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Energy saving potential of load sensing system with hydro-mechanical pressure compensation and independent metering

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ABSTRACT

This paper reports a systematic analysis of a load sensing system with hydro-mechanical pressure compensation and independent metering. In contrast to a conventional spool valve controlled load sensing system, the proposed load sensing system is characterised by both meter-in and meter-out pressure compensation. Quasi-static behaviour analysis is applied to three distinct load sensing systems: the meter-in pressure compensation system (MIPCS), meter-out pressure compensation system (MOPCS), and pressure compensation load sensing system (PCLSS). The energy usage equation shows that minimising pump supply pressure is the only way to ensure high energy saving efficiency; proper opening modes between the meter-in and meter-out orifices of the MIPCS and MOPCS are also obtained by deducing and analysing appropriate equations. Systems parameters are then kept constant as the pump supply pressure of the three systems are compared by varying the external force. Comparison results show that the pump supply pressure of both the MIPCS and MOPCS are lower than that of PCLSS, and that the optimal metering pressure compensation method is dependent on the working mode. Taken together, the results show that a load sensing system with independent metering offers more significant energy savings than the traditional load sensing system.

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1. Introduction

Electro-hydraulic control systems are commonly used in the mobile machinery because of the high power and force to weight ratio (Shenouda and Book 2005). Normally, proportional directional spool valves are used in the conventional electro-hydraulic control systems to control the desired flow direction and flow rate passing through the valves (Hu and Zhang 2002a). With this kind of valve, the meter-in and meter-out orifices are mechanically linked. The mechanical connection makes the system easy to control, but it also brings in some significant limitations. The proportional directional spool valve produces a number of types of pressure losses when achieving flow rate control performance, so the input pressure of the proportional spool valve has been definitely increased. The good motion control performance of the actuators can be achieved with such a system, but the limitations mainly result in lower efficiency (Tabor 2005a). Furthermore, the sliding spool of the conventional valve is specially designed and manufactured for different applications, so it cannot be interchangeable even if they are exactly the same size (Hu and Zhang 2003).

To overcome these shortcomings, a hydraulic system using four 2/2-valves independent metering was proposed by Jansson and Palmberg (1990). This four 2/2-valves configuration breaks the mechanical linkage and decouples the restriction. There are other available valve configurations for realising independent metering technology, as well, including two 3/3-valves; a combination of two 3/3-valves and one 2/2-valve; or five 2/2-valves, for example, each of which has different control logics and metering modes. With this configurations, the multiple functions including regeneration functionality, float functionality, energy saving functionality, and cavitations prevention functionality can be realised (Eriksson and Palmberg 2011).The independent metering valve configurations are designed as lower cost, and also can be interchanged with expensive servo valves (Liu and Yao 2006). Compared to the conventional electro-hydraulic control system, the most notable characteristic of the independent metering system is that the meter-in and meter-out orifices are decoupled, which reduces input power and thus achieves highly valuable energy savings (Shenouda and Book 2008).

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The independent metering technology exists for a long time, but it has not been applied widely in the actual mobile hydraulic systems on the machinery market. The main reason is that the independent metering system is a multiple input system which has two degrees of the control freedom. So when precise position control is desired in this system, it has been required the complex control algorithms and more additional expensive equipment.

In hydraulic system applications, especially in mobile machinery, the motion of actuators must be consistent with the operator commands, so it is necessary to control the constant flow passing through the control valves at various operating pressures. There currently exist two compensation methods to accomplish this: electro-hydraulic pressure compensation method, and the hydro-mechanical pressure compensation method. If using electro-hydraulic pressure compensation method in the independent metering system, it will be needed more sensors and appropriate control methods. Using five poppet type valves and additional pressure sensors, with incorporation of the pressure controller and adaptive robust controller in a single-rod cylinder hydraulic system, the precise motion and energy saving performance can be achieved as is explained in Yao and DeBoer (2002). The hybrid control algorithm has been applied successfully in a five-valve independent metering system for position control (Hu and Zhang 2002b). So the electro-hydraulic pressure compensation method can achieve the desired independent metering system performance, but is rarely used in real-world machines due to the method's high level of complexity in practice compared to what is possible theoretically and further, because the additional measurement points necessary drive down the overall system stability.

As opposed to the electro-hydraulic pressure compensation method, hydro-mechanical pressure compensation solutions react directly and rapidly on disturbance variables in the system (Sitte and Weber 2013). The flow regulating valves with hydro-mechanical pressure compensation method are commonly used in the conventional load sensing system to improve the controllability characteristics (Krns *et al.* 1996). A flow regulating valve is composed of a regulating valve and a hydro-mechanical pressure compensator valve. The regulating valve determines flow rate; the hydro-mechanical pressure compensator valve works to keep the pressure difference across the regulating valve constant. The mechanical connection of the meter-in and meter-out orifices in the flow regulating valve control system results in extra metering losses under some working conditions (Wang and Wang 2014). The pressure drop across the regulating valve is normally as a small constant value, most of the inlet-outlet pressure difference is consumed at the pressure compensator valve. While the inlet-out pressure difference across the regulating valve is reaching a high value, the cavitations phenomenon is easily appearing in the load sensing system (Suzuki and Urata 2008). However, Compared to the regulating valve, independent metering valve configurations can provide cavitations prevention functionality (Tabor 2005b).

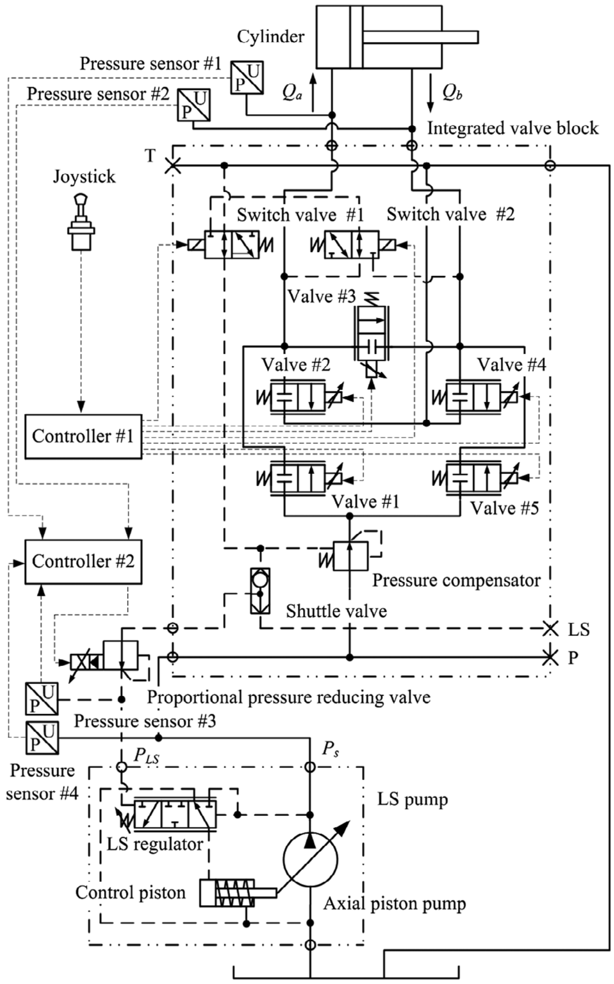

Figure 1. The schematic of load sensing system with hydro-mechanical pressure compensation and independent metering.

The above altogether suggests that designing a load sensing system that uses hydro-mechanical pressure compensation and independent metering technology is favourable compared to other available approaches. Compared to the conventional spool valve controlled load sensing system, for example, a load sensing system with independent metering has two pressure compensated methods: meter-in and meter-out, which of these is the optimal method for energy savings remains unclear, however.

Using different hydraulic circuits and appropriate components can realise the load sensing systems with both the meter-in and meter-out pressure compensation methods. Take the meter-in pressure compensation method for example: the schematic of a load sensing system with hydro-mechanical pressure compensation and independent metering is shown in Figure 1. The load sensing system is composed of a load sensing pump (LS pump), a proportional pressure reducing valve, a valve integrated valve block, one cylinder, two controllers, four pressure sensors, and a joystick. If more actuators are required to control in this system, it can be added the integrated valve blocks and joysticks. The LS pump is composed of a load sensing regulator (LS regulator), a control piston, and an axial piston pump, which is the common pump in the load sensing system of mobile machinery.

Compared to the conventional load sensing system, the most notable difference is that a proportional pressure reducing valve is added between the integrated valve blocks and LS pump. The integrated valve block is comprised of one shuttle valve, one pressure compensator, five 2/2-valves, and two switch valves. The controller #1 is to used to adjust the flow through the cylinder by controlling the five 2/2-valves and two switch valves.

If the system is in the extension working mode, Valves #1 and #4 are actuated to separately control the meter-in and meter-out orifices. Similarly, Valves #2 and #5 are actuated in the retraction working mode. Various combinations of the five 2/2-valves can realise regeneration function and multiple float functions. It is possible to remove the additional flow sensors and pressure sensors, however, which reduces the cost of the system substantially and makes the control algorithm for the five 2/2-valves much simpler; in effect, the whole independent metering system becomes more stable, more reliable, and cheaper if the number of pressure measurement points is reduced.

Now take the extension working mode for example: its working principle is that the flow is supplied at high pressure P_S from the LS pump through the pressure compensator and Valve #1 into head chamber of the cylinder, causing the piston to extend. This piston motion then forces the flow out of the rod chamber through Valve #4 at low pressure to the tank. The two switch valves are used to leading the working pressure into the control oil port of the pressure compensator and the comparing port of the shuttle valve.

The combination of the LS pump and the proportional pressure reducing valve allows the supply pressure from the LS pump to be controlled appropriately. Controller #2 is used to regulate the out port pressure of the proportional pressure reducing valve by controlling the supply pressure as close as possible to the minimum require supply pressure. While controlling the supply pressure, the signals from Pressure sensors #3 and #4 are closed loop control signals. The minimum required supply pressure can be calculated by system parameters, flow coefficient of the actuating valve, and the pressure from sensors #1 and #2. Controller #2 can indeed control the supply pressure based on the load pressure, but the control strategy of the supply pressure from the LS pump is so complicated that requires a separate research project in the future.

The control valve block in the conventional pressure load sensing system can be substituted easily by the proposed integrated valve block. In addition, the meter-out pressure compensation method can also be used in the integrated valve block.

This study focuses on the quasi-static behaviour of the systems and ignores the dynamic effects. The results certainly do contain some errors compared to the actual system, in other words, these results do not fully reflect the variations of the parameters in the real system.

The remainder of this paper is organised as follows. Section 2 introduces the main layout and the working principle of the independent metering system. Section 3 presents the two pressure compensation methods applied in the independent metering system. The quasi-static behaviour equations of the independent metering system with two pressure compensation methods and the pressure compensation load sensing system (PCLSS) are deduced in detail in Section 4, as well as the manner in which the proposed system minimises energy usage. Section 5, system parameters are kept constant and the energy saving performance of the meter-in pressure compensation system (MIPCS), meter-out pressure compensation system (MOPCS) and pressure compensated load sensing system are presented for the sake of comparison. Section 6 concludes the paper by discussing the applicability of a load sensing system with hydro-mechanical pressure compensation and independent metering.

2. System layouts and working principle

Many previous researchers have explored different hydraulic system layouts for realising independent metering systems, these layouts can roughly divided into two categories, as depicted in Figure 2. Layout design involves adding valves or changing the connections based on these two types of layouts. The four 2/2-valves control system and two 3/3-valves controlled system layouts are shown in Figure 2(a) and (b), both of them can realise extension and retraction working modes by different control logics. Compared to the two 3/3-valves layout, the four 2/2-valves layout has more functions such as different float characteristics and regeneration function.

Though the two main layouts have different kinds of valves and various connection formations, they share the characteristic feature of individual control of meter-in

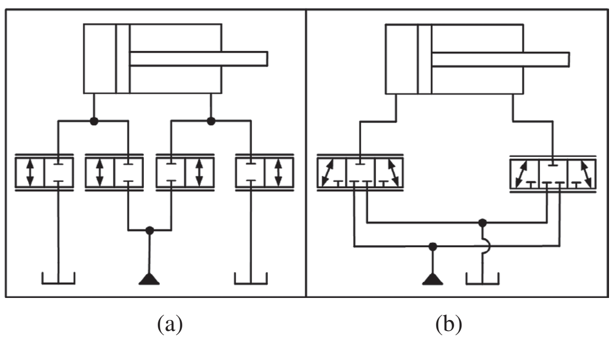

Figure 2. Two main kinds of hydraulic system layouts.
(a) Four 2/2-valves, (b) two 3/3-valves.

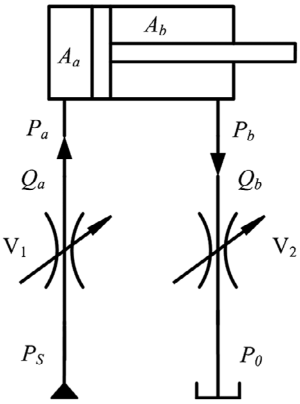

Figure 3. The schematic of independent metering system.

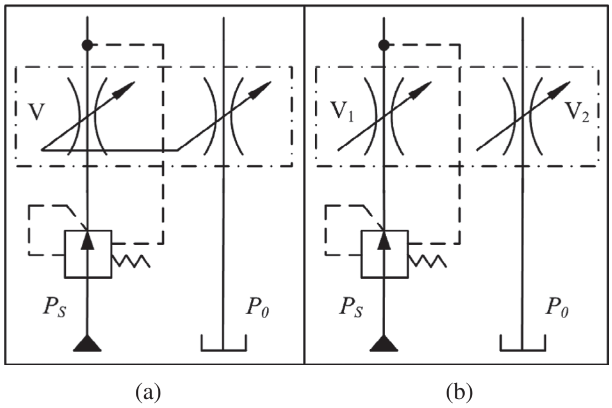

Figure 4. The pressure compensation principle.
(a) Pressure compensated load sensing system, (b) independent metering system with pressure compensation.

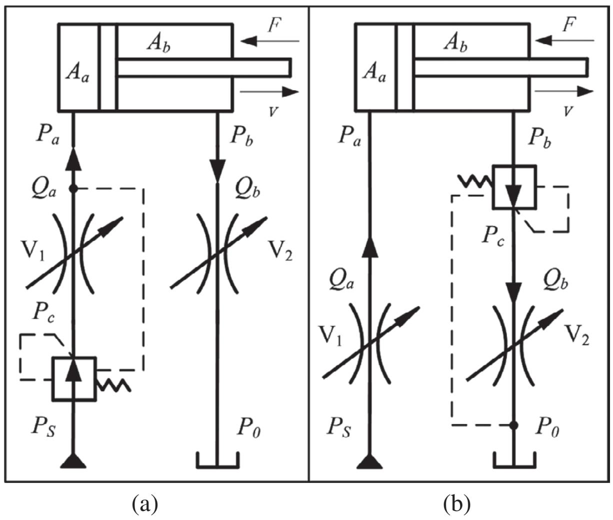

Figure 5. The two pressure compensation ways.
(a) The meter-in pressure compensation method, (b) the meter-out pressure compensation method.

and meter-out orifices. So both the layouts can be simplified as the same schematic of the independent metering system as is shown in the Figure 3. It is clear that the piston rod of the cylinder can be extended in this situation; this is called the extension working mode. The principle of the extension working mode is that flow Q_a at a supply pressure P_S travels through Valve V_1 to the head chamber A_a of the cylinder and is forced out of the rod chamber A_b of the cylinder through Valve V_2 to the tank at low pressure P_0. The other mode, retraction working mode, differs from extension working mode is that the flow is in the rod chamber A_b of the cylinder and out of head chamber A_b of the cylinder due to different valve control logic.

3. Independent pressure compensation systems

In order to obtain the constant flow necessary for optimal actuator function with various loads, pressure compensators are typically employed in hydraulic systems, and especially in pressure compensated load sensing system. In an independent metering system, the meter-in and meter-out orifices are regulated independently, so the pressure compensation method differs inherently from that appropriate for a pressure compensated load sensing system.

Figure 4(a) shows the pressure compensation principle of the pressure compensated load sensing system. It is clear that the meter-in and meter-out orifices are regulated simultaneously due to the mechanical connection of metering edges by a spool valve V, and the pressure compensator is kept the pressure difference across the valve V as constant, as a result that can be obtained the desired flow with a certain valve opening ratio.

Figure 4(b) shows the pressure compensation principle of independent metering system. It is obvious that the desired flow can be obtained by regulating the meter-in orifice with constant pressure difference across valve V_1. Simultaneously, the meter-out orifice of the valve V_2 can be regulated for the optimal system performance. It can be deduced that the pressure compensated load sensing system is a typical type of the independent metering system.

Unlike in the conventional pressure compensated load sensing system, where only the meter-in pressure compensation method is applicable, there are two pressure compensation methods appropriate for the independent metering system: meter-in pressure compensation method and meter-out pressure compensation method as is shown in the Figure 5(a) and (b). The pressure compensation method requires a pressure compensator and a regulating valve to regulate the flow, whether by meter-in or meter-out pressure compensation method.

4. Energy saving analysis

4.1. Flow regulating valve performance analysis

In the independent pressure compensation system, whether it uses meter-in or meter-out pressure compensation, the pressure compensator and regulating valve provide flow regulation via the working principle shown in Figure 6. Accordingly, it is necessary to analyse the

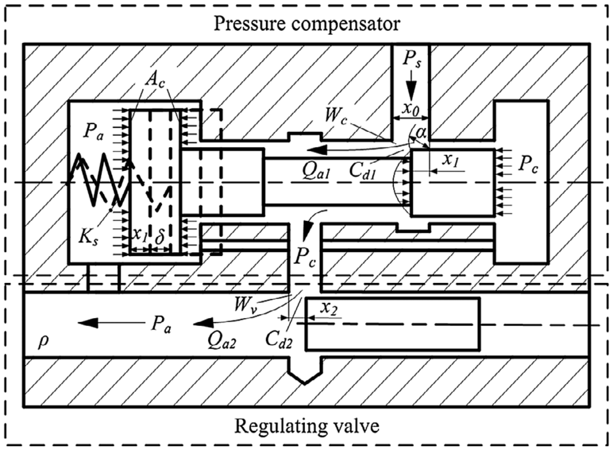

Figure 6. The working principle of the flow regulating valve.

characteristics of the flow regulating valve before analysing the energy saving performance of the independent pressure compensation system.

The flow regulating valve in the MIPCS performs similarly to that in the MOPCS. The fluid compressibility and the dynamic effects of the hydraulic system components are ignored in the following analysis.

As is shown in the Figure 6, the flows in the pressure compensator can be expressed as:

$$Q_{a1} = C_{d1} W_c (x_0 - x_1) \sqrt{\frac{2(P_s - P_c)}{\rho}} \tag{1}$$

where C_{d1} is the flow coefficient of the pressure compensator, W_c is the area gradient of the pressure compensator, x_0 is the spool pre-displacement of the pressure compensator, x_1 is the spool displacement of the pressure compensator, P_s is the supply pressure from pump, P_c is inlet pressure of the regulating valve, and ρ is the density of the hydraulic fluid.

The flows in the regulating valve can be expressed as:

$$Q_{a2} = C_{d2} W_v x_2 \sqrt{\frac{2(P_c - P_a)}{\rho}} \tag{2}$$

where C_{d2} is the flow coefficient of the regulating valve, W_v is the area gradient of the regulating valve, x_2 is the spool displacement of the regulating valve, and P_a is the outlet pressure of the meter-in valve.

The balance force of the spool in the pressure compensator can be expressed as follows:

$$\begin{aligned} &P_c A_c - P_a A_c - K_s(\delta + x_1) \\ &- 2C_{d1} W_c x_1 \cos\alpha (P_s - P_c) = 0 \end{aligned} \tag{3}$$

where A_c is the Effective area of the pressure compensator, K_s is the spring stiffness coefficient, δ is the spring pre-displacement, and α is the jet angle of the pressure compensator.

The pressure difference of the regulating valve can be obtained from the Equation (3):

$$P_c - P_a = \frac{1}{A_c}\left[K_s(\delta + x_1) + 2C_{d1} W_c x_1 \cos\alpha (P_s - P_c)\right] \tag{4}$$

The Equation (4) shows the influence factors of the pressure difference is composed of the spring force and the transient flow force, they are respectively influenced on the quasi-static and dynamic behaviour of the regulating valve.

Ignore the effect of the transient flow force, the Equation (4) can be simplified as:

$$P_c - P_a = \frac{K_s}{A_c}(\delta + x_1) \tag{5}$$

In order to keep the pressure difference ΔP as a constant value, the pressure compensator spool displacement x_1 must be less than the sum of the spring pre-displacement δ, so the pressure compensator spool displacement x_1 also can be considered as a constant value as its tiny change.

The flows in the pressure compensator Q_{a1} equals the flows in the regulating valve Q_{a2}:

$$Q_{a1} = Q_{a2} \tag{6}$$

Take the Equations (1), (2), (5) and (6) into derivation, the supply pressure P_s can be obtained:

$$P_s = \frac{K_s}{A_c}(\delta + x_1)\left[\frac{C_{d2}^2 W_v^2 x_2^2}{C_{d1}^2 W_c^2 (x_0 - x_1)^2} + 1\right] + P_a \tag{7}$$

The Equation (7) shows the relationship between the supply pressure P_s and the regulating valve outlet pressure P_a. In this equation, the regulating valve spool displacement x_2 is a variable value, and other parameters can be considered as constant values. The Equation (7) can be used in both of the meter-in and meter-out pressure compensation systems.

As is shown in the Figure 5(a), the spool displacement x_2 of regulating valves in the MIPCS can be obtained from the following equations:

$$x_2 = x_{\max} x_{\text{in}} \tag{8}$$

where x_{max} is the maximum spool displacement of the regulating valve, and x_{in} is the meter-in valve opening ratio.

Similarly, as is shown in the Figure 5(b), the spool displacement x_2 of regulating valves in the meter-out pressure compensated system can be obtained from the following equations:

$$x_2 = x_{max} x_{out} \tag{9}$$

where x_{out} is the meter-out valve opening ratio.

Assume that the intrinsic parameters of the valves in the meter-in and meter-out pressure compensation systems are same. So define the substituted parameters K_c and K_v as follows:

$$K_{\Delta p} = \frac{K_s}{A_c}(\delta + x_1) \tag{10}$$

$$K_v = \frac{C_{d2}^2 W_v^2 x_{max}^2}{C_{d1}^2 W_c^2 (x_0 - x_1)^2} \tag{11}$$

The relationship between the inlet and outlet pressure of the flow regulating valves in the meter-in compensation system can be obtained:

$$P_s = K_{\Delta p} K_v x_{in}^2 + K_{\Delta p} + P_a \tag{12}$$

In the MOPCS, the Equation (5) is as follows:

$$P_c - P_0 = \frac{K_s}{A_c}(\delta + x_1) \tag{13}$$

So the relationship between the pressure P_b and P_0 can be obtained:

$$P_b = K_{\Delta p} K_v x_{out}^2 + K_{\Delta p} + P_0 \tag{14}$$

4.2. Energy saving method

The energy usage of the hydraulic system can be calculated as follows (Yao 2009):

$$E = \int_{t_0}^{t_1} P_s(\tau) Q_s(\tau) \mathrm{d}\tau \tag{15}$$

where t_0 and t_1 are starting and the ending time of the task, and P_s and Q_s represent the supply pressure and the flow rate from the hydraulic pump.

From the Equation (15), there exist two ways to reduce the energy usage:

(a) reduce the pump supply pressure $P_{s(t)}$;
(b) reduce the flow rate $Q_{s(t)}$ from a pump.

If ignoring the fluid compressibility and oil leakage in the hydraulic system, the supply flow rate Q_s depends on the motion of actuator. So reducing the energy usage for energy saving have only path is that reduce the pump supply pressure P_s, whether it is in the meter-in or meter-out pressure compensation system.

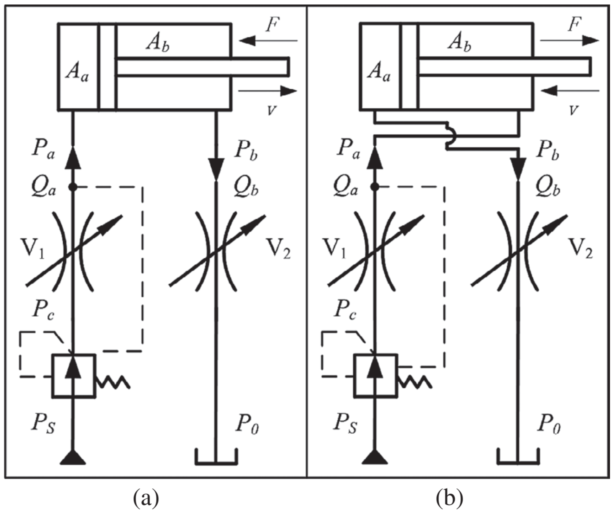

Figure 7. The meter-in pressure compensation system with two working modes.
(a) The extension working mode, (b) the retraction working mode.

4.3. The MIPCS performance analysis

In this section, the fluid compressibility and the dynamic effects are also ignored. The meter-in pressure compensated system with extension and retraction working modes can be seen in Figure 7. As is shown in the Figure 7, the supply flow rate Q_s equals the meter-in flow rate Q_a, so the method of realising energy saving is aimed at reducing the supply pressure P_s as much as possible.

Take the extension working mode for example, and it is similar with the retraction working mode. As is shown in the Figure 7(a), flows in regulating valve V_1 and V_2 can be expressed:

$$Q_a = C_{d2} W_v x_{max} x_{in} \sqrt{\frac{2(P_c - P_a)}{\rho}} \tag{16}$$

$$Q_b = C_{d2} W_v x_{max} x_{out} \sqrt{\frac{2(P_b - P_0)}{\rho}} \tag{17}$$

Flows in regulating valve V_1 and V_2 with the extension mode can be characterised by the following equations:

$$Q_a = A_a v \tag{18}$$

$$Q_b = A_b v \tag{19}$$

Take the Equations (16), (17), (18) and (19) into derivation, the results can be obtained:

$$\frac{Q_a}{Q_b} = \frac{x_{in}\sqrt{P_c - P_a}}{x_{out}\sqrt{P_b - P_0}} = \frac{A_a}{A_b} \tag{20}$$

By defining $\mu = \frac{x_{in}}{x_{out}}$, $R = \frac{A_a}{A_b}$ and the P_0 is the pressure of tank can be assumed as 0. Square the Equation (20) and rearranging the following expressions, the results are obtained:

$$P_b = \frac{\mu^2}{R^2}(P_c - P_a) \tag{21}$$

From the Equations (5) and (10), the Equation (21) can be simplified as:

$$P_b = \frac{\mu^2}{R^2}K_{\Delta p} \tag{22}$$

The hydraulic force can be expressed as follows:

$$F = P_a A_a - P_b A_b \tag{23}$$

Take the Equations (12), (22) and (23) into derivation, the results can be obtained:

$$P_a = \frac{\mu^2}{R^3}K_{\Delta p} + \frac{F}{A_a} \tag{24}$$

$$P_s = K_{\Delta p}\left(K_v x_{\text{in}}^2 + \frac{\mu^2}{R^3} + 1\right) + \frac{F}{A_a} \tag{25}$$

Use the same method, the results of the retraction working mode can be obtained as follows:

$$P_a = K_{\Delta p}R^3\mu^2 + \frac{F}{A_b} \tag{26}$$

$$P_b = K_{\Delta p}R^2\mu^2 \tag{27}$$

$$P_s = K_{\Delta p}\left(K_v x_{\text{in}}^2 + R^3\mu^2 + 1\right) + \frac{F}{A_b} \tag{28}$$

The supply pressure P_s in the MOPCS with extension and retraction working modes can be calculated by the Equations (25) and (28). From the two equations, it is clear that the parameters $K_{\Delta p}$, K_v, R, F, A_a and A_b are constant when the parameters of the actuator and the external force F are fixed. So the supply pressure P_s is related to the meter-in valve opening ratio x_{in}, meter-out opening ratio x_{out} and their ratio μ. The Equations (16) and (18) show that the meter-in valve opening ratio x_{in} is depended on the velocity of the actuator, so the way of reducing the supply pressure P_s is that decreasing the ratio μ. In this MOPCS, the ratio μ is variable as the meter-in and meter-out valve can be independently regulated. So the way of decreasing the ratio μ for realising energy saving is increasing the meter-out opening ratio x_{out} as much as possible.

4.4. The MOPCS performance analysis

The MOPCS with extension and retraction working modes as is shown in the Figure 8. The performance analysis method of the MOPCS is similar with the MIPCS.

As is shown in the Figure 8(a), flows in the regulating valve V_1 and V_2 can be expressed:

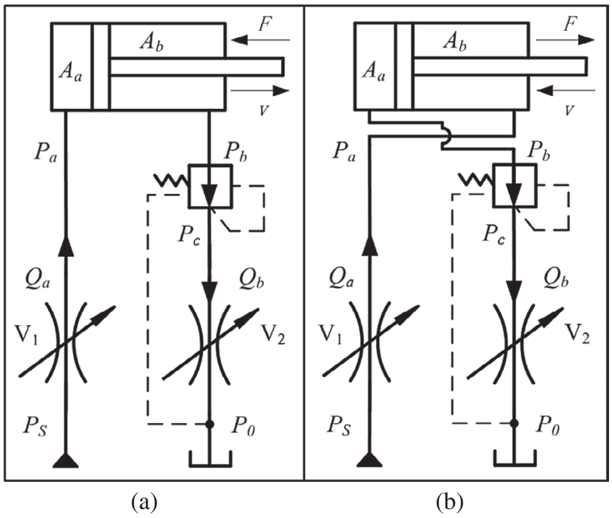

Figure 8. The meter-out pressure compensation system with two working modes.
(a) The extension working mode, (b) the retraction working mode.

$$Q_a = C_{d2}W_v x_{\max} x_{\text{in}}\sqrt{\frac{2(P_s - P_a)}{\rho}} \tag{29}$$

$$Q_b = C_{d2}W_v x_{\max} x_{\text{out}}\sqrt{\frac{2(P_c - P_0)}{\rho}} \tag{30}$$

The meter-in and meter-out flows can also be calculated by the Equations (18) and (19).

By defining the same parameters μ, R and P_0 as in the meter-in compensated system, take the Equations (13), (14), (18), (19), (29) and (30) into derivation, the results can be obtained:

$$P_s = \frac{R^2}{\mu^2}K_{\Delta p} + P_a \tag{31}$$

$$P_b = K_{\Delta p}K_v x_{\text{out}}^2 + K_{\Delta p} \tag{32}$$

The relationship between the pressure P_a and P_b also can be deduced by the Equation (13), so the pressure P_a and P_s can be calculated by the following equations:

$$P_a = \frac{K_{\Delta p}}{R}\left(K_v x_{\text{out}}^2 + 1\right) + \frac{F}{A_a} \tag{33}$$

$$P_s = \frac{K_{\Delta p}}{R}\left(K_v x_{\text{out}}^2 + \frac{R^3}{\mu^2} + 1\right) + \frac{F}{A_a} \tag{34}$$

Use the same method, the results of the retraction working mode also can be obtained as follows:

$$P_a = K_{\Delta p}R\left(K_v x_{\text{out}}^2 + 1\right) + \frac{F}{A_b} \tag{35}$$

$$P_s = K_{\Delta p}R\left(K_v x_{\text{out}}^2 + \frac{1}{R^3\mu^2} + 1\right) + \frac{F}{A_b} \tag{36}$$

The calculation of the pressure P_b in the retraction working mode is same as the extension working mode as is shown in the Equation (32).

In the meter-out pressure compensated system with the extension and retraction working modes, the Equations (34) and (36) can be calculated the supply pressure P_s. As mentioned above, the parameters $K_{\Delta p}$, K_v, R, F, A_a and A_b are constant. So the way of reducing the supply pressure P_s is increasing the ratio μ. Increasing the meter-in opening ratio x_{in} as much as possible can be obtained a higher ratio μ for the purpose of reducing the supply pressure for realising energy saving.

4.5. Pressure compensated load sensing system performance analysis

As is mentioned above, the pressure compensated load sensing system is a typical type of the independent metering system with the meter-in pressure compensation method. So the Equations (26), (27) and (28) also can be used in the pressure compensated load sensing system for calculations of the pressure P_a, P_b and P_s. Due to the mechanical connection of metering edges, the ratio μ is a constant value.

In the pressure compensated load sensing system, the regulating valve is symmetrical or asymmetrical, which means the maximum spool displacements of the meter-in and meter-out are same or different. In the most applications of the pressure compensated load sensing system, one of the characteristics is that the valve opening ratio μ is close to the area ratio R.

So suppose the valve opening ratio μ equals the area ratio R, the equations of the pressure P_a, P_b and P_s in the extension working mode can be obtained:

$$P_a = \frac{1}{R}K_{\Delta p} + \frac{F}{A_a} \tag{37}$$

$$P_b = K_{\Delta p} \tag{38}$$

$$P_s = K_{\Delta p}\left(K_v x_{\text{in}}^2 + \frac{1}{R} + 1\right) + \frac{F}{A_a} \tag{39}$$

The equations of the pressure P_a, P_b and P_s in the retraction working mode can also be obtained as follows:

$$P_a = K_{\Delta p}R + \frac{F}{A_b} \tag{40}$$

$$P_b = K_{\Delta p} \tag{41}$$

$$P_s = K_{\Delta p}\left(K_v x_{\text{in}}^2 + R + 1\right) + \frac{F}{A_b} \tag{42}$$

The Equations (37)–(42) shows that the pressure P_a, P_b and P_s are constant values in the pressure compensated load sensing system with assured valve control signals and fixed external force, whether it is in the extension working mode or retraction working mode.

Table 1. The constant parameters of the three systems.

Var.	Value	Units
A_a	0.0079	m^2
A_c	0.0001	m^2
K_s	1150	N/m
C_{d1}	0.7	–
C_{d2}	0.7	–
W_c	0.0023	m
W_v	0.0025	m
x_0	0.025	m
x_{max}	0.0075	m
δ	0.08	m
ρ	850	kg/m^3

5. Comparison results of energy analysis

5.1. Parameters setting

So in order to secure the most effective possible pressure compensation method for energy savings, it is necessary to carefully compare the performance of MIPCS, MOPCS, and pressure compensated load sensing system in the extension working mode and retraction working mode. The parameters are setting as follows:

(1) As listed in Table 1, all parameters of the three systems were standardised.
(2) The external force F and head chamber area A_a were kept constant, and the rod chamber area A_b was calculated according to different area ratios R.
(3) In the MIPCS, the meter-in valve opening ratio x_{in} was increased proportionally from 0 to 1 and the meter-out valve opening ratio x_{out} was kept constant at 1 to secure the lowest possible supply pressure P_s, as mentioned above.
(4) Conversely, for the MOPCS, the meter-out valve opening ratio x_{out} was increased proportionally from 0 to 1 and the meter-in valve opening ratio x_{in} was kept constant at 1.
(5) In the pressure compensated load sensing system, the meter-in valve opening ratio x_{in} and the meter-out valve opening ratio x_{out} was increased simultaneously, and the valve opening ratio μ equals the area ratio R.

In the load sensing system where joystick signals correspond to flow demands, actuator velocity can be easily controlled via valve openings with various external forces. During actual operation, actuator velocity is only related to valve opening ratio regardless of changes in external load forces, so selecting two typical external forces for the three systems was to ensure accurate calculations. The two typical external forces are the constant external force and the sine curve external force; adopting the constant external force best distinguishes the variations in supply pressure P_s will bring out the obvious comparison results for distinguishing the variations of the supply pressure P_s among the three systems, while the sine curve external force best represents actual system operation. By examining both typical external

Table 2. The variable parameters of the three systems.

Var.	MIPCS	MOPCS	PCLSS	Units
F	40,000	40,000	40,000	N
	Sine curve	Sine curve	Sine curve	N
R	1, 2 and 4	1, 2 and 4	1, 2 and 4	–
	0–5	0–5	0–5	–
x_{in}	0–1	1	$R \cdot x_{out}$	–
x_{out}	1	0–1	0–1	–

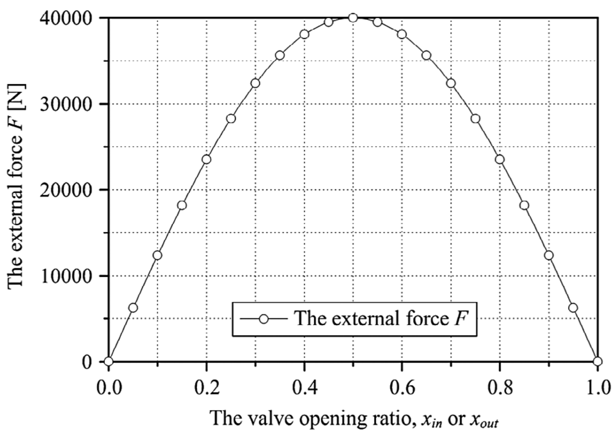

Figure 9. The variations of the external force F.

forces, the variation trends in supply pressure P_s can be fairly well elucidated.

The variable parameters can be seen in the Table 2. The setting of the external force F varied as sine curve as is shown in the Figure 9. The MIPCS, MOPCS, and pressure compensated load sense system are simplified as MIPCS, MOPCS, and PCLSS.

5.2. Comparisons with the same output velocity

In the actual hydraulic system, the actuator (whether the cylinder or motor) dimension is depending on the actual needs of the mechanical system. There are many types of actuators available; but the area ratio R is commonly from 0 to 5. In this study, the ratio R was set to 1, 2, and 4 for analysis.

Due to the existence of the pressure compensators in the MIPCS, MOPCS, and PCLSS, actuator velocity is proportional to the valve opening ratio under various external forces. Actuator velocity is dependent on the flow through the main control valve according to Equation (18). Additionally, the external force is not related to actuator velocity according to Equations (25), (28), (34), (36), (39), and (42).

The minimum required pressure can be obtained according to the constant parameters and external force by changing the valve opening ratios as listed in Table 2. The desired actuator velocity was set as the abscissa to describe the minimum required pressure in the figures below.

In the following analysis, supply pressure was regarded as a special parameter related to the energy saving characteristics; there is no separate discussion

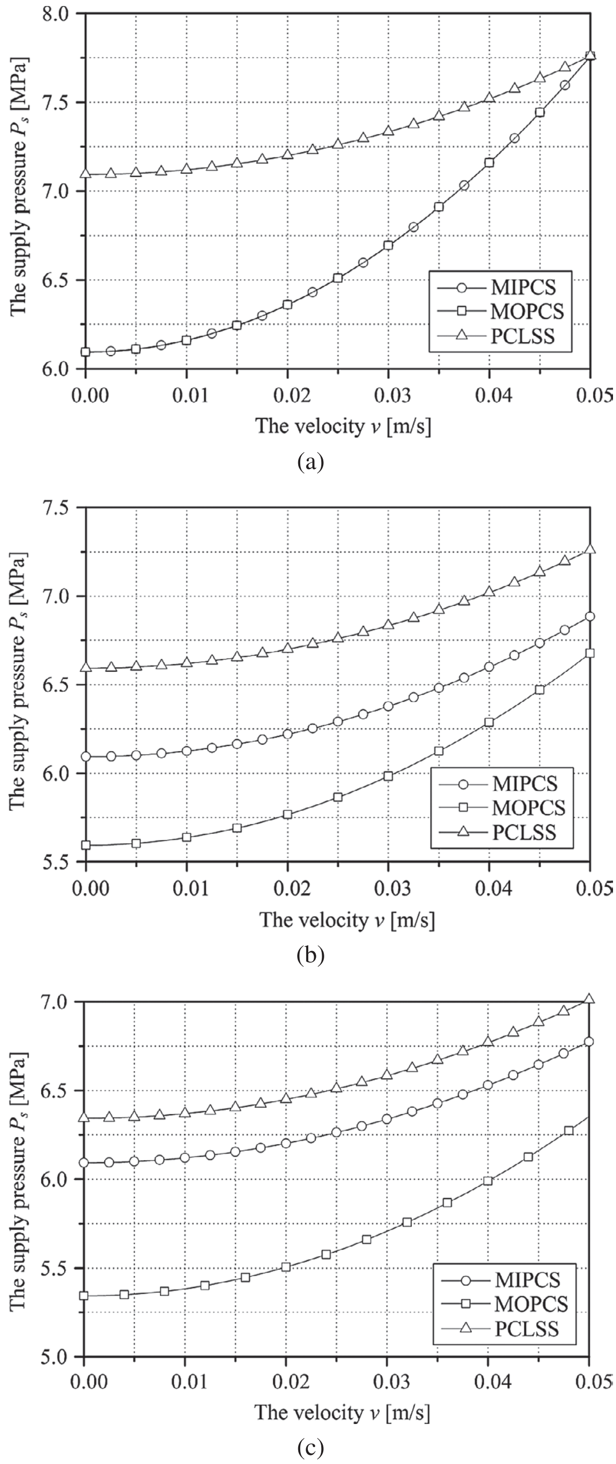

Figure 10. The comparison results of the extension working mode with the constant external force.
(a) The area ratio R equals 1, (b) the area ratio R equals 2, (c) the area ratio R equals 4.

here on how the supply pressure the system can be controlled, but the control method of the supply pressure is depicted in Figure 1 (and a brief description provided above in Section 1).

Many parameters, (e.g. Q_a, Q_b, P_a, P_b), but this primary focus of this study was energy saving performance, so the parameter P_s was given priority as it reflects energy savings according to the Equation (15). For

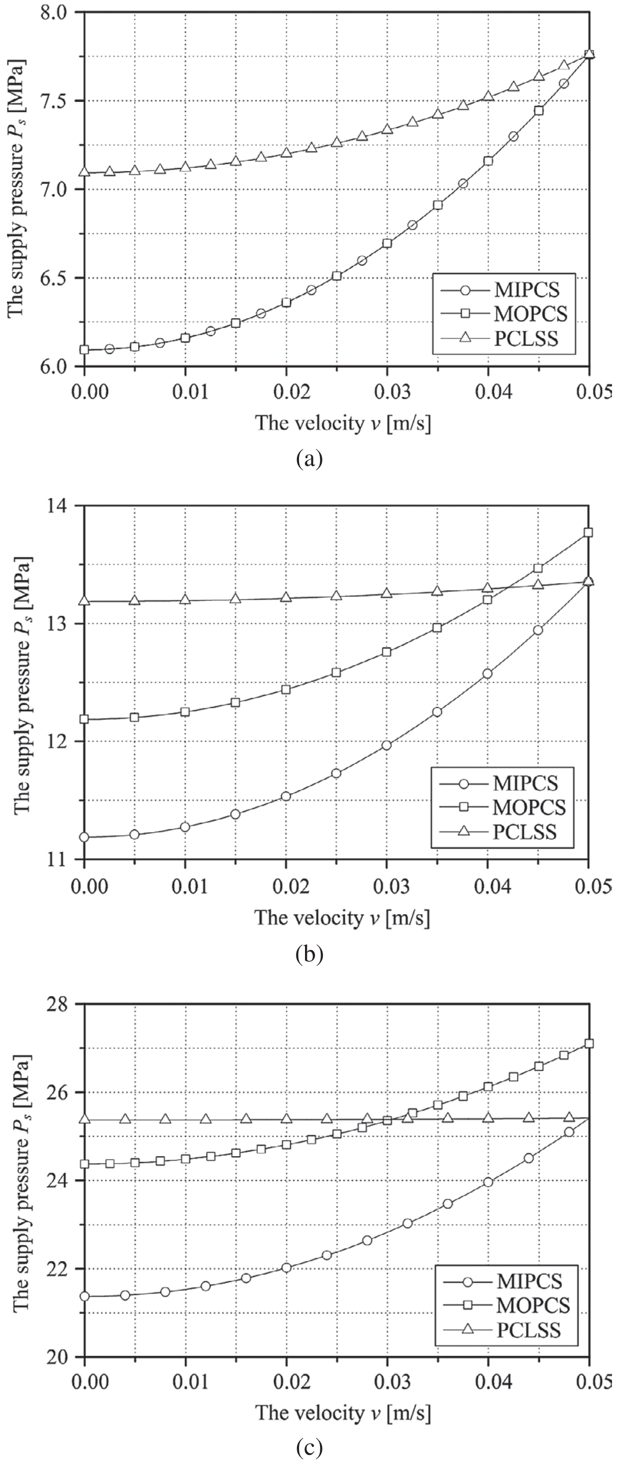

Figure 11. The comparison results of the retraction working mode with the constant external force.
(a) The area ratio R equals 1, (b) the area ratio R equals 2, (c) the area ratio R equals 4.

these calculations, external force F was kept constant at 40,000 N and the sine curve value varied. The comparison results with extension and retraction working modes can be seen in the Figures 10–13.

The Figures 10 and 11 reflects the constant external force F. The abscissas in the Figures 10 and 11 represent the variations of the output velocity v and the ordinates represent the variations of the supply pressure P_s. The

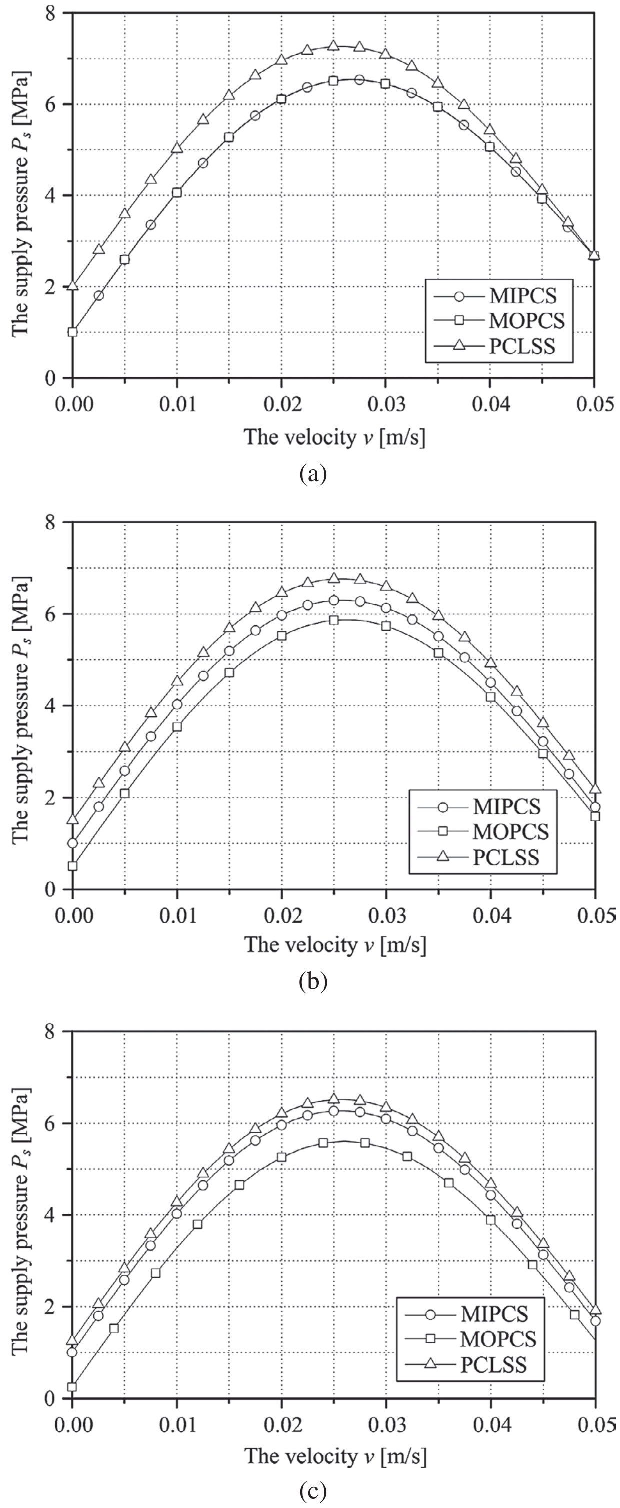

Figure 12. The comparison results of the extension working mode with the variable external force.
(a) The area ratio R equals 1, (b) the area ratio R equals 2, (c) the area ratio R equals 4.

external load force F and the output velocity v are same in this comparisons.

The Figures 12 and 13 reflects the variable external force F of the sine curve.

The Figures 10 and 12 show that in the extension working mode (whether the external force F is constant or variable), the MIPCS and MOPCS have lower supply

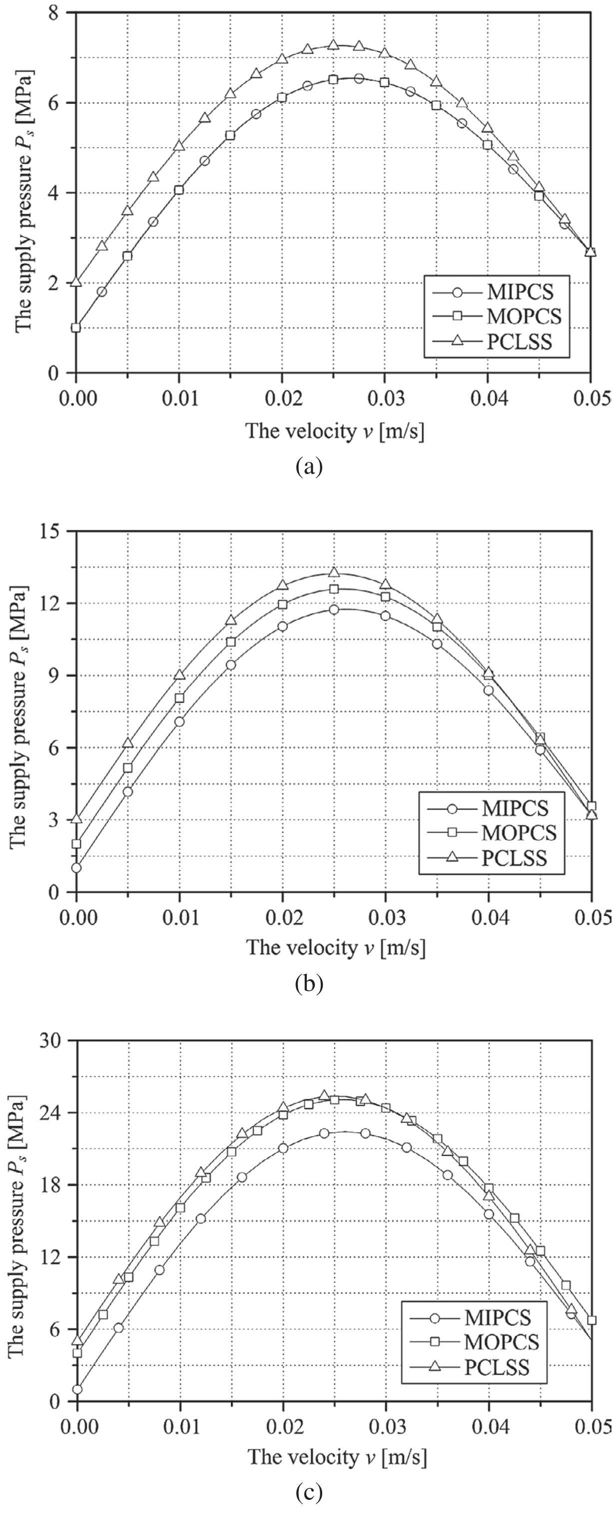

Figure 13. The comparison results of the retraction working mode with the variable external force.
(a) The area ratio R equals 1, (b) the area ratio R equals 2, (c) the area ratio R equals 4.

pressure P_s than the PCLSS. Furthermore, MOPCS have the lowest supply pressure P_s of the three systems. So the energy saving performance of the MOPCS with the extension working mode is the best system of the three systems.

The Figures 11 and 13 show that in the retraction working mode (whether the external force F is constant or variable), the MIPCS and MOPCS have lower supply pressure P_s than the PCLSS when the ratio R was set to 1 and 2. When the ratio R was set to 4, the PCLSS is lower than the MOPCS while the output velocity increased a certain value. Furthermore, MIPCS also have the lowest supply pressure P_s of the three systems. So the MIPCS with the extension working mode is the best system of the three systems.

From the Figures 10(a), 11(a), 12(a) and 13(a), it is clear that the supply pressure P_s of the three systems have the same variation trends when the area ratio R was set to 1.

5.3. Comparison with the same area ratio

In the actual design of the hydraulic system, the area ratio R is less than 5 and generally equals from 1 to 3. So in order to compare the characteristics of the three systems with different area ratios R, calculating the average supply pressure P_s with the extension and retraction working modes as is shown in Figures 14 and 15.

The Figure 14 shows that the average supply pressure P_s of the three systems are decreased when the area ratio

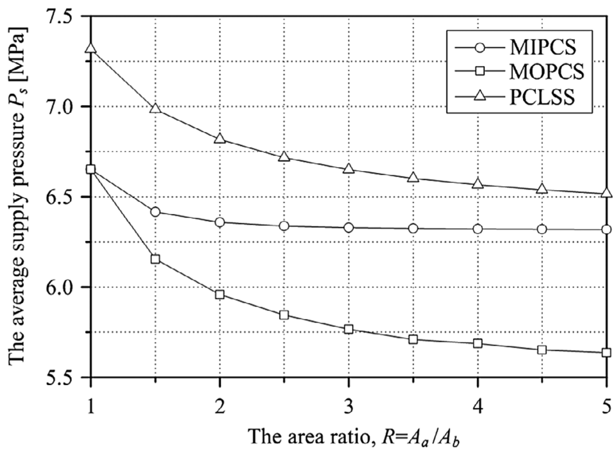

Figure 14. The average supply pressure of the extension working mode.

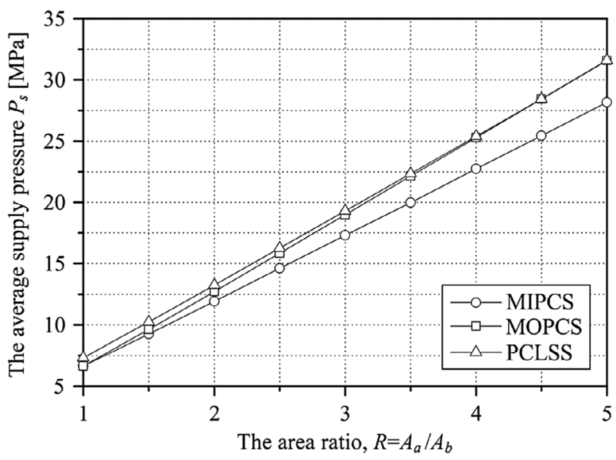

Figure 15. The average supply pressure of the retraction working mode.

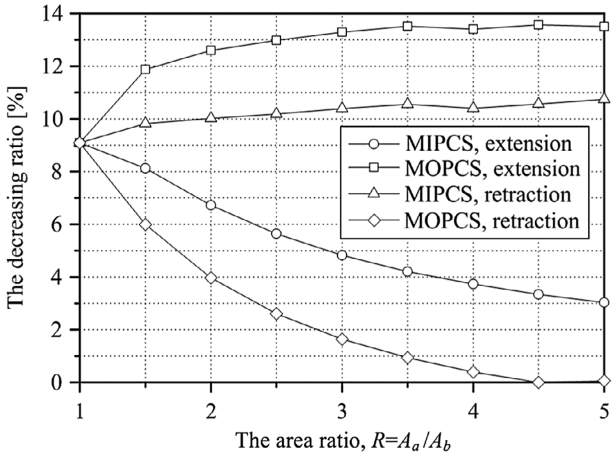

Figure 16. The decreased ratio of the average supply pressure.

is increasing from 1 to 5, and the MOPCS have the lowest average supply pressure P_s in the three systems.

The Figure 15 shows that the average supply pressure P_s of the three systems are increased when the area ratio is increasing from 1 to 5, and the MIPCS have the lowest average supply pressure P_s in the three systems.

It is also clear that the average supply pressure P_s of the PCLSS is higher than other systems in the Figures 11 and 12.

As is shown in the Figures 14 and 15, the average supply pressure P_s of the MIPCS and MOPCS are lower than the PCLSS, but it can't be shown the decreased range of the supply pressure P_s. So take the average supply pressure P_s of the PCLSS as a reference and calculate the decreased ratio of average supply pressure P_s of the MIPCS and MOPCS with extension and retraction working modes. The results of decreased ratio of the average supply pressure P_s can be seen in Figure 16.

The Figure 16 shows that the decreased ratios of the both systems in the extension working modes are higher than the retraction working modes, so the both the systems in the extension working modes have better energy saving performances than in the retraction working modes. It is also clear in the Figure 13 that the decreased ratio of the MOPCS in the extension working mode is higher than the MIPCS, but in the retraction working mode the decreased ratio of the MOPCS is lower than the MIPCS.

6. Discussion and conclusions

The independent pressure compensation principle of the independent metering system and its energy saving performances were investigated in this study. Compared to the conventional pressure compensated load sensing system, there are two pressure compensation methods. Both methods can obtain the desired proportional output velocity with external load disturbance.

The energy saving performance of the MIPCS, MOPCS, and pressure compensated load sensing system were compared according to the results of the equations discussed above; these results allowed us to determine the most energy efficient manner of opening the meter-in and meter-out orifices for optimal output velocity.

Comparison results have been obtained from the calculations. When the area ratio R in the hydraulic system equals 1, such as symmetrical cylinder or hydraulic motor, the energy savings of the meter-in and meter-out pressure compensation systems are same, in other words, the meter-in pressure compensation method is equivalent to the meter-out pressure compensated method in independent metering system. In the extension working mode, the optimal pressure compensation method is meter-out, because it can reduce the input power of the system considerably. In the retraction working mode, the optimal pressure compensation method is meter-in for the same reason. In short, the main working mode determines whether meter-in or meter-out pressure compression is better for the independent metering system.

Our results suggest that it is crucial to determine the actuator's main working mode (extension working mode or retraction working mode), prior to designing an independent metering load sensing system with either the meter-in or meter-out pressure compensation method. The comparison provided above indicated that the load sensing system with hydro-mechanical pressure compensation and independent metering offers more significant energy savings than the traditional load sensing system.

Dynamic effects and system stability were ignored for the purposes of this study, so the analysis results do not fully accurately reflect the actual system; similarly, because the system was simplified for the purposes of analysis no all variations in overall systematic parameters could be accounted for. The results can, regardless, be considered a reference for designing independent metering system, especially those that use the independent pressure compensation method. Compared to the pressure compensated load sensing system, the supply pressure control strategy of the load sensing system with hydro-mechanical pressure compensation and independent metering yet requires further research.

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Nomenclature

P_S	pump supply pressure (MPa)
P_0	pressure of the tank (MPa)
P_a	outlet pressure of the meter-in valve (MPa)
P_b	inlet pressure of the meter-out valve (MPa)
P_c	inlet pressure of the regulating valve (MPa)
A_a	the head chamber area (m^2)
A_b	the rod chamber area (m^2)
A_c	effective area of the pressure compensator (m^2)
Q_a	flow rate of metering in (L/min)
Q_b	flow rate of metering out (L/min)
Q_{a1}	flow rate of pressure compensator (L/min)
Q_{a2}	flow rate of regulating valve (L/min)
Q_s	the flow rate from the hydraulic pump (L/min)
F	external force (N)
v	the velocity of piston (m/s)
K_s	the spring stiffness coefficient (N/m)
$K_{\Delta p}$	the substituted parameter (–)
K_v	the other substituted parameter (–)
C_{d1}	flow coefficient of the pressure compensator (–)
C_{d2}	flow coefficient of the regulating valve (–)
W_c	area gradient of the pressure compensator (m)
W_v	area gradient of the regulating valve (m)
x_0	the spool pre-displacement of the pressure compensator (m)
x_1	the spool displacement of the pressure compensator (m)
x_2	the spool displacement of the regulating valve (m)
x_{max}	the maximum spool displacement of the regulating valve (m)
x_{in}	meter-in valve opening ratio (100%)
x_{out}	meter-out valve opening ratio (100%)
α	the jet angle of the pressure compensator (°)
δ	the spring pre-displacement (m)
μ	meter-in and meter-out ratio, x_{in}/x_{out} (–)
R	the area ratio, A_a/A_b (–)
ρ	the density of the hydraulic fluid (kg/m^3

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References

Eriksson, B. and Palmberg, J.O., 2011. Individual metering fluid power systems: challenges and opportunities. *Proceedings of the institution of mechanical engineers, part I: journal of systems and control engineering*, 225 (2), 196–211.

Hu, H. and Zhang, Q., 2002a. Development of a programmable E/H valve with a hybrid control algorithm. *SAE International Off-Highway Congress co-located with CONEXPO-CON/AGG*. Las Vegas, NV.

Hu, H. and Zhang, Q., 2002b. Realization of programmable control using a set of individually controlled electrohydraulic valves. *International journal of fluid power*, 3 (2), 29–34.

Hu, H. and Zhang, Q., 2003. Multi-function realization using an integrated programmable E/H control valve. *Applied engineering in agriculture*, 19 (3), 283–290.

Jansson, A. and Palmberg, J.O., 1990. *Separate controls of meter-in and meter-out orifices in mobile hyraulic systems. International Off-Highway & Power Plant Congress and Exposition*. Milwaukee, WI.

Krns, H.P., Jansson, A., and Palmberg, J.O., 1996. The design of pressure compensators for load sensing hydraulic system. *Control*, 96, 1996–1456.

Liu, S. and Yao, B., 2006. Automated onboard modeling of cartridge valve flow mapping. *IEEE/ASME transactions on mechatronics*, 11 (4), 381–388.

Shenouda, A. and Book, W., 2005. *Energy saving analysis using a four-valve independent metering configuration controlling a hydraulic cylinder. SAE Commercial Vehicle Engineering Conference*, Chicago, Illinois.

Shenouda, A. and Book, W., 2008. Optimal mode switching for a hydraulic actuator controlled with four-valve independent metering configuration. *International journal of fluid power*, 9 (1), 35–43.

Sitte, A. and Weber, J., 2013. Structural design of independent metering control systems. *In*: *11th Scandinavian international conference on fluid power*. Linköping: Linköping University Electronic Press, 261–270.

Suzuki, K. and Urata, E., 2008. Development of a water hydraulic pressure-compensated flow control valve. *International journal of fluid power*, 9 (3), 25–33.

Tabor, K.A., 2005a. *A novel method of controlling a hydraulic actuator with four valve independent metering using load feedback. SAE Commercial Vehicle Engineering Conference*, Chicago, Illinois.

Tabor, K.A., 2005b. *Optimal velocity control and cavitation prevention of a hydraulic actuator using four valve independent metering. SAE Commercial Vehicle Engineering Conference,* Chicago, Illinois.

Wang, T. and Wang, Q., 2014. An energy-saving pressure-compensated hydraulic system with electrical approach. *IEEE/ASME transactions on mechatronics*, 19 (2), 570–578.

Yao, B., 2009. Integrated mechatronic design of precision and energy saving electro-hydraulic systems. *In*: *Proceedings of the 7th international conference on fluid power transmission and control*. Hangzhou, 360–372.

Yao, B. and DeBoer, C., 2002. Energy-saving adaptive robust motion control of single-rod hydraulic cylinders with programmable valves. *In*: *Proceedings of the American Control Conference,* 2002 May 8–10. Anchorage, AK: IEEE.