Application Report
SPRAA76 – January 2005

DSP Control of Electro-Hydraulic Servo Actuators

Richard Poley
DSP Field Applications

ABSTRACT

Hydraulic actuators are characterized by their ability to impart large forces at high speeds and are used in many industrial motion systems. In applications where good dynamic performance is important it is common to contain the actuator in a servo loop comprising a feedback transducer and electronic controller. The majority of electronic servo-controllers used in these systems are analogue based implementations of the well-known PID type. However, the requirement to implement advanced control strategies has led to an increased interest in the use of digital signal processors (DSPs) in this field. One design approach which merits special consideration is the use of computer simulation software to model the hydraulic plant and electronic servo-controller, and to generate and test embedded code for the target DSP. This application report discusses some of the issues involved in controlling linear hydraulic actuators, and the suitability of the TMS320C28x™ DSP for such systems.

Contents

List of Figures

List of Tables

1 Introduction

The range of applications for electro-hydraulic servo systems is diverse, and includes manufacturing systems, materials test machines, active suspension systems, mining machinery, fatigue testing, flight simulation, paper machines, ships and electromagnetic marine engineering, injection moulding machines, robotics, and steel and aluminium mill equipment. Hydraulic systems are also common in aircraft, where their high power-to-weight ratio and precise control makes them an ideal choice for actuation of flight surfaces.

Although electrical motors are sometimes used in many of these applications, motion control systems requiring either very high force or wide bandwidth are often addressed more efficiently with electro-hydraulic rather than electromagnetic means. In general, applications with bandwidths of greater than about 20 Hz or control power greater than about 15 kW, may be regarded as suitable for servo-hydraulic techniques.

Apart from the ability to deliver higher forces at fast speeds, servo-hydraulic systems offer several other benefits over their electrical counterparts. For example, hydraulic systems are mechanically “stiffer”, resulting in higher machine frame resonant frequencies for a given power level, higher loop gain and improved dynamic performance. They also have the important benefit of being self-cooled since the driving fluid effectively acts as a cooling medium carrying heat away from the actuator and flow control components. Unfortunately hydraulic systems also exhibit several inherent non-linear effects which can complicate the control problem.

The vast majority of electronic closed loop controllers are based on simple analogue circuit designs offering robust, low cost implementations of the well known PID control strategy. This approach works well in systems with simple topology and limited bandwidth. However the growing use of complex control strategies, coupled with the need to support enhanced features such as data-logging and digital communications, has led to increased interest in the use of digital processors for control of hydraulic servo-systems. Nowhere is this more apparent than in the field of mechanical test equipment, where the use of a programmable digital processor allows the same servo controller to be used with a wide range of hydraulic systems.

This application report reviews some of the issues facing the hydraulic control engineer and discusses the suitability of high-speed DSPs for control of servo-hydraulic systems. The application report begins with an evaluation of the DSP for use in electro-hydraulic servo-controllers, and introduces the TMS320C28x family: a DSP platform optimized for digital control applications. Section 3 describes the principal components of the hydraulic system. In Section 4, mathematical models for the various plant elements are developed and using Simulink. Section 5 presents a case study of a hydraulic control system and deals with fitting real data to the model to validate its behavior. The application report concludes with a brief summary and discussion of the application and design process.

2 The C2000 DSP for Digital Control

2.1 The Benefits of DSP for Digital Control

Embedded digital controllers offer several important benefits to the electronic design engineer, including:

- immunity from errors arising from component tolerance, thermal drift and aging
- improved noise immunity
- ability to modify and store control parameters
- ability to easily implement digital communications
- system fault monitoring and diagnostic capabilities
- data logging capability
- ability to perform automated calibration

The performance of a high quality hydraulic actuator is very dependant on the servo-controller. DSPs lend themselves well to implementing real-time control algorithms, and have been widely used in high-performance digital controllers for many years.

In addition to the benefits above, a DSP allows the engineer to:

- implement advanced control strategies, including multi-variable and complex control algorithms using modern intelligent methods such as neural networks and fuzzy logic.
- perform adaptive control, in which the algorithm dynamically adapts itself to match variations in system behavior.
- implement complex topologies such as multi-axis control where synchronization of multiple force patterns is required.
- perform diagnostic monitoring, including frequency spectrum analysis to identify mechanical vibrations and predict failure modes.
- efficiently implement high-order digital filters including sharp cut-off notch filters to remove energy that would otherwise excite resonant modes and possibly lead to instability.
- reduce system cost by taking advantage of a rich integrated peripheral set to minimize component count and board size.

The use of digital controllers is sometimes avoided on the expectation that the user is obliged to invest time in learning new programming languages, and will face difficulties in testing and de-bugging the code. The Texas Instruments C28x™ DSP platform includes a high performance C compiler and an extensive set of software libraries to minimize development effort.

Recent developments in simulation software also enable designers to model fixed-point digital processors, and to automatically generate optimized source code which may be compiled and run on the target processor directly from the simulation environment. This “hardware-in-the-loop” approach enables the control algorithms generated by the model to be executed on a real target processor such as the C28x during simulation, and increases the level of design confidence. The ease with which control algorithms can be created and modified in this manner can save months of development time and leads to earlier error detection compared with traditional hand coding methods.

2.2 The TMS320C28x DSP Family

The choice of processor to implement a given control algorithm is influenced by many factors. The most basic and obvious is that the device must be able to compute the control algorithm swiftly enough to keep up with the real-time demands of the system. In many cases, such as the simple PID controller, the control task is relatively simple, but to implement more complex control strategies and in cases where additional processor tasks are to be performed, more CPU bandwidth is required and it is desirable to select a processor which is optimized to perform real-time computations. A multi-bus architecture and rich instruction set makes DSPs well suited to executing demanding real-time control algorithms. The TMS320C28x family from Texas Instruments represents the 'state-of-the-art' in control DSPs and is the ideal architecture for digital control applications.

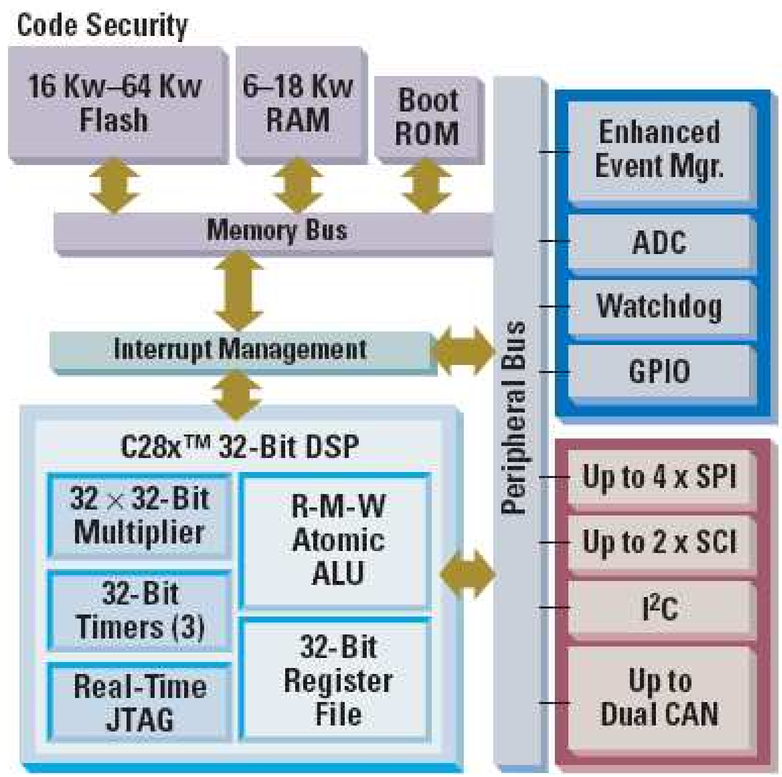

Figure 1. Block Diagram of the TMS320F280x DSP Family

The TMS320C28x DSP incorporates a high-performance 32-bit fixed-point DSP core, featuring a low latency interrupt mechanism, highly efficient instruction set including "atomic" instructions, and an execution pipeline for high-speed code execution from internal flash memory. The C28x has on-chip ROM and RAM memory blocks, and a rich set of integrated peripherals including a high speed A/D converter, several serial ports, and multiple PWM generation units.

Several optimized libraries are available for the C28x platform, including a set of C language peripheral header files, and an optimized mathematical library supporting 32-bit fixed point functions. The combination of a high performance DSP core, a rich integrated peripheral set, and an extensive library of optimized control algorithms renders the C28x an excellent choice for digital control applications.

3 Typical Hydraulic System Description

A typical position controlled hydraulic system consists of a power supply, flow control valve, linear actuator, displacement transducer, and electronic servo-controller. The servo controller compares the signal from the feedback displacement transducer with an input demand to determine the position error, and produces a command signal to drive the flow control valve. The control valve adjusts the flow of pressurized oil to move the actuator until the desired position is attained: a condition indicated by the error signal falling to zero. A force controlled hydraulic system operates in a similar way, except that the oil flow is adjusted to achieve an output force, measured by a suitable transducer.

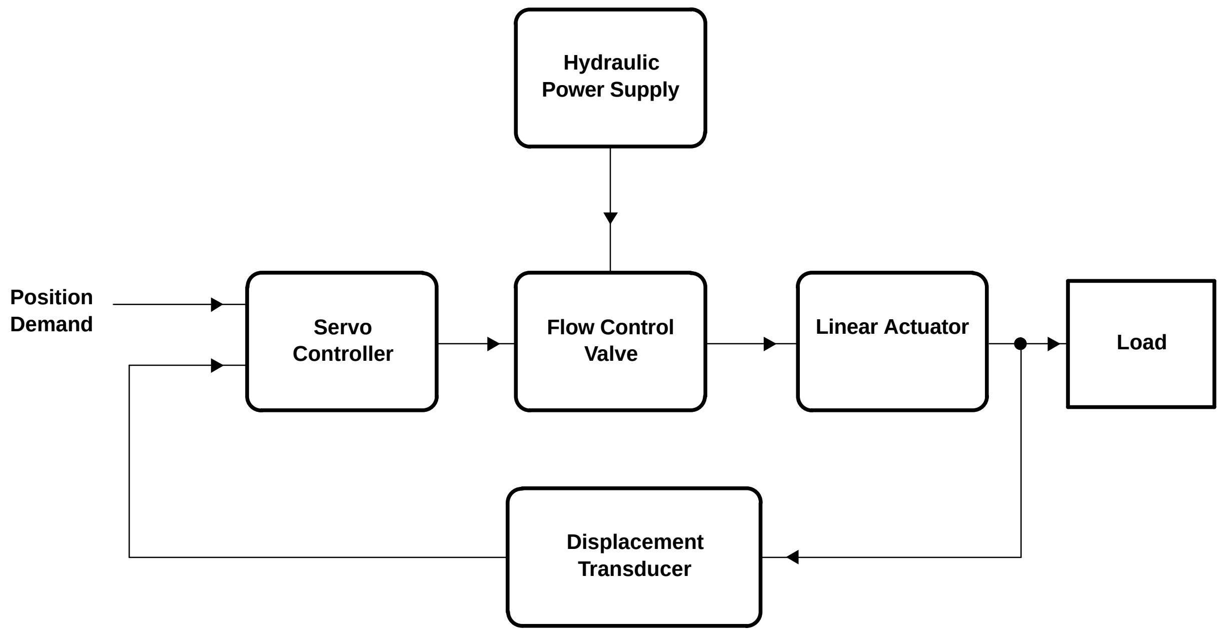

Figure 2. Block Diagram of a Position Controlled Hydraulic Servo System

The following subsections comprise a brief description of the principal hydraulic elements that make up a typical position controlled system.

3.1 Hydraulic Power Supply

All hydraulic systems require a supply of pressurized fluid, usually a form of mineral oil. The choice of system oil pressure depends on various factors. Low pressure means less leakage, but physically larger components are required to develop a given force. High pressure systems suffer from more leakage, but have better dynamic performance and are both smaller and lighter: significant advantages in mobile and aircraft applications. In many high performance systems 3,000 psi (approximately 210 bar) is a standard choice of system pressure.

Oil is drawn from a reservoir (tank) into a rotary vane or piston pump, driven at constant speed by an electric motor. The oil is driven at constant flow rate into an adjustable pressure relief valve, which regulates system pressure by allowing excess oil to return to the reservoir once a pre-defined pressure threshold has been reached. Pressurized hydraulic oil is carried to the servo-valve through a system of rigid or flexible piping, possibly fitted with electrically operated shut-off valves to control hydraulic start-up and shut-down sequences. Oil is returned from the valve to the tank through a low pressure return pipe, which is often fitted with an in-line heat exchanger for temperature regulation of the oil.

3.2 Flow Control Value

The electro-hydraulic flow control valve acts as a high gain electrical to hydraulic transducer, the input to which is an electrical voltage or current, and the output a variable flow of oil. The valve consists of a spool with lands machined into it, moving within a cylindrical sleeve. The lands are aligned with apertures cut in the sleeve such that movement of the spool progressively changes the exposed aperture size and alters differential oil flow between two control ports.

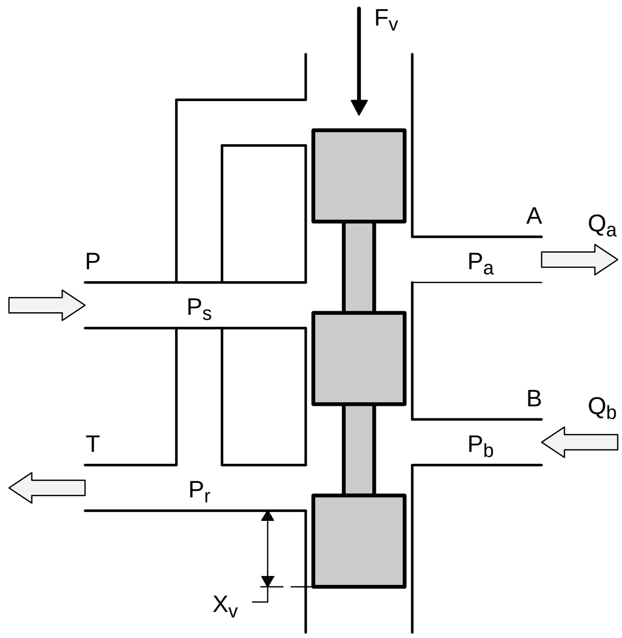

Figure 3. Diagram of Three Land, Four-way Flow Control Valve Spool

Figure 3 shows the spool configuration of a typical “3-4” flow control valve. The ports are labelled P (pressure), T (tank), and A and B (load control ports). The spool is shown displaced a small distance (x_v) as a result of a command force applied to one end, and arrows at each port indicate the direction of fluid flow which results. With no command force applied (F_v=0), the spool is centralized and all ports are closed off by the lands resulting in no load flow.

In the context of hydraulic servo-systems, flow control valves fall broadly into two main categories: proportional valves and servo-valves. Proportional valves use direct actuation of the spool from an electrical solenoid or torque motor, whereas servo-valves use at least one intermediate hydraulic amplifier stage between the electrical torque motor and the spool.

A major advantage of proportional valves is that they are largely unaffected by changes in supply pressure and oil viscosity. However, the relatively large armature mass and large time constant associated with the coil means that these valves generally have poorer dynamic performance compared with servo-valves of equivalent flow characteristics. In recent years, “servo-proportional” valves have begun to appear with shorter spool displacements and lighter spools, giving dynamic performance which approaches that of true servo-valves but at a much lower cost.

The basic servo-valve produces a control flow proportional to input current for a constant load. While the dynamic performance of a servo-valve is influenced somewhat by operating conditions (supply pressure, input signal level, fluid and ambient temperature and so on) a major advantage is that load dynamics do not affect stability, unlike single stage proportional valves. Servo valves usually have superior dynamic response, although their close internal machining tolerances make them relatively expensive and susceptible to contamination of the hydraulic fluid.

Two stage servo-valves may be further divided into nozzle-flapper and jet pipe types. Both use a similar design of electromagnetic torque motor, but the hydraulic amplifier circuits are radically different. Nozzle-flapper type servo-valves are currently by far the most common in high performance servo applications and the description which follows is based on this type of valve.

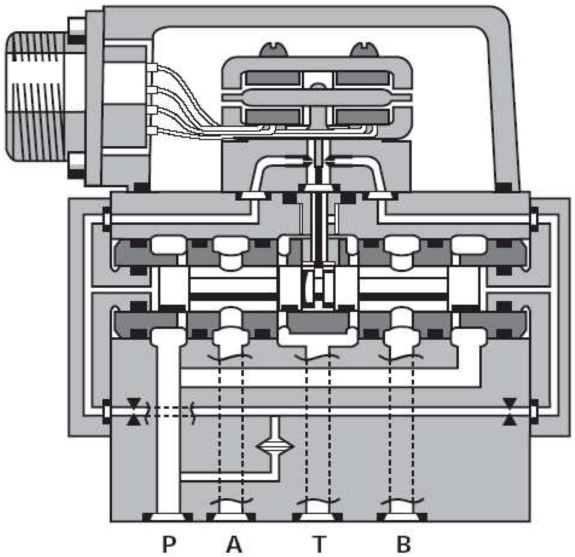

Figure 4. Cross Section of Nozzle-flapper Type Servo-valve (illustration courtesy of Moog)

A cross sectional view of a typical nozzle-flapper type servo-valve is shown in Figure 4. High pressure hydraulic oil is supplied at the inlet pressure port (P), and a low pressure return line to the oil reservoir is connected to the tank port (T). The two hydraulic control ports (A and B) carry the control oil flows to and from the load actuator.

3.3 Linear Hydraulic Actuator

A hydraulic actuator is a device which converts hydraulic energy into mechanical force or motion. Actuators may be divided into those with linear movement (sometimes called rams, cylinders or jacks), and those with rotary movement (rotary actuators and motors). Linear actuators may be further sub-divided into those in which hydraulic pressure is applied to one side of the piston only (single acting) and are capable of movement only in one direction, and those in which pressure is applied to both sides of the piston (double acting) and are therefore capable of controlled movement in both directions.

Linear actuators may also be classified as single-ended, in which the piston has an extension rod on one end only, or the double-ended type which have rods on both ends. Single-ended actuators are useful in space constrained applications, but unequal areas on each side of the piston results in asymmetrical flow gain which can complicate the control problem. Double-ended actuators have the advantage that they naturally produce equal force and speed in both directions, and for this reason are sometimes called *symmetric* or *synchronizing* cylinders.

Hydraulic motors are a separate class of actuator, in which the speed and direction of a rotating output shaft is regulated by the flow control valve.

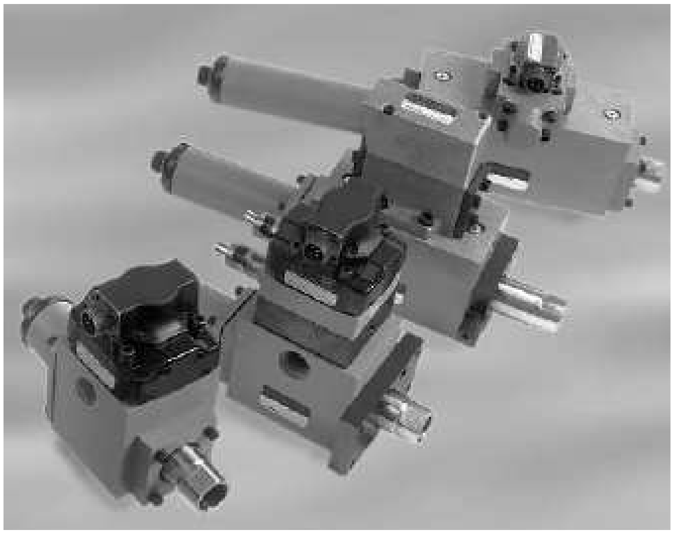

Figure 5. Linear Servo-Hydraulic Actuator Assemblies (illustration courtesy of Moog)

The description which follows is based on a linear, double-acting, double-ended actuator: a type used in many industrial applications. A cross section of such an actuator is shown in Figure 6. The actuator consists of a rod and central annulus, and incorporates low friction seals fitted to the piston annulus and at each of the cylinder end caps to minimize leakage. Control ports are drilled into each end of the cylinder to allow hydraulic fluid to flow in and out of the two chambers.

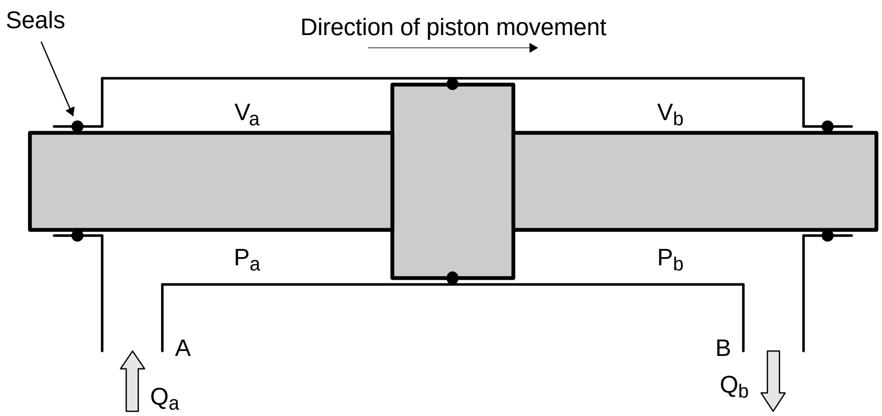

Figure 6. Cross-sectional Diagram of Double-ended, Double-acting Linear Actuator

The position of the piston is determined by the hydraulic fluid pressures in the chambers on either side of the central annulus, and may be adjusted by forcing fluid into one control port while allowing it to escape from the other. In the diagram above, hydraulic fluid is shown entering control port A while escaping from port B. This causes an increase in fluid pressure in the chamber to the left of the piston annulus, and a decrease in pressure in the right chamber. The net pressure difference exerts a force on the active area of the annulus which moves the piston to the right as shown. Adjustment of piston position is therefore a matter of controlling the differential oil flow between the two actuator control ports.

3.4 Displacement Transducer

Position transducers are usually collocated with the actuator, and often attached directly to the piston rod. Various types of feedback transducer are in use, including incremental or absolute encoders, inductive linear variable differential transformer (LVDT's) and rotary variable differential transformer (RVDT's), linear and rotary potentiometers, and resolvers. In industrial applications employing linear displacement control, the LVDT is a common choice of feedback transducer due to its accuracy and robustness. The transducer is usually selected such that its bandwidth is ten times or so higher than that of the servo-valve and actuator, and is often omitted from a first analysis of the system.

3.5 Servo Controller

The diagram below shows the layout of the servo-controller block.

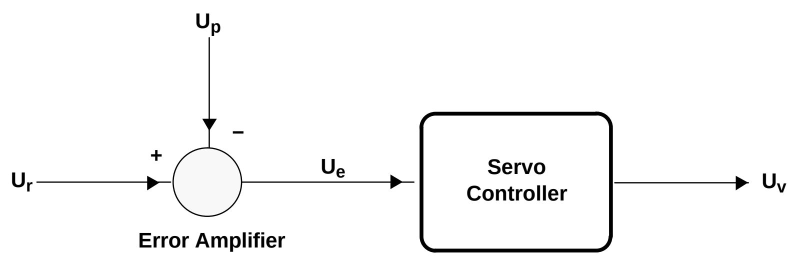

Figure 7. Block Diagram of the Analogue Controller

The error amplifier continuously monitors the input reference signal (u_r) and compares it against the actuator position (u_p) measured by a displacement transducer to yield an error signal (U_e).

$$u_e = u_r - u_p \tag{1}$$

The error is manipulated by the servo controller according to a pre-defined control law to generate a command signal (u_v) to drive the hydraulic flow control valve. Most conventional electro-hydraulic servo-systems use a PID form of control, occasionally enhanced with velocity feedback. The processing of the error signal in such a controller is a function of the proportional, integral, and derivative gain compensation settings according to the control law

$$u_v(t) = K_p u_e(t) + K_i \int u_e \, dt + K_d \frac{du_e}{dt} \tag{2}$$

where K_p, K_i, and K_d are the PID constants, u_e is the error signal and u_v is the controller output.

For information on implementing discrete controllers, the reader may find 17, 18, and 19 of Section 7 at the end of this application report helpful.

4 Hydraulic System Modelling

This section develops basic mathematical models of the hydraulic components described in Section 3, and makes use of Simulink™ modelling software from The Mathworks™ Inc. For further information on this software, the reader is referred to the references at the end of this application report to the training courses run by The Mathworks Inc. Examples of graphical models of some of the major hydraulic elements are shown in Appendix A.

4.1 Flow Control Servo-Valve

The two stage nozzle-flapper servo-valve consists of three main parts: an electrical torque motor, hydraulic amplifier, and valve spool assembly.

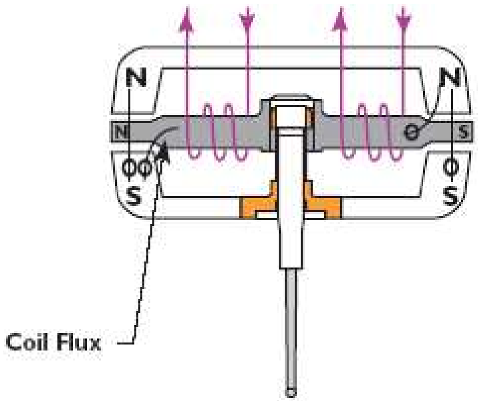

Figure 8. Valve Torque Motor Assembly (illustration courtesy of Moog)

The torque motor consists of an armature mounted on a thin-walled sleeve pivot and suspended in the air gap of a magnetic field produced by a pair of permanent magnets. When current is made to flow in the two armature coils, the armature ends become polarized and are attracted to one magnet pole piece and repelled by the other. This sets up a torque on the flapper assembly, which rotates about the fixture sleeve and changes the flow balance through a pair of opposing nozzles, shown in Figure 9. The resulting change in throttle flow alters the differential pressure between the two ends of the spool, which begins to move inside the valve sleeve.

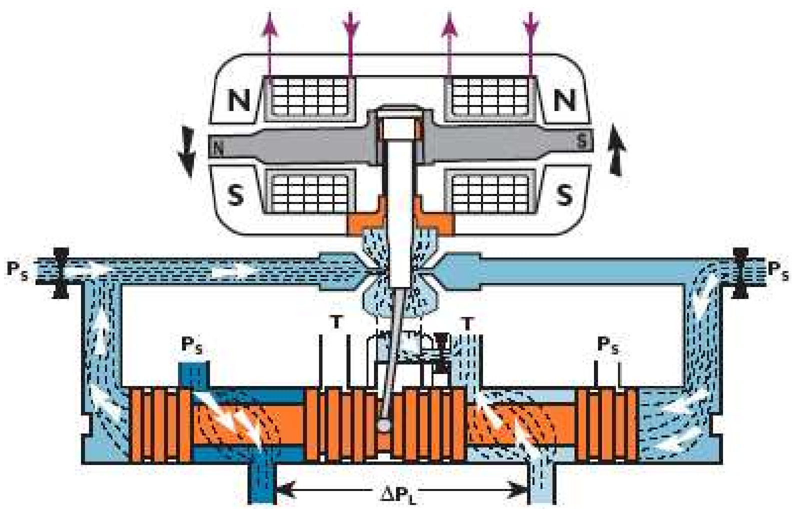

Figure 9. Valve Responding to Change in Electric Input (illustration courtesy of Moog)

Lateral movement of the spool forces the ball end of a feedback spring to one side and sets up a restoring torque on the armature/flapper assembly. When the feedback torque on the flapper spring becomes equal to the magnetic forces on the armature the system reaches an equilibrium state, with the armature and flapper centred and the spool stationary but deflected to one side. The offset position of the spool opens flow paths between the pressure and tank ports (P_S and T), and the two control ports (A and B), allowing oil to flow to and from the actuator.

4.1.1 Torque Motor

For simplicity, the electrical characteristics of the servo-valve torque motor may be modelled as a series L-R circuit, neglecting for the time being any back-EMF effects generated by the load. The transfer function of a series L-R circuit is

$$\frac{I(s)}{V(s)} = \frac{1}{sL_c + R_c} \tag{3}$$

where L_c is the inductance of the motor coil, and R_c the combined resistance of the motor coil and the current sense resistor of the servo amplifier. Values of inductance and resistance for series and parallel winding configurations of the motor are published in the manufacturer's data sheet.

The lateral force on the valve spool is proportional to torque motor current, but oil flow rate at the control ports also depends upon the pressure drop across the load.

4.1.2 Valve Spool Dynamics

A servo-valve is a complex device which exhibits a high-order non-linear response, and knowledge of a large number of internal valve parameters is required to formulate an accurate mathematical model. Indeed, many parameters such as nozzle and orifice sizes, spring rates, spool geometry and so on, are adjusted by the manufacturer to tune the valve response and are not normally available to the user.

Practically all physical systems exhibit some non-linearity: in the simplest case this may be a physical limit of movement, or it may arise from the effects of friction, hysteresis, mechanical wear or backlash. When modelling complex servo-valves, it is sometimes possible to ignore any inherent non-linearities and employ a small perturbation analysis to derive a linear model which approximates the physical system. Such models are often based on classical first or second order differential equations, the coefficients of which are chosen to match the response of the valve based on frequency plots taken from the data sheet.

A simple first or second order model yields only an approximation to actual behavior, however the servo-valve is not the primary dynamic element in a typical hydraulic servo system and is generally selected such that the frequency of the 90 degree phase point is a factor of at least three higher than that of the actuator. For this reason it is usually only necessary to accurately model valve response through a relatively low range of frequencies, and the servo-valve dynamics may be approximated by a second order transfer function without serious loss of accuracy.

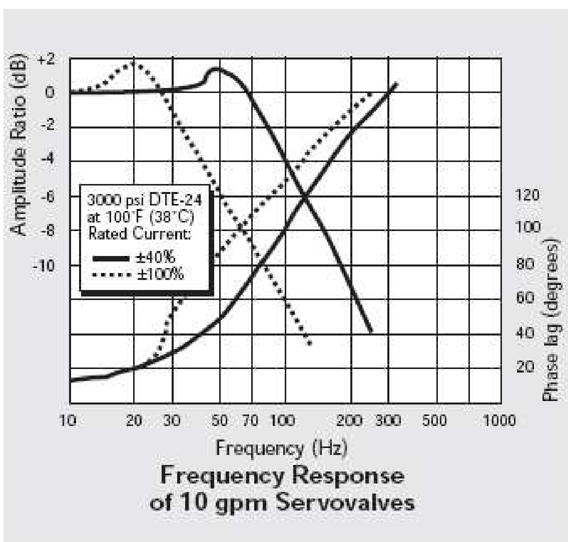

Figure 10. Typical Servo-valve Frequency Response Curve (illustration courtesy of Moog)

A typical performance graph for a high-response servo-valve is shown in Figure 10. Assuming a second order approximation is to be used, suitable values for natural frequency and damping ratio will need to be determined from the graph. Natural frequency (ω_v) can be read fairly accurately from the -3dB or 90 degree phase point of the 40% curve. Damping can be determined from an estimate of the magnitude of the peaking present. For an under-damped second order system, the damping factor (S_v) can be shown to be related to peak amplitude ratio (M_v) by the formula

$$M_v = \frac{1}{2\zeta_v\sqrt{1-\zeta_v{}^2}} \tag{4}$$

In this example, a reasonable estimate of peaking based on the 40% response curve would be about 1.5 dB, which corresponds to an amplitude ratio of about 1.189. A suitable value of damping determined iteratively from Equation 2 is about 0.48. Armed with these values, a simplified model of the servo-valve spool dynamics may be constructed. The input to the model will be the torque motor current derived from Equation 1 normalized to the saturation current obtained from the datasheet, and the output will be the normalized spool position.

4.1.3 Valve Flow-Pressure

The servo-valve delivers a control flow proportional to the spool displacement for a constant load. For varying loads, fluid flow is also proportional to the square root of the pressure drop across the valve. Control flow, input current, and valve pressure drop are related by the following simplified equation

$$Q_L = Q_R \times i_v^* \times x\sqrt{\frac{\Delta P_V}{\Delta P_R}} \tag{5}$$

In the above equation, Q_L, is the hydraulic flow delivered through the load actuator, Q_R the rated valve flow at a specified pressure drop (ΔP_R), and i^*_v is normalized input current. ΔP_V is the pressure drop across the valve given by $\Delta P_V = P_S - P_T - P_L$, where P_S, P_T, and P_L are system pressure, return line (tank) pressure, and load pressure respectively. Maximum power is transferred to the load when $P_L = 2/3\ P_S$, and since the most widely used supply pressure is 3,000 psi, it is common practice to specify rated valve flow at ΔP = 1,000 psi (approximately 70 bar). The static relationship between valve pressure drop and load flow is often presented in manufacturer's datasheets as a family of curves of normalized control flow against normalized load pressure drop for different values of valve input current as shown in Figure 11.

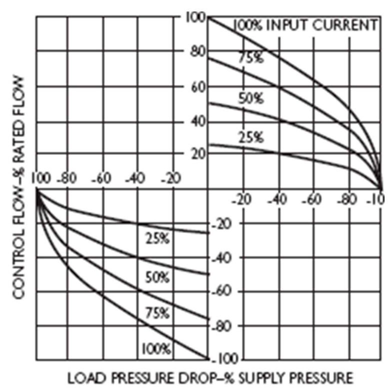

Figure 11. Servo-valve Flow-pressure curves (illustration courtesy of Moog)

The horizontal axis is the load pressure drop across the valve, normalized to 2/3 of the supply pressure. The vertical axis is output flow expressed as a percentage of the rated flow, Q_R. The valve orifice equation is applied separately for the two control ports to obtain expressions for oil flow into each of the two actuator chambers. Since load flow is defined as the flow *through* the load: $Q_L = Q_A = -Q_B$

A Simulink model of the servo-valve is shown in Appendix A. The inputs are command voltage from the amplifier, supply and return oil pressures from the hydraulic power supply (P_S and P_T), and load pressures from the actuator chambers (P_A and P_B). Outputs are the flows to each side of the piston (Q_A and Q_B), and the load flow (Q_L).

4.2 Linear Actuator

4.2.1 Cylinder Chamber Pressure

The relationship between valve control flow and actuator chamber pressure is important because the compressibility of the oil creates a “spring” effect in the cylinder chambers which interacts with the piston mass to give a low frequency resonance. This is present in all hydraulic systems and in many cases this abruptly limits the usable bandwidth. The effect can be modelled using the flow continuity equation from fluid mechanics which relates the net flow into a container to the internal fluid volume and pressure.

$$\Sigma Q_{in} - \Sigma Q_{out} = \frac{dV}{dt} + \frac{V}{\beta}\frac{dP}{dt} \tag{6}$$

The left hand side of the equation is the net flow delivered to the chamber by the servo valve. The first term on the right hand side is the flow consumed by the changing volume caused by motion of the piston, and the second term accounts for any compliance present in the system. This is usually dominated by the compressibility of the hydraulic fluid and is common to assume that the mechanical structure is perfectly rigid and use the bulk modulus of the oil as a value for β. Mineral oils used in hydraulic control systems have a bulk modulus in the region of 1.4 x 109 N/m. Equation 6 can be re-arranged to find the instantaneous pressure in chamber A as follows:

$$P_A = \frac{\beta}{V}\int\left(Q_A - \frac{dV_A}{dt}\right)dt \tag{7}$$

4.2.2 Piston Dynamics

Once the two chamber pressures are known, the net force acting on the piston (F_P) can be computed by multiplying by the area of the piston annulus (A_P) by the differential pressure across it.

$$F_P = (P_A - P_B)A_P \tag{8}$$

An equation of forces for piston motion can now be established by applying Newton’s second law. For the purposes of this analysis, it will be assumed that the piston delivers a force to a linear spring load with stiffness K_L, which will allow us to investigate the load capacity of the actuator later. The effects of friction (F_f) between the piston and the oil seals at the annulus and end caps will also be included. The resulting force equation for the piston is shown below and may be modelled in Simulink using two integrator blocks.

$$F_P = M_P\frac{d^2x_p}{dt^2} + F_f + K_L x_p \tag{9}$$

The total frictional force depends on piston velocity, driving force (F_P), oil temperature and possibly piston position. One method of modelling friction is as a function of velocity, in which the total frictional force is divided into static friction (a transient term present as the actuator begins to move), Coulomb friction (a constant force dependent only on the direction of movement), and viscous friction (a term proportional to velocity). Assuming that viscous and Coulomb friction components dominate, frictional force (F_f) can be modelled as

$$F_f = \frac{dx}{dt}F_{v0} + sign\left(\frac{dx}{dt}\right)F_{c0} \tag{10}$$

where viscous and Coulomb friction coefficients are denoted by F_{v0} and F_{c0} respectively. Frictional effects are notoriously difficult to measure and accurate values of these coefficients are unlikely to be known, but order of magnitude estimates can sometimes be made from relatively simple empirical tests. One test which can yield useful information is to subject the system to a low frequency, low amplitude sinusoidal input signal, and plot the output displacement over one or two cycles. A low friction system should reproduce the input signal, but the presence of friction will tend to flatten the tops of the sine wave as the velocity falls to a level blow that necessary to overcome any inherent Coulomb friction. In actuators fitted with conventional, PTFE-based bearings, friction is related fairly linearly to supply pressure and oil temperature and care should be taken to conduct testing under representative conditions.

In a first analysis, leakage effects in the actuator are sometimes neglected, however this is an important factor which can have a significant damping influence on actuator response. Leakage occurs at the oil seals across the annulus between the two chambers and at each end cap, and is roughly proportional to the pressure difference across of the seal. Including leakage effects, the flow continuity equation for chamber A is

$$Q_A - K_{La}(P_A - P_B) - K_{Le}P_A = \frac{dV_A}{dt} + \frac{V_A}{\beta}\frac{dP_A}{dt} \tag{11}$$

where K_{La} and K_{Le} are internal and external leakage coefficients respectively. The equation for chamber B is similar with appropriate changes of sign. It is a relatively simple matter to modify the model to compute the instantaneous chamber leakages and subtract them from the total input flow. The complete actuator model, including frictional and leakage effects, is shown in Appendix A.

4.3 Hydraulic Power Supply

The behavior of the hydraulic power supply described in Section 3.1 may be modelled in the same way as the chamber volumes: by applying the flow continuity equation to the volume of trapped oil between the pump and servo-valve. In this case, the input flow is held constant by the steady speed of the pump motor, and the volume does not change. The transformed equation is

$$P_S = \frac{\beta}{V_1}\int \left(Q_{pump} - Q_L\right)dt \tag{12}$$

This equation takes into account the load flow (Q_L) drawn from the supply by the servo-valve, and accurately models the case of a high actuator slew rate resulting in a load flow which exceeds the flow capacity of the pump. In such cases the supply pressure (P_S) falls, leading to a corresponding reduction in control flow (Equation 5) and loss of performance. The action of the pressure relief valve may be modelled using a limited integrator to clamp the system pressure to the nominal value.

5 Case Study

This section presents some typical values for a high-performance, symmetrical, linear actuator. Simulation results for an actuator based on these values are shown, followed by the results of some physical tests conducted on a similar real actuator.

5.1 Example System Data

The example data presented here is based on a symmetrical, double-ended actuator with a total stroke of around 100mm and an active piston area of around 1”. Piston mass may be specified by the manufacturer, or can be calculated from a knowledge of the geometry of the actuator. For a piston of this size, a mass of around 9 Kg would be typical.

Table 1. Actuator Data

Symbol	Description	
M_p	Mass of actuator pis- ton	9 Kg
$X_{p(max)}$	Total stroke of the piston	0.1 m
A_p	Active area of piston annulus	645 x 10^{-6} m ^{2}

Data for the servo-valve can be read directly or calculated from the manufacturer's data sheet as described in Section 4. The following values are based on the Moog 760-series valve.

Table 2. Servo-valve Data

Symbol	Description	
Q_r	Rated flow of valve at 70 bar pressure drop	0.63069×10^{-3} m^3/s
M_v	Ratio of peaking in servo valve frequency response	1.5 dB
S_v	Damping ratio of servo valve model	0.48
ω_v	Natural frequency of servo valve model	534 rad/s
L_c	Inductance of servo valve coil	0.59 H
R_c	Series resistance of torque motor circuit	100 Ω
$I_{v(sat)}$	Saturation current torque motor	0.02 A

The remaining data is usually available from knowledge of the system. Leakage and frictional effects could be modelled using coefficient values estimated from empirical data.

Table 3. Miscellaneous System Data

Symbol	Description	
B_e	Bulk modulus of hydraulic fluid	1.4×10^{-9} N/m^2
P_s	Supply pressure from hydraulic pump	2.1×10^{7} Pa
P_R	Pressure drop in return line to tank	0 Pa
Q_p	Maximum oil flow capacity of pump	1.67×10^{-3} m^3/s
V_t	Volume of trapped oil between pump and servo-valve	0.0005 m^3

Note:
The SI system of units (Kg, m, s) has been used throughout. The use of Imperial units (lb, in, s) is quite widespread in hydraulics and the user should keep this in mind when interpreting data sheets. A list of variables used in the model is shown in Appendix B.

5.2 Simulation Results

A simple test which yields useful information about the performance of the system is to apply a small step signal to the input and monitor the response. In Simulink, this can be simply achieved using a 'scope' block to monitor command and response signals. The position of the step input source and scope blocks are colored red in the system level block diagram in Appendix A, and a typical actuator step response is shown in Figure 12. The vertical axis is graduated in Volts measured at the error amplifier. The controller is scaled to a range of ±10V so for a 100mm stroke actuator, a step input of +2V corresponds to a piston displacement of +10mm from the central position.

Controller gain terms are adjusted and the step test repeated to tune the actuator response as required. The plot shown represents a satisfactory compromise between rise time and overshoot, and the corresponding controller settings would serve as a useful starting point for the control engineer when testing the real system.

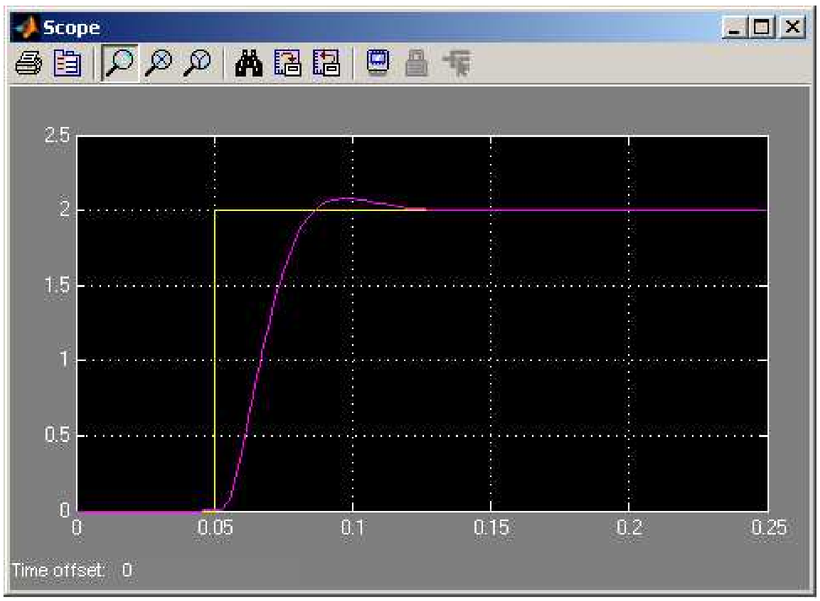

Figure 12. Simulated Actuator Step Response

The relationship between load pressure and load flow from the servo-valve was described in Section 4.2.1. By configuring the actuator chamber pressures as test points in Simulink, a 'floating scope' block may be used to monitor these and other signals during the step response test. The graph below shows the behavior of the chamber pressures during the step response simulation. Chamber B pressure (P_b) is shown as a continuous line, and chamber A pressure (P_a) a dashed line. The vertical axis is in Pascals (N/m). The slight asymmetry results from the change in chamber volumes as the piston is displaced to its new position.

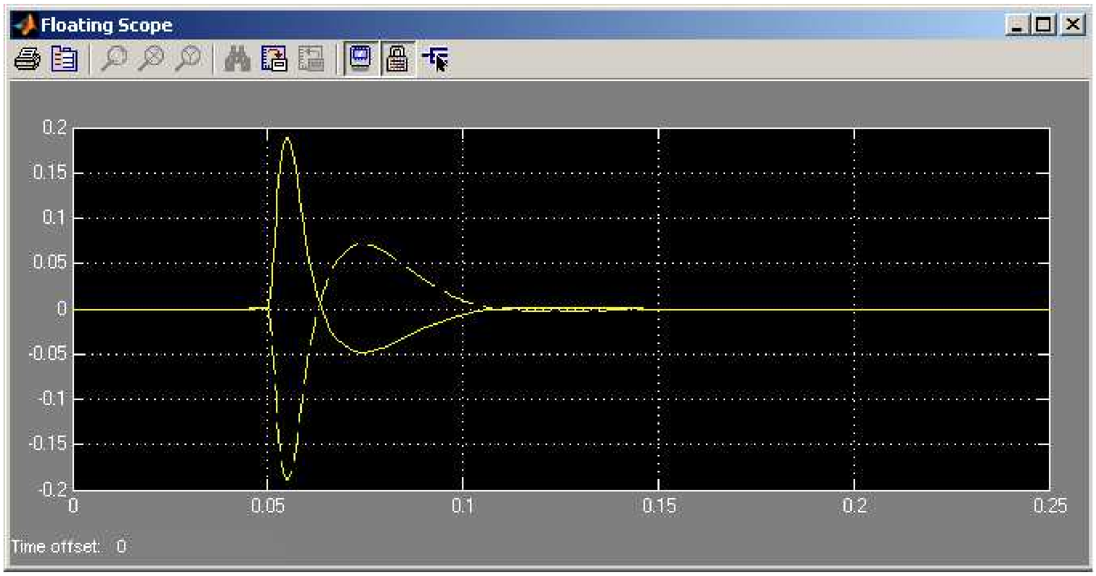

Figure 13. Step Response of Piston Chamber Pressures

5.3 Physical Test Results

Figure 14 shows the step response of a real high-performance linear actuator with dimensions similar to those given in Section 5.1. The sharp rising and falling edges and minimal overshoot represent the optimum response that can be obtained with a PID control strategy and a good quality actuator.

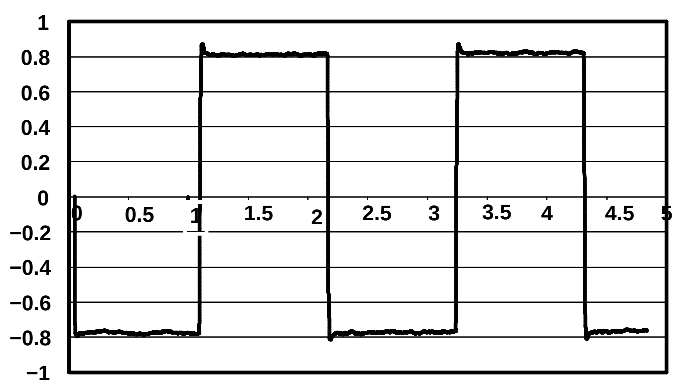

Figure 14. Step Response of High-Performance Linear Actuator

The dynamic performance of the system is limited by the capacity of the hydraulic power supply as well as the performance of the servo-valve. This is illustrated by Figure 15, in which a large sinusoidal command input is applied and the frequency gradually increased.

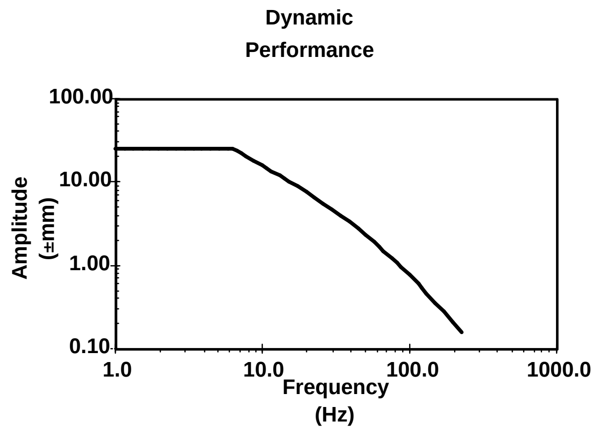

Figure 15. Frequency Response of High-Performance Linear Actuator

The curve shows two high frequency asymptotes: the first occurs at about 8 Hz and is caused by the limited flow capacity of the power supply (see Section 4.3). The second, at about 40 Hz, is the high frequency response limit of the servo-valve.

The importance of friction in a high performance actuator is demonstrated by the following two displacement/time graphs. Both show the behavior of a linear servo-actuator when subjected to a low-frequency, low amplitude sinusoidal command input. The first shows a low friction actuator, the second an actuator with higher friction caused by tighter piston and end cap oil seals.

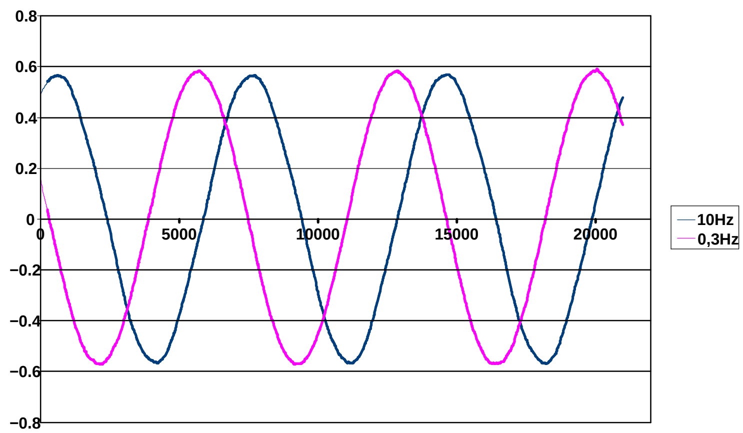

Figure 16. Low Frequency Test on Low Friction Linear Actuator

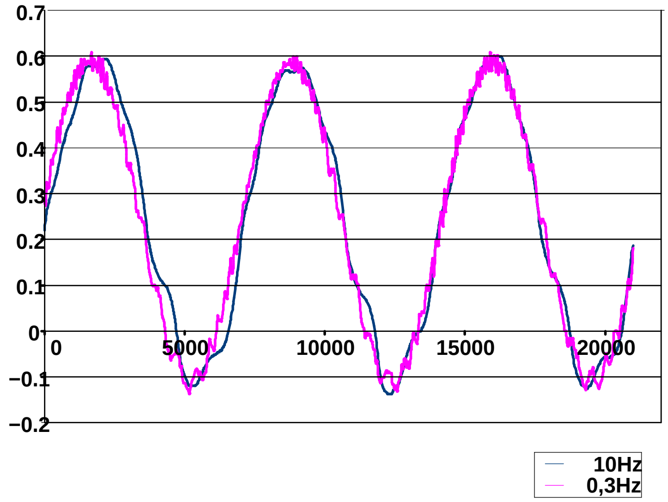

Figure 17. Low Frequency Test on Linear Actuator with Significant Friction

Response losses caused by friction in the actuator can be reduced to some extent by the addition of “dither” to the controller output. This is a relatively high frequency, constant amplitude oscillation, which keeps the valve spool in constant motion and reduces the break-away force needed to overcome any static friction present in the system. The use of Simulink to model and design the system allows enhancements such as this to be easily added to the digital controller.

6 Summary

This application report has examined various issues relating to the control of electro-hydraulic servo-actuators, and discussed the benefits of using a DSP for digital control in these systems. A basic mathematical model has been derived based on a high performance symmetrical linear actuator, but with relatively simple parametric changes could be modified to embrace asymmetric and rotary actuator types.

The principal non-linear effects in hydraulic systems arise from the compressibility of hydraulic fluid, the complex flow properties of the servo-valve, and internal friction in the actuator. These depend on physical factors which are difficult to measure accurately and for this reason simulation results should be supported by experimental testing whenever possible.

Conventional feedback control techniques work well in cases where dead reckoning of the above factors is possible or where their influence is sufficiently small that they can safely be ignored. However, for true high-performance control advanced digital control techniques are required and in this the DSP excels.

The servo-controller for systems such as those described in this application report could be implemented on a C28x DSP. The device has a high-performance A/D converter for sampling command and feedback signals, a 32-bit CPU core optimized for digital control, and an array of PWM channels capable of generating the servo demand signal for the flow control valve. A highly efficient C compiler and powerful code development and debug tools contribute to make this the platform of choice for many embedded control designers.

As pressure grows for faster time to market of new products with more stringent safety features, increasing emphasis is being placed on the use of high-level tools and software for application development. These afford a level of abstraction from the processor core, and permit rapid application development and re-use of material. For complex control applications, the ability to simulate controller and plant behavior at the design phase is invaluable, and the use of embedded auto-code generation and validation features affords the designer the ability to move quickly and easily from simulation to prototyping. This is a powerful advantage and we may expect the trend towards integrated system simulation and code development to continue in the coming years.

7 References

1. Herbert E. Merritt, Hydraulic Control Systems, Wiley, 1967
2. M.Jelali and A.Kroll, Hydraulic Servo Systems - Modelling, Identification & Control, Springer, 2003
3. Electrohydraulic Valves A Technical Look, Moog Technical Paper
4. Moog 760 Series Servovalves, product datasheet
5. R.H.Maskrey and W.J.Thayer, A Brief History of Electrohydraulic Servomechanisms, Moog Technical Bulletin 141, June 1978
6. T. P. Neal, Performance Estimation for Electrohydraulic Control Systems, Moog Technical Bulletin 126, November 1974
7. W.J.Thayer, Transfer Functions for Moog Servovalves, Moog Technical Bulletin 103, January 1965
8. J.C.Jones, Developments in Design of Electrohydraulic Control Valves, Moog Technical Paper, November 1997
9. D.C.Clarke, Selection & Performance Criteria for Electrohydraulic Servovalves, October 1969
10. M.D'Amore and G.Pellegrinetti, Dissecting High-Performance Electrohydraulic Valves, Machine Design, April 2001
11. D.DeRose, The Expanding Proportional and Servo Valve Marketplace, Fluid Power Journal, March/April 2003
12. D.DeRose, Proportional and Servo Valve Technology, Fluid Power Journal, March/April 2003
13. D. Caputo, Digital Motion Control for Position and Force Loops NFPA Technical Paper I96-11.1, April 1996
14. B.C.Kuo and F.Golnaraghi, Automatic Control Systems, Wiley, 2003
15. J.Schwarzenbach and K.F.Gill, System Modelling & Control, Edward Arnold, 1992
16. J.B. Dabney and T.L. Harman, Mastering Simulink, Pearson Prentice Hall, 2004
17. *Digital Control Applications With The TMS320 Family Selected Application Notes* (SPRA019)
18. *Implementation of PID and Deadbeat Controllers with the TMS320 Family* (SPRA083)
19. Gene F. Franklin, J. David Powell & Michael L. Workman, Digital Control of Dynamic Systems, Addison-Wesley, 1998
20. *TMS320F2812 Data Manual* (SPRS174)
21. *Signal Conditioning an LVDT using a TMS320F2812 DSP* (SPRA946)

Appendix A Simulink Models

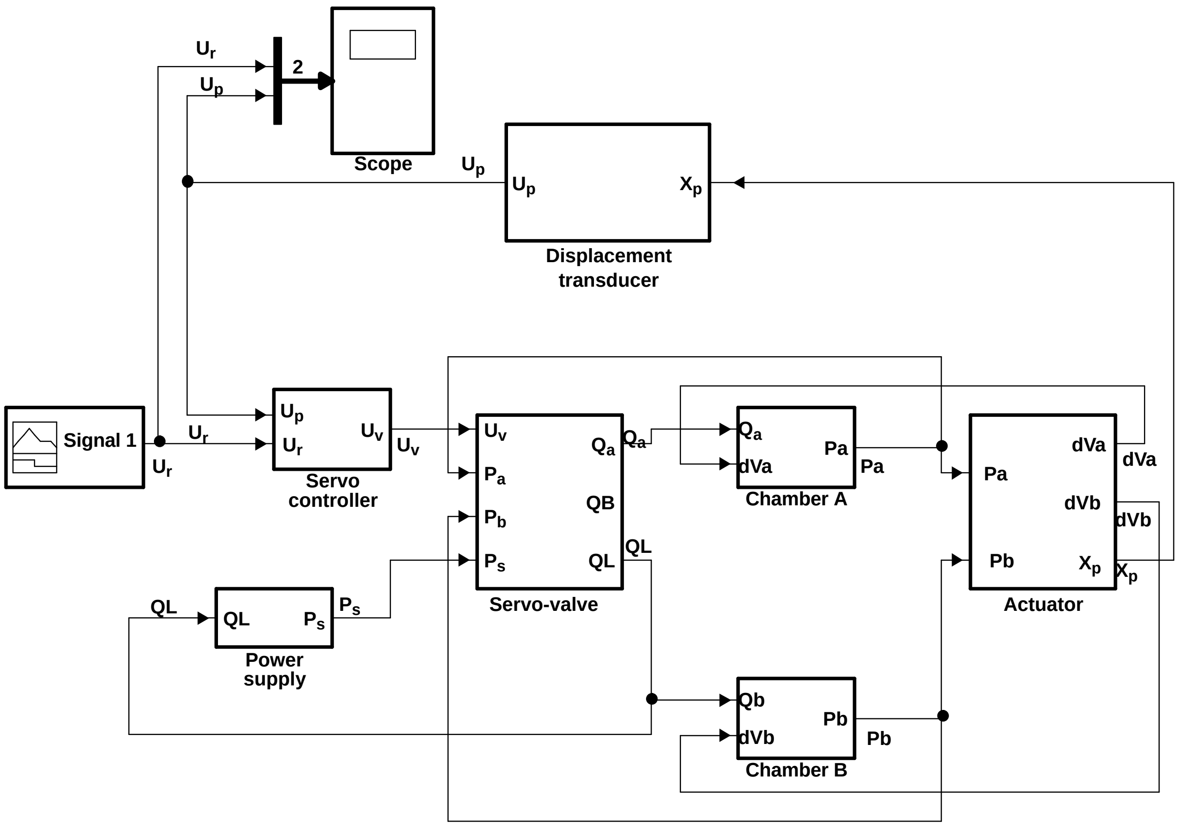

Figure A-1. Top Level System Diagram

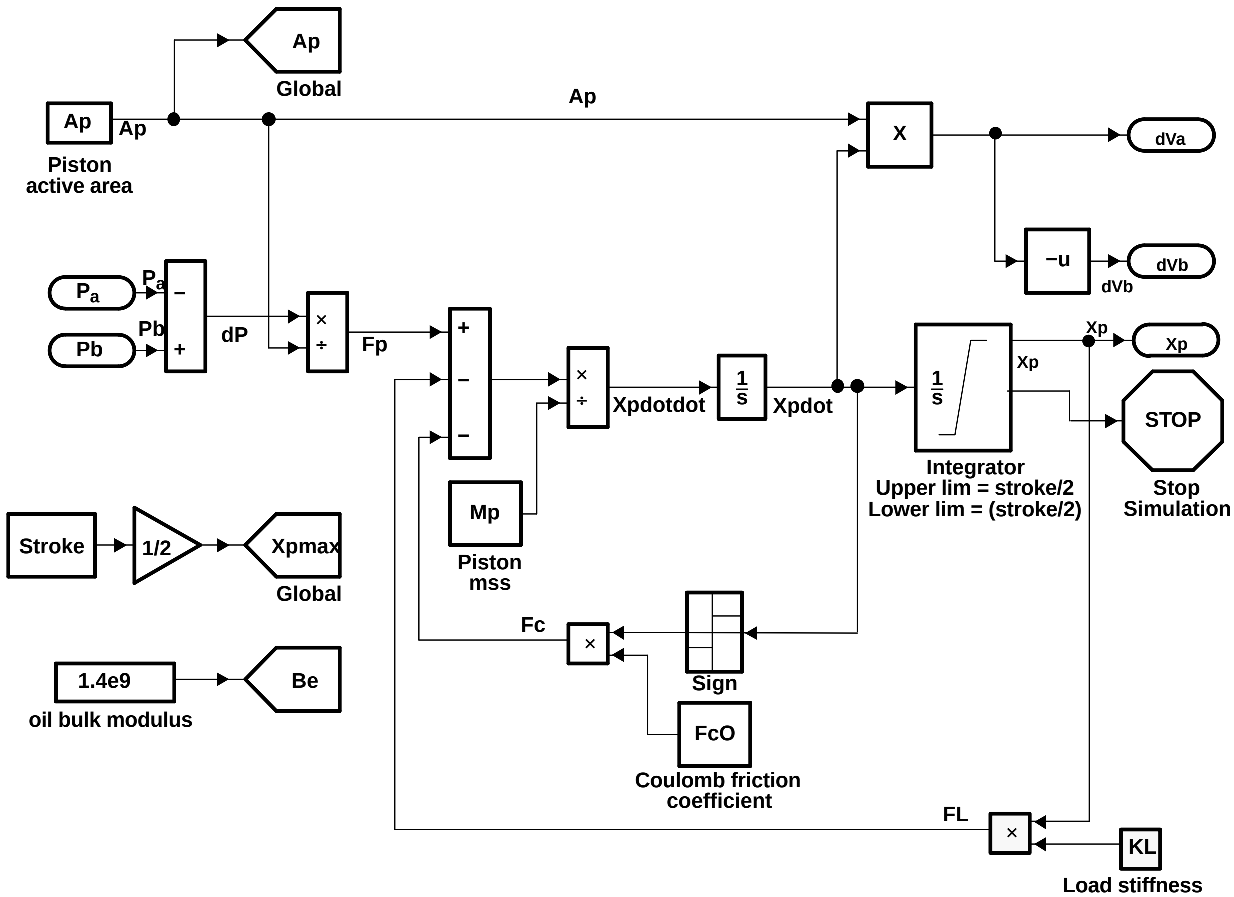

Figure A-2. Simulink Model of Hydraulic Actuator

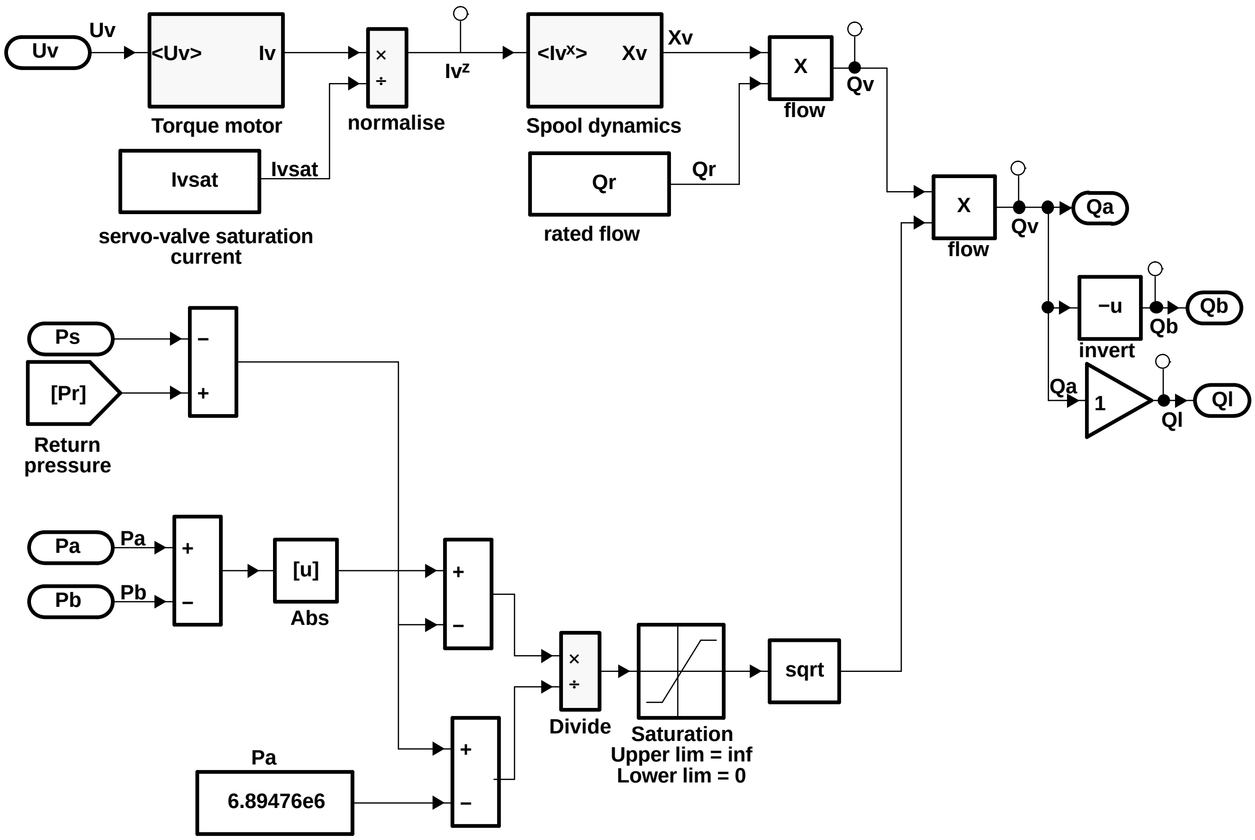

Figure A-3. Simulink Model of Servo-Valve

Appendix B List of Symbols

Symbol	Description	Units
A_P	Active area of piston annulus	m^2
B_e	Bulk modulus of hydraulic fluid	N/m^2
$dV_{A,B}$	Rate of change of volume of chamber A, B	m^3/s
F_p	Force generated across piston annulus	N
L_c	Inductance of servo valve coil	H
M_p	Mass of actuator piston	Kg
M_v	Ratio of peaking of servo valve frequency response	
$P_{A,B}$	Oil pressure at actuator port A, B	Pa
P_R	Pressure drop in return line to tank	Pa
P_S	Supply pressure from hydraulic pump	Pa
$Q_{A,B}$	Oil flow at servo-valve control port A,B	m^3/s
Q_L	Total oil flow through the load	m^3/s
Q_p	Maximum oil flow capacity of pump	m^3/s
Q_r	Rated flow of servo-valve at 70 bar press-ure drop	m^3/s
R_c	Series resistance of torque motor circuit	Ω
U_e	Error output from summing amplifier	V
u_p	Feedback signal from displacement trans-ducer	V
u_r	Command signal to servo loop	V
u_v	Command signal to servo valve	V
$V_{A,B}$	Volume of trapped oil in chamber A,B of the cylinder	m^3
V_t	Volume of trapped oil between pump and servo-valve	m^3
X_P	Displacement of piston relative to centre position	m
S_v	Damping ratio of servo valve model	
ω_v	Natural frequency of servo valve model	rad/s

IMPORTANT NOTICE

Following are URLs where you can obtain information on other Texas Instruments products and application solutions:

Products		**Applications**	
Amplifiers	amplifier.ti.com	Audio	www.ti.com/audio
Data Converters	dataconverter.ti.com	Automotive	www.ti.com/automotive
DSP	dsp.ti.com	Broadband	www.ti.com/broadband
Interface	interface.ti.com	Digital Control	www.ti.com/digitalcontrol
Logic	logic.ti.com	Military	www.ti.com/military
Power Mgmt	power.ti.com	Optical Networking	www.ti.com/opticalnetwork
Microcontrollers	microcontroller.ti.com	Security	www.ti.com/security
		Telephony	www.ti.com/telephony
		Video & Imaging	www.ti.com/video
		Wireless	www.ti.com/wireless

Mailing Address: Texas Instruments
Post Office Box 655303 Dallas, Texas 75265