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INFLUENCE OF OPERATING AND CONSTRUCTION PARAMETERS ON THE BEHAVIOR OF HYDRAULIC CYLINDER SUBJECTED TO JERKY MOTION

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KEYWORDS

Pilot operated check valve**,** Jerky motion, Modeling.

ABSTRACT

The present study is a step further in a planed investigation dealing with studying the dynamic behavior of hydraulic cylinder subjected to jerky motion during load lowering. It is directed to extend the use of a previously validated model to investigate theoretically the effect of some parameters, difficult to study experimentally, on the performance of the hydraulic cylinder and consequently on the related hydraulic system performance. These parameters include the pump flow rate, the spring stiffness of the pilot operated check valve, POCV, and meter-out throttle valve. In addition, an analysis is performed using the mathematical model to study the oscillation phenomena in cylinder piston side, when lowering loads. This includes studying the behavior of poppet displacement of the POCV, pressure at cylinder piston side and the cylinder piston displacement.

The results reviled that increasing the pump flow rate results in a decrease in the system pressure oscillation. By decreasing the magnitude of the spring stiffness, the pressure amplitude decreases. Meanwhile, increases the spring stiffness results in an increase in the pressure oscillation and amplitude. Using meter-out orifice with the POCV a deceleration is happened to the descending load which overcomes system pressure oscillation, but the operating pressure increases

INTRODUCTION

The hydraulic circuits which is used to raise and lower heavy loads need a valve (Pilot operated check valve, POCV, or pilot control counter balance valve, CBV) to hold the loaded cylinder when the hydraulic circuit is switched off (neutral position). The pilot signal of the check valve is either internally or externally. The internal pilot signal check valve system may produce some leakage if the pressure from the loaded cylinder overcomes the adjustment of the pilot signal pressure. The pilot check valve with external pilot signal does not cause any leakage but it causes a noticed unsteadiness oscillation while lowering heavy loaded cylinders only. A practical solution used for overcoming this phenomena is replacing the pilot operated check valve by a pilot control counter balance valve with external pilot signal. Although the pilot control counter balance valve solves this problem it results in leakage from the hydraulic circuit during neutral position. The present study is a continuation towards better understanding the previously explained problem.

Bendict and Gleed [1] considered the pressure losses involved in measuring flow rate. These losses can be large and affect the choice of pump or flow metering section to be used. Solutions are developed for predicting these losses for various fluid meter arrangements. Several methods are described for reducing fluid meter loss, including the use of a new step diffuser and conical diffuser.

Lequoc et al. [2] studied the frequency response analysis of a novel electro hydraulic servo system. A nonlinear-dimensional mathematical model has been derived for a conventional and new electro hydraulic servo configuration. In this configuration the return oil instead of flowing through the servo valve is discharged through a restrictor and the maximum discharge pressure is set by a relief valve. The theoretical and experimental analysis, of the proposed servo system subjected to step inputs, led to the conclusion that the new configuration would offer a higher steady actuator velocity, a lower percent overshoot and a shorter settling time.

Kim and Cho [3] studied the sub-optimal controller design method for the energy efficiency of a loading-sensing the system. Through a series of simulation studies and experiments, the effectiveness of the design method was illustrated by comparison with non-optimal cases. They found that the sub-optimal control system has much better control performance than several non-optimal control systems. In addition, the setting pressure is an important parameter which should be determined at the control performance and the energy efficiency of the system. Nogi et al. [4] investigated the performance of flow rate control valve using piezoelectric element. The valve has a horn and a ball which are vibrated by the piezoelectric element. The liquid flow is controlled by the valve opening time. The element is effective for decreasing the time required for the valve open in order to supply a voltage when the pulse width increases.

Tamure and Iwamoto [5] examined experimentally the factors which cause the errors in the flow rate measured using orifice plate meter under pulsating flow conditions. In the experiment a rotary valve of the butterfly valve type is used to generate the pulsating flow in the pipeline where the orifice plate is installed, and the pulsation frequency is changed. The flow rate under steady flow conditions is also measured by orifice flow meter in the line connected to the pulsating flow line via large damping tank.

Kassab et al. [6] investigated experimentally a phenomenon called jerky motion. This intermittent phenomenon is associated with the descending of high inertia loads carried by hydraulic equipment. Pilot operated check valves are used to hold the load position to avoid creep due to leakage in the hydraulic circuit. In practice, one way to eliminate the occurrence of the jerky motion, is replacing the pilot operated check valve by a pilot control counter balance valve. Therefore, they studied the internal and external performance of a hydraulic system using both types of valves one at a time. The obtained results revealed that: for the system controlled by pilot operated check valve, lowering high inertia loads result in intermittent piston motion (jerky motion) associated with considerable pressure oscillations inside the cylinder. Meanwhile, using pilot control counter balance valve instead of pilot operated check valve leads to a steady piston motion without pressure oscillations even for the case of lowering high inertia loads. For light loads, using any one of the two valves leads to a stable and steady performance, externally (no jerky motion) and internally (no pressure oscillations). Moreover, operation parameters and constructional coefficients, needed for theoretical calculations of the hydraulic system performance were obtained experimentally.

On the other hand, as a continuation of Kassab et al. [6] work, Swidan et al. [7] studied theoretically the internal and external performance of the hydraulic system before, during and after the occurrence of the jerky motion. Two systems were considered, one using pilot operated check valve and the other using pilot control counter balance valve. Mathematical models are deduced and computer simulation programs are developed and used to study the behavior of these systems. For the case of using pilot operated check valve, the results show that for the case of descending heavy loads piston motion is associated with pressure oscillations inside the cylinder. While for the case of pilot control counter balance valve the piston motion is regular and there is no pressure oscillation for all load categories, light or heavy. Comparison between the obtained theoretical results and sets of corresponding experimental results obtained by Kassab et al. [6] showed good agreement.

The present study is the third step in a planed investigation dealing with studying the dynamic behavior of hydraulic cylinder subjected to jerky motion during load lowering. The first step was the experimental study [6]. The second was the theoretical study and validation of the model through comparisons with the experimental results [7]. The third step (the present study) is directed to extend the use of the validated model to investigate theoretically the effect of some other parameters (difficult to study experimentally) which are not included in the experimental study performed by Kassab et al. [6]. These parameters include the effects of varying the pump flow rate, the spring stiffness of the pilot operated check valve, POCV, and the use of meter-out throttle valve, on the system behavior. In addition, an analysis is performed using the mathematical model to study the oscillation phenomena in cylinder piston side, when lowering loads. This includes studying the behavior of poppet displacement of the POCV, pressure at cylinder piston side and the cylinder piston displacement. The need for this analysis is raised from the feed backs obtained during the discussion of the results of Swidan et al. [7] as well as a recommendation for more deep inside analysis by experts in the related field. Moreover, the analysis may explain the inside picture of the presented results in the present study.

MODELING AND SIMULATION

The dynamic behavior of the studied system during the lowering mode is investigated in the present study. The basic construction and operational parameters during this mode are shown in Fig. 1. The DCV is represented by the two restrictions 3a and 3b. The following are the equations describing this operating mode.

Pump Flow Rate:

$$Q_p = Q_t - \left(P_p / R_L\right) \tag{1}$$

The theoretical flow rate Q_t and resistance to leakage R_L were evaluated experimentally by Swidan (2003). They were found to be:

$Q_t = 1.6 \times 10^{-4}$ m³/s and $R_L = 10^{12}$ Pa s/m³.

Continuity Equation Applied To Pump Delivery Line:

The pump delivery line connects the pump to the directional control valve, during the normal operating conditions and the relief valve is closed. The continuity equation applied to this line is:

$$Q_p - Q_{vi} = \left(V_p / B\right)\left(dP_p / dt\right) \tag{2}$$

Flow Through Directional Control Valve Restrictions:

$$Q_{vi} = C_{d1} A_{d1} \sqrt{2(P_p - P_{cp}) / \rho} \qquad \text{Or}$$

$$Q_{vi} = K_1 \sqrt{P_p - P_{cp}} \tag{3}$$

By neglecting the down stream pressure the flow through restriction 3b is given by:

$$Q_{\mathrm{cv}} = C_{d2} A_{d2} \sqrt{2\, P_{cv} / \rho} \qquad \text{Or}$$

$$Q_{\mathrm{cv}} = K_2 \sqrt{P_{\mathrm{cv}}} \tag{4}$$

Continuity Equation Applied To The Cylinder Piston Chamber:

$$Q_{vi} - A_p \frac{dy}{dt} - A_{sc} \frac{dx_c}{dt} = \frac{\left(V_{oc} + A_{sc} x_c + A_p y\right)}{B_e} \frac{dP_{cp}}{dt} \tag{5}$$

During load lowering the piston is driven by the pressure forces and constant weight of driven mass, hence the piston speed increases continuously. The pump flow rate becomes insufficient to fill the piston chamber. The pressure in this chamber drops below the atmospheric pressure. The dissolved air in the oil is released and oil evaporates to fill the excessive chamber volume. The piston chamber is thus filled with mixture of oil and gases. The bulk modulus of mixture should be recalculated to consider the high compressibility of gaseous portion of mixture. Therefore an expression for equivalent bulk modulus of elasticity is needed to consider the pressure of gases.

The following expression for the equivalent bulk modulus of elasticity was deduced [8].

$$B_e = \begin{cases} \dfrac{n(P_{cp} + 10^5)\,\mathrm{B}}{(1-\alpha)\,\mathrm{n}(\mathrm{P_{cp}} + 10^5) + \alpha B} & \text{for } \alpha > 0 \\ B & \text{for } \alpha \le 0 \end{cases} \tag{6}$$

Equation Of Motion Of Piston:

$$P_{cp}\,\mathrm{A}_p - P_{cr}\,\mathrm{A_r} + F - F_{Lm} = m_p \frac{d^2 y}{dt^2} + f_p \frac{dy}{dt} \tag{7}$$

Continuity Equation Applied To The Cylinder Rod Chamber:

$$A_r \frac{\mathrm{dy}}{dt} - Q_r = \frac{\left(V_{or} - A_r \times y\right)}{B} \frac{dP_{cr}}{dt} \tag{8}$$

Flow Rate Through The Check Valve:

$$Q_r = C_{dc}\,\mathrm{A}_{cvr} \sqrt{2(P_{cr} - P_{cv}) / \rho} \tag{9}$$

The restriction area of the check valve (A_{cvr}) is the side area of a truncated cone, Fig. 2. The following expressions are deduced for this area and the valve projection area (A_{cv}) subjected to pressure force. Details are given in [8].

$$A_{cvr} = \pi\left(\sqrt{R^2 + 2x\sqrt{R^2 - r^2} + x^2} - R\right) \times \left(\frac{\mathrm{(R)(r)}}{\sqrt{R^2 + 2x\sqrt{R^2 - r^2} + x^2}} + r\right) \tag{10}$$

$$A_{cv} = \pi\,\mathrm{h}^2 = \pi \frac{(R^2 r^2)}{R^2 + 2x\sqrt{R^2 - r^2} + x^2} \tag{11}$$

Equation Of Motion Of Check Valve Poppet:

$$\begin{aligned} &P_{cp}\,A_{sc} - P_{cv}\,\mathrm{A_{rc}} - (P_r - P_{cv}) A_{cv} + F_{\mathrm{cm}} = \\ &m_c\,\mathrm{x_c} + K_c\left(x_{oc} + x_c\right) + f_C x_c^{\circ} \end{aligned} \tag{12}$$

Continuity Equation Applied To The Check Valve Inner Chamber:

$$Q_r - Q_{cv} + A_{rc}\frac{dx}{dt} = \frac{V_{ocv} - A_{rc}x}{B}\frac{dP_{cv}}{dt} \qquad (13)$$

The hydraulic system with POCV is described mathematically by equations (1 to 13). The deduced mathematical model was used to develop a computer simulation program by using the MATLAB (SIMULINK) V. 6 package. The numerical values of constructional parameters were found by direct measurements of the real system geometry. The basic operational parameters, such as the pump resistance to leakage, were predicted experimentally [8]. The simulation program enables to calculate the transient performance of the system considering the different possible operating conditions, Fig. 3.

RESULTS AND DISCUSSIONS

Analysis Of Oscillation Behavior Of System Controlled By Pilot Operated Check Valve

Figure 4 represents the behavior of valve poppet displacement, cylinder piston pressure response, cylinder piston displacement and cylinder piston velocity. Zooming is taken on two cycles of oscillating wave. In addition the cycle is divided into different four periods to study the oscillation behavior and the relation between these four parameters with respect to each other.

Period (t_1): At the beginning of this period the pressure reaches its minimum value 9bar, while the valve poppet is closed completely. During this period the pressure is increasing, while the valve poppet is seated along the whole period. The cylinder piston velocity is decreasing at the beginning of the period until it stops completely, although the poppet is closed due to load inertia. Meanwhile during the rest of this period the cylinder piston velocity starts to act in the raising direction (opposite direction) due to reaction force effect.

Period (t_2): At the beginning of this period the pressure is increasing and reaches its maximum value 20 bar at the end of the period, and the poppet displacement increases too due to pressure increase. Meanwhile the velocity at the beginning of period continues acting in the opposite direction and at the rest of the period it starts to increase in the direction of lowering.

Period (t_3): At the beginning of this period the pressure reaches its maximum value and starts to decrease all over the period, due to the increase in cylinder piston velocity, while the valve poppet displacement is still increasing due to valve inertia and the high flow rate through its restriction area resulting from the cylinder piston high velocity.

Period (t_4): The pressure and the poppet displacement are still decreasing until they reach their minimum value at the end of this period. Meanwhile the cylinder piston velocity increases until it reaches its maximum value and then starts to decrease again due to the decrease in poppet displacement.

It was noticed that the cylinder piston (displacement and velocity) are the main parameters affecting the pressure variation and hence the opening and closing of the POCV during descending stroke.

Parametric Study

Some parameters are studied theoretically using the mathematical model, which is easier than studying them experimentally.

Pump Flow Rate:

Figure 5 represents the pressure performance of system loaded by 680 N, while using different flow rate values. Figure 5-a shows the results for the case of increasing pump flow rate from 9.6 lit/min, which is the used flow rate value in both experimental [6] and theoretical [8], by 25% and 50%, to be 12 lit/min and 14.4 lit/min respectively. It is noticed that by increasing pump flow rate the pressure oscillation decreases and the load starts to descend smoothly. These results confirm the explanation given by Swidan et al. [7] for system oscillation while lowering 610 and 680 N loads. As it was stated that the speed of cylinder piston increases rapidly by increasing the load and the pump flow rate becomes less than the rate of increase of piston chamber volume therefore the pressure drops in the pressure line and in pilot line causing the check valve to close and cylinder piston stop. Meanwhile the use of high pump flow rate, compensate the increase of piston chamber volume due to rapid speed while using loads 610 and 680 N. Consequently, the oscillation effect is disappeared.

Figure 5-b shows the results for the case of decreasing pump flow rate from 9.6 lit/min, which is the used flow rate, by 25% and 50%, to be 7.2 lit/min and 4.8 lit/min respectively. This figure shows that by decreasing pump flow rate the pressure oscillation is still found, but with lower amplitude. This phenomena may be because when the pressure reaches the cracking pressure value of pilot operated check valve (just starts to open), there is a tiny time delay until the valve is opened due to valve overlap distance between the valve piston and poppet, which is equal to 3.5 mm in the present study. At this tiny time delay period the cylinder piston is at rest (constant cylinder piston chamber volume). Consequently, the main factor affecting the pressure amplitude is the change of oil volume occupying the cylinder piston chamber. By decreasing the pump flow rate the change of oil volume occupying the cylinder piston chamber will decrease too, leading to the decrease of the pressure amplitude.

Spring Stiffness:

Figure 6 represents the pressure performance of system loaded by 680 N, while using different spring stiffness. By changing the spring stiffness, 110 kN, which is the value of spring stiffness used in the mathematical model, once increasing it by 100% to be 220 kN and the other by decreasing it by 50% to be 55 kN. It was noticed that by increasing the spring stiffness the oscillation amplitude increases due to the increase of force needed to open the valve. Meanwhile by decreasing the spring stiffness the load descends smoothly with small transient period. On the other hand, decreasing the spring stiffness results in a decrease in the operating pressure amplitude. This decrease leads to facilitate the probability of leakage from the valve restriction area. This leakage may results in descending the load while the system is switched off.

Using Meter-Out Throttle Orifice:

Figure 7 represents the pressure performance at 680 N load while using the same system with POCV, but after adding meter-out orifice of diameter, d =2.5 mm. It was noticed that by using the orifice no pressure oscillation is noticed in the system, but the operating pressure increases. The obtained results can be explained, as follows: By using the meter-out orifice it causes a deceleration to the descending load so the pump flow rate can overcome the increase in cylinder piston chamber due to the piston displacement.

One the other hand, the obtained results due to the presence of meter-out orifice, can help in understanding and explain the reason for the difference in response for system using POCV and system using CBV. It was noticed from the measured values of poppet construction [8], that the magnitude of the variation in restriction area due to poppet displacement is greater in case of POCV than that of CBV, Fig. 8. Meanwhile it can be noted that this throttling effect in case of CBV leads to stability of operating pressure due to the slow increase of the piston chamber volume. This effect is corresponding to the effect of using meter-out throttle orifice as shown in Fig. 7.

CONCLUSION

In the present study, a parametric study is performed for the hydraulic system controlled by pilot operated check valve, POCV, using a mathematical model. From the parametric study it was deduced that:

- By increasing pump flow rate the system pressure oscillation decreases.
- By decreasing the magnitude of the spring stiffness, the pressure amplitude decreases, but may lead to facilitate the probability of leakage from the valve restriction area.
- By increasing the spring stiffness the pressure oscillation and amplitude increases.
- By using meter-out orifice with the POCV a deceleration is happened to the descending load which overcomes system pressure oscillation, but the operating pressure increases.

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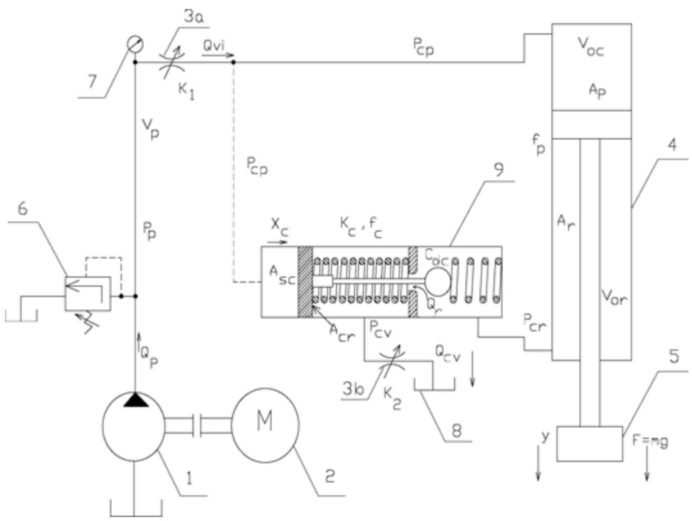


1. Gear pump	2. Electric motor	3. 4/3 DCV
4. Single acting cylinder	5. Load,	6. Relief valve
7. Pressure gauge	8. Tank	9. POCV
10. Pressure transducer		

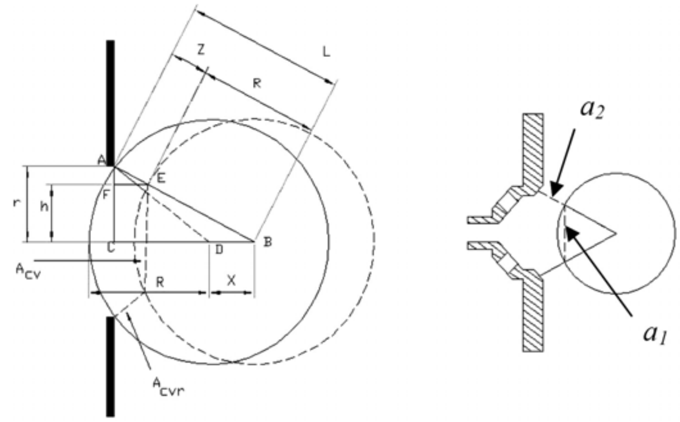


Fig. 2 Scheme of the check valve (a_1= A_{cv} , a_2= A_{cvr})

Fig. 1. Functional scheme of the system during lowering mode

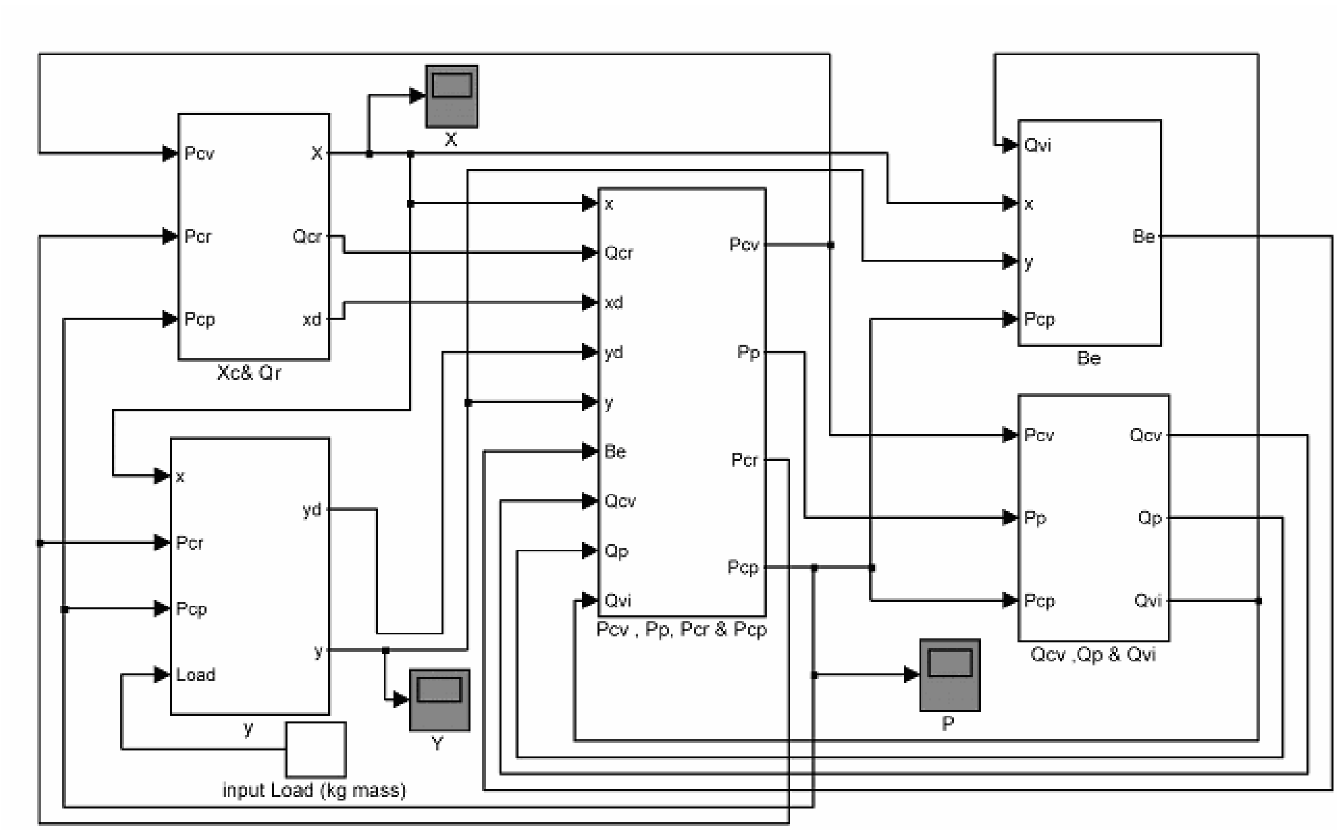


Fig. 3 First page of the SIMULINK program developed for the system with POCV.

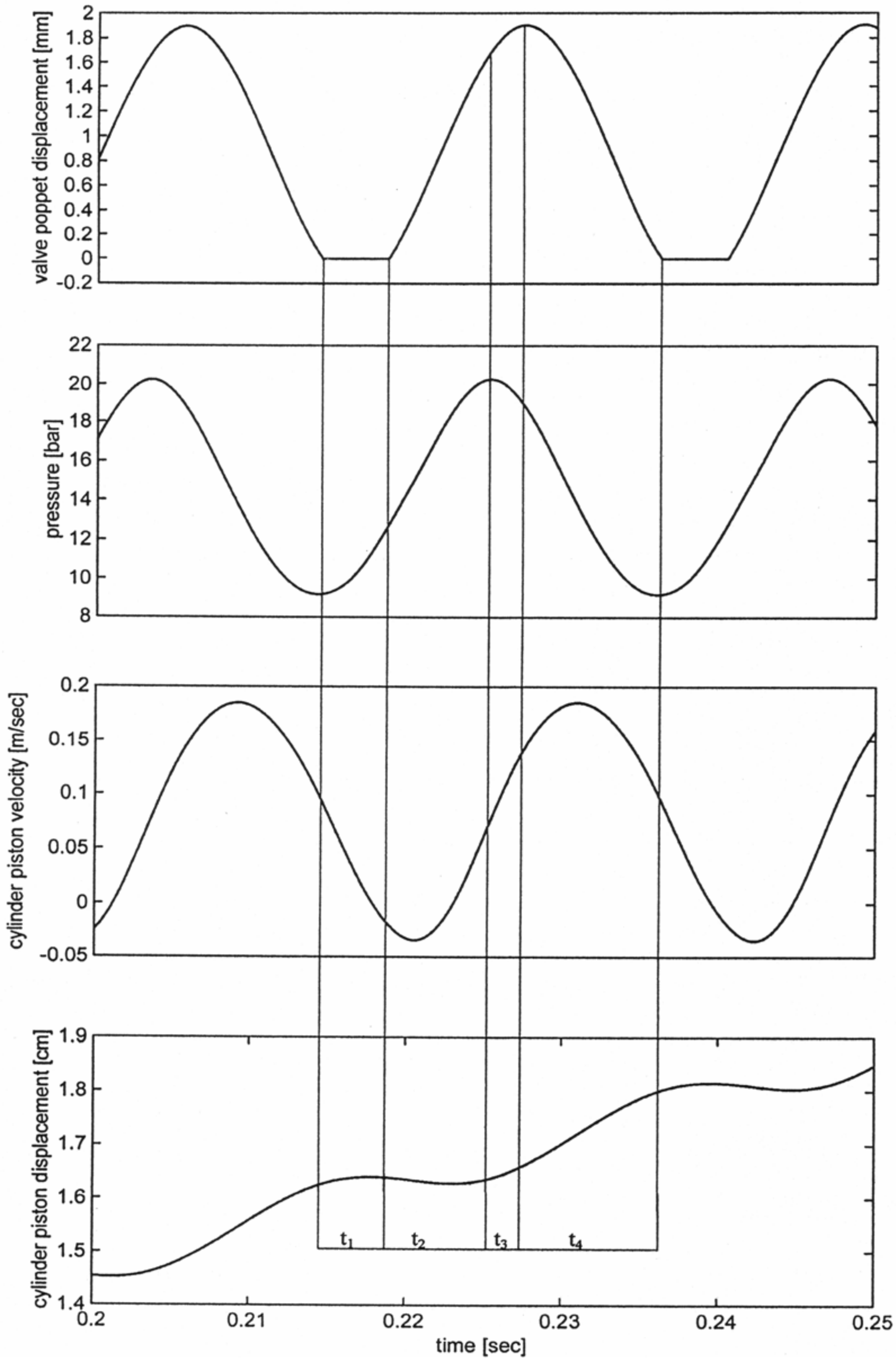


Fig.4 The relation ship between poppet displacement, cylinder piston pressure response, cylinder piston displacement and cylinder piston velocity during lowering with a load of 680 N.

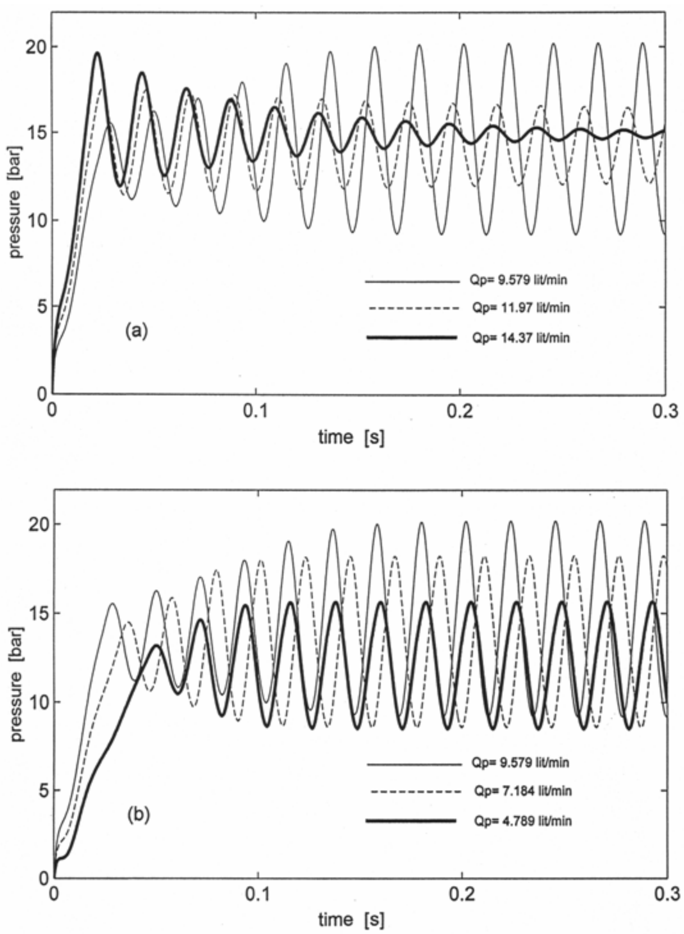


Fig. 5. Pressure transient response with respect to pump flow rate

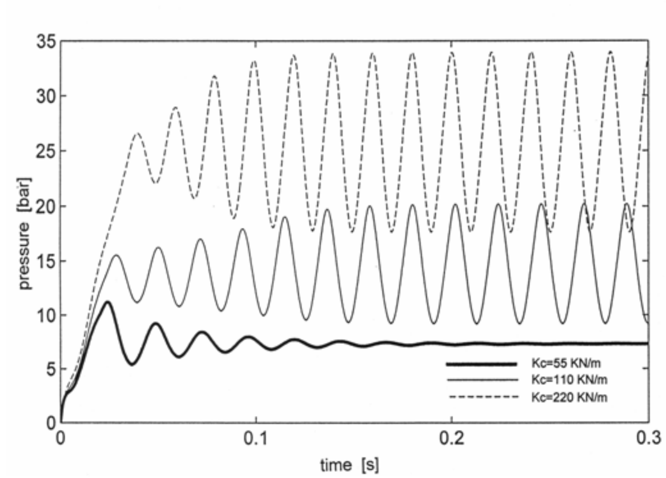


Fig. 6. Pressure response according to spring stiffness

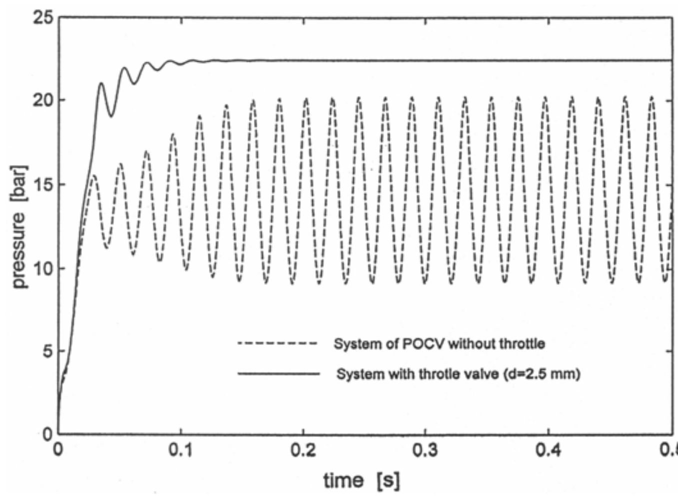
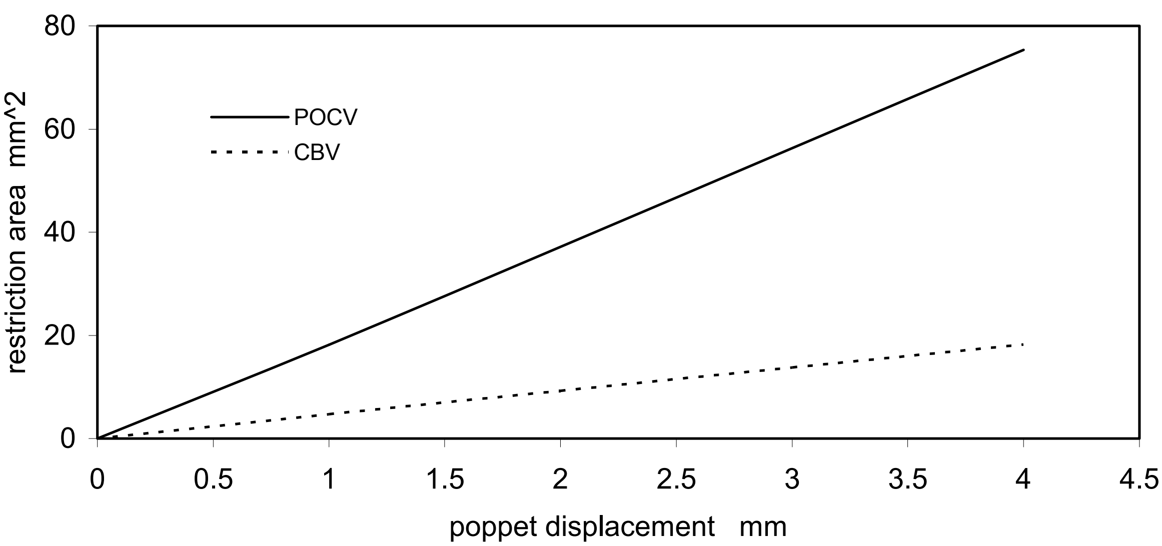


Fig. 7. Pressure response using meter out throttle orifice



Fig, 8. Response of restriction areas of POCV & CBV according to the poppet displacement