Journal of Physics: Conference Series

PAPER • OPEN ACCESS

Improving performance of pump-controlled hydraulic circuits for single-rod actuators: conceptual study

Improving performance of pump-controlled hydraulic circuits for single-rod actuators: conceptual study

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Abstract. Pump-controlled hydraulic circuits are more efficient than conventional valve-controlled ones. Pump-controlled hydraulic circuits for double rod cylinders are well developed and, presently, implemented in many applications including aviation. Nevertheless, current pump-controlled hydraulic circuits for single-rod cylinders encounter performance issues during specific operating conditions. Pressure and actuator velocity oscillations are encountered when operating in this critical zone. Different concepts, techniques and designs are proposed by researchers to overcome such vibration issues. In this research, three different concepts to overcome the reported oscillation problem in these circuits are presented; namely: (1) shifting of critical zone into lower loading ranges, (2) applying selective leakage, and (3) applying selective throttling. Simulation studies indicated that the new concepts alleviated the oscillation issue of the common pump-controlled circuits, and improved their performances. The second and third concepts, in particular, were capable of eliminating the whole critical zone. The first concept reduced the area of the critical zone in the load-velocity plane and lessened undesirable effect of oscillations. Simulation studies further proved the enhanced performance of circuits that applies these concepts as compared to previously-designed circuits.

1. Introduction

Hydraulic actuation systems are widely used in industry. They possess fast response and high power-to-weight ratio. Efficiency, however, is the main disadvantage of conventional valve-controlled hydraulic systems. Pump-controlled hydraulic circuits are more efficient than valve-controlled circuits, as they avoid the use of flow throttling valves and hence eliminate energy losses in valves. Presently, existing pump-controlled solutions for single rod cylinders encounter an undesirable performance during certain operating conditions. More specifically, these circuits experience oscillatory behavior when switching from assistive to resistive modes during actuator retraction. Many trials to deal with the issue using different concepts, techniques and designs are proposed by researchers to overcome such vibration issues.

Williamson and Ivantysynova [1] proposed utilizing a feedforward control of the actuator pressure, with the addition of an observer to provide sufficient leading time for controller. Wang et al. [2] implemented an extra control loop that comprises pressure sensors and two electrically-operated regulating valves to allow oil leakage during system oscillation. Caliskan et al. [3] proposed a modified version of the abovementioned circuit; they utilized a 3/3 open-center shuttle valve (OC-SHV) that compensates for the cylinder differential flow in addition to stabilizing the circuit through oil leakage. Imam et al. [4] proposed a circuit with a limited throttling valve that applies throttling effect over critical

 2616 (2023) 012015 doi:10.1088/1742-6596/2616/1/012015

regions and least throttling over other operating regions, thus maintaining efficiency. Imam et al. [5] shifted the critical zone into lower loading margin by utilizing two different charge pressures to both sides of the actuator. Experimental results showed an improved performance of the circuit.

Simulation studies, on conceptual level, of three methods used to improve the performance of pump-controlled hydraulic circuits are presented in this research. These concepts are: (1) shifting of the critical zone into lower loading ranges, (2) applying selective leakage and (3), applying selective throttling. Theoretical analysis and simulation studies that demonstrate the enhanced performance of the circuits as a result of applying the proposed methods are presented. Applying these concepts, individually or collectively, leads to different new pump-controlled circuits [6].

2. Concept I- shifting of critical zone

Figure 1(a) shows that the undesirable zone forms two regions: 5 and 6, located to the left and right-hand sides of a vertical line passing through the initial critical load F_{cr0} [4]. Critical load is identified as the applied external load that is equal to the actuator biased force when the pressure at both sides of the cylinder is equal to the charge pressure [2]. It is also noticed that the critical zone is asymmetric around the zero-load axes, which shiftes the oscillatory critical region 6 to a higher load margin on the load axis. Oscillations at higher load values, especially due to inertial loads, are more severe in amplitude and, consequently, destructive on the machine [5]. Shifting of the critical zone into a symmetric position around the zero-load axes reduces the load values seen by the circuit in region 6 and consequently reduces the magnitude and effect of oscillations (see Figure 1(b)). Note that the undesirable region 5 may reach higher load values, however, this can be tolerated since region 5 does not exhibit oscillatory behavior. Different strategies to shift the critical zone into a symmetric position around the zero-load vertical axes are proposed in this section. Simulations of a sample of these designs are carried out to show the improved performance of the proposed concept [6].

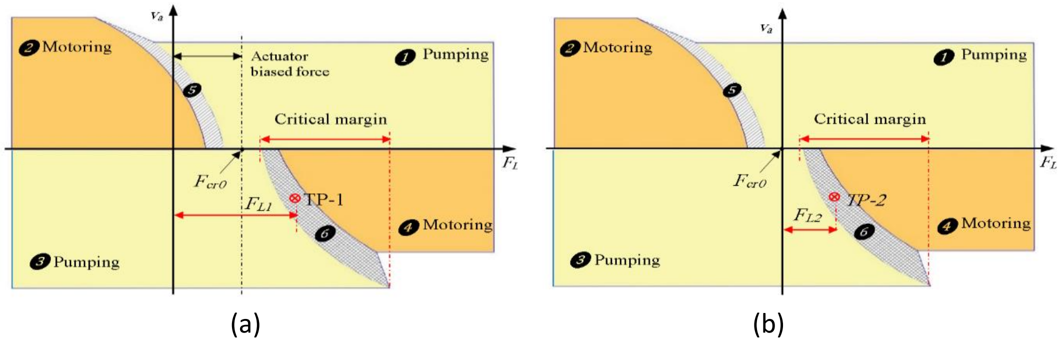


Figure 1. Construction of critical regions for, (a) conventional pump-controlled circuit; (b) circuit that applies shifting of the critical zone concept.

By ignoring the transmission lines losses, the critical load is defined as $F_{cr0} = A_A p_c (1 - \alpha)$ where α is the actuator area ratio ($\alpha = {}^{A_B}/_{A_A}$) [4]. There are two solutions to obtain zero-value critical load: (i) choosing $\alpha = 1$, which means utilizing a symmetric cylinder and (ii) reducing the charge pressure to zero gauge-pressure, i.e. $p_c = 0$ which means using a non-pressurized oil tank. Apparently, both solutions are not acceptable since this research is related to single-rod asymmetric cylinders. Furthermore, a low-pressure charging system is needed in the closed pump-controlled circuits in order to allow fast compensation for the differential flow of the single-rod actuators, avoid pump cavitation and supply low-pressure flow to the pump case for cooling and lubrication [6].

One feasible solution is to utilize two different low charge pressures in the circuit. In this solution, each side of the circuit is connected through the compensating valve to a separate charge pressure. Low pressure values are chosen such that, at zero-load, zero-velocity condition, the pressure-induced forces at each side of the actuator are equal ($A_A P_{cA} = A_B P_{cB}$) and consequently $F_{cr0} = 0$. In order to maintain proper operation of the circuit with two different charge pressures, asymmetric or biased compensating valves are used. This concept can be implemented on different pump-controlled circuits, including those utilizing pilot-operated check valves (POCVs) and those utilizes shuttle valves (SHV). Implementation is done through utilizing two different cracking pressures POCVs or biased shuttle valves [6].

Figures 2(a) and (b) show the commonly used circuits with one charge pressure and two identical pilot operated check valves and the proposed circuit with two charge pressures and two different pilot operated check valves, respectively. On the other hand, the commonly used circuits with one charge pressure and 3/3 shuttle valve and the proposed circuit with two charge pressures and biased 4/3 shuttle valve are shown figures 2(c) and (d).

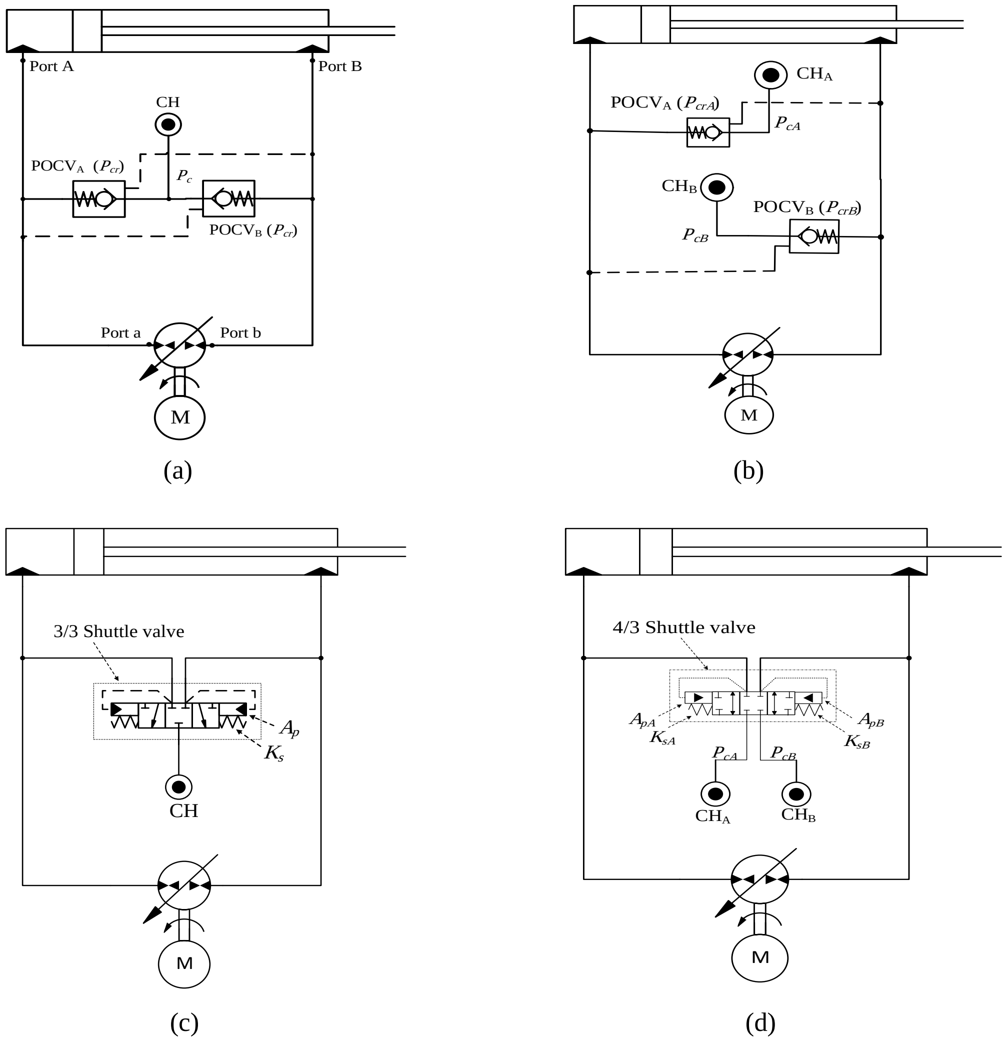


Figure 2. Circuits with: (a) one charge pressure and two identical pilot operated check valves; (b) two charge pressures and two different pilot operated check valves; (c) one charge pressure and a 3/3 closed-center shuttle valve (SHV); (d) two charge pressures and a 4/3 closed-center biased SHV.

2.1. Modeling and Simulation studies

Mathematical models of the conventional pump-controlled circuits with pilot-operated check valves (POCVs) and that with a 3/3 closed-center shuttle valve (CC-SHV) as well as proposed modified circuits with two charge pressures are detailed in our previous work [5]. Simulation parameters and values of the different components of the circuit are listed in Table 1. Simulation studies are done for two load-motion scenarios. Simulations are performed in two scenarios to show the effect of shifting the critical zone towards lower loading values on the performance of the proposed circuits. The first scenario is related to the circuit responses for a constant load and a step input control signal. The second scenario is related to the circuit responses for variable loading condition and a square input signal [6].

2.2. Constant loading simulations

Different simulation runs to evaluate circuit performance under different loading conditions are done. At each simulation run, a voltage step input signal is applied to the system at specific loading condition and actuator velocity is recorded versus time. Figure 3(a) shows the critical zone of the conventional circuit with two POCVs and proposed circuit with two charge pressures. Compared to conventional circuit, critical zone of the proposed circuit with shifting of critical zone concept is reduced and shifted about 1400 N towards less load margin. Figure 3(b) shows the critical zone of the conventional circuit with 3/3 CC-SHV and proposed circuit with the 4/3 biased CC-SHV and two charge pressures. Compared to circuit with 3/3 CC-SHV, critical zone of the proposed circuit is reduced and shifted about 1200 N towards less load margin. Figures 3(c) and (d) show the actuator velocity oscillation amplitudes versus the applied load at input signal of -4V for both conventional and modified circuits shown in figures 3(a) and (b). Figures 3(c) and (d) further prove the reduced critical zone and oscillation amplitudes. Reduced load values where oscillation occurs reduced the adverse effect of oscillationg load on the machine. Comparizon between figures 3(c) and (d) shows the reduced critical zone and oscillation amplitudes in circuit with POCVs as compared to that with CC-SHVs.

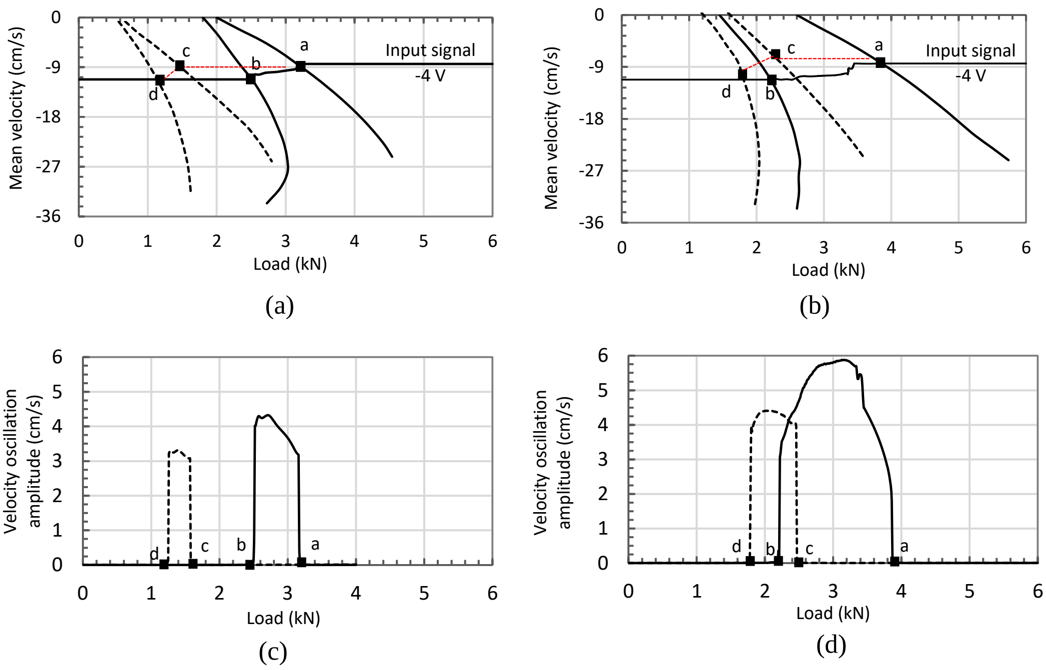


Figure 3. Simulation results for shifting of criitcal zone given data in Table 1: (a) critical zone of the conventional circuit with two POCVs (solid line) and proposed circuit with two charge pressures (dash-line) ; (b) critical zone of the circuit with 3/3 CC-SHV (solid line) and proposed circuit with two charge pressures and a biased 4/3 CC-SHV (dash-line); (c) velocity oscillation amplitude of the conventional circuit with two POCVs (solid line) and proposed circuit with two charge pressures (dash-line); (d) velocity oscillation amplitude of the circuit with a 3/3 CC-SHV (solid line) and proposed circuit with two charge pressures and a biased 4/3 CC-SHV (dash-line).

2.3. Variable loading simulations

Figures 4 and 5 show the simulation results of performance of the conventional circuit with a 3/3 CC-SHV and modified circuit with 4/3 biased CC-SHVs under low and high loading conditions, respectively. Figures 4(a) and 5(a) show the square input control signals. Figures 4(b) and 5(b) illustrate the applied load patterns. The actuator velocity versus time of the conventional and the proposed circuits are shown in figures 4(c) and (e) and 5(c) and (e), for low and high loads, respectively. It is clear that velocity oscillations occur during switching zone S43 whereas oscillations under low loading conditions are much sever when comaped to that ubder the high loading conditions. It is also noticed that oscillations in proposed circuits are much less in amplitude and duration as compared to its counterpart of the conventional circuits. Reduction in actuator oscillation amplitudes reduces the operator's inconvenience as well as the destructive effect on the machine. Figures 4(d) and (f) and 5(d) and (f) illustrate pressures at both sides of pump versus time of the conventional and the proposed for low and high loads, respectively. Similar to the velocity responses, pressure oscillations occurred in switching zone S43 under low loading condition are much higher that under higher loading conditions. Figures 4(d) and (f) as well as figure 5(f) show that pressure values cross the zero-pressure line during some operating conditions which can be attributed to simulation' imperfection. However in practical circuits this phenomenon will not happen due to the anti-cavitation valves fitted in many places of the circuit, mainly in the pump, that limit the pressure drop to some safe value (typically -0.02 to -0.05 MPa gage pressure). Different loading simulations show improved performance of the proposed circuits that incorporates shifting of critical margin concept when compared to that of the conventional circuits. Although this solution does not totally alleviate osillations, it reduces its undesirable effect and facilitaes the application of other solutions.

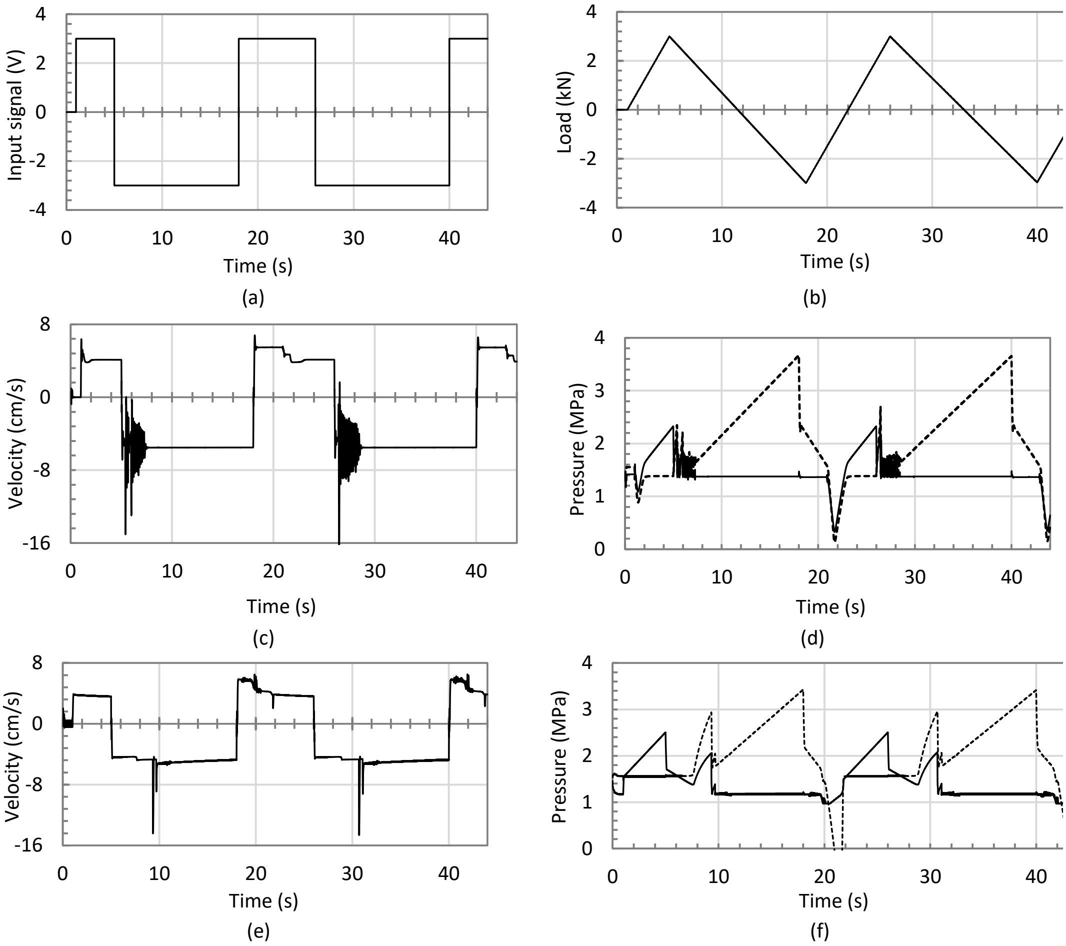


Figure 4. Simulation results for conventional circuit with 3/3 CC-SHV and modified circuit with 4/3 biased CC-SHV and two charge pressures under low loading conditions given data in Table 1: (a) input control signal; (b) applied load; (c) actuator velocity for conventional circuit; (d) pressures at pump ports *a* (solid line) and *b* (dotted line) for conventional circuit; (e) actuator velocity for modified circuit; (f) pressures at pump ports *a* (solid line) and *b* (dotted line) for modified circuit.

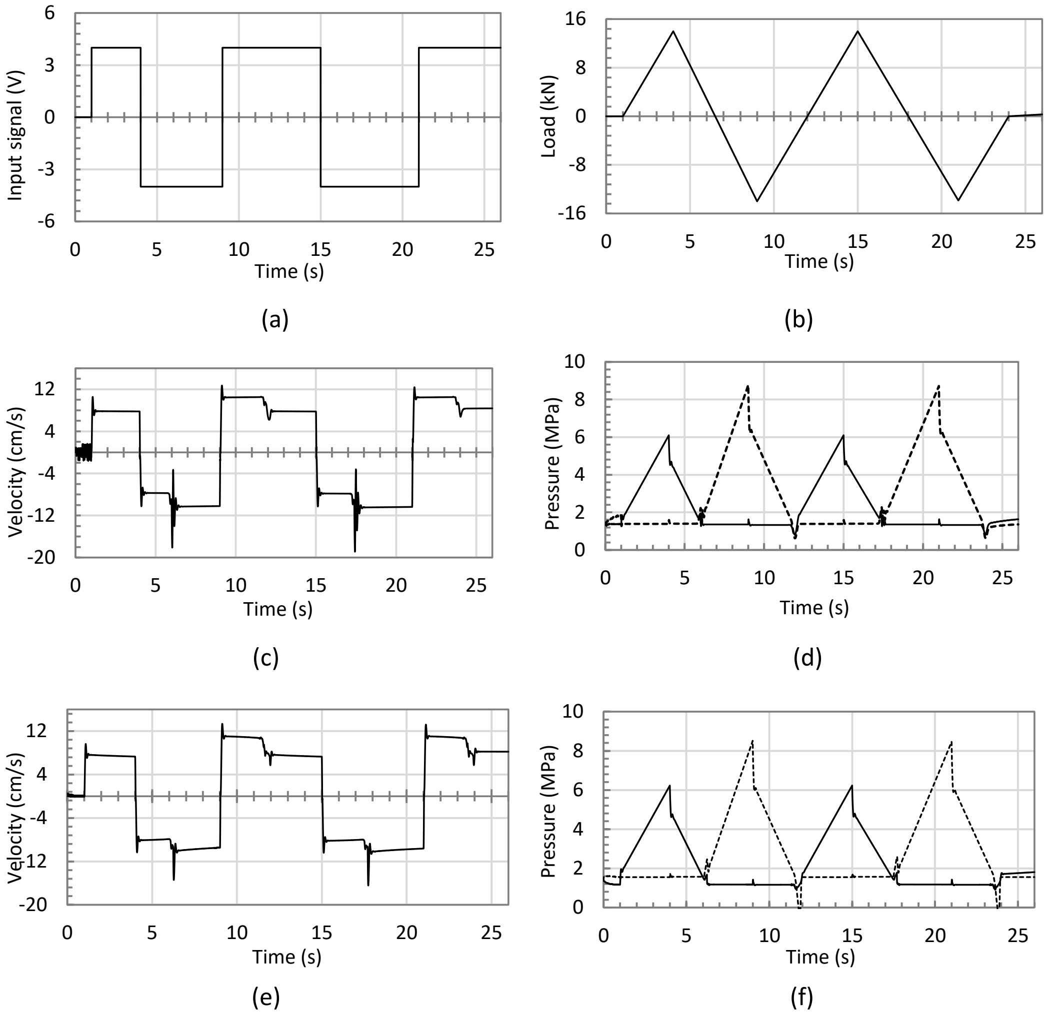


Figure 5. Simulation results for conventional circuit with 3/3 CC-SHV and modified circuit with 4/3 biased CC-SHV and two charge pressures under high loading conditions given data in Table 1: (a) input control signal; (b) applied load; (c) actuator velocity for conventional circuit; (d) pressures at pump ports *a* (solid line) and *b* (dotted line) for conventional circuit; (e) actuator velocity for modified circuit; (f) pressures at pump ports *a* (solid line) and *b* (dotted line) for modified circuit.

Table 1. Values of parameters used in simulations.

Parameter	Definition	Values
A_a	Area of piston cap side	31.67×10^{-4} [m^2]
A_b	Area of piston rod side	23.75×10^{-4} [m^2]
K_{oil}	Effective bulk modulus	0.689×10^9 [Pa]
M_{eq}	Equivalent mass	$0 - 400$ [kg]
A_{lk}	Valve leakage area	0 [m^2]
A_{max}	Max flow area in valve	25×10^{-6} [m^2]
P_{cr}	Cracking pressure	0.2 [MPa]
P_{mx}	SHV maximum opening pressure	0.5 [MPa]
P_c	charge pressure for conventional circuit	1.38 [MPa]
P_{cA}	Side-A charge pressure for proposed circuit	1.17 [MPa]
P_{cB}	Side-B charge pressure for proposed circuit	1.56 [MPa]

3. Concept II- selective leakage

The concept of applying leakage to stabilize pump-controlled circuits was introduced by Wang et. al. [2, 7] and Caliskan et. al. [3]. Both researchers applied this concept over circuits with shuttle valves. Wang et. al. [2, 7] used two extra electrically-controlled regulating valves to control leakage through an additional control loop. Caliskan et. al. [3] used an open-center shuttle valve to incorporate leakage control together with flow compensation. However, this solution does not cover the full undesirable margin and is limited to a specific critical speed [3]. It is seen from the comparative analysis in the previous section that the undesirable margin is smaller in size and exhibits lesser oscillation amplitudes in circuits with two POCVs as compared to that in circuits with SHVs. Thus, applying the concept of controlled leakage to the circuit with two POCVs is expected to give better results. In this section, applying leakage to stabilize pump-controlled circuits that utilize POCVs is proposed. Leakage control in pump-controlled circuits that utilize POCVs is achieved by adding flow control valves either in main flow lines or in differential flow lines, as can be seen in figures 6(a) and (b), respectively. Circuit with two controlled leakage valves next to the actuator ports is used as proof of concept [6].

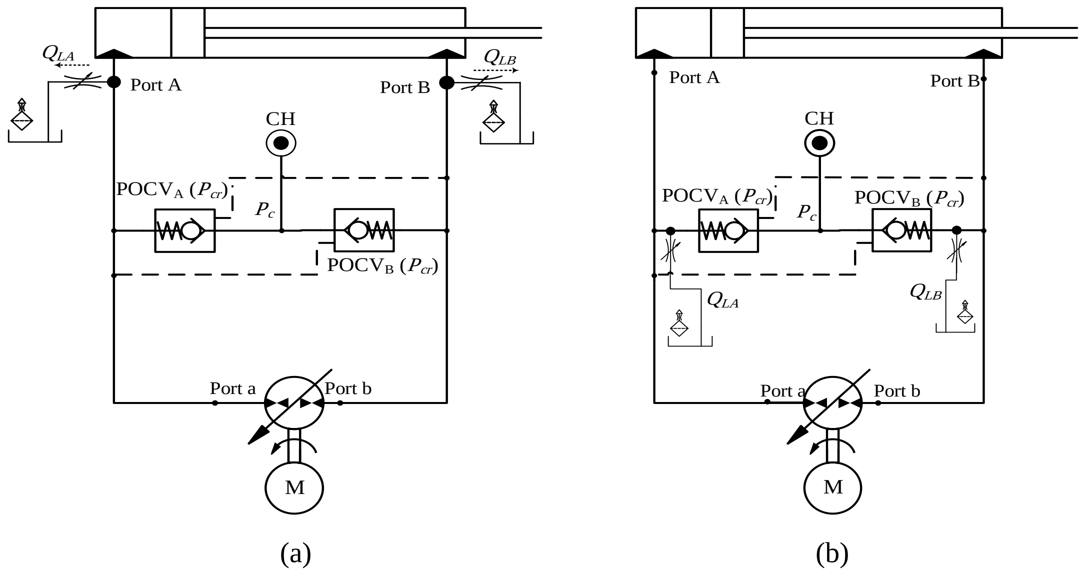


Figure 6. Using leakage to stabilize pump-controlled circuits that utilize POCVs: (a) leakege in main flow lines; (b) leakege in differential flow lines.

3.1. Modeling and Simulation Studies

Mathematical model of the circuit with leakage control is obtained by adding the mathematical model of flow through a leakage valve to the model of the conventional circuit with POCVs derived in [5]. Leakage through flow throttling valves is described as follows:

$$Q_{lA} = C_d \, A_{lA}\sqrt{\frac{2}{\rho}|p_A - P_c|} \; sign\,(p_A - P_c) \tag{1}$$

$$Q_{lB} = C_d \, A_{lB}\sqrt{\frac{2}{\rho}|p_B - P_c|} \; sign\,(P_c - p_B) \tag{2}$$

where Q_{lA} and A_{lA} and Q_{lB} and A_{lB} are flow and throttling areas in leakage valves at sides A and B of the circuit, respectively. Similar to the previous circuits, simulation studies are done for two load scenarios. The first scenario is related to constant load and a step input control signal, and the second scenario is related to variable loading condition and a square input signal. Simulation parameters for the circuit are listed in Table 1, while the selected valve leakage areas are listed in Table 2.

3.2. Constant loading simulations

Figure 7(a) shows the steady state velocity responses of the actuator at different loads and input control signals to the pump. It is clear that velocity increases gradually when switching between the 4th and 3rd quadrants. External load varies from 12000 N to 0 N with step of 10 N, while the input voltage values are -3 V, -4 V, -6 V and -8 V during simulations. The velocity dynamic responses at three test points TP1, TP2 and TP3 are shown in figures 7(b) - (d). Test points are located at the beginning, middle, and end of the switching zone S43 corresponding to external loads of 4200 N, 3000 N and 1700 N, respectively. Applied step input signal was -4 V. Figures 7(b) - (d) show the damped velocity responses for circuit with leakage control at all test points. Figure 7 shows that the circuit performance is oscillation-free over the tested zone that includes the switching zone S43.

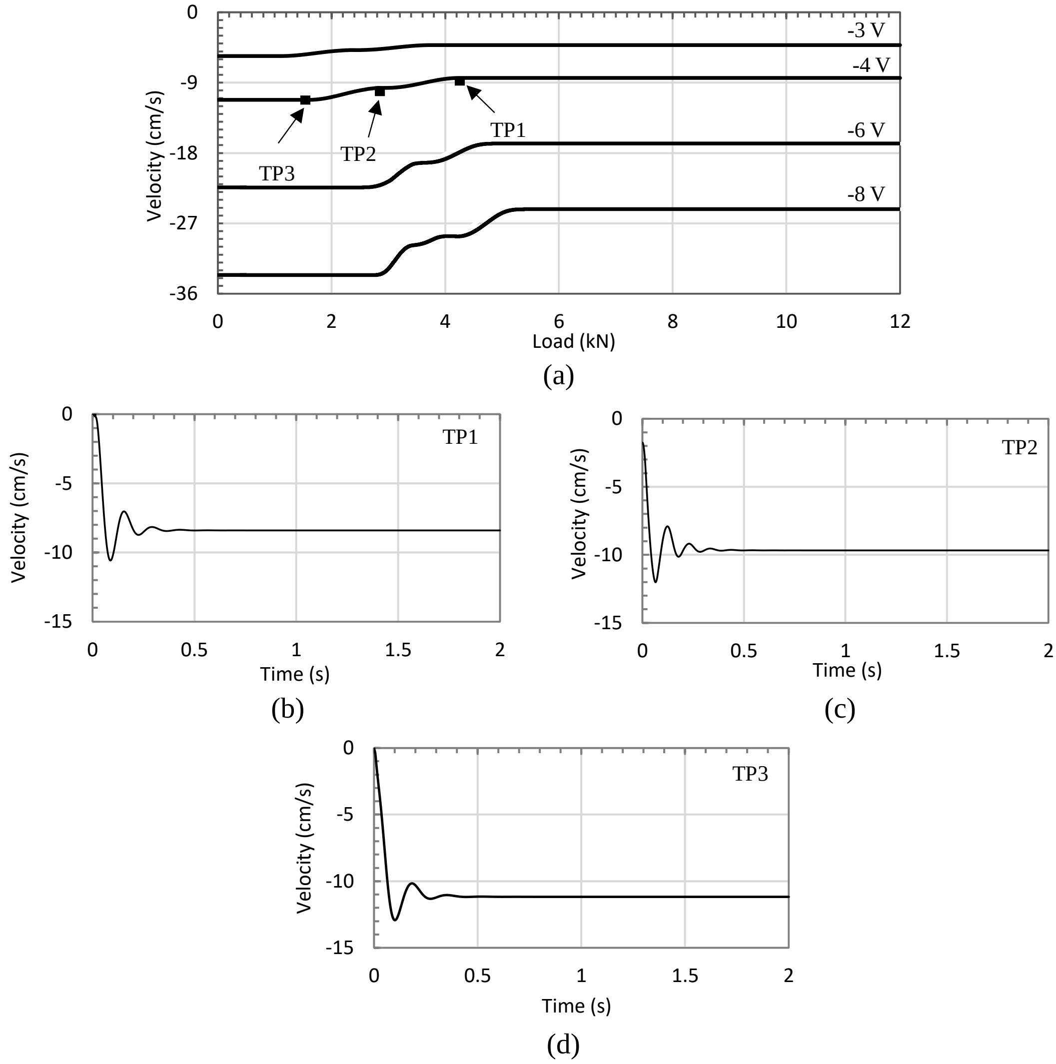


Figure 7. Simulation results for circuit with pilot-operated check valves and leakage valves, given data in Tables 1 and 2: (a) velocity steady state responses at different loads and step input signals; (b-d) velocity response at TP1-TP3 at beginning-middle and end of critical zone for -4 V step input and loads of 4200N, 3000 N and 1700 N.

Table 2. Leakage areas at different input signals to circuit.

Input signal (V)	Leakage area in Line A (mm^2)	Leakage area in Line B (mm^2)
-3	0.5	0.45
-4	1	0.9
-6	2.1	1.8
-8	3.2	2.7

3.3. Variable loading simulations

Simulation results of circuit performance under low and high loading conditions are shown in figures 8 and 9. Figures 8(a) and 9(a) illustrate the square input control signals under low and high loading conditions, respectively. Load patterns are shown in figures 8(b) and 9(b). Figures 8(c) and 9(c) show the actuator velocity versus time under both loading conditions. Figures 8(d) and 9(d) illustrate pressures at both sides of the pump versus time. It is clear from the velocity and pressure responses that the system is oscillation-free in all operating conditions.

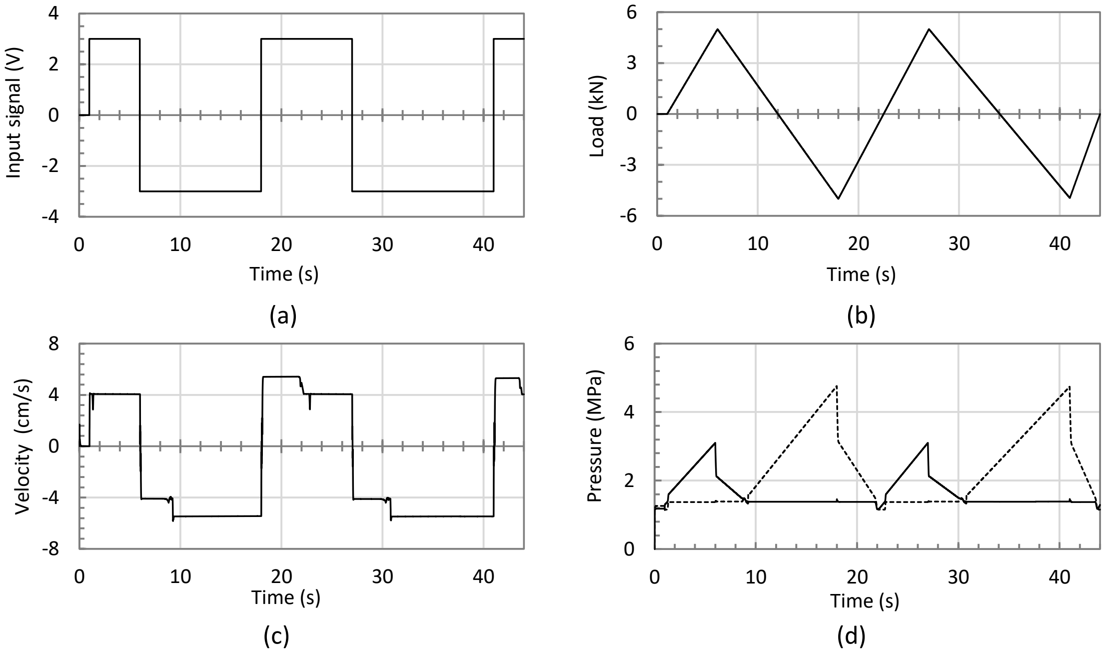


Figure 8. Simulation results for circuit with pilot-operated check valves and leakage control under low loading condition, given data in Tables 1 and 2: (a) input control signal; (b) variable load; (c) actuator velocity versus time; (d) pressures at pump ports *a* (solid line) and *b* (dotted line).

4. Concept III- selective throttling

The main idea behind this concept is to utilize flow throttling to dampen system oscillations. Throttling is applied only in undesirable regions where responses are prone to become oscillatory. In other operating regions, motion is throttle-less. Throttling of hydraulic fluid creates pressure difference (Δp) across the valve orifice maintaining increased pressure in cylinder chambers as compared to pump ports, which contributes towards a stiffer actuator [4, 8, 9]. The proposed concept allows circuits to have a comparable efficiency and energy regeneration ability to the conventional pump-controlled circuits in high loading conditions. It also achieves stability of the circuits with throttling valves under low loading conditions. This concept is implemented by proposing circuits with new compensating valve(s) to perform throttling along with flow compensation [10]. One other solution is obtained by adding a special valve(s) to the previously-designed pump-controlled circuits [4]. Both solutions have to meet the following requiremnts [4]: (1) applying the proper throttling over flow when the two pilot pressures to the compensating valve(s) are close to each other; (2) allowing free flow when the two pilot pressures are not close to each other and throttling is unnecessary. This concept is applicable to previously designed circuits with pilot-operated check valves and those with shuttle valves [4]. Figure 10 shows two different solutions to add the selective-throttling valves into circuit that utilises two POCVs [6].

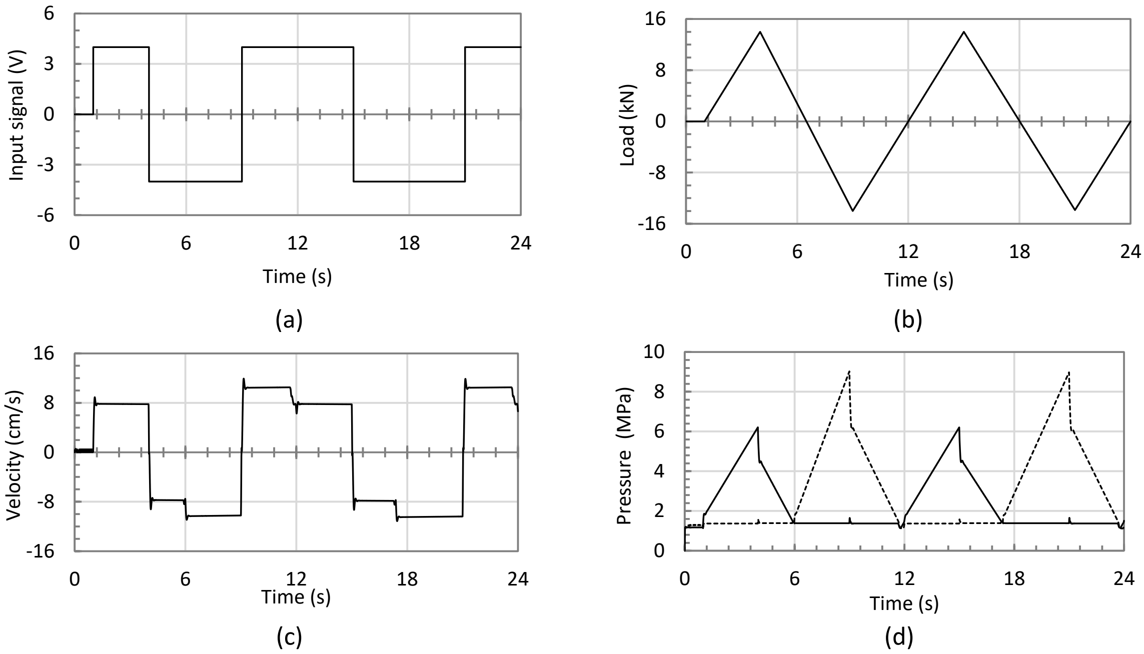


Figure 9. Simulation results for circuit with pilot-operated check valves and leakage control under high loading condition, given data in Tables 1 and 2: (a) input control signal; (b) variable load; (c) actuator velocity versus time; (d) pressures at pump ports *a* (solid line) and *b* (dotted line); (e) load-velocity plane; (f) pressure plane.

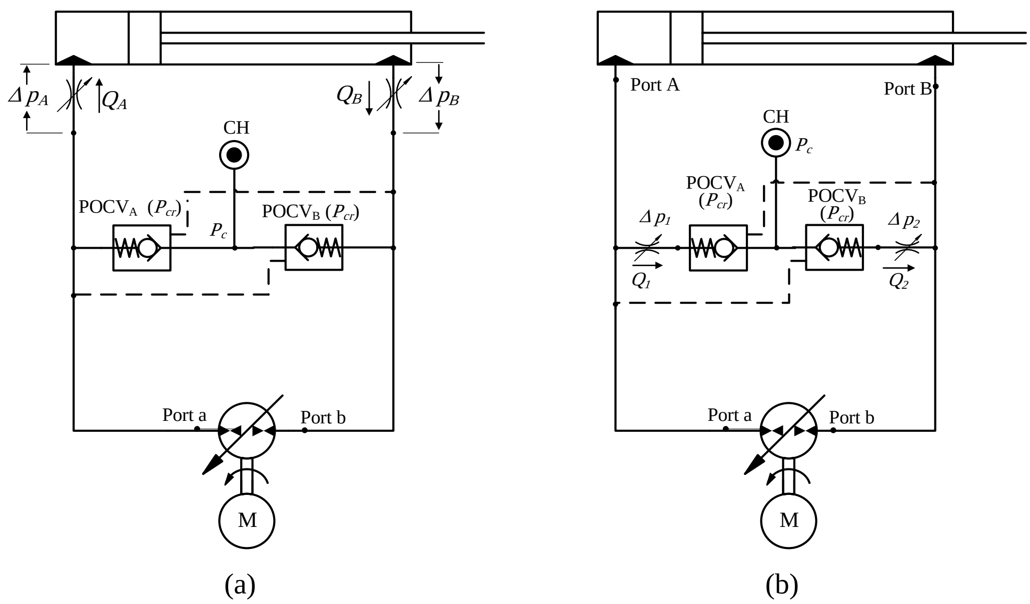


Figure 10. Proposed location for additional selective-throttling valve in the circuit that utilizes two POCVs, (a) in the main flow lines; (b) in the differential flow lines.

4.1. Modeling and Simulation Studies

Performance of circuit that utilizes two POCVs and equipped with two throttling valves next to the actuator ports (shown in Figure 10(a)) is addressed here. The mathematical model is obtained by adding throttling effect to mathematical model of the conventional circuit with POCVs derived in [5]. Figure 11 shows the circuit proposed in Figure 10(a) with notations in the first quadrant of operation. In this model two pressure variables p_{A1} and p_{B1} are added downward of the throttling valves. The mathematical model of the flow through throttling valves is described as follows:

$$Q_A = C_d\, A_{thA}\sqrt{\frac{2}{\rho}|p_{A1} - p_A|}\, sign\,(p_{A1} - p_A) \tag{3}$$

$$Q_B = C_d\, A_{thB}\sqrt{\frac{2}{\rho}|p_B - p_{B1}|}\, sign\,(p_B - p_{B1}) \tag{4}$$

where A_{thA} and A_{thB} are throttling areas in valves at sides A and B of the circuit, respectively. Simulations are used to show the damping effect of flow throttling on system performance. Similar to the previous circuits, simulation studies are done for two load-motion scenarios. The first scenario is related to constant load and a step input control signal, whereas the second scenario is related to variable loading condition and a square input signal. Simulation parameters for the circuit are listed in Table 1, whereas the proposed throttling areas of valve at different input signals to pump are listed in Table 3.

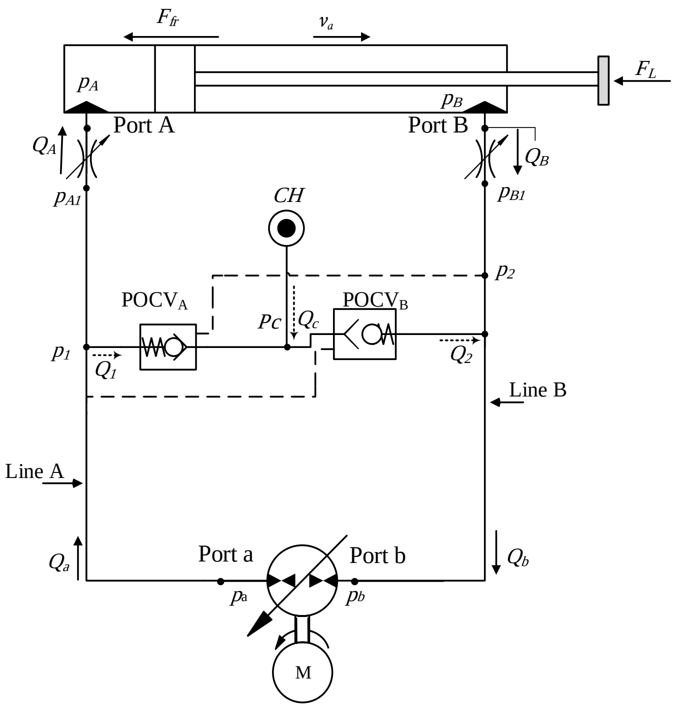


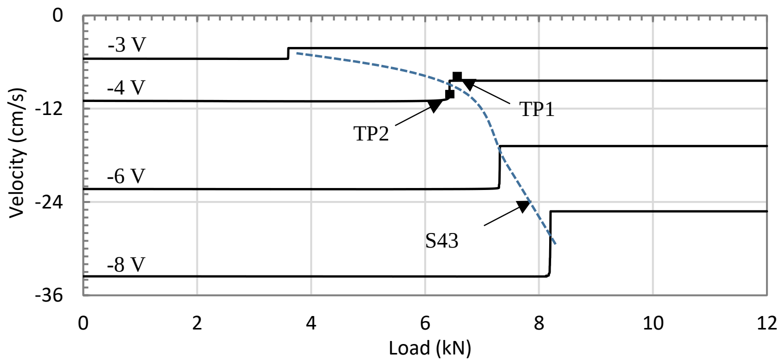
Figure 11. Schematic drawing of the circuit that utilizes POCVs equipped with limited throttling valves, in the first quadrant of operation.

4.2. Constant loading simulations

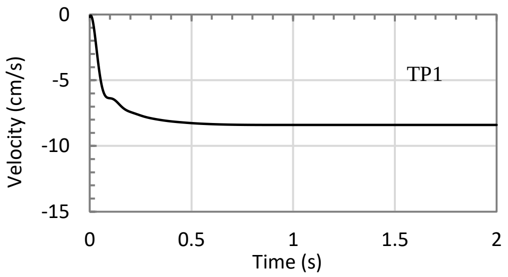
Simulations are conducted under different loading conditions. At each simulation run, a voltage step input signal is applied to the system at constant loading condition and actuator velocity versus time is recorded. At each of the four input signals (-3 V, -4 V, -6V and -8 V), the applied load varied from 0 N to 12,000 N with 2 N incremental step. Figure 12(a) shows the simulation results of the steady state actuator velocity at each of the different runs. Figure clearly shows that switching zone is narrowed and almost turned into a vertical line, while velocity abruptly increased when switching between the 4th and 3rd quadrants. Figure 12(b) and (c) show the damped velocity responses at test points TP1 and TP2 located just before and after switching line, at input signal of -4 V and loads of 6202 N and 6200 N, respectively. Abrupt changes in velocity responses that are notices in S43 do not indicate any performance issues. Note that Figure 12(a) is generated by combining steady state velocity responses of different simulation runs. Simulations that show realistic switching situation are illustrated in the second scenario in the following section. Simulation results showed the enhanced performance of the proposed circuit. Note that the throttling areas that alleviate oscillations at 4 different input signals are predicted based on the trial and error concept. Table 3 shows the input control signals and corresponding throttling areas at lines A and B of the circuit used in simulations.

Table 3. Throttling areas at different input signals to pump

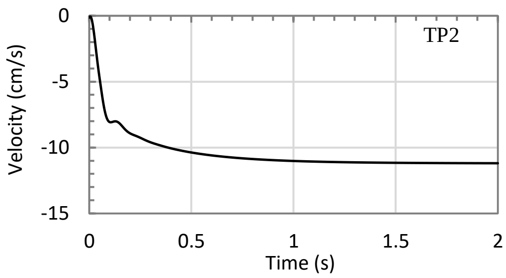
Input signal (V)	Throttling area in Line A (mm^2)	Throttling area in Line B (mm^2)
-3	9	6.75
-4	10	7.5
-6	20	15
-8	30	22.5



(a)



(b)



(c)

Figure 12. Simulation results for circuit with pilot-operated check valves and selective throttling valves, given data in Tables 1 and 3; (a) steady state velocity responses around switching zone S43 in the load-velocity plane; (b) velocity response at TP1 at load of 6202 N in the fourth quadrant; (c) velocity response at TP2 at load of 6200 N in the third quadrant of operation.

4.3. Variable loading simulations

Simulations conducted for variable load scenario are shown in figures 13 and 14. Figures 13(a) and 14(a) illustrate the square input control signals under low and high loading conditions, respectively. The applied load patterns are shown in figures 13(b) and 14(b), respectively. Figures 13(c) and 14(c) show the actuator velocity versus time in both loading conditions. Figures 13(d) and 14(d) illustrate the pressures at both sides of the pump versus time. It is clear from the velocity and pressure responses that the system is oscillation-free under all operating conditions. Simulations further proves the improved performance of the circuit and the smooth transitions in all switching zones.

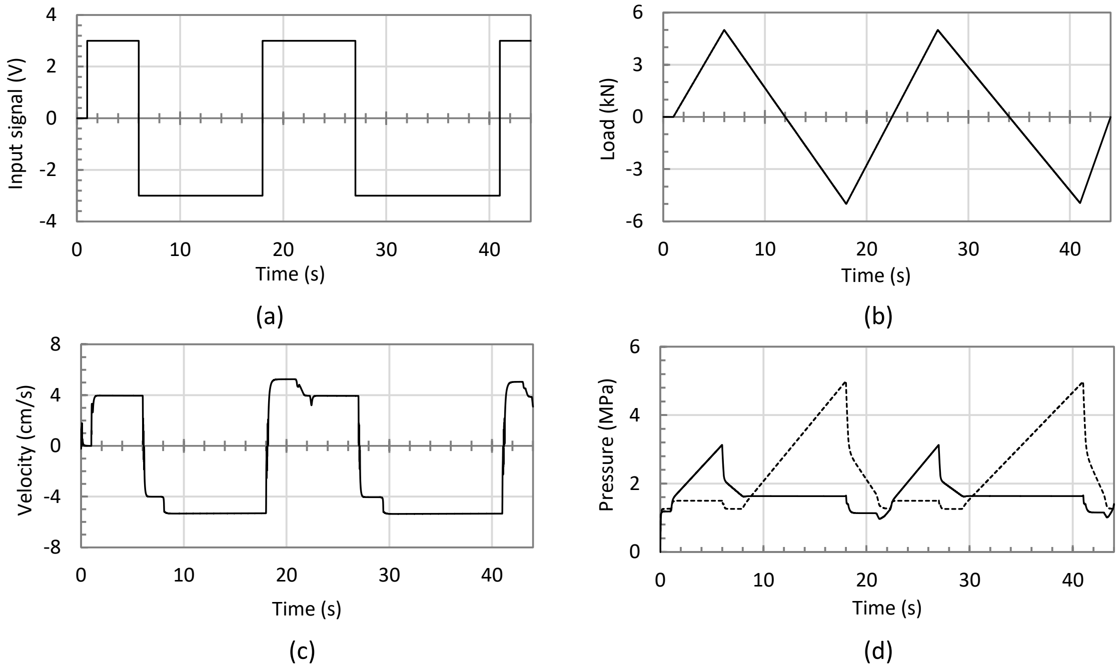


Figure 13. Simulation results for circuit with pilot-operated check valves and selective throttling valves at low loading conditions, given data in Tables 1 and 3: (a) input control signal; (b) applied load; (c) actuator velocity versus time; (d) pressure at pump ports a (solid line) and b (dotted line).

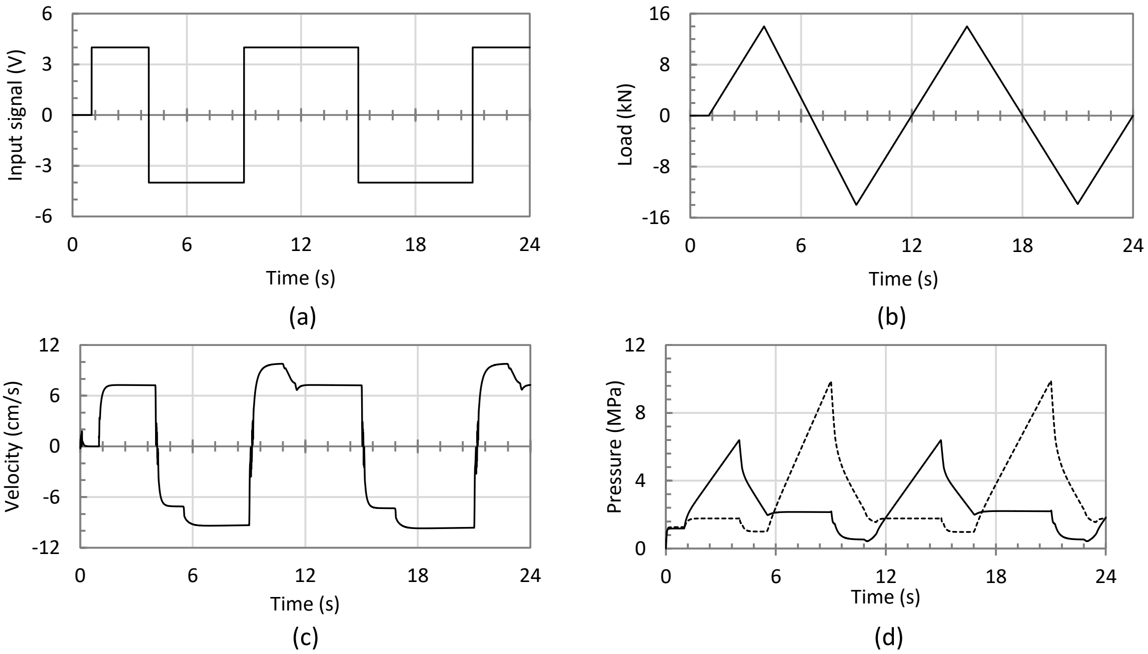


Figure 14. Simulation results for circuit with pilot-operated check valves and selective throttling valves at high loading conditions, given data in Tables 1 and 3: (a) input control signal; (b) applied load; (c) actuator velocity versus time; (d) pressure at pump ports a (solid line) and b (dotted line).

5. conclusion

Three concepts to improve performance of pump-controlled circuits were investigated. These concepts are: (1) shifting of the critical zone into lower loading ranges, (2) applying pre-designed selective leakage in critical zone, and (3) applying selective throttling over flow in critical zone. Performance enhancement due to application of the aforementioned concepts was demonstrated through simulation studies. Application of the second and third concepts totally removed the whole critical zone while application of the first method reduced the area and effect of oscillations. By applying the above-mentioned concepts, either individually or collectively, different pump-controlled circuits for single-rod cylinders will be created.

Additionally, the proposed concepts have minimal effect on system efficiency since they apply corrective actions (selective throttling or selective leakage) only in the critical loading zones, which already represent a small margine of the whole operating margine of the machine (typically 10%). On the other hand they allow more efficient throttle-less flow in the larger operational areas outside the critical loading zones. Furthermore, four-quadrant operation is fully retained whereby motoring of the pump in two quadrants can be used for energy regeneration purposes for optimal efficiency.

Consent, portions of this work were partially prsented in the first author's PhD thesis at University of Manitoba that can be reached at mspace.lib.umanitoba.ca.

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