Parameter estimation of soft switch and influence of critical parameters on the performance of a valve controlled hydraulic drive system

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Abstract. This study presents a technique to estimate the parameter values of the soft switch, which can be used for reduction of pressure surge due to abrupt mode shift or speed reversal in high pressure hydrostatic transmission. The estimation of critical parameters is essential for component design and fabrication thereafter. Different hydraulic systems should have different parameter values of the soft switch. It depends on the developed pressure in the switched volume and valve transition time. The throttling energy loss of an on/off valve controlled hydraulic system is influenced by different component parameters of the soft switch, which has also been investigated in this work.

1. Introduction

Power hydraulic system is inefficient as comparison to the other system (mechanical, electrical etc.) due to multiple sources of energy losses. In power hydraulics, different types of energy losses are frictional loss of fluid, leakage loss, fluid compressibility loss, accumulator hysteresis loss, pressure drop loss and throttling energy loss of an on/off valve. Among these, throttling energy loss is the major energy loss i.e. 60% of the total energy loss and rest are other losses [1-2]. This is caused by the transition of an on/off valve from one position to the other position on a hydraulic circuit. Sometime this loss is also called as valve transition time loss. Therefore, it is important to enhance the energy efficiency of the hydraulic system by minimizing the throttling energy loss. The energy efficiency of a hydraulic system can be increased by using a unidirectional 3-way rotary valve for pulse width modulation (PWM) which modulate rapid switching, as a result decreases the throttling energy loss [3]. The concept of 3-way rotary valve and its design, governing equitation and it dynamic model were validated experimentally by Tu *et al.* [4]. Later, extend the 3-way rotary valve concept into 4-way tandem rotary valve for virtually variable displacement pump/motors (VVDPM) hydraulic system [5]. However, the results shown that the efficiency of the VVDPM was decreased by 6%. In [6-7], authors were reported a switched inertance hydraulic system (SIHS) to control the supply flow and pressure. That system (SIHS) commenced a flow booster for investigate and it comprises of switching element, a capacitance and an inductance (inertance). The experimental results shown that the system efficiency can be increased significantly by using SIHS. For further development of SIHS, Yudell and Van de Ven [8] had applied power electronics concept i.e. analog of zero-voltage-switching concept on it. It allowed throttled fluid to store in a capacitive element and it bypassed through check valves in parallel

with the switching valves. It was observed that the system with soft switching boost converter provided 42% more efficient as comparison to the system of boost converter without soft switching. For further improvement of energy efficiency of the hydraulic system, Rannow and Li [9] had provided a concept of soft switching. In that concept, an additional component i.e. soft switch was incorporated into an on/off valve controlled hydraulic system. The main task of the soft switch was to store the high pressure switched volume working fluid into it, temporarily, when the valve was in transition mode. The soft switch is comprised of four parts: cylindrical chamber, one piston, one compression spring and an external locking/unlocking mechanism. Initially, the piston was locked at the middle of the soft switch. The simulation responses substantiated that the throttling energy loss and total energy loss of the system were reduced by 81% and 64%, respectively by implementing the soft switching concept. But it has a drawback on locking/unlocking mechanism. If the piston is not released or locked with appropriate timing, it may accumulate the throttling energy loss. Thus, the mechanism is highly sensitive and should operate with high precise timing to diminish the throttling energy loss. To overcome the drawback, the soft switch was redesigned by Ven de Ven [10] where high pressure signal in the switched volume was used to trigger the unlocking mechanism of the piston of the soft switch. Also, the redesigned soft switch piston was locked at the top of the cylindrical chamber which enhanced the storing capacity of the soft switch in comparison with the previous model. Author [10] had simulated and compared the responses of five different configured hydraulic systems, without and with soft switch. It was established that the throttling energy loss of a valve controlled hydraulic system was reduced by 66.1%. That article ([10]) used two different types of soft switch: one is locking soft switch and another is passive soft switch. Thereafter, Beckstrand *et al.* [11] validated the same theory of soft switching experimentally by incorporating locking soft switch only. But it could not provide the reason of eliminating of the passive soft switch from the system. Later, the influence of the passive soft switched is reported by Mahato *et al.* in [12] and remarked that additionally 3.25% throttling energy loss can be saved by discarding passive soft switch from the system. Thereafter, Mahato *et al.* [13] experimentally validated the locking soft switch performance regarding reducing the throttling energy loss by considering a typical hydrostatic drive system. Authors in [13] used multi-run- simulation technique to optimize the parameter values of the locking soft switch. The estimation of parameter values of the soft switch is highly important to operate the soft switch precisely. But all these studies have ignored this. Also, it is imperative for fabrication of the locking soft switch. Moreover, the influence of different parameters of the soft switch on the throttling energy loss of the hydraulic system has been ignored previously but same has been reported in this work.

   This article presents a method to estimate parameter values of the locking soft switch for a particular hydraulic system that is required to fabricate the soft switch. Also, it reports the influence of soft switch piston radius, spring stiffness and dynamic viscosity of different working fluids on the throttling energy loss of an on/off valve controlled hydraulic system.

2. Operation of an On/off valve controlled hydraulic system with locking soft switch

Hydraulic pump (fixed displacement) supplied an invariable flow (Q_{pump}) to the loading side of the hydraulic circuit through the load valve. The loading side flow is varied and it is controlled by the valves (load and tank valve). At one time only one valve is opened and other is in closed condition. Initially, the pump flow is passed through the tank valve. When the tank valve starts to close, a high pressure is developed into the system until the load valve will opened completely that means the locking soft switch starts working. Similarly, during the closing mode of the load valve, the locking soft switch is in working mode until the tank will opened fully. The detail working principle of the soft switch is addressed by Van de Ven in [10]. The soft switch facilitates the reduction of the valve transition time loss [9-13]. The locking soft switch is a combined mechanism of piston, compression spring, cylindrical chamber and two check valves. Referring to figure 1, one check valve restricts the flow from accumulator to the soft switch. This check valve used for safety purpose only to releases the high pressure from the rear end chamber of the soft switch. Other check valve allows the flow from the tank to the rear end chamber of the soft switch whenever the pressure at the rear end chamber is less than the tank pressure. This situation arises only when the cross drilled port of the piston coincides

with the annulus of the cylindrical chamber of the soft switch. In such situation soft switch releases the stored fluid into the main line.

The governing equations (steady state equations) of the said system have been already discussed in the article [12]. The pulse width modulation (PWM) of the valve is shown in figure 2. Referring to the figure 2, initially the tank valve is opened and the load valve is closed condition. The full flow of the pump is passed through the tank valve. The closing transition time period of the tank valve is t_1 and opening transition time period of the load valve is t_2. After certain period of operation of the load valve, the closing transition time period of the load valve is t_3 and thereafter subsequent opening transition time period of the tank valve is t_4. Thus, at point 7 (refer figure 2), the system completed one duty cycle.
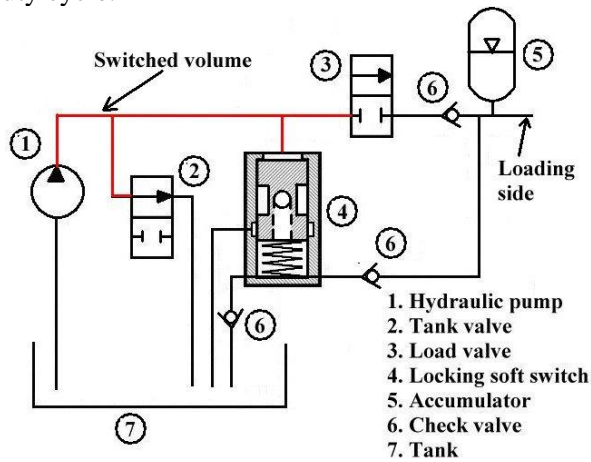

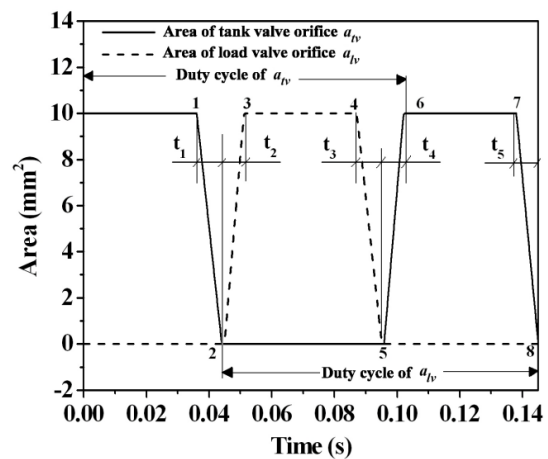Figure 1. On/off valve controlled hydraulic system with locking soft switch

Figure 2. Pulse Width Modulation (PWM) of the valve

3. Process of estimating the soft switch parameter values

In this section, the complete design of locking soft switch and its parameter values estimating process has been discussed which is useful for fabrication of the soft switch. In previous papers, this topic has been ignored. The detail sectional view of the locking soft switch is shown in figure 3.
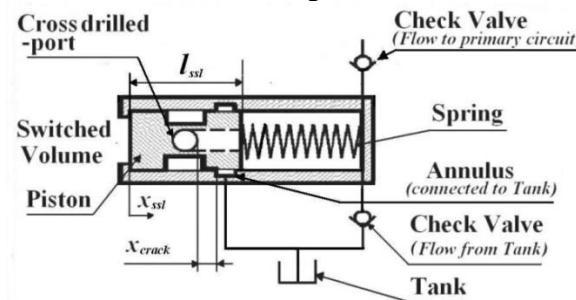

Figure 3. Detail cross-section view of the locking soft switch and it components

3.1. Parameter values estimation of the compression spring

The force applied on the soft switch piston is estimated from the dynamics of soft switch and is expressed in equation. (1)

$$\dot{v}_{ss} = \frac{P_{sv}a_{ss} - F_{pld} - k_s x_s - \mu_{ss}v_{ss}}{m_{ss}} \tag{1}$$

where, P_{sv} is the measured switched volume pressure; $a_{ss}\,(=\pi r_2)$ is the piston surface area: F_{pld}, k_s and x_s are the preload compressive force applied to the spring, spring stiffness, and spring displacement, respectively; m_{ss} is the piston mass; v_{ss} is the linear velocity of the piston and μ_{ss} is the frictional resistance between the piston and internal surface of the cylindrical chamber.

The wire diameter of the spring (d) is estimated from equation (2)

$$d = \sqrt{\frac{8k_{wf}Fc}{\pi\tau}} \tag{2}$$

where, F is applied force on the spring, c is spring index ($c = (D/d)$), whose numerical value to be preferred in the range 6 to 9 for close tolerance spring and where subjected to cyclic loading [14]. D is the mean coil diameter ($D = (D_i + D_o)/2$). D_i and D_o are the inside and outside diameter of the spring coil respectively. τ is the permissible shear stress of the spring and k_{wf} is the Wahl factor which is expressed as

$$k_{wf} = \frac{4c-1}{4c-4} + \frac{0.615}{c} \tag{3}$$

The number of the coil of the spring (n) is calculated by equation (4)

$$n = \frac{Gd^4}{8D^3k_s} \tag{4}$$

where, G is the modulus of rigidity of the spring.

The free length of the spring (l_f) is estimated by equation (5)

$$l_f = nd + (n-1)g_t + \delta \tag{5}$$

where, g_t is the gap between adjacent coils and δ is deflection of the spring when it's subjected to maximum load.

The pitch of the spring (P_i) is expressed as

$$P_i = \frac{l_f}{(n-1)} \tag{6}$$

Therefore, the total length of the soft switch (L_{ss}) is

$$L_{ss} = l_{ss} + l_f + 2s \tag{7}$$

where, l_{ss} and s are the length of the soft switch piston and wall thickness of the soft switch, respectively.

4. Parameter influences

The dynamic response is important for the hydraulic system performance. The influence of different soft switch parameter (piston radius, spring stiffness and dynamic viscosity of the working fluid) values on the dynamic responses of the soft switch has been analysed. The parameter values used for simulation are available in [12]. The effect on the throttling energy loss of the system with the variation of the piston radius is shown in figure 4.
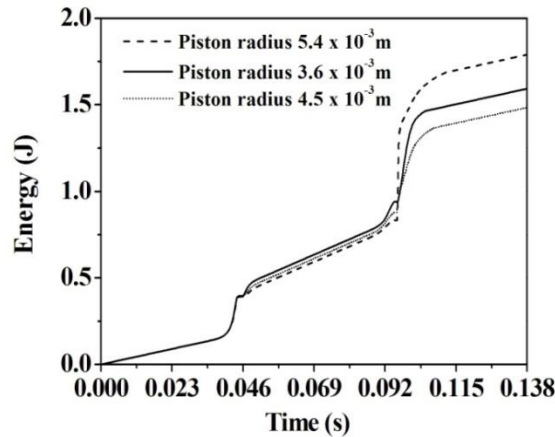

Figure 4. Comparison of throttling energy loss with the variation of piston radius

The throttling energy loss is minimum i.e. 1.483 J, when the piston radius is 4.5×10^{-3} m which is also the optimized parameter value of the piston radius as per the article [12]. Apart from that other values accumulate more throttling energy loss. Also, lower value of the piston radius needs more precise fabrication of the soft switch.

The influence of spring stiffness on the throttling energy loss of the system is shown in figure 5. The throttling energy loss is minimum i.e. 1.483 J, when the spring stiffness is 6276 N/m. This is increased with increasing the spring stiffness. Also, the throttling energy loss increases with decreasing the spring stiffness from its optimum value 6276 N/m. Thus, it can be said that all other values except the optimum value of the spring stiffness, accumulated more energy loss from the system.
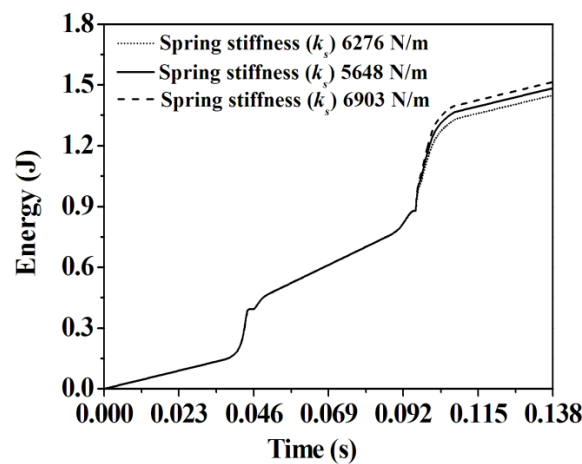

Figure 5. Comparison of throttling energy loss with the variation spring stiffness

Similarly, a comparison analysis is made by varying the dynamic viscosity of the different working fluids. However, it is assumed that the working fluid is incompressible and its properties do not change with temperature. The main purpose of this study is to analyze the effect of throttling energy loss with different working fluids as their dynamic viscosities are different. Figure 6 represents the affect on throttling energy loss with various dynamic viscosities working fluids. It is observed that lower dynamic viscosities working fluid provides better energy efficiency of the hydraulic system. The throttling energy loss increased with the higher dynamic viscosities working fluids.
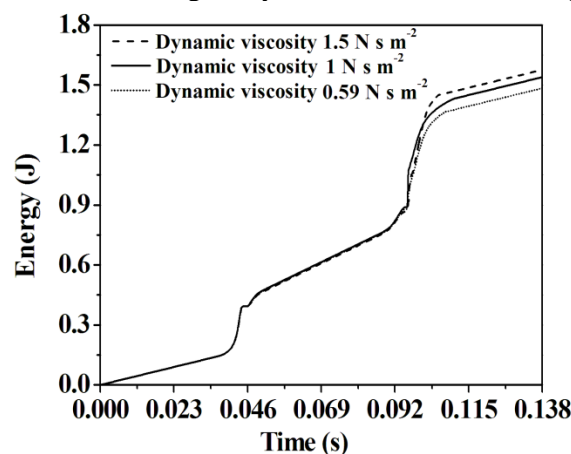

Figure 6. Comparison of throttling energy loss with various dynamic viscosities working fluids

5. Conclusions

The way of reduction of the throttling energy loss of the hydraulic system by using soft switching concept is an important and power full technique. It provides massive energy saving by incorporating a low-cost equipment i.e. soft switch into the hydraulic system before the on/off valve. The following conclusions are obtained from the present study.

1. The throttling energy loss of the system is higher except the corresponding optimized value of piston radius. Any other value of piston radius except its optimized valve can accumulate more throttling energy loss.

2. All other values of spring stiffness rather than its optimized valve can also accumulate higher throttling energy loss.

3. Higher values of dynamic viscosities of the working fluids increases throttling energy loss from the valve controlled hydraulic system.

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