PPU College Of Engineering and Technology

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Mechanical Engineering Department

Mechatronics Engineering Program

Bachelor Thesis

Graduation Project

Electro-Hydraulic Servo System

Project Team

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Electro-Hydraulic Servo System

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According to the directions of the project supervisor and by the agreement of all examination committee members, this project is presented to the department of Mechanical Engineering at College of Engineering and Technology, for partial fulfillment Bachelor of engineering degree requirements.

Supervisor Signature   Supervisor Signature

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Abstract

This project concerns of building an electro-hydraulic servo system model to be used in the industrial hydraulics lab to achieve the controlling of a hydraulic flow and pressure using a hydraulic servo valves, understanding the internal construction and the operational principles of servo hydraulic valves, understanding the mathematical model of a simple hydraulic system using hydraulic servo valve.

The mathematical model for the built system was constructed and the simulation for linear mathematical model was done, and showed that the instability of the system and large steady state error. So indeed the system need an external controller to be designed.

Dedication

To Our Beloved Palestine

To my parents and family.

To the souls that i love and can't see.

To all my friends.

To Palestine Polytechnic University.

To all my teachers.

May ALLAH Bless you.

And for you i dedicate this project

Acknowledgments

I could not forget my family, who stood beside me, with their support, love and care for my whole life; they were with me with their bodies and souls, and helped me to accomplish this project.

I would like to thank our amazing teachers at Palestine Polytechnic University, to whom i would carry my gratitude to my whole life. Special thanks for my supervisor Eng. Hussein Amro for his wise and amazing supervision, to Eng. Mohammad AL-Shareef for his special helping, and to Eng. Khaled Tamizi for his supervision.

Table of contents

List of Tables

Table of Figures

Introduction

1.1 Overview :

Although electrical drives have become increasingly popular for high-performance motion control, hydraulic servo-systems still find a wide variety of applications in present day industrial motion systems, for instance in machining plants, robotics, motion simulators and so on.

Owing to increased computer power and ongoing developments in control theory, expectations regarding modeling of non-linear dynamic behavior of hydraulic servo systems have increased. More detailed descriptions of dominant non-linear characteristics and relevant dynamics over wider frequency ranges have to be taken into account. The main non-linarites in hydraulic systems arise from the compressibility of the hydraulic fluid, the complex flow properties of the servo-valve and the friction in the hydraulic actuators. They depend on factors, which are difficult to measure or estimate online, such as viscosity and temperature. Therefore, conventional feedback control, which is easily tuned by hand, is only sufficient for some cases. Advanced control methods are necessary if high performance motion control is required.

1.2 Importance of the project :

Due to the need for the local industrial market to the use of modern technical means, and in view of the revolution that the world has witnessed in the field of servo Hydraulic Valves, this made us think about constructing a model for PPU students to be used in the industrial hydraulic Lab., that will help them in understanding how to control the rate of hydraulic flow and pressure of a hydraulic system through using servo hydraulic valve. Also the project aims to help the students to learn the operation principles of the servo hydraulic valves and to understand the mathematical model of a simple hydraulic system uses a servo valves.

1.3 Project Objectives:

The main purpose of the project is to build a hydraulic model uses servo hydraulic valve to control the flow and the pressure of a double acting cylinder moving a carriage, this model is to be used in the industrial hydraulic lab. In order to achieve the following learning objects:

- Understanding the internal construction and the operational principle of servo hydraulic valves.

- Understanding the mathematical model of a simple hydraulic system uses a hydraulic servo valve.
- Controlling a hydraulic flow and pressure using a hydraulic servo valve.

1.4 Literature review :

Electro hydraulic servomechanism is highly nonlinear with inherit parameter uncertainties. Various type of Sliding Mode Control based on Variable Structure Control has been proposed by researchers to control such a system. Some of the existing results will be briefly outlined in this section.

In [Rong-Fong Fung, 1997] a new technique of the variable structure controls applied to an electro hydraulic servo control system which is described by third order nonlinear equation with time-varying coefficient. A two-phase variable structure controller is designed to get the precise position control of an electro hydraulic servo system. A reaching law method is implements to the control procedure, which makefast response in the transient phase and good stability in the steady state of a nonlinear hydraulic servo system.

Sliding mode control with time-varying switching gain and a time-varying boundary level has been introduced in [L-C.Huang, 1996] to modify the traditional sliding mode control with fixed switching gain and constant width bounded layer to enhance the control performance of electro hydraulic position and different pressure. Under certain condition, for a time-varying switching gain and boundary layer, the combination of weighted position error and differential pressure can be asymptotically tracked even when the system is subject to parameters uncertainties. One of the important features is to use only one input to simultaneously control the angular position and torque of the electro hydraulic servo system in a different load condition. Byusing this technique, the high frequency and large amplitude of control input are attenuated.

1.5 Time Schedule

The project plan follows the time schedule, which includes the related tasks of study and system analysis.

Table (1.1): Tasks description

Task ID	Task Description
T4.1	Selecting the project
T4.2	Collecting information, literature review and related theory
T4.3	System analysis
T4.4	System design and simulation

Table (1.2) Timing Table

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
T4.1																
T4.2																
T4.3																
T4.4																

1.6 Outline of the chapters

This graduation project contains five chapters. Chapter 2 deals with the construction of servo hydraulic valve, operation, and control system. The basic structure, description of the components, characteristics and electrical measurements of servo hydraulic valve.

Chapter 3 deals with the mathematical modeling of the electro hydraulic servo valve control system, and present the controller design. And shows some of the simulation results using Matlab/Simulink.

Chapter 4 shows the construction of electro-hydraulic servo valve system, which contains principle of the system operation; explain the work of all its components, and method of work.

Chapter five shows the description of tuning PID controller experimentally , the experimental results ,and the simulation result for the Electro-Hydraulic servo system that was constructed .

Servo Hydraulic Valve Operation, Construction, and Control.

2.1 Introduction

"Servo hydraulics" is an established term. Nevertheless, opinions still vary as to its true meaning. For example, servo hydraulics may be expressed as 'closed loop electro-hydraulic control". This definition includes all closed loop control applications involving hydraulic devices. Operating in closed loop control means that operation is constantly monitored by means of measurement and deviations from required operation are automatically corrected. Servo valves were originally designed to be used in the aviation industry (or precise control of a variety of aircraft by means of small electrical input signals. Electrical or electronic control was changed to electro-hydraulic open loop and closed loop control due to high flight speeds and hence high applications. It was thus possible to offer the devices at prices acceptable to industry. Servo hydraulics comes under the broad heading of system technology. This means that all elements within the closed loop control must be considered. [1]

2.2 Basic structure of hydraulic servo-system

A hydraulic servo system is an arrangement of individual components, interconnected to provide a desired form of hydraulic transfer. The basic structure of hydraulic systems is shown in figure 2.1 and consists of

- Hydraulic power supply,
- Control elements (valves, sensors, etc.),
- Actuating elements (cylinder and/or motors), and
- Other elements (pipelines, measuring devices, etc.).

The basic concept of hydraulic system (a standard valve-controlled hydraulic system is used here as an example) is briefly described in the following:

- The pump converts the available (mechanical) power from the prime mover (electrical or diesel motor) to hydraulic power at the actuator.
- Valves are used to control the direction of pump flow, the level of power produced, and the amount of fluid and pressure to the actuator. A linear actuator (cylinder) or a rotary actuator (motor) converts the hydraulic power to usable mechanical power output at the point required.
- The medium, which is a liquid, provides direct power transmission, a lubrication of the hydraulic system components, a valves sealing, and a hydraulic system cooling.
- Connectors, which link the various system components, direct the power of the fluid under pressure, and return fluid-flow to tank (reservoir).

- Finally, fluid storage and conditioning equipment ensure sufficient quality and quantity, and cooling of the fluid.

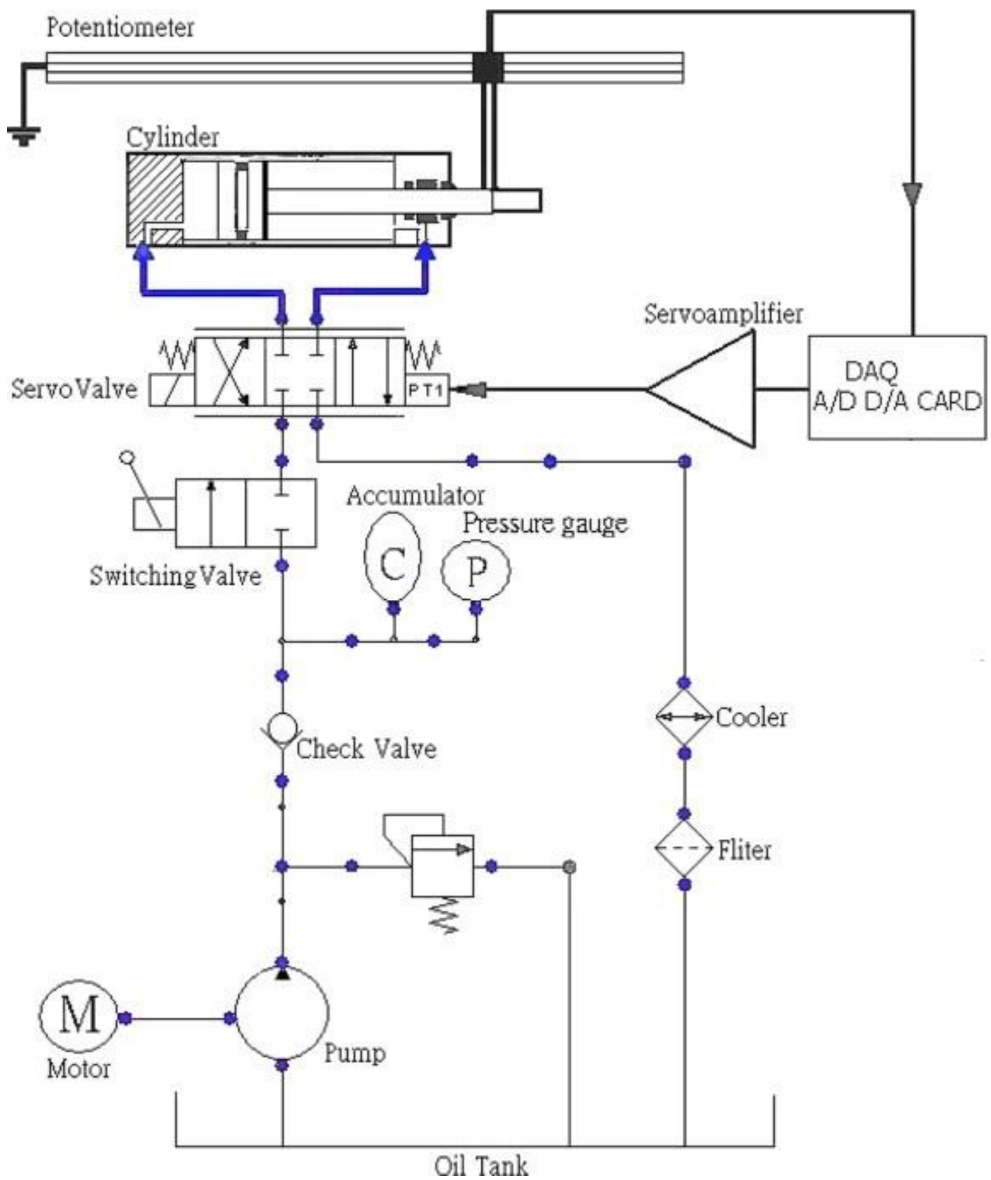


Figure 2.1: Basic structure of electro-hydraulic system [3]

2.3 Description of the components

This section provides a detailed description of various hydraulic components (pump, valve, actuators) and an explanation of their operation with an impartial evaluation divorced from the design and development philosophy of individual manufacturers.

2.3.1 Actuating elements

1- Hydraulic motor: converts hydraulic energy into mechanical energy. It uses a hydraulic pressure and flow to generate torque and rotation. it can use hydraulic motors for many applications, such as winches, crane drives, self-driven cranes, excavators, mixer and agitator drives, roll mills, etc.[3]

Figure 2.2: Hydraulic motor.

2- Hydraulic cylinders: convert hydrostatic energy into mechanical energy. They are considered as motors or actuators capable of producing work. However, hydraulic cylinders differ from hydraulic motors as they carry out a linear (translator) movement instead of a rotary movement. Thus, the cylinders are also referred to as linear motors.

Figure 2.3: Hydraulic cylinders.

2.3.2 Control elements

Basically there are four main categories of valves in hydraulics:

1- Pressure-relief valves: limit the maximum permissible system pressure, and divert some or all of the pumps flow to the tank when the pressure setting of the relief valve is reached. [3]

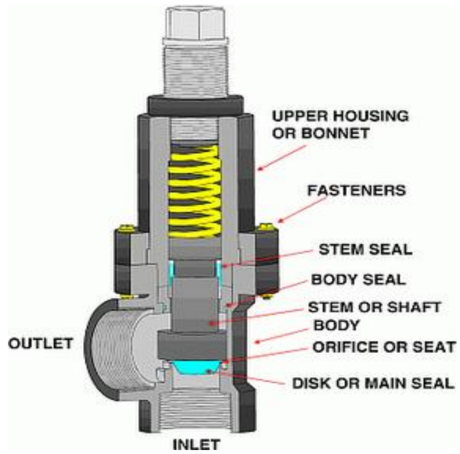


Figure 2.4: Pressure-relief valve.

2- Check valves are a very special type of directional control valve, as they only permit fluid flow in one direction while blocking flow in the reverse direction.

They can be divided into unloaded or spring-loaded check valves, and check valves for logic operations (OR, AND).[3]

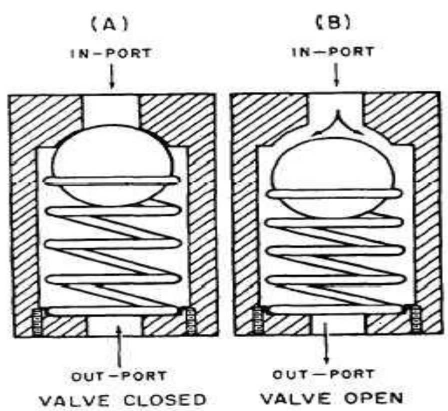


Figure 2.5: Automatic check valve.

3- Flow valves are used to control the rate of flow from one part of the hydraulic system to another, i.e., they limit the maximum speed of cylinders and motors, limit the maximum power available to sub-circuits by controlling the flow of them, or proportionally divide or regulate the pump flow to various branches of the circuit.[3]

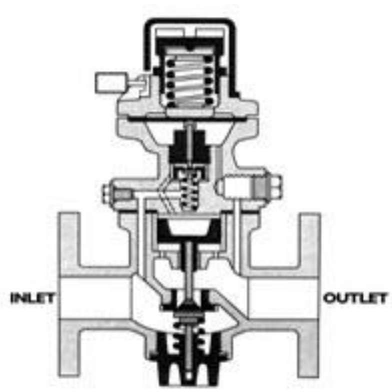


Figure 2.6: Regulator flow valve.

4- Directional valves are used as multi-polar switches. Before the advent of servo and proportional valves they were used to control the direction of actuator motion, selected alternative control circuits, and performed logic control functions. Nowadays, however, proportionally variable controls allow infinitely adaptable and quickly variable setting of actuators with regard to force, speed and stroke position.[3]

Figure 2.7: DCV 4way/3 position.

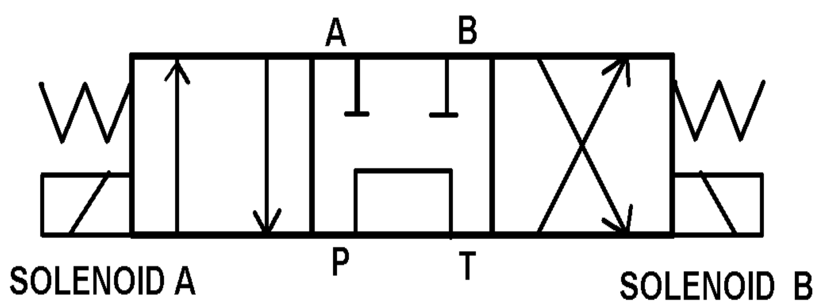


Figure 2.8: DCV 4way/3 position symbol.

2.3.3 Power Pumps

1- Hydraulic accumulator stores fluid under pressure and can serve a number of functions within a hydraulic system. It is an item that can provide years of trouble-free service.

Figure 2.9: Hydraulic accumulator.

Accumulators can take a specific amount of fluid under pressure and store it. The fluid is then released when its required to perform a specific task in the hydraulic system. Accumulators can provide several functions, such as:

•Energy storage

•Compensation of leakage oil

•Compensation of temperature fluctuations

•Emergency operation

•Cushioning of pressure shocks which may occur at sudden switching of the valves

•Dampening vibrations

•Swell compensator

2- Hydraulic pumps are sources of power for many dynamic machines. Hydraulic pumps are capable of pushing large amounts of oil through hydraulic cylinders or hydraulic motors. In this fashion, the pump converts the mechanical energy of the drive (i.e. torque, speed) into hydrostatic energy (i.e. flow, pressure).

Hydraulic pumps operate according to the displacement principle. This involves the existence of mechanically sealed chambers in the pump. Through these chambers, fluid is transported from the inlet (suction port) of the pump to the outlet (pressure port). The sealed chambers ensure that there is no direct connection between the two ports of the pump. As a result, these pumps are very suitable to operate at high system pressures and are ideal for hydraulics.[1]

Figure 2.10: hydraulic pump.

3- hydraulic filter helps to remove the particles and clean the oil on a continuous basis. The performance for every hydraulic filter is measured by its contamination removal efficiency, i.e. high dirt-holding capacities. Almost every hydraulic system contains more than one hydraulic filter.

The suction filter provides protection to the hydraulic pump from particles larger than 10 microns. You should use a suction filter if there is any likelihood of pump damage due to larger particles or pieces of dirt. For example, this may occur when it is difficult to clean the tank or if several hydraulic systems use the same tank for oil supply.

Figure 2.11: hydraulic filters.

4- Hydraulic Oil Coolers are suitable for heat transfer fluids, lubricants and quenching oils.

Figure 2.12: hydraulic oil.

2.4 Why choosing a servo valves:

The reasons for choosing a servo valves are their ability to control a mechanical parameters such as:

- Displacement or angle of rotation
- velocity or rotary speed
- force or torque

Or hydraulic parameters such as:

- flow
- Pressure.

To be able to control the parameters mentioned above, a suitable measuring devices are necessary to determine actual parameter measurements, therefore, servo hydraulics does not simply refer to individual hydraulic components, but to the interaction between applied closed loop control, power transmission hydraulics and data processing electronics.

In order to understand and assess closed loop electro-hydraulic control and its power limits, it is necessary to examine.

- Closed loop control
- Electronics
- Hydraulics and
- Measurement technology. [1]

2.5 Characteristics and operations of Servo Valves

Servo valves can be broadly classified either as single-stage, two-stage or three-stage. Single-stage servo valves consist of a torque motor or a linear force motor directly attached for positioning of the spool. Because torque or force motors have limited power capability, this in turn limits the hydraulic power capacity of single-stage servo valves. In some applications the single-stage concept may also lead to stability problems. This is the case if the flow forces acting on the spool are close to the force produced by the electro-magnetic motor. Flow forces are proportional to the flow and the square root of the valve pressure drop, which gives a limitation in hydraulic power. A single-stage servo valve with a linear force motor is shown in Figure 1. The valve illustrated in the figure is a valve, which employs just one linear force motor (Proportional magnet) to move the spool either side of the central position. The electrical signal from a position transducer is then used for closed loop control of the spool position.

Figure 2.13: Single-stage servo valve with electrical position feedback.

One of the most common types of servo valve is the two stages one. Either a flapper nozzle pilot stage or a jet pipe stage can be used in conjunction with an electric torque motor to control the main spool in the valve as illustrated in Figure (2.14).

Figure 2.14: 2-stage directional servo valve.

- And the 3-stage directional servo valves basically consist of the 1st stage control and the 2nd stage in the form of a flow amplifier stage in the 3rd stage for open loop flow control of the main oil flow and an inductive positional transducer whose core is secured to the control spool of the 3rd stage as illustrated in Figure (2.15).

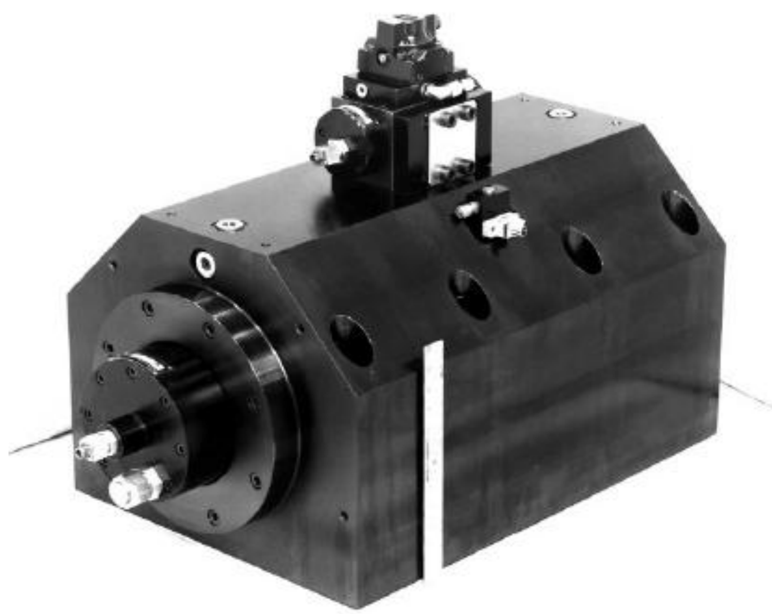


Figure 2.15: 3-stage directional servo valve.

In this project a two-stage servo valve that's called "Servo directional valve of 4-way design type 4WS.2E… that's shown in figure (2.16) will be used.

Figure 2.16: Rexroth servo valve type 4WSE2ED 10-51/30B11ET315K31EV.

2.5.1 Servo valve operation

Servo valves have been developed as industrial valves that comply with industry requirements for its reliability, interchangeability and easy servicing. They are of modular design. The term "servo" is used to describe a wide variety of functions. In general; this term refers to the function in which a small input signal produces a large output signal (amplifier).

The best known example is probably servo steering in a motor vehicle, where the steering wheel is moved with little effort to produce a large force on the wheels. The process is similar in servo hydraulics.

The servo valve in the form of an electrically controlled hydraulic amplifier is primarily used in closed loop control circuits, i.e. not only is an electrical input signal converted into a corresponding flow, but also deviations from preset speeds or positions are measured electrically and fed to the servo valve for correction.

2.5.1.1 Rexroth servo valve type 4WSE2ED 10-51/30B11ET315K31EV mechanical operation:

2-stage directional servo valve as shown in Figure 2.17 basically consist of:

- the 1ststage
- mechanical feedback (3) as a link between 1st and 2nd stages, and
- The2ndstage with interchangeable control sleeve (4) and control spool (5) which is coupled to mechanical feedback (3).

Control spool (5) is linked almost backlash-free to the torque motor (1) of the 1st stage by the mechanical feedback (3).The type of feedback used in this case depends on the torque balance at the torque motor (1) and feedback spring (3).This means, when a change in electrical input signal creates unequal torques, flapper plate (6) is first moved from the mid position between the control orifices. As a result, a pressure difference is produced which acts on both ends of the control spool Due to the pressure difference, the position of the control spool (5) changes As a result of this change, the feedback spring (3) bends until the flapper plate is pulled back to the center position to such an extent that the main spool stops moving and the torques are the same again. The spool stroke and hence flow, which are proportional to the input signal have therefore been reestablished. The two socket screws (8) (located left and right in the valve covers (9) may be used to move the position of the control sleeve (4) control land with respect to the control spool (5), in order to adjust the hydraulic null point. [1]

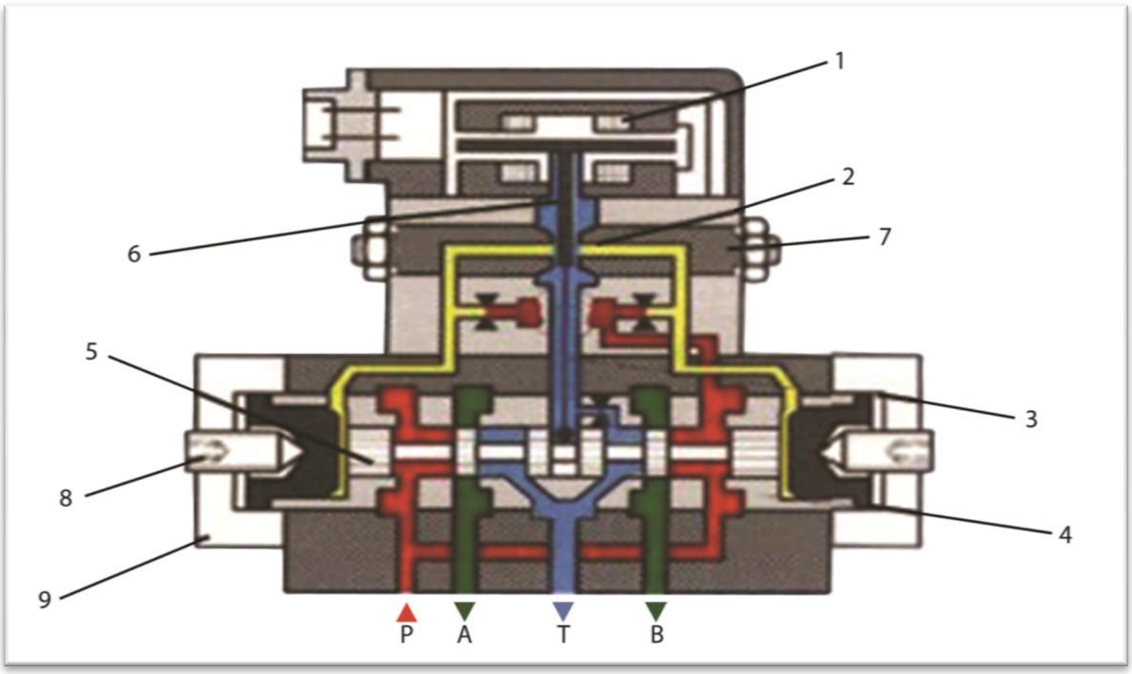


Figure 2.17: 2- Stage directional control valve with mechanical feedback. [1]

2.5.1.2 Rexroth servo valve type 4WSE2ED 10-51/30B11ET315K31EV Measurement and electrical operation

These 2-stage directional servo valves basically consist of:

- The 1st stage
- the 2nd stage with interchangeable control sleeve (3) and
- An inductive positional transducer (4) with its core (5) secured to the control spool (6).

The control spool (6) is coupled to the inductive positional transducer (4) by means of suitable electronics. A change in the position of the main spool (6). As well as a change in the command signal produce a differential voltage at the output when the core (5) is displaced within the coil of the positional transducer. The difference between command and feedback signals is measured by suitable electronic components and fed as a closed loop error to the first stage of the valve. This signal moves the flapper plate (7) from the mid-position between the two control orifices (8). As a result, a pressure difference is produced between the two control chambers (9) and (10). The control spool (6) with the attached core (5) of the inductive positional transducer (4) is shifted until the command signal is the same as the actual signal. When this is the case, the flapper plate returns to the center position. In closed loop control, control chambers (9) and (10) are pressure-balanced and the control spool is held in this controlled position. Due to the position of the control spool (6) with respect to the control sleeve (3) a control opening is produced to control flow; this is proportional to the command signal, in the same way that the spool stroke and flow are proportional to this signal. The valve frequency response is optimized by means of the electrical gain in the electronic control.[1]

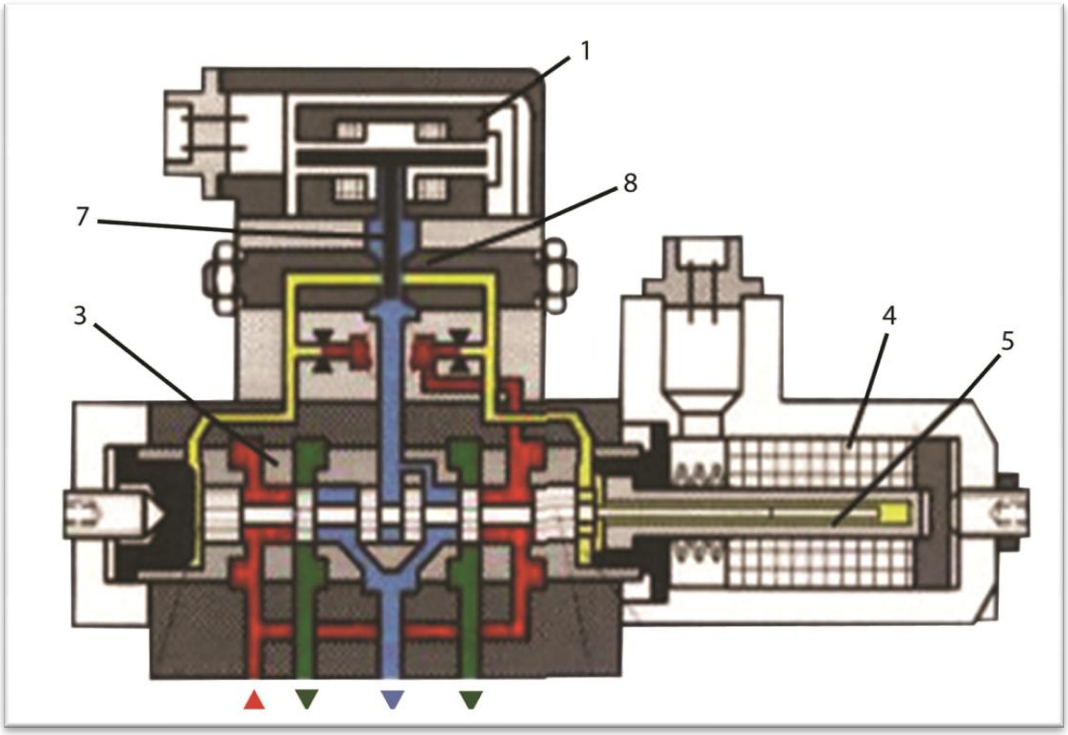


Figure (2.18): 2-stage directional servo valve with electrical feedback. [1]

2.5.2 Comparison between proportional valve and servo valve

The basic decision is whether to use servo valves or proportional valves. The main difference between them is how the spool is shifted. Proportional valves use an electric coil and magnet, like the voice coil of typical audio speaker, to directly move the spool. Servo valves use small torque motor to control hydraulic pressure, which in turn moves the spool (pilot-actuated).

The response of these two valve types differs because of the force available to shift the spool. Servo valves generally respond faster than proportional valves because of the ratio of hydraulic forces to the mass of the spool, although some proportional valves approach servo valve response time. Proportional valves must supply enough force to move the spool, the inline LVDT, and the solenoid core, as well as overcome spring centering forces.

The precise machining and small orifices associated with pilot-operated servo valves drive up the cost and make them more susceptible to contamination. In many applications, this has steered people away from servo valves and toward proportional valves. However, servo valves still have their place. For example, servo valves often work better on high-flow applications, where system pressure is available to move the spools and oppose the higher flow forces. In these types of applications, servo valves are a safer design choice and perform better because they have a faster and more linear response (and hence are easier to control).

In some cases, proportional valves will not have enough power to overcome the Bernoulli forces caused by high flows. In these cases, the valves will appear to lose control momentarily until the flow forces are reduced. While troubleshooting, there may be a tendency to fault the device controlling the valve instead of the valve itself. An oscilloscope or another diagnostic tool that can record control signals, spool positions, and actuator positions is available in these cases.

To solve this flow force problem, one can use a multiple-stage valve. Flow from a small pilot valve is used to control the spool position of the main spool.

Multi-stage valve are more expensive and can be much slower, as there are multiple stages causing phase delay. However, large valves require more force to quickly move the main spool than what an electric solenoid can provide alone. In these causes, the pilot valve enhances performance by directing oil pressure to move the main spool quickly. And servo can be used at a remote location to proportionally follow the angular position of a control knob. The connection between the two is not mechanical, but electrical or wireless.

Servos are commonly electrical or partially electronic in nature, using an electric motor as the primary means of creating mechanical force, though other types that use hydraulics, pneumatics or magnetic principles are available. A proportioning valve is a valve that relies on the laws of fluid pressure to distribute input forces to one or more output lines. A proportioning valve can increase or decrease forces for each output, depending on the cross-sectional surface areas of those output lines. [6]

CHAPTER ONE 1

Introduction

CHAPTER TWO 2

Servo Hydraulic Valve Operation, Construction, and Control

CHAPTER THREE **3**

Mathematical Model And Simulation results

CHAPTER FOUR 4

Electro-Hydraulic Servo System Construction

Mathematical Model And

Simulation results

3.1 Introduction

In this chapter the mathematical model of the project is discussed, where the position tracking performance of an electro-hydraulic servo (EHS) system using proportional-integral-derivative (PID), description of flow control valve, and how the valves responding to change in electrical input, and the effects of coils in torque motor, and represent servo valve spool dynamics through wider frequency range. The relationship between valve control flow and actuator champers pressure, also describe the piston motion and the total frictional force depend on piston velocity.

Simulation results show that the model has a bad tracking performance without using PID controller.

3.2 Modeling the EHS System

Dynamics equation of EHS system consists of servo valve and hydraulic actuator as illustrated in Figure (3.5).The hydraulic actuator motion is controlled by modulating the hydraulic oil flow from the cylinder chambers using a servo valve. The mass is attached with a spring and a damper that generates the counter force against the actuator.

3.2.1 Flow Control Servo Valve

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

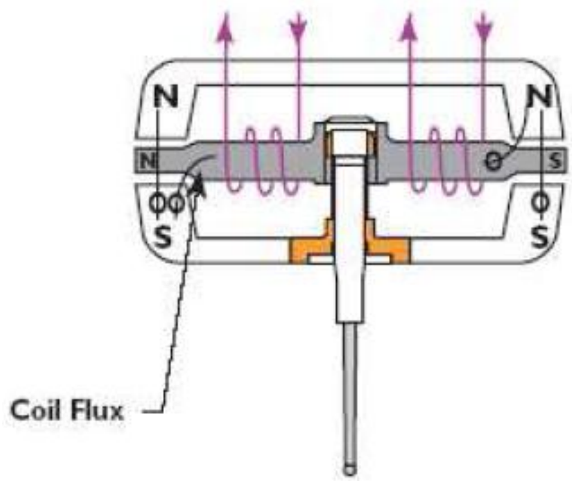


Figure (3.1): Valve Torque Motor Assembly. [4]

The motor consists of an armature mounted on a thin-walled sleeve pivot and suspended in the air gap of a magnetic field produced by a pair of permanent magnets. When current is made to flow in the two armature coils, the armature ends become polarized and are attracted to one magnet pole piece and repelled by the other. This sets up a torque on the flapper assembly, which rotates about the fixture sleeve and changes the flow balance through a pair of opposing nozzles, shown in Figure (3.2). The resulting change in throttle flow alters the differential pressure between the two ends of the spool, which begins to move inside the valve sleeve.

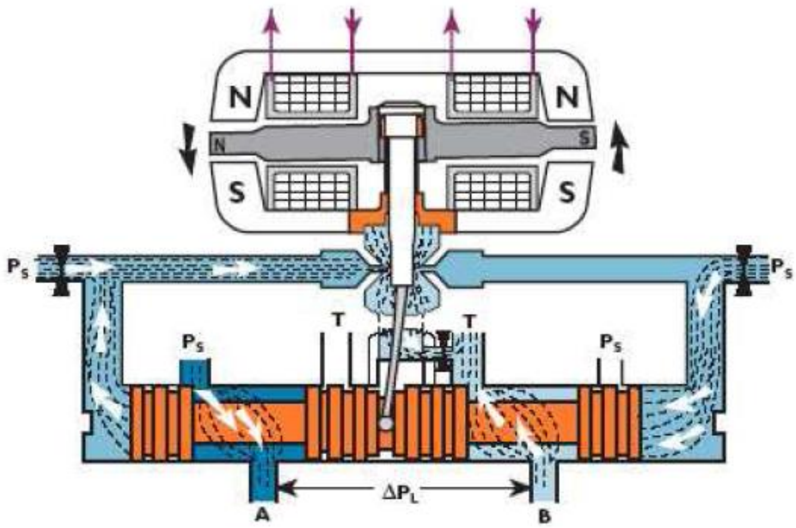


Figure (3.2): Valves Responding to Change in Electrical Input. [4]

Lateral movement of the spool forces the ball end of a feedback spring to one side and sets up a restoring torque on the armature/flapper assembly. When the feedback torque on the flapper spring becomes equal to the magnetic forces on the armature the system reaches an equilibrium state, with the armature and flapper centered and the spool stationary but deflected to one side. The offset position of the spool opens flow paths between the pressure and tank ports (P_s and T), and the two control ports (A and B) in the figure (3.2), allowing oil to flow to and from the actuator. [4]

3.2.1.1 Torque Motor

In the electronic design, the electrical characteristics of the servo-valve torque motor may be modeled as a series L-R circuit. The transfer function of a series L-R circuit is: [4]

$$\frac{I(S)}{V(S)} = \frac{1}{SL_f + R_f} \qquad (1)$$

Where The relationship between the time-varying voltage $v(t)$ across an inductor with inductance L and the time-varying current $i(t)$ passing through it is described by the differential equation:[4]

$$v(t) = L_f \frac{di(t)}{dt} \tag{2}$$

And

$$v(t) = i(t)R \tag{3}$$

$$v(s) = (sL_f + R_f)I(s) \tag{4}$$

L_f is the inductance of the motor coil, and R_f the combined resistance of the motor coil and the current sense resistor of the servo amplifier as shown in the figure below:

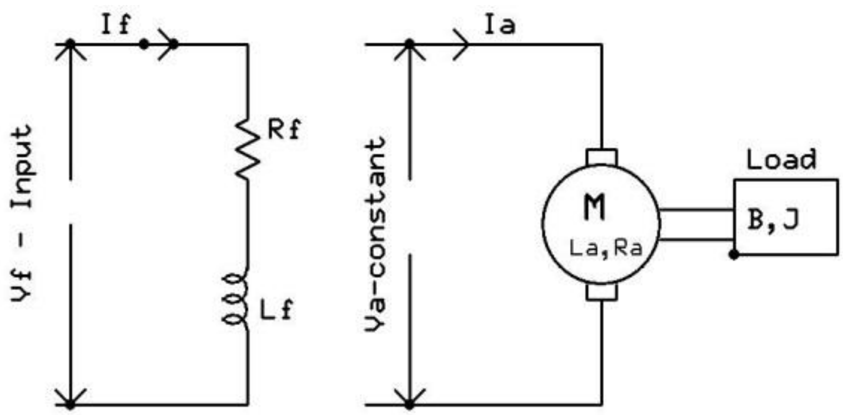


Figure (3.3): series L-R circuit of the servo-valve torque motor

Values of inductance and resistance for series and parallel winding configurations of the motor are published in the manufacturer's data sheet. The lateral force on the valve spool is proportional to torque motor current, but oil flow rate at the control ports also depends upon the pressure drop across the load.

3.2.1.2 Valve Spool Dynamics

In order to represent servo valve dynamics through a wider frequency range, a second order transfer function is used as approximation of the valve dynamics. The relation between the servo valve spool position x_v and the input voltage v can be written as:[5]

$$\frac{x_v}{v} = \frac{K_v}{s^2/\omega_v{}^2 + 2\xi_v S/\omega_v + 1} \tag{5}$$

Where k_v is the valve gain, ξ_v is the damping ratio of the servo valve and ω_v is the natural frequency of the servo valve.

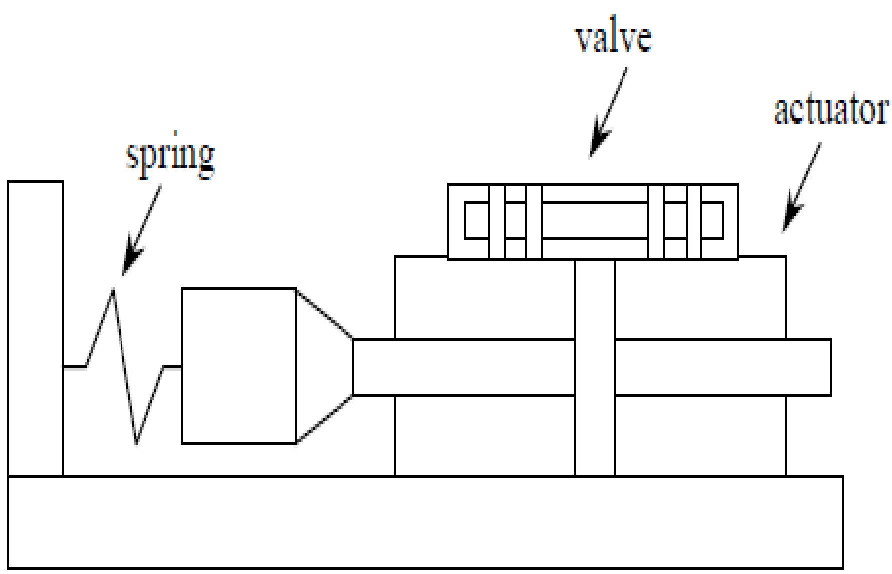


Figure (3.4): A schematic diagram of the electro-hydraulic system. [5]

3.2.2 Linear Actuator

3.2.2.1 Cylinder Chamber Pressure

The servo valve control the flows Q in each chamber in the actuator can be models from the orifice equations relates the spool valve position x_v , pressure difference P_L and servo valve gain K . For the ideal orifice equation:[2]

$$Q = K x_v \sqrt{\Delta P_L} \tag{6}$$

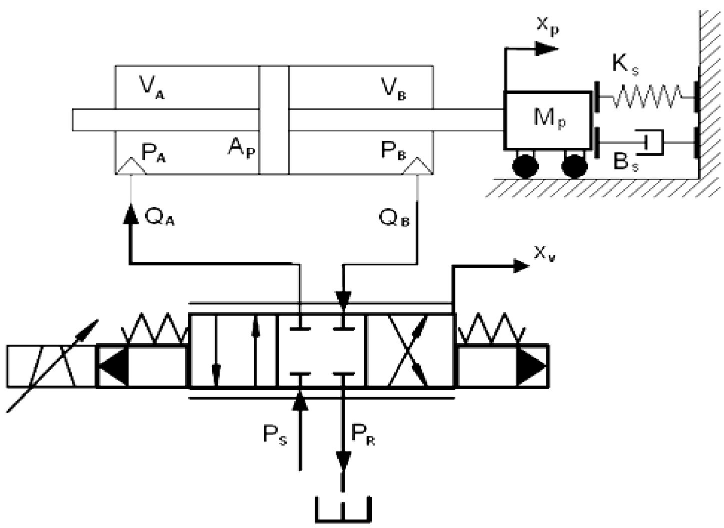


Figure (3.5): Electro-hydraulic servo system configuration. [2]

Neglecting leakage in the valve, the flow into sides 1 and 2 of the cylinder is:[2]

$$Q_A = \begin{cases} K_A x_v \sqrt{P_S - P_A} & x_v \geq 0 \\ K_A x_v \sqrt{P_A - P_R} & x_v < 0 \end{cases} \tag{7}$$

$$Q_B = \begin{cases} -K_B x_v \sqrt{P_B - P_R} & x_v \geq 0 \\ -K_B x_v \sqrt{P_S - P_B} & x_v < 0 \end{cases} \tag{8}$$

Where the coefficient gain is assumed to be $K = K_A = K_B$ for a symmetrical valve, P_R is the return pressure, P_S is a pumps pressure, and P_B , P_A are the pressure in each chamber.

The hydraulic actuator is modeled from the dynamics of volume of each chamber as follows:[2]

$$V_A = V_{line} + A_P(x_s + x_p) \tag{9}$$

$$V_B = V_{line} + A_P(x_s - x_p) \tag{10}$$

Where V_{line} is the volume of the pipeline and the zero position is located at the center of the cylinder, x_s is the total actuator displacement, x_p is the pistons displacement, and A_P is the pistons area.

3.2.2.2 Piston Motion

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

$$F_P = (P_A - P_B)A_P \tag{11}$$

An equation of forces for piston motion can now be established by applying Newton's second law. For the purposes of this analysis, it will be assumed that the piston delivers a force to a linear spring load with stiffness K_s, which will allow us to investigate the load capacity of the actuator later. The effects of friction F_f between the piston and the oil seals.

The total frictional force depends on piston velocity, driving force F_P, oil temperature and possibly piston position. One method of modeling friction is as a function of velocity, in which the total frictional force is divided into static friction, Coulomb friction and viscous friction. Assuming that viscous and Coulomb friction components dominate, frictional force F_f can be modeled as:[2]

$$F_f = \frac{dx}{dt}\alpha_2 + sign\left(\frac{dx}{dt}\right)\alpha_0 \tag{12}$$

Where viscous and Coulomb friction coefficients are denoted by α_2 and α_0 respectively. Frictional effects are notoriously difficult to measure and accurate values of these coefficients are unlikely to be known, but order of magnitude estimates can sometimes be made from relatively simple empirical tests. One test which can yield useful information is to subject the system to a low frequency, low amplitude sinusoidal Input signal, and plot the output displacement over one or two cycles. A low friction system should reproduce the input signal, but the presence of friction will tend to flatten the tops of the sine wave as the velocity falls to a level blow that necessary to overcome any inherent Coulomb friction. In a first analysis, leakage effects in the actuator are sometimes neglected, however this is an important factor which can have a significant damping influence on actuator response. Leakage occurs at the oil seals across the annulus between the two chambers and at each end cap, and is roughly proportional to the pressure difference across of the seal.

The total force generated from the actuator can be expressed in (10) from the overall dynamics equation of spring, damper, moving mass and friction. [2]

$$F_p = A_p(P_A - P_B) = M_p \frac{d^2 x_p}{dt^2} + B_s \frac{dx_p}{dt} + K_s X_p + F_f \qquad (13)$$

3.2.3 Hydraulic Power Supply

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

$$P_s = \frac{\beta}{V} \int \left(Q_{pump} - Q_L\right) dt \qquad (14)$$

where $Q_L = \mathrm{Q_A} - \mathrm{Q_B}$

This equation takes into account the load flow Q_L drawn from the supply by the servo valve, and accurately models the case of a high actuator slew rate resulting in a load flow which exceeds the flow capacity of the pump. In such cases the supply pressure P_s falls, leading to a corresponding reduction in control flow, and loss of performance. [4]

3.2.4 The systems model and response simulation

In this section the mathematical modeling of EHS system is discussed, where the system consists of electronic drive, hydraulic actuator, servo valve, and piston transducer. The mathematical model behavior of servo valve can be developed from the relationship between the displacement x_p and the input voltage v.

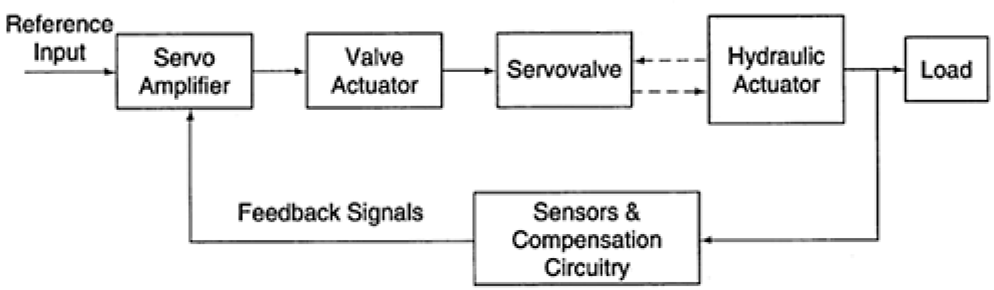


Figure (3.6): schematic diagram of EHS system.

3.2.4.1 Equivalent Transfer Function of EHS system

The transfer function of servo amplifier is the relationship between the input voltage and the output as it mentioned in torque motors section is bellow:

$$\frac{\mathrm{I(S)}}{\mathrm{V(S)}} = \frac{1}{\mathrm{S}L_f + \mathrm{R}_f} \tag{15}$$

Because of L_f <<<< R_f , so it is considered that the transfer function of servo amplifier is a constant gain.

The transfer function of a servo valve is the relation between the servo valve spool position x_v and the input voltage v can be written as it mentioned in section (3.2.1.2):

$$\frac{x_v}{v} = \frac{K_v}{s^2/\omega_v{}^2 + 2\xi_v S/\omega_v + 1} \tag{16}$$

From section (3.2.2.2), getting the total force generated from the actuator from the overall dynamics equation of spring, damper, moving mass and friction.

$$F_p = A_p(P_A - P_B) = M_p \frac{d^2 x_p}{dt^2} + B_s \frac{dx_p}{dt} + K_s X_p + F_f \tag{17}$$

In order to represent the actuator dynamics from the previous equation through a wider frequency range, a second order transfer function is obtained. The transfer function of actuator is the relation between the piston position x_p and the input force F_p can be written as

$$\frac{X_p}{F_p} = \frac{1}{M_P s^2 + B_S s + K_S} \tag{18}$$

The transfer function of the feedback transducer signal is the relation between the piston displacement and the input voltage v that was found as a constant gain by comparison the ratio between the two parameters.

The equivalent open loop transfer function of the whole system without controller is:

$$T(s) = \frac{K}{9s^4 + 2010s^3 + 2130s^2 + 2193s + 200} \tag{19}$$

As it shown in the figure (3.7) below:

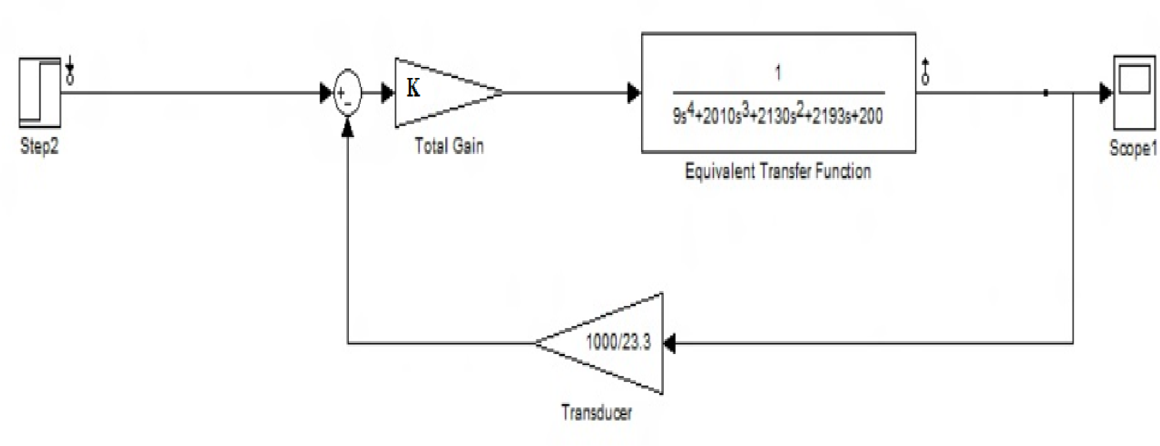


Figure (3.7): closed loop block diagram for EHS system without controller.

3.2.4.2 System's response

The exact total gain of the forward transfer function can be found practically , Appling Routh-Huriwtz criterion will give the range of this gain for stability which is 0 < K < 50 , at a specific gain of K=10 the response of the uncontrolled system is shown in the figure (3.8) below :

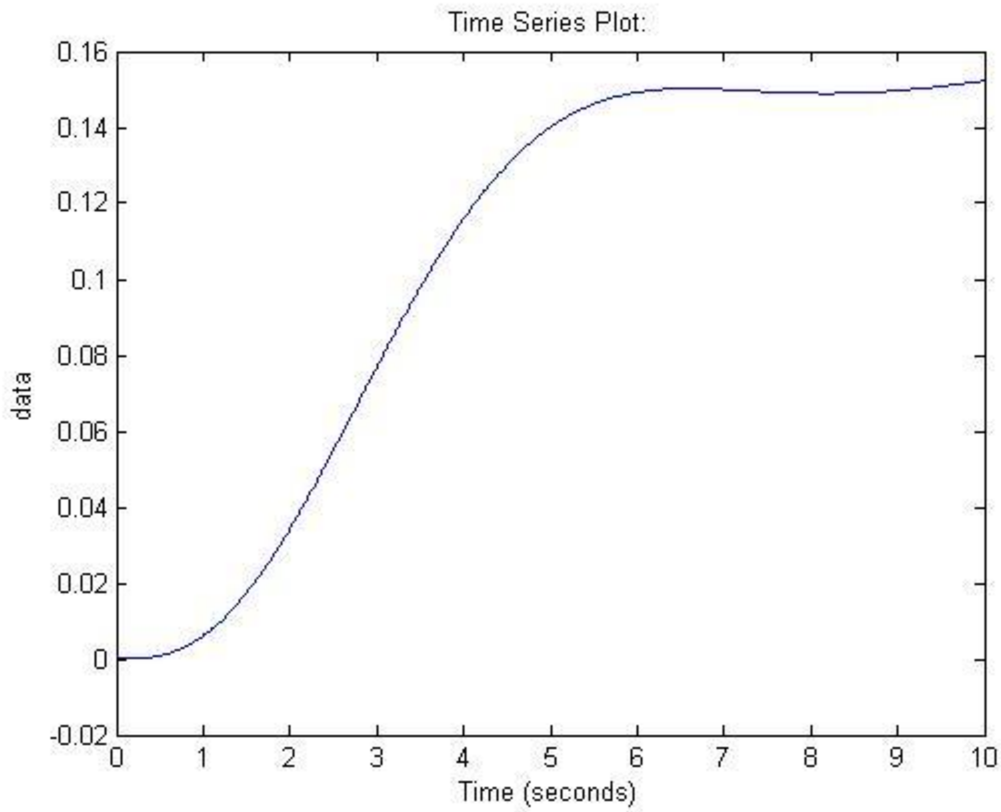


Figure (3.8): The response of the uncontrolled EHH System.

After knowing the exact value of the gain K , a PID controller is to be designed , after implementing this controller and due to system's variation a little tuning in the PID controller is required .

3.2.4.3 System's response using state feedback controller

By using state feedback controller, the system will be stable as shown in the figure below:

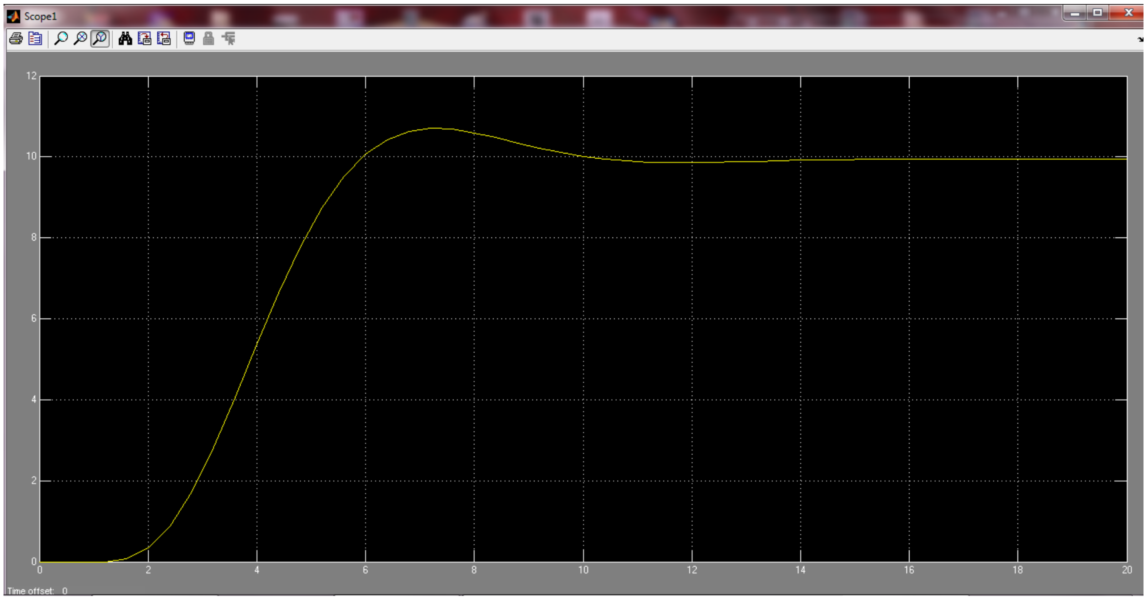


Figure (3.9): The response of the EHH system using state feedback controller.

The block diagram of the EHH system using state feedback controller is shown below:

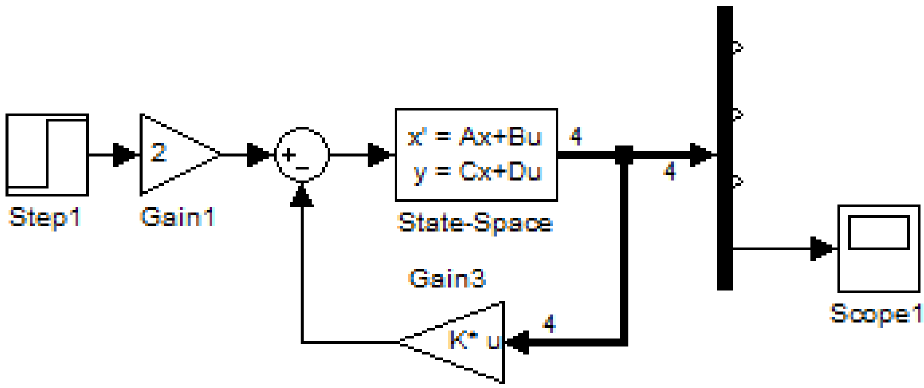


Figure (3.10): The schematic block diagram of the EHH system using state feedback Controller.

This system has only one output which is the last state, because of that the last output from the state space is only measured. So the system must has also three sensors to measure the remain outputs. Therefore, state feedback controller doesn't suitable for this system, so the PID controller must be constructed by experimental tuning in the last chapter.

Table 3.1: Parameters of EHS System

Hydraulic actuator and Servo valve parameters

Description	**Symbol**	**Value**
Total mass	Mp	9 kg
Damping coefficient	Bs	2000 Ns/m
Spring stiffness	Ks	10 Nm
Total actuator displacement	Xs	0.1 m
Piston area	Ap	654 x $10^{-6}\ m^2$
Servo valve damping ratio	ζ	0.48
Servo valve natural frequency	Ω	534 rad/s
Servo valve coil resistance	Rc	100 Ω
Servo valve coil inductance	Lc	0.02 A
Servo valve gain	K	2.38 x$10^{-5}\ m^{5/2}/Kg^{1/2}$
Bulk modulus of hydraulic fluid	β	1.4 x$10^{9}\ N/m^2$
Pump pressure	Ps	2.1 x10^{7} Pa
Return pressure	P_R	0 Pa

Table 3.2: Friction Parameters

Description	Symbol	Value
Stribeck velocity	V_{sk}	0.032 m/s
Coulomb friction	α_0	370 N
Stribeck friction	α_1	217 N
Viscous friction	α_2	2318 N/m/s
Bristles stiffness	σ_0	5.77 x 10^6 N/m
Bristles damping	σ_1	2.28 x 10^4 N/m/s

Electro-Hydraulic Servo System Construction

4.1 Introduction

This chapter discusses the hydraulic model constructed based on the mathematical model explained in chapter three, beginning with the structure design of the servo hydraulic system, principles of system's operation, explaining the characteristics and relationships, plotting curves for flow, pressure and frequency response, and how to use the project in the laboratories' experiments by the students.

4.2 Description of the project components

The project design is constructed and implemented as shown in Figure (4.1), which consists of:

- Iron table
- 2-stage servo valve
- Interface card
- Pump
- Tank
- Piston(cylinder)
- potentiometers
- Card holder
- Filter
- Operational amplifier
- Hydraulic pipes
- Accumulator
- Cooler

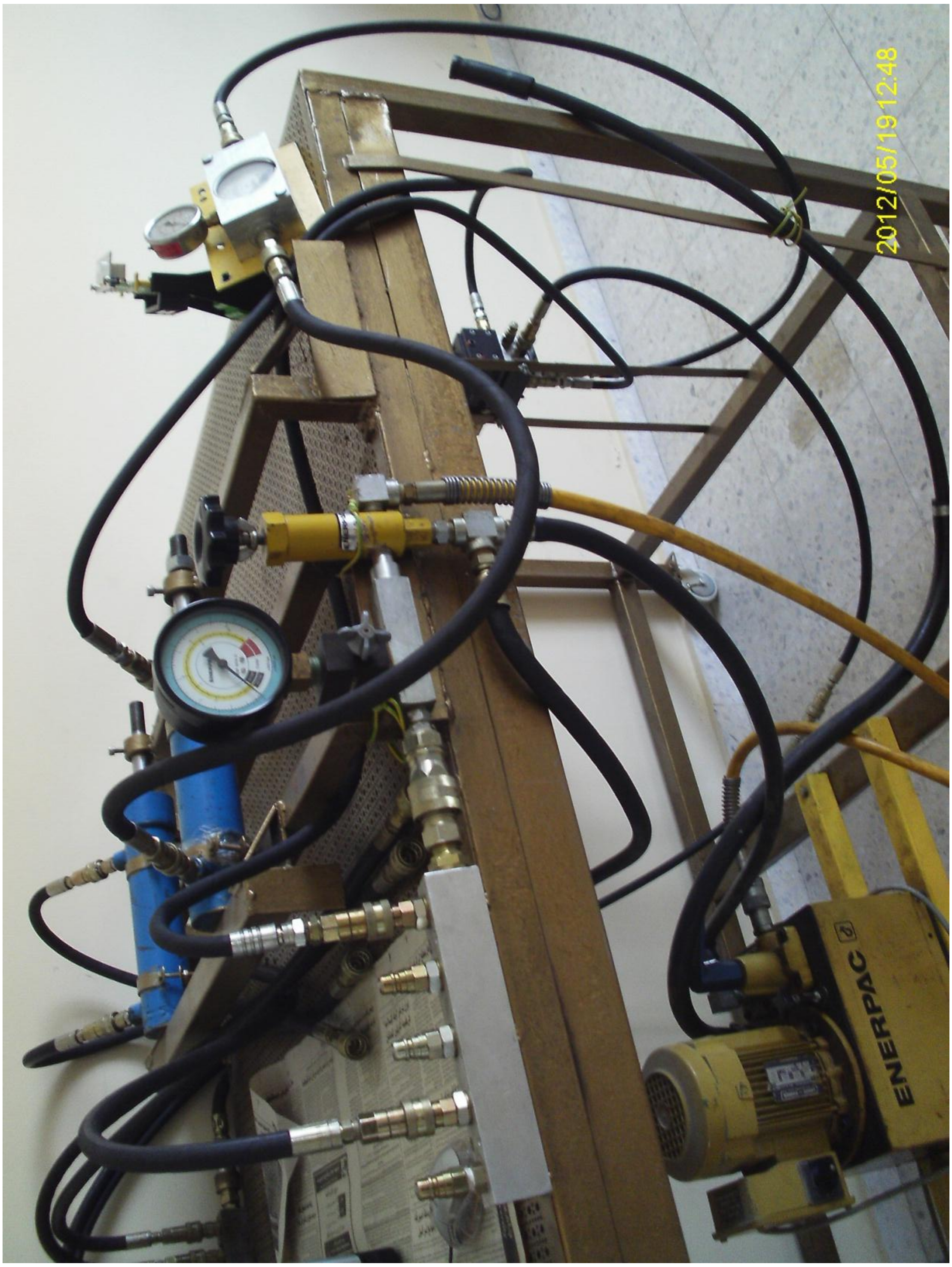


Figure (4.1): Servo Hydraulic System

Mainly, the project’s components are described in the previous chapter, but there are few components that are not, so they are briefly described as follows:

1. Interface card (Analogue amplifier Type VT-SR2):

Servo amplifier cards convert an input signal to a proportional current to drive the servo valve torque motor. On industrial hydraulics systems several servo amplifiers feature an integrated PID (proportional, integrator, and derivative) control circuit, which allows tuning for optimum performance of position control, or constant velocity circuits. With the added ability to combine the PI, PD, or PID controls for any circuit, a wider range of electro-hydraulic motion control sequences can be engineered, for sophisticated applications in automotive, plastics, machine tool, heavy industry and other applications. Its Amplify with high open-loop gain (operational amplifiers) are frequently encountered in analog signal processing circuits, since the application of negative feedback enables numerous transfer functions to be implemented.

Figure (4.2): Analogue amplifier Type VT-SR2. [8]

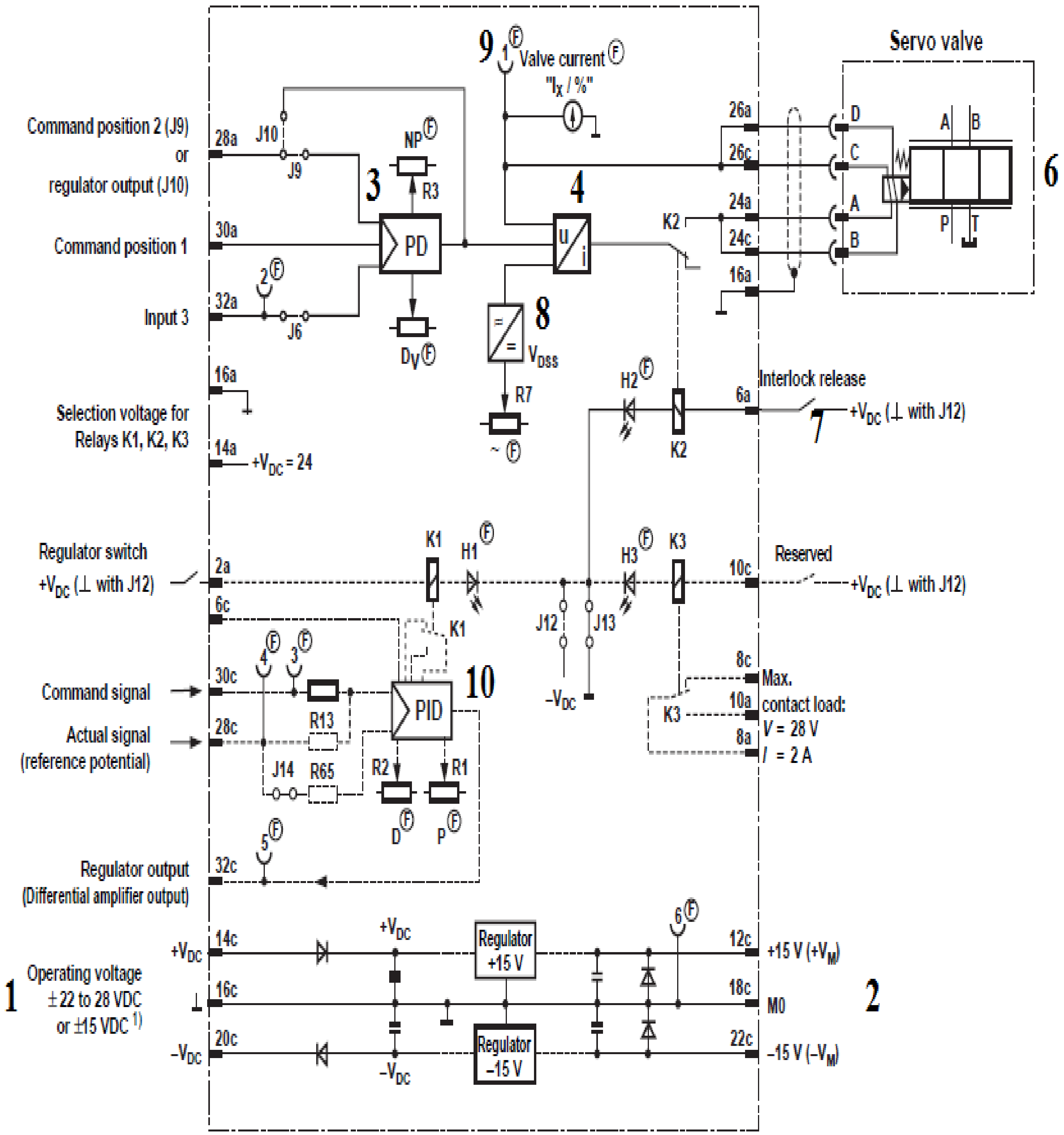


Figure (4.3): Connection manual of servo amplifier VT-SR2

The circuit for the servo amplifier is shown in Figure (4.3), where a smoothed DC voltage between ±22 V and ±28 V is necessary as the supply, and a stabilized voltage of+15 V is then produced from the supply voltage for the amplifier card, this voltage is used for the supply of external actuators such as potentiometers (tapping point at 12c (+15 V) and at 22c (-15 V) and the supply of internal operational amplifiers. Furthermore, two basic function groups should be considered:

a. Control for servo valve with output stage (4) and PD regulator (3). In the version without electrical feedback, the command signal is fed directly to the PD regulator (3), and then the PD regulator is used for the closed loop positional control of the valve itself, where the inductive positional transducer measures the position of the valve spool, and produces an AC voltage signal which varies in amplitude, dependent on the position of the valve spool. The valve position regulator (3) then compares the command signals at 28a (optionally at 30a) with the feedback from the valve spool measured at test point (2) or terminal (32a). Depending on the difference between the command signal and actual signal, the regulator (3) sends the output stage (4) a signal which it converts into a current For example, depending on system pressure, the signal from the output stage (4) may be fed to the contact at (7) and relay K2, this prevents the flapper jet system in the servo valve from being damaged. The flapper jet system may be damaged, if the servo valve is actuated without system pressure being present. Hence, it is good practice to interlock the servo valve by means of a pressure switch in the hydraulic system (input 6a). Other system dependent connections may be linked to this input. A dither current is superimposed upon the valve current by the oscillator (8). The measuring instrument (9) on the front panel of the amplifier indicates the valve current.

The two coils of a torque motor may be connected in three different configurations: parallel, series, and the so-called push-pull arrangement. These options are illustrated in Figure (4.4). The push-pull arrangement is the most common. In the arrangement, leads B and A are both connected to ground through the control circuit amplifier. Leads D and C are connected to separate output terminals on the command amplifier. When the voltage input to both coils is equal, the armature is centered. Increasing the voltage input to one coil, while simultaneously reducing the input to the other coil by the same amount, causes the armature to rotate. The voltage can be varied from zero to its maximum value for each coil, but the polarity is never reversed. This means that the position of the armature is determined by differential torque. When the voltage is the same to both coils, the torque is equal, and the armature is centered. Any change in voltage to either coil results in rotation of the armature.

This push-pull method of connecting the coils is preferred for at least three reasons: First, any changes in current as a result of voltage fluctuations, temperature changes, or other causes are canceled by the equal and opposite effects on the coils. Second, there is more stability in armature positioning because of the opposing torque. Third, the power consumption is lower than for the other two circuits.

In the parallel connection the direction of rotation depends on the polarity of the input signal. Rather than opposing each other (as in the push-pull circuit) the coils in the parallel circuit assist each other. That is, they both attempt to move either clockwise or counterclockwise. Reversing the polarity reverses the direction of rotation [37]

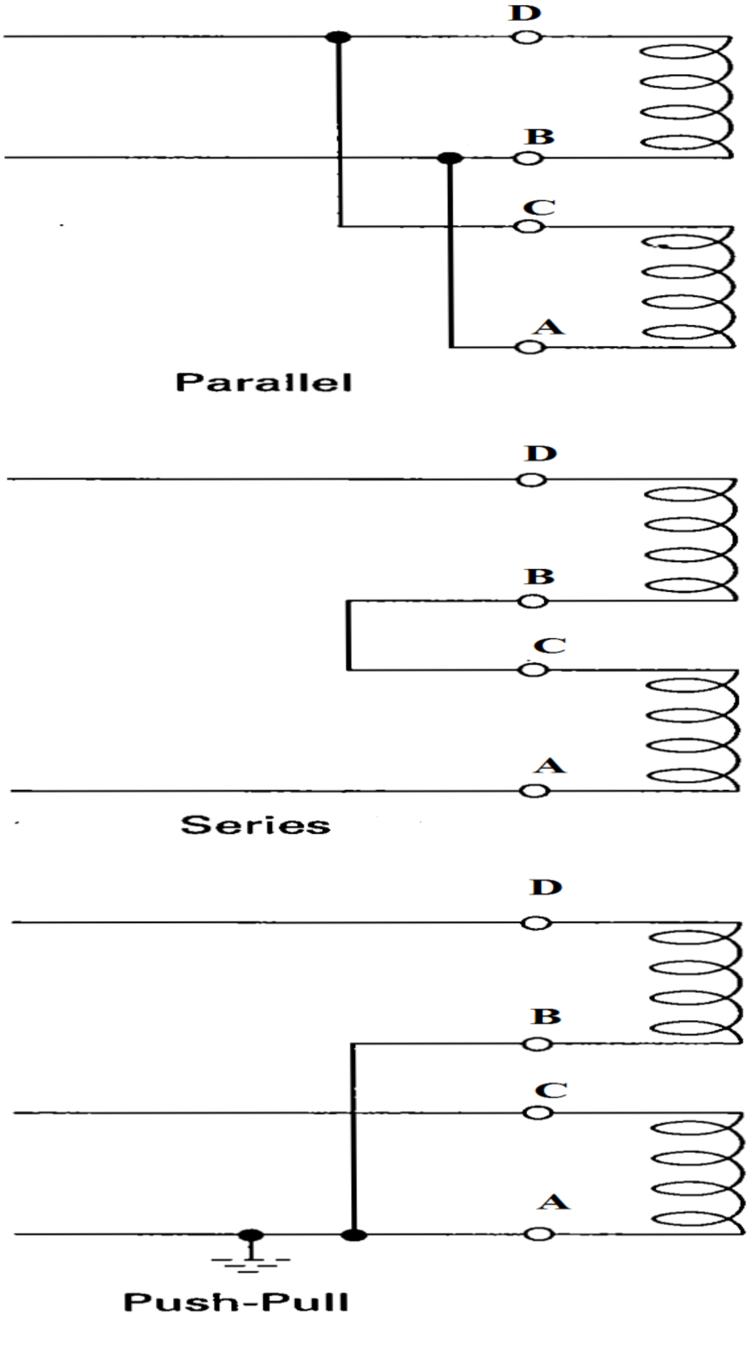


Figure (4.4): Torque motor coil can be connected in several combinations. Each causes different operation.

b. Second regulator (PID) (10) for a superimposed closed loop control circuit. This may be added on request. By using a suitable circuit. The control characteristic is achieved as required. The function sequence is briefly as follows: The PID regulator (10) compares the command signal applied at 30c with the actual value at 28c Depending on the difference, the regulator (at 32c) produces a voltage signal, and this signal may then be used to control the servo valve in 28c. Relay K1 is used to release the regulator (10) which may be selected at terminal 2a.

2. Card holder:

The VT-3002 card holder is available in either a 32-pin or 48-pin connection format. Individual screw terminals aid in robust field connections while the VT-3002 provides a stable platform to anchor field connections. Push buttons on each side permit releasing an amplifier board without incurring undue stress on an amplifier's faceplate.

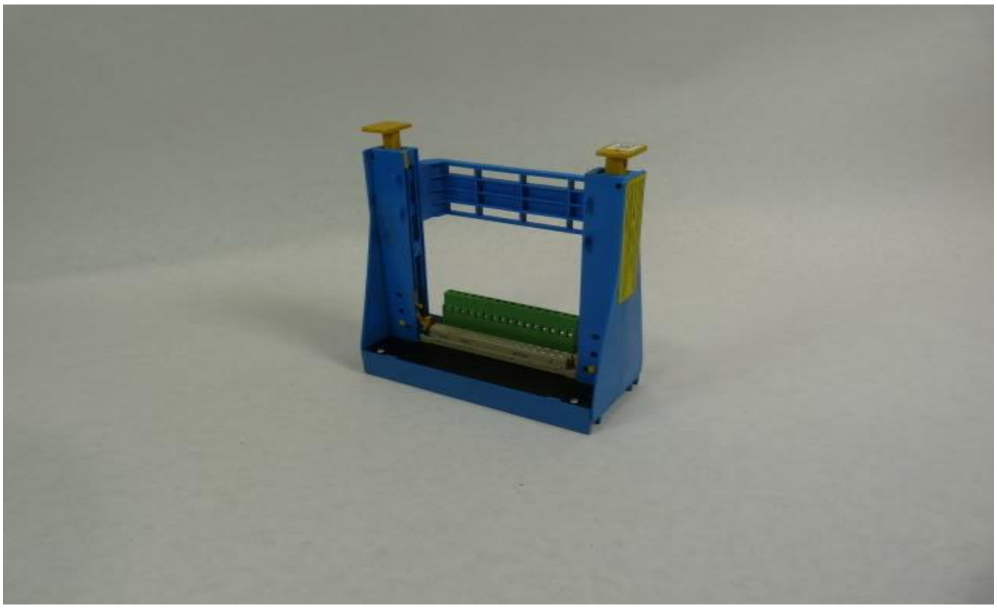


Figure (4.5): card holder.

3. Linear potentiometer:

A linear taper potentiometer has a resistive element of constant cross-section, resulting in a device where the resistance between the contact (wiper) and one end terminal is proportional to the distance between them. Linear taper describes the electrical characteristic of the device, not the geometry of the resistive element. Linear taper potentiometers are used when an approximately proportional relation is desired between shaft rotation and the division ratio of the potentiometer. [10]

Another way of determining the stroke during stroke-dependent deceleration is via a linear potentiometer as shown in Figure (4.6). Similarly, in this version, the stroke is measured via an analogue voltage and this signal is processed via an electronic amplifier. Since, in this case, the entire stroke is converted to a signal; it is possible to select any stroke via the electrical amplifier. The examples described so far are clearly all examples of open loop control. This means that the actual value, e.g. velocity of a cylinder, is not measured and is not compared with the command signal, hence all external disturbances naturally have an effect on results. If it is necessary to compensate for external disturbances. The system must be designed as a closed loop control circuit.

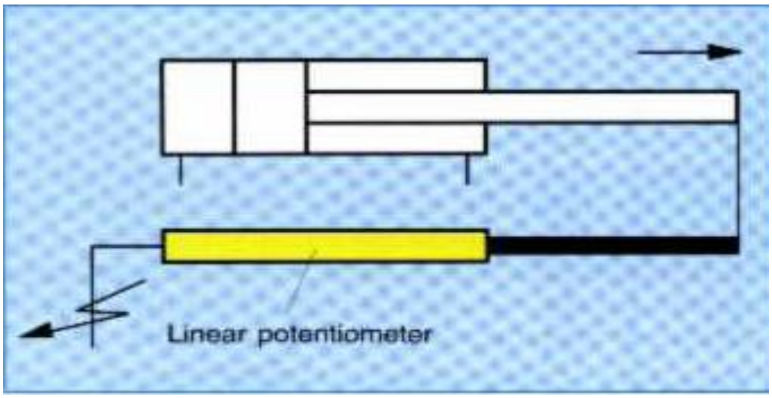


Figure (4.6): linear potentiometer. [1]

2.1 Conductive Plastic Potentiometer for Analogue Measurement

This is a positional transducer with resistor and collector lines made of conductive plastic (analogue measurement), and it has Lengths to be measured up to 1000 mm, it is simply mounted, robust, high precision industrial linear motion transducer, measurement range 25mm to 1000mm, high accuracy $\pm$ 1 % to $\pm$ 0.025%, excellent repeatability, essentially infinite resolution, and non sensitive to temperature variation.

Figure (4.7): Conductive plastic linear potentiometer.

3. Precision rotary potentiometers:

It is a component, a three-terminal resistor with a sliding contact that forms an adjustable voltage divider. If only two terminals are used, one end and the wiper, it acts as a variable resistor, so it's used to control the input voltage to this system.

Figure (4.8): rotary potentiometer.

5. Operational amplifier TL084:

The TL08x JFET-input operational amplifier family is designed to offer a wider selection than any previously developed operational amplifier family. Each of these JFET-input operational amplifiers incorporates well-matched, high-voltage JFET and bipolar transistors in a monolithic integrated circuit. The devices feature high slew rates, low input bias and offset currents, and low offset voltage temperature coefficient. Offset adjustment and external compensation [34]

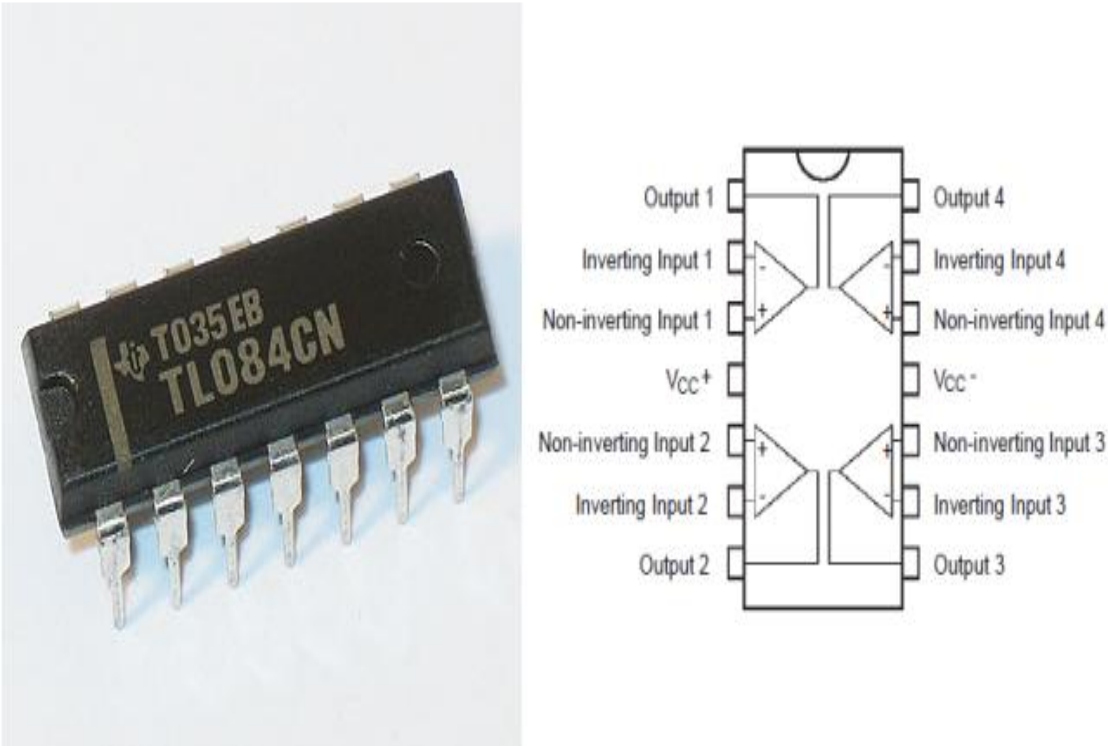


Figure (4.9): operational amplifier.

6. Capacitor:

A capacitor (originally known as condenser) is a passive two-terminal electrical component used to store energy in an electric field. The forms of practical capacitors vary widely, but all contain at least two electrical conductors separated by a dielectric (insulator); for example, one common construction consists of metal foils separated by a thin layer of insulating film. Capacitors are widely used as parts of electrical circuits in many common electrical devices. It's used to build pi controller for this system.[36]

Figure (4.10): capacitor.

7. Resistors:

A resistor is a passive two-terminal electrical component that implements electrical resistance as a circuit element. The current through a resistor is in direct proportion to the voltage across the resistor's terminals. Thus, the ratio of the voltage applied across a resistor's terminals to the intensity of current through the circuit is called resistance. It's used to build pi controller for this system.

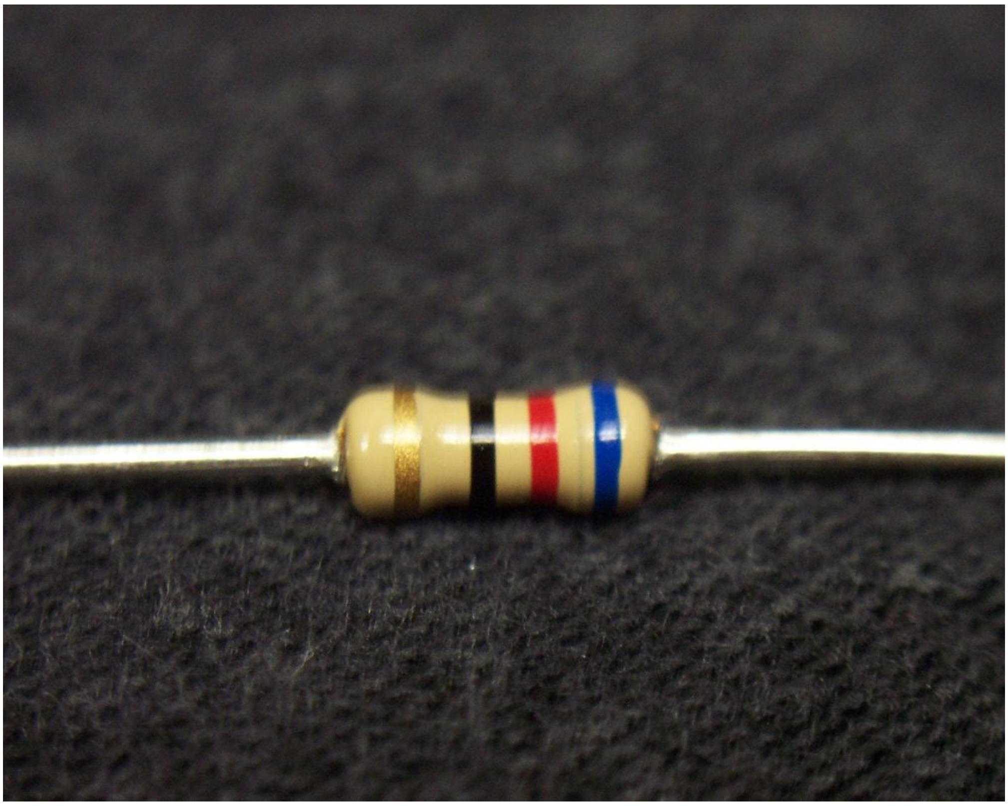


Figure (4.11): resistor.

8. Flow meter:

It’s an instrument for monitoring, measuring, or recording the rate of flow, pressure, or discharge of a fluid, as of a gaseous fuel.

Figure (4.12): flow meter. [13]

9. Pressure gauge:

Instrument for measuring the condition of a fluid (liquid or gas) that is specified by the force the fluid would apply, when at rest, to a unit area, such as pounds per square inch (psi) or Pascal's (Pa). The reading on the gauge, called the gauge pressure, is always the difference between two pressures. When the lower of the pressures is that of the atmosphere, the total (or absolute) pressure is the sum of the gauge and atmospheric pressures. [14]

Figure (4.13): Pressure gauge. [15]

4.3 Principles of system's operation

A command voltage is selected via the command voltage potentiometer. This voltage corresponds to the piston's position. The actual position of the piston is measured and fed back as a voltage by the feedback potentiometer. At the amplifier input the difference between these two voltages, i.e. the command/feedback signal difference or closed loop error is created. The error is amplified by the amplifier and is thus able to energize the servo valve coil. As a result, the servo valve opens and the piston moves. As the actual displacement changes, the actual voltage created by the feedback potentiometer also changes. The actual voltage gradually approaches the command voltage, until they are finally both equal when the required position has been reached During this process, the error becomes smaller and smaller, and despite amplification, less current is available to the servo valve coil. This means the servo valve gradually closes and therefore the piston slows down. When the required displacement has been reached, the error is zero and the servo valve closed. The variables disrupting open loop control no longer or scarcely effect closed loop control. This is an important feature of closed loop control and therefore also of servo hydraulics.

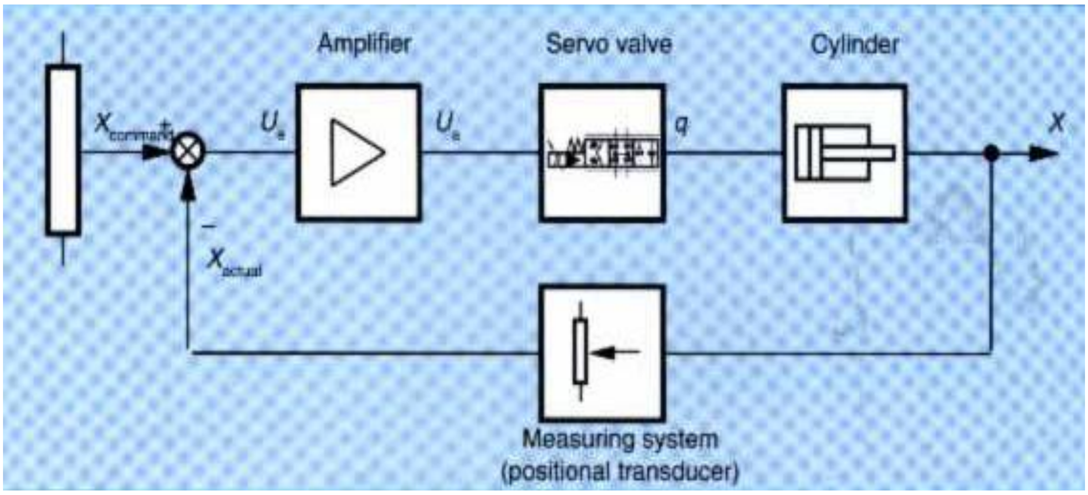


Figure (4.14): block diagram of a closed loop control circuit. [1]

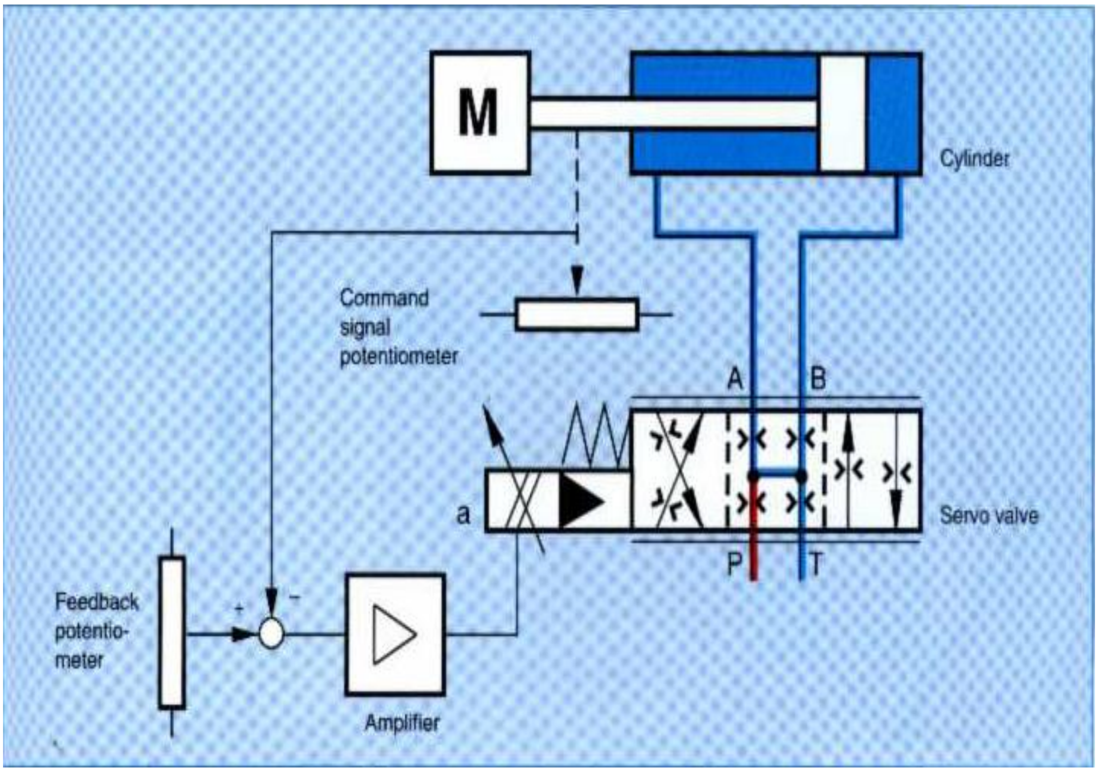


Figure (4.15): closed loop control circuit with servo valve. [1]

4.4 Static Technical Data

4.4.1 Nominal Flow

The nominal flow of a servo valve is normally quoted at a total pressure drop of 70 bars, However this does not mean that operation is limited to 70 bar pressure drop, any operating point (flow) may be used. The nominal flow is always with respect to full stroke operation of the servo valve. In the case of partial stroke operation the Row varies in proportion to the stroke. [1]

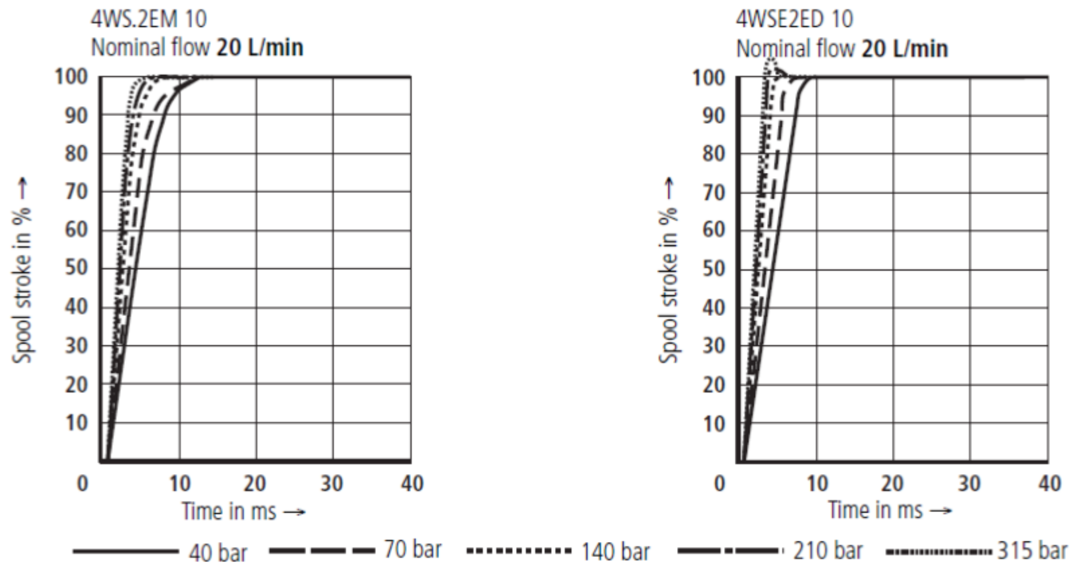


Figure (4.16): Transient functions with the 315 bar pressure stage, step response without flow. [7]

4.4.2 Flow Operation Curve

The operating curve for flow shows the relationship between valve flow and electrical input signal

A. B = Characteristic operating points

A = Null point operation

B = Operating point when open

To understand the importance of operating points in control systems refer to Figure (4.2) below: [1]

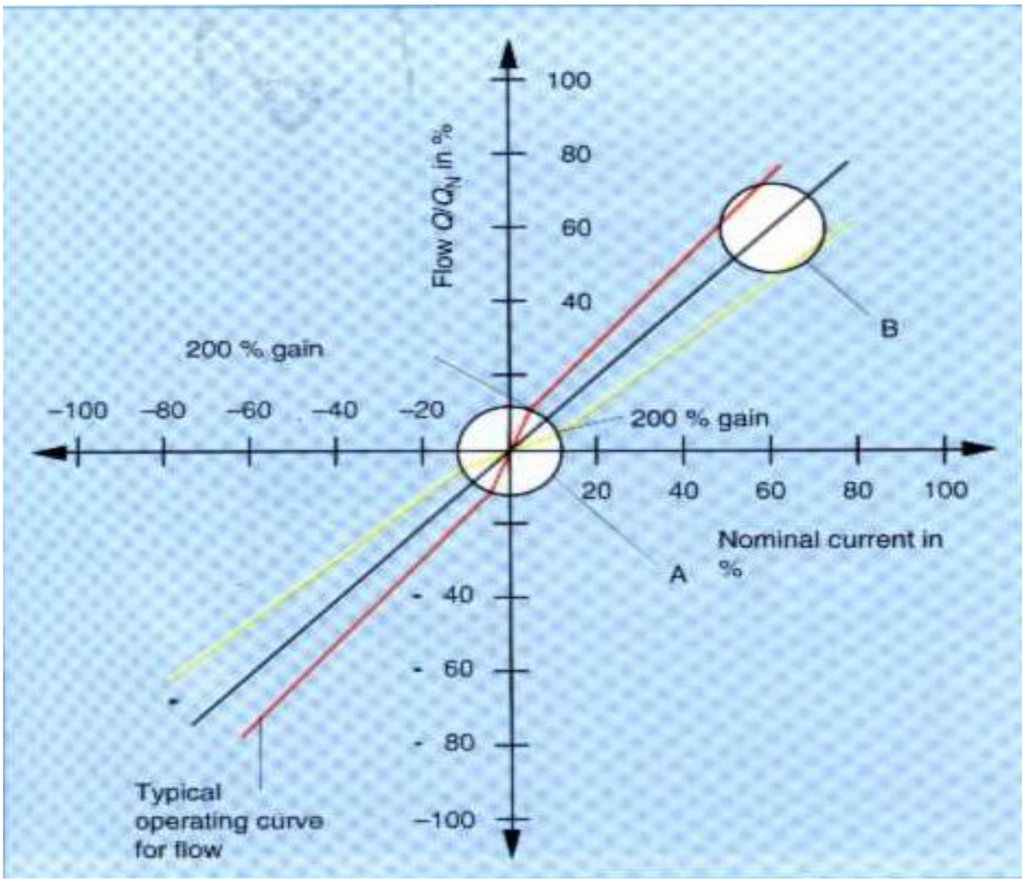


Figure (4.17): operating curves for flow. [1]

4.4.3 Flow-Load Characteristic

A hydraulic servo drive generally consists of a servo valve and a cylinder or motor as the actuator movements are effected by throttling the supply oil flow.

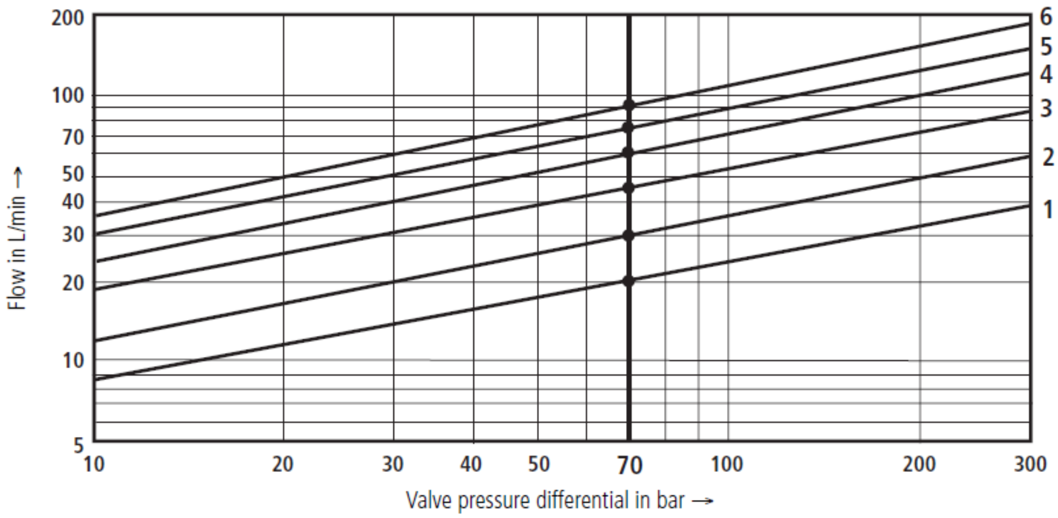


Figure (4.18): flow-load characteristics. [7]

Where Δp is the pressure difference, and the nominal flow as shown sequentially:

20 L/min $\widehat{=}$ curve 1	60 L/min $\widehat{=}$ curve 4
30 L/min $\widehat{=}$ curve 2	75 L/min $\widehat{=}$ curve 5
45 L/min $\widehat{=}$ curve 3	90 L/min $\widehat{=}$ curve 6

4.4.4 Pressure-Signal Characteristic.

Force is necessary to correct a drive. For this reason, the behavior of the output pressure with respect to the input signal is significant. This behavior is shown in the pressure-signal characteristic.

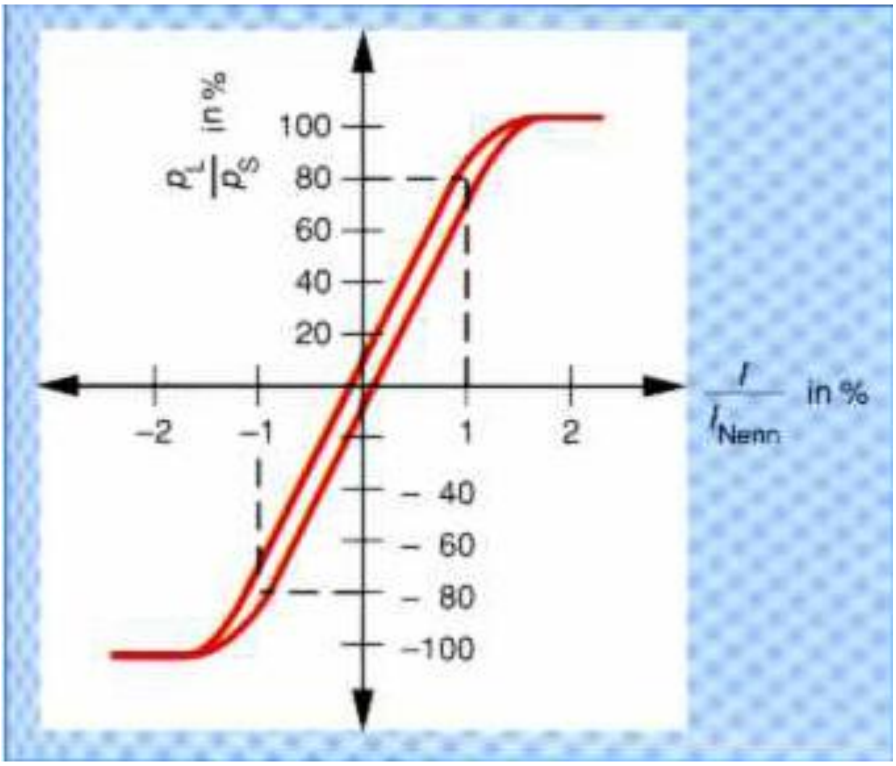


Figure (4.19): pressure signal characteristic. [1]

4.5 Dynamic Technical Data

The natural frequency of the drive and resulting total gain determine the closed loop control accuracy. The natural frequency of the drive is determined by the dynamics of the servo valve. An examination of the positioning time of the valve is not sufficient to describe its dynamic characteristics, the most common way of examining the dynamic characteristics is by looking at the frequency response. During this process, the servo valve is excited by sine waves and the reaction of the valve to these signals is monitored. The servo valve response (flow O) is also sinusoidal. However, its amplitude and phase are offset with respect to the excitation signal. Initially, the frequency of the applied signal is low and then gradually increased. As a result, as the frequency increases, the output amplitude decreases, and the movement of the valve lags more and more behind the input signal.

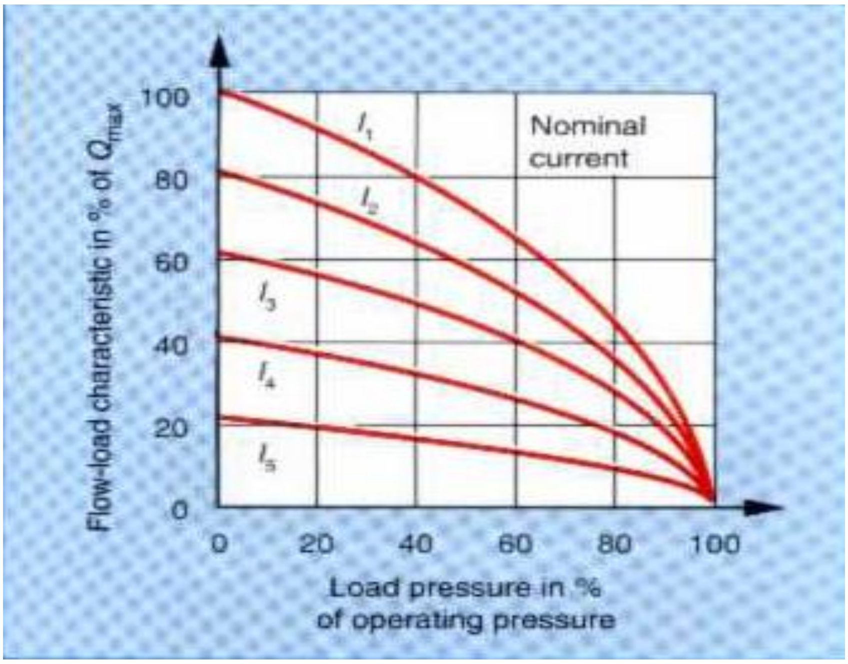


Figure (4.20): Flow-load characteristic. [1]

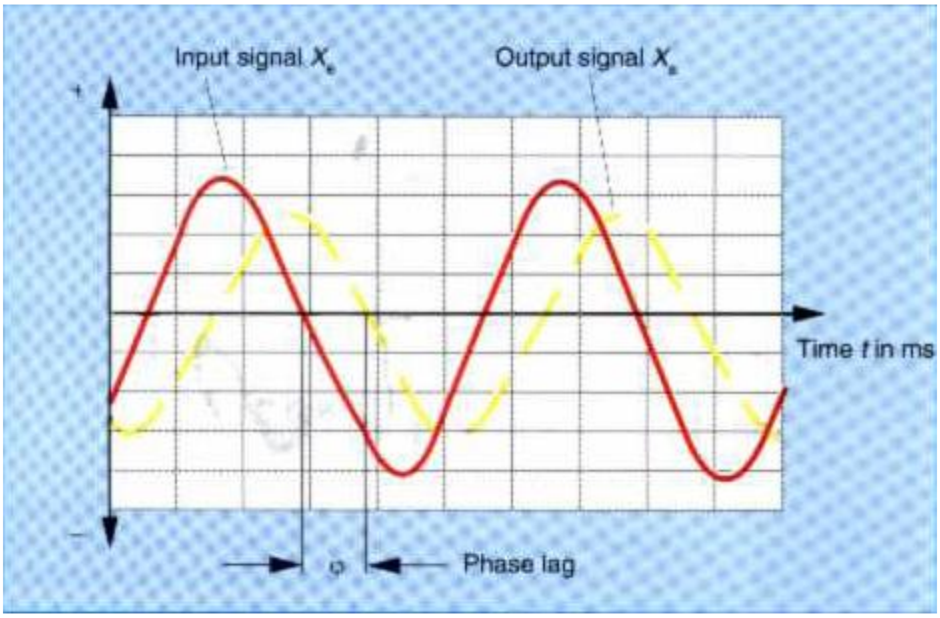


Figure (4.21): Frequency response curve. [1]

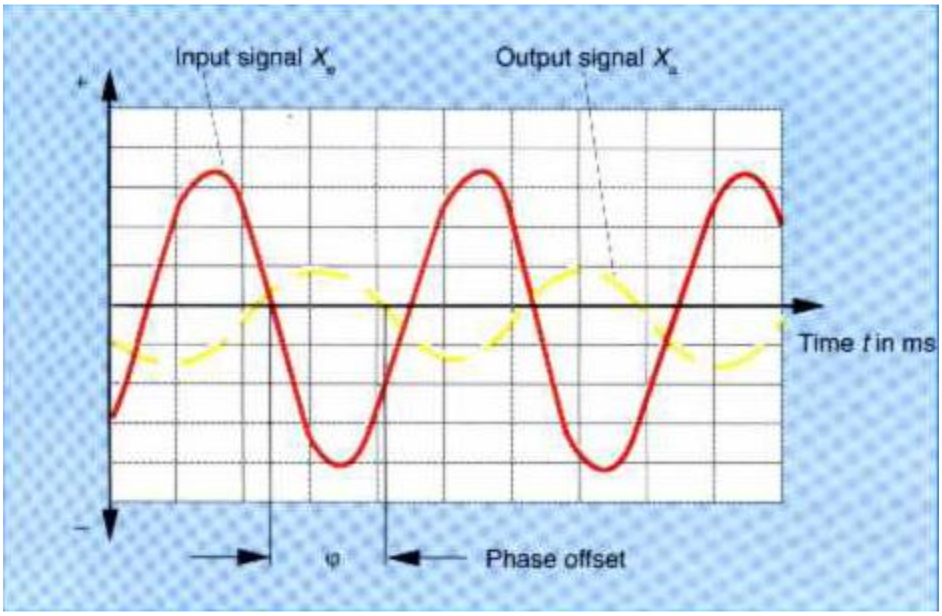


Figure (4.22): Frequency response curve. [1]

4.6 Application example

As an application of EHS is the injection plastic machine exists at Palestine Polytechnic University, it is shown in Figure (4.23) below:

Figure (4.23): Ningbo Shuangma Plastic Machine. [18]

This machine is used to produce variable types of plastic containers, which represents the interfacing between electrical, mechanical, and hydraulic system. All of its actuator movements are controlled by EHS.

Appendix

A

Servo directional valve of 4-way design
Type 4WS.2E…

Appendix

B

Analogue amplifier

Appendix

C

Precision Linear Transducers, Conductive Plastic, up to 1000 mm

Appendix

D

Card holder

Appendix

E

Compact power supply units

Appendix

F

Precision-Rotary-Potentiometers

Rexroth Bosch Group

Compact power supply units

RE 29929/03.09
Replaces: 03.05

VT-NE30, component series 2X
VT-NE31, component series 1X
VT-NE32, component series 1X

Table of contents

Features

- Power supply unit **VT-NE30** (unipolar output voltage):
  - For the voltage supply of amplifiers for proportional valves
  - For the voltage supply of proportional valves with integrated electronics
- Power supply unit **VT-NE31** (bipolar output voltage):
  - For the voltage supply of amplifiers for servo-valves
- Power supply unit **VT-NE32** (unipolar output voltage):
  - Same function as power supply unit VT-NE30
  - In addition, regulated output voltage for the supply of closed-loop control or evaluation electronics or accessory components such as pressure transducers, position measuring systems
- Compact design, connection via terminal strip
- Unregulated output voltages (filtered); with VT-NE32 additionally regulated output voltage
- Short-circuit protection by miniature fuses in output circuits; with VT-NE32 cut-off of the regulated output voltage after approx. 1.5 s in the case of overloading or short-circuit; restart by pressing the “reset” key
- Can be screwed on or snapped onto top hat rail EN 60715
- LED indicator lamps
- Conformity with EC Directives (CE mark)

Ordering code

VT– –1X/ *

Compact power supply unit, bipolar	**= NE31**
Compact power supply unit, unipolar and regulated	**= NE32**
1X =	Series 10 to 19 (10 to 19: unchanged technical data and pin assignment)
*	Further details in clear text

VT– –2X/ *

Compact power supply unit, unipolar	**= NE30**
2X =	Series 20 to 29 (20 to 29: unchanged technical data and pin assignment)
*	Further details in clear text

Technical data (for applications outside these parameters, please consult us!)

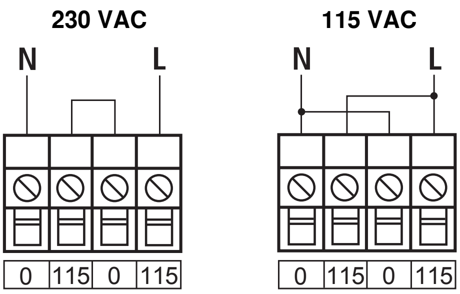
		VT-NE30	VT-NE31		VT-NE32	
			Output 1	Output 2	Output 1	Output 2
		Filtered	Filtered	Filtered	Filtered	Controlled
Operating voltage	U	230 VAC ±10 % or 115 VAC ±10 %				
Frequency	f	50/60 Hz				
Output	P_O	108 W	6 W	6 W	60 W	24 W
Output voltage (at U = 230 VAC):						
– No load	U_O	+32 VDC [1)]	+27 VDC [1)]	–27 VDC [1)]	+30 VDC [1)]	+24 VDC
– Full load	U_O	+26 VDC [1)]	+24 VDC [1)]	–24 VDC [1)]	+25 VDC [1)]	+24 VDC
Load current	I_{max}	4 A	0.25 A	0.25 A	2.5 A	1 A
Residual ripple content (full load)	U_{eff}	< 5 %	< 5 %	< 5 %	< 2 %	< 50 mVAC
Fuse	I_F	8 A slow-blow	0.8 A slow-blow	0.8 A slow-blow	4 A slow-blow	See features
Connection cross-section	A	max. 4 mm²				
Permissible ambient temperature range	ϑ	–20 to +50 °C				
Storage temperature range	ϑ	–20 to +70 °C				
Weight	m	3.1 kg	1.1 kg		2.2 kg	

[1)] In the case of deviations of ±10 % from operating voltage U = 230 VAC, the output voltage U_O changes by ±10 %.

Note:
For details regarding **environment simulation testing** in the fields of EMC (electromagnetic compatibility), climate and mechanical stress, see RE 29929-U (declaration on environmental compatibilty).

Terminal assignment

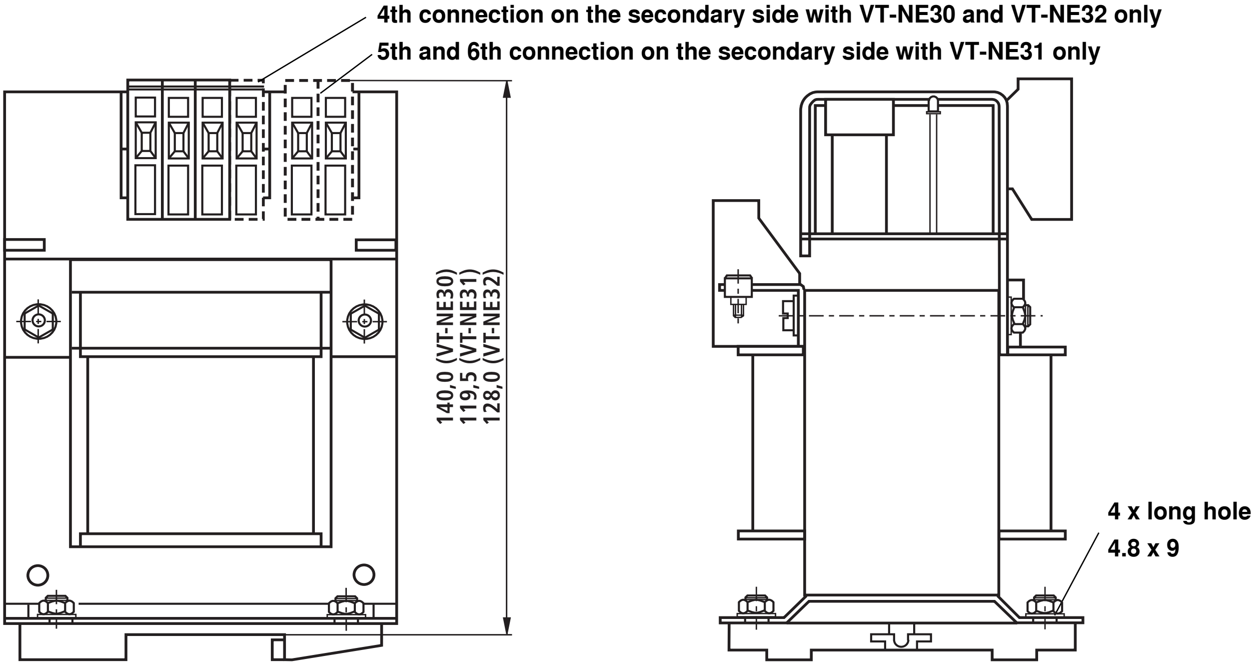
Connection of operating voltage



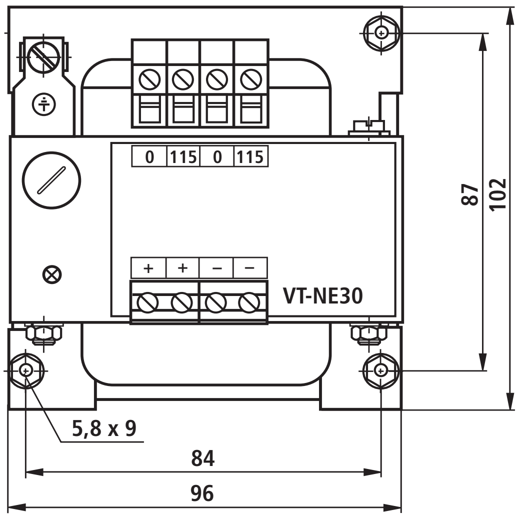
Compact power supply units VT-NE30 to VT-NE32 are ready for connection for an operating voltage of 230 VAC.

For screwed mounting, remove the plate for hat rail mounting.

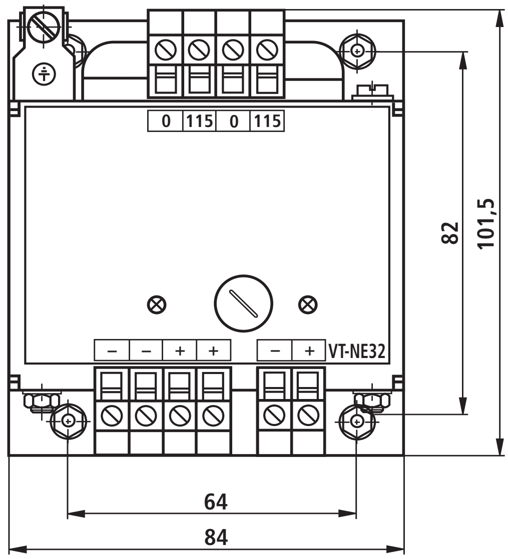
Unit dimensions (dimensions in mm)



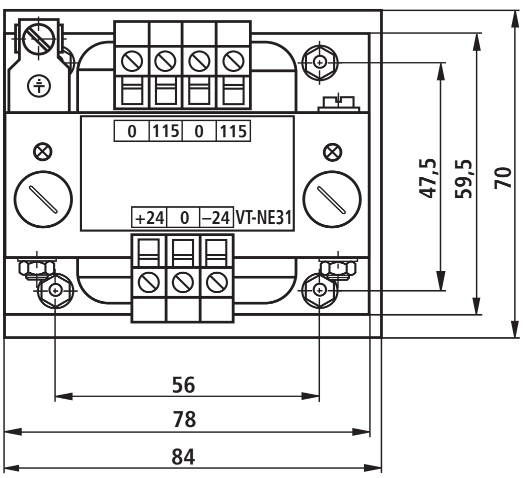
Top view VT-NE30



Top view VT-NE32



Top view VT-NE31



Notes

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Precision Linear Transducers, Conductive Plastic, up to 1000 mm

The 115 L is a simply mounted, robust, high precision industrial linear motion transducer.

FEATURES

- Measurement range 25 mm to 1000 mm
- High accuracy ± 1 % down to ± 0.025 %
- Excellent repeatability
- Essentially infinite resolution
- Non sensitive to temperature variations

RoHS COMPLIANT

ELECTRICAL SPECIFICATIONS

Theoretical Electrical Travel (TET) = E	From 25 mm to 1000 mm in increments of 25 mm
Independent Linearity (over TET) On Request	≤ ± 1 % ≤ ± 0.1 % ≤ ± 0.05 % for E ≥ 100 mm ≤ ± 0.025 % for E ≥ 200 mm
Actual Electrical Travel (AET)	AET = TET + 1.5 mm min.
Ohmic Values (R_T)	400 Ω/cm to 2 kΩ/cm
Resistance Tolerance at 20 °C	± 20 %
Repeatability	≤ ± 0.01 %
Maximum Power Rating	0.05 W/cm at 70 °C, 0 W at 125 °C
Wiper Current	Recommended: a few µA - 1 mA max. (continuous)
Load Resistance	minimum 10^3 x R_T
Insulation Resistance	≥ 1000 MΩ, 500 V_{DC}
Dielectric Strength	≥ 1000 V_{RMS}, 50 Hz
Protection Resistor	Integrated inside the transducer to protect against errors when setting up (short circuit)

MECHANICAL SPECIFICATIONS

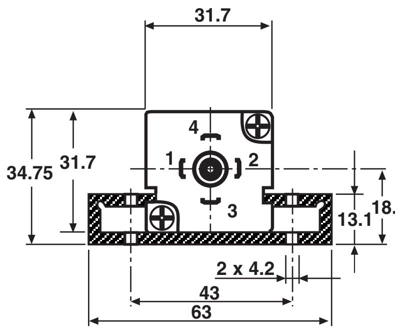
Mechanical Travel	E + 8 ± 2 mm
Housing	Anodized aluminum
Operating Force	7.5 N typical
Shaft (Free Rotation)	Stainless steel
Termination	Hydraulic type connector DIN 43650
Wiper	Precious metal multifinger
Mounting	Movable brackets

PERFORMANCE

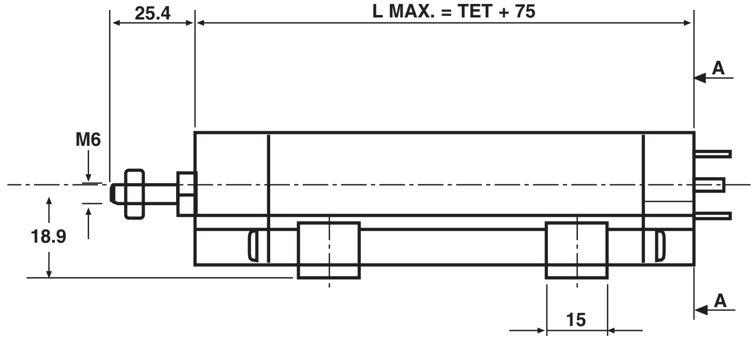
Operating Life	40 million cycles typical/1 Hz/T° = 20 °C ± 5 °C/80 % TET
Temperature Range	- 55 °C to + 125 °C
Sine Vibration on 3 Axes	1.5 mm peak to peak 0 - 10 Hz 15 g - 10 Hz - 2000 Hz
Mechanical Shocks on 3 Axes	50 g - 11 ms - half sine
Speed (max.)	8 m/s for f < 2 Hz; 3 m/s for f < 5 Hz

Precision Linear Transducers, Conductive Plastic, up to 1000 mm

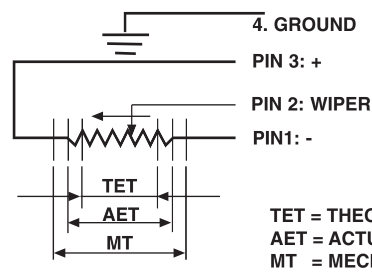
DIMENSIONS in millimeters, general tolerance ± 1 mm



VIEW A-A



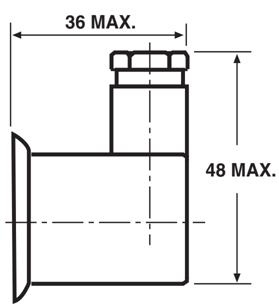
ELECTRICAL CONNECTIONS



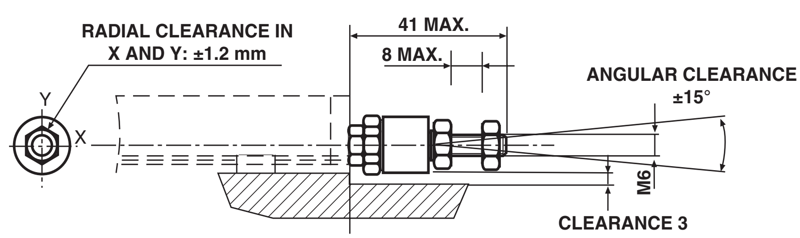
TET = THEORETICAL ELECTRICAL TRAVEL
AET = ACTUAL ELECTRICAL TRAVEL
MT = MECHANICAL TRAVEL

ACCESSORIES ON REQUEST DIMENSIONS in millimeters, general tolerance ± 1 mm

1) FEMALE CONNECTOR
Vishay's Reference: 3248610



2) SPECIAL BALL JOINT ON SHAFT
Vishay's reference: 323655



ORDERING INFORMATION/DESCRIPTION

REC	115	L	23	D	103	W...	e.
SERIES	MODEL	NUMBER OF TRACKS	THEORETICAL ELECTRICAL TRAVEL	LINEARITY	OHMIC VALUE	MODIFICATIONS	LEAD FINISH
		L = 1	Times 25 mm	A: ± 1 % D: ± 0.1 % E: ± 0.05 % F: ± 0.025 %	First 2 digits are significant numbers 3rd digit indicates number of zeros	Special feature code number	

SAP PART NUMBERING GUIDELINES

RE	115 L	23	D	103	W....
SERIES	MODEL	TET	LINEARITY	OHMIC VALUE	SPECIAL FEATURES

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