve pressure drop

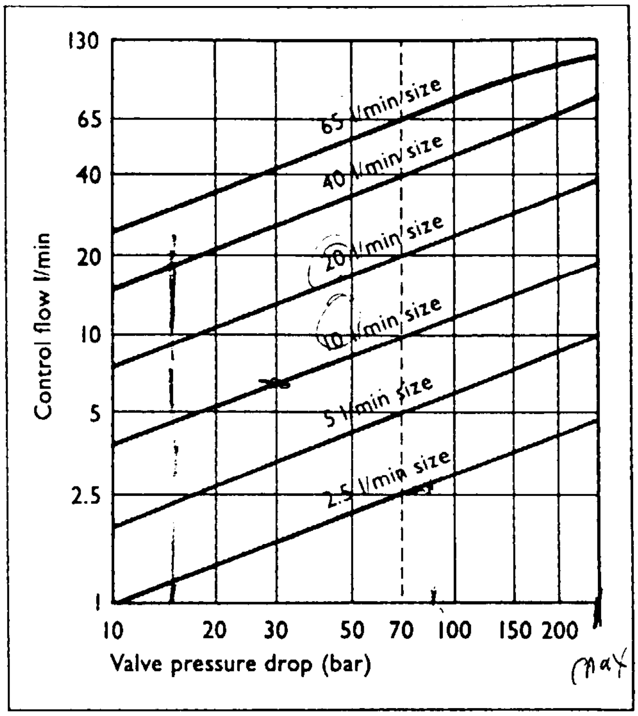
Internal leakage
Maximum internal leakage is determined by the valve rated flow. The maximum leakage at 210 bar supply for typical E760 Series Servovalves is shown on page 8.

Unless specified otherwise, all performance parameters are given for valve operation using petroleum based hydraulic fluid at 40°C and 35 cSt.

Change in control flow with current and load pressure



Change in control flow with valve pressure drop



Technical data

E760 Servovalves are intended to operate with constant supply pressure to the pilot stage.

Minimum pilot pressures required are 3 bar to move the spool with 15 bar as a recommended minimum. For low pilot pressure applications please consult Moog.

Pressure rating:

Standard	210 bar
High pressure (aluminium body)	315 bar
Stainless Steel Body Versions	420 bar

Proof pressures:
Pressure port P — 150% rated supply pressure
Control ports C1, C2 — 150% rated supply pressure
Return port R — 100% rated supply pressure up to maximum of 350 bar.

Fifth port. For supply pressures in excess of 350 bar use fifth port option, restricting pilot pressure to 350 bar.

Control flow with 70 bar pressure drop.

Standard response valves:
2.5–5–10–20–40–65–75 l/min

High response valves:
3.8–9.5–19–38.5 l/min

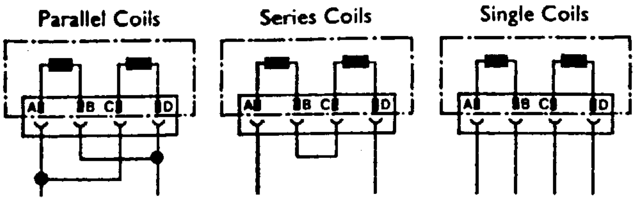
Super-high response valves:
3.8–9.5–19 l/min

For other flow rates please contact the factory.

Electrical characteristics

Rated current and coil resistance
A wide variety of coils is available for E760 Servovalves. Table 2 lists some of the more popular configurations and their parameters. Please contact the factory with other requirements.

Standard electrical configuration



Connector pins
External connections and electrical polarity for flow out of C2 are
single coils: A+, B−; or C+, D−
series coils: tie B to C; A+, D−
parallel coils: tie A to C and B to D; A & C+, B & D−

Coil connections
A four-pin electrical connector (that mates with an MS3106–14S–2S) is standard. All four torque motor leads are available at the connector so external connections can be made for series, parallel, or differential operation.

E760 Servovalves can be supplied on special order with other connectors or a pigtail. Also the coils can be wired internally for 2- or 3-wire operation.

Servoamplifier
The servovalve responds to input current, so a servoamplifier that has high internal impedance (as obtained with current feedback) should be used. This will reduce the effects of coil inductance and will minimize changes due to coil resistance variations.

Coil impedance
The resistance and inductance of standard coils are given in Table 1. The two coils in each servovalve are wound for equal turns with a normal production tolerance on coil resistance of ±12%. Copper magnet wire is used so the coil resistance will vary significantly with temperature. The effects of coil resistance changes can be essentially eliminated through use of a current feedback servoamplifier having high output impedance.

Inductance is determined under pressurized operating conditions and is greatly influenced by back emf's of the torque motor. These effects vary with most operating conditions, and vary greatly with signal frequencies above 100 Hz.

Electrical connections (standard coils)

Table 1	Parallel Coils	Series Coils	Single Coils
Coil Resistance ±10% (at 25°C)	40Ω	160Ω	80Ω
Rated Current	±40mA	±20mA	±40mA
Nominal voltage for rated current	±1,6V⎓	±3,2V⎓	±3,2V⎓
Inductance	0,18 Henry	0,66 Henry	0,22 Henry
Electrical Power	0,064 Watt	0,064 Watt	0,128 Watt
Connections for flow out Control Port C2	A and C(+) B and D(−)	A(+) D(−), B and C connected	A(+) B(−) or C(+), D(−)

Note: *Other coil ratings are available, including intrinsically safe versions. Consult Moog for details. See Table 2*

Coil ratings available for E760 Servovalves

Table 2 Nominal resistance per coil at 70°F (21°C) Ohms	**Recommended rated current—mA** Differential Parallel or Single Coil Configuration	Series Coils	**Approximate coil inductance—Henrys** Single Coils	Differential* Coils	Series Coils	Parallel Coils
22	200	100	0.07	0.10	0.21	0.06
80	40	20	0.22	0.34	0.66	0.18
200	15	10	0.72	1.1	2.2	0.59
500	15	7.5	1.3	2.1	4.1	1.1
1000	10	5	3.2	5.0	9.7	2.6

**Inductance per coil with differential operation (class A push-pull).*

Performance characteristics

Common performance characteristics

Resolution	<0.5%
Hysteresis	<3%
Null bias	<2%
Null shift with 55°C Temperature change	<2%
With change in supply pressure from 80% to 110%	<2%
Operating nominal temperature range	−54°C to +135°C
Fluid	Petroleum based hydraulic fluids 10 to 100 cSt at 38°C
Seal material	Buna N (Viton A, EPR available to special order)
Supply filtration recommended	15 µm absolute or better (ß 15 > 75)

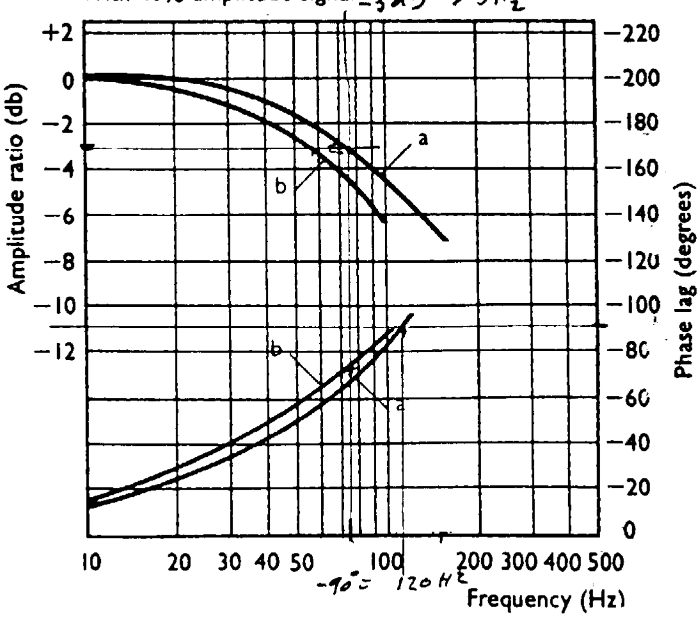
Standard valves

Model numbers

Control flow with Pv—70 bar (l/min±10%)	Standard pressure 210 bar 4 port	5 port	High pressure 315 bar 4 port	5 port	Internal leakage (max) 210 bar supply,
2.5	E760–400	E760–450	E760–500		<1.1
5	E760–401	E760–451	E760–501	E760–551	<1.2
10	E760–402	E760–452	E760–502	E760–552	<1.3
20	E760–403	E760–453	E760–503	E760–553	<1.5
40	E760–405	E760–455	E760–505	E760–555	<2.3
65	E760–406	E760–456	E760–506	E760–556	<2.3 VI
75	E760–410	E760–458			

Typical frequency response of standard response servovalves at 210 bar supply pressure

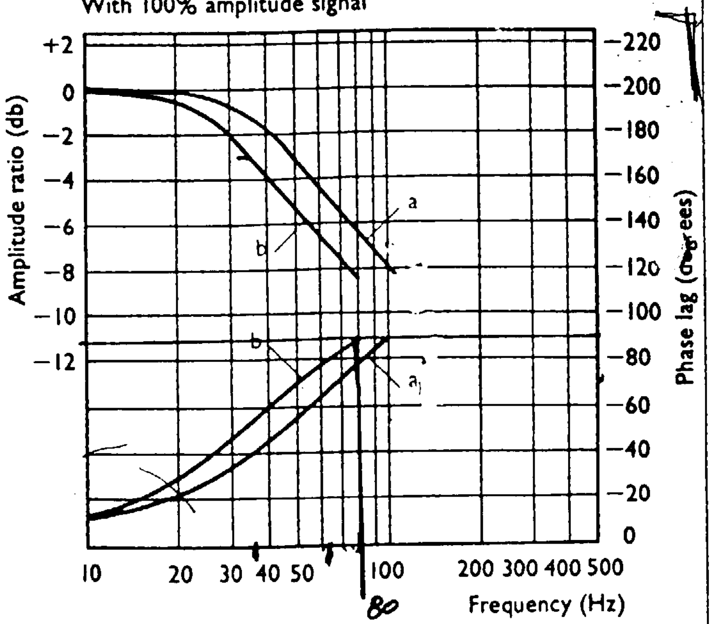
With 40% amplitude signal -3db 75Hz



a—Model E760–401, 402, 403, 405

b—Model E760–406

With 100% amplitude signal



a—Model E760–401, 402, 403, 405

b—Model E760–406

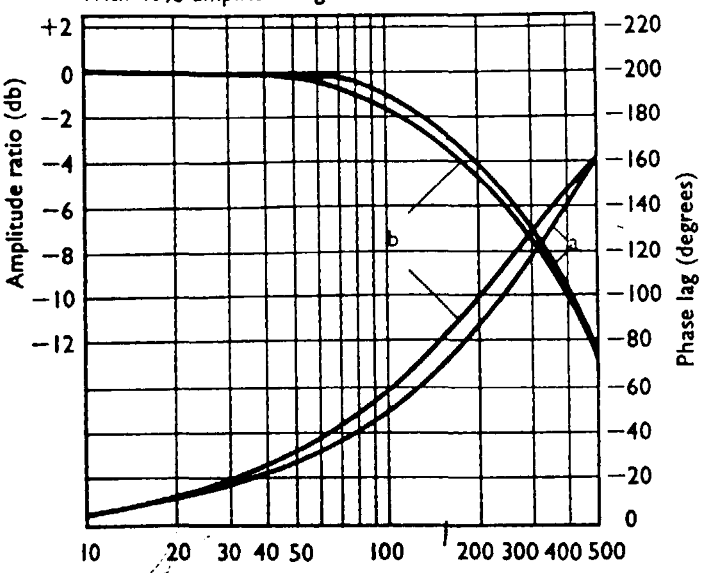
High response valves

Model numbers	Control flow (l/min ± 10%)	Standard pressure 210 bar	High pressure 315 bar
	3.85	E760–230	E760–530
	9.6	E760–231	E760–531
	19.2	E760–232	E760–532
	38.5	E760–233	E760–533 VZ — $1,525

XZ V3

Typical frequency response of high response servovalves with 210 bar supply pressure

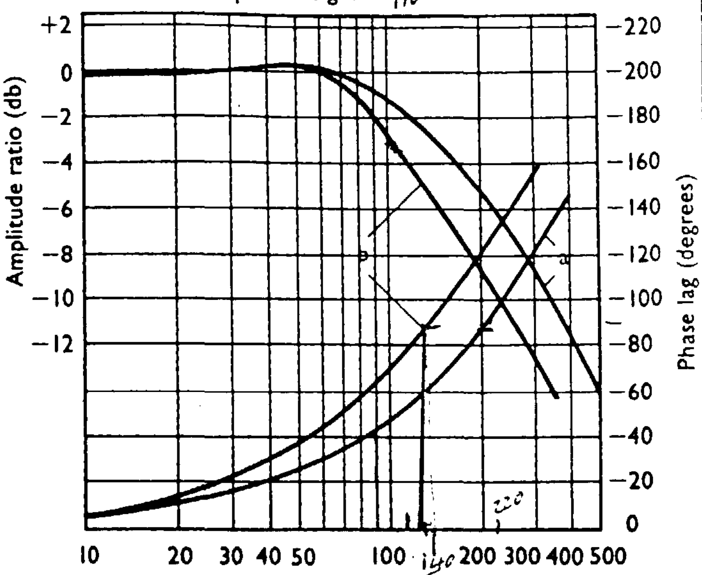
With 40% amplitude signal



a — Model E760–230, 231, 232 Frequency (Hz)

b — Model E760–233

With 100% amplitude signal 110



a — Model E760–230, 231, 232 15 Frequency (Hz)

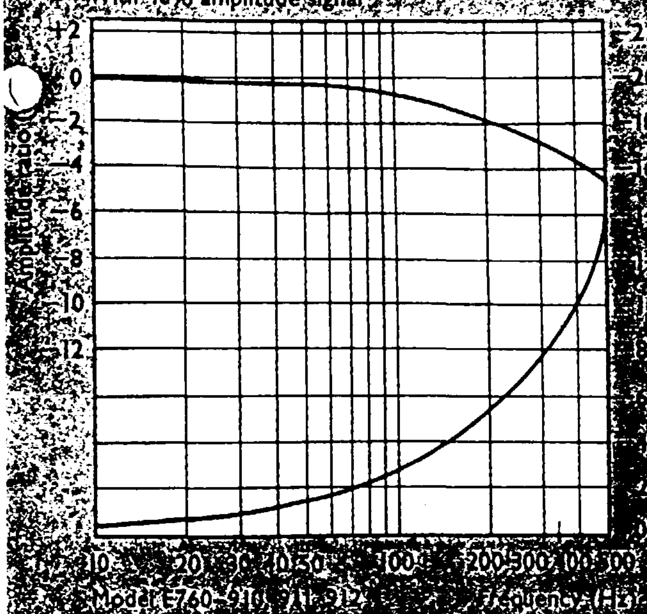
b — Model E760–233

Super-high response valves

Model numbers	Control flow (l/min ± 10%)	Standard pressure 210 bar	High pressure 315 bar
	3.85	E760–910	E760–920
	9.6	E760–911	E760–921
	19.2	E760–912	E760–922

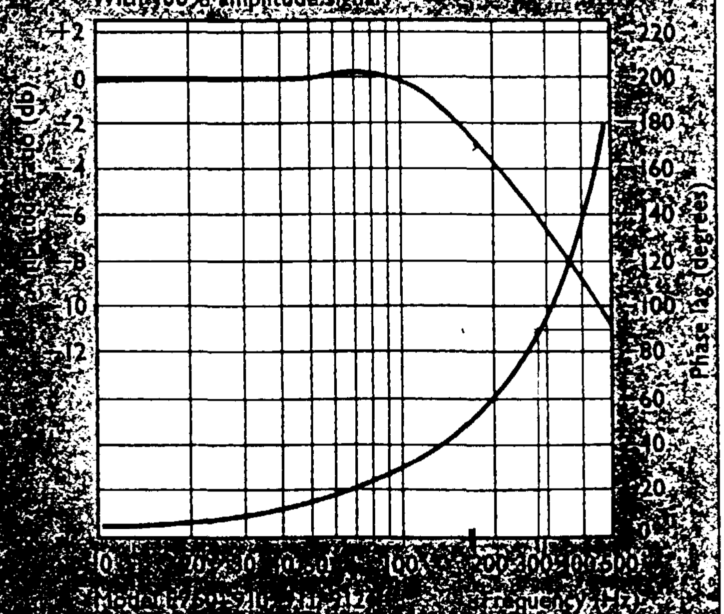
Typical frequency response of super-high response servovalves with 210 bar supply pressure

With 40% amplitude signal



Model E760–910, 911, 912 Frequency (Hz)

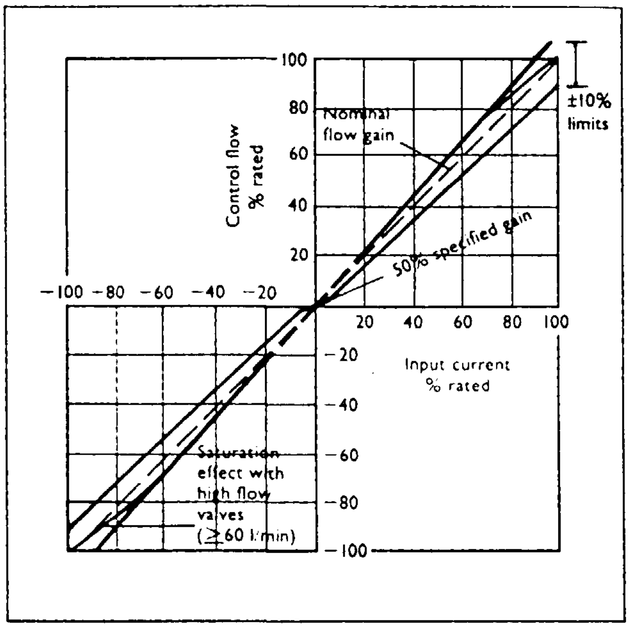
With 100% amplitude signal



Model E760–910, 911, 912 Frequency (Hz)

Performance characteristics

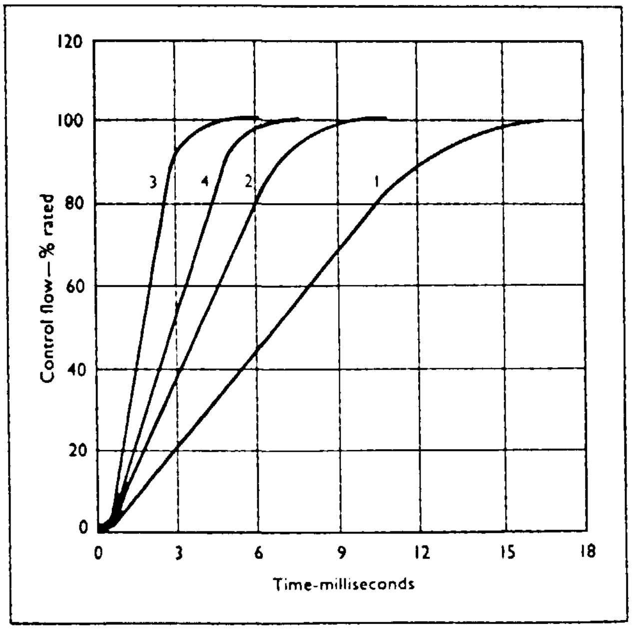
No-load flow gain tolerances



Flow gain

The no-load flow characteristics of E760 Servovalves can be plotted to show flow gain, symmetry and linearity. Typical limits (excluding hysteresis effects) are shown above.

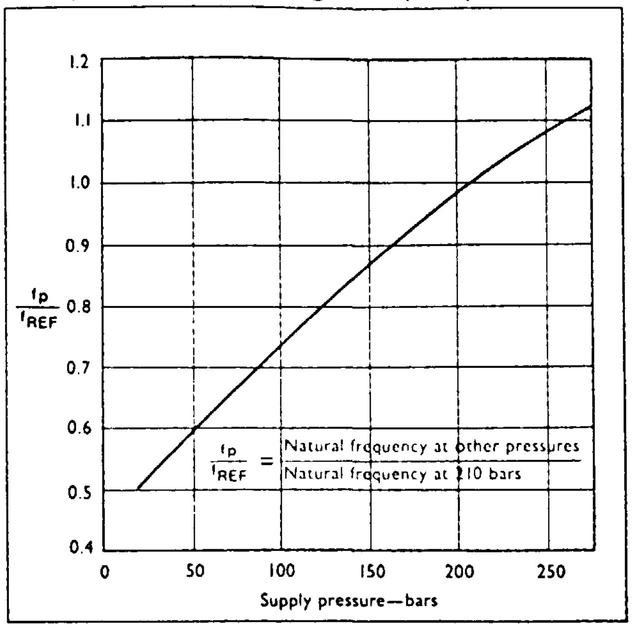
Step responses



Typical transient response of 760 Servovalves is shown above. The straight-line portion of the response represents saturation flow from the pilot stage which will vary with the square root of the change in supply pressure.

1 E760–406
2 E760–401/2/3/5
3 E760–230/1/2
4 E760–233

Frequency response change with pilot pressure



Null flow gain

Nonlinearity of control flow to input current in the null region is due to variations in the spool cut. Typically the valve flow gain about null (within ±5% of rated current input) may range from 50 to 100% of the specified flow gain. Special null cuts are available.

Spool driving forces

The maximum hydraulic force available to drive the second-stage spool will depend upon the supply pressure. The standard first-stage configuration for E760 Servovalves will produce a spool driving force gradient which exceeds 5N/% input current with 210 bar supply. The typical spool driving force with 210 bar supply is 700N.

Pressure gain

The blocked load differential pressure will change rapidly from one limit to the other as input current causes the valve spool to traverse the null region. Normally the pressure gain at null for E760 Servovalves exceeds 30% of supply pressure for 1% of rated current and can be as high as 100%.

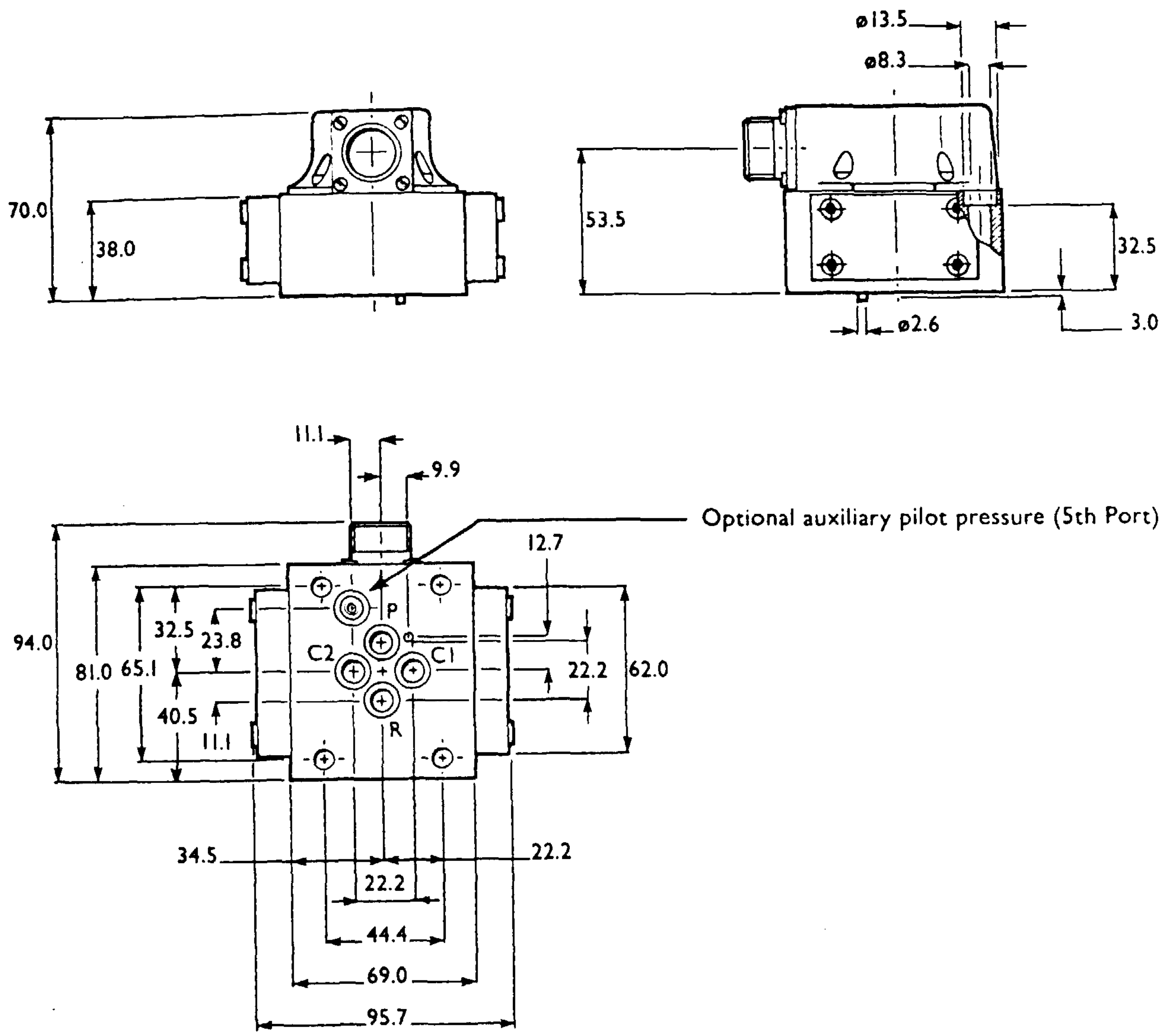
Null

Externally adjustable using 3/32" Allen key.

30% per 1% of supply

Installation

Installation diagram



All dimensions in mm.

Installation details

Suggested mounting screws: M8×45 Capscrews, torqued to 25 Nm.
Electrical connector is available at 90° to 180° to position shown.
Ports P, R, C1 & C2 ø9.5: O-rings 1,78mm section × 10,82 I.D.
Pilot port ø3: O-ring 1,78mm section × 9,25 I.D.
Hole for orientation pin in base ø3 min.
Mounting surface for valve requires a finish of Ra 1 µm or better, flat to within 0,02mm.
Electrical connector mates with MS3106–14S–2S or equivalent.
Null adjust: flow out C2 will increase with clockwise rotation of null adjust pin.
Compressed oil volume for one control port: 4,0ml.
Valve weight: 1,0 kg.

APPENDIX 2

Hydrosoft Performance simulation.

Appendix II. System 1: Single Moog 760 406 Valve.

System Parameters

```
system reference:ITERLRDA40 Closed loop system
valve operated system without flow feedback
4 way valve
Double ended Cylinder
 4 bore x  2.8 rod dia. x  7 stroke (cm)
inlet cylinder area= 6.408849 cm», outlet cylinder area= 6.408849 cm»
mass = 15 kg
coefficient of viscous damping =  0 kp/cm per sec
load = 2 kN
inlet/outlet orifice area ratio= 1
equivalent valve flow rating= 65.37013 L/min
actuator velocity= 170 cm/sec
inlet flow = 65.37013 L/min
supply pressure = 210 bar,  valve pressure drop= 173.9065 bar
port diameter= .95 cm
trapped volume= .2448619 L ( 4.486195E-02 :Actuator +  .2 :Piping)
actuator shunt coefficient = 0 L/min per bar
bulk modulus= 13793 bar
WH0=125.01hz
DF0=  0.45
K0/K= 1
```

Steady State Errors

```
STEADY-STATE ERRORS
SPECIAL FEATURES
VALVE PARAMETERS MODEL:E760-406
Flow Gain= 1.024524 L/min per % of command signal
Pressure Gain= 63 bar per % of command signal
Hysteresis= 2 % of command signal units
Enter YES(Y) for mode update?
POSITION CONTROL
Load Error @  2 kN= 9.051324E-02 mm
Hysteresis Error= .365454 mm
Velocity Error @  170 cm/sec= 11.65896 mm
VELOCITY CONTROL
Load Error @  2 kN= 9.051324E-02 mm/sec
Hysteresis Error= .365454 mm/sec
Acceleration Error @  170 cm/sec»= 11.65896 mm/sec
```

Closed loop frequency response

```
Natural frequency WR= 26.18hz            CL Damping factor DFR=  0.38
Band width at 4 dB attenuation WC= 40.62hz
Max.Overshoot at  0.021sec = 27.02%
Step response time=  0.012sec
system reference:DX
```

Appendix II, System 1: Single Moog 760 406 Valve.

Open Loop system parameters

```
system reference:DX          Closed loop system
Components selected:E760-406
OPEN LOOP SYSTEM PARAMETERS
Frequencies in hz:
WH0=125.01
WH1= 80.00
WH2=%100000.00
WH3=%100000.00
Damping factors:
DF0=  0.45
DF1=  1.32
DF2=  0.00
DF3=  0.00
Time constants in seconds:
t0= 0         t1= 0          t2= 0          t3= 0          t4= 0
free integrator:SINGLE
loop gain=145.84  1/sec
```

Appendix II, System 1: Single Moog 760 406 Valve.

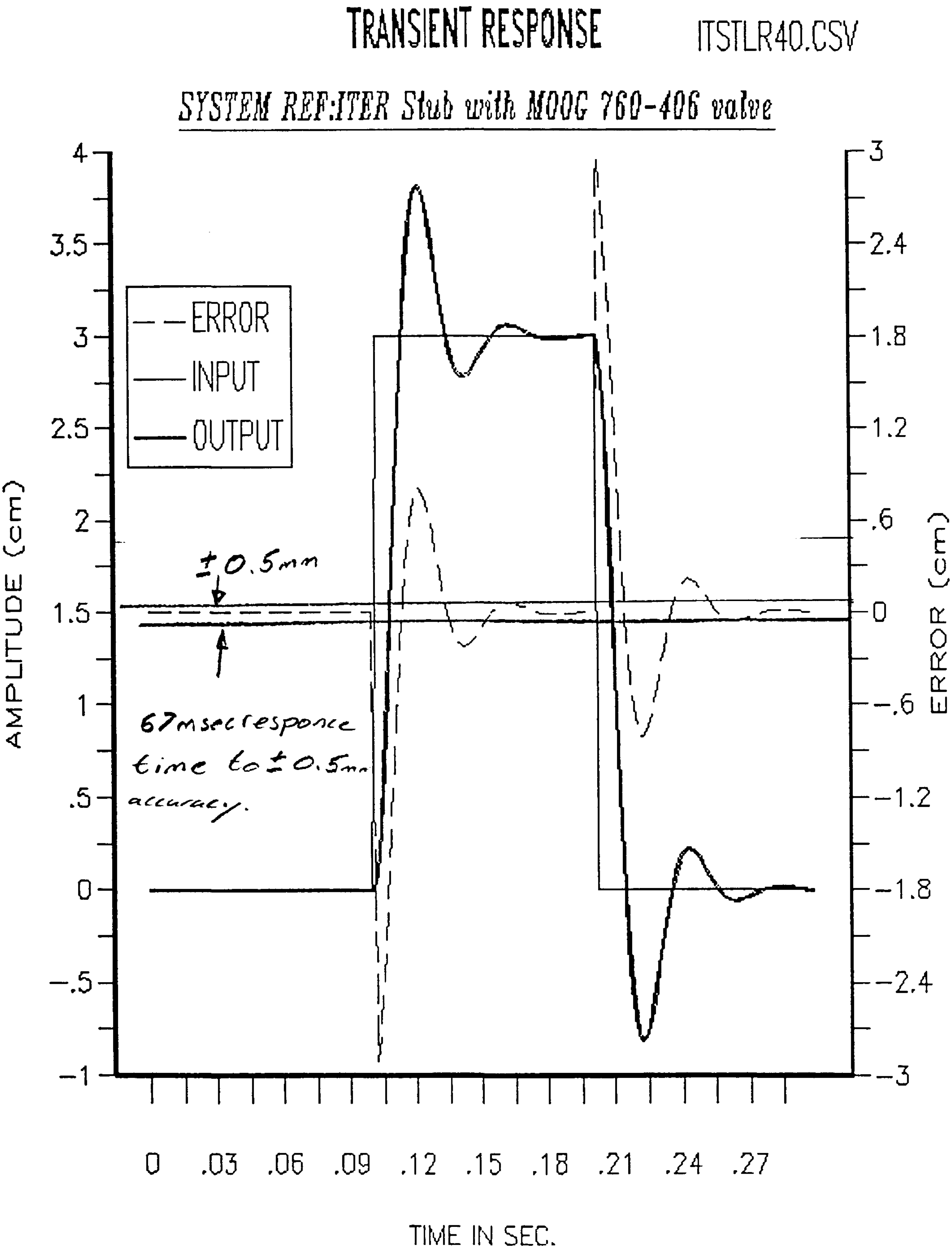


Fig. II. 1.1

Appendx II. System2: Single Moog 233 Valve with LVDT.

System Parameters

```
system reference:S760-233   Closed loop system
valve operated system without flow feedback
4 way valve
Double ended Cylinder
 4 bore x  2.8 rod dia. x  7 stroke (cm)
inlet cylinder area= 6.408849 cm», outlet cylinder area= 6.408849 cm»
mass = 15 kg
coefficient of viscous damping =  0 kp/cm per sec
load = 2 kN
inlet/outlet orifice area ratio= 1
equivalent valve flow rating= 57.67953 L/min
actuator velocity= 150 cm/sec
inlet flow = 57.67953 L/min
supply pressure = 210 bar,  valve pressure drop= 174.9887 bar
port diameter= .95 cm
trapped volume= .2448619 L ( 4.486195E-02 :Actuator +  .2 :Piping)
actuator shunt coefficient = 0 L/min per bar
bulk modulus= 13793 bar
WH0=125.01hz
DF0=  0.39
K0/K= 1
```

```
Valve Model:E760-233
Enter Security Code for update of model?

     VALVE LIMIT SETTINGS:
       Valve datum flow rate= 38.5 L/min, Valve pressure drop @
       datum flow rate= 70 bar, Valve overlap= 0 %
    Pressure Gain = 30 % of supply pressure, hysteresis = 1 % of command signa
       Model Code:E760-233
```

Steady State Errors

```
STEADY-STATE ERRORS
SPECIAL FEATURES
VALVE PARAMETERS MODEL:E760-233
Flow Gain= .6087188 L/min per % of command signal
Pressure Gain= 63 bar per % of command signal
Hysteresis= 1 % of command signal units
Enter YES(Y) for mode update?
POSITION CONTROL
Load Error @  2 kN= 4.367135E-02 mm
Hysteresis Error= 8.816317E-02 mm
Velocity Error @  150 cm/sec= 8.353957 mm
VELOCITY CONTROL
Load Error @  2 kN= 4.367135E-02 mm/sec
Hysteresis Error= 8.816317E-02 mm/sec
Acceleration Error @  150 cm/sec»= 8.353957 mm/sec
```

Appendx II. System2: Single Moog 233 Valve with LVDT.

Open Loop system parameters

```
system reference:DX          Closed loop system
Components selected:E760-233     LVDT
OPEN LOOP SYSTEM PARAMETERS
Frequencies in hz:
WH0=125.01
WH1=140.00
WH2=400.00
WH3=%100000.00
Damping factors:
DF0=  0.39
DF1=  0.87
DF2=  0.71
DF3=  0.00
Time constants in seconds:
t0= 0          t1= 0          t2= 0          t3= 0          t4= 0
free integrator:SINGLE
loop gain=179.59  1/sec
'C' to continue 'any key' for frequency domain access
```

Closed loop frequency response

```
Natural frequency WR= 46.06hz  *          CL Damping factor DFR=  0.42
Band width at 4 dB attenuation WC= 76.80hz
Max.Overshoot at  0.012sec = 23.21%
Step response time=  0.007sec
system reference:DX
```

Appendx II. System2: Single Moog 233 Valve with LVDT.

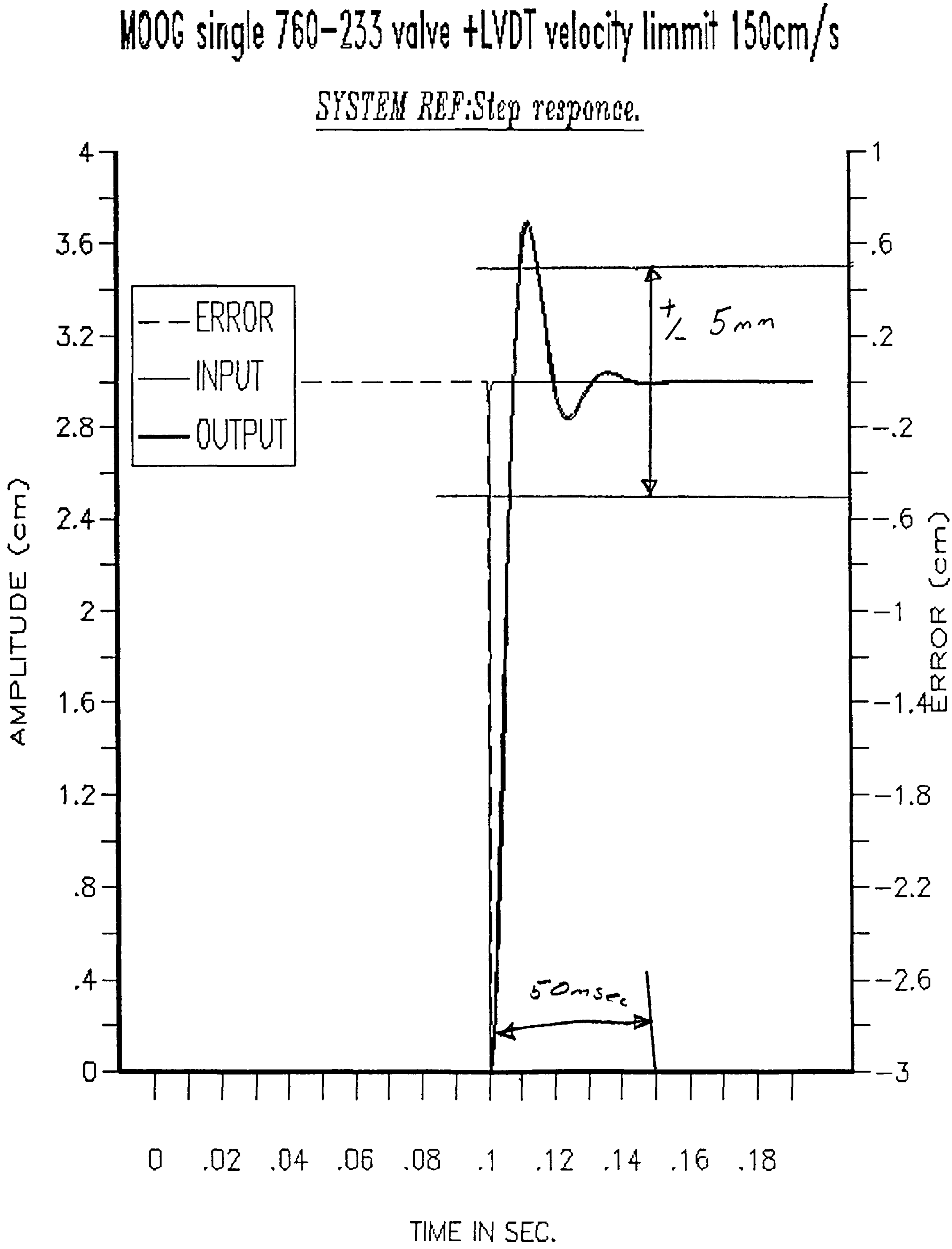


Fig. II. 2.1

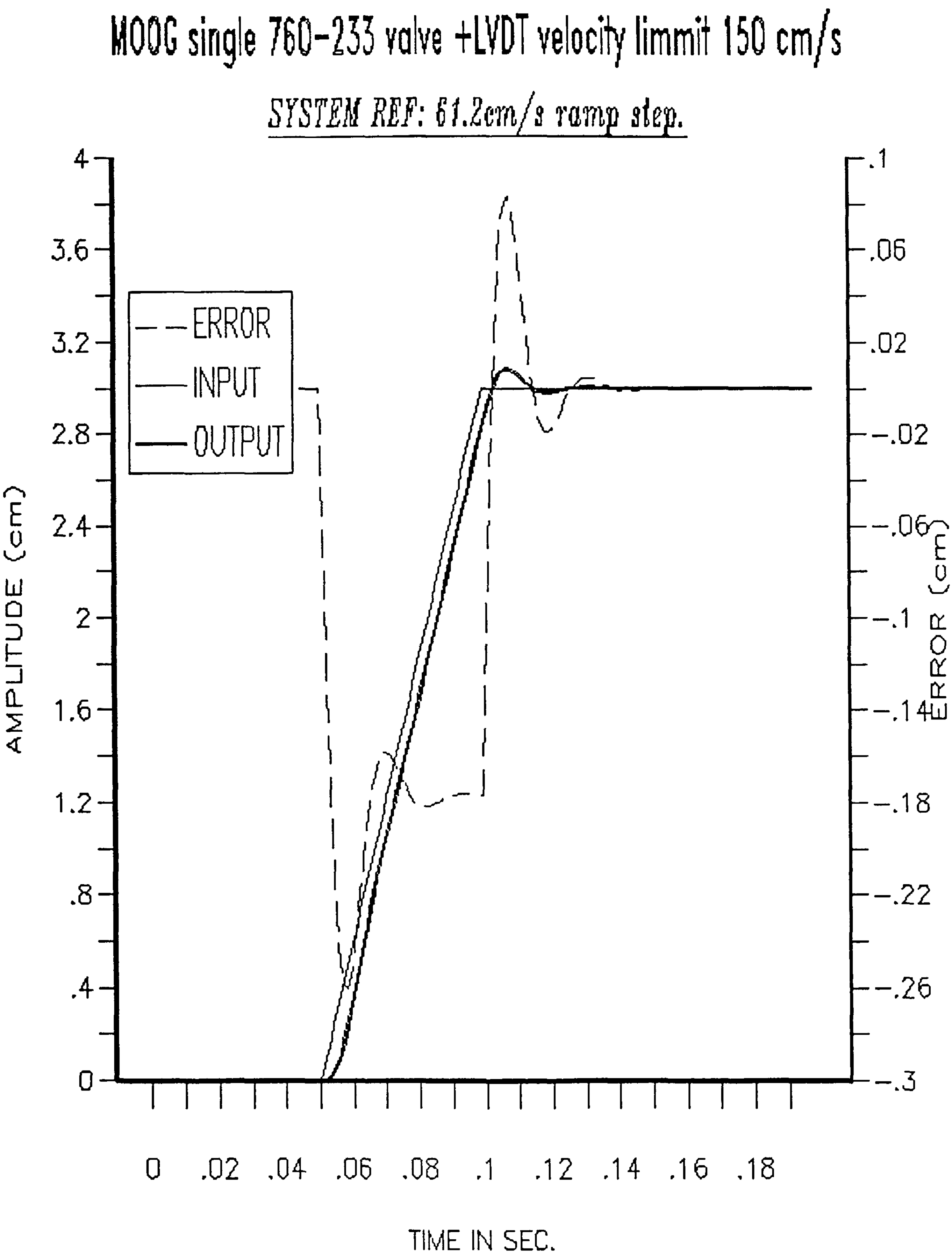

MOOG single 760-233 valve +LVDT velocity limmit 150 cm/s
SYSTEM REF: 61.2cm/s ramp step.
ERROR
INPUT
OUTPUT
AMPLITUDE (cm)
ERROR (cm)
4
3.6
3.2
2.8
2.4
2
1.6
1.2
.8
.4
0
.1
.06
.02
-.02
-.06
-.1
-.14
-.18
-.22
-.26
-.3
0
.02
.04
.06
.08
.1
.12
.14
.16
.18
TIME IN SEC.

Fig. II. 2.2

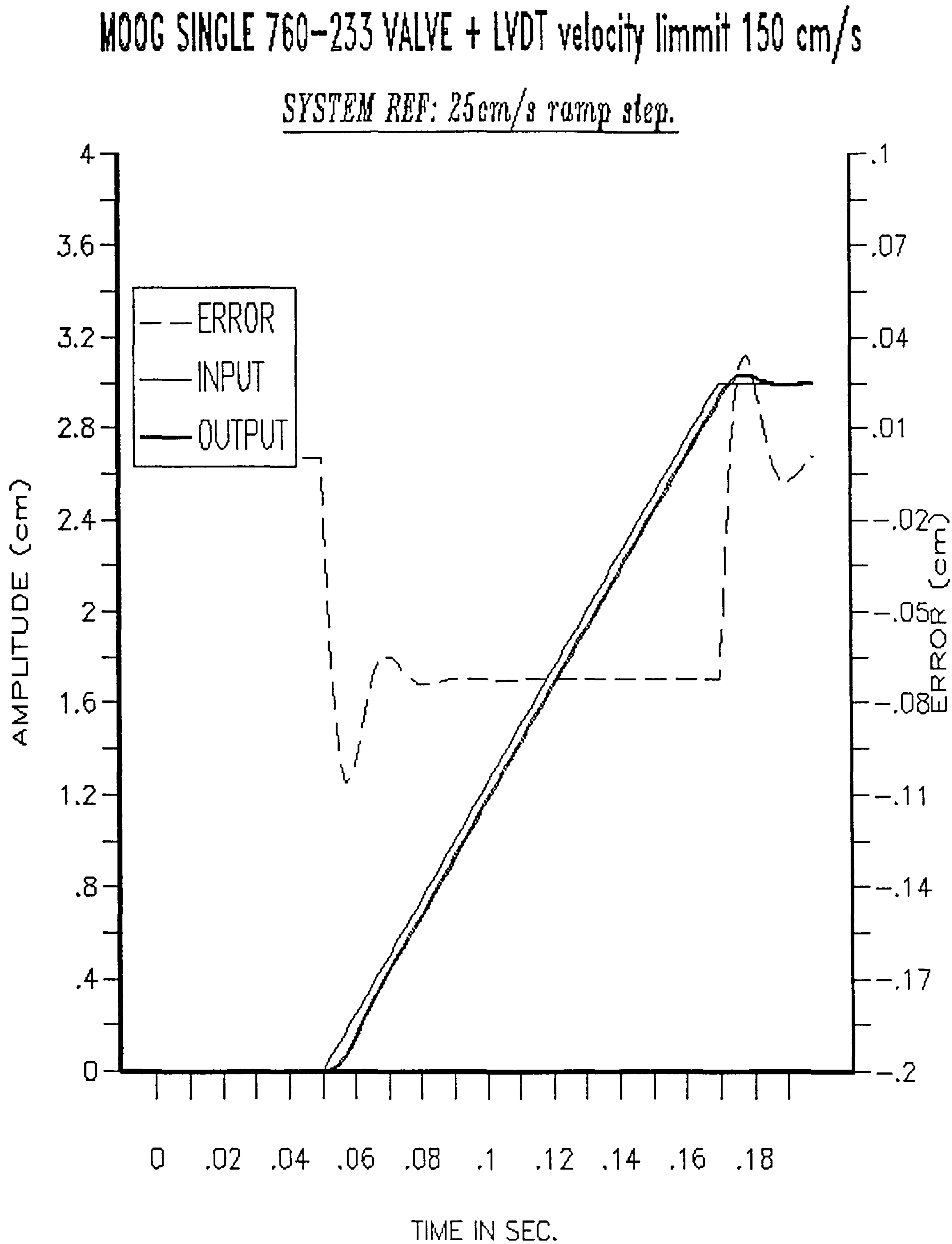

MOOG SINGLE 760-233 VALVE + LVDT velocity limmit 150 cm/s
SYSTEM REF: 25cm/s ramp step.
ERROR
INPUT
OUTPUT
AMPLITUDE (cm)
ERROR (cm)
TIME IN SEC.
4
3.6
3.2
2.8
2.4
2
1.6
1.2
.8
.4
0
.1
.07
.04
.01
-.02
-.05
-.08
-.11
-.14
-.17
-.2
0
.02
.04
.06
.08
.1
.12
.14
.16
.18

Fig. II. 2.3

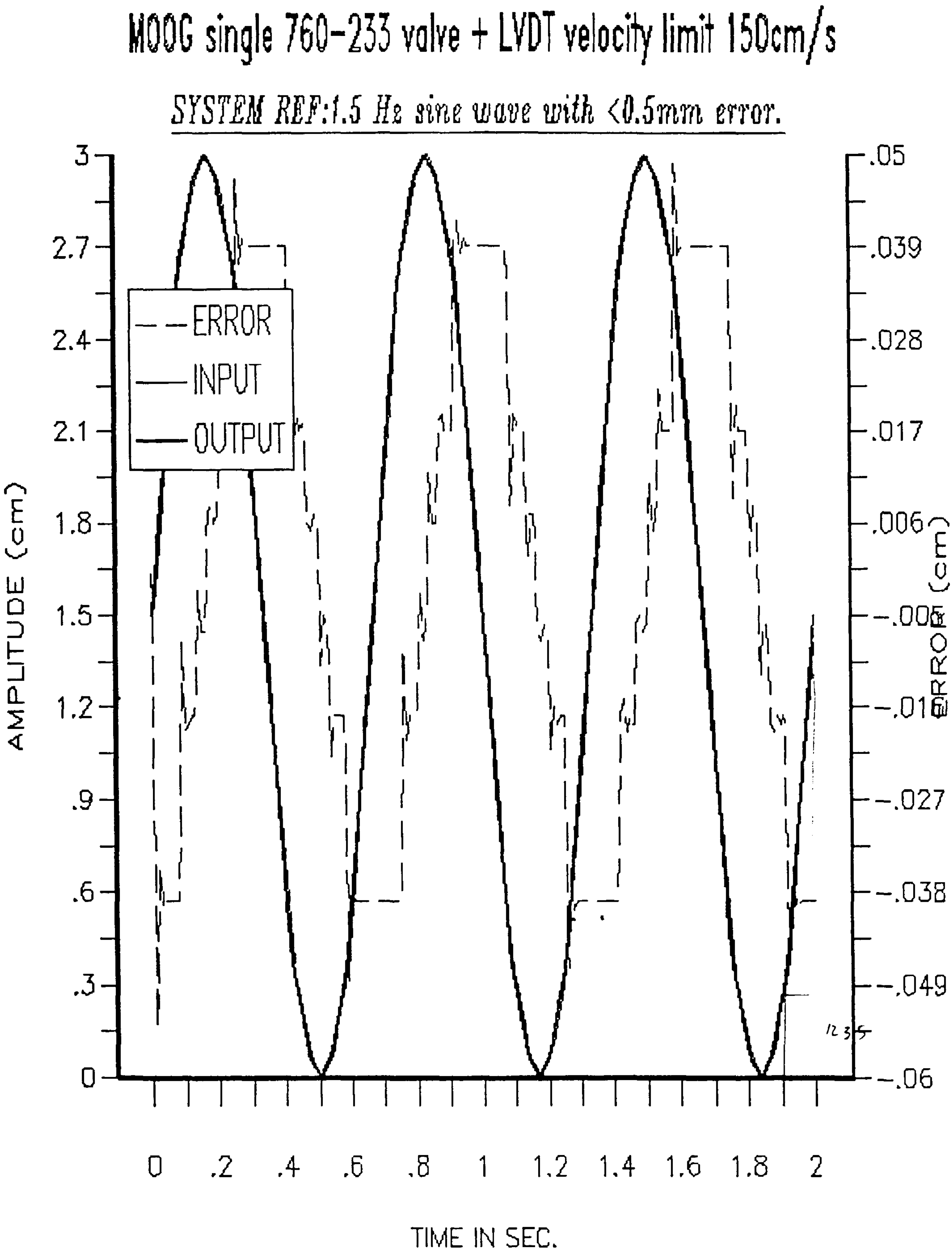

MOOG single 760-233 valve + LVDT velocity limit 150cm/s
SYSTEM REF:1.5 Hz sine wave with <0.5mm error.
ERROR
INPUT
OUTPUT
AMPLITUDE (cm)
ERROR (cm)
3
2.7
2.4
2.1
1.8
1.5
1.2
.9
.6
.3
0
.05
.039
.028
.017
.006
-.005
-.016
-.027
-.038
-.049
-.06
0
.2
.4
.6
.8
1
1.2
1.4
1.6
1.8
2
TIME IN SEC.

Fig. II. 2.4

Appendx II. System 3: Twin Moog 233 Valve

System Parameters

```
 system reference:ITERHRDA40 Closed loop system
 valve operated system without flow feedback
 4 way valve
 Double ended Cylinder
  4 bore x  2.8 rod dia. x  7 stroke (cm)
 inlet cylinder area= 6.408849 cm», outlet cylinder area= 6.408849 cm»
 mass = 15 kg
 coefficient of viscous damping =  0 kp/cm per sec
 load = 2 kN
 inlet/outlet orifice area ratio= 1
 equivalent valve flow rating= 65.37013 L/min
 actuator velocity= 170 cm/sec
 inlet flow = 65.37013 L/min
 supply pressure = 210 bar,  valve pressure drop= 173.9065 bar
 port diameter= .95 cm
 trapped volume= .2448619 L ( 4.486195E-02 :Actuator +  .2 :Piping)
 actuator shunt coefficient = 0 L/min per bar
 bulk modulus= 13793 bar
 WH0=125.01hz
 DF0=  0.45
 K0/K= 1

VALVE LIMIT SETTINGS:
  Valve datum flow rate= 77.02 L/min, Valve pressure drop @
  datum flow rate= 70 bar, Valve overlap= 0 %
Pressure Gain = 30 % of supply pressure, hysteresis = 3 % of command signa
  Model Code:2E760-233
```

Steady State Errors

```
STEADY-STATE ERRORS
SPECIAL FEATURES
VALVE PARAMETERS MODEL:2E760-233
Flow Gain= 1.213982 L/min per % of command signal
Pressure Gain= 63 bar per % of command signal
Hysteresis= 3 % of command signal units
Enter YES(Y) for mode update?
POSITION CONTROL
Load Error @  2 kN= .0760336 mm
Hysteresis Error= .460487 mm
Velocity Error @  170 cm/sec= 8.265385 mm
VELOCITY CONTROL
Load Error @  2 kN= .0760336 mm/sec
Hysteresis Error= .460487 mm/sec
Acceleration Error @  170 cm/sec»= 8.265385 mm/sec
```

Appendx II. System 3: Twin Moog 233 Valve

Open Loop system parameters

```
system reference:DX
Components selected:2E760-233
OPEN LOOP SYSTEM PARAMETERS
Frequencies in hz:
WH0=125.01
WH1=140.00
WH2=%100000.00
WH3=%100000.00
Damping factors:
DF0=  0.45
DF1=  0.90
DF2=  0.00
DF3=  0.00
Time constants in seconds:
t0= 0          t1= 0          t2= 0          t3= 0          t4= 0
free integrator:SINGLE
loop gain=205.72  1/sec
'C' to continue 'any key' for frequency domain access
```

Closed loop frequency response

```
system reference:DX          Closed loop system
Components selected:2E760-233
OPEN LOOP SYSTEM PARAMETERS
Frequencies in hz:
WH0=125.01
WH1=140.00
WH2=%100000.00
```

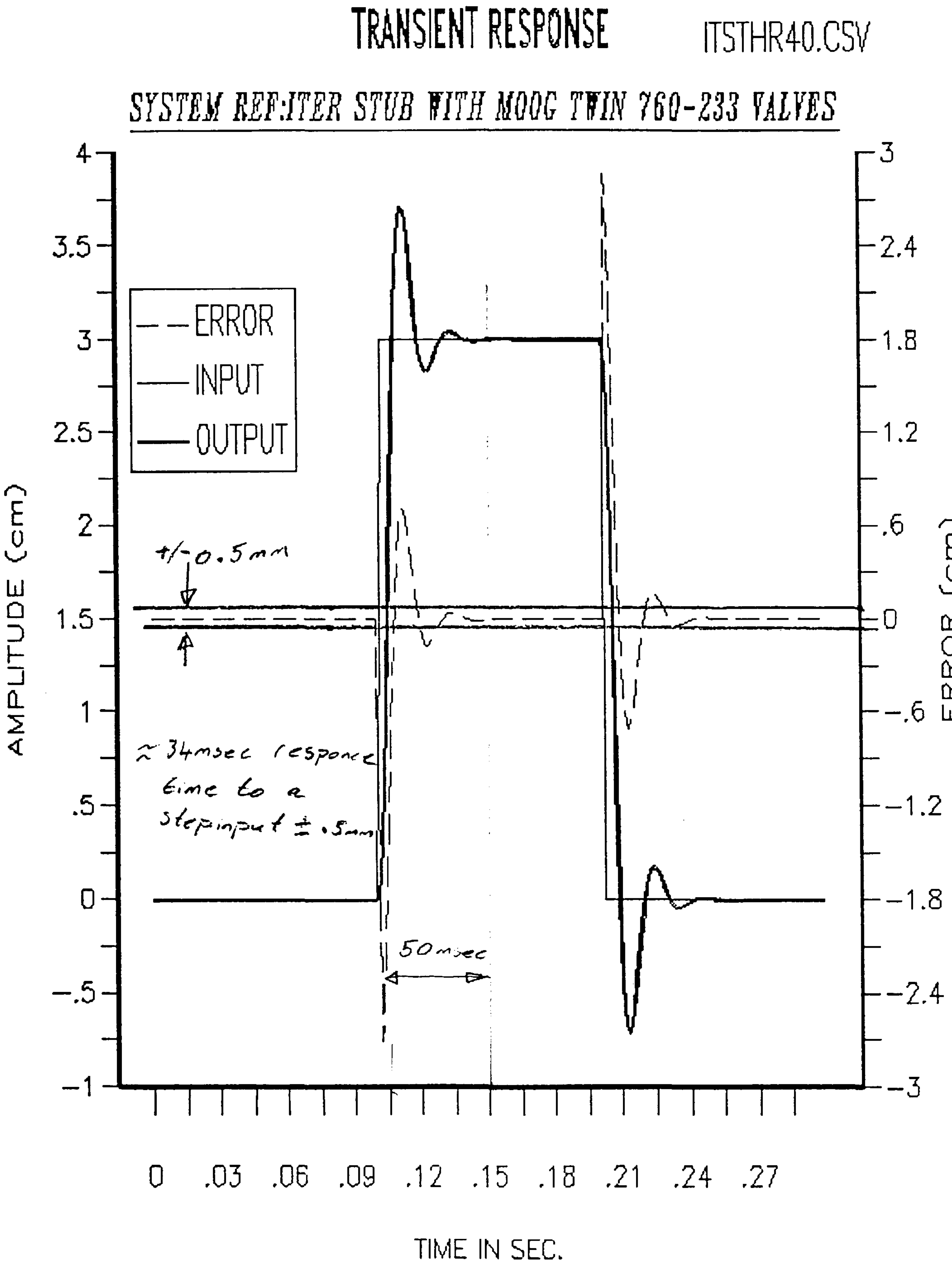

TRANSIENT RESPONSE
ITSTHR40.CSV
SYSTEM REF:ITER STUB WITH MOOG TWIN 760-233 VALVES
ERROR
INPUT
OUTPUT
AMPLITUDE (cm)
ERROR (cm)
4
3.5
3
2.5
2
1.5
1
.5
0
-.5
-1
3
2.4
1.8
1.2
.6
0
-.6
-1.2
-1.8
-2.4
-3
+/-0.5mm
≈ 34msec responce time to a stepinput ± .5mm
50msec
0
.03
.06
.09
.12
.15
.18
.21
.24
.27
TIME IN SEC.

Fig. II. 3.1

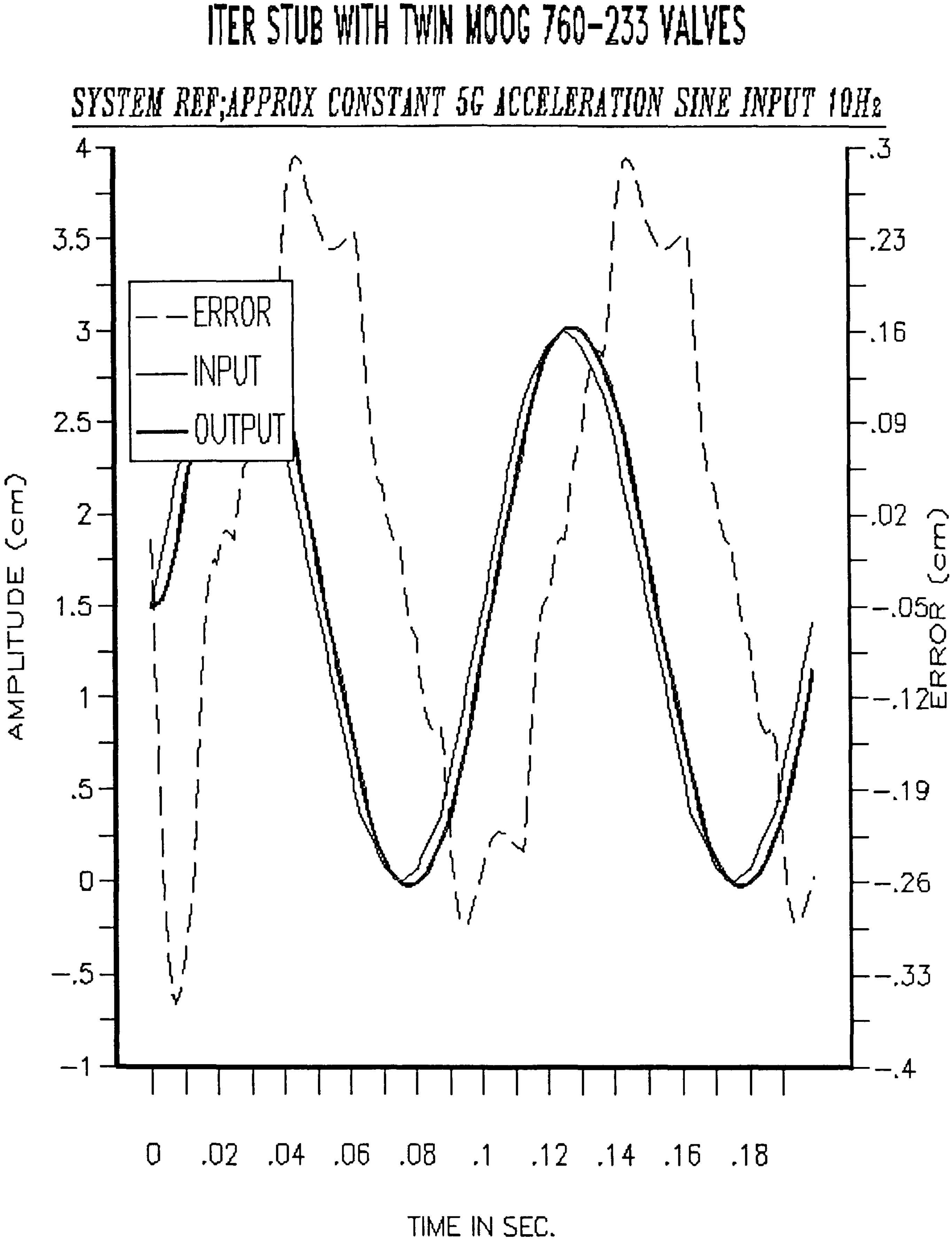

ITER STUB WITH TWIN MOOG 760-233 VALVES
SYSTEM REF;APPROX CONSTANT 5G ACCELERATION SINE INPUT 10Hz
ERROR
INPUT
OUTPUT
AMPLITUDE (cm)
ERROR (cm)
4
3.5
3
2.5
2
1.5
1
.5
0
-.5
-1
.3
.23
.16
.09
.02
-.05
-.12
-.19
-.26
-.33
-.4
0
.02
.04
.06
.08
.1
.12
.14
.16
.18
TIME IN SEC.

Fig. II. 3.2

Appendx II. System 3: Twin Moog 233 Valve

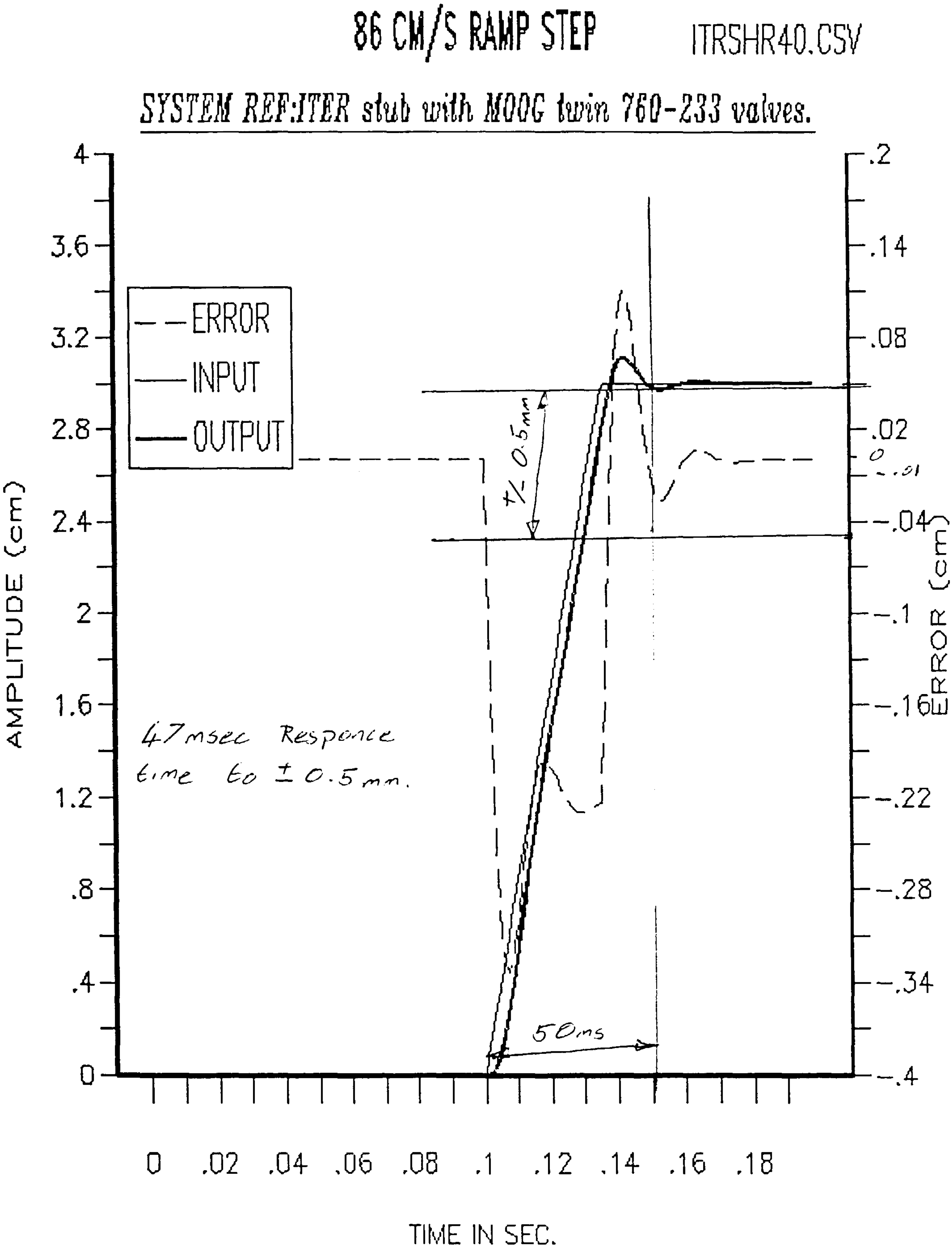


Fig. II. 3.3

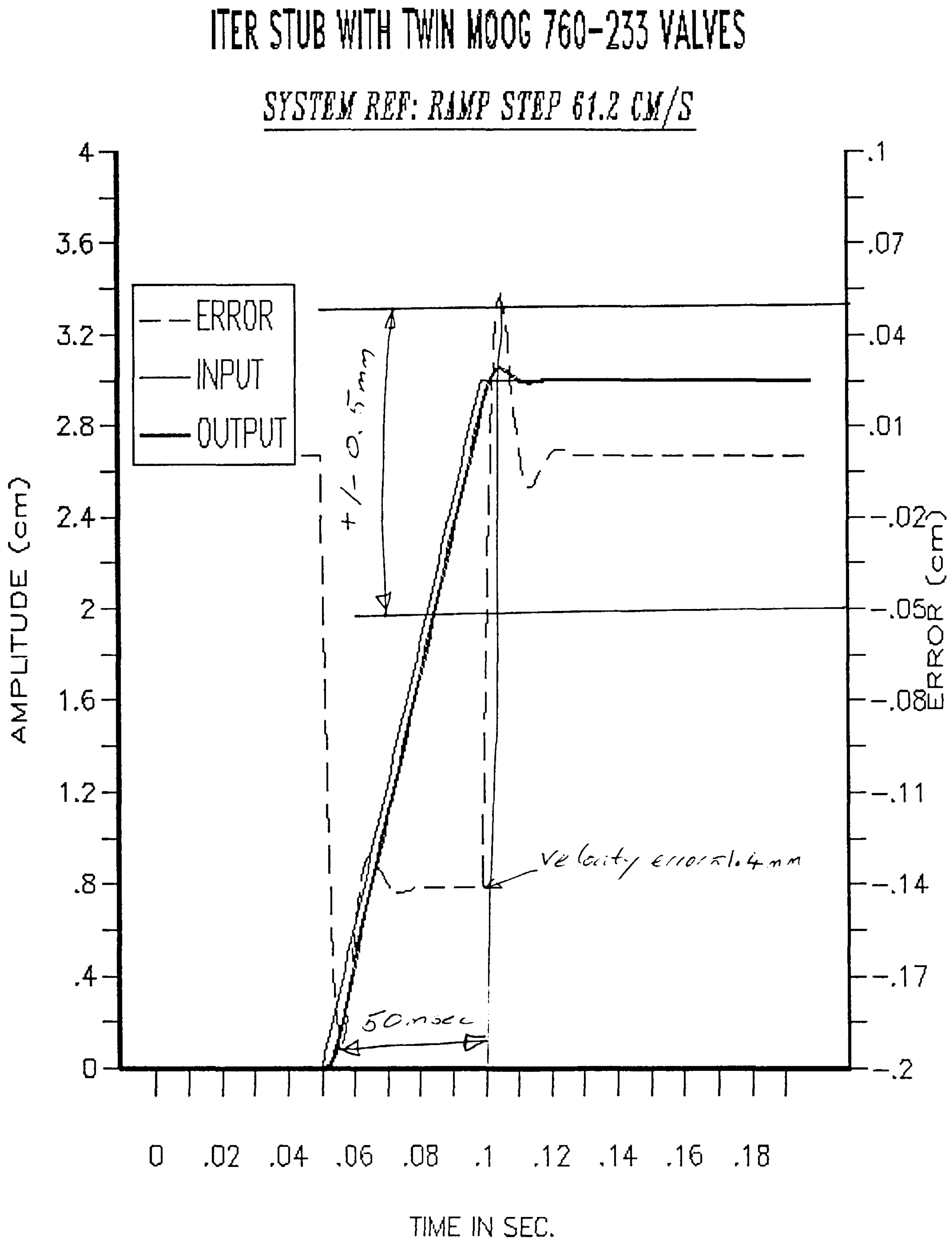

ITER STUB WITH TWIN MOOG 760-233 VALVES
SYSTEM REF: RAMP STEP 61.2 CM/S
ERROR
INPUT
OUTPUT
AMPLITUDE (cm)
4
3.6
3.2
2.8
2.4
2
1.6
1.2
.8
.4
0
ERROR (cm)
.1
.07
.04
.01
-.02
-.05
-.08
-.11
-.14
-.17
-.2
0
.02
.04
.06
.08
.1
.12
.14
.16
.18
TIME IN SEC.
+/- 0.5mm
velocity error 1.4mm
50msec

Fig. II. 3.4

MOOG TWIN 760-233 VALVE WITH 400 hZ LVDT, STEP RESPONCE.

SYSTEM REF:ITER STUB DRIVER WITH FEEDBACK TRANSDUCER.

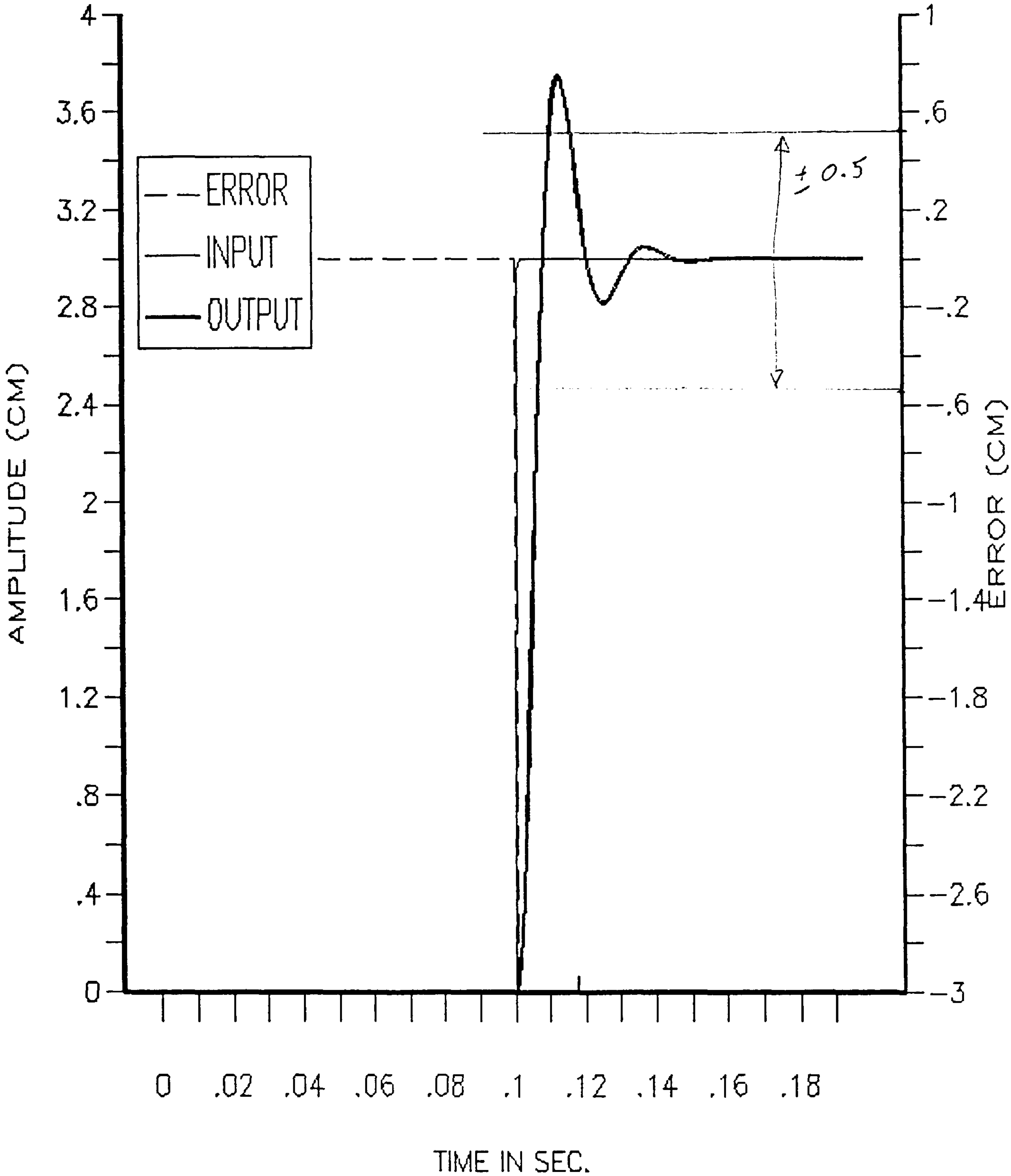


Fig. II. 3.5

Appendx II. System 3: Twin Moog 233 Valve +LVDT

System Parameters

```
system reference:LVDTHRDA40 Closed loop system
valve operated system without flow feedback
4 way valve
Double ended Cylinder
 4 bore x  2.8 rod dia. x  7 stroke (cm)
inlet cylinder area= 6.408849 cm», outlet cylinder area= 6.408849 cm»
mass = 15 kg
coefficient of viscous damping =  0 kp/cm per sec
load = 2 kN
inlet/outlet orifice area ratio= 1
equivalent valve flow rating= 65.37013 L/min
actuator velocity= 170 cm/sec
inlet flow = 65.37013 L/min
supply pressure = 210 bar,  valve pressure drop= 173.9065 bar
port diameter= .95 cm
trapped volume= .2448619 L ( 4.486195E-02 :Actuator +  .2 :Piping)
actuator shunt coefficient = 0 L/min per bar
bulk modulus= 13793 bar
WH0=125.01hz
DF0=  0.45
K0/K= 1

VALVE LIMIT SETTINGS:
  Valve datum flow rate= 77.02 L/min, Valve pressure drop @
  datum flow rate= 70 bar, Valve overlap= 0 %
Pressure Gain = 30 % of supply pressure, hysteresis = 3 % of command signa
  Model Code:2E760-233
```

Steady State Errors

```
STEADY-STATE ERRORS
SPECIAL FEATURES
VALVE PARAMETERS MODEL:2E760-233
Flow Gain= 1.213982 L/min per % of command signal
Pressure Gain= 63 bar per % of command signal
Hysteresis= 3 % of command signal units
Enter YES(Y) for mode update?
POSITION CONTROL
Load Error @  2 kN= 8.680607E-02 mm
Hysteresis Error= .525729 mm
Velocity Error @  170 cm/sec= 9.436428 mm
VELOCITY CONTROL
Load Error @  2 kN= 8.680607E-02 mm/sec
Hysteresis Error= .525729 mm/sec
Acceleration Error @  170 cm/sec»= 9.436428 mm/sec
To obtain Amplifier Gain, enter Feedback Transducer Gain in Volt/cm for posit
 control and Volt per cm/sec for velocity control? 1
Amplifier Gain= 57.06362 % of command signal units per volt
```

Appendx II. System 3: Twin Moog 233 Valve +LVDT

Open Loop system parameters

```
system reference:DX          Closed loop system
Components selected:2E760-233      LVDT
OPEN LOOP SYSTEM PARAMETERS
Frequencies in hz:
WH0=125.01
WH1=140.00
WH2=400.00
WH3=%100000.00
Damping factors:
DF0=  0.45
DF1=  0.90
DF2=  0.71
DF3=  0.00
Time constants in seconds:
t0= 0         t1= 0         t2= 0         t3= 0         t4= 0
free integrator:SINGLE
loop gain=180.19  1/sec
'C' to continue 'any key' for frequency domain access
```

Closed loop frequency response

```
Natural frequency WR= 44.12hz             CL Damping factor DFR=  0.40
Band width at 4 dB attenuation WC= 73.08hz
Max.Overshoot at  0.012sec = 24.94%
Step response time=  0.007sec
system reference:DX
```

Appendix II. System 4: Three Moog 912 Valves

System Parameters

```
system reference:ITERSHRDA40                Closed loop system
valve operated system without flow feedback
4 way valve
Double ended Cylinder
 4 bore x  2.8 rod dia. x  7 stroke (cm)
inlet cylinder area= 6.408849 cm», outlet cylinder area= 6.408849 cm»
mass = 15 kg
coefficient of viscous damping =  0 kp/cm per sec
load = 2 kN
inlet/outlet orifice area ratio= 1
equivalent valve flow rating= 65.37013 L/min
actuator velocity= 170 cm/sec
inlet flow = 65.37013 L/min
supply pressure = 210 bar,  valve pressure drop= 173.9065 bar
port diameter= .95 cm
trapped volume= .2448619 L ( 4.486195E-02 :Actuator +  .2 :Piping)
actuator shunt coefficient = 0 L/min per bar
bulk modulus= 13793 bar
WHO=125.01hz
DFO=  0.45
KO/K= 1

VALVE LIMIT SETTINGS:
  Valve datum flow rate= 57.6 L/min, Valve pressure drop @
  datum flow rate= 70 bar, Valve overlap= 0 %
Pressure Gain = 30 % of supply pressure, hysteresis = 3 % of command signa
  Model Code:3E760-912
```

Steady State Errors

```
STEADY-STATE ERRORS
SPECIAL FEATURES
VALVE PARAMETERS MODEL:3E760-912
Flow Gain= .9078861 L/min per % of command signal
Pressure Gain= 63 bar per % of command signal
Hysteresis= 3 % of command signal units
Enter YES(Y) for mode update?
POSITION CONTROL
Load Error @  2 kN= 5.029203E-02 mm
Hysteresis Error= .3045867 mm
Velocity Error @  170 cm/sec= 7.310342 mm
VELOCITY CONTROL
Load Error @  2 kN= 5.029203E-02 mm/sec
Hysteresis Error= .3045867 mm/sec
Acceleration Error @  170 cm/sec»= 7.310342 mm/sec
```

Appendix II. System 4: Three Moog 912 Valves

Open Loop system parameters

```
system reference:DX          Closed loop system
Components selected:3E760-912
OPEN LOOP SYSTEM PARAMETERS
Frequencies in hz:
WH0=125.01
WH1=280.00
WH2=%100000.00
WH3=%100000.00
Damping factors:
DF0=  0.45
DF1=  1.00
DF2=  0.00
DF3=  0.00
Time constants in seconds:
t0= 0         t1= 0         t2= 0         t3= 0         t4= 0
free integrator:SINGLE
loop gain=232.59  1/sec
'C' to continue 'any key' for frequency domain access
```

Closed loop frequency response

```
Natural frequency WR= 63.55hz            CL Damping factor DFR=  0.46
Band width at 4 dB attenuation WC=102.20hz
Max.Overshoot at  0.009sec = 19.41%
Step response time=  0.005sec
system reference:DX
```

Appendix II. System 4: Three Moog 912 Valves

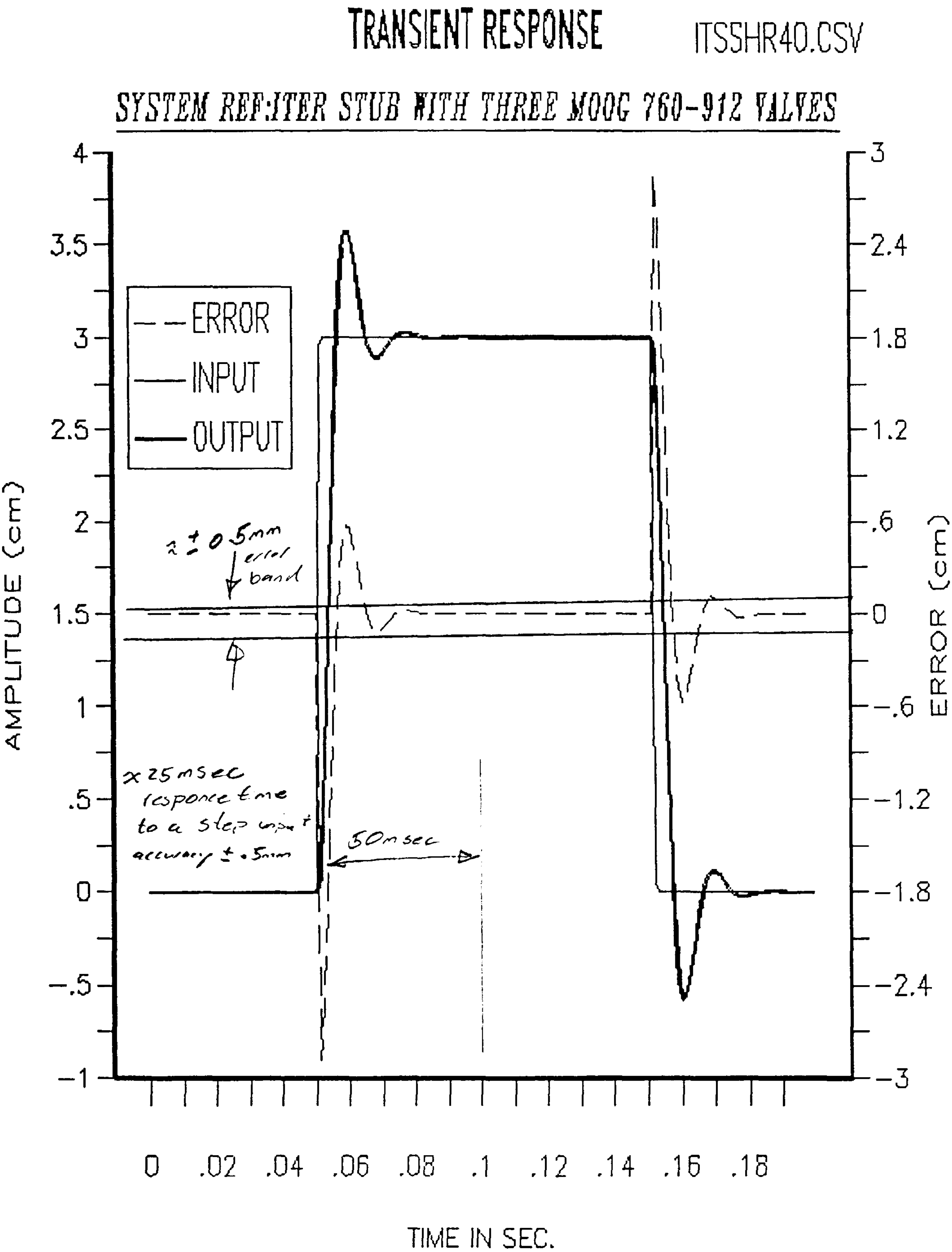


Fig. II. 4.1

173 cm/s saw tooth, 173 cm/s to step.ITRSHR40

SYSTEM REF:ITER STUB WITH 3 MOOG 760-912 VALVES

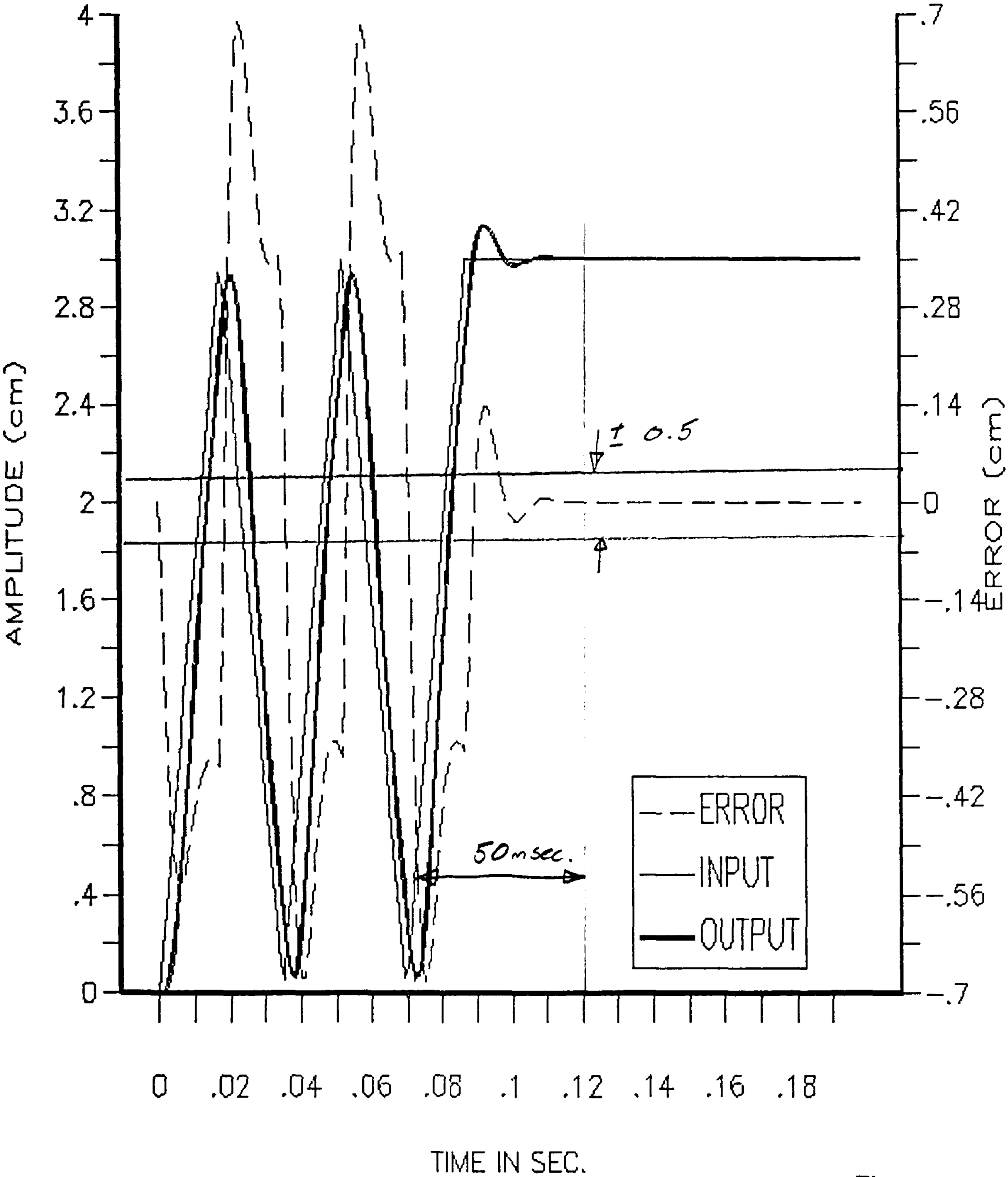


Fig. II. 4.2

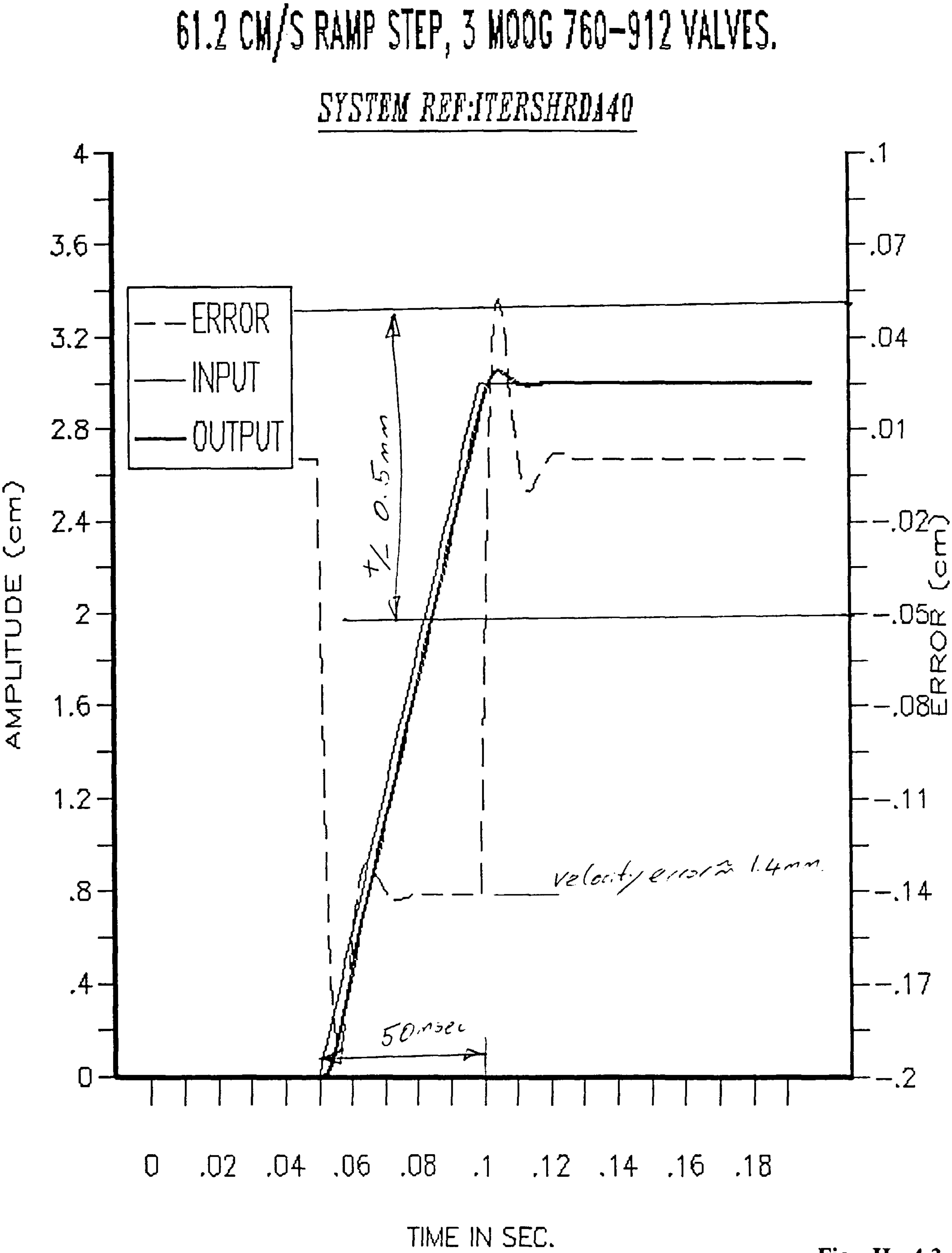

61.2 CM/S RAMP STEP, 3 MOOG 760-912 VALVES.
SYSTEM REF:ITERSHRDA40
ERROR
INPUT
OUTPUT
AMPLITUDE (cm)
ERROR (cm)
4
3.6
3.2
2.8
2.4
2
1.6
1.2
.8
.4
0
.1
.07
.04
.01
-.02
-.05
-.08
-.11
-.14
-.17
-.2
+/- 0.5mm
velocity error ≈ 1.4mm
50msec
0
.02
.04
.06
.08
.1
.12
.14
.16
.18
TIME IN SEC.

Fig. II. 4.3

Appendix II. System 5: Three Moog 912 Valves+LVDT
Large cylinder for maximum hydraulic stiffness and response

System Parameters

```
system reference:OPTIMUM    Closed loop system
valve operated system without flow feedback
4 way valve
Double ended Cylinder
 6 bore x  2.8 rod dia. x  7 stroke (cm)
inlet cylinder area= 22.11681 cm», outlet cylinder area= 22.11681 cm»
mass = 15 kg
coefficient of viscous damping =  0 kp/cm per sec
load = 2 kN
inlet/outlet orifice area ratio= 1
equivalent valve flow rating= 82.27438 L/min
actuator velocity= 62.00001 cm/sec
inlet flow = 82.27438 L/min
supply pressure = 210 bar,  valve pressure drop= 193.2164 bar
port diameter= .95 cm
trapped volume= .2048177 L ( .1548177 :Actuator +  .05 :Piping)
actuator shunt coefficient = 0 L/min per bar
bulk modulus= 13793 bar
WH0=471.71hz
DF0=  0.20
K0/K= 1
- - - -

VALVE LIMIT SETTINGS:
  Valve datum flow rate= 57.6 L/min, Valve pressure drop @
  datum flow rate= 70 bar, Valve overlap=-3 %
Pressure Gain = 30 % of supply pressure, hysteresis = 2 % of command signa
  Model Code:3E760-912
```

Steady State Errors

```
STEADY-STATE ERRORS
SPECIAL FEATURES
VALVE PARAMETERS MODEL:3E760-912
Flow Gain= .9290909 L/min per % of command signal
Pressure Gain= 63 bar per % of command signal
Hysteresis= 2 % of command signal units
Enter YES(Y) for mode update?
POSITION CONTROL
Load Error @  2 kN= 2.272229E-03 mm
Hysteresis Error= 3.166031E-02 mm
Velocity Error @  62.00001 cm/sec= 1.401818 mm
VELOCITY CONTROL
Load Error @  2 kN= 2.272229E-03 mm/sec
Hysteresis Error= 3.166031E-02 mm/sec
Acceleration Error @  62.00001 cm/sec»= 1.401818 mm/sec
To obtain Amplifier Gain, enter Feedback Transducer Gain in Volt/cm for positi
 control and Volt per cm/sec for velocity control? 1
Amplifier Gain= 631.7057 % of command signal units per volt
```

Appendix II. System 5: Three Moog 912 Valves+LVDT
Large cylinder for maximum hydraulic stiffness and response

Open Loop system parameters

```
Components selected:3E760-912     LVDT
OPEN LOOP SYSTEM PARAMETERS
Frequencies in hz:
WH0=471.71
WH1=280.00
WH2=400.00
WH3=%100000.00
Damping factors:
DF0=  0.20
DF1=  1.00
DF2=  0.71
DF3=  0.00
Time constants in seconds:
t0= 0         t1= 0         t2= 0         t3= 0         t4= 0
free integrator:SINGLE
loop gain=442.37  1/sec
```

Closed loop frequency response

```
loop gain determined by phase margin= 442.3705 1/sec     52.91572 dB
Natural frequency WR= 92.73hz            CL Damping factor DFR=  0.37
Band width at 4 dB attenuation WC=151.76hz
Max.Overshoot at  0.006sec = 29.13%
Step response time=  0.003sec
system reference:DX
```

Appendix II. System 5: Three Moog 912 Valves+LVDT
Large cylinder for maximum hydraulic stiffness and response

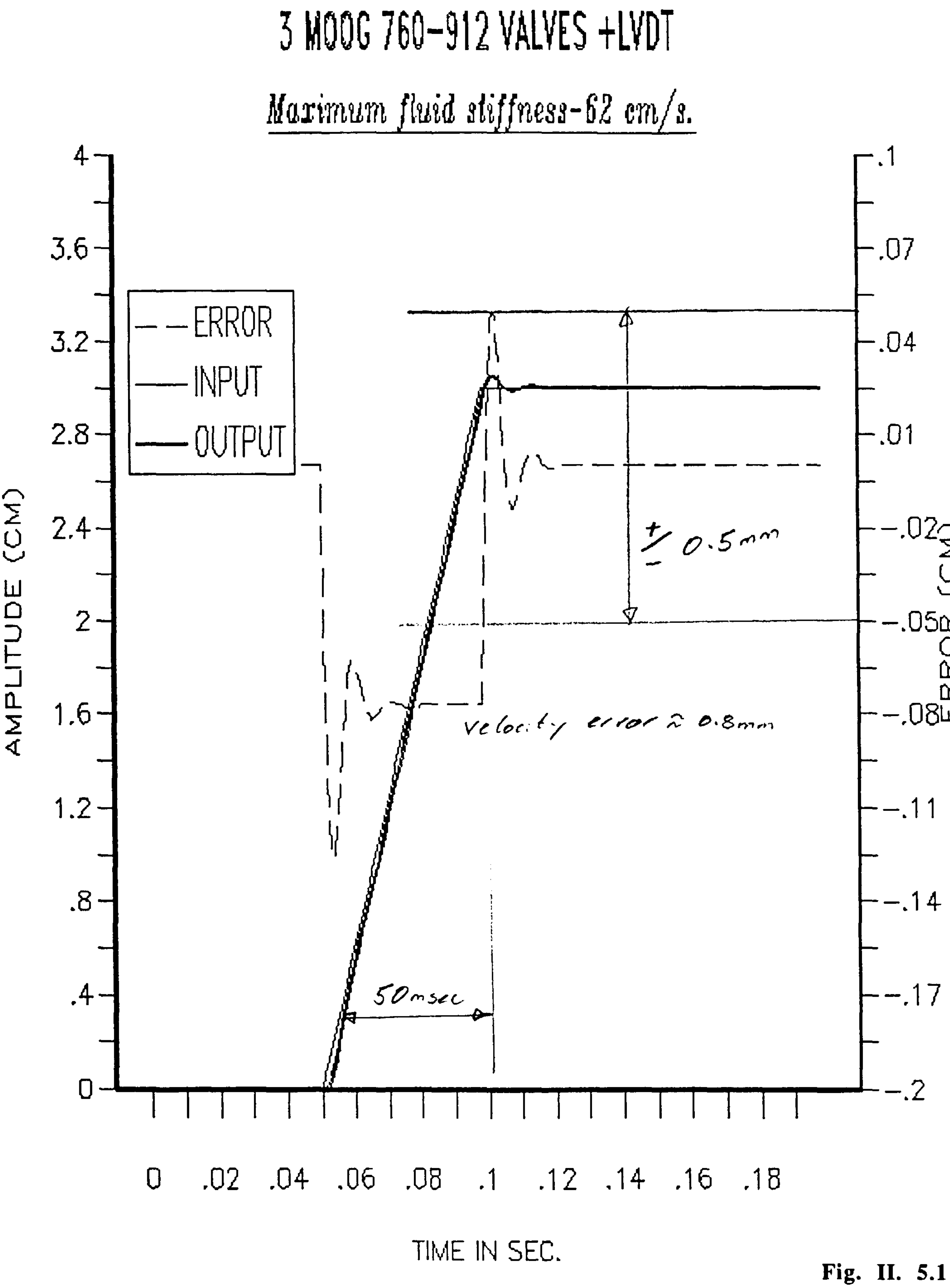


Fig. II. 5.1

Appendix II. System 5: Three Moog 912 Valves+LVDT
Large cylinder for maximum hydraulic stiffness and response

3 MOOG 760-912 VALVES+LVDT

Maximum fluid stiffness

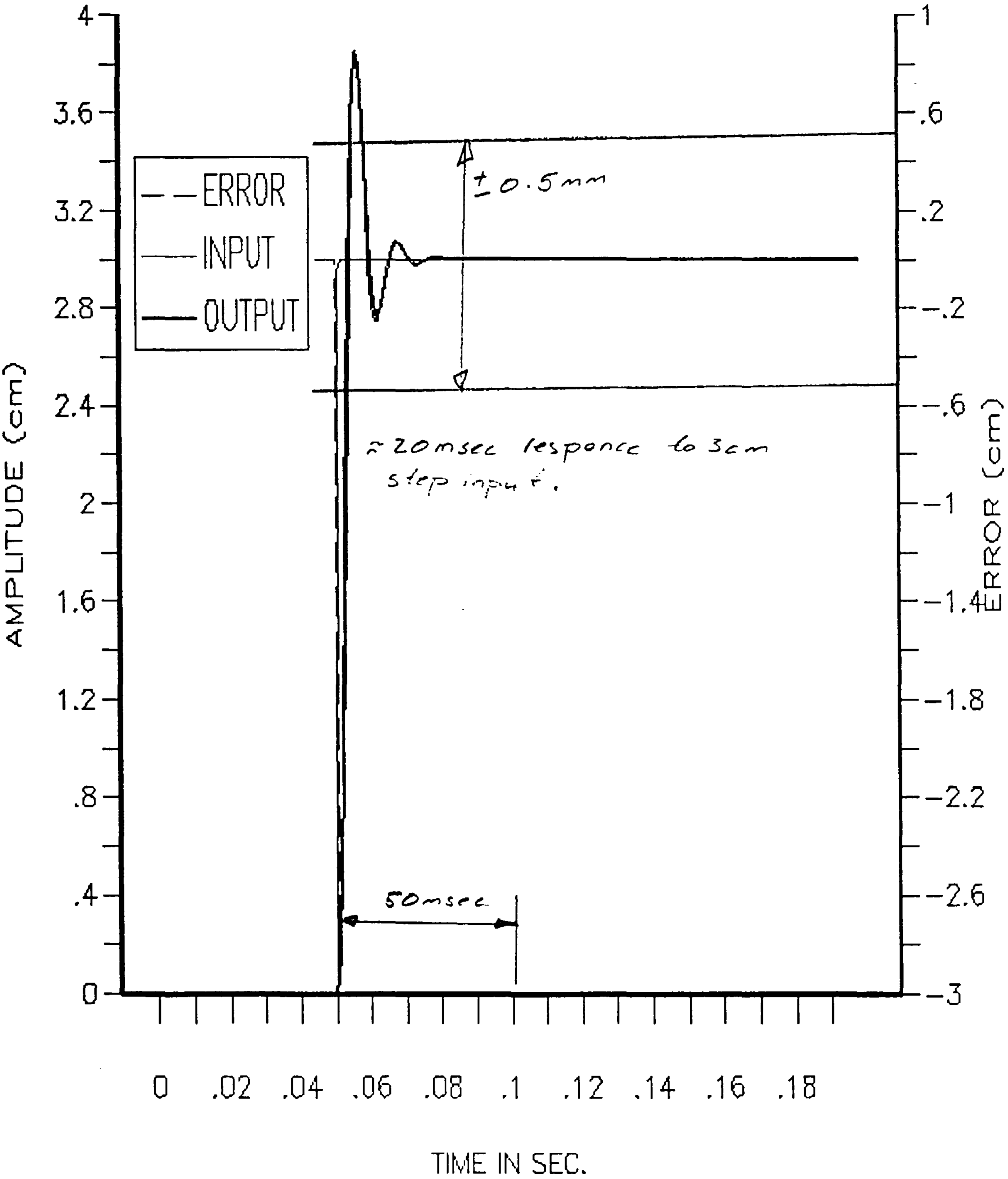


Fig. II. 5.2