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POLO REGIONALE DI LECCO MASTER OF SCIENCE IN MECHANICAL ENGINEERING



DYNAMIC ANALYSIS OF CLOSED LOOP HYDRAULIC SYSTEM WITH 5/3 WAY CUSTOM VALVE USING PID CONTROLLER

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Abstract:-

This project deals with dynamic analysis of hydraulic unit by matlab simulation. It directly relates with machine response of the various component. In the case of the hydraulic systems, which are discussed in this paper, the actual input is some kind of prime mover such as an internal combustion engine or an electrical motor. The speed and torque of the prime mover is converted to the hydraulic power parameters at the pump, the directional control of the hydraulic power is provided by a valve, while the output can be the force and velocity of the double acting cylinder.

To analyze piston rod displacement, pressure and force acting in the system. Dealing with complicated system which contain customized valve, power supply unit and PID controller. The hydraulic system is described mathematically in preparation for modeling. At first, the model with high degree of detail, but their non-linear character is created. The linear model which used in design of an appropriate control system which enables the accurate control of velocity, force and pressure of the system.

An input signal of sine wave to pass through the actuating device of customized valve. Using closed loop control system with PID regulation, which is used to tune the performance of system along the transfer function of system. One of the complicated task taken in this part, design and performance of the high response servo valve (Moog D661). And it is tuned by PID controller with appropriate parameters.

The principle of closed loop hydraulic system was analyzed and the total pressure control valve was discussed in detail. There are two methods were used to compute the whole system natural frequency based on building hydraulic system model in frequency domain. The other is Laplace transform method in which we got the relationship between the whole system and every part's of natural frequencies. So, our final result to determine the transfer function of the closed loop of hydraulic system.

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1. Introduction:

This chapter will be an introduction to this thesis. At first, a short introduction will be given to a review of dynamical analysis of hydraulic system (Presses) and modelling of hydraulic system using simulink in matlab will be carried out.

The hydraulic press, invented by Englishman John Brahmah, was one of the first work-able pieces of machinery developed that used hydraulics in its operation. It consisted of a plunger pump piped to a large cylinder and a ram. This press found wide use in England because it provided a more effective and economical means of applying large forces in industrial uses.

Today, hydraulic power is used to operate many different tools and mechanisms. In a garage, a mechanic raises the end of an automobile with a hydraulic jack. Dentists and barbers use hydraulic power, through a few strokes of a control lever, to lift and position their chairs to a convenient working height. Hydraulic doorstops keep heavy doors from slamming. Hydraulic brakes have been standard equipment on automobiles, since the 1930s. Most automobiles are equipped with automatic transmissions that are hydraulically operated. Power steering is another application of hydraulic power. Construction workers depend upon hydraulic power for the operation of various components of their equipment. For example, the blade of a bulldozer is normally operated by hydraulic power.

There are many hydraulic systems in the engineering applications composed of pumps, motors, pipes, pistons, valves, filters, and accumulators that use nearly incompressible fluids such as water or hydraulic oil. Such systems are found in machine tools, earth-moving equipment, power transmissions, and aircraft control surface servomechanisms. Generally, these systems have high pressures and low fluid velocities so that the static pressure dominates the dynamic pressure. The performance of a hydraulic control system is strongly influenced by the dynamic characteristics of its control valves. Almost every hydraulic system is equipped with a pressure regulator system to maintain the working pressure of the system at a predetermined level.

Hydraulic actuators are used in a lot of different mechanical applications like wind mills, industrial production such as robots and processing machinery, construction machines such as cranes and excavators, just to name a few. The term Hydraulic actuators cover a group of components in machine design which can create a rotating or linear motion by utilizing a pressurized fluid. Actuators generating rotation are called motors while linear hydraulic actuators are called cylinders. The focus in this thesis will be on hydraulic cylinders.

Hydraulic actuators are widely applied in hydraulic servo-systems, which is a system that is made up of several individual hydraulic components such as pump, valves, actuators, sensors and the connecting elements in-between. These components are interconnected so they can perform a pre-defined task through the hydraulic transfer.

Hydraulic Press unit:



2. Concept for determining the dynamic analysis of hydraulic system

2.1 Hydraulic System Design and Analysis Process

Once the design goals are established for a given hydraulic system, the actual design process can be initiated. The generalized hydraulic system design and analysis process is illustrated in Fig. 2.1. From the design goals, the design concept to be integrated into the hydraulic system must be established and a system schematic must be developed along with the operational specifications for the system. Once these tasks are completed the designer will enter into the component sizing and selection process. In the past when the sizing and selection phase was completed, the components would be purchased and a prototype system constructed. The system performance was not simulated and the success of a particular system was mainly a result of the experience of the designer and luck. The actual performance characteristics were evaluated through laboratory and field tests using the system prototype.

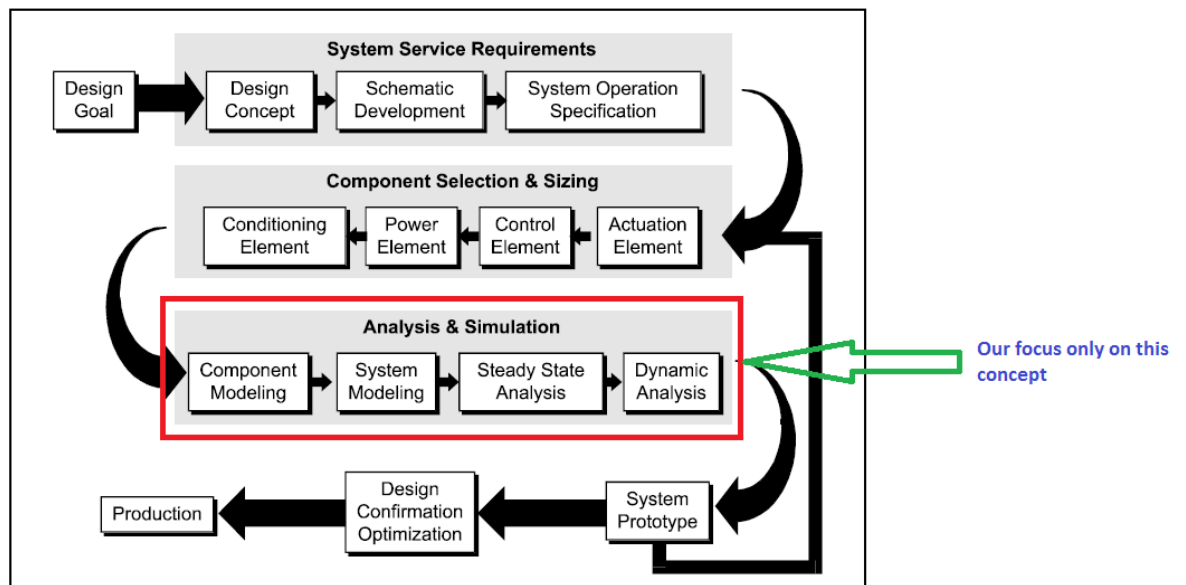


Fig 2.1 Hydraulic System Design and Analysis Process

2.2 Three concepts deal with modeling and simulation

Our thesis deals with only the Analysis and simulation concept, which are discussed

1. Modeling the hydraulic system in Matlab-Simulink.
2. Evaluating the theoretical analysis of the actuating system.
3. Determine the Simulation result of hydraulic system.

3. Hydraulic system and experimental setup

This chapter gives an introduction to the hydraulic system and the experimental setup with regards to components and transducers.

3.1 Hydraulic system design

This section describes the design of the hydraulic system with regards to the chosen components. All of the components have been found in the hydraulics circuit.

Specifications for the hydraulic system are given in section 3.1.2.

3.1.1 Hydraulic components

The following components are chosen to be used in the hydraulic system:

The datasheets of the cylinder and servo valve are found on the attached. The rated cylinder size, which is 80mm. The maximum velocity of the chosen cylinder is 0.3 m/s which results in a flow of 40 l/min. A pressure relief valve and a pump station will be specified, when the overall system is constructed, as these are important in the following analysis of the hydraulic system.

3.1.2 Schematic diagram of Hydraulic system

The configuration of the hydraulic system is shown in figure 4.1.

Component	Type
Cylinder	ALTOF Ø 80/56Mx170-MF3/N/S
Servo Valve	Moog D661-4519C
Pressure relief valve	Rexroth DBDS6K-1X/100/200
Variable displacement pump (Q=40l/min)	Rexroth
Solenoid valve	4/3 direction control valve – Rexroth

Table 3.1: Components of the hydraulic system.

PRESS CYLINDER

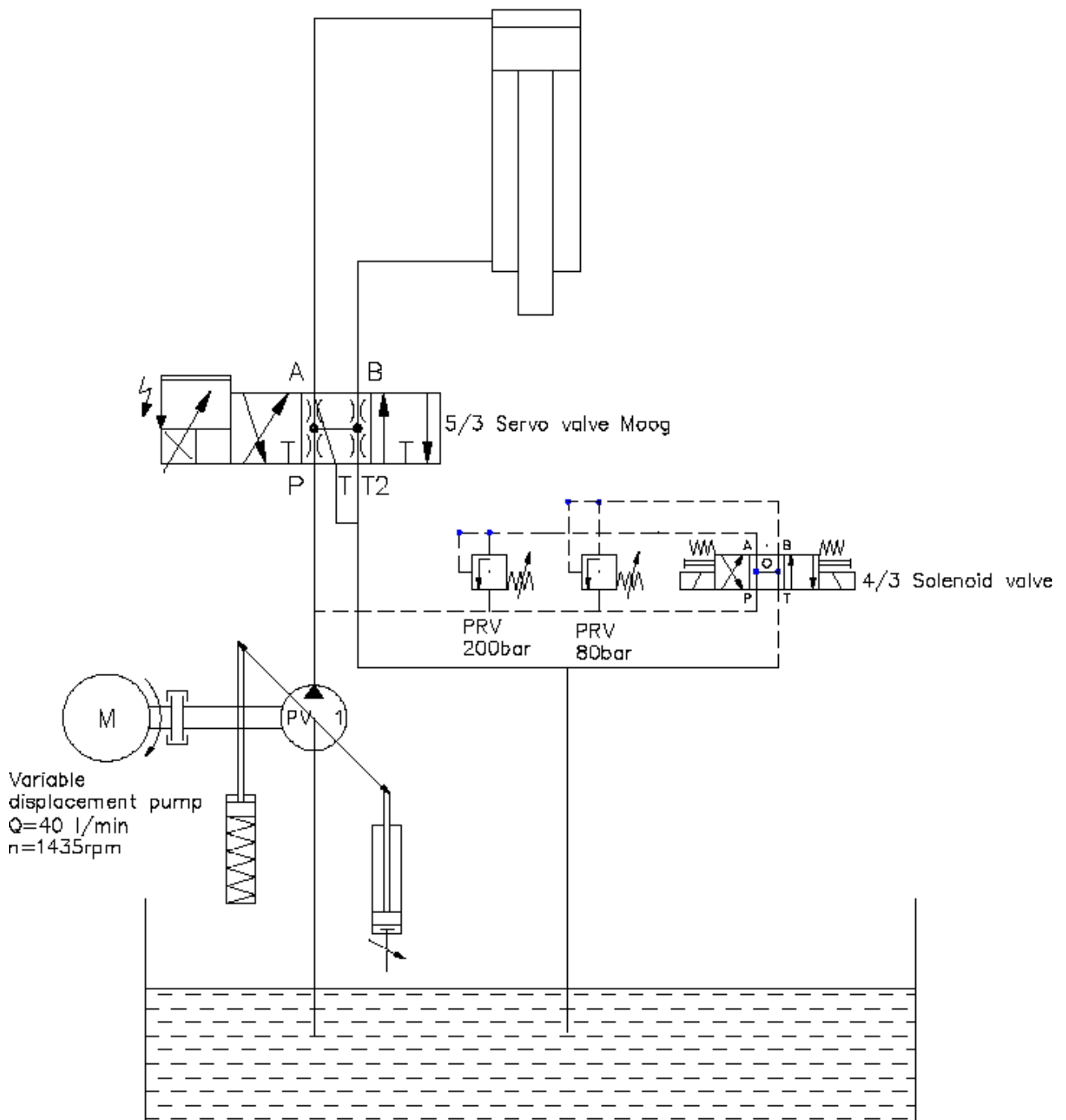


Fig 3.1 Schematic diagram of hydraulic press machine

3.2 Basic hydraulics and components:

3.2.1 Principles of hydraulics:

Hydraulics is the science of transmitting force and/or motion through the medium of a confined liquid. In a hydraulic device, power is transmitted by pushing on a confined liquid. Figure 1-1 shows a simple hydraulic device. The transfer of energy takes place because a quantity of liquid is subject to pressure. To operate liquid-powered systems, the operator should have a knowledge of the basic nature of liquids. This chapter covers the properties of liquids and how they act under different conditions.

Hydraulic System:

A hydraulic system contains and confines a liquid in such a way that it uses the laws governing liquids to transmit power and do work. This chapter describes some basic systems and discusses components of a hydraulic system that store and condition the fluid. The oil reservoir (sump or tank) usually serves as a storehouse and a fluid conditioner. Filters, strainers, and magnetic plugs condition the fluid by removing harmful impurities that could clog passages and damage parts. Heat exchanges or coolers often are used to keep the oil temperature within safe limits and prevent deterioration of the oil. Accumulators, though technically sources of stored energy, act as fluid storehouses.

3.2.2 Hydraulic components:

- Variable displacement pump
- Double acting cylinder
- 5/3 servo-valve
- 4/3 Solenoid valve
- Pressure relief valve
- Pipes or hoses
- Tank or reservoir

Variable displacement pump:

When in neutral, oil is pumped until the pressure rises to a predetermined level. A pressure-regulating valve allows the pump to shut off by itself and maintain this pressure to the valve. When the control valve is operating, oil is diverted from the pump to the bottom of a cylinder. The drop in pressure caused by connecting the pumps pressure line to the bottom of the cylinder causes the pump to go back to work, pumping oil to the bottom of the piston and raising the load. When the valve moves, the top of the piston connects to a return line, which allows the return oil that was forced from the piston to return to the reservoir or pump. When the valve returns to neutral, oil is trapped on both sides of the cylinder, and the pressure passage from the pump is dead-ended. After this sequence, the pump rests. Moving the spool in the downward position directs oil to the top of the piston, moving the load downward. The oil from the bottom of the piston is sent into the return line.

Double acting cylinder:

Double acting cylinders have two opposing effective areas which are of the same or different sizes. They are fitted with 2 pipe ports which are isolated from each other. By feeding fluid via ports "A" or "B", the piston may transfer pulling and pushing forces in both stroke directions. This type of cylinder may be found in nearly all types of application.

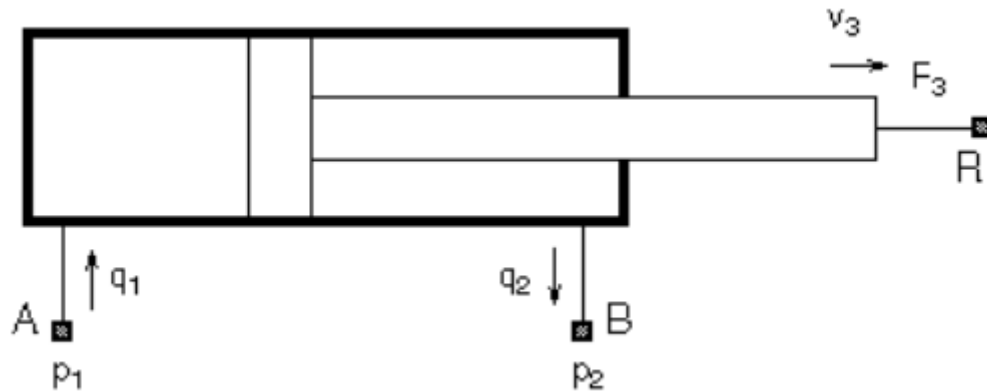


Fig 3.2 Double acting single rod cylinder

5/3 Servo-valve (Moog):

Servo valves and Servo-Proportional Valves are electro-hydraulic operate by transforming a changing analogue or digital input signal into a smooth set of movement in a hydraulic cylinder. Servo valves can provide precise control of position, velocity, pressure and force with good post movement dampening characteristics. A low voltage is used control the servo valve. The control voltage is passed into an amplifier which provides the power to alter the valve's position. The valve will then deliver a measured amount of fluid power to an actuator. The use of a feedback transducer on the actuator returns an electrical signal to the amplifier to condition the strength of the voltage to the servo valve.

Many of the servo valves in use today are manufactured by Moog Inc. A typical example of a servo valve is the Moog servo valve D661 series. An example of servo valve use is in Blow Molding and hydraulic press. The details of servo valve are explained below.

4/3 solenoid valve:

A solenoid valve is an electromechanical valve for use with liquid or gas. The valve is controlled by an electric current through a solenoid: in the case of a two-port valve the flow is switched on or off; in the case of a three-port valve, the outflow is switched between the two outlet ports. Multiple solenoid valves can be placed together on a manifold. Solenoid valves are the most frequently used control elements in fluidics. Their tasks are to shut off, release, dose, distribute or mix fluids. They are found in many application areas. Solenoids offer fast and safe switching, high reliability, long service life, good medium compatibility of the materials used, low control power and compact design.

Pressure relief valve:

Pressure relief valves are used in hydraulic system to limit the system pressure to a specific set level. If this set level is reached, the pressure relief valve is activated and feeds the excess flow (difference between pump and actuator flow) from the system back to the tank. This valve is always arranged as a bypass valve. The pressure relief valve is also known as a safety valve.

4. Dynamical model of fluid-mechanical system

This chapter will describe the mathematical modeling of the dynamics of the system in question. At first the modeling detail with regards to the system will be defined by bounding the region of importance in order to establish a sufficient model. Thereafter the assumptions preceding the mathematical descriptions will be mentioned and finally the mathematical models will be presented.

The purpose of this model is to describe the dynamics of the system and how these dynamics affects the output when a specific input is given. Further on, this model will be linearized and the linearized model will be used to evaluate system behavior with regards to stability and design specific closed-loop controllers.

The dynamic part of the system consist of a hydraulic cylinder which mounted in the Press machine. The cylinder is controlled by servo valve (Moog) and solenoid valve, which controls the fluid flow supplied to the cylinders. The fluid is supplied from tank to variable displacement pump, by supply variable pressure with actuating conditions of 80 and 200 bar. This variation can be critical view of dynamical analysis of fluid-mechanical system. For this system it will be necessary to describe the dynamics and properties of the servo valve and cylinder with regards to flow, pressure and force.

The following assumptions have been made in order to simplify the modeling of the system:

1. The hydraulic units are mounted completely rigid.
2. The varying supply pressure is either 80 or 200 bar.
3. Leakage flow is present in the cylinders or valves.
4. Constant Bulk Modulus.
5. Tank pressure is constant.

In figure an illustration of the parts in the system, which have to be described mathematically, are shown. The designations indicated by figure will be used to set up the equations for the system.

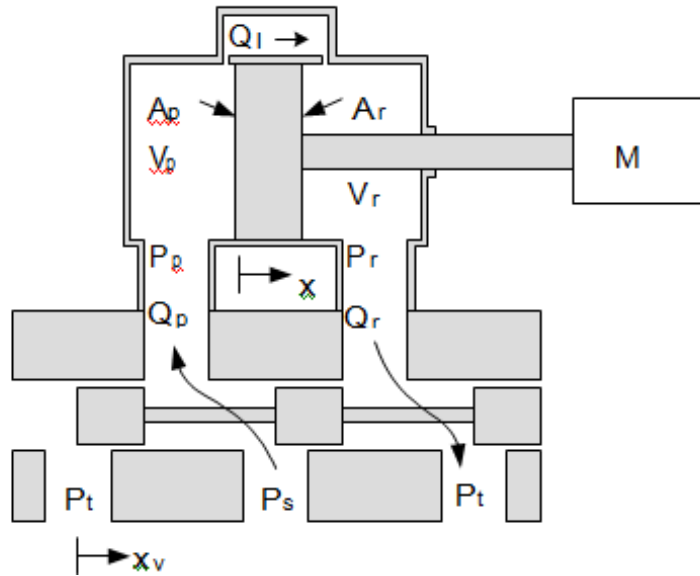


Fig 4.1 illustrates between the connection of cylinder and servo valve

4.1 Friction characteristics:

The total friction in a given situation can be divided up in different types of friction which are characterized by the velocity state at which they act and their dependence of this state. Furthermore, certain phenomenon exist which characterize the nature of friction. These frictions and effects according to [Armstrong-Hélouvy, 1991] are:

Static Friction: The force necessary to initiate motion from rest.

Kinetic Friction: Friction independent of the magnitude of velocity. Also referred to as Coulomb friction.

Viscous Friction: Friction directly dependent of velocity as this friction is zero at zero velocity.

Dahl Effect: A friction phenomenon which arises from the elastic deformation of bonding sites between two surfaces which are locked in static friction.

Stribeck Effect or Stribeck Friction: A phenomenon occurring when using fluid lubrication where the friction is decreasing with increasing velocity at low velocities.

Break-away Force: The amount of force required to overcome static friction.

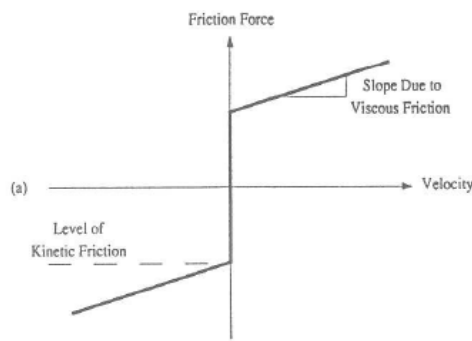


Fig 4.2 Kinetic and viscous friction [Armstrong-Hélouvy 1991]

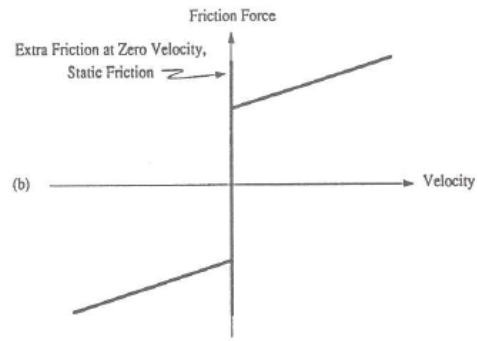


fig 4.3 Static, kinetic and viscous friction combined

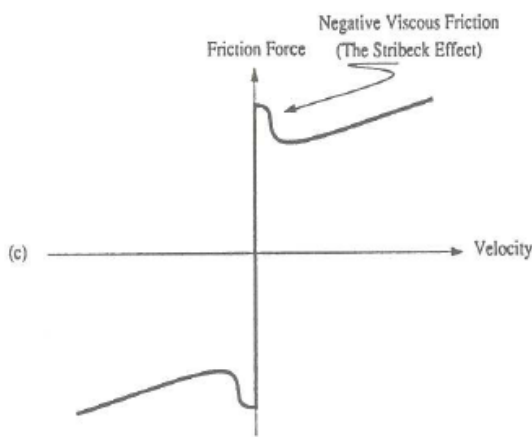


Fig 4.4 kinetic and viscous friction curve (Stribeck friction)

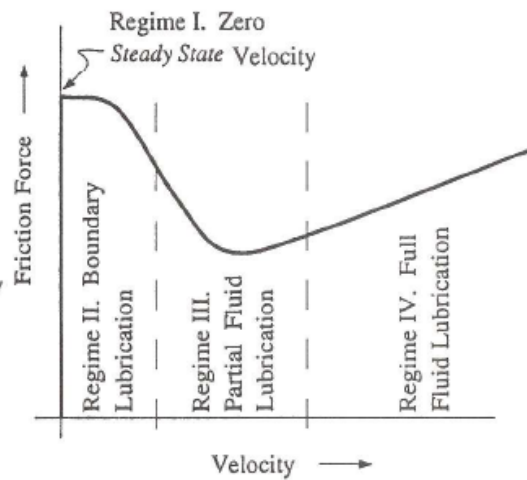


Fig 4.5 The generalized stribeck illustrating frictions depends of velocity

As seen in figure 1.1-1.3, friction is considered a function of velocity. As mentioned earlier the kinetic friction is independent of velocity and always present. On the contrary, the viscous friction is proportional to the velocity and it occurs in fluid lubricated interfaces (figure 1.1). As figure 1.2 illustrates, in some cases the break-away force, which is the force necessary to initiate motion, is larger than the force needed to sustain motion because of the static friction. The static friction and the Dahl effect are closely correlated as the Dahl effect is a consequence of the static friction and the asperities of the surfaces in the interface. Static friction is the greatest cause to stick-slip motion which is explained later. Another consequence of correlation between the Dahl effect and the static friction is position-dependence of the static friction. An in depth explanation of the Dahl effect and static friction can be found in [Armstrong-Hélouvy et al., 1994]. The Stribeck effect is illustrated in figure 1.3 which suggests that the drop from the static friction doesn't happen instantaneously whereby the friction decreases with increasing velocity for low velocities. The Stribeck curve in figure 1.4 gives a closer look at friction at low velocity and show the three moving regimes, of the four in total, which contribute to the dynamics a controller confronts as the system accelerates away from zero velocity [Armstrong-Hélouvy, 1991].

The Stribeck curve is a representation of the friction in a system which is lubricated with grease or oil, as is the case in most mechanical systems. The curve illustrates how the different regimes of lubrication change according to velocity and how this affects the friction. The lubrication regimes provides a physical explanation for the friction phenomena, but this will not be covered in depth here. For more information see [Armstrong-Hélouvry, 1991; Armstrong-Hélouvry et al., 1994]

From figure 1.1-1.4 it is apparent that all the different kinds of friction, except viscous, are discontinuous when the velocity is zero. This property along with the Stribeck effect causes nonlinearity and the consequences of this, with respect to servo systems, will be discussed in the following section.

4.1.2 Friction in Servo systems:

Upper and lower bounds

Friction brings both positive and negative traits into a servo system. Friction can bring damping into a system which otherwise would be unstable. This damping is provided at all frequencies both under and over the bandwidth of the control. Besides playing a role in the dynamics of the system, friction affects the speed and power and thereby limiting the overall performance. Most often systems are assessed at their upper performance bounds with regards to maximum speed, maximum force and so on. Just as much as friction affects the upper bounds of performance, it affects the lower bounds, as very small displacements and corresponding low velocities can be unobtainable because of friction and its non-linear nature at low velocity. The non-linearity causes a periodic process of sticking and slipping motion called stick-slip which limits the lower bounds with regards to minimum achievable displacement and minimum sustainable velocity. [Armstrong-Hélouvry, 1991]

Frictional lag:

So far, friction have been assumed to have a steady dependence of velocity as shown by the Stribeck curve. It has been assumed that if the velocity changes then the friction will change simultaneously. Though, experiments by [Hess and Soom, 1990] have shown that there is a lag in the change of the friction, which is designated as frictional lag. Frictional lag is illustrated in figure 1.5.

Furthermore, frictional lag creates hysteresis, which is the separation between the friction levels during acceleration and deceleration. The hysteresis loop is shown in figure 1.6 and it is greatly affected by the oscillatory frequency, as the hysteresis is greatest when the oscillating period is short relative to the frictional lag. The existence of frictional lag and hysteresis indicates a necessity for dynamic friction models which describe these phenomena. [Armstrong-Hélouvry, 1991; Olsson et al., 1997]

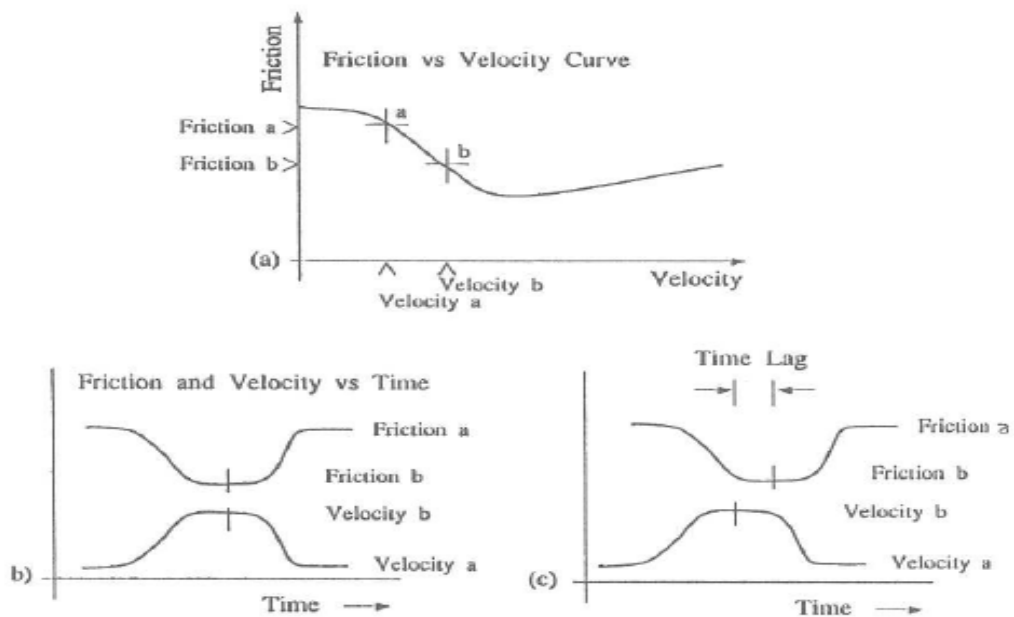


Fig 4.6 a) and b) Friction as a function of velocity (stribeck curve) independent of time; c) The friction lags behind the velocity changes. [Armstrong-Hélouvy, 1991]

Mechanical considerations and experimental measurement of friction

The total friction in a given system may be the result of similar friction levels in interfaces between the parts which makes up the mechanical system. If that is the case, it can be impossible to separate the different contributions and an overall friction model is necessary to describe the friction in the entire mechanism. In other cases, the primary source of friction might originate from a single interface as can be the case when using a hydraulic cylinder.

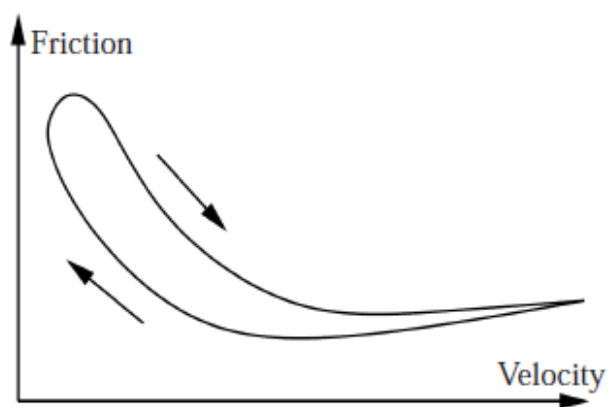


Fig 4.7: The friction-velocity hysteresis loop reported in [Hess & soom 1990]. The shorter the period of the oscillation the wider the loop.

4.2 Modeling of cylinders:

Modeling of the cylinder refer to determining the pressure in the cylinder chamber along with describing the motion of the piston. The pressures are expressed via the flow continuity equations while the motion of the system is given by Newton's II law of motion.

4.2.1 Continuity flow for the cylinder:

Consider as an incompressibility flow for the cylinder is expressed by the flow continuity equations given by

$$Q_p - Q_l = \frac{dv_p}{dt} + \frac{V_p}{\beta} \cdot \frac{dp_p}{dt} \quad (4.1)$$

$$Q_l - Q_r = \frac{dv_r}{dt} + \frac{V_r}{\beta} \cdot \frac{dp_r}{dt} \quad (4.2)$$

$$V_p = V_{p0} + A_p y \quad (4.3)$$

$$V_r = V_{r0} - A_r y \quad (4.4)$$

Therefore

$$Q_p = \frac{dv_p}{dt} + \frac{V_p}{\beta} \cdot \frac{dp_p}{dt} \quad (4.5)$$

$$- Q_r = \frac{dv_r}{dt} + \frac{V_r}{\beta} \cdot \frac{dp_r}{dt} \quad (4.6)$$

- Q_p : flow into the test-cylinder on the piston side([m3/s])
- Q_l : leakage flow over the cylinder piston ([m3/s])
- Q_r : flow out of the test-cylinder on the rod side ([m3/s])
- V_p : volume on the piston side ([m3])
- V_r : volume on the rod side ([m3])
- V_{p0} : initial volume on the piston side ([m3])
- V_{r0} : initial volume on the rod side([m3])
- A_p : piston area ([m2])
- A_r : roside piston area ([m2])
- P_p : pressure acting in the piston side chamber ([m3])
- P_r : pressure acting in the rod side chamber ([m3])
- y : displacement of the piston from the initial position ([m])
- \dot{y} : velocity of piston ([m/s])
- β : bulk modulus of the fluid ([bar])

By neglecting the leakage flow, Q_l , the pressure gradient of the rod- and piston side can be expressed from equation 4.5 and 4.6

$$Q_p = A_p \dot{y} + \frac{V_0}{\beta} \frac{dP_p}{dt} + \frac{A_p}{\beta} \frac{dP_p}{dt} \quad (4.7)$$

$$-Q_r = -A_r \dot{y} + \frac{V_0}{\beta} \frac{dP_r}{dt} + \frac{A_r}{\beta} \frac{dP_r}{dt} \quad (4.8)$$

Equation of motion:

The system equation of motion is given by Newtons II law of motion from the force acting on the interconnected pistons:

$$m_{eq} \ddot{y} = P_p A_p - P_r A_r \quad (4.9)$$

\ddot{y} = Acceleration of masses ([m/s²])

m_{eq} = Total mass of the moving bodies ([kg])

4.3 Valve linearity:

Valves are usually non-linear. That is, the flow through the valve is not the same as the valve position. Several types of valves exist:

- Linear
- Same gain regardless of valve position
- Equal Percentage
- Low gain when valve is nearly closed
- High gain when valve is nearly open
- Quick Opening
- High gain when valve is nearly closed
- Low gain when valve is nearly open

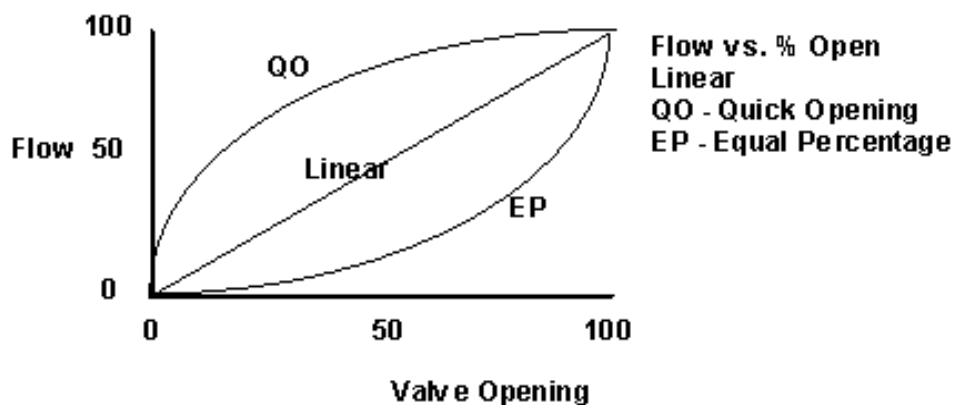


Fig 4.8 Characteristics of valve opening

As we will see later, including the valve, is very important to the tuning of the loop. If the controller is tuned for one process gain, it may not work for other process gains.

4.4 Linearized equation of servo valve:

Servovalve:

A servovalve 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 modeling complex servovalves, 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.

As mentioned, subsystems utilize a MOOG D661 servovalve. The flow through the orifice in the servovalve is determined from the valve-input, which is equivalent to a displacement of the valve spool, and the pressure drop over the valve. The input is a +/-10VDC voltage (10V corresponds to 100% valve opening) where a positive voltage corresponds to a displacement in the positive direction and vice versa.

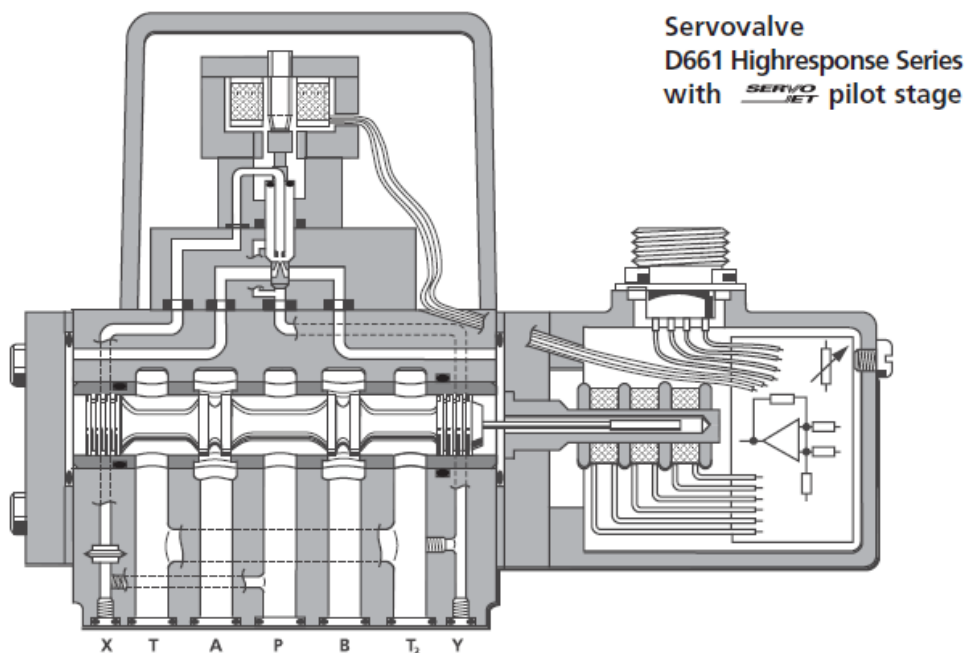


Fig 4.9 D661 High response series of servo valve (moog)

For this subsystems, the modeling of the servovalve is almost similar and the following expressions are therefore written in a general manner. The only variation in the modeling is a reversed configuration of the equations in (2.7)- (2.10) with regards to the sign of u_v when modelling the load-servovalve.

For an input voltage $u_v > 0$ the static flow through the valve is given as:

$$Q_p = K_T \cdot u_v \sqrt{\frac{2}{\rho} (P_s - P_p)} \quad (2.10)$$

$$Q_r = K_T \cdot u_v \sqrt{\frac{2}{\rho} (P_r - P_t)} \quad (2.11)$$

When $u_v < 0$ the flow is given as:

$$Q_p = K_T \cdot u_v \sqrt{\frac{2}{\rho} (P_p - P_t)} \quad (2.12)$$

$$Q_r = K_T \cdot u_v \sqrt{\frac{2}{\rho} (P_s - P_r)} \quad (2.13)$$

Where

Q_p : flow at the inlet orifice of the valve	([m ³ /s])
Q_r : flow at the exit orifice of the valve	([m ³ /s])
K_T : Static valve gain	([-])
u_v : input voltage to the servovalve	([V])
ρ : density of the oil	([kg/m ³])
P_s : supply pressure	([P a])
P_t : tank pressure	([P a])
P_p : pressure acting between the orifice, and the piston side	([P a])
P_r : pressure acting between the orifice, and the rod side	([P a])

The equations in (2.7) & (2.8) are static models for the flow through the valve. The valve gain, K_T is calculated from the related flow, which is found in the data sheet for MOOG D661. This data sheet can be found on the attached below. The data sheet contains the valve flow diagram illustrated in figure 2.2. The data sheet states that the rated pressure drop is $\Delta P_N = 35$ bar per meeting land for the rated flow $Q_N = 40$ l/min (fig. 2.2) at 100% command signal (i.e. 10 VDC = 100% valve opening).

Applying these values in equation (2.7) and the valve gain K_T is found to be:

$$K_T = \frac{Q_N}{U_{100\%} \sqrt{\frac{2\Delta P_N}{\rho}}} \quad (2.14)$$

$$K_T = \frac{(40 \text{ l/min}) \left(\frac{1 \text{ m}^3 \cdot \text{min}}{60000 \text{ l.s}} \right)}{10 \text{ V} \sqrt{\frac{2.35 \cdot 10^5 \text{ Pa}}{870 \frac{\text{Kg}}{\text{m}^3}}} = 7.43 \cdot 10^{-7} \frac{\text{m}^2}{\text{V}} \quad (2.15)$$

5/3 Directional Proportional valve:

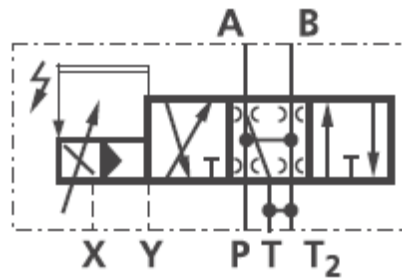


Fig 4.10 Schematic diagram of 5/3 directional proportional valve

Flow diagram of MOOG D661 series:

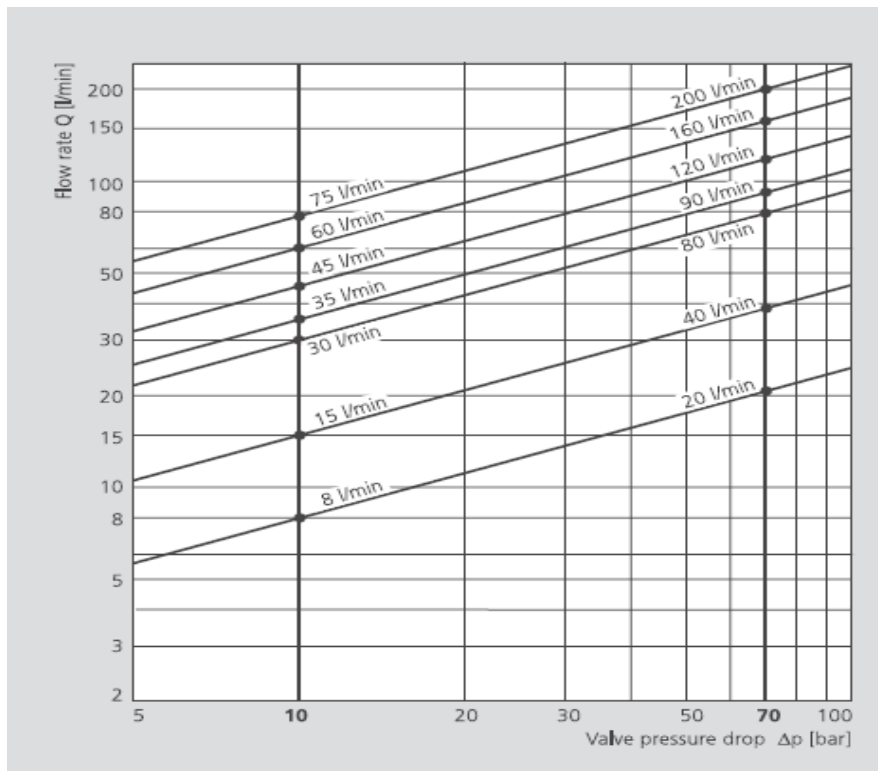


Figure 4.11 valve flow diagram from the data sheet of the MOOG D661 servovalve. [MOOG- Controls]

The dynamic of the servovalve is determined from its both step response and frequency response which is also to be found in the data sheet [MOOG-controls] are shown below in fig 4.12

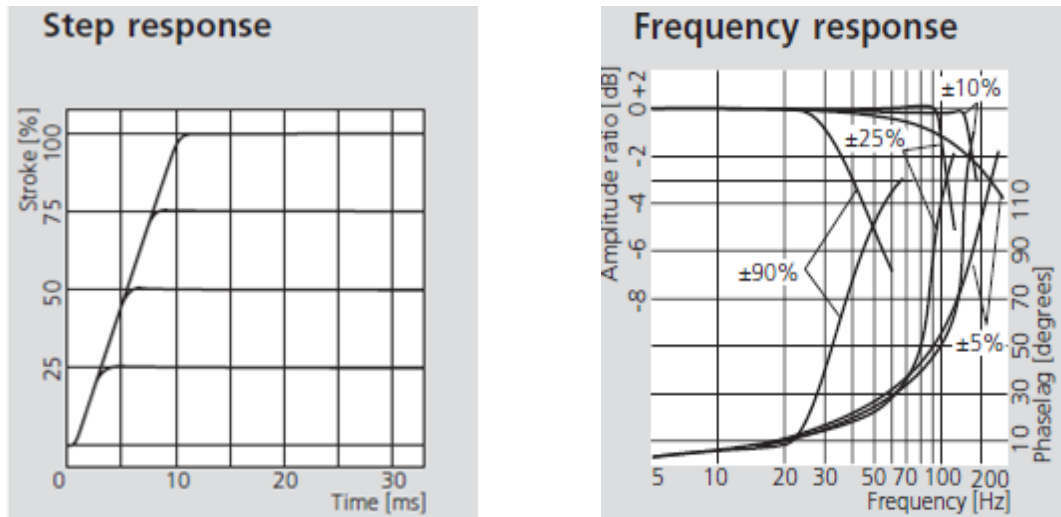


Figure 4.12 Step response and Frequency response at flow rate of 40/80 l/min.

The frequency response indicates that the dynamics can be estimated by using a second order transfer function. Signals used, are considered small in the $\pm 10\%$ range. For performing the estimation, the transfer function of a second-order system is considered.

$$G(s) = \frac{K}{\frac{1}{\omega_n^2}s^2 + \frac{2\zeta}{\omega_n}s + 1} \quad (2.16)$$

Where,

- K: gain
- ζ : damping of the system dynamics
- ω_n : eigen frequency of the system dynamics ([rad/s])

From the figure 2.5 it is clearly seen that the gain is $k=1$ as the amplitude is zero for low frequencies. Furthermore, the lack of a resonance peak indicates that the damping is in the interval $0.7 \leq \zeta \leq 1$. ζ is estimated to be 1, which is also the most conservative, with regards to fast performance. The eigen frequency of the system is found as the frequency at -90 degrees phase lag.

5. Linear Model of the system:

This chapter describes the creation of the Linear model. The description will contain the linearisation of the equations from the previous chapter. The linearised equations will be Laplace-transformed and used to create the block diagram from which the transfer functions of the linear system will be derived. Along with the linearisation will follow an assessment of the operating point around which the linearisation of the systems equations will be done. Furthermore, a comparison between the output of the linear and non-linear mode will be made in order to show agreement between the two models.

5.1 Linearisation of the equations of the system

The equation of the subsystem will linearised, below listed the previous equation will be linearised.

$$Q_p = A_p \dot{y} + \frac{V_0}{\beta} \frac{dP_p}{dt} + \frac{A_p}{\beta} \frac{dP_p}{dt} \quad (5.1)$$

$$-Q_r = -A_r \dot{y} + \frac{V_0}{\beta} \frac{dP_r}{dt} + \frac{A_r}{\beta} \frac{dP_r}{dt} \quad (5.2)$$

Subtract the equation (5.1) & (5.2)

$$Q_p + Q_r = (A_p + A_r) \dot{y} + \frac{V_0}{\beta} \left(\frac{dP_p}{dt} - \frac{dP_{pr}}{dt} \right) + \frac{y}{\beta} (A_p \frac{dP_p}{dt} + A_r \frac{dP_r}{dt}) \quad (5.3)$$

$$Q_p = K_T \cdot u_v \sqrt{\frac{2}{\rho} (P_s - P_p)} \quad (5.5)$$

$$Q_r = K_T \cdot u_v \sqrt{\frac{2}{\rho} (P_r - P_t)} \quad (5.6)$$

Equation (5.1) and (5.2) are linearised by neglecting the control volumes dependence of the position (x) of the piston. Doing this, can be justified by the assumption that $A_p \frac{dP_p}{dt}$ and $A_r \frac{dP_r}{dt}$ is only a small displacement difference around the operating point (small signals).

$$Q_p + Q_r = (A_p + A_r) \dot{y} + \frac{V_0}{\beta} \left(\frac{dP_p}{dt} - \frac{dP_{pr}}{dt} \right) \quad (5.7)$$

$$\left(\frac{dP_p}{dt} - \frac{dP_{pr}}{dt} \right) = \dot{P}_L$$

$$Q_p + Q_r = (A_p + A_r) \dot{y} + \frac{V_0}{\beta} \dot{P}_L \quad (5.8)$$

Equation (5.8) is linearised by using Taylor series expansion for a system with two inputs and one output, $Q_L = f(x, P_L)$, Taylor series equations can be written as:

$$Q_L = Q_{L0}(x_o, P_{L0}) + \frac{\partial Q_L}{\partial x} | (x, x_0) + \frac{\partial Q_L}{\partial P_L} | (P_L, P_{L0}) \quad (5.9)$$

$Q_L = Q_L(x, P_L)$ is a non linear function

where,

$$K_q = \frac{\partial Q_L}{\partial x} | U_v = \bar{U}_v ; P_p = \bar{P}_p \quad (5.9)$$

$$K_c = -\frac{\partial Q_L}{\partial P_L} | U_v = \bar{U}_v ; P_r = \bar{P}_r \quad (5.10)$$

$$Q_L = K_q x - K_c P_L \quad (5.11)$$

Substitute the equation (5.11) in (5.8)

$$K_q x - K_c P_L = (A_p + A_r) \dot{y} + \frac{V_0}{\beta} \dot{P}_L \quad (5.12)$$

Linearised equation:

$$\frac{V_0}{\beta} \dot{P}_L + K_c P_L = K_q x - (A_p + A_r) \dot{y} \quad (5.13)$$

5.2 Complete Hydro-Mechanical equation:

The equation of motion will be linearised by neglecting friction as, only changing variables are considered in the linear model. Thus the equation of hydro-mechanical equation becomes:

$$m_{eq} \ddot{y} = P_p A_p - P_r A_r \quad (5.14)$$

$$\frac{V_0}{\beta} \dot{P}_L + K_c P_L = K_q x - (A_p + A_r) \dot{y} \quad (5.15)$$

6. Simulation Using Simhydraulics in Matlab:

6.1 Introduction of simhydraulics:

SimHydraulics software is the extension of Simscape platform product, expanding its capabilities to model and simulate hydraulic elements and devices.

SimHydraulics software requires these products:

- MATLAB
- Simulink
- Simscape

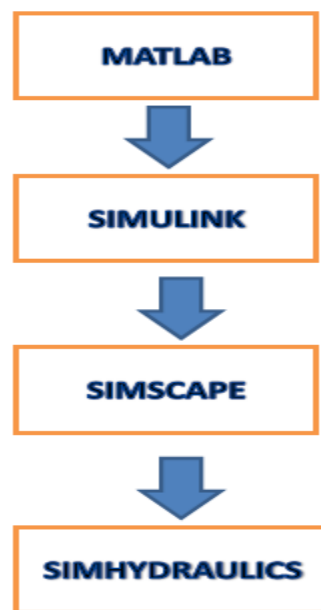


Fig.6.1 Procedure to model the simhydraulics simulation

6.2 Overview of MATLAB Product:

MATLAB is a high-level technical computing language and interactive environment for algorithm development, data visualization, data analysis, and numeric computation. Using the MATLAB product, you can solve technical computing problems faster than with traditional programming languages, such as C, C++, and Fortran.

You can use MATLAB in a wide range of applications, including signal and image processing, communications, control design, test and measurement, financial modeling and analysis, and computational biology. Add-on toolboxes (collections of special-purpose MATLAB functions, available separately) extend the MATLAB environment to solve particular classes of problems in these application areas.

MATLAB provides a number of features for documenting and sharing your work. You can integrate your MATLAB code with other languages and applications, and distribute your MATLAB algorithms and applications.

Features include:

- High-level language for technical computing
- Development environment for managing code, files, and data
- Interactive tools for iterative exploration, design, and problem solving
- Mathematical functions for linear algebra, statistics, Fourier analysis, filtering, optimization, and numerical integration
- 2-D and 3-D graphics functions for visualizing data
- Tools for building custom graphical user interfaces

6.2.1 Simulink:

Simulink is a graphical extension to MATLAB for the modeling and simulation of systems. In Simulink, systems are drawn on screen as block diagrams. Many elements of block diagrams are available (such as transfer functions, summing junctions, etc.), as well as virtual input devices (such as function generators) and output devices (such as oscilloscopes). Simulink is integrated with MATLAB and data can be easily transferred between the programs. In this tutorial, we will introduce the basics of using Simulink to model and simulate a system.

Simulink is supported on Unix, Macintosh, and Windows environments, and it is included in the student version of MATLAB for personal computers. For more information on Simulink, contact the MathWorks. The idea behind these tutorials is that you can view them in one window while running Simulink in another window. Do not confuse the windows, icons, and menus in the tutorials for your actual Simulink windows. Most images in these tutorials are not live - they simply display what you should see in your own Simulink windows. All Simulink operations should be done in your Simulink windows.

The Simulink Library Browser window should now appear on the screen. Most of the blocks needed for modeling basic systems can be found in the subfolders of the main "Simulink" folder (opened by clicking on the "+" in front of "Simulink"). Once the "Simulink" folder has been opened, the Library Browser window should look like:

6.2.1.1 Basic Elements

There are two major classes of elements in Simulink: blocks and lines. Blocks are used to generate, modify, combine, output, and display signals. Lines are used to transfer signals from one block to another.

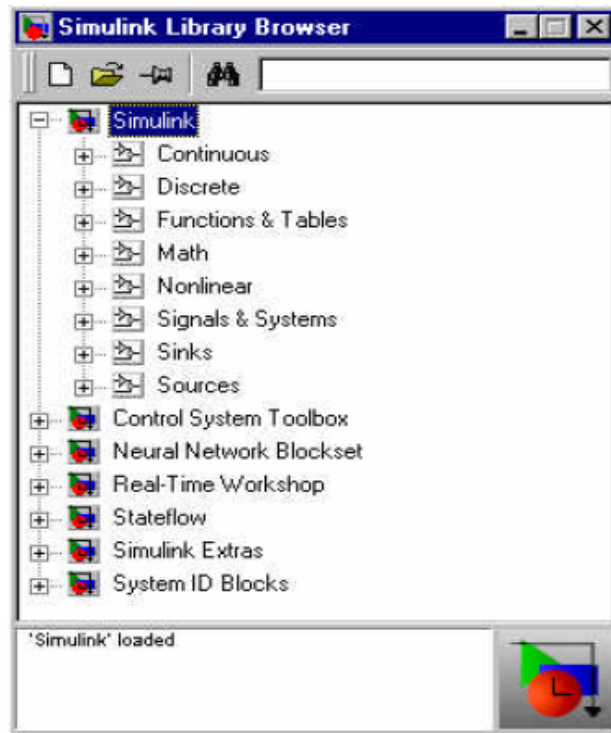


Fig 6.2 Simulink library browser window

The subfolders underneath the "Simulink" folder indicate the general classes of blocks available for us to use:

- Continuous: Linear, continuous-time system elements (integrators, transfer functions, state-space models, etc.)
- Discrete: Linear, discrete-time system elements (integrators, transfer functions, statespace models, etc.)
- Functions & Tables: User-defined functions and tables for interpolating function values
- Math: Mathematical operators (sum, gain, dot product, etc.)
- Nonlinear: Nonlinear operators (coulomb/viscous friction, switches, relays, etc.)
- Signals & Systems: Blocks for controlling/monitoring signal(s) and for creating subsystems
- Sinks: Used to output or display signals (displays, scopes, graphs, etc.)
- Sources: Used to generate various signals (step, ramp, sinusoidal, etc.)

6.2.2 Simscape:

Simscape software extends the Simulink product line with tools for modeling and simulating multidomain physical systems, such as those with mechanical, hydraulic, pneumatic, thermal, electrical, and electromagnetic components. Unlike other Simulink blocks, which represent mathematical operations or operate on signals, Simscape blocks represent physical components or relationships directly. With Simscape blocks, you build a model of a system just as you would assemble a physical system.

Simscape models employ a Physical Network approach to model building: components (blocks) corresponding to physical elements such as pumps, motors, and op-amps, are joined by lines corresponding to the physical connections that transmit power. This approach lets you describe the physical structure of a system rather than the underlying mathematics. From your model, which closely resembles a schematic, Simscape technology automatically constructs equations that characterize the behavior of the system. These equations are integrated with the rest of the Simulink model.

Simscape software runs within the Simulink environment and interfaces seamlessly with the rest of Simulink product family and with the MATLAB® technical computing environment.

6.2.2.1 Simscape Block Libraries

Simscape foundation libraries contain a comprehensive set of basic elements and building blocks, such as:

- Mechanical building blocks for representing one-dimensional translational and rotational motion.
- Electrical building blocks for representing electrical components and circuits.
- Magnetic building blocks that represent electromagnetic components.
- Hydraulic building blocks that model fundamental hydraulic effects and can be combined to create more complex hydraulic components.
- Pneumatic building blocks that model fundamental pneumatic effects based on the ideal gas law.
- Thermal building blocks that model fundamental thermal effects.
- Physical Signals block library that lets you perform math operations on physical signals, and graphically enter equations inside the physical network.

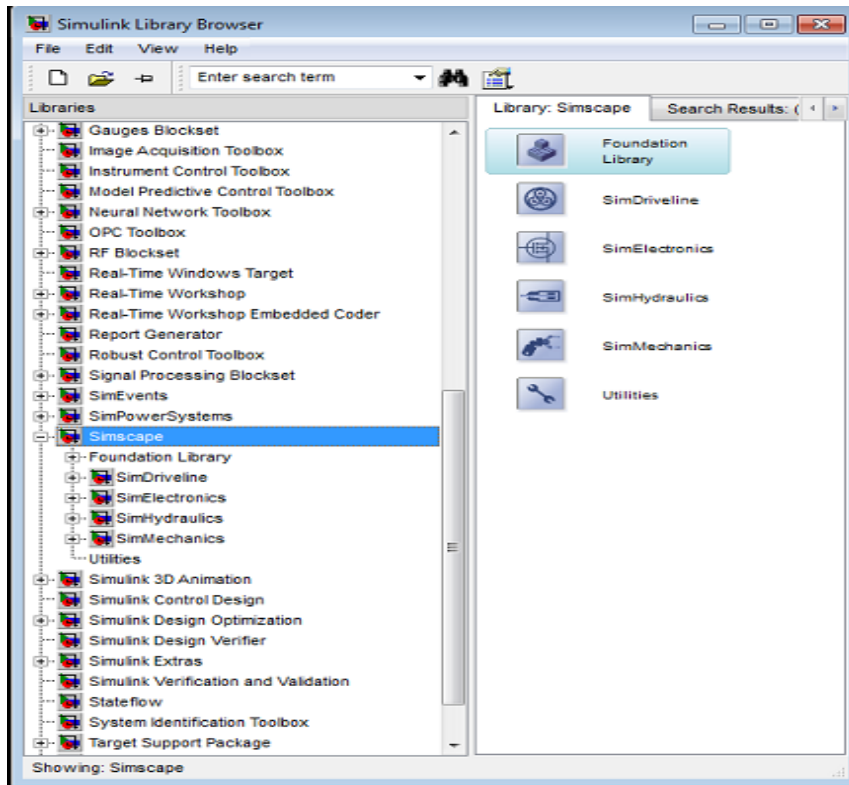


Fig 6.3 Simscape library window

6.3 Simhydraulics:

Simscape modeling environment provides the Physical Network approach for modeling and solving systems under design as one-dimensional networks. SimHydraulics software utilizes these basic modeling principles and contains a library of specialized hydraulic blocks that seamlessly interact with basic Simscape blocks.

SimHydraulics models are essentially Simscape block diagrams. When building a SimHydraulics model, you use a combination of SimHydraulics blocks with the blocks from the Simscape Foundation and Utilities libraries. Each SimHydraulics diagram must have at least one Solver Configuration block from the Simscape Utilities library. You can use basic hydraulic, electrical, and one-dimensional translational and rotational mechanical elements from the Simscape Foundation library and directly connect them to SimHydraulics blocks. You can also use basic Simulink blocks, such as sources and scopes, but you need to connect them through the Simulink-PS Converter and PS-Simulink Converter blocks from the Simscape Utilities library. You can also use these converter blocks to specify the desired input and output signal units. For more information on specifying units in SimHydraulics diagrams, see “Physical Units” in the Simscape documentation.

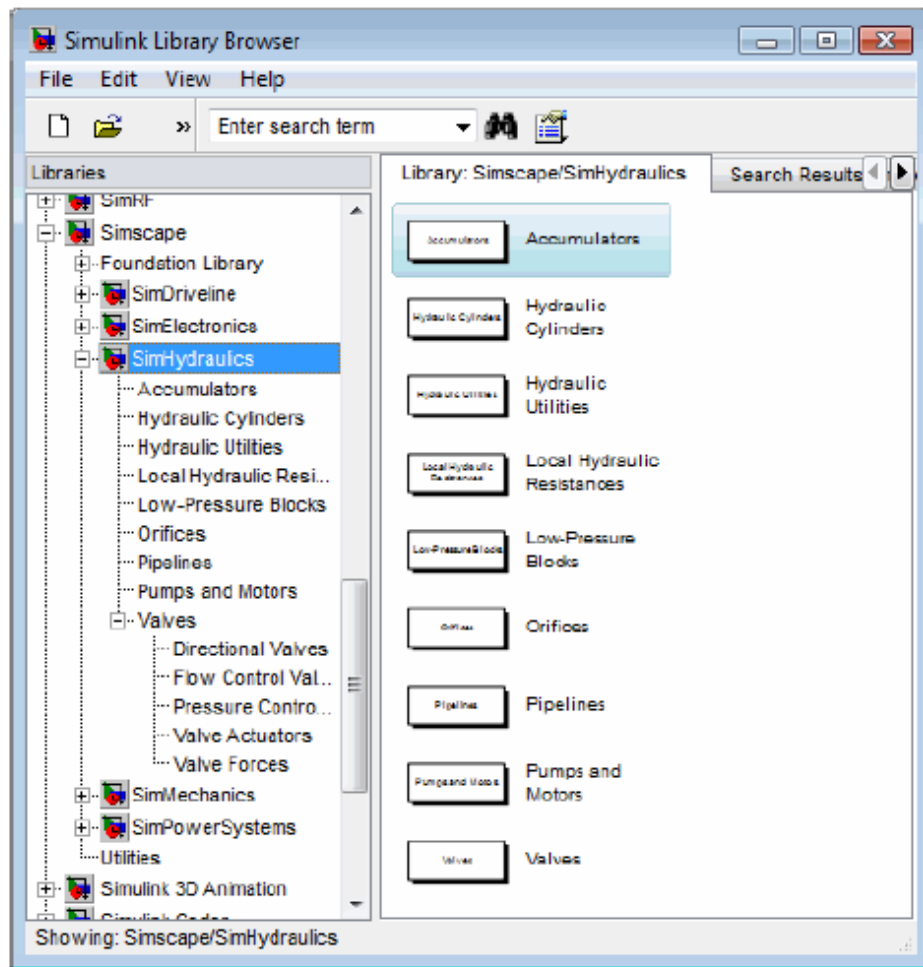


Fig 6.4 Simhydraulic library window

6.3 Running of Simhydraulics Models:

SimHydraulics software gives you multiple ways to simulate and analyze hydraulic power and control systems in the Simulink environment. Running a hydraulic simulation is similar to running a simulation of any other Simscape model. See “Model Simulation” in the Simscape documentation for a discussion of the following topics:

- Explanation of how SimHydraulics software validates and simulates a model
- Specifics of using Simulink linearization commands in SimHydraulics models
- Generating code for SimHydraulics models
- Restrictions and limitations on using Simulink tools in SimHydraulics models

All these aspects of simulating SimHydraulics models are exactly the same as for Simscape models.

7. Modeling of hydraulic systems:

7.1 Introducing the main three libraries in Simscape modeling:

SimHydraulics software uses the Simscape library as its main library. Simscape modeling environment provides the Physical Network approach for modeling and solving systems under design as one-dimensional networks.

SimHydraulics software utilizes these basic modeling principles and contains a library of specialized hydraulic blocks that seamlessly interact with basic Simscape blocks. When modeling hydraulic power and control systems, you use the following Simscape libraries:

- Foundation library — Contains basic hydraulic, mechanical, and physical signal blocks
- SimHydraulics library — Contains advanced hydraulic diagram blocks, such as valves, cylinders, pipelines, pumps, and accumulators
- Utilities library — Contains essential environment blocks for creating physical Networks models

You can combine all these blocks in your SimHydraulics diagrams to model hydraulic systems. You can also use the basic Simulink blocks in your diagrams, such as sources or scopes.

7.2 Building of hydraulic model from hydraulic schematic circuit:

The following schematic represents the model are about to build. It contains a double-acting hydraulic cylinder with single rod, which is controlled by an high response servo-valve (D661) Moog. The cylinder drives a load consisting of a mass, damping, and preloaded spring.

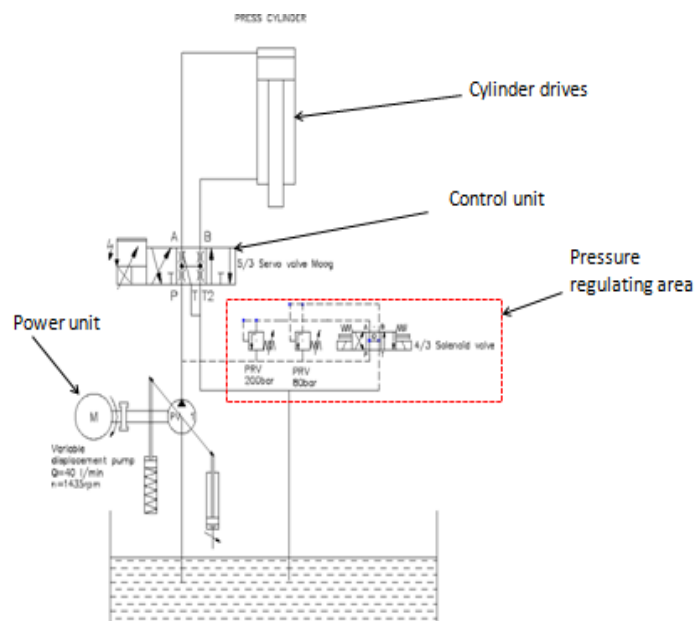


Fig 7.1 Schematic diagram of hydraulic circuit

The power unit consists of a motor, a variable-displacement pump, and motor. Pressure regulating devices which are the two pressure relief valves and 4/3 solenoid valve. Depending on its characteristics, such a power unit can be modeled in a variety of ways, as described in “Modeling Power Units”. In this example, the pump unit is assumed to be powerful enough to maintain two variable pressures at the valve inlet. Therefore, we are going to represent it in the diagram by a Hydraulic Pressure Source block.

7.3 Modeling of Power unit in Simhydraulic :

To create an equivalent SimHydraulics diagram for power unit, follow these steps below:

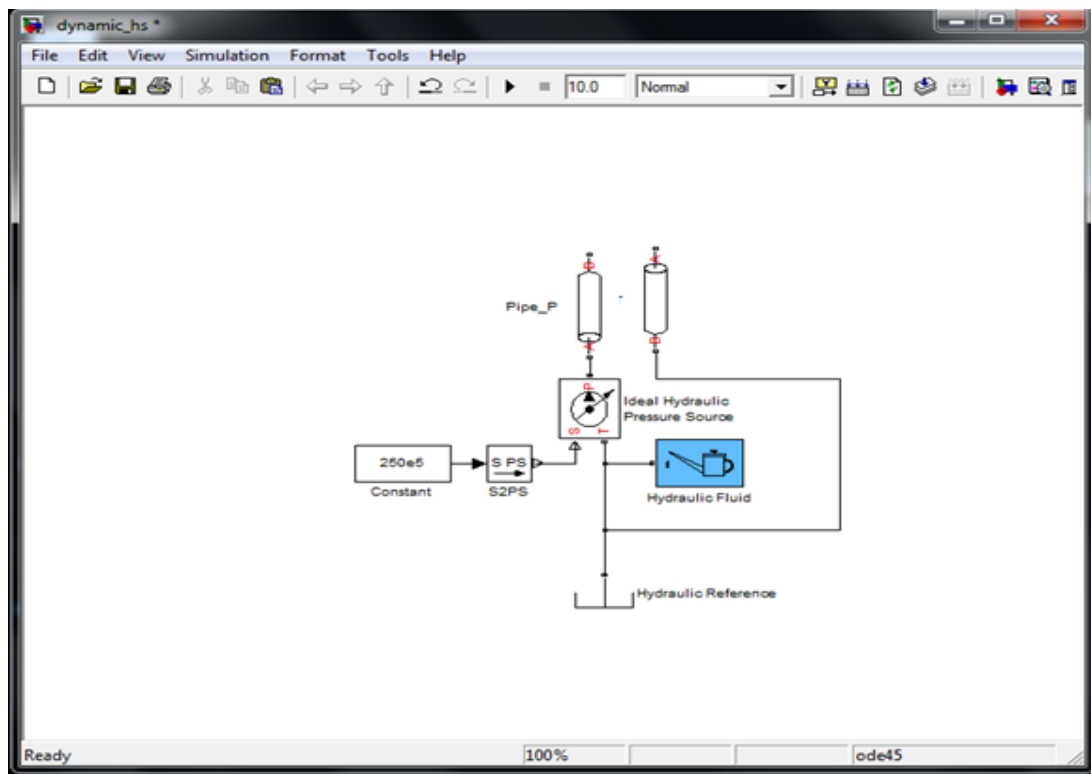


Fig 7.2 Modeling of Power unit in Simhydraulic

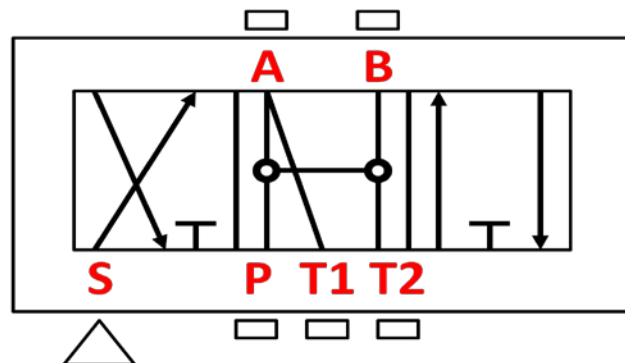
1. Open the Simscape and Simulink block libraries, as described in “Introducing the SimHydraulics Block Libraries”
2. Create a new model. To do this, click the New button on the Library Browser’s toolbar or choose New from the library window’s File menu and select Model. The software creates an empty model in memory and displays it in a new model editor window.
3. Open the Simscape > Foundation Library > Hydraulic > Hydraulic Sources library and drag the Hydraulic Pressure Source block into the model window.
4. Open the Simscape > Foundation Library > Hydraulic > Hydraulic Element library and place the Hydraulic reference block into the model window.

5. To model the hydraulic fluid, open the Simscape > SimHydraulics > Hydraulic Utilities library and Place the hydraulic fluid block into the model window.
6. Input pressure is 250 bar, assigned the pressure through constant block into the model. Open the simulink>Sources library and place the constant block into the window.
7. To convert the unitless simulink input signal to a physical signal. To assign the simulink to physical block open the simscape>Utilities library and the place the simulink input signal to a physical signal block.
8. The connection pipe lines are two types used in this hydraulic system. One is rigid and other is flexible in the system. Parameter of the pipes details will listed later. To place the pipe lines into the model window are open the Simscape>Simhydraulics>found the hydraulic pipeline in Pipelines library.

7.4 Designing of Customized 5/3 directional proportional valve (Moog)

The purpose of custom this valve is very important task into this system. There is no more block similar of 5/3 directional valve in simscape valves library. So we analyzed and custom the valve to the system.

MOOG:
5 way 3 position
Directional valve



Throughout SimHydraulics libraries, hydraulic ports are identified with the following symbols:

- P — Pressure port
- T, T₂ — Return (tank) port
- A, B — Actuator ports
- S — Signal port

The 5-way valve in its simplest form is built of six variable orifices. In the equivalent circuit, they are named P-A, P-B, B-T₂, T₁₁-T₁, T₂₁₁-T₂, A-T₁, P-T₁₁, P-T₂₁₁. The Variable Orifice block, which is the most generic model of a variable orifice in the SimHydraulics libraries, is used in this particular example. You can use any other variable orifice blocks if the real valve design employs a configuration backed by a stock model, such as an orifice with round holes or rectangular slots, poppet, ball, or needle. In general, all orifices in the model can be simulated with different blocks or

with the same block, but with different way of parameterization. For instance, six orifices can be represented by their pressure-flow characteristics (for details, see the Variable Orifice block reference page).

Use the valve for applications with two actuators, each being controlled by a valve of this type. When both valves are in neutral position, the pump is unloaded. If any of the valves is shifted from neutral, the diverting line is cut off and the respective actuator is fed at fuel pump pressure. The figure shown below the configuration of 5/3 directional proportional valve. This valve unloads the pump in neutral position (P-T11, P-T211) and requires six variable orifice blocks. The Orifice with Variable Area Round Holes blocks have been used as a variable orifice in this model. Port T corresponds to an intermediate point between ports P and T2.

All blocks are controlled by the same position signal, provided through the physical signal port S, but the Orifice orientation parameter in the block instances is set in such a way that positive signal at port S opens the orifices colored blue in the block diagram (orifices P-A, B-T2) and closes the orifices colored yellow (orifices A-T1, P-B).

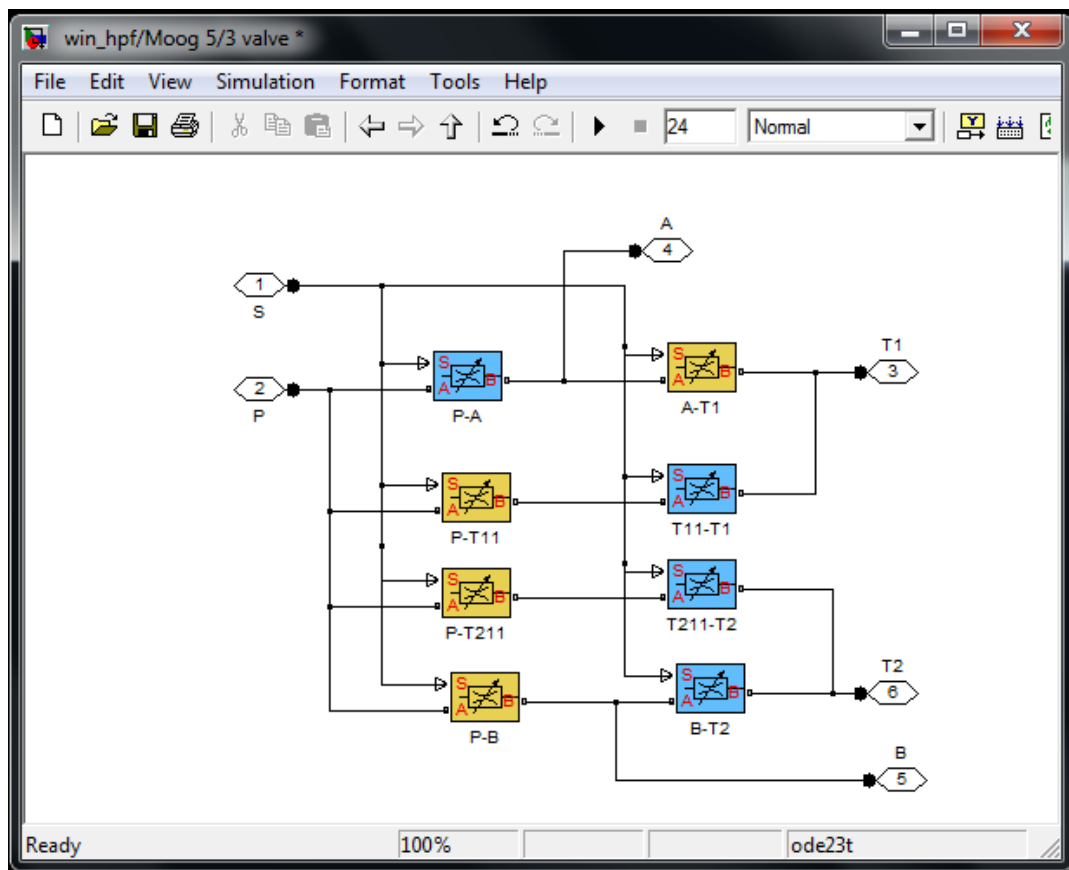


Fig 7.3 Modeling of Custom valve in simhydraulic

As a result, the openings of the orifices are computed as follows

$$h_{PA} = h_{PA0} + x$$

$$h_{PB} = h_{PB0} - x$$

$$h_{BT2} = h_{PBT20} + x$$

$$h_{AT1} = h_{AT10} - x$$

$$h_{PT11} = h_{PT110} - x$$

$$h_{T11T1} = h_{T11T10} + x$$

$$h_{PT211} = h_{PT2110} - x$$

$$h_{T211T2} = h_{T211T20} + x$$

where ,

h_{PA}	Orifice opening for the Variable Orifice P-A block
h_{PB}	Orifice opening for the Variable Orifice P-B block
h_{AT1}	Orifice opening for the Variable Orifice A-T1 block
h_{BT2}	Orifice opening for the Variable Orifice B-T2 block
h_{PT11}	Orifice opening for the Variable Orifice P-T11 block
h_{T11T1}	Opening for the Variable Orifice T11-T1 block
h_{PT211}	Orifice opening for the Variable Orifice P-T211 block
h_{T211T2}	Orifice opening for the Variable Orifice T211-T2 block
h_{PA0}	Initial opening for the Variable Orifice P-A0 block
h_{PB0}	Initial opening for the Variable Orifice P-B0 block
h_{AT1}	Initial opening for the Variable Orifice A-T1 block
h_{BT2}	Initial opening for the Variable Orifice B-T2 block
h_{PT11}	Initial opening for the Variable Orifice P-T11 block
h_{T11T1}	Initial opening for the Variable Orifice T11-T1 block
h_{PAT211}	Initial opening for the Variable Orifice P-AT211 block
h_{T211T2}	Initial opening for the Variable Orifice T211-T2 block
x	Control member displacement from initial position

7.5 Signal Builder

The Signal Builder block displays and allows you to create or edit interchangeable groups of signal sources and quickly switch the groups into and out of a model.

Signal groups can greatly facilitate testing a model, especially when you use them with conjunction with Simulink Assertion blocks and the Model Coverage Tool from the Simulink Verification and Validation. For a description of the Model Coverage Tool, see the Simulink Verification and Validation User's Guide

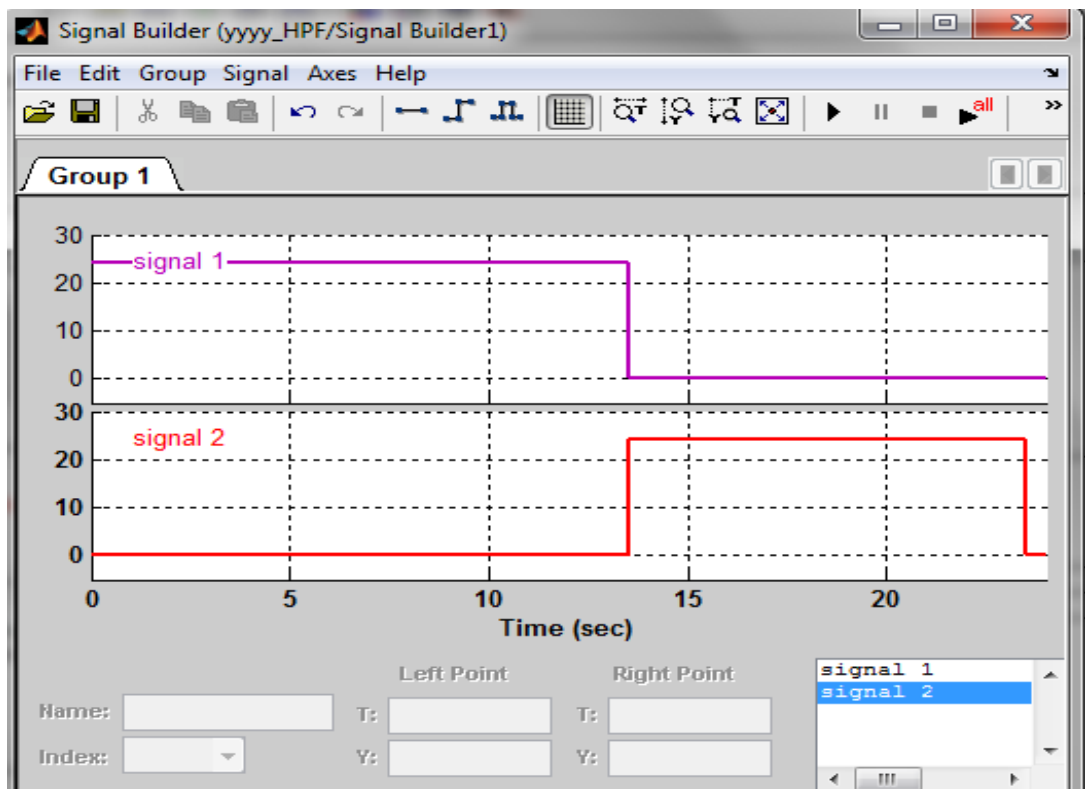


Fig 7.4 Signal builder in simhydraulic

Signal-1 (Operating time 0sec-13.5secs)

Our aim is to operate hydraulic press in low pressure and less force for first 13.5 seconds, so to obtain this operating condition we build signal which is to actuate 80bar pressure relief valve .Input pressure 250bar from hydraulic power source is now connect 80bar PRV(Pressure relief valve), so that excess pressure will drain to tank and our system will work on low pressure.

Signal-2 (Operating time 13.5secs-23.5secs)

This signal is build to operate hydraulic system with high pressure and high force.

During this condition signal-2 will actuates position B of directional valve and in turns pressure port with 250bar inlet pressure is connected with 200bar pressure relief valve , so that excess pressure will be drain to the tank and the system will work on 200bar pressure for the given operating time.

7.6 Double acting single rod cylinder in Simulink:

This block represents a double-acting hydraulic cylinder. The model of the cylinder is built of the following building blocks: Translational Hydro-Mechanical Converter, Piston Chamber, Translational Hard Stop, and Ideal Translational Motion Sensor. The rod motion is limited with the mechanical Translational Hard Stop block.

Connections R and C are mechanical translational conserving ports corresponding to the cylinder rod and cylinder clamping structure, respectively. Connections A and B are hydraulic conserving ports. Port A is connected to chamber A and port B is connected to chamber B. The block directionality is adjustable and can be controlled with the Cylinder orientation parameter.

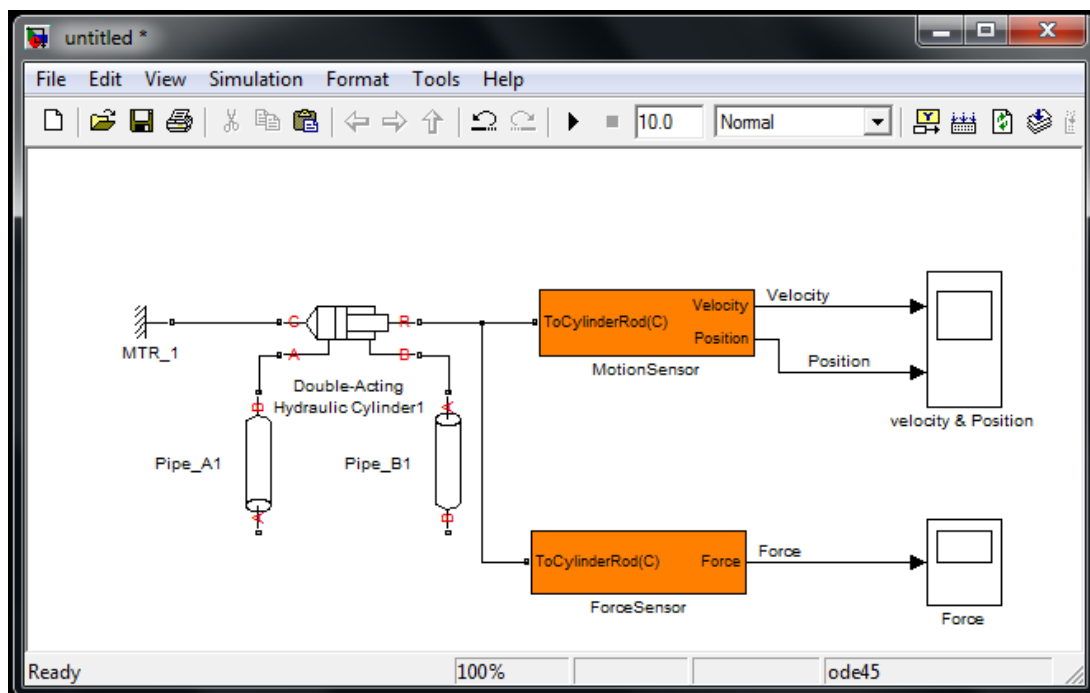


Fig 7.5 Modeling of Double acting cylinder in simhydraulic

7.6.1 Cylinder and valve Area Parameter:

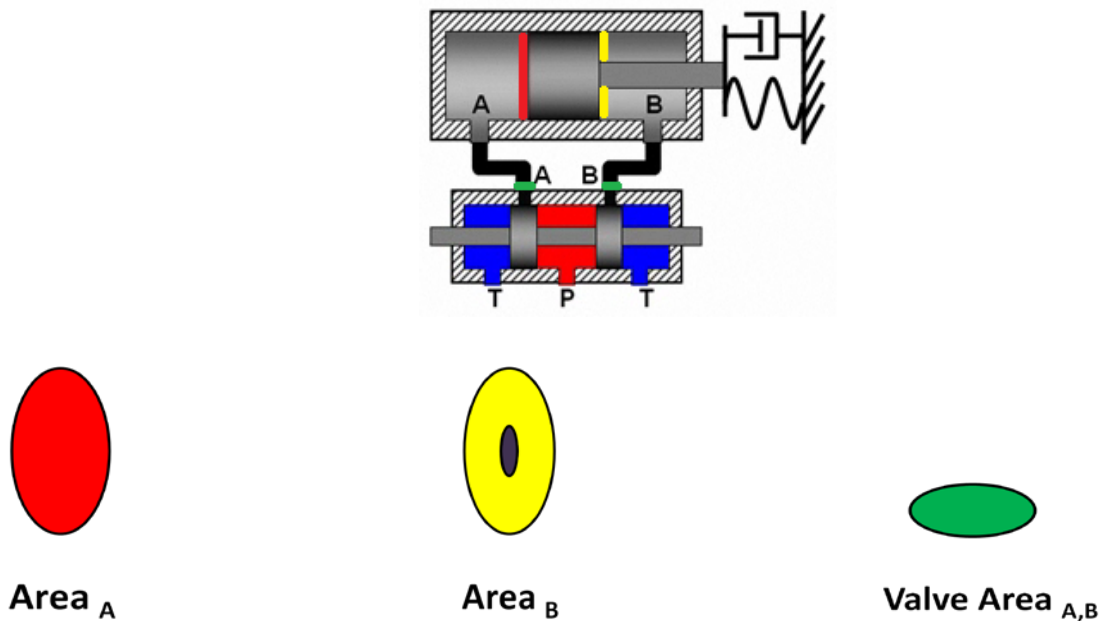


Fig 7.6 Cylinder and valve area parameters

Area parameter:

$$\text{Piston Area } A = 5.026 \times 10^{-3} \text{ m}^2$$

$$\text{Rod Area } B = 2.563 \times 10^{-3} \text{ m}^2$$

$$\text{Valve area } A,B = 1.038 \times 10^{-4} \text{ m}^2$$

Model the mechanical load for the cylinder. Open the Simscape > Foundation Library > Mechanical > Translational Elements library and add the Mass, Translational Spring, Translational Damper, and Translational Friction blocks to the diagram.

To indicate that the cylinder case is fixed, connect port C of the Double-Acting Hydraulic Cylinder block to one of the Mechanical Translational Reference blocks. To rotate the Mechanical Translational Reference block, select it and press Ctrl+R. You can also shorten the block name to MTR to make the diagram easier to read.

Connect the other blocks to port R of the Double-Acting Hydraulic Cylinder block, which are listed below.

1. Motion sensor
2. Force sensor

Motion sensor:

In motion sensor is connected to cylinder rod with the ideal translation sensor block, PS-Simulink converter, velocity and position scope.

Ideal translation sensor:

The block represents an ideal mechanical translational motion sensor, that is, a device that converts an across variable measured between two mechanical translational nodes into a control signal proportional to velocity and position. The sensor is ideal since it does not account for inertia, friction, delays, energy consumption, and so on.

Connections R and C are mechanical translational conserving ports and connections V and P are physical signal output ports for velocity and position, respectively. The block positive direction is from port R to port C.

PS-Simulink converter:

Converts the input Physical Signal to a unitless Simulink output signal.

The unit expression in 'Output signal unit' parameter must match or be commensurate with the unit of the Physical Signal and determines the conversion from the Physical Signal to the unitless Simulink output signal.

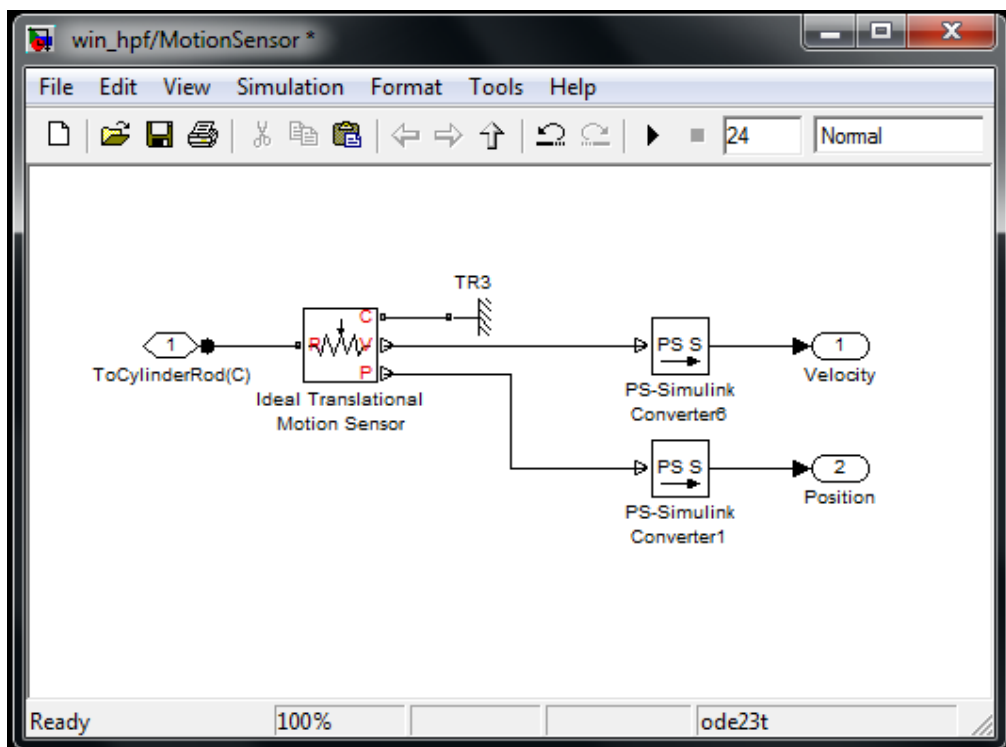


Fig 7.7 Modeling motion sensor connected with Double acting cylinder in Simhydraulic

Force sensor:

In force sensor is connected to cylinder rod with the Ideal force sensor, Mass, Translational damper, Translational friction, PS-Simulink converter.

Ideal Force Sensor:

The block represents an ideal force sensor, that is, a device that converts a variable passing through the sensor into a control signal proportional to the force with a specified coefficient of proportionality. The sensor is ideal since it does not account for inertia, friction, delays, energy consumption, and so on.

Connections R and C are mechanical translational conserving ports that connect the sensor to the line whose force is being monitored. Connection F is a physical signal port that outputs the measurement result. The sensor positive direction is from port R to port C

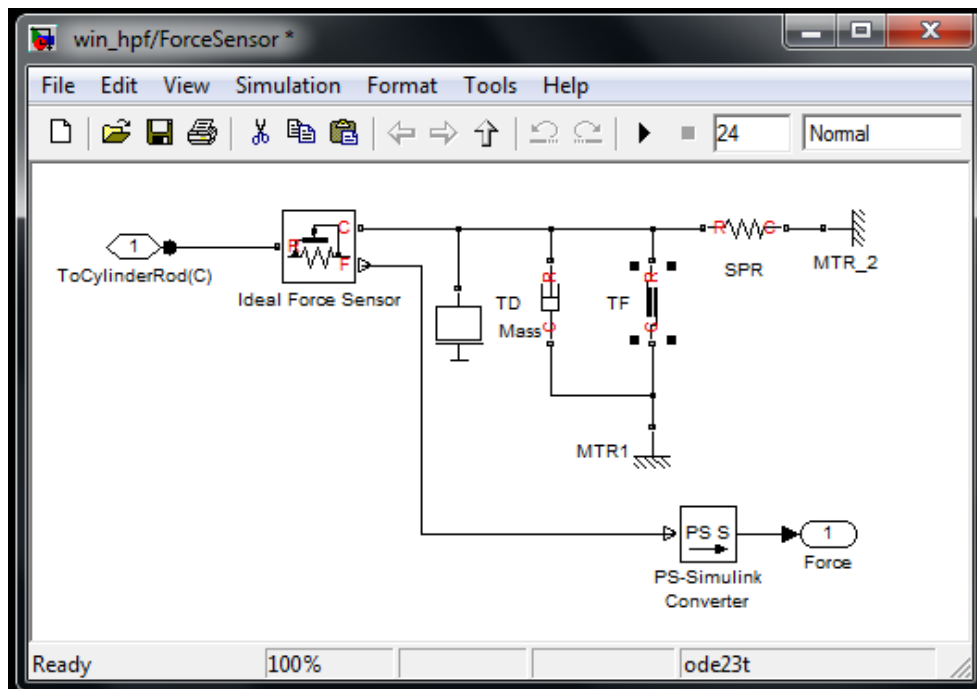


Fig 7.8 Modeling of Force sensor connected with Double acting cylinder in Simhydraulic

Translational Damper:

The block represents an ideal mechanical translational viscous damper.

Connections R and C are mechanical translational conserving ports, with R representing the damper rod, while C is associated with the damper case. The block positive direction is from port R to port C.

Translation Friction:

The block represents friction in the contact between moving bodies. The friction force is simulated as a function of relative velocity and assumed to be the sum of Stribeck, Coulomb, and viscous components. The sum of the Coulomb and Stribeck frictions at zero velocity is often referred to as the breakaway friction.

Connections R and C are mechanical translational conserving ports. The block positive direction is from port R to port C. This means that if port R velocity is greater than that of port C, the block transmits force from port R to port C.

Mass:

The block represents an ideal mechanical translational mass.

The block has one mechanical translational conserving port. The block positive direction is from its port to the reference point. This means that the inertia force is positive if mass is accelerated in positive direction.

Parameter:

Mass = 30Kg

Initial velocity = 0 m/s

8. Control system analysis and Design

8.1 PID controller:

A **proportional–integral–derivative controller (PID controller)** is a generic control loop feedback mechanism (controller) widely used in industrial control systems – a PID is the most commonly used feedback controller. A PID controller calculates an "error" value as the difference between a measured process variable and a desired setpoint. The controller attempts to minimize the error by adjusting the process control inputs.

The PID controller calculation (algorithm) involves three separate constant parameters, and is accordingly sometimes called **three-term control**: the proportional, the integral and derivative values, denoted P , I , and D . Proportional(P) depends on the *present* error, Integral (I) on the accumulation of *past* errors, and Derivative (D) is a prediction of *future* errors, based on current rate of change. The weighted sum of these three actions is used to adjust the process via a control element such as the position of a control valve, or the power supplied to a heating element.

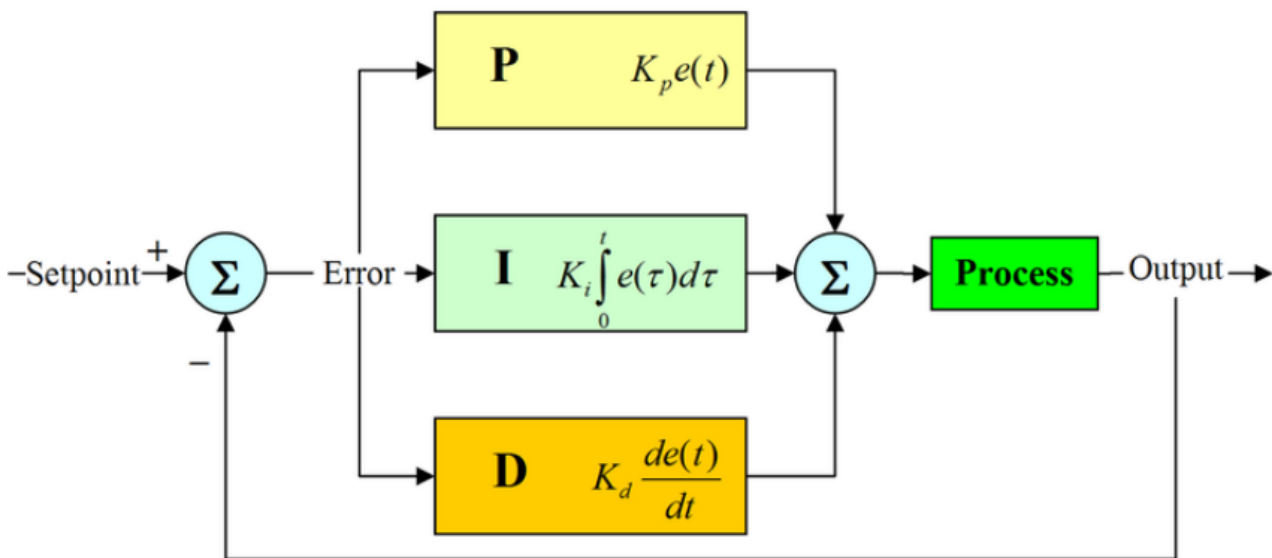


Fig. 8.1 PID controller

For most practical processes being controlled with a PID controller, these two wishes cannot be achieved simultaneously. In other words,

- The faster responses, the worse stability, and
- The better stability, the slower responses.

For a control system, it is more important that it has good stability than being fast. So, we specify:

Acceptable stability (good stability, but not too good as it gives too slow responses)

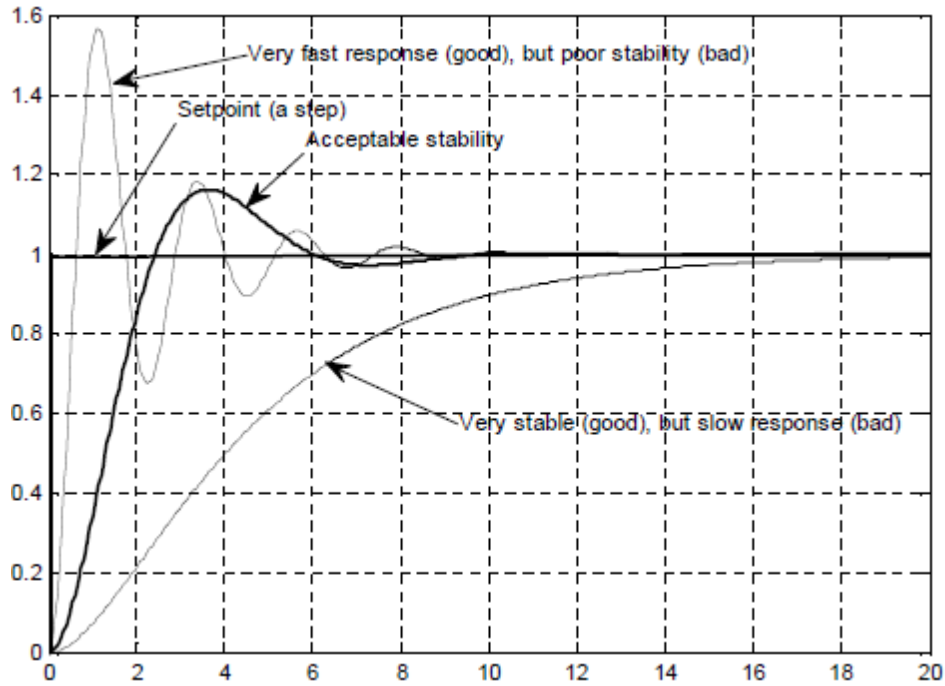


Fig. 8.2 In controller tuning we obtained acceptable stability of the control system

PID in closed loop hydraulic unit:

$$U(t) = k_p e(t) + k_i \int_0^t e(\tau) d\tau + k_d \dot{e}(t) \quad (8.1)$$

$$F(s) = U(s) = K_p E(s) + K_i \frac{E(s)}{s} + K_D s E(s) \quad (8.2)$$

$$F(s) = U(s) = K_p \left(1 + \frac{1}{T_i s} + T_d s \right) E(s) \quad (8.3)$$

$$F(s) = U(s) = R(s) E(s) \quad (8.4)$$

8.2 Transfer Function of Hydraulic System:

From Simhydraulics we designed the hydraulic circuit using with various hydraulic component blocks. It shown below,

A **transfer function** (also known as the **system function** or **network function**) is a mathematical representation, in terms of spatial or temporal frequency, of the relation between the input and output of a linear time-invariant system. With optical imaging devices, for example, it is the Fourier transform of the point spread function (hence a function of spatial frequency) i.e. the intensity distribution caused by a point object in the field of view.

8.2.1 Laplace transformation of linear model:

Before constructing the block diagram, the linear model will be laplace transformed from the timedomain to the frequency domain. Laplace transforming the set of equations for the subsystem and suitably arranging them, gives:

$$\frac{V_0}{\beta} s P_L(s) + K_c P_L(s) = K_q X(s) - (A_p + A_r) s Y(s) \quad (8.5)$$

The Laplace transform of the equation of motion are:

$$m_{eq} s^2 Y(s) = P_L(s)(A_p - A_r) \quad (8.6)$$

To find the transfer function of $G(s)$

$$G(s) = \frac{Y(s)}{X(s)}$$

$$P_L(s) = \frac{K_q X(s)}{\left(\frac{V_0}{\beta} s + k_c\right)} - \frac{(A_p + A_r)}{\left(\frac{V_0}{\beta} s + k_c\right)} s Y(s)$$

From equation (8.6) when consider the damping and spring stiffness in the hydraulic modeling, so the transfer function of $G(s)$

$$G(s) = \frac{Y(s)}{X(s)} = \frac{K_q (A_p - A_r)}{\left\{ (m_{eq}^* s^2 + C^* s + K^*) \left(\frac{V_0}{\beta} s + K_c \right) \right\} + (A_p^2 - A_r^2) s} \quad (8.7)$$

To find the closed loop transfer function, mathematical model is

$$H(s) = R(s)G(s) = R(s) \frac{Y(s)}{X(s)}$$

$$= K_p \left(1 + \frac{1}{T_i s} + T_d s \right) \frac{K_q (A_p - A_r)}{\left\{ (m_{eq}^* s^2 + C^* s + K^*) \left(\frac{V_0}{\beta} s + K_c \right) \right\} + (A_p^2 - A_r^2) s}$$

$$H(s) = \frac{K_p \left(s + \frac{1}{T_i} + T_d s^2 \right) K_q (A_p - A_r)}{s \left[\left\{ (m_{eq}^* s^2 + C^* s + K^*) \left(\frac{V_0}{\beta} s + K_c \right) \right\} + (A_p^2 - A_r^2) s \right]}$$

..... (8.8)

From the equation (8.8) we can find the poles and zeros

- Pole in the origin
- One pair of complex conjugate poles
- One real negative pole
- One pair of complex conjugate zeros

8.3 Block diagram of Hydro-Mechanical of the closed loop system:

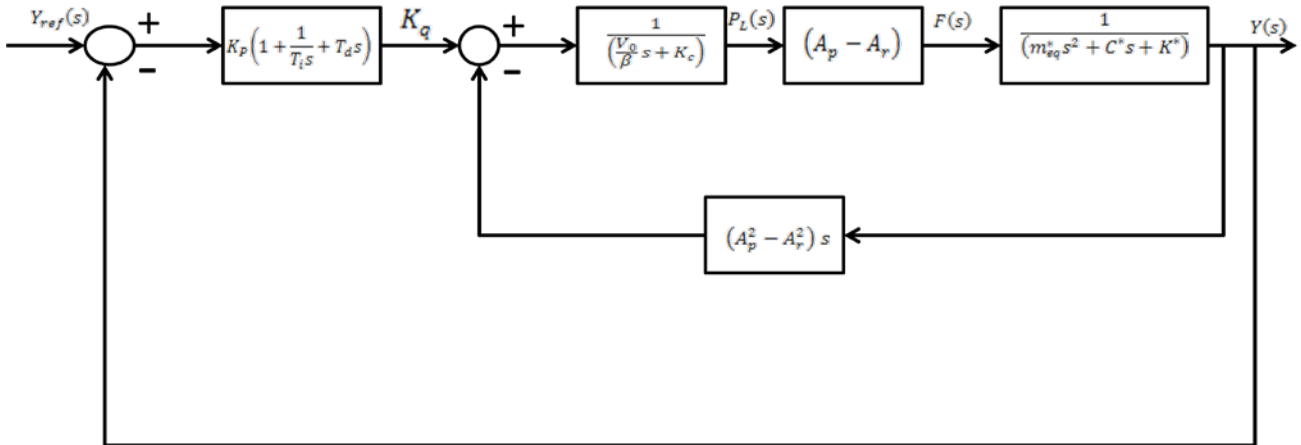


Fig. 8.3 Block diagram of the linear model

8.4 Simulation of the hydraulic system:

While Analyzing the structure and establishing dynamic working mathematical model and simulation model of priority valve lays a solid foundation for the optimization modeling of closed loop hydraulic system. To running the simulation file which input sine wave signal and to derive the output of force signal is shown in fig 8.4

Input of Frequency = 0.1 rad/sec

Constant Amplitude = 1

Magnitude = 543.5678

Phase = -1.8161

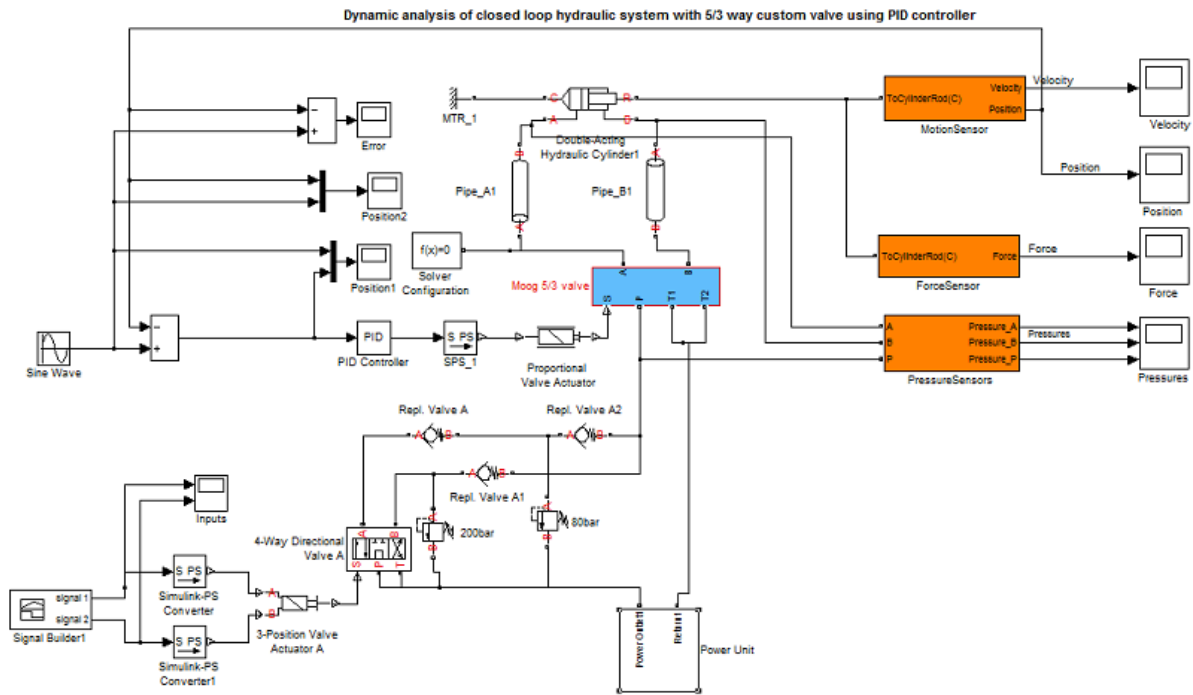


Fig 8.4 Modeling of closed loop hydraulic system with 5/3 custom valve using PID controller

8.5 Transfer function in Matlab:

The following below command are used to plot the transfer function of the hydraulic system. Here to plot the magnitude and phase with various frequency of the input signal.

```
close all
clear all
clc
```

```
[fname,pname] = uigetfile('*.fig','Load figure');
open([pname fname])
```

```
und = findstr(fname, '_');
fr = str2num(fname(und+1:end-4));
```

```
aa = get(gca);
bb = (aa.Children(1));
cc = get(bb);
x = cc.XData;
xx = 0:0.001:max(x);
yy = interp1(cc.XData,cc.YData,xx);
[mod,fas,frq] = ffg(yy,xx(2)-xx(1));
frq = frq*2*pi;
```

```
tfa = interp1(frq,mod,fr)
tff = interp1(frq,fas,fr)
```

```
%figure,plot(frq*2*pi,mod),grid
%figure,plot(frq*2*pi,fas),grid
```

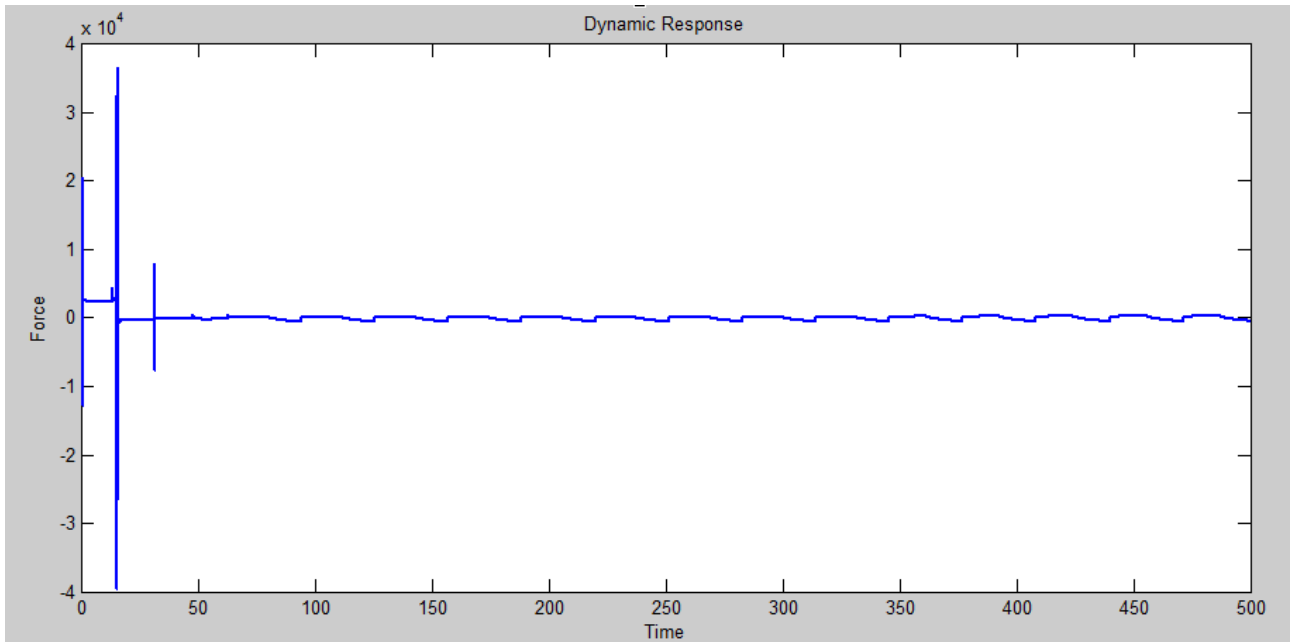


Fig 8.5 Plot graph for dynamic response of Force

To find the Magnitude and Phase of hydraulic system is used to increased value of frequency 0.1 to 10 rad/sec.

Frequency (rad/sec)	Amplitude	Magnitude	Phase
0.2		351.6131	-2.0356
0.4		242.9328	-2.0568
0.5		192.1376	-2.0558
0.7		146.2743	-1.9907
1.0		107.2543	-2.0041
1.3		101.9799	-2.0435
1.5		116.8649	-2.0530
1.6		118.5670	-2.1137
1.8		109.3407	-2.6080
1.9		135.7403	-2.3751
2.1		144.8846	-2.5866
2.2		181.4717	-2.7278
2.5		164.7039	-2.9661
3		119.0063	1.5115
4		97.8590	-1.0316
5		104.5885	2.1717
6		84.3295	2.8813
7		92.6361	2.7271
8		68.6595	2.5388
9		70.0963	2.6975
10		57.6504	2.5437

Table 8.1 Magnitude and Phase of closed loop transfer function

8.6 Dynamic response of the closed loop hydraulic system

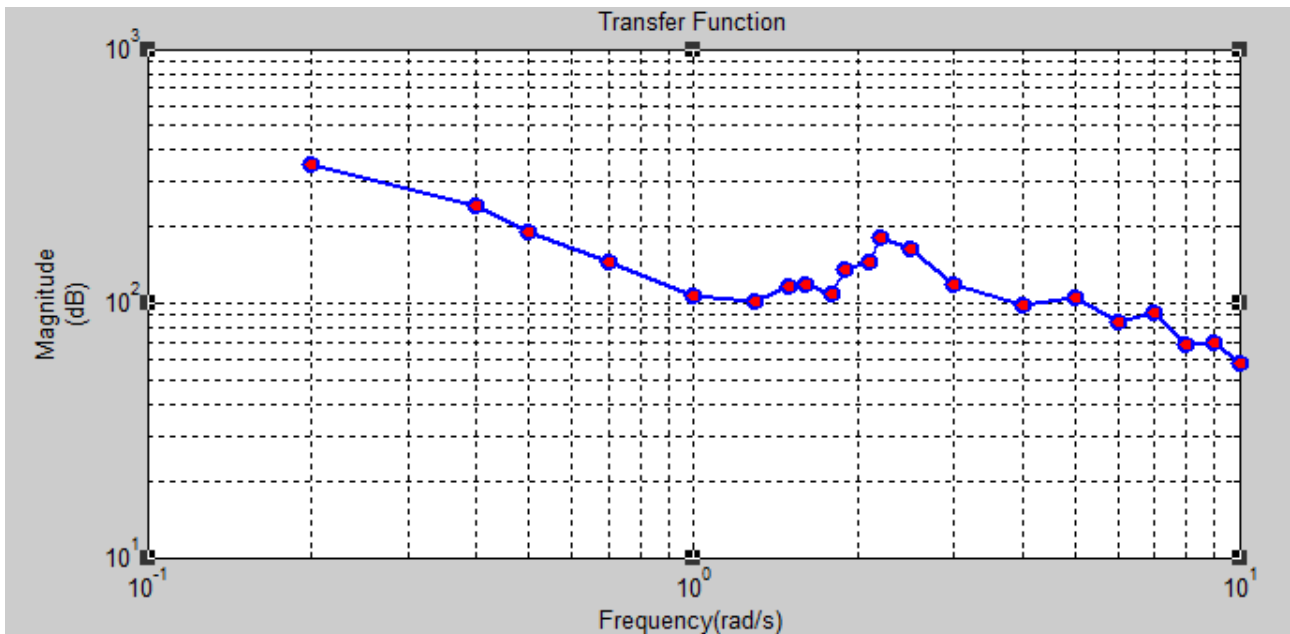


Fig 8.6 Plot graph dynamic response for Magnitude and Frequency (rad/sec)

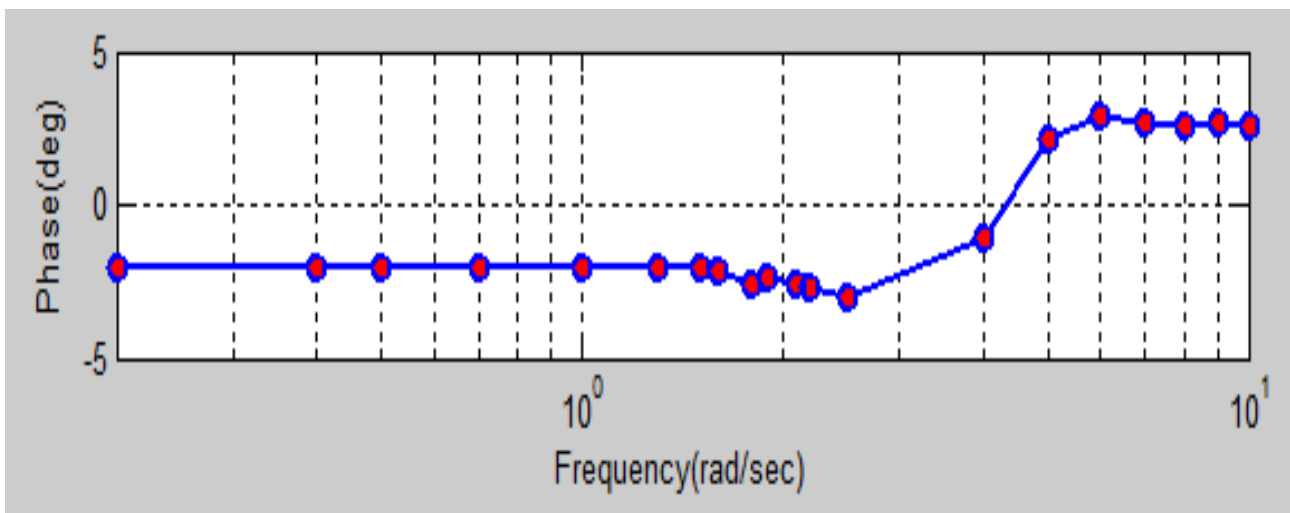


Fig 8.7 Plot graph dynamic response for Phase and Frequency (rad/sec)

9. Conclusion

In this paper we presented the design, analysis, and simulation of the hydraulic system. A dynamic model of a closed hydraulic system by utilizing the MATLAB-SIMULINK software, the effects of various design parameters on the overall response of the system are investigated. The verification of the simulation results with the experimental studies justifies the proposed dynamic model. Several various nonlinearities of the system are taken into account through the modeling process. For the finding the transfer function of closed loop hydraulic system with increasing the frequency and constant amplitude of given input signal. By simulating the MATLAB-SIMULINK we achieved the smooth gradual decreases the magnitude of the output force. There's a pole in the origin as expected by the model and the resonance peak at 2.1-2.2 is the mechanical resonance of the whole system.

10. References:

Jill E. Krutz, David F. Thompson, Gary W. Krutz, and Randall J. Allemang. Design of a hydraulic actuator test stand for non-linear analysis of hydraulic actuator systems. Automation Technology for Off-Road Equipment, Proceedings of the July 26-27, 2002 Conference, pages p. 169–183, 2002.

Mohieddine Jelali and Andreas Kroll. Hydraulic Servo-systems - Modelling, Identification and Control. Springer, 1. edition, 2003. ISBN: 1-85233-692-7.

Basic Principles and Components of Fluid Technology, BOSCH REXROTH by H. Exner, R. Freitag, Dr.-Ing H. Geis, R. Lang, J. Oppolzer, P. Schwab, E. Sumpf, U. Ostendroff Hydromatik GmbH, Ulm M. Reik HYDAC GmbH, Sulzbach

Adrian Bonchis, Peter I. Corke, and David C. Rye. A pressure-based, velocity independent, friction model for asymmetric hydraulic cylinders. 1999 IEEE International Conference on Robotics and Automation, pages p. 1746–1751, 1999.

Chunxing Wang. hydraulic control system. Beijing, CA China Machine Press 1999.

H. E. Merritt, Hydraulic Control Systems, John Wiley & Sons Inc., 1967.

Huang, C. L., C. H. Lan and Y. C. Wu (1994). The position control of electrohydraulic servomechanism via a novel variable structure control. Mechatronics 4(4), 369–391.

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