SIMULATION OF ELECTRO-HYDRAULIC SERVO ACTUATOR

A Thesis submitted in partial fulfillment of the requirements for the degree of

Master of Technology in Mechanical Engineering
(Machine Design and Analysis Specialization)

By

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BONAFIDE CERTIFICATE

This is to certify that the thesis entitled "SIMULATION OF ELECTRO-HYDRAULIC SERVO ACTUATOR", submitted by Mr. Vijaya Sagar Tenali for the award of the degree of Master of Technology (Machine design & Analysis) of National Institute of Technology is product of research work carried out by him under my/our guidance. Mr.Vijaya Sagar Tenali has worked on the above problem at Hydraulics group, RWRDC, HAL Bangalore and this has reached the standard of fulfilling the requirements and the regulation to the degree. The contents of this thesis, in full or in part, have not been submitted to any other university or institution for the award of any degree or diploma.

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CONTENTS

ABSTRACT

Hydraulic actuators are used in many applications like aircraft flight control, machinery and automobiles etc. This actuator when coupled with a feedback system is called a Servo Actuator. The response of the hydraulic actuator with time is significant particularly when the actuator is used for flight control operations. So finding the time response of the particular hydraulic actuator much before its actual operation will be very helpful for the designer for analyzing the performance of the system. This also helps the designer to arrive at optimum design parameters of the hydraulic actuator. In this thesis a position control electro-hydraulic linear actuator is selected. This actuator is used for controlling the movements of the helicopter. Mathematical modelling of the hydraulic actuator and its components is done and based on the mathematical equations Matlab/Simulink models of the actuator and its components were made and the time response of the linear actuator is obtained by using Matlab/Simulink Software. The time response graphs which are obtained in this simulation are found to be in good compromise with the time response graphs of Moog experimental time response graphs.

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NOMENCLATURE

A_P	=	Active area of the piston annulus
Be	=	Bulk Modulus of the hydraulic fluid
dV_A	=	Rate of change of volume of chamber A
dV_B	=	Rate of change of volume of chamber B
F_P	=	Force generated across piston annulus
L_C	=	Inductance of the servo valve
M_P	=	Mass of the actuator piston
P_A	=	Oil Pressure at actuator port A
P_B	=	Oil Pressure at actuator port B
P_R	=	Pressure drop in return Line to tank
P_S	=	Supply pressure from hydraulic pump
Q_A	=	Oil flow at servo valve at servo valve port A
Q_B	=	Oil flow at servo valve at servo valve port B
Q_L	=	Total flow through the load
Q_P	=	Maximum oil flow capacity of the pump
Q_r	=	Rated flow of servo valve at 70 bar pressure drop
R_C	=	Series resistance of the Torque motor circuit
U_e	=	Error output from summing Amplifier
U_p	=	Feedback signal from displacement transducer

U_v	=	Command signal to servo valve
V_A	=	Volume of trapped oil in chamber A of the Cylinder
V_B	=	Volume of trapped oil in chamber B of the cylinder
V_t	=	Volume of trapped between pump and servo valve
X_p	=	Displacement of the piston relative to centre position

1. INTRODUCTION TO HELICOPTER MOTION CONTROL

A Helicopter is a Rotorcraft that derives its lift from one or more power driven rotors. (A rotor is a system of rotating aerofoil.). Helicopters offer a facility to move from one place to other, which are remotely located, and not having well laid out runway which are otherwise required for fixed wing planes. Helicopters serve various purposes from to civil transport, ambulance, police, and forest fire prevention to sophisticated military application.

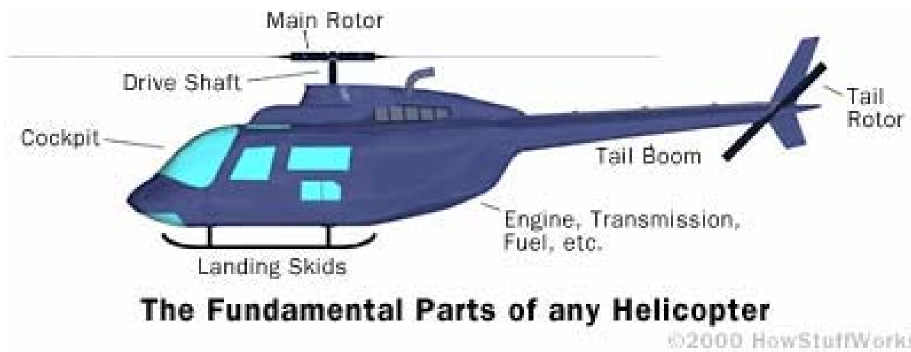

Figure 1.1

Generally one or two turbo shaft engines provide the power to the rotor system through a speed reduction gearbox to run the rotor at a constant speed. A rotor at the tail with a moment arm provides anti torque to the main body of the helicopter called the Fuselage. Accessory driven gear box provides facility for Hydraulic pump, electrical generator and other utility power outlets at various speeds. Modern helicopters utilize digital display and communication for information transfer from one system to other and there by extracting the optimum performance out of the flight vehicle.

Helicopters have two type of flight control mechanism to control the pitch angle of four main Rotor blades.

1. Differential control (Pitch & Roll)
2. Collective pitch control
3. Yaw control

In Differential control opposite blades being tilted in opposite direction.i.e. clockwise & anticlockwise. Hence Differential lift force creates the couple, which in turn tilts the plane of rotation. This changes the direction of resultant force which results turn of Helicopter itself. Pitch & Roll controls are achieved through this mode only.

In collective control, all the four blades are tilted n the same direction. i.e. pitch angle of all the blades being changed uniformly. The result is Change in the lift force. There is no change in the direction of force. Only magnitude gets changed.

Yaw control requires, tail rotor blade pitch control. By increasing the pitch angle, the thrust force produced by the tail rotor blades is increased which in turn creates a moment and hence helicopter turns in the lateral direction.

Three types of flight Control:

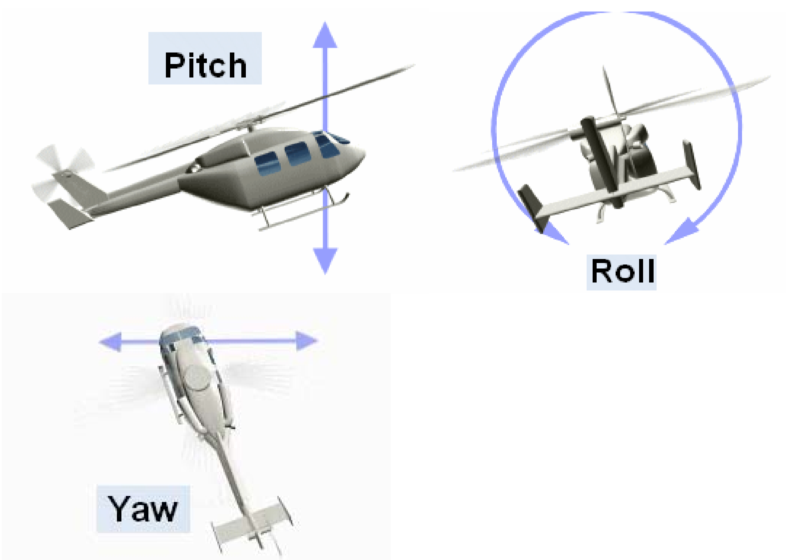

Figure 1.2

Differential Control

Figure 1.3

Collective control

Collective Control

Figure 1.4

1.1 COMPARATIVE STUDY BETWEEN FIXED WING (AEROPLANE) & ROTARY WING(HELICOPTER)

Sl. No.	Characteristics	Fixed Wing Aircraft	Rotary Wing Aircraft
1	Lifting device	Fixed Wing	Rotating wings (or blades)
2	Flight	Controlled by changing the angle of ailerons, elevators and rudders.	Controlled by varying the pitch of the rotating wings.
3	Controlling the aircraft motion	Easy	Difficult as more complex parts are used.
4	Condition before obtaining lift	Certain linear velocity should be reached	Certain rotational speed of the blades has to be obtained.
5	Maximum forward velocity reached	Very high	Comparatively low.
6	Power generated	By jet engine	By gas turbines.
7	Hovering	Not possible normally.	Possible, also sideward and rearward flights possible.
8	Area required for landing and take off	Large	Less
9	Operation in remote and difficult areas	Not possible.	Possible
10	Handling qualities	Better	Prone to vibration & noise. Unstable.

Table 1.1

1.2 ROLE OF HYDRAULICS IN HELICOPTER FLIGHT CONTROL

Nowadays Helicopter flight control in the most common configurations is realized by collective and cyclic variation of the angle of attack of each rotor blade. The Collective blade control pitches the rotor blades to equal angles of attack around their longitudinal axis, changing the rotor thrust at constant rotor speed. Yaw and roll control is realized via cycle blade motion by changing the angle of attack of every rotor blade locally and periodically during one revolution.

The control of the rotor blade angles for small helicopters is done manually by the pilot with the help of push-pull rods .But for a bigger helicopter the aerodynamic forces acting on the rotor blade are so high that it becomes impossible to control the rotor blade angles manually for the pilot. So for medium and large helicopters the control of the rotor blade angles is done with the help of hydraulic Actuators. These hydraulic actuators help in controlling the roll, pitch and collective Movements of the helicopter.

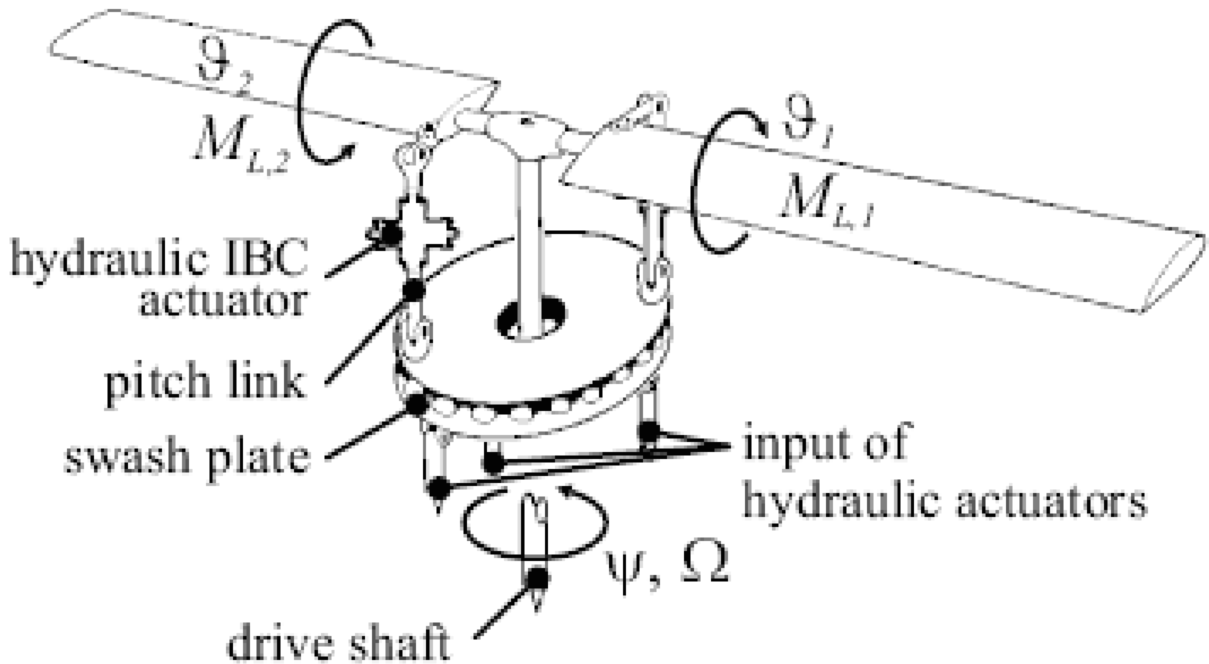

Figure 1.5 Helicopter flight Control system

2. INTRODUCTION

Many mobile, airborne and stationary applications employ hydraulic control components and servo systems. Hydraulic servo systems can generate very high forces, exhibit rapid responses, and have a high power-to -Weight ratio compared to other technologies. On the other hand, they exhibit a significant nonlinear behavior due to the nonlinear flow/pressure characteristics, oil compressibility, time varying behavior, nonlinear transmission effects, flow forces acting on spool and friction, which is not only largely uncertain but is greatly influenced by external load disturbances.

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 molding machines, robotics, and steel and aluminum 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.

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.

Aerodynamic control surfaces are deployed in helicopters for generating control forces and moments. In general, the commands issued for this purpose activate an actuation system, which carries out the task of control surface deployment through a series of actions. In case of electro-hydraulic actuation system1, hydraulic power, in conjunction with a servo valve, is used to generate the requisite forces and the motion.

The desired motion is achieved through a closed loop feedback control system that senses the actual deflection and corrects it until the desired position is reached. In recent times, there has been a trend towards designing higher agility aerospace vehicles resulting in larger bandwidths, as well as higher actuation rates, of actuation systems.

Electro hydraulic Servo System (EHSS) is a closed loop control system, which is usually applied as an actuator unit to drive an object such as a rudder or vane. Depending on variable to be controlled, it can be a position, velocity or force control system. Electro hydraulic servo systems have the advantages of, precise and fine control, high power to weight ratio, small size, good load matching, high environmental stiffness, fast dynamic performances and wide adjustable speed range. Large inertia and torque loads can be handled with high accuracy and very rapid response. All these advantages are suitable for aerospace and missile applications.

2.1 HYDRAULIC SYSTEM DESCRIPTION

The following subsections comprise a brief description of the Principal Hydraulic Elements that make up a typical position controlled system.

1) Hydraulic Power Supply

2) Flow Control Value

3) Linear Hydraulic Actuator

4) Displacement Transducer

5) Servo Controller

2.1.1Hydraulic 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 bars) 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.

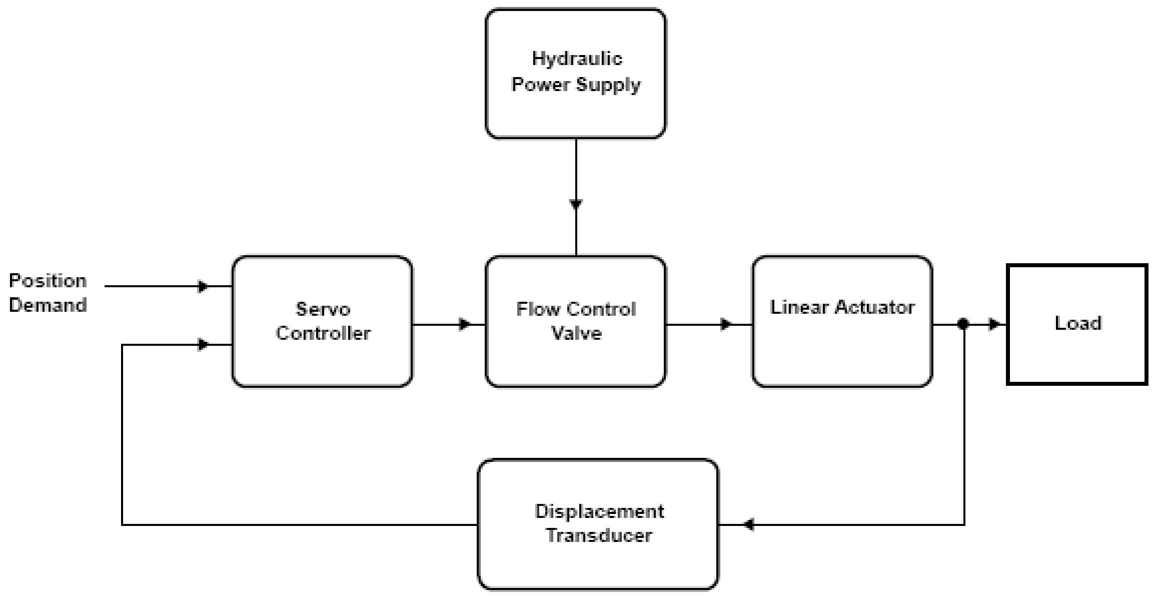

Figure 2.1 Block diagram of a position controlled Hydraulic Servo system

2.1.2 Hydraulic Supply Pressure Selection:

One of the first steps in design of a hydraulic control system is to select supply pressure. Many considerations favor a large supply pressure. Power is the product of pressure and flow. As supply pressure is increased, less flow is required to provide a given power. Smaller Pump, lines, valves, oil supply etc are then possible. Faster response is often possible because of small oil volume and higher bulk modulus. Thus high pressure in EHSS improves power to weight ratio and reduce size of the components and overall weight. The development of aerospace, aviation and missiles technology require, light weight, small volume, high pressure, high power and multi redundant and intelligent control.

2.1.3 Flow Control Valve:

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.

The vast majority of flow control servo valves in existence employ a double flapper nozzle pilot stage and a single spool boost stage. A stiff feedback spring is generally used to provide feedback from the boost stage to the pilot. These types of servo valves tend to be difficult to manufacture and expensive. A less conventional, less costly type of flow control servo valve utilizes a two-spool boost stage and a flapper nozzle pressure control pilot. Because a feedback wire between the nozzle flapper pilot and the boost stage is not needed, assembly is simplified. The two spools in the boost stage are spring-loaded and meter flow into and out of the valve separately. The main advantages of the two-spool/ pressure control pilot designs are 1) ease of manufacturing 2) lower costs 3) higher degree of adjustment; and 4) Greater safety.

Figure 2.2 shows the spool configuration of a typical "3-4" flow control valve. The ports are labeled P (pressure), T (tank), and A and B (load control ports). The spool is shown displaced a small distance (xv) 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 (Fv=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.

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.

A cross sectional view of a typical nozzle-flapper type servo-valve is shown in figure 2.3. 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.

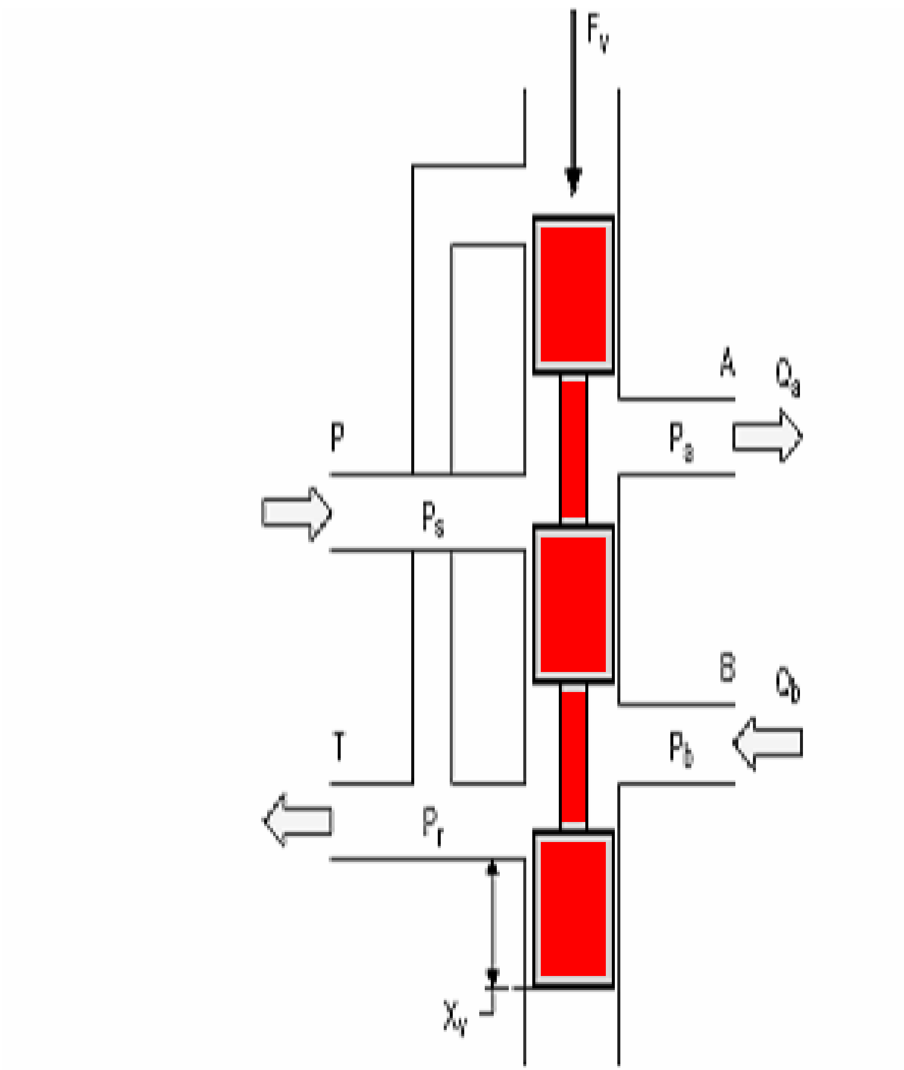

Figure 2.2 Diagram of Three Land, Four Way Flow Control Valve

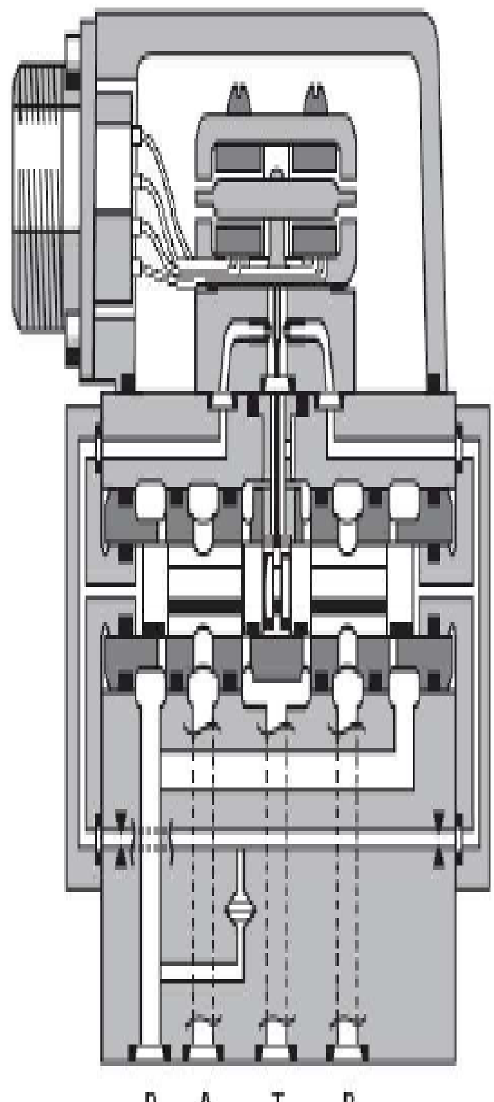

Figure 2.3 showing Cross Section of Nozzle Flapper Type Servo valve

2.1.4 Linear Hydraulic Actuator:

Linear actuators are the devices for converting fluid power into linear motion. They may be used to exert a force, to hold or clamp, and to initiate or stop motion. All linear actuators are some modification of an air or hydraulic cylinder and may be either single or double acting .the single acting cylinder receives power at one end only and is returned to its original position by gravity or by spring action, while double acting cylinder is powered in both directions. Double-acting cylinders permit more complete control of movement. Ram is a form of single acting cylinder in which the piston rods are of the same diameter.

The double rod cylinder has a rod attached to both sides of the piston. This type of cylinder is center-mounted and is normally used when the same task is performed at the either end on staggered cycles. Obviously the force and speed will be the same at either end.

2.2. Selection of Hydraulic Actuator

Actuator is the key component of hydraulic servo system.

1) Its size should be large enough not only to handle the loads expected during duty cycle but also ensure the required load velocity. It is also important not to oversize actuators so that the flow required for maximum velocity is kept to a minimum. Otherwise, the hydraulic power supply becomes bulky with large no load power losses.
2) The load should be properly matched to the output of the system. Load matching effectively utilizes the output power of hydraulic source and improves performance of the hydraulic system.

3) Other important Actuator performance characteristics, which might be important, are speed, range, stiffness of the smoothness (i.e., absence of velocity variations at operating speeds).
4) Reliability
5) Backlash
6) Pressure rating
7) Cost

2.3 DESCRIPTION OF THE ACTUATOR:

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 2.4. 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.

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.

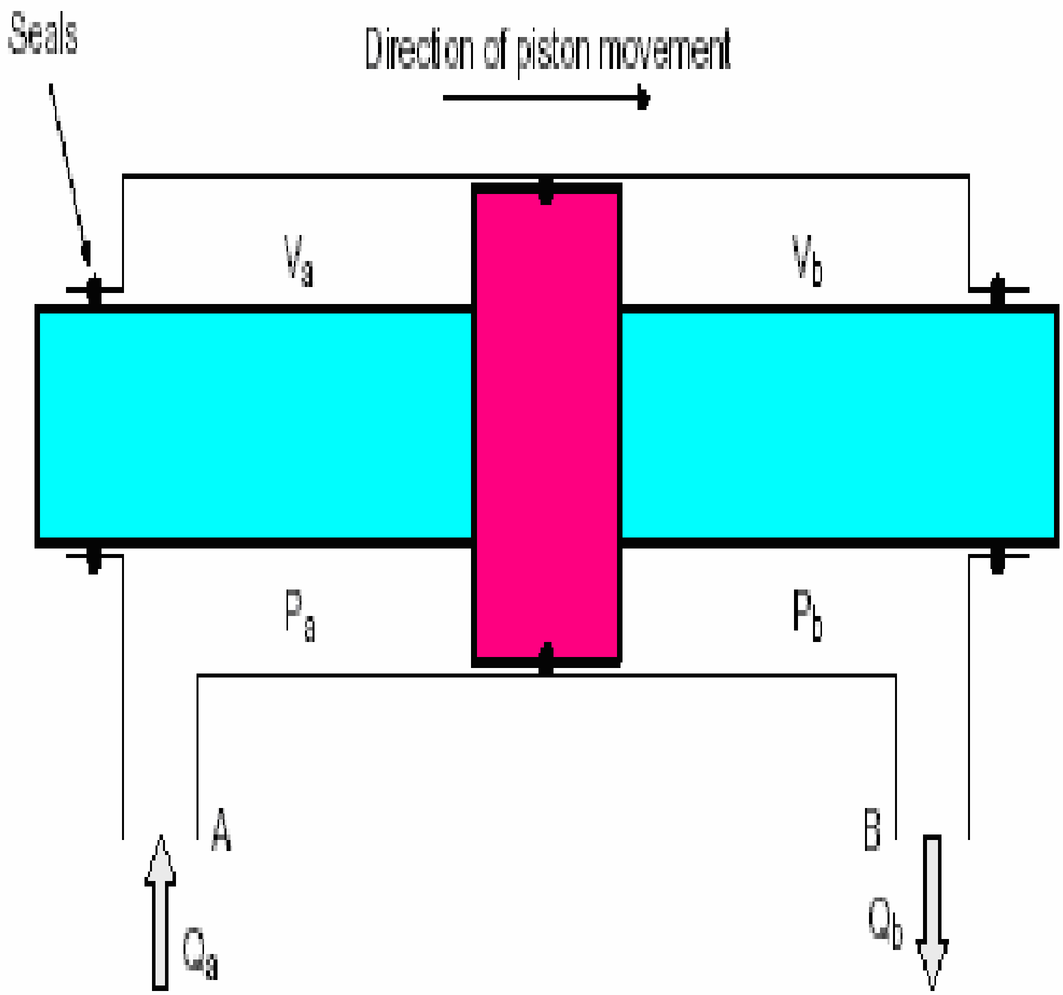

Figure 2.4 showing Cross-sectional Diagram of Double-ended, Double-acting Linear Actuator

2.4 DEFINITIONS

Servomechanism - A continuously acting, bidirectional closed-loop control system.

Servo valve - A device used to produce hydraulic control in a servomechanism.

Electro hydraulic Servo valve - A servo valve which produces hydraulic control in response to electrical signal inputs; sometimes called a transfer valve.

Electro hydraulic Flow Control Servo valve - A servo valve designed to produce hydraulic flow output proportional to electrical current input.

2.4.1 VALVE NOMENCLATURE

1. Hydraulic Amplifier -A fluid valving device which acts as a power amplifier, such as a sliding spool, or a nozzle flapper, or a jet pipe with receivers.
2. Stage - The portion of a servo valve which includes a hydraulic amplifier. Servo valves may be single stage, two stage, three stage, etc.
3. Output Stage -The final stage of hydraulic amplification used in a servo valve, usually a sliding spool.
4. Port -A fluid connection to the servo valve; for example, a supply port, a return port, or control port
5. Three-Way Valve - A multi-orifice fluid control element with supply, return and one control port arranged so that valve action in one direction opens the control port to supply and reversed valve action opens the control port to return.
6. Four-Way Valve - A multi-orifice fluid control element with supply, return and two control ports arranged so that valve action in one direction simultaneously opens control port #1 to supply and control port #2 to return. Reversed valve action opens control port #1 to return and control port #2 to supply.

2.4.2 ELECTRICAL INPUT CHARACTERISTICS

1. Torque Motor - The electromechanical transducer commonly used with the input stage of a servo valve. Displacement of the armature of the torque motor is generally limited to a few thousandths of an inch.
2. Input Current -The current which is required for control of the valve, expressed in ma. For three and four lead coils, input current is generally the differential coil current, expressed in ma.
3. Rated Current - The specified input current of either polarity to produce rated flow,
4. Current must be associated with a specific coil connection.

2.4.3 STATIC PERFORMANCE CHARACTERISTICS

1. Control Flow, also called Load Flow or Flow Output the fluid flow passing through the valve control ports, expressed in cis or gpm. In testing a four-way servo valve, flow passing out one control port is assumed equal to the flow passing in the other. This assumption is valid for no-load valve testing with a symmetrical load
2. Rated Flow - The specified control flow corresponding to rated current and specified load pressure drop, expressed in cis or gpm. Rated flow is normally specified as the no-load flow.
3. No-Load Flow - The servo valve control flow with zero load pressure drop, expressed in cis or gpm.
4. Power Output -The fluid power which is delivered to the load, expressed in hp.
5. Flow Saturation - The condition where flow gain decreases with increasing input current. Flow saturation may be deliberately introduced by mechanical limiting of the valve range, or may be the result of increasing pressure drops along internal fluid passages.
6. Flow Limit - The condition wherein control flow no longer increases with increasing input current.

2.5. ABOUT SIMULINK

Simulink is a software package for modeling, simulating, and analyzing dynamical systems. It supports linear and nonlinear systems, modeled in continuous time, sampled time, or a hybrid of the two. Systems can also be multi rate, i.e., have different parts that are sampled or updated at different rates.

For modeling, Simulink provides a graphical user interface (GUI) for building models as block diagrams, using click-and-drag mouse operations. With this interface, we can draw the models just as we would with pencil and paper. This is a far cry from previous simulation packages that require us to formulate differential equations and difference equations in a language or program. Simulink includes a comprehensive block library of sinks, sources, linear and nonlinear components, and connectors. We can also customize and create our own blocks.

Models are hierarchical, so we can build models using both top-down and bottom-up approaches. We can view the system at a high level, and then double-click on blocks to go down through the levels to see increasing levels of model detail. This approach provides insight into how a model is organized and how its parts interact.

After we define a model, we can simulate it, using a choice of integration methods, either from the Simulink menus or by entering commands in MATLAB's command window. The menus are particularly convenient for interactive work, while the command-line approach is very useful for running a batch of simulations. Using scopes and other display blocks, we can see the simulation results while the simulation is running. In addition, we can change parameters and immediately see what happens, for "what if" exploration. The simulation results can be put in the MATLAB workspace for post processing and visualization.

Model analysis tools include linearization and trimming tools, which can be accessed from the MATLAB command line, plus the many tools in MATLAB and its application tool boxes. And because MATLAB and Simulink are integrated, we can simulate, analyze, and revise our models in either environment at any point.

We can build models from scratch, or take an existing model and add to it. Simulations are interactive, so we can change parameters and immediately see what happens. We can have instant access to all of the analysis tools in MATLAB, so we can take the results and analyze and visualize them.

With Simulink, we can move beyond idealized linear models to explore more realistic nonlinear models, factoring in friction, air resistance, gear slippage, hard stops, and the other things that describe real-world phenomena. It turns our computer into a lab for modeling and analyzing systems that simply wouldn't be possible or practical otherwise, whether the behavior of an automotive clutch system, the flutter of an airplane wing, the dynamics of a predator-prey model.

3. LITERATURE SURVEY

3.1 INTRODUCTION

Until now a lot of work has been done on control, operation and testing of hydraulic systems. With the evolution of computer simulation techniques, this process has become much simpler. There are many reports describing field experience related to analyzing hydraulic actuators using Simulink software. In this chapter works published in a wide spectrum of journals have analyzed and the goals for the present study have been given a firm foundation with the information derived from the survey. They are presented in the subsequent sections.

3.2 SIMULATION OF HYDRAULIC ACTUATORS

The mathematical model for the hydraulic System is made with the help of system characteristics and its behavior. With the help of these mathematical models various hydraulic systems have been analyzed by using softwares like MATLAB, SIMULINK, and SIMULATIONX etc.

Pramod[1] studied the effect of non-linearities in the configuration design of Digital Auto Pilot (DAP) in launch vehicles. An electro hydraulic actuator model of a launch vehicle control system is considered for analysis of non-linearities. Various non-linear effects like saturation (in current and stroke limit), dead zone and coulomb friction are taken into account. DAP, which is an interface between the guidance system and control system, is designed to cater to the model (linear/ non-linear) adopted for the actuator. In the actuator alone case, without considering the total flight regime and vehicle model, the performance is found to be satisfactory for linear as well as non-linear actuator models. In the actuator–vehicle combination, when the simulation is carried out for the total flight regime considering the vehicle model, the performance of the linear / nonlinear actuator model is dependent on DAP configuration this study brings out the fact that the DAP configuration is specific to the actuator model, so that satisfactory performance of launch vehicle control system can be ensured only by choosing proper configuration for DAP, based on consideration of non-linearities in actuator model.

Evangelos Papadopoulos[2] presented an optimal hydraulic component selection for electro hydraulic systems used in high performance servo tasks. Dynamic models of low complexity are proposed that describe the salient dynamics of basic electro hydraulic equipment. Rigid body equations of motion, the hydraulic dynamics and typical trajectory inputs are used in conjunction with optimization techniques, to yield an optimal hydraulic servo system design with respect to a number of criteria such as cost, weight or power. The optimization procedure employs component databases with real industrial data, resulting in realizable designs.

Edson Roberto [3] studied the problem of experimental control of hydraulic actuators is considered. To deal with mechanical and hydraulic uncertainties a different controller is synthesized: a back stepping controller. Experimental results of both implementations are analyzed in the context of practical difficulties, mainly the measurement of acceleration. These results illustrate the main features of these controllers when applied on a hydraulic actuator.

Kexiangwei [4] developed a fluid power control unit using electro rheological fluids. Electro Rheological (ER) fluids can change their rheological properties when subjected to an electrical field. By using ER fluids as the working medium in fluid power systems, direct interface can be realized between electric signals and fluid power without the need for mechanical moving parts in fluid control unit. The pressure drop and flow rate can be directly controlled through the change of applied electric fields. This paper investigates the design and controllability of ER fluid power control system for large flows. The design criterion for an ER valve is proposed and four ER valves are manufactured based on this criterion. A fluid control unit consisting of an ER valves bridge circuit is constructed, the characteristics of which are theoretically and experimentally investigated. The results show that the ER fluid control units have better controllability for fluid power control.

Holger Berndt [5] presented an interactive design and simulation platform for flight vehicle systems development. Its “connect-and-play” capability and adaptability enable “on-line” interaction between design and simulation during the integrated development. As a case study, the implementation of the proposed platform and an aircraft flight control system development example are demonstrated on an experimental test bed including a real time Systems simulator.

P. G. Jayan[6] in his paper simulation work for a typical fighter aircraft’s flight control system was carried out using MATLAB© as basic platform. Altitude, Mach number, angle of attack, elevator command and rudder command are the inputs for the simulation. These are obtained from tests conducted on hydraulic system test rig. Simulation results are verified with tests conducted on test rig.

Anderson [7] in his paper presented a nonlinear dynamic model for an unconventional, commercially available electro hydraulic flow control servo valve is presented. The two stage valve differs from the conventional servo valve design in that: it uses a pressure control pilot stage; the boost stage uses two spools, instead of a single spool, to meter flow into and out of the valve separately; and it does not require a feedback wire and ball. Consequently, the valve is significantly less expensive. The proposed model captures the nonlinear and dynamic effects. The model has been coded in Matlab/Simulink and experimentally validated.

Ashok Joshi [8] in his papers presented the effects of servo valve nonlinearity, actuation compliance and friction related nonlinearity on the dynamics of a flight control surface, during its deployment through an electro-hydraulic actuation system. Starting from the pilot command, a realistic model of the electro-hydraulic actuation system is evolved, which includes the command lags, servo valve nonlinearity, actuation chain compliance and friction nonlinearity. A realistic mathematical model for the control surface motion, under the action of the actuator forces and the aerodynamic and inertia forces is postulated, using subsonic incompressible aerodynamics.

Peter Rowland, M. Longvitt, Keith Austin and Irfan Bhatti [9] this paper describes about modular design approach for modeling of large and complex hydraulic systems. Using this creation and analysis of large hydraulic models can be avoided. It will reduce run time, editing and results can be manages easily. Each complex model is divided into small systems and each system was modeled using standard pressure and flow source models as boundary conditions. Later subsystem could be linked together the boundary condition models removed and the desired analyses completed. For accurate simulation of landing gear model interaction between hydraulic and mechanical systems is required. This allows better modeling of both gear deployment time and pressure time history in hydraulic system.

Sreeraj P.N.[10] in this simulation work for typical aircraft's steering system and wheel brake system is carried out using SIMULINK of MATLAB© software. By taking the wheel slip ratio, torque exerted on the wheel, hydraulic amplifier flow gain, natural frequency of armature as inputs simulation of wheel speed and stopping distance, steering angle, rack position was carried out and these results are validated with the specified test results.

Panagiotis [11] in his technical paper presented a model-based controller applied to a fully detailed model of an electro hydraulic servo system aiming at improving its position and force tracking performance. Fluid, servo valve, cylinder and load dynamics are taken into account. Simulation results show the strategy to be promising in controlling hydraulic servo actuators. It also compares its position tracking performance to that of a classical linear controller, using intensive simulations.

Ing T. Hong, Richard K.Tessmann [12] Response time is the time gap between input and output commands. The Authors describe the importance of dynamic analysis for calculating system response and importance of it for hydraulic systems. A simple case study of servo control valve is taken and its response time is calculated.

Paul J.Heney[13] describes about challenges in aircraft hydraulic system compared to industrial and mobile hydraulics. Aircraft hydraulic system will be operating at higher pressures compared to many industrial applications. So, designing high pressure reliable system is challenging. Selection of hydraulic fluid is difficult because it should be able to operate in wider range of temperatures and leakage is also main concern in selecting the fluid. To increase the reliability of hydraulic system redundancy should be maintained. Majority of aircrafts will have three or four redundant hydraulic systems, which are geographically separate in many cases.

Ming Yuan –Tsuei[14] this describes about application of computer programs for system design and analysis. It also talks about features of HyPneu© software and about the two sections of it HPCAD and HPMGR. Different case studies are taken and these are simulated with HyPneu©.

P.Krus, A.Jansson and J.O.Palmberg[15] this paper describes about use of computer simulation for optimization. Optimizing total number of parameters of all components in a system is too large to be handled by numerical computation. A new approach is adopted here by introducing performance parameters which uniquely define the components. In aircraft design it is very important that system is optimized with respect to different aspects such as performance and weight. Using an optimization strategy and a simulation model of the system, it is possible to use a computer to optimize the system globally once the system layout is established.

Joseph N.Demarchi and John Ohlson[16] this paper describes about development of 8000 psi aircraft light weight hydraulic systems as compared to the present 3000 psi system. Use of high operating pressures for aircraft hydraulic system provides significant reduction in both weight and volume. Computer simulation of these systems was carried out to determine effect of dynamic stability of a flight control actuator system with reference to elevated hydraulic pressure. Later actual hardware was designed and tested.

4. MATHEMATICAL MODELING OF THE HYDRAULIC SYSTEM

Mathematical models are developed for various components of the hydraulic system in this chapter. Mathematical modeling involves in representing the hydraulic system components in the form of equations. These mathematical models help in representing the hydraulics system components in Simulink Software. This mathematical modeling is done by considering the component properties such as flow properties, functional properties, characteristics of the component (like electrical characteristics etc).

4.1. MATHEMATICAL MODELING OF FLOW CONTROL SERVO VALVE:

The flow control valve considered in this case study is a two-stage nozzle flapper servo valve. It consists of the following elements

1. Electrical torque motor
2. Hydraulic amplifier
3. Valve spool assembly

4.1.1 Torque motor: 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 4.1. 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.

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 centered and the spool stationary but deflected to one side. The offset position of the spool opens flow paths between the pressure and tank ports (Ps and T), and the two control ports (A and B), allowing oil to flow to and from the actuator.

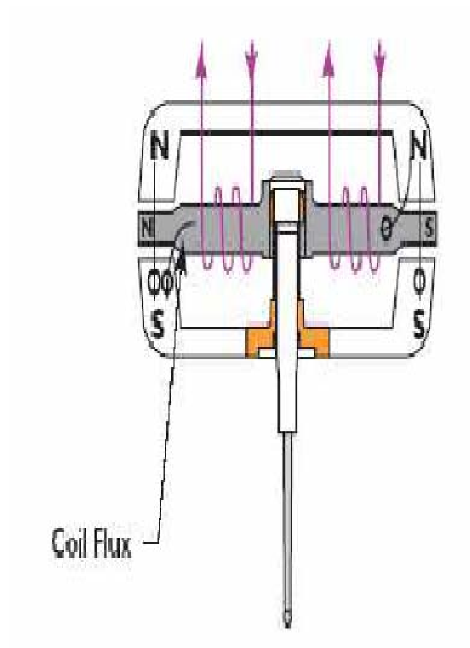

Figure 4.1 Valve Torque motor Assembly

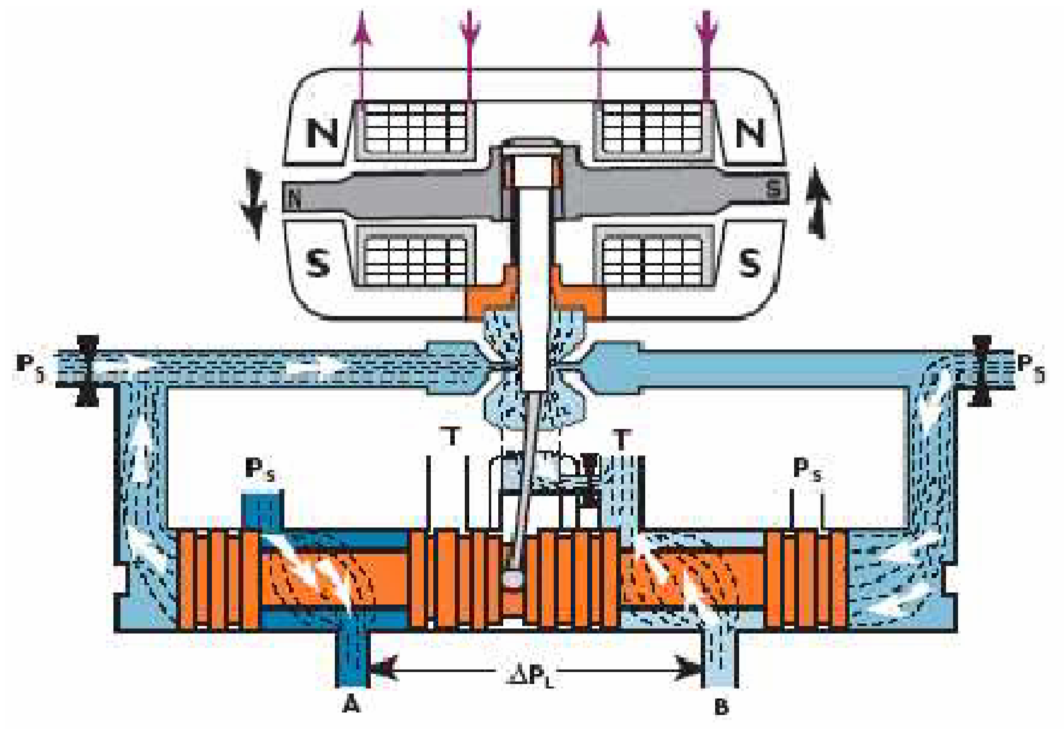

Figure 4.2 Valve responding to change in Electrical input

By considering the electrical characteristics of the servo valve Torque motor the torque motor may be considered as a series Inductance (L) – Resistance(R) circuit.

Neglecting the back EMF generated by the load. The transfer function of a series L-R circuit is given by

$$\frac{I(s)}{V(s)} = \frac{1}{sLc + sRc} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(1)$$

Where Lc is the inductance of the motor coil,

Rc is the combined resistance of the motor coil and the current sense resistor of the servo amplifier.

The above values of inductance and resistance for series and parallel winding configurations of the motor are published in the manufacturer's data sheet.

Modelling 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

$$Q_L = Q_R \times i_{v*} \times x \sqrt{\frac{\Delta P_v}{\Delta P_R}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(2)$$

Where QL, is the hydraulic flow delivered through the load actuator

Q_R the rated valve flow at a specified pressure drop (ΔP_R)

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 $\Delta P = 1{,}000$ 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.

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 figure. 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 MODELLING 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.

$$\sum Q_{in} - \sum Q_{out} = \frac{dV}{dt} + \frac{V}{\beta}\frac{dP}{dt} \qquad \text{……………… (3)}$$

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 b. Mineral oils used in hydraulic control systems have a bulk modulus in the region of 1.4 x 109 N/m. Equation 3 can be re-arranged to find the instantaneous pressure in chamber A as follows:

$$P_A = \frac{\beta}{V}\int \left(Q_A - \frac{dV_A}{dt}\right) dt \qquad \text{..…………………… (4)}$$

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.

$$\mathbf{F_P = (P_A - P_B)A_P} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\mathbf{(5)}$$

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.

$$F_p = M_P \frac{d^2 x_p}{dt^2} + F_f + K_L x_p \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

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

$$F_f = \frac{dX}{dt} F_{V0} + \text{sign}\left(\frac{dx}{dt}\right) F_{C0} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

Where F_{V0} is the viscous friction Coefficient

F_{C0} is the coulomb friction coefficient

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

$$Q_A - K_{La}(P_A - P_B) - K_{Le} P_A = \frac{dV_A}{dt} + \frac{V_A}{\beta}\frac{dP_A}{dt} \qquad \text{……………..} \ (8)$$

Where KLa 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.

4.3 MODELLING OF HYDRAULIC POWER SUPPLY

The behavior of the hydraulic power supply described earlier 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

$$\mathbf{P_s} = \frac{\beta}{V_1}\int\left(Q_{pump} - Q_L\right)dt \qquad \text{………………………… } \mathbf{(9)}$$

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 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.

4.4 MODELLING OF SERVO 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).

$$\mathbf{U_e = U_r - U_p} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \textbf{(10)}$$

The error is manipulated by the servo controller according to a pre-defined control law to generate a command signal (Uv) 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

$$U_v\,(t) = K_p\, U_e\,(t) + K_i \int U_e\, dt + K_d \frac{dU_e}{dt} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \textbf{(11)}$$

Where K_p, K_i, and K_d are the PID constants, U_e is the error signal and Uv is the controller output.

5. NUMERICAL SIMULATION DATA USED IN THE PRESENT STUDY

5.1 ACTUATOR DATA

Mass of actuator piston	$M_p = 9$ Kg
Total stroke of the piston	$X_{P(max)} = 0.1$ m
Active area of the piston annulus	$A_P = 645\times 10^{-6}$ m^2

5.2 SERVO VALVE DATA

Rated flow of valve at 70 bar pressure drop	$Q_r = 0.63069\times10^{-3}$ m^3/s
Inductance of servo valve coil	$L_c = 0.59$H
Series resistance of torque motor circuit	$R_c = 100\ \Omega$
Saturation current of torque motor	$I_{v(sat)} = 0.02$ A

5.3 HYDRAULIC SYSTEM DATA

Bulk modulus of the hydraulic fluid	$B_e = 1.4\times10^{-9}$ N/m^2
Supply pressure from Hydraulic Pump	$P_s = 2.1\times 10^{7}$ Pa
Pressure drop in return line to tank	$P_R = 0$ Pa
Maximum oil flow capacity of the pump	$Q_P = 1.67\times10^{-3}$ m^3/s
Volume of the trapped oil between the Pump Servo Valve	$V_t = 0.0005$ m^3

6. SIMULINK MODELS

Simulink models have been made by utilizing the mathematical models of the subsystems. The Figure 6.1 represents the simulink model of top level diagram of the hydraulic system. A scope block is connected to monitor the time response of the hydraulic actuator. the connections to the various blocks in the model have been made by considering the equations obtained in chapter 4 mathematical modelling.

Figure 6.2 represents the simulink model of hydraulic actuator system. A scope block is connected to monitor the time response of the hydraulic actuator. The Connections to the various blocks in the model have been made by considering the Equations 3, 4, 5, 6,7and 8 which are obtained in chapter 4 mathematical modelling.

Figure 6.3 represents the simulink model of servo valve system. A scope block is connected to monitor the time response of the servo valve. The connections to the various blocks in the model have been made by considering the Equations 1 and 2 which are obtained in chapter 4 mathematical modelling.

Figure 6.4 represents the Simulink model of piston chamber 'A' of the actuator. A scope block is connected to monitor the Time response of the piston chamber 'A'. The Connections to the various blocks in the model have been made by considering the Equations 3 and 4 which are obtained in chapter 4 mathematical modelling.

Figure 6.5 represents the Simulink model of piston chamber 'B' of the actuator. A scope block is connected to monitor the time response of the piston chamber 'B'. The connections to the various blocks in the model have been made by considering the Equations 3 and 4 which are obtained in chapter 4 mathematical modelling. Simulink model of piston chamber 'B' of the actuator is much similar to the Simulink model of piston chamber 'A' of the actuator.

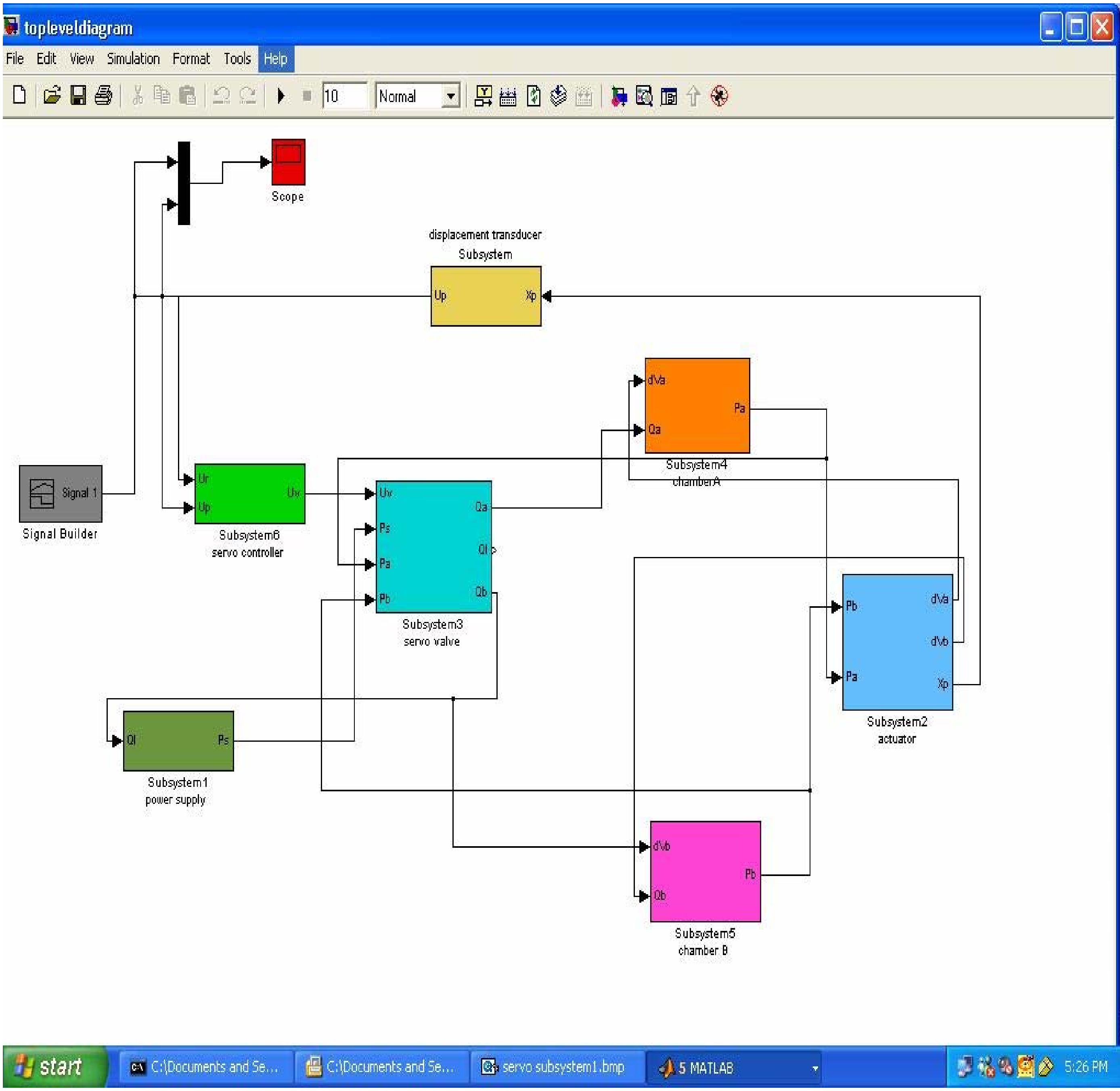

Figure 6.1 Simulink Model of Top level Hydraulic System

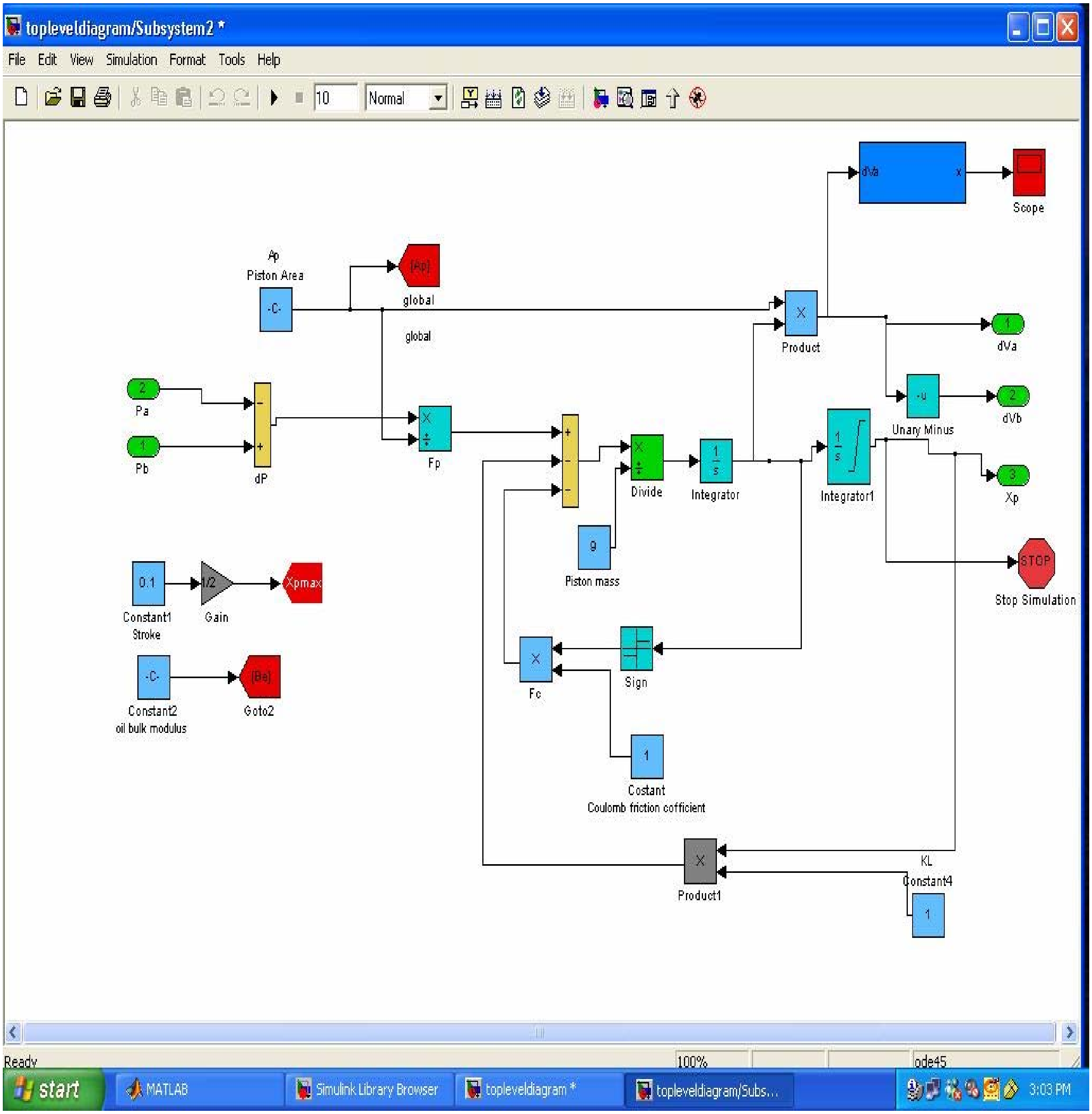

Figure 6.2 Simulink Model of Hydraulic Actuator

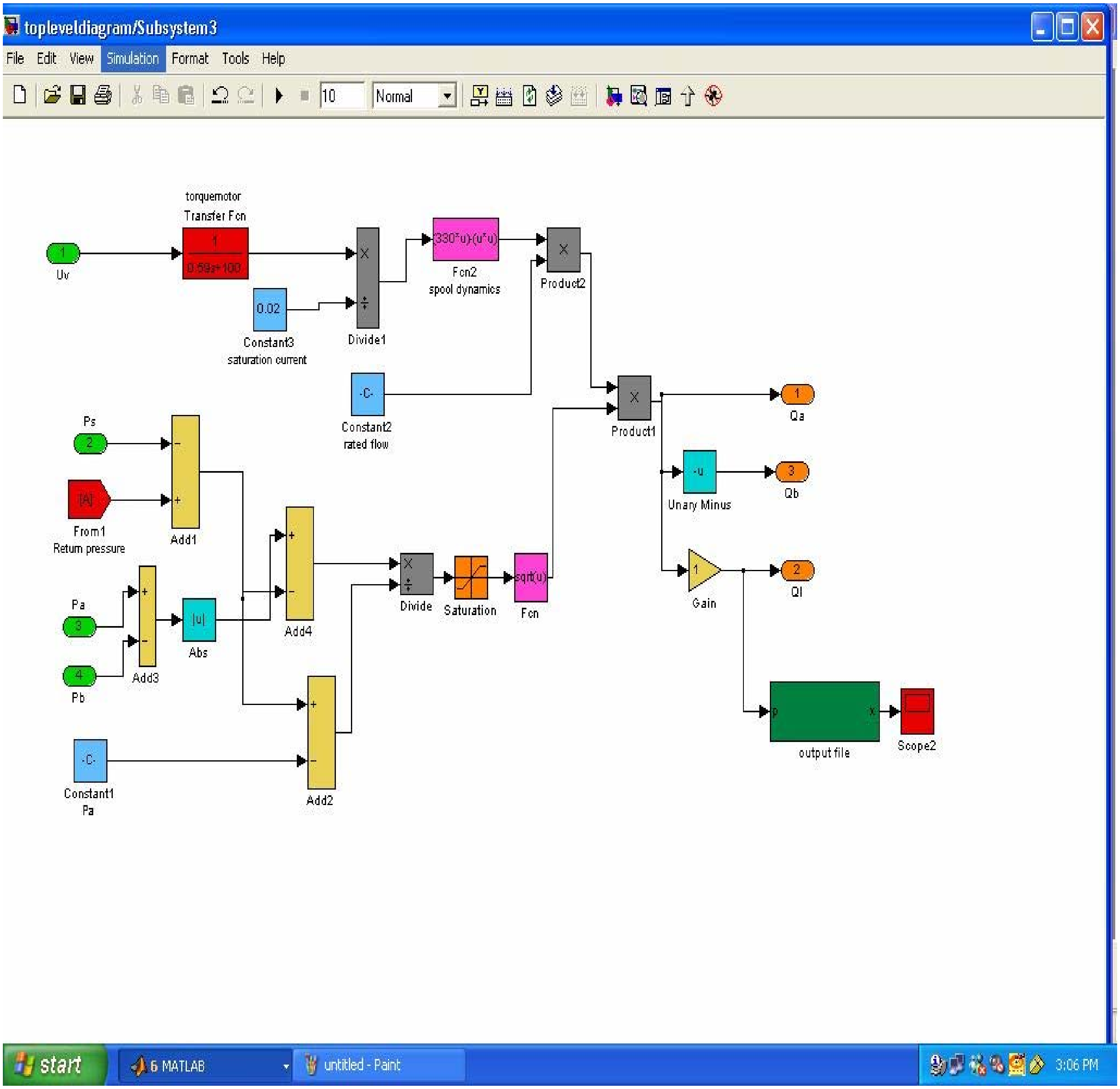

Figure 6.3 Simulink model of Servo Valve

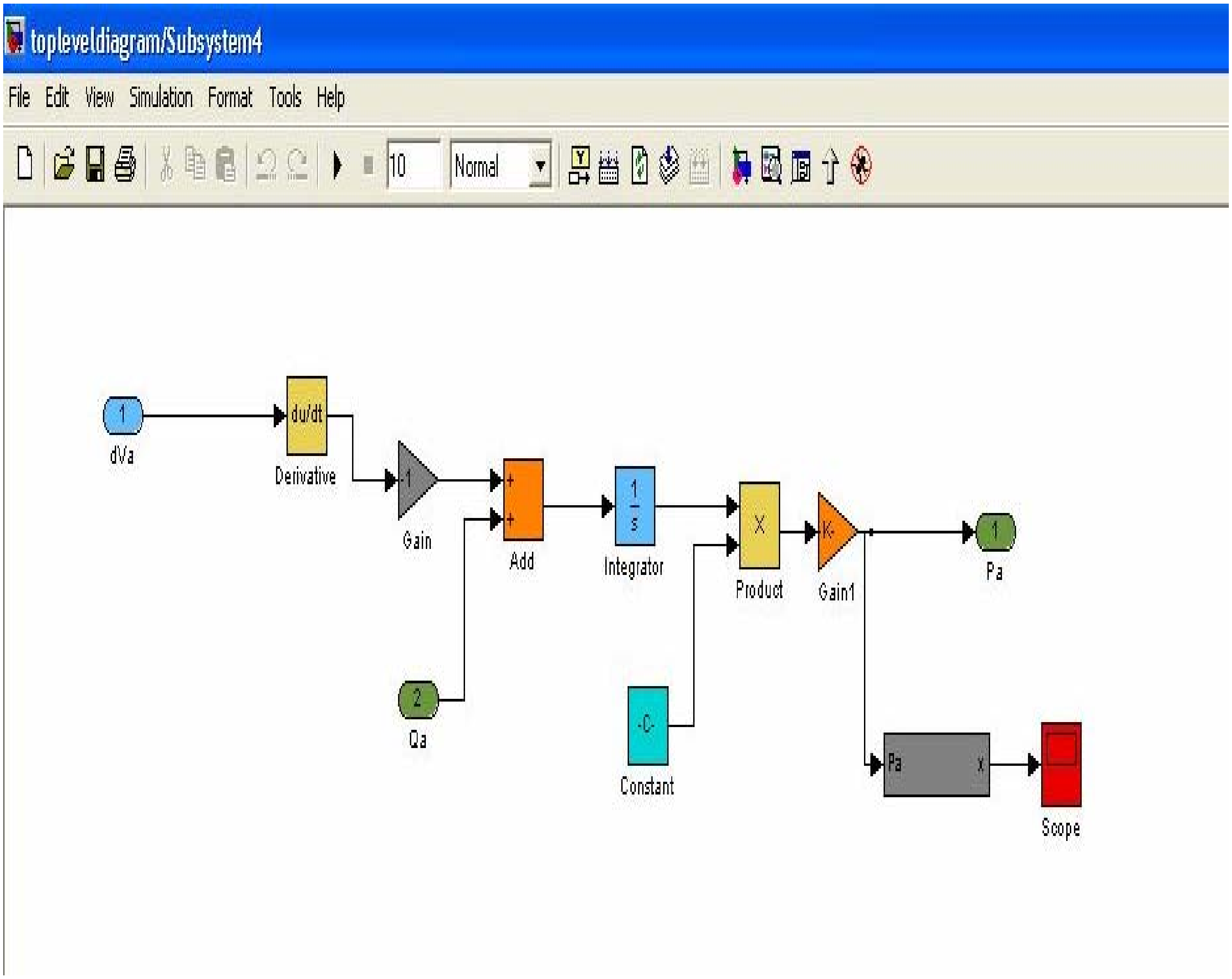

Figure 6.4 Simulink Model of Piston Chamber ‘A’ of the Actuator

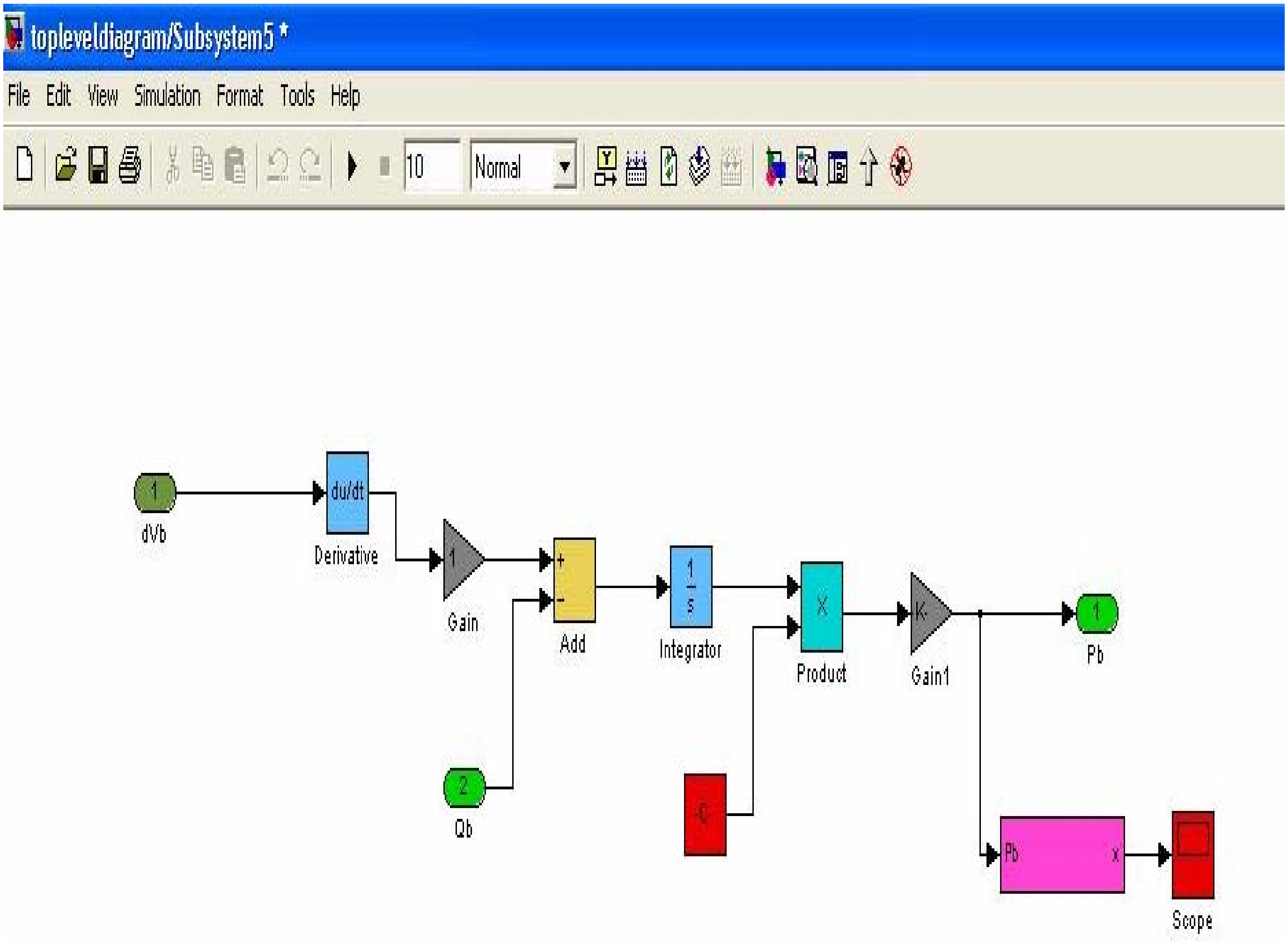

Figure 6.5 Simulink Model of Piston Chamber ‘B’ of the Actuator

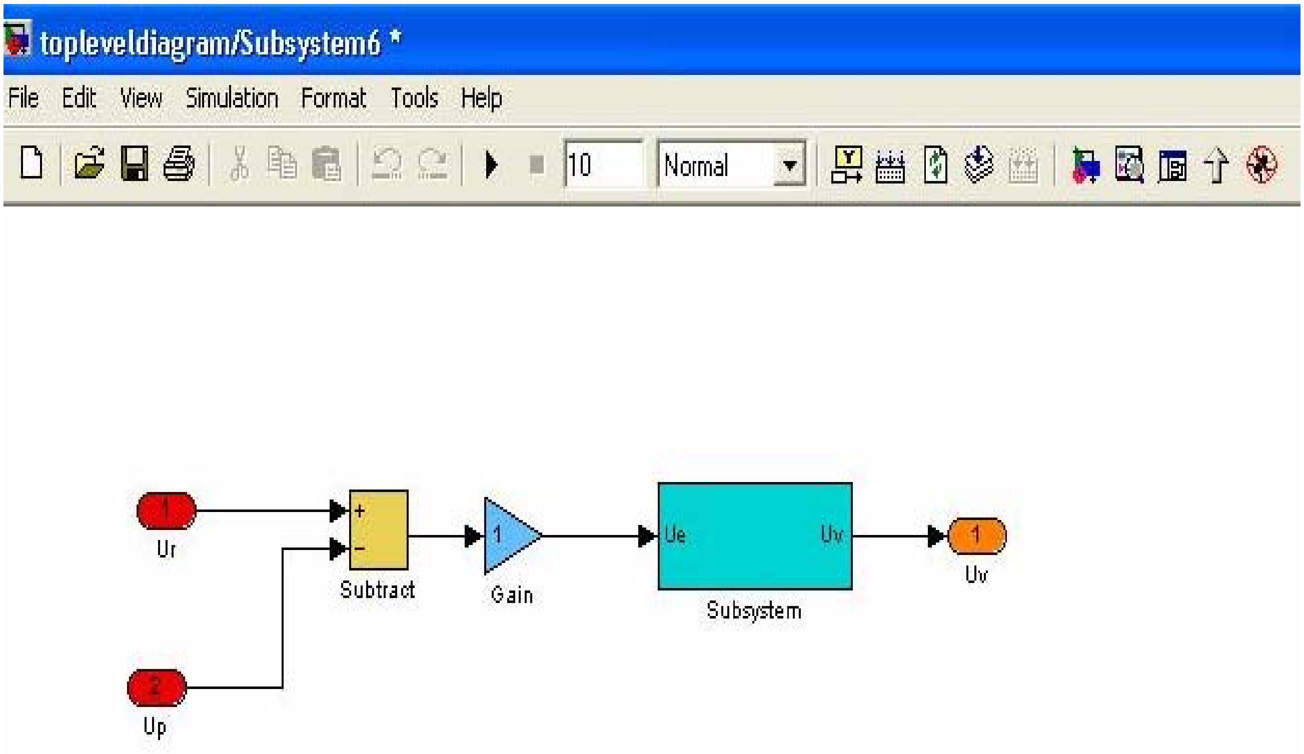

Figure 6.6 Simulink Model of Servo Controller

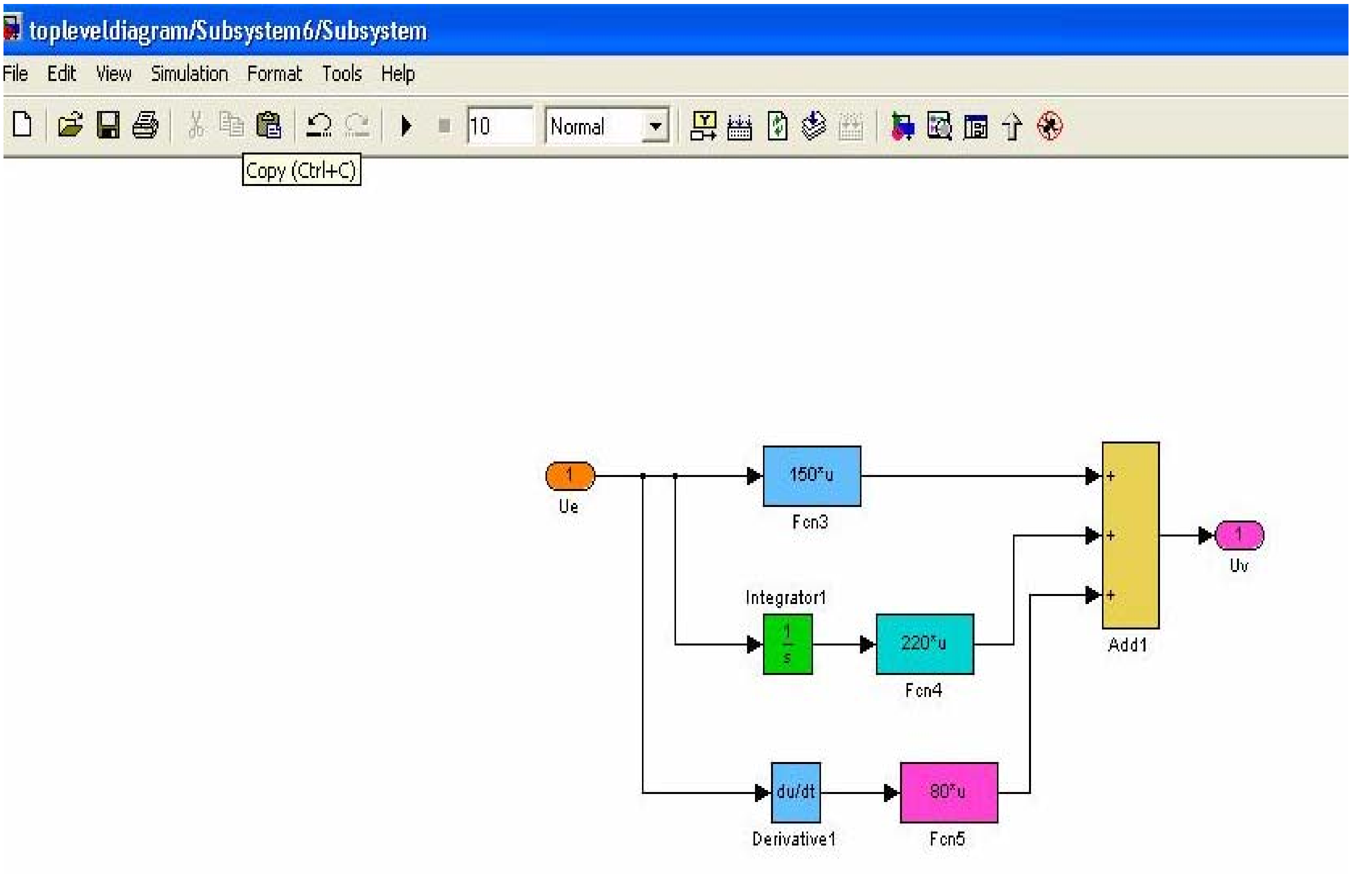

Figure 6.7 Simulink Model of Servo Controller Subsystem

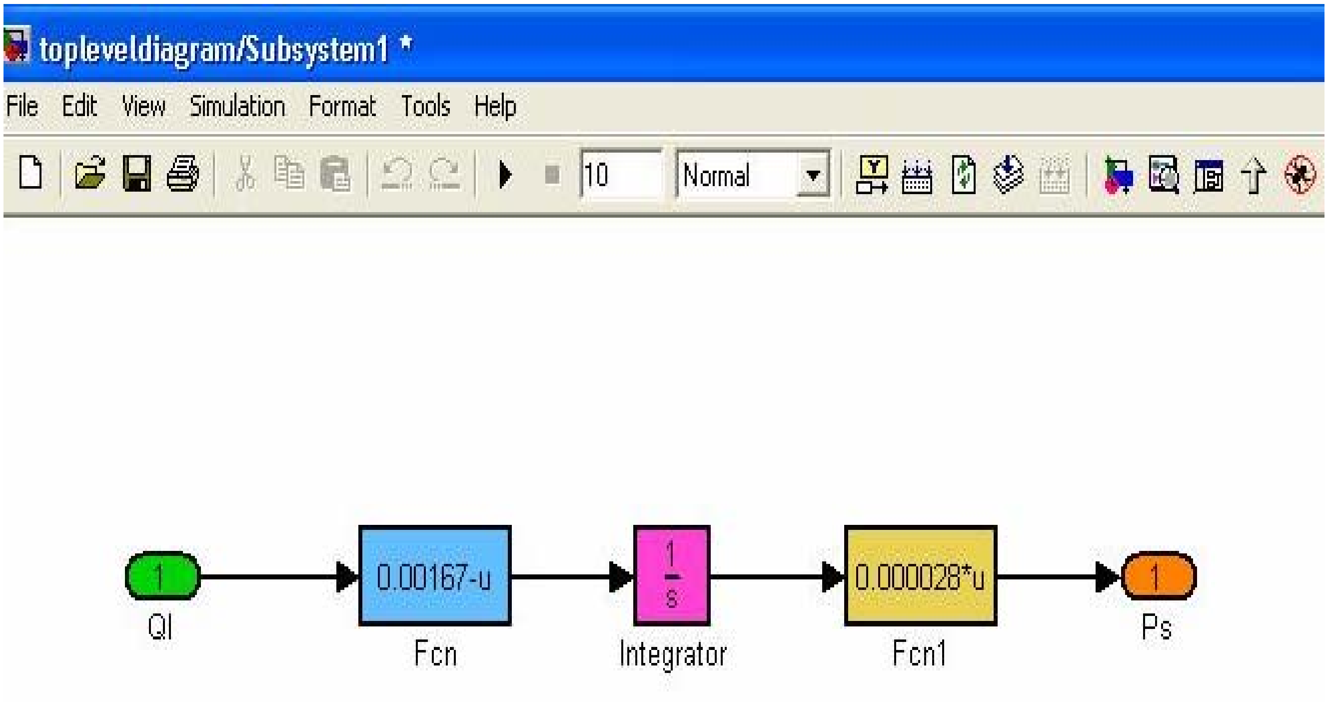

Figure 6.8 Simulink Model of Pressure Supply Subsystem

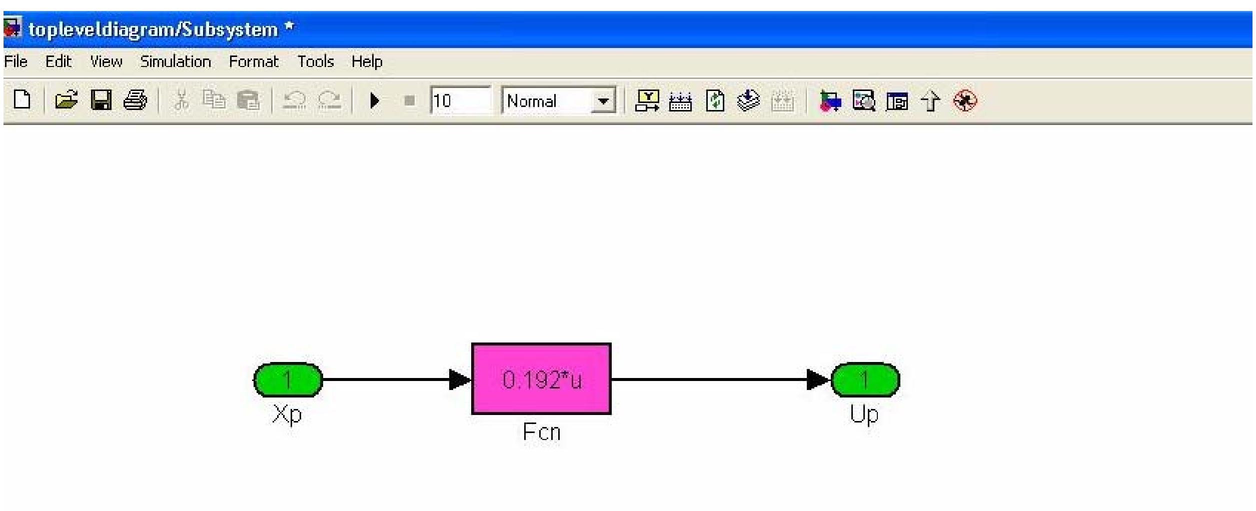

Figure 6.9 Simulink Model of LVDT

7. RESULTS AND DISCUSSION

In this case Study a step signal is given as input signal. The actuator that is considered here is a Moog electro-hydraulic actuator that is used for helicopter flight control. The time response of a particular system is obtained in MATLAB/SIMULINK software with the help of a scope block. The time response of a system is the behavior of the system with respect to time. The time response is a significant parameter for evaluating the system performance. Time response is a plot between time on X axis and Amplitude on Y axis. The time response of an actuator, which is used in helicopter flight control system, should be high for effective flight Control of the helicopter. As shown in Figures the models of the hydraulic actuator, servo valve, servo controller, piston chambers and pressure supply were made in MATLAB/SIMULINK software.

In the top level diagram actuator model shown in figure 6.1 a scope block is connected to obtain the time response of the system. Figure 7.1 shows the time response of the Actuator. The actuator system attains the maximum amplitude in 0.4 seconds (approx).The time response graph that is obtained in the scope represents a satisfactory compromise between rise time and overshoot. This time response graph, which is obtained, is compared with the experimental time response graphs of Moog electro-hydraulic actuator, which are shown in figure 7.6, and both the graphs were found to be in good compromise.

In the chamber 'A' model which is shown in figure 6.4 a scope block is connected to obtain the time response of the system Figure 7.2 shows the time response of the actuator. In the simulation time response graph of the chamber 'A' model the system attains the peak value in a very short period of time 0.2 seconds (approx) and attains the minimum position in 0.4 seconds (approx). The system rises again in amplitude and tends to attain the stable condition. The figure 7.7 shows the experimental time response of Moog actuator. The experimental and the simulation graphs were found to be in satisfactory compromise.

Figure 6.5 shows the MATLAB/SIMULINK model of the chamber ‘B’ subsystem. A Scope block, which is shown in SIMULINK model, helps to find the time response of the chamber ‘B’. In the simulation graph which is shown in the figure 7.3 is almost symmetrical to the time response of chamber ‘A’. Slight asymmetry results from the change in chamber volumes as the piston is displaced to its new position. The figure 7.7 shows the experimental time response of Moog actuator. The experimental and the simulation graphs were found to be in satisfactory compromise.

Figure 6.2 shows the MATLAB/SIMULINK model of the actuator subsystem. A scope block, which is shown in SIMULINK model, helps to find the time response of the actuator for a ramp input. The figure 7.4 shows the time response of the actuator. The sharp rising and falling edges and minimal overshoot represent the optimum response.

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Figure 6.3 shows the MATLAB/SIMULINK model of the servo valve subsystem. A scope block, which is shown in SIMULINK model, helps to find the time response of the servo valve. The figure 7.5 shows the response of the servo valve. The system rises to maximum amplitude in 0.15 seconds (approx) and then reduces to a minimum value of amplitude in 0.4 seconds and again the amplitude of the system increases in magnitude and at 0.8 seconds the system tends to attain stable condition.

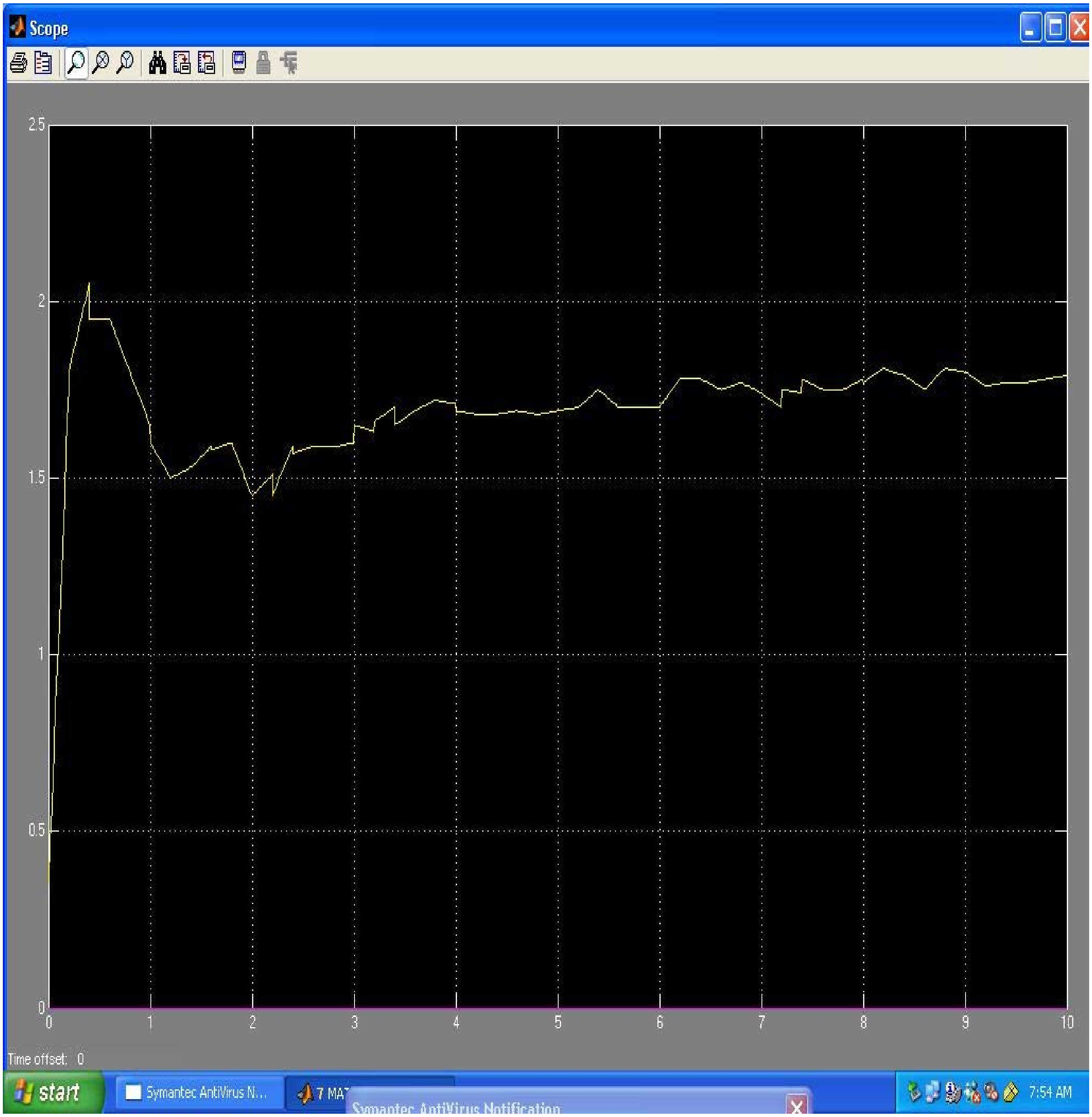

Figure 7.1 Time Response of the Linear Hydraulic Actuator

Time in seconds on Xaxis

Amplitude on Yaxis

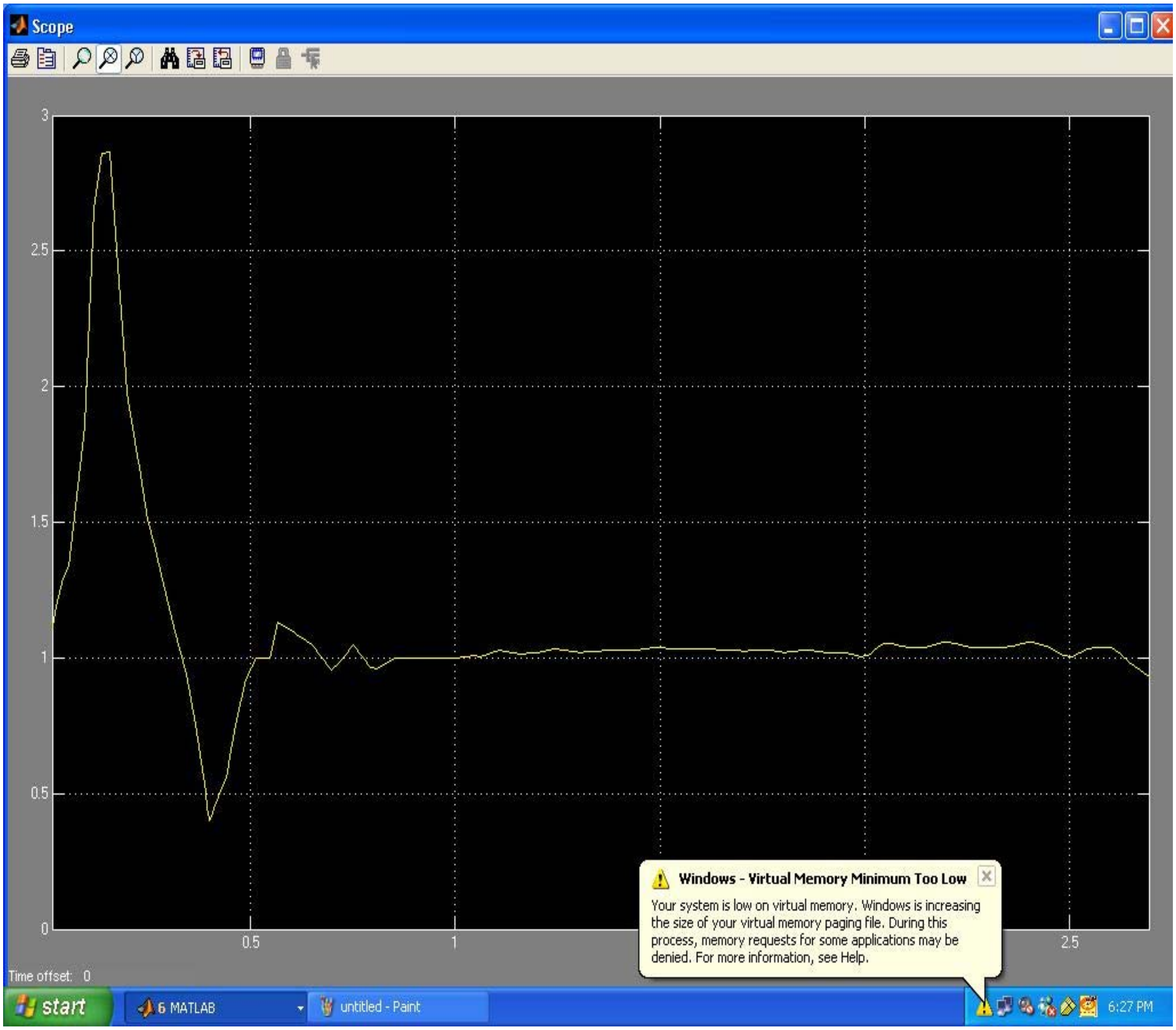

Figure 7.2 Time Response of the Piston Chamber' A' of the Actuator

Time in seconds on Xaxis

Amplitude on Yaxis

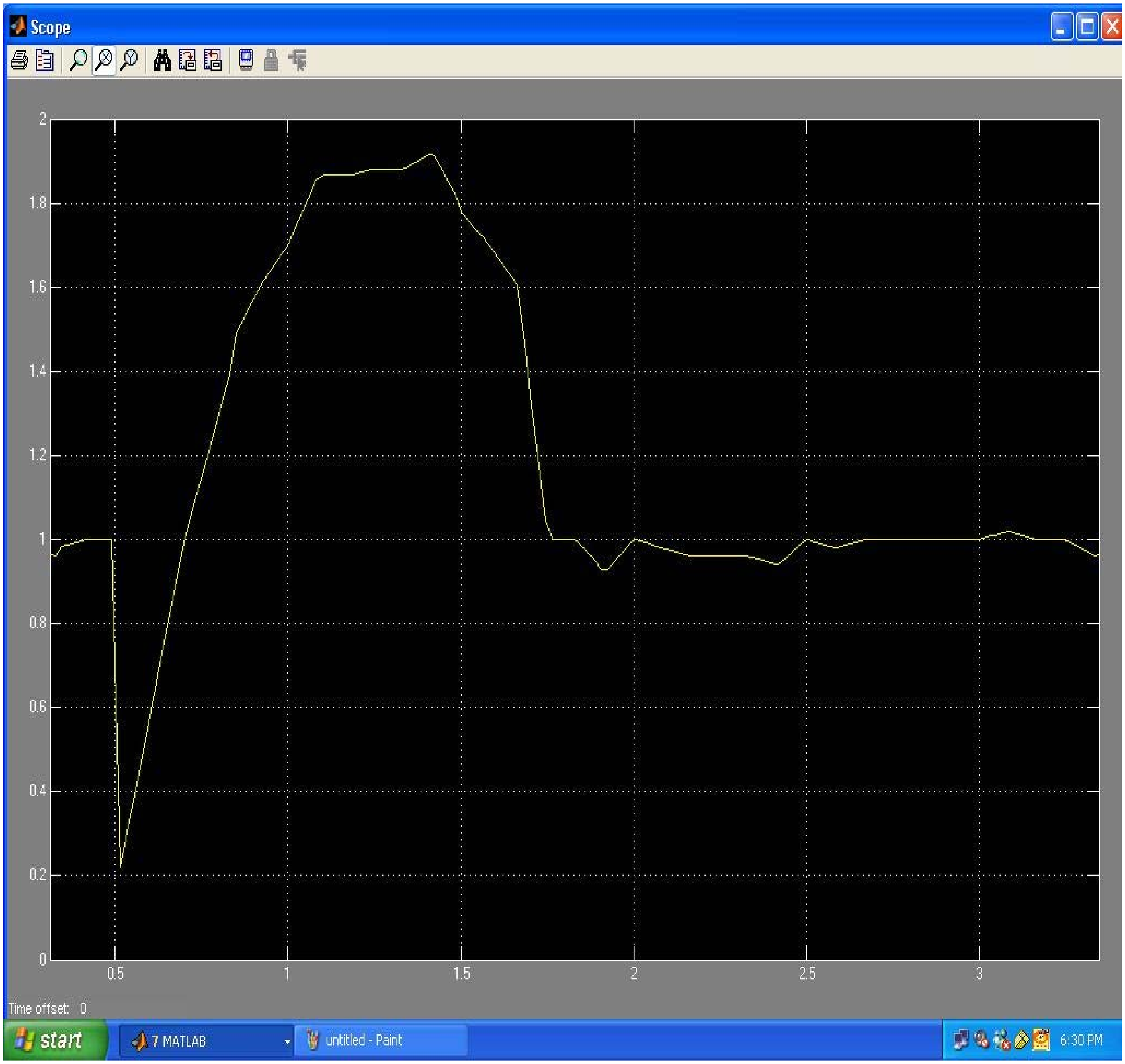

Figure 7.3 Time Response of the Piston Chamber'B' of the Actuator

Time in seconds on Xaxis

Amplitude on Yaxis

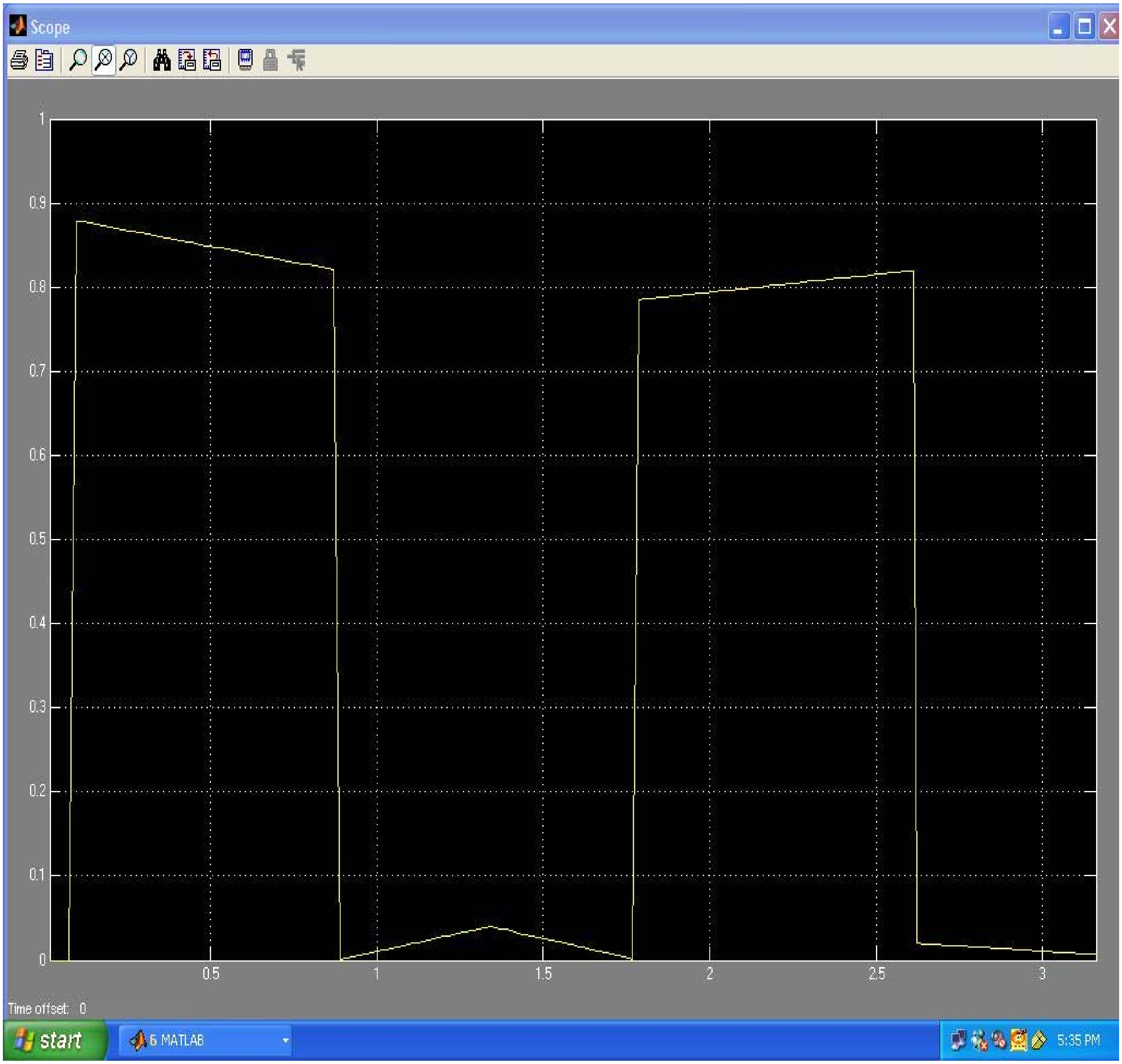

Figure 7.4 Time Response of the Actuator

Time in seconds on Xaxis

Amplitude on Yaxis

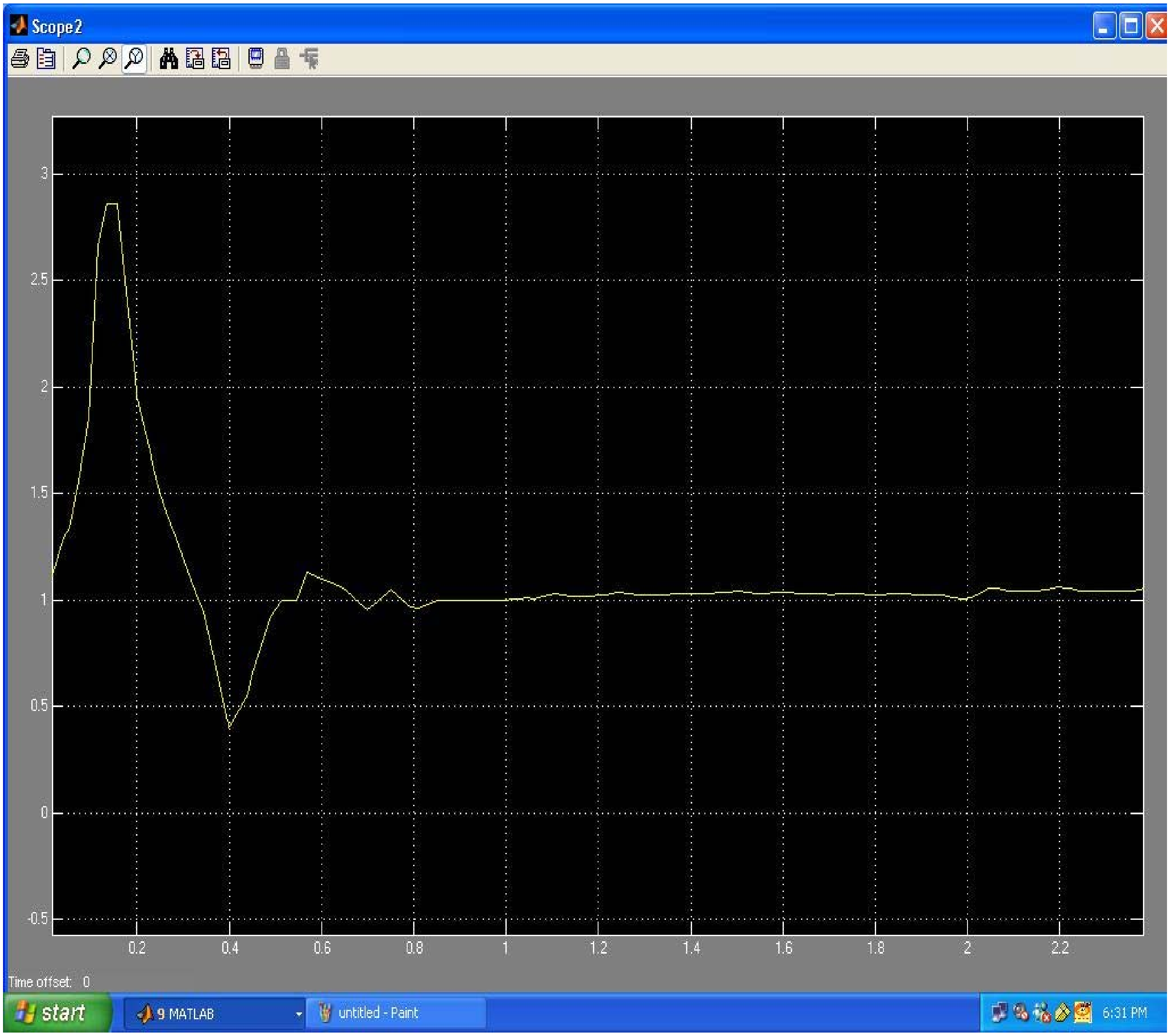

Figure 7.5 Time Response of the Servo Valve

Time in seconds on Xaxis

Amplitude on Yaxis

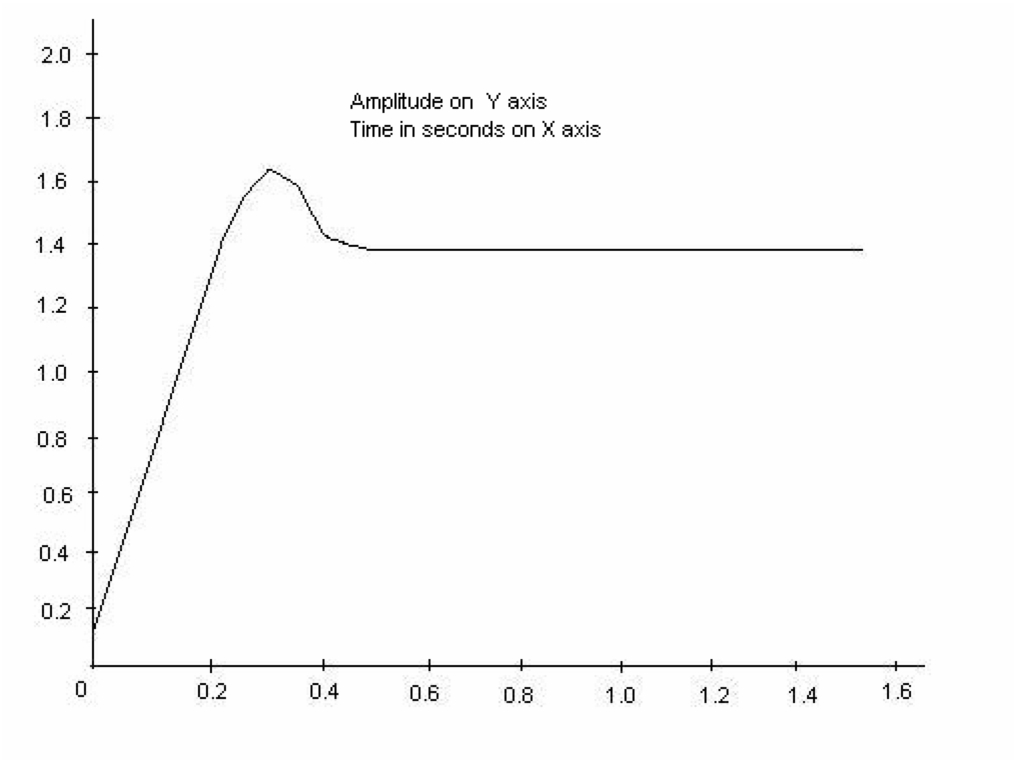

Figure7.6 Experimental time response graph of a hydraulic linear actuator (Courtesy Flight Test Centre, RWRDC)

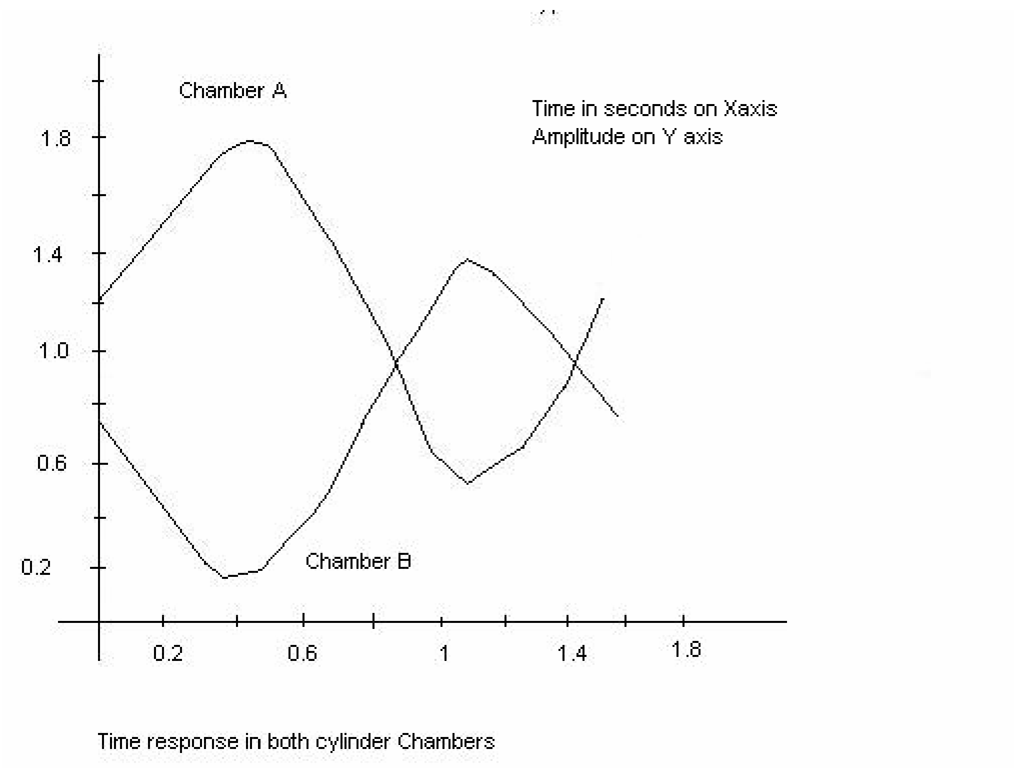

Figure 7.7 experimental time response in chambers A and B (courtesy Flight Test centre, RWRDC**)**

8. CONCLUSIONS

Mathematical models have been developed for the hydraulic system components like hydraulic actuator, servo valve, piston chambers, and servo controllers by considering the system requirements, system characteristics, fluid flow properties. By using these mathematical models

MATLAB/SIMULINK models have been made for the hydraulic system components .This time response is a very significant factor when the system considered is a critical system like a flight control actuator.

1. These Simulink models of hydraulic actuator function like a virtual hydraulic actuator where in we can obtain the behavior of the system with respect to time without physically testing the component.
2. The time response of hydraulic actuator, servo valve is obtained from the MATLAB/SIMULINK software.
3. With the help of these MATLAB/SIMULINK models the performance of the hydraulic system components, sub systems like servo controller; servo valve etc can be monitored.
4. By varying the subsystem parameters like pressure, active annulus area, stroke length, control current etc the designer can arrive at optimum parameters of the hydraulic actuator.
5. With the help of these MATLAB/SIMULINK models of electro hydraulic servo actuator the time response of the hydraulic actuator can be obtained without physically testing the actuator.
6. The time responses of the hydraulic actuator, servo valve and piston chamber are compared with the MOOG hydraulic actuator data (courtesy Flight Test Centre, RWRDC). The time response graphs which are obtained by this simulation of electro-hydraulic actuator are found to be coinciding with the experimental time response graphs of Moog electro hydraulic actuator.

9. SCOPE FOR FUTURE DEVELOPMENT

The Simulation of Hydraulic Actuator can also be done with the help of Softwares like HYPNEU, SIMULATIONX etc.

1. The Simulation using MATLAB/SIMULINK can also be done for finding the Time response of other hydraulic System components like Pump.
2. This Simulation using MATLAB/SIMULINK can also be applied for finding the Time response of the Aircraft Flight control surfaces like Ailerons, Elevators and Rudder etc.
3. The Time response of the Aircraft wheel brake System can also be obtained by using this approach of MATLAB/SIMULINK Simulation
4. Using this approach of MATLAB/SIMULINK Simulation can also be done for finding the Time response of Aircraft Landing Gear.
5. By this approach of simulation the behavior of the Rotary Hydraulic Actuator with respect to time can be found out.

10. REFERENCES

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