Minimum Effort Hydraulic System Regulation

ألتحكم بنظام محرك هيدروليكي باستخدام نظام تحكم ذات حد أدنى لإستهلاك الطاقة

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Abstract

An investigation into the electrohydraulic performance of long- stroke, loaded, spool valve-controlled, high-pressure, cylinder type servomechanism is presented herein. In addition to spool valve, a needle valve is also included in the hydraulic system so as to create oil leakage across the piston. The leakage flow across the piston is considered so as to produce damping to piston motion. To accommodate performance requirements; single input–multiple output modeling and regulation is proposed. In the study, oil compressibility, the inertia in piston motion and the state equations derived in linearized form are considered. PQR technique is used to derive the transfer function of the open loop system. The applied voltage on the spool and needle valves is considered the single input, while the load position and the leakage flow across the piston are considered as the multiple out-puts. Minimum effort is obtained by optimizing the pole locations of the single input – multiple output system. Optimum locations of the poles are determined by using root locus method and by creating a controller generated zero. The responses are analyzed according to two cases; when the volumes of the cylinder chambers are equal and when they are unequal. Simulation of output responses is shown in order to determine how far the volumes can be considered so as to validate output responses. The control technique used in the application dissipates minimum energy minimizing heat generation, wear and maintenance cost.

خلاصة البحث

يقوم ألبحث بدراسة مستفيضة لأداء جهاز تحكم بالموقع (سيرفوميكانزم) يعمل بالنظام الكهروهيدروليكي حيث يتألف من إسطوانة و مكبس هيدروليكيين و صمامين كهربائيين ألأول هو صمام بكرة يتحكم بإتجاه تدفق السائل ألهيدروليكي من وإلى الإسطوانة, والثاني هو صمام يحتوي على إبرة من اجل التحكم بالسائل الهيدروليكي المتسرب عبر مجرى بين غرفتي الإسطوانة. فمن أجل الإحاطة بأداء النظام تم إعتباره ذات دخل منفرد وخرج متعدد وذلك من أجل إتاحة عملية النمذجة والتحكم بالنظام موضع الدراسة.

و من الجدير بالذكر فقد تم اعتبار السائل الهيدروليكي قابل للإنضغاط , و حركة ألمكبس بها عطالة, وان معادلات الحالة بوضع خطي, وقد تم توظيف تقنية خاصة تستخدم المصفوفات من أجل إشتقاق دالة التحويل لنظام الدائرة المفتوحة. أما الجهد الكهربائي المطبق على مدخل الصمامين الكهربائين ففد تم إعتباره بالدخل المنفرد في حين إعتبار موقع المكبس و تدفق الزيت الهيدروليكي عبر ألمكبس بالمخارج ألمتعددة.

لقد تم في هذا البحث الوصول الى الإستهلاك الأدنى للطاقة من خلال تحسين موقع أقطاب دالة ألتحويل للنظام وحيد الدخل متعدد المخارج, إن ألموقع الأمثل لهذه الأقطاب قد تم الوصول له من خلال طريقة "المحل الهندسي للجذور" وكذلك من خلال ايجاد وحساب صفر التحكم لنظام الدائرة المغلقة.

أما إستجابات النظام فقد تم تحليلها تبعا لحالتين مختلفتين: الأولى عندما يكون حجم غرف الإسطوانة الهيدروليكية متساو والثانية عندما يكون غير متساو, فبالتالي فقد تم إستعراض نتائج محاكاة استجابات النظام من اجل الحكم الى أي مدى يمكن للسيرفوميكانزم ان يعمل عند أحجام إسطوانة مختلفة ألتي عندها يمكن له ان يعطي نتائج صالحة ومعتمدة.

إن طريقة ألتحكم المتبعة في هذا التطبيق أعطت نتائج تدلل على أنها ألأمثل من حيث استهلاك الحد الأدنى من الطاقة وانها تقلل سخونة ألجهاز والبلى وكلفة ألصيانة.

Acknowledgments

At the end of this work I thank Professor Robert Whalley who contributed a wonderful guidance in the completion of this work, and special thanks to Dr. Alaa' Abdul Ameer because of the excellent support he has offered.

List of Notations and Abbreviations:

a_1: Coefficient of the reduced polynomial of Leakage flow transfer function.

a_2: Coefficient of the reduced polynomial of piston position transfer function.

A: State space matrix

A_p: Piston area (in^2)

b_1: Coefficient of the reduced polynomial of Leakage flow transfer function.

b_2: Coefficient of the reduced polynomial of piston position transfer function.

B: State space matrix

C: State space matrix

c_p: Friction coefficient $(N/in/\sec)$

D: State space matrix

$E(t)$: Control energy cost function

F: Piston force (N)

F_c: Friction force (N)

$g_{ij}(s)$: Numerator of elements of transfer function matrix

$g_1(s)$: Leakage flow input- output system description

$g_2(s)$: Piston velocity input- output system description

$G_1(s)$: Leakage flow input- output system description

$G_2(s)$: Load position input- output system description

$\tilde{G}_1(s)$: First order reduced Leakage flow input- output system description

$\tilde{G}_2(s)$: First order reduced load position input- output system description

i_l: Current flow in the needle valve coil (Amp)

i_v: Current flow in the spool valve coil (Amp)

h_1: Feedback gain from output -1

h_2: Feedback gain from output-2

I: Identity matrix

J: Performance Index

K_1: Spool valve flow gain $(in^3/\sec/in)$

K_2: Flow – pressure coefficient in the cylinder $(in^3/\sec/psi)$

k_1: Forward path gain

k_2: Forward path gain

k_d: Needle valve flow gain $(in^3/\sec/in)$

k_L: Flow – pressure coefficient across the piston $(in^3/\sec/psi)$

K_L: Needle displacement – current coefficient (in/Amp)

K_v: Spool displacement – current coefficient (in/Amp)

L: Piston stroke (in)

L_l: Inductance of the needle valve coil (H)

L_v: Inductance of spool valve solenoid coil (H)

M: Load mass (lb)

m: Number of inputs of multivariable system

λ_i : Open loop poles vector

P_0 : Return pressure (psi).

p_1 : Coefficient of numerator polynomial of root locus transfer function.

p_2 : Coefficient of numerator polynomial of root locus transfer function.

P_1 : Pressure in hydraulic cylinder compartment 1 (psi)

P_2 : Pressure in hydraulic cylinder compartment 2 (psi)

P_L : Load pressure (psi)

P_s : Supply pressure (psi)

q : Potential leakage flow $(in^3/\sec)$

q_1 : Oil flow between spool valve and the cylinder $(in^3/\sec)$

q_2 : Return oil flow between cylinder and spool valve $(in^3/\sec)$

q_n : Oil flow part of leakage flow equilibrium across the piston $(in^3/\sec)$

q_l : Leakage flow $(in^3/\sec)$

q_v : Oil flow in the spool valve $(in^3/\sec)$

r : Number of out puts of multivariable system

$r(s)$: Reference input signal

R_l : Resistance of the needle valve coil (Ω)

R_v : Resistance of spool valve solenoid coil (Ω)

u_l : Voltage applied on the needle valve $(volt)$

u_v : Voltage applied on the spool valve solenoid $(volt)$

u_1: Input -1 of the control system

u_2: Input -2 of the control system

$u(t)$: Input vector

$u(s)$: Laplace transformed Input vector

$v(t)$: Piston velocity $(in/\sec)$

V: Total cylinder volume (in^3)

V_0: Initial oil volume (in^3)

V_1: Volume of cylinder chamber-1 (in^3)

V_2: Volume of cylinder chamber-2 (in^3)

$x(t)$: States vector

y: Spool valve displacement (in)

$y(s)$: Laplace transformed output vector

$y(t)$: Output vector

z: Load displacement (in)

β: Bulk modulus (psi)

ε: Error signal

$\Delta(s)$: Denominator of the open loop transfer function

δ_i: Load disturbance signal

MVC: Minimum-Variance Control

GPC: Generalized Predictive Control

LQG: Linear Quadratic Gaussian

MRAC: Model reference adaptive control

SMC: Sliding mode control

GA: Genetic Algorithm

HS: Hydraulic system

HAS: Hydraulic Actuation System.

PAA: Parameter adaptive algorithm

PI: Performance index

PID: Proportion integral derivative

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Chapter One

Introduction

1.1 Study Background

A servo mechanism can be defined as (Welbourn 1963, p. 1): “a servo-mechanism or servo-motor of any form of mechanism or motor which produces power amplification between the input and the output”.

The development of the hydraulic power systems developed by man through the history started nearly 350 years ago. In 1738, Bernoulli published his book “Hydrodynamica”, which included his kinetic-molecular theory of gases, the principle of jet propulsion, and the rule of the power transformation.

By mid of nineteenth century, fluid power played an important role in both the industrial and civil fields. In England, for example, many cities had central industrial hydraulic distribution networks, supplied by pumps driven by steam engines. Prior to adaptation of Electrical energy, power fluid occupied a great space in competition with other energy sources in the city of London.

Company of “London hydraulic power” managed to provide a hydraulic power to many of applications such as port cranes, bridges and building elevators for major cities in the world. In the thirties of last century, through the revolution period of hydraulic power, a power source of water, pumping of an average of a 12 m3/min underneath London’s roads, was powering and running almost everything in the city requires up and down movement (Rabie 2009).

The marriage between electronics and hydraulic power systems has led to many powerful and precise control systems, saving much energy and money. This concept is applied in the application of electrohydraulic proportional and servo systems. These systems have the same advantages as hydraulic power systems, particularly the maximum power to- weight ratio and the high stiffness of hydraulic actuators. They also have the same advantages as electronic controllers, mainly in regard to high controllability and precision.

Fluid power engineering has four classes of control valves that use electric controllers. They include the following.

• Ordinary or switching valves: Widely used to turn valves on and off.

• Electrohydraulic proportional valves: Used usually in open loop control systems. They are controlled electronically to produce an output pressure or flow rate proportional to the input signal. They offer advantages such as control reversal, the step-less variation of the controlled parameters, and the reduction of the number of hydraulic devices required for particular control jobs.

• Digital valves: Wherein a microprocessor sends discrete signal pulses to a stepping motor, which in turn positions the control element of a valve.

• Electrohydraulic servo-valves: Usually used in closed-loop (feedback) control systems.

The “servo concept” is a widely used expression. Taken alone, it indicates a system in which a low power input signal is amplified to generate a controlled high power output or signal. An input signal of low power—for example, 0.08 watts—can provide analog control of power reaching more than 100 kW. Electrohydraulic servo systems provide one of the best controllers from the point of view of precision and speed of response. They are used to control almost all hydraulic and mechanical parameters, such as the pressure, pressure difference, angular speed, displacement, angular displacement, strain, force, etc. Electrohydraulic servo-valves were used in military equipment as early as the 1940s. Hydraulic servo actuators have been used also widely in aerospace and marine applications, to control the rotating blades, pitch angles, thrust deflectors, and the displacement of different control surfaces, such as rudders, ailerons, and elevators. (Rabie 2009).

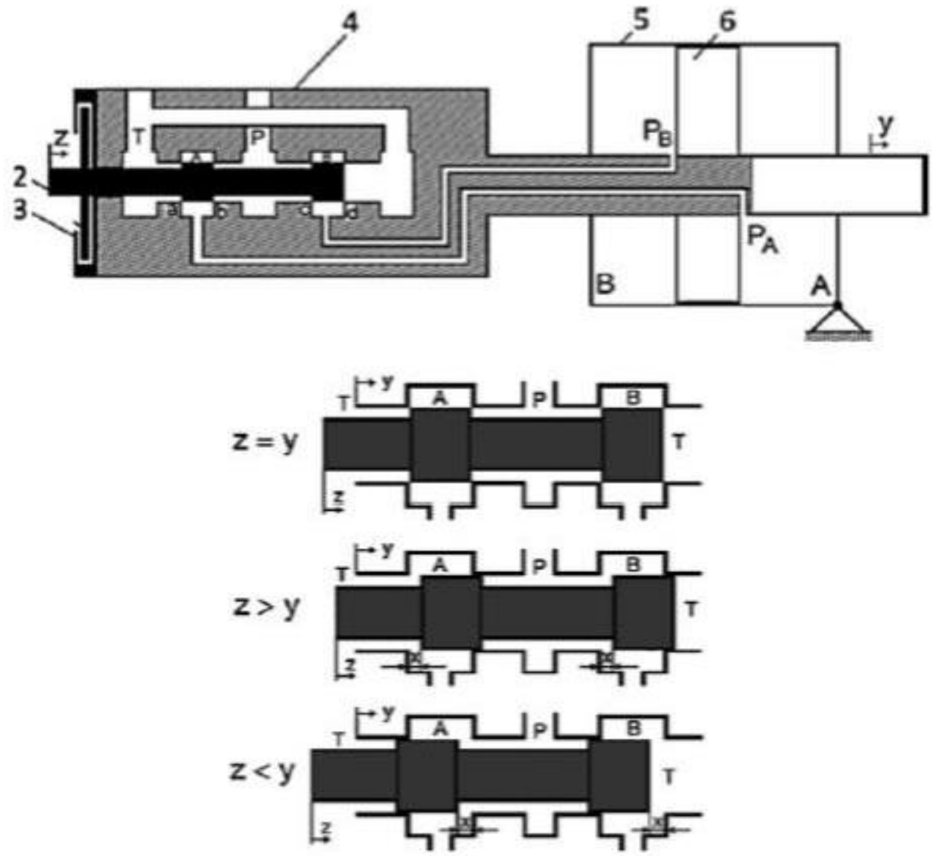
In modern control theory the term servomechanism or servo is restricted to feedback control systems in which the controlled variable is a mechanical position or time derivatives of position, e.g., velocity and acceleration. A servo system uses a hydraulic cylinder, moves the load and creates the load displacement. The output (controlled) and desired (reference) positions are measured and compared in order to produce an error signal which is a measure of discrepancy between the input and output signals. The actuating signal is applied to the process (transfer function of the plant) so as to influence the output signal in a manner which tends to reduce the error (Nagrath and Gopal 2007).

One of the classified hydraulic actuators is a hydraulic cylinder, performing linear motion. The hydraulic cylinders convert the hydraulic power into mechanical power, performing rectilinear motion. The pressure of the input oil is converted into the force acting on the piston. Hydraulic servo actuators (HSAs) are used to precisely control the displacement in a wide range of equipment. Generally, a hydraulic servo actuator consists of a hydraulic actuator controlled by a directional control valve with a range of positions with a feedback arrangement**. Fig (1.1)** and **(1.2)** show typical constructions of hydraulic servo actuators.

As the spool valve is displaced to the right as per **Fig 1.1**, the high pressure line (P) can be connected with left hand side of the cylinder compartment (B) which is a reaction of the spool valve displacement to position z, therefore the oil can flow from the high pressure (P) to increase the oil pressure PB in cylinder compartment (B). The right piston compartment (A) is connected simultaneously with the return line (T). The pressure, PA, decreases and the pressure difference (PB − PA) acts to drive the piston (6) to the right.

As a feedback control; the body of the directional control valve (4) is rigidly attached to the piston rod and they move as one body. This displacement causes a gradual decrease in the spool valve opening distance, throttling area, inlet flow rate, and piston speed. Finally, when the total piston displacement equals that of the spool, the spool valve ports are almost closed. The pressure difference in the piston chambers produces a force equal to the loading force. The fluid flow to the cylinder chambers is cut and the piston is stopped. The HSA is operated by displacing the spool relative to the sleeve (or relative to the valve body). In the steady state, for the piston to stop moving, the spool should be brought to its neutral position. Therefore, the feedback acts to bring the spool to the neutral position when the piston is displaced to the required position. (Rabie 2009).

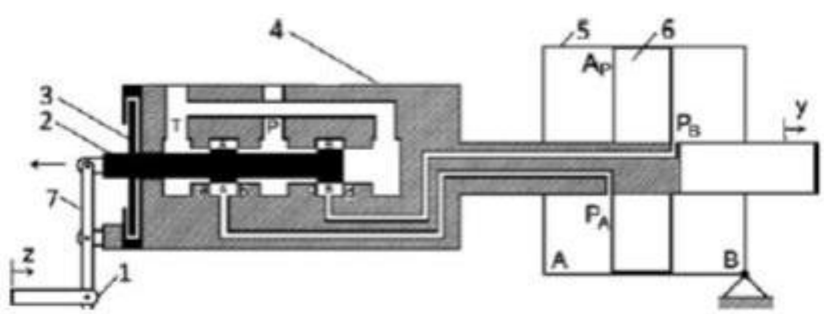
HAS is naturally a multivariable system. Such system is known that an input in a system can influence one or other outputs in the same system, hence these systems are usually process wise complicated when the controlling process has to be designed, the difficulty is arising because of the complicated interfaces between the inputs and outputs of the system, such case can lead to unwanted response from the system itself. Most of today's industries processes are multivariable systems, they become so, due to the need to obtain power efficiency, products of quality features and friendly to environment (Rames and Sujatha 2012)



2. Spool, 3. Spool displacement limiter, 4. Directional control valve, 5. Hydraulic cylinder,

6. Piston, z = Spool displacement, y = Piston displacement, x = Valve opening distance.

Figure 1.1 structure of hydraulic servo actuator



1. Control rod (input displacement), 2. Spool, 3. Spool displacement limiter,

4. Directional control valve, 5. Hydraulic cylinder, 6. Piston, 7. Feedback rod

Figure 1.2 Typical design of an HSA with mechanical feedback linkage.

The HAS in this study represents a single input multiple output system, and consist of:

- Servo valve (spool valve) as a torque motor with the role of controlling the oil flow.
- Hydraulic cylinder which moves to displace the load.
- Needle valve which controls the oil leakage across the piston, the valve is acting as the damping agent on the output response.

The single input hereby is the common voltage applied on the spool valve and the needle valve. The multiple outputs comprise load displacement (z) and the leakage flow, **Fig. (1.3)** illustrates the Hydraulic Actuator subject of the study:

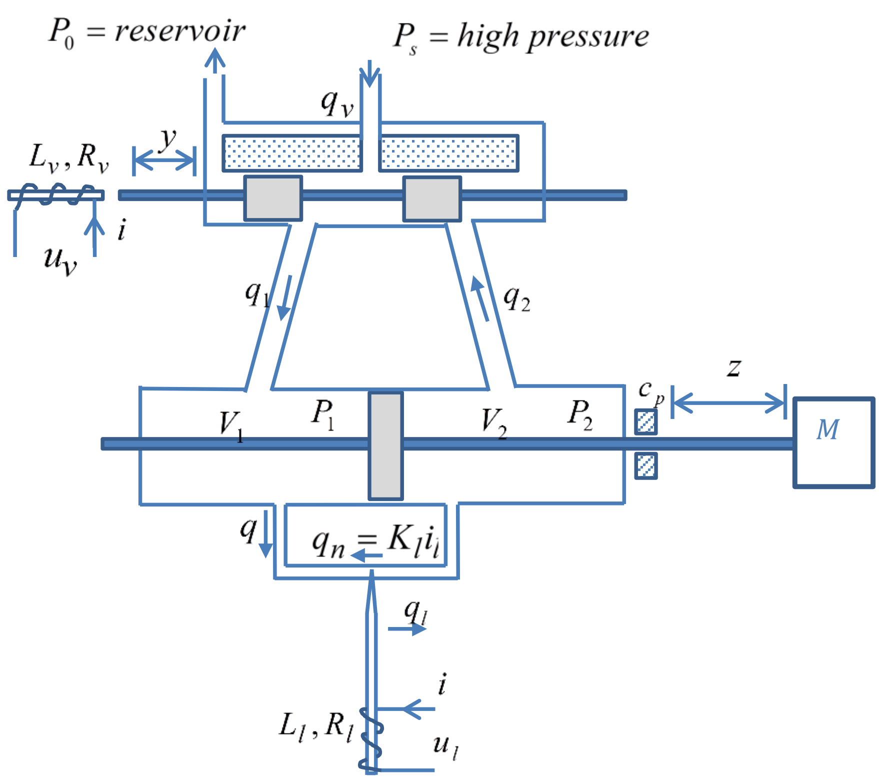


Figure (1.3) hydraulic servo actuator

1.2 Assertion of work problem

Controlling the hydraulic actuations system (HAS) is the major problem of the study, this problem will be analyzed and examined. The control technique which will be employed for this system is the optimization of pole assignment with least energy consumption for single input, multiple output hydraulic servomechanisms. The followed control strategy will exhibit particular processes to calculate and to identify the particular values of the forward and feedback gains which optimize the pole locations of the closed loop system in such a manner that the piston velocity and position responses are improved in terms of the dynamics and the steady state. The analysis considers first that the volumes of the cylinder chambers are equal so that unequal volumes can create nonlinearity so that the output responses cannot be analytically predicted, the behaviour of actuator at unequal volumes of the cylinder chambers will be analyzed too.

1.3 Study Aims and Intentions

This study work is driven by the need to take a full benefit of applying the latest researches of least efforts theory controlling single input multiple output controller design emphasizing the optimization of the closed loop pole locations with least consumed energy. Such theory has been presented by Whalley and Ebrahimi (2000) when they applied the least efforts controller on a mixing-tank liquid level control model. The control strategy for this work is an extension of their work by using the same methodology of establishing the performance index (J) and calculating the minimum values of it, in order to determine (n) value. More details can be found in the research methodology chapter.

The suggested controlling technique is going to solve the standard control problems such as stability of the system at closed loop mode and improving responses of the system in the transient and steady state stages. One more important aim of the resource is to determine how far the location of the piston can be offset from the middle of the stroke so that the responses can be validated accordingly. The simulation and observation of the responses can serve in such an assessment.

1.4 Dissertation and study structure

Basically, a literature review in chapter two is an overall presentation of some relevant studies worked by some researchers in the past in the area of modeling and controlling hydraulic servo actuators.

In chapter three, the control methodologies will be presented, it will include also analyzing the structure of the hydraulic system actuation, deriving the mathematical model and the transfer function of the multivariable system, moreover, the parameters of the controller will be computed and the responses to unit step will be simulated and shown.

Chapter four deals with the application of control theory and with the observation resulted from the simulation of system.

Chapter five goes in details through the obtained results presenting the benefits of the new proposed control methodology

Chapter Two

Literature review

2.1 Introduction:

Hydraulic actuator studies have increased significantly in the last decade with so many researchers contributing to the analysis, design and selection of control techniques for hydraulic actuators.

In the 1850s the transmission of power using compressed fluids began when the pressurized liquid has been employed to run hoists, lifters, press machines and extruders. The standard open loop system was the control method which has been followed during this time. However, Brown in 1870 has developed the closed loop control in fluid power systems when he proposed a mechanical feedback obtained from the ship rudder in order to confirm the required position of the piston in a cylinder. A spool vale has been used to achieve the desired piston position (Conway 1953).

The employment of the water throughout the years of the century as a transmission intermediate was decreased; alternatively, hydraulic oil has been developed despite the fact that there were some researches recently on using the water hydraulics owing to environmental considerations. In I906, replacing the electrical system used for the purpose of lifting warship gun by a hydraulic system was considered as the first use of this type of systems (Pippenger 1992).

The use of hydraulic power control systems since that time have been expanded and developed significantly, especially when electrohydraulic servo systems have been introduced in the 1940s, the use of these powerful equipments have developed widely and nowadays are used extensively in so many fields and sectors, for example, but not limited to transport, construction, agriculture, aviation and materials handling. Controlling fluid power components and systems involve real challenges despite of the wide range of usages, (Edge 1997).

In 1965 Nikiforuk and Westlund have turned to a hydraulic servomechanism controlled by a spool valve, these equipments are in general widely used and characterized that they are loaded with immense loads and usually powered by high amplitude input signal values. As per the authors, logical methods mostly used for the purpose of analyzing such systems are not enough satisfactory, the evaluations of available literatures show obviously that almost all of the applicable relevant studies were confined to the work on responses related to open loop system, while a small percentage of these works have limited interest to the respond to the hydraulic systems in the closed-loop mode including system dynamic behaviour when the servomechanism is loaded with slight loads. It is worthwhile to mention that compressibility and oil leakage were highly ignored due to fact which states that their impacts can make the analysis so difficult to understand, such ignorance in turn will push the study approach to resort to linearize the mathematical model in order to obtain accepted solutions. Conversely, although linearizing the mathematical model can consider compressibility and oil leakage; the investigation will be limited to small input signals and small loads so that the followed approximation in this regard will make the practical systems far from reality (Nikiforuk and Westlund 1965).

2.2 Example of power fluid actuation:

Fig. 2.1 illustrates the layout graph of an electrohydraulic servo system. The controller processes the variance value which is resulted from subtracting the actual position from the desired position of the load so that the signal from the comparison is employed to drive the electrohydraulic valve. The valve plays the role of controlling the actuator by controlling the flow of the hydraulic oil (or its pressure) flowing to the actuator that moves the load. The physical parameters of the hydraulic actuator such as force, torque, acceleration and velocity can be controlled thereafter, a proportional, integral and derivative (PID) controller may be employed to give a satisfactory degree of performance.

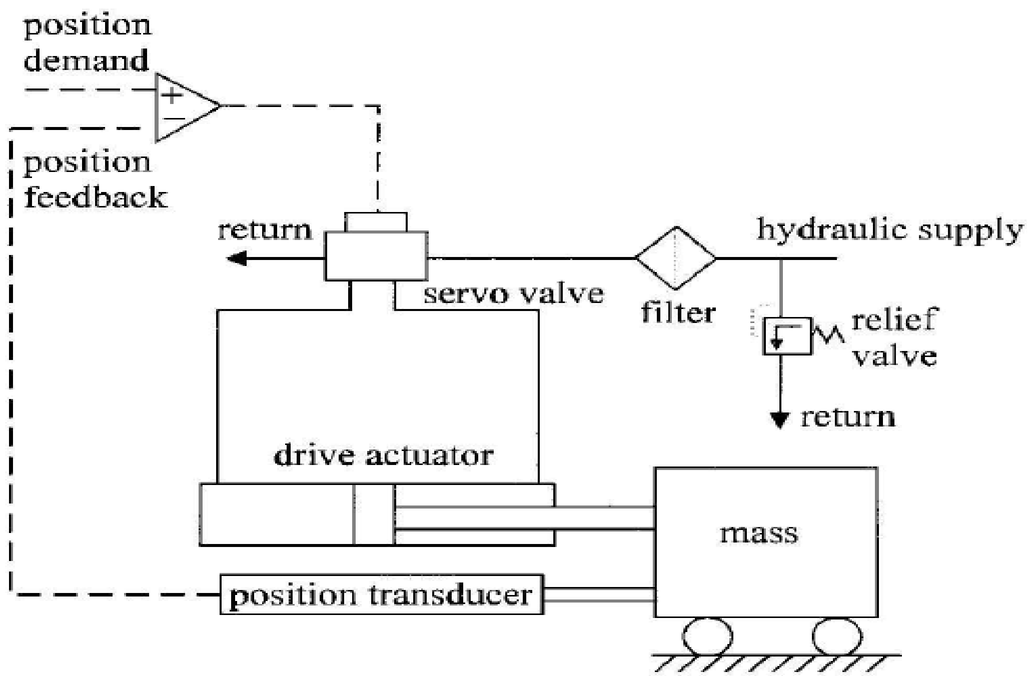


Figure 2.1 Position control system with electro-hydraulic actuator

On the other hand, improving the performance of electro-hydraulic actuators continued through many works and researches which took place so as to improve many aspects in the hydraulic actuators, for example, but not limited to deal with the application complication, enhancement of efficiency, and being a good alternative to the similar electrical drives. Such aspects have shown very good progress due to the so many works committed to develop the hydraulic applications and the progress in control methods and by analyzing and implementing them. Nonetheless, there were many challenges affected the works and the researches employed to address these important aspects, these challenges impact the transient and the steady state responses of the hydraulic actuators (Edge 1997).

2.3 Main challenges of designing power Fluid devices and systems.

The features of nonlinearity of the major parts of the hydraulic actuator including valves are very important aspects during the analysis stage. These valves can exhibit a non-linearity in the flow-pressure expression which accordingly demonstrates saturation, hysteresis, and under-lap (or over-lap). A system where a hydraulic cylinder is employed for positioning a load, can show variations in stiffness when the stroke travel changes. This in turn will cause changes in dynamic behavior. When a hydraulic cylinder is used to drive an object, the travel direction will govern the characteristic of the load movement itself.

Likewise; multi-axis structures can reveal complicated cross-coupling interfaces. Moreover hydraulic oil temperature also can play a significant role relating to the behaviour of the system.

Therefore these aspects identify the cost to be invested so as to reach a quick reaction to inputs, high ratio of power output divided by the mass, then, for hydraulics, it is interesting to reach a system of low elasticity features. The above mentioned issues are very important at the design stage of the control system (Edge 1997).

Carrington and Martin presented in 1965 a study regarding some problems of Electro-Hydraulic Servomotors. The study addressed the region of threshold conditions where the non-linearities of actual servomechanisms have this effect. For small input signals, there is no response, and for small disturbances and errors there is no correction. If the input signal or disturbance is increased sufficiently the servo will try to respond or correct itself, even though, the results from one test to another may not be consistent.

There was an attempt to bring together some of the known facts associated with this region and the techniques used for improvement. The authors in the study have talked about many improving aspects for servomechanism, and discussed in detail the improvements of: **specific basic design aspects including friction on the spool valve, electro-hydraulic valves, potentiometers, anti-backlash gears, loop gains, orifice bleed valve under-lap, pressure feedbacks** and **electrical networks.** (Carrington and Martin 1965).

With presentation of investigational as well as theoretic results; D. McCloy and H. R.Martin offered a study pertaining to linearized analysis to address the main reason of the instability impact of flow of a valve or Bernoulli forces. As the oil starts to flow in a hydraulic spool valve, the flow itself can establish a power which can act in the opposite direction of the spool motion, such power is created by the amount of variation in the oil energy through the valve, and it is generally identified to be flow force. As the pressure in the cylinder drops to zero, when large acceleration is needed; cavitation occurs. Linearized theory has been used to prove that when the load is from inertia; flow-force effects can prompt instability which can be verified by experimental evidence. The authors found that the phenomenon of power caused by the fluid flow can exhibit the following impacts on the dynamics of the system, the static gain of the open loop of the spool valve and its stability are decreased. Once the forces of the flow are produced at a particular relay current, this means that there is less valve movement as a force of Bernoulli acts against the motive force of the relay, justifying the reduced open-loop static gain. This effect has been used to obtain the amount of the force of the flow.

If the characteristic of the current of the relay is considered when the supply of the oil is on and when it is off, plus when the openings of the valve are linked with each other, the proportion of slop relation when oil supply is on, to slop relation when oil supply is off, can give that magnitude of the flow force

The load just bounces at the range of natural frequency on the oil pillar at which the values of compressibility flow as well as the displaced flow are tinny. Consequently when the flow value is very small through the valve, it will make Bernoulli force small as well and the ratio of valve displacement to relay current is extreme. Usually, at the range of natural frequency, the output of the hydraulic actuator will be great at small values of damping and greater at increasing the gain of the valve which in turn reduces the open loop system stability.

The authors have recommended a technique to overcome the reaction force, they proposed employing a powerful solenoid in such a manner the gain of the relay amplifier can increase which in turn lowers the flow factor so that reducing the influence of the de-stabilizing factor caused by the flow force, this can be achieved by including a viscous load, such load can get the procedure of a damper which resists motion via viscous friction or by utilizing an orifice bleed.

It has been presented at the outcome of cavitation discussions that even when the valve is lapped at zero, the pressures in the hydraulic cylinder will be very high and big inertia loads can cause this pressure in the cylinder. It is meaningful stating that the study can be an instruction and a base for further detailed studies (McCloy and Martin 1963).

D. Mccloy in 1969 made a further contribution when he turned to the cavity of the fluid and its impact on the dynamic response to step inputs, step inputs are applied on a valve driving hydraulic actuator, and the system is modeled by the non-linear cavity equations exist on both sides of the actuator piston. Meanwhile, compressibility and leakage effects would make this doubtful in practice. In this particular study, the concentration applied on the cavitation, while other fluid features such as compressibility and leakage have been neglected to make the problem responsive to analysis. The impact of cavitation on dynamic response, and the possible effects of cavitation damage have been addressed by the author in his research. He aimed to examine the effect of cavitation to see if there is any reason for confining a servo to work in a non-cavitating mode.

As the movement of the spool valve may be in either direction, to drive the piston motion, one side of the piston is connected to the tank and the pressure in the relevant cylinder chamber drops rapidly to oil vapor pressure (assumed zero) in an attempt to make this side full which consequently forms a cavity in the cylinder chamber.

With the assumption that the fluid oil is incompressible; the author confirmed that cavitation has been shown to happen, and the piece-wise non-linear equations describing the response have been solved. Under certain conditions cavities could exist on both sides of the cylinder at the same time. The author has compared the behaviour of system with cavity formation with behaviour of a system in which the oil is able to resist negative pressures. It has been found that the piston velocity and displacement responses are approximately same, and the major dissimilarities found in the responses lies in the pressure waveforms but the similarities of the responses of velocity and displacement on both systems can lead to the fact that using a mathematical model that does not consider cavitation, does not lead to large errors.

The author reached a conclusion that there is no enough justification as to why hydraulic servomechanism must not be run in a cavitating mode. In the meantime the response is slightly affected and cavitation damage is questionable under cavitating conditions and can be sufficiently described by a simple model that does not take cavitation into consideration. Oil damage by oxidation is possible but this may already be present as a result of orifice cavitation and de-aeration (McCloy 1969).

2.4 Overview of earlier studies of servomechanism

R. Butler in 1959 has worked on analyzing a hydraulic relay which consists of a double-acting ram to a flow of oil of constant-pressure which is also controlled by means of a small double-acting pilot valve. **Fig. 2.2.a** illustrates the configuration of the hydraulic relay studied by the author, he has shown also the velocity and displacement responses for four different inputs including step input, ramp input, sinusoidal input and general input, he has derived comprehensively the motion equation and considered all the possible properties of both hydraulic and components.

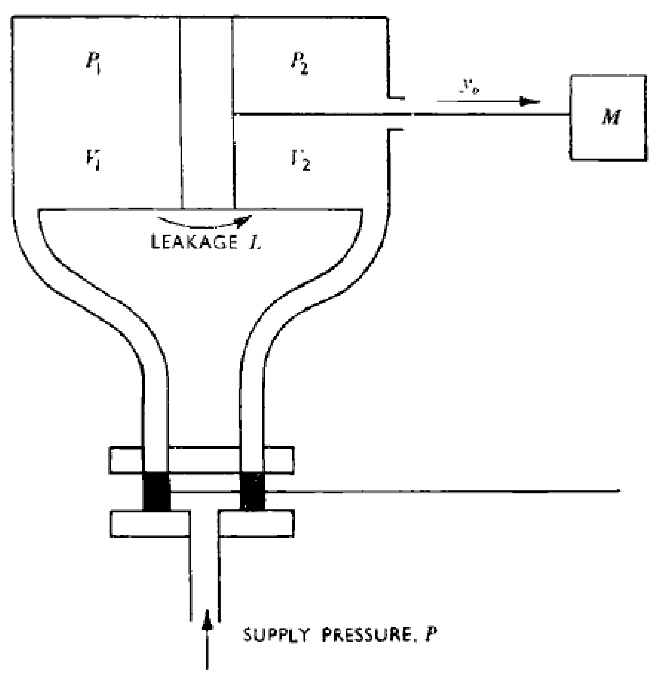


Figure 2.2.a scheme of hydraulic relay

The functions and the characteristics of these properties have been appreciated in the motion equation derivation, which added a valuable contribution in recognizing the operation, design and the performance of the hydraulic relay. At that point, the work was addressing the features of hydraulic relay without considering the control strategy to be followed so as to optimize the relay performance (Butler 1959).

D. E. Turnbull worked in 1959 on the behaviour of a loaded hydraulic servomechanism. As the required force to transport the object is formed by a pressure's fall through the actuator, such a fall makes the speed of actuator responses go down, these are the piston speed as well as the fluid speed through the ports of the valve which is relational to the square root of the pressure's fall through the valve. This fall drop in the pressure is equivalent to the variance amongst the overall pressure's fall through the whole process and that requisite producing movement of the load. The output's speediness is governed by the load and by which the non-linearity arises. **Fig. 2.2.b** shows the hydraulic actuator which is the subject of the study. The pressure falls and the arrows show motion to the right.

The author has examined the impacts of numerous sorts of load on the transient response of the system when step and sinusoidal inputs are applied.

The examination has been applied on the separate and joined impacts of several familiar types of loads , for example when a constant load is load to the system, the time constant merely increases which consequently reduces the speed of the response. The load by the way does not create non linearity in the system, while in the orifice as well as viscous loads it gives a significant impact on the primary response speed which is always less than unity. In general the existence of a load on the output rises the response's time but the level of the rise is subject to the definition used (Turnbull 1959).

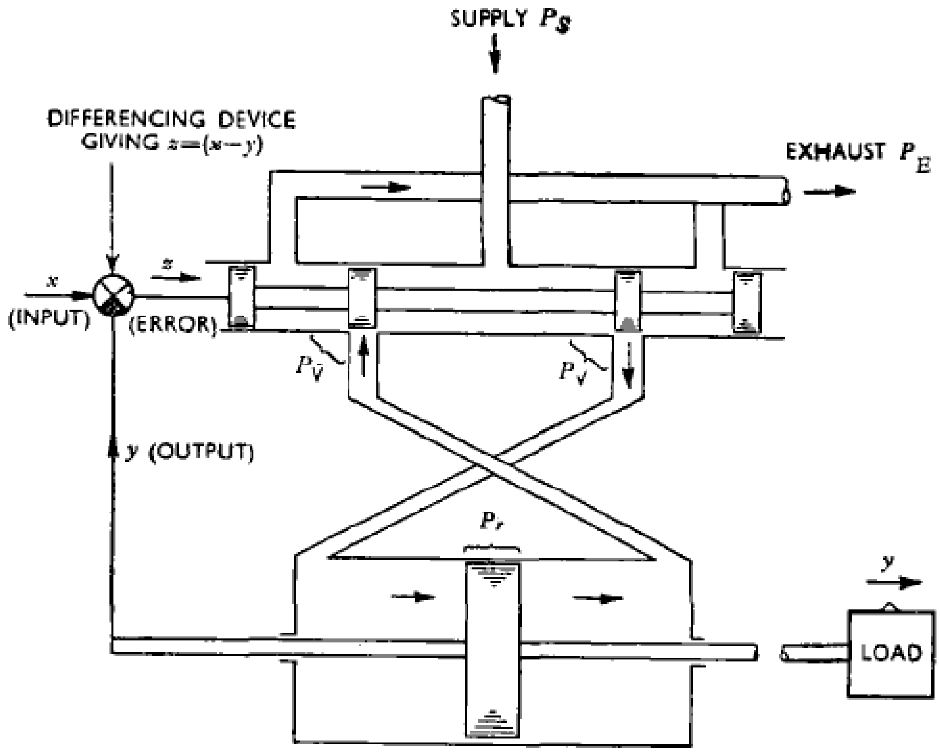


Figure 2.2.b “The hydraulic actuator”

Nikiforuk and Westlund worked on responses of the big-magnitude of a input signal having the shape of a sine curve and input signal of the transient phase of a loaded high-pressure hydraulic servomechanism under closed loop control taking in consideration that the oil is compressible and oil leakage in the system analysis . The analysis of the study was not only considering the small magnitude of input signals and small loads.

The studied system is a structure of electronic amplifier, dual stage electric spool valve and piston equipped hydraulic cylinder, whereas, the load is a combination of coulomb friction, mass, and viscous damping, **Fig. 2.4.c** shows schematic diagram of the study’s servomechanism. In order to assess the closed loop frequency response; the authors have used load loci to describe the function procedure.

However, the curve of the load as well as linearization procedure which was made up of pieces are employed to identify the dynamic and the transient behaviour of the closed loop system. The analysis was broadly more than enough to consider the inclusion of known and unknown characteristics of valve pressure-flow.

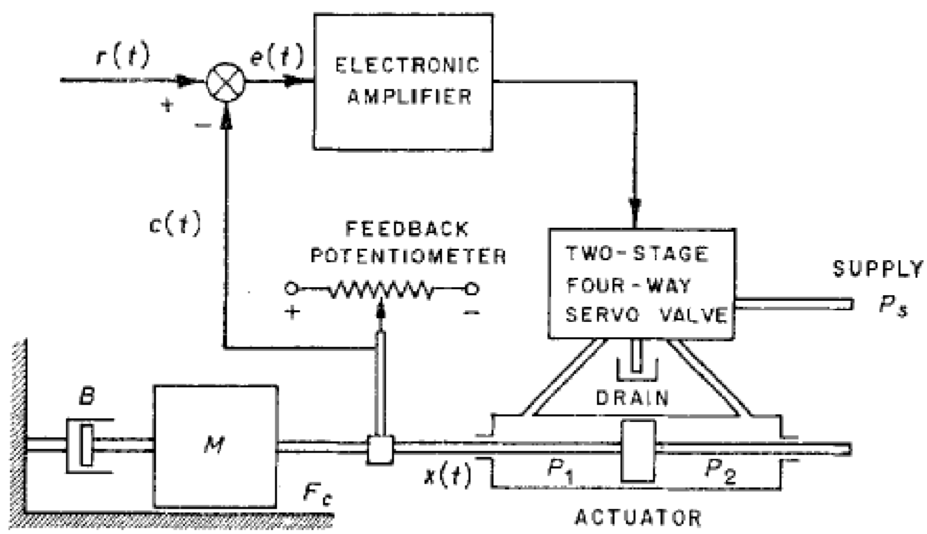


Figure 2.4.c "Schematic diagram of servomechanism"

The authors have reached a conclusion that valve-flow gains $\left(\frac{\partial q_v}{\partial i}\right)$ and $\left(\frac{\partial q_v}{\partial P_L}\right)$ are according to the test results are the most leading factors of the system non-linearities which in turn can cause degradation in system performance. The effect of both factors on the sinusoidal response of the servomechanism is a waveform; such response is promptly distorted to a triangular shape as an outcome of flow saturation, but a precise of prediction of the system response employing linear techniques was only possible with small masses and small input signals.

Therefore the function of the valve is limited to a narrow area near the beginning of its flow pressure relation curve, where the valve-flow gains are approximately linear. On the other hand; enlarging the input signal out of this region can lead to saturation in the flow because of the increase in the valve current. In the same way; increasing the load also at smaller value of valve currents has shown that flow saturation can happen. Consequently; under such operating condition; the analysis of linear techniques caused poor predictions of system performance. The significance of the study is that it has demonstrated a good correlation between the predicted results and the experimental results. (Nikiforuk and Westlund 1965)

R. Bell and A. De Pennington in 1969 worked on improving the response of the servomechanism, (hydraulic cylinder drive type) using derivative signals, by means of introducing an active damping on the system response, this can take place by employing the acceleration and pressure transducer signals. Compressibility of the trapped volume of fluid in the cylinder chamber was always a limitation and restriction of such drives. When it is used in position control system, this condition leads to compromises in the design of this class of drive, between the choice of effective area to ensure adequate stiffness for a given stroke, and the amount of power that must be applied to reach the desired maximum velocity. Nevertheless, since the servomechanism is a hydraulic cylinder drive; the problem of the damping response should also be considered. As the dynamic response of a cylinder drive is significantly affected by the mechanical damping at the load, which is in this study is close to zero, the load is supported by anti-friction bearings. The need of alternative damper arises to optimize the performance of the position control, they have employed acceleration and pressure transducer signals, to introduce active damping into drives, by including minor loop compensation. It is worthy to say that the approach was based on the use of linearized model.

The study showed that employing acceleration output signal, as a minor feedback loop can give effective damping for servomechanisms of electrohydraulic cylinder drives. The experimental work presented that the results of this analysis can be used with confidence in determining the feedback conditions for a particular system. (Bell and Pennington, 1969).

Armstrong and McCloy presented a development of an old published study dealing with the design of a positioning system employing a hydraulic servomechanism by using an ordinary linear control systems including linear feedback. Many control methods have been presented aiming to optimize the time response; the study used an analytical method in order to propose a controller which can improve the time response, to achieve that, an advanced simulation programs including improving the some factors settings through sub routines, have been used to reach the best improved controller design, the behaviour of the final projected controller has been compared with the optimum behaviour resulted in a traditional proportional controller

Conventionally; the servomechanism have been invented and designed for the purpose of moving an object from a location to another. Generally the accuracy of the motion control can be achieved by employing a control system with a feedback technique. Furthermore; the speed of the response of the controller can be enhanced by designing a network of compensator when response time is more important than other aspects, hence the authors have demonstrated a wide range of discussions pertaining to optimizing the time response of the system.

The authors derived three mathematical models. Model (A) which was considered as the simulation model in which they assumed that cavity exists (The reservoir pressure to which the oil returns and the vaporized liquid can be considered as zero), oil is compressible and the load consists of viscous friction and inertia. In model (B) the authors have added the assumption that there are no dynamics in the valve response and the load represents inertia only. As the actuator can exhibit lopsidedness due to the non-center piston location, this situation can also be ignored. In last model C, the hydraulic fluid has been considered as incompressible.

A modest mathematical model has been followed in model (B) and (C) for the analysis of response time optimization so that the mathematical equations can be analytically explained. The conclusion obtained from this study showed that if the oscillation amplitude of output signals for the hydraulic servomechanism in undamped mode can be considered as an indication for the response time speed, then the modest mathematical model used to represent the system can be effectively employed to explain and predict the behaviour of the control system, this prediction can happen for wide variety of factors' values. In the meanwhile it should be emphasized that control functions must be obtained from logged steady-state error values of the system and should not be attained from that modest mathematical models. The ideal execution of the controller can be considered as comparable in on-off mode with the proportional controller. The routine of the control protocol in the on-off mode is in a manner that the valve ports are completely opened as a response of step input and completely closed when the input is zero at the time when the object reaches its desired location. When the oil is incompressible, this situation is accurate as fully opened ports can generate maximum acceleration while ports shutting can make the load to stop immediately. Therefore; when the cost is reduced in designing and building on-off controller, the uneasiness in obtaining accurate desired load position is understood and reasonable. In the meanwhile, the rapid closure of the valve ports can create a high fluid pressure at the side of the piston so that oscillation in the output response can occur with high amplitude; hence the employment of quick response device by using such controller is not necessarily a bonus

As alternative plan, another control system was presented to enhance the behaviour of the on-off controller by switching the spool valve from a positive unit step to the opposite extreme position with negative unity step, this procedure can be executed for some time prior to valve shutdown at zero input, and at the time the load reaches the rest position and the motion braking time in such procedure will be minimum. However, the occurrence of cavity due to valve shut down and attained response times weaken the reliability of such controller so that the advantages of reaching zero load velocity prior to valve shutdown are eliminated.

The authors also discussed double genre control employing a role of modest feedback donated as (u_f), by this feedback signal , a very good controller behaviour can be reached by means of successive of control signals and consequently, the speed of the load can be decelerated to in the shortest period, therefore no cavitation will happen provided that the load pressure is of value (-1) during deceleration period, the oscillation in the response can be seen despite the fact that this controller reduces the response oscillation, this occurs because the fluid pressure generated from the pressure difference in both cylinder chambers is not zero when the spool valve shuts down, hence a practical solution is to propose a controller which can decrease the load speed and pressure to a value of zero just before the spool valve shuts down, this in turn will make sure that the oscillation in response will not take place, it should be mentioned that although the oscillation is eliminated, the deceleration of the load is not reached to reasonable maximum.

Meanwhile, further interventions on the study made by researchers mentioned that the above discussed comparisons, exhibit the power of the double approach controller in which the load slowness range can stop cavity occurrence. Relative stability in each controller could be also an additional aspect of comparison. It is advisable to mention that best response time can be the worse system in terms of stability so that it is wide-open to small parametric variations causing destabilization. (Armstrong and McCloy1973).

In the end of the sixties of last century and along the seventies decade till early years of eighties, an investigation of conference proceedings and journal papers on works regarding electrohydraulic system control was presented. The researcher De Pennington and his colleagues in last years of the last century sixties and earlier years of the seventies have presented an innovative research on enhancing the transient behaviour of a hydraulic servomechanism. They considered the importance of allowing a precise dynamic characteristic modeling of a servo-valve when assessing the position system performance. They have also launched the basics of the system in terms of acceleration, transient acceleration and transient pressure feedback as well. These control techniques were useful in improving the responses of control systems but they have not been considered broadly possibly due to the cost of components such as transducers (Bell and Pennington1969), (Pennington, Marsland and Bell 1971), (Pennington, Mannetje and Bell 1974).

2.5 Design stage:

The design phase starts from the specification and the setting and selecting the components. K. Edge has set a procedure to design a hydraulic actuator. **Fig. 2.5** shows the flow chart followed in the design process (Edge 1997). In some specific cases when modest loops are employed, the whole structure can be proposed and built, in the meanwhile the control system can also be calibrated live and instantly aiming to get the fulfilled specifications.

A more systematized method comprises of establishing an optimal a scientific system model by deriving the relevant mathematical set of equations, such dedicated model can give a clear indication on the control strategy. Valuable understanding of system in the transient phase including the dynamics of the system could be gained by analyzing the mathematical model which represents the performance of individual elements of the system and approximating the non-linear equations by a well knows linearization technique. Such linearization approach would be valid to most of the hydraulic devices and systems. Nevertheless; attention must be given when implementing the linearized analysis for controller design and should be very wise to measure the response of a non-approximated system in the non-linearity form via a simulation program. More control techniques can be assessed at earlier stage of the controller procedure by the current simulation tools, hence is it very important to open the systems architecture in order to allow the whole integration of the different packages used.

Even when simulation is confirmed as procedure in the design phase, the probability in getting a failure in matching the wanted specifications in the final proposed system is very high, this might be because of poorness in the model, a wrong adaptation of the procedure and may be non-awareness of certain parameters. Load as an example is one of the problems in this regard (Edge 1997).

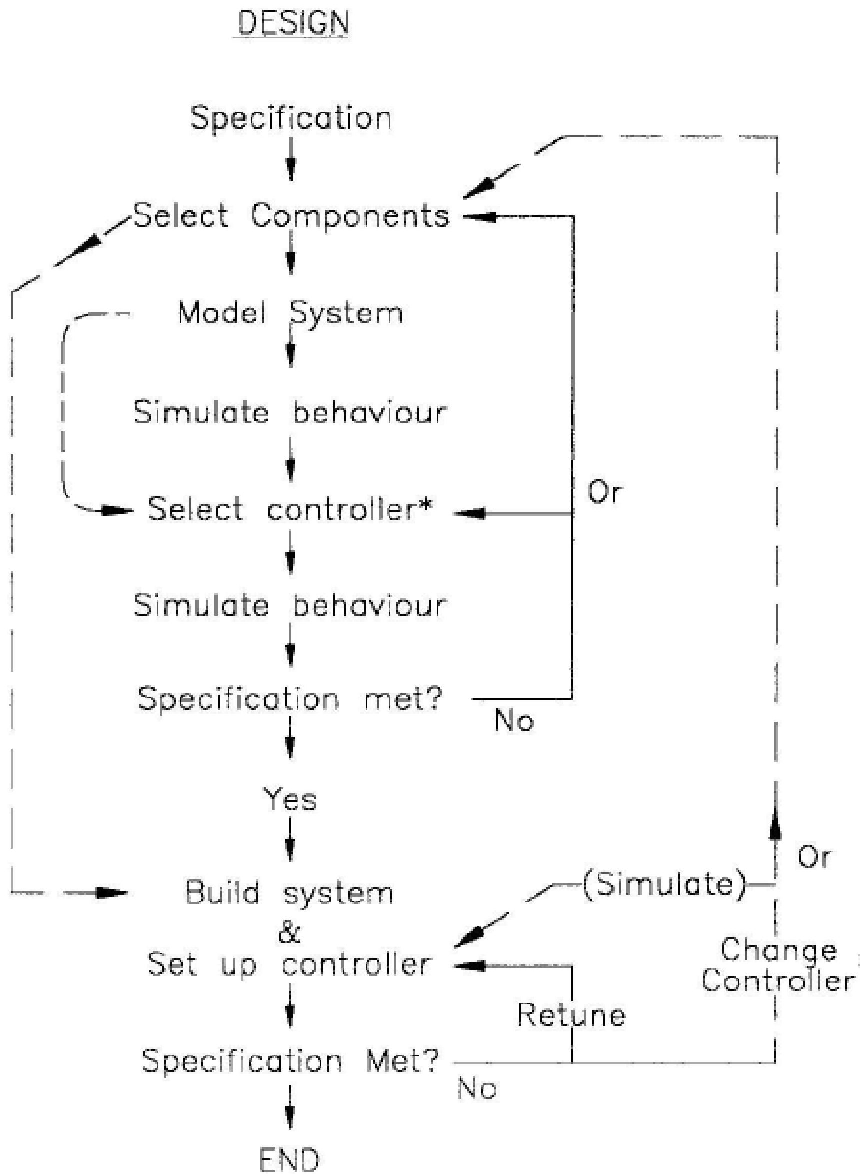


Figure 2.5 “Flowchart of design process”

2.6 Overview of Control Strategies

Control techniques and methods have been investigated and presented recently in so many works where the microprocessor has been employed, these control approaches can be outlined by many techniques.

2.6.1 PID control

Fig. 2.6.1.a shows the basic components of PID controller, by this control structure it can assure that only at high frequencies, the derivative controller can properly work. In the meantime and during the valve saturation time the wind up of the integrator can be disallowed by usually incorporation of a loop of anti-windup

There is an intrinsic integral behaviour when converting the velocity to position in the transfer function of the open loop system when considering the application as a position controller employing a hydraulic actuator

However; because of the nonlinearity in the valve function the steady state error of the system will be quite significant so that this response drift is established because of the nature of the hydraulic actuator and because of maintaining the fixed applied force, hence integral control can be considered in order to reach a minimum steady state error. As a result; there is a double integral caused by the plant and the controller, this will cause the response to get much more oscillatory in the transient phase, therefore the derivative controller can be employed to increase the nature frequency, but can be harmful for slightly damped systems (Edge 1997).

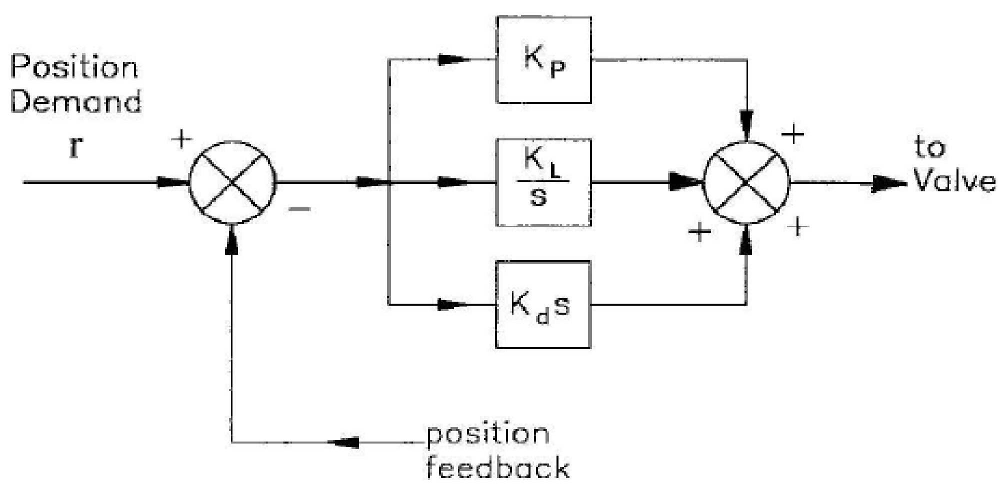


Figure 2.6.1.a block diagram for PID control system

In the meantime; with velocity controller systems, the PID controllers are generally arranged in a little different routine **(Fig. 2.6.1-b).**

Doebelin has discussed in 1985 such control approach, the control plan was executed by adding a proportional element in the feed forward path and including a semi-derivative in the a feedback to circumvent the requirement of differentiating the signal in the feedback loop (Doebelin 1985).

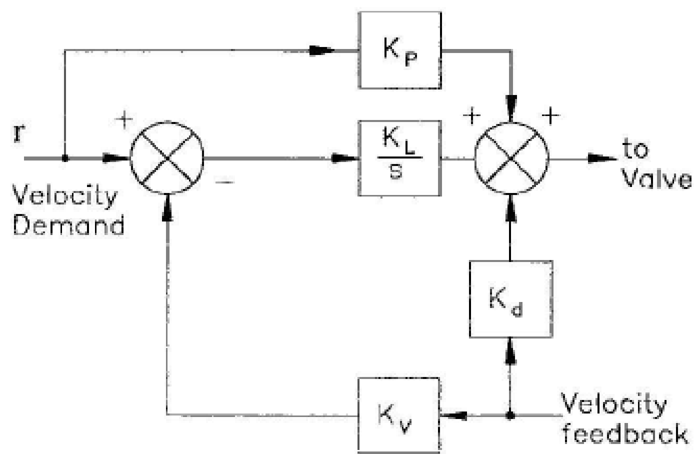


Figure 2.6.1.b block diagram for PID control system in velocity control

in 1995 three authors presented a work regarding the powerful proportional integral derivative controller, in their contribution they proposed to use a digital employment to control the shift in a hydraulic a swashplate piston pump., in the study it has been emphasized that the limitation in speed and torque are very important for the proper functional collaboration between the engine and the pump in which perfect mechanical performance can be achieved for the studied reference.

(Paoluzzi, Zarotti and Ferretti 1995)

2.6.2 State feedback control

State feedback is one of the controllers used and employed by many of authors. This control methodology has been used in many industrial applications, for example in **Fig. 2.6.2.a;** the physical parameters relevant to a positioning system such as acceleration and velocity are measured or generated by differentiation operation and fed back for the purpose of position control.

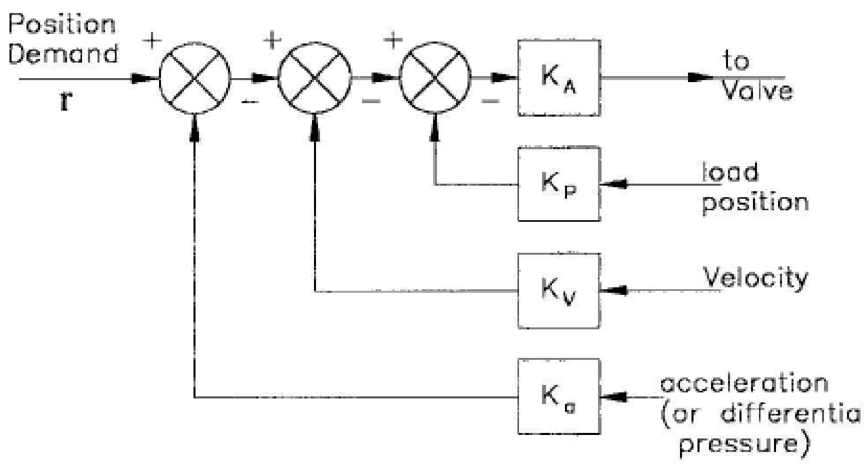


Figure 2.6.2.a Scheme of State feedback control

Full state feedback is utilized by commanding the input vector (u) which can be proportional to the state vector (x) so that:

$$u = -K x$$

(K) in the equation is the gain fed back to the control process. Using the arrangement of full state feedback or the methodology of locating poles in desired locations in order to identify and find the gains of feedback have been tackled and studied by several investigators. (Edge 1997)

The concept of pole placement controllers has been also presented by Vaughan and Plummer in 1991. The authors have investigated about the elements which can impact and influence the methodology of assigning the poles in the desired location; they applied their study on a hydraulic servomechanism driven by a digital computer controlling spool valve. A well-designed fixed gain of pole assignment system has been revealed, the approach has shown that the effect caused by major changes in the fluid pressure and the natural frequency of the hydraulic actuator is negligible and can yield to consistent performance. (Vaughau and Plummer1991)

Mare and Laffitte in 1995 have provided a valuable analysis of several methodologies of pole assignment in discrete-time representation which is controlling hydraulic servomechanism. A robust pole placement technique has been defined and presented as a perfect controller; this controller has been applied on a system of a simple dual poles system (Mare and Laffitte1995).

Ionnidis and Nguyen in, 1986 explained that when proposing a controller, the state variables can be identified by employing an observer or by applying a differentiation procedure on the signal of position, but utilizing such procedures can produce problems. An observer which should be linear can be integrated in the system, nonetheless, errors can be generated due to the non-linearity existence, it has to be noted that when the differentiation procedures become more complicated than the backward differentiation, this situation can cause a more phase lags in the feedback signal and this in turn can produce unreliable performance (Iounidis and Nguyeu1986).

The optimal control methodology was followed by many of researchers who were interested to apply this technique on hydraulic fluid applications including hydraulic servomechanism, in most of the relevant works, the gains of state feedback have been calculated for the purpose of minimizing the cost by minimizing the performance index, the optimization can take place in adopting a linear quadratic expression (Edge 1997).

2.6.3 Non-linear correction

As it has been proved that the characteristics of a spool valves are not linear; these characteristics vary with pressure, they can also vary with valve opening which is relational to the input voltage applied at the spool valve. The variation in fluid pressure can lead to variation in fluid flow and consequently these changes can influence the damping of the system.

In the meanwhile; operating conditions vary significantly the damping of the load; this non-linearity is possible to be compensated by simple algorithm providing the valve differential pressure is measured. One of the options is to work on the flow-pressure relation characteristics and to identify an area on the characteristics which shows linearity, the area can be governed by the a boundary confirmed by the maximum drive signal, by this procedure the induced damping can be eliminated practically.

The flow pressure relation is:

$$Q = ku\sqrt{\Delta P}$$

If $u = \dfrac{u_{ref}}{\sqrt{\Delta P}}$

Then $Q = ku_{ref}$

Damping can be regenerated in systems with slightly damped loads using the controller and via employing a feedback signal consists of differential fluid pressure or load acceleration in such a manner the non-linearity can be compensated through employing acceleration or differential pressure feedback, having the non-linearity compensated, a fixed damping on the response can be implemented, it is necessary to mention that such implementation is subject to specific limitation (Edge 1997).

Such concept of the nonlinear control has been also approached by (Backe1993), (Plummer1995) and (Watton 1996) as variations of the concept. Plummer included additional compensation in order to deal with the impacts of valve under- lap.

Although these methods are known as powerful and easy to implement, they have been taken in consideration as practical techniques only in the recent days (Edge 1997).

(Whiting and CotteH 1995) presented a general survey of a control system pertaining to a hydraulic servomechanism which can be programmable; they have described a compensator which can be characterized by being a non-linear, this compensator is employed to solve some unwanted behavioral issues pertaining to the spool valve and the hydraulic cylinder, the authors have explained that such impact can produce a variation in the gain which is dedicated by the direction of the stroke. (Whiting and CotteH 1995).

2.6.4 Adaptive control

Adjustment of controller parameters can be achieved by implementing a sort of automation so that the control system can react to the dominant conditions of the plant and adapt the operational changes automatically, such technique is a peer to non-linear correction concept. The technique can be implemented by employing a prior knowledge of the factors governing the performance which can lead to gain scheduling. It can be also implemented by using a self-adaptive control scheme (Edge 1997).

2.6.4.1 Gain scheduling

A Gain scheduling approach for improving the system performance in spite of its simplicity is the basis for the technique of loop gain scheduling, the loop gain can be scheduled according to the system factors which are governing the performance or the behaviour of the system response. However; in order to implement this technique, an awareness of the way how these factors impact the performance and the measurements of these factors are very important. The hydraulic piston travel can govern the scheduling of the feedback gain which can be used in the control system of the hydraulic servomechanism which employs the state feedback. A nonlinear observer can be utilized alternatively (Edge 1997).

A control system has been proposed by Virtanen in 1993 in which modeling the observer has been altered, this has been done by employing a prior awareness as per location of the load in the hydraulic servomechanism. Acceptable transient behaviour has been observed including a good transient behaviour at the end of the piston travel, while the behaviour of the system which is equipped with linear observer was insufficient (Virtanen 1993).

2.6.4.2 Self-adaptive systems

Further controlling technique is pertaining to self-adaption so that proposing a practical controller can be done without the concept of “prior awareness” mentioned earlier, in this technique, the control system can be planned to adjust itself in which a consistent dynamic performance can be implemented in the attendance of changing parameters. Such changes can happen due to many reasons, such as changes in the load, unexpected external source of inputs, variation in fluid pressure plus non-linearities relating to some system components(Edge 1997).

When applying a self-adaptation strategy (Fig. **2.6.4.2.a**), using the input-output information of application system, the system factors can be recognized instantly by the aid of the mathematical model of the system.

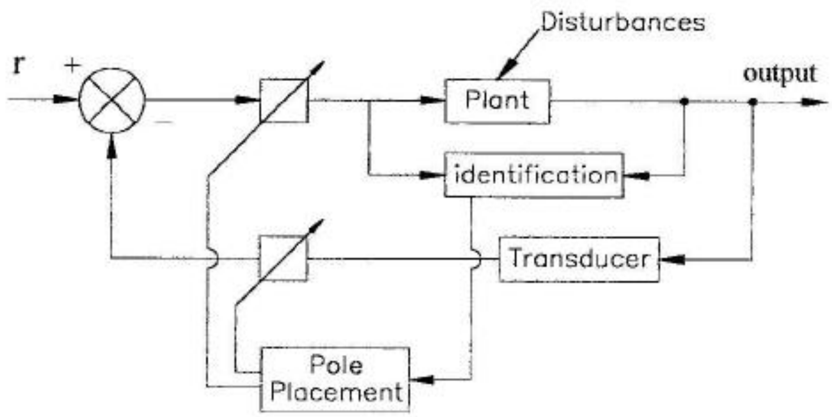


Figure 2.6.4.2.a Self-tuning control

According to the recognized system factors the control factors are informed by a proper plan process. (Fig. **2.4.4.2. a**) shows a pole placement scheme despite of other design methods which are implemented such as minimum-variance control (MVC), generalized predictive control (GPC) and linear quadratic Gaussian LQG control. The control system has been performed by modifiable digital filters (Edge 1997).

(Vaughan and Whiting 1986) have presented a study of on-line self-tuning controller employed for a hydraulic system. They proposed a pole placement methods for a hydraulic servomechanism and explained that when there is an unexpected variation in the fluid pressure, the system can be self-familiarized with that variation and recover an acceptable behaviour, although relatively slowly. The authors have indicated to a problem of short of permanent observation and execution for the self- adaptation which can reflect on factors reorganization, they have suggested alternatively to shut down the procedure of self-adjustment and correction in the time when the input signals variations are taking long time.

Another method which is similar to self-tuning control called "Model reference adaptive control (MRAC)" but is not a method that needs to recognize the values of the factors of the control system, hence this technique can be characterized as a technique controls directly more than indirectly and shown in **Fig. 2.6.4.2-b**. In this procedure, the desired system performance can be represented by a transfer function in general; the reference model as well as plant both work under the influence of similar input and desired signals so that the dissimilarity between the response of the reference model and the original system model is called "the modeling following error", this error is employed by a procedure termed as parameter adaptive algorithm (PAA) which in turn fine-tune the system factors aiming to reduce the error.

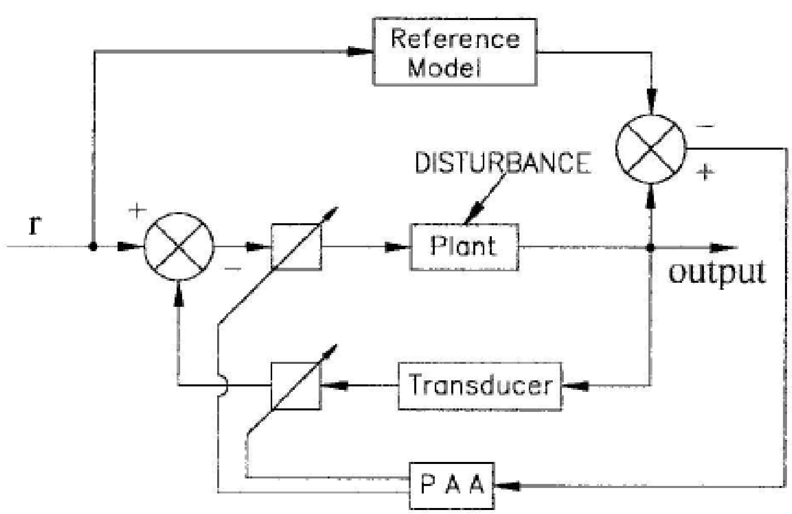


Figure 2.6.4.2.b Model reference adaptive control

The control procedure can be normally implemented by integrating adaptable digital filters, moreover, analogue filters employment is also possible (Edge 1997).

Beside to self-tuning and Model reference adaptive control, examination and proposing of extra adaptive control patterns have also taken place. For example; adaptive feed forward control has been presented by (Nakao and Urata 1994), adaptive model-based actuator control has been presented by (Andersen, Conrad, Hansen and Zhou 1994) and non-linear adaptive control has been presented by (Scavarda 1993).

2.6.5 Robust control

The term robust control has been defined by. Dorf and Bishop, they have explained that the system under robust control can remain offering an accepted behaviour when changes are occurred in the system itself (Dorf and Bishop 1995).

Variable structure and H-schemes are considered as robust control system. Sliding mode control (SMC) was discussed by Vaughan and Gamble in 1992, it was a particular sort of changeable construction of a control system in terms of contents, the control technique was magnificently considered content wise, and it was also seen operative for valve-controlled servomechanism.

In principle, it is a converting process that can push the system to operate in the transient phase of the system response with single order below the state space of the system; this reducing process will impact the performance to be not influenced by the disturbances and non-linearity. To reach such superior properties, rebuilding the states of the output with single order below system dynamics is required, for the situation of valve controlled position assembly presented by the authors, it is mainly third order, so that the feedback of physical factors expressing the load positioning process like position speed and acceleration are demanded**, Fig. 2.6.5** outlines the control technique.

When a step input is applied; the comparator in turn power the coil of the valve with complete voltage value, energizing the solenoid to be operated at a full operation extend in order to reach the wanted location. When the signals of feedback summation reach above the demanded level; then full opposite polarity of the power will take place applying the reversed power on the coil of the solenoid. When this situation happens, the comparing element will in a rapid manner swap between the polarities of the voltage applied on the solenoid. The arrangement is now can be understood that it slides alongside a phase plane space confirmed by the gains K_v and K_A. By identifying the values of these two factors, they state closed loop behaviour.

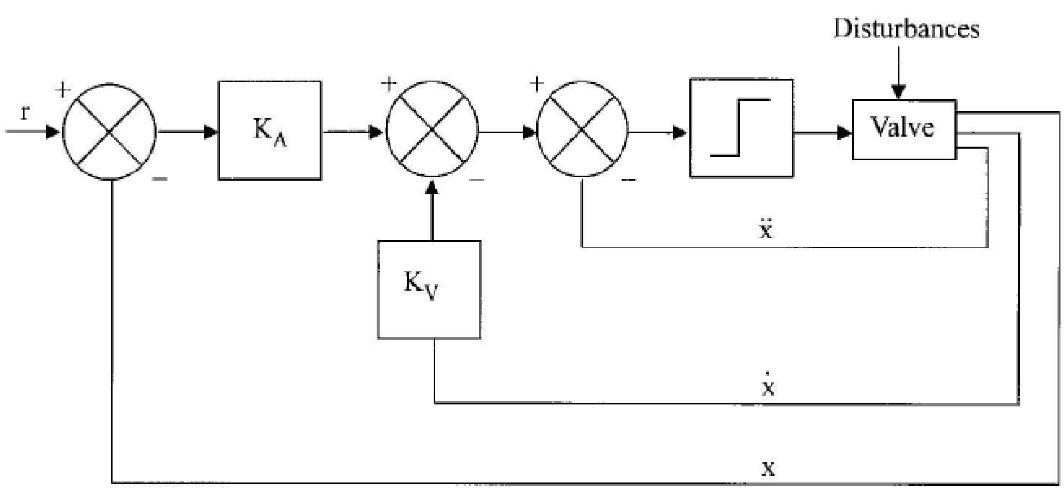


Figure 2.6.5 Simplified sliding control scheme for valve control

But the rapid switching process is considered as the main disadvantage, and creates a phenomenon which can be unfavorable to behaviour of certain systems. Vaughan and Gamble highlighted issue of the very high sampling amounts which are required to their model; they have employed digital-analogue convertor so that the higher frequencies swapping were restricted to the analogous accelerations feed-back loop systems (Vaughan and Gamble1992).

In the meanwhile; researches of SMC at the system level pertaining to hydraulic systems driven by electrical valves were published in order to cover the electro-hydraulic systems. Becker's worked in 1995 on a study pertaining to servomechanism using a hydraulic actuator; it was a prominent work when he studied the employment of swapping valve. Therefor the higher swapping amounts usually looked to be a drawback of Sliding Mode Control (Becker 1995).

(Piche and Virvalo 1991) based their work on dealing with the problem of how to design a reliable hydraulic servomechanism of a control system of lower order. The authors have drawn the attention that a linear proportional feedback control system is unique in terms of being sensitive and robust at a very wide range of frequency. A perfection for certain extend of frequencies would accordingly be decadence at specific frequencies, what was prominent hereby is the phrase "linear controller", in spite of that the paper researchers show worthy behaviour once the controller is matched to a proportional controller, it a sort of attention-grabbing by getting the comparing process with a non-linear controller and with an adaptive controller

2.6.6 Genetic algorithms

It is quite attractive to tune the controllers off-line by means of a genetic algorithm (GA) method, this procedure is employed to make the cost as less as possible. In addition to being widely used; the computer program of genetic algorithm to optimize the control process is not specified for particular industrial process implementation, the employment of it was to ease selection of specific hydraulic components for the purpose of fulfilling the requirements of the customer, nonetheless; employment a control system as a component of the procedure is normal completion of the whole process. For certain control systems relating to some application, to optimize the process would means to make the budget cost minimum. On the other hand; employment of a genetic algorithm must add value in the case where the controller is ready to be able to deal with the nonlinearity feature (Edge 1997).

A valuable review of such theory has been presented by Donne, Tilley and Richards in 1995. The authors have worked on a process how to optimize the factors of a compensating element with a feature of traditional phase lag; the plant was pertaining to hydraulic servomechanism integrated for the implementation of a press. . They examined the case when the elapsed time required to displace an object is relatively as per exact requirement. The results obtained were as per expectation; the researchers highlighted how much difficult to define the features in mathematic expressions. The major objective of the study was to show that the integration of the genetic algorithm in designing a hydraulic power industrial implementation employing hydraulic power is more importance than addressing a controller details. In considering the result of the study, the researchers did not justify why they selected a specific compensator, they did not also give any comment if alternative compensator can offer perfect behaviour (Donne, Tilley and Richards 1995).

Another methodology has been followed by Handroos and Halme in 1995; they managed to use genetic algorithm to implement live exploration about the factors of the state feedback control system used for servomechanism. They have presented that when searching the control process at different nodes the gains obtained in a parameter scheduling control system are easily confirmed. The researchers reach a conclusion that an important continuity of the study can be applying the procedure of fine-tune through fuzzy and neural controls (Handroos and Halme 1995).

2.6.7 Fuzzy logic control

It is a controlling technique which has been developed based on the fact that digital computers can only deal with precise data, hence for data where there is no precise judgment such as hot, red, tall, deep, far, etc. Humans are capable to deal with imprecise data better than the digital computer and therefore, fuzzy logic is a tool to make the digital computer capable to solve imprecise problems by coding the imprecise data before it is fed to the controller.

A structural scheme of a fuzzy control arrangement is presented in **Fig. 2.6.7**, as the diagram shows, measured amounts pertaining to quality of system's performance are at the beginning transformed to expressions of language, so that inputs are categorized, for instant "very far", "far", "near" and "very near". This process can be termed as "fuzzification". Consequently a protocol of response can be followed regarding the reaction to be taken place so that it can be expression of language as well, in other meaning; the controller is a sort of non-linear control system (Edge 1997).

Fuzzy logic controller has been employed by some researchers of fluid power in the field of hydraulic position servos; Liu and Dransfield 1993 have contributed in giving a meaningful overview about the fuzzy logic control, while (Klein and Backe 1995) presented a valid descriptive system model

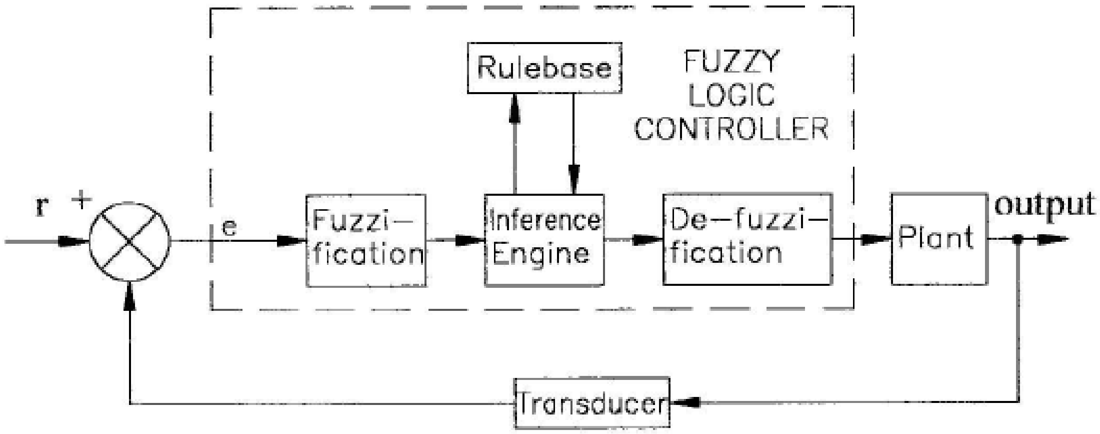


Figure 2.6.7 Fuzzy logic controller

Shih and Liaw 1993 and Niemela and Virvalo 1995 have approached control systems of the model shown as per **Fig. 2.6.7** which has been applied on power fluid servomechanism. Meanwhile; the control system proposed by Shih and Liaw employed the error signal resulted from position and the velocity of the load piston in the procedure of fuzzification; the researchers presented reliable behaviour in the existence of variation in the factors of the actuator components.

Niemela and Virvalo discussed that control systems of the model approached by Shih and Liaw can't certainly offer responses which are more reliable than state control system, they also argued that these models don't assure reliable transient and steady state responses. Alternatively, Niemela and Virvalo proposed a fuzzy state control model represents a controller of combination of fuzzy logic with traditional state feedback. One analogous system has been discussed by Liu and Dransfield in 1993, even though an extra investigative methodology has been implemented in comparison with the model proposed by Niemela and Virvalo, employment of investigative approach can be always inappropriate for certain on-off systems implementation. Moreover, to confirm regarding adopting of a rule base when it is formed through aspect of loads and components sizes for example is a non-obvious action.

A remarkable notification can be obtained to know that fuzzy logic can offer a magnificent contribution in the electrohydraulic models when they are designed in ‘open-loop' mode while the loop can be closed manually (Edge 1997).

2.6.8 Neural control

When motivation has taken place to avoid the mathematical modeling, neural control systems arose accordingly; several methods were presented in the range of hydraulic control systems. A neural system can represent a nonlinearity outlining role in the middle of input and output parameters in its simplest form. **Fig. 2.6.8** illustrates a model of a typical neural network, the inputs can be represented in a layer and can be communicated to the outputs layer, and the transmission can be conducted via a concealed layer, so that this layer can give a single or couple of outputs in an implementation of controller. The layers or sheets consist of a single or many of neurons; such neuron is arranged as per the illustration in **Fig. 2.6.8.** A weighing and summation process by means of normally a sigmoid equation take place with a bias signalized information in every neuron, the received data could be data of inputs sheet or data of other sheets. A procedure of training is utilized to perform the function of nonlinearity with the layer of inputs and outputs in order to get the selected weightiness. Awareness of the inputs and outputs data is necessary, whereas algorithmic process alters the weighted terms to reach the least error. The information confirmed and utilized to govern the training process should be representing the functional array of the signals; if this is not the case the whole system needs to go through estimation course so that it may lead to sever inconsistent responses (Edge 1997).

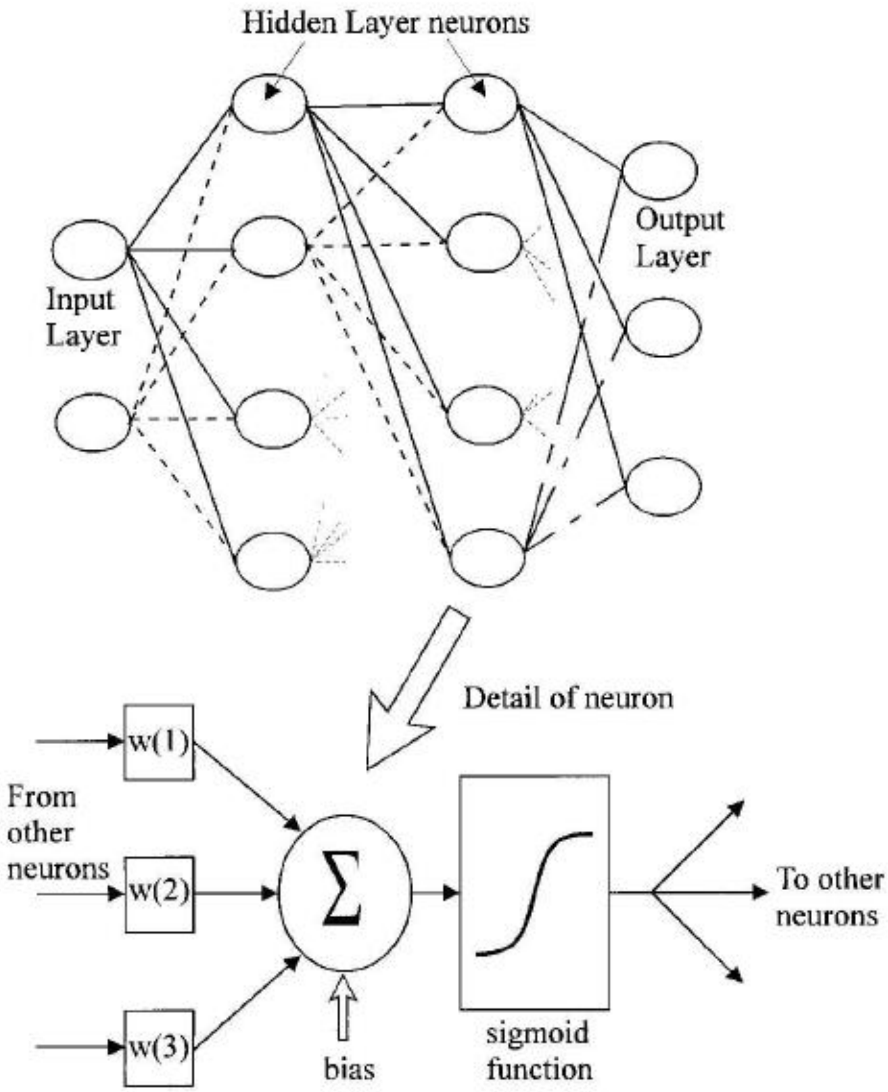


Figure 2.6.8 Example of a typical neural network

(Liu and Dransfield 1993) have approached an overview of method of neural complexes when studying control systems. Typically, the neuron complex could be utilized to be the control technique; it could be also employed to be modeling method of the transient operations in the system. Information of system dynamics will be processed when inputs are fed in the form continuous time beside their differentiations or discrete form including the present and past data. The systems in which the complex is used to act as a control organism, the training process can be applied on the complex to follow the reversed system scheme by feeding reference information to the system which in turn gives the wanted output.

Meanwhile; when the complex is employed as a modelling technique, it then acts as a non-linear model in a Model Reference Adaptive control implementation or to support in the training procedure for a network of neural control (Liu and Dransfield 1993)

Sanada, Kitagawa and Pingdong 1993 have studied in Japan a simple adaptive neural controller when they approached a valve controlled hydraulic motor speed control system. The authors have employed one layer complex as a preliminary investigational work study in which the neural networks are shown with deprived of function of non-linearity. They have studied dual structures by which an excellent behaviour has been shown for sinusoidal demand.

Shih and Lee have approached a neuron-fuzzy controller; they have highlighted the importance of the period to be spent on the methodology of trying and failing in order to select the factors of fuzzy control system. However; in order to bridge this gap, the authors have suggested a control system analogous to the one in **Fig. 2.6.7**, and concerned with the information guidelines extracted from the trained neuron with investigational information in the non-live mode. Excellent system behaviour has been obtained with the application of fluid power servomechanism. Selecting the factors employed in the training of the network is considered as a main difficulty (Shih and Lee 1995).

2.7 Summery

To sum up; there were so many controllers and regulation techniques followed by the researcher to control the performance of hydraulic servomechanism, these varied in terms of methodology of modeling, assumptions followed and the control strategy employed.

Therefore; the work in this dissertation will extend the previous works by integrating a second electrical actuator which governs leakage flow across the piston, the valve which is called in this application a needle valve will produce damping in the system response for the purpose of optimizing the performance. However; the control technique to be followed in this dissertation is pole assignment optimization with least energy for single input multiple outputs system.

Chapter Three:

Research Methodology

3.1 basic layout of hydraulic servomechanism

Welbourn in 1963 explained the basic function of the hydraulic servomechanism when he turned to steering gear of ship application. The input to the unit is produced by moving mechanically a lever which is connected to the shaft of the spool valve, while the output (which is the piston rod movement), is connected to the input as a feedback through a mechanical connection. **Fig. 3.1** illustrates the basic structure of servomechanism studied by Welbourn. The lands of the valve are assumed to cover the ports exactly, without over-lap or under-lap. Suppose that the whole unit is initially in an equilibrium position at rest, with the valve shut, and the end of the lever is moved upwards through a small amount (x), with (z) held fixed. This will result in the lever pivoting initially about (z) and the valve will open a small amount (y).

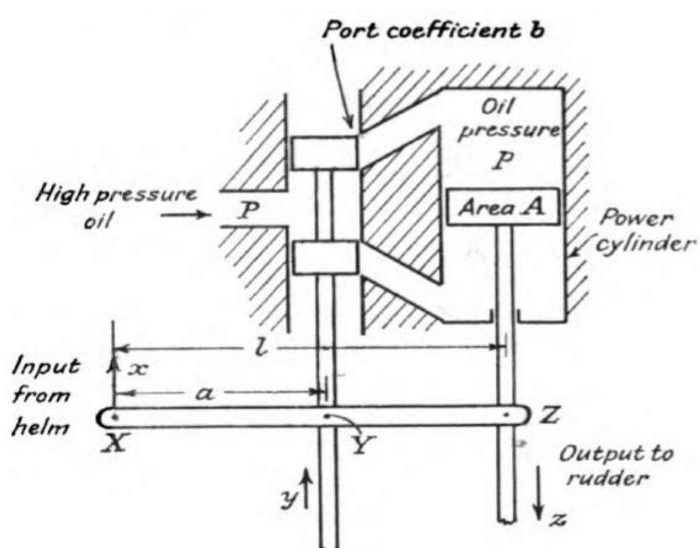


Figure 3.1 The basic structure of servomechanism

As soon as the valve starts to open, oil will start to flow into the top of the cylinder and out of the bottom of it moving the piston. As a result, the point (z) will start to move, and this will tend to shut the valve again, since the lever will now pivot about (X). As the valve shuts, the oil flow through it will become smaller and smaller, after sometime the valve will be completely shut, and the mechanism will have come to rest in a new position.

The writer has analyzed the function of the unit physically, and then this physical reality has been idealized, this methodology has been followed so as to give a mathematical expression to what is happening, in order to express the behaviour of the servomechanism in general terms. He has used two feedback loops to control the performance of the hydraulic servomechanism for the application of steering gear of ships (Welbourn 1963).

3.2 Initial assumptions

A simple servomechanism can be analyzed according to a couple of simplifying assumptions in order to derive the mathematical model and the transfer function of the system in the approximated forms.

The servomechanism is assumed that it works in the faulty condition as the hydraulic systems age with time; they work under high oil pressure that creates wear and heat. One effect can impact the operation of the system is that the hydraulic actuator is air leakage to the inside of the hydraulic circuits which makes the oil getting high compressibility (low bulk modulus value). The compressibility of the oil depends mainly on the degree of aeration in it. The assumption of compressibility means that the mass of oil in the cylinder depends not only on the volume of the cylinder, but also on the pressure it works under. An assumption is further made that the pressure in the cylinder never drops so far that cavitation occurs in the oil (Welbourn 1963).

The liquid compressibility as per (Hodges 1996), rules the power consumed and impact the duration of time the hydraulic pump takes in order to produce the powerful pressure, compressibility is also has an obvious impact on the decompression of the hydraulic fluid and the power released from this process.

The volume of a hydraulic fluid can be shrinking when it is exposed to a pressure, this process can be expressed in the following relation:

$$\text{Compressibility} = -\frac{\Delta V}{\Delta P V_0}$$

To represent the reduction in the volume; the negative sign is included in the equation. In physics the compressibility of a liquid can be expressed by a term called Bulk Modulus which is the inverse of compressibility, hence:

Bulk modulus can be expressed as:

$\beta = -\frac{V_0 \Delta P}{\Delta V}, (V_0)$ is the initial volume value, since $(\frac{V_0}{\Delta V})$ is a non-unit fraction term, the unit of bulk modulus becomes the same unit of hydraulic pressure so that high value of compressibility matches the low bulk modulus rate and vice versa. (Hodges 1996)

The bulk modulus equation can be written in the following representation:

$$DP = -\beta \frac{DV}{V_0} \qquad (3.1)$$

Compressibility as per Nikiforuk and Westlund is confirmed in the analysis based on the fact that its value can be estimated including cylinder dilation and dissolved air effects (Nikiforuk and Westlund 1965).

On the other hand, further assumptions which are considered in the approximation are aspects related to piston volume, piston area and the load inertia. The volume and the area of the piston are calculated by neglecting the rod area and volume. The output piston has inertia, the consequence of this assumption is that the difference in oil pressure across the power piston must be taken into account, and also that the rate of flow of oil through the valve ports is no longer proportional only to their opening, but also to the pressure difference across them.

It is meaningful to mention that when pressures of more than a thousand pounds per square inch are used, trouble is frequently experienced with the valves of such servos in such a manner they go into self-sustained oscillation at audible frequency. This is called valve squeal or valve scream. A servo-valve for an aircraft system working at about 4000 Ibf/in2 has been heard making a sharp howl at a frequency of a few hundred cycles per second. The effects were being produced by different mechanisms, and probably both linear and also non-linear effects were involved (Welbourn 1963).

3.3 Hydraulic servomechanism basic model

The hydraulic servomechanism subject of this thesis is generally similar to what has been studied by (Welbourn 1963). It is of pressures of more than a thousand pounds per square inch, these equipments are used in many industries such as aircrafts, ships, earth moving equipments, etc.,

Fig. 3.2 shows a simple representation of electro-hydraulic servomechanism in open loop mode.

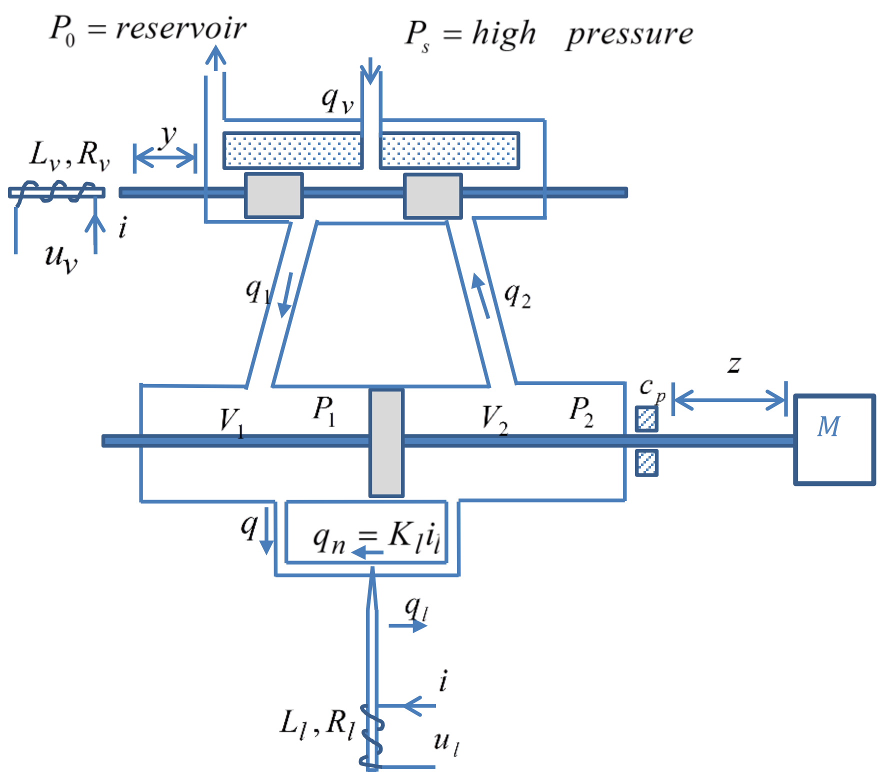


Figure 3.2 Electrohydraulic servomechanism in open loop mode

The unit has been equipped with torque motor coils energized by electrical current creates magnetic forces on ends of armature. The armature controls the oil flow in the spool valve so that the electrical signal (u_v) applied on the coils is considered as the first input of the hydraulic servomechanism. The second torque motor coils is controlling the oil flow across the piston by means of needle valve so that the electrical signal (u_l) applied on the coils is considered as the second input of the hydraulic servomechanism.

It has to be noted that the needle valve has been incorporated in the system in order to introduce damping to the piston movement; the damping in the piston movement reduces the shock forces on the piston and decreases the oscillation at the very beginning of energizing the hydraulic actuator.

3.4 Linearized and simplified dynamic model

The governing volume-pressure equation of cylinder chamber can be written the following representation:

$$f(V,P) = Q - \left(\frac{V}{A_p}P + Dz\right)\frac{DA_p}{\beta} - K_2 P = 0 \tag{3.2}$$

Equation (3.2) is a nonlinear equation, the linear approximation of $f(V,P)$ at the point (V_1, P_1) is:

$$L(V,P) = f(V_1,P_2) + \left.\frac{\partial f}{\partial V}\right|_{(V_1,P_1)} \Delta V + \left.\frac{\partial f}{\partial P}\right|_{(V_1,P_1)} \Delta P \tag{3.3}$$

At small changes in (V) and (P) which means $(z) \simeq 0$; the linearized functions becomes for cylinder chamber -1:

$$L(V,P) = f(V_1.P_1) = q_1 - \frac{DV_1}{\beta}P_1 = K_2 P_1 \tag{3.4}$$

Similarly for chamber -2:

$$q_2 - \frac{DV_2}{\beta}P_2 = K_2 P_2 \tag{3.5}$$

All the analytical forthcoming equations which model the hydraulic servomechanism including the differential equations will be considered as linear equations. Linearity has been obtained by linearizing the flow gain relation of spool displacement with valve flow and flow-pressure coefficients. However, internal and external leakage flow in the spool valve and the hydraulic cylinder are very small and assumed negligible, in the meanwhile the working temperature variation is assumed is very small as big variation in temperature will affect the density and the compressibility of the oil so that consequently, this will make the linearization near the nominal working ambient temperature not correct.

The flow rate equations through the ports which have non-linear terms can be linearized in the following forms assuming that the flow through the ports to be directly proportional both to the amount which they are open, and also to the pressure drop across them

$$q_1 = K_1 y - K_2 p_2 \tag{3.6}$$
$$q_2 = K_1 y + K_2 p_2 \tag{3.7}$$

where (K_1) is the flow gain and (K_2) is the flow-pressure coefficients

3.5 Mathematical model methodology

A similar explanation of the mathematical model of a hydraulic actuator has been explained by (Whalley R. 2012). The mechanical part of the hydraulic servomechanism can be expressed by equation which governs the motion of the load caused by load pressure and according to Newton second law:

$$M \frac{d2z}{dt^2} + c_p \frac{dz}{dt} = A_p P_L \tag{3.8}$$
$$Dv = -\frac{c_p}{M} v + \frac{A_p}{M} P_1 - \frac{A_p}{M} P_2 \tag{3.9}$$

Where the force is: $F = A_p P_L$ and the load pressure is: $P_L = P_1 - P_2$

The hydraulic servomechanism in this study has two inputs; these are the voltage applied at the torque motor coils of the spool valve, and the voltage applied on the needle valve (Walley R. 2012).

The voltage current relation at the spool valve can be explained in the following equation:

$$u_v = R_v i_v + L_v \frac{di_v}{dt} \tag{3.10}$$

$$i_v = \frac{u_v}{(R_v + L_v D)} \quad ; \quad \text{where} \quad D = \frac{d}{dt} \tag{3.11}$$

The displacement of the spool (y) is proportional to the applied current at the spool valve:

$$y = K_v i_v \tag{3.12}$$

So that substituting equation (3.11) in (3.12) yields:

$$y = K_v \frac{u_v}{(R_v + L_v D)} \tag{3.13}$$

Consequently; the spool displacement will cause the oil to flow through the ports depending on the port geometry. The flow is assumed to be directly proportional to the amount which the ports are open:

$$q_v = K_1 y \tag{3.14}$$

Plugging (3.13) in (3.14) yields

$$q_v = K_1 K_v \frac{u_v}{(R_v + L_v D)} \tag{3.15}$$

rearranging equation (3.15) leads to:

$$Dq_v = -\frac{1}{\tau_v} q_v + \frac{K_1 K_v}{L_v} u_v \tag{3.16}$$

The needle valve which is equipped in the servomechanism is analogous to the function and the structure of the universal ones. , these equipments have been designed for the purpose of controlling the movement of fluids into fragile instruments which might be broken due to unexpected liquid billow caused by liquid pressure, they are also employed to regulate the function of the fluid control in a system where the flow reaches the end of the process so the movement needs to be paused gradually, they are used also in systems where the regulation of fluid flow can accurately performed (Keith Mobley 2000).

Hence; it has been decided that the needle valve is used in the hydraulic servomechanism of this study so as to produce damping in the piston response and to reduce the shock forces.

The voltage current relation at the needle valve can be explained as follow:

$$i_l = \frac{u_l}{(R_l + L_l D)} \tag{3.17}$$

The opening amount of the needle valve (d) is proportional to the applied current at the needle valve:

$$d = i_l K_l \tag{3.18}$$

$$d = K_l \frac{u_l}{(R_l + L_l D)} \tag{3.19}$$

Opening the needle will cause the oil to leak through the stem of the valve in a flow which is proportional to the current through the coil of the needle valve, as per the following equation:

$$q_n = k_d d \tag{3.20}$$

Exchanging equation (3.19) with equation (3.20)

$$q_n = k_d K_l \frac{u_l}{(R_l + L_l D)} \tag{3.21}$$

$$Dq_n = -\frac{1}{\tau_l} q_n + \frac{k_d K_l}{L_l} u_l \tag{3.22}$$

The flow across the piston owing to pressure difference can be written as follow:

$$q = k_l(P_1 - P_2) \tag{3.23}$$

Moreover; the flow equilibrium through the needle valve:

$$q_l = q - q_n \tag{3.24}$$

By substituting equation (3.23) in equation (3.24); the leakage flow can be written as follow:

$$q_l = k_l P_1 - k_l P_2 - q_n \tag{3.25}$$

Accordingly; the flows through the cylinder chambers can be represented by another set of equations and can be determined as the following flow equilibrium in each chamber:

$$q_1 + q_l = -ADz + q_v \tag{3.26}$$

$$q_2 - q_l = ADz - q_v \tag{3.27}$$

Where (Dz) is the piston velocity which can also donate as (v) and (ADz) is the volumetric flow.

3.6 State Space representation

As per the control engineering, the servomechanism will be represented by a state space equations, it is nothing but a set of mathematical equations which can represent the model of the system, state space is a combination of three sets, inputs, outputs and state variables, all the sets are expressed together by first order differential equations. It is also featured by time-domain style and can give an appropriate and powerful procedure to exhibit and to evaluate multiple inputs multiple outputs related systems. The state space method will describe the interior structure of the hydraulic servomechanism, the state space representation informs the dynamics of the system as the first order differential equations which are part of this representation, in other words state space representation resulted from the performance of a system when dynamic response is generated from applying inputs signals and resulted also from output equations formed from the impact of the inputs on the state variables, it is worthwhile to mention that state variables can be considered as outputs of the system (Whally and A. Ameer 2011).

3.6.1 State equations

In overall situation the representation the state equations that form (n) equations are:

$$\begin{aligned} \dot{x}_1 &= f_1(x,u,t) \\ \dot{x}_2 &= f_2(x,u,t) \\ &\vdots \\ \dot{x}_n &= f_n(x,u,t) \end{aligned} \tag{3.28}$$

In matrix form:

$$\frac{d}{dt}\begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & \cdots & a_{1n} \\ a_{21} & a_{22} & \cdots & a_{2n} \\ \vdots & \vdots & \vdots & \vdots \\ a_{n1} & a_{n2} & \cdots & a_{nn} \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix} + \begin{bmatrix} b_{11} & \cdots & b_{1r} \\ b_{21} & \cdots & b_{2r} \\ \vdots & \vdots & \vdots \\ b_{n1} & \cdots & b_{nr} \end{bmatrix}\begin{bmatrix} u_1 \\ \vdots \\ u_r \end{bmatrix} \tag{3.29}$$

which can be summarized as:

$$\dot{x} = Ax + Bu \tag{3.30}$$

where (n) is number of the state variables and (r) is number of inputs (Whally and A. Ameer 2011).

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3.6.2 Output equations

The output of the system is defined to be any system variable of interest. An arbitrary variable in a system of (m) order with (r) inputs may be written**:**

$$\begin{bmatrix} y_1 \\ y_2 \\ \vdots \\ y_m \end{bmatrix} = \begin{bmatrix} c_{11} & c_{12} & \cdots & c_{1n} \\ c_{21} & c_{22} & \cdots & c_{2n} \\ \vdots & \vdots & \vdots & \vdots \\ c_{m1} & c_{m2} & \cdots & c_{mn} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix} + \begin{bmatrix} d_{11} & \cdots & d_{1r} \\ d_{21} & \cdots & d_{2r} \\ \vdots & \vdots & \vdots \\ d_{m1} & \cdots & d_{mr} \end{bmatrix} \begin{bmatrix} u_1 \\ \vdots \\ u_r \end{bmatrix} \tag{3.31}$$

which can be summarized as:

$$y = Cx + Bu \tag{3.32}$$

In many of applications (D) matrix is a zero matrix which in turn simplifies the output expression into (Whally and A. Ameer 2011):

$$y = Cx \tag{3.33}$$

3.6.3 Obtaining A, B, C, D matrices from PQR form of Hydraulic Servomechanism:

The hydraulic servomechanism has a vector of two inputs, the voltage (u_v) applied on the spool valve and (u_l) applied on the needle valve:

$$u = [u_v \quad u_l]^T \tag{3.34}$$

The output vector is the piston velocity (v) and the leakage flow (q_l) across the piston, so that; the output vector is:

$$y = [v \quad q_l]^T \tag{3.35}$$

However; the state vector is:

$$x = [q_v \quad P_1 \quad P_2 \quad q_n \quad v]^T \tag{3.36}$$

It should be noted that as the output is velocity, a simple integrator can be added to the system at the velocity output in order to obtain the position of the piston.

To illustrate the analysis process; the set of state equations representing the hydraulic servomechanism are equations (3.4), (3.5), (3.9), (3.16), (3.22), (3.25), (3.26) and (3.27) respectively. Nonetheless; to achieve the convenience of P, Q, and R method; the last equation to be considered is:

$$v = v \tag{3.37}$$

The multivariable system of the hydraulic servomechanism with two inputs and two outputs can be represented in **table -1**(Whally and A. Ameer 2011).

Table 1: Multivariable System Representation of state and outputs equations

Derivatives					Outputs					State variables						Inputs	
$\dot{q}$	$\dot{P}_1$	$\dot{P}_2$	$\dot{q}_n$	$\dot{v}$	q_1	q_2	q_l	v	=	q_v	P_1	P_2	q_n	v	+	u_v	u_l
1	0	0	0	0	0	0	0	0		$\frac{-1}{\tau_v}$	0	0	0	0		$\frac{K_1K_v}{\tau_v}$	0
0	$\frac{V_1}{\beta}$	0	0	0	-1	0	0	0		0	$-K_2$	0	0	0		0	0
0	0	$\frac{V_2}{\beta}$	0	0	0	-1	0	0		0	0	$-K_2$	0	0		0	0
0	0	0	1	0	0	0	0	0		0	0	0	$\frac{-1}{\tau_l}$	0		0	$\frac{k_dK_l}{L_l}$
0	0	0	0	1	0	0	0	0		0	$\frac{A}{M}$	$\frac{-A}{M}$		$\frac{-c_p}{M}$		0	0
0	0	0	0	0	1	0	1	0		1	0	0	0	$-A$		0	0
0	0	0	0	0	0	1	-1	0		-1	0	0	0	A		0	0
0	0	0	0	0	0	0	1	0		0	k_l	$-k_l$	-1	0		0	0
0	0	0	0	0	0	0	0	1		0	0	0	0	1		0	0

In order to obtain A, B, C, D matrices; the following raw operations can be carried out on the table 1:

- Add raw (8) to raw (7)
- Multiply raw (8) by (-1) and add it to raw (6)
- Add raw (7) to raw (3)
- Add raw (6) to raw (2)
- Divide raw (2) by $(\frac{V_1}{\beta})$ and raw (3) $(\frac{V_2}{\beta})$,

The outcome of the above raw operations can lead to obtain PQR matrices, table-2 represent the said matrices.

Table 2: PQR matrices

Derivatives					Outputs					State variables						Inputs	
$\dot{q}_v$	$\dot{P}_1$	$\dot{P}_2$	$\dot{q}_n$	$\dot{v}$	q_1	q_2	q_l	v	=	q_v	P_1	P_2	q_n	v	+	u_v	u_l
1	0	0	0	0	0	0	0	0		$\frac{-1}{\tau_v}$	0	0	0	0		$\frac{K_1K_v}{\tau_v}$	0
0	1	0	0	0	−1	0	0	0		$\frac{\beta}{V_1}$	$\frac{\beta}{V_1}(-K_2-k_l)$	$\frac{\beta}{V_1}k_l$	$\frac{\beta}{V_1}$	$\frac{-A\beta}{V_1}$		0	0
0	0	1	0	0	0	−1	0	0		$\frac{-\beta}{V_2}$	$\frac{\beta k_l}{V_2}$	$\frac{\beta}{V_2}(-K_2-k_1)$	$\frac{-\beta}{V_2}$	$\frac{A\beta}{V_2}$		0	0
0	0	0	1	0	0	0	0	0		0	0	0	$\frac{-1}{\tau_l}$	0		0	$\frac{k_dK_l}{L_l}$
0	0	0	0	1	0	0	0	0		0	$\frac{A}{M}$	$\frac{-A}{M}$	0	$\frac{-c_p}{M}$		0	0
0	0	0	0	0	1	0	1	0		1	$-k_l$	k_l	1	$-A$		0	0
0	0	0	0	0	0	1	−1	0		−1	k_l	$-k_l$	−1	A		0	0
0	0	0	0	0	0	0	1	0		0	k_l	$-k_l$	−1	0		0	0
0	0	0	0	0	0	0	0	1		0	0	0	0	1		0	0

$[P]$ $[Q]$ $[R]$

From the table -2 the PQR representation can be written as follows:

$$[P]\begin{bmatrix} \dot{q}_v \\ \dot{P}_1 \\ \dot{P}_2 \\ \dot{q}_n \\ \dot{v} \\ q_1 \\ q_2 \\ q_l \\ v \end{bmatrix} = [Q]\begin{bmatrix} q_v \\ P_1 \\ P_2 \\ q_n \\ v \end{bmatrix} + [R]\begin{bmatrix} u_v \\ u_l \end{bmatrix}$$

The aim is to reach a case that [P] matrix is converted to [I] matrix which is the outcome of the above rows operations conducted on table 1- in order to reach table -2, hence:

$$\begin{bmatrix} \dot{q}_v \\ \dot{P}_1 \\ \dot{P}_2 \\ \dot{q}_n \\ \dot{v} \\ q_1 \\ q_2 \\ q_l \\ v \end{bmatrix} = [P]^{-1}[Q]\begin{bmatrix} q_v \\ P_1 \\ P_2 \\ q_n \\ v \end{bmatrix} + [P]^{-1}[R]\begin{bmatrix} u_v \\ u_l \end{bmatrix}$$

Then:

$$\begin{bmatrix} \dot{x} \\ \cdots \\ y \end{bmatrix} = \begin{bmatrix} A & \vdots & B \\ \cdots & \cdots & \cdots \\ C & \vdots & D \end{bmatrix}\begin{bmatrix} x \\ u \end{bmatrix}$$

where:

$$\begin{bmatrix} A \\ \\ \cdots \\ \\ C \end{bmatrix} = P^{-1}Q \quad , \quad \begin{bmatrix} B \\ \\ \cdots \\ \\ D \end{bmatrix} = P^{-1}R$$

Based on that; A, B, C, D matrices can be identified as follow

$$A = \begin{bmatrix} -\frac{1}{\tau_v} & 0 & 0 & 0 & 0 \\ \frac{\beta}{V_1} & \frac{\beta}{V_1}(-k_2 - k_l) & \frac{\beta}{V_1}k_l & \frac{\beta}{V_1} & -A\frac{\beta}{V_1} \\ \frac{-\beta}{V_2} & \frac{\beta}{V_2}k_l & \frac{\beta}{V_2}(-k_2 - k_l) & \frac{-\beta}{V_2} & \frac{\beta}{V_2}A \\ 0 & 0 & 0 & \frac{-1}{\tau_l} & 0 \\ 0 & \frac{A}{M} & -\frac{A}{M} & 0 & \frac{-c_p}{M} \end{bmatrix}$$

$$B = \begin{bmatrix} \frac{k_1 K_v}{L_v} & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & \frac{k_d K_l}{L_l} \\ 0 & 0 \end{bmatrix} \quad , \quad C = \begin{bmatrix} 0 & k_l & -k_l & -1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \quad , \quad D = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix}$$

So that the hydraulic servomechanism can be represented in the state space form as follow:

$$D\begin{bmatrix} q_v \\ P_1 \\ P_2 \\ q_n \\ v \end{bmatrix} = \begin{bmatrix} -\frac{1}{\tau_v} & 0 & 0 & 0 & 0 \\ \frac{\beta}{V_1} & \frac{\beta}{V_1}(-k_2-k_l) & \frac{\beta}{V_1}k_l & \frac{\beta}{V_1} & -A\frac{\beta}{V_1} \\ \frac{-\beta}{V_2} & \frac{\beta}{V_2}k_l & \frac{\beta}{V_2}(-k_2-k_l) & \frac{-\beta}{V_2} & \frac{\beta}{V_2}A \\ 0 & 0 & 0 & \frac{-1}{\tau_l} & 0 \\ 0 & \frac{A}{M} & -\frac{A}{M} & 0 & \frac{-c_p}{M} \end{bmatrix} \begin{bmatrix} q_v \\ P_1 \\ P_2 \\ q_n \\ v \end{bmatrix} +$$

$$+\begin{bmatrix} \frac{k_1 K_v}{L_v} & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & \frac{k_d K_l}{L_l} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} u_v \\ u_l \end{bmatrix} \qquad (3.38)$$

while the output equations can be represented as:

$$\begin{bmatrix} q_l \\ v \end{bmatrix} = \begin{bmatrix} 0 & k_l & -k_l & -1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} q_v \\ P_1 \\ P_2 \\ q_n \\ v \end{bmatrix} \qquad (3.39)$$

3.7 System conversion from State Space into Laplace representation:

$$L\begin{pmatrix} \dot{x} = Ax + Bu \\ y = Cx + Du \end{pmatrix} \equiv$$

$$sx(s) - x(0) = Ax(s) + Bu(s)$$

$$y(s) = Cx(s) + Du(x) \tag{3.40}$$

$$\left(sI - A\right)x(s) = x(0) + Bu(s)$$

$$x(s) = \left(sI - A\right)^{-1}\left[x(0) + Bu(s)\right]$$

If $x(0) = 0$ and D matrix = null

$$x(s) = \left(sI - A\right)^{-1} Bu(s) \tag{3.41}$$

Plugging (3.41) in (3.40) yields:

$$y(s) = C(sI - A)^{-1} Bu(s) \tag{3.42}$$

Hence the transfer function of multiple input multiple outputs servomechanism system can be represented in a matrix of functions:

$$G(s) = \frac{y(s)}{u(s)} = C(sI - A)^{-1} B = \frac{\begin{bmatrix} g_{11} & g_{12} \\ g_{21} & g_{22} \end{bmatrix}}{\Delta(s)} \tag{3.43}$$

Where the input and output vectors are respectively:

$$u = \begin{bmatrix} u_v & u_l \end{bmatrix}^T$$

$$y = \begin{bmatrix} v & q_l \end{bmatrix}^T$$

Hence:

$$\begin{bmatrix} q_l \\ v \end{bmatrix} = \frac{\begin{bmatrix} g_{11} & g_{12} \\ g_{21} & g_{22} \end{bmatrix}}{\Delta(s)} \begin{bmatrix} u_v \\ u_l \end{bmatrix} \tag{3.44}$$

On the other hand; $\Delta(s)$ is the determinant of the matrix $(sI - A)$, the determinant is also is known as the characteristic equation of the multivariable system.

The hydraulic servomechanism pertinent data are given in table-3:

Table-3: Hydraulic servomechanism pertinent data

Spool displacement- current gain (K_v)	=	6	in/mA
Spool valve flow gain (K_1)	=	37	In3/sec/in
Spool valve inductance coil (L_v)	=	0.4	H
Spool valve coil resistance (R_v)	=	24	Ohm
Piston area (A)	=	5.4	in2
V1 volume in chamber (1)	=	17	in3
V2 volume in chamber (2)	=	17	in3
Pressure flow coefficient (K_2)	=	0.35	in3/sec/psi
Flow gain of the needle valve (k_d)	=	0.81	in3/sec/in
Needle displacement- current gain (K_l)	=	1.43	in/mA
Pressure flow coefficient of the needle valve (k_l)	=	0.001	in3/sec/psi
Needle valve coil resistance (R_l)	=	36	Ohm
Spool valve inductance coil (L_l)	=	0.6	H
Bulk modulus (β)	=	100	Pound/in2 (psi)
Viscous friction (c_p)	=	298	N/in/sec
Load mass (M)	=	43	lb

The bulk modulus has been considered with low value in order to examine the system at faulty conditions which can take place when the system gets air in the hydraulic lines, so the system becomes spongier, this value selection will also show the impact of unequal piston position on the at later stage.

By substituting the above parameters in A, B, C matrices of the equation (3.43); the elements of the transfer function matrix can be obtained as follow:

$$\frac{g_{11}}{\Delta(s)} = \frac{6.529s^3 + 450.5s^2 + 3615s + 5590}{s^5 + 131.1s^4 + 4968s^3 + 4.476x10^4 s^2 + 1.526x10^5 s + 1.655x10^5}$$

$$\frac{g_{12}}{\Delta(s)} = \frac{-1.931s^4 - 137.2s^3 - 1358s^2 - 4809s - 5305}{s^5 + 131.1s^4 + 4968s^3 + 4.476x10^4 s^2 + 1.526x10^5 s + 1.655x10^5}$$

$$\frac{g_{21}}{\Delta(s)} = \frac{820s^2 + 5.089x10^4 s + 1.013x10^5}{s^5 + 131.1s^4 + 4968s^3 + 4.476x10^4 s^2 + 1.526x10^5 s + 1.655x10^5}$$

$$\frac{g_{22}}{\Delta(s)} = \frac{2.852s^2 + 177s + 352.3}{s^5 + 131.1s^4 + 4968s^3 + 4.476x10^4 s^2 + 1.526x10^5 s + 1.655x10^5}$$

where

$$\Delta(s) = s^5 + 131.1s^4 + 4968s^3 + 4.476x10^4 s^2 + 1.526x10^5 s + 1.655x10^5 \qquad (3.45)$$

As per equation (3.44); the output equations can be obtained as follow:

$$q_l = u_v \frac{g_{11}}{\Delta(s)} + u_l \frac{g_{12}}{\Delta(s)}$$

$$v = u_v \frac{g_{21}}{\Delta(s)} + u_l \frac{g_{22}}{\Delta(s)}$$

The blocks diagram of the open loop multiple inputs multiple output system can be shown in (**Fig. 3.3**) as follow:

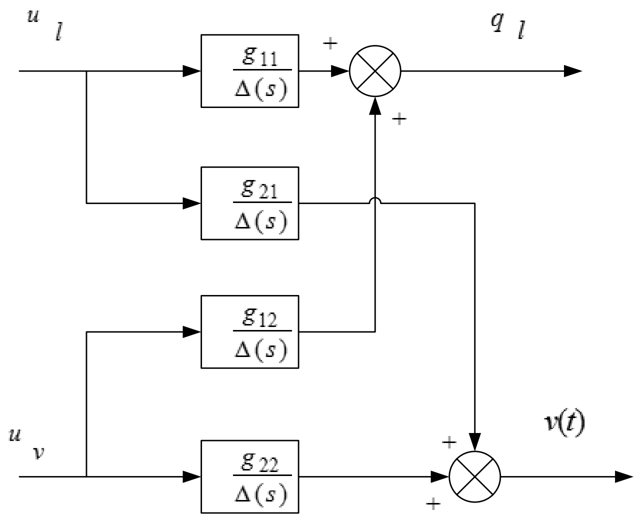


Figure 3.3 blocks diagram of the open loop of multiple input multiple output

However; in order to employ the effect of the needle valve, it should be energized at the same time the spool valve is energized so that the complete functions of the servomechanism including the damping function caused by the leakage flow can be obtained; therefore both the needle and the spool valves must be energized by the same power supply at the same time.

Let $u_v = u_l = \varepsilon$

$$q_l = \left(\frac{g_{11}(s) + g_{12}(s)}{\Delta(s)}\right)\varepsilon = g_1(s)\varepsilon$$

$$v = \left(\frac{g_{21}(s) + g_{22}(s)}{\Delta(s)}\right)\varepsilon = g_2(s)\varepsilon$$

Then the system can be considered as single input multiple output servomechanisms as per the following block diagram in **(Fig. 3.4)**:

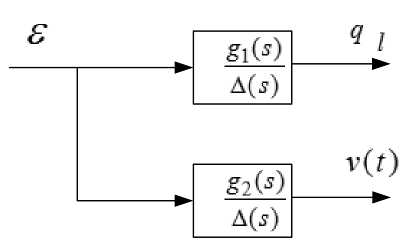


Figure 3.4 block diagram of single input multiple output system

So that:

$$g_1(s) = \frac{g_{11}(s) + g_{12}(s)}{\Delta(s)} = \frac{-1.931s^4 - 130.6s^3 - 907.9s^2 - 1194s + 284.6}{s^5 + 131.1s^4 + 4968s^3 + 4.476x10^4 s^2 + 1.526x10^5 s + 1.655x10^5} \tag{3.46}$$

$$g_2(s) = \frac{g_{21}(s) + g_{22}(s)}{\Delta(s)} = \frac{822.8s^2 + 5.106x10^4 s + 1.016x10^5}{s^5 + 131.1s^4 + 4968s^3 4.476x10^4 s^2 + 1.526x10^5 s + 1.655x10^5} \tag{3.47}$$

Fig. (3.5) illustrates the detailed block diagram of the Hydraulic Servomechanism for the open loop system:

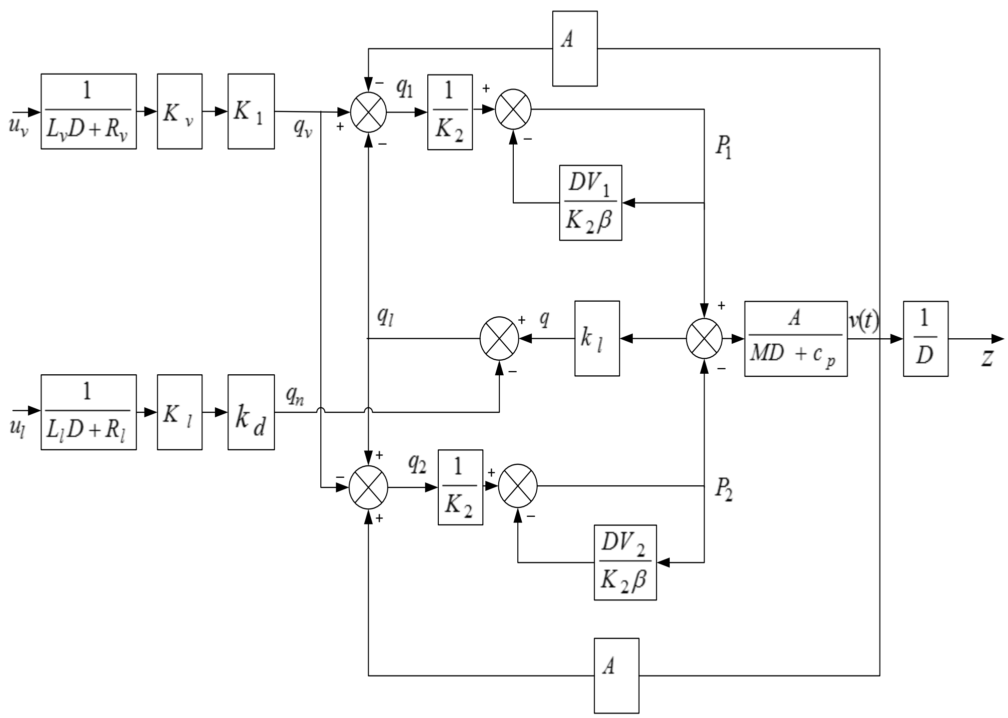


Figure 3.5 Hydraulic servomechanism block diagram in open loop mode

Fig.s (3.6), (3.7), (3.8), (3.9), (3.10) and (3.11) illustrate the response of the valve flow (q_v), leakage flow across the piston (q_l), oil pressure at cylinder chamber (1) (P_1), oil pressure at cylinder chamber (2) (P_2), oil load pressure (P_L) and piston velocity (v) respectively:

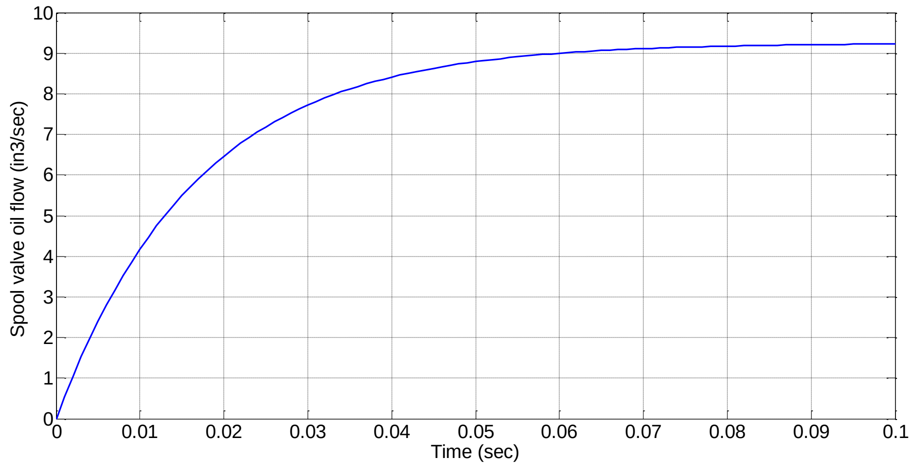


Figure 3.6 step response of valve flow (q_v),

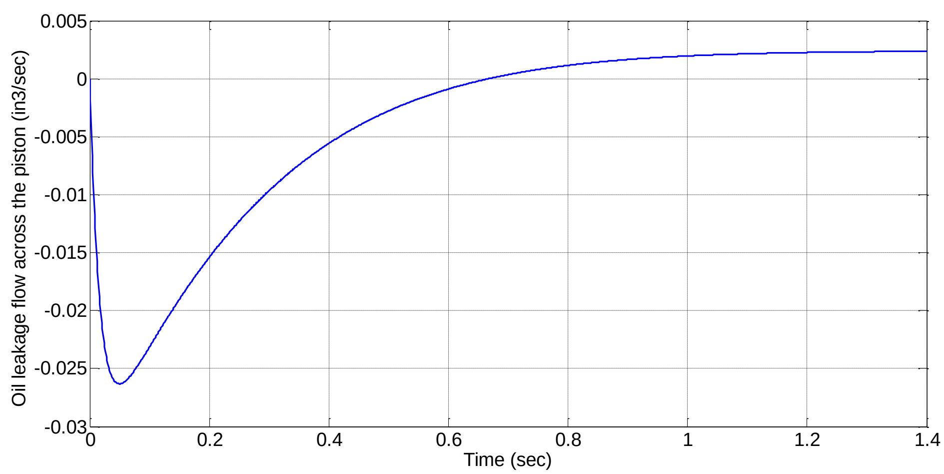
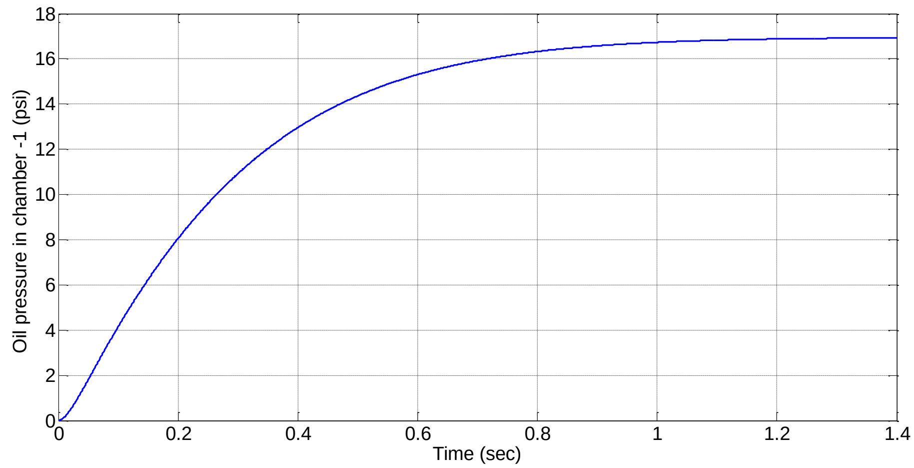
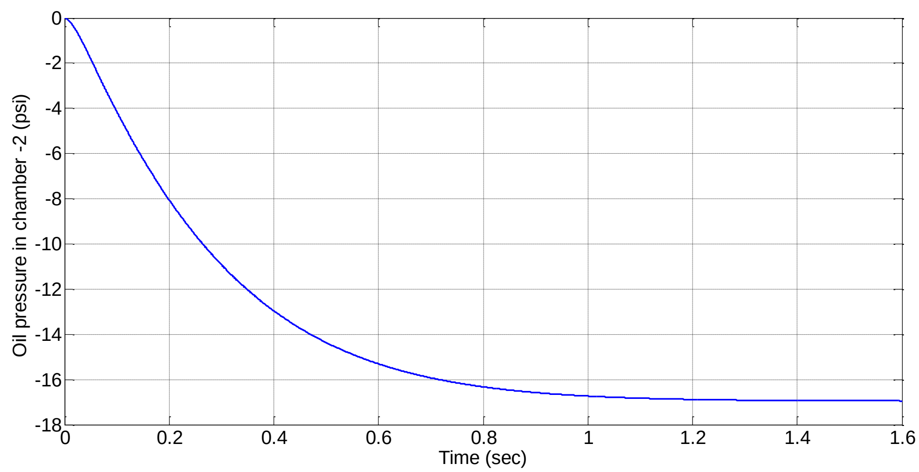


Figure 3.7 Step response of leakage flow across the piston (q_l)



Figures 3.8 Step response of oil pressure at cylinder chamber (1) (P_1)



Figures 3.9 Step response of oil pressure at cylinder chamber (1) (P_2)

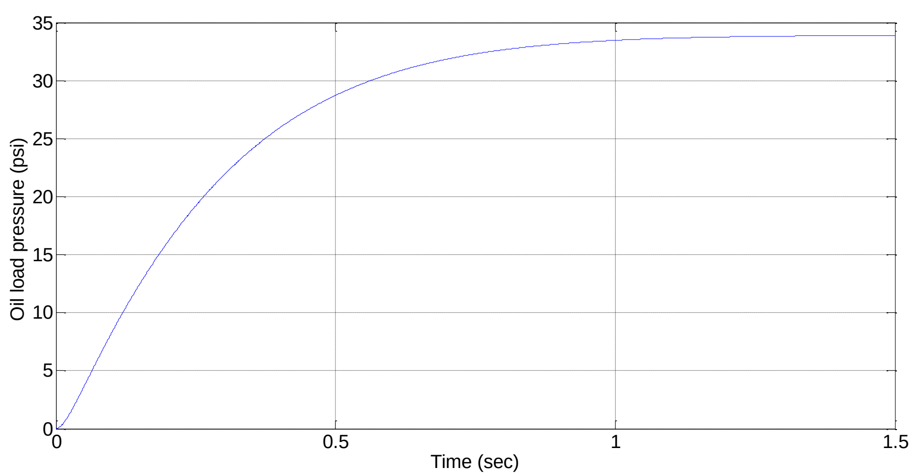


Figure 3.10 Step response of oil load pressure (P_L)

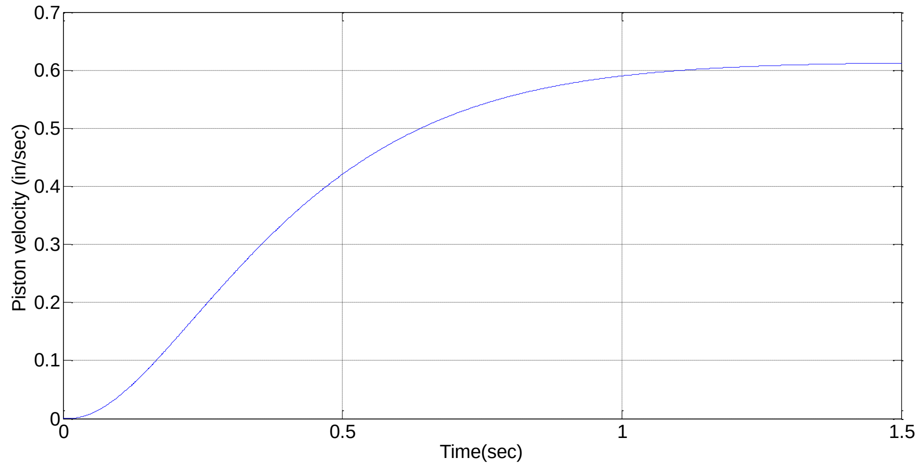


Figure 3.11 Step response of piston velocity (v)

3.8 Control strategy:

The system block diagram in closed loop with the feedback and forward gains can be represented in **Fig. (3.12),** disturbances are also shown:

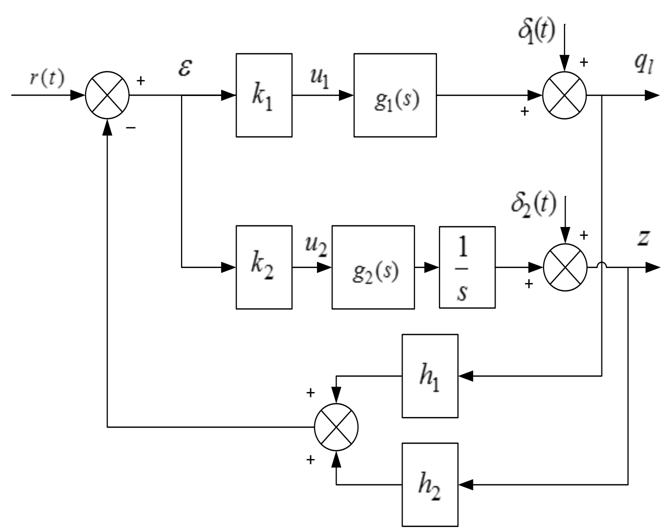


Figure 3.12 Single input multiple output of a hydraulic servomechanism in closed loop

The control technique is employing "optimization of pole assignment with least energy consumption" of a single input multiple output of a hydraulic servomechanism. This control strategy goes through particular processes to calculate and to identify the particular values of the forward and the feedback gains illustrated in **Fig. (3.12)**, the resulted gains must be optimizing the poles location of the closed loop system in such a manner the outputs of the system responses are improved and the least energy consumption.

The integrator added after at the piston velocity in **Fig. (3.12)** is going to give the piston position response, consequently; the transfer function $G_2(s)/s$ can be arranged as follow:

$$G_2(s) = \frac{g_2(s)}{s} = \frac{5.064x10^4 s^2 + 6.105x10^7 s + 1.306x10^{10}}{s^6 + 131.1s^5 + 4968s^4 + 4.476x10^4 s^3 + 1.526x10^5 s^2 + 1.655x10^5 s} \qquad (3.48)$$

However; the control procedure requires having common characteristic equation for both transfer functions, hence the $G_1(s)$ can be also written as follow:

$$G_1(s) = g_1(s)\frac{s}{s} = \frac{-390.7s^5 - 4.475x10^4 s^4 - 2.651x10^4 s^3 + 1.87x10^6 s^2 + 6.129x10^5 s}{s^6 + 131.1s^5 + 4968s^4 + 4.476x10^4 s^3 + 1.526x10^5 s^2 + 1.655x10^5 s} \qquad (3.49)$$

Where the common characteristic equation is:

$$\Delta(s) = s^6 + 131.1s^5 + 4968s^4 + 4.476x10^4 s^3 + 1.526x10^5 s^2 + 1.655x10^5 s \qquad (3.50)$$

As per equation (3.50) which shows the characteristic equation of the open loop hydraulic servomechanism; the poles of the open loop system can be found by calculating the roots of the characteristic equation:

The calculated poles are shown in vector (λ_i):

$$\lambda_i = \begin{bmatrix} -60 + 8.1\text{x}10^{-7}i \\ -60 - 8.1\text{x}10^{-7}i \\ -4.5 + 1.44i \\ -4.5 - 1.44i \\ -2.06 \\ 0 \end{bmatrix}$$

Most of the poles of the open loop are located on left of the imaginary axis of the (s) plane except one pole which is located at the origin. This pole makes the system marginally stable and is going to ramp the output without limitation unless proper feedback is employed in order to limit the piston position.

Fig. (3.13.a) illustrates the position response for non-optimal position response with random values of forward and feedback gains normal unity feedback, The rise time of the piston position response is approximately (3.7 sec), while **Fig. (3.13.b)** illustrates the block diagram of closed loop of the non-optimal control**.**

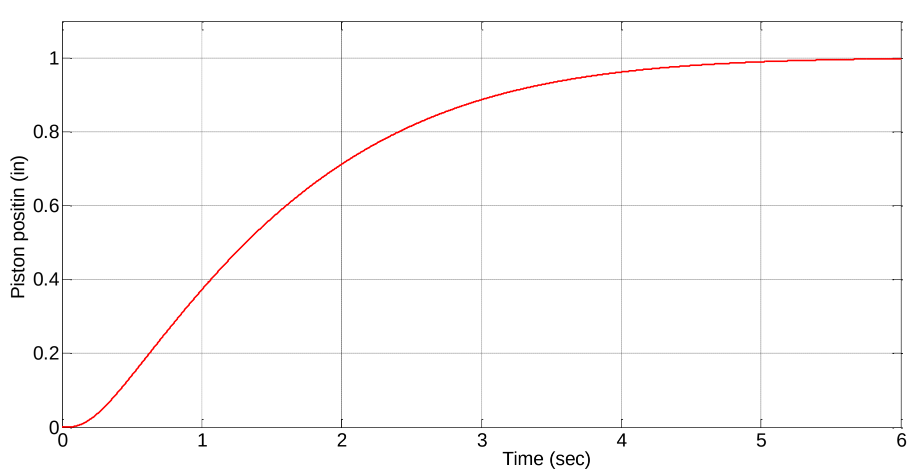


Figure 3.13.a Piston position response with normal unity feedback (non-optimal)

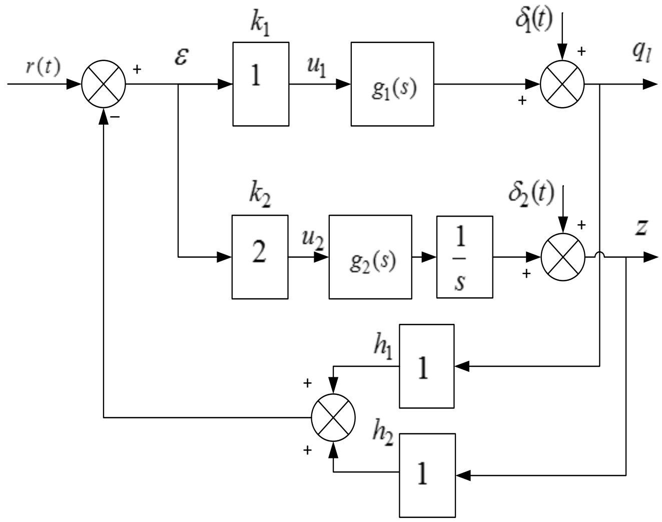


Figure 3.13.b block diagram of the closed loop system with non-optimal control

On the other hand; the control strategy used in this study also limits the piston position and optimizes the location of the closed loop poles. An investigation into such control strategy will be presented.

3.9 Theory of optimization of pole assignment:

3.9.1 Derivation of the closed loop characteristic equation

The procedure of pole assignment optimization goes through the following derivations based on block diagram in **Fig. (3.12)**

In matrix representation, the output of closed loop system on single input multiple outputs can be derived as follow:

The input signals can be written as per the following equation

$$\begin{bmatrix} u_1(s) \\ u_2(s) \end{bmatrix} = \varepsilon(s) \begin{bmatrix} k_1 \\ k_2 \end{bmatrix} \tag{3.51}$$

While the output signals can be written as per the following equation

$$\begin{bmatrix} q_l \\ \\ z \end{bmatrix} = \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} u_1(s) \\ \\ u_2(s) \end{bmatrix} + \begin{bmatrix} \delta_1(s) \\ \\ \delta_2(s) \end{bmatrix} \tag{3.52}$$

Equation (3.51) in (3.52) yields:

$$\begin{bmatrix} q_l \\ \\ z \end{bmatrix} = \varepsilon \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} + \begin{bmatrix} \delta_1(s) \\ \\ \delta_2(s) \end{bmatrix} \tag{3.53}$$

In the meanwhile the error signal (ε) can be calculated as per the following equation:

$$\varepsilon(s) = r(s) - \begin{bmatrix} h_1(s) & h_2(s) \end{bmatrix} \begin{bmatrix} q_l \\ \\ z \end{bmatrix} \tag{3.54}$$

Equation (3.53) in (3.54) yields:

$$\begin{bmatrix} q_l \\ \\ z \end{bmatrix} = \left(\left(r(s) - \begin{bmatrix} h_1(s) & h_2(s) \end{bmatrix} \begin{bmatrix} q_l \\ \\ z \end{bmatrix} \right) \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} \right) + \begin{bmatrix} \delta_1(s) \\ \\ \delta_2(s) \end{bmatrix} \tag{3.55}$$

By arranging equation (3.55); equation (3.56) can be obtained

$$\begin{bmatrix} q_l \\ \\ z \end{bmatrix} \left(I + \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} \begin{bmatrix} h_1(s) & h_2(s) \end{bmatrix} \right) =$$

$$= r(s) \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} + \begin{bmatrix} \delta_1(s) \\ \\ \delta_2(s) \end{bmatrix} \tag{3.56}$$

The output equation of the single input multiple outputs can be written as per equation (3.56) as follow:

$$\begin{bmatrix} q_l \\ \\ z \end{bmatrix} = \left(I + \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} \begin{bmatrix} h_1(s) & h_2(s) \end{bmatrix} \right)^{-1} \left(r(s) \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} + \begin{bmatrix} \delta_1(s) \\ \\ \delta_2(s) \end{bmatrix} \right) \tag{3.57}$$

Rearranging Equation (3.57) would give equation (3.58)

$$\begin{bmatrix} q_l \\ \\ z \end{bmatrix} = \frac{adj\left(I + \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} \begin{bmatrix} h_1(s) & h_2(s) \end{bmatrix} \right) \left(r(s) \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} + \begin{bmatrix} \delta_1(s) \\ \\ \delta_2(s) \end{bmatrix} \right)}{\det\left(I + \begin{bmatrix} G_1(s) & 0 \\ \\ 0 & G_2(s) \end{bmatrix} \begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} \begin{bmatrix} h_1(s) & h_2(s) \end{bmatrix} \right)} \tag{3.58}$$

So that the denominator of equation (3.58) $[\det(I + GKH)]$ is the characteristic equation of the closed loop system:

$$\det(\bullet) = 1 + h_1 k_1 g_1(s) \frac{s}{s} + h_2 k_2 \frac{g_2(s)}{s}$$

$$\Rightarrow \det(\bullet) = 1 + h_1 k_1 g_1(s) \frac{s}{s} + h_2 k_2 \frac{g_2(s)}{s} \tag{3.59}$$

3.9.2 Optimization requirements:

The classical approach to control system design is based on time domain and frequency domain analysis such as root locus, Bode and Nyquist techniques. Nevertheless, the modern approach of design is based on problem formulation in time domain with performance index (PI) which is optimal. A quantifiable value of behaviour of a system is necessary for parameter setting optimization of a control system and design of optimum systems. In general to design a control system or to improve its design, a performance index (PI) attains an extremum, normally a minimum value. In order to design a system with performance indices optimally, the optimality criterion and optimal design approach are to be adopted. Optimal control theory provides a simple and powerful tool for designing SISO as well as MIMO systems with optimal (PI) (Srivastava, MC Srivastava and Bhatnagar 2009).

The optimization concept is taking place according to (Whalley and Ebrahimi 2000) when the control efforts at time (t) are proportional to a system of two inputs at the forward path $u_1(t)$ and $u_2(t)$ with two outputs $y_1(t)$ and $y_2(t)$ as per the following equation (Whalley and Ebrahimi 2000):

$$\left(|k_1h_1|+|k_2h_1|\right)y_1(t)+\left(|k_1h_2|+|k_2h_2|\right)y_2(t) \tag{3.60}$$

So that the cost of the energy consumed in the control process can be expressed as follow:

$$E(t)=\int_{t=0}^{t=T_f}\left(\sum_{i=1}^{2}k_i^2\sum_{j=1}^{2}h_j^2y_j^2\right)dt \tag{3.61}$$

Formerly, for random variations in the converted output vector $y(t)$, subsequent to random disturbance variations

$$J = \sum_{i=1}^{2} k_i^2 \sum_{j=1}^{2} h_j^2 \qquad (3.62)$$

in turn can play a significant role in minimizing the energy of control expressed in equation (3.61).

If the relationship:

$$\begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} = \begin{bmatrix} 1 \\ \\ nk_1 \end{bmatrix} \qquad (3.63)$$

is considered , then performance index relation (3.62) can be expressed as per (Whalley and Ebrahimi 2000):

$$J = (k_1^2)(1+n^2)(h_1^2)(h_2^2) \qquad (3.64)$$

Referring to the characteristic equation (3.59) which includes the forward as well as feedback gains; these gains can be optimized in such a manner they can be calculated based on the Performance Index (J) extermum value in equation (3.64). Consequently; the extremum value of (J) is going to match specific value of the gain ratio (n).

By identifying (n) value; the forward and feedback gains can be also identified, so that in order to get the optimization results, a very important requirement pertaining to reducing complexity of the numerators of $G_1(s)$ and $G_2(s)$ by reducing their order to first order equations should be achieved, numerator reduced order of $G_1(s)$ and $G_2(s)$ can be written as follow:

$$\tilde{G}_1(s) = \frac{b_1 + a_1}{\Delta(s)} \text{ and } \tilde{G}_2(s) = \frac{b_2 + a_2}{\Delta(s)} \qquad (3.65)$$

It should be emphasized that the order reduction should maintain the same dynamics of the original equation of higher order, therefore; reduced $\tilde{G}_1(s)$ with first order numerator polynomial is:

$$\tilde{G}_1(s) = \frac{284.6s}{s\Delta(s)}$$

$$\tilde{G}_1(s) = \frac{284.6s}{s^6 + 131.1s^5 + 4968s^4 + 4.476x10^4 s^3 + 1.526x10^5 s^2 + 1.655x10^5 s} \quad (3.66)$$

and reduced $\tilde{G}_2(s)$ with first order polynomial is:

$$\tilde{G}_2(s) = \frac{5.106x10^4 s + 1.016\text{x}10^5}{s\Delta(s)}$$

$$\tilde{G}_2(s) = \frac{5.106x10^4 s + 1.016\text{x}10^5}{s^6 + 131.1s^5 + 4968s^4 + 4.476x10^4 s^3 + +1.526x10^5 s^2 + 1.655x10^5 s} \quad (3.67)$$

Fig. (3.14) and **Fig. (3.15)** show the simulation responses of reduced numerator transfer functions $\tilde{G}_1(s)$ and $\tilde{G}_2(s)$ with comparison to original transfer function responses of $G_1(s)$ and $G_2(s)$ respectively. The Figures show obviously that both responses are absolutely overlapping.

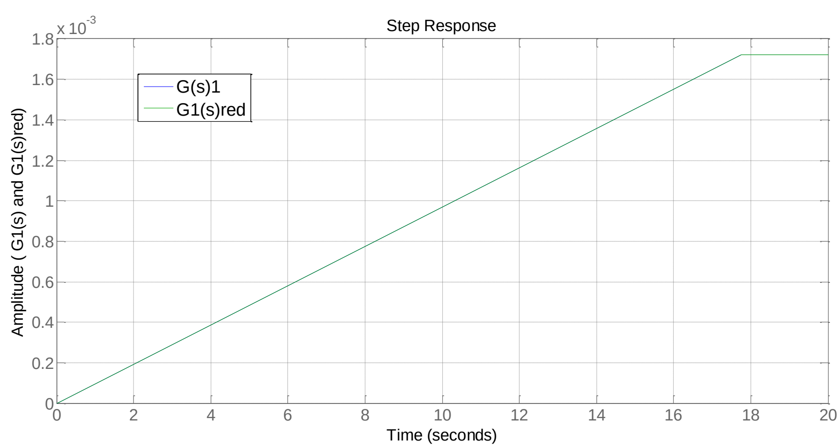


Figure 3.14 Oil leakage response in open loop system for $\tilde{G}_1(s)$ and $G_1(s)$

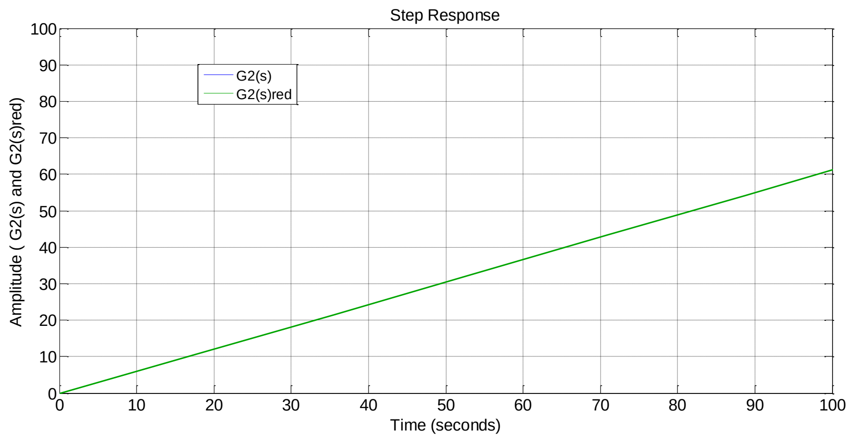


Figure 3.15 Piston position response in open loop system for $\tilde{G}_2(s)$ and $G_2(s)$

The coefficients of the order reduced numerators are:

$$\begin{bmatrix} a_1 \\ \\ b_1 \end{bmatrix} = \begin{bmatrix} 0 \\ \\ 284.6 \end{bmatrix} \quad \text{and} \quad \begin{bmatrix} a_2 \\ \\ b_2 \end{bmatrix} = \begin{bmatrix} 1.016x10^5 \\ \\ 5.106x10^4 \end{bmatrix} \tag{3.68}$$

Accordingly; equation (3.59) can be written in the following representation:

$$\det(\bullet) = 1 + \begin{bmatrix} h_1 & h_2 \end{bmatrix} \frac{\begin{bmatrix} (b_1 s + a_1)K_1 \\ \\ (b_2 s + a_2)K_2 \end{bmatrix}}{s\Delta(s)} \tag{3.69}$$

Considering $(K_1 = 1)$ yields:

$$\det(\bullet) = 1 + \frac{(b_1 s + a_1)h_1 + (b_2 s + a_2)nh_2}{s\Delta(s)} = 1 + \frac{p_1 + p_2 s}{s\Delta(s)} \tag{3.70}$$

Then, equation (3.70) becomes the characteristic equation of the single input multiple output servomechanism system.

Considering root locus definition; the characteristic equation in (3.70) can be arranged as follow:

$$-1 = p_2 \frac{\frac{p_1}{p_2} + s}{s\Delta(s)} \tag{3.71}$$

The poles of the denominator $(s\Delta(s))$ are:

$(0\ ,\ -60+8.11x10^{-7}i\ ,\ -60-8.11x10^{-7}\ ,-4.5+1.44i\ ,\ -4.5-1.44i\ ,\ -2)$

and shown on the s-plane as per **Fig. (3.16):**

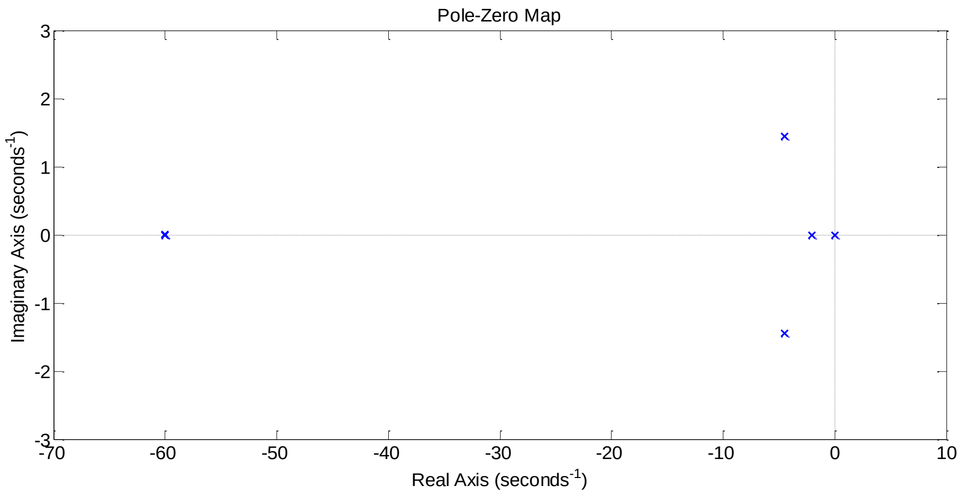
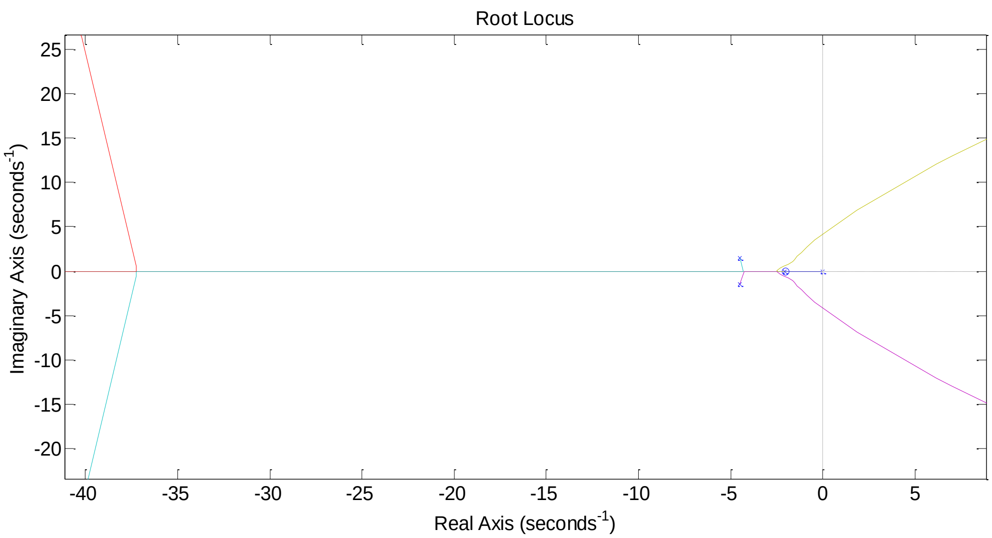


Figure 3.16 poles map of the characteristic equation

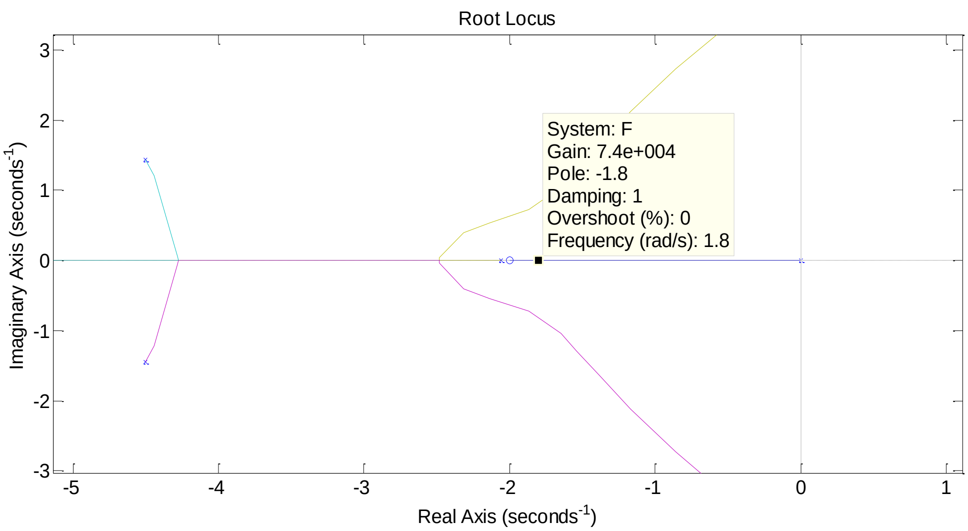
Referring to equation (3.71); the root locus gain is (p_2), while the controller generated zero is (p_1 / p_2) so that suitable $(p_1 + p_2 s)$ function based on beneficial dynamic responses can be identified from the root locus plot of equation (3.71) which is illustrated in **Fig. (3.17-a** and **3.17-b).**

Values of (p_1) and (p_2) can be identified as follow:

The controller generated zero has been selected with a value of (-2) so that it can attract the pole at the origin, with the pole resided at $(s=-1.8)$ on the real axis, the relevant gain at $(s=-1.8)$ is (gain=$7.42\text{x}10^4$).



Figures 3.17.a root locus of closed loop system



Figures 3.17.b root locus of closed loop system

Consequently; the values of (p_1) and (p_2) can be identified as:

$$p_2 = 7.42x10^4 \tag{3.72}$$

$$\frac{p_1}{p_2} = 2$$

$$p_1 = 14.84x10^4 \tag{3.73}$$

Recalling equation (3.70), it can be written in the following form:

$$\frac{[1 \quad s]}{\Delta(s)}\begin{bmatrix} a_1 & na_2 \\ b_1 & nb_2 \end{bmatrix}\begin{bmatrix} h_1 \\ h_2 \end{bmatrix} = \frac{[1 \quad s]}{\Delta(s)}\begin{bmatrix} p_1 \\ p_2 \end{bmatrix} \tag{3.74}$$

so that equation (3.75) can be attained:

$$\begin{bmatrix} a_1 & na_2 \\ b_1 & nb_2 \end{bmatrix}\begin{bmatrix} h_1 \\ h_2 \end{bmatrix} = \begin{bmatrix} p_1 \\ p_2 \end{bmatrix} \tag{3.75}$$

By substituting equations (3.72), (3.73) and (3.68) in equation (3.75); equation (3.76) can be reached

$$\begin{bmatrix} h_1 \\ h_2 \end{bmatrix} = \begin{bmatrix} 0 & n1.016x10^5 \\ 284.6 & n5.104x10^4 \end{bmatrix}^{-1}\begin{bmatrix} 14.84x10^4 \\ 7.42x10^4 \end{bmatrix} \tag{3.76}$$

Substituting equation (3.76) in (3.64) with considering $(k_1 = 1)$ yields:

$$J(n) = \frac{1.769n^4 - 4.16n^3 + 4.215n^2 - 4.16n + 2.446}{1.166n^2} \tag{3.77}$$

To find the minimum value of $J(n)$, its derivative must be equated to zero in order to find the values of (n) which make $\frac{dJ(n)}{dn}$ zero.

Hence:

$$\frac{dJ}{dn} = \frac{4.126n^5 - 4.853n^4 + 4.853n^2 - 5.707n}{1.361n^4} = 0$$

Equating $(\frac{dJ}{dn} = 0)$ tells that the values at which performance index $J(n)$ can be at low and high peak values when:

$$4.126n^5 - 4.853n^4 + 4.853n^2 - 5.707n = 0 \tag{3.78}$$

Roots of equation (3.78) are

$$\Rightarrow \begin{bmatrix} n_1 \\ n_2 \\ n_3 \\ n_4 \\ n_5 \end{bmatrix} = \begin{bmatrix} 0 \\ -1.055 \\ 1.176 \\ 0.5277 + 0.914i \\ 0.5277 - 0.914i \end{bmatrix} \tag{3.79}$$

The performance index (J) plot as per gain ration (n) is represented in **Fig. (3.18),** it reveals that the minimum value of (J) is achieved when $(n_3 = 1.176)$.

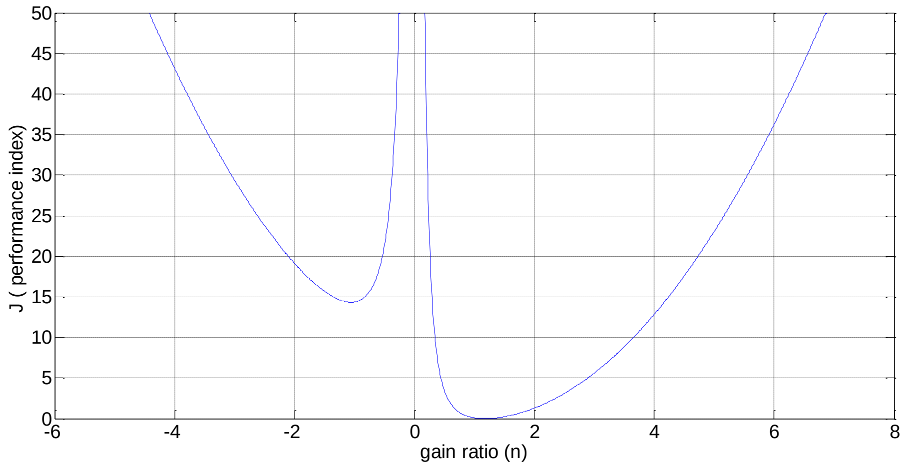


Figure 3.18 Performance Index (J) against gain ratio (n)

so that the

$$J(1.176) = 1.3x10^{-15} \tag{3.80}$$

By substituting $(n = 1.176)$ value in equation (3.76); the values of the feedback gains (h_1) and (h_2) can be identified as:

$$\begin{bmatrix} h_1 & h_2 \end{bmatrix} = \begin{bmatrix} -1.23 & 1.23 \end{bmatrix} \tag{3.81}$$

In the meanwhile the forward gains can be computed by equation (3.63) equation to yield:

$$\begin{bmatrix} k_1 \\ \\ k_2 \end{bmatrix} = \begin{bmatrix} 1 \\ \\ 1.176 \end{bmatrix} \tag{3.82}$$

The block diagram of the closed loop system with identified least effort parameters is shown in **Fig. 3.19**

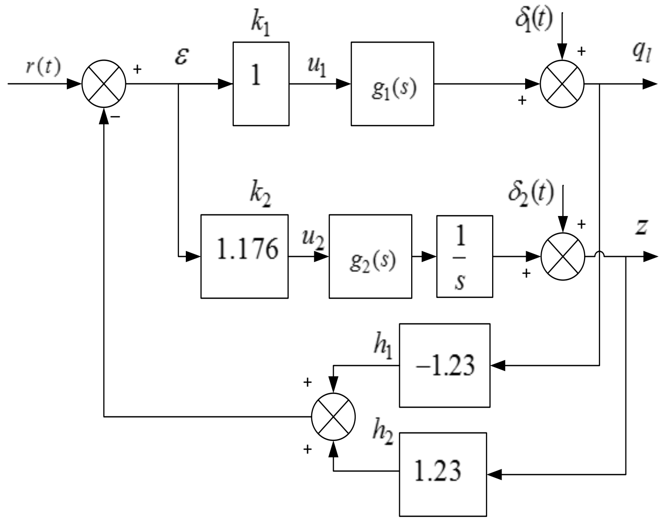


Figure 3.19 block diagram of the closed loop system with the optimization parameters

For the sake of comparison **Fig. 3.20** shows the block diagram of the closed loop system in non-optimal control in which the values of the forward and feedback gains selected for a stable system as $(k_1 = 1\ ,\ k_2 = 2\ ,\ h_1 = h_2 = 1)$

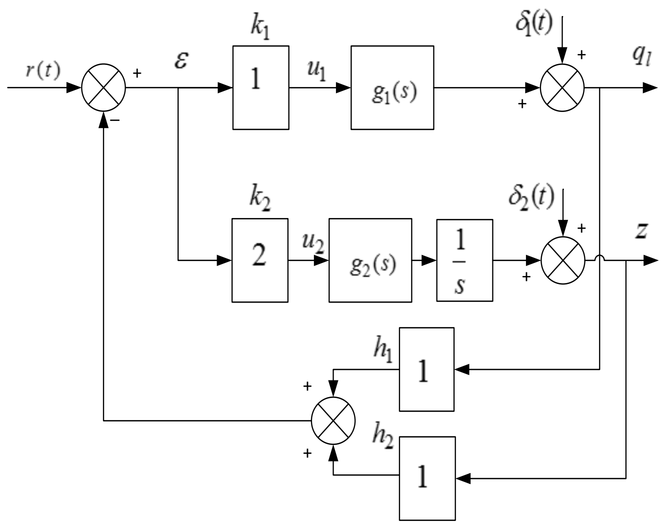


Figure 3.20 block diagram of the closed loop system with non-optimal control

The step response for the piston position with poles location optimization control, in comparison with position response with non-optimal close loop control is shown in **Fig. (3.21)**, it is obviously shown that the response of the optimal control is faster than the non-optimal one.

On the other hand leakage flow in optimal and non-optimal control is shown in **Fig. (3.22),** the response of leakage flow in optimal control is improved over the non-optimal one as well, and the leakage flow is acting an important role by producing and adding extra damping to reduce the heavy oscillation during the transient so that it is not affecting the steady state of the piston position.

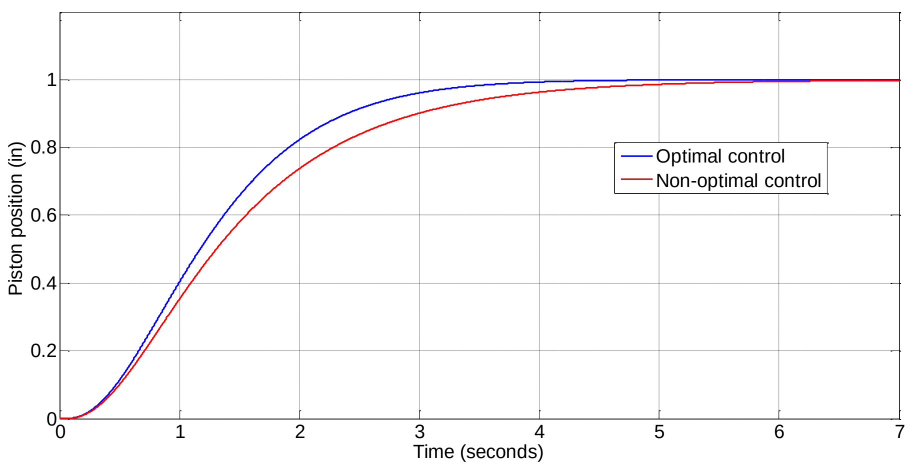


Figure 3.21 Piston position step responses of optimal and non-optimal control

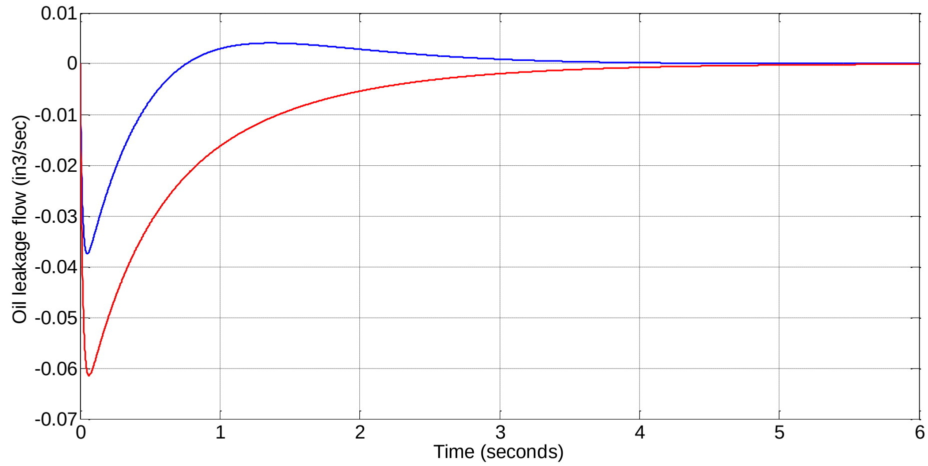


Figure 3.22 Step response of the leakage flow in optimized and non-optimized control

On the other hand the system is also behaves well when disturbances are applied at the system, **Fig. 3.23** shows the behaviour of the system when a step disturbance is acting on the load connected to the piston at idle mode (input signal $r(t) = 0$), the disturbance has been rejected and the control system managed to return back the response to the setting point of zero.

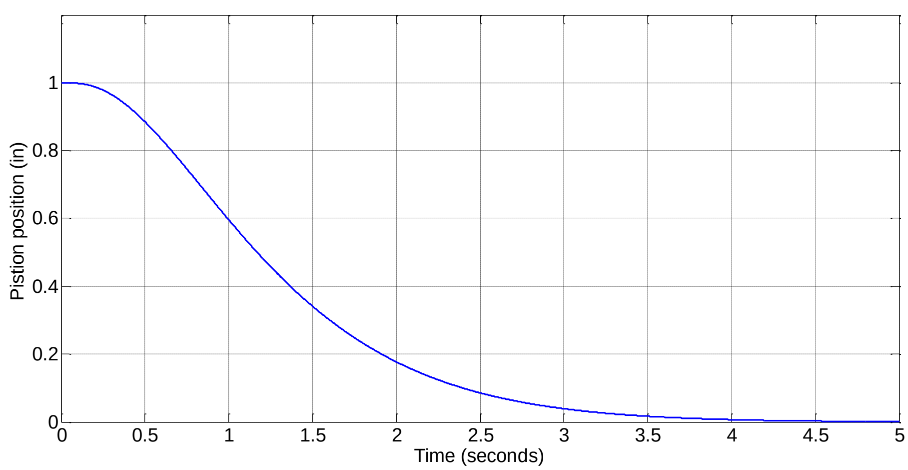


Figure 3.23 Piston position response when step disturbance is applied on the piston load

3.10 Piston responses at different chambers volumes

The previous analysis had considered that the changes in chamber volume-1 and volume of chamber-2 up to now are very small when the piston is moving in either ways. Such slight changes are still within the range of linearization validation. However, when the piston is not located at the middle of the cylinder, non-linearity can be established and the response cannot be analytically predicted, but it can be simulated, the non-linearity is happening due the product term of $(\frac{V}{A_p}P)$ in equation (3.5) so that the changes in $(\frac{V}{A_p})$ and (P) are not slight and must be considered, the simulation of the piston responses can help to confirm how far the unequal chambers volumes of the cylinder with acceptable piston position responses.

Simulation of the servomechanism at different chambers volumes is shown in **Fig. (3.24)**

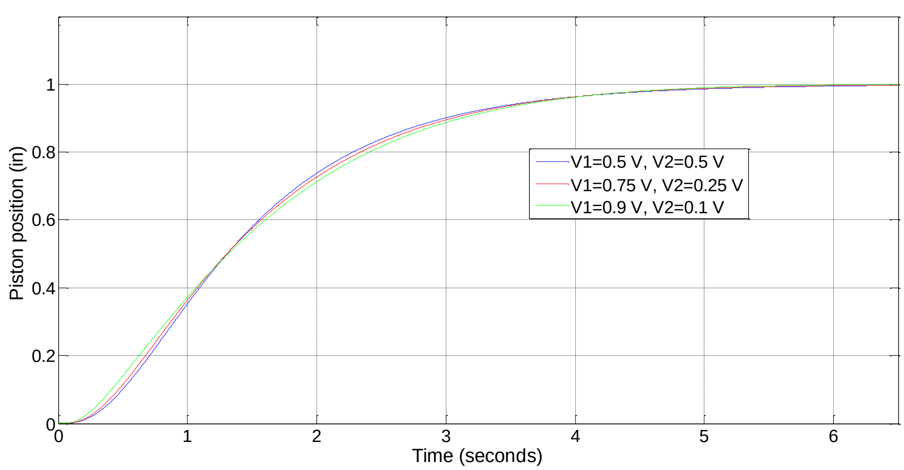


Figure 3.24 Piston position step responses in optimal form with different cylinder chamber volumes

According to **Fig. 3.24**; it can be noticed that the piston position is getting analogous responses and with slight changes at different chamber volumes; therefore a conclusion can be reached that the unequal volumes are not significantly impacting the piston position responses so that the servomechanism can work properly and with stability regardless how much the difference in chamber volumes.

3.11 Control Energy costs

So far; the paper has been intending to demonstrate an optimal control strategy for the Hydraulic system actuator so that the proposal of the controller is a challenge to come across a group of provisions which describe the general presentation of the system regarding specific computable measures, such as the behaviour processes related to dynamic responses to step input (rise time, steady state error, etc.). Other anticipated feature of a performance index is the least energy consumption that focuses on the possibility to minimize the energy cost incurred by the system Energy consumption costs can be calculated as per the following equation (Nagrath and Gopal 2007):

$$E(t) = \int_{0}^{T_F} u_1^2(t) + u_2^2(t)dt \tag{3.83}$$

If the expenses of consuming control energy according to **Fig. 3.25** when suppressing random noise disturbances matching equation (3.83) are computed with $r(t) = 0$, at this point, proportional energy dissipated records in **Fig. (3.26)** can be accomplished (Whalley and Ebrahimi 2004).

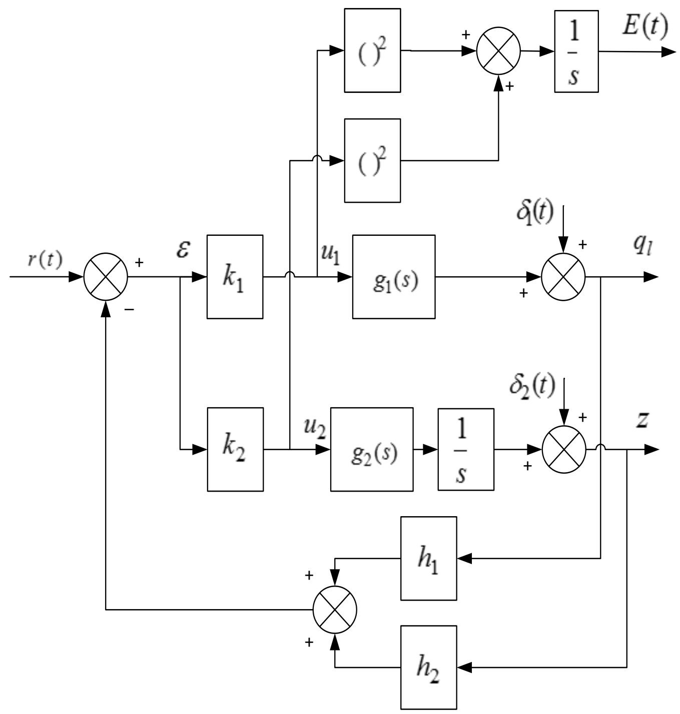


Figure 3.25 Block diagram of energy consumption calculation

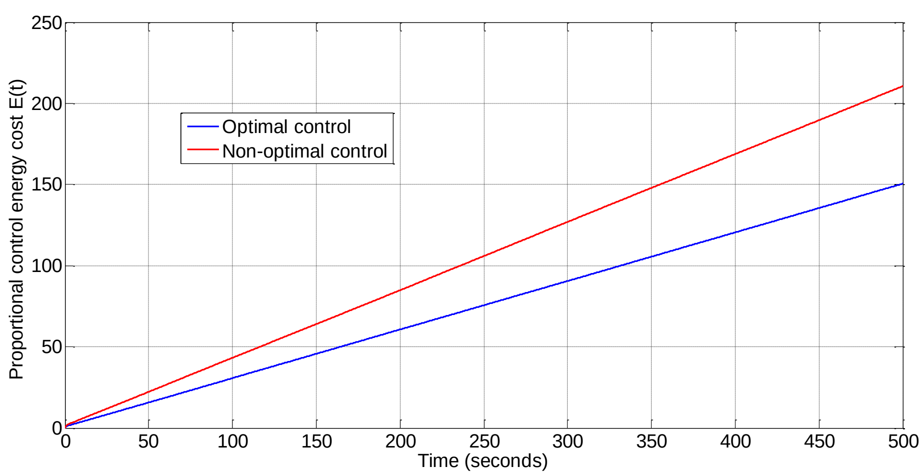


Figure (3.26) Proportional control energy costs following random disturbances on $\delta_1(t)$ and $\delta_2(t)$ for optimal and non-optimal position control

As **Fig. 3.26** illustrates, a substantial increased divergence expense related to non-optimal control with comparison with sustained cost of the optimal control, demonstrating effectiveness thereby.

Chapter Four

Results discussions

Hydraulic actuation systems are broadly employed in so many applications related to engineering and development, for instance robotics, earthmoving machines, motion control machines and air crafts component.

In aircraft applications; the hydraulic systems play a very important role. They can be found in most naval and civil aircraft and accomplish numerous tasks. Hydraulic system represent a key equipment in operating flying controller, activating the complex of wheels by which the aircraft lands and moves on the ground, powering the speediness brakes, operating fixed and rotary wing foldaway devices, and in supplementary systems. Power fluid systems are observed to have pros and cons when employed in the aerospace industry, in terms of pros, they are less weight, easy to install, simple to inspect and trouble shoot and least maintenance checkups ., In the meanwhile the efficiency of their function is very high with insignificant power loss. Nevertheless, the most disadvantage of the fluid power systems is fluid leakage through the inner and outer loops which in turn lead to get air and contaminants in the hydraulic loops so that system becomes not functioning properly.

Generally; Hydraulic systems age with time as they work under high oil pressure condition that creates wear and heat. One effect that can impact the operation of the system is that the hydraulic actuators are air leakage to the inside of the hydraulic circuits which makes the oil compressible.

As mentioned in chapter II, there were many researches and studies to implement several control techniques. The control strategy followed in this contribution considers the faulty conditions of the hydraulic system. Minimum efforts controller has considered the nonlinearity come from the unequal cylinder chamber volumes which intern locates the piston away from the middle if the cylinder due high working pressure, the followed technique has shown acceptable responses with non-linearity as per responses in **Fig. (3.24)** so that the HAS can work at any piston location with controlled and stable responses

The feedback controller with least efforts concept employed in this study which was not used before in the HSAs, gives least energy dissipation. According to **Fig. (3.26),** the least efforts controller consumes less energy than the non- optimal controller. With less energy consumption, the tear and the wear factors will be reduced significantly and can raise the reliability and the life cycle of the HASs. Consequently; using different values of gains in the forward and feedback paths makes the energy consumption worse and, wear out the system more.

On the other hand the least effort controller has also improved the response of the system by improving the damping condition and by reducing the rise time of the load position response, **Fig. (3.21) and Fig. (3.22)** illustrate the comparison between the least effort (optimal control) controller and the non-optimal controller and show the difference in the speed of the output responses to the favor of the least efforts controller.

Other advantages of the least efforts controller are the deterministic step disturbance when an external disturbance and random noise are applied on the controlled outputs. For aircraft application, when for example the air craft flies across the Atlantic, it will be disturbed by ambient changes coming from wind, dusting and pressure changes so the system rejects these disturbances and the ailerons are always in the proper position so that the stability of the aircraft is always maintained. **Fig. (3.23)** shows the response when step disturbance is applied on the ailerons, the drawing illustrates the successful rejection of the disturbance as it returns back the aileron to the desired location. Moreover when the random noise is applied on the hydraulic actuator generating amplitude of small values; these are the amount of energy being absorbed by the control system (absolute amount of control energy) in order to suppress the noise and to return the system to quiescence with the least energy consumed.

Chapter Five

Conclusion

The progress of the hydraulic actuation system was investigated in terms of structure, design and control. The hydraulic servo system was considered as a device employing a hydraulic cylinder to move the load. The control techniques of the hydraulic actuator system is based on the understanding of an output (controlled) and desired (reference) position are measured and compared in order to produce an error signal which is a measure of the discrepancy between the input and output signals. The actuating signal is applied to the process so as to influence the output signal in a manner which tends to reduce the error.

The marriage between electronics and hydraulic power led to many powerful and precise control systems. In this regard fluid power engineering was reviewed with the classes of control valves that use electronic valve controllers.

In this contribution the hydraulic actuator represents a single input multiple output system and consists of a servo valve (spool valve) and a torque motor with the role of controlling the oil flow to a hydraulic cylinder to build the oil pressure inside the cylinder chamber, the piston inside the cylinder moves the load by the influence of the oil pressure. A needle valve which controls the oil leakage across the piston acts as a damping on the output response. The whole hydraulic actuator was analyzed and the mathematical model of the system was derived based on linearization approximating the process with some important assumptions allowing a transfer function of the single input multiple outputs system to be derived. The PQR matrix technique was used to obtain the input-output system description.

As unequal cylinder chamber volumes creates non linearity; the problem statement of the study which is “how far can the responses be valid when the hydraulic cylinder piston is not located at the middle of the cylinder” was discussed and investigated.

The control technique employed for hydraulic actuator system was the optimization of pole assignment with least energy consumption. In formulating a theory enabling single input multiple output regulation, the controller parameters were calculated achieving the least energy consumption. The control theory was employed and root locus methods were applied to attain the transient response to design a controller with minimum dissipated efforts.

The simulation results have confirmed the following outcomes

1. Least effort control improved the response of the system by improving the transient and steady state characteristics. i.e. damping condition ,reduced rise time of the load position and the leakage flow response
2. Minimum least effort technique has shown good responses with non-linearity so that the hydraulic actuator system can work at any piston location with controlled and stable responses
3. The rule of employing the minimized efforts produces from the aims of performing minimum control components operations, reducing heat, maintenance and wear expenses. Furthermore, in various industries when a control system is designed the availability of energy is so limited, beside this, the whole control system components which consume power are labeled with power rate which has to be not surpassed. The basis for the use of minimum control creates from objectives of achieving least actuator activity, minimizing the generation of heat, wear and maintenance costs. In addition, in many applications, the energy available for

regulation is strictly limited and all of the actuators have, in any case, prescribed power ratings which should not be exceeded (Whalley and Ebrahimi 2006).

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Appendix

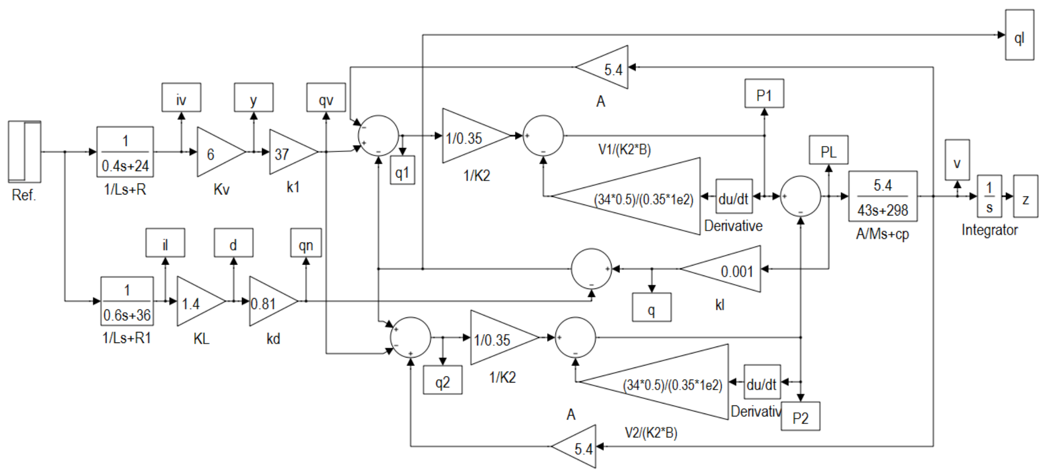

Figure A.1 Simulation of servomechanism in open loop system

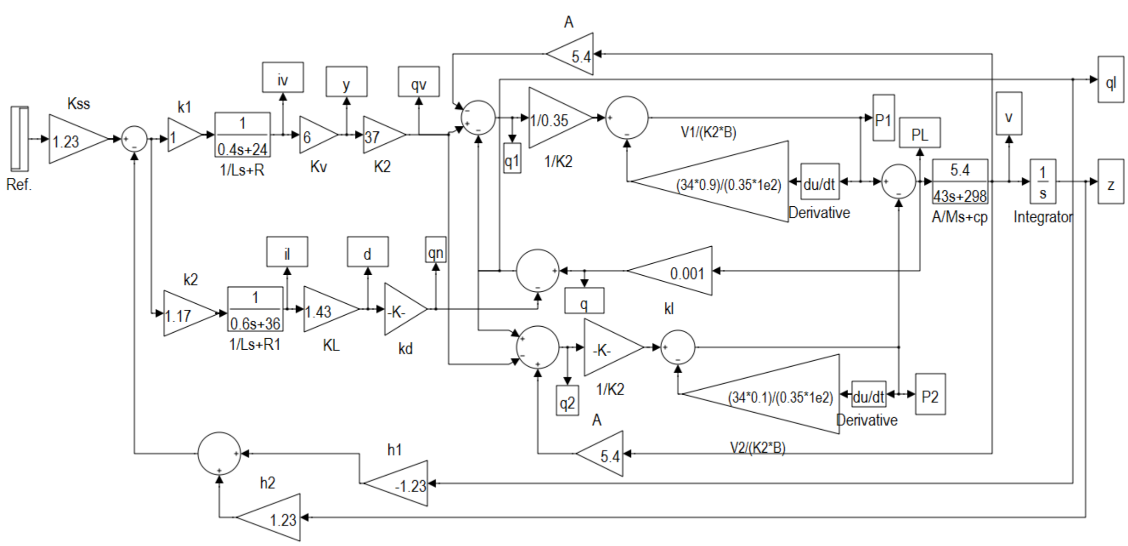

Figure A.2 Simulation of closed loop system with least effort controller

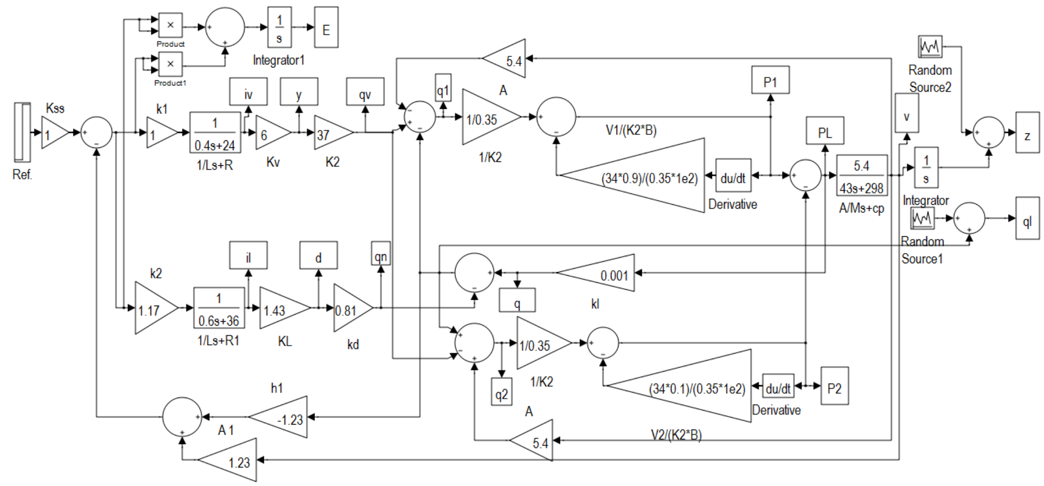

Figure A.3 Simulation of closed control system with less energy dissipation

Research methodology m.file :

```
%this program is formulated to derive the transfer function%
%of the servomechanism in open loop mode%
clear all
clc
syms s k1 k2 KL Kv M A tauv taul cp beta V1 V2 kL n m K1 K2;
Kv=6;k1=37;k2=0.35;
beta=1e2;
Rv=24;Lv=0.4;tauv=Lv/Rv;
Rl=36;Ll=0.6;taul=Ll/Rl;
kL=0.001;KL=1.43;kd=0.81;
M=43;A=5.4;V1=17;V2=17;cp=298;
%the output time constant is M/cp%
a=[-1/tauv 0 0 0 0;beta/V1 (beta/V1)*(-k2-kL) beta*kL/V1 beta/V1 -
A*beta/V1;-beta/V2 beta*kL/V2 (beta/V2)*(-k2-kL) -beta/V2 beta*A/V2;0 0
0 -1/taul 0;0 A/M -A/M 0 -cp/M];
C=[0 kL -kL -1 0;0 0 0 0 1];
b=[(k1*Kv)/(Lv) 0;0 0;0 0;0 (kd*KL)/(Ll);0 0];
I=[1 0 0 0 0;0 1 0 0 0;0 0 1 0 0;0 0 0 1 0;0 0 0 0 1];
A=(s*I)-a;
C11=det([A(2,2) A(2,3) A(2,4) A(2,5);A(3,2) A(3,3) A(3,4) A(3,5);A(4,2)
A(4,3) A(4,4) A(4,5);A(5,2) A(5,3) A(5,4) A(5,5)]);
C12=-1*det([A(2,1) A(2,3) A(2,4) A(2,5);A(3,1) A(3,3) A(3,4)
A(3,5);A(4,1) A(4,3) A(4,4) A(4,5);A(5,1) A(5,3) A(5,4) A(5,5)]);
C13=det([A(2,1) A(2,2) A(2,4) A(2,5);A(3,1) A(3,2) A(3,4) A(3,5);A(4,1)
A(4,2) A(4,4) A(4,5);A(5,1) A(5,2) A(5,4) A(5,5)]);
C14=-1*det([A(2,1) A(2,2) A(2,3) A(2,5);A(3,1) A(3,2) A(3,3)
A(3,5);A(4,1) A(4,2) A(4,3) A(4,5);A(5,1) A(5,2) A(5,3) A(5,5)]);
C15=det([A(2,1) A(2,2) A(2,3) A(2,4);A(3,1) A(3,2) A(3,3) A(3,4);A(4,1)
A(4,2) A(4,3) A(4,4);A(5,1) A(5,2) A(5,3) A(5,4)]);
C21=-1*det([A(1,2) A(1,3) A(1,4) A(1,5);A(3,2) A(3,3) A(3,4)
A(3,5);A(4,2) A(4,3) A(4,4) A(4,5);A(5,2) A(5,3) A(5,4) A(5,5)]);
```

```
C22=det([A(1,1) A(1,3) A(1,4) A(1,5);A(3,1) A(3,3) A(3,4) A(3,5);A(4,1)
A(4,3) A(4,4) A(4,5);A(5,1) A(5,3) A(5,4) A(5,5)]);
C23=-1*det([A(1,1) A(1,2) A(1,4) A(1,5);A(3,1) A(3,2) A(3,4)
A(3,5);A(4,1) A(4,2) A(4,4) A(4,5);A(5,1) A(5,2) A(5,4) A(5,5)]);
C24=det([A(1,1) A(1,2) A(1,3) A(1,5);A(3,1) A(3,2) A(3,3) A(3,5);A(4,1)
A(4,2) A(4,3) A(4,5);A(5,1) A(5,2) A(5,3) A(5,5)]);
C25=-1*det([A(1,1) A(1,2) A(1,3) A(1,4);A(3,1) A(3,2) A(3,3)
A(3,4);A(4,1) A(4,2) A(4,3) A(4,4);A(5,1) A(5,2) A(5,3) A(5,4)]);
C31=det([A(1,2) A(1,3) A(1,4) A(1,5);A(2,2) A(2,3) A(2,4) A(2,5);A(4,2)
A(4,3) A(4,4) A(4,5);A(5,2) A(5,3) A(5,4) A(5,5)]);
C32=-1*det([A(1,1) A(1,3) A(1,4) A(1,5);A(2,1) A(2,3) A(2,4)
A(2,5);A(4,1) A(4,3) A(4,4) A(4,5);A(5,1) A(5,3) A(5,4) A(5,5)]);
C33=det([A(1,1) A(1,2) A(1,4) A(1,5);A(2,1) A(2,2) A(2,4) A(2,5);A(4,1)
A(4,2) A(4,4) A(4,5);A(5,1) A(5,2) A(5,4) A(5,5)]);
C34=-1*det([A(1,1) A(1,2) A(1,3) A(1,5);A(2,1) A(2,2) A(2,3)
A(2,5);A(4,1) A(4,2) A(4,3) A(4,5);A(5,1) A(5,2) A(5,3) A(5,5)]);
C35=det([A(1,1) A(1,2) A(1,3) A(1,4);A(2,1) A(2,2) A(2,3) A(2,4);A(4,1)
A(4,2) A(4,3) A(4,4);A(5,1) A(5,2) A(5,3) A(5,4)]);
C41=-1*det([A(1,2) A(1,3) A(1,4) A(1,5);A(2,2) A(2,3) A(2,4)
A(2,5);A(3,2) A(3,3) A(3,4) A(3,5);A(5,2) A(5,3) A(5,4) A(5,5)]);
C42=det([A(1,1) A(1,3) A(1,4) A(1,5);A(2,1) A(2,3) A(2,4) A(2,5);A(3,1)
A(3,3) A(3,4) A(3,5);A(5,1) A(5,3) A(5,4) A(5,5)]);
C43=-1*det([A(1,1) A(1,2) A(1,4) A(1,5);A(2,1) A(2,2) A(2,4)
A(2,5);A(3,1) A(3,2) A(3,4) A(3,5);A(5,1) A(5,2) A(5,4) A(5,5)]);
C44=det([A(1,1) A(1,2) A(1,3) A(1,5);A(2,1) A(2,2) A(2,3) A(2,5);A(3,1)
A(3,2) A(3,3) A(3,5);A(5,1) A(5,2) A(5,3) A(5,5)]);
C45=-1*det([A(1,1) A(1,2) A(1,3) A(1,4);A(2,1) A(2,2) A(2,3)
A(2,4);A(3,1) A(3,2) A(3,3) A(3,4);A(5,1) A(5,2) A(5,3) A(5,4)]);
C51=det([A(1,2) A(1,3) A(1,4) A(1,5);A(2,2) A(2,3) A(2,4) A(2,5);A(3,2)
A(3,3) A(3,4) A(3,5);A(4,2) A(4,3) A(4,4) A(4,5)]);
C52=-1*det([A(1,1) A(1,3) A(1,4) A(1,5);A(2,1) A(2,3) A(2,4)
A(2,5);A(3,1) A(3,3) A(3,4) A(3,5);A(4,1) A(4,3) A(4,4) A(4,5)]);
C53=det([A(1,1) A(1,2) A(1,4) A(1,5);A(2,1) A(2,2) A(2,4) A(2,5);A(3,1)
A(3,2) A(3,4) A(3,5);A(4,1) A(4,2) A(4,4) A(4,5)]);
C54=-1*det([A(1,1) A(1,2) A(1,3) A(1,5);A(2,1) A(2,2) A(2,3)
A(2,5);A(3,1) A(3,2) A(3,3) A(3,5);A(4,1) A(4,2) A(4,3) A(4,5)]);
C55=det([A(1,1) A(1,2) A(1,3) A(1,4);A(2,1) A(2,2) A(2,3) A(2,4);A(3,1)
A(3,2) A(3,3) A(3,4);A(4,1) A(4,2) A(4,3) A(4,4)]);
Adj=transpose([C11 C12 C13 C14 C15;C21 C22 C23 C24 C25;C31 C32 C33 C34
C35;C41 C42 C43 C44 C45;C51 C52 C53 C54 C55]);
g=C*Adj*b;
den=sym2poly(det(A));
g1=sym2poly(g(1,1));
g2=sym2poly(g(1,2));
g3=sym2poly(g(2,1));
g4=sym2poly(g(2,2));
G11=tf(g1,den)
G12=tf(g2,den)
G21=tf(g3,den)
G22=tf(g4,den)
format long;
PP=pole(G11);
den1=[den(1,1) den(1,2) den(1,3) den(1,4) den(1,5) den(1,6) 0];
%-------------------------------------------------------------------%
num1=[g2(1,1) g1(1,1)+g2(1,2) g1(1,2)+g2(1,3) g1(1,3)+g2(1,4)
g1(1,4)+g2(1,5) 0];
G1=tf(num1,den1);
%-------------------------------------------------------------------
num1red=[num1(1,5) num1(1,6)];
G1red=tf(num1red,den1);
```

```
%-------------------------------------------------------------------------%
num2=[g3(1,1)+g4(1,1) g3(1,2)+g4(1,2) g3(1,3)+g4(1,3)];
G2=tf(num2,den1);
%-------------------------------------------------------------------------%
num2red=[num2(1,2) num2(1,3)];
G2red=tf(num2red,den1);
%-------------------------------------------------------------------------%
POLES=pole(G1);
%-------------------------------------------------------------------------%
NUM=[1 2];
F=tf(NUM,den1);
rlocus(F);
%-------------------------------------------------------------------------%
syms n h1 h2 p1 p2;
p2=7.36e4;p1=p2*NUM(1,2);
a1=num1(1,6);b1=num1(1,5);a2=num2(1,3);b2=num2(1,2);
H=([n*b2 -n*a2;-b1 a1]/((n*a1*b2)-(n*a2*b1)))*([p1;p2]);
%-------------------------------------------------------------------------%
format long;
K1=1;K2=n*K1;
J=((H(1,1)+H(2,1))^2)*(1+n^(2));
[N,D]=numden(J);
NN=sym2poly(N);
DD=sym2poly(D);
G=tf(NN,DD);
n=-1000:0.01:1000;
num=[NN(1,1).*n.^4+NN(1,2).*n.^3+NN(1,3).*n.^2+NN(1,4).*n.^1+NN(1,5)];
den=[DD(1,1).*n.^2+DD(1,2).*n.^1+DD(1,3)];
X=num./den;
plot(n,X); grid;
[q,d]=polyder(NN,DD);
tf(q,d);
P=roots(q)
J1=polyval(NN,P(1,1))/polyval(DD,P(1,1))
J2=polyval(NN,P(2,1))/polyval(DD,P(2,1))
J3=polyval(NN,P(3,1))/polyval(DD,P(3,1))
J4=polyval(NN,P(4,1))/polyval(DD,P(4,1))
J5=polyval(NN,P(5,1))/polyval(DD,P(5,1))
%After simulating the above commands, J is minimum at P(3,1)%
n=P(3,1);
[j,i]=numden(H(2,1));
h2=j/(polyval(sym2poly(i),n))
h1=H(1,1)
```