IMPROVEMENTS IN SERVO HYDRAULIC ACTUATORS FOR SEISMIC SHAKE TABLES

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ABSTRACT

Seismic shake table are used to investigate the exact effects of earth quake loading on specimen. This type of testing eliminates the uncertainty in numerical predictions of complex stress and strain in any part of specimen subjected to earth quake time histories in three orthogonal directions. Shake tables are becoming an an important tool for conducting experimental research in the areas of structural identification, assessment, and control. The purpose of this paper is to improve the tracking characteristics of a shake table by improving the bandwidth of servo hydraulic actuator that drives the shake table. A three stage approach was adopted for the development of high performance electro hydraulic servo linear actuators. Firstly by use of frictionless hydraulic linear actuators, secondly by incorporating advance controllers and finally by improvements in hydraulic system. The developed actuators have a resolution of 5micron and flat response up to 50Hz in bode plot thus capable of high fidelity reproduction of earthquake time histories.

INTRODUCTION

Nuclear power reactors and its various systems are designed to withstand many operational loads and environmental conditions over decades of safe and economic use. One of the environmental conditions is earthquakes. Although the present day numerical simulation techniques can predict the behavior of various structures and components subjected to earth quakes, the complex nature of material and variety of potential failure modes, restricts the reliability of numerical simulation. Regulatory authorities therefore enforce tests on real structures and components to validate structural integrity.

Using shake table one can investigate the exact effects of earth quake loading on specimen in all three degrees of freedom. This type of testing eliminates the uncertainty in numerical predictions of complex stress and strain in any part of specimen subjected to earth quake. Hence shake tables have become an important tool for conducting experimental research in the areas of structural identification, assessment, and control.

Fluid power is used in a diverse range of applications and offers many advantages over other types of motive force. With fluid power systems, a single source of fluid pressure can power many axes or fluid power devices, much smaller in size and weight nevertheless can develop high forces an torque, often quieter and cooler, a big advantage in cases of stalling were electric motors will over heat and fail, very smooth and efficient for linear movement and has no backlash problems. Thus hydraulic actuators are best suited for shake table application.

To improve the tracking characteristics of a shake table, the bandwidth of servo hydraulic actuator that drives the shake table had to be improved. Conventional hydraulic actuators have bandwidth less than 10Hz. A three stage approach was adopted for the development of high performance electro hydraulic servo linear actuators. Firstly by use of frictionless hydraulic linear actuators, secondly by incorporating advance controllers and finally by improvements in hydraulic system

FRICTIONLESS HYDRAULIC SERVO ACTUATOR (FHLA):

The actual output produced by any motion generating device is usually defined by the instantaneous force and the rate of change of forces. In the case of a hydraulic system, these parameters correspond to the pressure and rate of change of pressure of hydraulic fluid. Accurately and smoothly controlled output force of the motion device will result in equally accurate and smooth control of acceleration, provided that the output force is used solely for the production of acceleration and only a small proportion for overcoming friction. Of the forms of friction, ‘stiction’, or the non-linear force present at the motion reversal point, is of far greater importance. Coulomb friction in conventional hydraulic actuators easily reaches from 5% to 15% of the maximum available load capacity. Friction in hydraulic actuators not only reduce force generated (and power loss) but also reduces the control accuracy and bandwidth of hydraulic actuators.

A literature survey was done to find the various hydrostatic bearings available. Three types of hydrostatic bearing are generally being used.

Mahajm [1] has investigated both theoretically and experimentally the phenomenon of hydraulic locks in conical hydrostatic bearings. Hydraulic lock is sticking or locking of pistons in its enclosing cylinder due to an axial pressure gradient in the fluid contained in the clearance space surrounding the piston lands. If there are machining imperfections in the piston lands or cylinder bore, the flow of fluid in the clearance space is diverging or converging; an asymmetric distribution of pressure will result and will be accompanied by a transverse thrust on the piston. If the clearance is divergent (Fig. 1a) in the direction of flow, the piston will be forced against the cylinder wall (hydraulic lock); if convergent (Fig. 1b), the action of any thrust arising will be to exert a centralizing affect. The author has experimentally verified for both the above cases. For a case of flow in diverging clearance, transverse thrust {locking force/coefficient of friction (0.24)} of 1080N for d=54.6mm, l=42.9, c=0.023mm, t=0.046mm was observed for differential pressure of 3MPa. Mahajm [1] has investigated only simple conditions of hydrostatic bearing in the mathematical analysis.

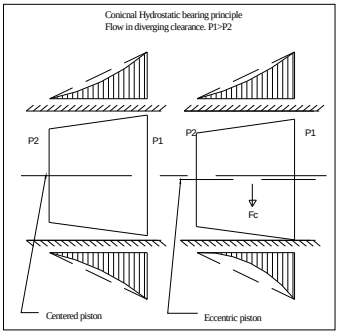

Fig. 1a: Flow in diverging clearance

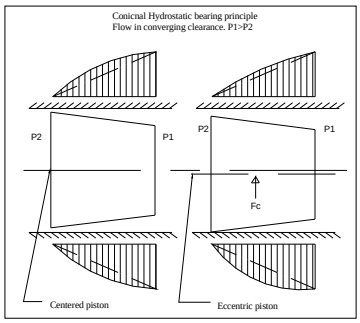

Fig. 1b: Flow in converging clearance

In the discussions of [1], a special hydrostatic bearing as shown in the Fig. 1c was investigated. The net centralizing force produced in this kind of bearing is less than the conical counter part because the centralizing forces are produced only at two discrete parts. Where as in conical hydrostatic bearings the centralizing forces are produced throughout the bottom half of the piston.

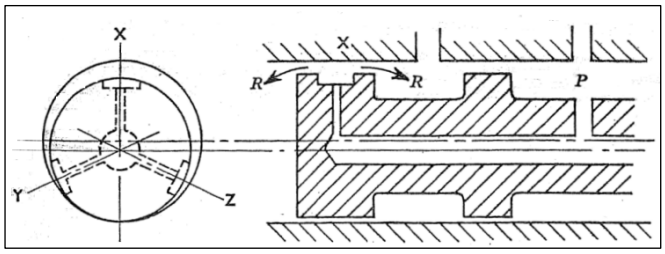

Fig. 1c: Special hydrostatic bearing

Elliot [2] has considered step piston for hydrostatic bearings as shown in Fig. 1d. The advantage in using step piston is that it is easy to manufacture and the simple pressure distribution, Fig. 1e, around the piston makes mathematical analysis easy. But the pressure gradient distribution around the piston is less, hence giving less centering force when compared to conical hydrostatic bearings. Since our objective is to design hydraulic linear actuators that withstand maximum lateral forces, conical hydrostatic bearing was selected as the type of bearing in hydraulic linear actuator. Also the effect of velocity of piston on the centering force has not been considered by Elliot [2]. A stepped bearing is more sensitive for clearance variation and occurring clearance differ considerably from the calculated values due to manufacturing tolerances.

Viersma [3] has investigated conical hydrostatic bearing with all practical conditions considered in the mathematical analysis. Even the effect of velocity on the centering force has been considered. So this was selected for hydrostatic bearing in high performance electro hydraulic servo linear actuators. The two limitations of single hydrostatic bearings when used in hydraulic cylinders are:

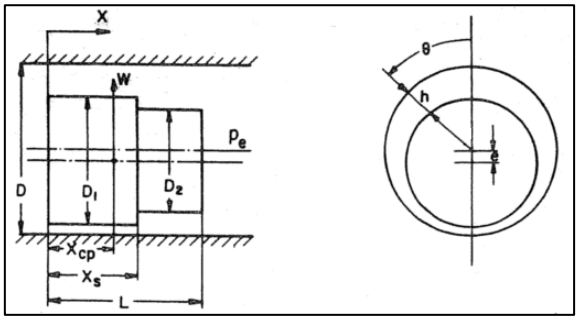

Fig. 1d: Stepped hydrostatic bearings

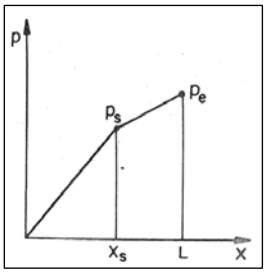

Fig. 1e: Pressure variation over length

1) The bearing provides centering force only when pressure at the wide end of gap is greater than the pressure at the narrow end of the gap. Otherwise, decentering forces will occur. Hence it is not possible to use single hydrostatic bearings in symmetric hydraulic cylinders.
2) Velocity of piston directly influences the maximum lateral load carrying capacity of hydraulic cylinder. Therefore requiring a certain lateral load carrying capacity in hydraulic cylinder limits the maximum allowable velocity.

Both the above limitations can be overcome by double bearing which consists of two identical single bearing with opposite tapers, installed side by side. A double conical hydrostatic bearing [3] has been used to completely eliminate this coulomb friction and hence improve the performance of hydraulic actuator. Two ton frictionless hydraulic linear actuator was designed for operating pressure of 20MPa, maximum lateral force = 2500N, stroke-200mm, bore-50mm and piston rod-35mm. The taper and length of the piston and rod end bearings was optimized to develop maximum centering force.

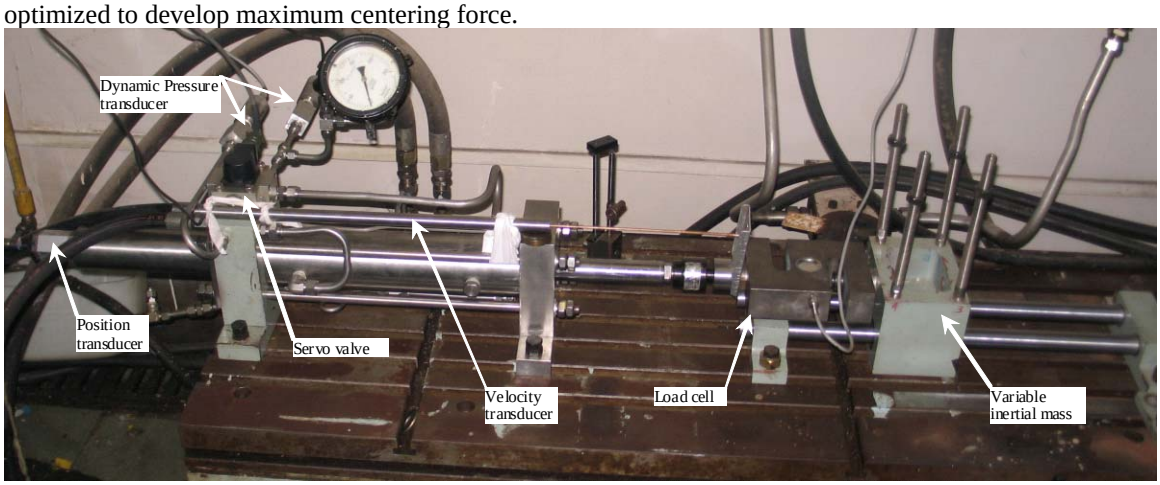

Fig. 2: Experimental test facility for frictionless hydraulic linear actuator.

FHLA was fabrication with strict inspection on dimensions and assembled in clean environment. The frictionless condition was achieved and verified by experiments in the test facility [4] as shown in Fig. 2. Experiment [4] was conducted to measure the stiffness in the floating condition.

Also seen in the Fig. 2 are sensors to measure three states (displacement of piston, velocity of piston & dynamic differential pressure across piston in hydraulic cylinder) of hydraulic actuator, servo valve and inertial mass loading system mounted co-axially with hydraulic actuator on a T-slot table. Since no elastomeric seal was used, excellent dynamic response was achieved.

ADVANCE CONTROL ALGORITHMS

Many industrial controllers for hydraulic servo system achieve high bandwidth with fixed gain control laws by over-sizing the cylinder diameter in order to increase the effective stiffness of the fluid in cylinder. This requires larger and more costly components and higher fluid-flow rates in order to move a load at given speed. A better approach to obtaining a fast response is to model the dominant dynamics of the system, and then to use this knowledge of the dynamics in the controller design. The advantage of such an approach is that, to achieve a given bandwidth, the mechanical components are smaller, the required flow rates are less and overall system is therefore much less expensive.

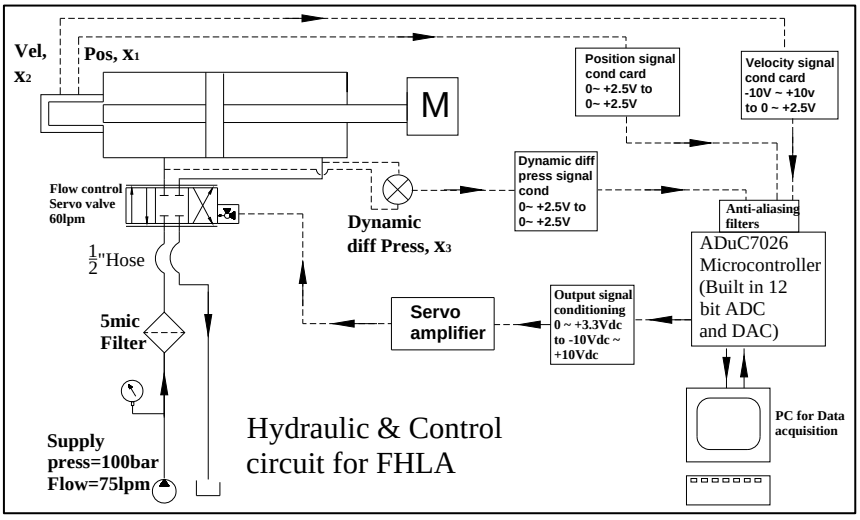

Fig. 3: Schematic of hydraulic and control used for experiment

State feedback controller

Linear hydraulic actuators having fast response with precise motion control are required in applications like seismic shake table and material testing. With the design objective of improved control bandwidth, velocity feedback has been used in simple proportional position control. An experimental test set up as shown in Fig. 3 was built. Mathematical model of each component of hydraulic system was developed; starting from the first principles [5] and for some components, empirical values from the data sheet provided by the manufacturer was used.
A fifth order linear state space model of servo hydraulic linear actuator was derived and validated with experiments.

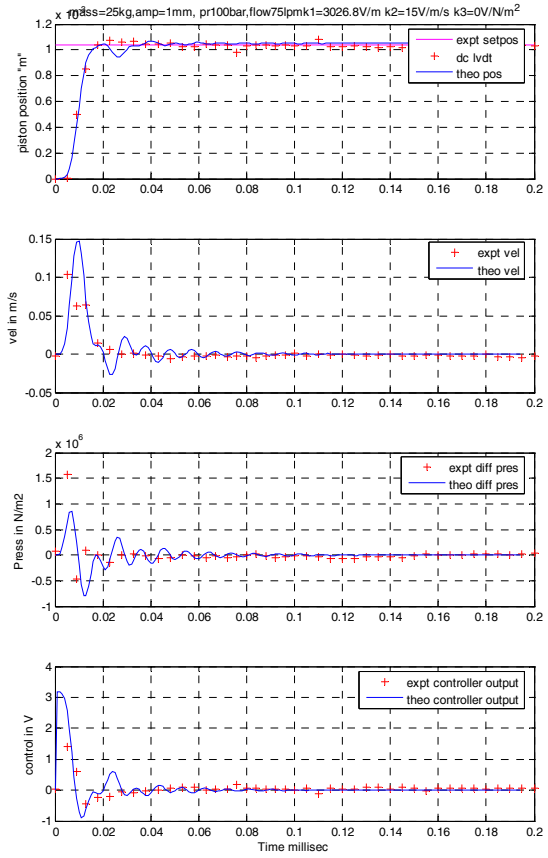

Fig. 4: Variation of states for velocity feedback in proportional control

For a step response (Fig.4) a close match obtained between theoretical prediction (continuous blue line) and experimental results (red plus sign) can be clearly seen for displacement, velocity, dynamic differential pressure and control output of servo hydraulic actuator.

Design and analysis of various controllers was studied using Eigen value map. With increase in proportional position gain, the complex poles due to valve dynamics moves closer to the imaginary axis and reduces the damping. The increase in proportional gain results in overshoot in time response for step input and introduces peaking in frequency response which is undesirable. The increase in gain has no effect on complex poles due to hydraulic linear actuator dynamics. Further improvement in performance cannot be achieved by simply increasing the proportional gain.

Using velocity feed back in proportional position control, we can control the movement of the complex poles due to the servo valve. The step response Fig.4 of proportional position control with velocity feedback was simulated in MATLAB in discrete domain. All the simulation has been supported by experimental results. The servo hydraulic actuator developed has a position resolution of 4micron, rise time 15millisec and bandwidth 40Hz.

Pole placement compensator

Further improvement in the bandwidth was attained by use of pole placement compensator. The design of compensator was carried out in SISOTOOL of MATLAB. The undesired open loop poles can be compensated by adding additional zeros and then shape the closed loop root locus to get the wanted closed loop behavior. To further improve the bandwidth with no peaking, we have used a compensator designed by pole placement technique. To get high bandwidth, the dominant closed loop poles cannot be moved very far away from imaginary axis as this requires large control values.

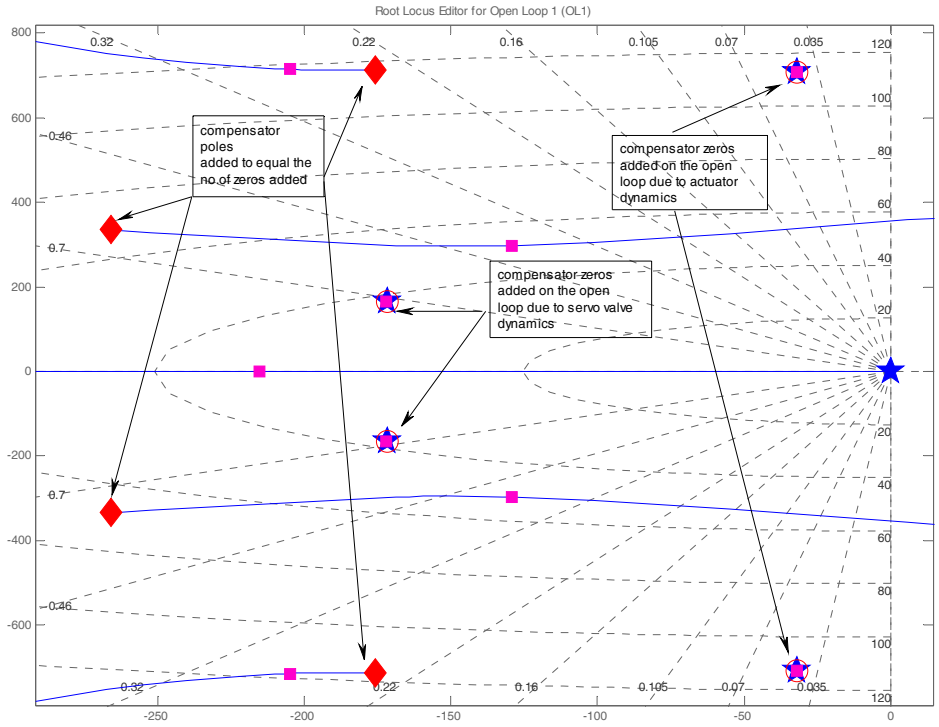

Fig. 5: Compensator design by pole placement analysis

The undesirable effects of poles due to servo valve and actuator were compensated by placing two pairs of complex zeros exactly on the locations of undesirable complex poles, Fig.5. Since compensator transfer function with only complex zeros cannot be realized, two complex poles must be added, Fig.5. The placement of poles and zeros was optimized in the root locus map to get the maximum band width with no peaking and with limited control currents to servo valve. The best optimized location of the pole and zeros of compensators are as shown in the Fig.6.

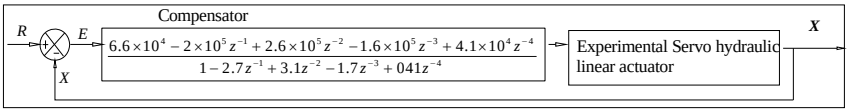

Fig. 6: Discrete pole placement Compensator

So with this limited control current available to servo valve the compensator was optimized. The optimized compensator was discretised, Fig.5 and loaded in digital controller. Comparison of pole placement compensator was made with optimized proportional controller, gain Kp= 19536 in step responses and bode plots.

A step response theoretically contains infinite range of frequencies hence a system subjected to step response is excited by all the frequencies. In the step response, Fig.6a the contribution from servo valve dynamics and actuator dynamics results in over shoot of 24% . The rise time is 7millisecs and settling time of 52millisecs.The -3dB cut off in bode plot, Fig.7 is occurring at 44Hz in the case of proportional controller and with +2.8dB peaking. The bandwidth of this proportional controller cannot be further improved by increasing gain as this would result in more peaking in bode and large overshoot in step response.

Where as the -3dB cut off in bode plot is occurring at 56Hz in the case of pole placement compensator and with no peaking in bode plot. We are able to use high gain (66804) due to increased damping in the system and thus obtain high bandwidth with no peaking as observed in simple proportional controller. The rise time is 6millisecs and settling time of 29millisecs as seen in the step response, *Fig(4c)* . There is significant improvement in settling time

The improvement of its performance (-3dB at 56Hz) was experimentally demonstrated. In the bode plot Fig.7 peaking observed in proportional controller is due to undesired poles of servo valve. The improved performance can be clearly seen in bode plot of pole placement compensator where the reduced damping due to undesired poles of servo valve have been compensated.

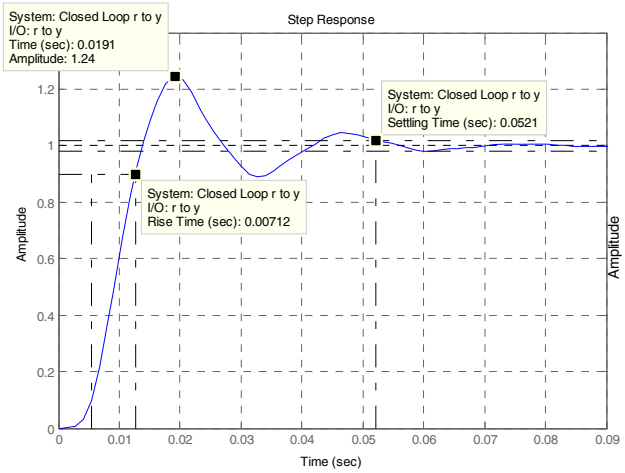

Fig. 6a: Step response of proportional controller

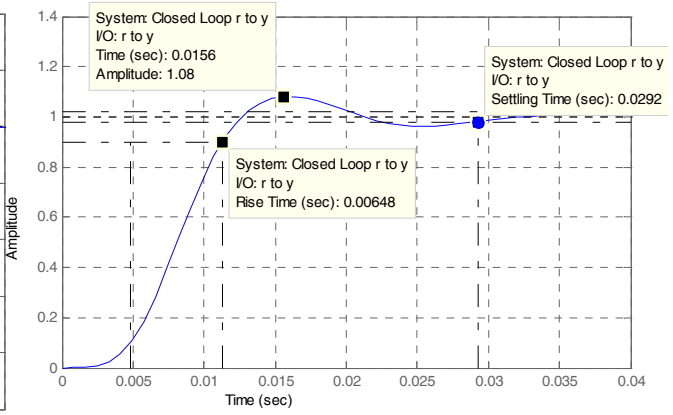

Fig. 6b: Step response for pole placement compensator

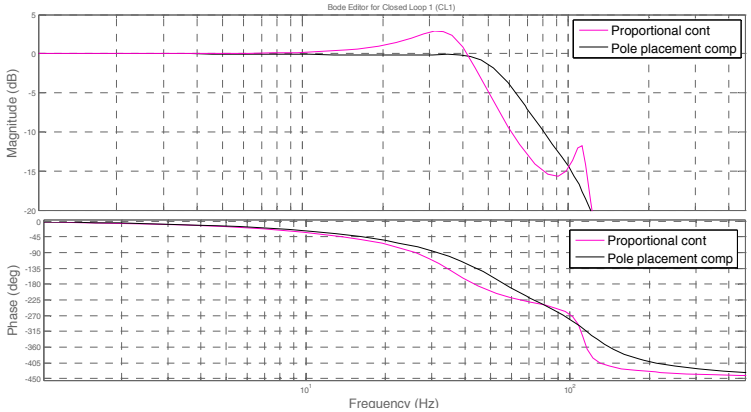

Fig. 7: Bode plot analysis of proportional controller & pole placement compensator

Indigenous Digital Controller

In house developed servo-controller card (Fig.8) 42MHz, 32 bit ARM processor with RS485, SPI, I2C, RS232 Interface with MODBUS support and 12 Bit 1MSPS ADC & DAC, EEPROM 62KB and digital controller currently running at 1KHz was used. Advanced control algorithm was developed in-house and programmed into these controllers.

The design of control cards has gone through various stages of design, development and upgradation. In first stage 8bit controllers were used. In second stage, to improve speed and algorithm efficiency, 32 bit ARM controllers were adopted. Also advanced communication buses such as USB, CANBUS were incorporated. For the third stage we are developing expertise in DSP processors which would be used in applications requiring high end data computations. Currently six- axis floating point DSP based multi-axis controller is under the final stage of development. This is aimed to solve many kinematic equations in real time required for the robotic and shake table applications. Fast communication buses and other advanced features of this controller are helpful in achieving real-time requirements of the application.

Fig.8: Indigenous servo controller

These controllers have features of anti-aliasing filter, galvanic isolation and input amplifiers. Provision for optical isolation of digital inputs has been provided. Digital filtering of all inputs makes the control more stable. Servo valve drives using push pull amplifiers have been used and in applications were power saving is required, PWM scheme is being used. These indigenous controllers have drastically reduced the cost and provide faster data handling and control capabilities.

HYDRAULIC SYSTEM IMPROVEMENTS:

It is very important to supply clean, cool and pressurized hydraulic fluid to the EHSLA. This is provided by hydraulic power supply system, Fig.9. To achieve and maintain the above performance of EHSLA, it is required to maintain cleanliness (NAS5) of hydraulic oil (ENKLO-68) and increase the bulk modulus of fluid.

Fig. 9: Hydraulic power supply system, Electrostatic oil cleaner, Vacuum dehydration unit

In addition to use of 3micron mechanical filter in hydraulic power supply system, electrostatic oil cleaner and magnetic filter were used. Mechanical filter are effective in removing particles greater than 5microns whereas electrostatic oil cleaner (Fig.9) is effective in removing particle lesser than 5microns. Ferrous and other magnetic

Fig. 10: Particle counter and ferrography

particles are removed by magnetic filter. To monitor the actual cleanliness achieved by the above filtration system, automatic particle counters (Fig.10) was used and ferrography (Fig.10) was used for qualitative analysis of hydraulic system.

Air present in hydraulic oil reduces the effective bulk modulus which directly influences the natural frequency of EHSLA. Effective bulk modulus of hydraulic oil was improved by removing air from oil by degassing using vacuum dehydration unit (Fig.9) specially developed for this purpose in FPL. The percentage of air in oil was reduced from 1.2% to 0.4% and thus improving the effective bulk modulus from $4.68x10^8 N/m^2$ to $8.54x10^8 N/m^2$ and the percentage of water in oil decreased from 63% to 30% of saturation level.

CONCLUSION

Indigenous electro hydraulic servo linear actuators for demanding requirement of seismic shake table having good precision and high bandwidth requirements has been developed by incorporating many improvements in conventional hydraulic actuators. Dominant dynamics of the EHSLA was modelled, advance control algorithms were designed and implemented in indigenous servo controller cards. Performance of these actuators was experimentally evaluated and compared with simple proportional controller.

Digital compensator have been developed that overcomes the limitations like the maximum gain limit in proportional controllers, low bandwidth of commercially available servo valves, limited control current to servo valve. And still obtain high band width with no peaking. The bandwidth of the electro hydraulic servo linear actuators has been improved to 56Hz by the use of advanced compensator. In this process, better understanding of the controller dynamics was gained both in simulation and in experimental investigations. These actuators can be confidently used for shake table applications.

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