A RESEARCH ON ENERGY EFFICIENCY IMPROVEMENT OF WATER HYDRAULIC MOTOR SYSTEM

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ABSTRACT

Improving energy efficiency in hydraulic systems is very important problem, especially in water hydraulic systems with lower energy efficiency than oil hydraulic ones. This research proposes a method to reduce the supply energy of a water hydraulic motor system based on reducing the supply pressure in the system. The simulation results showed that the proposed system could raise the energy efficiency nearly 3 times with the same control response quality to the conventional servo motor system.

1. INTRODUCTION

Nowadays, water hydraulics still faces with some main difficulties for widening application. First, the initial cost of water hydraulic components is normally more expensive than oil hydraulic ones; this property can be compensated by using very cheap pressure medium (water), much reducing insurance and disposal fees if the utilized period is long enough. The second challenge is that the control performances of water hydraulic systems are effected by nonlinearity; strong friction and considerable leakage than oil hydraulics. This can be overcome by using advanced control methods or by improving the performance of water hydraulic devices. Very important challenge for not only water hydraulics but also oil hydraulics is low energy efficiency of hydraulic systems, from 6% - 40% for oil hydraulics depending on applications (Wang & Wang, 2013) and even lower in water hydraulics.

Researching on energy-saving in hydraulic systems is very attractive. There are some trends for saving energy such as using load sensing system or controlling the velocity of the prime mover connected to a hydraulic pump for matching the required energy supply; find the way to recover potential energy of cylinder or braking energy of hydraulic motor in accumulator or electric circuit etc.

Water hydraulic variable displacement pump does not exist in market so far; therefore, cannot applying load sensing system in water hydraulic systems. Recovering potential or kinetic energies of cylinder or motor is possible as shown in (Inoguchi et al. 2012), (Pham et al. 2013). However, the control performance of such systems were limited and the recovered energies were relatively small in comparison with the total energy consumption in such systems.

This paper proposes a novel system with the idea of reducing the supply pressure to be matched with the load pressure by using a direct connection between a hydraulic pump and motor, the pressure between these devices is controlled by fluid control valves for releasing the surplus flow to reservoir. By this method, the energy efficiencies raised three times than a compared conventional system.

2. SYSTEM MODELLING

To examine the influence of characteristics of the components to the performance of the systems and to get the simulation results for evaluating and comparing sketchily about the control performance and energy efficiency of the proposed system to a conventional servo motor system, the simulated was constructed before doing experiment in the next steps. This section introduces a proposed water hydraulic circuit for motor system and a conventional servo motor system circuit for comparison, mathematical model of important devices in these both systems.

2.1. Proposed System and Servo Motor System

Figure 1 shows the block diagram of the proposed system, which includes following main elements. A fixed displacement pump P connected to an engine E; a hydraulic motor M is connected to a flywheel FW, which is considered as a rotational load of the system. Two solenoid On/Off valves SV_1 and SV_2 are assembled to two

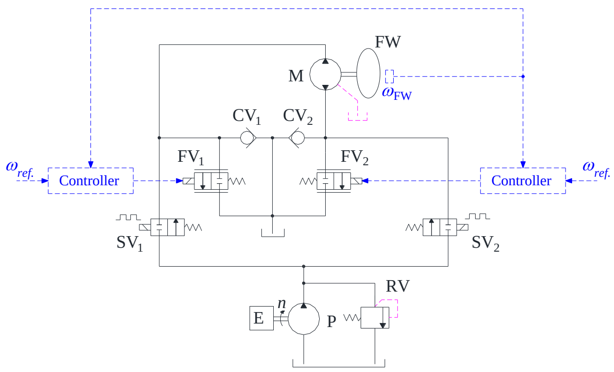


Fig. 1 Proposed water hydraulic circuit for motor system.

path from the pump P to the motor M for changing the rotational direction of the flywheel FW. These two paths are connected to two check valves CV_1 and CV_2, which permit the fluid flows only one direction. Two flow rate control valves FV_1 and FV_2 are branch-connected at one end thereof to check valves CV_1 and CV_2. These fluid control valves are main devices to control the rotational velocity of the flywheel.

The operation of the system can be explained as follows; based on the reference phases as shown in Fig. 3(a). In the first cycle corresponding to the clockwise rotational direction of the flywheel, the On/Off valve SV_1 is opened, the valve VS_2 is closed. In the acceleration and constant velocity phases, the fluid control valve FV_2 is opened fully and the control valve FV_1 is used to control the velocity of the flywheel to track the reference. In the deceleration phase, the valve FV_1 is closed, the valve FV_2 is used to control the velocity of the flywheel. Meanwhile, the suction fluid supplied to the motor flows through the check valve CV_1. In the second cycle corresponding to the anti-clockwise rotational direction of the flywheel, the operation of the system can be described same as above, only the On/Off valve SV_2 is opened instead and the valve SV_1 is closed.

Figure 2 shows the block diagram of a conventional water hydraulic servo motor system with the main components are as follows. A fixed displacement pump P connected to an engine E; A pressure relief valve RV is assembled on the output port of the hydraulic pump P. A hydraulic motor M with flywheel FW is driven by a flow control valve FV (servo valve). For easy comparison, the characteristics of the engine E, pump P, hydraulic motor M, and flywheel FW were chosen same as in the proposed system.

2.2 Mathematical model

2.2.1 Flow control valve.

In this simulation, the behavior of the flow control valves FV_1 and FV_2 are considered as the behavior of the servo valve FV after reducing two ports and one position to become two ports, two positions. The mathematical model of the servo valve FV (as well as the two valves FV_1 and FV_2) is as follows.

The spool valve displacement $x_v(t)$ is related to the control input $u(t)$ by a first-order system given by

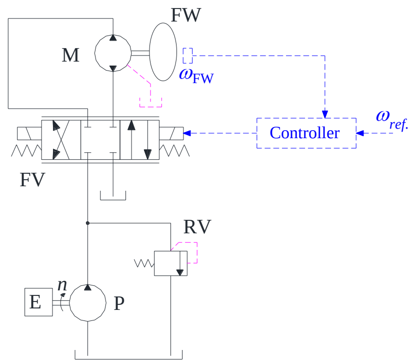


Fig. 2 Conventional water hydraulic servo motor system.

$$\tau_v \dot{x}_v(t) = x_v(t) + k_v u(t) \tag{1}$$

where τ_v and k_v are the time constant and gain of the servo valve dynamics, respectively.

The leakage of the servo valve is quite small and it is neglected in this research. Hence, the servo valve flows Q_1 and Q_2 can be calculated as follows (Yao et al., 2000):

$$Q_1 = \begin{cases} k_{q1} x_{ve} \sqrt{|P_s - P_1|}, & \text{for } x_{ve} \geq 0 \\ k_{q1} x_{ve} \sqrt{|P_1 - P_r|}, & \text{for } x_{ve} < 0 \end{cases} \tag{2}$$

$$Q_2 = \begin{cases} k_{q2} x_{ve} \sqrt{|P_2 - P_r|}, & \text{for } x_{ve} \geq 0 \\ k_{q2} x_{ve} \sqrt{|P_s - P_2|}, & \text{for } x_{ve} < 0 \end{cases} \tag{3}$$

where P_1, P_2, P_s, and P_r are the pressures of the two ports of the motor, the supply pressure, and the reservoir pressure, respectively; k_{q1} and k_{q2} are the flow gain coefficients of the servo valve.

2.2.2. On/Off Valve

The flow rate $Q_{On/Off}$ through On/Off valve can be calculated by the orifice equation as follows (Jelali & Kroll, 2004):

$$Q_{On/Off} = \alpha_d A_{On/Off} \sqrt{\frac{2}{\rho} \Delta p} \tag{4}$$

where α_d is the discharge coefficient, $A_{On/Off}$ is the opening area of the On/Off valve, ρ is the mass density of water, and Δp is the pressure difference between the input and output ports of the On/Off valve.

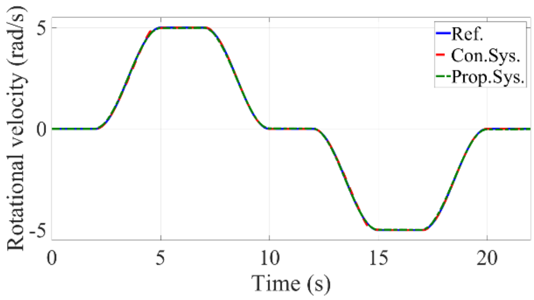
2.2.3. Hydraulic Motor

Newton's second law is applied to obtain the torque balance equation for the motor,

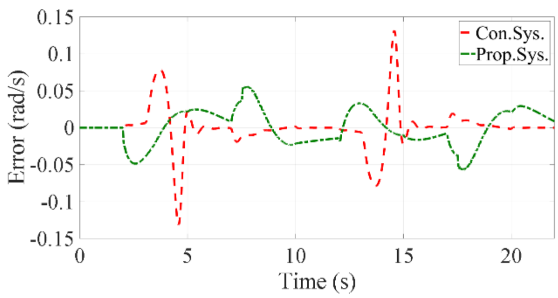
$$I_{fw}\dot{\omega} + T_f = T_m \tag{5}$$

where I_{fw} is the moment of inertia of the flywheel, T_f the friction torque, and T_m the torque of the motor.

The torque of the motor can be expressed by



(a)



(b)

Fig. 3 Control response (a) and tracking error (b).

$$T_f = T_v\omega + \mathrm{sign}(\omega)\left[T_{c0} + T_{s0}\exp\left(-\frac{|\omega|}{c_s}\right)\right] \quad (6)$$

where T_v, T_{c0}, T_{s0}, and c_s are the coefficients for viscous friction, Coulomb friction, and static friction, respectively.

$$T_m = \eta_m \frac{D_m}{2\pi}(P_1 - P_2) \quad (7)$$

where η_m is the estimated overall of volumetric and mechanical efficiency of the motor and D_m is the displacement volume of the motor.

Based on the continuity equation, the pressure dynamics in the motor chambers are given by

$$\dot{P}_1 = \frac{E}{V_0}\left[Q_1 - \frac{D_m}{2\pi}\omega - C_{Li}(P_1 - P_2)\right] \quad (8)$$

$$\dot{P}_2 = \frac{E}{V_0}\left[-Q_2 + \frac{D_m}{2\pi}\omega + C_{Li}(P_1 - P_2)\right] \quad (9)$$

where E is bulk modulus of water assumed to be constant value and C_{Li} the internal leakage flow coefficient. Both chamber volumes have been assumed to be equal to V_0.

3. SIMULATION RESULTS AND DISCUSSION

3.1 Control Performance

Figure 3 shows control performances (Fig. 3(a)) and tracking errors (Fig. 3(b)) of both conventional servo motor system and proposed system. It is easy to see that both system operated well with very small error (less than 2%). Only conventional servo motor system produced a little bit bigger overshoot (±0.131 rad/s for the maximum reference velocity of ±5 rad/s). The bigger overshoot than that in proposed system was due to the conventional system worked with higher supply pressure while the supply pressure was much smaller in proposed system as shown in Fig. 4.

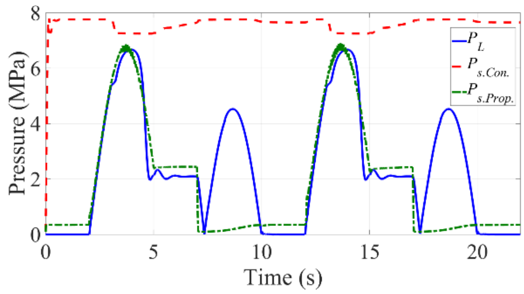


Fig. 4 Load pressure and supply pressures in conventional system and proposed system.

3.2 Energy Efficiency

Figure 4 shows the load pressure in conventional system ($P_{L.Con.}$), which is derived in Eq. (10), and supply pressures in both conventional servo motor system and proposed system with much reduction in the proposed system can be seen clearly. In the deceleration phases, which are from 7 to 10 sec and from 17 to 20 sec, the supply pressure in the proposed system reduced to around atmospheric pressure because this system using the meter out method to brake the flywheel. No supply energy was required in the deceleration phases.

$$P_{L.Con.} = |P_{1.Con.} - P_{2.Con.}| \quad (10)$$

Figure 5 shows the supply flow rate, and the flows through the hydraulic motor M in conventional and proposed systems. The maximum value of flow rates in both systems are at the constant (velocity) phases, the supply flow was chosen to be very near this value. Only very small gap exists to compensate for the leakage in devices. In the constant phases or steady state, no flow rate was required. However, the supply responses delivered the same value of flow rate for all the working time. It made the huge loss in conventional system. In the proposed system, at these time, the supply pressure automatically reduced to very small value because of the full open of the flow control valve FV_1 or FV_2; as a results, the energy loss in these periods was diminished drastically.

The energy consumptions of conventional servo motor system ($E_{Con.}$) and proposed system ($E_{Prop.}$) can be calculated by following equations.

$$E_{Con.} = \frac{1}{\eta_P}\int_{t=t_{start}}^{t=t_{end}} P_{s.Con.} Q_s dt \quad (11)$$

$$E_{Prop.} = \frac{1}{\eta_P}\int_{t=t_{start}}^{t=t_{end}} P_{s.Prop.} Q_s dt \quad (12)$$

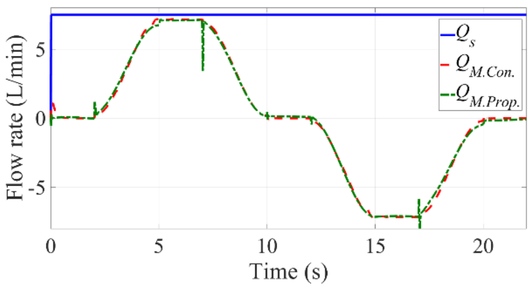


Fig. 5 Supply flow and flows through the motor M in conventional system and proposed system.

where $P_{s.Con}$, $P_{s.Prop}$ are the supply pressures in conventional and proposed systems, respectively; η_P is the efficiency of the hydraulic pump, getting the value of 85% in this research; t_{start} and t_{end} are the starting and finishing times of simulation.

The effective energy (E_{eff}) for driving the hydraulic motor can be considered as the supply energy to the motor ports, which is calculated via load pressure and flow through the hydraulic motor M, multiplies with the energy efficiency of the hydraulic motor η_M, getting the value of 90% in this research. Based on Fig. 5, the flows through the hydraulic motor M in both systems are approximately equal. Thus, the effective energy to both systems is assumed to be derived by following equation.

$$E_{eff.} = \eta_M \int_{t=t_{\text{start}}}^{t=t_{\text{end}}} P_{L.Con} Q_{M.Con.} dt \tag{13}$$

where $P_{L.Con}$ and $Q_{M.Con}$ are the load pressure and flow through the hydraulic motor M in conventional system.

The energy efficiencies ($\eta_{Con.}$ and $\eta_{Prop.}$) of both systems are derived as

$$\eta_{Con.} = \frac{E_{eff.}}{E_{Con.}}, \quad \eta_{Prop.} = \frac{E_{eff.}}{E_{Prop.}} \tag{14}$$

The energy consumptions, effective energy, and the energy efficiency of the both systems are listed in Table 1. Based on this table, the energy efficiency nearly tripled when the proposed system was used.

Table 1 Energy efficiency

Systems	Supply energy	Effective energy	Energy efficiency
Conventional	10.58 kJ	2.53 kJ	23.91 %
Proposed	3.66 kJ		69.13 %

CONCLUSIONS

In this research, the proposed system and conventional servo motor system for comparison was simulated successfully, this is the first step to prepare for building the experimental system based on the proposed circuit. The simulation results showed that the control response of the both systems are almost the same quality with small error (less than 2%), the proposed system can slightly reduce overshoot.

The most contribution of the proposed system is on energy efficiency. Due to the huge reduction in supply pressure by directly connecting between hydraulic pump and hydraulic motor via only On/Off valves, the energy consumption of the proposed system was diminished drastically. Applying the proposed system, the energy efficiency of the whole system can be raised around three times.

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