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 applications1) and even lower in water hydraulics.

Researching on energy-saving in hydraulic systems become important requirement for researchers. 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.

One of important applications of water hydraulics is designed for hygiene solution in food processing industry, which cannot be solved by bio-oil or oil based hydraulic systems. Compare with pneumatic solutions, water hydraulic solutions have the significant advantages of easy to flush and clean according to the requirements and regulations in the food processing industry, lighter weight of devices due to higher power density, much higher efficiency and saving energy costs2).

Recently, water hydraulics has been applied in meat slicer3), 4). Water hydraulic pushing cylinder system is very important for achieving precise weight and shape of meat slices. The pushing cylinder system used servo valve for controlling the pressure at the end of the piston rod which was equivalent to force control for the piston rod. The supply pressure used in such system was produced by a fixed displacement pump connected to an electric motor. A relief valve was used to adjust the supply pressure level as shown in paper3). The energy loss due to surplus pressure in such kind of pushing cylinder system was enormous because the value of the required load pressure was changed its value depending on the reference force which was a sinusoid signal with the frequency of 1 to 2 Hz while the supply pressure was around the value higher than maximum value of the load pressure.

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Energy Saving for Water Hydraulic Pushing Cylinder in Meat Slicer

Water hydraulics is developed toward widening application especially in food processing systems. Water hydraulic pushing cylinder system used in meat slicer is recently paid attention by both academic and industrial researchers in Japan. Due to very fast working cycle (0.5-1 seconds per cycle corresponding to 120-60 meat slices per minute), in conventional systems, methods to make a supply pressure track a load pressure for saving energy cannot be used. Supply pressure is normally set around a constant value by a relief valve. Thus, the energy loss in such system is huge because of much surplus supply pressure. This research introduces a novel energy-saving water hydraulic pushing cylinder system in meat slicer, which uses a 2/2 flow control valve for controlling the pressure in high-pressure chamber of the cylinder and 3 On/Off valves mainly for adjusting working direction of the piston rod. This system made the supply pressure nearly equal to the load pressure. As a result, the energy consumption reduced much, the reduction is approximately 50% based on the simulation result and control performance is acceptable.

**Keywords:** Water Hydraulics, Energy Saving, Meat Slicer, Pushing Cylinder, Control Performance

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There is so far no way to reduce the supply pressure adapted to load pressure. A water hydraulic variable displacement pump has been not existed in the market, so load sensing method cannot be applied in water hydraulic systems. Another reason is that the load pressure was changed its value very quickly, a load sensing system does not response fast enough.

In this study, a novel water hydraulic pushing cylinder system is introduced. The main purpose is to reduce supply pressure to be nearly equal to load pressure. This leads to the energy loss can be reduced drastically. The simulation results showed that proposed system can raise the energy efficiency around 50% and the control response was acceptable.

The rest of this paper is organized as follows. Section 2 introduces the simulation modelling of conventional system and proposed system. Section 3 compares the force control responses and energy consumptions in both conventional and proposed pushing systems.

2. Nomenclature

| Symbol | Description |
|--------|-------------|
| $A_A, A_B$ | Ring area in chamber A and B |
| $A_{On/Off}$ | Opening area of On/Off valve |
| $b_l, b_r$ | Left and right break-points |
| $C_{lb}, C_{lb}$ | Hydraulic capacitances |
| $C_i$ | Internal leakage coefficient |
| $D_b, D_r$ | Diameters of bore and rod |
| $E_{Conv.}, E_{Prop.}$ | Energy consumptions |
| $F_{fr}, F_{prop}$ | Friction and external forces |
| $m_t$ | Total mass |
| $m_l, m_p$ | Load and piston masses |
| $m_A, m_B, m_l, m_p$ | Masses of fluid in two chambers |
| $P_A, P_B$ | Pressures in two chambers |
| $P_s$ | Supply pressure |
| $P_r$ | Reservoir pressure |
| $Q_A, Q_B$ | Flow rate in two chambers |
| $Q_s$ | Supply flow rate |
| $Q_{On/Off}$ | Flow rate through On/Off valve |
| $x_{sv}, x_{ve}$ | Servo valve spool positions |
| $\alpha_d$ | Discharge coefficient |
| $\rho$ | Mass density of water |
| $\tau_v$ | Time constant of servo valve |

3. System Modelling

To examine the influence of characteristics of the components to the performance of the systems and to get the

![Diagram](image)

**Fig.1** Schematic diagrams of the conventional pushing cylinder system (a) and proposed system (b)
and a conventional servo cylinder system for comparison, mathematical model of important devices in these both systems.

3.1 Proposed System and Conventional System

Figure 1 shows the schematic diagrams of the conventional water hydraulic servo cylinder system (Fig. 1(a)) and proposed water hydraulic pushing cylinder system (Fig. 1(b)). Conventional pushing cylinder system consists of the main elements as follows. A fixed displacement pump connected to an electric motor. High pressure port of the pump is connected to a relief valve for setting the working value of the supply pressure. A servo valve is connected to a symmetric cylinder for controlling the force generated by piston rod to track a given reference. Proposed pushing cylinder system used in meat slicer includes following elements. A fixed displacement pump connected to an electric motor and the high-pressure port of the pump is connected to a relief vale. These devices used in this system are same as in the conventional system. However, the relief valve in the proposed system is only used for safety purpose. Two solenoid On/Off valves are assembled to two paths from the hydraulic pump to cylinder for changing the working direction of the piston rod. These two paths are connected to a fluid control valve and another On/Off valve. The main working direction is from left to right, the movement of the piston rod track to a sinusoid signal with the frequency of 1 Hz corresponding to 60 meat slices per minute. In this working direction, On/Off valves 1 and 3 are open and On/Off valve 2 is closed. The force generated by the piston rod is controlled via a 2-way, 2-position flow control valve C/V. The movement from right to left of the piston rod is only performed after a working circle and the piston rod backs to the initial position. In this direction, On/Off valve 1 and 3 are close, On/Off valve 2 is open, and the movement of the piston rod is controlled via flow control valve C/V.

3.2 Flow Control Valve

In this simulation, the behavior of the 2-way 2-position flow control valve C/V in proposed system (Fig. 1(b)) are considered as the behavior of the 4-way 3-posision servo valve in the conventional system (Fig. 1(a)) after reducing two ports and one position to become two ports, two positions. The mathematical model of the servo valve (as well as the valves C/V) 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:

\[
\tau_v x_v(t) = x_v(t) + k_v u(t)
\]

where \( \tau_v \) and \( k_v \) are the time constant and gain of the servo valve dynamics, respectively.

In addition, to minimize the valve leakage losses which increase with wear and tear of spool and valve body, the spool is designed to have overlap so that for a range of the spool positions \( x_v \), there is no fluid flow. This leads to the dead-zone between the spool position \( x_v \) and the effective spool position \( x_{ve} \) shown in Fig. 2. The dead-zone nonlinearity of the spool position with break-points \( b_l < 0 \) and \( b_r > 0 \) as shown in Fig. 2 can be described by

\[
x_{ve} = \begin{cases} 
    x_v - b_r, & \text{if } x_v \geq b_r \\
    0, & \text{if } b_l < x_v < b_r \\
    x_v - b_l, & \text{if } x_v \leq b_l 
\end{cases}
\]

The leakage of the servo valve is quite small and it is neglected in this research. Hence, the servo valve flows \( Q_A \) and \( Q_B \) in the two chambers can be calculated as follows:

\[
Q_A = \begin{cases} 
    k_{qA} x_{ve} \sqrt{P_s - P_A}, & \text{for } x_{ve} \geq 0 \\
    k_{qA} x_{ve} \sqrt{P_s - P_A}, & \text{for } x_{ve} < 0 
\end{cases}
\]

\[
Q_B = \begin{cases} 
    k_{qB} x_{ve} \sqrt{P_s - P_B}, & \text{for } x_{ve} \geq 0 \\
    k_{qB} x_{ve} \sqrt{P_s - P_B}, & \text{for } x_{ve} < 0 
\end{cases}
\]

where \( P_s, P_A, P_B, P_{ve} \) and \( P_r \) are the pressures of the two chambers of the cylinder, the supply pressure, and the reservoir pressure, respectively; \( k_{qA} \) and \( k_{qB} \) are the flow gain coefficients of the servo valve.
3.3 On/Off Valve

The flow rate $Q_{\text{On/Off}}$ through On/Off valve can be calculated by the orifice equation as follows:

\[
P_{\text{A}} = \frac{1}{C_{hA}} \left( Q_{\text{A}} - A_{\text{A}} x_t - Q_{\text{in}} - Q_{\text{leA}} \right) \quad (7)
\]

\[
P_{\text{B}} = \frac{1}{C_{hB}} \left( -Q_{\text{B}} + A_{\text{B}} x_t - Q_{\text{in}} + Q_{\text{leB}} \right) \quad (8)
\]

where $P_A$ and $P_B$ are the pressures in the two chambers, $C_{hA}$ and $C_{hB}$ are the hydraulic capacitance in the two chambers, $Q_A$ and $Q_B$ are the flow rate to the two chambers, $Q_{\text{in}}$ and $Q_{\text{leA}}$ and $Q_{\text{leB}}$ are the external leakages of the two chamber – in this research, the external leakages were considered to be approximately zero, $Q_{\text{in}}$ is the internal leakage.

\[
Q_{\text{le}} = C_{i} (P_{\text{A}} - P_{\text{B}}) \quad (9)
\]

here, $C_{i}$ is the internal leakage coefficient.

Based on Newton’s second law, the piston motion is described as below equation.

\[
m t = m_{i} + m_{p} + m_{A,\beta} + m_{B,\beta} + F_{fr}(x_{t}) - F_{ext} \quad (10)
\]

where $m_{i}$ is the total mass described in Eq. (11), $F_{fr}$ is the friction force showed detail in Eq. (14), $F_{ext}$ is the external force.

\[
m_{i} = m_{i} + m_{p} + m_{A,\beta} + m_{B,\beta} \quad (11)
\]

where $m_{i}$, $m_{p}$, $m_{A,\beta}$, and $m_{B,\beta}$ are the masses of the load, piston and fluid in the two chambers, which are shown as follows:

\[
m_{A,\beta} = \rho (V_{\text{pl,A}} + V_{A0} + x_{t} A_{A}) \quad (12)
\]

\[
m_{B,\beta} = \rho (V_{\text{pl,B}} + V_{B0} + x_{t} A_{B}) \quad (13)
\]

where $\rho$ is the density of the fluid, $V_{\text{pl,A}}$ and $V_{\text{pl,B}}$ are the volumes of the connected pipe to the chambers A and B, $V_{A0}$ and $V_{B0}$ are the initial volumes of the chambers A and B, respectively.

The friction force equation is based on Stribeck friction as below.

\[
A_{L} = A_{B} = \frac{\pi D_{r}^{2} - \pi D_{b}^{2}}{4} \quad (6)
\]

where $A_{L}$ and $A_{B}$ are the ring areas in two chambers, $D_{r}$ and $D_{b}$ are the diameters of bore and rod, respectively.
\[ F_s(x_t) = \alpha x_t + \text{sign}(x_t) \left[ F_{s0} + F_{s0}e^{c_s} \right] \tag{14} \]

where \( \alpha, F_{s0}, F_{s0}, \) and \( c_s \) are the corresponding viscous, Coulomb, and static friction coefficients.

4. Simulation Results

This research is the first time for introducing the new idea of a novel pushing cylinder system. The aim of the system is to raise the energy efficiency via reducing the energy loss because of huge surplus supply pressure. In this section compares both control performances and energy efficiencies of conventional and proposed systems together.

4.1 Control Response

The main contribution of this research is to find the way for improving the energy efficiency of the water hydraulic pushing cylinder system used in meat slicer. However, the control response is the most important factor for assuring the system work well and it should be considered first.

Figure 4 shows the forces generated by piston rod in conventional system and proposed system. It is easy to realize that the generated forces in both systems can track quite well to the given reference. In the upward phase, the proposed system showed faster tracking. Only around minimum value of the reference (0 N at the times of 1, 2, and 3 seconds), the force of the piston rod in the proposed system could not reduce to be zero, error still existed. However, this error was not big and therefore did not effect to the working of the meat slicer – it was not enough to move the bar of meat because it was much smaller than friction force.

4.2 Energy Efficiency

In this subsection, the energy consumptions of the both conventional and proposed water hydraulic pushing cylinder system are analyzed in detail. Both systems used the same supply response including an electric motor connected to a fixed displacement pump. The relief valve operated with different functions, it was used to set the maximum value for the supply pressure in conventional system (\( P_{s,\text{Conv.}} \)) and used for protecting the proposed system against the supply pressure (\( P_{s,\text{Prop.}} \)) getting high value in very rare case if the On/Off valves 1 and 2 cannot do it function. The both systems had the same supply flowrate; thus, the energy consumptions of these systems only depended on the supply pressures.

Figure 5 shows the load pressure (\( P_L \)), which is derived in Eq. (15), supply pressures in conventional and proposed systems. In the conventional system, the supply pressure was adjusted by the relief valve to change around a constant value and keep bigger than load pressure. It leaded to the gap between the supply pressure in the conventional system and the load pressure was too big. As a result, the energy loss because of the surplus pressure became very big value. This problem is still challenge for conventional water hydraulic pushing cylinder system in meat slicer because the supply pressure requires changing the value very quick (from 60 to 120 cycles per minute). This problem cannot solve by using load sensing method because of the slow response of the set of the electric motor and hydraulic pump. Because of the very small pressure drop over the On/Off valve 1, the supply pressure in the proposed system was only slightly bigger than the load pressure. Therefore, the energy loss due to the surplus pressure in the proposed system was very small, it was approximately equal to zero.

\[ P_L = |P_A - P_B| \tag{15} \]

The energy consumptions of conventional servo motor system (\( E_{\text{Conv.}} \)) and proposed system (\( E_{\text{Prop.}} \)) can be calculated by following equations.
where $P_{s,\text{Conv.}}, P_{s,\text{Prop.}}, Q_{s,\text{Conv.}}, Q_{s,\text{Prop.}}$ are the supply pressures and supply flow rates of conventional and proposed systems, respectively; $\eta_P$ is the efficiency of the hydraulic pump, getting the value of 85% in this research; $t_{\text{start}}$ and $t_{\text{end}}$ are the starting and finishing times of simulation.

Figure 6 shows energy consumptions in conventional and proposed systems. Corresponding to the working time of 3 seconds, the energy consumption in the conventional system was 129.8 J and much reduced in the proposed system which only consumed 65.0 J for 3-second operation. That means by using proposed system, energy efficiency of the water hydraulic pushing system increased by nearly 50%.

5. Conclusions

In this research, the proposed system and conventional system which was used for comparison were simulated successfully. This is the first step to prepare for constructing the experimental system based on the proposed circuit.

The simulation results showed that the control responses in the both systems could track well to the reference, slightly faster in proposed system. The proposed system could not track to the reference with the value of 0 N at the time 0, 1, 2, and 3 second. However, the error was not big and it did not effect to the working property of the pushing system.

By using the proposed system, the supply pressure much reduced. It leaded to the energy loss due to surplus pressure decreased drastically. As a result, the energy efficiency in the proposed system raised nearly 50%.

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