Energy Consumption and Dynamic Behavior Analyses in Single-Rod Hydraulic Actuation of Mobile Machines

by

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ABSTRACT

This thesis investigates the performance of single-rod hydraulic actuation of mobile machines. The performance is evaluated by studying the energy consumption and the dynamic performance of such actuation systems. Presently, valve-controlled systems control actuators of mobile machines. However, they are inefficient in terms of energy consumption. Therefore, low-energy consumption pump-controlled systems have been proposed. This thesis studies the energy consumption of pump-controlled systems developed by Wendell and Rahmfeld-Ivantysynova and valve-controlled systems. This thesis also compares the dynamic performance of the above pump-controlled systems with the load sensing valve-controlled system. A set of evaluation tests is conducted on single-rod hydraulic actuations that are simulated using the Automation Studio® software. Based on a cycle test, the energy consumption of Wendell's pump-controlled system is 60% lower than that of the load sensing system. However, the above pump-controlled system leads to the poor dynamic behavior of a single-rod cylinder.

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NOMENCLATURE

P_a	Piston side pressure of cylinder	[bar]
P_b	Rod side pressure of cylinder	[bar]
P_p	Pump pressure	[bar]
P_{pa}	Pump pressure at piston side of cylinder	[bar]
P_{pb}	Pump pressure at rod side of cylinder	[bar]
P_l	Load pressure	[bar]
P_r	Relief valve input pressure	[bar]
P_{min}	Accumulator minimum pressure	[bar]
P_{max}	Accumulator maximum pressure	[bar]
P_e	Accumulator pre-charge pressure	[bar]
P_{acc}	Accumulator pressure	[bar]
ΔP_{LS}	Differential pressure between the pump pressure and the load pressure in load sensing system	[bar]
$P_{poc1}, P_{poc2}, P_{poc3}$	required pilot pressure value to open a pilot-operated check valve	[bar]
Q_a	Flow rate at piston side of cylinder	[L/min]
Q_b	Flow rate at rod side of cylinder	[L/min]
Q_p	Pump flow rate	[L/min]
Q_{pa}	Pump flow rate at piston side of cylinder	[L/min]
Q_{pb}	Pump flow rate at rod side of cylinder	[L/min]
Q_r	Relief valve flow rate	[L/min]
Q_{acc}	Accumulator flow rate	[L/min]
Q_{poc1}, Q_{poc2}	Pilot-operated check valve flow rate	[L/min]
Q_c	Check valve flow rate	[L/min]
P_{loss}	Power loss	[kW]
P_{load}	Power requirement to move an actuator	[kW]
P_{eng}	Engine power transmission	[kW]
P_{in}	Input power	[kW]
P_v	Directional valve power loss	[kW]
P_r	Relief valve power loss	[kW]
E_{eng}	Engine energy transmission	[kJ]
E_p	Pump output energy	[kJ]
E_{pl}	Pump energy loss	[kJ]
E_{load}	Useful work to move an actuator	[kJ]
E_v	Directional valve energy loss	[kJ]
E_r	Relief valve energy loss	[kJ]
x	Cylinder displacement	[m]
v	Cylinder speed	[m/s]
F	External force	[kN]
$F(t)$	Switching force	[kN]
M	Load	[kg]

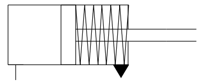
θ	Angle between rod end of cylinder and horizontal line	[degree]
V_x	Volume of displaced gas in accumulator	[L]
V_a	Total gas volume in accumulator	[L]
ω_{eng}	Engine speed	[rpm]
ω_{max}	Engine maximum speed	[rpm]
A_a	Piston side area of cylinder	[m^2]
A_b	Rod side area of cylinder	[m^2]
D_p	Pump displacement	[cm^3/rev]
C	Pilot ratio of pilot-operated check valve	

ABBREVIATIONS

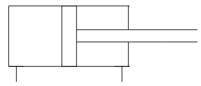
OC	Open-center
CP	Constant-pressure
LS	Load sensing
ELS	Electrical load sensing
PL	Pressure limiting
POC	Pilot-operated check
TPC	Two-position circulation
HHV	Hydraulic hybrid vehicle
EHV	Electrical hybrid vehicle
HPA	High-pressure accumulator
LPA	Low-pressure accumulator

HYDRAULIC SYMBOLS

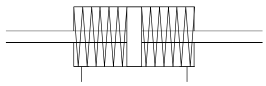
Actuator



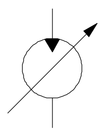
Single-acting single-rod cylinder with spring return



Double-acting single-rod cylinder

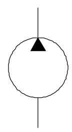


Double-acting double-rod cylinder with spring return

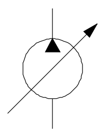


Unidirectional variable-displacement hydraulic motor

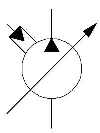
Pump



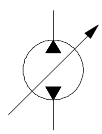
Unidirectional fixed-displacement pump



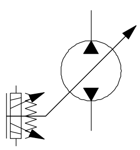
Unidirectional variable-displacement pump



Load sensing pump

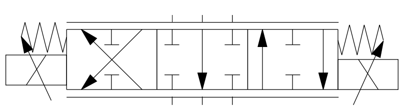


Bidirectional variable-displacement pump

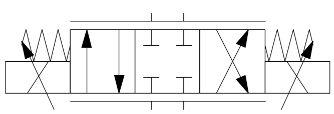


Bidirectional variable-displacement pump electrically controlled

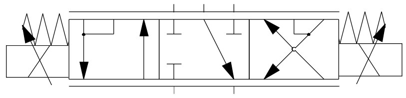
Proportional directional valve



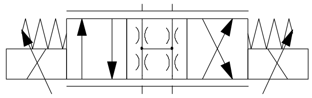
6 port / 3 position open-center electrically controlled



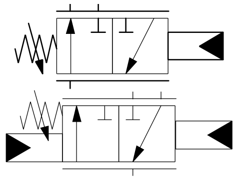
4 port / 3 position closed-center electrically controlled



5 port / 3 position load sensing electrically controlled

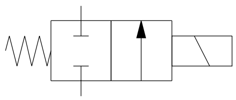


4 port / 3 position all ports open-center servo electrically controlled

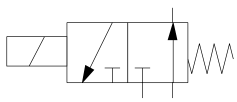


3 port / 2 position hydraulically controlled

Directional valve

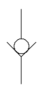


2 port / 2 position normally closed electrically controlled

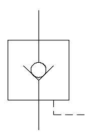


3 port / 2 position electrically controlled

Flow valve

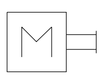


Check valve



Pilot-operated check valve

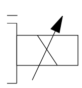
Power transmission



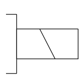
Engine

Shaft

Valve actuation



variable solenoid



On-off solenoid

Hydraulic pilot

Spring return

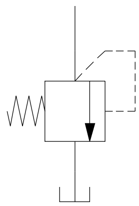
Variable spring return

Line

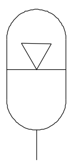
Main

Pilot

Frame containing several components



Relief valve



Gas-loaded accumulator

Tank

1. INTRODUCTION

1.1. Background

Hydraulic actuators are essential parts of most mechanical systems in industry. A linear hydraulic actuator, also called a hydraulic cylinder, is a device that is used to give a linear force through a linear stroke [1]. There exist two types of hydraulic cylinders: single-rod and double rod. A double-rod hydraulic cylinder has two rods; each connected to the piston of the cylinder. Double-rod hydraulic cylinders are used when equal pulling and pushing speed of the cylinder, without any special control components, is desirable. However, double-rod hydraulic cylinders require too much space which makes them not suitable for mobile machines [2]. A single-rod hydraulic cylinder has one rod that is usually connected to movable mechanical parts such as an excavator bucket. Having one rod, the material and sealing components of the single-rod cylinder are cheaper than that of the double-rod cylinder. At constant pressure, a single-rod hydraulic cylinder provides a stronger hydraulic force during the cylinder extension than during the cylinder retraction [3]. However, since more fluid is required to fill the piston side of the single-rod hydraulic cylinder, cylinder extension is considerably slower [4].

1.2. Problem Statement

One of the high-demand applications of single-rod hydraulic cylinders is in mobile machines. Mobile machines such as excavators, lift-trucks and wheel loaders are widely used in construction fields. Hydraulic systems, currently used in these mobile machines, are valve-controlled. In these hydraulic systems, a directional valve controls the cylinder speed, and a pump generates flow to the system. The throttling of the directional valve causes energy loss of the pressurized fluid coming from the pump. On the other hand, high energy consumption and wide usage of such hydraulic systems in the construction field cause environmental pollutions and high fuel consumption. Therefore, new hydraulic circuits have been designed in such a way that the directional valve is eliminated from the system and a pump-controlled drive system controls a single-rod cylinder. Depending on the pump operation, there are two major types of pump-controlled systems: closed-circuit and open-circuit. If the pump does not have pre-defined high and low pressure compartments, it is said to work in a closed-circuit. When the pump only operates against high pressure on one side it is working in an open-circuit [5]. Pump-controlled systems evaluated in this thesis are closed-circuit types. In some pump-controlled systems, in the case of lowering a load, the pump serves as a hydraulic motor, causing the system to generate the energy rather than consuming it. Thus, the energy consumption of such pump-controlled systems is considerably less than that of valve-controlled systems. However, a pump-controlled system may cause poor dynamic behavior of a cylinder. For example, in the pump-controlled displacement actuation developed by Rahmfeld and Ivantysynova [2], the actuator speed showed oscillatory behavior during changing modes of the pump. Solutions for this problem were proposed with a controlling feedback method [6].

1.3. Objectives of the Thesis

In recent research in the field of mobile hydraulics, the energy consumption of the valve-controlled systems including load sensing (LS) [7], and pump-controlled systems such as

Rahmfeld-Ivatysynova's [2] and Wendell's [8] has been studied. However, there are few evaluations on dynamic performance of such hydraulic systems. Although running a hydraulic system with less energy consumption is desirable, the dynamic performance of hydraulic systems is also important. Poor dynamic behavior of the actuator may contribute to the reduction of the quality of tasks and consequently the productivity. Based on the above discussion, the objectives of this research are:

(i) To evaluate the energy consumption of single-rod hydraulic actuation of mobile machines including three types of commonly-used valve-controlled systems and two types of proposed pump-controlled systems. The evaluations are based on the simulation studies on a single-rod hydraulic cylinder using Automation Studio software. The chosen pump-controlled systems, developed by Rahmfeld-Ivantysynova [2] and Wendell [8], are designed for the purpose of controlling the actuator of mobile machines. These two systems have already been examined on Bobcat 435 (Rahmfeld-Ivantysynova's system) and Caterpillar 330B excavators (Wendell's system).

(ii) To evaluate the dynamic performance of Rahmfeld-Ivantysynova's and Wendell's pump-controlled systems and a typical LS valve-controlled system. The LS system is recognized as a standard system in today's machines when efficiency is a priority. Here, the LS system is considered as a baseline for comparison with the above low-energy consumption pump-controlled systems. The evaluations are done by visually observing the cylinder speed.

1.4. Scope of the Thesis

In Chapter 2, a literature review on the hydraulic actuation of mobile machines is presented. Typical valve-controlled systems currently used in mobile machines such as Open-Center (OC), Constant-Pressure (CP) and LS systems, and two types of pump-controlled systems proposed by Rahmfeld-Ivantysynova and Wendell are described. Hydraulic components of the above systems are specified for further simulations and evaluations.

In Chapter 3, energy terminologies related to hydraulic systems such as energy storage, energy consumption and energy recovery are defined. The energy consumption of the studied hydraulic systems (OC, CP, LS, Rahmfeld-Ivantysynova's and Wendell's systems) are calculated and compared based on a test condition. A valve-controlled system having the ability to recover energy is introduced, modified and compared with the same system without energy recovery.

In Chapter 4, four case studies are presented to evaluate the dynamic performance of the LS valve-controlled system, and Wendell's and Rahmfeld-Ivantysynova's pump-controlled systems. Based on case study conditions, each hydraulic system is simulated using Automation Studio® software. For each case study, discussions relating to the dynamic performance of hydraulic systems are presented. The purpose of this chapter is to examine how well the pump-controlled systems mentioned above can control a single-rod hydraulic cylinder. Evaluation case studies are designed in such a way that the drawback of each system can be shown. Conclusions are presented in Chapter 5.

2. ACTUATION CIRCUITS

Mobile machines such as excavators, lift-trucks and wheel loaders, are widely used in construction and agricultural fields. Using pressurized fluid in the single-rod cylinder, mobile machines are capable of lifting heavy loads and querying rocks. Each single-rod hydraulic cylinder in a mobile machine is controlled by a hydraulic system. Currently, valve-controlled systems are used to control actuators of mobile machines. These systems cause high energy consumption of mobile machines. The concept of pump-controlled system can be used to lower energy consumption. This chapter describes both valve-controlled and pump-controlled hydraulic systems used in mobile machines.

2.1. Valve-Controlled Actuations

In the field of mobile hydraulics, several types of valve-controlled systems have evolved over the years. They can be classified into two major types: open-center and closed-center. An open-center system in the simplest way has a fixed-displacement pump, a directional valve, and a relief valve (Fig. 2.1). This system is called an open-center system, because in the neutral position of the directional valve, the entire constant flow from the pump releases to the tank through the directional valve. Therefore, power used in moving the high volume of oil, is completely dissipated as heat [9, 10]. According to Fig. 2.1, as the directional valve spool moves to the left position, constant fluid from the pump goes to the piston side of the cylinder and extends the cylinder. When the directional valve spool moves to the right position, the cylinder retracts by the differential pressure between the two lines connected to the cylinder. When the pump pressure (P_p) exceeds the cracking pressure of the relief valve (P_r), the relief valve opens to release the fluid to the tank, and maintain the system pressure at P_r. Note that if the neutral position of the directional valve was closed, the fixed-displacement pump would be always under load and caused high energy consumption. The load-hold check valve is to prevent undesirable load sinking [11].

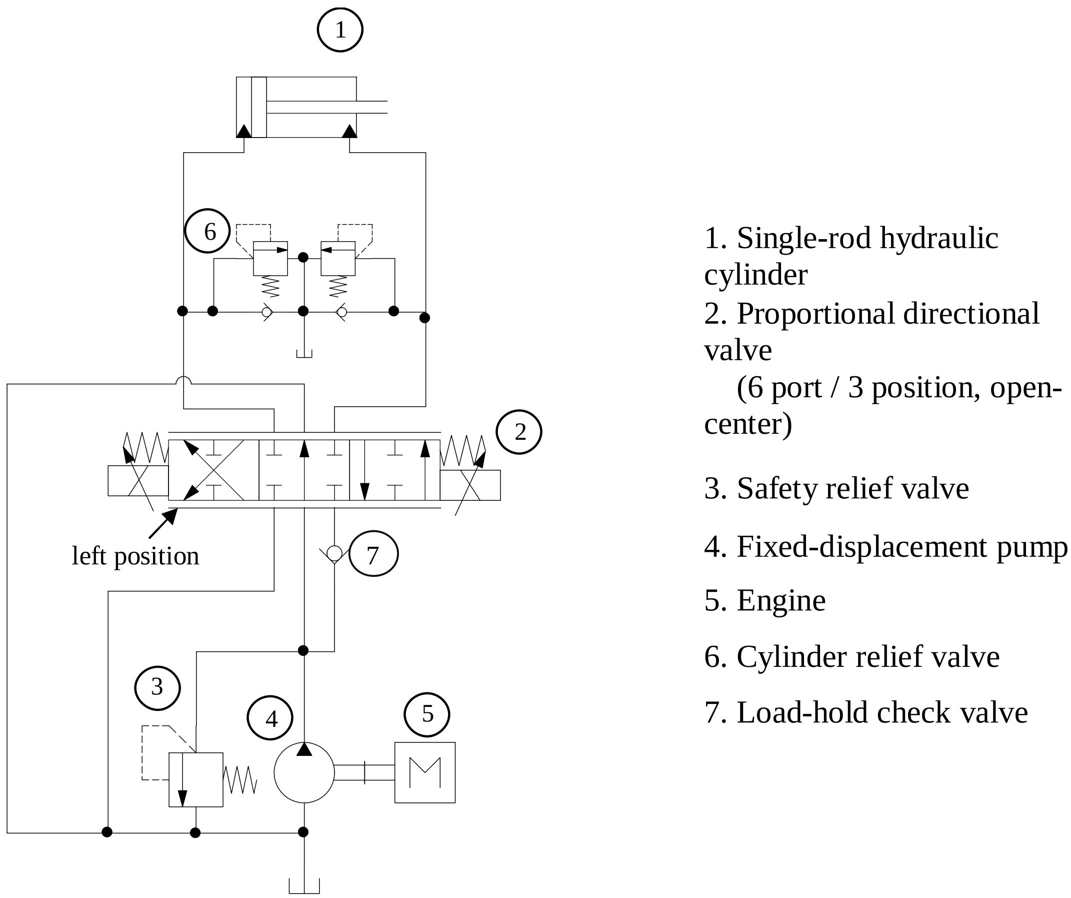
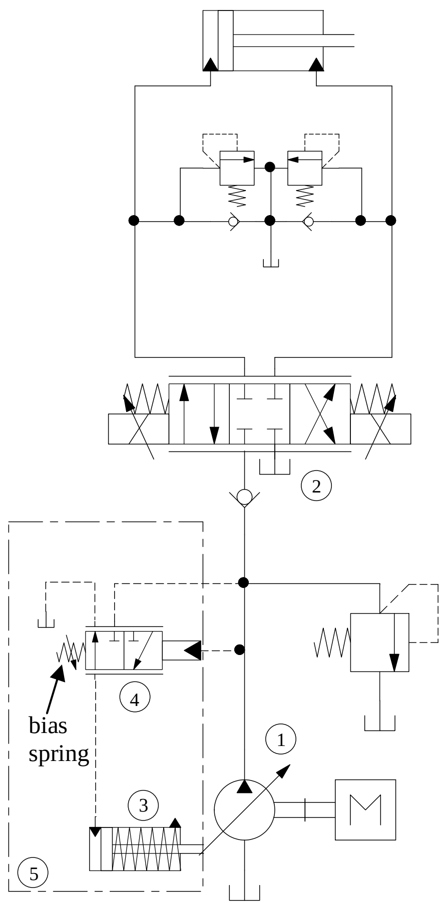


Fig. 2.1. Typical open-center (OC) hydraulic system.

In a closed-center system, the neutral position of the directional control valve is closed, requiring the use of a variable displacement pump to have the minimum displacement. Closed-center systems are classified into pressure compensated or constant-pressure (CP) and the load sensing (LS) [10]. In a CP system, the variable displacement pump provides the flow only required to maintain P_p at the pressure setting (see Fig. 2.2). The pressure setting is adjusted by the bias spring of the pressure compensator valve. In the case of exceeding the set pressure, the pressure compensator valve moves to the right position. Then, the servo piston extends and reduces the pump flow to maintain P_p at the pressure setting.

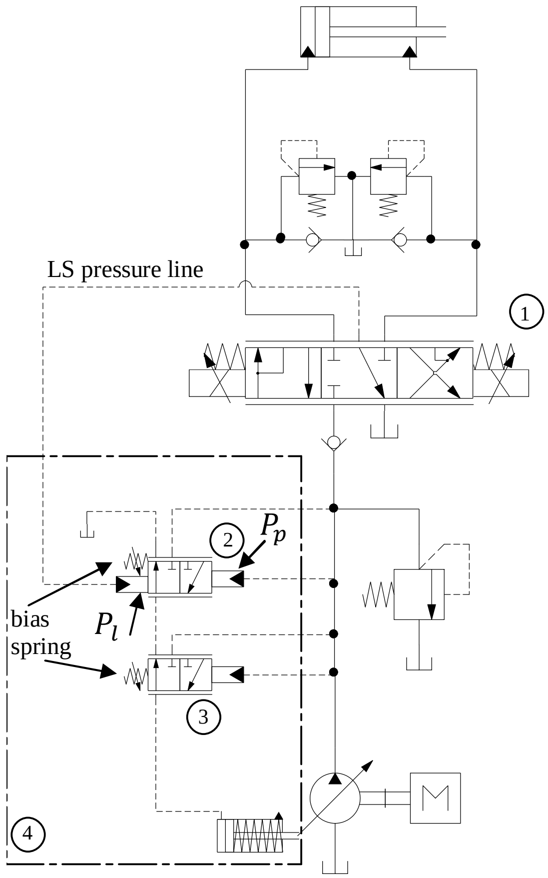


1. Variable displacement pump
2. Proportional directional valve (4 port / 3 position, closed-center)
3. Servo piston
4. Pressure compensator valve
5. Pressure compensator system

Fig. 2.2. Constant-pressure (CP) hydraulic system.

The LS system is a closed-center system with a control technique in which the displacement of a variable pump is adjusted by the feedback from the actuator load pressure (P_l). In fact, the LS control system regulates the pump flow, based on the cylinder load requirement. A simple LS valve-controlled system with one cylinder is shown in Fig. 2.3. The LS control system includes an LS valve, a pressure limiting (PL) valve and a servo piston. The LS valve adjusts the angle of the swash plate at any spool position of the directional valve in order to keep the differential pressure between P_p and P_l at a constant value $(P_p - P_l = \Delta P_{LS})$. ΔP_{LS} is set by the bias spring of the LS valve. The PL valve operates in the maximum pressure limiting mode, and only determines the maximum value of the system pressure [12]. When the directional valve is closed, the cylinder does not move and the pump pressure is equal to ΔP_{LS}. At this low pressure, the circuit consumes very little power. In the case of opening the directional valve to change the position of the cylinder, the LS pressure line from the port of the directional valve transmits the cylinder load pressure to the LS control system. Then, the LS control system adjusts the pump swashplate to maintain the pump pressure to approximately 10 to 20 bar above the load pressure [13].

In the LS control system, the LS pressure line transmits P_l to the pilot line of the LS valve and brings LS valve to the left position. As a result, a path is generated for amount of released flow from the piston side of the servo piston. Then, the retraction of the servo piston increases the pump flow rate. Since the LS valve remains on the left position and the pump flow is tuned on, P_p increases. When the pump pressure pilot line connected the LS valve becomes more than LS valve bias spring plus the load pressure, the LS valve moves to the right position. As a result, the servo piston extends and reduce the pump flow to maintain P_p at ΔP_{LS} higher than P_l.



1- Proportional directional valve ,
(5 port / 3 position Load Sensing)

2. LS valve

3. Pressure Limiting (PL) valve

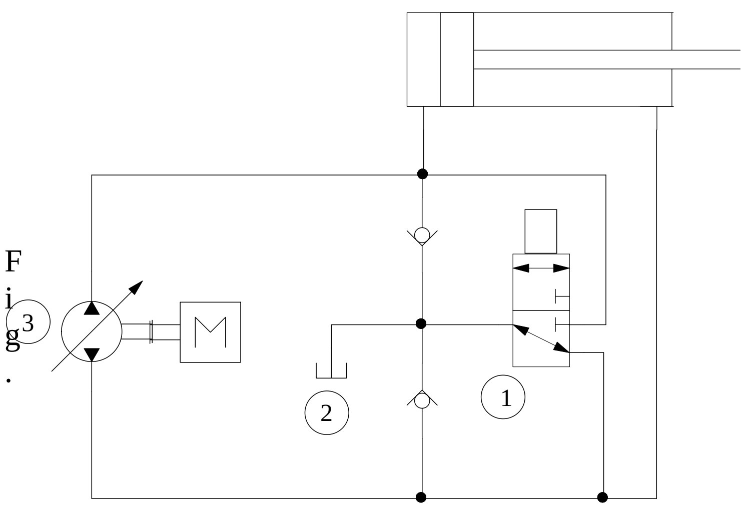
4. LS control system

Fig. 2.3. Load sensing (LS) hydraulic system.

Consider the situation where P_p exceeds the limited pressure adjusted by the bias spring of the PL valve. In this case, the PL valve moves to the right position and opens the flow path to the servo piston chamber. The servo piston extends to reduce the pump displacement. Thus, pump pressure cannot exceed set pressure of the PL valve. Bias pring pressure of the PL valve is usually set to less than the limited pressure in order to avoid getting closer to this critical value.

2.2. Pump-Controlled Actuations

One of the first designs for pump-controlled systems, is the diagram shown in Fig. 4 and developed by Hewett [14]. In this pump-controlled system, the bidirectional variable-displacement pump controls the speed of the cylinder by changing its swash plate (when it is coupled with an engine) or its shaft speed and direction (when it is coupled with an electrical motor). The two position circulation (TPC) valve [14] serves to compensate or remove (depending on the operational mode) the differential flow between the rod side and the piston side of the cylinder, in order to balance the flow on both sides of the pump. The valve may be activated hydraulically or electrically (not shown in the circuit). The two anti-cavitation check valves prevent the system from the negative pressure at pump lines.



1. Two position circulation valve
2. Hydraulic fluid supply (tank)
3. Bidirectional variable-displacement pump

Fig. 2.4. Pump-controlled hydraulic system developed by Hewett.

The direction of the external force applied to the cylinder and velocity of the cylinder can divide the cylinder operating space into four different load quadrants modes [15]. Fig. 2.5 shows these modes for Hewett system. For instance, extension of the cylinder against an external force is called extend resistive [16]. The load pressure originated from the external force is shown by a thick line. According to the flow circulation shown by the arrows, in both extend and retract resistive modes, the pump pressurizes the fluid on the cylinder side to move the cylinder in opposite direction of the external force. However, during extend and retract overrunning modes, external force and cylinder velocity are at the same direction, allowing the pressurized fluid of the cylinder sides to run the pump. In addition, the system is in low energy consumption, since the power source to run the pump is provided by the external force rather than the prime mover. In extend resistive and overrunning modes, the differential flow between the piston side and the rod side of the cylinder enters the pump lines thorough the TPC valve. The differential flow is directed to the hydraulic fluid supply when the hydraulic system is in extend overrunning and resistive modes.

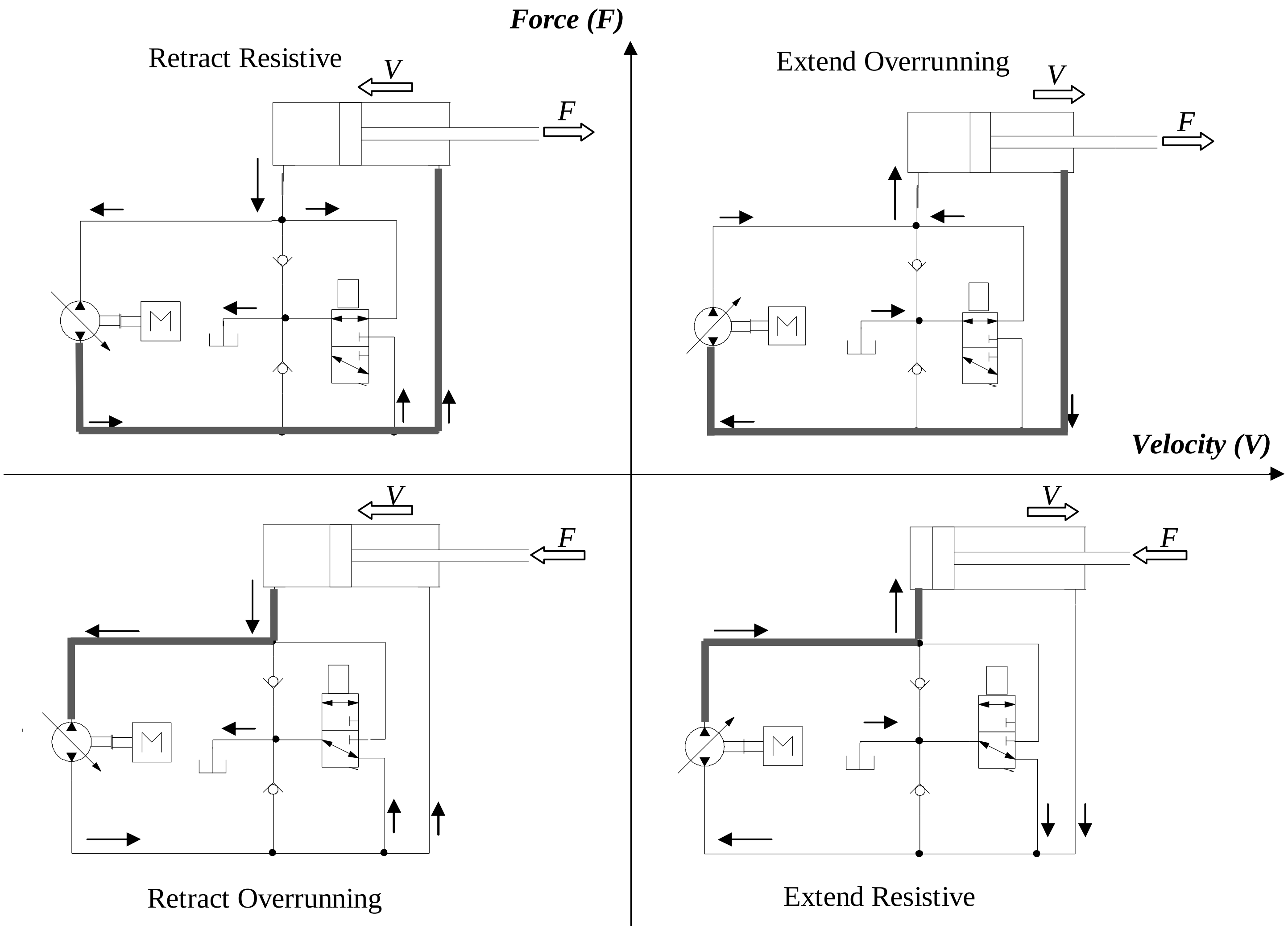


Fig. 2.4. Flow circulation of Hewett system in four-quadrant modes in V-F plane.

Rahmfeld and Ivantysynova [2] designed a pump-controlled hydraulic system called the "Energy Saving Pump-Controlled Displacement Actuation" (DC system) (see Fig. 2.6). The system was tested on a Bobcat 435 excavator and showed that the system has 34.3% fuel saving compared to the LS hydraulic system [2]. In this pump-controlled system, pilot-operated check (POC) valves serve to compensate or remove the differential flow between the piston side and rod side of the cylinder.

1. Pilot-operated check (POC) valve
2. Cylinder relief valve
3. Low-pressure system

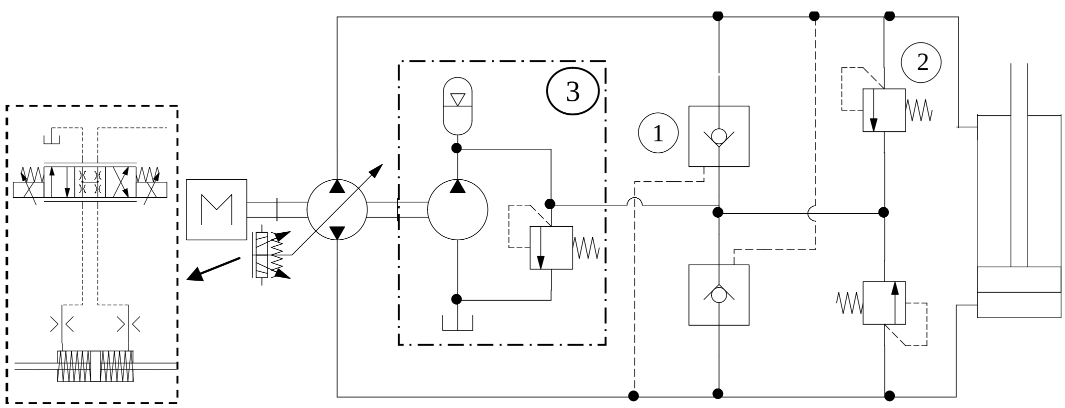


Fig. 2.5. Pump-controlled hydraulic system developed by Rahmfeld and Ivantysynova.

A POC valve is a check valve which can be opened by an external pilot pressure (see Fig. 2.7). It blocks the flow in direction A, like a standard check valve. But, it can release the flow in that direction once an adequate pilot pressure (P_{poc}) is applied. Free flow is allowed in the reverse direction, B.

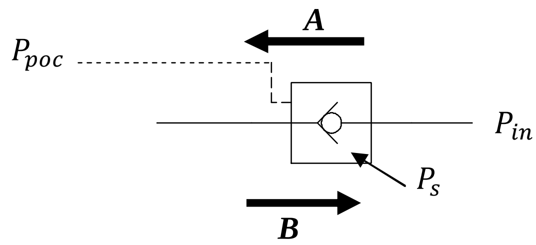


Fig. 2.6. Pilot-operated check valve hydraulic symbol.

P_{poc} is given by [17]:

$$P_{poc} = \frac{P_{in}}{C} + P_s$$

where P_{in} is the back pressure of the POC valve, P_s is the required pressure to press the spring of the POC valve, and C is the pilot ratio of the POC valve (see Fig. 2.7). The low-pressure system provides additional fluid for the hydraulic systems to prevent the pump cavitation. This system normally contains a relief valve, a charge pump and an accumulator (see Fig. 2.8). The cracking pressure of the relief valve in the low-pressure system affects the stiffness of the hydraulic cylinder. The higher cracking pressure provides more stiffness for the system while causing higher energy consumption of the system.

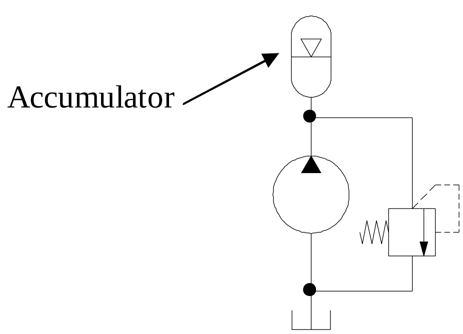


Fig. 2.7. Typical low-pressure system.

The flow circulation of the DC system is depicted in Fig. 2.9. The load pressure is shown by a thick line. During retract resistive and overrunning modes, POC valves remove the differential flow from pump lines to the low-pressure system. During extend overrunning and resistive modes, however, POC valves enter the differential flow to pump lines.

Note that Figs. 2.5 and 2.9 cannot show the functionality of the pump whether it works as either a pump or a motor. There is another four-quadrant arrangement in which four-quadrant modes are defined based on pump operational modes. In this arrangement, the cylinder operating space is divided into four-quadrant of a plane, with the cylinder differential pressure $(P_a - P_b)$ in horizontal axis and the cylinder velocity in vertical axis [6, 2]. Fig. 2.9 shows the operational mode of the DC system in $(P_a - P_b)$-V plane. As observed, during motoring modes, the load pressure runs the bidirectional pump. As a result, the bidirectional pump operates as a motor and produces torque to the prime mover. Note that in the DC system, $P_a - P_b$ is the differential pressure between the pump inlets, as well. Thus, the pump mode is determined by having the sign of the differential pressure and the cylinder velocity. Under this arrangement, the ability of a pump-controlled system, in changing the pump mode, can be shown. However, this arrangement cannot be applicable for pump-controlled systems in which the bidirectional pump serves only as a pump. An example of these systems is the Wendell circuit described next.

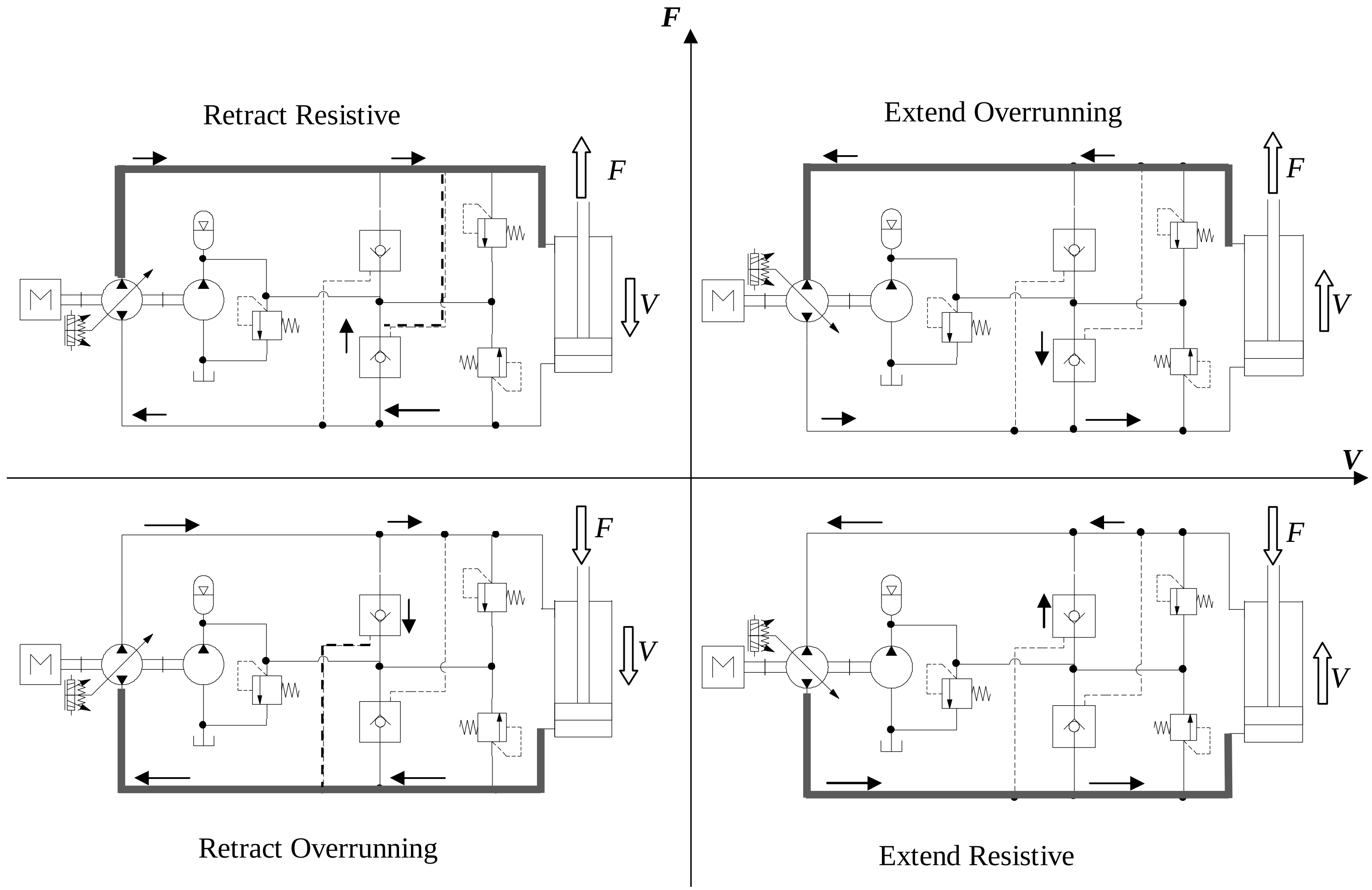


Fig. 2.8. Flow circulation of DC pump-controlled system in four-quadrant modes in F-V plane

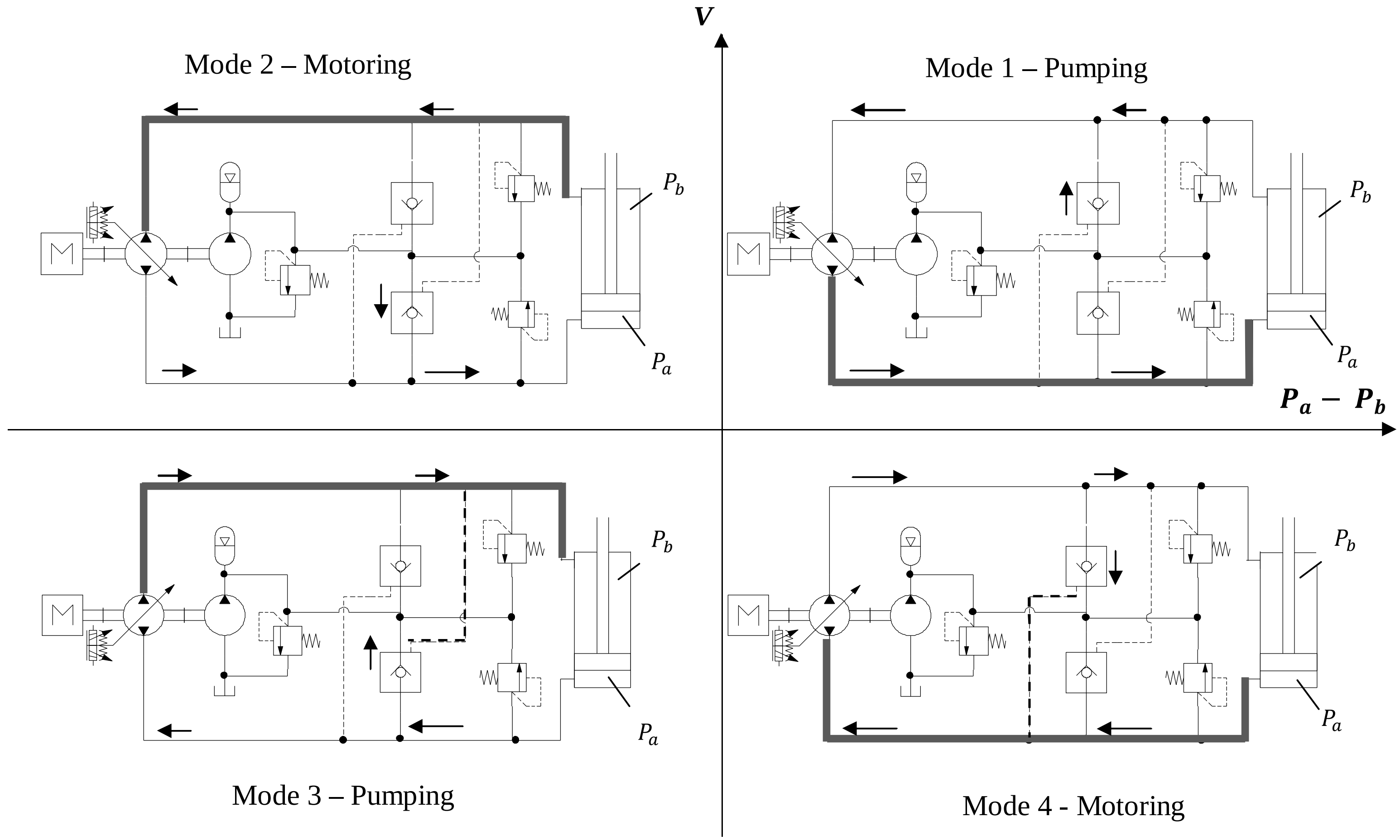


Fig. 2.9. Flow circulation of DC pump-controlled system in four-quadrant modes in $(P_a - P_b)$-V plane.

Wendell [12] presented a pump-controlled hydraulic system equipped with energy storage components. The system is called hydraulic regenerative. The system was experimentally evaluated on a Caterpillar 330B. Results indicated that power consumption reduces to 46% as compared to the conventional excavator design [12]. The major difference between this design and the DC circuit is to have POC valves at the both sides of the cylinder. These valves are the same as the POC valves in the DC circuit. However, these valves at the cylinder sides are used to prevent the cylinder from uncontrolled movements; especially when the cylinder carries a load [18]. With reference to Fig. 2.12, during extend and retract overrunning, the POC valves connected to the both sides of the cylinder do not allow the load pressure to run the pump. As a result, in both modes, the pump still serves as a pump and cannot produce additional power for energy recovery purposes[1].

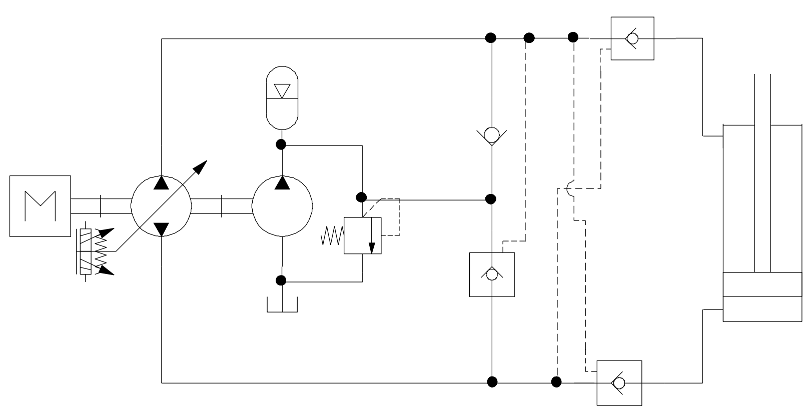


Fig. 2.10. Pump-controlled hydraulic system developed by Wendell.

[1] For more information about energy recovery, refer to section 3.1.3.

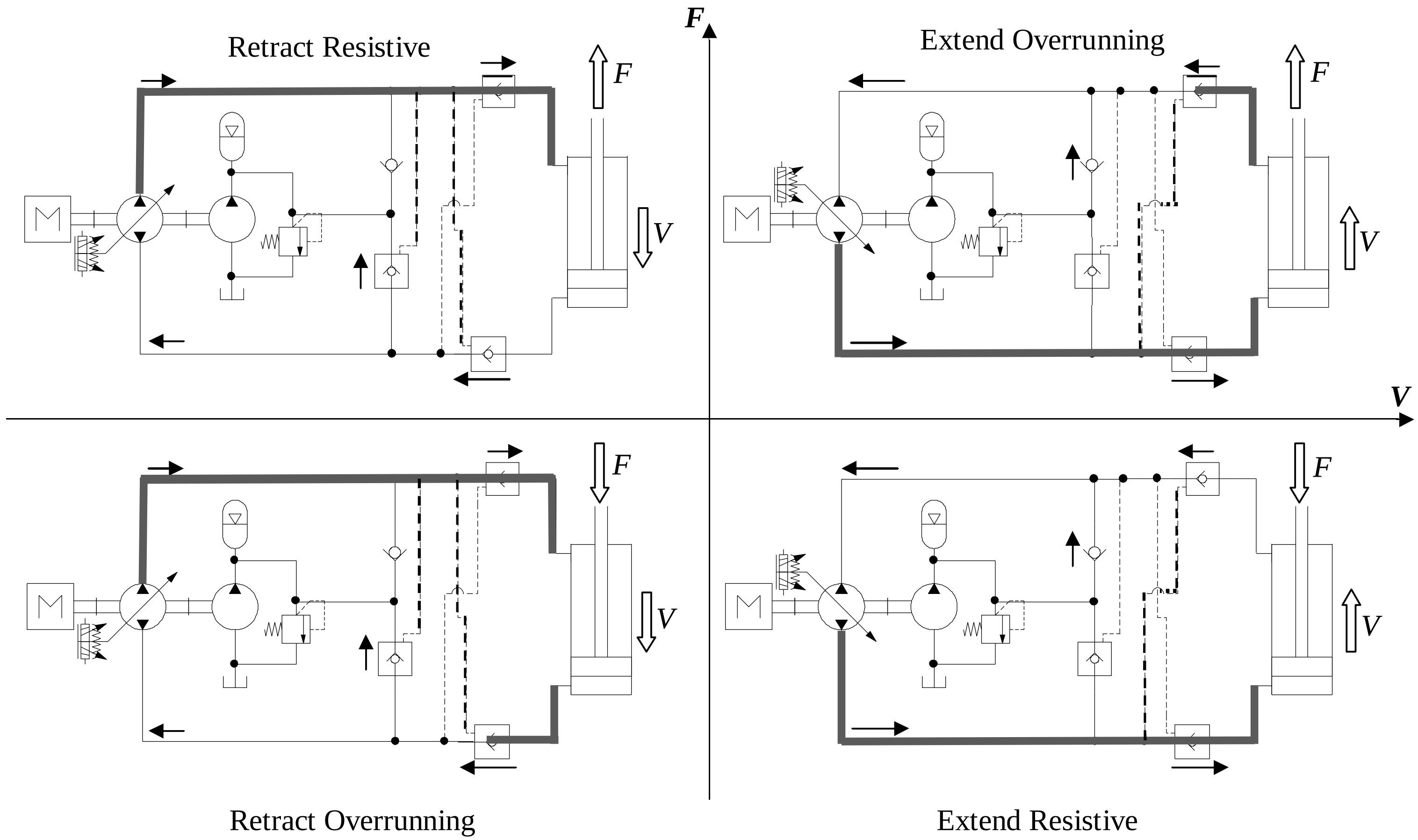


Fig. 2.11. Flow circulation of Wendell pump-controlled system in four-quadrant modes in F-V plane.

2.3. Parameters of Hydraulic Components

In Section 2.2, different types of hydraulic systems were described. Component sizing of these hydraulic systems is an essential part of the design that becomes more important when comparison of energy consumption and dynamic performance of such hydraulic systems are considered. To compare responses, the component parameters having the same task must be the same size. For example, the size of the variable-displacement pump in the LS system must be equal to the size of the fixed-displacement pump in the OC system. Table 2.1 shows the general specifications of the components used in all hydraulic systems introduced in Section 2.2. The system parameters are chosen from the component specifications of Bobcat 435 mini-excavator. Moreover, the cylinder size is the same as the size of the cylinder boom of Bobcat 435 mini-excavator [19].

Table 2.1. Specifications of hydraulic components used in simulation.

	Specification	Value
Engine [19]	Power (P_{eng})	49 hp
	Maximum speed (ω_{max})	2200 rpm
Pump [19, 20]	Displacement (D_p)	45 cm^3/rev
	Volumetric efficiency (η_{pv})	96 %
	Mechanical efficiency(η_{pm})	91.67 %
	Maximum flow ($D_p \omega_{max} \eta_{pv}$)	95.05 L/min
Cylinder [19]	Piston diameter	9.525 cm
	Rod diameter	5.08 cm
	Stroke	67.86 cm
Cylinder relief valves	Cracking pressure	300 bar (4351.13 psi)
Fluid: NUTO FG-32 [21]	Kinematic viscosity	0.000031 m^2/s
	Density	852 kg/m^3

Figure 2.13 shows typical friction as a function of the cylinder speed which is applied to the hydraulic cylinder during simulation in Automation Studio® software.

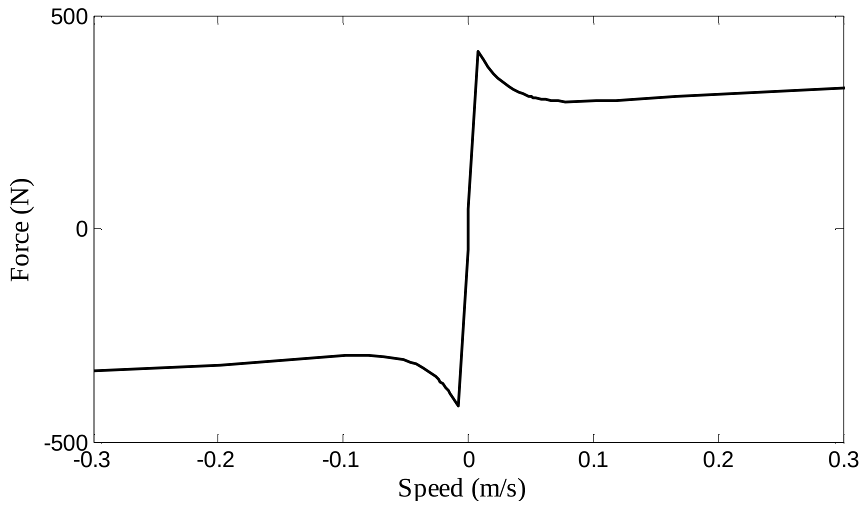


Fig. 2.12. Friction model for single-rod hydraulic cylinder.

Table 2.2 defines the set pressures of valve-controlled systems. The maximum pump pressure of the CP system can be adjusted by the bias spring of the pressure compensator valve (Fig. 2.2) while the LS pump maximum pressure is adjusted by the bias spring of the PL valve (Fig. 2.3). ΔP_{LS} represents the differential pressure between pump and load pressures in the LS system. This value is chosen high enough to overcome the pressure drop between the upstream and downstream of the directional valve. In the cases of sudden pressure boost, the safety relief valve (Fig. 2.3) opens to maintain the pump pressure at 200 bar level which is cracking pressure of the safety relief valve.

Table 2.2. Valve-controlled system pressure settings.

Valve-controlled type	Safety relief valve cracking pressure (bar)	Maximum pump pressure (bar)	ΔP_{LS} (bar)
LS	200	150	20
CP	200	150	-
OC	200	-	-

The directional valve for each valve-controlled system (OC, CP or LS) must have the same characteristic curve, during the operation of the cylinder, in order to make the valve-controlled systems comparable. Fig. 2.14 shows the characteristic curve of OC, CP or LS directional valves. As seen, they are chosen asymmetric, because in a single-rod hydraulic actuation, an asymmetric directional valve causes lower energy consumption than a symmetric directional valve [22]. In order to have a solid comparison between a low-energy consumption pump-controlled system and a typical valve-controlled system, the directional valve should not be sized in such a way that it causes high power loss. The maximum flow passing through directional valves is 120 L/min which is enough for a system in which the maximum pump flow is 95 L/min.

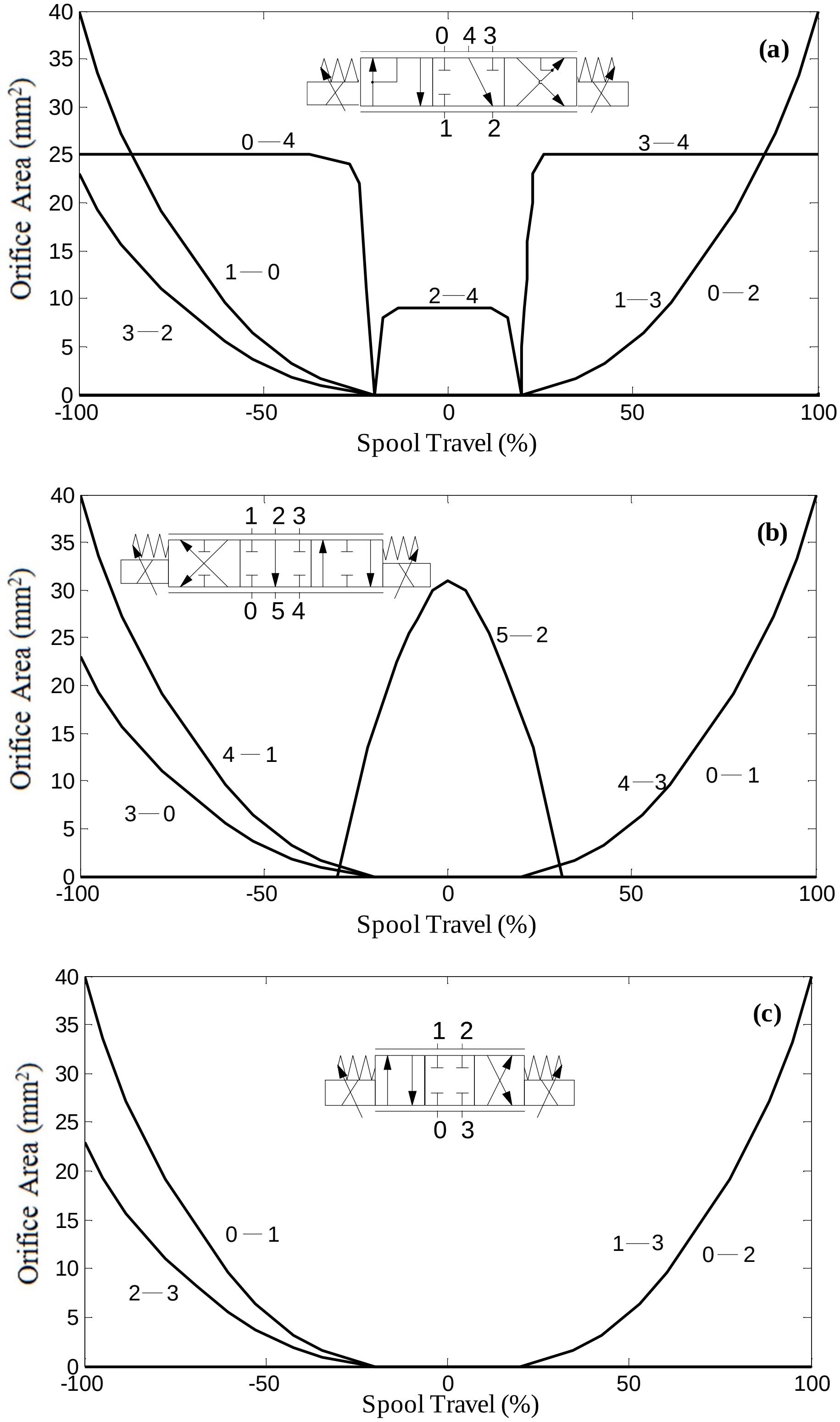


Fig. 2.13. Chracteristic curve of directional valve in (a) LS system, (b) OC system and (c) CP system.

Table 2.3 defines the specification of the pump-controlled system components shown in Fig. 2.10. In the low-pressure system, V_a, the size of the accumulator, is the total gas volume when the accumulator is empty. The accumulator is chosen 5 L, the standard size for an accumulator [23]. V_x is the total volume of the displaced gas in the accumulator which is either expanded or compressed by the pressurized fluid. In other words, the volume of the displaced gas is equal to the volume of the fluid stored or discharged. In the low-pressure system (Fig. 2.14), the maximum volume stored or discharged by the accumulator is determined by the differential volume between rod-side and piston side of the cylinder. In the accumulator, the heat process is assumed to be polytrophic. In other words, the ratio of specific heat (γ) is 1.25 as the fluid is being filled or replenished in the accumulator. P_{max} is the maximum working pressure of the system applied to the accumulator. In the low-pressure system (Fig. 2.14), P_{max} is the relief valve cracking pressure. P_{min} is the minimum working pressure of the system applied to the accumulator. P_e is the pre-charge pressure of the accumulator which is normally equal to $0.9P_{min}$. The accumulator is charged to reach P_e before it is used in a hydraulic system. In Table 2.3, P_e is calculated using the following equation:

$$P_e = \left(\frac{V_a}{V_x}\right)^{\gamma} \frac{\left(1 - \left(\frac{P_{min}}{P_{max}}\right)^{\frac{1}{\gamma}}\right)^{\gamma}}{P_{min}} \tag{2.1}$$

Table 2.3. Pump-controlled system specifications.

System	Component	Specification	Value
DC system	POC1	Cracking pressure	0.69 bar (10 psi)
		Pilot ratio	5
	POC2	Cracking pressure	0.69 bar (10 psi)
		Pilot ratio	5
Wendell system	POC1	Cracking pressure	0.69 bar (10 psi)
		Pilot ratio	3
	C	Cracking pressure	0.69 bar (10 psi)
	POC3	Cracking pressure	0.34 bar (5 psi)
		Pilot ratio	3
	POC4	Cracking pressure	0.34 bar (5 psi)
		Pilot ratio	3
Low-pressure system	Charge pump [19]	Displacement	15 cm^3/rev
	Accumulator	Pre-charge pressure (P_e)	8.7 bar
		Volume (V_a)	5 L
		V_x	1.38 L
		P_{min}	9.7 bar
		P_{max}	15 bar
	Relief valve	Cracking pressure (P_r)	15 bar

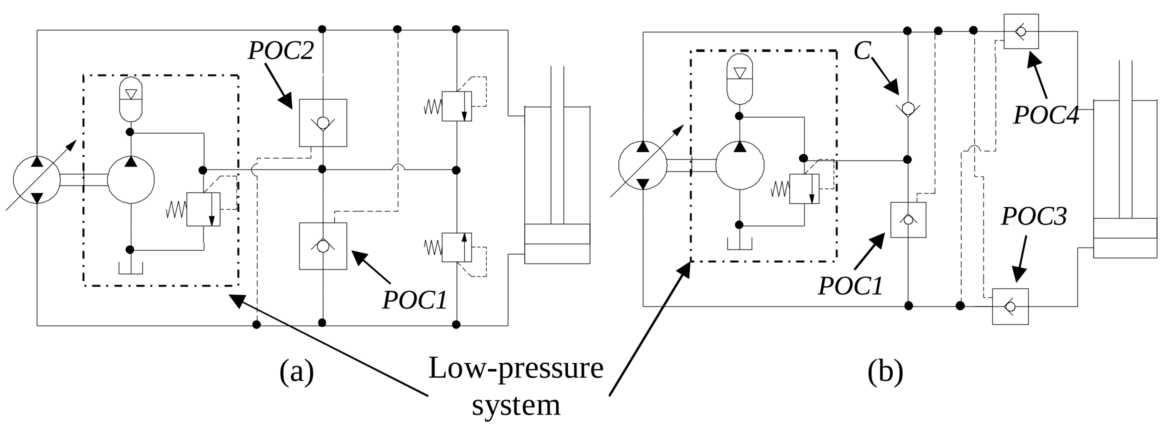


Fig. 2.14. Pump-controlled systems: (a) DC system; (b) Wendell system.

3. ENERGY CONSUMPTION

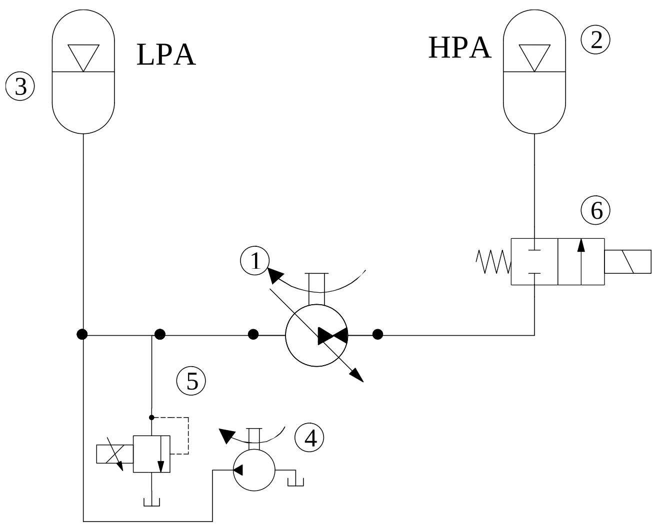
3.1. Background

During the last decade, fluid power systems have extensively been studied in terms of energy usage. As a result, some major fields were developed including energy storage, energy consumption and energy recovery.

3.1.1. Energy Storage

In hydraulics, energy storage is accomplished by a device called accumulator. Hydraulic accumulators store the hydraulic fluid under pressure. Pressure is commonly supplied through a bag, diaphragm or piston by either a spring, or pressured gas embedded in the accumulator. Energy storage systems can provide additional power leveling for the engine. This can result in a significant reduction in the required engine and pump size as well as improving overall efficiency and durability of the engine [8]. Note that an accumulator employed to store low-pressurized fluid is not an energy storage device.

Vehicles equipped with hydraulic energy storage systems, have the ability to store kinetic energy while braking and then reuse it for subsequent acceleration. Such vehicles are commonly called hydraulic hybrid vehicles (HHV), since they consist of two propulsion energy sources, the energy from the engine and the energy from the accumulator [8]. Usually, the hydraulic circuit of the HHV comprises of a high-pressure accumulator (HPA), a low-pressure accumulator (LPA), a pump and a charge pump (Fig. 3.1). In this configuration, the pump unit can operate as a pump, causing the vehicle to decelerate while transferring the fluid from the LPA to HPA. It operates as a motor, taking the fluid from HPA. Thus, when the vehicle decelerates, the HPA pressure rises and the LPA pressure falls, and converse as the vehicle accelerates. Because the pump is typically a high speed axial piston unit, it requires a charge pressure, typically about 10 bar, at its inlet. The HPA is considered as an energy storage device. Energy stored in the HPA scales off for longer term by the shut-off valve. The LPA contains low-pressurized fluid; therefore, it does not store the energy [21].



1. Variable displacement pump
2. High-pressure accumulator (HPA)
3. Low-pressure accumulator (LPA)
4. Charge pump
5. Proportional relief valve
6. Shut-off valve

Fig. 3.1. Energy storage system of a HHV [24].

3.1.2. Energy Consumption

Energy consumption is in proportion to the amount of the energy that a system uses to do a task. In hydraulics, power loss (P_{loss}) in a hydraulic circuit can originate from valve metering losses, pump losses, leakage, and line pressure drop. It is calculated according to following equation [25]:

$$P_{loss} = \sum_{i=1}^{n} (P_{in} - P_{mech}) \tag{3.1}$$

where P_{in} is the input power including engine power (P_{eng}) and P_{mech} donates the power consumed to run mechanical systems such as actuators and pumps.

To evaluate the energy consumption in a hydraulic system, energy consumption of each component in the circuit must be calculated. The following texts show how to calculate the energy consumption of the components commonly used in hydraulic circuits including engine, pump, directional valve, relief valve and hydraulic cylinder.

(i) Engine

The engine power (P_{eng}) is the product of the torque placed on the engine and the engine speed [26]:

$$P_{eng} = M_{eng}\omega_{eng} \tag{3.2}$$

where M_{eng} and ω_{eng} are engine torque and speed, respectively.

The energy transmitted by the engine is therefore calculated as follows:

$$E_{eng} = \int P_{eng}\, dt \tag{3.3}$$

(ii) Pump

The output energy of the pump is given by [13]:

$$E_p = \int Q_p P_p dt \tag{3.4}$$

where P_p and Q_p are the pump output pressure and the flow rate, respectively (see Fig. 3.2).

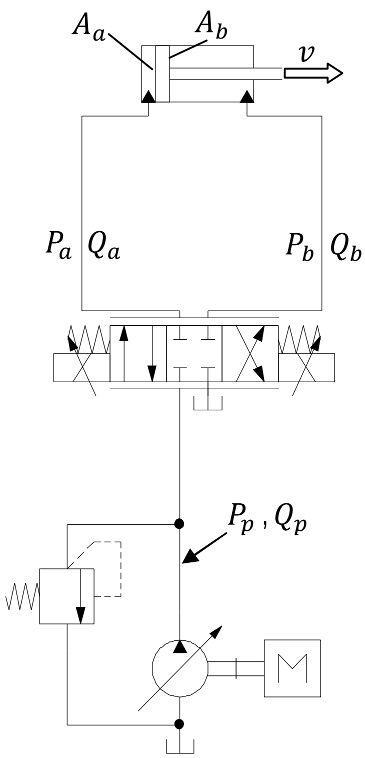


Fig. 3.2. Typical valve-controlled system

The pump loss is defined as the difference between engine input energy and pump output energy:

$$E_{pl} = E_{eng} - E_P \tag{3.5}$$

The pump efficiency (η_p) is given by [13]:

$$\eta_p = \frac{E_P}{E_{eng}} \tag{3.6}$$

(iii) Hydraulic cylinder

Load power (P_{load}) is the power required to move a hydraulic cylinder which is calculated from the cylinder speed and the load pressure applied to the cylinder [26]:

$$P_{load} = (P_a\ A_a - P_b\ A_b)v \tag{3.7}$$

where P_a and P_b are the piston side and rod side pressures of the cylinder, respectively. A_a and A_b represent the piston side area and rod side area of the cylinder, respectively. The cylinder speed is v (see Fig. 3.2). When P_{load} is positive, it is considered as the useful power that the system consumes to move the load.

The useful work to move an actuator is expressed by:

$$E_{load} = \int P_{Load} dt \quad \text{when } P_{load} \geq 0 \tag{3.8}$$

In case of having negative load, the work is performed due to the load weight that helps the cylinder move. That is, the pump consumes low energy to fill the cylinder with the fluid rather than overcoming the load weight.

(iv) Directional valve

The directional valve metering loss consists of the flow meter from the pump to the actuator and the flow meter from the actuator to the tank [26, 27]:

$$P_v = (P_p - P_a)Q_a + P_b Q_b sgn(Q_b) \quad \text{when cylinder extends} \tag{3.9}$$

$$P_v = (P_p - P_b) Q_b + P_a Q_a sgn(Q_a) \quad \text{when cylinder retracts} \tag{3.10}$$

where Q_a is the flow rate of the piston chamber and Q_b represents the flow rate of the rod chamber (see Fig. 3.2). P_v donates the power loss of the directional valve. The first term in either equation (3.9) or (3.10) is the power loss from the pump to the actuator. The second terms represent the power loss from the actuator to the tank.

It is assumed that:

$$Q_a > 0 \quad , \quad Q_b < 0 \quad \text{when cylinder extends} \tag{3.11}$$

$$Q_a < 0 \quad , \quad Q_b > 0 \quad \text{when cylinder retracts} \tag{3.12}$$

This inequality originates from the sign of the cylinder velocity. When the cylinder extends, the cylinder velocity is assumed to be positive, and it is negative when the cylinder retracts. Thus, in equations (3.9) and (3.10), $(P_p - P_a)Q_a$ and $(P_p - P_b)\ Q_b$ are always positive. Since, the total power loss of a directional valve is the summation of meter loss from the pump to the actuator and actuator to the tank, the terms $P_b\, Q_b$ and $P_a Q_a$ should be always positive. That is, both $P_b\, Q_b$ and $P_a Q_a$ are multiplied by a sgn function to keep them always positive. $sgn(x)$ is defined as below:

$$sgn(x) = \begin{cases} 1 & x > 0 \\ 0 & x = 0 \\ -1 & x < 0 \end{cases} \tag{3.13}$$

The directional valve energy loss (E_v) is given by:

$$E_v = \int \mathrm{P}_v\, dt \tag{3.14}$$

(v) Relief valve

Relief valve power and energy losses are given by:

$$\mathrm{P}_r = P_r\ Q_r \tag{3.15}$$

$$E_r = \int \mathrm{P}_r\, dt \tag{3.16}$$

where P_r is the relief valve input pressure, Q_r represents the relief valve flow rate (Fig. 3.3).

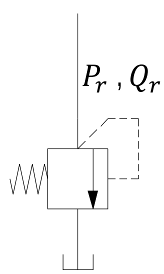


Fig. 3.3. Relief valve hydraulic symbol.

Cyclic Efficiency is defined as the ratio of total useful work done by hydraulic actuators to the total energy demanded by the hydraulic system from the engine in a working cycle [26, 22]:

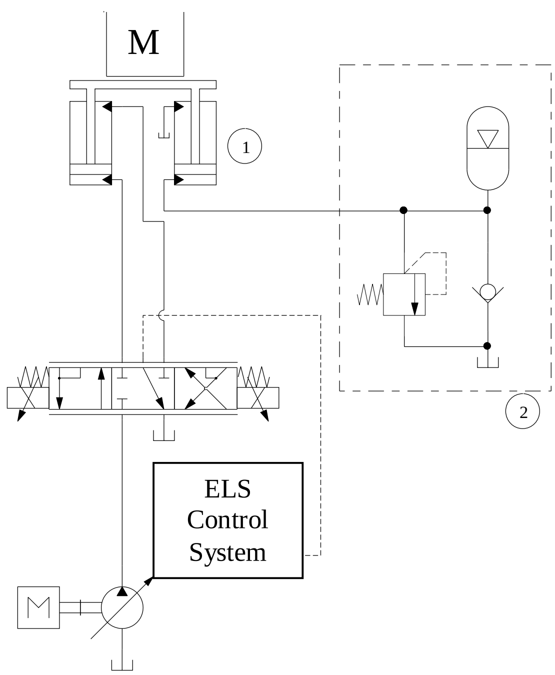
$$\eta_{cy} = \frac{\int \mathrm{P}_{load} dt}{\int \mathrm{P}_{eng}\, dt} \text{ when } \mathrm{P}_{load} \geq 0 \tag{3.17}$$

This value cannot represent the specification of a hydraulic system, since it is dependent on cyclic test conditions such as load and system parameters. In this thesis, η_{cy} is used for comparison between efficiency of different types of hydraulic systems based on the same test conditions.

3.1.3. Energy Recovery

Energy recovery is a method to reduce the energy consumption of a system by recovering the dissipated energy, and reutilizing it for the same system. In an excavator, for example, the heavy boom requires the actuator large force to balance the gravitational force. An energy recovery system stores the potential energy when the boom moves down and reuses it when the boom moves up. There are two types of energy recovery: the hydraulic energy recovery and the electrical energy recovery. In hydraulic energy recovery, the accumulator stores high pressures so that it is immediately available as the source energy. The advantage of using hydraulic accumulator is straightforward interface; they can be integrated in to hydraulic circuit by a variable pump. In electrical energy recovery, the battery is the main device of storing electrical energy in mobile vehicle. The main problem in electrical systems is that they cannot be charged into large power which is always required in a hydraulic vehicle [28].

Liang and Vivarlo [29] studied the energy recovery on a crane, where an electrical load sensing (ELS) valve-controlled system controls the speed of the cylinder. In this system, an electrical controller adjusts the swash plate of the pump by the load pressure feedback signal. The energy storage system is a split power unit, which includes a balance cylinder, an accumulator, a check valve and a relief valve. The balance cylinder is a separate cylinder that moves with the actual working cylinder (Fig. 3.4). When the crane lowers with load, the accumulator is charged. Potential energy of the crane and the load are thus recovered in the form of hydraulic energy. When the crane arm moves up, the saved energy is reutilized. With this hydraulic energy recovery method, the efficiency is 7% higher than that of ELS system without energy recovery [29].



1. Balance cylinder
2. Energy storage system

Fig. 3.4. Electrical Load Sensing (ELS) system with energy recovery developed by Liang and Vivarlo.

3.2. Evaluation of the Energy Consumption

3.2.1. Energy Consumption Test

In order to evaluate the energy consumption in a hydraulic system, a cyclic test must be set. The employed cyclic test, considered in this project, is to move the cylinder up and down with a constant load applied to it (Fig. 3.5a). This test simulates the condition of a hydraulic cylinder in a lift-truck. The load weight compresses the fluid at the piston side of the cylinder. The system is an open-loop one. The joystick directly controls the swash plate of the pump or the spool of the directional valve (Fig. 3.5b). The constant load on the cylinder is coming from 1142 kg. The load value corresponds to maximum gravitational load applied to the boom of a Bobcat 435 excavator machine.

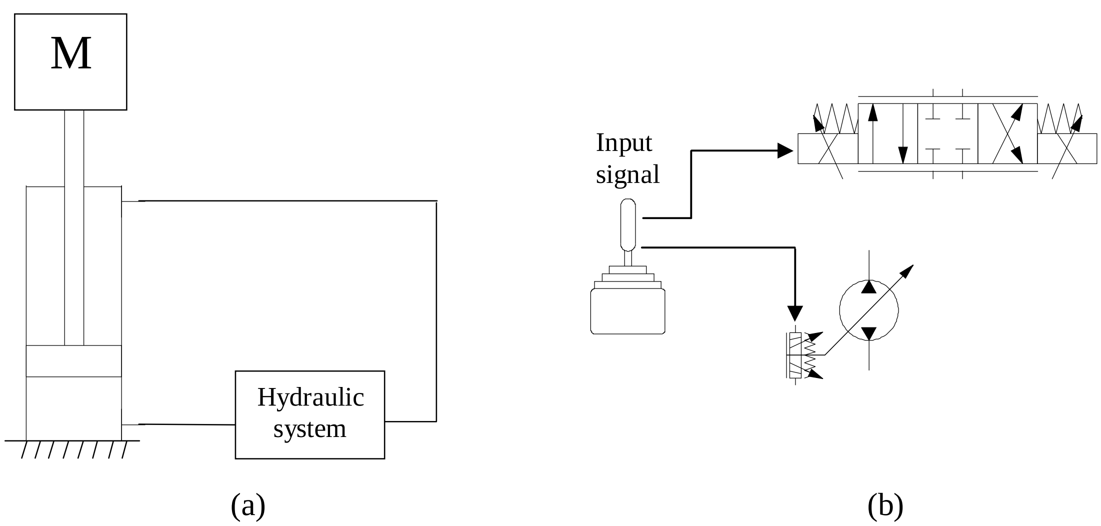


Fig. 3.5. Energy consumption test: (a) physical system; (b) control system.

Figure 3.6 shows the schematics of the mentioned hydraulic systems with their pressure and flow variables. The LS system variables can be used for CP and OC systems as well.

These variables can describe the response of the mentioned hydraulic systems when they control a single-rod hydraulic cylinder.

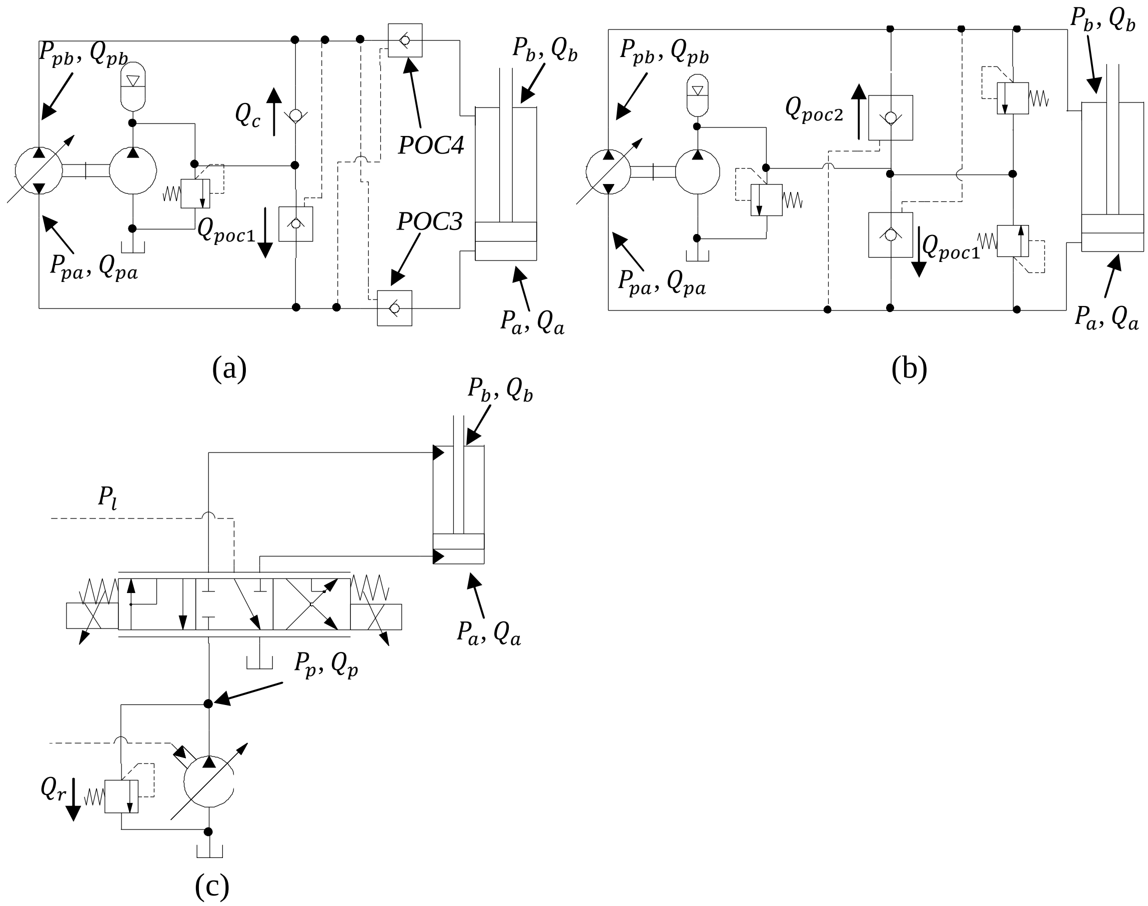


Fig. 3.6. Schematics of (a) Wendell, (b) DC, and (c) LS hydraulic systems and corresponding variables. P_{pa} is pump pressure at piston side of cylinder; Q_{pa} is pump flow rate at piston side of cylinder; P_{pb} is pump pressure at rod side of the cylinder; Q_{pb} is pump flow rate at rod side of cylinder; Q_{poc1} is flow rate of POC valve at cylinder piston side in DC and Wendell systems; Q_{poc2} is flow rate of POC valve at cylinder rod side in DC system; Q_c is flow rate of check valve in Wendell system.

As observed in Fig. 3.7, the flow, discharging from a bidirectional pump, is assumed to have negative sign, while the flow, entering the pump, is positive. The flow exiting from a check valve is also assumed to be positive. At the opposite direction, the flow entering a

POC valve is negative. The sign of the cylinder flow and speed were discussed in Section 3.1.2.

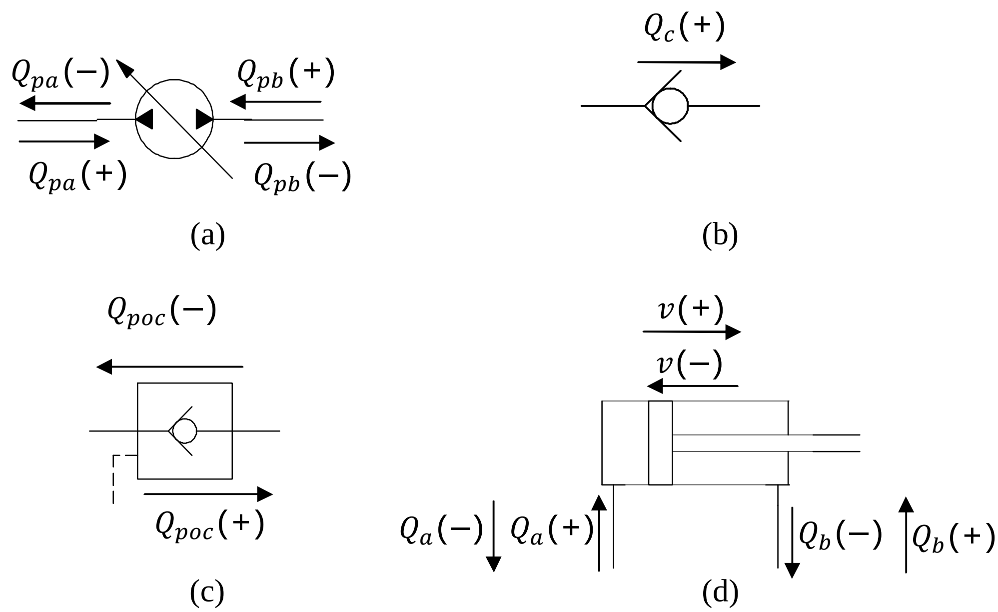


Fig. 3.7. Flow sign of (a) bidirectional pump, (b) check valve, (c) pilot-operated check valve, and (d) cylinder.

3.2.2. Response of Valve-Controlled Systems

In the OC system, as seen in Fig. 3.8.g, the pump flow (Q_p) has the constant value of 95 L/min during movement of the cylinder. In the case of holding the cylinder, the fluid discharging from the pump releases to the tank through the center position of the directional valve. The pump pressure, P_p, and flow, Q_p, remain at 13.3 bar and 95 L/min, respectively (Figs. 3.8e and 3.8g). However, during running the cylinder, the pump pressure maintains at 200 bar which is the cracking pressure of the relief valve. This valve-controlled system has an excellent dynamic performance, since the cylinder speed profile is the same as the joystick input signal (see Figs. 3.8a and 3.8b).

In the CP system, during moving the cylinder, P_p is almost constant at 150 bar (Fig. 3.9e). The pressure compensator system adjusts the swash plate of the pump to keep P_p at 150 bar. Small fluctuations in P_p occurs once the pressure compensator is changing the pump swash plate. The CP system is a closed-center system, because during inactivity of the cylinder, $Q_p = 0$ L/min and the directional valve in the center position blocks the fluid discharging from the pump (shown in Fig. 3.9g).

In the LS system, as shown in Fig. 3.10e, P_p is always 20 bar higher than the load pressure (P_l). The LS control system adjusts this differential pressure (ΔP_{LS}). When the cylinder extends, P_l is equal to the piston side pressure (P_a). As the cylinder retracts, P_l is the rod side pressure (P_b) (see Fig. 3.10d). When the hydraulic cylinder is inactive, the directional valve is closed and the LS line is connected to the tank. P_l is zero and consequently, P_p remains constant at 20 bar. Smooth displacement and the constant speed of the cylinder during moving up and down the load indicates the good performance of the LS valve-controlled system (Figs. 3.10b and 3.10c). Comparing Fig. 3.8b with Figs. 3.9b and 3.10b, the cylinder controlled by OC system is shown slightly better in terms of dynamic response than the one controlled by LS and CP systems.

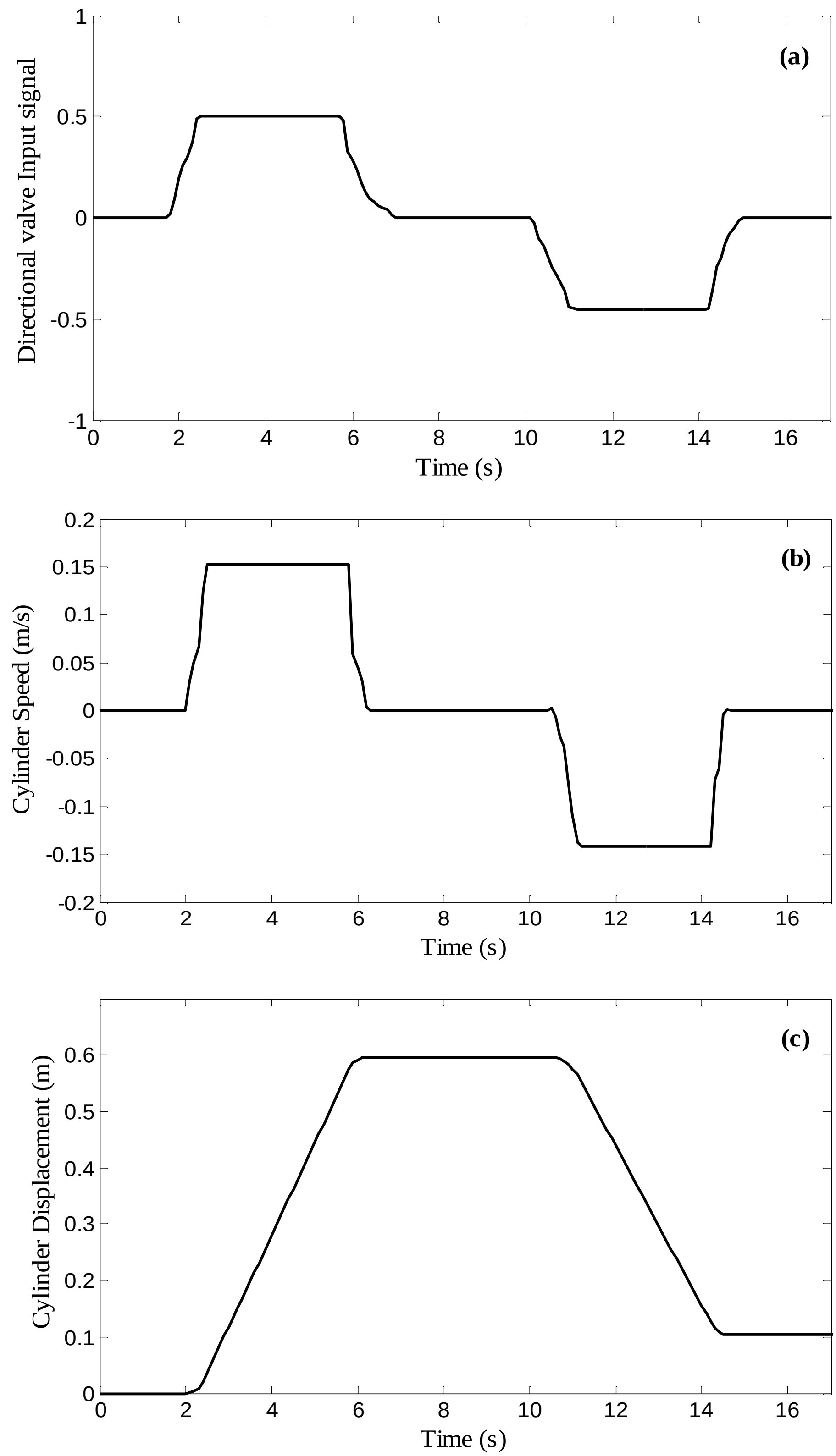
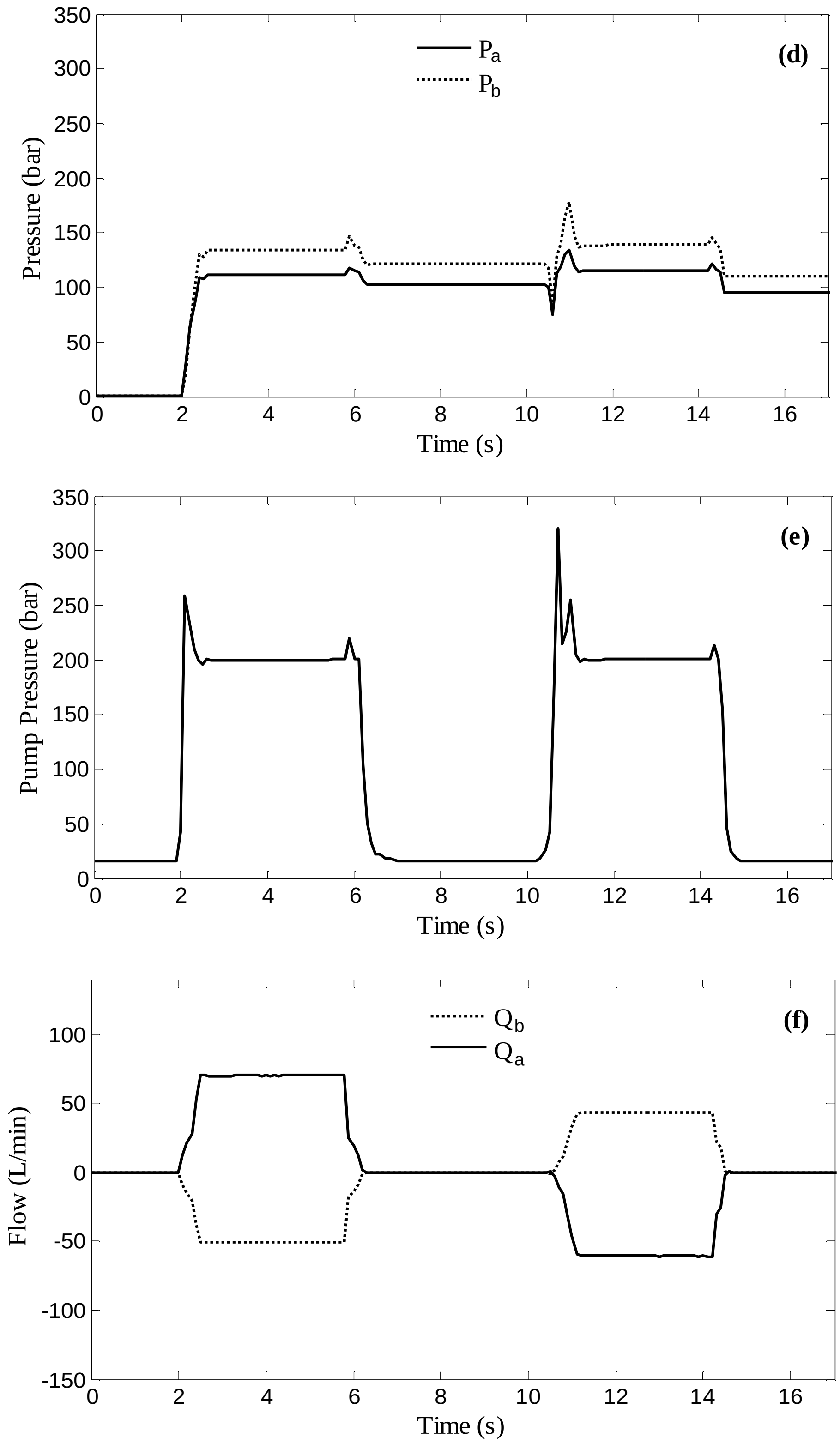


Fig. 3.8. Response of open-center (OC) system (energy consumption test).



(d)
Pa
Pb
Pressure (bar)
Time (s)
(e)
Pump Pressure (bar)
Time (s)
(f)
Qb
Qa
Flow (L/min)
Time (s)

Fig. 3.8. Contd.

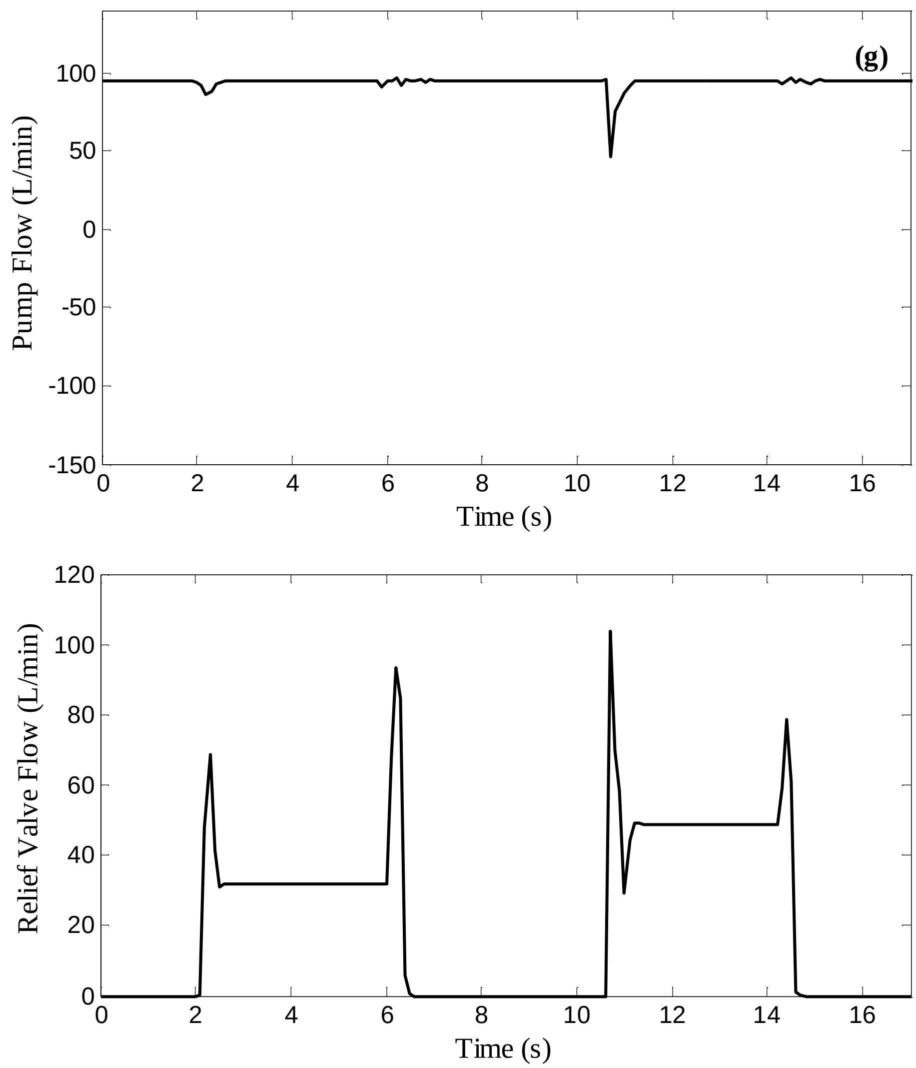

(g)
100
50
0
-50
-100
-150
Pump Flow (L/min)
0
2
4
6
8
10
12
14
16
Time (s)
120
100
80
60
40
20
0
Relief Valve Flow (L/min)
0
2
4
6
8
10
12
14
16
Time (s)

Fig. 3.8. Contd.

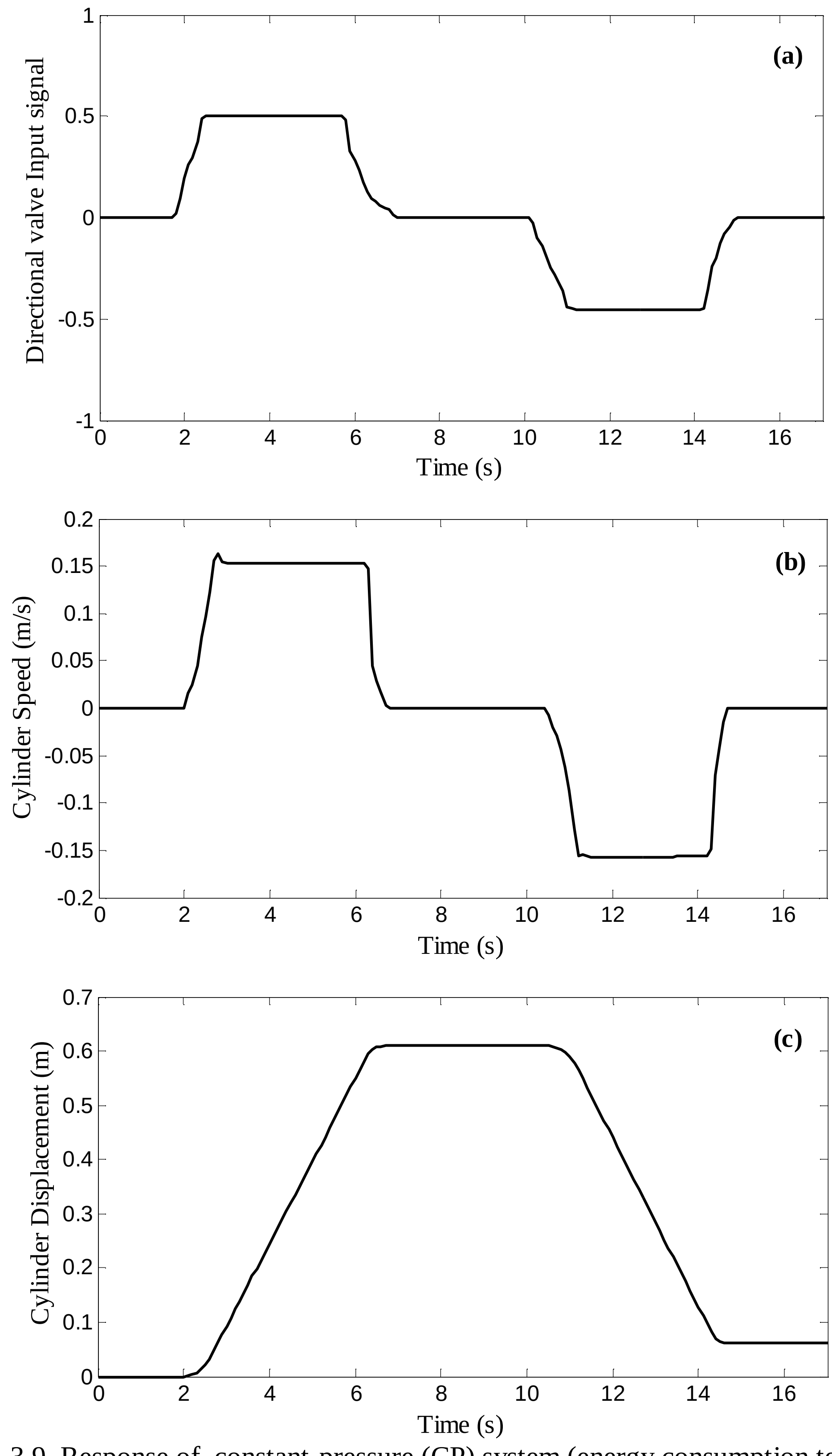


Fig. 3.9. Response of constant-pressure (CP) system (energy consumption test).

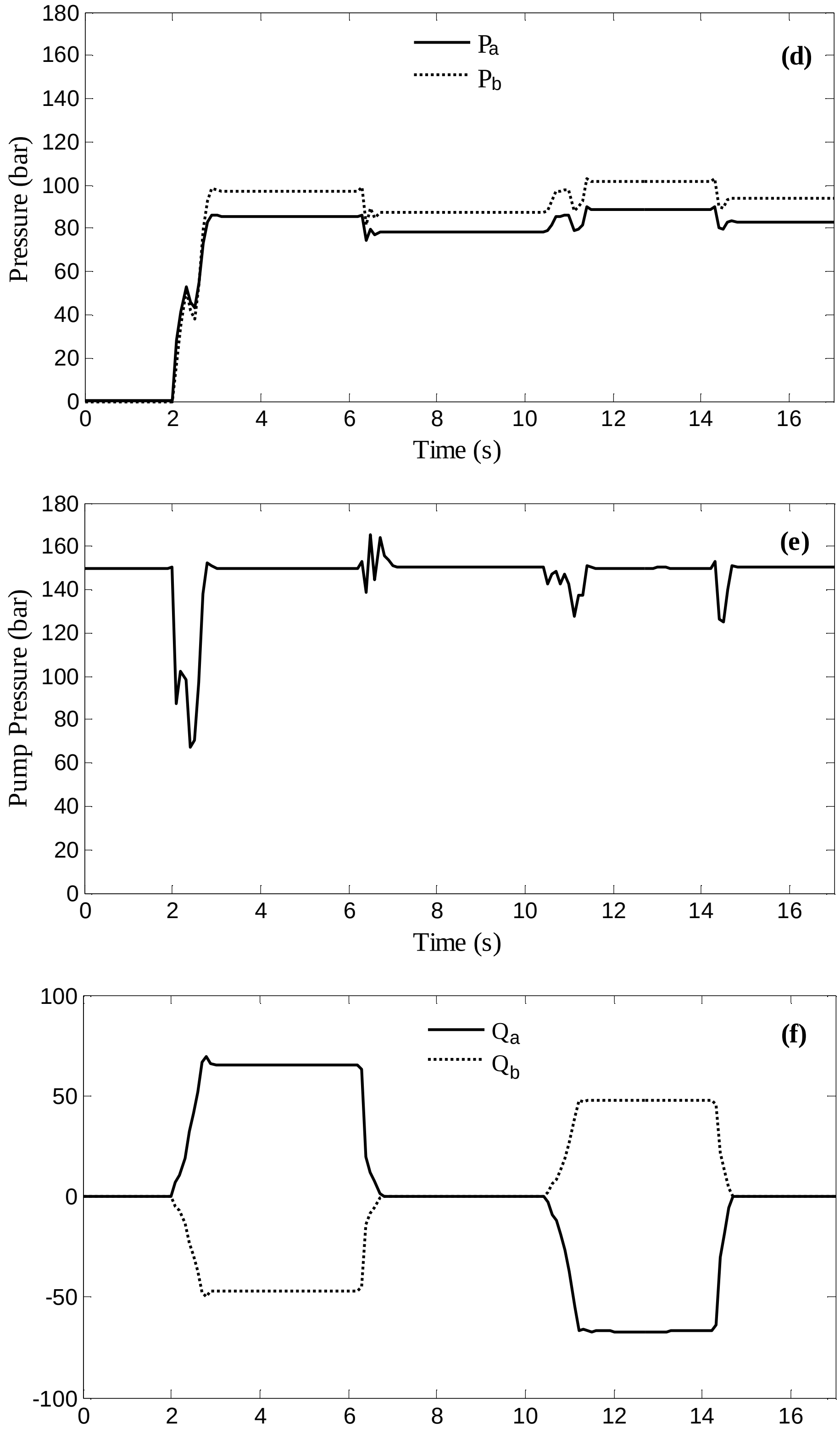


Fig. 3.9. Contd.

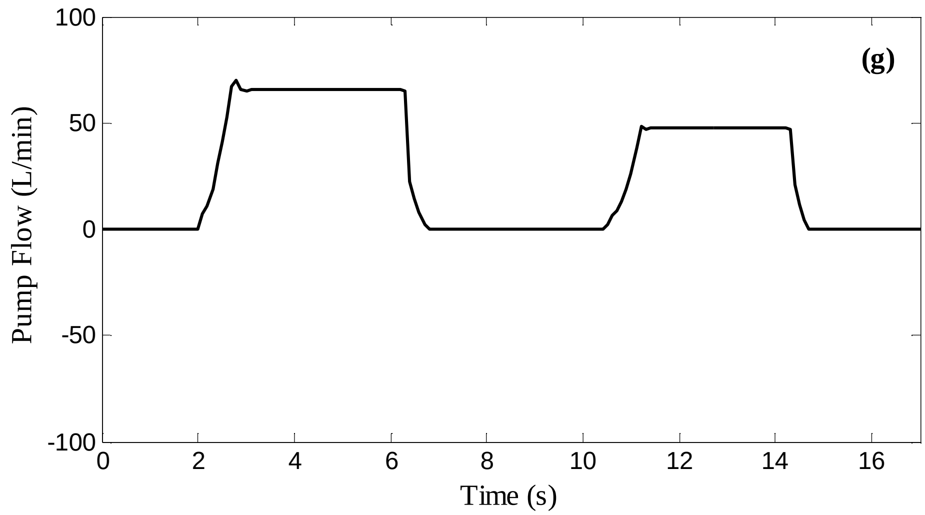


Fig. 3.9. Contd.

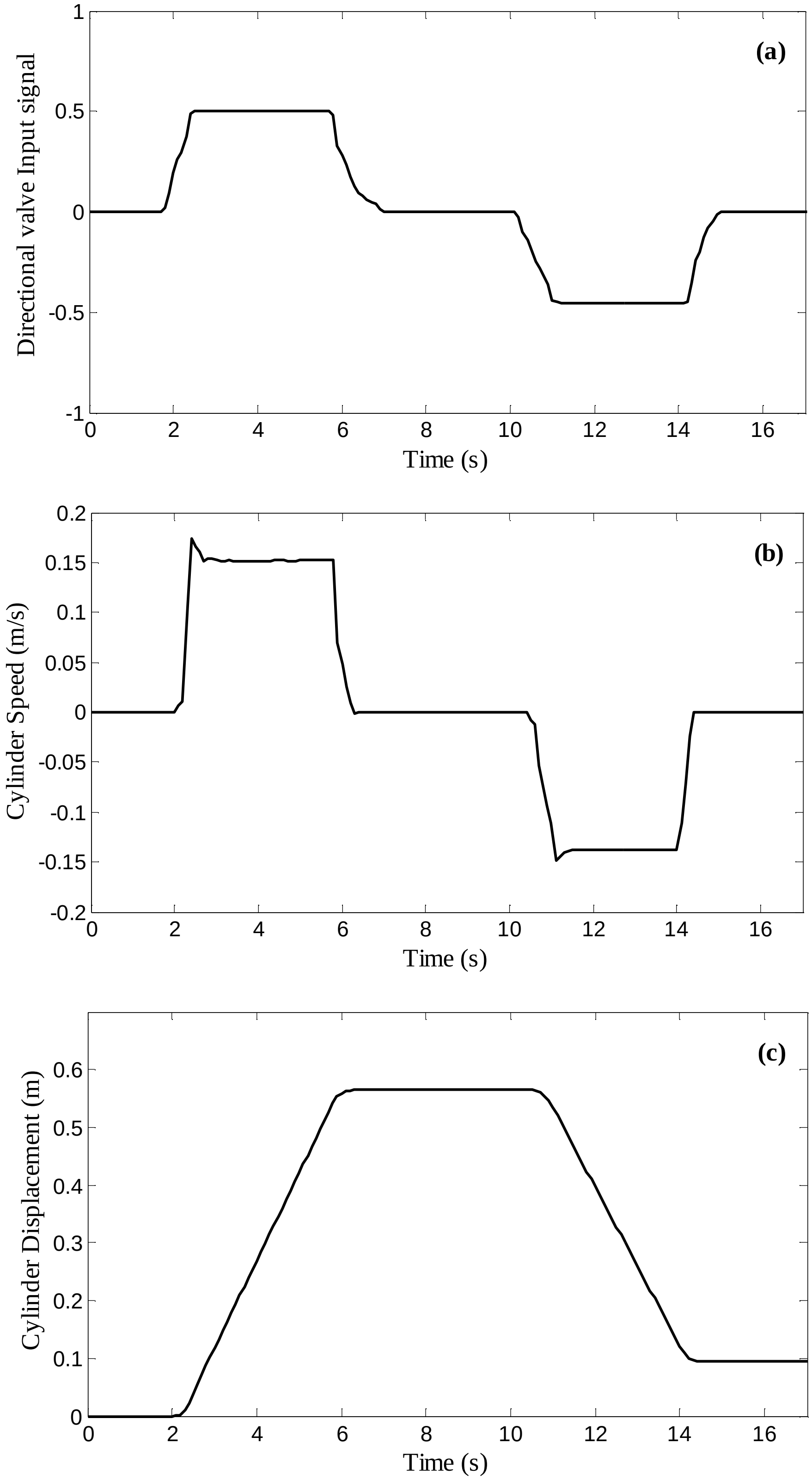


Fig. 3.10. Response of load sensing (LS) system (energy consumption test).

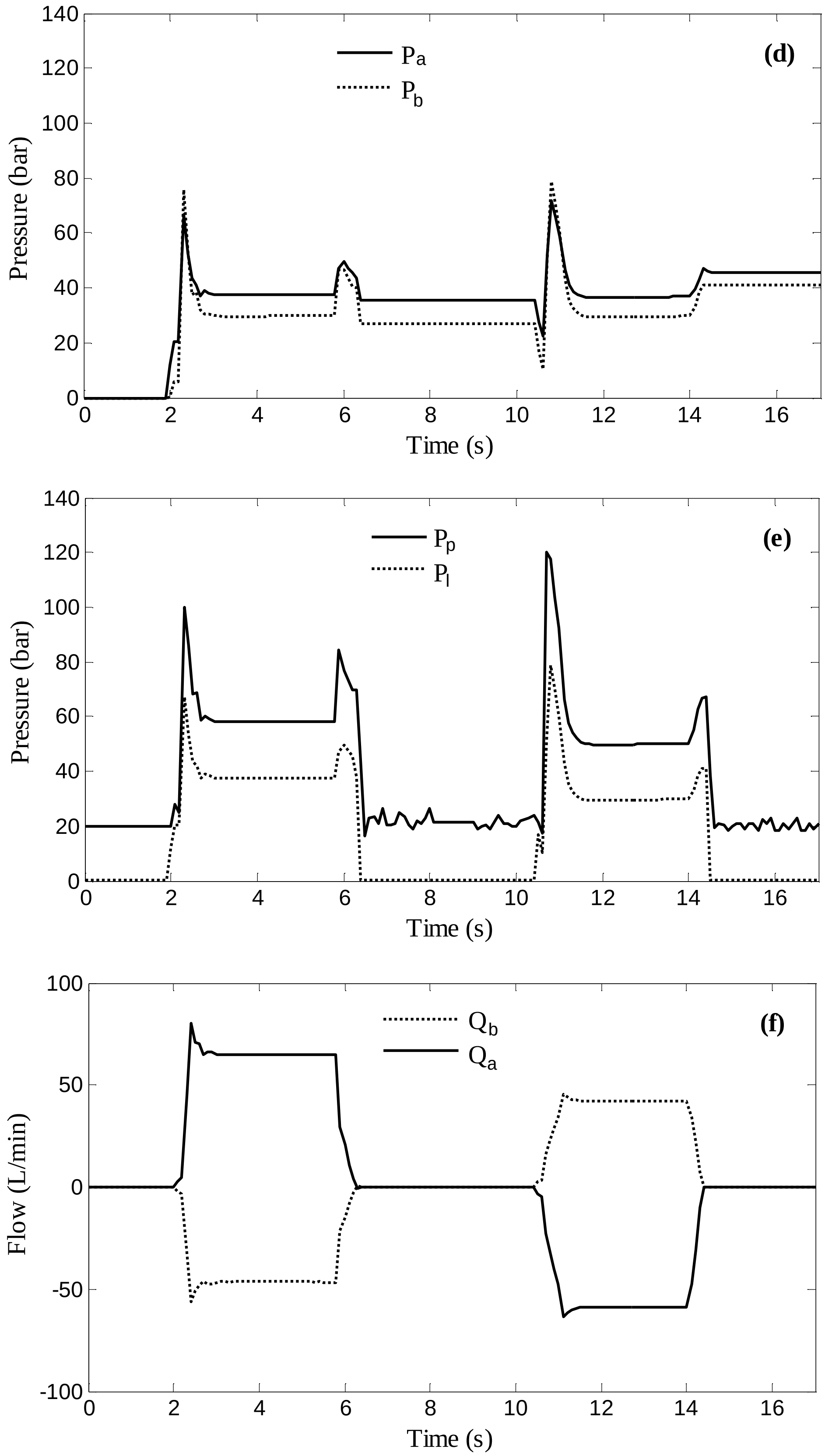


Fig. 3.10. Contd.

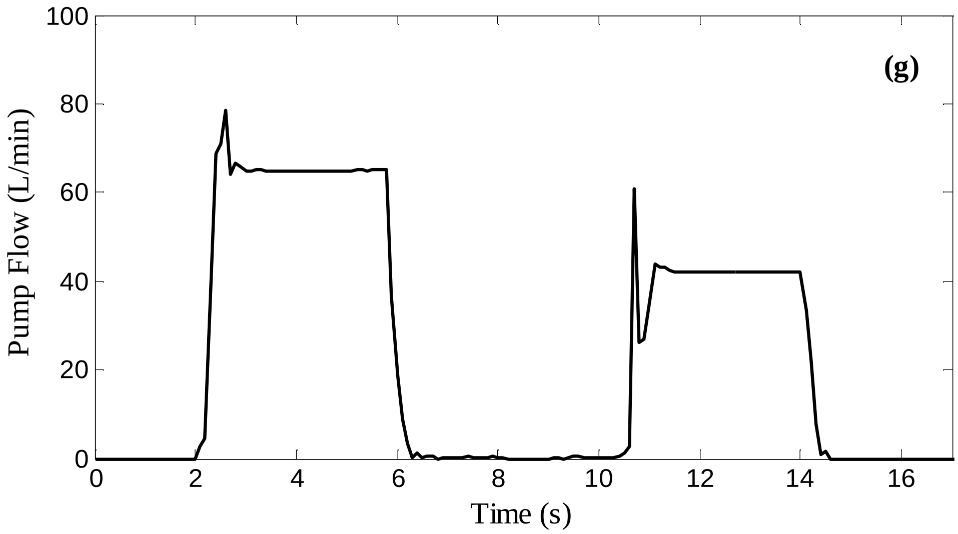


Fig. 3.10. Contd.

3.2.3. Response of Pump-Controlled Systems

In the DC system, during the cylinder extension, the cylinder speed is constant (see Fig. 3.13b). With reference to Fig. 3.13g, Q_{poc2} is positive. The POC valve connected to the rod side of the cylinder (POC2) compensates for the differential flow between the piston and rod chamber of the cylinder ($Q_a - Q_b$).

During the cylinder retraction, with reference to Fig. 3.13b, the cylinder speed fluctuates at the beginning of the cylinder retraction. This is related to influence of the system load inertia on the cylinder motion. Indeed, the load weight influences on the cylinder speed before the pump displacement takes the control of the cylinder. At this moment, the load weight causes reduction in P_b. Since the POC2 pilot line keeps the POC2 open, amount of flow through the POC2 release to the rod side of the cylinder, as the pump starts the flow. Fig. 3.13g shows this sudden flow entrance by positive sign. Once the pump takes the control of the cylinder, the flow is removed from the rod side of the cylinder through the POC2. During cylinder extension and retraction, The pump flow at the piston side of the cylinder (Q_{pa}) is same as Q_a, since the POC valve connected to the piston side of the cylinder (POC1) is closed (see Figs. 3.13f and 3.13g). More details on flow circulation of this hydraulic system are shown in Fig. 2.9 (see modes 1 and 4). The effect of load inertia on the cylinder dynamic behavior becomes more evident when increasing load on the cylinder. Fig. 3.11 shows improper cylinder displacement controlled by the mentioned hydraulic system while moving 3000 kg load instead of 1142 kg.

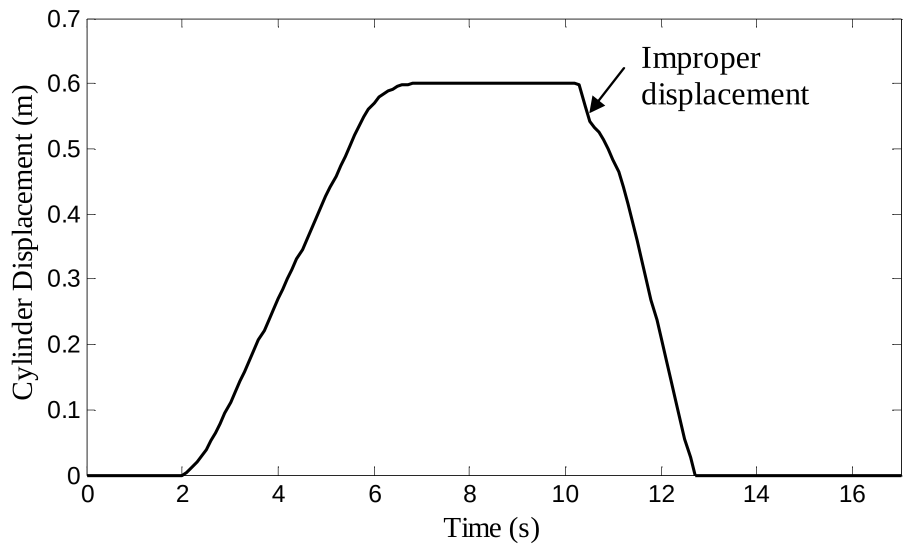


Fig. 3.11. Cylinder dispalcement under 3000 kg load, controlled by DC pump-controlled system.

In Wendell system, cylinder pressures are higher compared to the DC system (see Fig. 3.14d). During the cylinder extension, the load weight builds the piston side pressure up to the required pilot pressure to open the POC4 (P_{poc4}). Then, the flow, coming out of the rod side of the cylinder, is directed to the inlet of the pump thorough the POC3. During the cylinder retraction, the load weight causes reduction in P_b which becomes lower than the required pilot pressure to open the POC3 (P_{poc3}). As a result, the POC3 becomes closed until the pump increases P_b up to P_{poc3}.

The POC3 and POC4 make the cylinder displacement independent of the load value. Fig. 3.12 shows the cylinder displacement in the case of moving 3000 kg load. As observed, the cylinder displacement is same as the one holding 1142 kg load (Fig. 3.14c). As compared to Fig. 3.11, the influence of system inertia on the cylinder, controlled by Wendell system, is lower than the one controlled by DC system. However, the POC3 causes continued oscillations in the cylinder displacement during moving down the load (see Fig. 3.12).

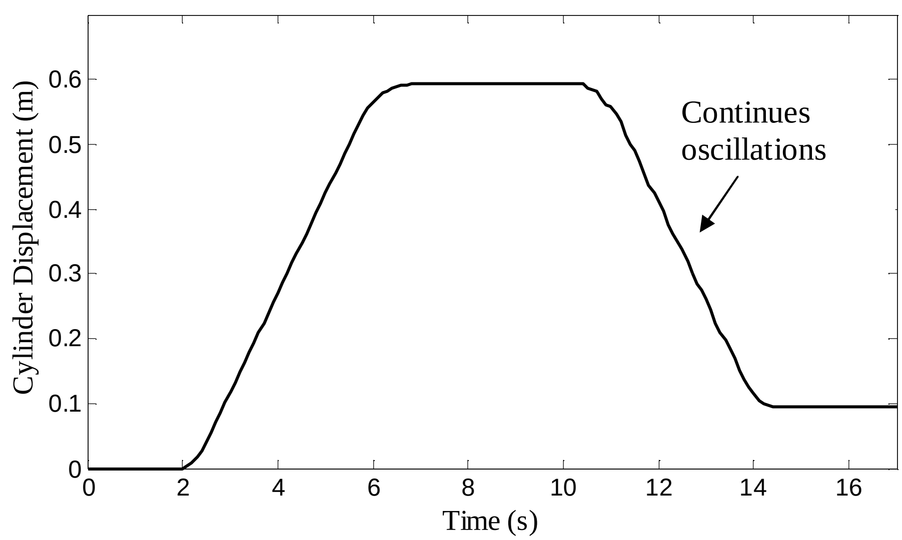


Fig. 3.12. Cylinder displacement under 3000 kg load controlled by Wendell system.

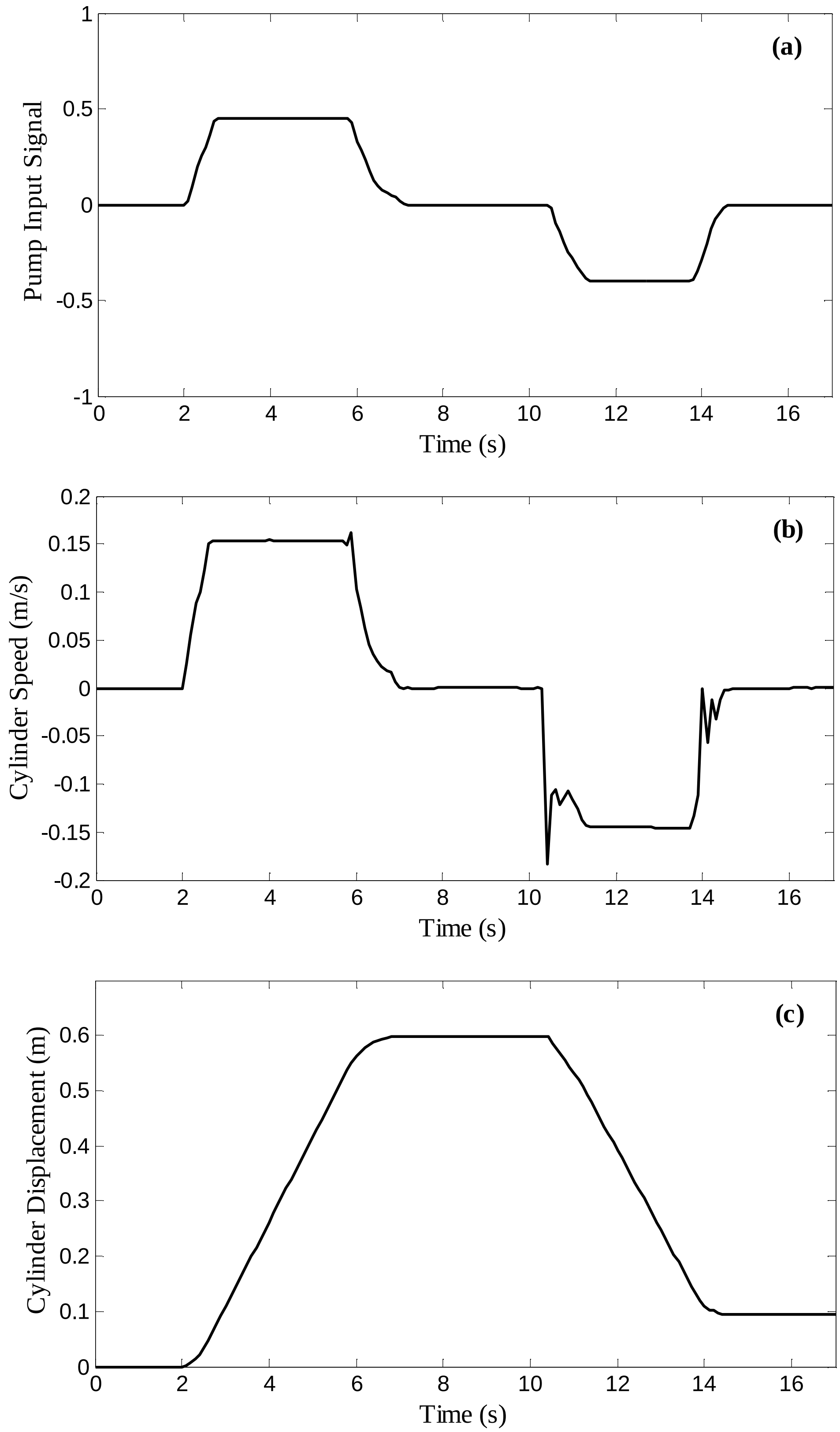


Fig. 3.13. Response of DC system (energy consumption test).

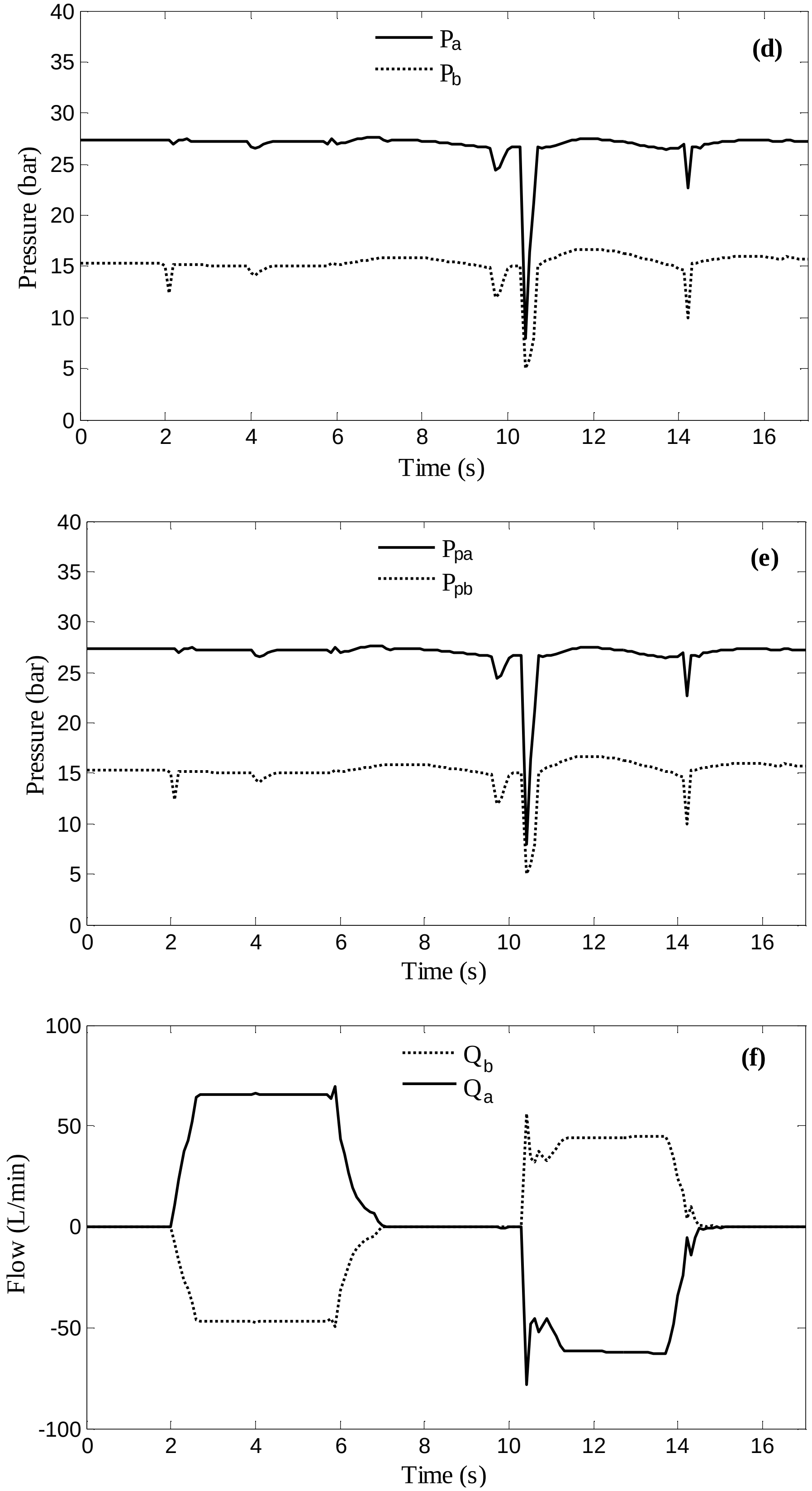


Fig. 3.13. Contd.

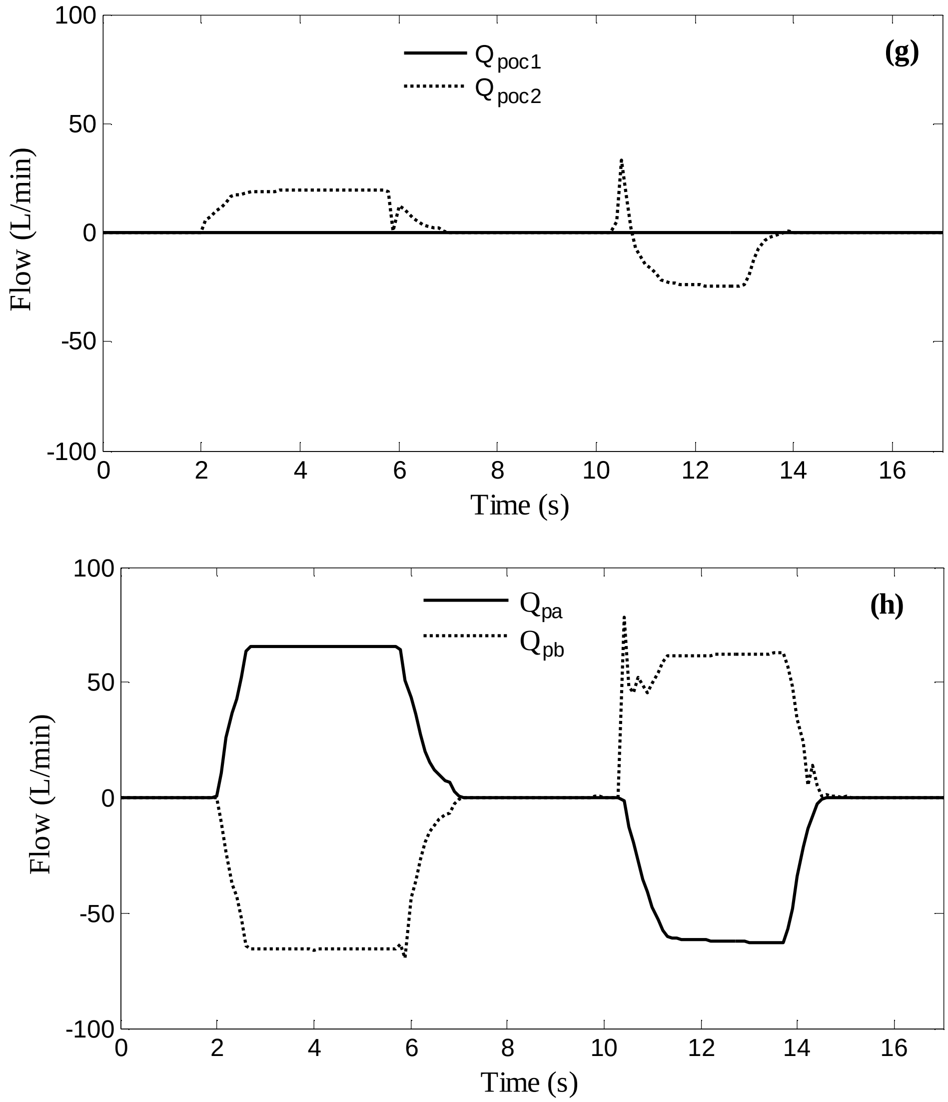


Fig. 3.13. Contd.

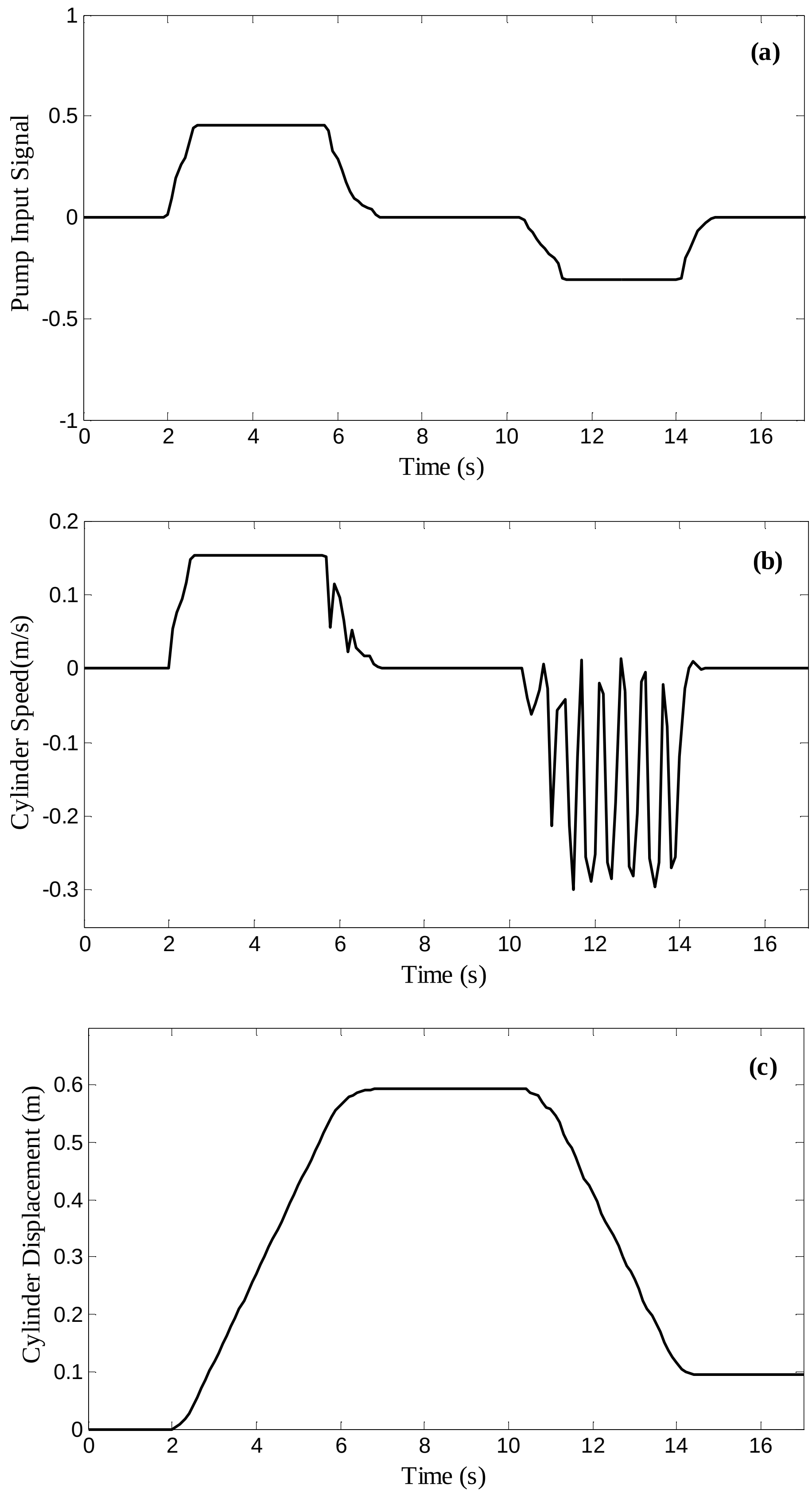


Fig. 3.14. Response of Wendell system (energy consumption test).

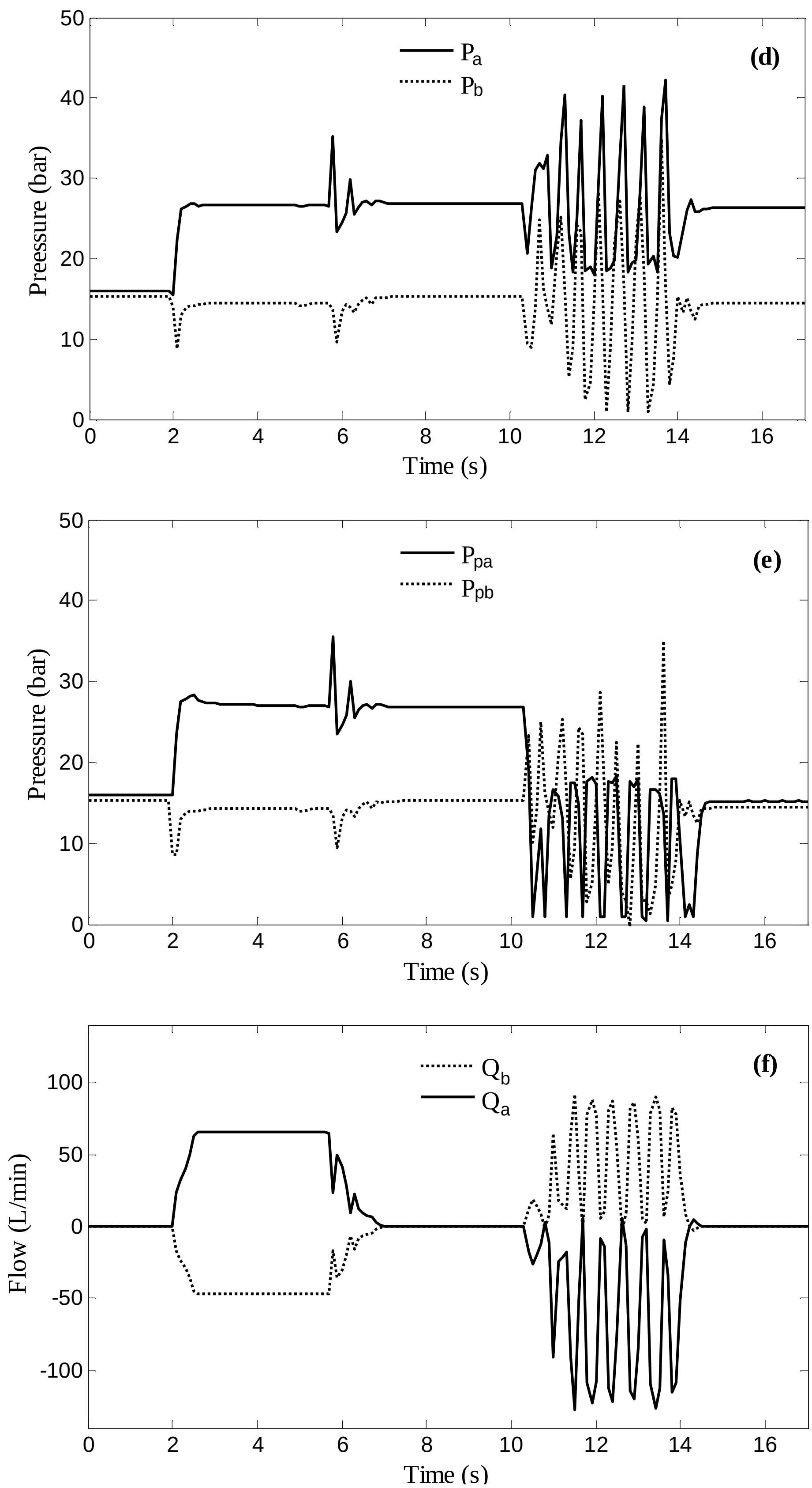


Fig. 3.14. Contd.

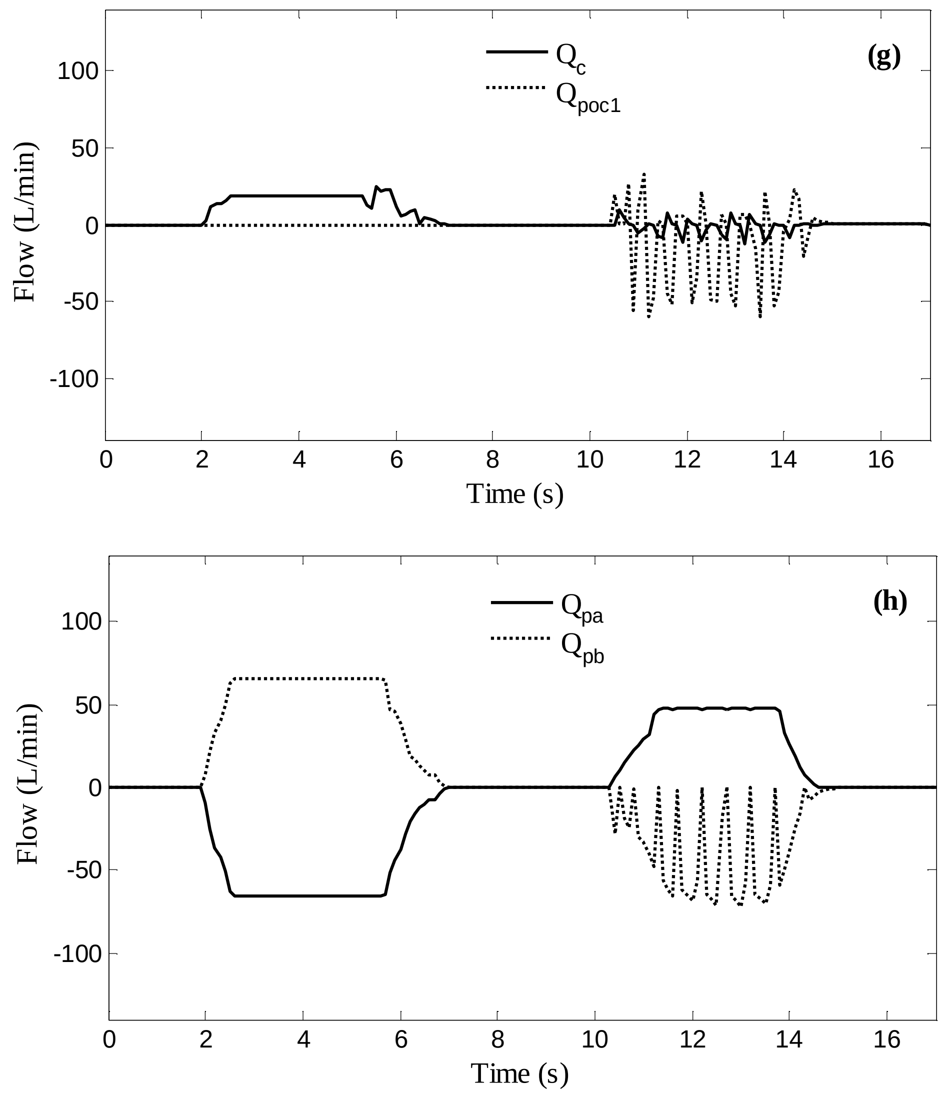


Fig. 3.14. Contd.

3.2.4. Discussions

Based on the equations defined in Section 3.1.2, the directional valve loss for valve-controlled systems, pump loss, useful work and the efficiency of each hydraulic system were calculated and shown in the Table 3.1. More details on calculation of the energy are given in Appendix A. As seen in Table 3.1, the LS system has the highest efficiency among valve-controlled systems, because the LS pump displacement is adjusted by the load pressure. The CP system showed higher efficiency than the OC. Efficiency of the

OC system is very low (2.15%), since the pump produces constant flow to the system. As seen in Fig. 3.15, the OC system shows considerable power consumption, when the cylinder was inactive. LS system exhibits lower power consumption than the OC and CP systems.

All valve-controlled systems have the considerable energy loss due to the directional valve inefficiency. High difference in power consumption of the LS system and pump-controlled systems is shown in Fig. 3.16. As observed, during moving down the load, the power consumption of the Wendell system is higher than that of the DC system. The reason is that the POC3 of Wendell system does not allow the gravitational load pressure to help the pump run. In addition, in Wendell system, the pump must consume the energy to increase P_b up to P_{poc3} to open the POC3, during moving down the load. Fig. 2.12 visualizes this behavior (during retract overrunning mode).

Table 3.1. Results of energy consumption of hydraulic systems.

Hydraulic system	Type	Pump loss (kJ)	Directional valve loss (kJ)	Relief valve loss (kJ)	Useful work (kJ)	E_{eng} (kJ)	η_{cy} (%)
Valve-controlled systems	OC	42.62	128.5	112.38	6.6	306.00	2.15
	CP	26.23	51.4	0	6.6	108.61	6.08
	LS	13.84	39.43	0	6.6	54.42	12.13
Pump-controlled systems	Wendell	8.50	-	0	5.57	22.22	24.10
	DC	3.37	-	0	7.13	14.23	50.10

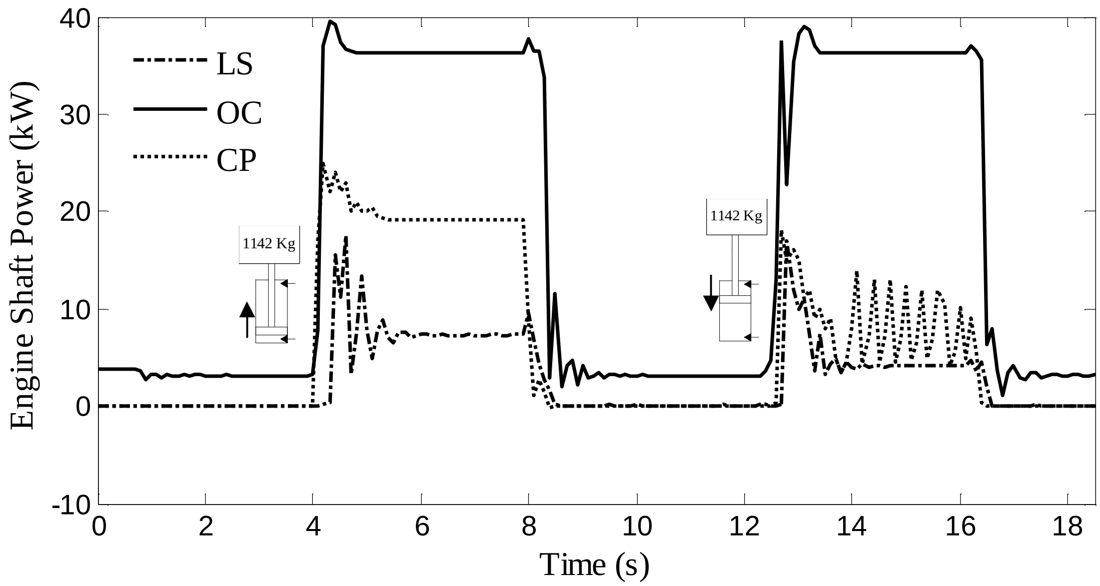


Fig. 3.15. Power consumptions of valve-controlled systems.

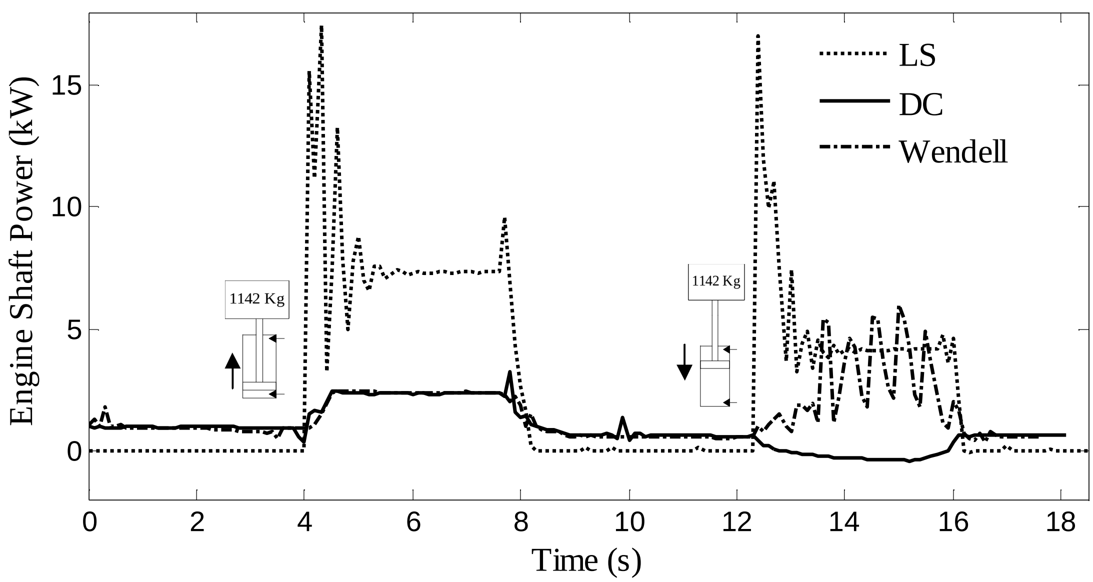


Fig. 3.16. Power consumptions of LS, DC and Wendell systems.

Figure 3.17 shows the Sankey diagram of DC and LS systems, during moving up the load. In the DC system, the useful work is made from the energy consumption of the pump and the low-pressure system. More detailed information about the Sankey diagram is given in Appendix B. Energy distributions for OC, CP, LS, and DC systems, during moving up the load ($4\ s < t < 8\ s$), are shown by the charts in Fig. 3.18. Note that the line energy loss is assumed to be negligible. While a few percentage of the energy consumption of valve-controlled systems transformed into useful work, the useful work in the DC system is 64.8% of the engine total energy transmission.

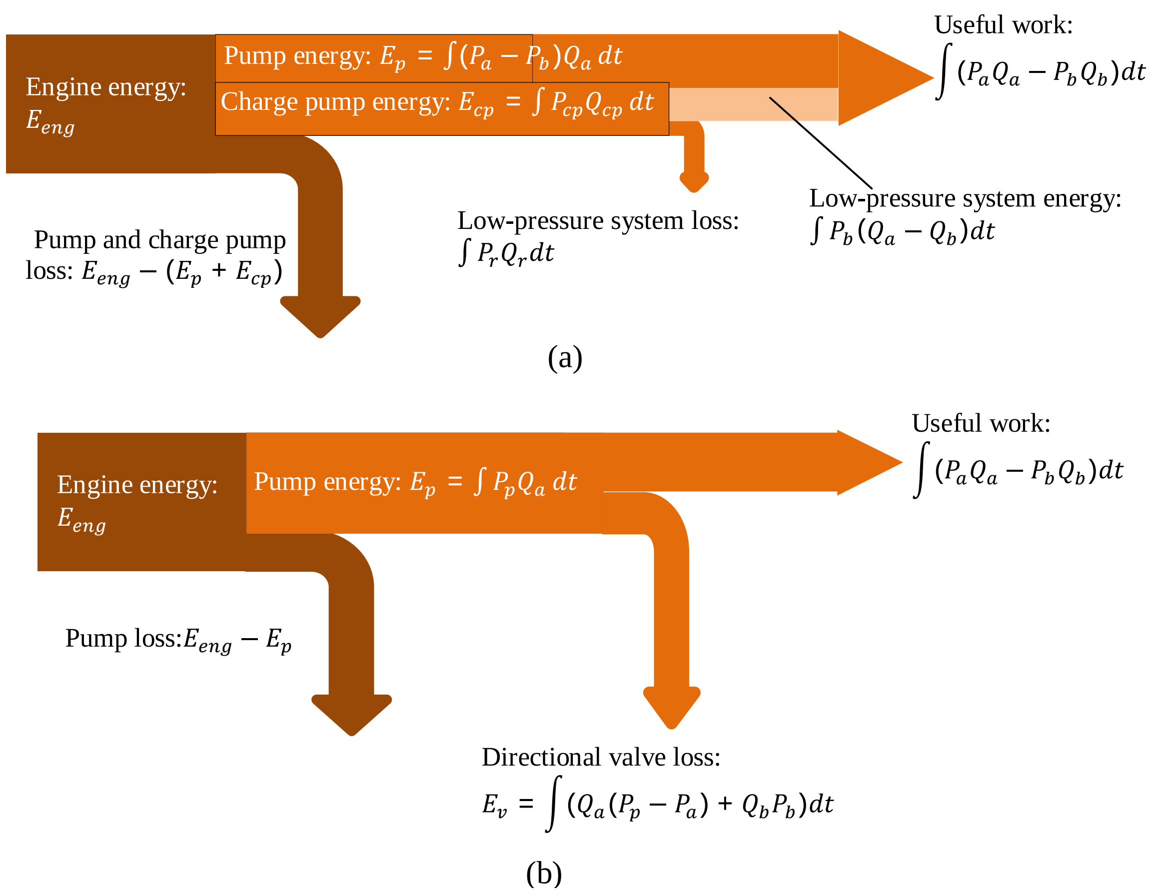


Fig. 3.17. Sanky diagram of (a) DC system and (b) LS system, during moving up the load.

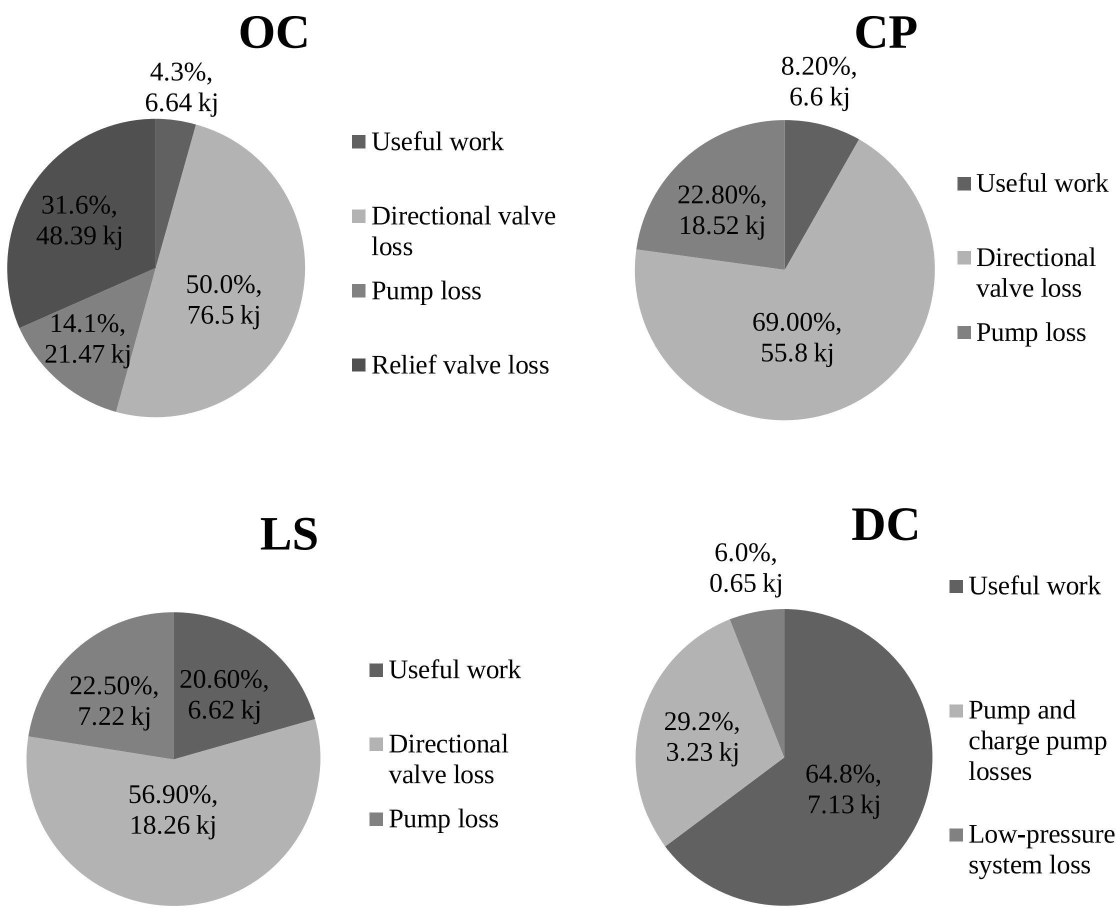


Fig. 3.18. Measured energies and distributions for OC, CP, LS, and DC systems, during moving up the load.

3.2.5. Summary

The energy consumption of the valve-controlled systems is considerably high because of the energy loss in the directional valve. Among the tested valve-controlled systems, the LS system exhibited lower energy consumption, because in the LS system, the pump flow rate is regulated by the load requirement. Pump-controlled systems evaluated in this thesis, showed lower energy consumption than the LS valve-controlled system. The energy consumption of the Wendell system was higher than that of the DC system. In the

Wendell system, POC3 prevented the load pressure to run the pump and consequently increased the energy consumption of this system.

3.3. Energy Recovery in LS Valve-Controlled Systems

The LS valve-controlled system has lower energy consumption among the other types of valve-controlled systems. However, the directional valve in the LS system still causes undesirable energy consumption. Recently, some LS circuits incorporate electrical energy recovery methods to reduce the energy consumption of the system. Fig. 3.19 shows a typical example that is taken from the design from Naruse *et al* [30]. This electrical energy recovery system was designed for an excavator. The system has the ability to recover the potential energy of the load, transform it into electrical energy and reutilize it as a power source for the electrical motor. With reference to Fig. 3.19, in moving down the load, the on-off valve moves to the left position so that the pressurized fluid from the piston side of the cylinder is guided to the variable pump. The pump runs the generator, and produces electrical energy. Next, the electrical energy is stored in battery and consumed by the electrical motor.

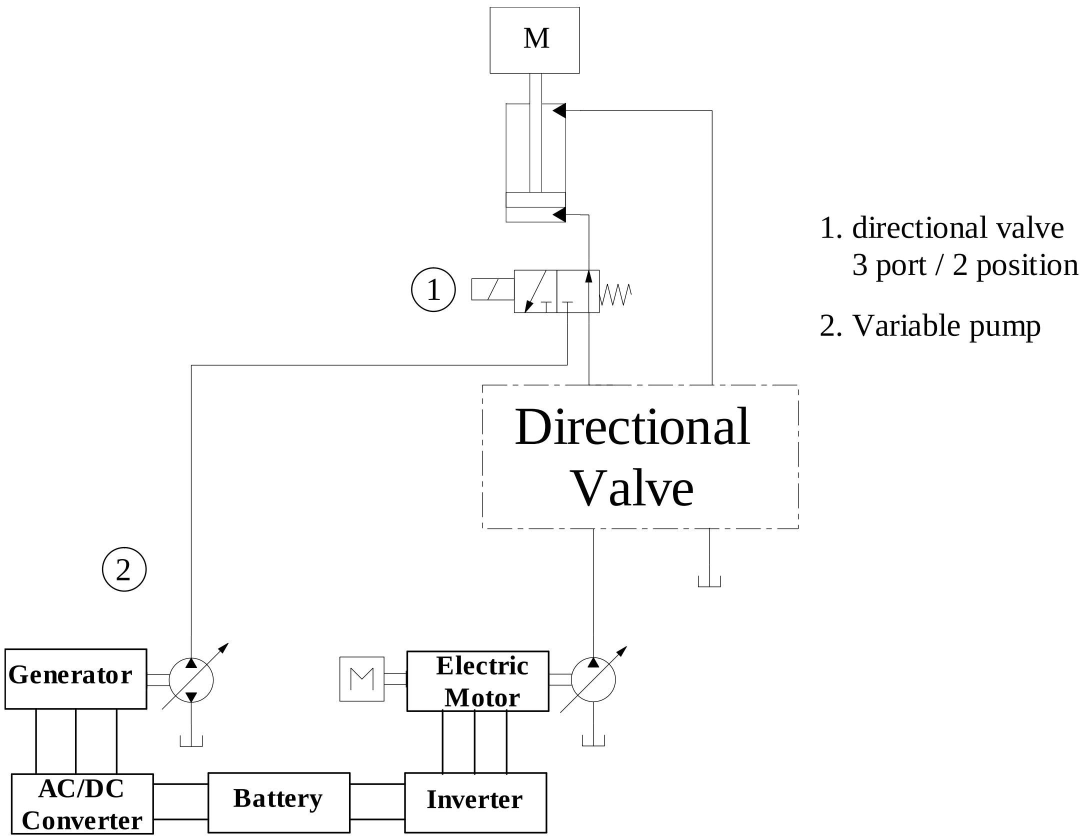


Fig. 3.19. Hydraulic system with electrical energy recovery controlling a single-rod hydraulic cylinder.

The circuit, shown in Fig. 3.19 is modified with hydraulic energy recovery components. Fig. 3.20 shows schematics of the LS valve-controlled system equipped with the hydraulic energy recovery. In this design, a hydraulic motor is coupled with the main pump. A high-pressure accumulator (HPA) is connected to the inlet of the hydraulic motor and main pump through a directional valve (V2). The pressurized fluid, coming from the piston side of the cylinder, runs the hydraulic motor. If the pressurized fluid, entering the hydraulic motor, is high enough to be stored in the accumulator, V1 closes to guide the pressurized fluid to the accumulator. During the next working cycle, V2 moves to the right position in order to release the accumulator fluid to the inlet of the main pump. The recovered piston side pressurized fluid becomes a power source to run the main pump.

If the piston side pressure is not high enough for energy recovery, V1 moves to the right position to release the pressurized fluid into the tank. Since the hydraulic motor is coupled with the main pump, portion of the power required to run the main pump is provided by torque transmission of the hydraulic motor. As a result, the gravitational energy is regenerated to the system.

With this method, the energy consumption of the entire systems will significantly decrease. This hydraulic system is simulated based on the energy consumption test defined in Section 3.2. During moving up the load, the system works the same as the LS system without energy recovery. During moving down the load, because the load pressure at the piston side chamber is not high enough to be stored in the accumulator, the V1 is opened to release the fluid, discharging from hydraulic motor to the tank.

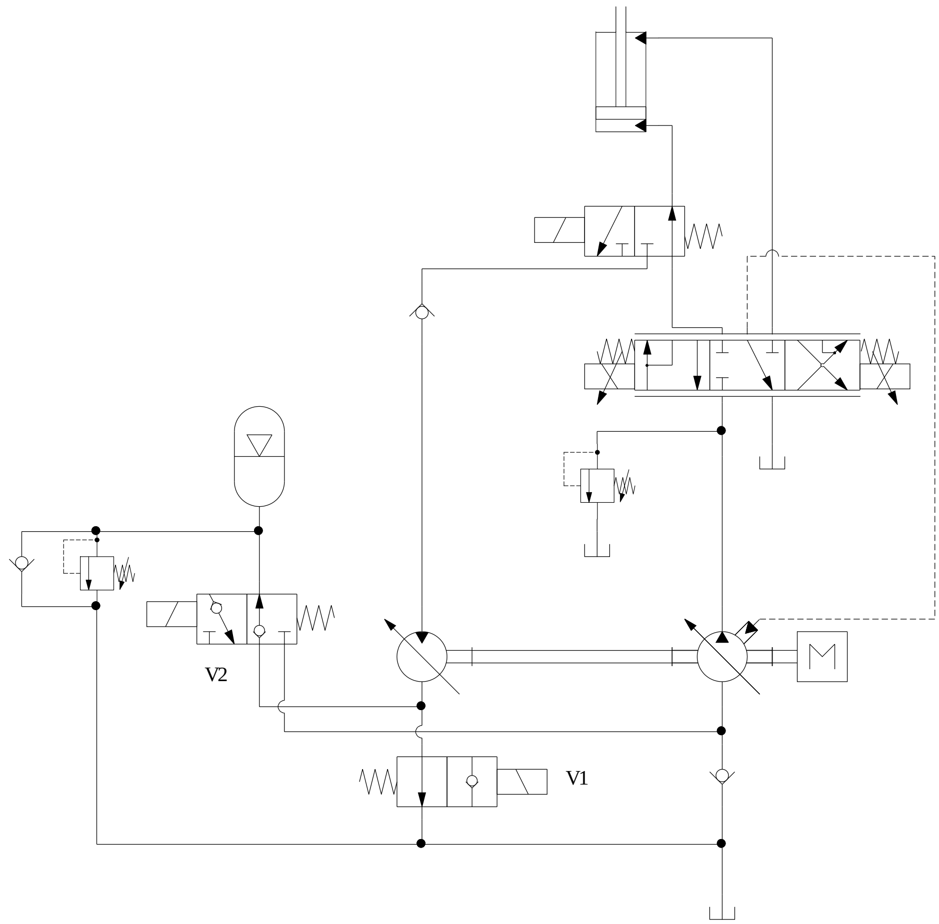


Fig. 3.20. Load Sensing (LS) system with hydraulic energy recovery.

The control of the hydraulic motor and directional valve are performed by only one joystick. The hydraulic motor receives the joystick signal only when V2 is open. During moving down the load, valve and hydraulic motor signals are adjusted in such a way that the cylinder moves down smoothly with constant speed as the hydraulic motor and directional valve control the displacement of the cylinder (see Figs. 3.21b and 3.21c).

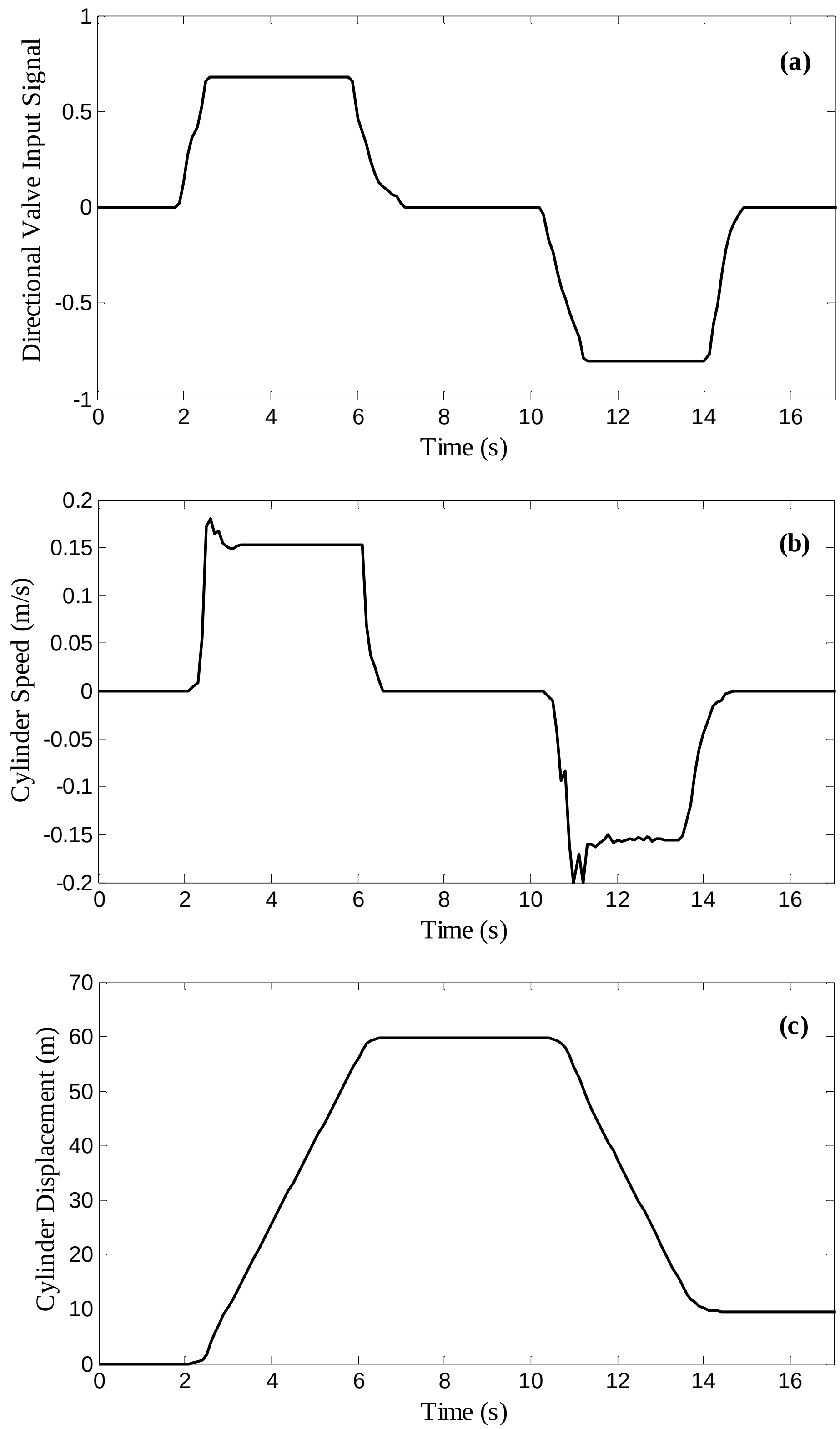


Fig. 3.21. Response of LS with hydraulic energy recovery (energy consumption test).

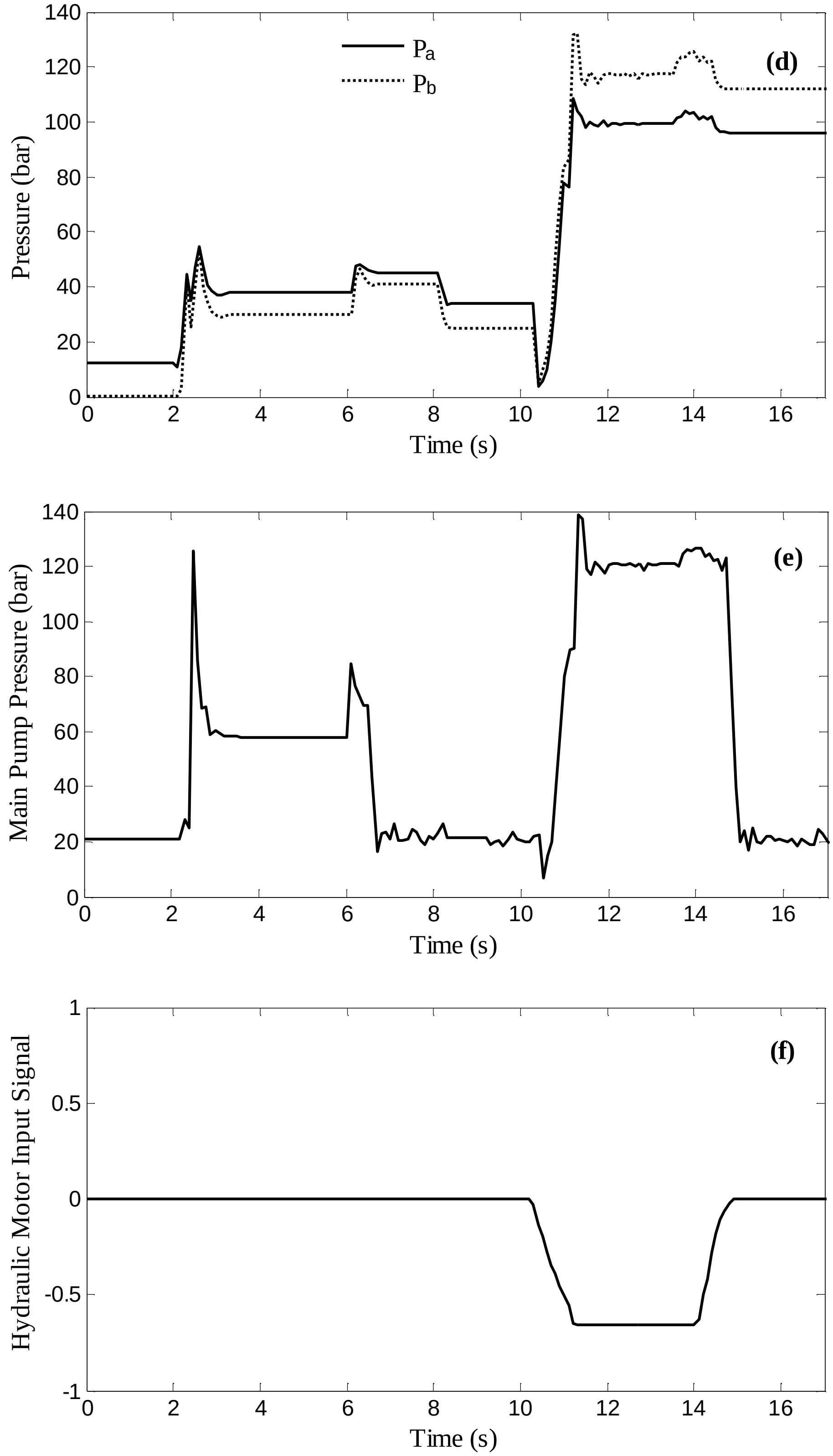


Fig. 3.21. Contd.

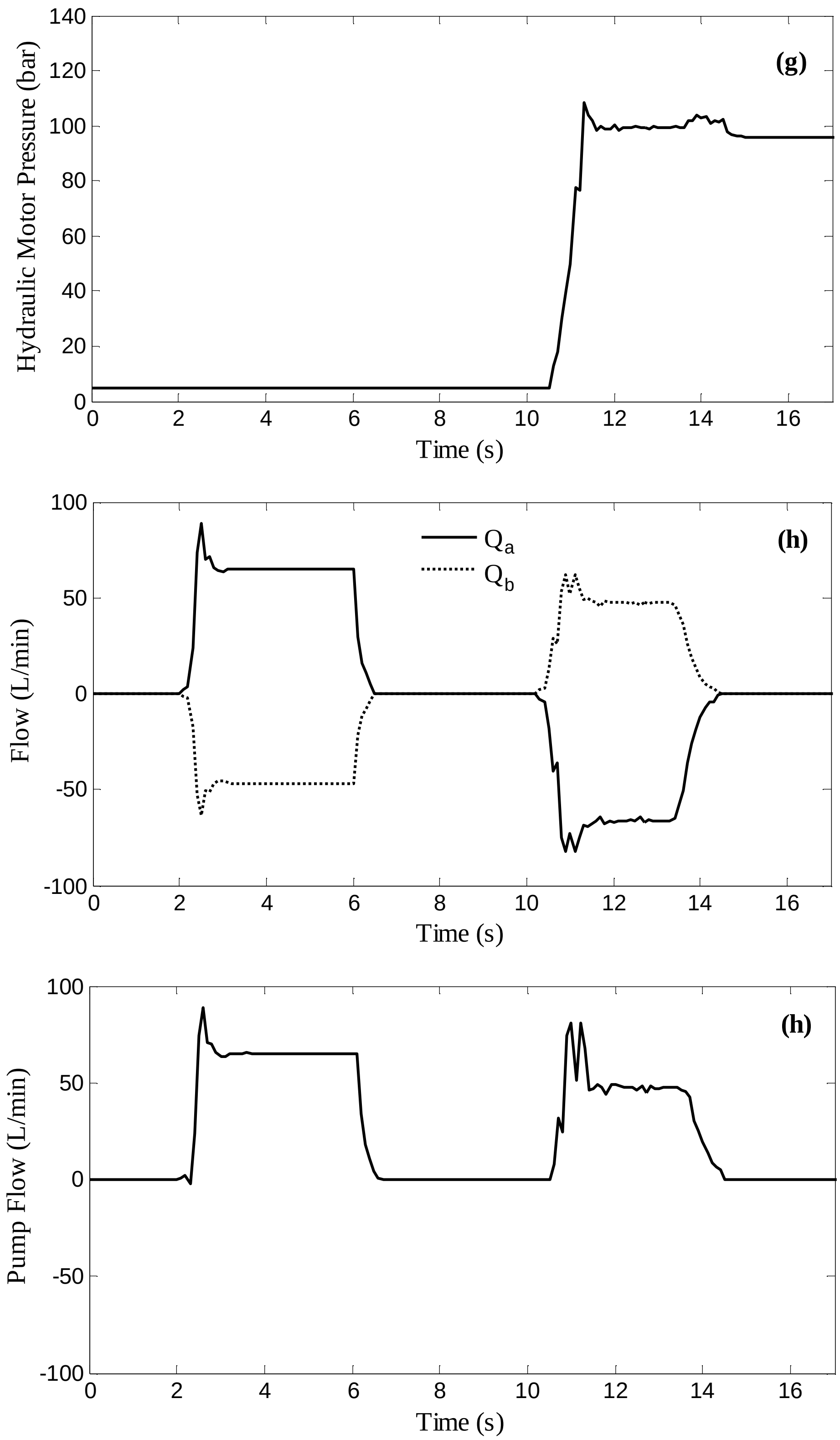


Fig. 3.21. Contd.

Comparison between power consumptions of the LS and LS with hydraulic energy recovery are shown in Fig. 3.22. As observed, during moving down the load, the engine shaft power of the LS system with energy recovery is lower than the typical LS system. Results indicated that the efficiency of the LS and LS with energy recovery were 12.13% and 19.5 %, respectively.

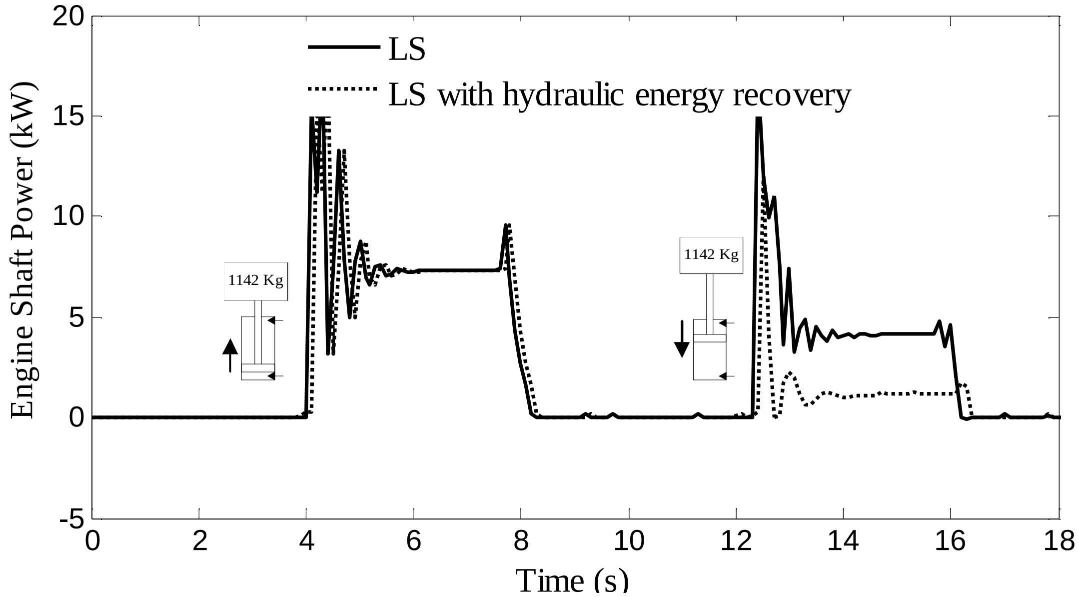


Fig. 3.22. Power consumption of LS and LS with hydraulic energy recovery.

4. DYNAMIC BEHAVIOUR ANALYSES

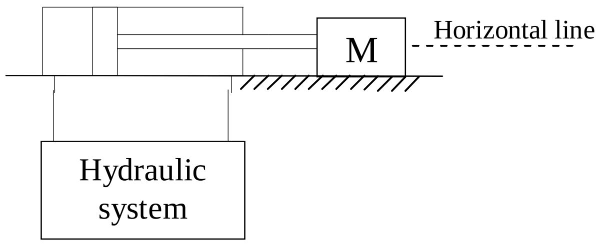
4.1. Case Studies

In order to evaluate the dynamic performance of the mentioned hydraulic systems, four case studies are designed (see Table 4.1). In all case studies, the joystick controls the spool displacement of the directional valve (in valve-controlled systems) or the swash plate of the pump (in pump-controlled systems) as shown in Fig. 3.5b. As mentioned in Section 3.2.1, the operator manipulates the joystick in such a way that the cylinder extends, and retracts with preferably constant speed at most times.

In the first case study, the cylinder extends and retracts along the horizontal line and moves the load of 1142 kg. As seen, no effective force is applied to the rod end of the cylinder. The friction between load and ground is assumed to be negligible. The purpose is to simulate the high inertia load as in a mobile machine. In Case study 2, the gravitational force is applied to the rod end of the cylinder, when rod side faces downward. In Case study 3, the cylinder extends to move the load from the beginning (0%) to the entire cylinder stroke (100%) while θ changes proportionally from 60° to -60°. During the retraction of the cylinder, θ rotates from -60° to 60° which is proportional to the cylinder displacement.

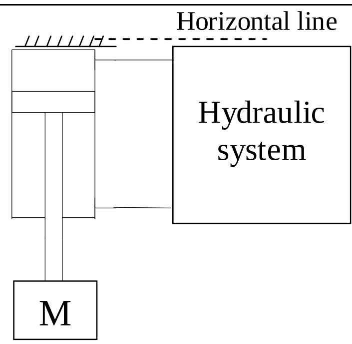
Table 4.1. Case studies

Case study 1



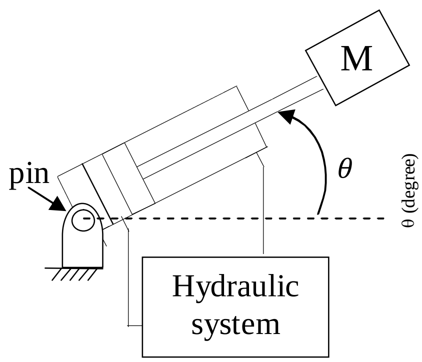
Cylinder extends and retracts a 1142 kg load along the horizontal line.

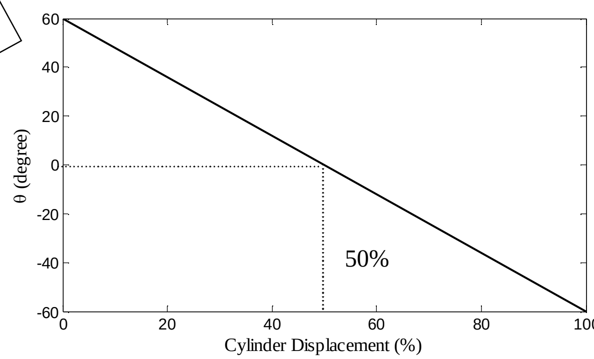
Case study 2



Gravitational force is applied to rod end of cylinder, when rod side faces downward with a 1142 kg load.

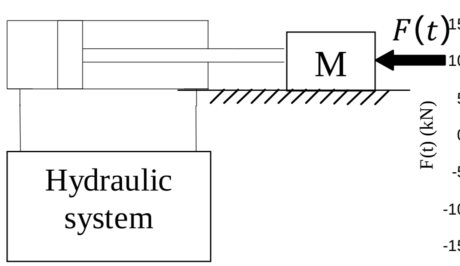
Case study 3

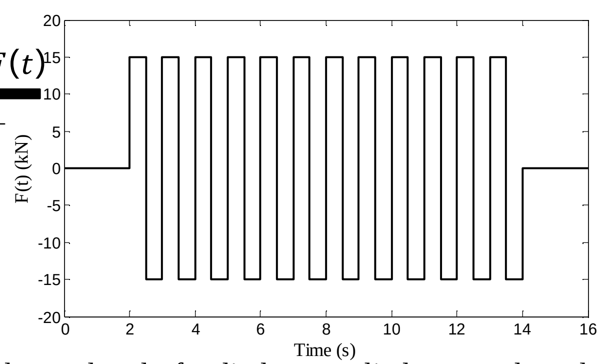




Cylinder extends to move a 1142 kg load from beginning (0%) to entire cylinder stroke (100%) while θ changes proportionally from 60° to - 60°.

Case study 4





A switching force is applied to rod end of cylinder as cylinder extends and retracts a 1142 kg load on horizontal line. $F(t)$ changes periodically over time from -15 kN to 15 kN.

In Case study 3, the simultaneous angular and the linear displacement of the cylinder create four-quadrant modes shown in

Fig. 4.1. During the cylinder extension, the system is in extend resistive mode. After passing 50% of the cylinder displacement ($\theta = 0°$), the system enters extend overrunning mode in which the effective force ($Mg \sin\theta$) and the cylinder velocity are at the same direction. The system operates under retract resistive mode, when the cylinder retracts and rotates from $\theta = -60°$ to $\theta = 0°$. Retraction and rotation of the cylinder, from $\theta = 0°$ to $\theta = 60°$, bring the system into retract overrunning mode. This case study evaluates whether a hydraulic system can control the cylinder properly at the moment of changing the modes. For instance, the movement of excavator stick brings the hydraulic system in into four-quadrant modes.

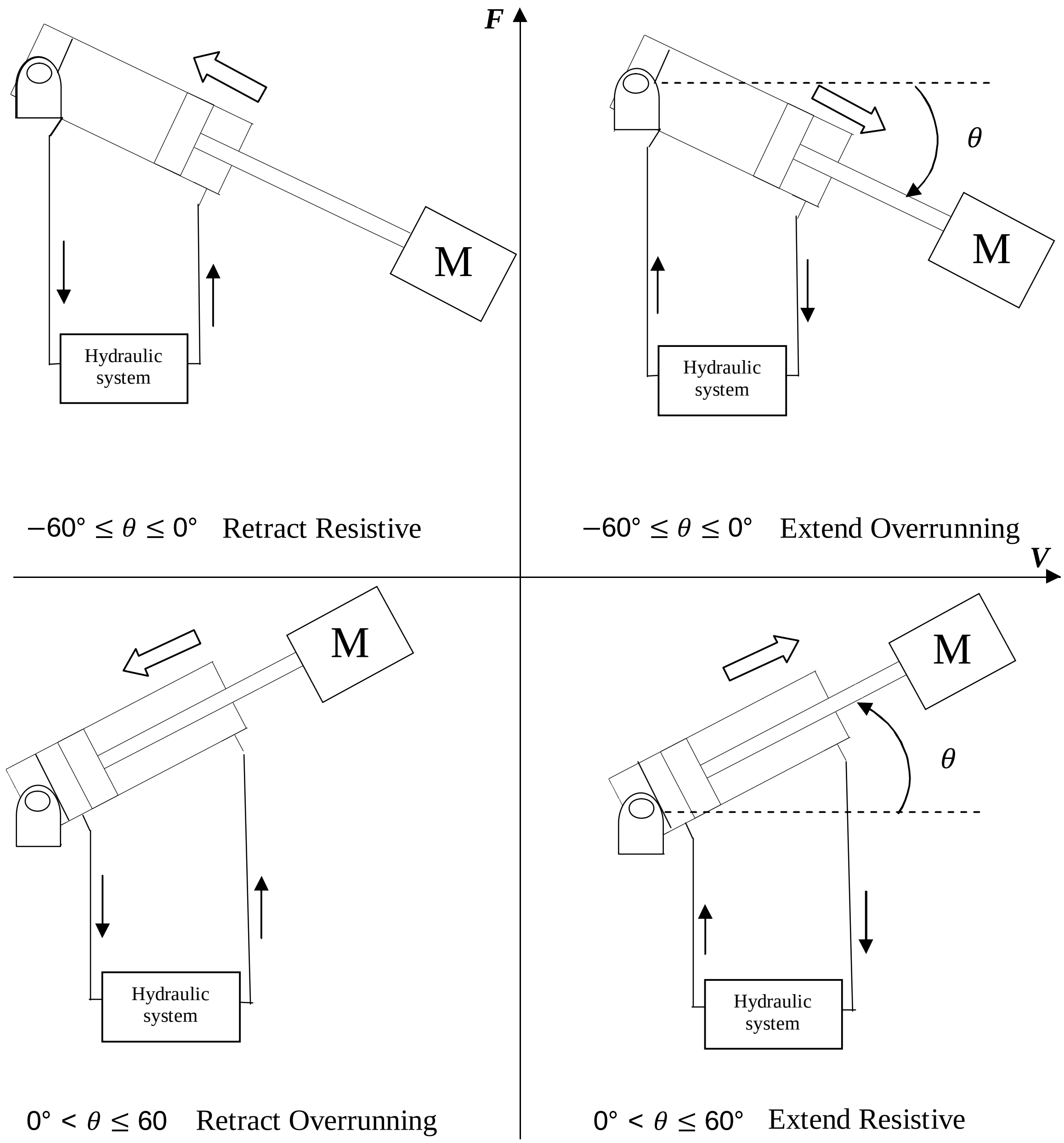


Fig. 4.1. Configuration of Case study 3 in four-quadrant modes.

In Case study 4, a switching force is applied to rod end of the cylinder as the cylinder extends and retracts on the horizontal line. $F(t)$ changes periodically over time from -15 kN to 15 kN. Dynamic performance of hydraulic systems based on this case study condition can show whether a hydraulic system can control a single-rod cylinder properly under unpredictable forces.

4.2. Performance of LS Valve-Controlled System

The LS valve-controlled system is widely applied in heavy equipment industry which confirms the high dynamic performance of such a hydraulic system. It has lower energy consumption among the other valve-controlled systems as also confirmed in Section 3.2.4. Therefore, in dynamic behavior analysis of a single-rod hydraulic cylinder, the performance of the LS system is selected as a baseline for comparison with the pump-controlled systems [7, 2]. In all case studies, the LS valve-controlled system controls the cylinder speed and displacement properly. The throttling of the directional valve in the LS system eliminates the influence of the system inertia and unpredictable forces on the cylinder dynamic behaviour.

(i) Case study 1

With reference to Figs. 4.2a, 4.2b and 4.2c, the cylinder extends and retracts smoothly as the joystick controls the directional valve. The cylinder speed is almost the same as the joystick signal. At the beginning of the stroke, there is an abrupt increase in the cylinder speed. This is related to the sudden increase in the pump pressure by the LS control system once the directional valve starts to open (see Figs. 4.2b and 4.2e). Since the area of the piston side (A_a) is larger than the area of the rod side of the cylinder (A_b) and there is no effective force on the cylinder, P_a is lower than P_b (Fig. 4.2d). P_p is 20 bar higher than P_l, because ΔP_{LS} is set at 20 bar level (see Table 2.2). At the end of the stroke, once the directional valve starts to close, P_p increases until the LS control system

(Fig. 2.3) reduces Q_p and maintains P_p at 20 bar level (Fig. 4.2e). Note that during the cylinder extension, P_a is equal to P_l while in the case of retraction of the cylinder, P_b is the same as P_l. Q_a is higher than Q_b, because the volume of the piston side chamber is higher than the rod side chamber. The plot of the cylinder flow is similar to that of the cylinder speed, since the flow is the product of the area of the cylinder which is a constant value, and the cylinder speed (Fig. 4.2f). The actuating force of the cylinder, $P_aA_a - P_bA_b$, is shown in Fig. 4.2g.

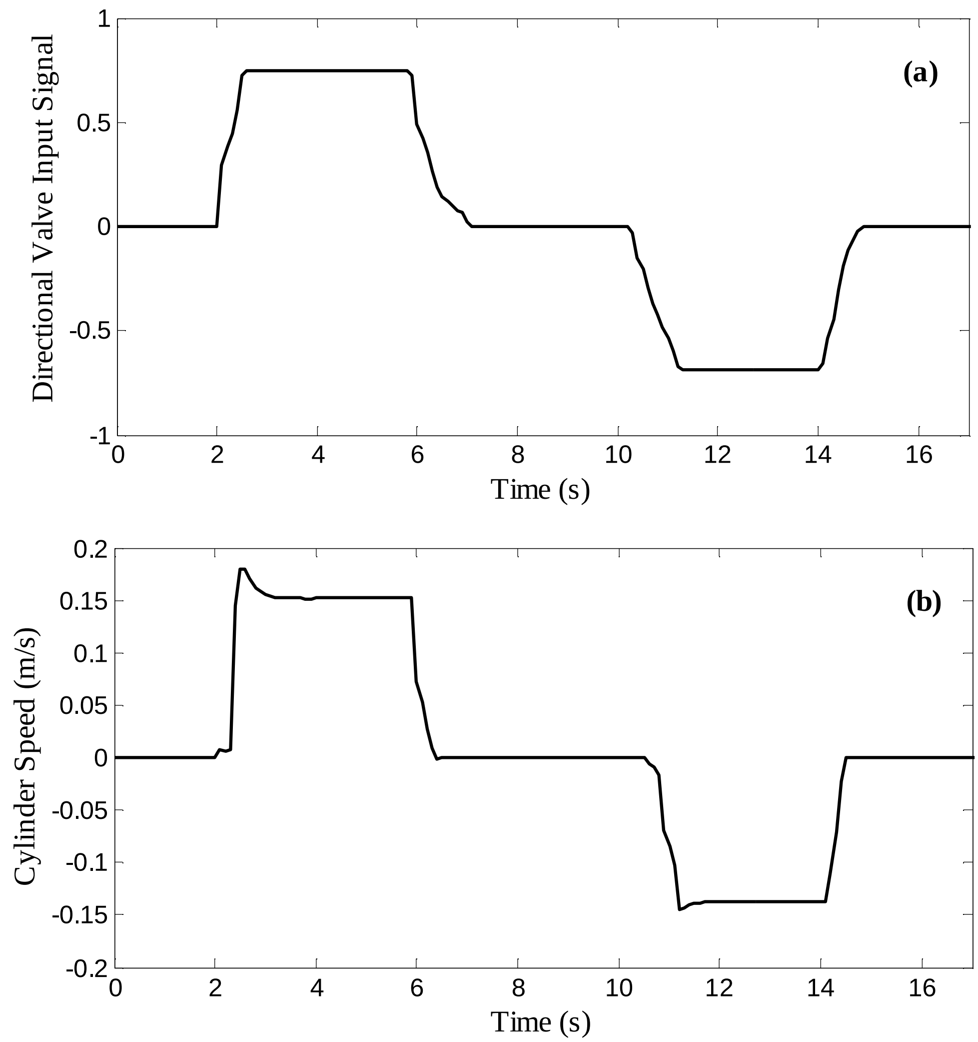


Fig. 4.2. Response of load sensing (LS) system (Case study 1).

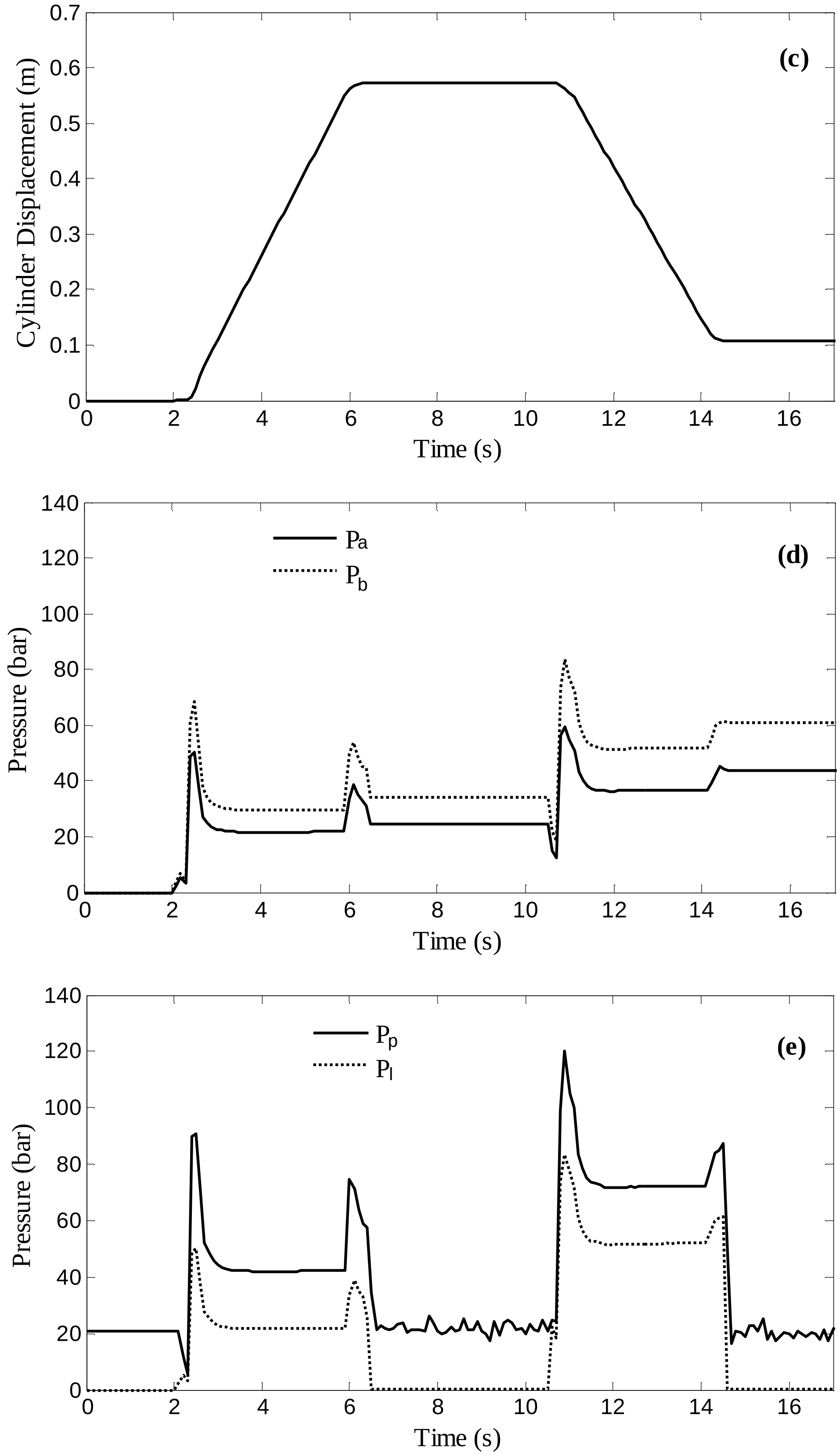

(c)
Cylinder Displacement (m)
Time (s)
(d)
Pa
Pb
Pressure (bar)
Time (s)
(e)
Pp
Pl
Pressure (bar)
Time (s)

Fig. 4.2. Contd.

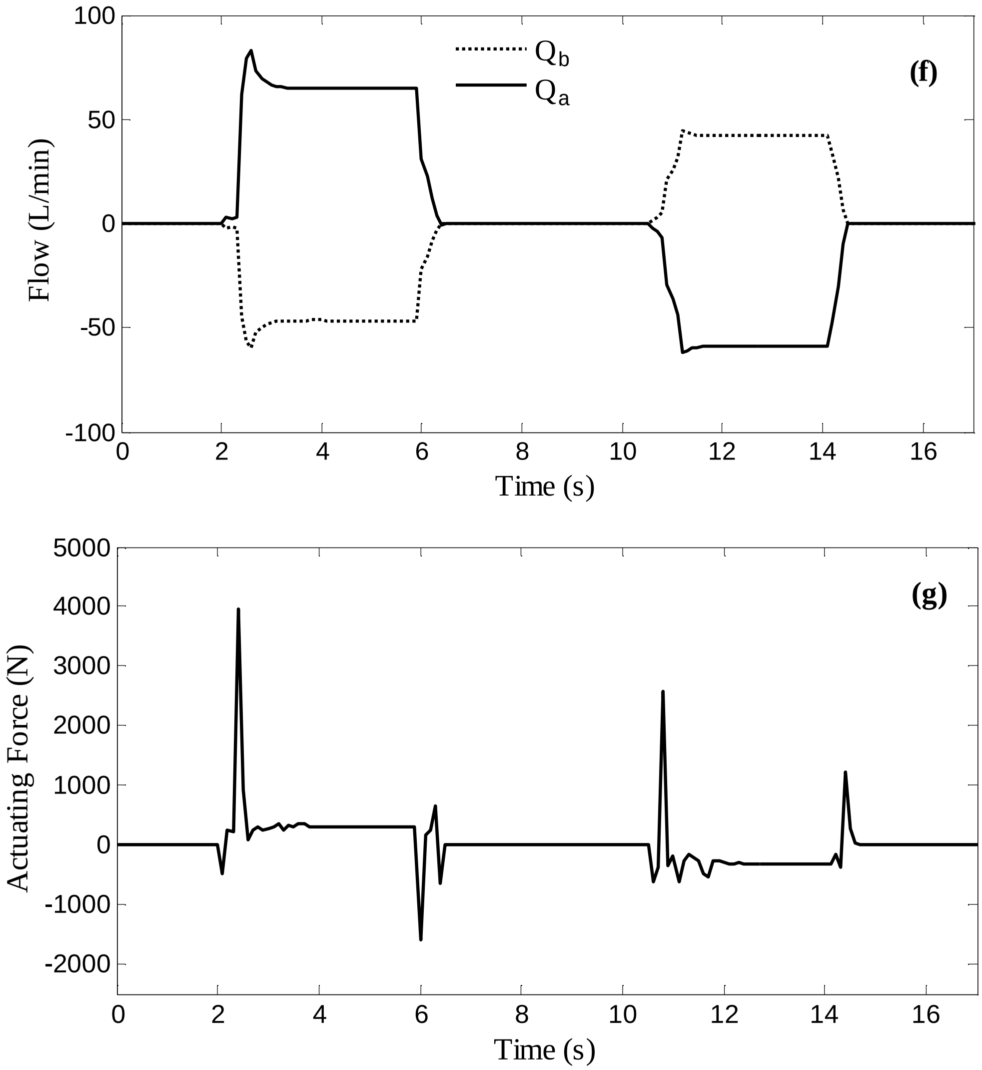


Fig. 4.2. Contd.

(ii) Case study 2

According to Figs. 4.3b and 4.3c, smooth displacement and speed of the cylinder indicate good dynamic performance of the LS system. The throttling of the directional valve at the tank side eliminates influence of the load on the cylinder motion. During the cylinder extension, the load weight causes reduction in P_a. As a result, P_p remains low at 30 bar. However, during the cylinder retraction, the pump pressure rises to 95 bar to boost P_b and lift the load (see Figs. 4.3d and 4.3e).

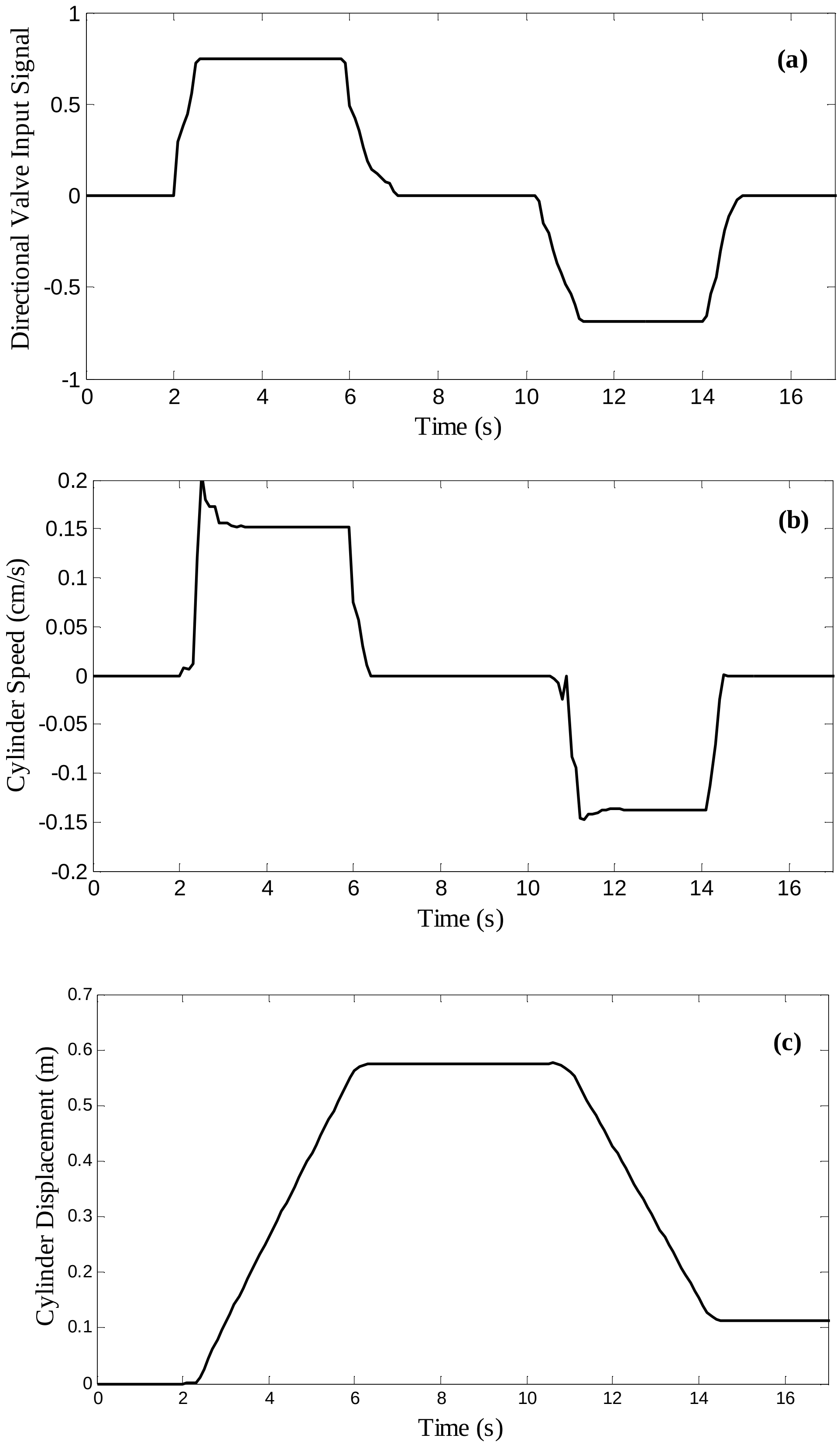
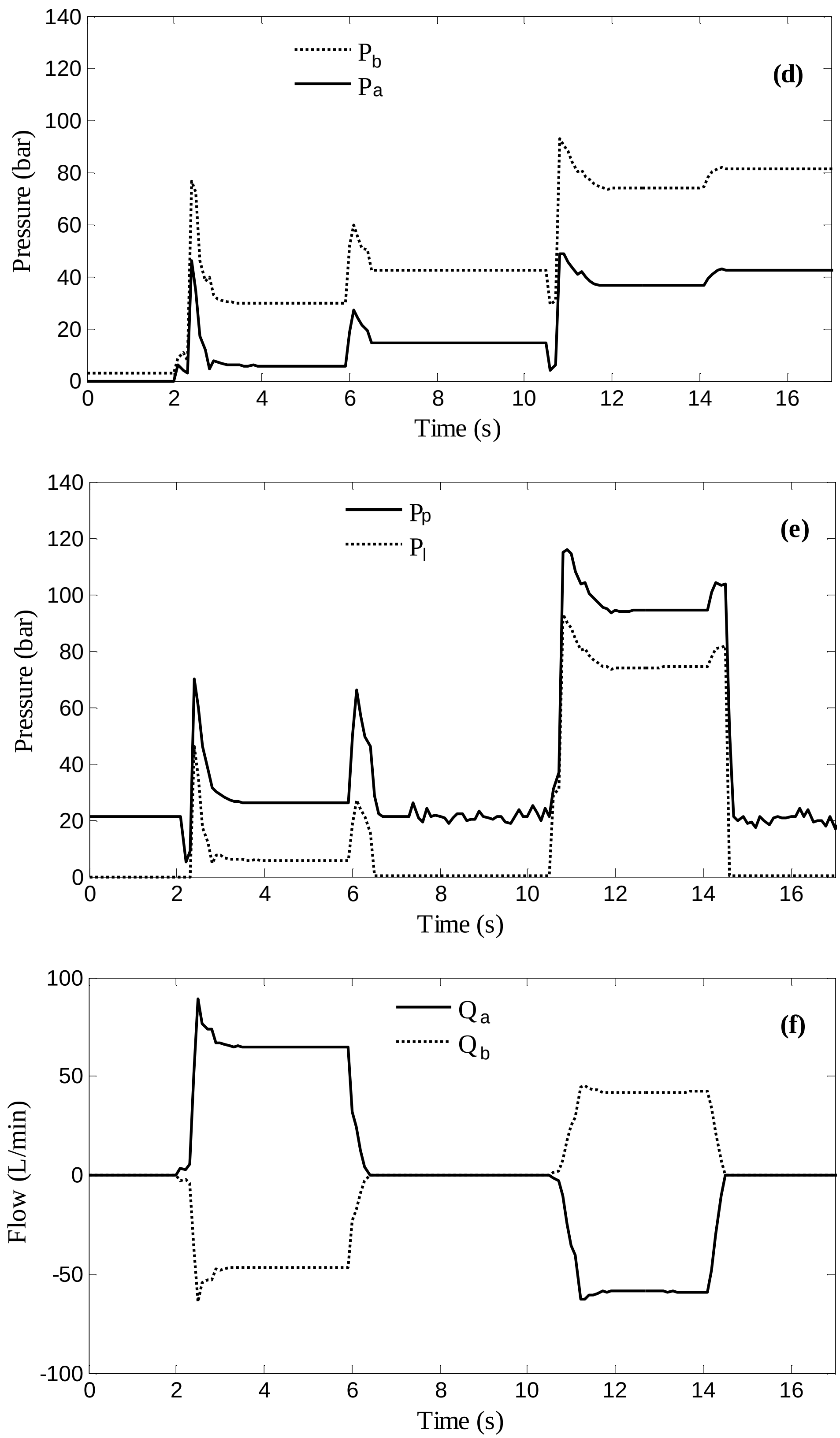


Fig. 4.3. Response of load sensing (LS) system (Case study 2).



(d)
Pb
Pa
Pressure (bar)
Time (s)
(e)
Pp
Pl
Pressure (bar)
Time (s)
(f)
Qa
Qb
Flow (L/min)
Time (s)

Fig. 4.3. Contd.

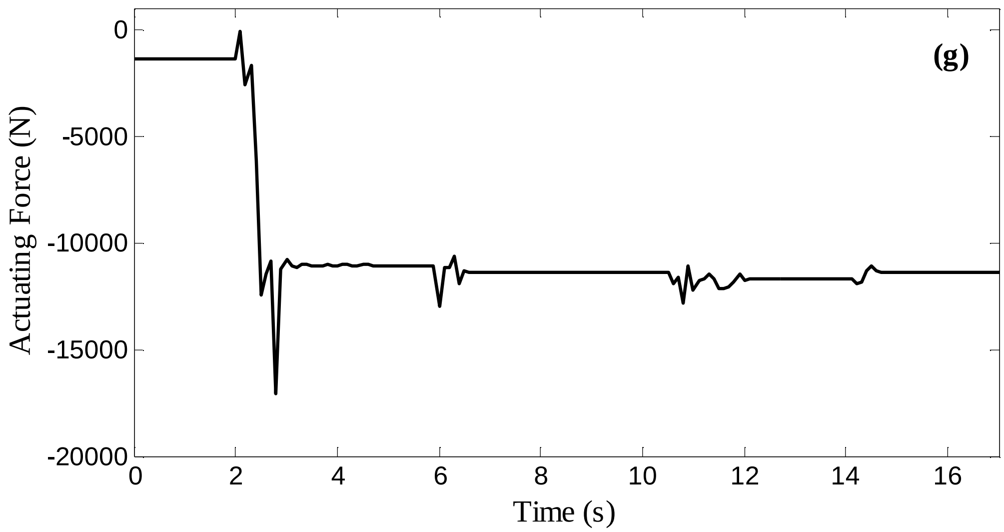


Fig. 4.3. Contd.

(iii) Case study 3

The linear cylinder speed is constant while the cylinder turns the load about the pin axis (Figs. 4.4b and 4.4i). As shown in Fig. 4.4c, the cylinder displacement is smooth. During the cylinder extension, θ rotates from 60° to -40° proportionally to the cylinder displacement (Fig. 4.4g). As θ decreases, P_a decreases gradually from 40 bar to 11 bar. This behavior originates from the reduction in the effective force ($Mg\ \sin\theta$) applied to the cylinder (Fig. 4.4h). Based on the value of P_a, the LS control system gradually reduces P_p and maintains at 20 bar higher than P_a (Figs. 4.4d and 4.4e). Note that, during the cylinder extension, P_l is equal to P_a. During the cylinder retraction (*i.e.*, rotation from θ = -40° to θ = 42.7°), P_l is equal to P_b. Consequently, when P_b decreases, P_p reduces and keeps pump pressure 20 bar higher than P_b.

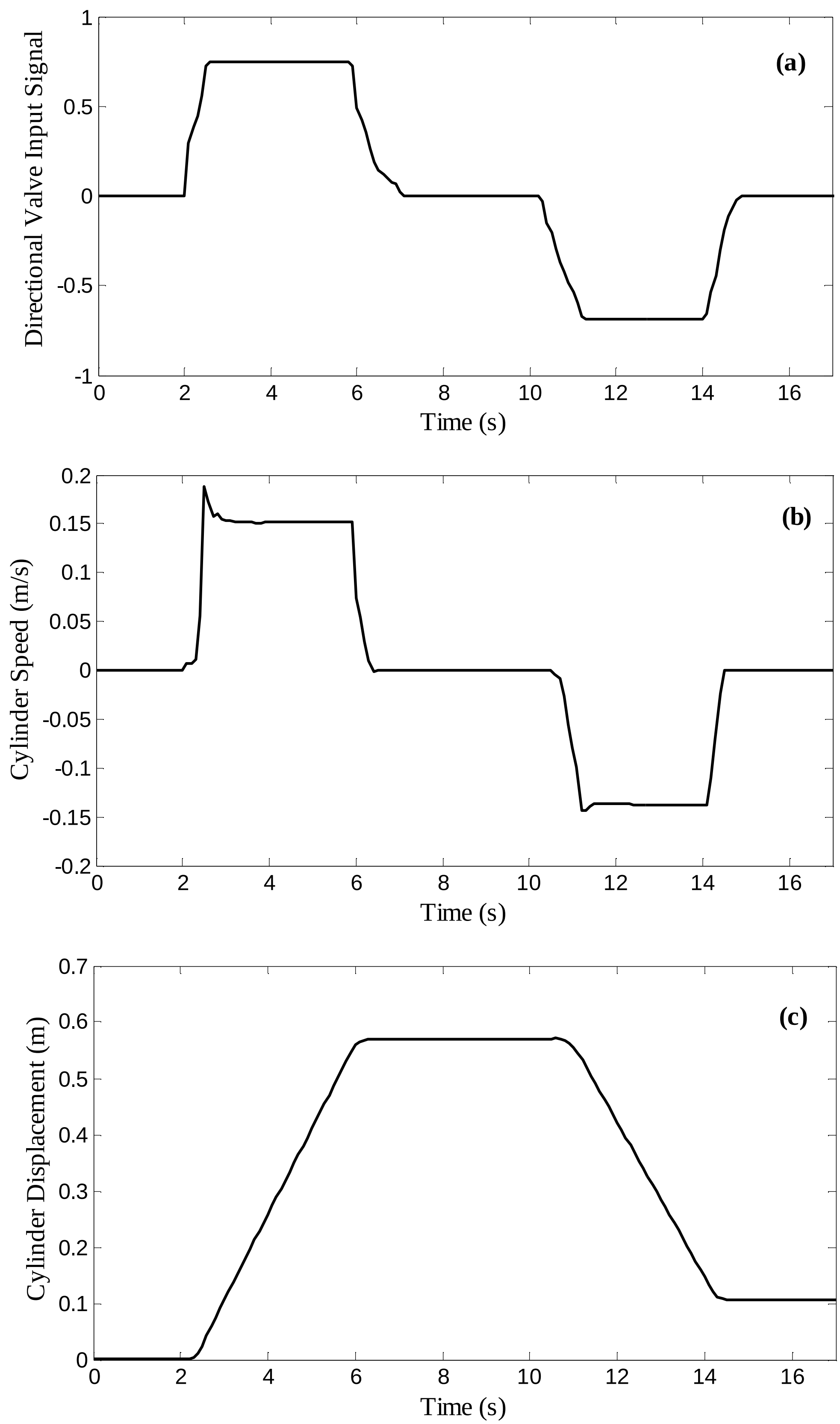


Fig. 4.4. Response of load sensing (LS) system (Case study 3).

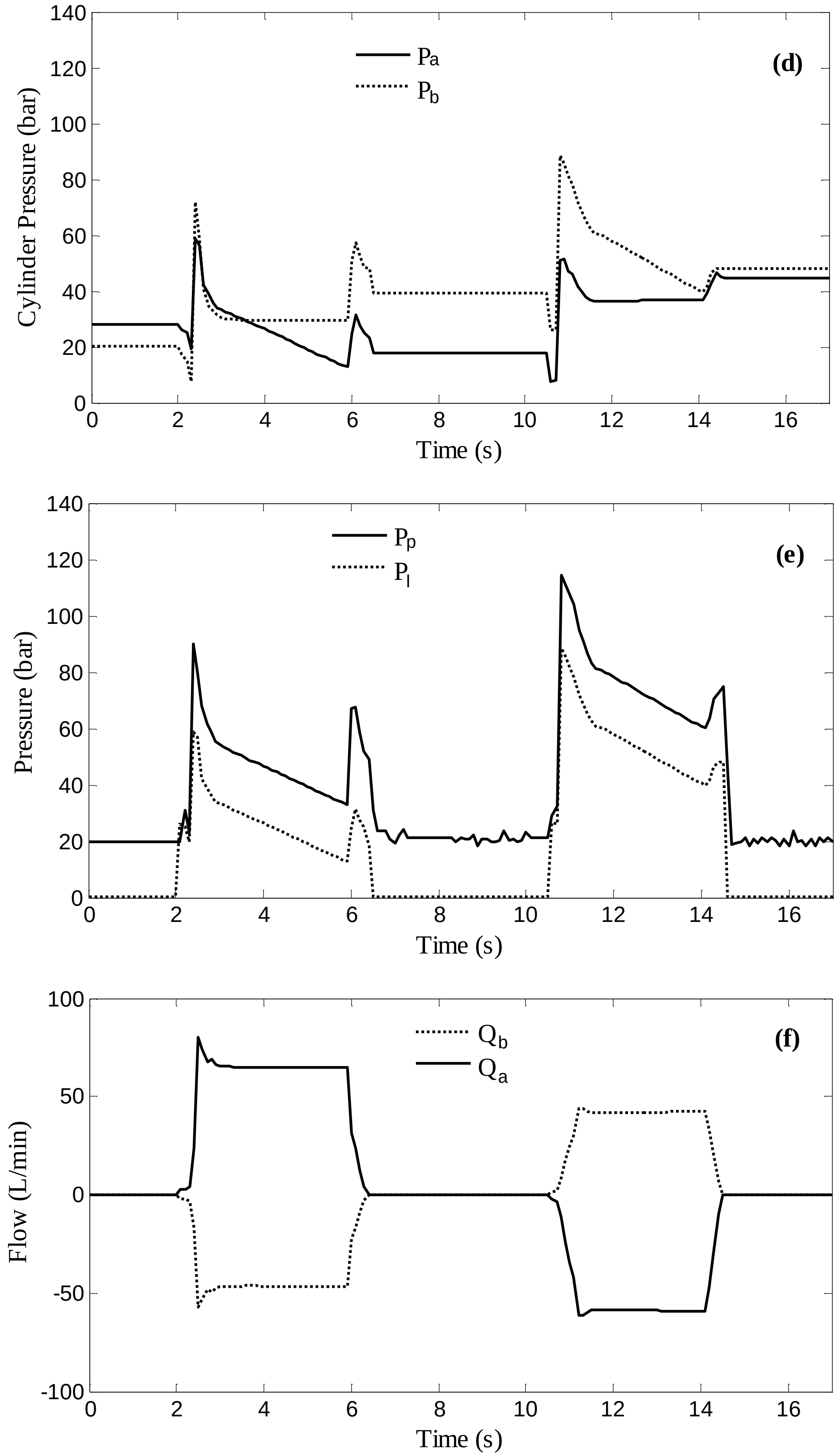

Pa
Pb
(d)
Cylinder Pressure (bar)
Time (s)
Pp
Pl
(e)
Pressure (bar)
Time (s)
Qb
Qa
(f)
Flow (L/min)
Time (s)

Fig. 4.4. Contd.

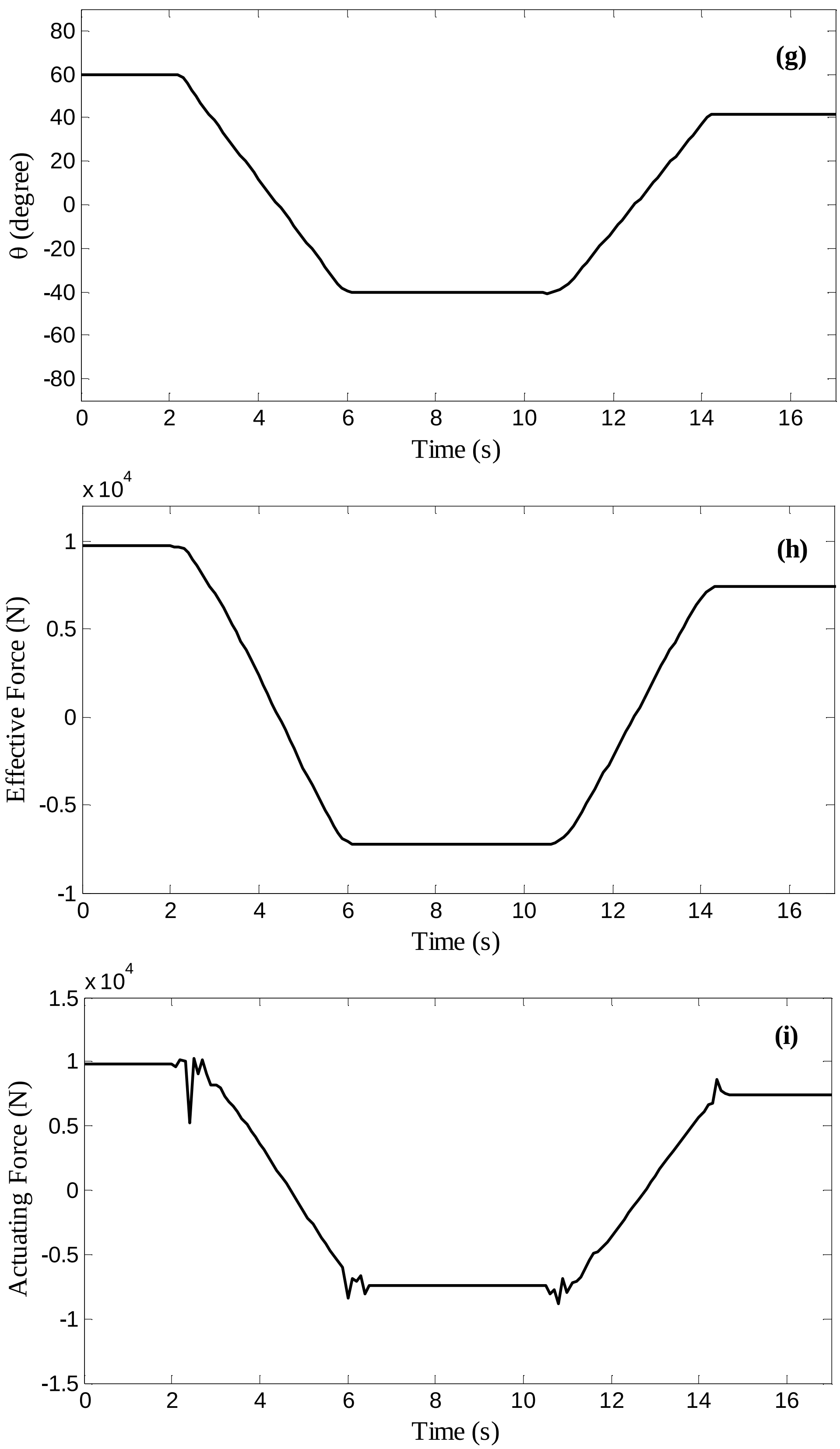

(g)
θ (degree)
80
60
40
20
0
-20
-40
-60
-80
0
2
4
6
8
10
12
14
16
Time (s)
x 10^4
(h)
Effective Force (N)
1
0.5
0
-0.5
-1
0
2
4
6
8
10
12
14
16
Time (s)
x 10^4
(i)
Actuating Force (N)
1.5
1
0.5
0
-0.5
-1
-1.5
0
2
4
6
8
10
12
14
16
Time (s)

Fig. 4.4. Contd.

(iv) Case study 4

With reference to Figs. 4.5b and 4.5c, the cylinder speed is constant except for some moments when the direction of the switching force ($F(t)$) changes by the time. The throttling of the directional valve provides high stiffness for the cylinder to eliminate the influence of $F(t)$ on the cylinder motion. The periodic behavior of $F(t)$ causes oscillation in the cylinder pressure, as shown in Fig. 4.5d. Since $A_a > A_b$, the average rod side pressure is higher than the average piston side pressure (Fig. 4.5d). P_p oscillates while maintaining its value 20 bar higher than P_l (Fig. 4.5e).

During the cylinder extension, P_a shows more oscillations than P_b, because the piston side chamber is connected to the pump line in which P_p is oscillating by P_l (Fig. 4.5d). However, the rod side chamber is connected to the tank line, which has a constant low pressure. Similarly, during the cylinder retraction, the piston side chamber is connected to the tank line through the directional valve.

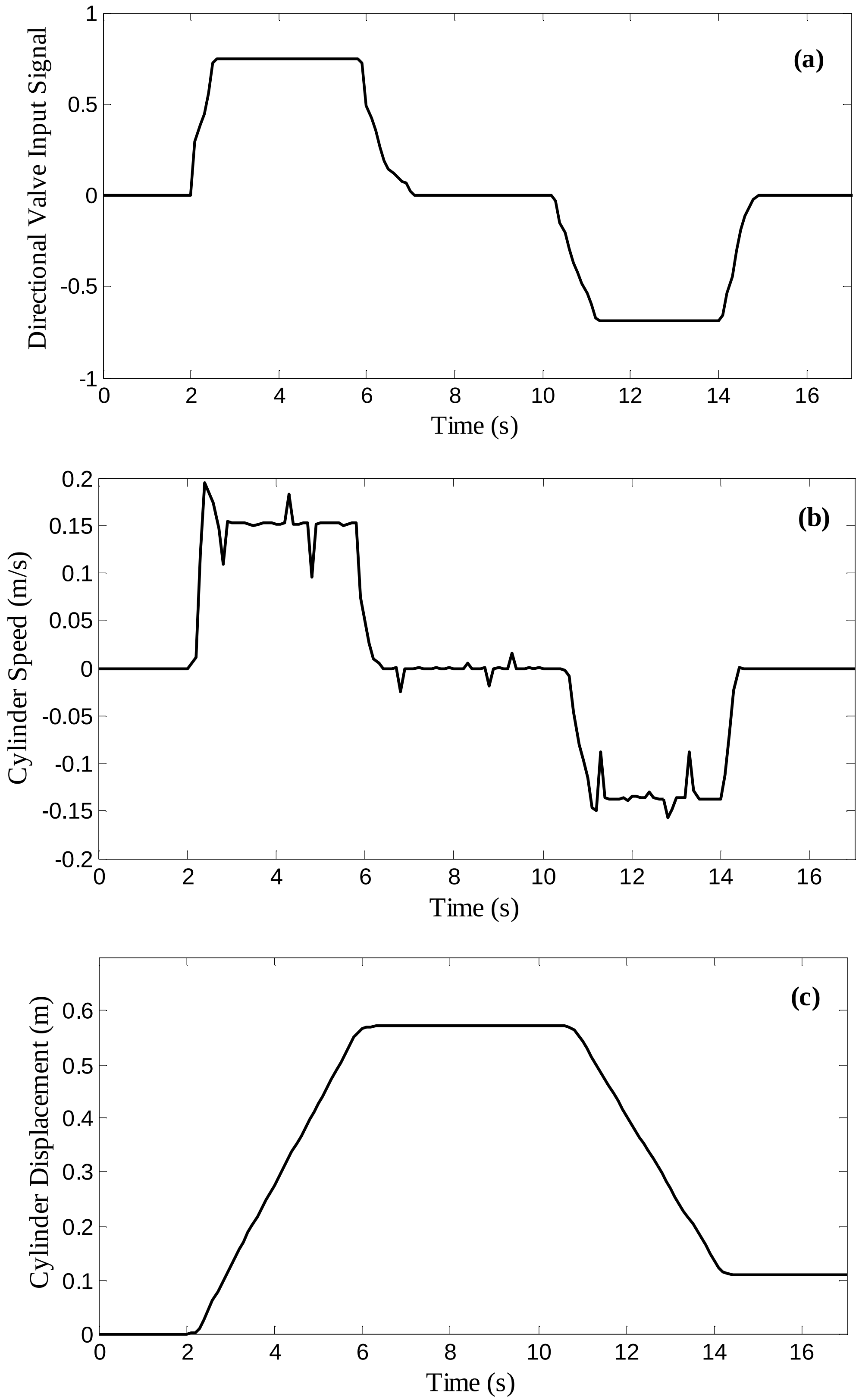


Fig. 4.5. Response of load sensing (LS) system (Case study 4).

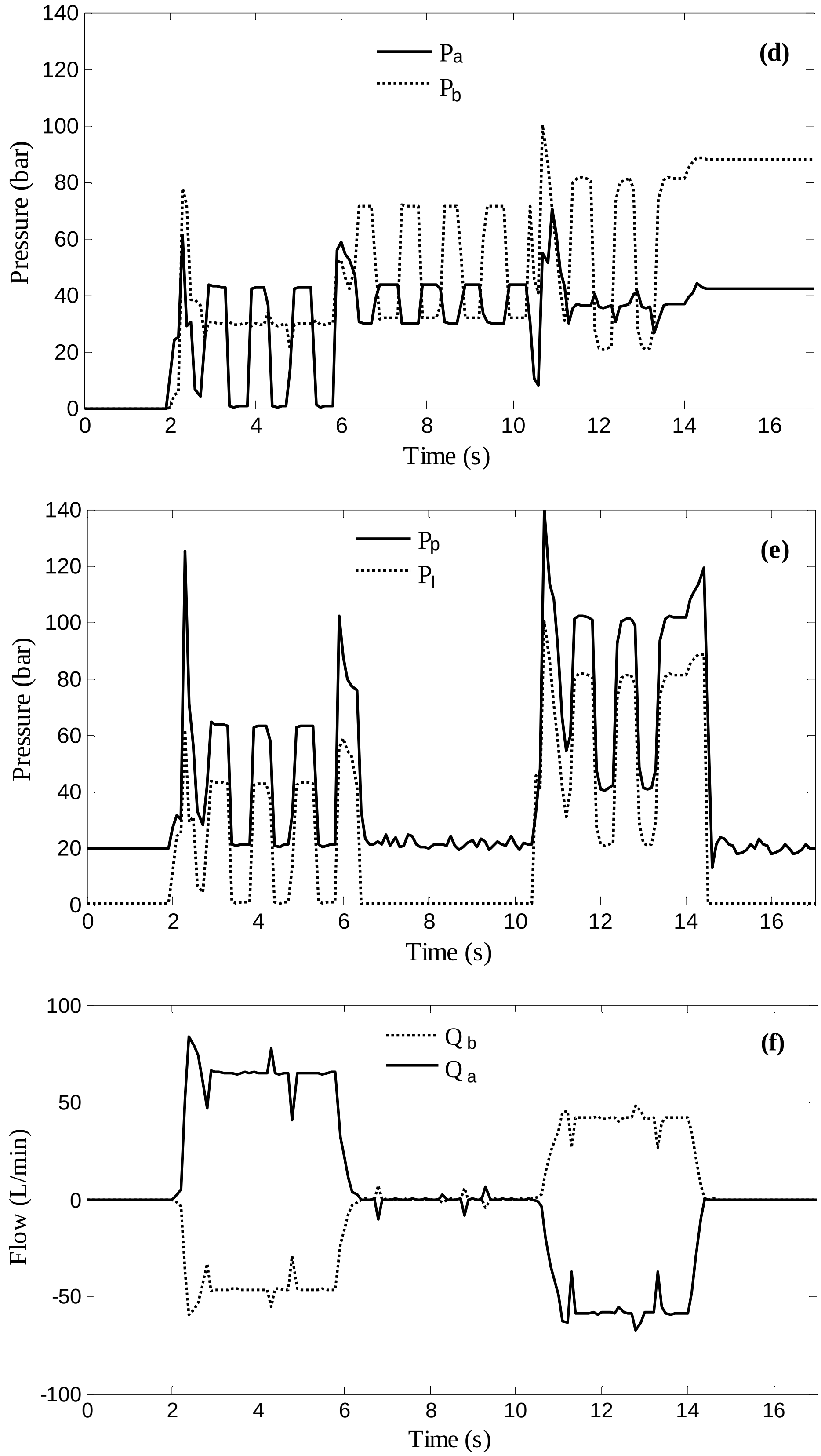

(d)
Pa
Pb
Pressure (bar)
Time (s)
(e)
Pp
Pl
Pressure (bar)
Time (s)
(f)
Q b
Q a
Flow (L/min)
Time (s)

Fig. 4.5. Contd.

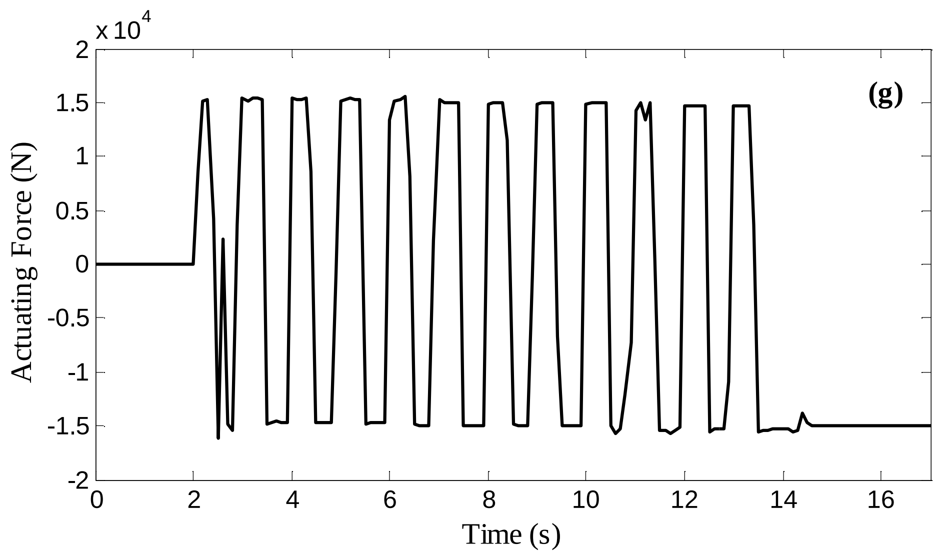


Fig. 4.5. Contd.

4.3. Performance of DC Pump-Controlled System

(i) Case study 1

The cylinder extends and retracts as the joystick signal changes the swash plate angle of the pump. During the cylinder extension, the cylinder speed is constant (Fig. 4.6b). Since Q_p is the same as Q_b (Figs. 4.6f and 4.6h), the pump does not provide enough flow for the piston side of the cylinder. In this case, POC1 pilot line opens the POC1 to compensate for the differential flow between piston side and rod side of the cylinder ($Q_a - Q_b$). Fig. 4.6g shows $Q_a - Q_b$ passing through the POC1 valves. In this figure, the flow rate is positive when it is removed from the pump lines connected to both sides of the cylinder. The flow is negative when it enters pump lines through a POC valve. At the end of the stroke, during the cylinder extension, joystick signal reduces the swash plate angle of the pump to stop the cylinder smoothly. However, the system inertia, including masses of the load and the cylinder, causes abrupt fluctuation in the cylinder speed (Fig. 4.6b).

During the cylinder retraction, the system inertia affects the cylinder speed at the beginning of the cylinder displacement (Fig. 4.6b). The cylinder displacement is smooth (Fig. 4.6c). P_b increases up to P_{poc1} to open the POC1 (Fig. 4.6d). As a result, Q_a is divided into two flows; one enters to the low-pressure system through the POC1 and the other enters to the pump inlet (see Figs. 4.6g and 4.6h). The hydraulic system works in modes 2 and 3, during the cylinder extension and retraction, respectively. To see the flow circulation during these modes, refer to Fig. 2.9.

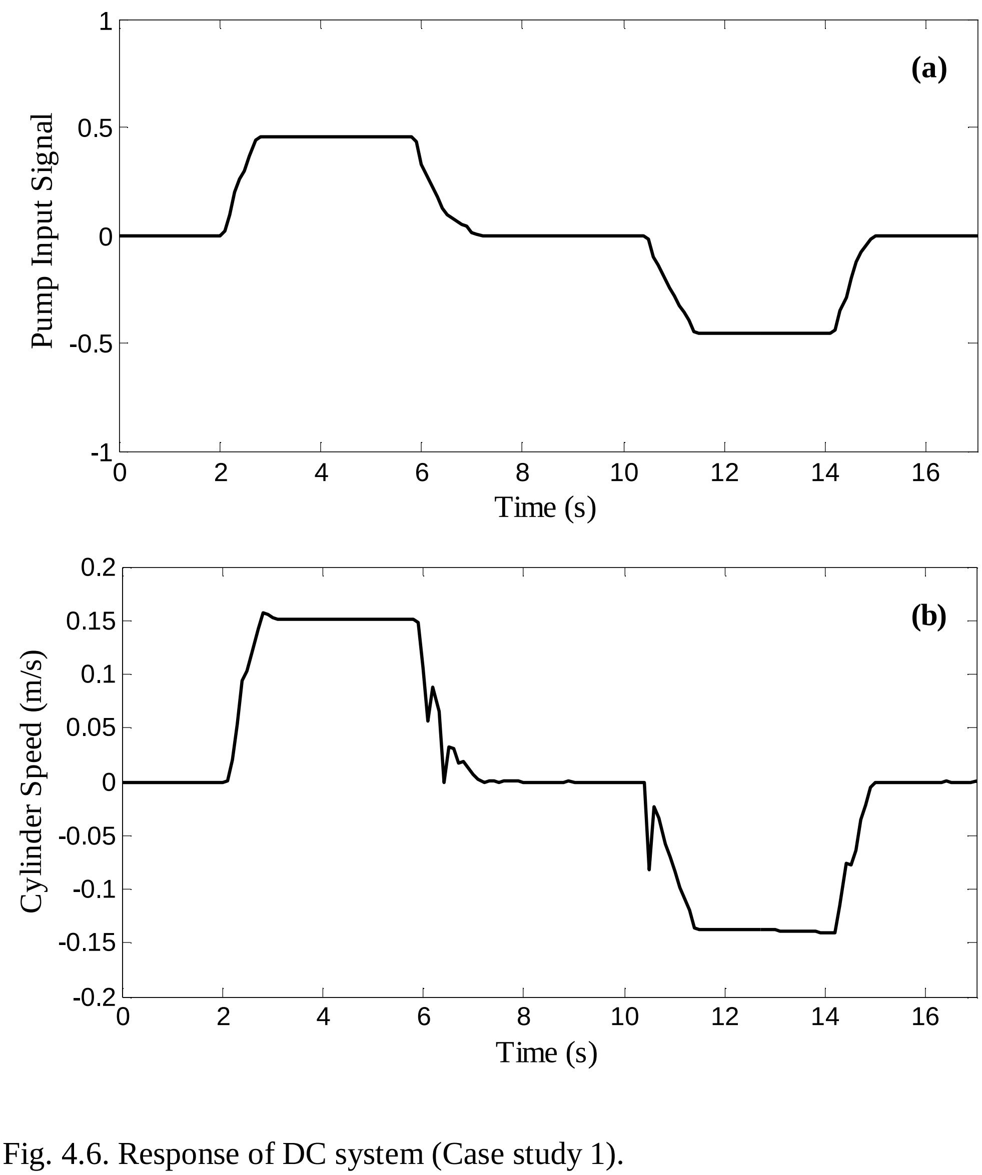


Fig. 4.6. Response of DC system (Case study 1).

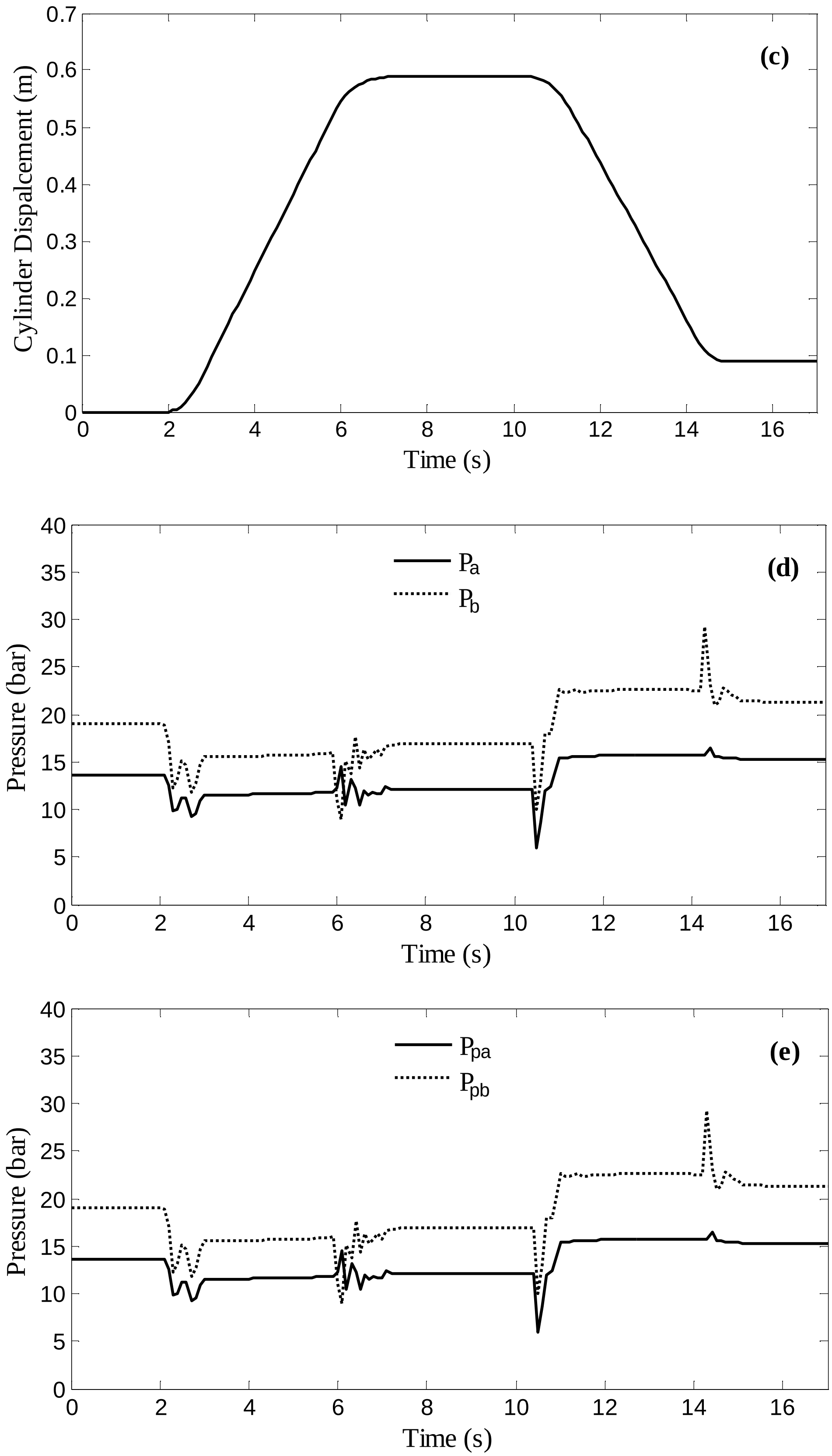

(c)
Cylinder Displacement (m)
Time (s)
(d)
Pa
Pb
Pressure (bar)
Time (s)
(e)
Ppa
Ppb
Pressure (bar)
Time (s)

Fig. 4.6. Contd.

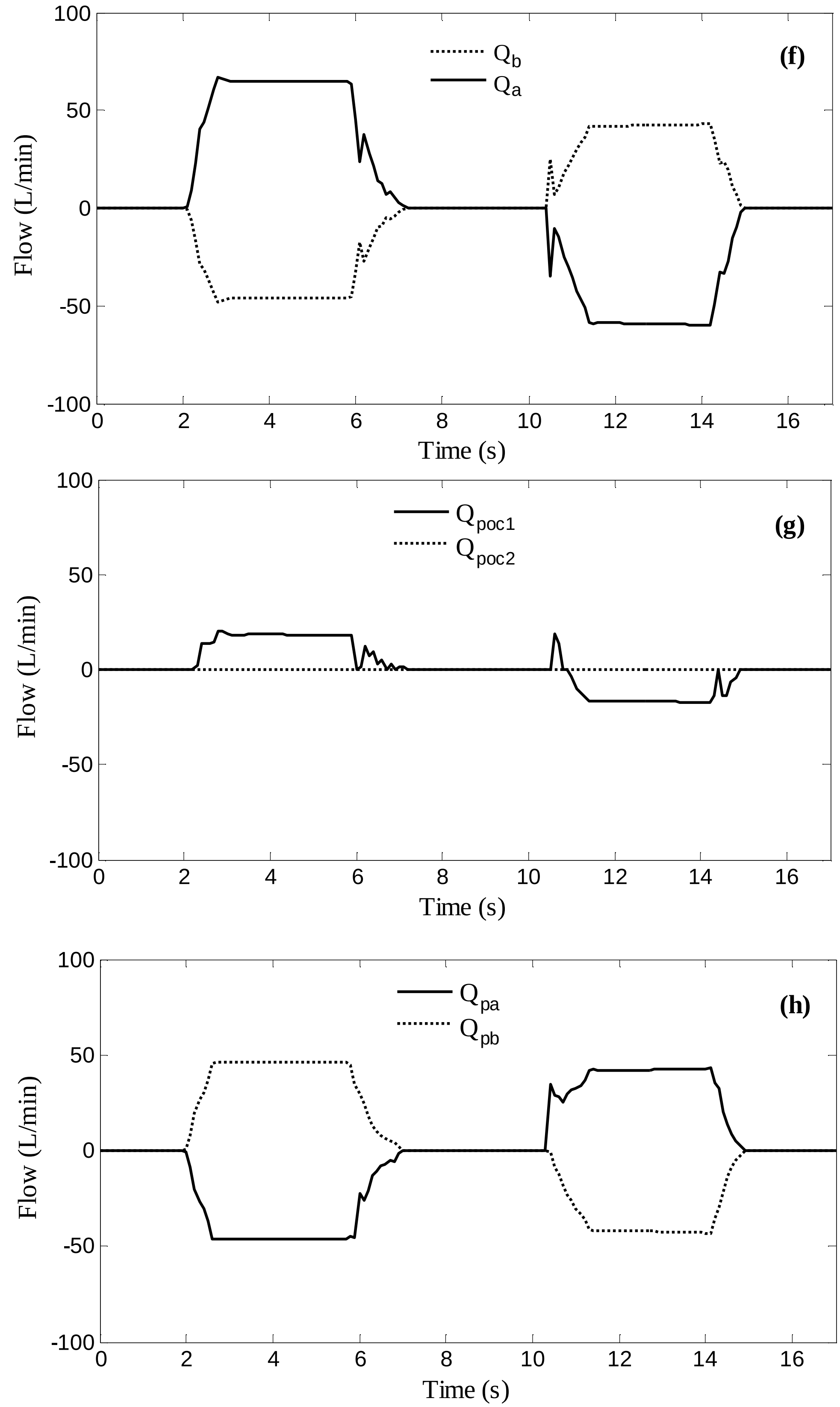

(f)
Q_b
Q_a
(g)
Q_poc1
Q_poc2
(h)
Q_pa
Q_pb
Flow (L/min)
Time (s)
100
50
0
-50
-100
0
2
4
6
8
10
12
14
16

Fig. 4.6. Contd.

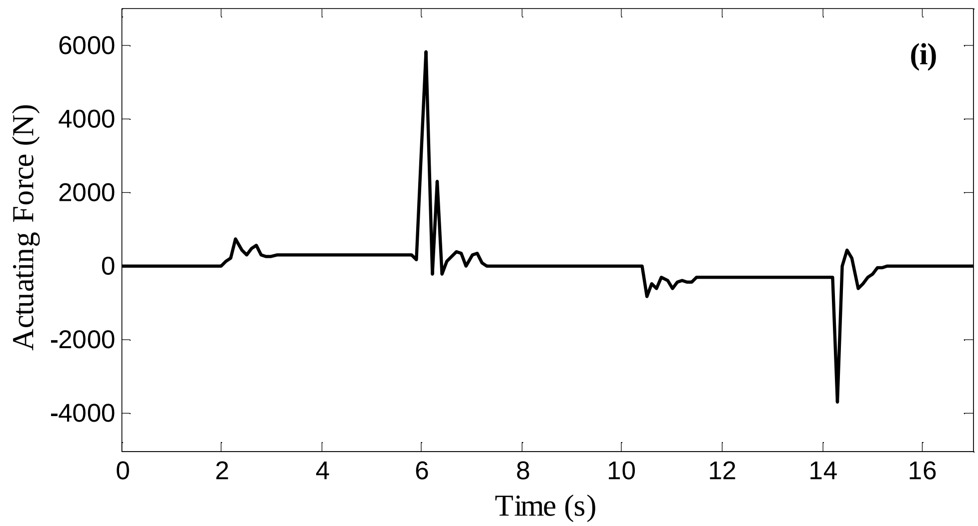


Fig. 4.6. Contd.

(ii) Case study 2

With reference to Fig. 4.7c, the cylinder displacement is smooth during the operation. During the cylinder extension, there is fluctuation in the cylinder speed at the end of the stroke (Fig. 4.7b). This is related to influence of the system load inertia on the cylinder motion. The load weight influences on the cylinder speed before the pump displacement takes the control of the cylinder. During the cylinder retraction, the system inertia, once again, influences the cylinder speed at the beginning of the stroke (see Fig. 4.7b). Tendency of the load to extend the cylinder generates the low pressure at the piston side, and causes the flow from the low-pressure system to enter the piston side through the POC1 (Fig. 4.7g). This happens prior to controlling the cylinder by the pump.

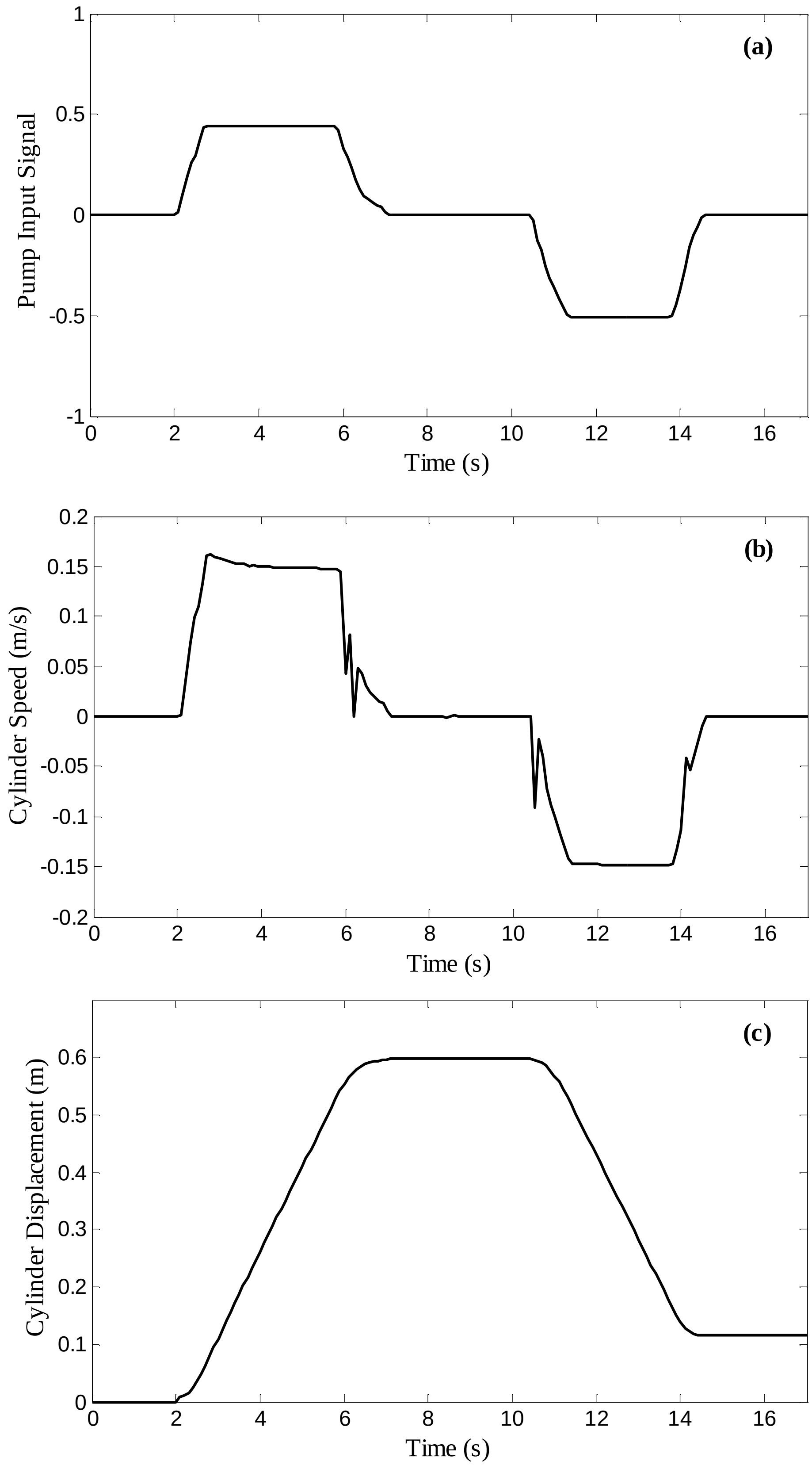
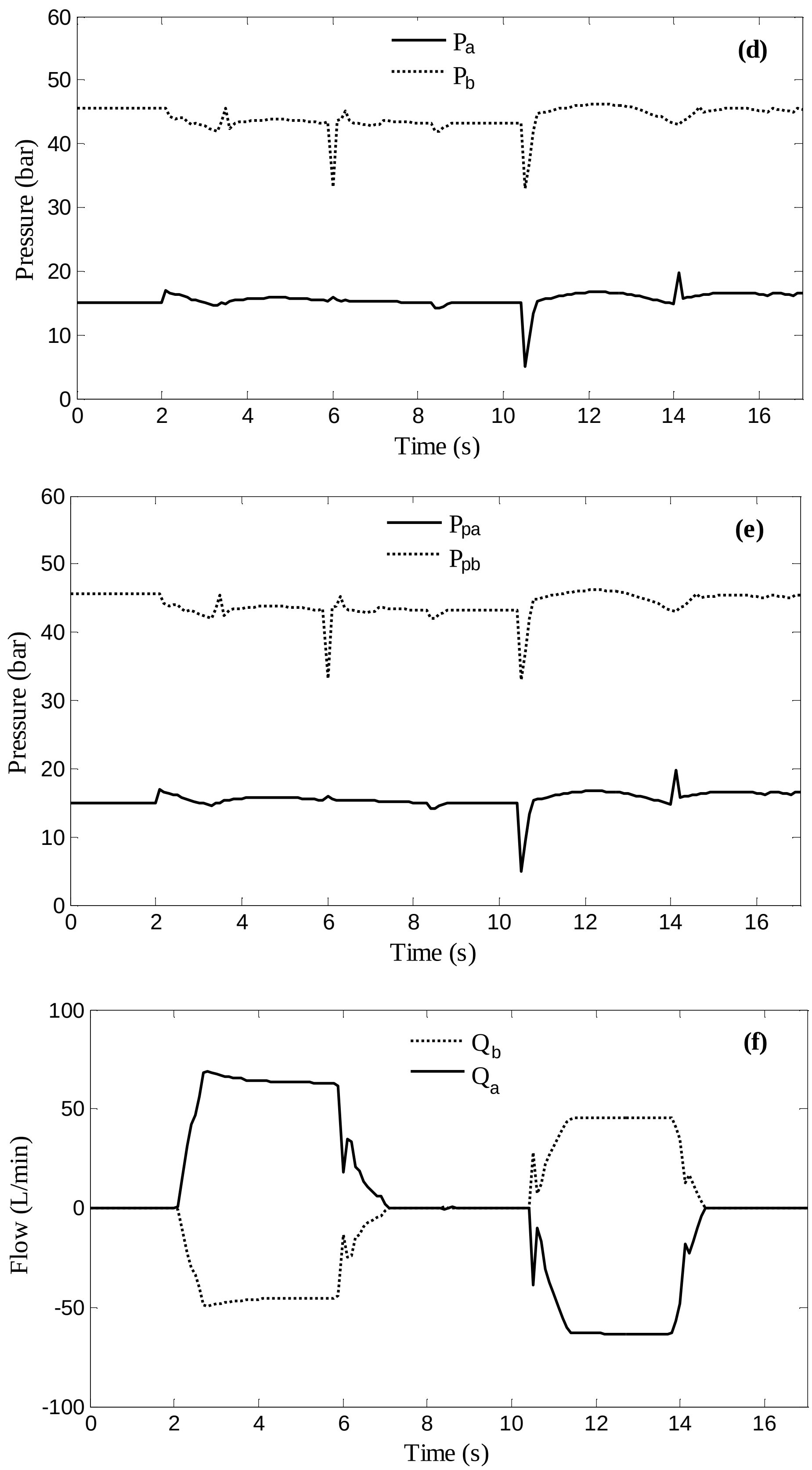
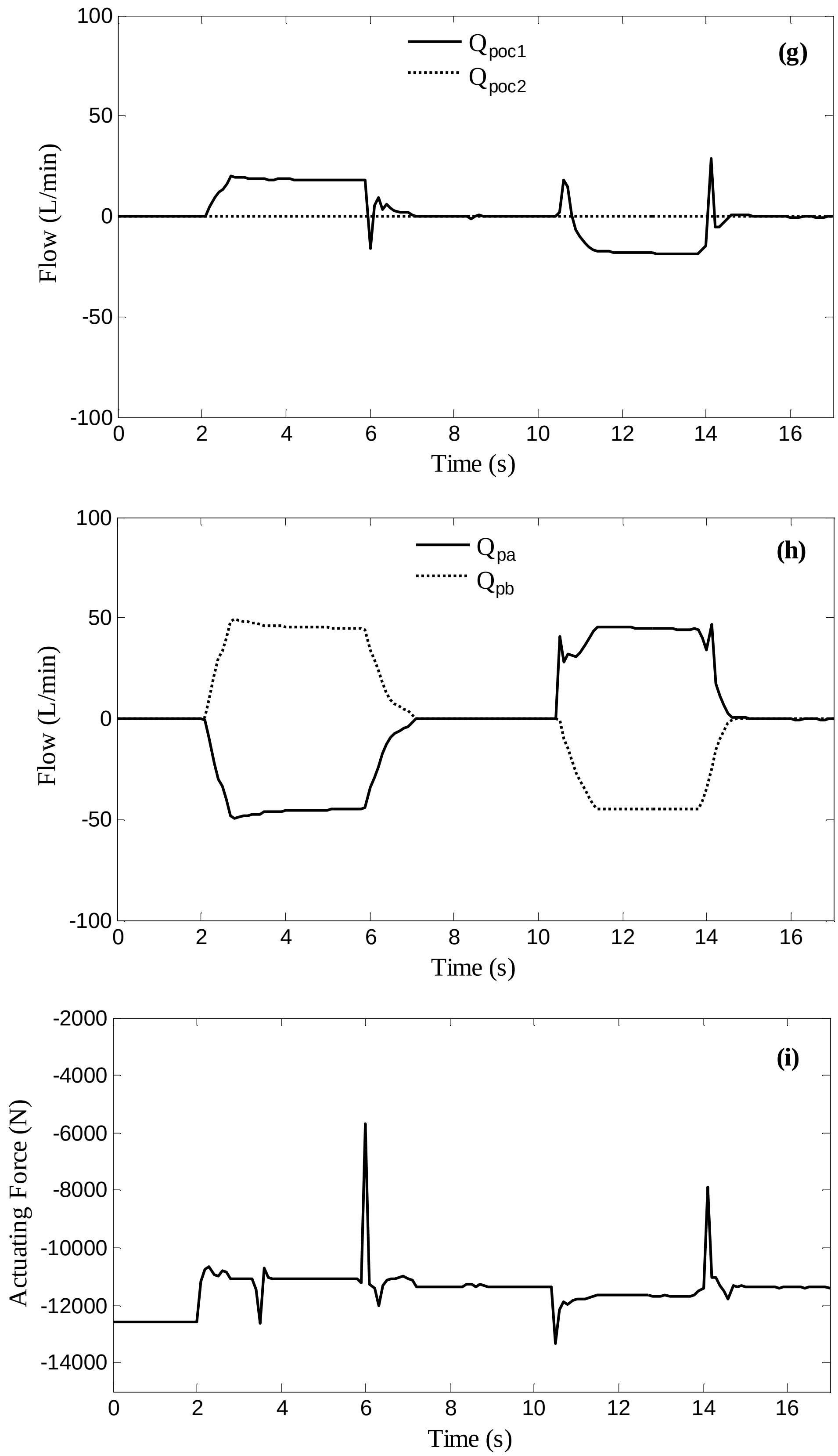


Fig. 4.7. Response of DC system (Case study 2).



(d)
P_a
P_b
Pressure (bar)
Time (s)
(e)
P_{pa}
P_{pb}
Pressure (bar)
Time (s)
(f)
Q_b
Q_a
Flow (L/min)
Time (s)

Fig. 4.7. Contd.



(g)
Q_poc1
Q_poc2
Flow (L/min)
Time (s)
(h)
Q_pa
Q_pb
Flow (L/min)
Time (s)
(i)
Actuating Force (N)
Time (s)

Fig. 4.7. Contd.

(iii) Case study 3

During the cylinder extension, and rotation from $\theta = 60°$ to $\theta = 0°$, the hydraulic system is in mode 1 (see Fig. 2.9). With reference to Fig. 4.8g, $Q_a - Q_b$ is compensated by the flow passing through the POC2. As the cylinder crosses the horizontal line, at $\theta = 0°$, POC2 closes and POC1 opens. During $\theta < 0°$, the hydraulic system enters mode 2 in which the pump (bidirectional pump) serves as a motor. The pump flow is almost the same in modes 1 and 2 (Fig. 4.8h). In mode 1, the entire pump flow enters the piston side $(Q_{pa} = Q_a)$ while, in mode 2, the entire pump flow enters the rod side chamber of the cylinder $(Q_{pb} = Q_b)$. Because $A_a > A_b$, the cylinder speed in mode 1 is lower than the cylinder speed in mode 2 (Fig. 4.8b). The cylinder speed is under the influence of the effective force $(Mg \sin\theta)$, during the cylinder extension. As the effective force reduces, the cylinder velocity increases.

During the cylinder retraction, the hydraulic system is in mode 3 as the cylinder turns the load from $\theta° = -42°$ to $\theta = 0°$. In this mode, the pump generates the flow required for the rod side chamber of the cylinder $(Q_{pb} = Q_b)$. $Q_a - Q_b$ is removed to the low-pressure system through the POC1 (Fig. 4.8g). As seen in Fig. 4.8b, the cylinder speed oscillates, after the system mode changes from 3 to 4, when the cylinder passes the horizontal line $(\theta = 0°)$. During the mode changing, rapid changing of the flow rate causes oscillation in the cylinder pressures. Because, in mode 4, the cylinder pilot pressure controls the POC valve opening, POC valves frequently close and open. This closing and opening POC valves, cause oscillations in the cylinder speed. The POC valve characteristic such as pilot ratio and pressure drop do have effect on instability, but their contributions are minor compared to the mentioned factor [6].

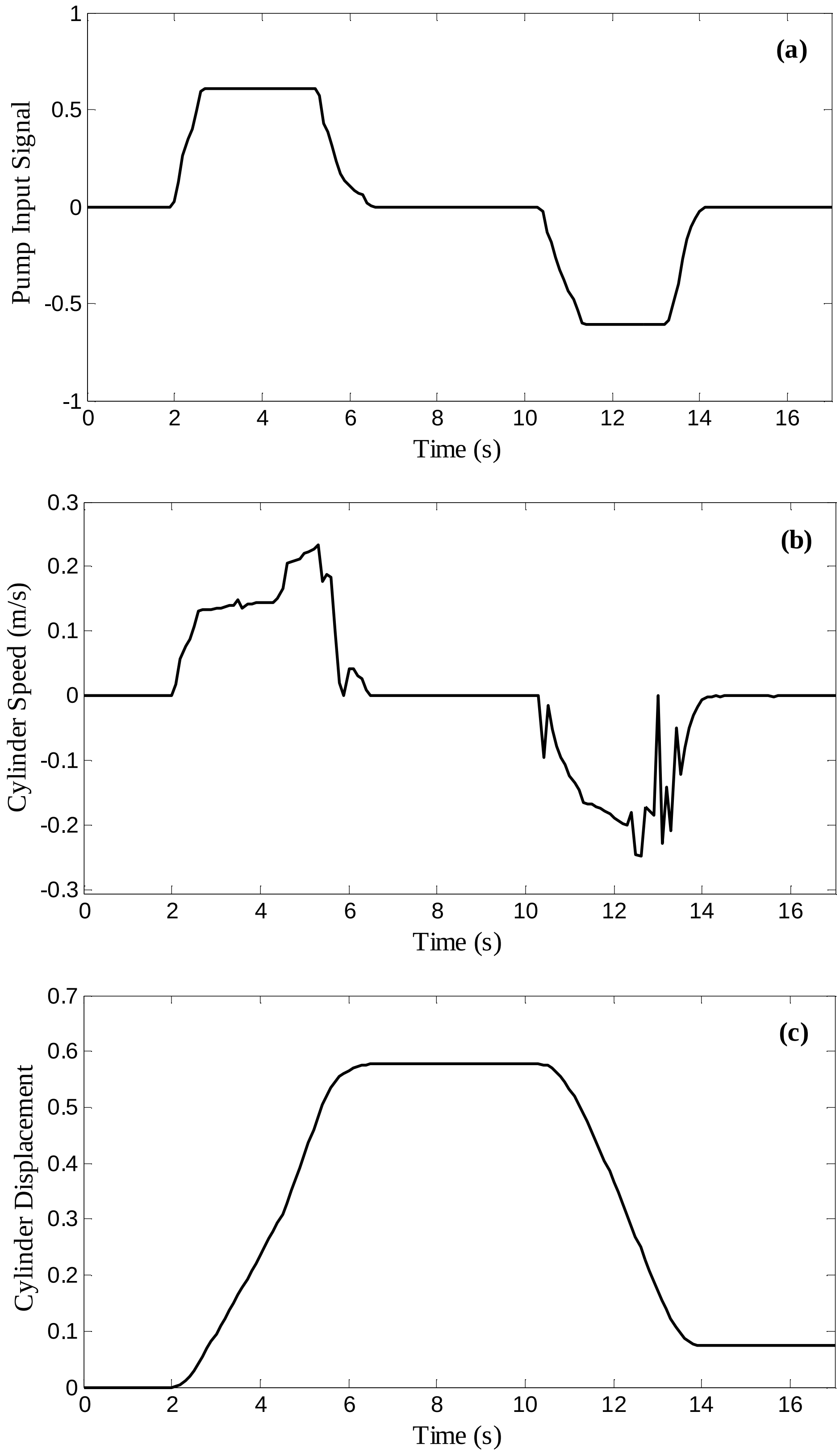


Fig. 4.8. Response of DC system (Case study 3).

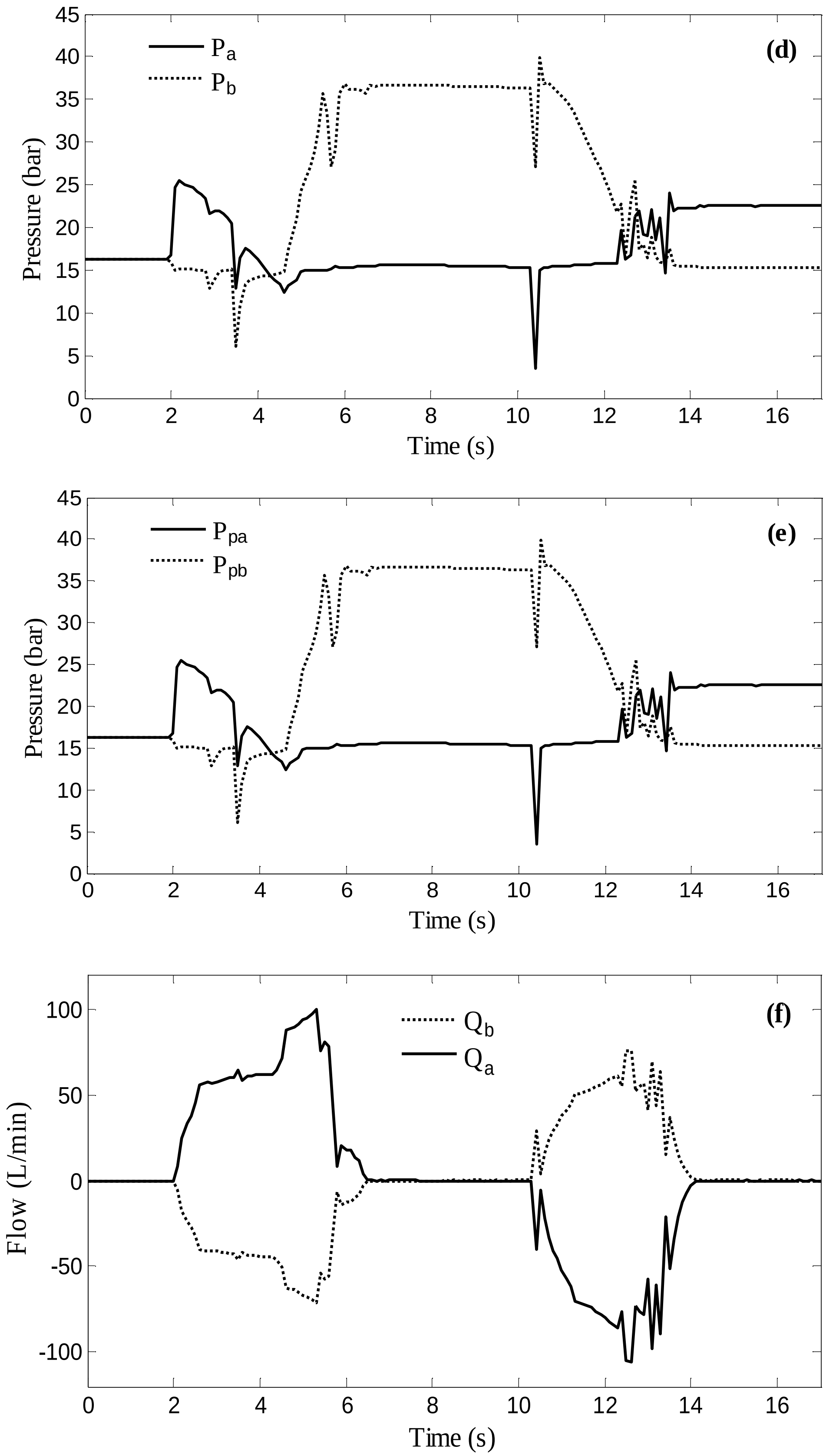
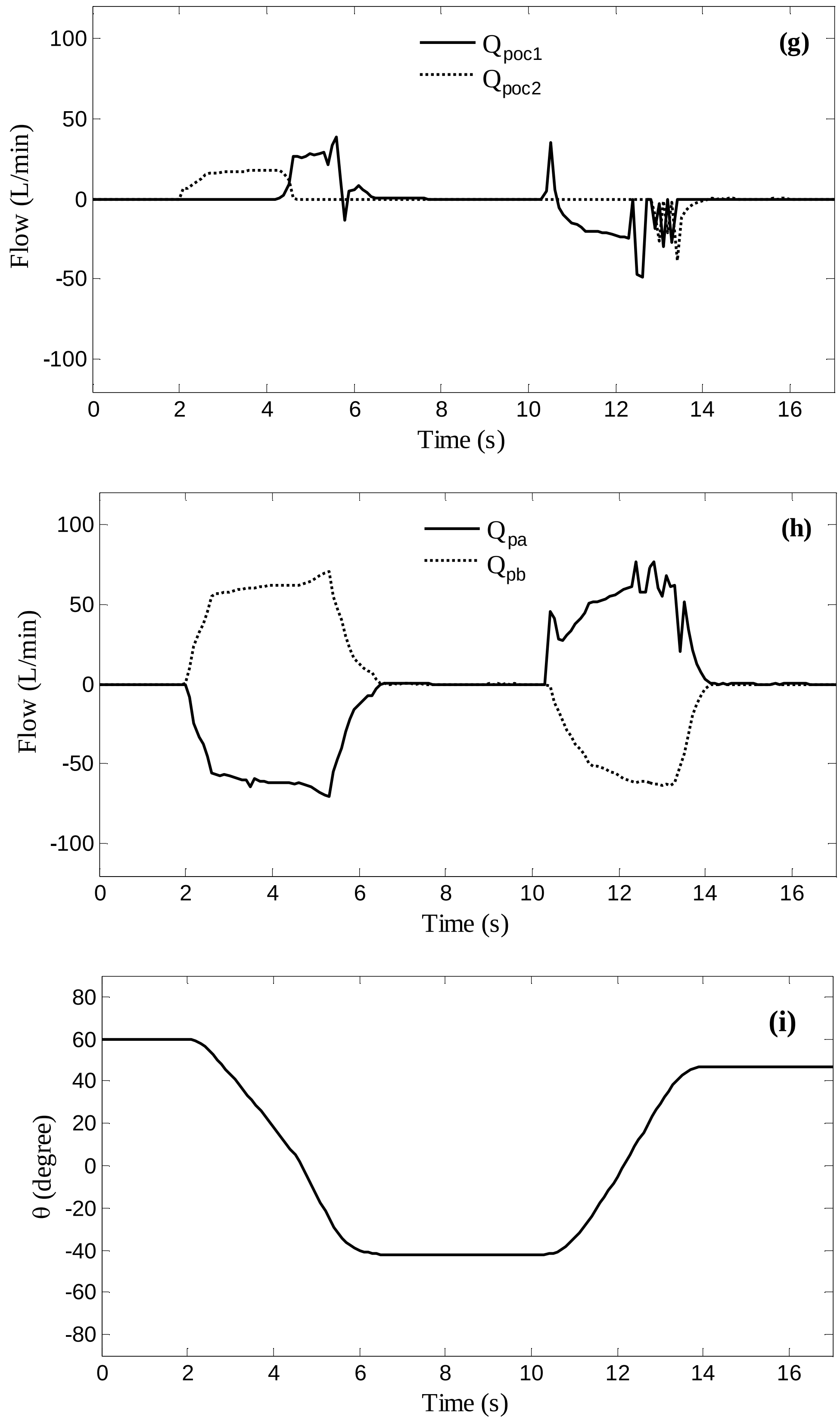

(d)
P_a
P_b
Pressure (bar)
Time (s)
(e)
P_{pa}
P_{pb}
Pressure (bar)
Time (s)
(f)
Q_b
Q_a
Flow (L/min)
Time (s)

Fig. 4.8. Contd.



Q_poc1
Q_poc2
(g)
Flow (L/min)
Time (s)
Q_pa
Q_pb
(h)
Flow (L/min)
Time (s)
(i)
θ (degree)
Time (s)

Fig. 4.8. Contd.

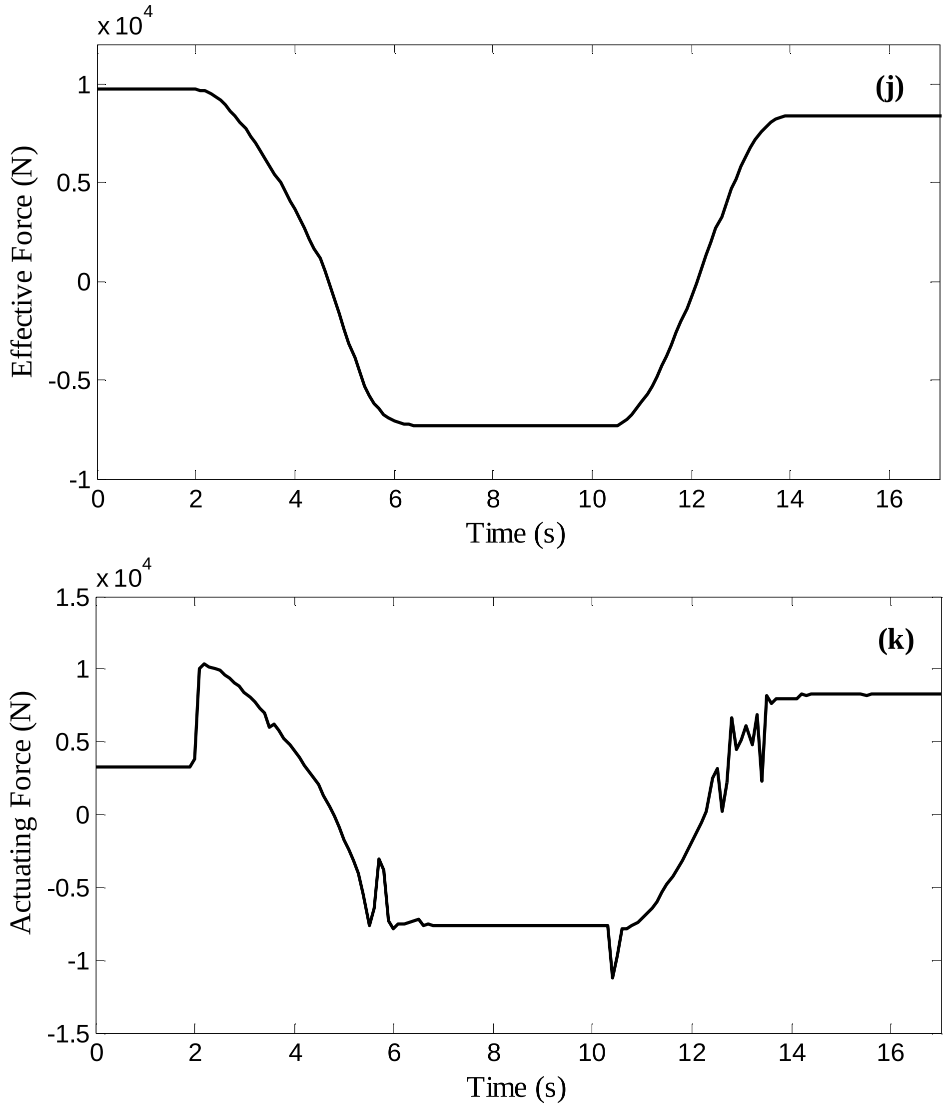


Fig. 4.8. Contd.

(iv) Case study 4

Due to the effecting $F(t)$, cylinder speed is shown to have oscillatory behavior (Fig. 4.9b). As seen in Fig. 4.9c, the cylinder displacement is not smooth. In fact, the hydraulic system cannot eliminate the effect of such an unpredictable force on the cylinder motion. During the cylinder extension, since $A_a > A_b$, the amplitude of oscillation in P_b is more than that of P_a (Fig. 4.9d). A sudden increase in the cylinder speed happens when the cylinder differential pressure $(P_b - P_a)$ becomes maximum (*e.g.*, at t = 3.7 s). At those moments, because P_a is very low, considerable low-pressurized fluid enters the system through the POC1 (Fig. 4.9g).

The pump cannot totally hold the cylinder while the cylinder is inactive (5.5 s $< t <$10.5 s). Any impulse from $F(t)$ to the cylinder compresses the fluid in one of the cylinder chambers. As the direction of $F(t)$ changes, the compressed fluid expands and pushes the cylinder into the opposite direction. Therefore, the cylinder displacement changes when the direction of $F(t)$ changes periodically (Fig. 4.9c).

During the cylinder retraction, high-pressurized flow must be removed through the POC valves. This happens when the cylinder pressures become equal or more than P_{poc} of the POC valves. When the cylinder pressure becomes lower than the P_{poc} of the POC valves, the POC valves get closed. As a result, the flow amount exited to the low-pressure system from through the POC valves during the cylinder retraction is lower than the flow amount entered to the pump line, during the cylinder extension (see Fig. 4.9g).

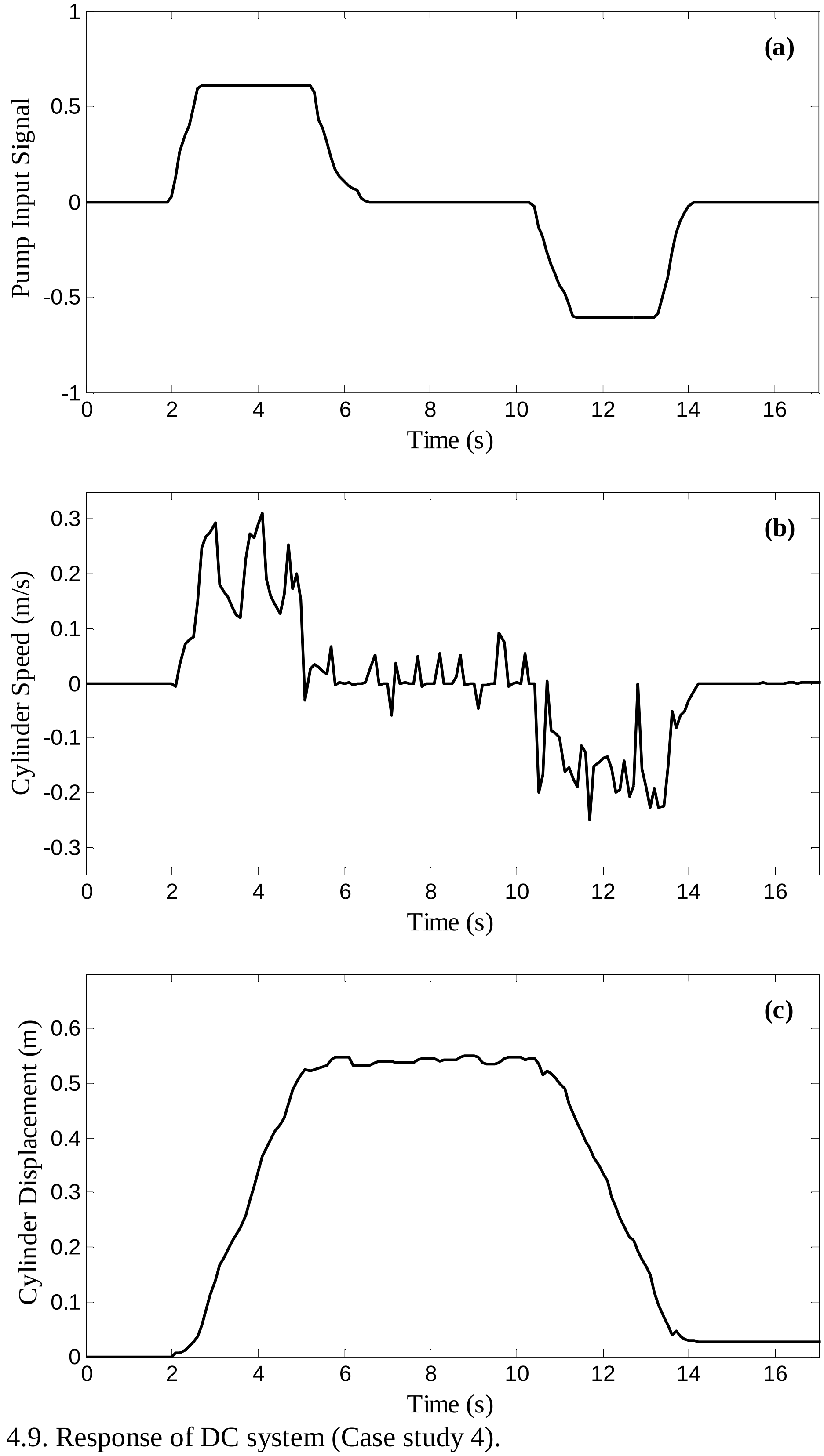
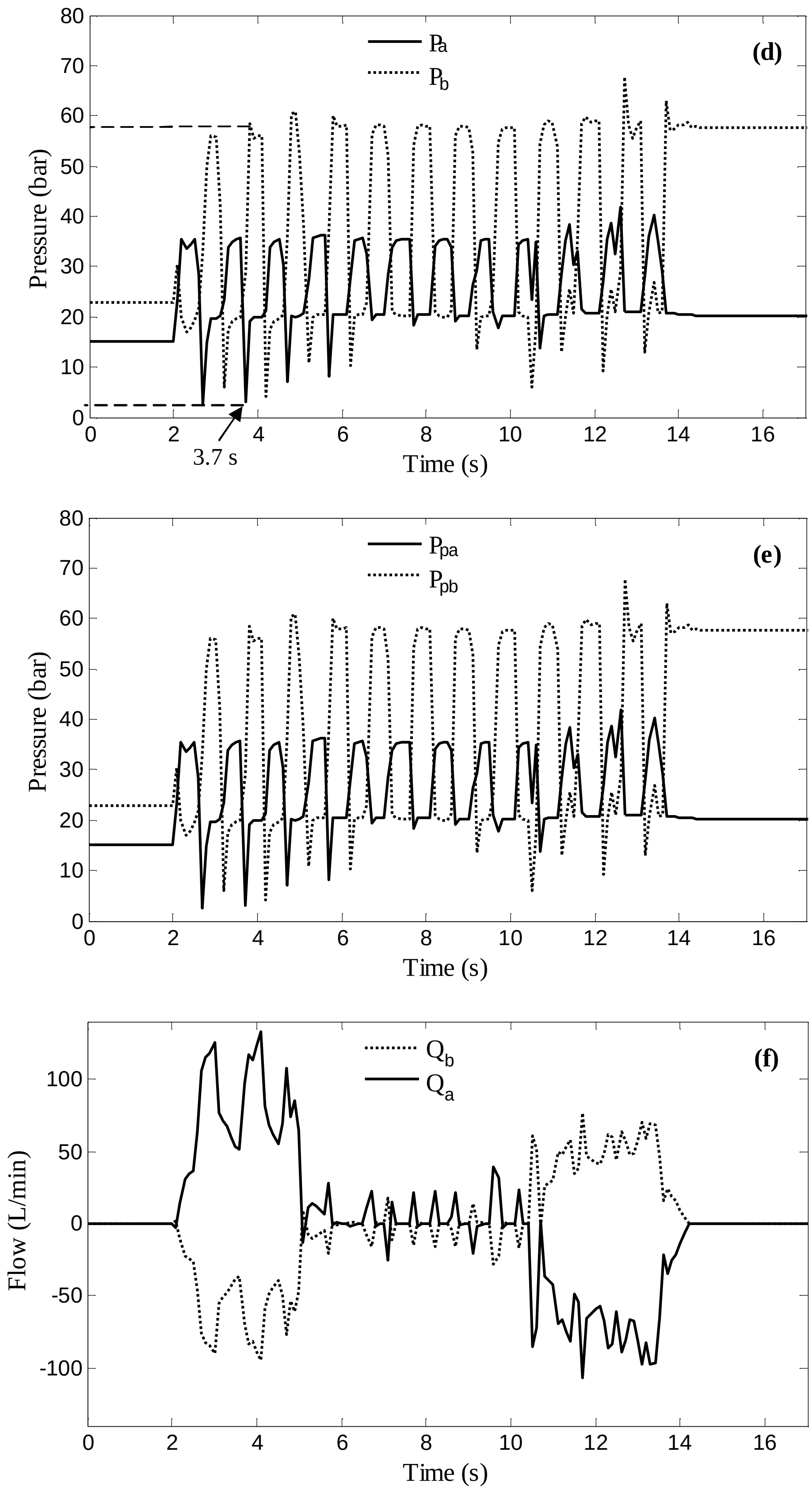
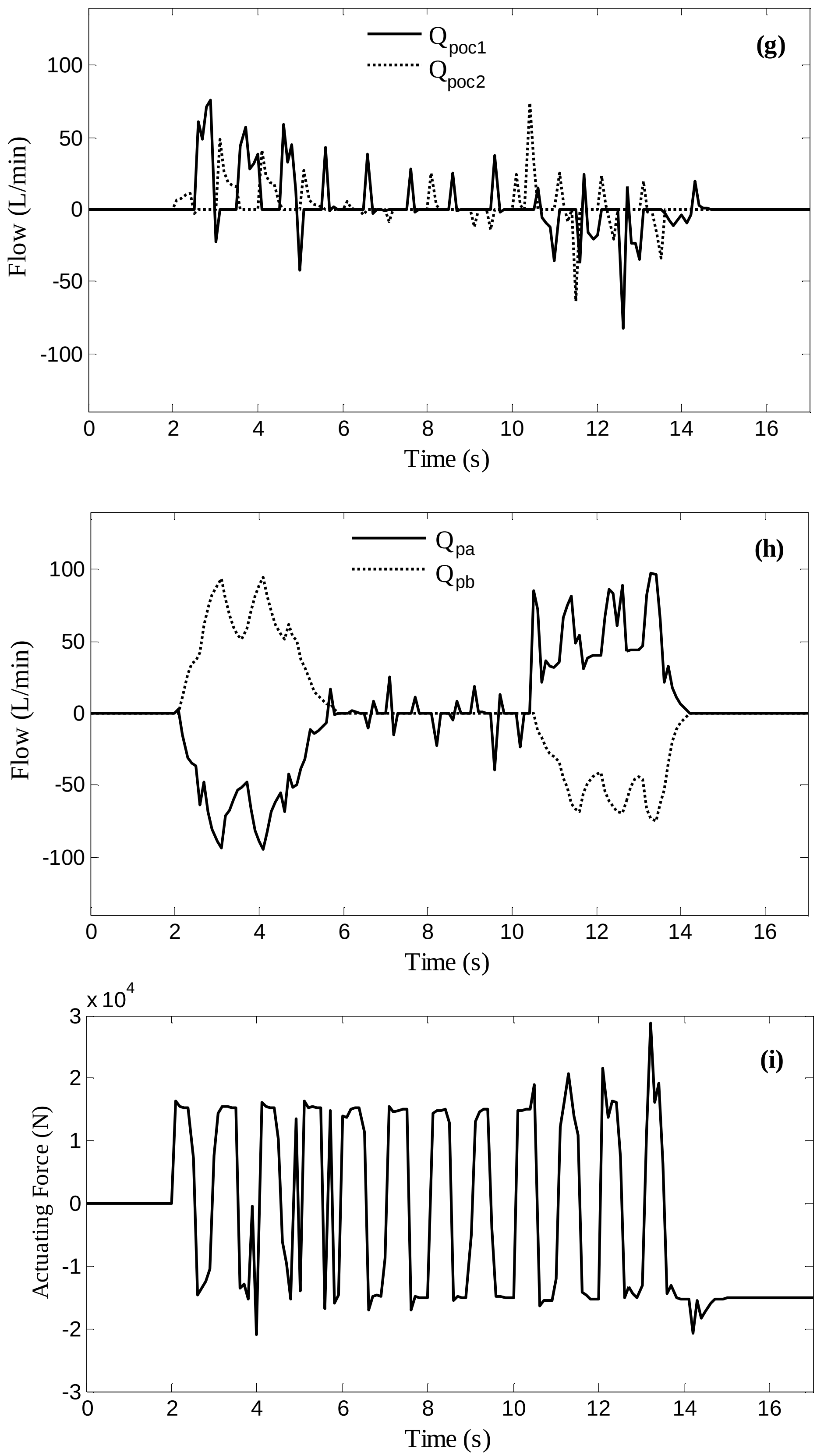


Fig. 4.9. Response of DC system (Case study 4).



(d)
Pa
Pb
Pressure (bar)
Time (s)
3.7 s
(e)
Ppa
Ppb
Pressure (bar)
Time (s)
(f)
Qb
Qa
Flow (L/min)
Time (s)

Fig. 4.9. Contd.



Q_poc1
Q_poc2
(g)
Flow (L/min)
Time (s)
Q_pa
Q_pb
(h)
Flow (L/min)
Time (s)
x 10^4
(i)
Actuating Force (N)
Time (s)

Fig. 4.9. Contd.

4.3.1. Discussions

The dynamic performance of the DC system was observed. In case studies 1 and 2, the system inertia caused abrupt fluctuations in the cylinder speed, when the pump began moving the cylinder. In Case study 3, system inertia and mode changing contributed to the oscillation of the cylinder speed. In Case study 4, direction changes in the switching force led to oscillation of the cylinder speed. Overall, according to the results, the DC pump-controlled system could not eliminate the influence of unpredictable forces and system inertia on the cylinder motion.

4.4. Performance of Wendell Pump-Controlled System

Wendell pump-controlled system has two POC valves on both sides of the cylinder (POC2 and POC3 in Fig. 3.6). Although, these POC valves prevent from uncontrolled movement of the cylinder, they interfere with the pump-controlled system and lead to oscillation of the cylinder speed.

(i) Case study 1

During the cylinder extension, the cylinder speed shows oscillatory behavior (Fig. 4.10b). This behavior is due to the frequent closing and opening the POC4. At the beginning of the cylinder displacement, POC4 is closed preventing the cylinder to move. During this period, P_a rises up to P_{poc4} to open the POC4. Once POC4 opens via the pilot line, the pressurized fluid exits from the rod side of the cylinder with high flow rate (Fig. 4.10b). The sudden increase in flow Q_b into the pump line reduces P_a to be lower than P_{poc4} (Fig. 4.10d). Consequently, POC4 closes and blocks the path of rod side fluid to the pump line for another time. This scenario is repeated during the cylinder extension and causes oscillations in the cylinder speed and pressure (Figs. 4.10b and 4.10d).During the cylinder retraction, the speed is constant except for the end of the cylinder retraction (Fig. 4.10b). The abrupt fluctuation in the cylinder speed originates from the effect of system inertia which was discussed in Section 3.2. P_b is high enough to open the POC3 via the pilot line. The rod side of the cylinder usually provides higher pressure than the piston side (if no load is applied to the cylinder). Thus, the hydraulic system does not take time to build the cylinder pressure as experienced during cylinder extension. $Q_a - Q_b$ is removed from the piston side through the POC1 (Fig. 4.10g).

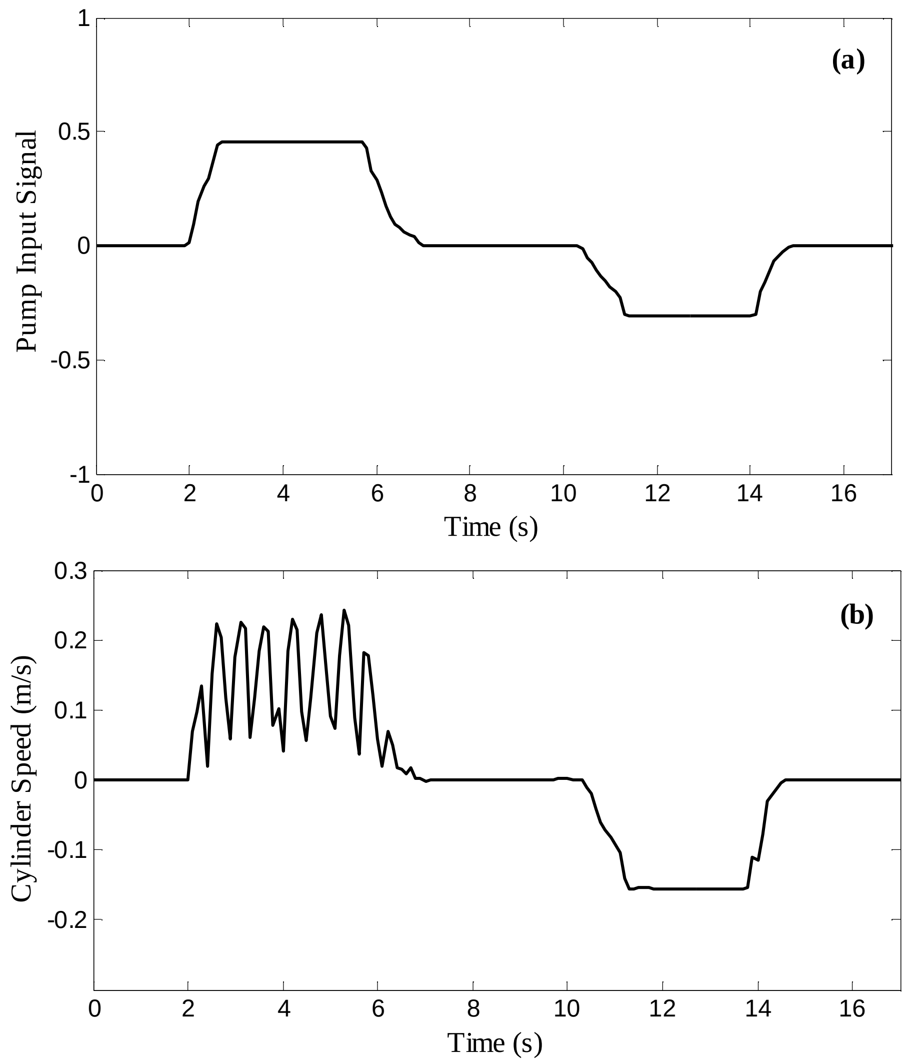


Fig. 4.10. Response of Wendell system (Case study 1).

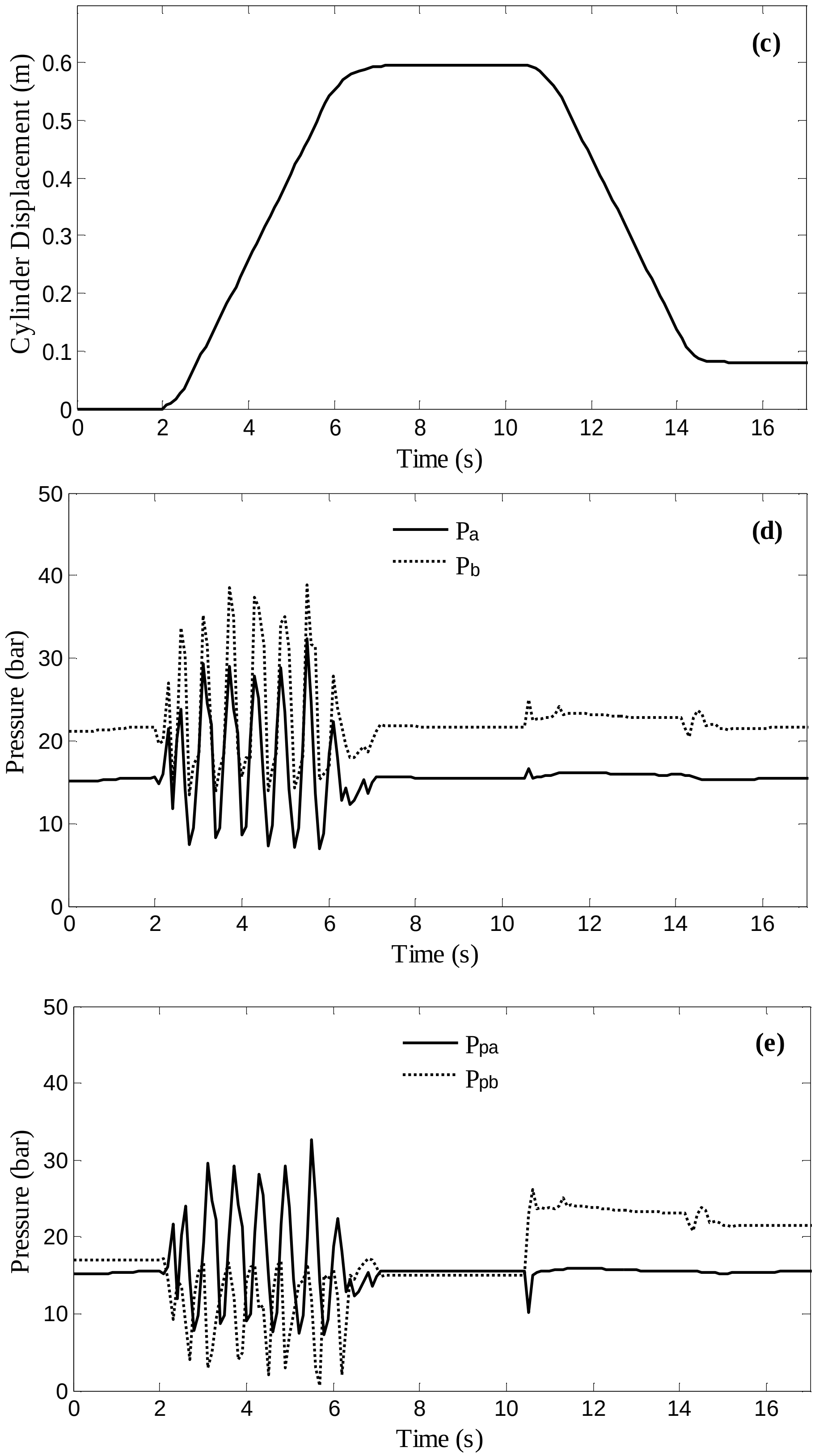


Fig. 4.10. Contd.

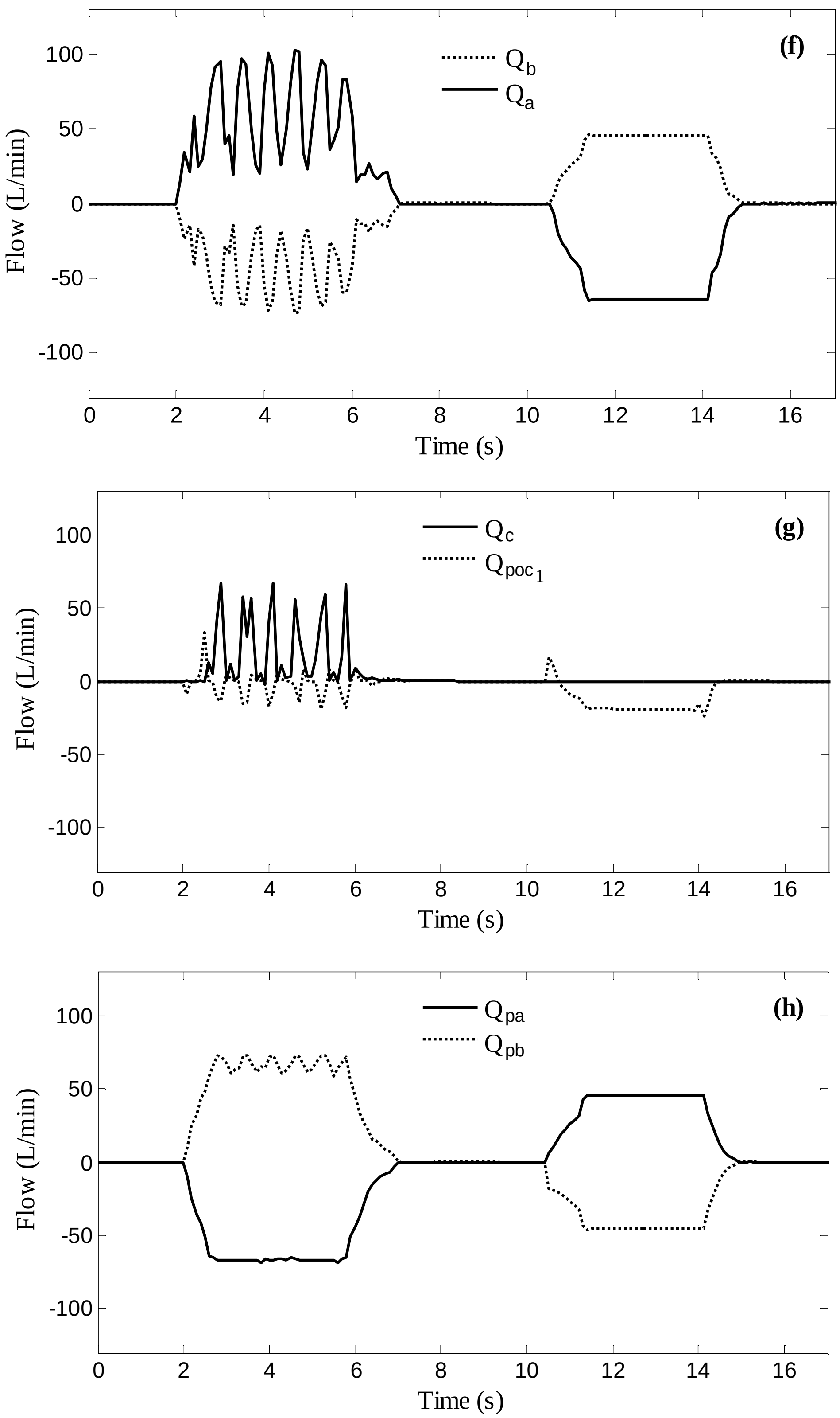


Fig. 4.10.Contd.

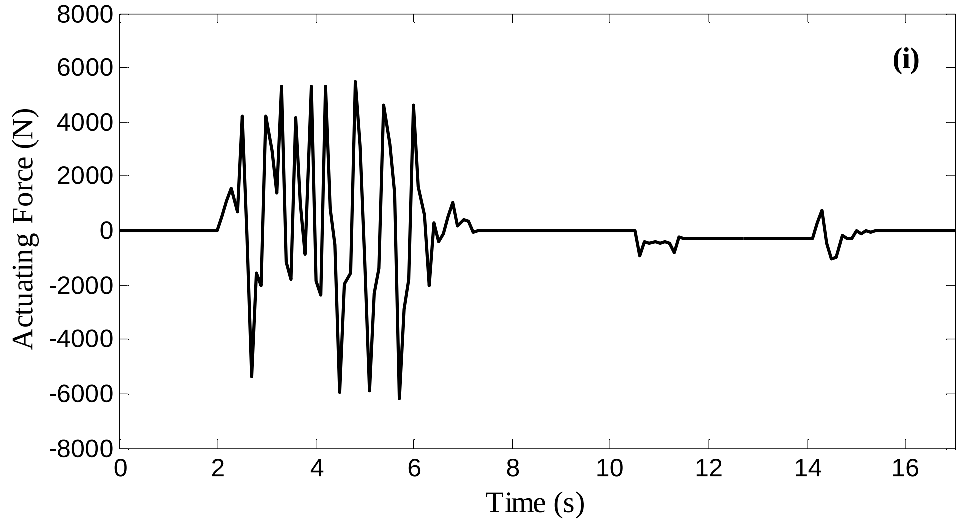


Fig. 4.10. Contd.

(ii) Case study 2

During the cylinder extension, the cylinder pressures and the cylinder speed oscillate (Figs. 4.11b and 4.11d). This behavior is related to opening and closing the POC4 which was discussed earlier. Since the effective force is higher than the one in Case study 1, the oscillation of the cylinder speed is relatively higher. Furthermore, by comparing Figs. 4.10c and 4.11c, the cylinder displacement in Case study 1 is observed smoother than the one in Case study 2. Therefore, the higher load causes more oscillation in the cylinder speed. During the cylinder retraction, cylinder displacement is smooth (Fig. 4.11c).

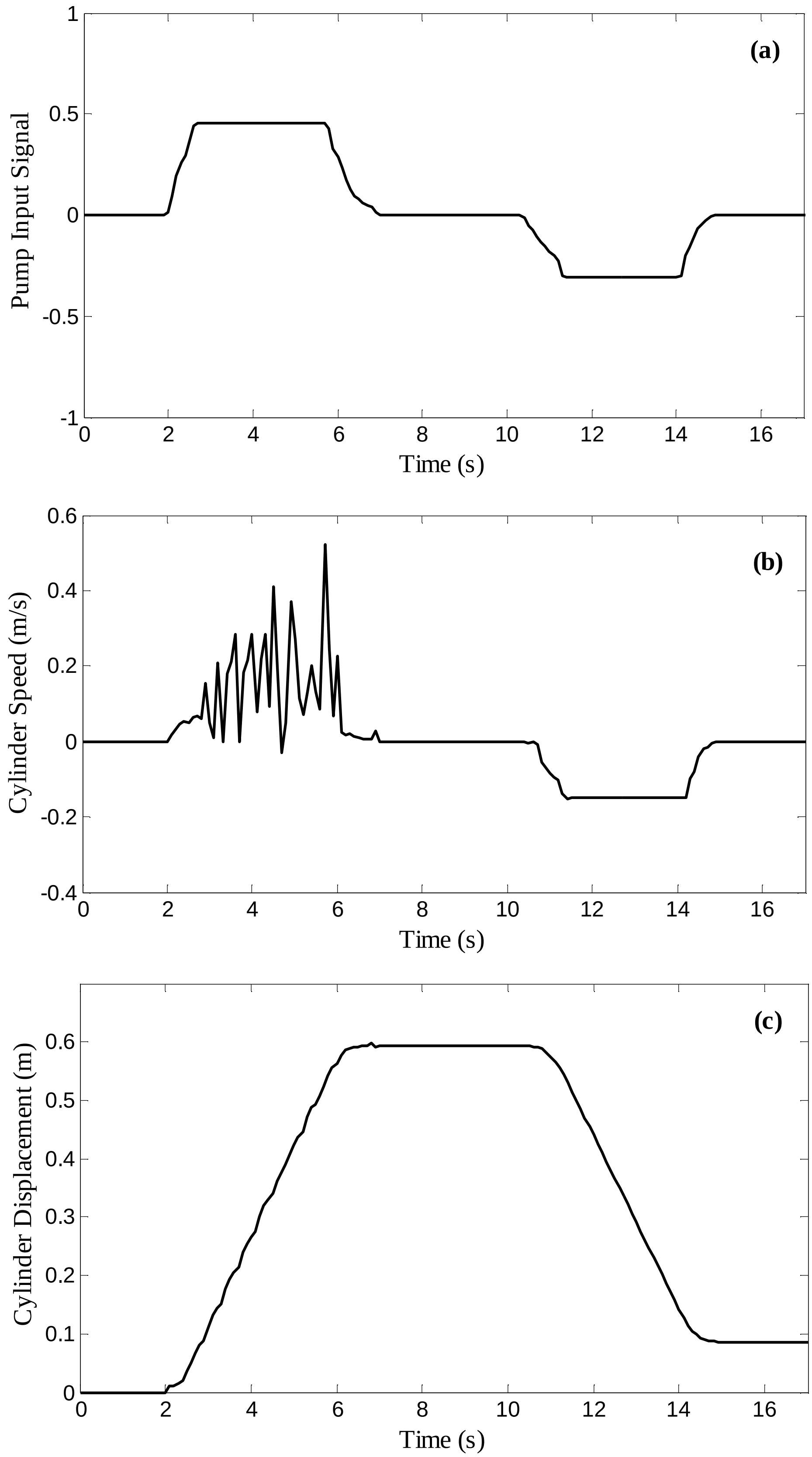


Fig. 4.11. Response of Wendell system (Case study 2).

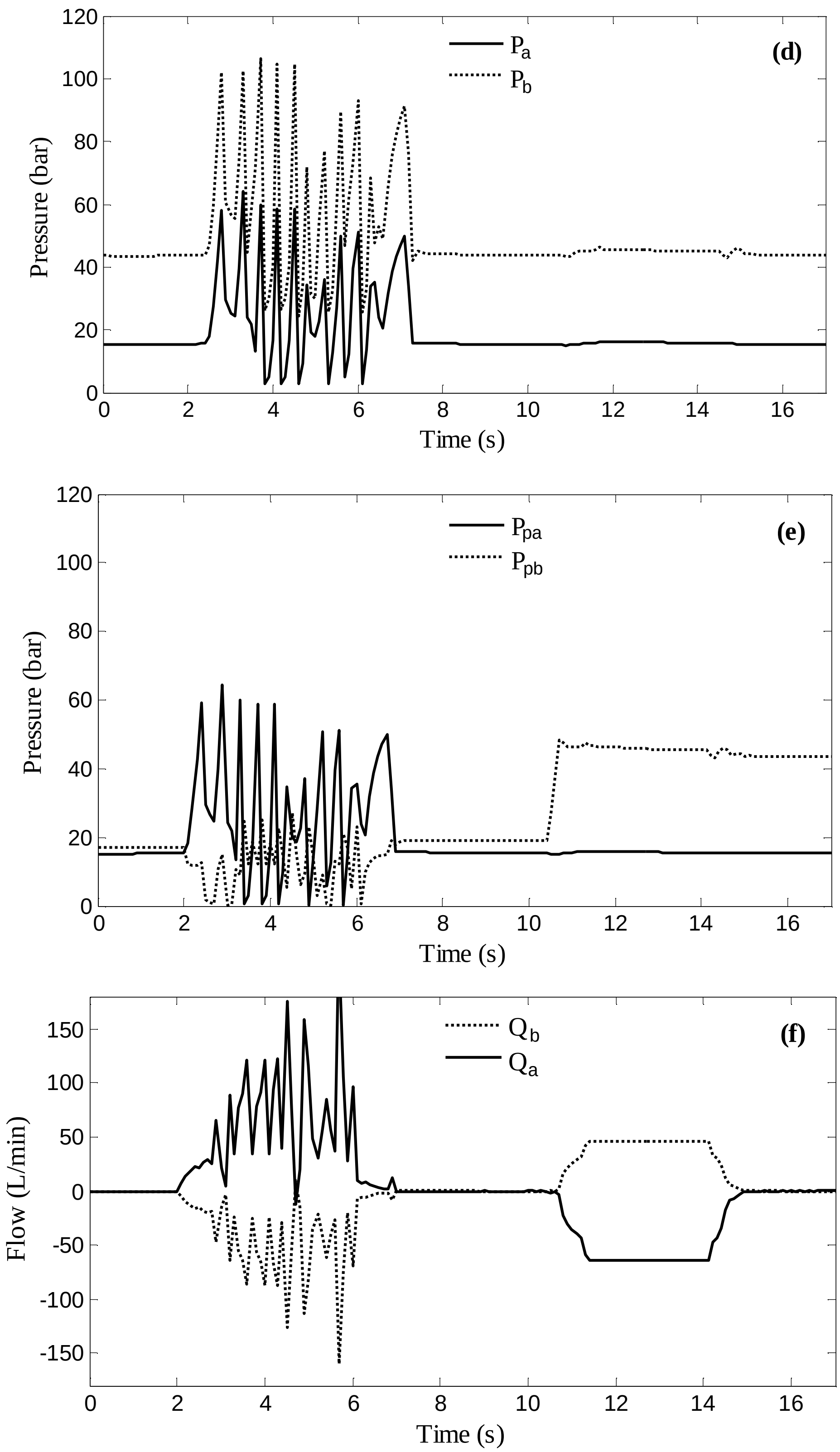


Fig. 4.11. Contd.

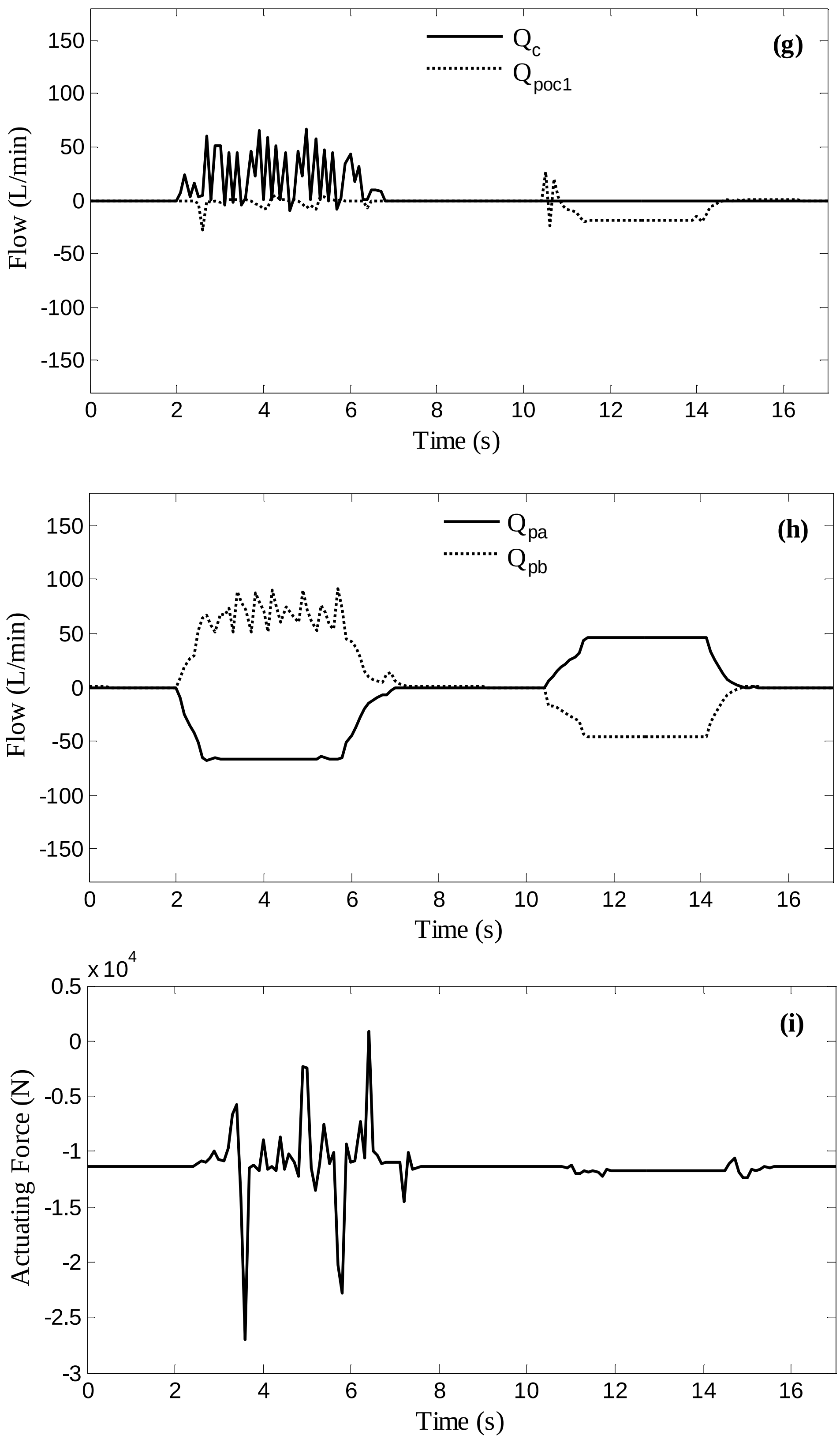


Fig. 4.11. Contd.

(iii) Case study 3

During the cylinder extension and rotation from $\theta = 60°$ to $\theta = 20.2°$, the cylinder speed is proper relatively (Figs. 4.12b and 4.12i). P_a is high enough to keep the POC4 open $(P_a \geq P_{poc4})$. When θ= 20.2°, the POC4 closes, since the effective force $(Mg \sin\theta)$ is not heavy enough to provide enough pilot pressure. The POC4 is kept closed until the pump flow increases P_a up to P_{poc4} to open POC4. As Q_b passes through the POC4, P_a reduces and becomes lower than P_{poc4}. As a result, the POC4 gets closed for another time. During 20.2° $< \theta <$-39°, because of closing and opening the POC4, cylinder speed and pressure show oscillatory behavior (Figs. 4.12b and 4.12d).

During the cylinder retraction, and rotation from $\theta =$-39° to $\theta = 30°$, cylinder speed is constant. P_b is high enough to keep the POC3 open via the pilot line. However, during the cylinder rotation from $\theta = 30°$ to $\theta = 40°$, the cylinder speed oscillates. As observed before, this behavior originates from the insufficiency of P_b to keep the POC3 open ($P_b \leq P_{poc3}$). Since the effective force tends to assist in retraction of the cylinder, P_b decreases and becomes lower than P_{poc3}. The POC3 gets closed until P_b increases by the pump flow.

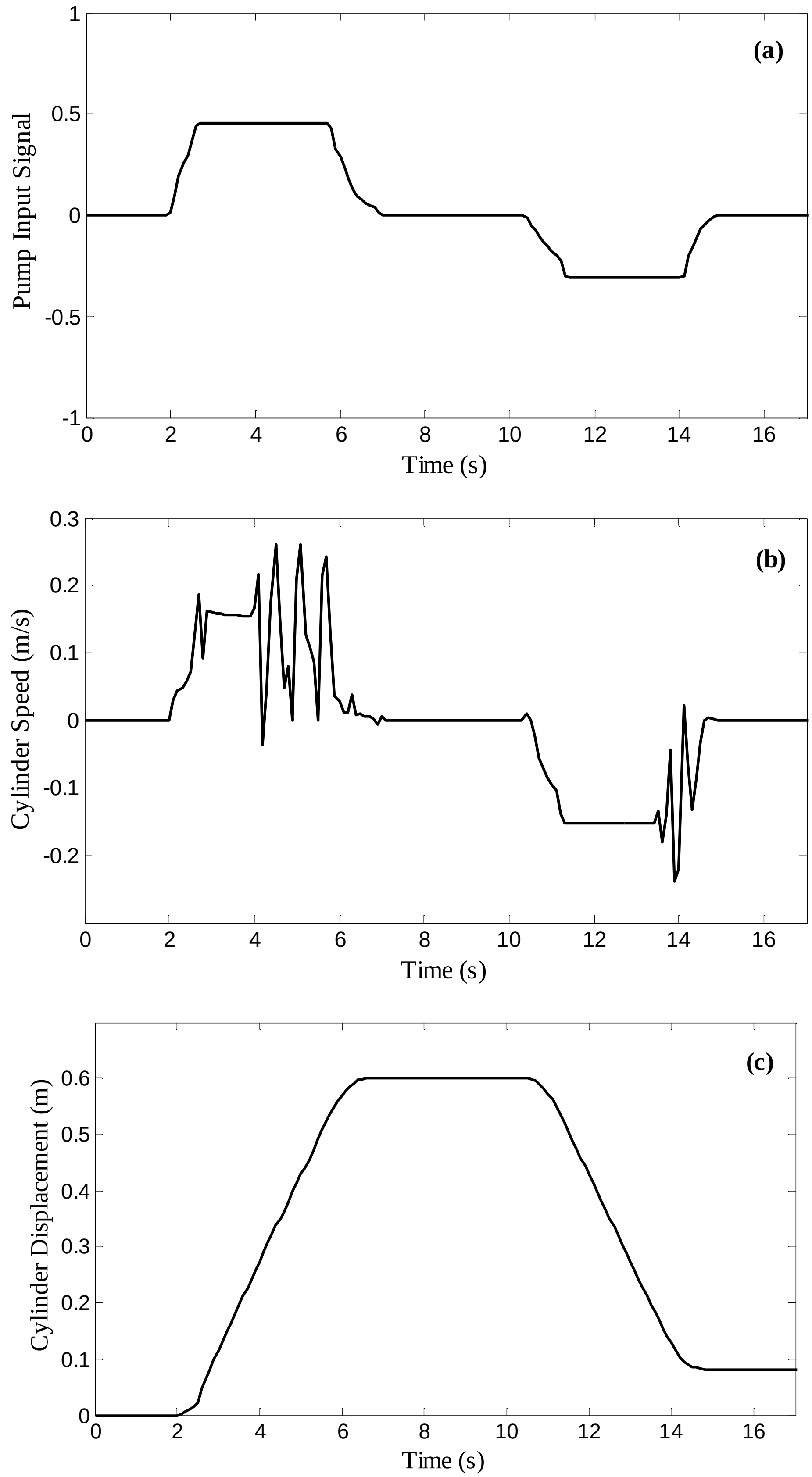


Fig. 4.12. Response of Wendell system (Case study 3).

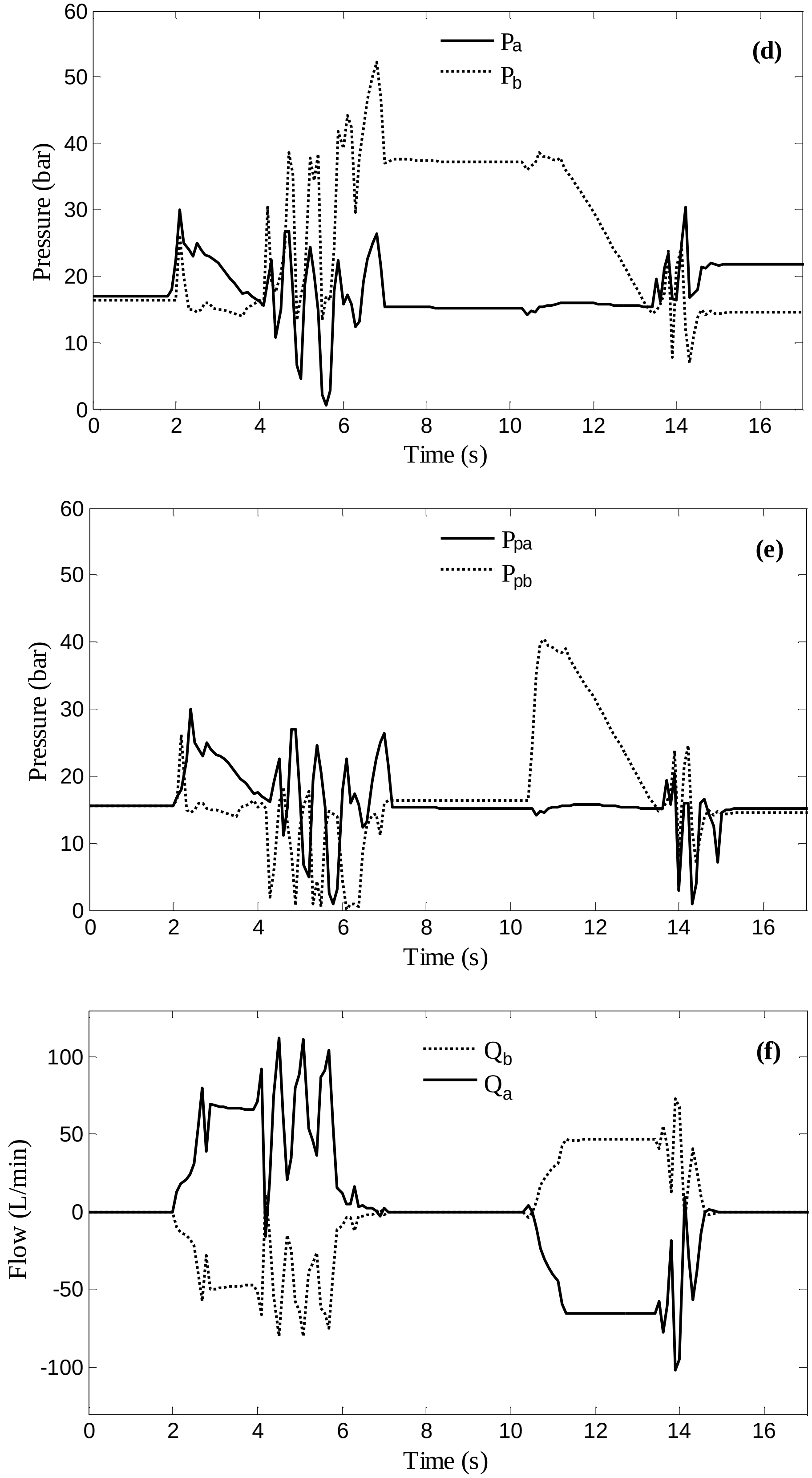


Fig. 4.12. Contd.

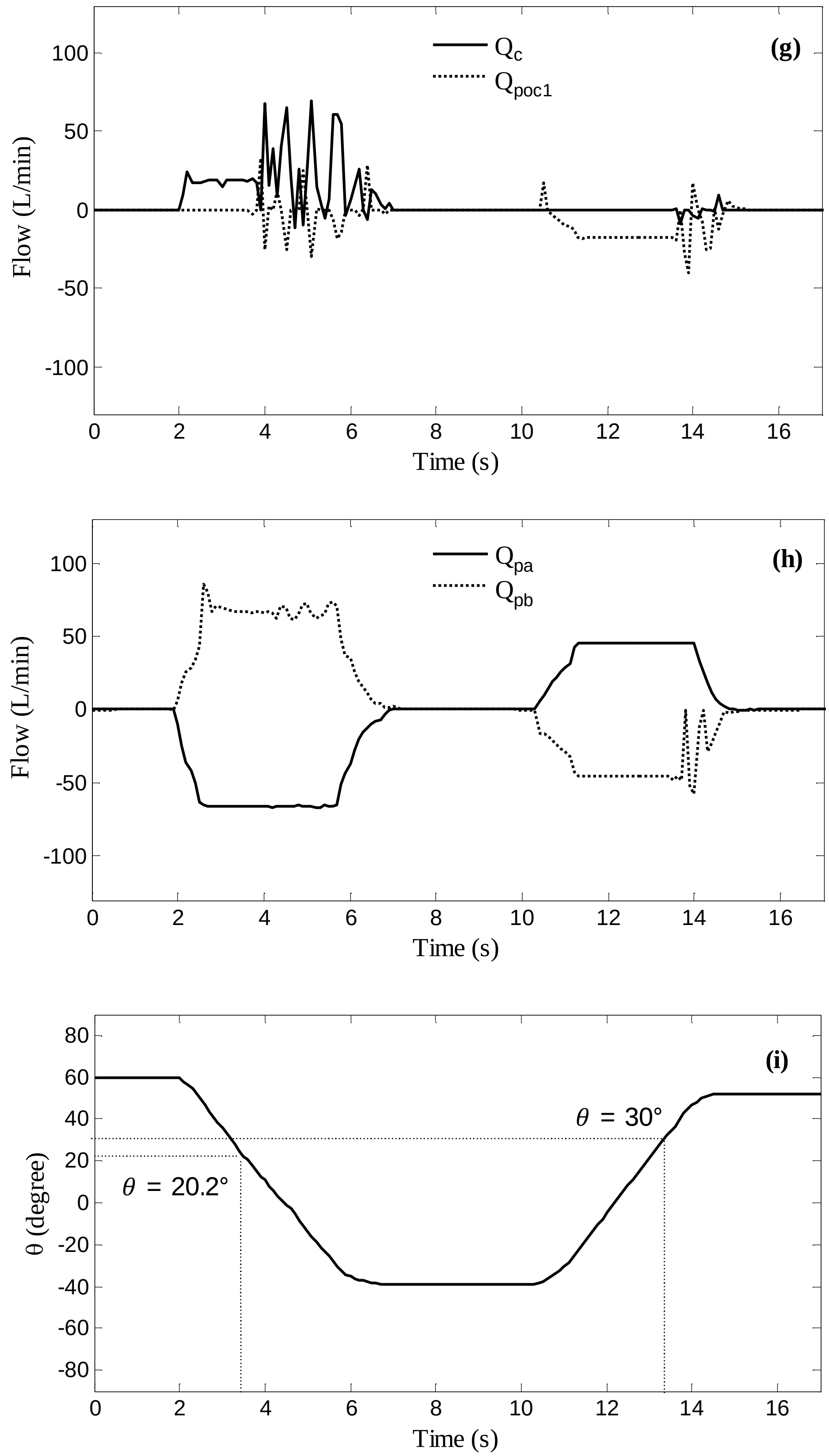


Fig. 4.12. Contd.

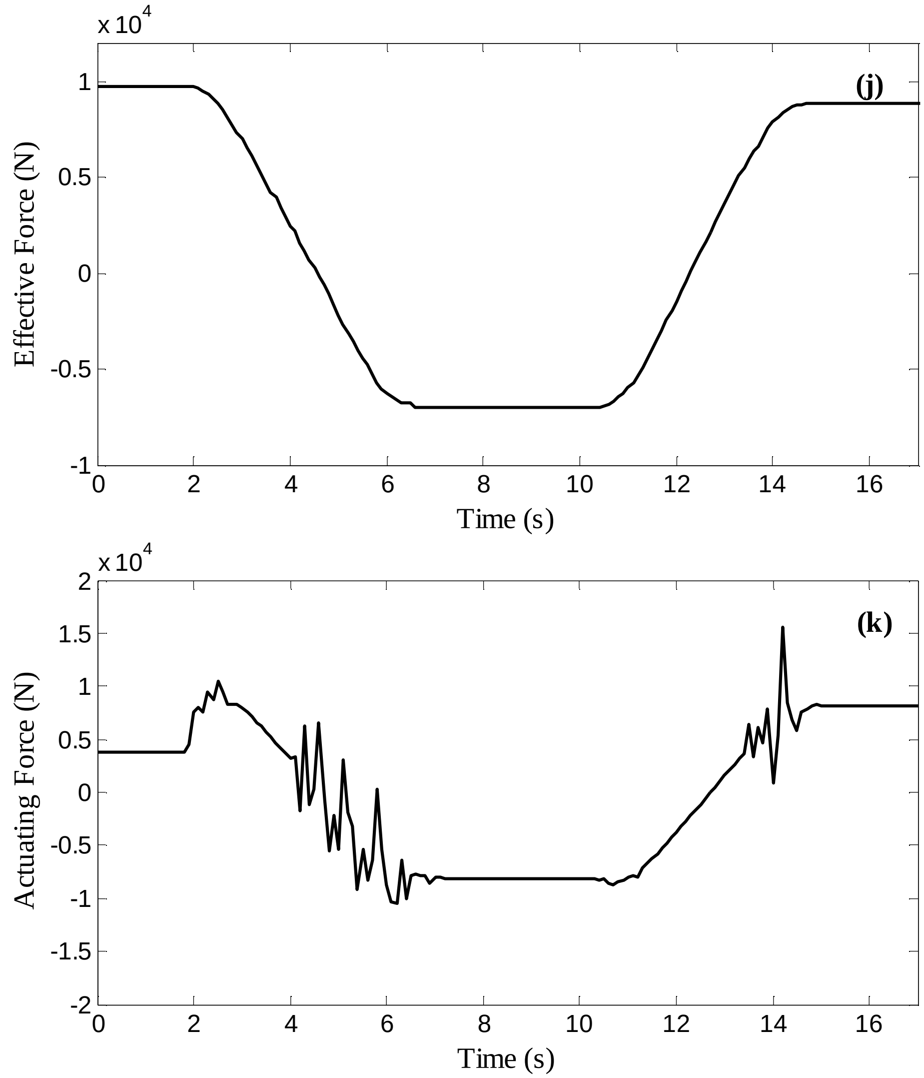


Fig. 4.12. Contd.

(iv) Case study 4

As seen in Fig. 4.13b, the cylinder speed oscillates, during the operation. The cylinder displacement rises up along a straight line; however, some small chattering is observed during the motion (Fig. 4.13c). During $6\ s < t < 11\ s$, the cylinder displacement is smooth. POC2 and POC3 prevent from uncontrolled movement of the cylinder, originating from $F(t)$. In this case study, compared to the DC response, the dynamic performance of this system is better.

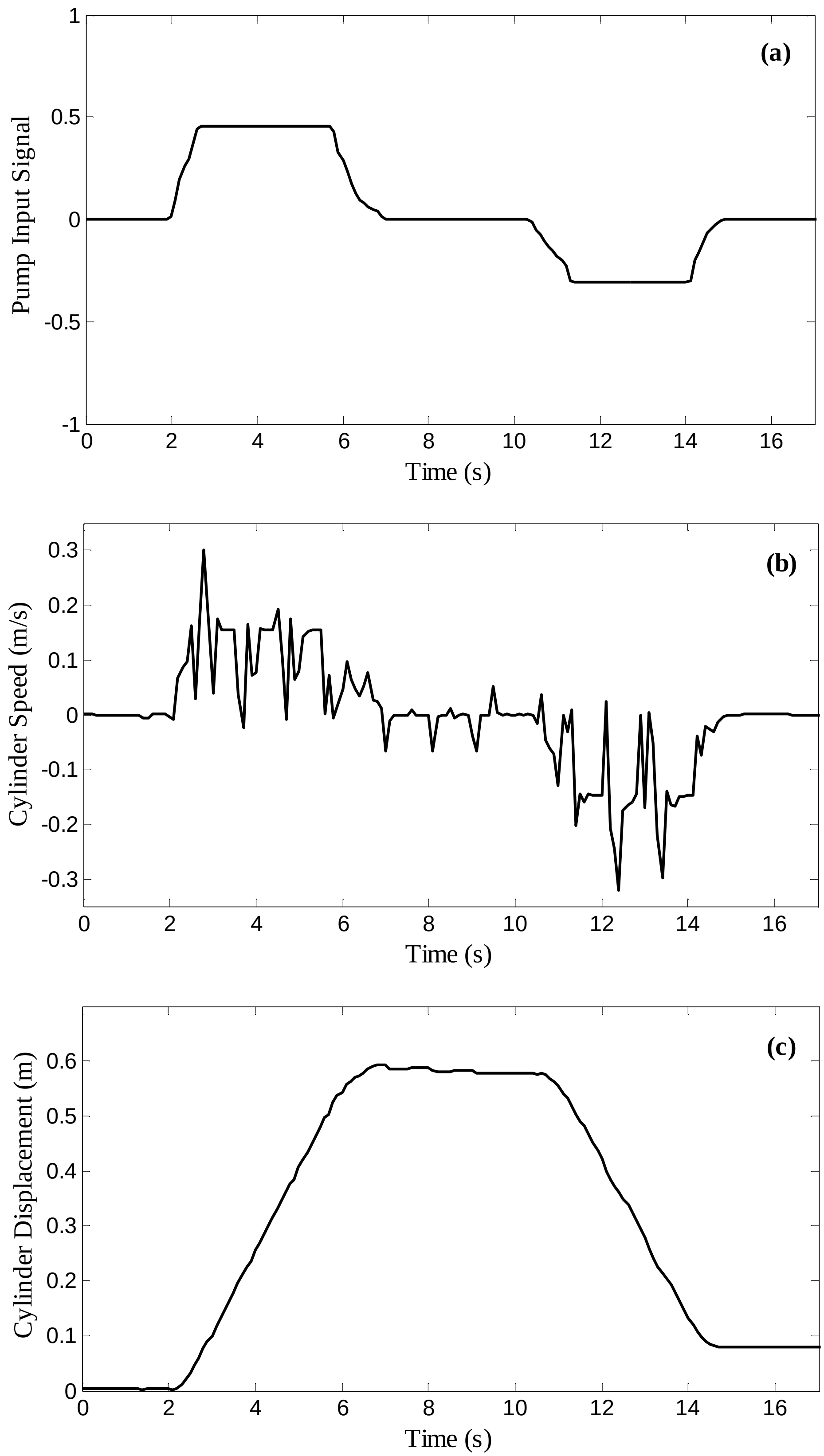


Fig. 4.13. Response of Wendell system (Case study 4).

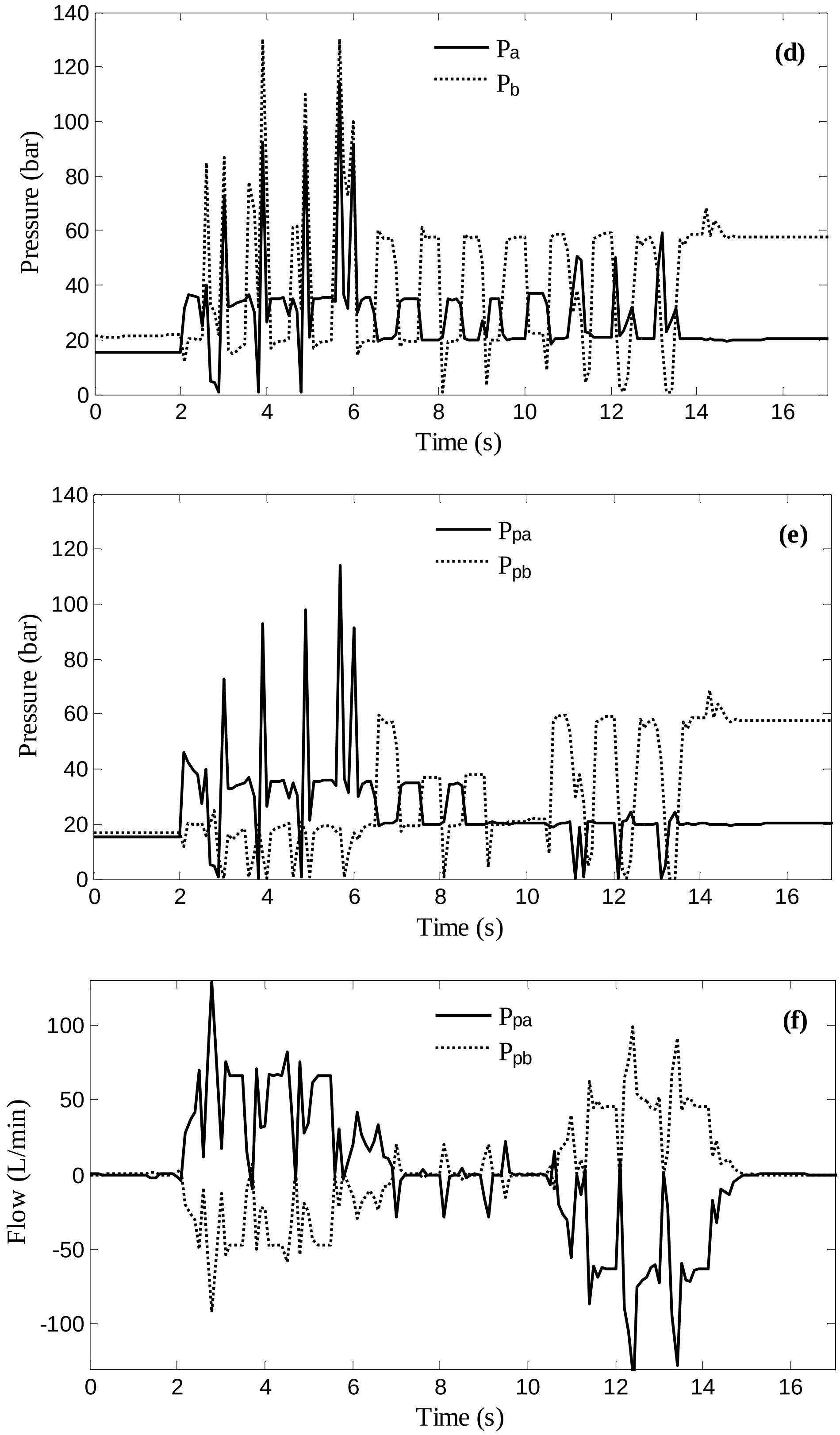


Fig. 4.13. Contd.

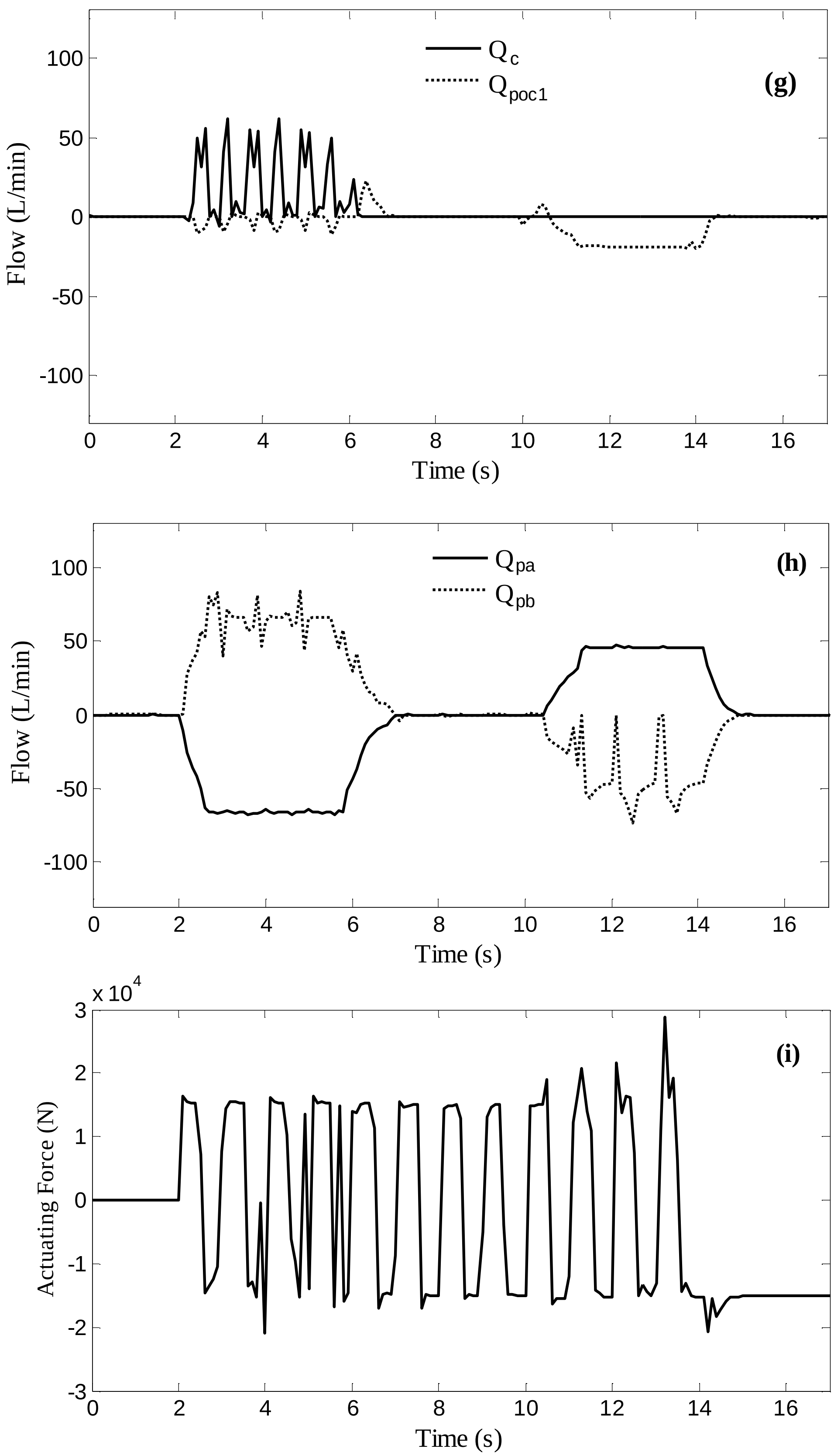


Fig. 4.13. Contd.

4.4.1. Discussions

The dynamic performance of the Wendell pump-controlled system was observed. In all case studies, POC3 and POC4 caused oscillations in the cylinder speed. However, they prevent from uncontrolled movement of the cylinder, as seen in Case study 4. In case studies 1 and 2, the system inertia did not influence on the cylinder speed. However, the system inertia caused fluctuation in the speed of the cylinder controlled by the DC circuit. The performance of the Wendell system in reducing the influence of unpredictable forces on the cylinder dynamic behaviour was better than that of the DC system, as seen in case studies 3 and 4.

4.5. Summary

In this chapter, four case studies were presented to investigate the performance of the LS valve-controlled system and two types of pump-controlled systems (DC and Wendell). The evaluation was based on the simulation studies carried out using Automation Studio® software. For all case studies, the LS valve-controlled system showed good performance in controlling the cylinder. Based on dynamic behavior analysis, the DC pump-controlled system could not eliminate the influence of the system inertia on the cylinder motion. This behavior was observed in case studies 1 and 2, when the pump began moving the cylinder. Moreover, cylinder speed showed oscillatory behaviour after the pump changed the operational mode (Case study 3). In all case studies, in Wendell pump-controlled system, POC valves connected to both sides of the cylinder caused the cylinder speed to become oscillatory. However, these valves prevent uncontrolled displacement of the cylinder, as observed in Case study 4. As compared to the DC system, Wendell's system had better performance in eliminating the influence of system inertia and external force on the cylinder motion (Case study 4).

5. CONCLUSIONS

5.1. Contribution of This Thesis

This thesis evaluated the energy consumption and dynamic performance of hydraulic systems in mobile machines. The energy consumption of three types of valve-controlled systems (open-center, OC, constant-pressure, CP, and load sensing, LS) and two types of pump-controlled systems (displacement-controlled actuation, DC, developed by Rahmfeld-Ivantysynova and hydraulic regenerative developed by Wendell) were evaluated. The evaluation was based on the simulation of a cyclic test using Automation Studio software. The dynamic performance of the LS valve-controlled system as well as Wendell and DC pump-controlled systems were evaluated, based on case studies which were also simulated using Automation Studio.

On the energy consumption front, energy distributions were graphically presented by Sankey diagram. Result of power consumption and flow circulation of the Wendell pump-controlled system showed that pilot-operated check valves at both sides of the cylinder prevent the bidirectional pump to work as a hydraulic motor. As a result, the energy recovery may not be obtained considering the check valves at the sides of the cylinder in Wendell's circuit, are pilot-operated. Energy consumption of the Wendell pump-controlled system was also compared, for the first time, with that of the DC pump-controlled systems. Based on a typical test designed in this thesis (extending and retracting under a constant load), the energy consumption of Wendell system was shown to be 59% higher than that of the DC pump-controlled system. The OC valve-controlled system exhibited the highest energy consumption (306 kJ). The energy consumption of

CP, LS, DC, and Wendell systems were also found to be 40.6%, 17.7%, 7.3%, and 4.6% lower than that of OC system, respectively. Among the chosen hydraulic systems, the DC pump-controlled system showed the lowest energy consumption (14.2 kJ).

On the dynamic performance front, four cases studies were developed. Case study one was designed to investigate the influence of system inertia on the response. Case study two was designed to evaluate the effect of gravitational load on the response of the single-rod hydraulic actuation system. Case study three was to evaluate the performance when the actuator operational mode changes. Case study four was designed to evaluate the response of the system given unpredictable force is applied to the rod end of the single-rod hydraulic cylinder. To the best of the author's knowledge, prior to this work, no other study evaluated the dynamic performance of hydraulic systems using the mentioned cases studies. Based on the results from these case studies, the LS valve-controlled system showed good performance in controlling the cylinder. Using the DC pump-controlled system, the system inertia exhibited abrupt fluctuations in the cylinder speed, when the cylinder began moving or changing the direction. Further, cylinder speed showed oscillatory behaviour when the actuator operational mode changed from extend resistive to extend overrunning. Wendell pump-controlled system also showed oscillations, in all case studies. However, the pilot-operated check valve at the cylinder side in Wendell system, prevent uncontrolled movement of the cylinder, which is described.

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APPENDICES

Appendix A: Calculation of Energy Consumption

1. Equations

System power loss	$$P_{loss} = \sum_{i=1}^{n}(P_{in} - P_{mech})$$
Engine power transmission	$$P_{eng} = M_{eng}\omega_{eng}$$
Engine energy transmission	$$E_{eng} = \int P_{eng}\, dt$$
Pump output energy	$$E_p = \int Q_p P_p dt$$
Pump energy loss	$$E_{pl} = E_{eng} - E_P$$
Load power	$$P_{load} = (P_a\, A_a - P_b\, A_b)v$$
Useful work	$$E_{load} = \int P_{Load} dt \quad \text{when } P_{load} \geq 0$$
Directional valve power loss	$$P_v = (P_p - P_a)Q_a + P_b\, Q_b sgn(Q_b) \quad \text{when cylinder extends}$$ $$P_v = (P_p - P_b)\, Q_b + P_a Q_a sgn(Q_a) \quad \text{when cylinder retracts}$$
Directional valve energy loss	$$E_v = \int P_v\, dt$$
Relief valve power loss	$$P_r = P_r\, Q_r$$
Relief valve energy loss	$$E_r = \int P_r\, dt$$
Cyclic efficiency	$$\eta_{cy} = \frac{\int P_{load} dt}{\int P_{eng}\, dt} \quad \text{when } P_{load} \geq 0$$

2. Calculations

This section explains how to calculate the pump output energy and directional valve loss in a typical LS valve-controlled system using Automation Studio, Excel, and Matlab.

2.1. *Calculation of Pump Output Energy:*

1. Extract the data of the pump pressure and flow from Automation Studio to Excel (see Fig. A.1.):

2. In Excel, multiply pump pressure with pump flow using the following equation:

$$Power(t) = P_p(t)Q_p(t)$$

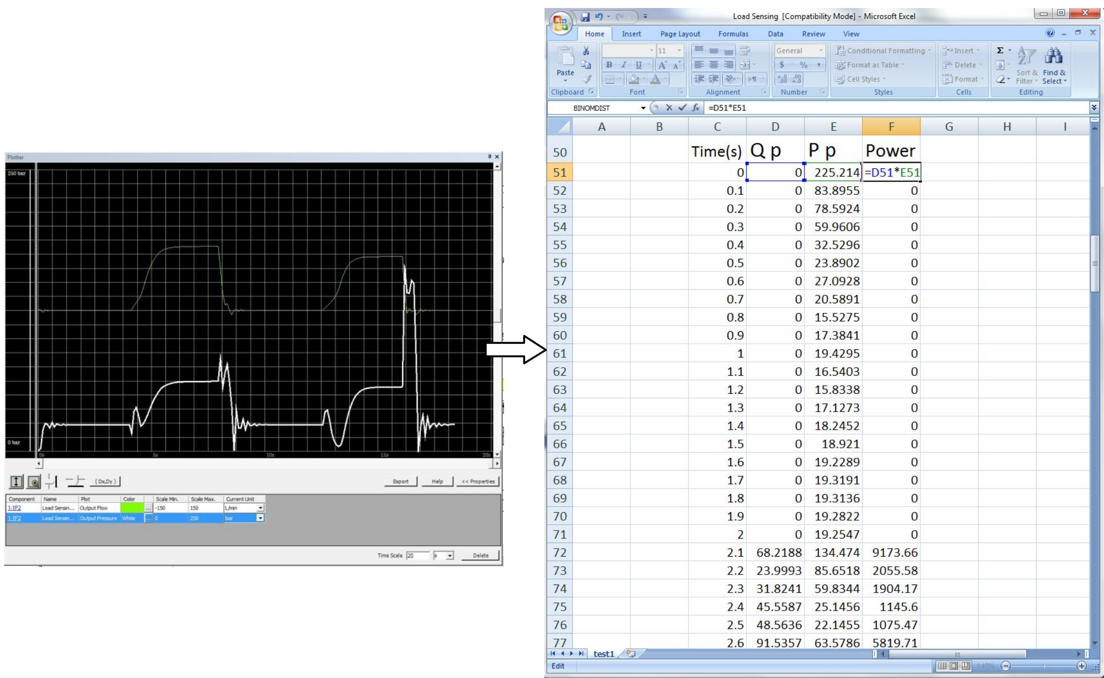

Fig. A.1. Screenshot of plot in Automation Studio (Left) and worksheet in Excel (Right).

3. Open the Excel File in Matlab and name it as “Power”:

Power= xlsread('Load Sensing.xls');

4. In Matlab, claculate the pump output energy using *Trapz* command as bellow:

% Comment: $E_p = \int Power(t)dt$

```
Ep = trapz(Power(:,3),Power(:,6))
= 40.58 kj
```

where time and power is the third and sixth column of the “Load Sensing” excel file, respectively.

$Z = trapz(X, Y)$ Computes the integral of Y with respect to X using trapezoidal integration [31].

2.2. *Calculation of Directional Valve Energy Loss:*

1. Extract the data of Q_a, P_a, Q_b, P_b , and P_p from Automation Studio to Excel.

2. Calculate $P_v(t)$ in Excel using the following equations introduced in Section 3.1.2:

$P_{v1} = (P_p - P_a)Q_a + P_b Q_b sgn(Q_b)$ when cylinder extends

$P_{v2} = (P_p - P_b) Q_b + P_a Q_a sgn(Q_a)$ when cylinder retracts

$P_v(t) = P_{v1}(t) + P_{v2}(t)$

3. Save the Excel file as “Valve Loss”. In the Excel file, time is in column A and $P_v(t)$ is in column B.

4. Open the Excel file in Matlab and calculate the valve energy loss using *Trapz* command.

```
ValveLoss = xlsread('Valve Loss.xls');
% Comment: Ev = ∫ Power(t)dt
   Ev = trapz(ValveLoss(:,1),ValveLoss(:,2))
      = 39.43 Kj
```

Appendix B: Sankey Diagram

Sankey diagram is a widely used flow visualization tool which represents visually various energy inputs, outputs, and losses, so that overall energy improvements can be done in accordance with priority or preference. Also, it can be used to visualize material balance, and cost transfer. The Sankey diagram can help in locating dominate contribution of an energy component to an overall flow of energy. Captain Henry Matthew Sankey (1853-1925), an Irish engineer who worked on steam engine improvements, was the first to show that the transfer of energy can be represented by a flow diagram [32]. Since, dawning the Sankey diagram by hand is time-consuming, some softwares such as E! Sankey, S. Draw, Sankey Editor and Sankey Helper are designed [33]. There is a specific way in which a Sankey diagram is to be drawn [34]:

- The input energy is from the left of the diagram
- The useful energy is to the right.
- All energy loss is made to go vertically down.
- Width of the flow arrows proportional to the flow quantity

Fig. B.1 illustrates the Sankey diagram of a typical system consuming energy [32]. As observed, the energy loss of the component 1 is more than that of the component 2. Also, most of the input energy is converted into useful energy.

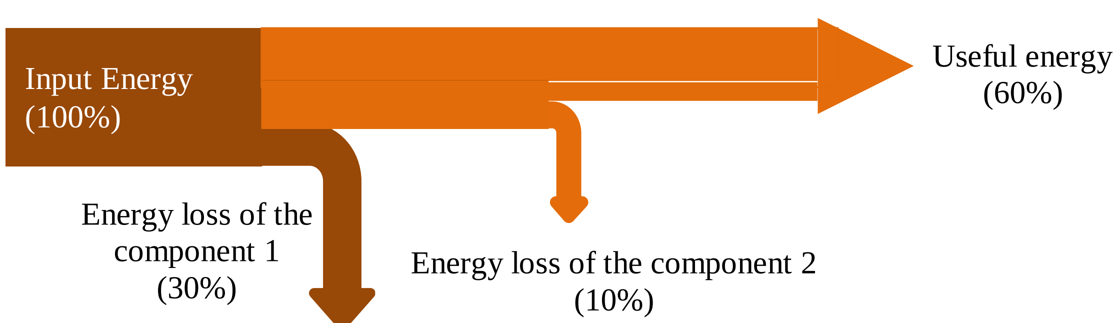

Fig. B.1. Sankey diagram of a typical system consuming energy.