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الحركة التزامنية لاسطوانات هيدروليكية متعددة باستخدام صمام تقسيم الجريان

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كجزء من متطلبات نيل شهادة الماجستير في علوم الهندسة الميكانيكية

من قبل

المهندس سعد حنون سعدون البزوني

بكالوريوس في الهندسة الميكانيكية (2007)

بأشراف

أ.م. د. رافع عباس البلداوي **د. يحيى عبد الله فرج**

1435 هـ 2014 م

بِسْمِ اللّٰهِ الرَّحْمنِ الرَّحِيمِ

﴿شَهِدَ اللَّهُ أَنَّهُ لاَ إِلَهَ إِلاَّ هُوَ وَالْمَلائِكَةُ
وَأُوْلُواْ الْعِلْمِ قَائِمًا بِالْقِسْطِ لاَ إِلَهَ إِلاَّ هُوَ
الْعَزِيزُ الْحَكِيم﴾

صَدَقَ اللّٰهُ العَظيم

سُورَةُ آل عِمرانَ(الآية 18)

Abstract

A common problem in hydraulic applications is the synchronization of multi cylinders that must match each other's position. Stroke length depends on the volume of oil delivered to the cylinders, and moving them together requires equal oil flow to each cylinder as well as diameters .Oil leakages, pump slip, changing workloads, and varying friction loads could also affect oil delivery among other factors. Flow divider valve has been used to control the motion of dual actuators. The system allows monitoring displacement, pressure and flow rate synchronously. The position readings measured by position sensors ,pressure readings measured by pressure transmitters and flow readings measured by flow sensors. A control system has been designed and manufactured for this purpose. It consists of three main parts; measurements sensors, control board, and data show. Three values of oil operating temperatures (40, 45,50) ^{o}C have been used to show the effect of the temperature. Also, the compressibility factor has been studied. Different weights have been used to produce different pressures (4,10) bar. Matlab/Simulink has been used to represent the mathematical model of the system. Both, a physical Simulation and circuits drawing have been achieved by Automation studio.

A comparison study has been done among the conventional, flow control valve and flow divider valve synchronous circuits for different experimental cases. These cases include: no load no linkage; no load with linkage; load in the center; off center load and finally with separate load. The results show a percentage error (11.1%) for conventional system, (1.6%) for FCV and (0.2%) for FDV for the case no load no linkage.

Acknowledgment

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Saad

2014

Dedication

To the soul of Dr. Ali Muhawesh

To my Mother and Father

To my friend Jaffer

To my wife and my kids

Jaafer and Fatimah

Saad

2014

Examining Committee Certificate

We certify that we have read this thesis entitled **(Motion Synchronization for Multi Hydraulic Cylinders by Using Flow Divider Valve)** and as an examining committee, examined the student **(Saad Hanoon Saadoon)** in its contents and that in our opinion it meets the standard of a thesis for the Degree of Master of Science in Mechanical Engineering.

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Date: 2 / 2 / 2014

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List of abbreviations

Abbreviation	Mean
ADC	Analogue to digital Convertor
AS	Automation Studio
EHV	Electro-hydraulic valve
FCV	Flow control valve
FDV	Flow divider valve
FVC	Frequency -to- Voltage Converter
GUI	Graphical user interface
LVDT	Linear Variable Differential Transformer
PRV	Pressure relief valve

Nomenclature

symbol	Description	unit
$A_{1,2,3,4}$	Orifice area in FCV	m^2
A_L	Area of load orifice	m^2
A_o	Area of fixed orifices	m^2
A_p	Cylinder piston side area	m^2
A_r	Cylinder rod side area	m^2
A_{ref}	pressure relief valve area	m^2
A_s	spool end area for FDV	m^2
A_v	Area of variable orifices	m^2
C_d	Discharge coefficient of FCV	dimensionless
C_L	Discharge coefficient for load orifice	dimensionless
C_{do}	Discharge coefficient for the fixed orifices	dimensionless
C_{dv}	Discharge coefficients for variable orifices	dimensionless
D	Piston diameter	m
d	rod diameter	m
d_f	Filter diameter	m
D_p	Volume displacement rate of the hydraulic pump	m^3/ rad
E_{cu}	Cumulative error	dimensionless
E_{ss}	Steady state error	dimensionless
F_{hs}	Hydrostatic force	N
F_{hd}	Hydrodynamic force	N
F_s	Spring force	N
f_f	FCV viscous factor	N.sec/m
f_p	Piston viscous factor	N.sec/m
f_r	Pressure relief valve viscous factor	N.sec/m

f_s	FDV viscous factor	N.sec/m
g	gravitational acceleration	m/sec^2
k_e	Elasticity coefficient of pipe	m^3 /Pa
K_f	Spring constant of FCV	N/m
K_r	Spring constant of PRV	N/m
K_s	Spring constant of FDV	N/m
M	External mass	kg
M_f	Mass of FCV spool	kg
M_r	Mass of PRV spool	kg
M_s	Mass of FDV spool	kg
N	Angular speed of the hydraulic pump	rad /sec.
pa	Supply pressure to FDV	Pa
p_{a1}, p_{a2}	Pressure between fixed and variable orifices in FDV	Pa
p_i , p_p	Inlet and outlet pressures of hydraulic pump	Pa
p_L	Pressure drop across the load	Pa
p_p	Pressure of the hydraulic pump	Pa
p_r	Case drain pressure of the hydraulic pump	Pa
p_s	Supply pressure to FCV	Pa
p_o	Pressure tank	Pa
Q_a	Supply flow rate of FDV	m^3/sec.
Q_{a1}, Q_{a2}	Flow rate of FDV chambers	m^3/sec.
Q_{c1}, Q_{c2}	Return flow rate to the DCV	m^3/sec.
Q_L	Flow through DCV	m^3/sec.
Q_{line}	Flow rate in any hydraulic line	m^3/sec.
Q_{L1}, Q_{L2}	Flow rate of FDV outports	m^3/sec.
$Q_{p\ ideal}$	Ideal flow rate of the hydraulic pump	m^3/sec.
$Q_{p\ atcuale}$	Actual flow rate of the hydraulic pump	m^3/sec.

Q_{r1}, Q_{r2}	Return flow rate from the hydraulic cylinders	m^3/sec.
Q_{ref}	Flow through the pressure relief valve	m^3/sec.
Q_s	Supply flow rate to the DCV	m^3/sec.
$Q_{1,2,3,4}$	Flow through directional valve	m^3/sec.
t	Time	Sec.
t_{cum}	Time used when calculating cumulative error	Sec.
V_{a1}, V_{a2}	Fluid volume in FDV champers	m^3
V_L	Fluid volume in the hydraulic line to FDV	m^3
V_{L1}, V_{L2}	Fluid volume in hydraulic lines to the actuators	m^3
V_{line}	Fluid volume in any hydraulic line	m^3
V_r	Fluid volume in return hydraulic line to DCV	
V_{r1}, V_{r2}	Hydraulic oil volume in return lines from cylinders	m^3
V_s	hydraulic oil volume in pump line	m^3
v	Piston speed	m/sec.
w	area gradient	m
x_{p1}, x_{p2}	Cylinder1 and Cylinder2two position	m
x_s	Spool divider valve position	m
x_v	Spool directional valve position	m
z	Spring pressure relief valve position	m
η_{vp}	Volumetric efficiency of the hydraulic pump	dimensionless
ρ	Density of the hydraulic oil	kg/m^3
ν	Kinematic viscosity	m^2/sec.
μ	Absolute viscosity of the fluid	Pa.sec.
β	Effective bulk modulus	pa

Chapter One

Introduction

1.1 Fluid Power System

Fluid power is the transmission of forces and motions using a confined pressurized fluid. In hydraulic fluid power systems, the fluid is oil, while it is air in pneumatic fluid power systems. Hydraulic systems are any systems using pressurized oils to generate power by converting mechanical energy to hydraulic energy, and then back to mechanical energy. As shown in Figure (1-1).

The principle reason for converting to fluid energy is the convenience of transferring energy to a new location. The dynamic performance is superior when compared to electrical or mechanical drive systems in large power drive systems .For those systems which require an output power larger than 10 kW and a fast response speed,hydraulic drive systems are often the appropriate choice.

Hydraulic systems are especially suitable for those operations characterized by sudden loading; frequency stopping and starting; reversibility and speed variations. All these could cause sharp peak, cyclic and fluctuating power demands. These advantages make them very popular in applications such as in aircrafts, mobile equipments, lifting machines and forest machines [1, 2].

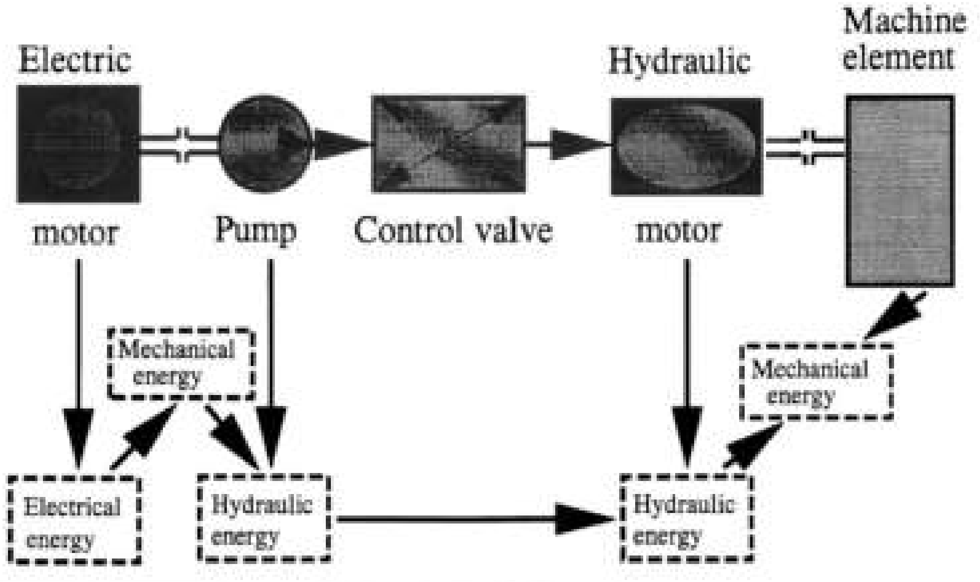


Fig. (1-1) Energy conversion in hydraulic system [3].

1.2 Hydraulic System Units

The physical appearance of hydraulic power systems varies considerably, depending on the type of used oil, application and power output. However, hydraulic system is structured using component groups which perform various system functions. The structure of typical hydraulic power systems involves five component groups as shown in Figure (1-2).

Each component group has primary tasks to perform that relate to one of the five system functions. However, each group also performs secondary tasks relating to one or more of the other functions. Figure (1-3) shows the components together with their relationships to each other. These hydraulic components are:-

1. **Hydraulic Power Supply Unit:** The power unit group of components deals primarily with the energy-conversion function of the system. The unit consists of a prime mover, pump, and reservoir. The prime mover is the source of energy for the system. The energy produced by the prime mover turns the pump, which produces oil

flow that transmits energy through the system. The reservoir serves as a storage unit for the oil system, heat dissipation and cleanness of the operating oil.

2. **Hydraulic Power Control Unit:** Three different types of valves are required to perform the oil control function in a hydraulic power system which consist of valves to control oil direction, pressure, and flow rate.
3. **Hydraulic Transmission Unit:** Hydraulic distribution is the primary function of the conductors group (transmission unit) of components. Pipes, hoses, and tubes serve as the conductors that confine and carry hydraulic oil between the reservoir and other system components.
4. **Hydraulic Output Device:** An actuator to convert oil pressure into mechanical force or torque to do useful work. Actuators can either be cylinders to provide linear motion or, hydraulic motors to provide rotary motion.
5. **The Hydraulic Conditioning Group:** The hydraulic conditioning group (the accessories) involves maintaining and conditioning hydraulic system. This requires removal of dirt and moisture from the oil and also, assuring proper operating temperature. Specially designed components are available to perform these tasks. Such as filters, heat exchangers, accumulators etc.

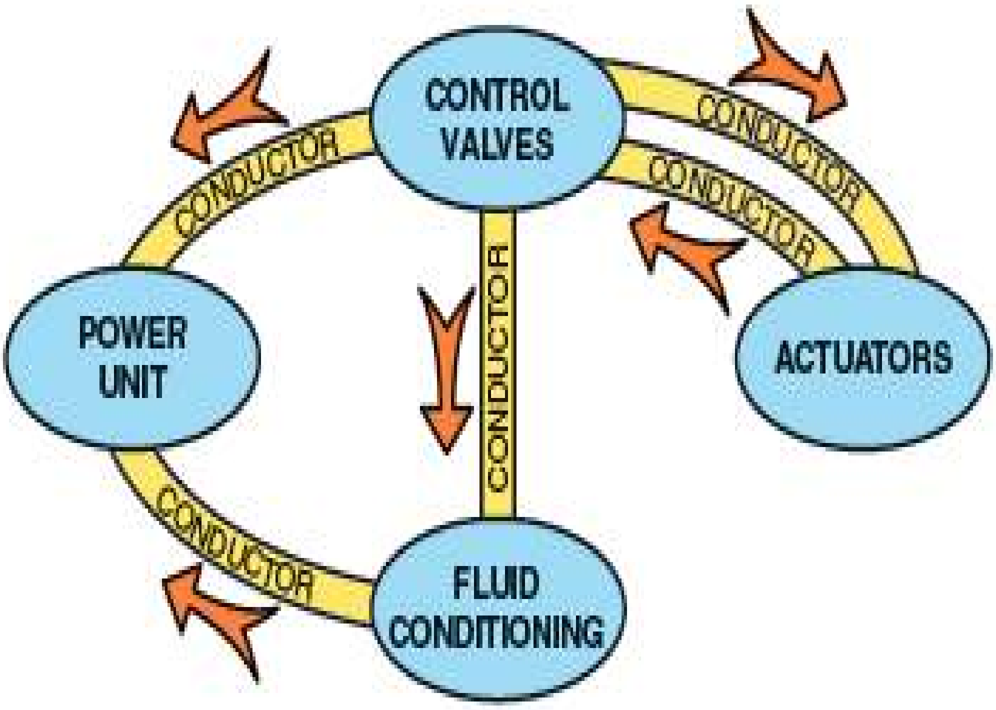


Fig. (1-2): The structure of hydraulic power systems [4].

1. Hydraulic reservoir
2. Hydraulic pump
3. Electric motor
4. Hydraulic valves
5. Conductors
6. Hydraulic actuator

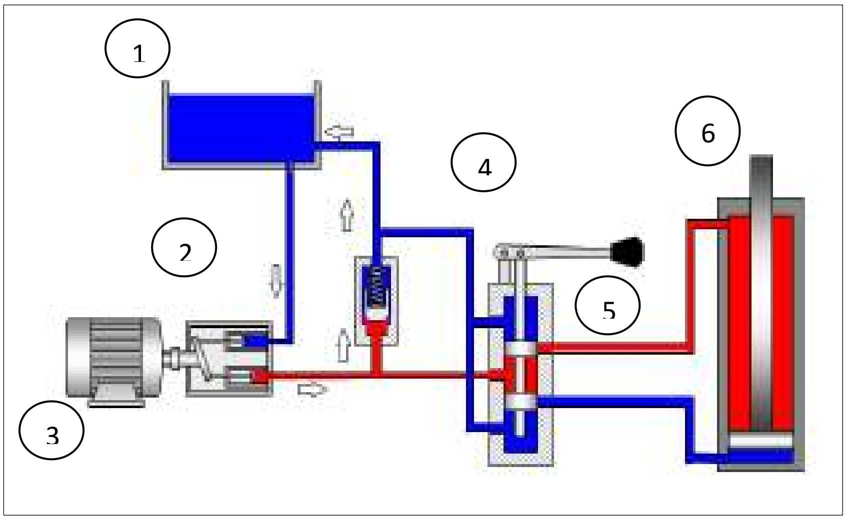


Fig. (1-3) Basic components in a hydraulic system [5].

1.3 Flow Divider Valve Principle

Flow divider valves are designed to divide the flow from a single source to supply two circuits which could be used to operate two actuators at equal speed. These valves are supplied with orifices that give a 50:50 split up to a 90:10 split of flow, generally in increases of 10%. The basic structure of a spool-type divider is composed of a pair of calibrated metering orifices, two centering springs, a compensating spool, and a pair of variable orifices which are adjusted by the movement of the compensating spool. A schematic diagram for the flow divider is shown in Figure (1-4). Equally division ratio flow divider valve is considered in this study.

Since identical metering orifices are used, the division flows, Q_{a1} and Q_{a2} are equal. Flow enters into Port 1 and exits from both, Port 2 and 3. The key component of the flow divider, as with most valves, is the spool. There is a passage drilled down the spool, so that fluid enters the passage and splits to flow along the passage in both directions (to the right and to the left) [6].

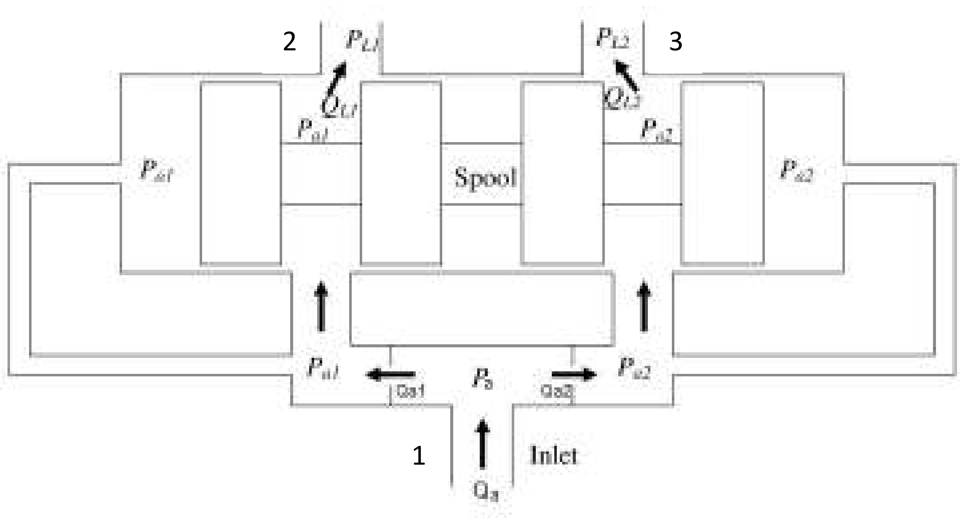


Fig. (1-4) Schematic diagram of flow divider valve [7].

Fluid entering the inlet port goes left and right through the fixed orifices, and then out from outlets 2 and 3. When either outlet encounters more backpressure than the other does, the high-pressure side forces the spool towards the low pressure side until pressures on both sides equalize. Equal pressure drop across both orifices produces equal flow.

For example, considering figure (1-4):

1. If the flow through branch 1(Q_{a1}) increases.
2. P_{a1} will decrease (as P_a is initially unchanged).
3. The pressure imbalance between P_{a1} and P_{a2} will force the spool to the left.
4. Thereby closing the left orifice and opening the right one.
5. Then the flow through branch 1(Q_{a1}) decreases.
6. Which increases P_{a1} and decreasing P_{a2} until equilibrium is regained, at which point, the ratio of outlet flows has been restored.

It should be noted that the right orifice opens as the left orifice closes, increasing its flow such that, although the magnitudes of the flows may have changed, the ratio has not [7].

1.4 Synchronization Motion

A common problem in hydraulic applications is the synchronization of multi cylinders that must match each other's position. Some machines with multiple cylinders require that cylinder strokes should be synchronized perfectly for the machine to operate properly. If all the loads, line sizes and lengths, and friction of the cylinders and machine members are identical, they may stroke synchronously. While line sizes and lengths and machine loading can be controlled to some extent, oil leakage and friction changes momentarily. Thus, when cylinders have to stroke together, some methods must be used to synchronize them.

There are four basic approaches to solve the synchronization problem of multiple actuators. These are:-

- Conventional hydraulic circuit (without flow control valve).
- Hydraulic circuit with flow control valve.
- Flow divider valve circuit.
- Hydraulic circuit with electro-hydraulic valve.

Synchronization by conventional hydraulic circuit is accomplished by using tee connection Figure (1-5). This can be used in the cases of identical loads, separate loads and also, when the accuracy is not important such as when it is used for lubricating purposes.

Flow control valve is used when there is no need to high accuracy and changing the speed of the actuator frequently. However a basic solution just on a manual flow control basis, Figure (1-6) is not ideal. This is owing to the piston movement occurs in function of the instantaneous founded reactions.

In the tee connection and flow control valve circuits even if the cylinders are equal sized and identical loads are used, the effect of the common flow would be that the one that found less opposite force would be the first rising. So, there would be a lag from the cylinder having the maximum opposite force till the one that had the minimum, which would result in an impossible synchronism.

Synchronization by using flow divider valve, Figure (1-7), that keeps the same cylinder velocity by maintaining the same flow rate on the cylinders. This is the simplest one but depends on the flow divider performance.

The most accurate way to synchronize hydraulic cylinders is with electro-hydraulic valves (EHV), such as servo valves as shown in Figure

(1-8). Servo valves independently control each cylinder with electronic position feedback, and compare each actuator's position with all others. This is the most expensive way to synchronize cylinders but the most accurate [8].

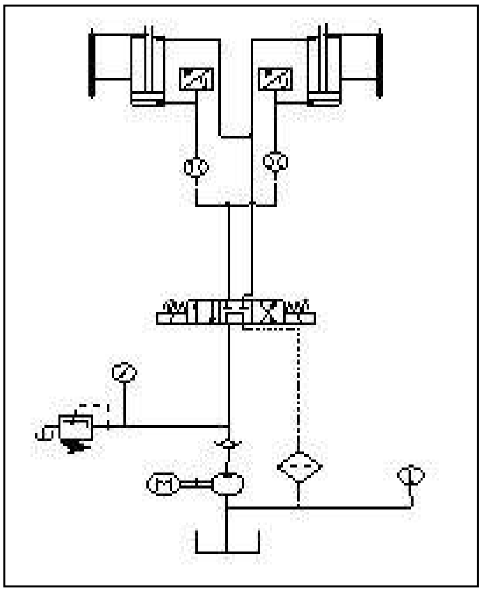


Fig. (1-5) Conventional synchronization circuit.

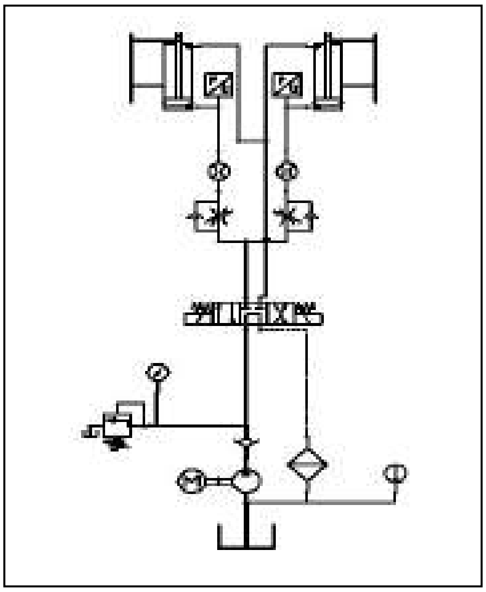


Fig. (1-6) Synchronization circuit with FCV.

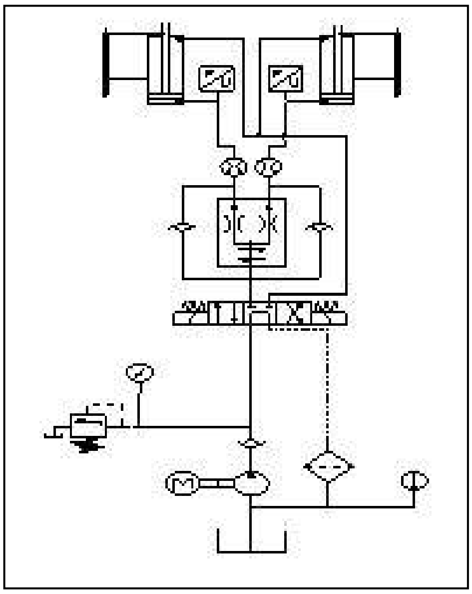


Fig. (1-7) Synchronization circuit with FDV.

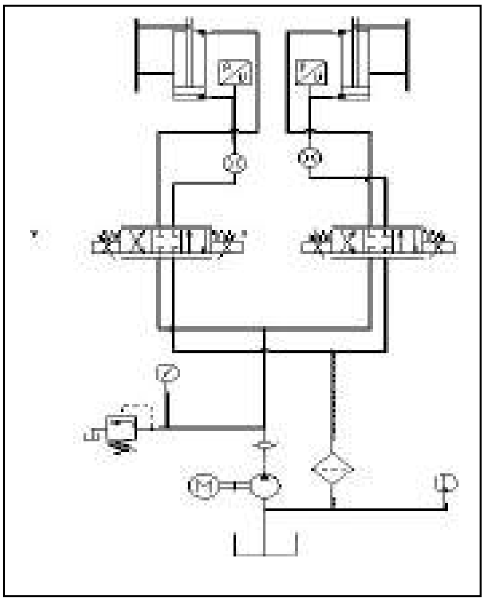


Fig. (1-8) Synchronization circuit with EHV.

1.5 Applications of Synchronization Motion

There are many applications to synchronization motion can be found in the industrial process. These are for example, in the water gates Figure (1-8), in the movable roofs Figure (1-9), in the Movable structures Figure (1-10) and some bridges Figure (1-11)[9].

Fig. (1-8) Radial gates synchronization system.

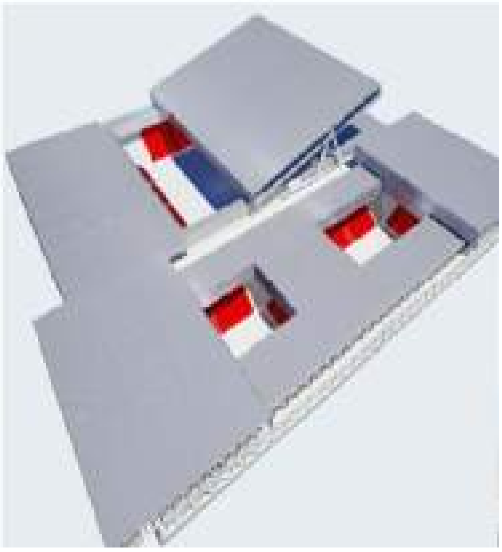


Fig. (1-9) Movable roofs synchronization system

Fig. (1-10) Movable structures synchronization system

Fig. (1-11) Bridges synchronization system

1.6 Aim of the Work

The studies related to the theory of flow divider valve are mature. This is in contract to its applications. So as a result, this topic will be focused on:-

1. **Simulation analysis** which includes:
   - **i.** Derivation of the mathematical model of the basic hydraulic components for the synchronization hydraulic system with flow divider valve and carried them out by using program (Matlab/Simulink package V7.1).
   - **ii.** Simulation of the synchronization hydraulic system using a special program for hydraulic simulation analysis (Automation Studio package V5.2). The Simulation is done under different temperatures and pressures.

2. **Experimental work** which includes the design and analysis of synchronization hydraulic circuits by using conventional, flow control valve and flow divider valve circuits and experiment tests on the test rig for the system at different temperatures and pressures.
3. **Comparison study between the three approaches of synchronization system** which includes the cases of: no load no linkage; no load with linkage; load in the center; off center load and finally with separate load.

Chapter Two

Literature Survey

2.1 Introduction

This chapter reviews and explains some of the previous work related to both, the experimental and theoretical works on the flow divider valve. Also, this chapter reviews and explains some of studies concerned the hydraulic synchronization motion.

2.2 Hydraulic Flow Divider Valve

Kwan (1979) [10]: Presented the first theoretical models which describe the accuracy of the flow divider valve. He showed that, increasing the effective area of the spool will reduce division error.

Chan (1980) [11]: adopted Kwan's basic design concept and discovered that if a thin rim was machined into the orifice, the flow forces could be greatly reduced and this was able to reduce flow division error greatly. The previous design modifications successfully improved the steady state accuracy. As the flow requirements were changed, the error increased consequently; which would not satisfy the all purpose industrial requirements. Moreover, due to the complexity structure of some modifications, their commercial applications would be limited. Therefore, the traditional spool type divider valve is still widely used in hydraulic systems.

Guo (1987) [12]: Both Kwan and Chan had studied the valve dynamic behavior by transforming the nonlinear governing equation into linear equation to simplify the simulation. However, the nonlinearities of the governing equations can not easily be approximated by a linear model without introducing considerable errors. By 1987 digital computers had developed to the point where numerical solutions to the non-linear dynamic equations could be found. Guo (1987) derived a dynamic model of a standard spool type flow divider valve and simulated it by using the Bond Graph technique. He evaluated the valve dynamic performance by the cumulative error. He proved that the steady and dynamic accuracies were strongly dependent on the pressure drops across the fixed metering orifices.

Zhang et al. (1993) [13]: The previous researchers had established that the steady state and dynamic performance of a typical flow divider valve was strongly dependent on the pressure drop across the fixed orifice. They showed that the performance could be improved for low flow rates by using small orifices. However, this meant very large pressure losses at large flows. Zhang et al. presented an "auto-regulator", a pressure controlled pair of orifices that replaced the fixed orifices. Thus a reasonable pressure drop over a wide range of flows had been maintained.

Tae and Tae Young (2000) [14]: The effect of friction force on the dynamic characteristics of a flow divider valve was studied by the researchers. A numerical analysis was carried out to show this effect. The viscous friction force acting on the spool is considered analyzing the viscous flow in the clearance gap between the spool and sleeve. They deduced that, dynamic characteristics are highly affected by the viscous friction force whose magnitude is relatively small compared with other fluid forces. They showed that a thick fluid film lead to low friction. Therefore, they present numerical scheme which can be used generally in designing and performance evaluation of the hydraulic spool valve.

Travis (2004) [7]: proposed an adjustable-ratio flow divider valve. The variable ratio flow divider is similar to the typical fixed-ratio flow divider each consists of two fixed orifices and two variable orifices. The difference between the two is the pilot stage, composed of two hydraulic bridges, which allows the ratio of outlet flows to be changed as shown in Figure (2-1). This valve attempts to split one input flow into two output flows in a predetermined ratio, independent of load pressure or total flow. The ratio of outlet flows can be set by the angular position of the spool, driven by a stepper motor or other low-power input. He also derived complex steady and dynamic models to investigate the effects of the physical parameters. The prototype's performance was experimentally examined.

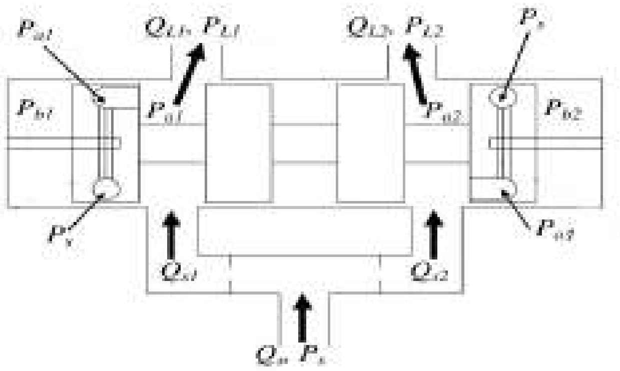


Fig. (2-1) Schematic diagram of an adjustable-ratio flow divider valve.

Norense (2006)[15]: Considered the reaction (flow) force associated with a conventional spool land flow divider valve and one with a rim machined into it, and a modified form of the rimmed land referred to as a "sharp edge tapered rim spool land" as shown in Figure (2-2). The rim and the sharp edge tapered rim were specially designed geometrical changes to the lands of the standard spool. This object was to reduce the large flow forces inherent in the standard spool valve.

Computational Fluid Dynamics (CFD), which was (CFX) program, was used to describe the fluid mechanics associated with the flow forces. As it provided a detailed structure of the flow through the valve, and to identify the flow mechanism whereby flow forces are reduced by the machining of a rim and tapered rim on the land of the spool. He made a comparison study among the three types and deduced important result. The result is the sharp tapered rim valve provides the largest reduction in flow forces.

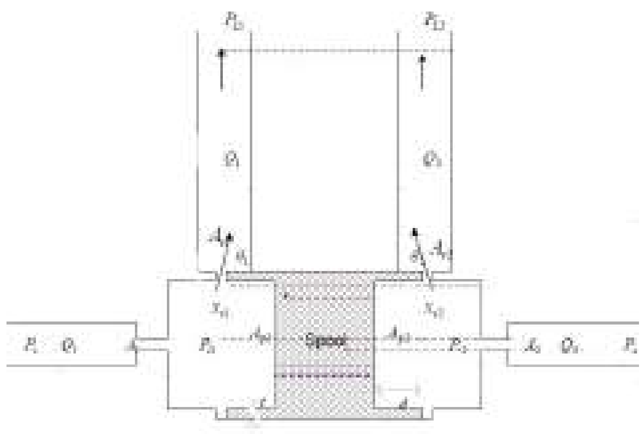


Fig. (2-2) Schematic diagram of a thin rim spool flow divider valve

Minter (2009) [16]: In his study, the steady and dynamic performances of a flow divider valve were simulated numerically by solving the characteristic equations using computer program a FORTRAN. The parameters studied in this research were centering spring constant, compensating spool mass and metering orifice area.

The simulation results show that flow division error increases with increasing the load pressure differential, centering spring constant, and metering orifice area. Even though, decreasing the spring force or the metering orifice area can reduce division error, the spring force still needs to be large enough to overcome the spool static friction and the orifice area cannot be too small to lose energy efficiency.

Dynamic division error increases with increasing load pressure differential and metering orifice area but with decreasing spool mass. The centering spring constant has no obvious effect on the valve dynamic response.

2.3 Hydraulic Synchronization Motion

Edward and Pawel (2000) [17]: Presented an analysis for the control of heavy duty machines unit with hydraulic actuator. They studied the performance for every actuator by analyzing both, the displacement and the pressure of the hydraulic system using flow divider valve with non uniform load. They used variable displacement pump and regeneration circuit. The results, which were presented in their work, allow determining the influence of constructional parameters on flow characteristic and positioning accuracy. Those parameters were: shape and dimension of throttle nozzles, spring stiffness, spool motion damping, and volumes of valve champers. A mathematical model was carried out by Matlab/Simulink program.

Hong and George (2001) [18]: presented a nonlinear control algorithm to address the motion synchronization problem for two cylinders of electro-hydraulic (EH) system. A two step design approach was applied such that it utilized a linear multi input multi output (MIMO) robust control technique to design an outer loop motion synchronization controller. A nonlinear single input single output (SISO) perturbation observer based pressure/force controller is designed for each of the lift cylinder. This is because the inner loop controller handle the nonlinearities associated with the EH actuators. Experimental results on a 2-cylinder system were presented to verify the effectiveness of the proposed approach. The experimental displacement difference is less than 0.3 (mm) under a load of 1587 (kg).

Nicolae et al. (2004) [19]:

Presented the theoretical and experimental researches performed by the authors in order to design a digital control systems for synchronizing hydraulic cylinders. A mathematical approach was developed for obtaining a mathematical model describing the whole control system. The dynamic performance of this system was studied by Matlab / Simulink software.

They studied the evaluation of the two electrohydraulic synchronization methods which they are:

a) The analog servocontrolers of the two servo-systems receive the same reference signal from the control unit. The position error between the two rods is used for generating two additional control signals for the two servocontrolers as shown in Figure (2-3).

b) One of the servo systems is designed as a “leader ”and the other have to follow the movement of the first as shown in Figure (2-4). They concluded that the best dynamic precision is obtained with common reference.

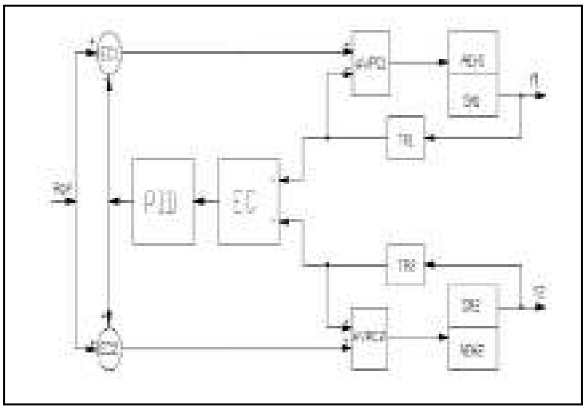


Fig. (2-3) Control diagram with common reference signal

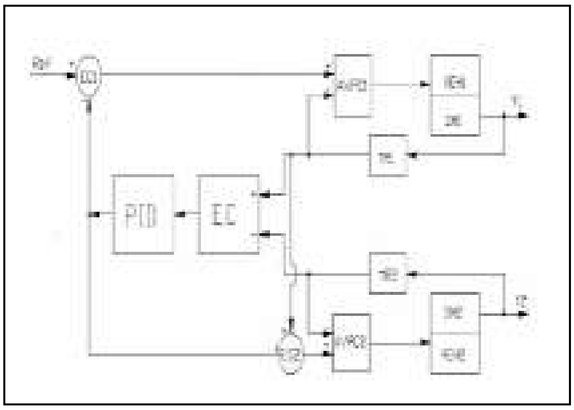


Fig. (2-4) Control diagram with a leader servo system

Hugo (2010) [20]: Discussed the development of an automatic and synchronized system. He begun at actuators connected in parallel and in series, which represents the first and easiest method to achieve hydraulic synchronism. Then, he discussed the synchronization by individual flow control valve on each cylinder and with flow divider valve. Finally, he discussed the control feedback theory in servovalve in the synchronized system and built an experimental system deducing that the (EH) system is the most accurate one. The system which he built allows controlling pressures and movements synchronously and accompanied by position readings measured by position sensors for four actuators. The simulation model was built using the Simhydraulics toolbox from Simulink.

Seong et al. (2011) [21]: presented a method to achieve a synchronous positioning objective for a dual-cylinders electro-hydraulic system with friction characteristics. The control system consists of a VSC (Variable Structure Controller) for each of the hydraulic cylinders and a PID (Proportional-Integral-Derivative) feedback controller. The PID controller is used for controlling the non-synchronous error generated by both cylinders when motion synchronization is carried out. To enhance the position-tracking performance of the individual cylinders, a friction characteristic was simulated based on the estimated friction force. Both, the simulation and experimental results show that the proposed method can effectively achieve the objective of position synchronization in the dual cylinders of electro-hydraulic system.

2.4 Summary

Table (2-1) and Table (2-2) shows summary of studies related with flow divider valve and with hydraulic synchronization system.

Table (2-1): Summary of studies related with flow divider valve

Year	Author	Work	Objective
1979	Kwan	theoretical model	Presented theoretical model which describes the accuracy of a flow divider valve.
1980	Chan	theoretical model	Discovered that if a thin rim was machined into the orifice, the flow forces could be reduced and this was able to reduce flow division error.
1987	Guo	theoretical model	Developed numerical solutions to the non-linear dynamic equations of a typical flow divider valve.
1993	Zhang et al.	theoretical model	Presented an “auto-regulator”, a pressure controlled pair of orifices that replaced the fixed orifices, thereby maintaining a reasonable pressure drop over a wide range of flows
2000	Tae and Tae Young	theoretical model	studied the effect of friction force on the dynamic characteristics of a flow divider valve
2004	Travis	theoretical model& experimental work with matlab	Proposed a new type of hydraulic valve: an adjustable-ratio flow divider valve
2006	Norense	theoretical model with CFX	considers flow force associated with a conventional spool land FDV and one with a rim machined into it and sharp edge tapered rim spool land
2009	Minter	theoretical model with Fortran	The steady and dynamic performances of a flow divider valve are simulated numerically by solving the characteristic equations

Table (2-2): Summary of studies related with hydraulic synchronization system.

Year	Author	Work	Objective
2000	Edward and pawel	theoretical model& experimental work with matlab/ Simulink	Presented an analysis of control of heavy duty machines unit with hydraulic actuator. They studied the performance for every actuator by analysis the displacement and the pressure of the hydraulic system by using (FDV) with non uniform load
2001	Hong and George	Theoretical model and experimental work	Presented a nonlinear control algorithm to address the motion synchronization problem for a 2- cylinder electro-hydraulic (EH) system.
2004	Nicolae et al.	theoretical model& experimental work with matlab/ Simulink	Presented the theoretical and experimental researches performed by the authors in order to design a digital control systems for synchronizing hydraulic cylinders.
2010	Hugo	theoretical model& experimental work with Simhydraulics	discussed the development of an automatic and synchronized system
2011	Seong et al.	theoretical model& experimental work	Presented a method to achieve a synchronous positioning objective for a dual-cylinder electro-hydraulic system with friction characteristics.
2014	Saad S. Saadoon	theoretical model& experimental work with Simulink &AS	Theoretically, presented a mathematical model to the synchronization system carrying it by Simulink and simulate the system by AS. Experimentally, presented a comparison study among conventional, FCV and FDV systems.

Chapter Three

Mathematical Models and Simulations

3.1 Introduction

This chapter is concerned with building Mathematical Models of the hydraulic gear pump, directional control valve, flow divider valve, hydraulic actuators and both, supply and return lines. A control system for measurement instruments has been considered. Also it deals with the Automation Studio and Matlab Simulink programs.

3.2 General Simplifying Assumptions.

During the process of the system modeling the following simplifying assumptions are made:-

1. Both, the internal and external leakages of the hydraulic pump are assumed to be linear functions to the pressure difference across the pump.
2. The inlet of the pump; the return line to the tank and the pressure drain case are all open to atmospheric environment (i.e. their values are zero).
3. There is no slip between the electric motor and the hydraulic pump (i.e. same rotation speed).
4. The hydraulic transmission lines connecting between the system are stiff. Their expansion due to pressure changes is negligible. Consequently, the dynamics effect resulting from the hose elasticity is neglected.
5. Both, Flow divider valve and the system are in thermal equilibrium condition.
6. Resistance of fluid flow in both, valve channels and conduits are negligible.
7. The bodies of valves have large stiffness.

3.3 Mathematical Model of the Hydraulic System.

The well known and quite common hydraulic circuit what can be found in many basic books is consisting of a tank with oil, pump, valve and actuator which are connected with hydraulic transmission lines. For technical hydraulic description of running process, values like pressure and flow are used in special points (especially on the input and output) of these elements. When starting to build a computer model, one must like to have a such model which mirror the shape and connection of physical object. In real object, there are many of hoses and many important values, which are pressure, displacement, flow and temperature. In modeling, those values can be treated as signals which pass through elements and parts of circuit. Nomenclature for the parameters and states is listed on page ix.

3.3.1 The Mathematical Model of Fixed Displacement Pump

Since the rotation speed of the electric motor is equal to the rotation speed of the hydraulic pump, the ideal flow rate of the pump is given by [22]:-

$$Q_{p\ ideal} = D_p . N \qquad (3.1)$$

Where: D_p Volume displacement rate of the hydraulic pump.

N Angular speed of the hydraulic pump.

The actual flow rate of the pump ($Q_{p\ actual}$) is less than the ideal flow rate. This is due to both, the fluid leakage and fluid compression. This is shown in the schematic diagram of the power unit for fixed displacement pump in Figure (3-1). The external gear pump operation is shown in Figure (3-2). the continuity equations for inlet and outlet flow rates can be written in the following form [22].

$$D_p . N - C_{ip} (P_p - P_i) - C_{ep} (P_p - P_r) - \frac{V_p}{\beta} \frac{d(p_p - p_i)}{dt} = Q_{p\ actual} \qquad (3.2)$$

Where: C_{ip}, C_{ep} Internal and external leakage coefficient of the hydraulic pump m^3/sec Pa.

(P_i) and (P_r) inlet and return pressures, are assumed to be zero, equation (3.2) can be written as:-

$$D_p. N - C_{ip} P_p - C_{ep} P_p - \frac{V_p}{\beta}\frac{d P_p}{dt} = Q_{p\ actual} \quad \ldots\ldots(3.3)$$

The leakage flow of the pump (including the internal leakage and the external leakage flow) can be approximated by :-

$$(C_{ip} - C_{ep}) P_p = C_{tp} P_p \quad \ldots\ldots (3.4)$$

Where: C_{tp} Total leakage flow coefficient of hydraulic pump m^3/ sec Pa

Equation (3.3) can be written as:-

$$D_p. N - C_{tp} P_p - \frac{V_p}{\beta}\frac{d P_p}{dt} = Q_{p\ actual} \quad \ldots\ldots (3.5)$$

Equation (3.5) represents unsteady state equation for the fixed displacement pump. The steady state condition is:-

$$D_p. N - C_{tp} P_p = Q_{p\ actual} \quad \ldots\ldots (3.6)$$

or

$$Q_{p\ actual} = \eta_{vp}\ Q_{p\ ideal} = \eta_{vp} D_p. N \quad \ldots\ldots(3.7)$$

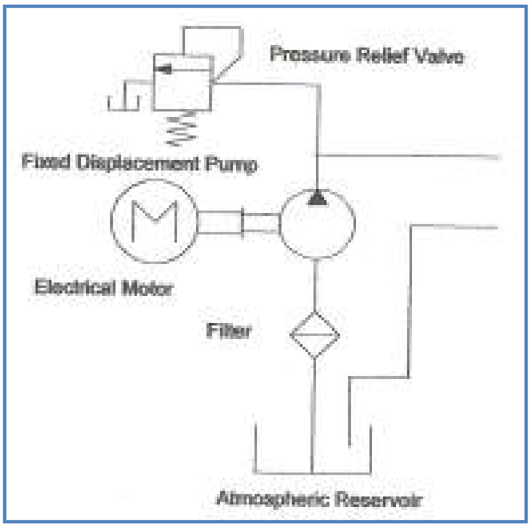


Fig. (3-1) Schematic diagram of power unit.

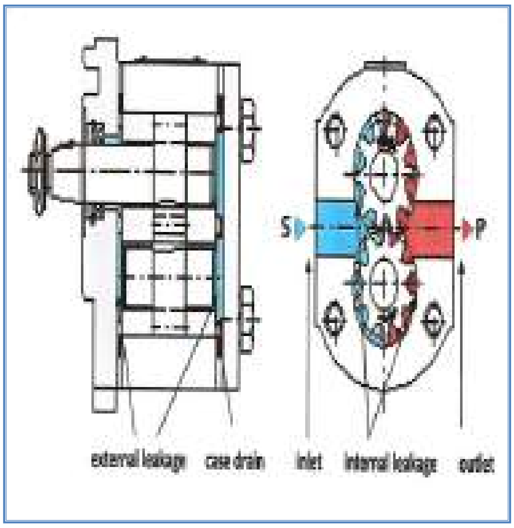


Fig. (3-2) External gear pump operation.

3.3.2 The Mathematical Model of the Pressure Relief Valve

Pressure relief valves are probably the most important of the pressure limiting devices. This is because every pump should have the protection that this valve offers. Pressure relief valve (PRV) automatically creates a passage way between the pressure line and reservoir. This passage will be large enough to divert all the excess fluid supplied by the pump back to the reservoir when the pressure exceeds a chosen value. The equation of motion is given by [23]:

$$Mr\frac{d^2zp}{dt^2} + f_r\frac{dz}{dt} + k_r z = p_S * A_{ref} \quad \text{……..……………………………..(3.8)}$$

Where: Mr Pressure relief valve spool mass, f_r Pressure relief valve viscous factor, Kr Spring constant of PRV

And the flow rate throughout pressure relief valve is given by:

$$Q_{ref} = C_r.A_{ref}\sqrt{\frac{2}{\rho}(P_S - P_o)} \quad \text{………...…………………...…………..(3.9)}$$

Where: C_r Discharge coefficient of PRV, P_o Tank pressure

A_{ref} pressure relief valve orifice area

3.3.3 The Mathematical Model of the Supply Hydraulic Line to Directional Control Valve

The continuity equation is applied to transmission lines between the hydraulic pump and hydraulic actuator. The hydraulic line is shown in Figure (3-3).The equation of flow in any line in synchronization system is given by the following form [24]:-

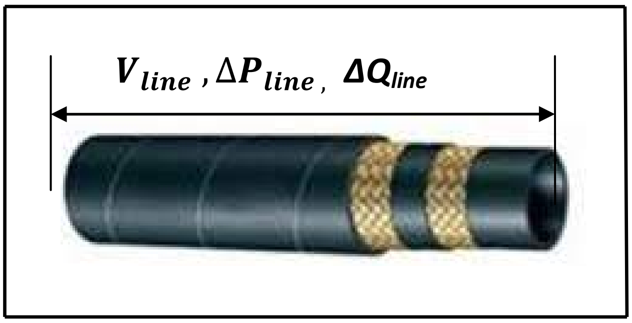


Fig. (3-3): Hydraulic transmission line [22]

$$(ke + \frac{V_{line}}{\beta})\frac{d\,P_{line}}{dt} = \Delta Q_{line} \quad \text{...} \quad (3.10)$$

Where: *ke* Elasticity coefficient of pipe, βBulck modulus of the fluid

Equation (3.10) simplified to:-

$$(\frac{V_{line}}{\beta})\frac{dP_{line}}{dt} = \Delta Q_{line} \quad \text{...} \quad (3.11)$$

Then, the mathematical model of supply hydraulic line to directional control valve is given by:-

$$\frac{dPs}{dt} = \frac{\beta}{Vs}(Qp - Qs) \text{...} (3.12)$$

Whrer: $Q_{p,}$ Q_p pump and supply to DCV flow rate

3.3.4 The Mathematical Model of the Directional Control Valve

Valves are the most important mechanical (or electrical) link to the fluid interface in hydraulic systems. The flow through the valve is described in the equation of flow through the orifice which takes into account the pressure drop direction [25].Simplified-functional diagram of the valve is presented in Figure (3-3).

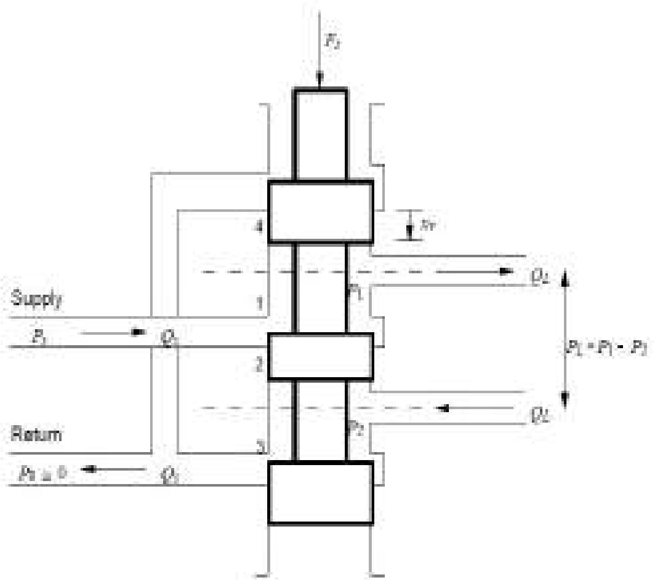


Fig. (3-4) Typical three-land-four-way spool valve [15].

Referring to Figure (3-4), for steady-state flow analysis:

$Q_L = Q_1 - Q_4$.. (3.13)

$Q_L = Q_3 - Q_2$.. (3.14)

Flow through the valve orifices is described by the orifice equation [26]. Therefore;

$$Q_1 = C_d . A_1 \sqrt{\frac{2}{\rho}(p_s - p_1)} \quad (3.15)$$

$$Q_2 = C_d . A_2 \sqrt{\frac{2}{\rho}(p_s - p_2)} \quad (3.16)$$

$$Q_3 = C_d . A_3 \sqrt{\frac{2}{\rho}(p_2 - p_o)} \quad (3.17)$$

$$Q_4 = C_d . A_4 \sqrt{\frac{2}{\rho}(p_1 - p_o)} \quad (3.18)$$

Assume that the valve orifices are matched and symmetrical, which is the case for most of the manufactured spool valves [27]. Matched orifices mean that

$A_1 = A_3,\ \ A_2 = A_4,$

And symmetrical orifices mean that

$A_1(x_v) = A_2(-x_v),\ A_3(x_v) = A_4(-x_v).$

Matched and symmetrical orifices indicate that

$$A_1(x_v) = A_3(x_v) = A_2(-x_v) = A_4(-x_v) = A(x_v), \quad (3.19)$$

Thus, only one orifice area (A) needs to be defined. If the orifice areas are linear with valve stroke, only one defining parameter is required: the width of the slot in the valve sleeve (w). w is the rate of change of orifice area with stroke, and is called the area gradient. The relationship between w and A is given by

$$A = w.\ x_v \quad (3.20)$$

If the orifices are matched and symmetrical, then

$$Q_2 = Q_4 \quad (3.21)$$

$$Q_1 = Q_3 \quad (3.22)$$

Substituting (3.15) and (3.17) into (3.22) yields

$$P_S = P_1 + P_2 \quad (3.23)$$

Which together with the definition of P_L,

$$P_L = P_1 - P_2 \quad (3.24)$$

Yielding:-

$$P_1 = (P_S + P_L)/2 \quad (3.25)$$

$$P_2 = (P_S - P_L)/2 \quad (3.26)$$

Furthermore, assume that the valve geometry is ideal, i.e., the orifice edges are perfectly square with no rounding and there is no radial clearance between the spool and the sleeve. As a result, the leakage flows (Q_2 and Q_4 when x_v is positive, Q_1 and Q_3 when x_v is negative) are zero. Then combining (3.13), (3.15) and (3.25) yields:-

$$Q_L = C_d . A_1 \sqrt{\frac{1}{\rho}(p_s - p_L)} \text{ for } x_v > 0 \quad \ldots\ldots (3.27)$$

For a negative valve displacement Q_L = -Q_4, and (3.18) and (3.25) gives:-

$$Q_L = -C_d . A_2 \sqrt{\frac{1}{\rho}(p_s + p_L)} \text{ for } x_v < 0 \quad \ldots\ldots (3.28)$$

Since the valve is symmetrical and assume that rectangular ports are used with an area gradient w, then (3.27) and (3.28) can be combined to give.

$$Q_L = C_d . A_1 \sqrt{\frac{1}{\rho}(p_s - p_L)} - C_d . A_2 \sqrt{\frac{1}{\rho}(p_s + p_L)} \quad \ldots\ldots (3.29)$$

And in same way Q_s can be obtained as follows:-

$$Q_s = C_d . A_1 \sqrt{\frac{1}{\rho}(p_s - p_L)} + C_d . A_2 \sqrt{\frac{1}{\rho}(p_s + p_L)} \quad \ldots\ldots (3.30)$$

The equation of spool motion is given by:-

$$M_f \frac{d^2 x_v}{dt^2} + f_f \frac{dx_v}{dt} + k_f . x_v = Ff \quad \ldots\ldots (3.31)$$

Where:M_f spool mass of FCV, f_f FCV viscous factor k_f FCV spring constant

3.3.5 The Mathematical Model of the Supply Hydraulic Line to Flow Divider Valve

Fluid flowing to flow divider valve with volumetric flow rate Q_a, is divided into two fluxes with volumetric flow rates Q_{a1}, and Q_{a2}. The equation of flow balance is [6]:

$$\frac{dPa}{dt} = \frac{\beta}{Va}(Q_a - Q_{a1} - Q_{a2}) \quad \ldots\ldots (3.32)$$

3.3.6 The Mathematical Model of the Flow Divider Valve

Simplified, functional diagram of the valve is presented in Figure (3-5)

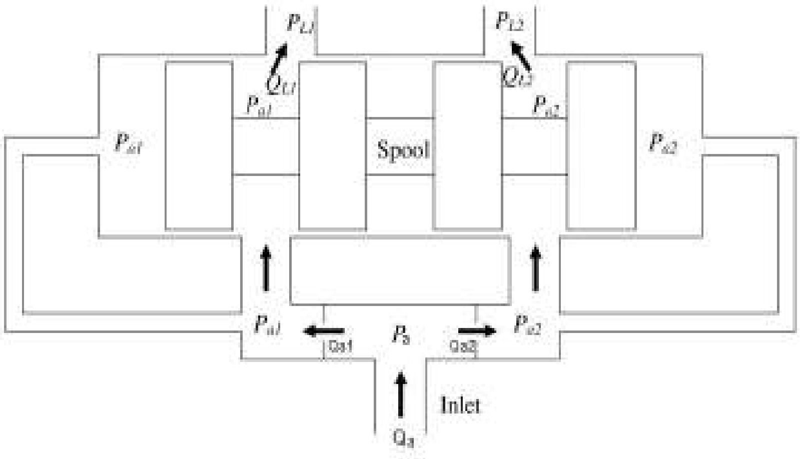


Fig. (3-5) Schematic diagram of flow divider valve [7].

Referring to Figure (3-5) the following equations represent the mathematical model for flow divider valve [16, 17].

The flow through the fixed orifices is given by equations 3.33 and 3.34.

$$Q_{a1} = C_{do1}.A_{o1}\sqrt{\frac{2}{\rho}(P_a - P_{a1})} \quad (3.33)$$

$$Q_{a2} = C_{do2}.A_{o2}\sqrt{\frac{2}{\rho}(P_a - P_{a2})} \quad (3.34)$$

The flow through the variable orifices is described by equations 3.35 and 3.36.

$$Q_{L1} = C_{dv1}.A_{v1}\sqrt{\frac{2}{\rho}(P_{a1} - P_{L1})} \quad (3.35)$$

$$Q_{L2} = C_{dv2}.A_{v2}\sqrt{\frac{2}{\rho}(P_{a2} - P_{L2})} \quad (3.36)$$

The intermediate pressures (between the fixed and variable orifices) are given by equations 3.37 and 3.38. These pressures are channeled directly to the spool ends. So, the intermediate pressures are given by:-

$$\frac{dPa1}{dt} = \frac{\beta}{Va1}(Q_{a1} - Q_{L1}) \quad \text{..} (3.37)$$

$$\frac{dPa2}{dt} = \frac{\beta}{Va2}(Q_{a2} - Q_{L2}) \quad \text{..} (3.38)$$

And now, the mathematical model of flow divider valve spool motion. The force balance equation for the spool's linear motion is given by equation 3.45. This includes forces due to the end pressures, flow forces and viscous friction. The flow force is not due to the classical form of Bernoulli forces, but is due to the pressure profile acting on the area of the spool face.

The following forces act on spool with mass M_s: hydrostatic forces, hydrodynamic forces, spring forces and viscous friction force between bush and spool, as shown in Figure (3-6) [17].

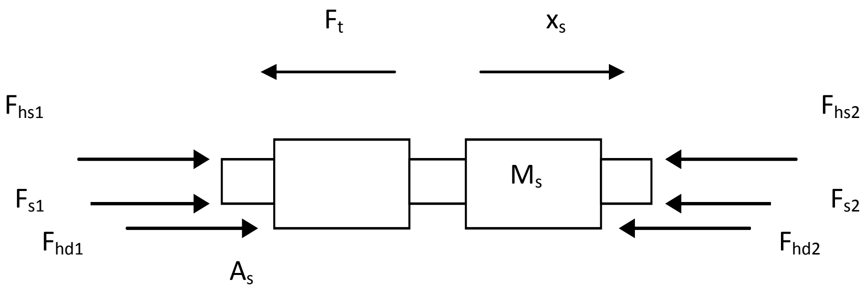


Fig. (3-6) Forces acting on spool valve.

$$\Delta F_{hs} = F_{hs1} - F_{hs2} = p_{a1}.A_s - p_{a2}.A_s \quad \text{..} (3.39)$$

$$\Delta F_S = F_{s1} - F_{s2} = -2\, k_s.\, x_s \quad \text{..} (3.40)$$

$\Delta F_{hd} = F_{hd1} - F_{hd2}$ .... (3.41)

$F_{hd1} = 0.36 Q_{L1}\sqrt{2\rho(p_{a1} - p_{L1})}$ .... (3.42)

$F_{hd2} = 0.36 Q_{L2}\sqrt{2\rho(p_{a2} - p_{L2})}$ .... (3.43)

$F_t = f_s \frac{dxs}{dt}$ .... (3.44)

$Ms\frac{d^2xs}{dt^2} + f_s\frac{dxs}{dt} + 2k_s.x_s = \Delta F_{hs} + \Delta F_{hd}$ .... (3.45)

Whrer: Ms FDV spool mass, f_s FDV viscous factor, k_s FDV spring constant

It is useful to derive equation (3.42), for a fluid flowing between two locations (p_{a1}) and (p_{L1}) as shown in Figure (3.5), for there to be a change in velocity between sections p_{a1} and p_{L1}, a force must be applied to the fluid. Newton's Second Law indicates that force is required to achieve a change in momentum. In general

$F_{hd} = \dot{m}.v_{L1} - \dot{m}.v_{a1}$ ....(3.46)

The magnitude of the velocity downstream of the orifice can be considered to be much greater than the upstream velocity. Thus:

$F_{hd} = \dot{m}.v_{L1}$ ....(3.47)

Applying Bernoulli equation and ignoring change in height yields:

$\frac{v_{a1}^2}{2} + \frac{p_{a1}}{\rho} = \frac{v_{L1}^2}{2} + \frac{p_{L1}}{\rho}$ .... (3.48)

Again applying the" up and downstream" conditions will give:-

$v_{L1} = \sqrt{\frac{2}{\rho}(p_{a1} - p_{L1})}$ ....(3.49)

But the mass flow rate is:-

$$\dot{m} = \rho . Q_{L1} \quad \text{.......(3.50)}$$

From equations (3.47) and (3.50):-

$$F_{hd} = (\rho . Q_{L1}) . v_{L1} \quad \text{.......(3.51)}$$

From equations (3.49) and (3.51) yields:-

$$F_{hd} = (\rho . Q_{L1}) \sqrt{\frac{2}{\rho}(p_{a1} - p_{L1})} \quad \text{.......(3.52)}$$

The angle at which the stream is discharged from the orifice will vary according to the orifice geometry and it has been considered 69^{o}. Thus the axial force required to change the momentum of the fluid passing through the valve orifice is [28]:-

$$F_{hd} = Q_{L1}\sqrt{2\rho(p_{a1} - p_{L1})}\ cos\theta \quad \text{.......(3.53)}$$

Where: θ is the angle at which the stream is discharged from the orifice in degree.

3.3.7 Flow Division and Cumulative Error

Flow division error with ratio of output flows 50-50 is given by [6, 29]:-

$$\%E_{ss} = \frac{Q_{L1} - Q_{L2}}{Q_a} \times 100\% \quad \text{.......(3.54)}$$

Flow division error is proportional to the pressure drop across the fixed orifice.

It is noticed that when a flow divider is used to synchronize two hydraulic actuators, there can be a significant positional error accumulated before the system reaches steady state. This is due to the large temporary flow dividing error that occurs before the valve can compensate. This cumulative error is an important measure of flow divider performance. In other words, the position error accumulated by the load in the time it takes the valve to respond to a disturbance, for a 50:50 divider ratio is defined as [7]:-

$$E_{cu} = \int_0^{t_{cum}} (Q_{L2} - Q_{L1}) dt \qquad (3.55)$$

3.3.8 The Mathematical Model of the Supply Hydraulic Line to Actuators

To cylinders line with adequate volumes flow fluid of volumetric flow rate $Q_{L1}$and Q_{L2} from flow divider valve, whilst the quantity of fluid flowing out is dependent on both, piston rod velocity and areas of pistons. It is described by equation [22]:-

$$\frac{dPL1}{dt} = \frac{\beta}{V_{L1}} (Q_{L1} - A_p \frac{dxp1}{dt}) \qquad (3.56)$$

$$\frac{dPL2}{dt} = \frac{\beta}{V_{L2}} (Q_{L2} - A_p \frac{dxp2}{dt}) \qquad (3.57)$$

3.3.9 The Mathematical Model of Double Acting Single Rod Hydraulic Cylinder

Double acting hydraulic cylinders have two opposing effective areas which are of different sizes. They are fitted with two ports (Port1 & Port2) which are isolated from each other. By feeding oil via ports (Port1 or Port2), the piston may transfer pulling and pushing forces in both stroke directions as shown in Figure (3-7) hydraulic cylinder schematic diagram [30,31].

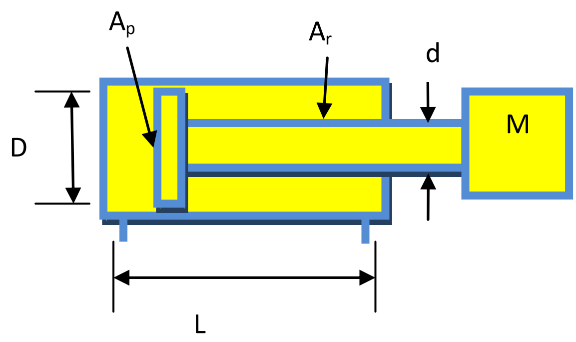


Fig. (3-7) Hydraulic cylinder schematic diagram.

$$M\frac{d^2x_{p1}}{dt^2} + f_p\frac{dx_{p1}}{dt} = \left(P_{L1}A_p - P_{r1}A_r\right) \quad (3.58)$$

$$M\frac{d^2x_{p2}}{dt^2} + f_p\frac{dx_{p2}}{dt} = \left(P_{L2}A_p - P_{r2}A_r\right) \quad (3.59)$$

As a general rule, the circular area [A_p] is calculated from the diameter [D] using the following formula [32]:

$$A_p = \frac{\pi}{4}(D)^2 \quad (3.60)$$

And the area at the rod end is calculated from the following formula:

$$A_r = \frac{\pi}{4}(D-d)^2 \quad (3.61)$$

3.3.10 The Mathematical Model of the Return Hydraulic Lines from Actuators [33]

$$\frac{dPr1}{dt} = \frac{\beta}{V_{r1}}(Q_{r1} - Q_{c1}) \quad (3.62)$$

$$\frac{dPr2}{dt} = \frac{\beta}{V_{r2}}(Q_{r2} - Q_{c2}) \quad (3.63)$$

$$Q_{r1} = A_r\frac{dxp1}{dt} \quad (3.64)$$

$$Q_{r2} = A_r\frac{dx_{p2}}{dt} \quad (3.65)$$

3.4 Control System for Measurement Instruments

The primary objectives of this section are to show the required components for monitoring and analyzing the pressure, flow rate and displacement of the hydraulic system. A control system has been designed and manufactured for this purpose. It consists of three main parts; pressure, flow rate and position sensors, control board, and data show.

3.4.1 Measurement Instruments

These include pressure, flow rate and position sensors that give necessary information about the synchronization hydraulic system behavior.

I. Pressure Sensor

The function of a pressure transmitter is to measure and convert pressure into an analog or digital electrical signal, from which a reading can ultimately be derived. The operating principles of the pressure sensors are similar, there is a silicon chip mounted inside a reference chamber. On one side of the chip is a reference pressure. This reference pressure is either a perfect vacuum or a calibrated pressure, depending on the application. On the other side is the pressure to be measured. The silicon chip changes its resistance with the changes in pressure. When the silicon chip flexes, Figure (3-8), with the change in pressure, the electrical resistance of the chip will change. This change in resistance alters the voltage signal, as the pressure rises, the voltage signal increases. Finally, the output signal is transferred to the control board [34].

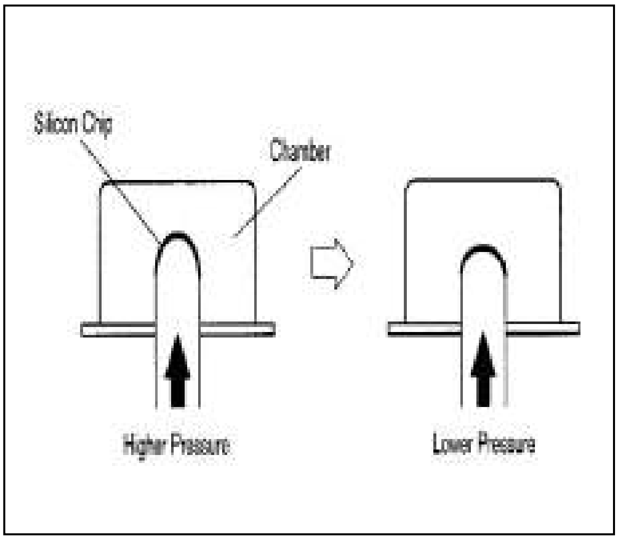
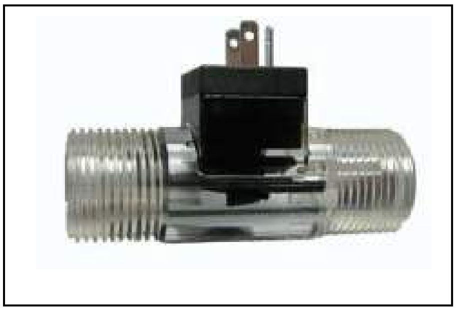


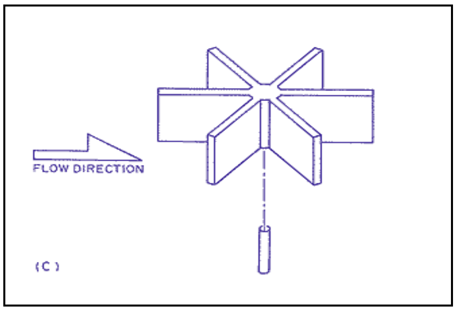
Fig.(3-8,a) Photo of pressure sensor

Fig.(3-8,b)Pressure sensor principle working[34]

II. Flow Sensor

The Principle working of the flow sensor is that the flow causes the bladed rotor of the flow sensor to turn at an angular velocity directly proportional to the velocity of the fluid measured. As the blades pass beneath a magnetic pickup coil, a frequency signal is generated. Each pulse is equivalent to a discrete volume of fluid. The frequency pulse is directly proportional to both, the rotor angular velocity and the flow rate. The waves are readily transmitted to a local or remote electrical instrumentation as shown in Figure (3-9). Finally, the output signal is transferred to the control board [35].





Fig(3-9,a) Photo of flow sensor

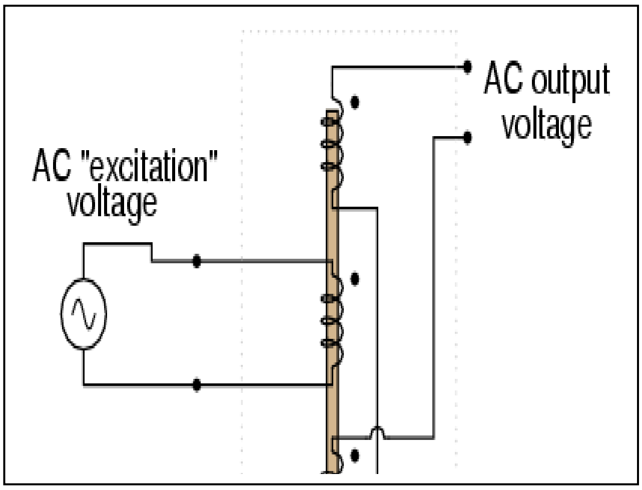
Fig(3-9,b) flow sensor principle working[35]

III. Position Sensor(LVD)

LVDTs are accurate transducers which are often used in industrial and scientific applications to measure small displacements. The letters LVDT are an acronym for Linear Variable Differential Transformer. This is a common type of electromechanical transducer that can convert the rectilinear motion of an object to which it is coupled mechanically into a corresponding electrical signal.

An LVDT consists of a central primary coil wound over the whole length of the transducer and two outer secondary coils. A magnetic core is able to move freely through the coil. The primary windings are energized with a constant amplitude AC signal. This produces an alternating magnetic field which induces a signal into the secondary windings. The strength of the signal is dependent on the position of the core in the coils. When the core is placed in the centre of the coil the output will be zero. Moving the coil in either direction causes the signal to increase and the output signal is proportional to the displacement as shown in Figure (3-10). The output signal is then transferred to the control board [36].

Fig(3-10,a)Photo of position sensor



Fig(3-10,b) Position sensor principle working[36]

3.4.2 Control Board

The control board includes electronic parts such as microcontroller, frequency-to-voltage convertor, USB UART CLIC and other parts as illustrated in Figure (3-11) and Figure (3-12). A current adaptor supplies the board with DC 5V for operation.

I. Microcontroller

Microcontrollers are "special purpose computers" which are dedicated to one task and run one specific program through CPU (central processing unit) that executes programs. A microcontroller type **PIC16F877A** has been used. It considers the main brain of the control system which controls all operations and data acquisition.

The microcontroller has a programming memory of 8 Kbytes. Also a data memory of 368 Bytes microcontroller has been customized to understand the BASIC programming language. To transfer the data with the other devices, a 5 I/O ports, handles both input and output

The pressure and position sensors signals are received by an analog-to-digital converter (abbreviated ADC) which is a device which converts a continuous physical quantity (usually voltage) to a digital signal that represents the quantity's amplitude., 2 outputs pins for the start button, 2 output pins to send the digital signal to the USB UART CLICK [37].

II. USB UART CLICK

It is a device that transmits the data from the microcontroller and sends them to the personal computer.

III. Frequency -to- Voltage Converter (FVC)

Because of the output of the flow sensors are a frequency signals, a two Frequency -to- Voltage Converter has been used. FVC voltage is linearly proportional to the control frequency [38].

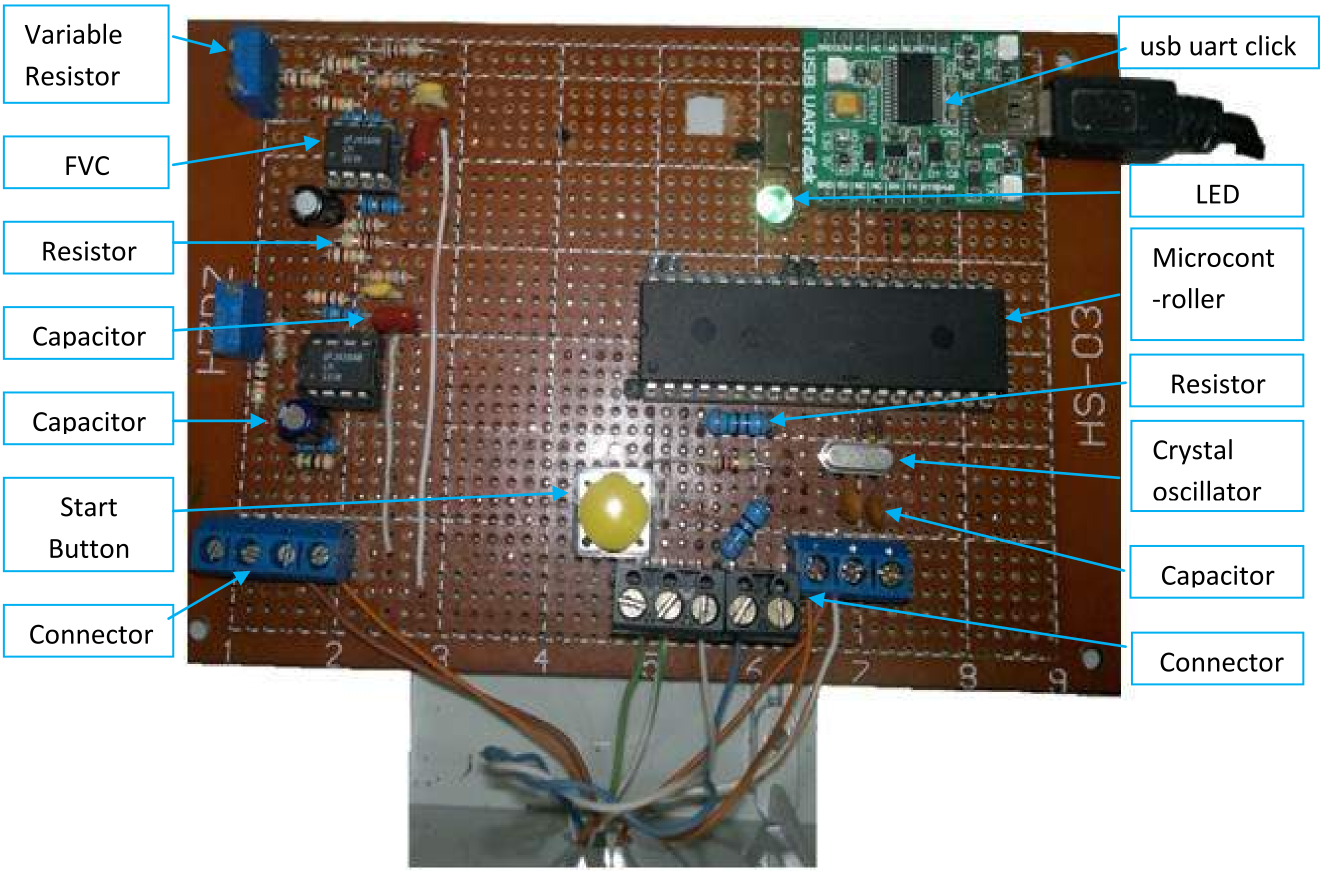
3.4.3 Principle Operation of Control System

As the prime mover is energized, the hydraulic pump rotates in angular speed equal to that of the prime mover. Now, the tandem directional control valve diverts the oil into the reservoir. Before actuation the DCV, the Start Button should be press to record the data. The hydraulic oil will then be pumped through the transmission lines to the actuators where the flow and pressure sensors are installed and immediately sensed the flow rate and the pressure. When the hydraulic cylinders begin to rise, the position sensors, which are mechanically connected to the cylinders, also rise and sense the displacement.

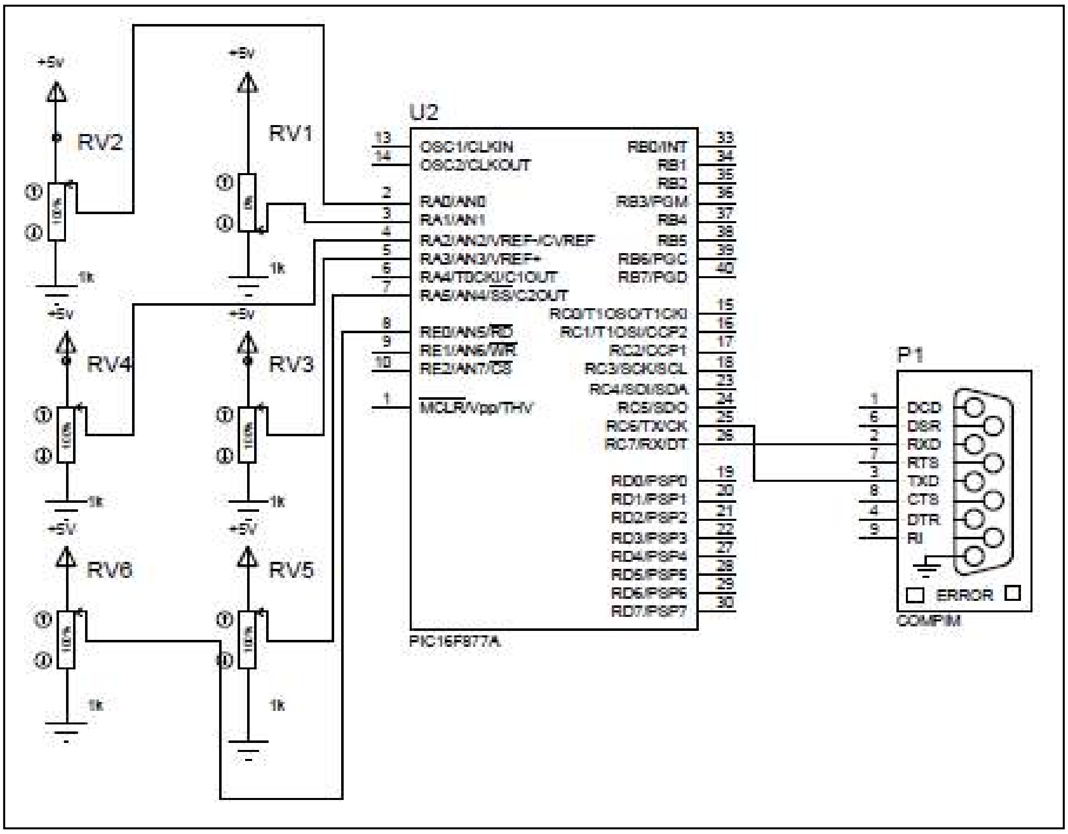
The position and pressure sensors directly send their signal to the ADC in the Microcontroller since their output signal is voltage. While the flow sensors send their signal to the FVC since their output signal is frequency, and then back to the ADC.

After converting the physical quantity to a digital number, the resulting signal will be sent to the USB UART CLICK. By which the computer receives the data and show them momentarily in the screen as shown in Figure (3-13). In addition, save the data to plot them in special software program.

The microcontroller code has been written by mickro C program.



Fig(3-11) Control board photo



Fig(3-12) Control board schematic diagram

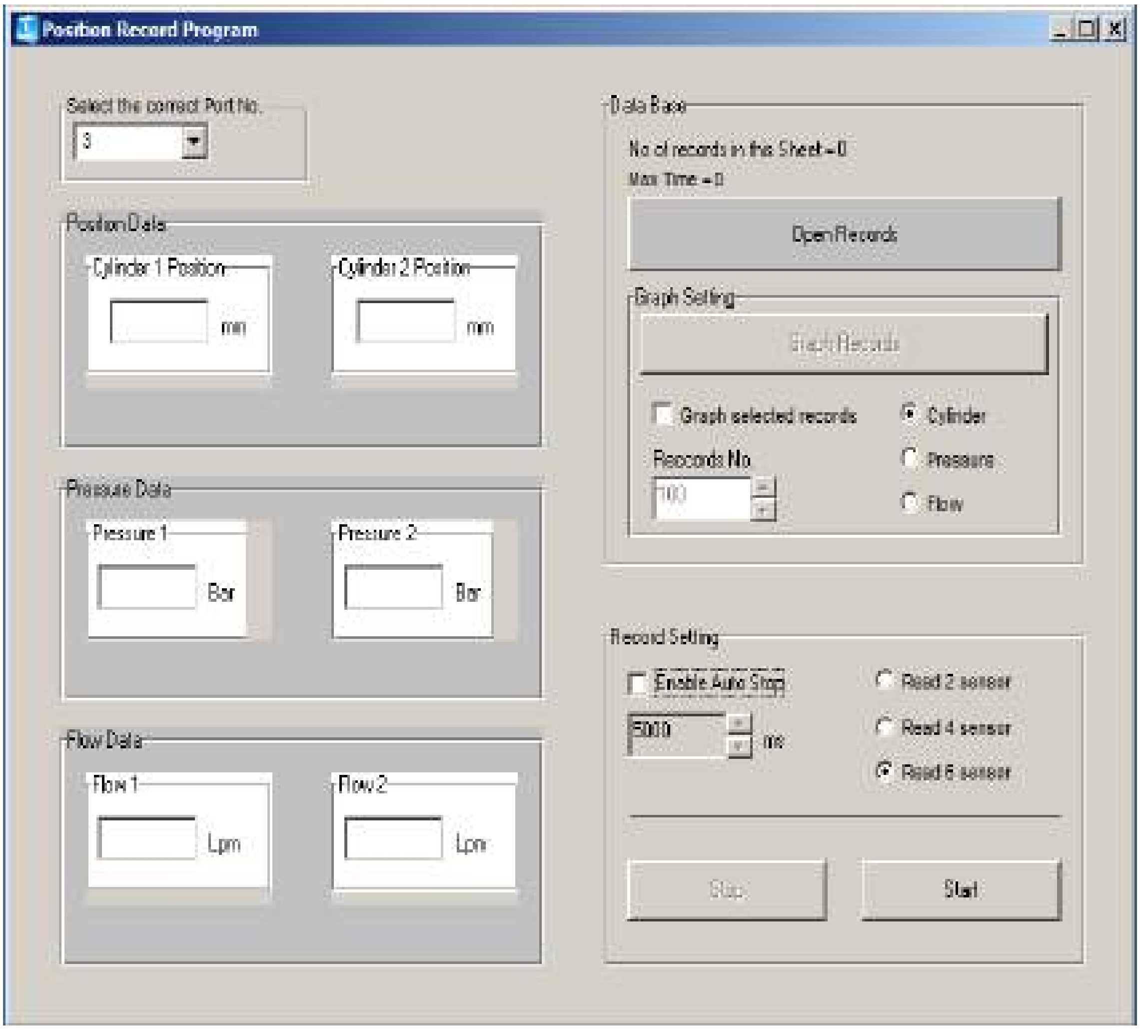


Fig. (3-13) The window of the GUI software package.

3.5 MATLAB SIMULINK

MATLAB, which stands for MATrix LABoratory, is a technical computing environment for high performance numeric computation and visualization.

Simulink is a software package for modeling, simulating, and analyzing dynamical systems .It supports linear and nonlinear systems, modeled in continuous time, sampled time, or a hybrid of the two.

For modeling, Simulink provides a graphical user interface (GUI) for building models as block diagrams, using click-and-drag mouse operations. With this interface, the models can be drawn just as we would with pencil and paper. This is a far from previous simulation packages that require to formulate differential equations in a language or program. Simulink includes a comprehensive block library of sinks, sources, linear and nonlinear components, and connectors. We can also customize and create our own blocks [39].

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

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

MATLAB and Simulink are integrated, so our models can be simulated, analyzed and revise in either environment at any point. An instant access to all of the analysis tools in Matlab can be obtained. So the results can be analyzed and visualized [40, 41].

3.6 Simulink Model for the Hydraulic System Units

Following, the Simulink model to the hydraulic system units in detail starting from figure (3-15) to figure (3-27). The governing equations are mentioned together with the Simulink model that related to it. The mathematical model was carried out with Matlab/ Simulink version 7.1 on compatible PC with Windows 7 operating system. The values of the synchronization system parameters used in the model were mentioned in appendix C.

Simulink modeling of the hydraulic system with flow divider valve is shown in Figure (3-27) while Figure (3-14) shows the structure of synchronization system model in Simulink.

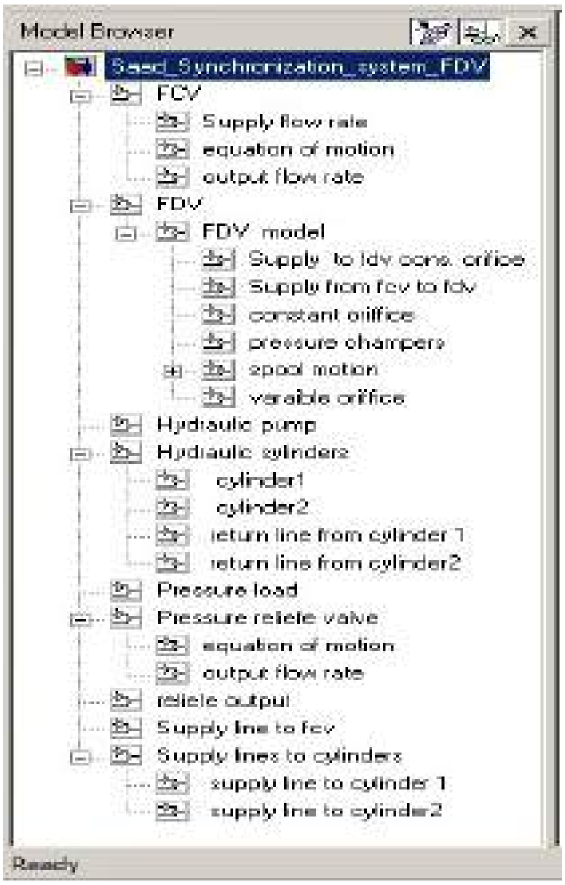


Fig. (3-14) Structure of synchronization system model in Simulink.

$$Q_{p\ ideal} = D_p . N \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.1)$$

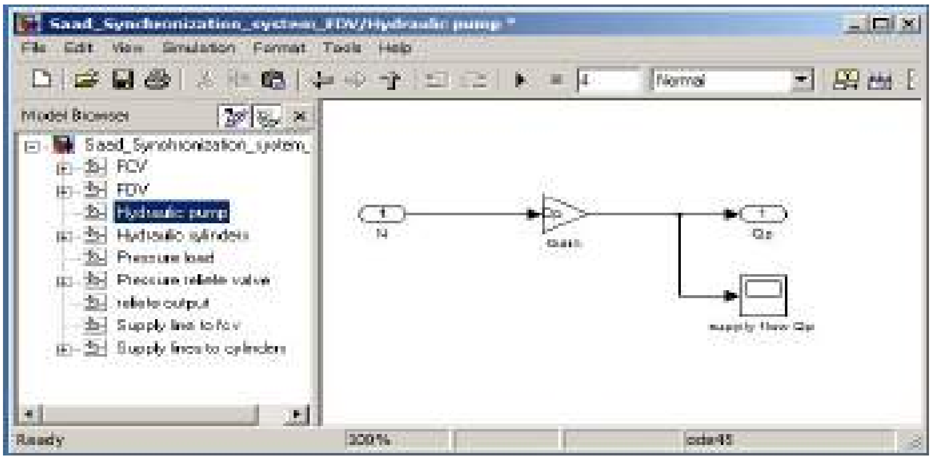


Fig. (3-15) Simulink model for hydraulic pump.

$$Mr\frac{d^2zp}{dt^2} + f_r\frac{dz}{dt} + k_r z = p_S * A_{ref} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.8)$$

$$Q_{ref} = C_r . A_{ref}\sqrt{\frac{2}{\rho}(P_S - P_o)} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.9)$$

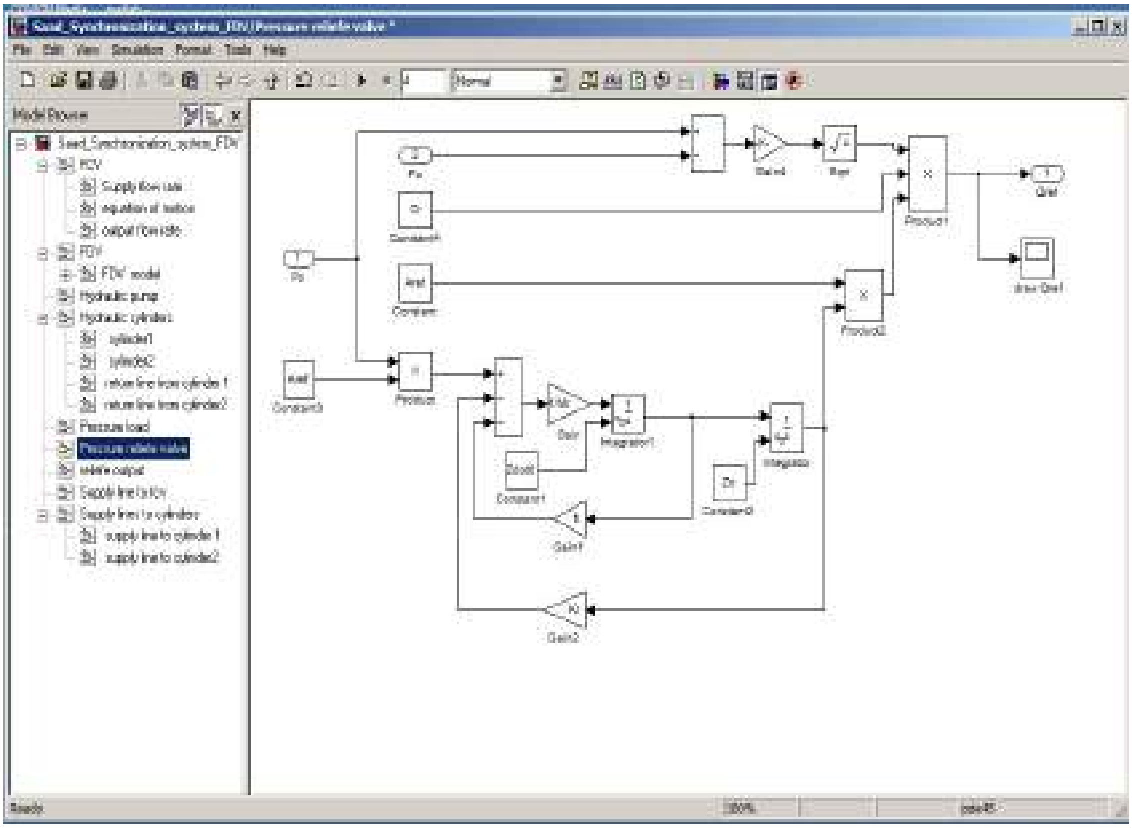


Fig. (3-16) Simulink model for PRV.

$$\frac{dPs}{dt} = \frac{\beta}{Vs}(Qp - Qs) \qquad \text{......(3.12)}$$

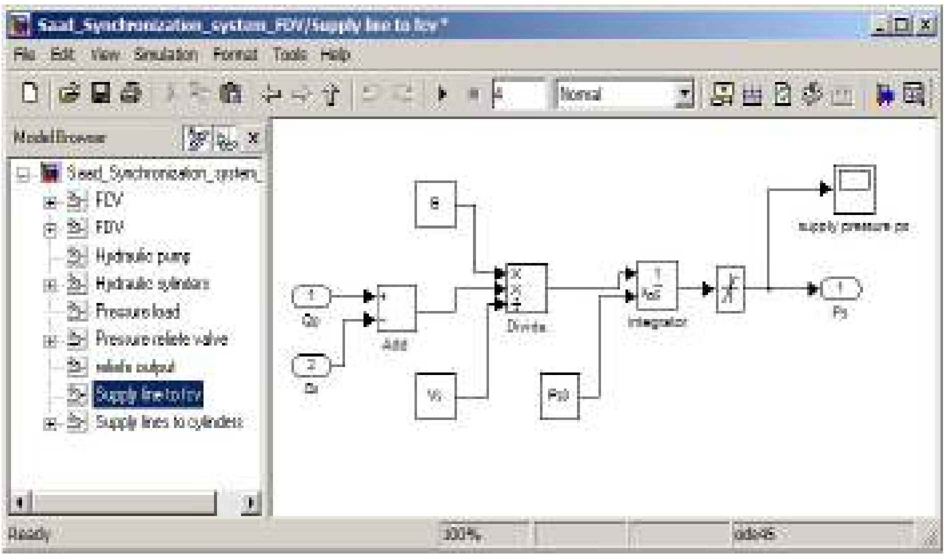


Fig. (3-17) Simulink model for the supply line to FCV.

$$Q_L = C_d.A_1\sqrt{\frac{1}{\rho}(p_s - p_L)} - C_d.A_2\sqrt{\frac{1}{\rho}(p_s + p_L)} \qquad \text{......(3.29)}$$

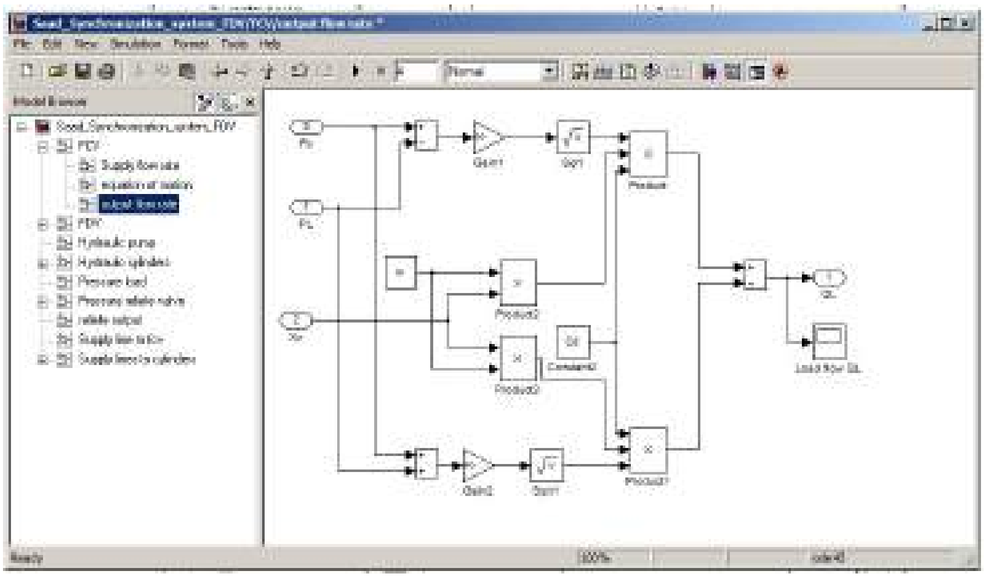


Fig. (3-18) Simulink model of output flow rate for FCV.

$$Q_s = C_d . A_1 \sqrt{\frac{1}{\rho}(p_s - p_L)} + C_d . A_2 \sqrt{\frac{1}{\rho}(p_s + p_L)} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.30)$$

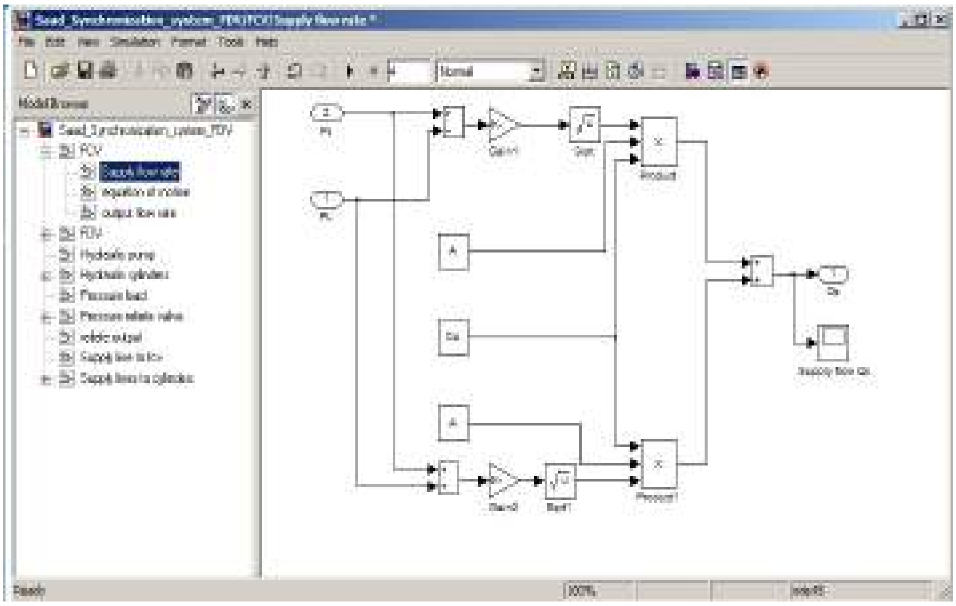


Fig. (3-19) Simulink model for supply flow rate to FCV.

$$M_f \frac{d^2 x_v}{dt^2} + f_f \frac{dx_v}{dt} + k_f . x_v = Ff \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.31)$$

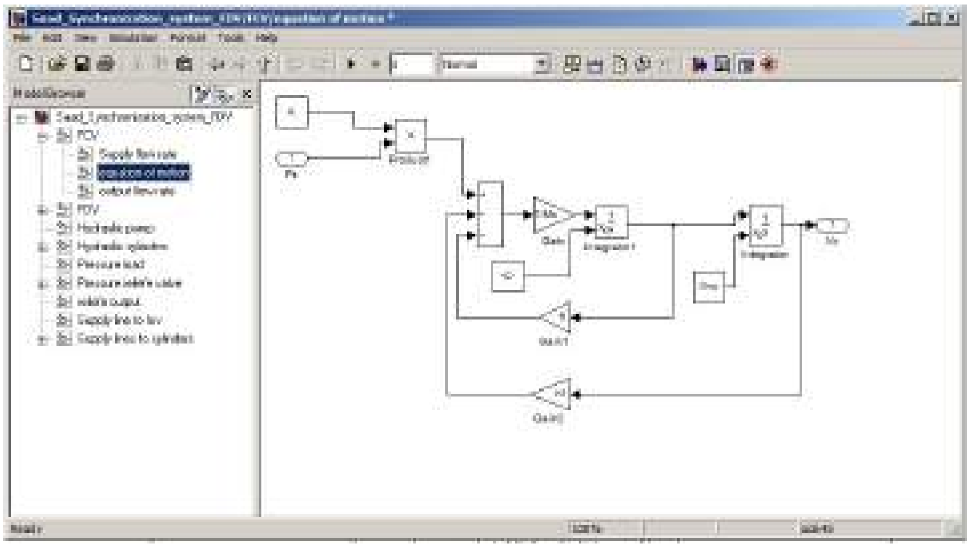


Fig. (3-20) Simulink model for equation of motion to FCV spool.

$$\frac{dPa}{dt} = \frac{\beta}{Va}(Q_a - Q_{a1} - Q_{a2}) \qquad (3.32)$$

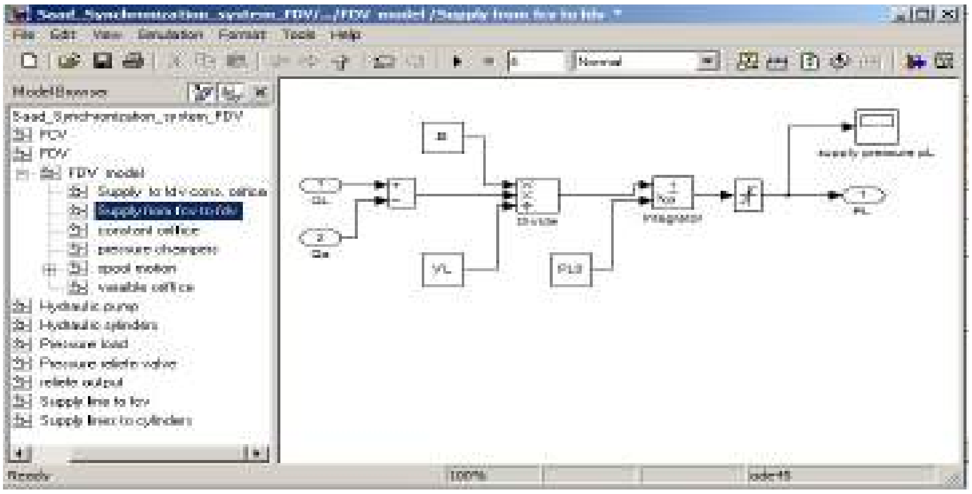


Fig. (3-21) Simulink model for supply line to FDV.

$$Q_{a1} = C_{do1}.A_{o1}\sqrt{\frac{2}{\rho}(P_a - P_{a1})} \qquad (3.33)$$

$$Q_{a2} = C_{do2}.A_{o2}\sqrt{\frac{2}{\rho}(P_a - P_{a2})} \qquad (3.34)$$

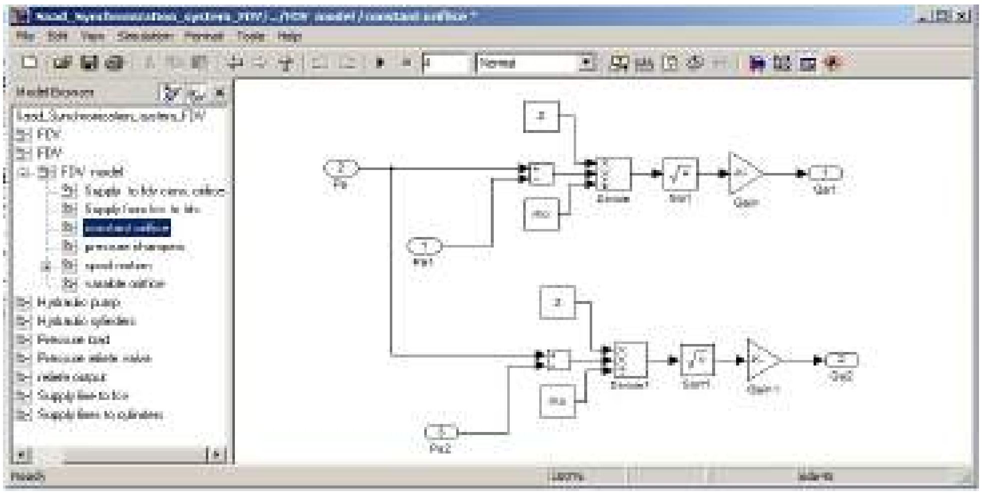


Fig. (3-22) Simulink model for constant orifice in FDV.

$$Q_{L1} = C_{dv1}.A_{v1}\sqrt{\frac{2}{\rho}(P_{a1} - P_{L1})} \quad (3.35)$$

$$Q_{L2} = C_{dv2}.A_{v2}\sqrt{\frac{2}{\rho}(P_{a2} - P_{L2})} \quad (3.36)$$

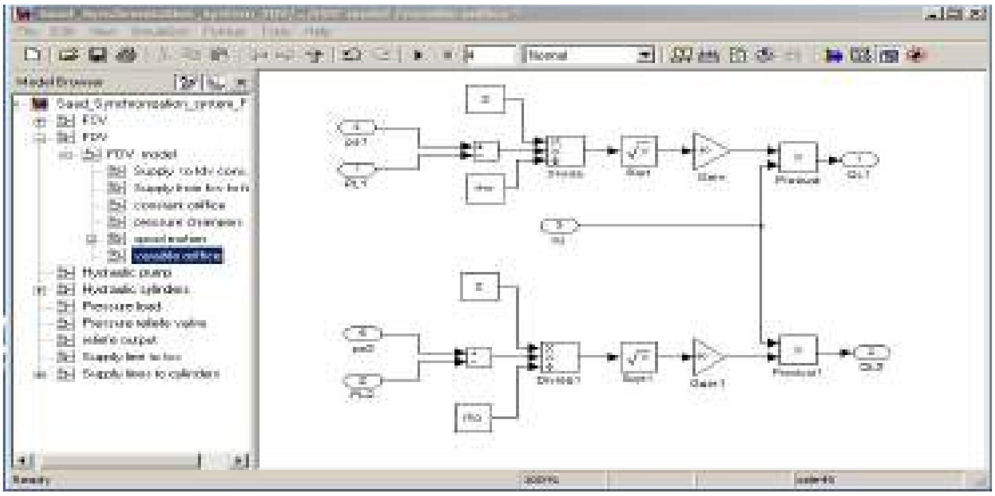


Fig. (3-23) Simulink model for variable orifice in FDV.

$$Ms\frac{d^2xs}{dt^2} + f_s\frac{dxs}{dt} + 2k_s.x_s = \Delta F_{hs} + \Delta F_{hd} \quad (3.45)$$

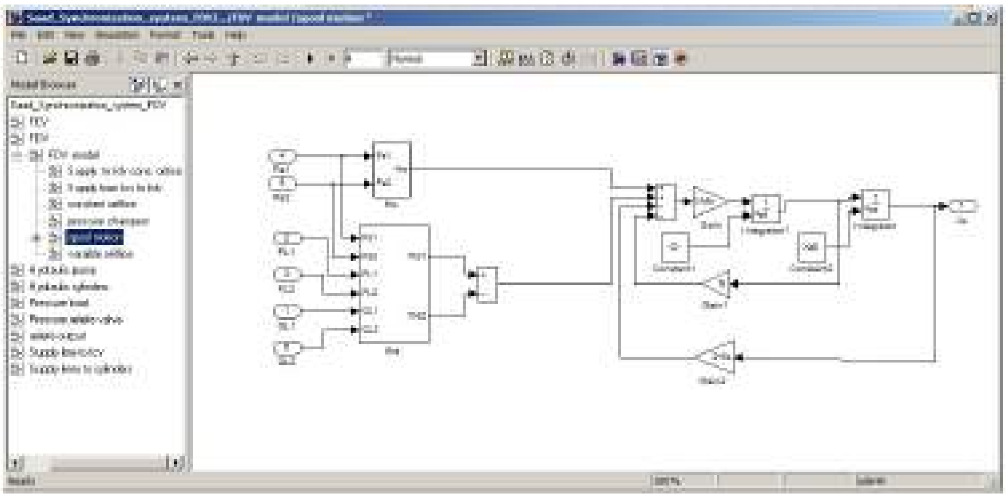


Fig. (3-24) Simulink model for spool motion for FDV.

$$\frac{dPL1}{dt} = \frac{\beta}{V_{L1}}\left(Q_{L1} - A_p\frac{dxp1}{dt}\right)\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.56)$$

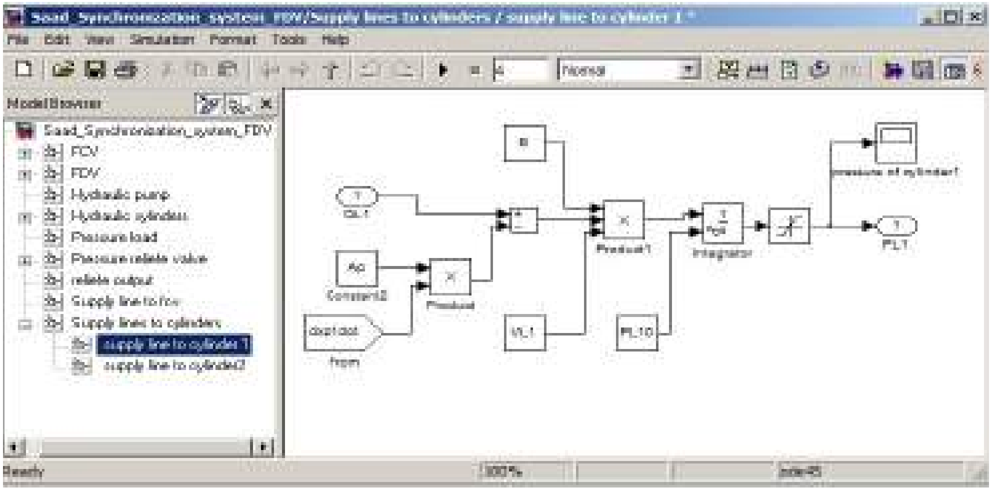


Fig. (3-25) Simulink model for supply line to cylinder1.

$$M\frac{d^2x_{p1}}{dt^2} + f_p\frac{dx_{p1}}{dt} = \left(P_{L1}A_p - P_{r1}A_r\right)\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.58)$$

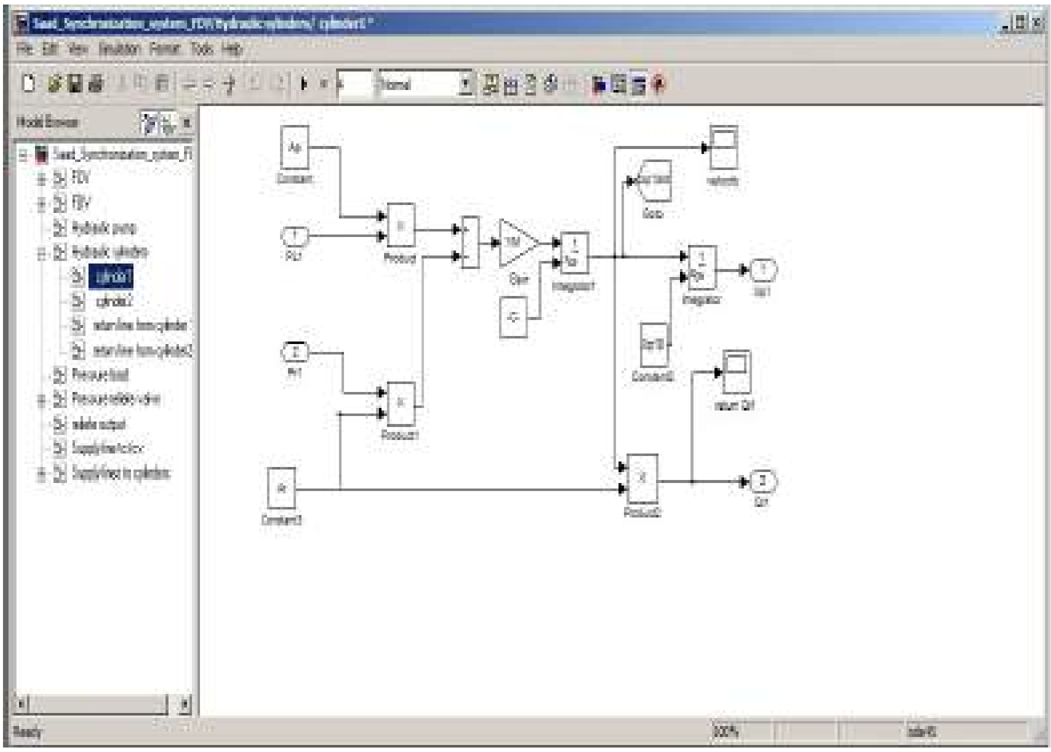


Fig. (3-26) Simulink model for cylinder1.

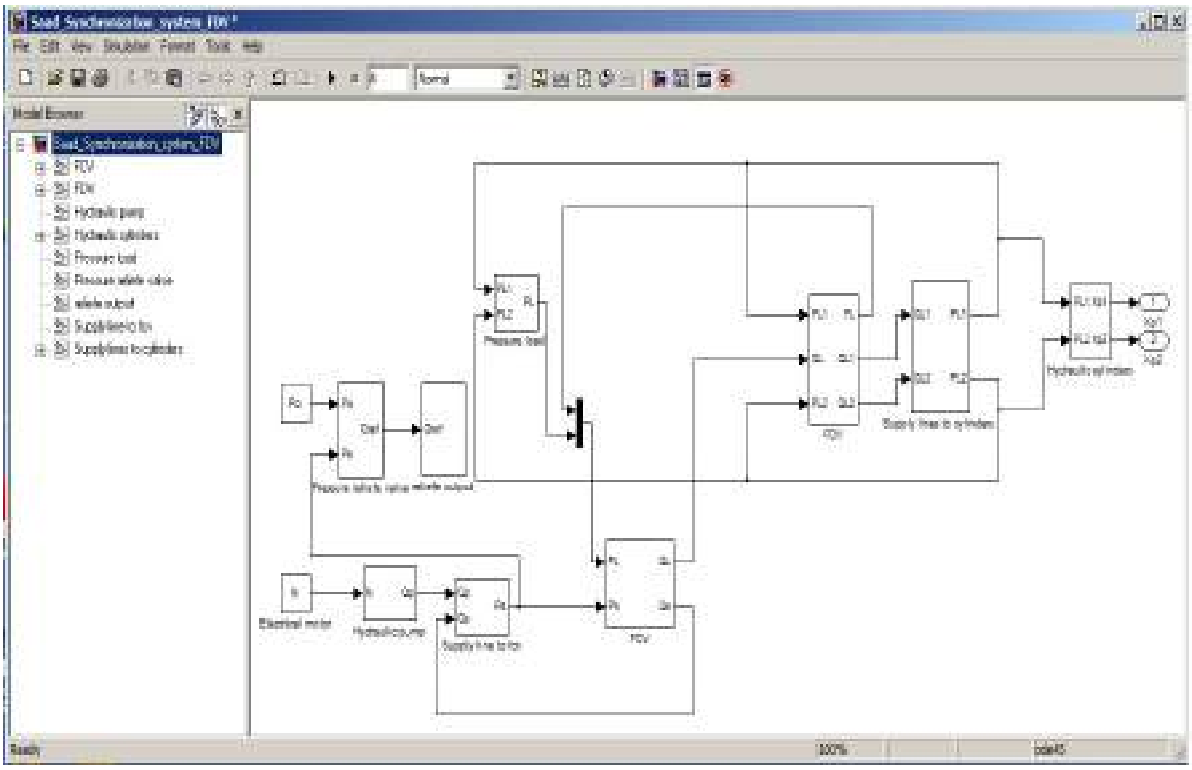


Fig. (3-27) Simulink modeling of the synchronization hydraulic system with FDV.

3.7Automation Studio

Automation Studio is a completely integrated software package that allows users to design, simulate and animate circuits consisting of various automation technologies. It is the ideal CAD and simulation tool for teachers, students and engineers. In addition to its standard Hydraulics and Ladder Logic libraries, Automation Studio further supports the following technologies: Pneumatic, Digital Electronics [42]. The main window of this package and some components are shown in Figures (3-28) and (3-29). All components and parameters of the hydraulic system were selected and the connection between them represented as in the actual system in the experimental work. The simulation properties of any component like the displacement of the hydraulic cylinders could be shown as a relation between the displacements (mm) with time (sec.) in the plotter window. The simulation result details for hydraulic system are given in chapter five.

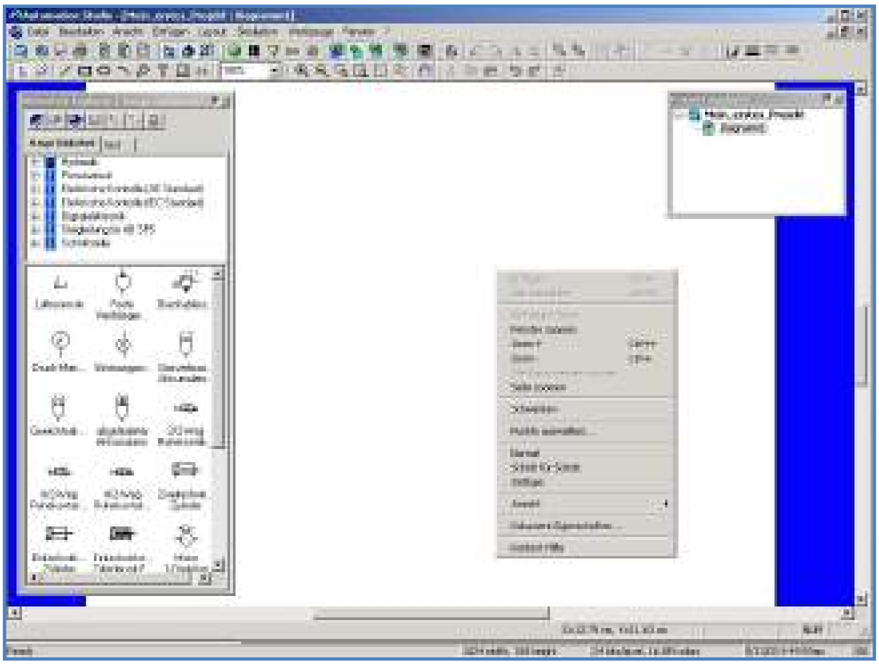


Fig. (3-28) main window of Automation Studio package.

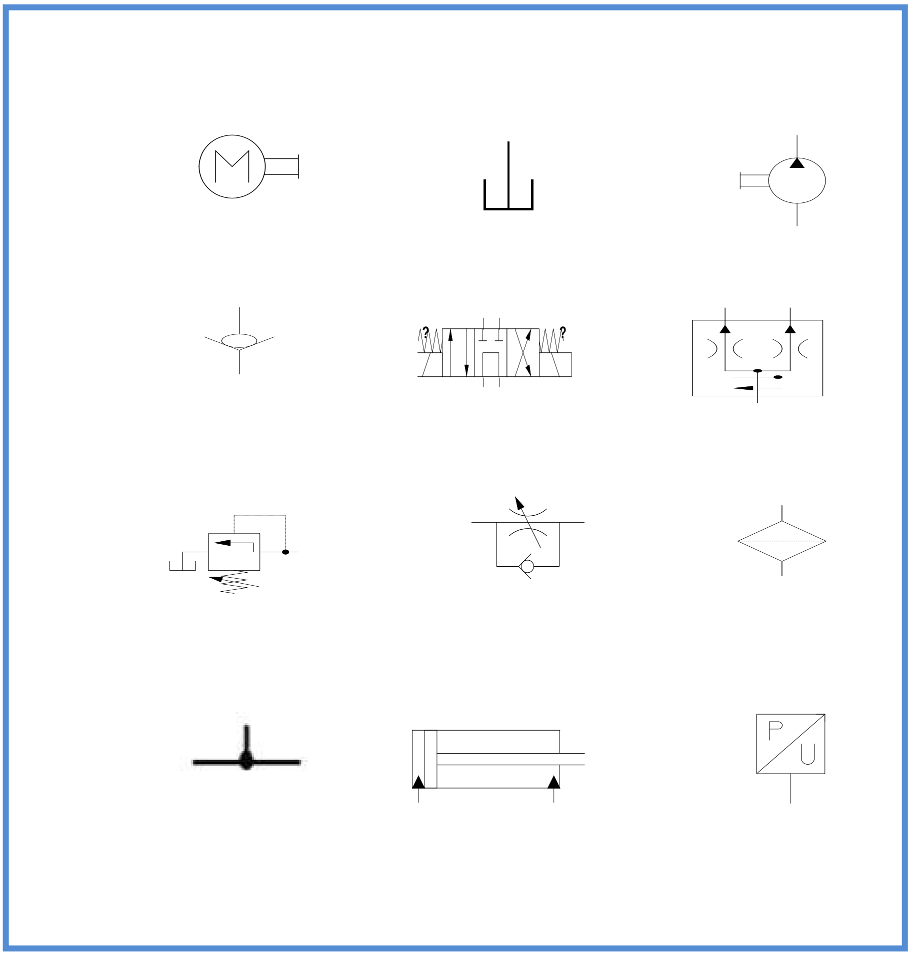


Fig. (3-29) Automation studio components figures.

3.8 AS Program Steps to Build and Simulate a Hydraulic System.

Following the first steps to build and simulates AS project beginning from Figure (3- 30) to Figure (3-37).

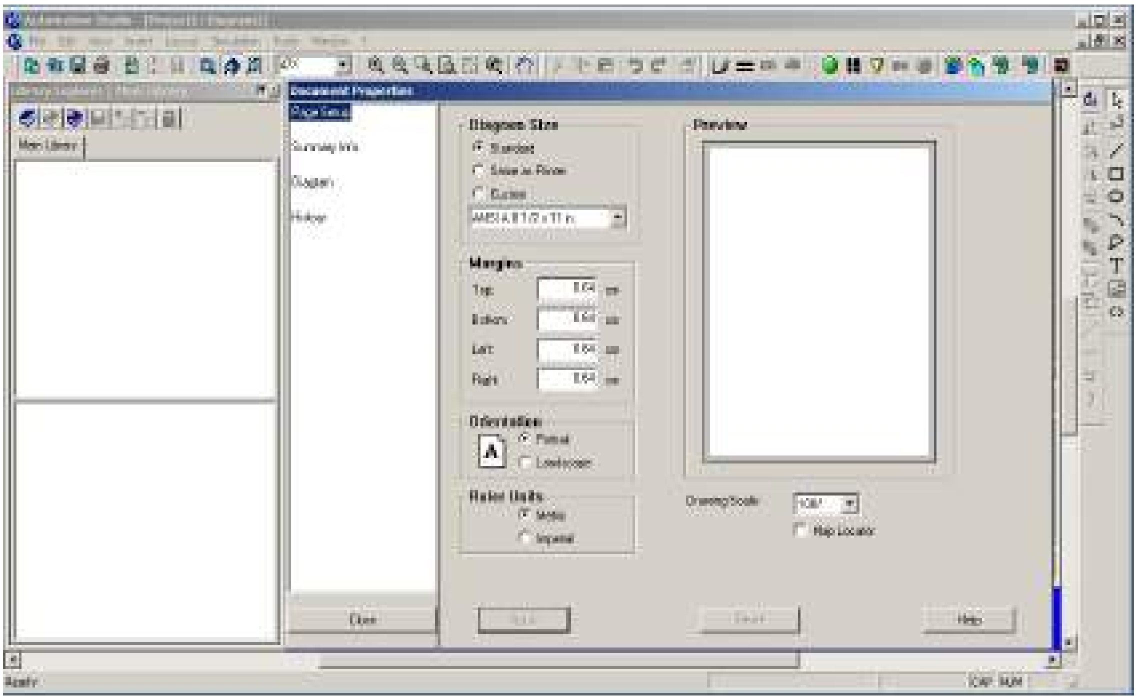


Fig.(3-30) Opening new project

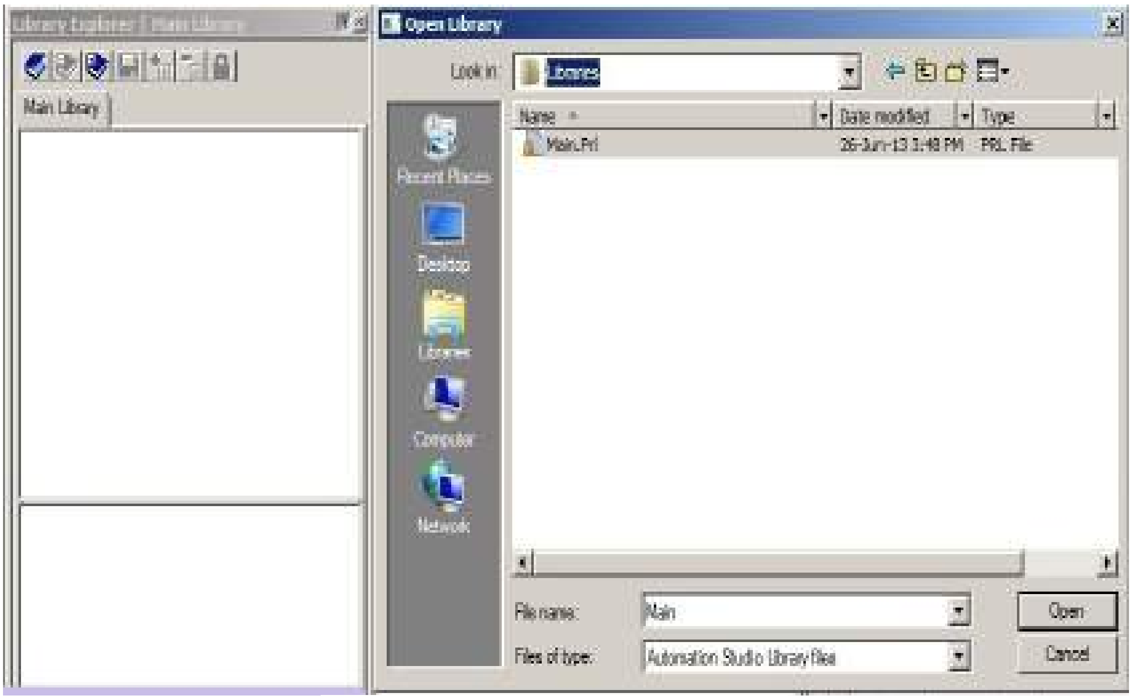


Fig. (3-31) Opening main library

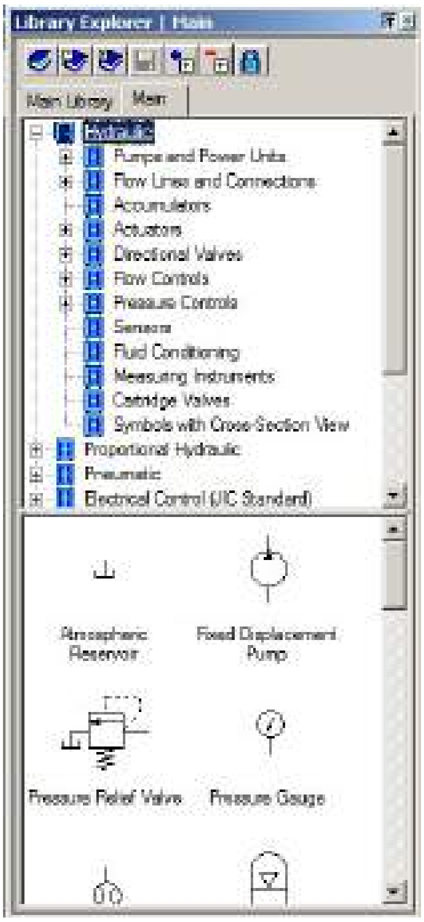


Fig (3-32) Library explorer

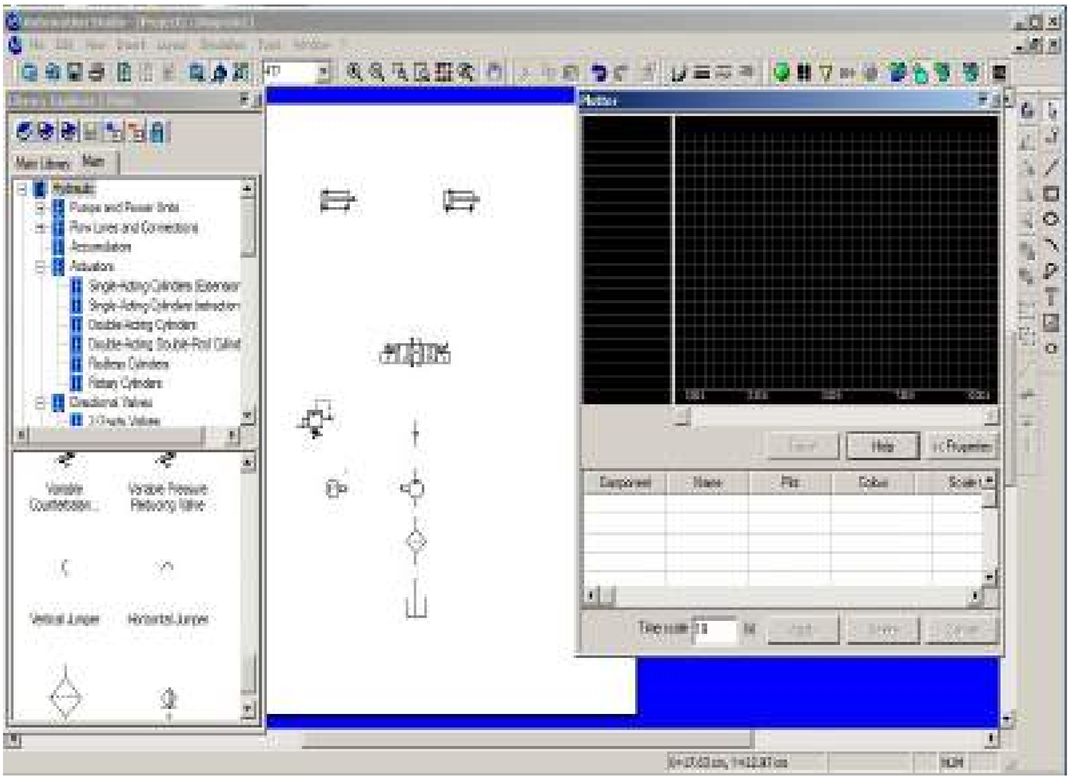


Fig. (3-33) Dragging the hydraulic components

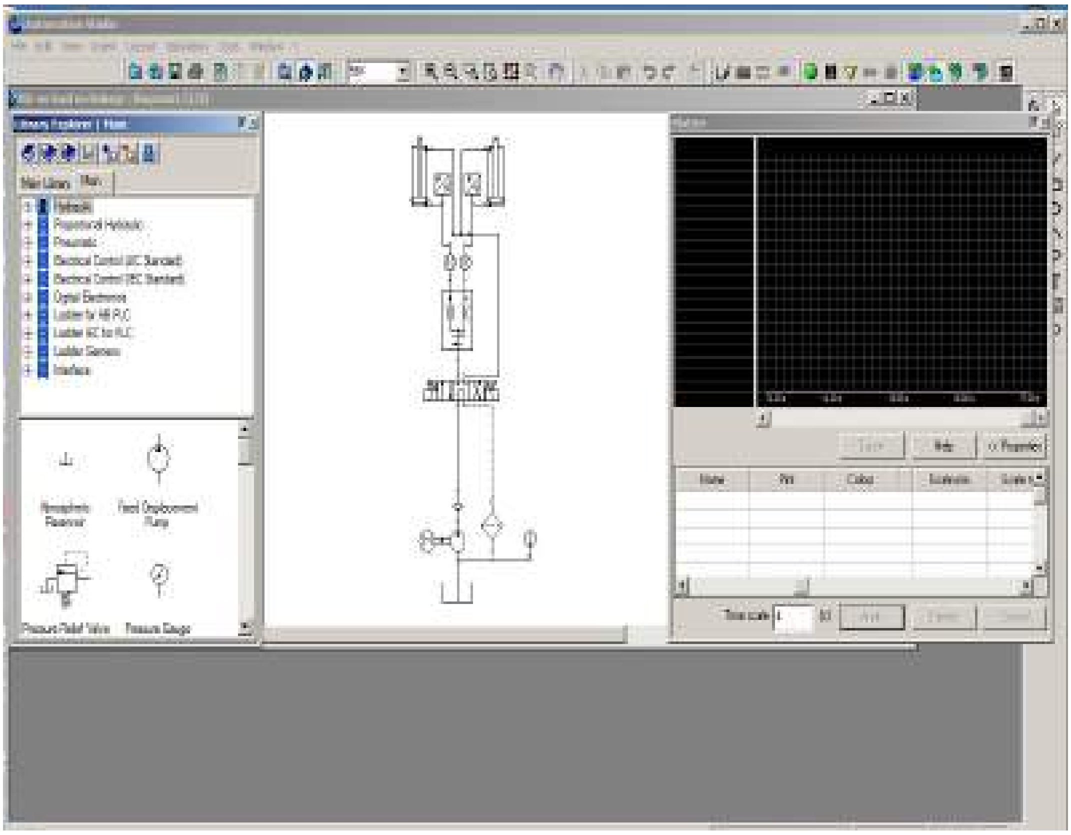


Fig.(3-34) Connecting lines between the components

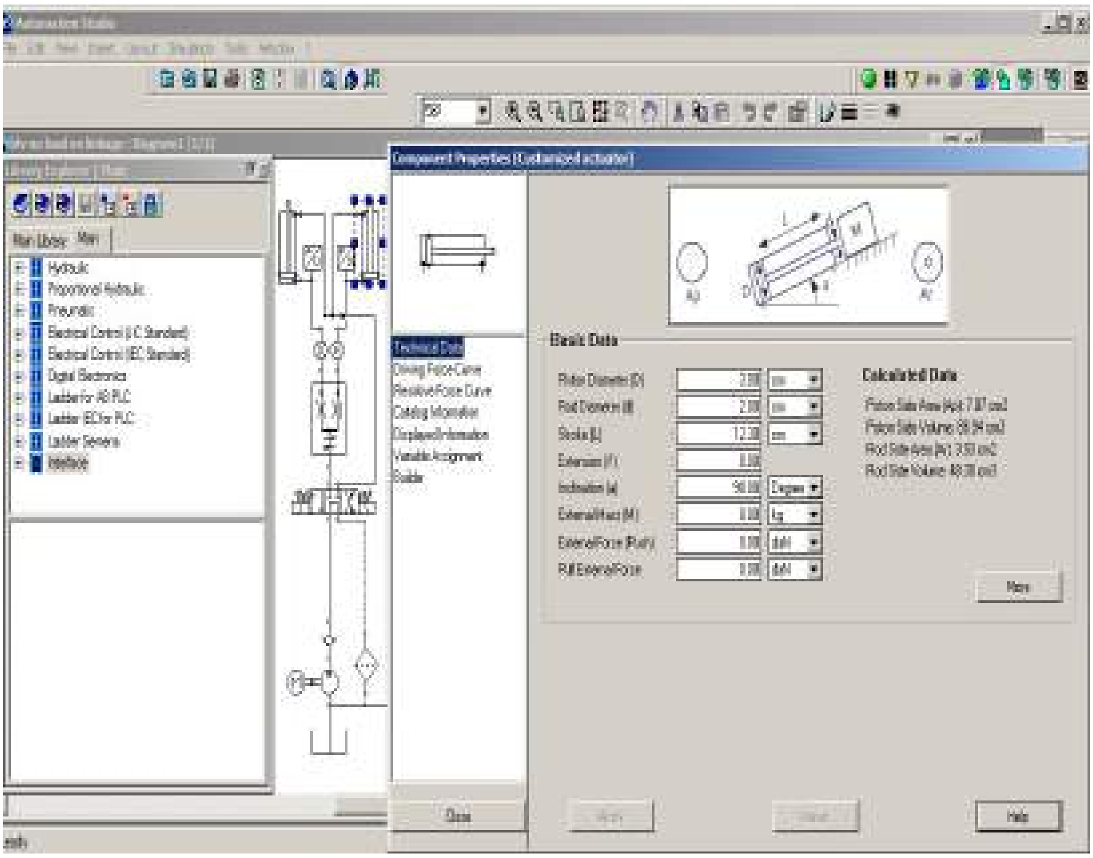


Fig. (3-35) components properties

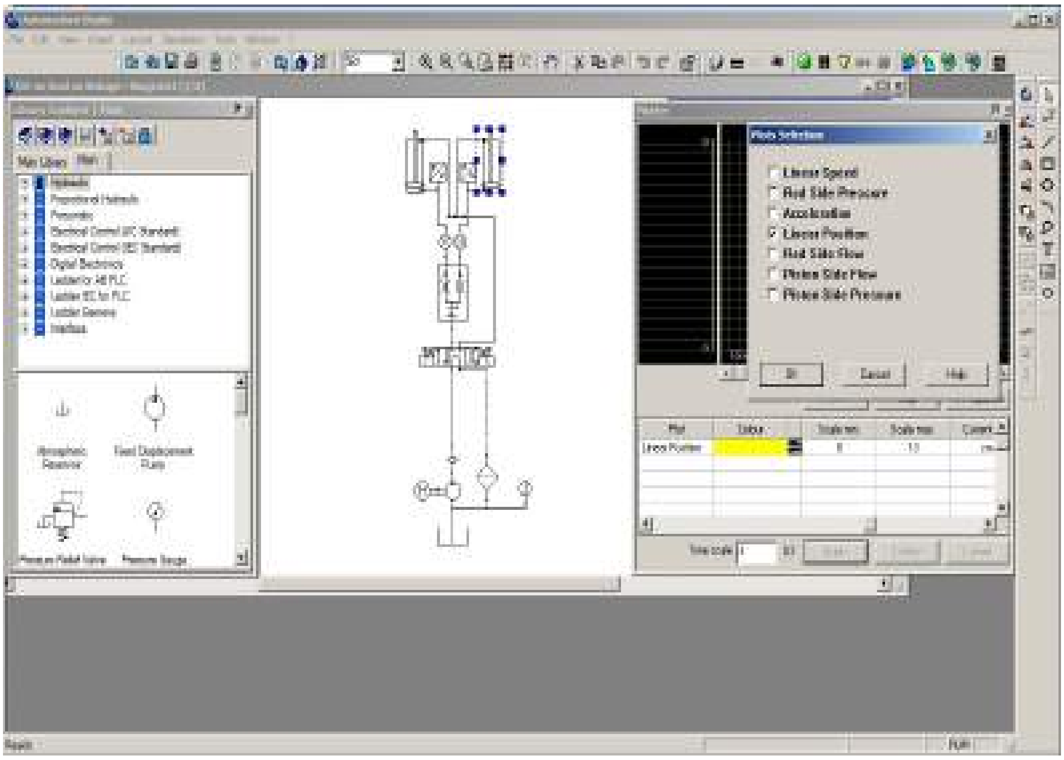


Fig.(3-36) Simulations procedures

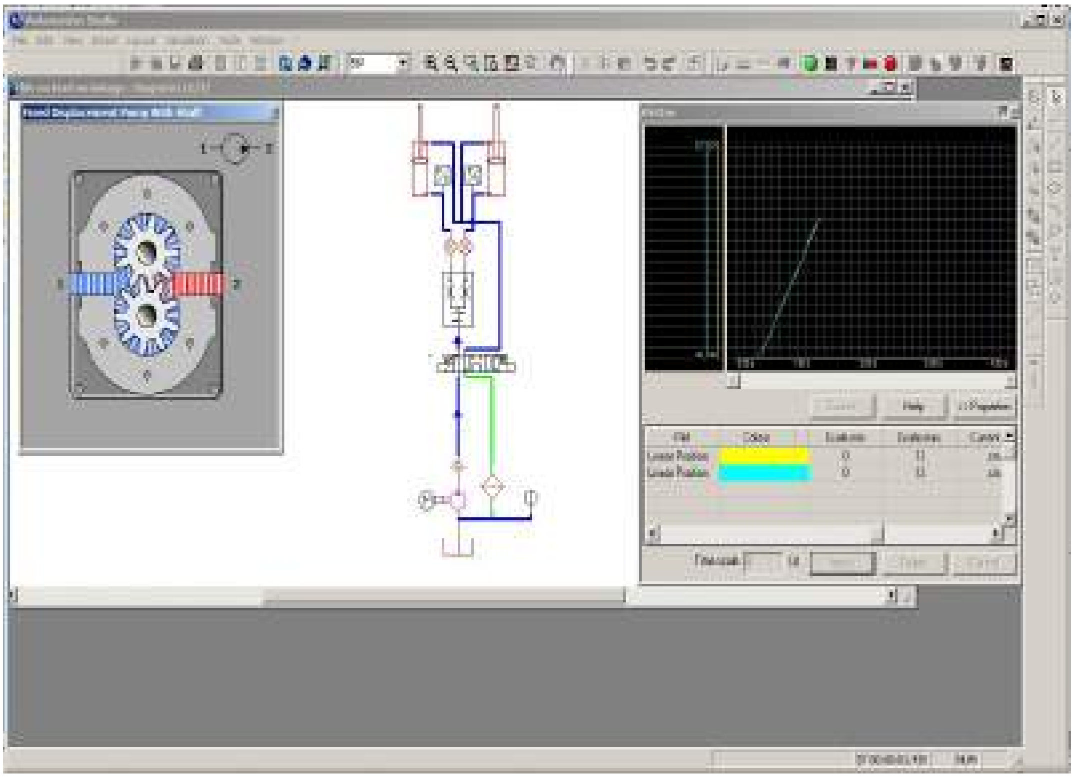


Fig. (3-37) Simulations and pump animation

Chapter Four

Experimental Work

4.1 Introduction

This chapter presents the experimental work, which includes construction the test rig and practical experiments, to study the synchronization motion of hydraulic linear actuators.

4.2 Experimental Apparatus

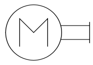
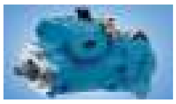
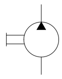
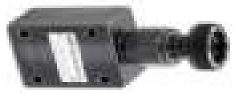
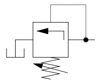
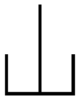
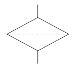
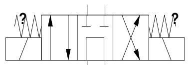
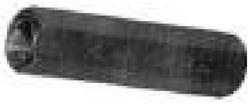
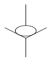
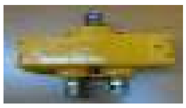
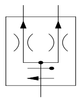
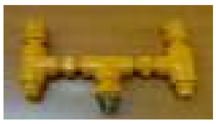
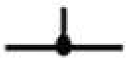
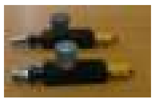
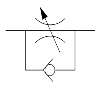
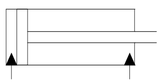
The experimental apparatus of the present work was a modification to an existing setup in the hydraulic and fluid laboratory, at the Mechanical Engineering Department in Al-Mustansiriyah University, to be suitable for the objective of present work. The modifications which have been accomplished on the available experimental apparatus consisted of adding the following components:

1. Flow Divider Valve
2. Two Identical Cylinders
3. Two Flow Control Valves
4. Two Pressure Transmitters
5. Tow Flow Sensors
6. Two Position Sensors
7. Two Check Valves
8. Digital Multimeter
9. Electrical Heater
10. Measurement Control System

4.3 Characteristics of the Hydraulic Components

Table (4-1) shows the Characteristics of the hydraulic components with detailed physical features of the components.

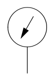
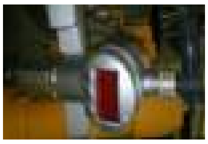
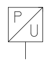
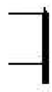
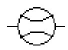
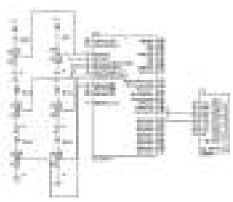
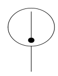
Table (4-1) Component descriptions.

Name	Description	Picture	symbol
Electrical Motor	.55Kw, 1410 RPM		
Hydraulic Gear Pump	9 (lpm)		
Pressure Relief Valve	150 (bar)		
Hydraulic Reservoir	13.2489 (Liter) HLP32		
Hydraulic Filter	$d_{f=}$ 40 mm, H=70 mm		
Directional Control Valve	3 Positions 4 ways		
Check valve	Ball type		
Flow divider valve	50:50 division ratio		
Tee connection	2 output ports 9.525 (mm)		
Flow control valve	56 (lpm)		
Hydraulic Cylinders	D=30 mm, d=20 mm, L=123mm		

4.4 Measurement Instruments

To measure pressure, flow rate and displacement, a measurement instruments have been utilized. These are pressure transmitters, flow sensors and position sensors as shown in Table (4-2). All the measurement instruments have been calibrated and the results are shown in appendix A.

Table (4-2) Measurement Instruments.

Name	Description	picture	Symbol
Pressure gage	100 (bar)		
Pressure transmitters	250 (bar)		
Position sensors	L=123 (mm)		
Flow sensors	32 (lpm)		
Control board	6 sensors		
Digital multimeter	12 ports		

4.5 The Design Requirements

To design a synchronization hydraulic circuit, necessary components must be added to and deleted from the existing test bench in the lab. Figure (4-1) shows the test bench before the modifications.

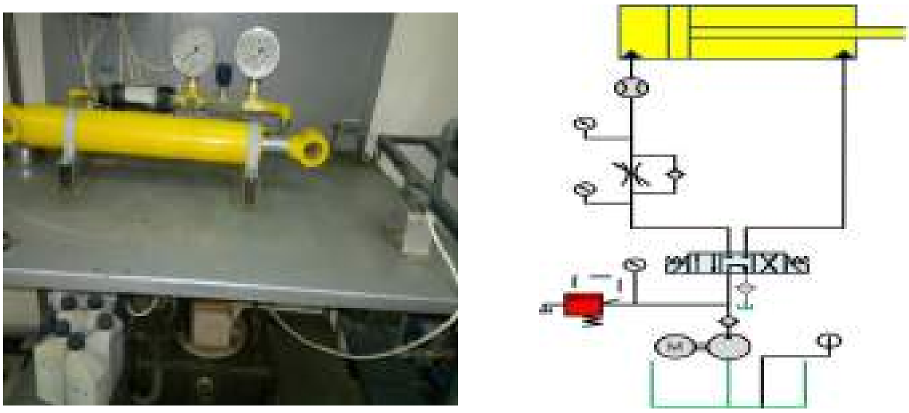


Fig. (4-1) Photo and schematic diagram of the test bench before the modifications [43]

Besides that, important parts have been manufactured and these are:

- Linkage: This is a rood made of stainless steel which is using for connecting the two hydraulic cylinders to synchronize them, as shown in Figure (4-2).
- Load loader: This is used for lifting center and off center loads as shown in Figure (4-3).
- Separate load loader: In the case of using separate loads, a special loader can be used as shown in Figure (4-4).

After the modifications, the test bench became as shown in Figure (4-5).

Fig. (4-2) Photo of the linkage

Fig. (4-3) Photo of the load loader

Fig. (4-4) Photo of the separate loader

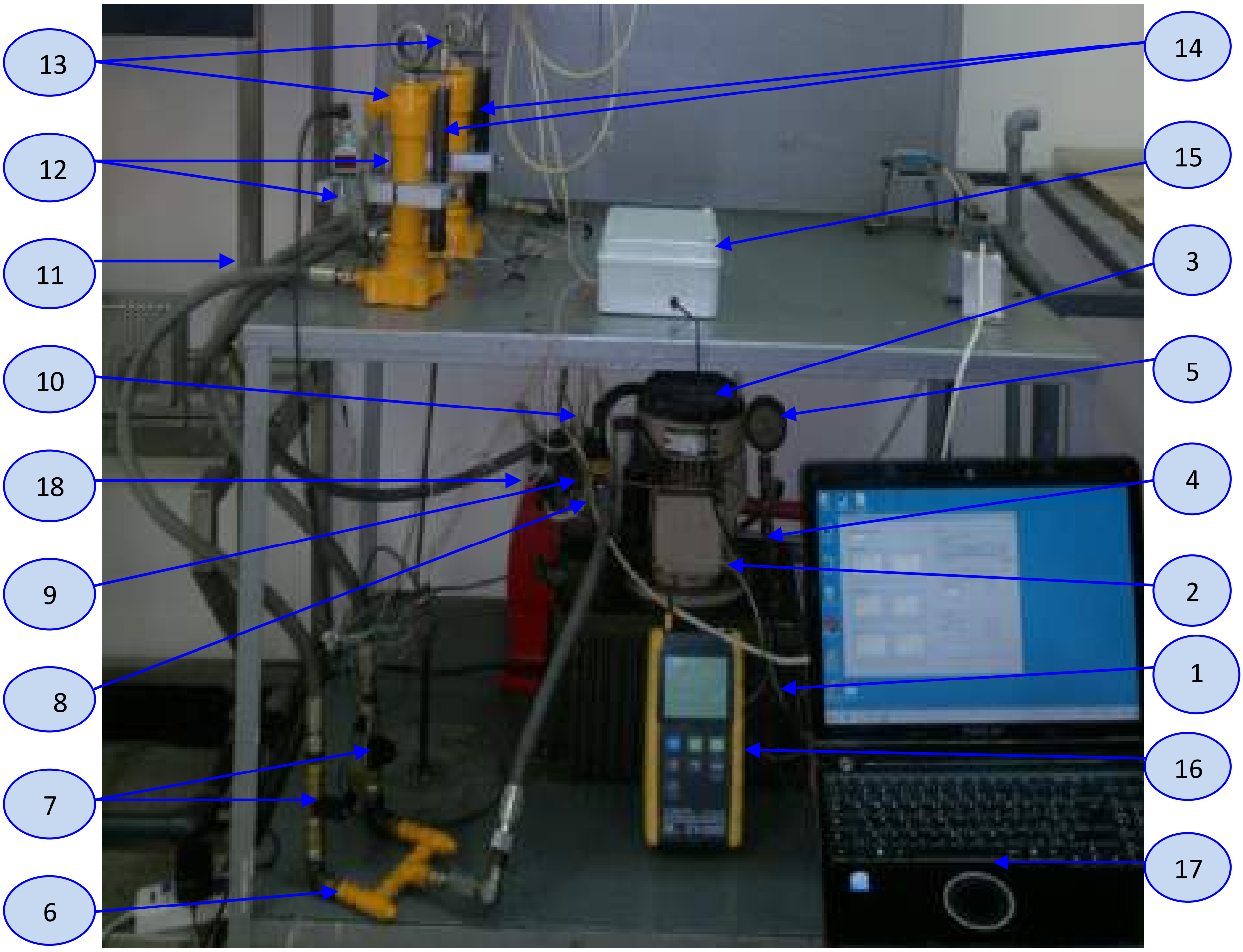


Fig. (4-5) The experimental apparatus for Tee connection (no load no linkage).

Table (4-3) Component designations.

NO.	Component designation	NO.	Component designation
1	Hydraulic reservoir	10	Hydraulic filter
2	Hydraulic pump	11	Hydraulic hoses
3	Electrical motor	12	Pressure transmitters
4	Pressure relief valve	13	Hydraulic cylinders
5	Pressure gage	14	Position sensors
6	Tee connection	15	Control board
7	Flow sensors	16	Digital multimeter
8	Check valve	17	Computer (Laptop)
9	Directional control valve	18	Electrical heater

4.6 Experimental Procedure

On the commencing of the tests, the following instructions must be done:

1) Running the experimental apparatus until it reaches the temperature steady state.

2) Setting the pressure relief valve by making the piston of cylinder reaches to the end of the stroke (extending or reacting). At the same time, the pressure gage of the pressure relief valve should be taken.

3) Determination and fixing the number of cycle of flow control valve.

4) Running the Digital Multimeter to read the following:

i. Temperature of ambient.

ii. Temperature of hydraulic oil by installs the sensor of thermocouple of the digital multimeter inside the hydraulic reservoir.

The experimental work includes designing synchronization hydraulic circuits by using a conventional, flow control valve and flow divider valve circuits.

These circuits were tested experimentally under different values of temperatures (40, 45, 50) ^{o}C and pressures with ambient temperature at (30 ^{o}C). First, the tee connection has been used without any additions like linkage or load at temperature (40 ^{o}C). Secondly, the test has been repeated at (45 ^{o}C) and finally, it has been worked out at (50 ^{o}C). When separate loads are used, the tests will be done as mentioned above. These temperatures ranges have been repeated for flow control valve tests and flow divider valve tests. The other tests have been accomplished at 40 ^{o}C. Figure (4-21) shows the steps of experimental work.

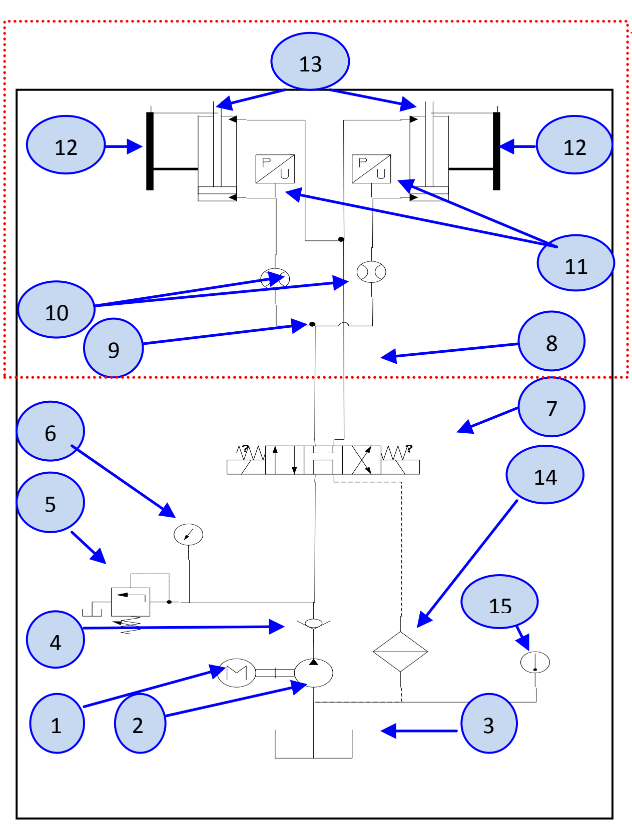
The pressure is according to the used weight. Because of the limitation of the test rig, the used weight has a limited value. In the linkage case, the weight is 57.7 kg, while in the separate load case the weight is 144.2 kg, or rather 72.1 kg on each cylinder.

4.6.1 Synchronization Motion by conventional circuit

Various experimental tests can be applied by using tee connection which are listed below:

Table (4-4) Tee Connection cases

Case No.	Tee Connection	Figure No.
1	no load, no linkage	(4-5),(4-6)
2	no load with linkage	(4-7)
3	center load with linkage	(4-8)
4	off center load with linkage	(4-9)
5	separate load	(4-10)



15	Digital multimeter	12 port	1
14	Hydraulic Filter	$d_{f=}$ 40 mm, H=70 mm	1
13	Hydraulic Cylinders	D=35mm, d=20 mm, L=123mm	2
12	Position Sensors	L=12 3 (mm)	2
11	Pressure Transmitters	250 (bar)	2
10	Flow Sensors	32 (lpm)	2
9	Tee Connection	Ports 9.525 (mm)	1
8	Hydraulic Hoses	413 (bar)	1
7	Directional Control Valve	3 Positions 4 ways	1
6	Pressure gage	100 (bar)	1
5	Pressure Relief Valve	150 (bar)	1
4	Check valve	Ball type	1
3	Hydraulic Reservoir	13.2489 (Liter)	1
2	Hydraulic Gear Pump	9.0850 (lpm)	1
1	Electrical Motor	.55Kw, 1410 RPM	1
Part	**Name**	**Description**	**Qty.**

Fig. (4-6): Schematic diagram of experimental apparatus for conventional circuit

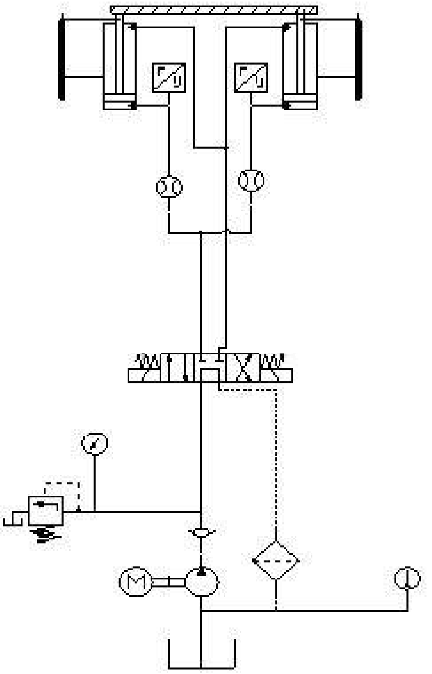


Fig. (4-7) The use of tee connection (no load with linkage).

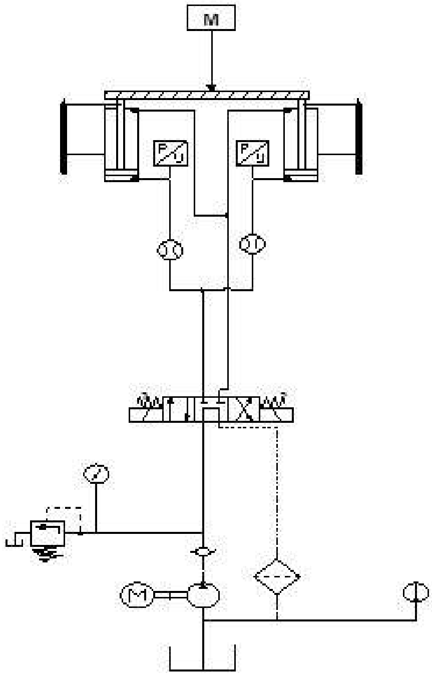


Fig. (4-8) The use of tee connection (center load with linkage).

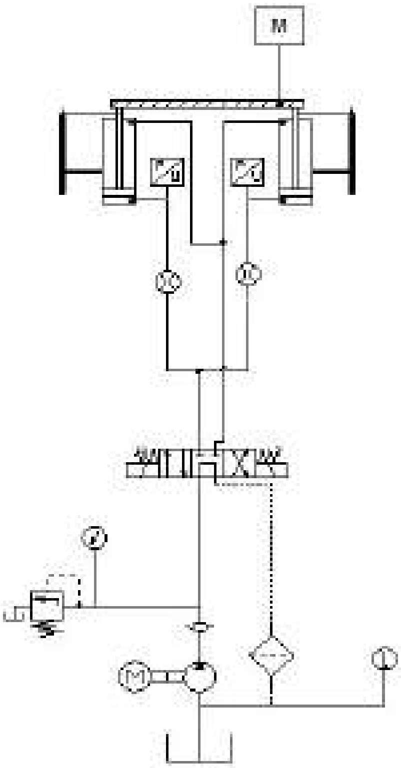


Fig. (4-9) The use of tee connection (off center load with linkage)

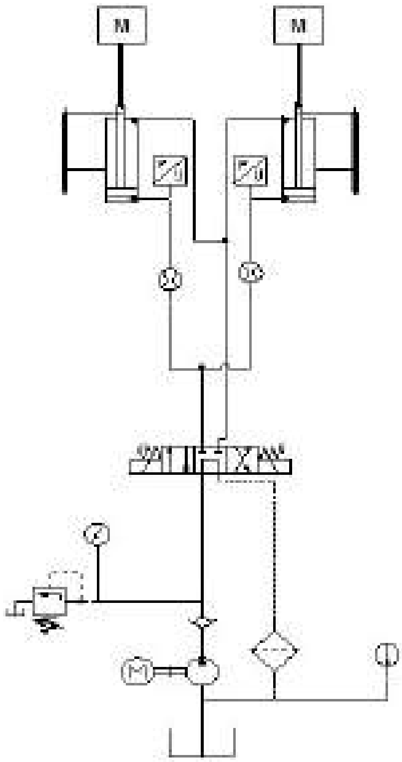


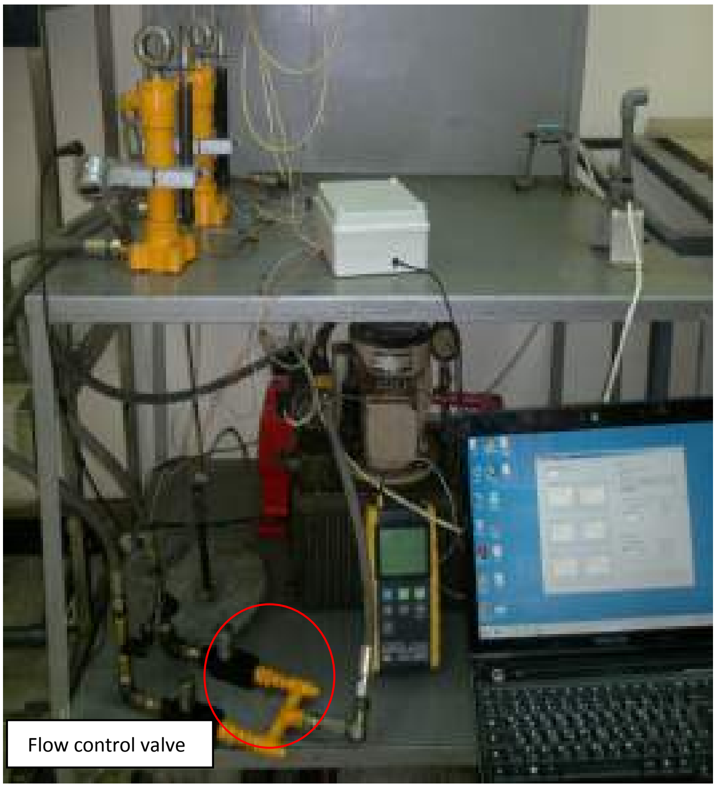
Fig. (4-10) The use of tee connection (separate load)

4.6.2 Synchronization Motion by Using Flow Control Valve circuit

This procedure will be accomplished by using the same hydraulic circuit which is shown in Figure (4-5). However, no addition except of the two throttles checks in the lines connecting the tee with hydraulic cylinders. Different testes can be applied by using flow control valve (throttle check) which are listed below.

Table (4-5) Flow control valve cases

Case No.	Flow control valve	Figure No.
1	no load no linkage	(4-11)
2	no load with linkage	(4-12)
3	center load with linkage	(4-13)
4	off center load with linkage	(4-14)
5	separate load	(4-15)



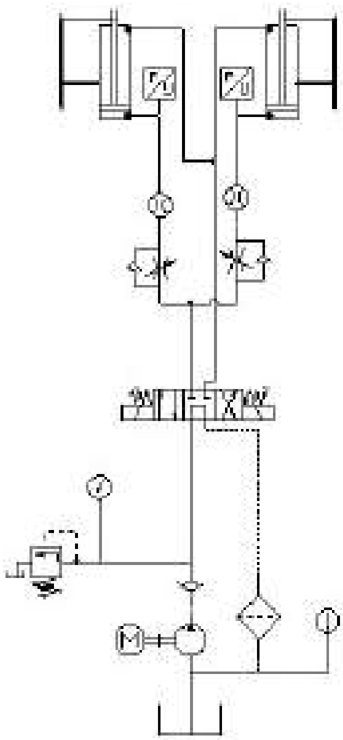


Fig. (4-11): The experimental apparatus to FCV (no load no linkage.) with the schematic diagram.

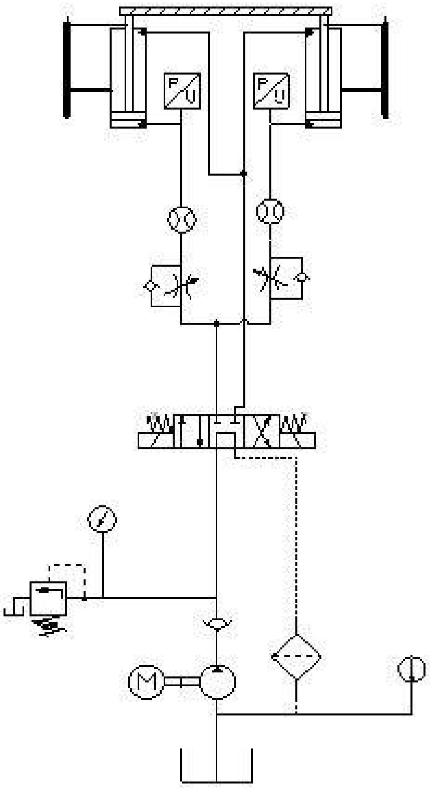


Fig.(4-12) The use of FCV (no load with linkage)

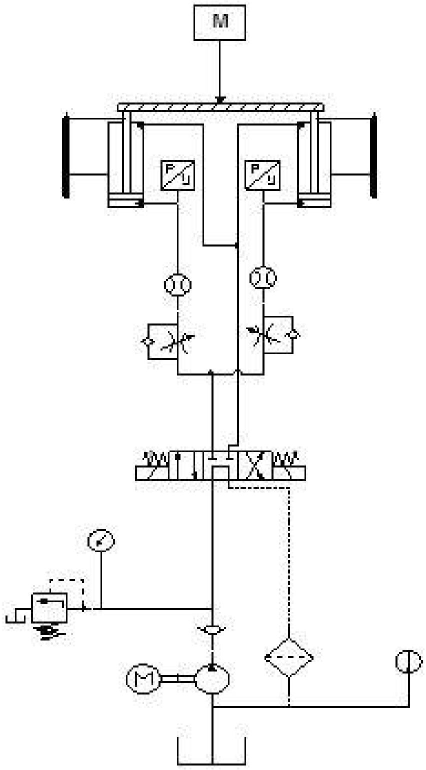


Fig. (4-13) The use of FCV (center load with linkage).

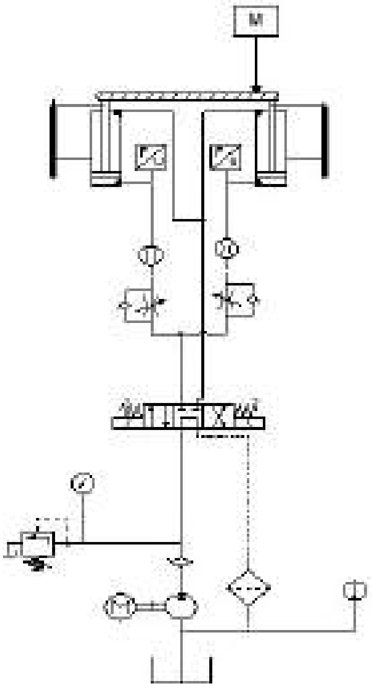


Fig. (4-14)The use of FCV (off center load with linkage)

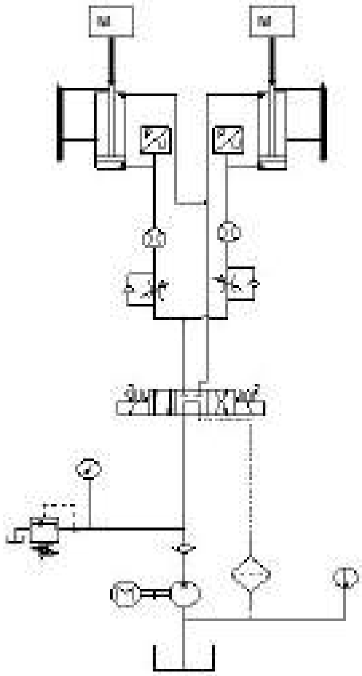


Fig. (4-15) The use of FCV (Separate load)

4.6.3 Synchronization Motion by Using Flow Divider Valve circuit

A flow divider valve and two check valves (these are used because the divider can't allow the oil to flow in the reverse case) were added to the circuit which is shown in Figure (4-5). Different testes can be applied by using flow divider valve which are listed below.

Table (4-6) Flow divider valve cases

Case No.	Flow divider valve	Figure No.
1	no load no linkage	(4-16)
2	no load with linkage	(4-17)
3	center load with linkage	(4-18)
4	off center load with linkage	(4-19)
5	separate load	(4-20)

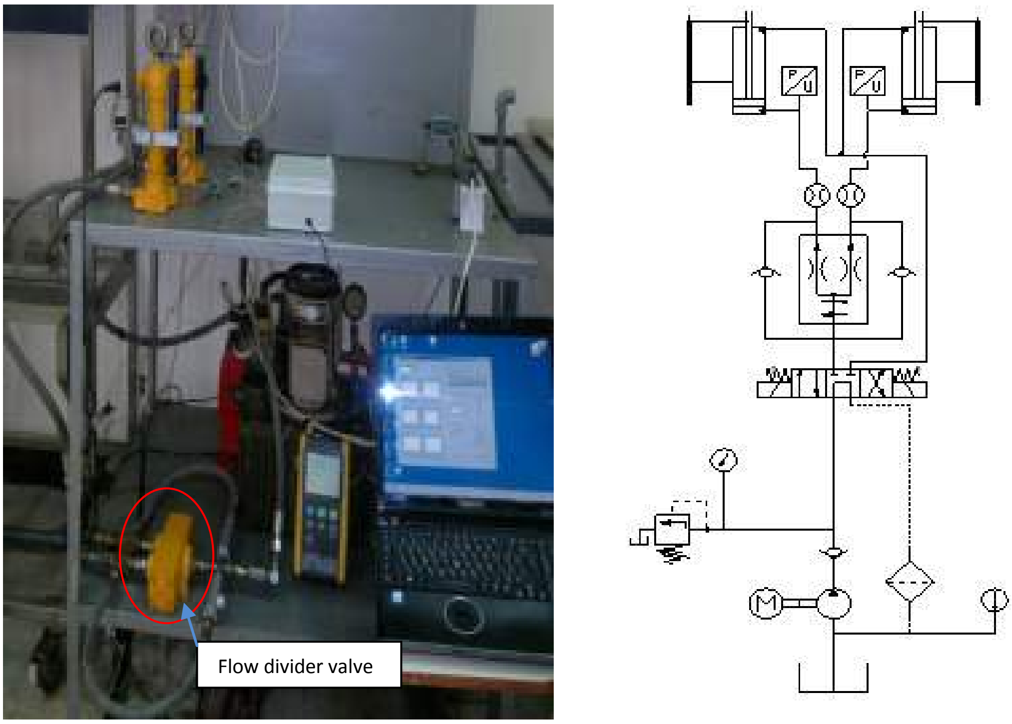


Fig. (4-16): The experimental apparatus for FDV (no load no linkage) with the schematic diagram.

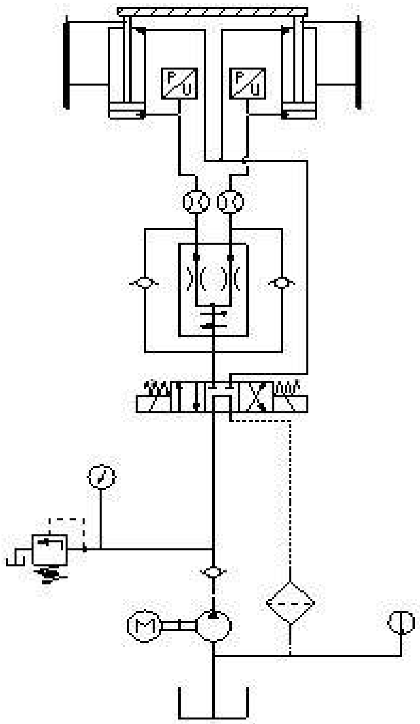


Fig. (4- 17) The use of FDV (no load with linkage).

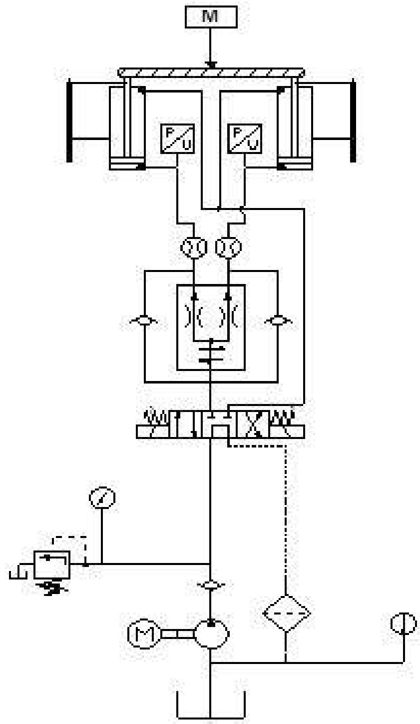


Fig. (4-18) The use of FDV (center load with linkage).

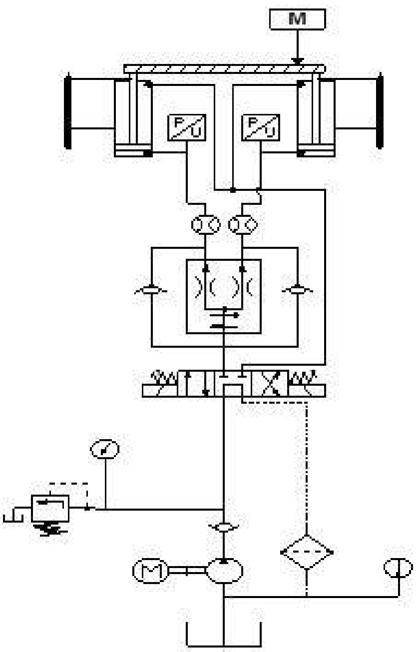


Fig. (4-19) The use of FDV (off center load with linkage).

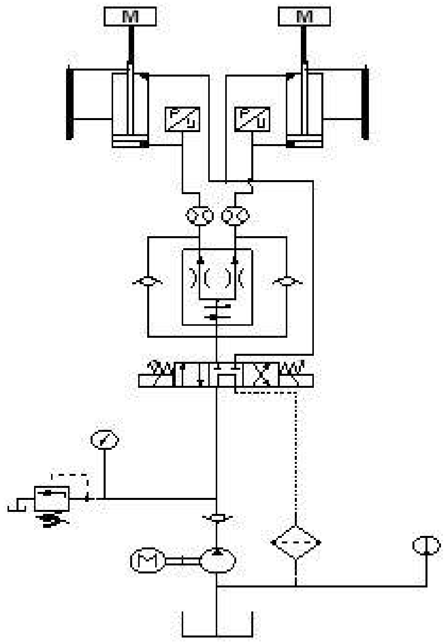


Fig. (4- 20) The use of FDV (separate load).

4.7 Steps of the Experimental Work

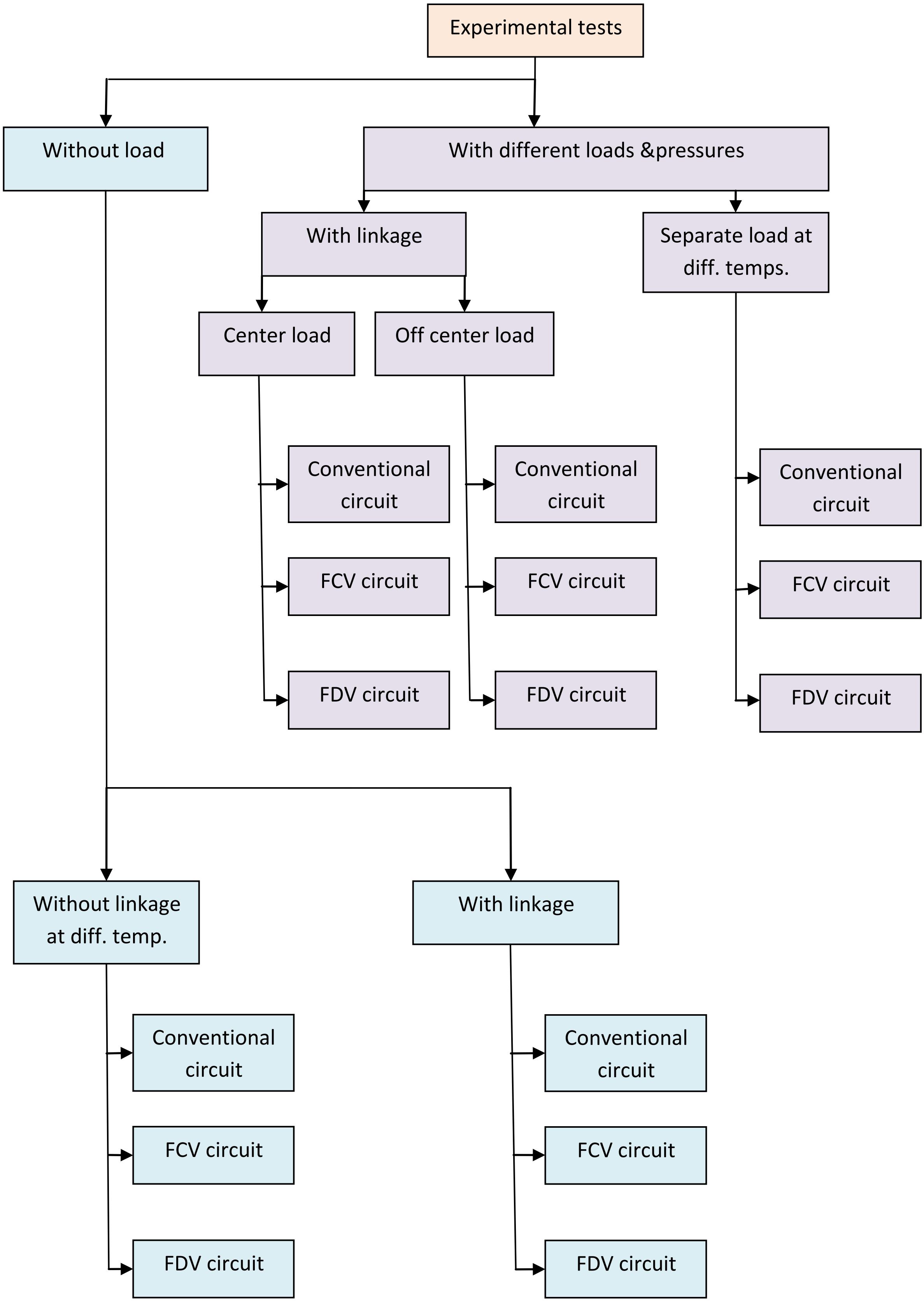


Fig. (4-21) Steps of the experimental work

Chapter Five

Results and Discussions

5.1 Introduction

This chapter shows the results and discussions of both, the experimental work and theoretical simulation. The results have been presented as graphical relations.

5.2 Simulation Results

Theoretical part includes both, Matlab/Simulink and Automation Studio simulations. These are discussed as follows:-

5.2.1 Simulink Results

In order to ensure that the mathematical model to the synchronization system was adequate, it was carried out using a Matlab/Simulink computer program. The parameters that have been used in Simulink program mentioned in Matlab m. files in appendix (C). The system that was presented experimentally in Figure (4-16) and its mathematical model in Figure (3-27) was undertaken a numbers of simulations.

In Figure (5-1) the Simulink results of pressures, flow rates and piston rod displacements during lifting variable loads. Begin at first without load, secondly at 720 (kg), and finally with 1081 (kg) which represents Figures a, b and c respectively. These are a distributed uniformly values that produce corresponding pressures 0, 50, 75 bar in series. These different theoretically loads are to show the compressibility effect.

From the results of Simulink, it can be shown that position, flow rate and pressure curves are perfectly identical. This is because of the mathematical signal which has no to do with the physical effects such as friction, leakage and among other factors. Also, there is no difference in the three cases (a, b, c), owing to the tiny compressibility of oil turns out to be important when the pressure is high, up to 5,000 psi (about 345 bar). Typical oil will decrease about 0.5% in volume for every 1000 psi (about 70 bar) increase in pressure [1].

5.2.2 Automation Studio Simulation Results

After the mathematical model was carried out, a simulation by Automation Studio has been done for the experimental validation. Three cases have been studied concerning with flow divider valve system. These are no load at different temperatures, even load and uneven load circuits.

Case1: Flow Divider Valve Circuits with No Load

The hydraulic synchronization system was built and simulated with Automation Studio package V5.1 as shown in figures (3-37) and (4- 16). The aim is to simulate flow rate, displacement and pressure versus time at different cases.

Figure (5-2) shows position, flow rate and pressure versus time at different operating temperatures for flow divider valve circuit. The error is nearly diminished and this is obvious on the displacements curves which are identical. There is a tiny increase in the pressure along the extending stroke which then goes to the set value when the cylinders reach to the full stroke. This is because in the experimental tests the pressure sensors can’t read the tinny values in reverse of the AS.

To show the effect of the temperature of the working hydraulic oil on the synchronization motion, three operating oil temperatures were used (40, 45, 50) ^{o}C (a, b, c) respectively. The result shows that there was no effect of the oil temperature on the synchronization motion.

Case2: Flow Divider Valve Circuits with Even Load.

Figure (5-3-a) is a repeat for case1 at 40 ^{o}C for comparison purposes with the following case. Figure (5-3-b) shows position, flow rate and pressure versus time for flow divider valve circuit. The net load which is used in this circuit is 144.2 (kg) on each cylinder 72.1 (kg). There is an overshot in the flow rate and then reaching to the steady state value. Also there is an overshot in the pressure at the beginning of the extending stroke. Then it remains at the steady state value and then goes to the set value when the cylinders reach to the full stroke. These overshoots are owing to the time that required for FDV spool to response for a sudden change in the flow rate.

Case3: Flow Divider Valve Circuits with Uneven Load.

Figure (5-3-c) shows position, flow rate and pressure versus time for flow divider valve circuit. The net load which is used in this circuit is 57.7 (kg), or rather 10 (kg) on cylinder1 and the rest on cylinder2 .The percentage error increased slightly and this appears on the displacements curves.

There is an overshot in the flow rate at both, the beginning of the extending stroke and in the end of the stroke. There is an increase in the flow rate of cylinder2 and a decrease in flow rate in cylinder1 till it reaches to zero. This is because when cylinder1 reached to the end, the entire flow rate goes to the cylinder2 port and this requires time to make flow divider valve to response to this rapid change. Also there is an overshot in the pressure at the beginning of the extending stroke which then remains at the steady state value and the goes to the set value when the cylinders reach to the full stroke.

5.3 Experimental Results

Many cases have been studied with each hydraulic system in the synchronizations circuits. The cases from (1- 3) and from (7 -9) have been worked out under three values of oil operating temperatures (40, 45, 50) ^{o}C. The other cases have been accomplished at operating oil temperature at 40 ^{o}C at an ambient temperature (30 ^{o}C). This is because the earlier results showed the temperature variation effect is insignificant.

Case1: Synchronization Motion by Conventional Circuit (No Load No Linkage).

Figure (5-4) represents the behavior of the hydraulic system when tee connection was used to synchronize dual cylinders. It presented a tee connection position, flow rate and pressure versus time at different operating temperatures with no load no linkage. The full stroke to both cylinders is 123 (mm) and the time which was required to reach the full stroke is 2.5 (sec.), while the set pressure is 5 (bar). Inlet flow rate was 9 (lpm). These parameters have been used in the theoretical analysis in Matlab Simulink and Automation Studio. The flow rate starts from zero at beginning of the directional control valve actuation reaching its steady state value which is about 4 (lpm) for cylinder1 and about 5(lpm) for cylinder2. When cylinder2 reaches the end of the stroke, its flow rate goes to zero, while the flow rate of cylinder1 goes to the maximum value which is about 9 (lpm). Similarly, the displacement curves begin to rise after DCV actuation and then continue in increasing manner until the hydraulic cylinders reach their full stroke. Then a constant value at 123 mm after 2.5 (sec.) has been reached and has been kept constant. This explanation is applicable on all cases including the figures from (5-4) to (5-13). Because there is no load applied on the hydraulic actuations, the pressure maintains a constant value near zero line which is hardly noticeable, and then increases to the set point when the hydraulic cylinders reach their limit.

The percentage error in the flow rate can be found by using equation (3.54) and was found to be (11.1%). This is properly because either of the friction of the cylinders or owing to the external and internal leakage of the hydraulic components especially in tee connection. This difference in the flow rate causes difference in the hydraulic cylinders displacements.

The results in Figure (5-4,a,b,c) show that there is no obvious effect of the oil temperature on the synchronization motion.

Case2: Synchronization Motion by Using Flow Control Valve Circuit (No Load No Linkage).

Figure (5-5) represents the behavior of the hydraulic system when flow control valve (throttle check) was used to synchronize dual cylinders. It presented a throttle check position, flow rate and pressure versus time at different operating oil temperatures with no load no linkage. The function of the throttle checks is for passing a limited flow rate according to the predetermined set value. Here, maximum values of 4.5(lpm) have been set. The percentage error is (1.6%), and this ratio is appropriate for some applications which do not require high accuracy. This approval appeared obviously on the displacement curve. The error is due to the pressure losses across the throttles. Besides, the throttle checks cycles are not perfectly identical.

The results in Figure (5-5,a,b,c) show there is no obvious effect of the oil temperature on the synchronization motion.

Case3: Synchronization Motion by Using Flow Divider Valve Circuit (No Load No Linkage).

Figure (5-6) represents the behavior of the hydraulic system when flow divider valve was used to synchronize dual cylinders. It presented divider valve position, flow rate and pressure versus time at different operating oil temperatures with no load no linkage. The percentage error is (0.2%) and this is excellent enhancement in comparison with previous results without FDV. the small error is due to the flow division error beside the friction factor in the spool .This approval appears obviously on the displacements curves which appears to be nearly identical.

The results in Figure (5-6,a,b,c) show there is no obvious effect of the oil temperature on the synchronization motion.

Case4: Synchronization Motion by Using Tee Connection, Flow Control Valve, Flow Divider Valve Circuits (Linkage without Load)

A comparison study between the three types of synchronization systems are shown in Figure (5-7) where linkage without load was used. (a) tee connection (b) throttle check (c) flow divider valve. The percentage error of tee connection was (2.2%) and for throttle was (1.8%). There is a little difference in throttle check curves from case2. Besides, an enhancement in tee system owing to the existing of the linkage for both systems which forces any possible lagging hydraulic cylinder. Also there is a tiny difference in the pressure curves. The change in the flow divider system is not sensible since the divider overcomes the linkage effect. So, the percentage error will be remained at (0.2%) and the pressure curves are identical.

Case5: Synchronization Motion by Using Tee Connection, Flow Control Valve, Flow Divider Valve Circuits (Linkage with Center Load).

Figure (5-8) shows a comparison study among the three types of synchronization systems where linkage with center load was used, (a) tee connection (b) throttle check (c) flow divider valve. The used load is 57.7(kg). In the tee connection case, there is a clear difference in the flow rate. Consequently, the increasing displacement curves have significant differences. Also there is variation in the pressure curve which starts at the zero point in the beginning of the actuation of the directional control valve reaching 4 (bar). It remains on this constant value through the extending stroke time. It rises then to the set pressure which have been 5 (bar). Besides that, there is a difference in the pressure between the two curves. The percentage error for tee connection was (3.3%). For the flow control valve circuit, the percentage error was (2.2%) which is apparently affected the displacement curves. For flow divider valve circuit, the percentage error was (0.4%) which is slightly affecting the displacement curves. The increase in the error in the three systems is owing to the existence of the load. The pressure curves in FDV on the both branches are nearly identical.

Case6: Synchronization Motion by Using Tee Connection, Flow Control Valve, Flow Divider Valve Circuits (Linkage with off Center Load).

The uneven load has a significant effect on the hydraulic cylinder displacement. The cylinder which is opposing less load rises first and the other one will be lagged in spite of the existing of the linkage. For this reason, the curves of flow rates, positions and pressures have all increased in difference, and hence the accuracy is decreased. In the tee connection case, the percentage error was (6.7%). While with flow control valve, the percentage error was (4.4%). For flow divider valve circuit, the percentage error was (0.7%). These results can be shown in Figure (5-9) where linkage with off center load was utilized. (a) tee connection (b) throttle check (c) flow divider valve.

Case7: Synchronization Motion by Using Conventional Circuit (Separate Load).

To show the effect of separate load (i.e., load without linkage) on the system, an addition of loads can be used. The load which was used was 144.2 (kg), or rather 72.1 (kg) on each actuator, so the corresponding pressure is 10 (bar). It is noticeable to mention that the area of the hydraulic cylinder which has been used in Matlab/Simulink, and also Automation Studio was calculated independent on this pressure as have been mentioned in appendix (C). The set pressure is 11 (bar) at ambient temperature 30 ^{o}C.

Figure (5-10,a,b,c) represents the behavior of the hydraulic system when tee connection was used to synchronize dual cylinders. It presented the tee connection position, flow rate and pressure versus time at different operating oil temperatures with separate load. The error was increased so that the percentage error became (13.3%) which is of a significant value. So the increasing linear positions curves differ corresponding to the flow rate changes. The pressure difference also increased. This is because that, when the tee connection was tested without linkage and load had percentage error (11.1%). So the addition of the load increased this error. The result shows that there was no obvious effect of the oil temperature on the synchronization motion.

Case8: Synchronization Motion by Using Flow Control Valve Circuit (Separate Load).

When flow control valve was used to synchronize dual cylinders, Figure (5-11,a,b,c) represents the results of its effect on the hydraulic system. It presents flow control valve position, flow rate and pressure versus time at different operating temperatures with separate load. The error increased so that the percentage error became (3.3%) which is significant value. So, the increasing linear position curves differ corresponding to the flow rate changes. The pressure difference also increased owing to the pressure loss across the valves orifices. The result shows there was no obvious effect of the oil temperature on the synchronization motion.

Case9: Synchronization Motion Using Flow Divider Valve Circuit with Separate Load.

Figure (5-12,a,b,c) represents the behavior of the hydraulic system when flow divider valve was used to synchronize dual cylinders. It presents flow divider valve position, flow rate and pressure versus time at different operating oil temperatures with separate load. The error was maintained at low rates values so that the percentage error was (0.4%). This is a small value and so the increasing linear position curves differ corresponding to the flow rate changes. The pressure difference also increased slightly because FDV overcomes flow division error in high quality. The result shows that there was no obvious effect of the oil temperature on the synchronization motion.

5.4 Comparison Among Experimental Work and Simulation of Automation Studio for FDV

A comparison between experimental and simulation results for flow divider valve system shows convergence in results clearly. Figure (5-6) is convergent to figure (5-2) for relationship between position, flow rate and pressure with time at no load case at different temps where the error is (0.2%). Figure (5-9-c) is convergent to Figure (5-3-c) for relationship between position, flow rate and pressure with time at uneven load where the error is (0.6%). Figure (5-12) is convergent to Figure (5-3-b) for relationship between position, flow rate and pressure with time at even load where the error is (0.4%).

The experimental results represent the real behavior of the system under many factors such as internal and external leakage, friction effects between the mechanical parts and the hoses difference length. The simulation results represent the simulation process for the system which depends on input data for all components in the simulation program. There is a little difference in the results between experimental and simulation results.

Figure (5-13, a, b, c) shows the comparison among AS, Simulink and the experimental results for FDV no load case. There is a convergence in displacement and pressure curves, while in flow rate curve there is a little difference between the Simulink and AS results. This is because the Simulink deals with a mathematical signal, while AS deals with physical signal.

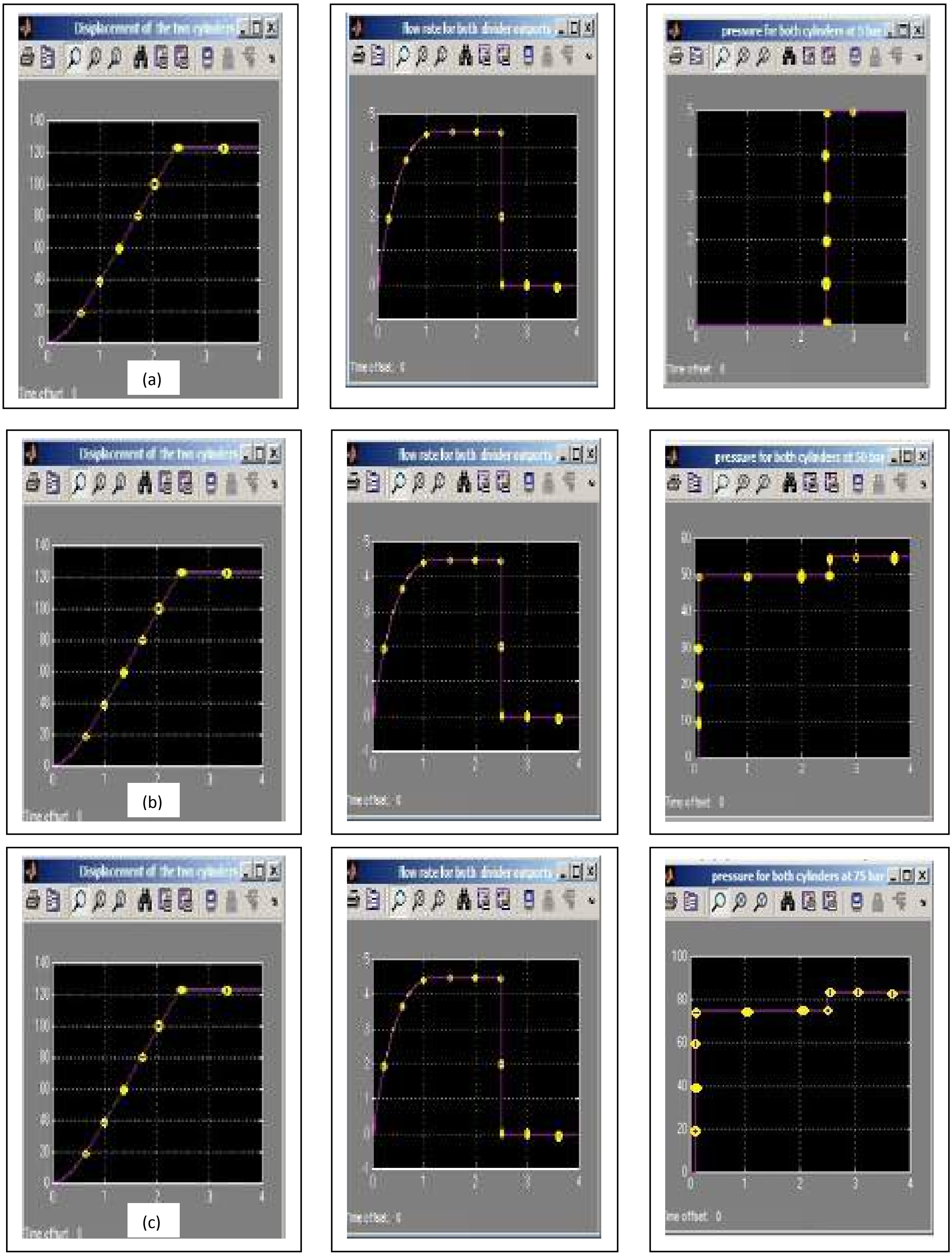


Fig. (5-1) Simulink results for flow divider valve position, flow rate, pressure versus time at different operating pressures. (a) Without load (b) at 50 bar(c) at 75 bar

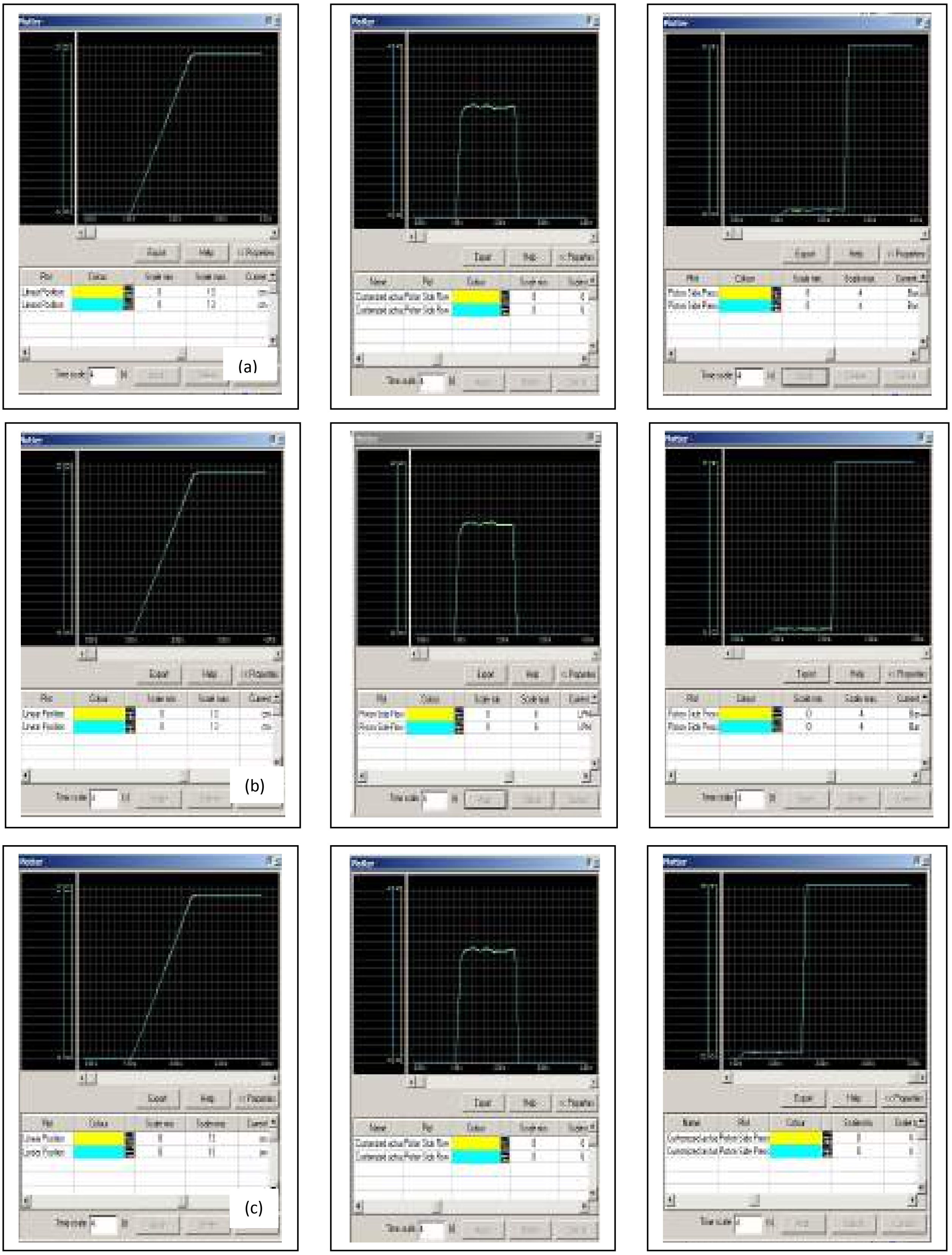


Fig. (5-2) AS results for flow divider valve position, flow rate, pressure versus time at different operating temperatures with no load. (a) at 40 ^{o}c , (b) at 45 ^{o}c (c) at 50 ^{o}c

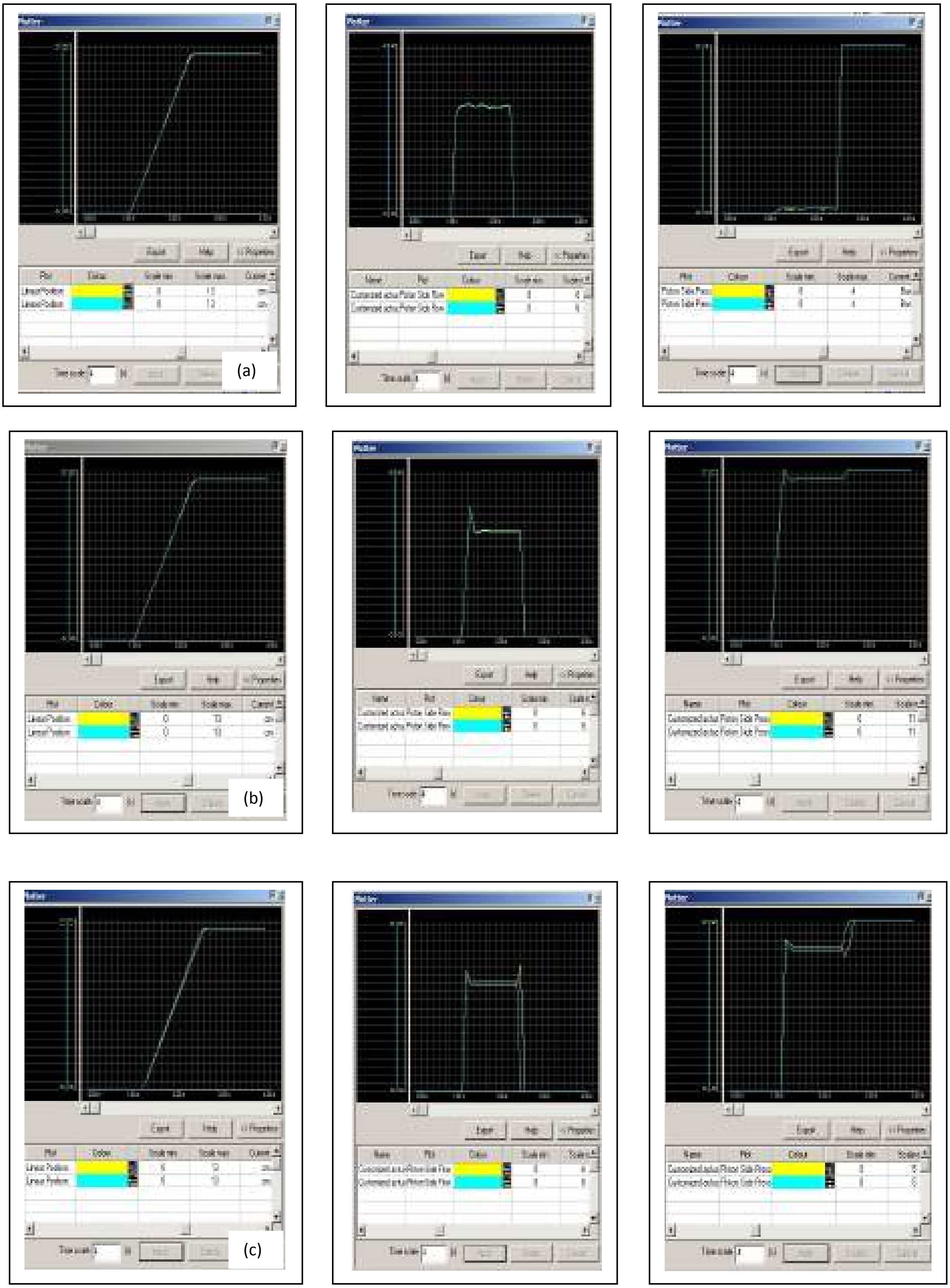


Fig. (5-3) AS results for flow divider valve position, flow rate, pressure versus time at different loads. (a) No load, (b) Even load (c) Uneven load

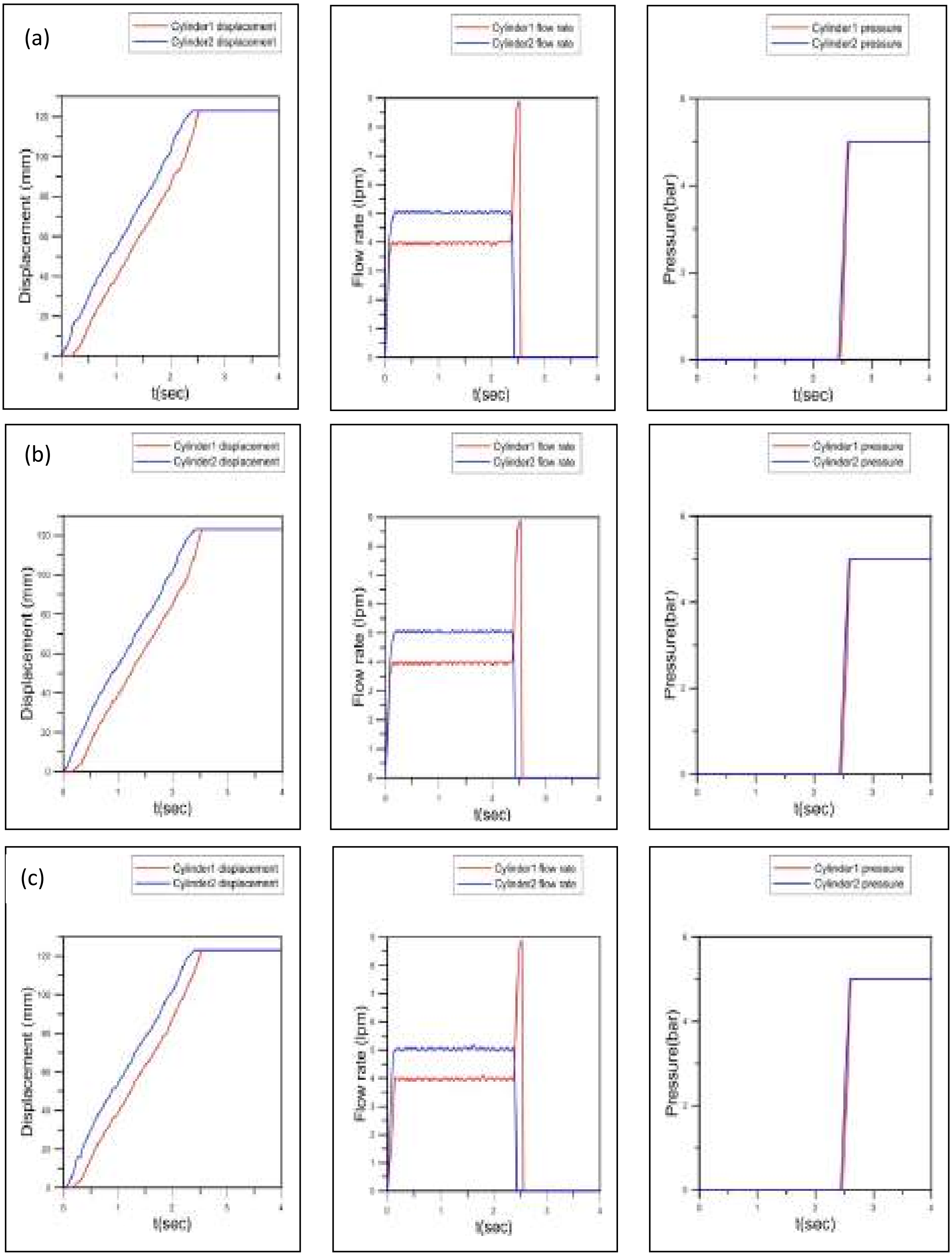


Fig. (5-4) Tee connection position, flow rate, pressure versus time at different operating temperatures with no load no linkage.(a) at 40 ^{o}c , (b) at 45 ^{o}c (c) at 50 ^{o}c

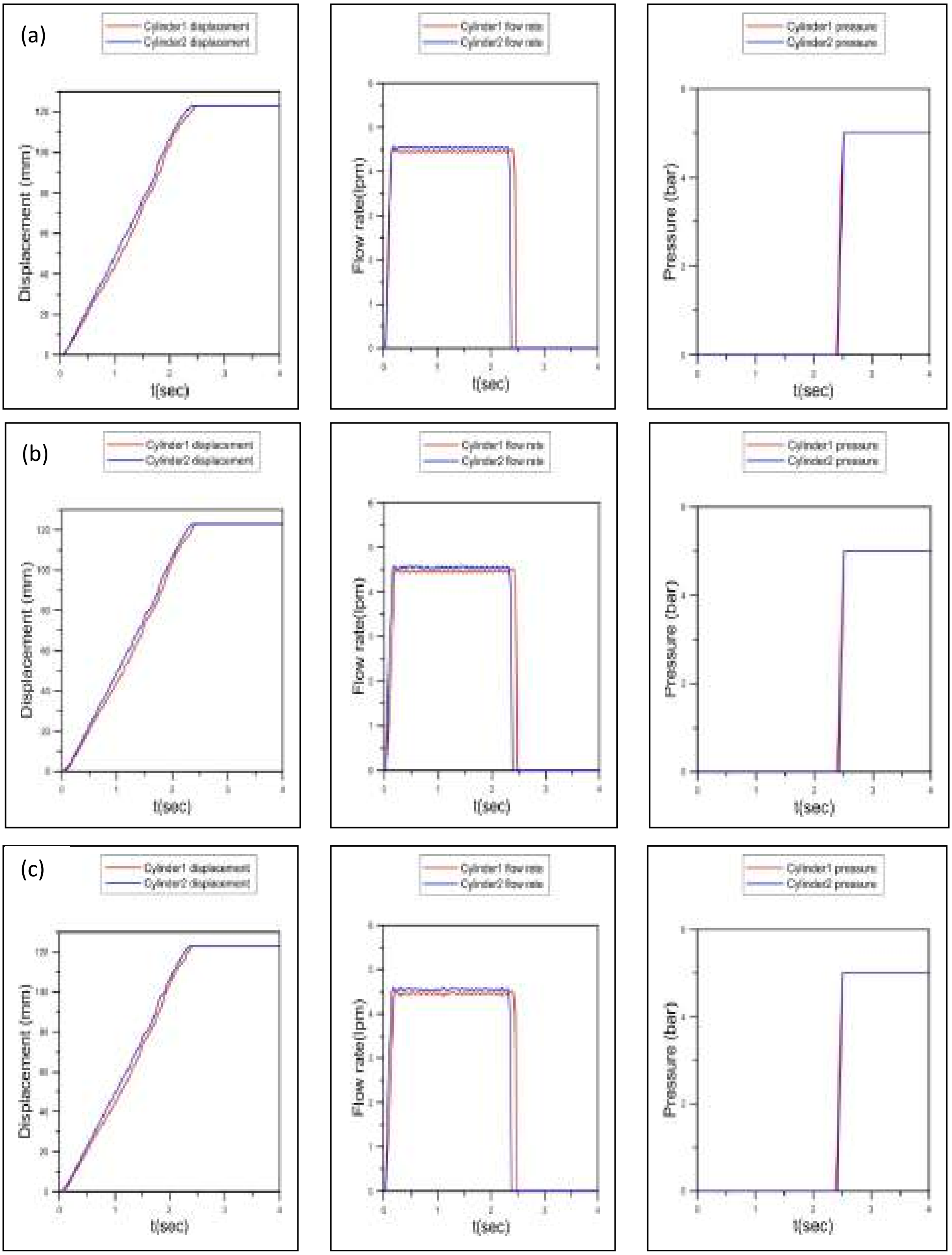


Fig. (5-5) Throttle position, flow rate, pressure versus time at different operating temperatures with no load no linkage.(a) at 40 ^{o}c , (b) at 45 ^{o}c (c) at 50 ^{o}c

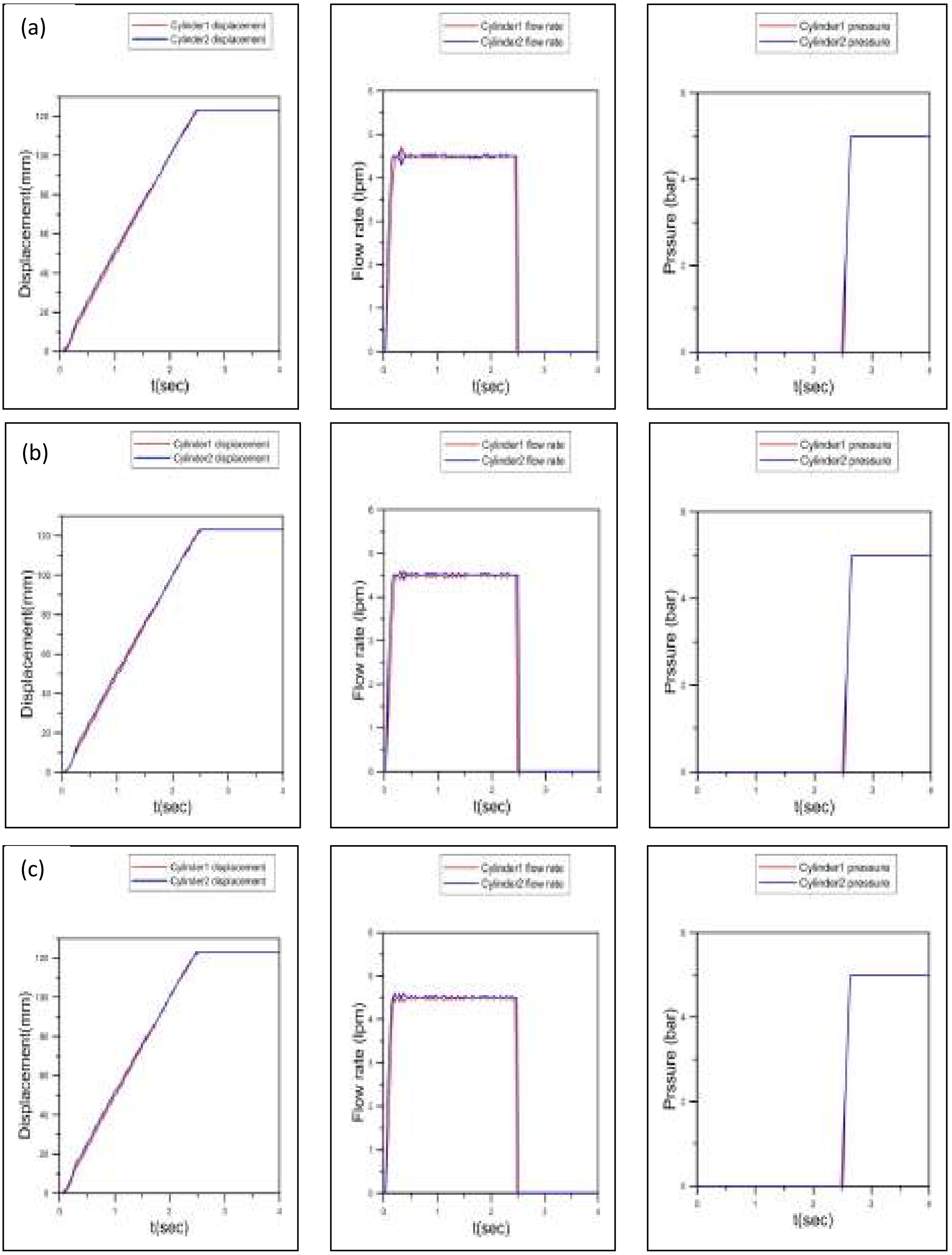


Fig. (5-6) Flow divider valve position, flow rate, pressure versus time at different operating temperatures with no load no linkage.(a) at 40 °C , (b) at 45 °C (c) at 50 °C

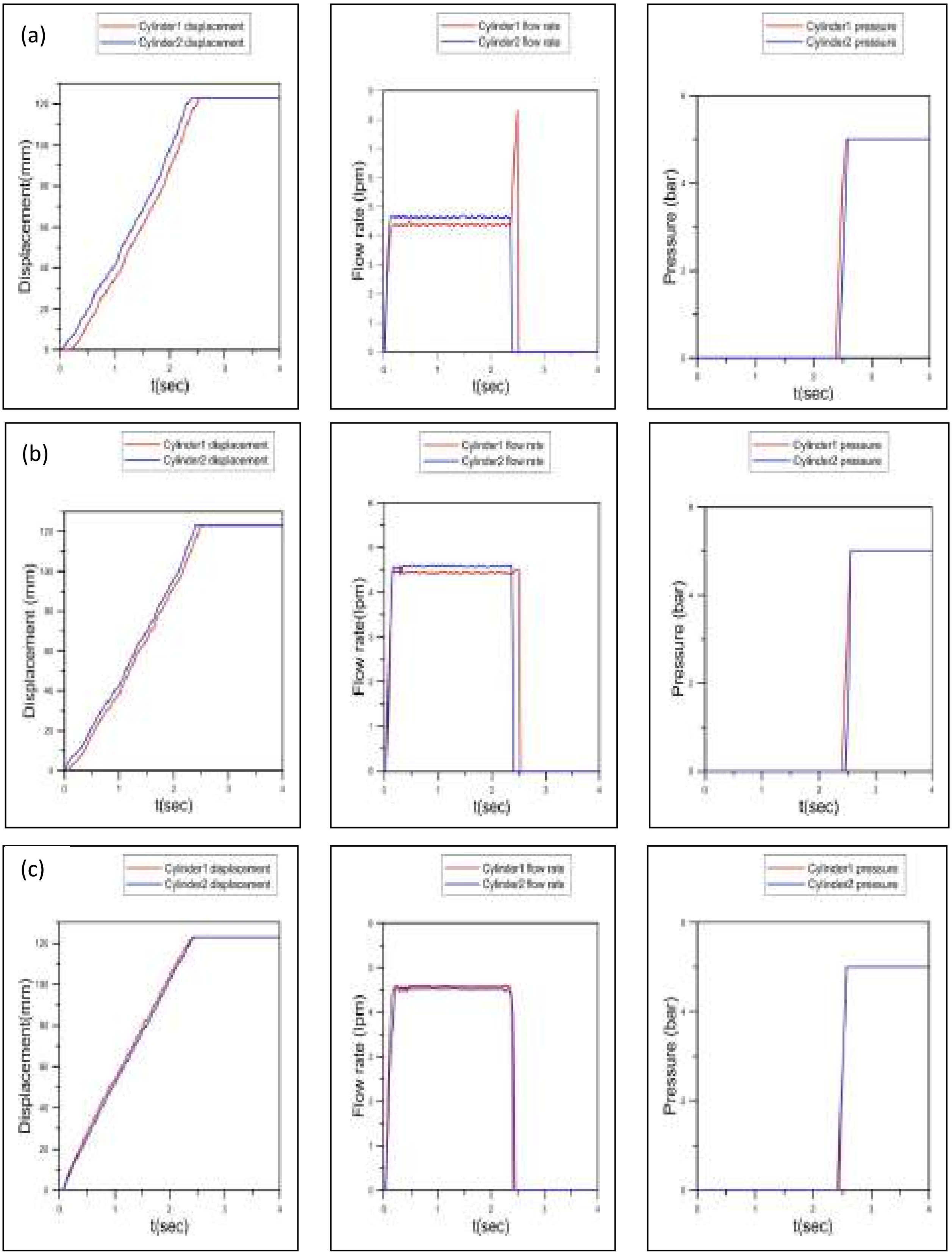


Fig. (5-7) Position, flow rate, pressure versus time with linkage without load. (a) Tee connection, (b) Throttle check (c) Flow divider valve.

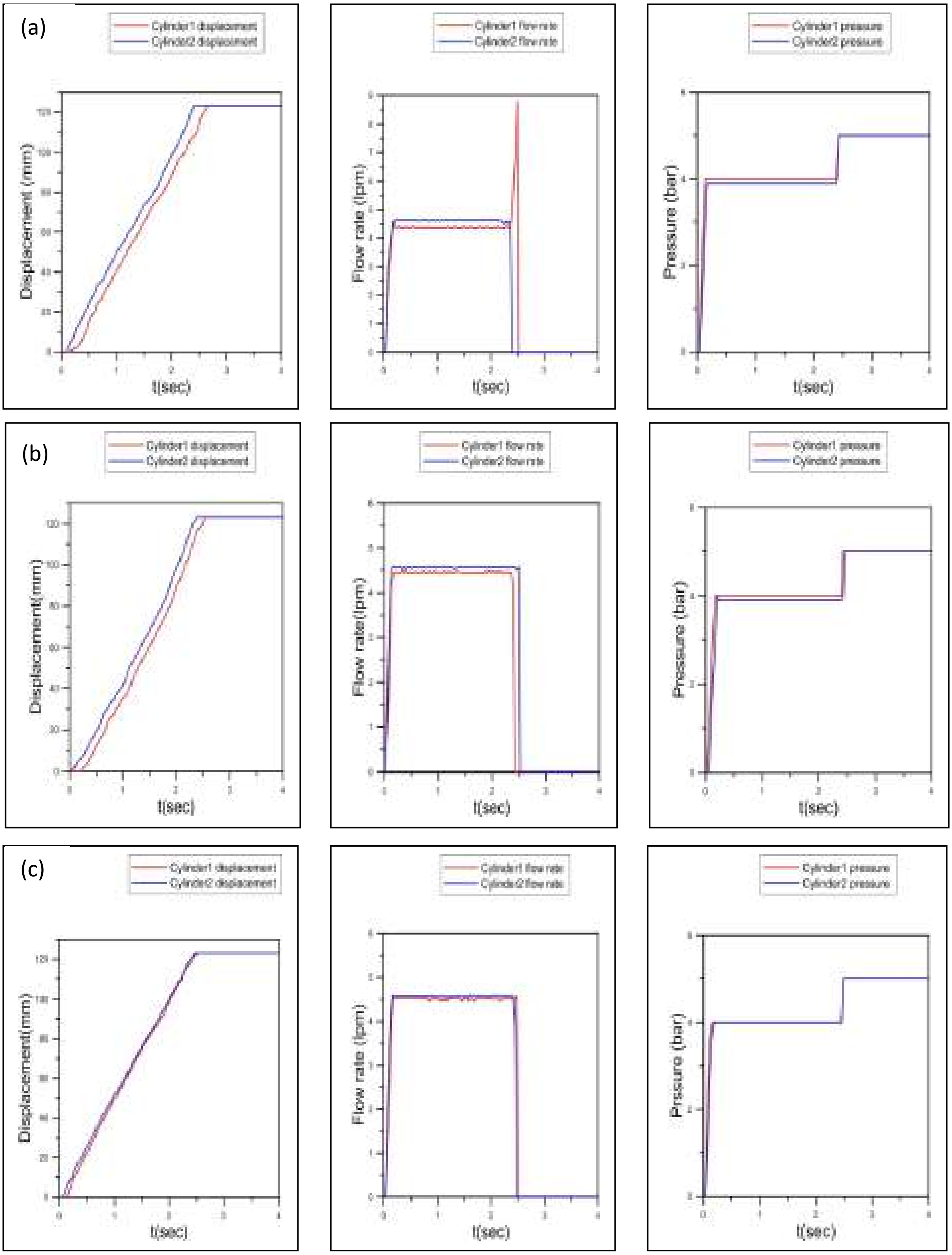


Fig. (5-8) Position, flow rate, pressure versus time with linkage with center load. (a) Tee connection, (B) Throttle check (C) Flow divider valve

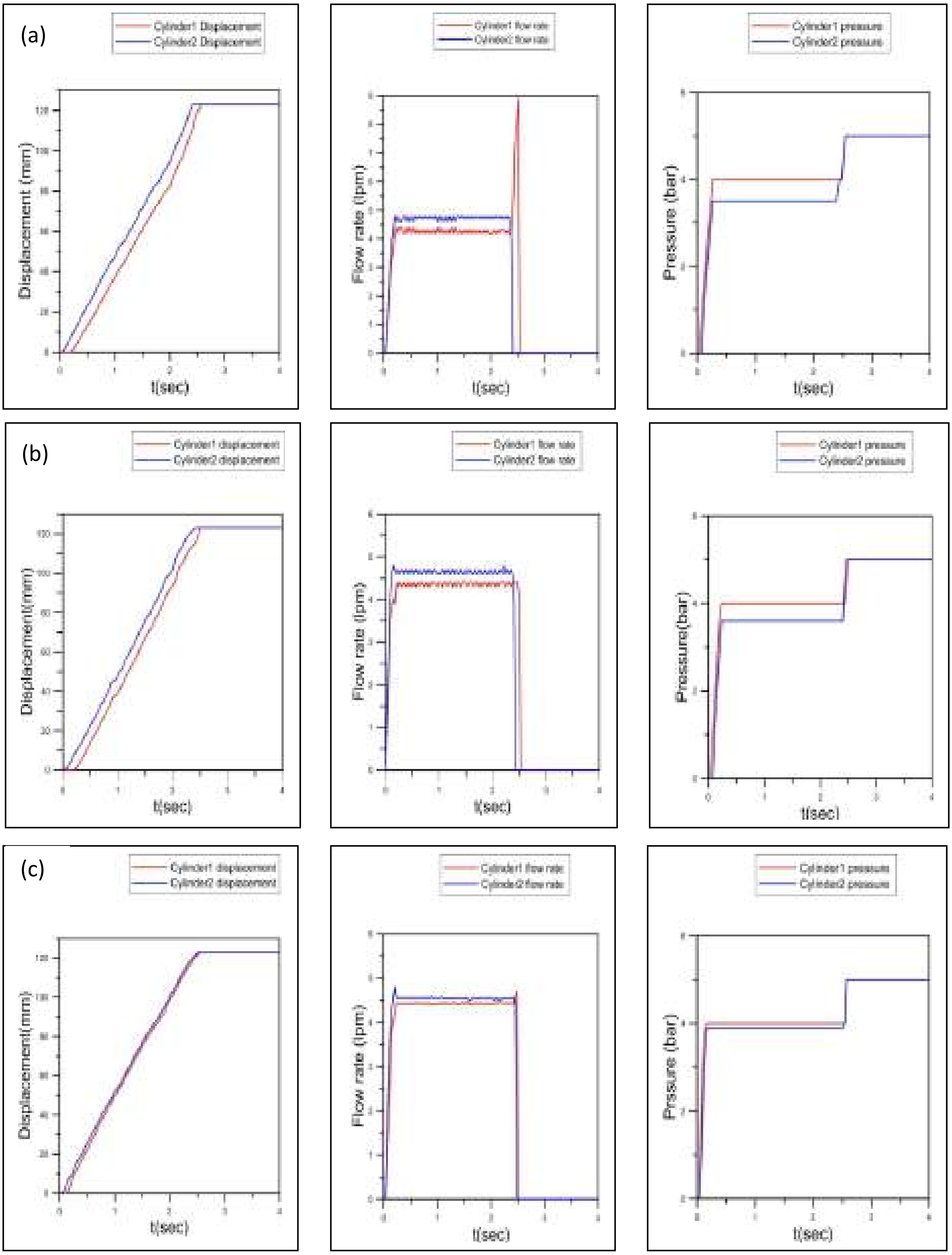


Fig. (5-9) Position, flow rate, pressure versus time with linkage with off center load. (a) Tee connection, (b) Throttle check (c) Flow divider valve.

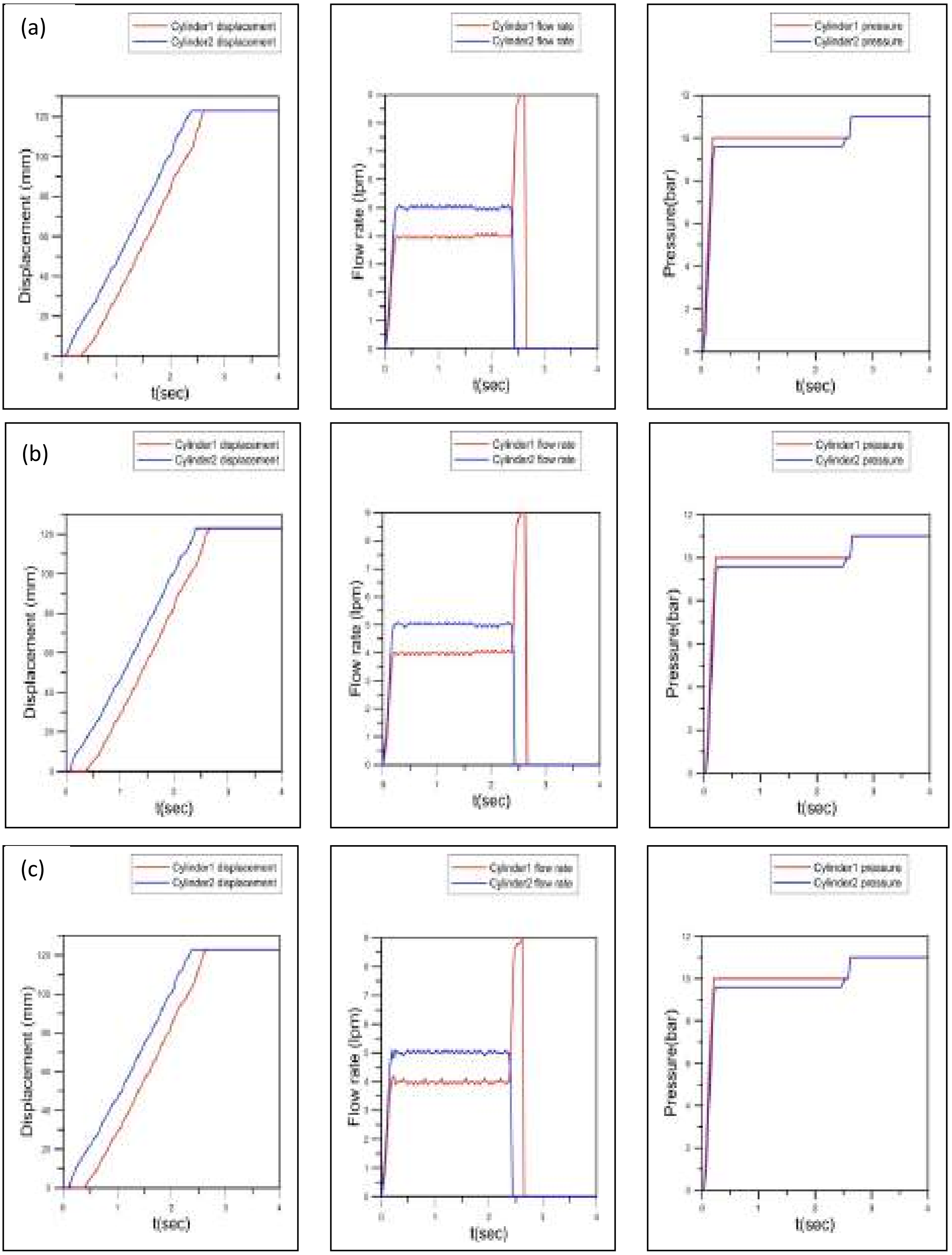


Fig. (5-10) Tee position, flow rate, pressure versus time at different operating temperatures with separate load.(a) at 40 ^{o}C , (b) at 45 ^{o}C (c) at 50 ^{o}C

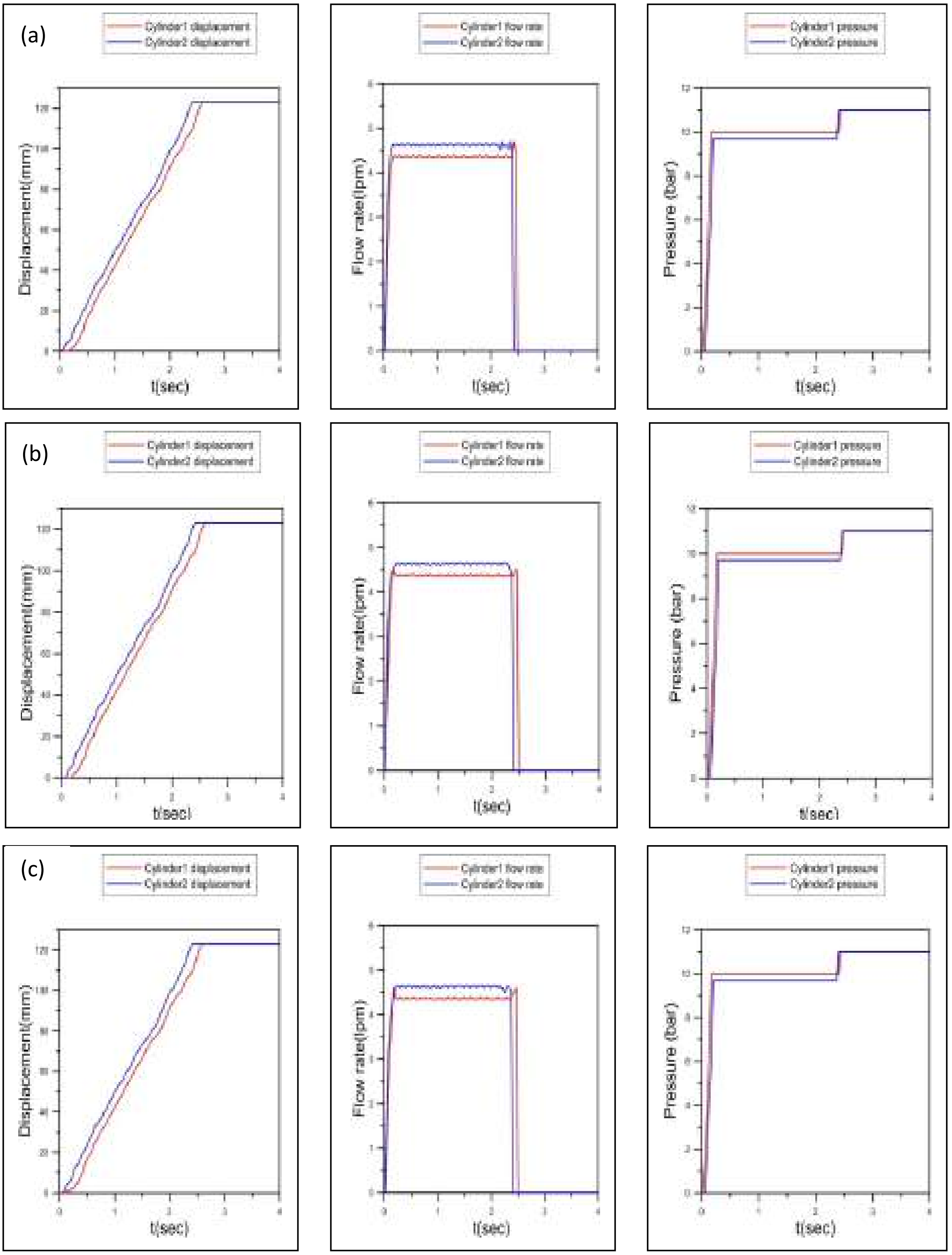


Fig. (5-11) Throttle position, flow rate, pressure versus time at different operating temperatures with separate load (a) at 40 ^{o}C (b) at 45 ^{o}C (c) at 50 ^{o}C

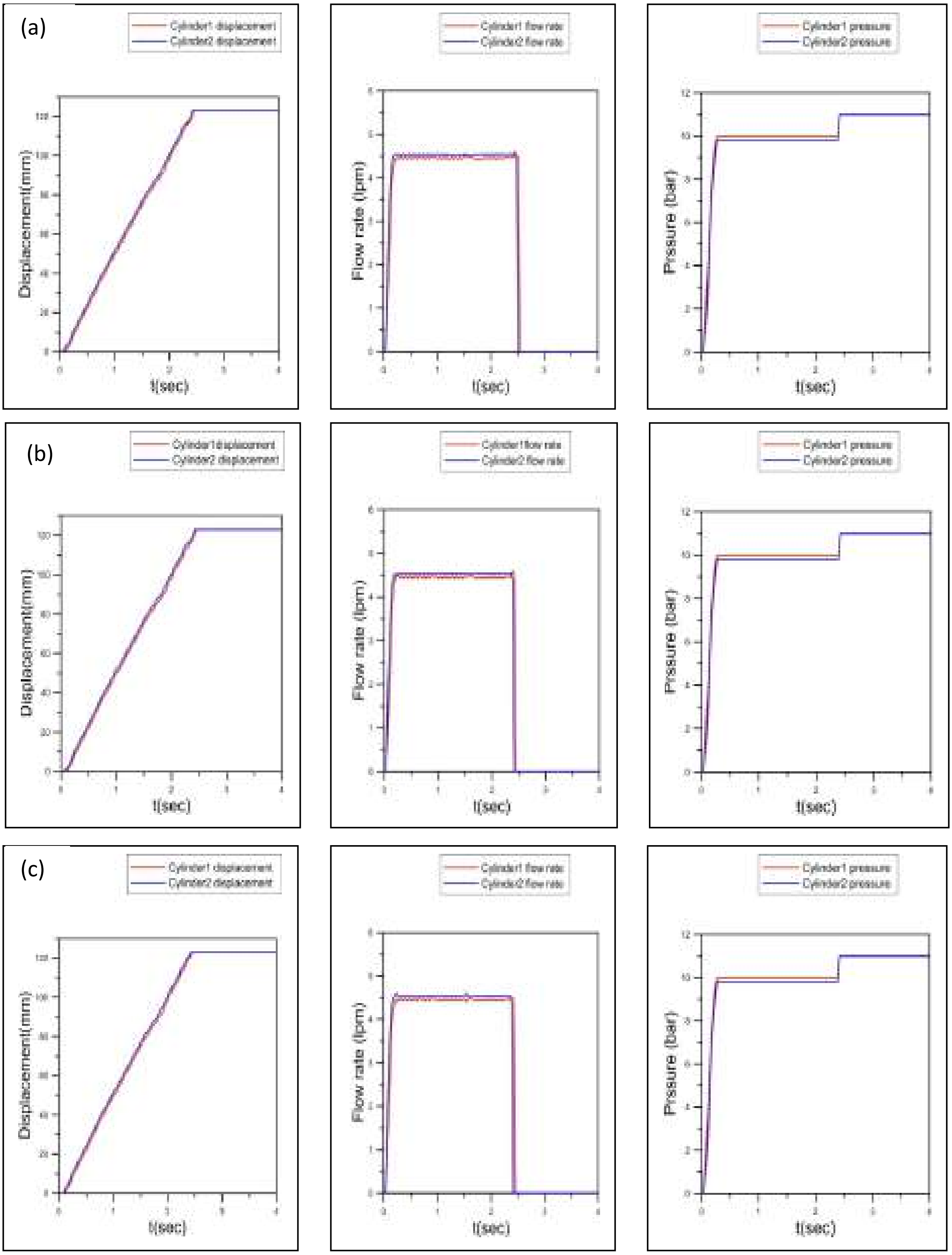


Fig. (5-12) Flow divider valve position, flow rate, pressure versus time at different operating temperatures with separate load. (a) at 40 °C (b) at 45 °C (c) at 50 °C.

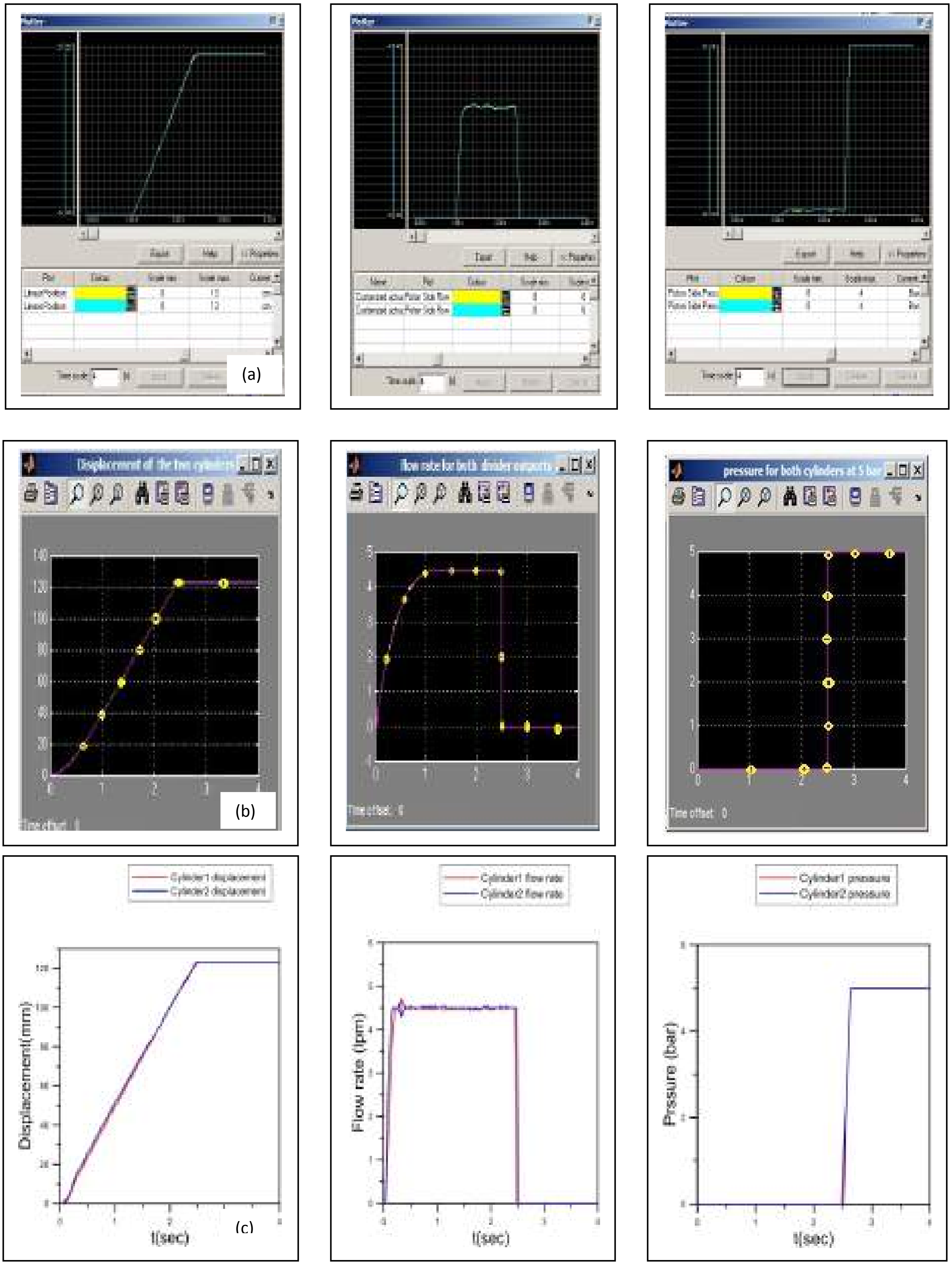


Fig. (5-13) Results for flow divider valve position, flow rate, pressure versus time at no loads. (a) AS results (b) Simulink results (c) Experimental results

Chapter Six
Conclusions and Recommendations

6.1 Conclusions

Both, the present theoretical simulation analysis and the experimental investigations reveal many conclusions. These are summarized as follows:-

1. One of the best ways to synchronize multiple hydraulic cylinders is by using flow divider valve which shows a high accuracy for moving elements of output devices.
2. Increasing or decreasing the operating hydraulic oil temperature within a range of ($\pm 5^{o}$c) from the optimum temperature doesn't affect the synchronization system.
3. The compressibility factor of hydraulic oil has no affect on the synchronization system. This is because the operating pressure is less than 1000 psi.
4. Increasing the loads or uneven loads on the experimental part of the hydraulic actuator leads to an increase in the displacement error.
5. Both, the experimental results and simulation using Automation Studio program are obviously convergence. This leads to the possibility of utilizing this program for testing, analyzing and designing of any hydraulic system. The (AS) package can be used as an educational system for the design of hydraulic systems.

6.2 Recommendations

The most important recommendations which must be taken into considerations for future work to study the synchronization system in different applications of hydraulic system as follows:-

1. Increasing the range of the operating hydraulic oil temperature over than 60 ^{o}c to in order to show its affect in high ranges.
2. Another field of research that could be investigated is that of divider/combiner valves so as to get rid of using two check valves for the return line in the extending stroke.
3. Using electro hydraulic valves instead of flow divider valve for more accurate position of synchronization of hydraulic actuators. Especially one servo valve for cylinder and making this cylinder fallow up the other one.

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Appendix A

Calibration Curves

The measurement instrument such us pressure sensors , thermometer ,flow sensors and position sensors should be calibrated before taking any measurements process to avoid measurements containing very large offset, gain and linearity errors.

A-1 Flow Sensors

The flow sensor was calibrated by plotting its reading against the really measured flow rate. This done by accumulating hydraulic oil in graduated cylinder of (1 Letter) volume and dividing this reading on the filling time measured by stop watch as shown in Figures (A -1) and (A -2).

A-2 Position Sensors

The position sensor was calibrated by electronic digital caliper by affirming the electronic digital caliper together with the position sensor on the cylinder rod. Then move the free portion to the beginning of the cylinder stroke, setting this point as the zero point (start point). Start moving the cylinder and recording the displacement by the position sensor to compare it with the digital recorded, as shown in Figures (A-3) and (A-4).

A-3 Temperature Measurement Accuracy

Table (A-1): Accuracy of the temperature

Range	accuracy	resolution
-20 ^{o}C-400 ^{o}C	± (1%)	.1^{o}C
400^{o}C-1000^{o}C	± (1.5%)	1^{o}C

A-4 Calibration of Flow Control Valve

The flow control valves which are used in the present work have seven cycles for each one from closure to full opening. The flow characteristics of the control valve which is shown in Figure (A-5) have been adjusted at six cycles in order to have 4.5 (lpm) the error was 1.5%.

A-5 Calibration of Flow divider valve

Flow divider valve has been tested in Al-Fedaa Company to apply high pressure and high flow rate. It is calibrated by measuring flow rate for each port. This done by connects flow meters in the output ports of the FDV. The flow division ratio is (0.1%) as shown in Figure (A -6).

A-6 Pressure gage

The pressure gage with the range of (0-100) bar was calibrated by compare its reading with the pressure sensor which in its rule was calibrated in the Labs of Central Organization for Standards and Quality Control (C.O.S.Q.C). A copy of calibration certificate for this pressure sensor is shown in Figure (A-9). This sensor Contain LCD display, so we can compare its reading with the pressure gage as shown in Figure (A-7).

A-7 Pressure sensors:

The calibration of the pressure sensors were done in Labs of Central Organization for Standards and Quality Control (C.O.S.Q.C). A copy of calibration certificate for these pressure sensors are shown in Figure (A-8).

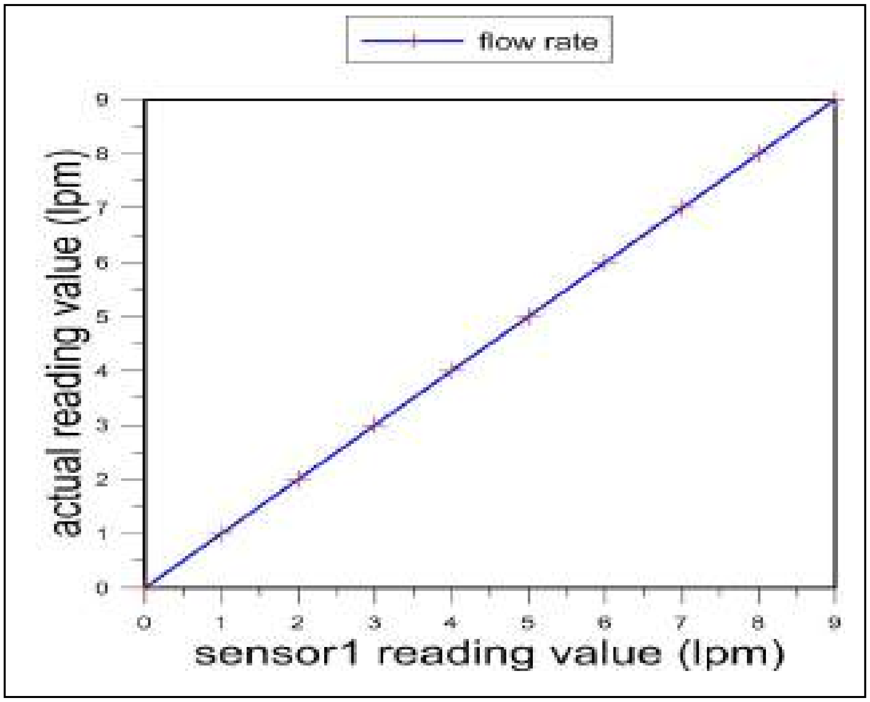

Fig. (A -1): Calibration curve of flow sensor1.

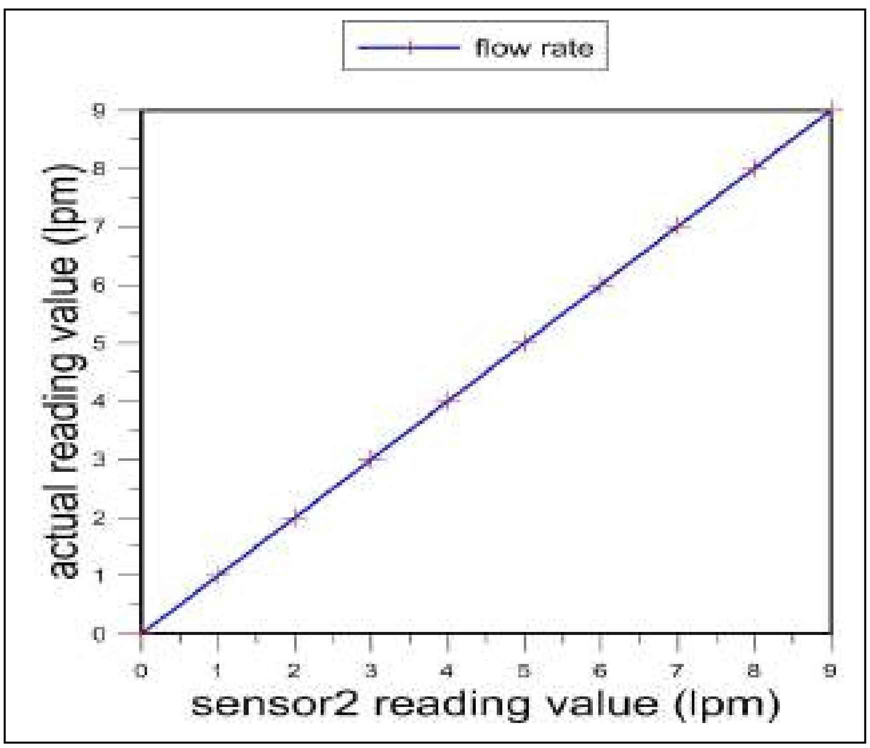

Fig. (A -2): Calibration curve of flow sensor2.

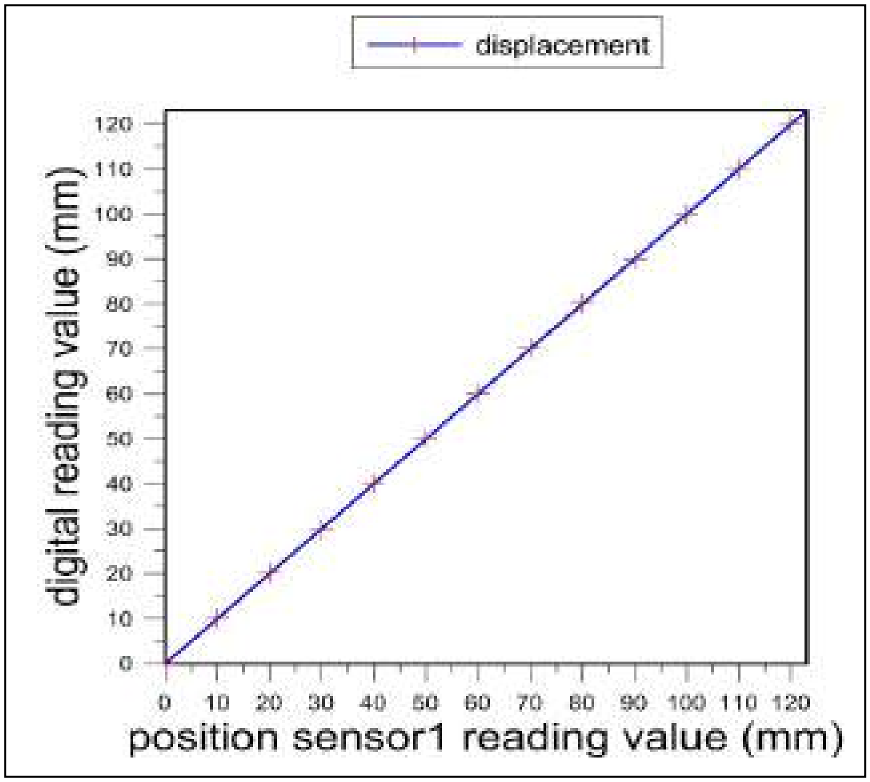

Fig. (A-3): Calibration curve of position sensor1

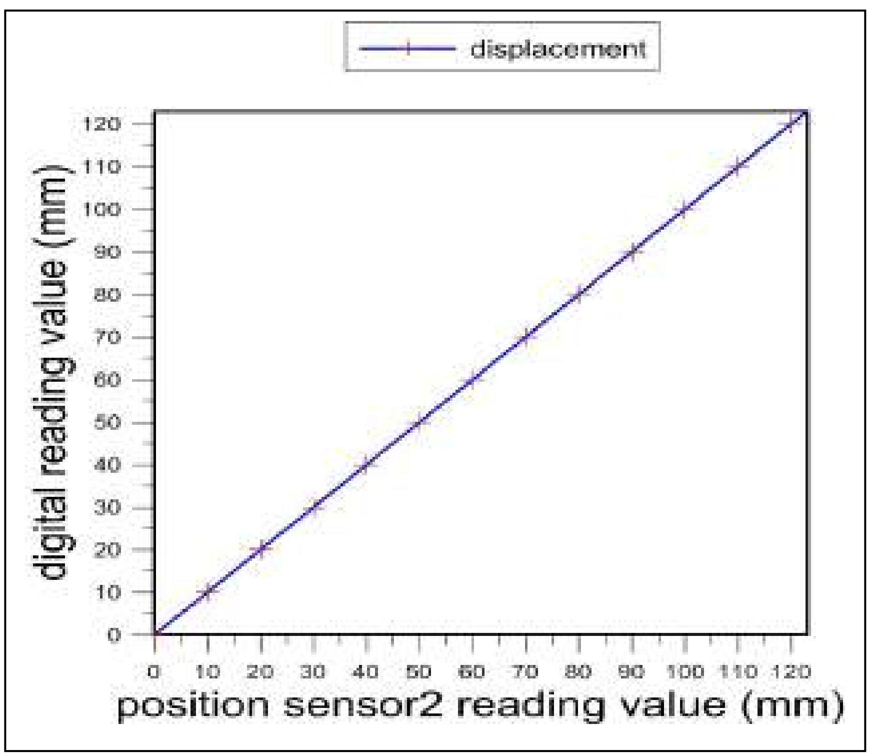

Fig. (A-4): Calibration curve of position sensor2

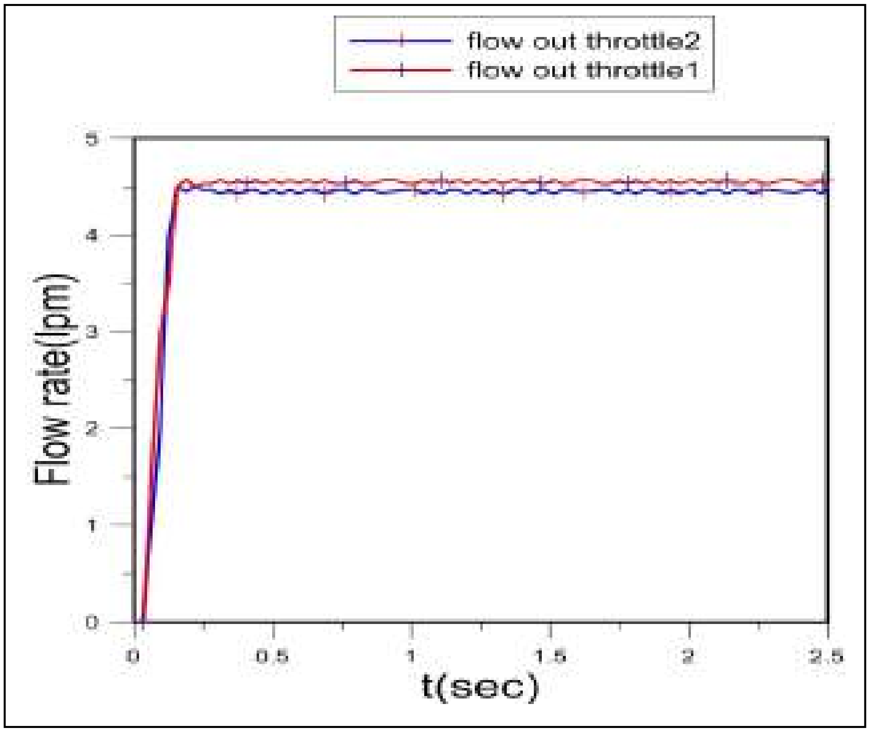

Fig. (A-5): Calibration curves of flow control vales

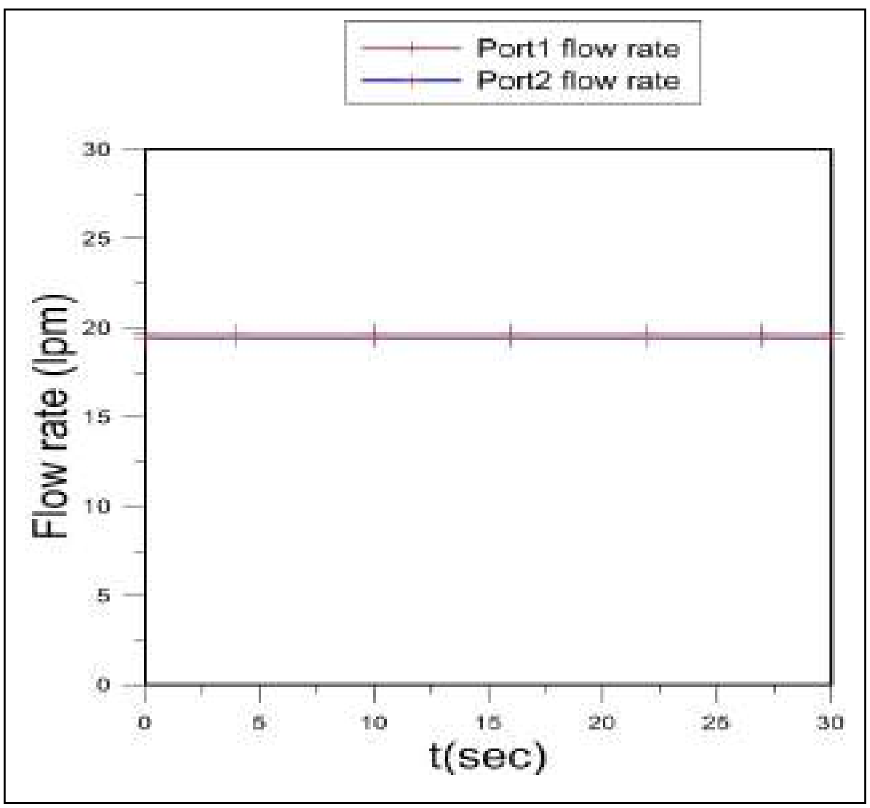

Fig. (A-6): Calibration curve of the flow divider valve

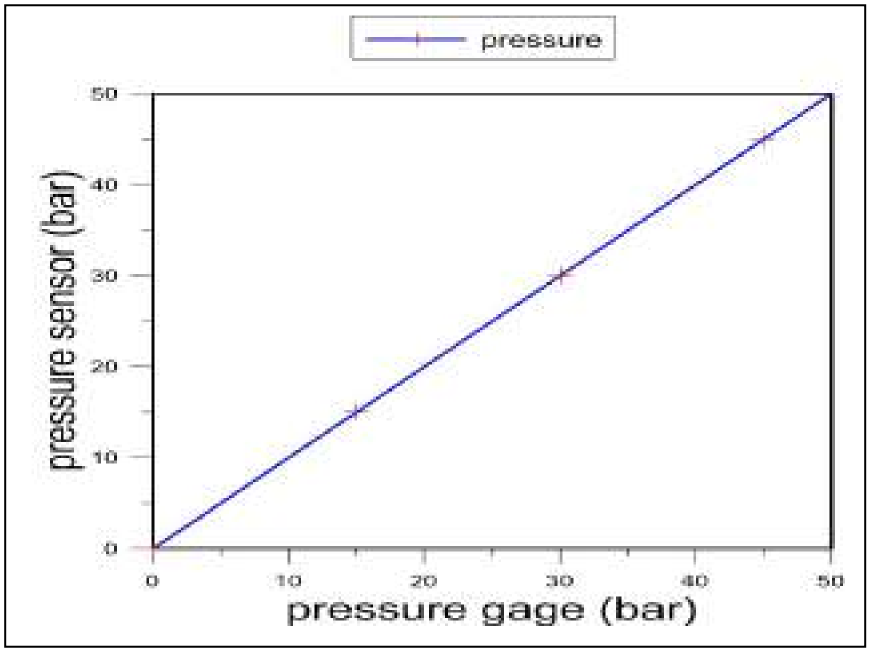

Fig. (A-7): Calibration curve of the pressure gage

Central Organization for Standardization and Quality Control

الجهاز المركزي للتقييس والسيطرة النوعية

CALIBRATION CERTIFICATE

شهادة معايرة

Calibration Results (Pressure Gauge)

Results

Central Organization for Standardization and Quality Control

الجهاز المركزي للتقييس والسيطرة النوعية

CALIBRATION CERTIFICATE

شهادة معايرة

Calibration Results (Pressure Gauge)

Results

Fig. (A-8) : Copies of the calibration certificates of the pressure sensors

Appendix B

Sample of calculations

B-1 Calculation of the Area of the Hydraulic Cylinder

$$A_p = \frac{M.g}{p} \quad \text{(B.1)}$$

$$A_p = \frac{72.1 * 9.81}{1000425.743} = 0.000707\ m^2 = 7.07\ cm^2$$

$$A_p = \frac{\pi}{4} D^2$$

$$D = 3\ cm$$

$$A_r = \frac{\pi}{4}(D^2 - d^2)$$

$$A_r = 3.93\ cm^2$$

$A_{P=}$piston side area m^2, g=gravitational acceleration (m/s^2)

M=external mass (kg), p =piston side pressure (pa)

D = piston diameter (cm), d=rod side diameter (cm)

Where: d=2(cm) experimentally measured

B-2 Calculation of the Volume of the Hydraulic Cylinder

Vcyl. =A_p.L

Vcyl.=7.07*12.3=86.961 cm^3

B-3 Calculation of the Volume of Oil in the Hoses

- Length of hose1 from pump to DCV. (10 cm) , diameter (0.9525 cm)
- Length of hose2 from DCV to FDV. (40 cm) , diameter (0.9525 cm)
- Length of hose3 from FDV to cylinder. (75 cm) , diameter (0.9525 cm)
- Length of hose4 from cylinder to Tee. (50 cm) , diameter (0.9525 cm)
- Length of hose5 from Tee to DCV. (75 cm) , diameter (0.9525 cm)
- Total length of fitting connection. (20 cm) , diameter (0.9525 cm)
- Length of the flow sensor. (12 cm) , diameter (0.9525 cm)

$A_h = \pi/4 * d_h^2$ .... (B.2)

$A = 0.712557\ cm^2$

$V = A*L$ .... (B.3)

d_h = hose diameter (cm) , L = Hose length (cm) , A_h =Hose area

$V_s = (0.712557)\ (10) = 7.12557\ cm^3$

$V_L = (0.712557)\ (40+20) = 42.75344\ cm^3$

$V_{L1} = V_{L2} = (0.712557)\ (75+22) = 69.11807$ cm3

$V_{r1} = V_{r2} = (0.712557)\ (50) = 35.62786962$ cm3

$V_r = (0.712557)\ (50) = 53.44180444$ cm3

B-4 Calculation of the Total Volume of Oil in the Hoses

$V_t = 2*V_{cyl.} + V_Z + 2*V_{L1} + 2*V_{r1} + V_r$

$= 2*86.961 + 42.75344 + 2*69.11807 + 2*35.62786962 + 53.44180444$

$V_t = 479.609\ cm^3$

Where: V_t =Total hydraulic oil volume circulating in the hydraulic system cm^3

Appendix C

Matlab Simulink m. files

This appendix includes copy of the Matlab m-files which has been used to set the base parameters for the Simulink model. Some values come from experimentally work, other researches and data sheets.

```
% This sets the base parameters for ALL simulations

Dp=1E-6;              % m^3/rad
N=1410;               % RPM
B=7E+6;               % N/m^2
rho=873;              % kg/m^3
Vs=7E-6;              % m^3
Va=7E-6;              % m^3
Va1=7E-7;             % m^3
Va2=7E-7;             % m^3
VL=42.7E-6;           % m^3
VL1=69.1E-6;          % m^3
VL2=VL1;              % m^3
Vr1=35.6E-6;          % m^3
Vr2=Vr1;              % m^3
X0=0;                 % m
Xp1dot0=0;            % m/sec.
Xp10=0;               % m
Xp2dot0=0;            % m/sec.
Xp20=0;               % m
Xs0=0;                % m
Xvdot0=0;             % m/sec.
Xsdot0=0;             % m/sec.
Xvo=0;                % m
Zo=0;                 % m
Zdot0=0;              % m/sec.
Ap=7.07E-4;           % m^2
Ar=3.93E-4;           % m^2
A=0.7E-4;             % m^2
Aref=0.5E-4;          % m^2
```

```
As=0.5E-4;            % m^2
Ao1=.07E-4;           % m^2
Ao2=.07E-4;           % m^2
Cd=.61;               % dimensionless
Cdo1=.6;              % dimensionless
Cdo2=Cdo1;            % dimensionless
Cdv1= .6;             % dimensionless
Cdv2=Cdv1;            % dimensionless
Cr=.62;               % dimensionless
ff=80;                % N.sec/m
fp=150;               % N.sec/m
fr=0;                 % N.sec/m
fs=56.4;              % N.sec/m
Ps0=0;                % N/m^2
Ps=4E+5;              % N/m^2
Pa0=0;                % N/m^2
Pa10=0;               % N/m^2
Pa20=Pa10;            % N/m^2
PL0=0;                % N/m^2
PL10=0;               % N/m^2
PL20=PL10;            % N/m^2
Pr10=0;               % N/m^2
Pr20=0;               % N/m^2
Po=0;                 % N/m^2
Kf=18 E+3;            % N/m
Kref=20+3;            % N/m
Ks=16 E+3;            % N/m
M=750;                % kg
Mr=.1;                % kg
Ms=.2;                % kg
w=0.01;               % m
```