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Simulation of Dynamics of a Compound Pressure Relief Valve

Basavaraj V Hubballi[1] and Vilas B.Sondur[2]

[1]Jain College of Engineering, T S Nagar, Belagavi and India.

[2]Sondur Academy, Hindawadi, Belagavi and India.

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ABSTRACT

Pressure relief valves uncertainty and parameter variations are of major significance in the hydraulic system operations. The use of one such valve, compound pressure relief valve is the subject of this paper. From the deduced differential equations, MATLAB-SIMULINK simulation model developed. The pressure response of a valve with damping spool pilot poppet has been investigated and compared with the experimental results. Also simulated the effects of some critical parameters on the valve dynamic performance.

Introduction

Hydraulics system deals with the generation, control and transmission of power by the use of pressurized liquids. Hydraulic circuits system complexity depends upon a variety of factors ranging from end-product stability, responsiveness, cost sensitivity and energy requirements. From simple hydraulic components to computer controlled electro hydraulic systems, stabilized output is essential. . It is a multi-disciplinary field; create challenges in efficiency, components design, integration and compactness, environmental impact, user-friendly and energy-efficient applications [1, 2, 3, 4, 5].

Precision pressure regulation is the key operation in this system. All most all circuits contain pressure relief valve for limiting maximum system pressure. This valve is a necessary safety element that protects the hydraulic system and its component from excessive pressure. When the relief valves are incorrectly designed, selected and operated, instability of the system may occur.

The basic function of a pressure relief valve is to sense line pressure with a spring loaded surface area, and open a flow path sufficiently to bleed oil to prevent higher pressure than set point.

Several designs accomplish this task nicely. The main difference between these designs is that they are either direct acting or pilot operated, and that the surface area used to sense pressure or to react it is used indifferent configurations.

Poppet type valve has less leakage than the spool valve and is less to clog with dirt. The relatively large tolerance makes it economic to manufacture and also better transient response than spool type .The direct acting relief valve operates with a spring to pre-load the poppet of the valve. The use of such valve is gradually decreasing due to its poor pressure override characteristics. [6-12].

As a consequence of this there is a lot of waste of energy which is converted into heat which increases working temperature of the hydraulic oil and decrease the efficiency of the hydraulic system In order to improve such characteristics, a pilot stage valve is introduced; it is also called a two stage pressure relief valve or compound pressure relief valve. The main advantage of this valve is its improved controllability and stability compared with the single stage pressure relief valve. Hence these relief valves are preferred to maintain a constant pressure in sophisticated hydraulic control systems [13].

CFD numerical analysis software are now generally used to model complex flow patterns in the valves and many other aspects of engineering, and now to a high degree of accuracy for both laminar and turbulent flow conditions. It is an effective way to investigate the static characteristics of the valve. [14, 15, 16].

Determination of the discharge coefficient of the valve orifices can be done with a CFD software, although there are still some uncertain issues like exact geometry mesh, real upstream and downstream conditions, and a great deal of effort is also placed on experimental determination[17,18].

Valve static characteristics can be improved with some modification, but it can result negative effect on dynamic characteristics. Therefore it is necessary to examine static and dynamic characteristics simultaneously to get optimal performance of the valve [19, 20].

A number of different approaches have been taken by various researchers for investigating the dynamic behaviour of various designs of two stage pressure relief valve with the conical poppet type pilot valve. Many authors have used different techniques like MATALB-SIMULINK, Root locus and Bode plot, AMESim,, BondGraph, CFD to analyze response of specific valve design[21-29]. Also some of they have validate the model results with the experimental techniques.

The different design of relief valve can lead to different static and dynamic characteristics. From the dynamic point of view, conical poppet with spring is considered as a spring-

Tele: +91 9448342032
E-mail address: bvhubliabhi@gmail.com

mass system, which causes oscillations when it moves. These oscillations affect the pressure and can be eliminated by damping. This damping spool stabilizes the poppet operation during opening and closing.

Different ways in which damping can be achieved are,

a) Damping spool and orifice to the spool chamber
b) Damping spool with flat one or two surfaces
c) Damping spool with large tolerance play.

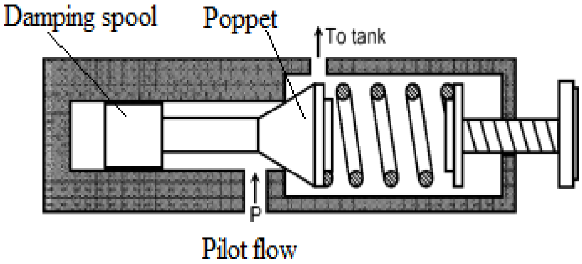

Fig 1. Pilot valve poppet damping spool.

In the pressure relief valve design, there are very small clearances between moving elements, necessary to minimize leakages while maintaining sufficient lubrication, and control areas necessary to either create pressure drops and/or direct a variable flow rate to different output ports. Hence, there is a need to determine static pressure–flow characteristic curves of such valves as accurately as possible.

The present work intends to analyze the dynamic characteristics of a compound pressure relief valve of Polyhydron Ltd Belagavi, Karnataka India, renowned manufacturers of hydraulic components. Since, there has not been any significant investigation done on this type of valve assembly with the pilot valve poppet damping spool with two surfaces type Fig.1.

Simulation model developed in the MATLAB/SIMULINK software from the deduced mathematical equations. It considered various effects such as coupled main and pilot valve dynamics, flow reaction forces, fluid compressibility and variable orifice flow. The results comprise of the valve transient response with respect to the step input flow rate. The effects of pilot valve spring pre-compression, poppet damping spool clearance, as well as the effect of change of input fluid chamber volume of main valve have been studied Also investigated the dynamic characteristics through experiment measurement and results shows good agreement with the simulation model. This model confirms the suitable to assess the effects of other critical parameters on the valve performance.

2. Description of the valve

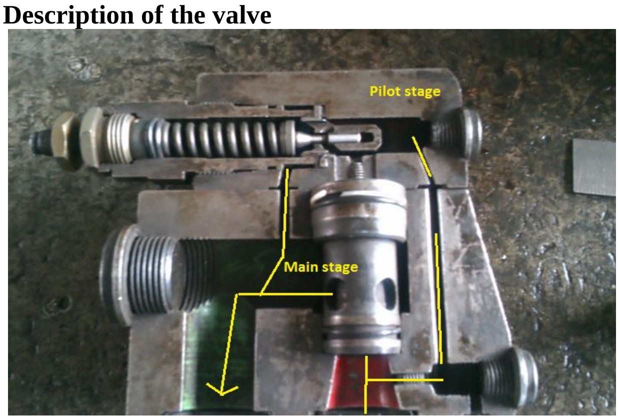

Fig 2. Sectional view of the valve.

Fig.2. represents the sectional view of the compound pressure relief valve. This valve consists of a main stage and pilot stage loaded by the different stiffness springs. The main valve is designed with a relatively large diameter of the main poppet and a lesser spring stiffness of the main valve, which decreases the pressure override. The pre-compression force can be overcome by a pressure difference of about 3.5–10 bar between the input pressure and the spring chamber pressure. The pilot valve consists of conical poppet with damping spool loaded by high stiffness spring. In order to avoid pressure peaks, damping incorporate by integrated damping spool with pilot poppet.

Fig.3. represents the schematic diagram of the valve. This valve system consists of three subsystems: main valve, pilot valve and fixed orifices. In normal position both pilot poppet and main cylindrical poppet are closed under the influence of the springs and there is a balance of forces at the main valve cylindrical poppet. Sometimes a small differential area is to keep the main-valve poppet closed. When the pressure in front of the pilot poppet reach higher value than the preset pilot spring force, the pilot poppet gets opened and oil begins to flow through the variable orifice. Because of restrictors, there is a rapid drop in pressure Psc, the main valve poppet is displaced, The pressure in the upper part of the main poppet is maintaining approximately constant by the pilot poppet. With further increase of the inlet pressure, the pressure drop continues to increase. Once the fluid force acting at the bottom of the main valve overcomes the fluid plus spring force, at which the main valve is opening and the major flow, occurs.

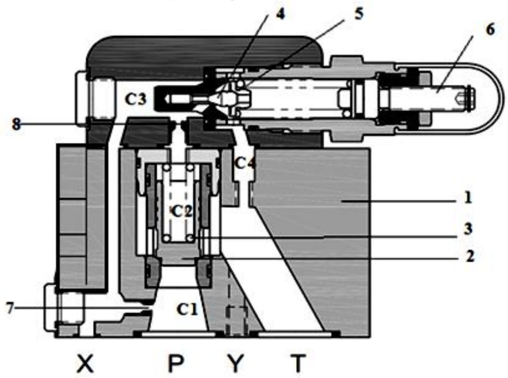

Fig 3. Schematic diagram of valve

(1-Valve housing, 2-Main cylindrical poppet, 3-Main valve spring, 4-Pilot poppet, 5-Pilot spring, 6-Screw, 7-Orifice, 8-damping orifice, P-Pressure line port, X,Y-Pilot port)

Nomenclature

m_m =Mass of the main valve cylindrical poppet
P_1 = Pressure in front of cylindrical poppet
d_m=Diameter of cylindrical poppet
A_m = Area of cylindrical poppet
A_{mo} = Main valve orifice area
A_{SC} = Area of spring chamber
b_m= Damping coefficient
k_m= Spring stiffness
y_0 = Spring pre-compressed length
C_{fm} = Main valve Flow factor
y= Cylindrical poppet displacement
p_{SC} = Spring Chamber pressure
m_p = Mass of the pilot valve conical poppet
p_3 = Pressure infront of conical poppet
d_p = Diameter of conical poppet
A_p = Area of conical poppet
A_{po}= Pilot valve orifice area
b_m = Damping coefficient
k_p = Spring stiffness
xo = Spring pre-compressed length
C_{fp} = Pilot valve Flow factor

x = Conical poppet displacement
V_1 = Volume of fluid chamber at 1
V_2 = Volume of fluid chamber at 2
V_3 = Volume of fluid chamber at 3
V_{SC} = Volume of spring fluid chamber
V_{DC} = Volume of damping spool fluid chamber
Q_1 = Input flow to Main valve
Q_2 = Flow into pilot valve chamber
Q_3= Flow into pilot valve
Q_4 = Pilot Valve relief flow to tank
Q_5 =Main valve relief flow to tank
Q_{SC} = Flow from spring chamber
Q_{DC} = Flow from damping spool chamber

2. Mathematical Model

The basic reasons for developing a mathematical model for a fluid power system is to understand regard the system function and also it helps to allows evaluation of system operation. The modeling of a valve dynamics mainly follows the approaches discussed in the investigations [19-24].

For dynamic analysis, valves are often modeled assuming the valve is comprised of fluid chambers separated by orifices. There are mainly three types of equations used: equations of motion, continuity equations, and orifice equations. These equations will provide a complete model for most fluid power systems.

The amount of fluid move from the nose side to the back side of a poppet depending on the resultant force of the mass inertia, the damping force, the spring force, the static pressure forces and the flow force. The size of the poppet opening and the pressure drop across the poppet head determinates the amount of the load that the system can be taken. While solving the system equations numerically, various pressure-flow characteristics across the valve ports and the orifices are taken into consideration.

Equation of motion

The equations of motion define the movement of the mass elements; also take into account the physical damping, spring constant and flow reaction of the system. The dynamic equations results from noting that masses subjected to unbalanced forces must accelerate according to Newton's Second Law, F = ma. The forces on the moving parts of fluid power circuit components may result from pressure differences, viscous friction, coulomb friction, gravity, and flow forces.

Continuity equation

The continuity equation governs the conservation of mass in a fluid volume. The total flow into the fluid chamber must equal the total flow out from chamber and plus the accumulation within the chamber as stated in Equation.

$$Q_{in} = Q_{out} + Q_{accum} \tag{1.1}$$

where *Qin*, *Qout*, *and Qaccum* represent the flow in and out of the fluid chambers and the accumulation within the chambers respectively

Orifice equation

Most hydraulic systems are designed for turbulent flow. Valves are fundamentally variable orifices where the area of the orifice depends on the poppet position. The orifice equation, as stated in equation, defines the relationship between the pressure drop across the orifice (ΔP) to the flow through the orifice (Q),

$$Q(x,p) = C_d A_o(x)\sqrt{\Delta P \frac{2}{\rho}} \tag{1.2}$$

Where A_o is the cross sectional area of the orifice, ρ is the density of the fluid, C_d is the discharge coefficient, x-displacement, Δp-pressure drop.

Flow through fixed orifices

Considering the flow through capillary tube stated as

$$Q = \frac{\pi d^4}{128\upsilon\rho l}\Delta P \tag{1.3}$$

d - diameter of orifice, μ = $\vartheta\rho$ = absolute viscosity of oil, l-length of orifice, Δp-pressure drop.

Flow through the annular area

To determine the flow through the annular area, considering the equation of flow between two parallel plates:

$$Q = \frac{\pi d h^3}{12\upsilon\rho l}\Delta P \tag{1.4}$$

d- damping spool diameter, h-clearance, μ = $\vartheta\rho$ = absolute viscosity of oil, l -length of damping spool, ΔP - Pressure drop

Seat reaction forces

Before opening of the outlet port and after complete opening of the port, the reaction force Fr_s acts on the poppet. In the event of the opening of the valve port (i.e., 0< X<Xmax), Frs=0;

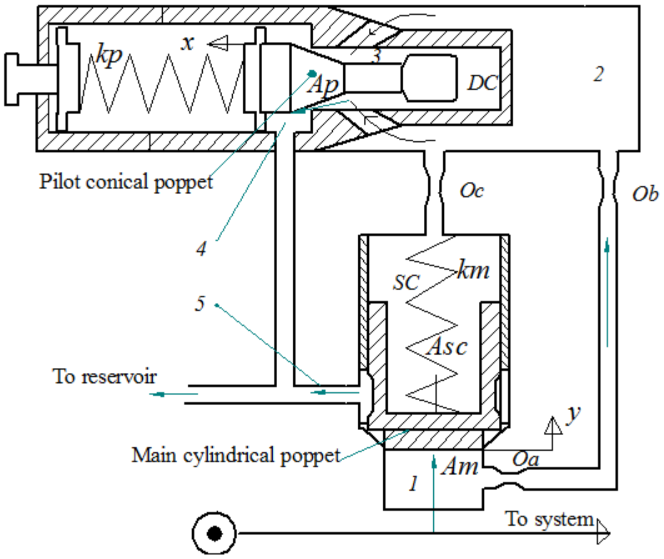

Fig 4. Schematic diagram of the valve.

For the analysis, following assumptions were made:

a) A constant source of supply to the valve
b) Properties of the fluid are constant.
c) Coefficient of discharge is constant for variable orifices.
d) Change of the chamber volume by displacement of the poppet is not considered.
e) Coulomb friction acting neglected.
f) All springs are assumed to be linear,
g) Outlet pressure is assumed to be zero.

2.1 Equation of motion for Pilot Valve Poppet:

Considering that the total mass of the moving parts m_p is equal to the mass of the poppet plus one-third mass of the spring and pad, the differential equation of the dynamic behaviour is derived as follows:

Hydraulic force = Mass acceleration force + Damping force + spring force + Flow reaction force + Seat reaction force

$$F_{hyd} = F_a + F_b + F_{fr} + F_{sr} \tag{1.5}$$

$$p_3 A_p = m_p \ddot{x} + b_p \dot{x} + k_p\left(x_o + x\right) + C_f p_3 x + F_{sr} \tag{1.6}$$

$$C_{fp} = C_d{}^2 \pi d_p \sin 2\theta$$

2.2 Equation of motion for Main Valve Poppet:

Hydraulic force = Mass acceleration force + Damping force + spring force + Flow reaction force + Spring chamber force

$$F_{hyd} = F_a + F_b + F_s + F_{sc} + F_{fr} + F_{sr} \tag{1.7}$$

$$p_1 A_m = m_m \ddot{y} + b_m \dot{y} + k_m\left(y_o + y\right) + C_{fm} p_1 y + p_{sc} A_{sc} \tag{1.8}$$

$$C_{fm} = C_d{}^2 \pi d_m \sin 2\theta$$

Due to compressibility of the fluid and elasticity of the hoses, tubes, the flow rate at the test valve can be different from that one at the exit to the pump. To model the compressibility effects, a hypothetical chamber is added whose volume is equal to the total volume of oil in front of the test valve system. For each of the fluid chambers, equations of continuity considering compressibility of fluid are formulated. as below.

The left side of the above equation is the volume change of the control chamber due to compressible flow in terms of time. The flow rates on its right side are caused by flows either entering or exiting the control chamber due to its pressure drop effect. The last item on the right side is the volume change of the control chamber due to boundary change. Where V is the control chamber volume, which is assumed as constant by ignoring the volume change caused by the movement of the poppet.

Pressure transient at fluid chamber 1

$$\frac{dp_1}{dt} = \frac{\beta}{V_1}\left(Q_1 - Q_2 - Q_5 - A_m \dot{y}\right) \tag{1.9}$$

Pressure transient at fluid chamber 2

$$\frac{dp_2}{dt} = \frac{\beta}{V_2}\left(Q_2 + Q_{sc} - Q_4\right) \tag{1.10}$$

Pressure transient at spring fluid chamber SC

$$\frac{dp_{sc}}{dt} = \frac{\beta}{V_{sc}}\left(A_m \dot{y} - Q_{sc}\right) \tag{1.11}$$

Pressure transient at fluid chamber 3

$$\frac{dp_s}{dt} = \frac{\beta}{V_3}\left(Q_3 - Q_{dc} - Q_4 + A_s \dot{x} - A_p \dot{x}\right) \tag{1.12}$$

Pressure transient at damping fluid chamber V_{DC}

$$\frac{dp_{dc}}{dt} = \frac{\beta}{V_{dc}}\left(Q_{dc} - A_s \dot{x}\right) \tag{1.13}$$

Equation of flow through variable orifice

Across Main valve cylindrical poppet

$$Q_5 = C_d A_{mo} \sqrt{\frac{2}{\rho}\left(p_1 - p_5\right)} \tag{1.14}$$

$$A_{mo} = \pi y \sin\phi \left\{d_m - y \sin\phi \cos\phi\right\}$$

Across Pilot valve conical poppet

$$Q_3 = C_d A_{po} \sqrt{\frac{2}{\rho}\left(p_3 - p_4\right)} \tag{1.15}$$

$$A_{po} = \pi x \sin\theta \left\{d_p - x \sin\theta \cos\theta\right\}$$

Flow through fixed orifices (O_a and O_b)

$$Q_2 = \frac{\pi d_o{}^4}{128 \upsilon \rho l}\left(p_1 - p_2\right) \tag{1.16}$$

Flow through damping spool chamber

$$Q_{dc} = \frac{\pi d h^3}{12 \upsilon \rho l}\left(p_3 - p_d\right) \tag{1.17}$$

3. Experimental study

Some of the parameters associated with the system equations derived above are measured and others are estimated suitably. The complete list of parameters measured and assumed in the studies is listed in the following Table 1.Figure 5, depicts the layout of an experimental setup to test the performance of a valve and Fig.5. shows the hydraulic circuit of the setup. The setup is divided into hydraulic part and electrical part. The hydraulic part mainly includes power pack with a test valve and the electrical part includes data acquisition and sensors to acquire the pressure and flow signals.

The measurement instruments were previously calibrated. The data acquisition and the signal processing were preformed with the help of Digital Storage Oscilloscope. Pressure transmitter Model S-10 manufactured by WIKA and Flow sensor Model VS 1 manufactured by VSE is used for pressure and flow rate respectively.

The experiments were carried out by varying pressure. To examine the influence of this parameter, the following series of experiments were carried out. Figure 8 shows a steady state relationship between the main valve and pilot valve pressures. In these figures, six series of experimental and corresponding simulation results are plotted. For verification of the valve performance, there are several issues to take into account, especially the valve adjustment. The measurement of the valve displacements during valve operation is quite complex, because no additional leakage or friction should be incorporated into the system. The transient response of the solenoid DCV is actually negligible compared with that of the valve since the solenoids cores dynamics are treated separately.

Compound Pressure Relief Valve tested on the experimental setup to investigate the transient response of the valve. For transient response of the specified valve a steady state flow of 9 l/min was set. When solenoid directional control valve is energized, the pressure built in the line of pressure relief valve. Once the pressure exceeds the set pressure, test valve gets cracked and releases the flow back to the reservoir. Referring to the Fig.6. Pressure - flows are measured through pressure transducer and flow sensor.

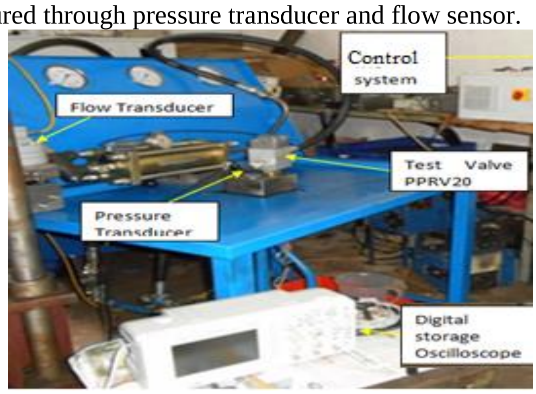

Fig 5. Experimental setup with test valve.

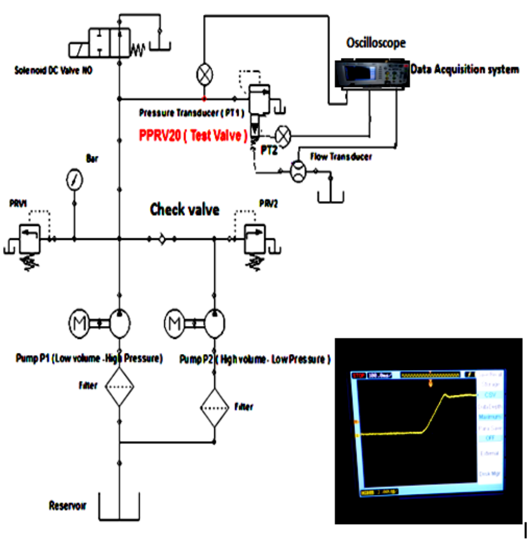

Fig 6. Experimental setup Hydraulic circuit with DSO.

4. Simulation Model

Based on the deduced mathematical equations of the compound pressure relief valves, the simulation model in Simulink block set program of Matlab has been obtained. Input to the simulation model is of step input flow. On a real situation of the relief valve usage ,input flow rate to the valve varies from zero to the rated flow.

The influence of the geometry on the valve dynamic characteristics has been examined by means of the simulation model. Under the same operating conditions the performance of the valve is changed, by altering the different parameters. The simulation results are presented through the pressure transient response.

The deduced equations 1.1 to 1.17 are solved numerically using MATALB/SIMULINK block set program. The solver used here is ODE23s (stiff/Mod. RosenBrock) method. The time step for calculation was variable step. The vibration disappeared after 0.2 s with all of the conditions. The calculation of the response was made until 0.5 s. The pressures and the valve displacement, flow was almost settled down after 0.3 s in most cases.

The effects of the variations of various parameters on the response of the system are obtained through simulation. Settling time was determined as the time until the main valve displacement was settled in the settled value ±2 % of step width, where the step width is the difference between the initial- and settled values of the main valve displacement. As shown in Fig. 7, pressure increment Po, pressure overshoot P1, peak rising time T1, damping ratio rd and period of damped vibration Td were found from waveform of supply pressure.

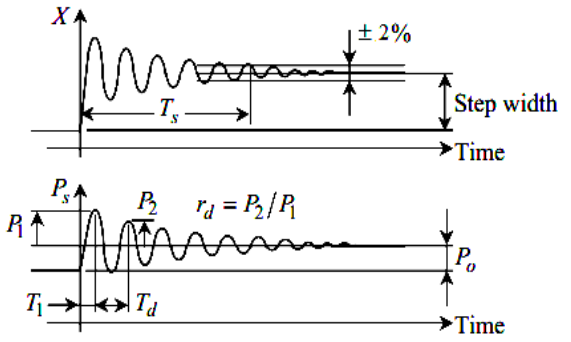

Fig 7. Measures of step response.

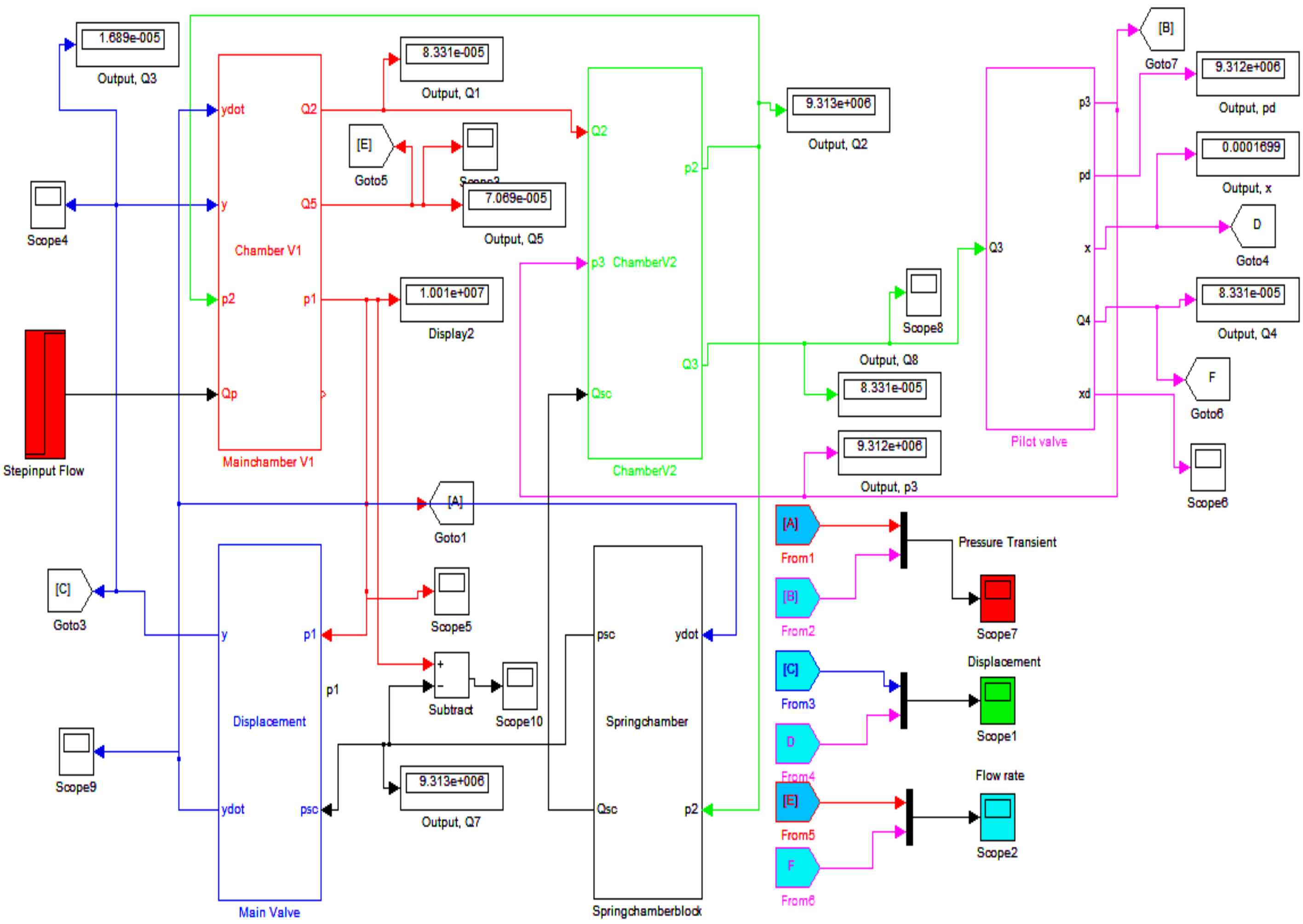

Fig 14. Simulink Simulation Model of Valve.

Table 1. Reference parameters and properties of fluid.

Q_1= 1.54e-3m^3/sec	d_m=14mm	b_p=160Ns/m	ρ=850kg/m^3	K_m=6700N/m	K_p=62100 N/m
β=1.5e9 Pa	b_m=750NS/m	m_{ep}=13.5gm	C_d=0.62	m_m =37.98gm	d_m=5mm

5. Results and discussion

5.1 Transient Response

Through experimental study, the pressure transient response of the valve for different pre-compression of the pilot valve spring observed. The results of the experimental investigation and the solution of the simulation model are presented in Fig.8-13. With the increase in the pre-compression of the pilot spring, the steady state valve pressure increases. On the other hand, the peak time and the settling time of the response also increases.

It can be noticed that the pressure in front of the pilot valve are lower than the pressure of opening of the main valve. It is important to analyze the values of the main valve pressure P1, spool chamber pressure Pd and pilot valve pressure P3. The difference between simulation and experiment becomes greater with pilot valve lower pressure. The main cause of the difference is uncertain estimate of the Coulomb friction, damping coefficient and flow force the simulation model.

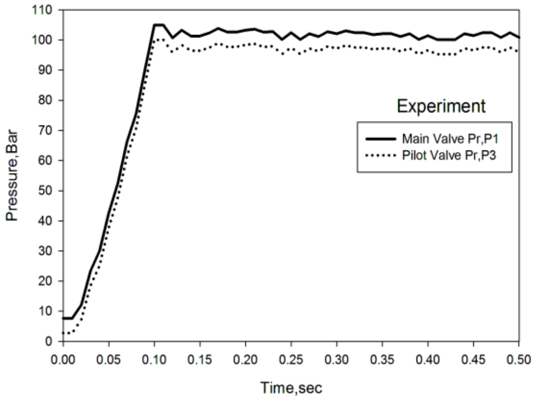

Fig 8. Transient response for 100 bar

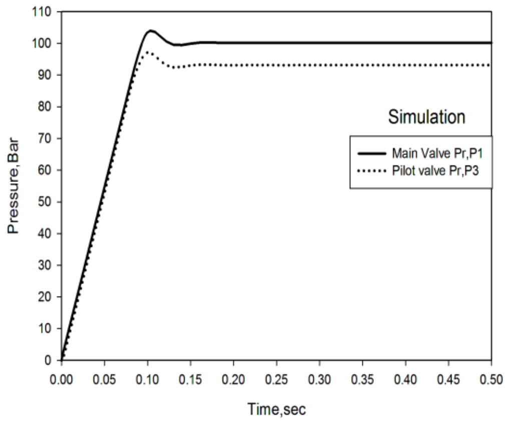

Fig 9. Transient response for 100 bar

5.2 Pressure-Flow Characteristics

Flow and pressure oscillations in the valve cause unwanted vibrations and noises, which affects its proper functioning and other system elements. These unwanted appearances can be minimized by selecting proper valve geometry.

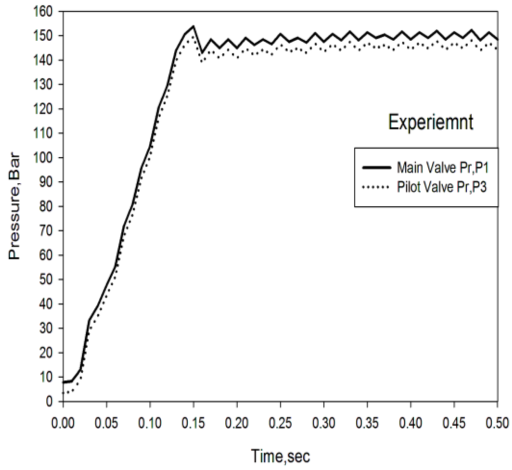

Fig 10. Transient response for 150 bar

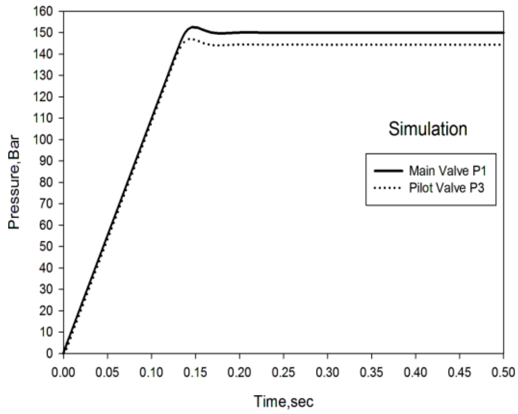

Fig 11. Transient response for 150 bar

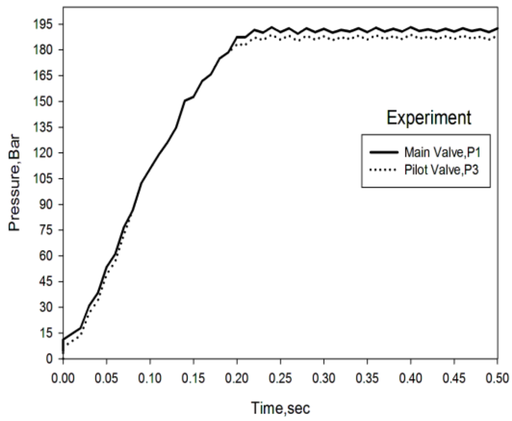

Fig 12. Transient response for 192 bar

Keeping flow constant, for different pressure settings, the performance of the valve is observed. When compared these graphs to the type of valve without damping spool pilot poppet shows less amount of dynamic overload (pressure override) exist in this valve. In the systems where overload cannot be accepted it is preferred to use this kind of compound pressure relief valve. Graphs show that, it has good pressure-flow regulations.

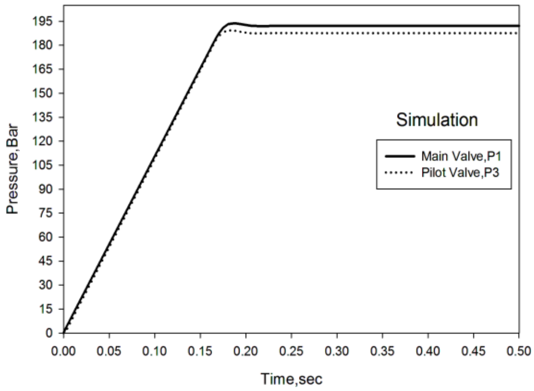

Fig 13. Transient response for 192 bar

To understand and make use of the response information of a valve is very much necessary while observing the response of a whole hydraulic system. The dynamic behaviour of the valve is dependent on its design, the operating state and the hydraulic system itself.

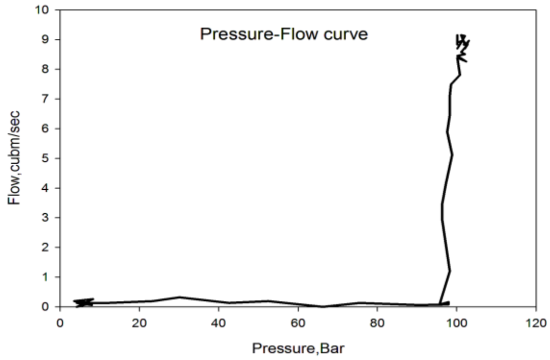

Fig 15. Pressure-Flow static curve for 100bar

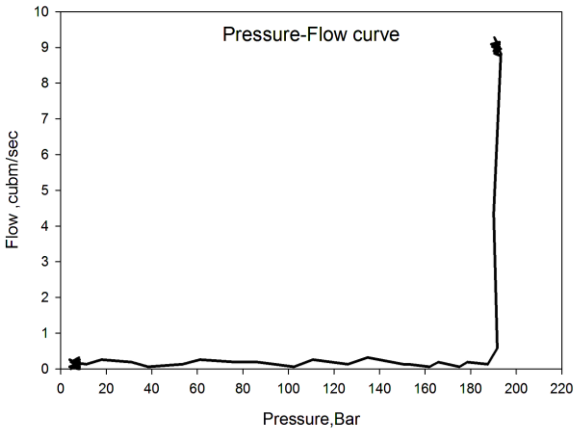

Fig 16. Pressure-Flow static curve for 192bar

The dependence of pressure on flow may be used to view the entire range of applications of a pressure relief valve. Experiment data Fig.15. and Fig.16. indicates that, it has good pressure-flow regulations over wide range. Pressure override ranges from 4 bar to 9 bar exist in this valve.

5.3 Pressure drop

The fig. shows the plot of main valve versus pilot valve transient response curve. It is observed that, there is a reduction in the pressure, as pilot valve cracks and later it stabilizes to the set pressure.

It has been observed that from the graph Fig. 11. that the pilot valve cracks first and then main valve opens to relieve the major flow to tank. The flow started to occurs as pilot valve cracks and then flow is stabilized i.e. full flow when main valve opens.

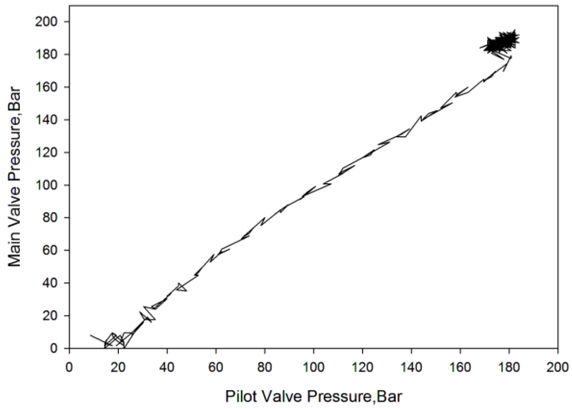

Fig 17. Main valve pressure V/s Pilot valve pressure

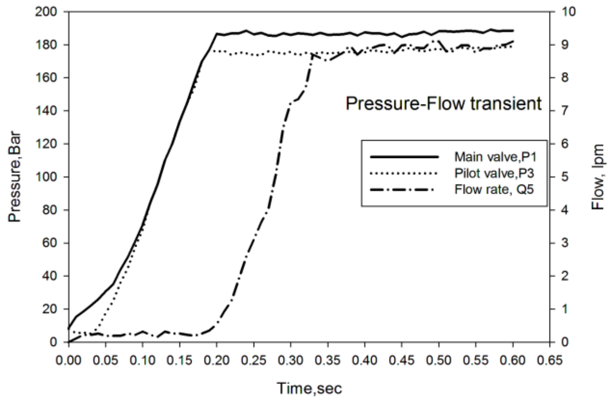

Fig 18. Pressure-Flow Transient response

5.4 Parametric Analysis by Simulation Model

Numerical simulation is a simple and effective method for analyzing valve dynamics. The effects of the variations of various parameters on the response of the valve are obtained through simulation analysis.

5.4.1 Pilot Valve damping spool clearance effect.

Figure shows the influence of the damping spool clearance, which indicates that the clearance has large influence on the stability. Small clearance made the overshoot, peak rising time and period of damped vibration increases slightly. In contrast, if clearance was large, the overshoot, peak rising time became small and the settling time became longer. To summarize, there exists an optimal value for the damping orifice diameter. In addition, its small vibration has large influence on the stability of the valve. This implies that machining error has large influence on stability.

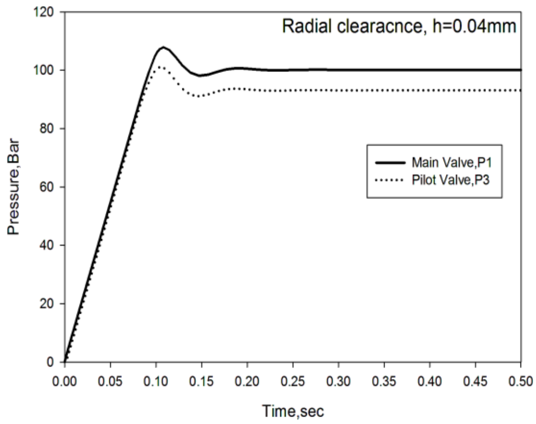

Fig 19. Effect of radial clearance h=0.04mm for 100 bar

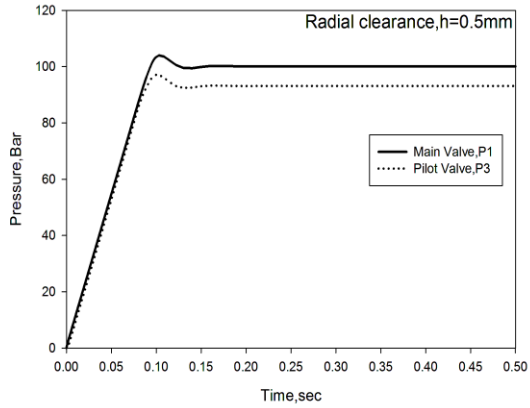

Fig 20. Effect of radial clearance h=0.05mm for 100 bar

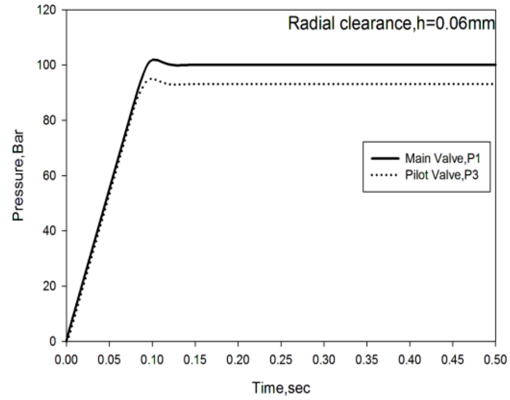

Fig 21. Effect of radial clearance h=0.06mm for 100 bar

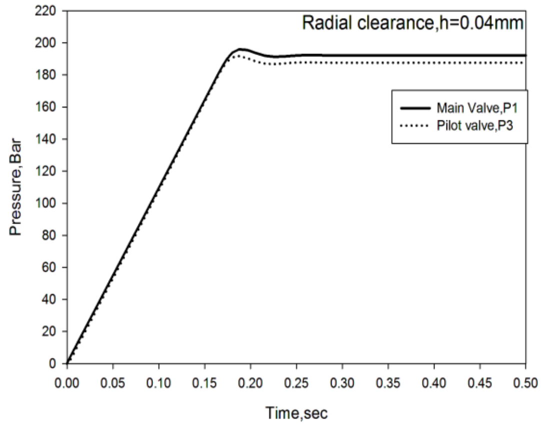

Fig 22. Effect of radial clearance h=0.04mm for 192 bar

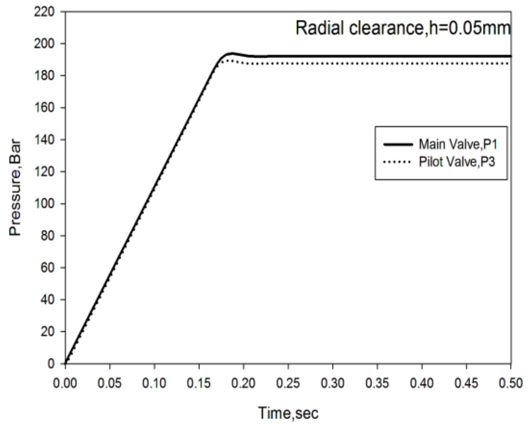

Fig 23. Effect of radial clearance h=0.05mm for 192 bar

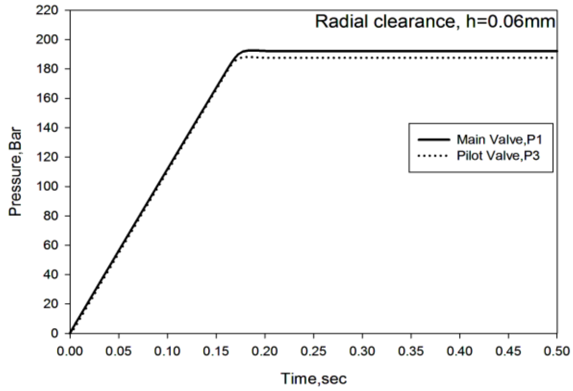

Fig 24. Effect of radial clearance h=0.06mm for 192 bar

5.4.2 Effect of inlet volume V1

For inlet volume of 1.5e-3 m3/sec, the settling time is around 100 ms for pressure of 100 [bar] and around 160 ms for pressure of 192 [bar]. Similarly for inlet volume of 2e-3 m3/sec, the settling time is around 150 ms for pressure of 100 [bar] and around 200 ms for pressure of 192 [bar]. For higher inlet volume of oil the transient response is slower. The pressure overshooting and frequency of oscillation is different for different inlet volume of oil.

Fig indicates the time for raising and peak is more for the larger inlet volume when compared with the lower inlet volume. But overshoot is more in case of low inlet volume. As can be seen at fig.4, a very fast response for lower inlet volume of oil is obtained.

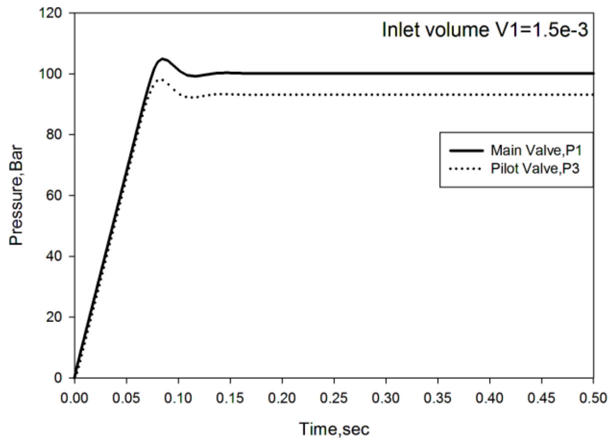

Fig 25. Effect of Inlet volume, V1=1.5e-3m^3/sec for 100bar

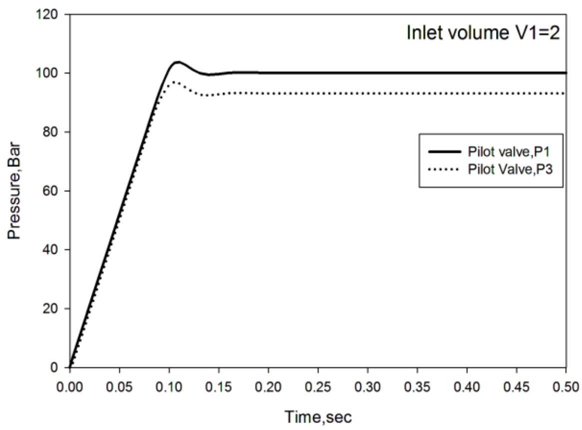

Fig 26. Effect of Inlet volume, V1=2e-3m^3/sec for 100bar

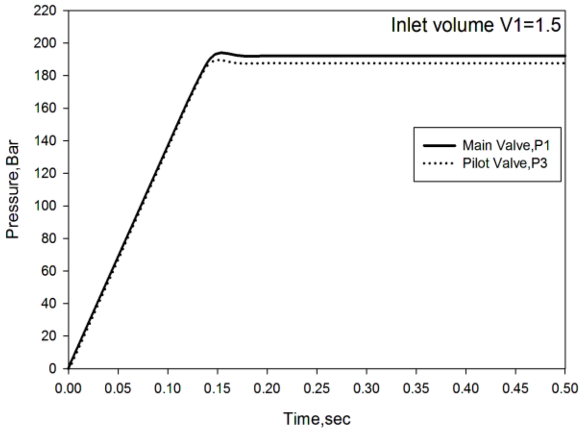

Fig 27. Effect of Inlet volume, V1=1.5e-3m^3/sec for 192bar

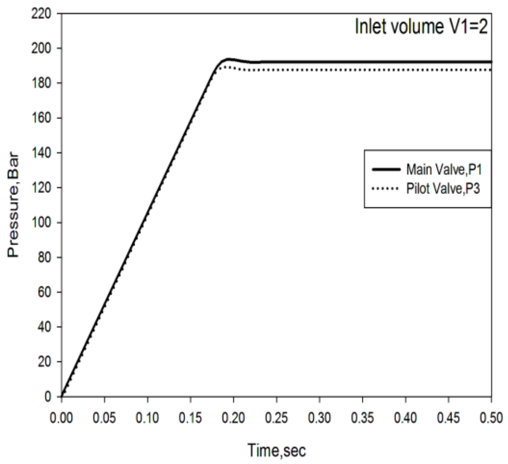

Fig 28. Effect of Inlet volume, V1=2e-3m^3/sec for 192bar

6. Conclusion

Based on the results, the following conclusions made.

(a) Simulated transient response of a valve model matches with the experimental results. By using this model it is possible to select the optimal setting points of the valve with respect to the hydraulic system operation.

(b) Transient response .is significantly influenced with the variation of the pre-compression of the pilot valve spring. There is a rise of steady-state pressure of the valve working range with the increase in the value.

(c) Overshoot and pressure override is much less than the other design of pilot relief valves.

(d) The overshoot and the peak time of the transient response are influenced by the radial clearance of the pilot poppet damping spool.

(e) Inlet volume to the valve results in additional effects on its response.

Overall it is summarized that, compound pressure relief valves pilot poppet damping spool with flat surface design gives more stability than compared to the valve studied in [19-24]. Results confirm the simulation model suitable to assess the performance as well as to choose the optimal parameters of the valve before manufacturing and installation in the hydraulic system.

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