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THEORETICAL AND EXPERIMENTAL INVESTIGATIONS OF DYNAMIC BEHAVIOR OF AN ELECTRO-HYDRAULIC DIRECTIONAL PROPORTIONAL VALVE

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ABSTRACT

Hydraulic systems are widely used in many fields. Controlling of hydraulic systems is carried out through several techniques. Proportional technique is considered one of the important and advanced techniques used to control hydraulic systems. Through this paper a complete model for a directional proportional valve has been built up using SIMULINK program to investigate the dynamic behavior of this class of valves. Experimental work has been conducted to measure the force exerted by the proportional solenoid and the flow rate through the directional proportional valve at different values of solenoid current. The experimental results have been used to validate the simulation program. The dynamic behavior of the electro-hydraulic proportional valve could be investigated using the obtained simulation program.

KEY WORDS

Hydraulic, System, Valves, Directional, Proportional, Solenoid, Control, Simulation, Modeling, Investigation, Program, Measurement Pressure, Flow rate and SIMULINK.

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1. INTRODUCTION

Fluid power is involved in virtually every phase of industry, including manufacturing, transportation and construction. Hydraulic systems are required in many applications. However, the real world of fluid power lies in the domain of electric and electronic command, control, and sensing. In many cases, simple manual control systems are totally adequate for the operation, even where a considerable amount of complexity is required. The use of electro-hydraulic enhances operator safety and reduces operator fatigue while providing improved controllability. The ability of achieve automated step less control of pressure and flow rate in fluid power systems has undergone major development in the past years.

Electro-hydraulic servo valves were invented in the late 1930s as a high technology solution to the motion control which needs very high cost. The mid 1980s saw the practical introduction of proportional valves as a viable and reasonably priced alternative to servo valves. Step less control of pressure or flow rate was not that critical to the operation of a machine, preset pressure or flow control valves could be achieved by having a bank of preset valves. The appropriate valve would be connected into the circuit via the actuation of a solenoid valve. For example, three discrete pressures could be achieved by having two pilot relief valves connected in parallel to the vent port of a vent-able pilot operated relief valve. These two valves will be isolated from the pilot operated relief valve by two-way normally closed direction control valve operated with solenoid. By individually actuating the two way valves, three different pressures could be achieved.

If the need existed for achieving varying pressure to an actuator in an effort to control force or torque, one needs to either a machine operator turns an adjustment knob, stroke a lever, or a mechanical means has to be designed to get a mechanical input device or linkage varies the setting of the valve. The same need holds true if the flow has to be varied. All these problems have been solved by the invention of servo valves. But due to the high cost of servo valves as a result of the extremely high manufacturing precision, the need of moderate cost alternative technology appeared. The invention of proportional valves, some kind, solved this problem.

Electro-hydraulic proportional valves (EHPVs) move up from the relatively low-tech world of simple on-off controls into a higher technological plane at which more sophisticated valves are operated by electronics rather than just electrical switching. The presences of electro-hydro proportional valves (EHPVs) fill the great gap between the on-off valves and servo valves. The great advantage of EHPVs, relatively to the on-off valves, is the flexibility in system design and operation as well as the decrease in fluid power circuit complexity for process requiring multiple speeds or multiple force output. There are two definitions for the proportional valve; firstly, it is any valve produces an output proportional to an electronic control input; secondly, it is any valve operates by proportional solenoid rather than on-off solenoid or torque motor. There are three categories of proportional valves: flow proportional valve, pressure proportional valves, and directional proportional valves. The electro-hydraulic system, under investigation, encloses a directional proportional control valve, so the theoretical and experimental investigations of that type of proportional valves will be discussed in details.

2. EXPERIMENTAL INVESTIGATION

All experiments in the present work have been carried out by using a hydraulic test bench, shown in Fig. 1, at the hydraulic laboratory of the mechanical power and energy department in the military technical college, the hydraulic test bench consists of the following parts: Oil tank (1) of capacity 20 liters contains a hydraulic gear pump, electrical motor, oil filter, and level indicator; Four Tank ports (2); Four Pump ports (3), Grid panel (4); Pressure Gauge (5); Loads of a vertical hydraulic cylinder (6); Vertical hydraulic cylinder (7).

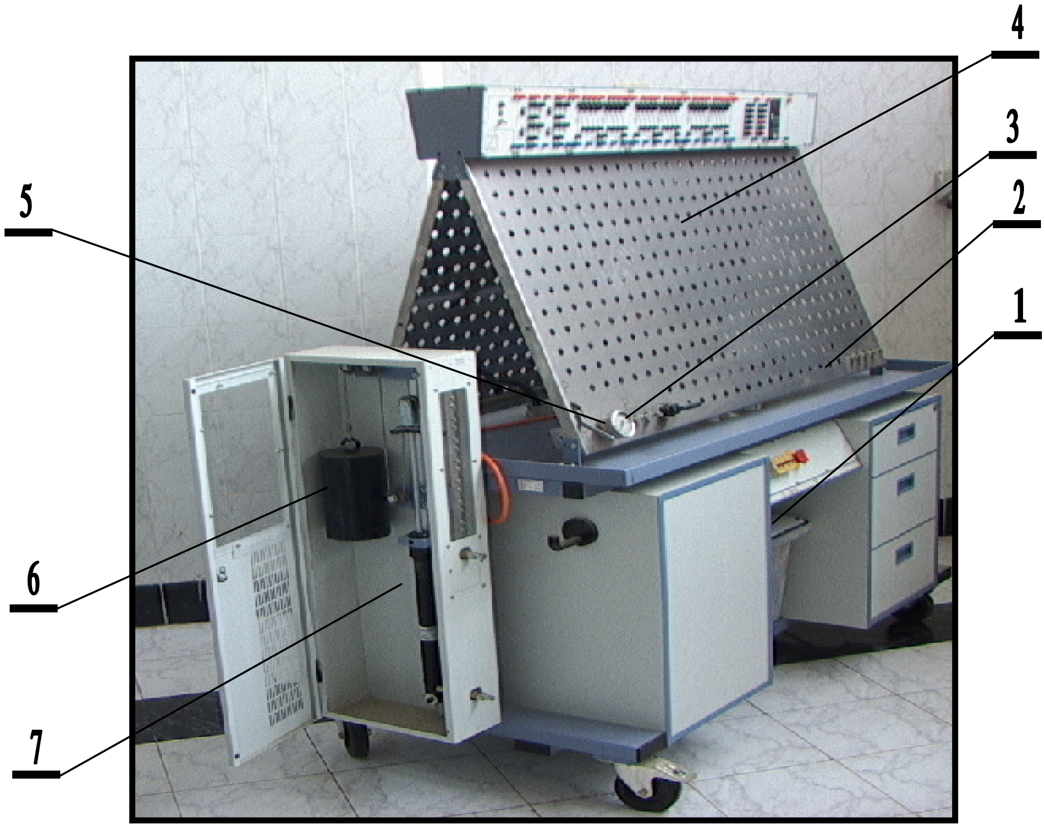

Fig.1. Hydraulic test bench.
Oil tank (1), Tank ports (2), Pump ports (3), Grid Panel (4), Pressure gage (5), load (6), Hydraulic cylinder (7).

Two experiments have been performed:

1- Measurement of the force exerted by the proportional solenoid versus solenoid current.
2- Measurement of flow rate through the directional proportional valve under constant pressure difference versus solenoid current.

2.1. Experiment #1: Force Measurement of Proportional Solenoid Versus Current at Different Air Gaps

Modeling of the direction proportional valve has been performed in two steps. At 1st step, modeling of the proportional solenoid and at 2nd step, modeling of the motion of spool inside the valve body. The mathematical model of the proportional solenoid is difficult to be obtained, because of the following reasons:

1- Lake of information about the proportional solenoid components (i.e. number of turns of winding – Permeability of core material).

2- No accurate mathematical model simulates the hysteresis in its operation.

3- No mathematical model simulates the friction value during armature motion.

The variation of Solenoid force with current (I) and air gap distance (x) have been measured in order to overcome the difficulty of obtaining a mathematical model for the proportional solenoid. This method provided us with the physical values of the solenoid force considering all nonlinear electromagnetic and mechanical effects occur during the solenoid operation. In order to perform the solenoid force measurement, a special hydraulic bench has been established as shown in **Fig2.** It composes of the following: A proportional solenoid (1); A special fixture to fix the solenoid, containing a screw over the end of the armature to control the air gap distance inside the solenoid (2); Weights from 1 gram to100 gram (3); Filler sheets of variable thickness (4); Current regulator (5).

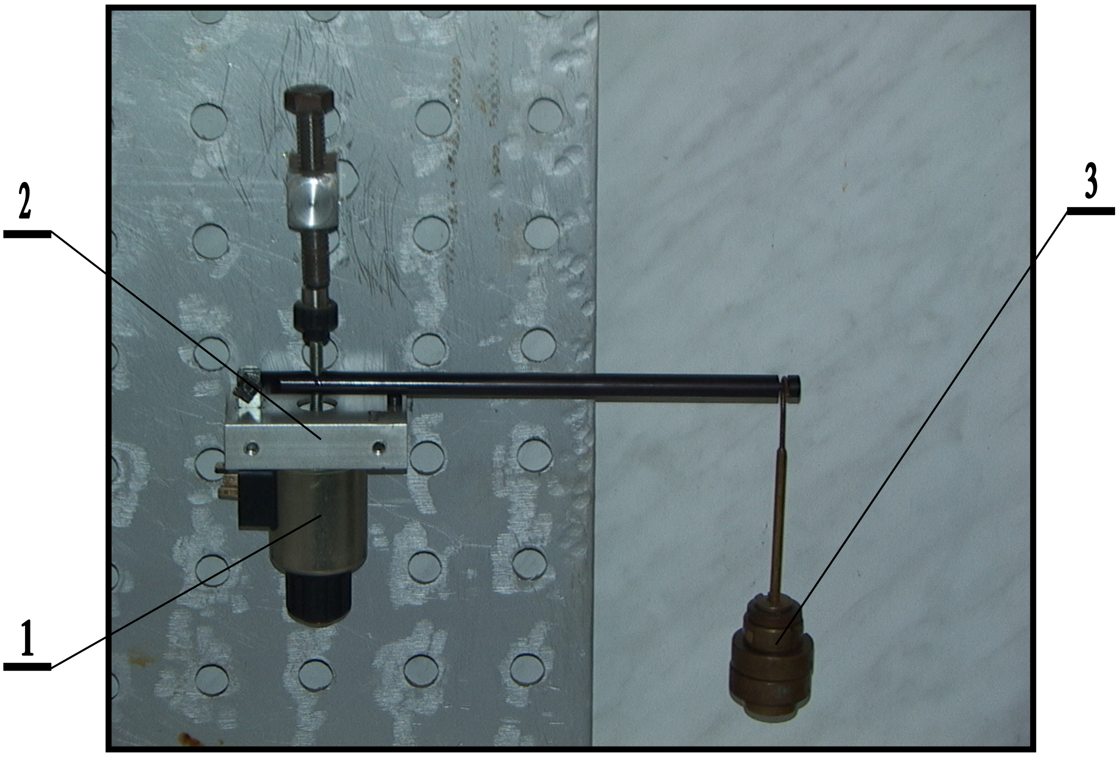

Fig.2. Solenoid force measurments.
Proportional solenoid (1), Special fixture (2), Weight (3).

2.1.1. Measurement Procedures

The following steps are carried out:

1. Switching on the power supply of the main test bench.
2. Adjusting the air gap (x) to 1mm by setting the air gap at zero and then using the 1mm filler sheet to adjust the distance between the screw and the surface of the armature at the value of 1 mm.
3. Initiating the proportional solenoid by the current regulator with a specific current, then the solenoid armature moves the distance of 1 mm with the lever.
4. Placing weights gradually at the end of the lever until the lever return to its initial position, so that the air gap returns to zero again.
5. Computing the solenoid force by taking moment of the forces about the hinge.
6. Repeating step (3) and (4) at different values of current.
7. Repeating step (2), (3), (4) and (5) at different values of air gap distance.

As mention before there are two ways to adjust the current regulator:
To demonstrate the hysteresis occurrence in the proportional solenoid, the pump flow rate versus solenoid current is measured by increasing the current value from zero in steps of 0.05 A till the maximum value (0.8A), then the flow rate is measured by decreasing the current value from 0.8A in steps of 0.05 A till the minimum value of current (0 A).

2.1.2. Measurement Results

The solenoid forces have been measured at 5 air gaps; (1mm; 2mm; 3mm; 4mm and 5mm) and the solenoid current has been varied starting from zero to 0.8 A with step of 0.05 A at each value of air gab. The measured results are shown in Table 1. below and are plotted at figures 3.7. These results have been inserted in the form of a look-up table shown in the theoretical simulation program of the spool motion.

Table 1. Solenoid forces versus input current at different air gaps

Current [A] / x [mm]	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.55	0.6	0.65	0.7	0.75	0.8
Air gap = 1	0.0	0.0	0.0	0.31	1.44	2.56	4.03	5.39	6.3	7.77	8.82	9.73	11.06	12.84	14.15
Air gap = 2	0.0	0.0	0.0	1.65	3.06	4.23	5.41	7.12	8.74	10.31	11.95	13.81	15.54	17.58	19.85
Air gap = 3	0.0	0.0	3.32	6.12	9.05	11.14	13.45	16.87	19.51	21.9	24.54	27.1	29.66	31.52	32.8
Air gap = 4	0.0	0.0	5.39	11.64	17.58	21.53	25.17	28.75	32.25	35.86	39.27	43.06	46.33	49.05	51.82
Air gap = 5	0.0	0.0	11.27	15.98	20.43	24.09	28.62	32.52	36.91	40.36	44.63	48.68	52.61	55.75	59.93

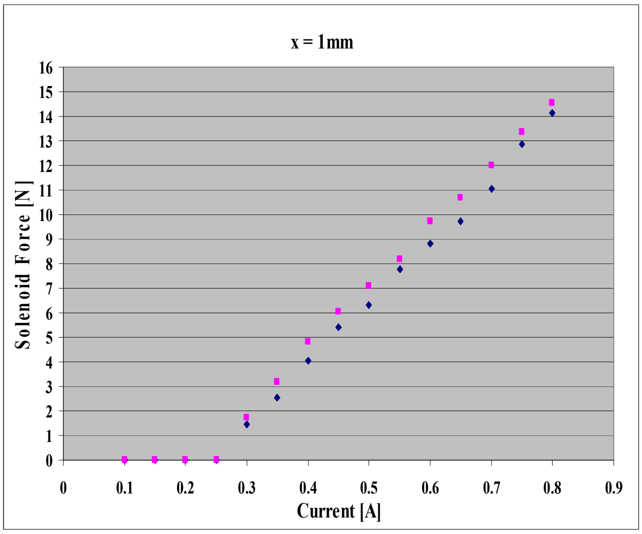

Fig. 3. Force current relation at solenoid air gap of 1mm

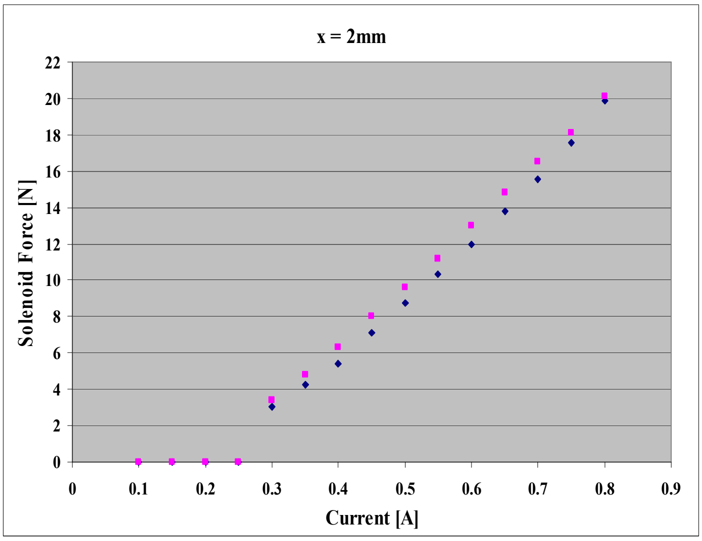

Fig. 4. Force current relation at solenoid air gap of 2mm

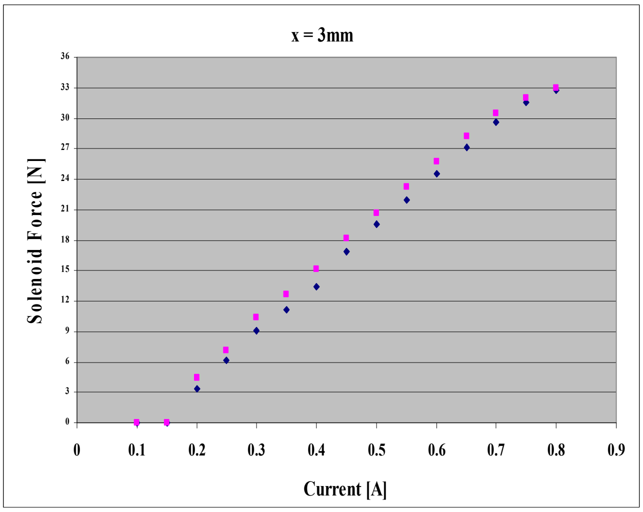

Fig. 5. Force current relation at solenoid air gap of 3mm

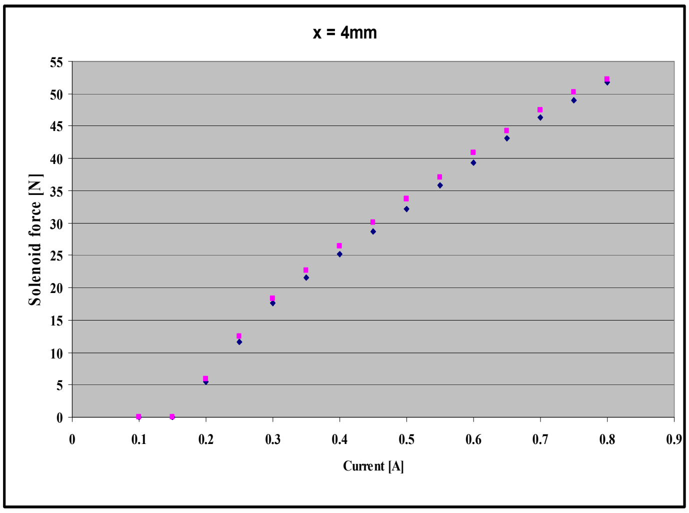

Fig. 6 Force current relation at solenoid air gap of 4mm

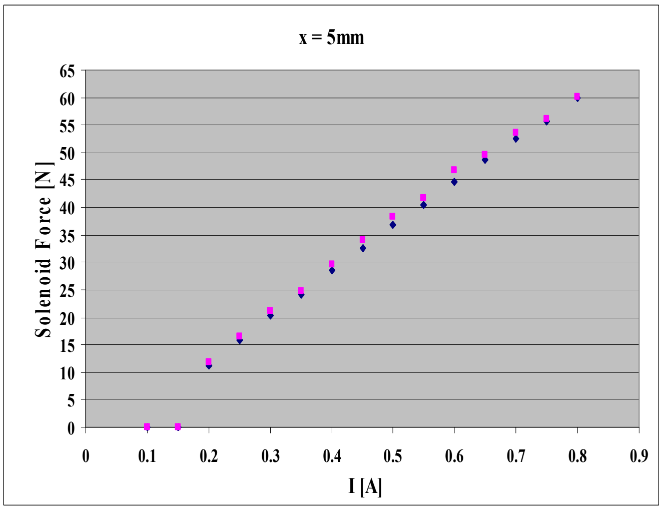

Fig. 7. Force current relation at solenoid air gap of 5mm mm.

2.2. Experiment #2: Measurement of flow rate through directional proportional valve versus solenoid current under fixed valve pressure drop

The aim of this experiment is to measure the variation of the discharged flow rate of a 4/3 directional proportional valve versus the electrical current of the solenoid which controls the value spool displacement. The flow rate through valve restriction area could be estimated by the well known relation: $Q = C_d A(x)\sqrt{\frac{2\Delta p}{\rho}}$

Where, C_d is the discharge coefficient of area, $A(x)$; of the valve corresponding to spool position; Δp, is the pressure drop across the proportional valve; is the hydraulic oil density at operating temperature.

The proportional valve opening area, $A(x)$, is function in the valve spool displacement (x) which is function in the electrical current operating the solenoid. Therefore, the flow rate varies with both the solenoid electrical current and the pressure drop across the proportional valve. Hence, in order to determine the effect of solenoid current on the flow rate quantity, the pressure drop across the valve should be kept constant during measurement, so that the flow rate becomes only dependent on the solenoid current.

2.2.1. Hydraulic circuit setup

The hydraulic bench has been set as shown in **Fig 8.** It contains the following elements: The main hydraulic test bench which includes: Solenoid current regulator and readout (1); Left solenoid (2); 4/3Direction proportional valve (3); Right solenoid (4); Inlet pressure transducer (5); Flowmeter (6); Flowrate readout (7); Pressures readout (8); Throttle valve (9); Outlet pressure transducer (10); Relief valve (11), they have been used to measure the pressure difference across the valve, while the throttle valve has been used to keep this pressure difference constant. **Fig. 9** demonstrates the circuit diagram of the hydraulic system which has been used to measure the flow rate of the proportional valve versus the solenoid electrical current at fixed pressure drop across the valve.

2.2.2. Measurement procedures

After connecting all the described elements to the hydraulic bench, the following steps have been performed:

1. Switching on the pump of the hydraulic bench.
2. Initiating one of the two solenoids of the direction proportion valve by the current regulator.
3. Assigning a pressure difference, Δp, by adjusting the throttle valve until the difference of readings of the two pressure transducer reaches the assigned value then measuring the pump flow rate, via the flow meter display units.

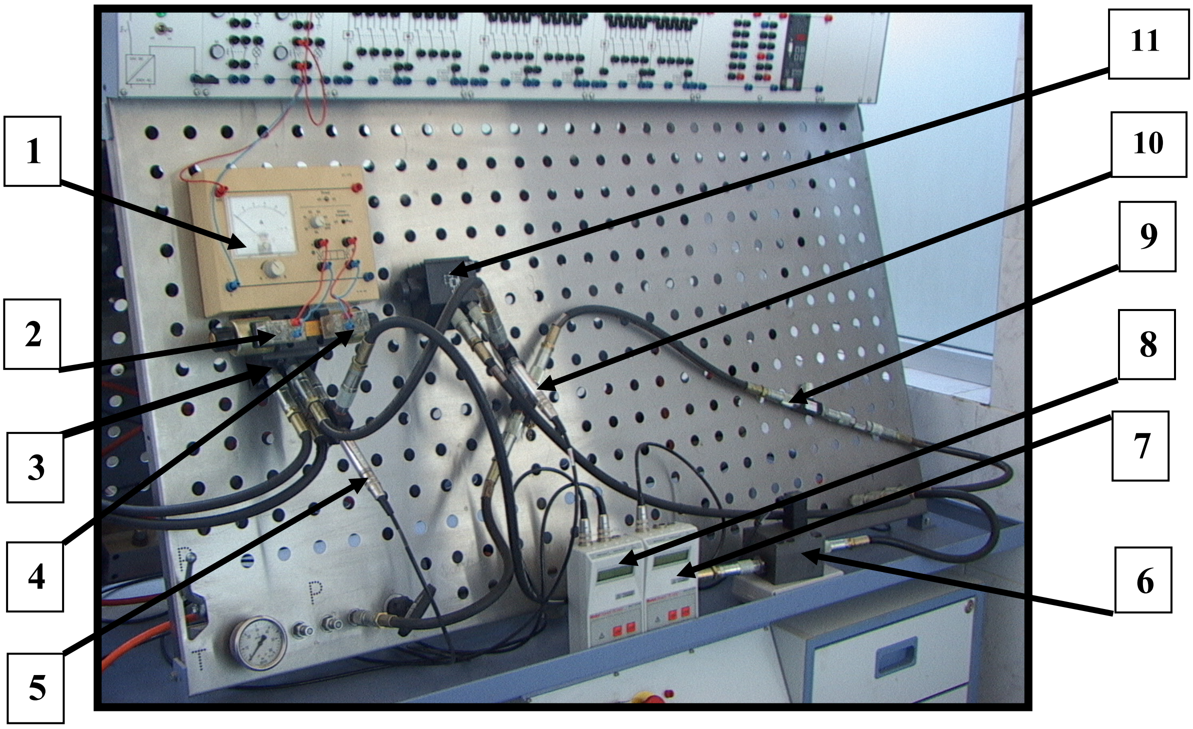

Fig.8. Measurment of valve flow rate versus solenoid current under fixed valve pressure drop. Solenoid current regulator and readout (1); Left solenoid (2); 4/3 Direction proportional valve (3); Right solenoid (4); Inlet pressure transducer (5); Flowmeter (6); Flowrate readout (7); Pressures readout (8); Throttle valve (9); Outlet pressure transducer (10); Relief valve (11).

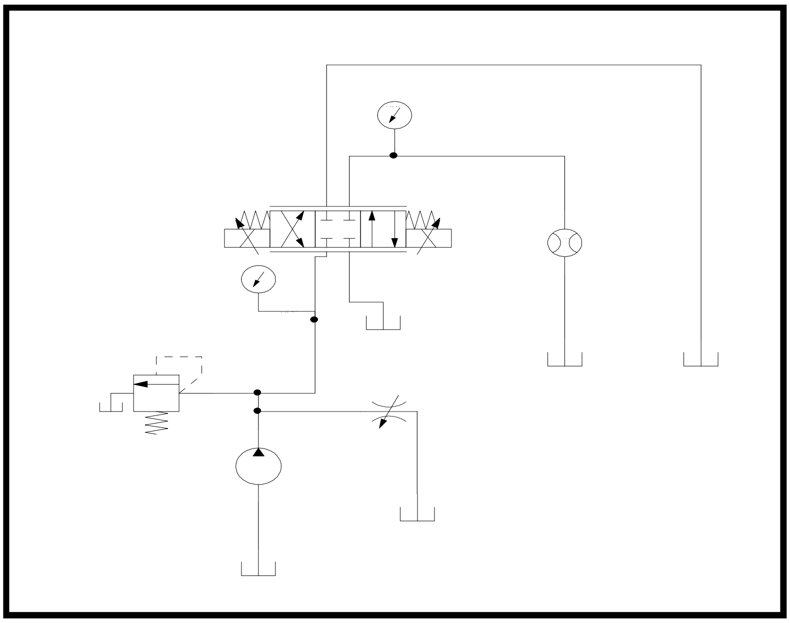

Fig. 9. Hydraulic Circuit of Measurment of valve flow rate versus solenoid current under fixed valve pressure drop.

4. Repeating steps (2) and (3) at different values of solenoid current while keeping the pressure difference constant at the adjusted value.
5. Repeating previous steps at another value of pressure difference across the proportional valve.

To demonstrate the hysteresis occurrence in the proportional solenoid, the pump flow rate versus solenoid current has been measured by increasing the current value starting from **zero** in steps of **0.05 A** till the maximum value has been reached **(0.8A)**, then the flow rate has been measured by decreasing the current value starting from **0.8 A** in steps of **0.05 A** till the minimum value has been reached (**0 A**).

2.2.3. Measurement Results

The valve flow rate versus solenoid current measurements have been performed at two values of pressure difference **8 bar** and **10 bar**. The variation of current has been from **zero** to **0.8 A** with steps of **0.05 A** at each value of pressure difference. The hysteresis in the solenoid behavior is obvious as shown in **Fig 10** at Δp **= 8 bar** and in **Fig 11** at Δp **= 10 bar**; as getting two values of flow rate for the same value of solenoid current.

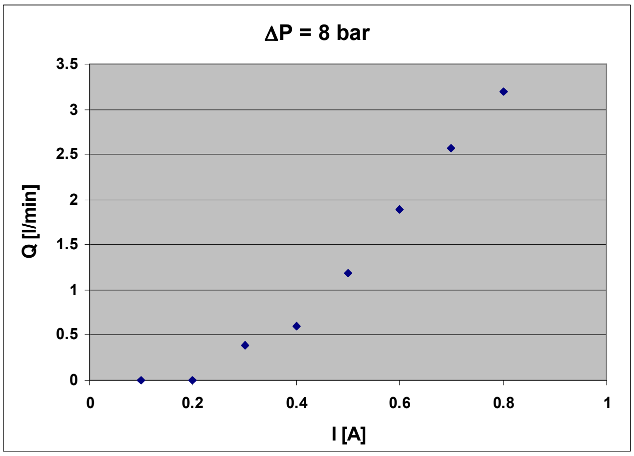

Fig.10. Measured valve flow rate versus solenoid current at p = 8 bar.

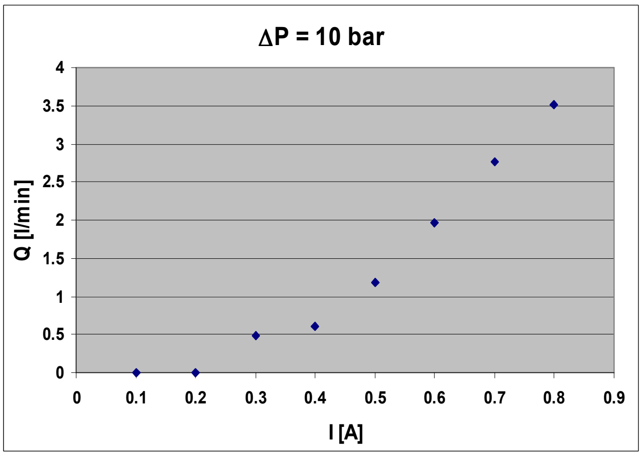

Fig.11. Measured valve flow rate versus solenoid current at p = 10 bar.

3. THEORETICAL INVESTIGATION

In order to obtain a simulation program for the electro-hydraulic directional proportional valve, the valve has been divided into two main parts: the proportional solenoid; and the valve block.

3.1. Proportional solenoid

To simulate the performance of the proportional solenoid valve, it is essential to investigate the effect of magnetic material properties, mass of the moving part, number of coil turns, and geometry in its design. Since all these data were unknown, then alternately the solenoid force which operates the valve spool according to the solenoid electric current has been measured experimentally as mentioned above. Eventually, using the experimental measured values of solenoid force in the theoretical model of the proportional valve restricts the obtained model to be applied only on that type of proportional solenoid. In spite of this drawback, there is a great advantage of using the experimental measured values of solenoid force in the theoretical model, by considering all kinds of losses and non linearity effects exist during solenoid operation, i.e including: eddy current; hysteresis; friction; etc. in the obtained theoretical model.

3.2. Valve block

The simulation model of valve block is carried out by considering the interaction between the equation of motion of the valve spool and the fluid flow across the orifices of the valve.

- **Equation of motion of the valve spool:** $F_s = m\frac{d^2x}{dt^2} + f\frac{dx}{dt} + kx$

Where: $\mathbf{F_s}$ is the proportional solenoid force [**N**]; **m** is the mass of moving parts [**kg**] **m=spool mass+0.5(spring mass)= [19.58 + (0.5 * 1.972)] * 10^{-3} = 0.020566 kg; f** is the friction coefficient [**50: 100 N.sec/m**]; **k** is the return spring constant [**24500 N/m**]; and **x** is the spool displacement [**m**].

The values of $\mathbf{F_s}$ which were shown in (**Table 1.)** have been inserted in the form of a **look-up-table** in the theoretical simulation model of the spool motion using SIMULINK program as shown in **Fig. 12**.

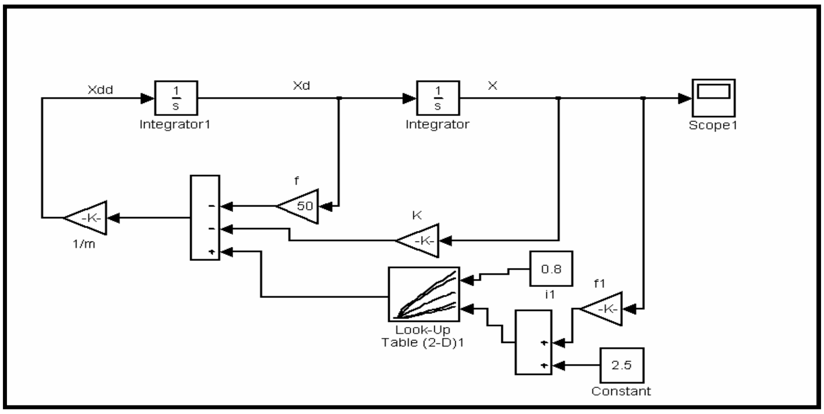

Fig.12. Modeling of the spool motion using SIMULINK program

Fluid Flow equation across the valve orifices: $Q = C_d(w \cdot x)\sqrt{\frac{2\Delta p}{\rho}}$

Where: $\mathbf{C_d}$ is the flow discharge coefficient; **w** is the width of the orifice [**m**]; ρ is the fluid density [**875 kg/m^3**]; Δ**p** is the pressure difference across the valve [**N/m^2**]. The modeling of the flow equation through valve orifices using SIMULINK program is shown in **Fig 13.**

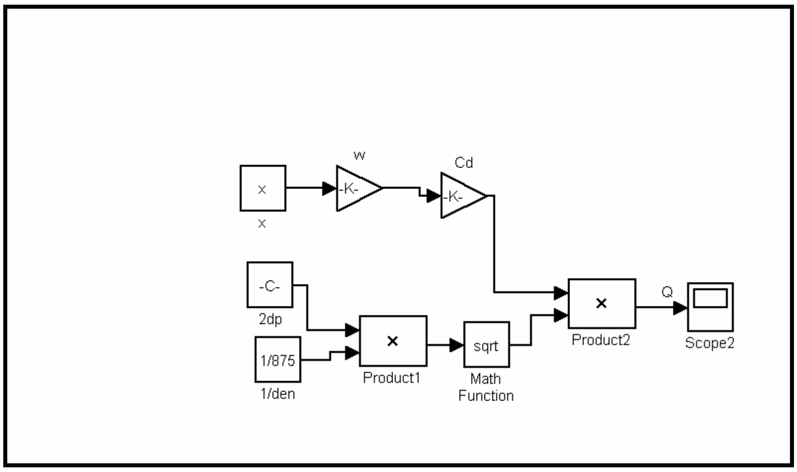

Fig.13. Modeling of the flow equation through valve orifices using SIMULINK program

By appending the obtained two models, in Fig. 11 and in Fig. 12, the complete theoretical simulation model of the directional proportional valve has been obtained as shown in **Fig 14.**

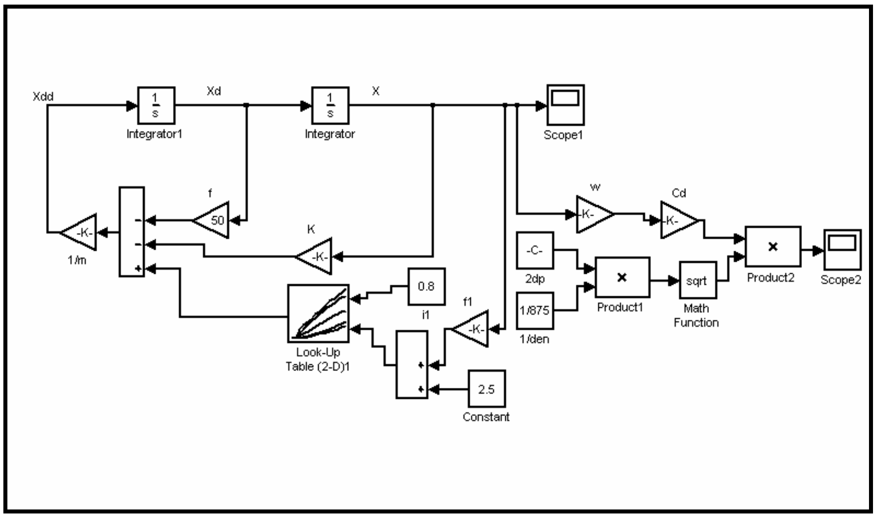

Fig.14. Simulation model of the directional proportional valve

4. VERIFICATION OF THE SIMULATION PROGRAM

In order to verify the obtained theoretical simulation model, the program has been run by inputting various values of solenoid current into the program and calculating the corresponding flow rate values of the valve while the pressure drop across the valve has been kept constant (**p = 8 bar** and **p = 10 bar**).

The relation has been curve fitted and the corresponding polynomial has been plotted as shown in **Fig 15** for the case of **p = 8 bar** and in **Fig 16** for the case of **p = 10 bar.** The experimental results represented in **Fig 10** and **Fig 11** have been inserted in **Fig 15** and **Fig 16** respectively. It is obvious from these plots that the theoretical simulation model of the proportional valve could be accepted since the correlation coefficient for the polynomial is equal to 0.98 and the standard deviation, has been calculated and found equal to 0.213 which is an allowable value.

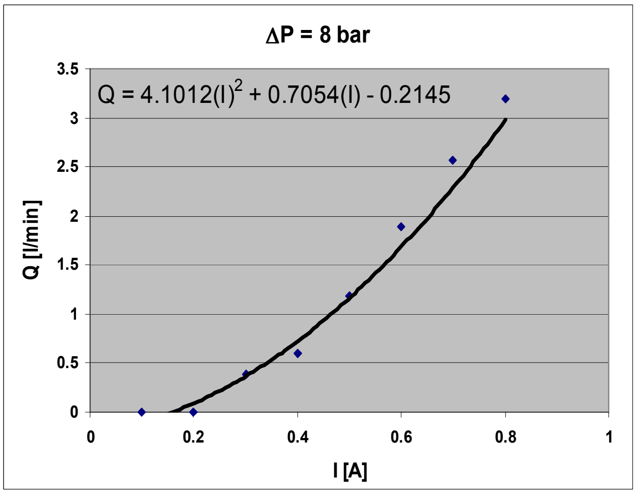

Fig. 15. Comparison between experimental and simulation results when valve pressure drop p = 8 bar

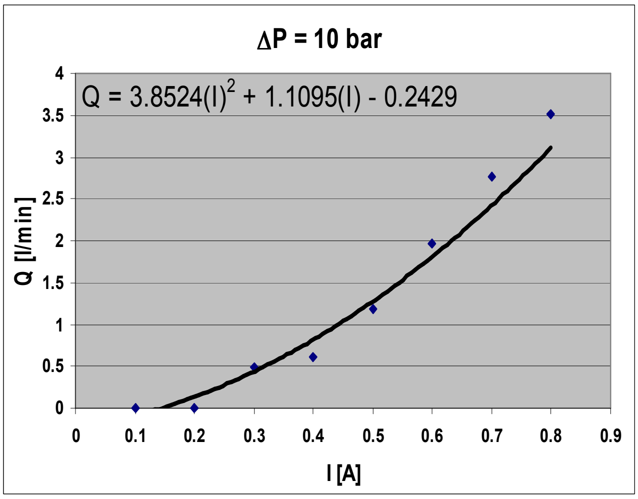

Fig. 16. Comparison between experimental and simulation results when valve pressure drop p = 10 bar

5. CONCLUSION

The model of the directional proportional valve is carried out by dividing the valve into two main parts, the proportional solenoid and the valve block. Due to some considerations, the mathematical model of the proportional solenoid could not be obtained. So, the force exerted by the proportional solenoid has been measured experimentally at different values of solenoid current and various air gab distances. The measured values of solenoid force are introduced into the valve model in a form of LOOK-UP table. The mathematical model of the valve spool is obtained and simulated by using SIMULINK program. The validation of the model is carried out by comparing the results of measured values of flow rate with the corresponding values obtained from simulation program under constant pressure difference. The comparison proved the validation of the theoretical model. The obtained theoretical model of the directional proportional valve will be used in future work to simulate an electro-hydraulic control system uses this type of valves.

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