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IMPLEMENTATION OF SINGLE FEEDBACK CONTROL LOOP FOR CONSTANT POWER REGULATED SWASH PLATE AXIAL PISTON PUMPS

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Abstract

Variable displacement pumps are often used in both industrial applications and mobile hydraulic machinery. In such pumps, flow rate is dictated by the system requirements. Mathematical model has been previously developed to simulate the dynamic performance of the electrically controlled constant power regulated swash plate axial piston pump with conical cylinder blocks. The pump is currently equipped with a double negative feedback control loop with an inner control loop to control the position of proportional valve using PID controller. Consequently, the proportional valve distributes the control pressure across the two sides of a control piston that is mechanically attached to the pump swash plate in order to change the pump flow rate. The outer control loop is used to control the pump flow rate in accordance with the system pressure change in order to keep the constant power operation using PD controller. For the convenience of pump constant power operation, PD controller is tuned to keep limited power shocks on the pump drive motor during the transient periods. The selected PD parameters result in relatively reduced settling time. Consequently swash plate steady state vibration appears.

Purpose of this paper is to investigate features of the pump performance in view of an alternative control scheme. Counting on the relatively good open loop static characteristics of the proportional valve, a control scheme with a single control feedback loop is proposed to simplify the currently used control scheme. Using such single feedback control loop reduces the pump production cost and leads to have less responsive system that suppresses the steady state vibration of the swash plate. Simulation results are verified experimentally and qualitatively compared with the results when the original control scheme is used. Results are presented and discussed.

Keywords: axial piston pump, constant power, PID, fuzzy

1 Introduction

Swash plate piston pump with the conventional cylindrical piston arrangement was extensively studied in the last two decades. Akers and Lin (1987) applied an optimal control theory to determine the design parameters of the pressure regulator of an axial piston pump, which incorporates a single stage electro-hydraulic servo-valve. A theoretical comparative study of the dynamic characteristics of three different types of hydro-mechanical constant-pressure regulators for swash plate pumps was carried out by Mohamed (1989). The system with the most suitable characteristics was determined. Kaliafetis and Costopoulos (1994) studied both theoretically and experimentally the static and dynamic characteristics of a standard variable geometric volume swash plate pump with constant pressure regulator. It was shown that the operating conditions are very crucial for the pump dynamic behaviour and that the dynamic performance is improved when the setting pressure is decreased. Modelling of a swash plate piston pump with the conventional cylindrical piston arrangement was carried out by Manring and Johnson (1996). They studied the effect of some design parameters on the pump performance. Swash plate pumps with conical cylinder blocks have been recently studied in view of their improved suction characteristics and some static and dynamic benefits. For this type of pumps, forces and moment acting on the swash plate were studied theoretically by Kassem and Bahr (2000). It was shown that the component of the moment that must be overcome by the pump control system is periodic and has an average value, which acts in the direction that decreases the swash plate inclination angle.

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Kassem and Bahr (2001) developed a mathematical model to describe the dynamic performance of the swash plate pumping mechanism that is controlled using double negative feedback control loop. They proposed a fuzzy logic controller to replace the conventional PD controller currently in use. Results show that the fuzzy logic controller could fairly replace the PD one. Bahr, Svoboda and Bhat (2002a) validated the mathematical model derived in (2001) based on the agreement between the simulation results and experimentally measured pump response to the stepwise change of the load pressure.

The scope of the work in this paper is using the pump model presented in (Kassem and Bahr, 2001) and (Bahr, Svoboda and Bhat, 2002) to simulate the pump performance in case of using single feedback control loop and verifying the simulation results experimentally. In view of different control scheme, features of interest of the pump performance are then discussed and qualitatively discussed.

2 Pump Mathematical Model

As presented in (Kassem and Bahr, 2001) and (Bahr, Svoboda and Bhat, 2002), a mathematical model was developed to describe the dynamics of a variable geometric volume swash plate axial piston pumping mechanism with conical cylinder block shown in Fig. 1. The model initially calculates the piston stroke and absolute acceleration during its general space motion as function of the driving shaft rotation angle. Pump flow rate and the piston chamber pressure were then found by solving the continuity equation for the control volume inside each piston chamber. Forces and moments acting on the swash plate are calculated based on the knowledge of the piston chamber pressure and the piston absolute acceleration.

As shown symbolically in Fig. 2, the pump control unit is composed of an open centre proportional directional valve and symmetric hydraulic control cylinder. When the valve solenoid receives a control signal above zero in an open loop, a proportional electromagnetic force acts on the valve spool and causes it to move against its return spring. A simple second order differential equation is used to describe the dynamics of the proportional valve spool displacement. Control piston of the symmetric hydraulic control cylinder is modelled as physical integrator. Pressure difference across the control piston is resulted as a consequence of the movement of the valve spool. Pressure difference across the control piston is then calculated by applying the continuity equation in its side chambers. A second order differential equation is used to represent the control piston dynamics. Control piston is attached mechanically to the pump swash plate. The model is further developed to include the dynamics of the swash plate considering all the moments acting on it that should be overcome by the control unit

Inner feedback control loop is used for the accurate positioning of the proportional valve spool using displacement sensor and PID controller. Control pressure across the control piston, due to movements of the proportional valve spool, can be found by solving the continuity equation for the variable control volumes of the control piston side chambers. Second order differential equation is used to represent the angular momentum of the swash plate swivelling motion considering the spring, damping and force effects of the attached control piston. Outer feedback control loop is used to control the swash plate swivelling angle using displacement sensor and PD controller.

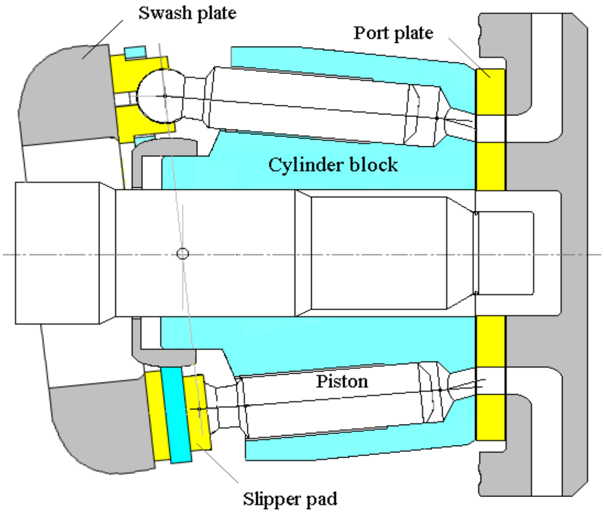


Fig. 1: *Variable geometric displacement volume swash plate axial piston pumping mechanism with conical cylinder block*

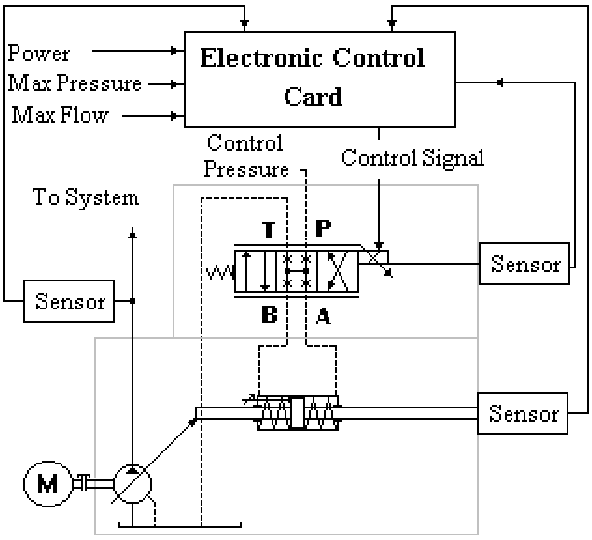


Fig. 2: *Symbolic representation of the pump and the control unit*

System pressure is permanently monitored, using pressure sensor, and fed back to an arithmetic logic unit. The arithmetic logic unit provides the swivelling angle reference value based on the requirements of the pump constant power operation. Static characteristic limits; namely power, maximum pressure and maximum flow that should be respected by the control unit are fed tom the control card electronically. Pump and

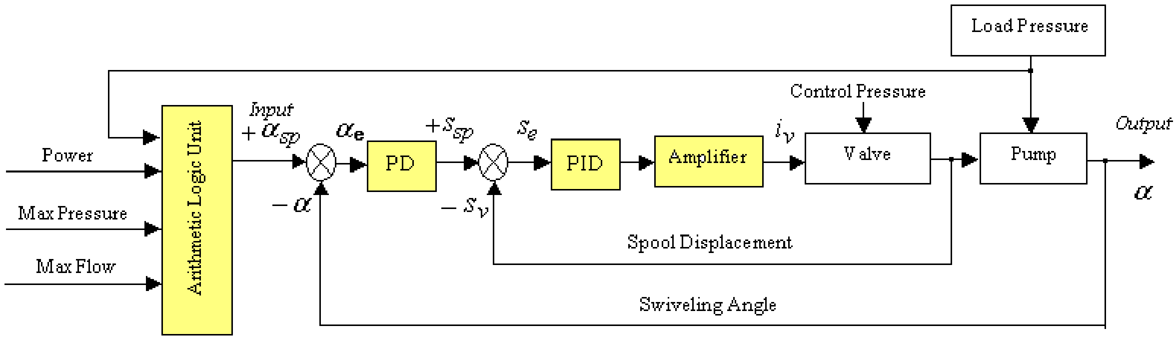


Fig. 3: *Block diagram represents Matlab/Simulink program constructed to simulate the pump performance in double feedback control loop*

valve controllers and the arithmetic-logic unit are built on the electronic card shown in Fig. 2. All the design parameters of the pump and the proportional valve are presented in the references (Kassem and Bahr, 2001) and (Bahr, Svoboda and Bhat, 2002).

3 Pump Dynamic Performance in Double Feedback Control Loop

A software based on Matlab/Simulink was developed to simulate the dynamic performance of the pump in constant power operation. Figure 3 shows, in block diagram form, the simulation subsystems constructed and integrated to simulate the reality of the pump as accompanied with the control unit. The figure shows that the swash plate swivelling angle is controlled, as currently used in practical applications, using double negative feedback control loops. Simulation program contains also units that represent the load pressure, control pressure and the arithmetic logic unit. Computer runs were carried out to simulate the dynamic performance of a 9-pistons pump that has 40 cm^3/rev geometric displacement volume. The design parameters of the pump, subject of the study, are shown in (Kassem and Bahr, 2001) and (Bahr, Svoboda and Bhat, 2002). The pump is manufactured by *Bosch Rexroth Corporation* and has the constructional dimensions and operational conditions shown in the data sheet reported by the manufacturer in (1997, ordering code "A4VSO40HS3U/10RPPB13NOON"). The swash plate is assumed to be initially at a minimum inclination angle that satisfies the pump self-lubrication requirements. Then, the pump is subjected to a step change in the load pressure with a value, which according to the constant power operation drives the swash plate to a corresponding percentage of its maximum inclination angle; namely 25%, 50%, 75% and 100%. At a certain time, after reaching the steady state, the load pressure was assumed to increase in a stepwise manner to its maximum value that swings the swash plate again to the minimum position. Results are presented and discussed in section 6.

The empirical-analytical method “Ultimate Sensitivity” introduced by Ziegler–Nichols in (Franklin, Powell and Abbas, 2002) is used to parameterize the PID valve controller. Using such approach, the proportional gain is increased until the system started to be marginally stable and continuous oscillation appears with amplitude limited by the proportional valve saturation. The corresponding gain is defined as K_u (called ultimate gain) and the oscillation period is P_u (called ultimate period). Then the tuning parameters are selected as follows. The valve control signal that is coming out of the PID controller equals $K_p[1 + 1/T_iS + T_dS]$, where $K_p = 0.6K_u$, $T_i = 0.5P_u$ and $T_d = 0.125P_u$. The parameters of the PID valve controller are further tuned based on the valve performance reported by the manufacturer. It was found to be $K_p = 1$, $T_i = 0.01$ and $T_d = 0.001$.

In constant power regulated pumps, when the loading pressure increases, the pump flow rate should be decreased quickly to avoid having power shocks on the prime mover. So, when there is a high demand to protect the prime mover against the power shocks, PD parameters should be tuned to decrease the pump settling time in order to keep limited power shocks on the prime movers during the transient periods. In this view, the pump control signal that is coming out of the PD controller equals $K_p[1 + 1/T_iS + T_dS]$, where $K_p = 1$, $T_i = \infty$ and $T_d = 0.02$.

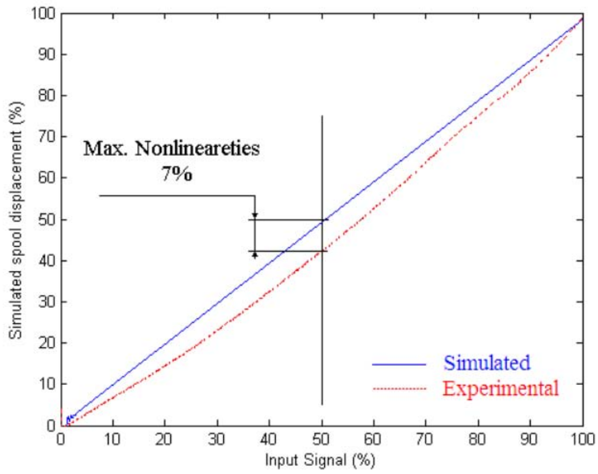


Fig. 4: *Open loop static characteristic of the proportional valve*

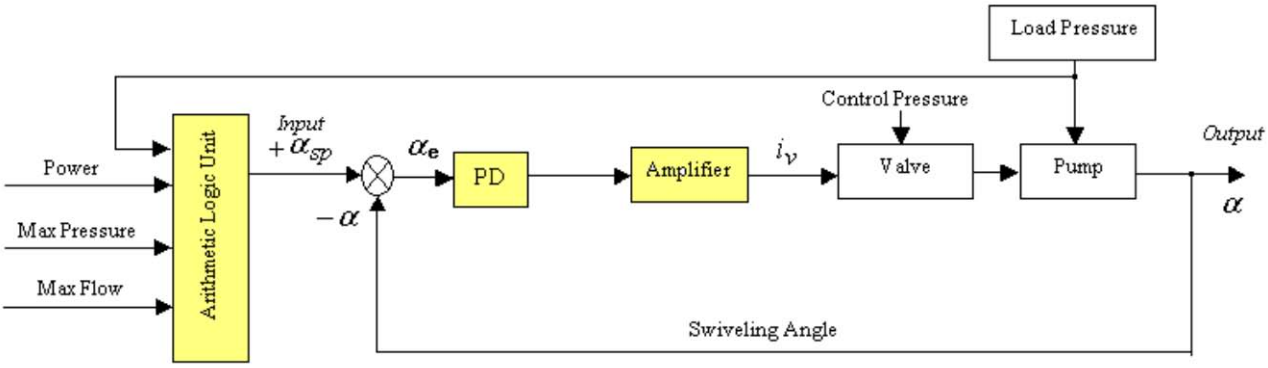


Fig. 5: *Block diagram represents Matlab/Simulink program modified to simulate the pump performance in single feedback control loop*

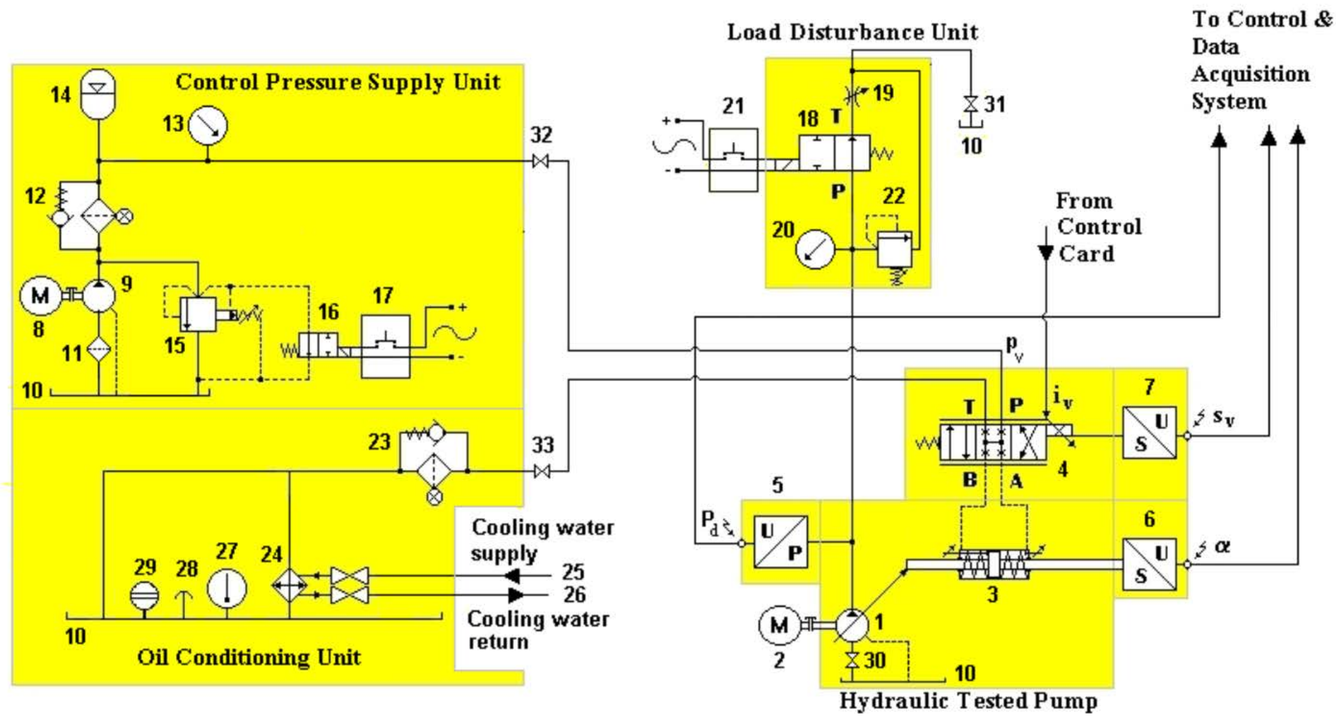


Fig. 6: *Circuit diagram of the hydraulic test bed*

In order to have relatively more robust control action in case of deteriorated operating conditions, fuzzy logic controller is constructed and used in (Kassem and Bahr, 2001) and (Bahr, Svoboda and Bhat, 2002) to replace the currently used PD controller in double feedback control loop. Inference unit of the fuzzy controller is constructed based on the change and the rate of change in the error of the swash plate inclination angle to give the equivalent control action of the PD controller. Simulation of the pump dynamic performance with fuzzy controller was repeated under the same loading conditions. Results are presented and discussed in section 6.

4 Implementation of a Single Feedback Control Loop Instead of Double

When the electrical feedback line of the proportional valve spool is removed, the spool moves in an open loop against the side spring. The valve open loop static characteristics of the proportional valve is simulated and measured experimentally, results are presented in Fig. 4. The Figure shows that the valve open loop static characteristic practically experiences nonlinearity of maximum 7 %.

The spool displacement non-linearity could be referred mainly to the electro-magnetic characteristics of the proportional solenoid. Counting on the shown open loop static characteristics of the proportional valve, a single feedback control loop, shown in Fig. 5, that contains PD controller is proposed to replace the double feedback control loop currently in use. The PD controller in this case is further tuned to suite the working conditions in the proposed single control loop. The best performance of the pump was found at $K_p = 1$, $T_i = \infty$ and $T_d = 0.01$. Computer runs were carried out to simulate the dynamic performance of the same pump, under the same loading conditions, with the proposed single feedback loop. Analytical findings are verified experimentally. Results are presented and discussed in section 6.

5 Experimental Measurements

As shown in the Fig. 6, the hydraulic test bed consists basically from the tested pump 1, which is coupled with electric motor 2. The swash plate is rigidly connected to a built in symmetrical hydraulic cylinder 3, which drives the swash plate to change its inclination angle. Position of the piston of the symmetrical hydraulic cylinder is controlled by means of a hydraulic proportional valve 4, which is integrated with the pump. The tested pump is equipped with three transducers; a pressure transducer 5 that senses the pump delivery pressure, LVDT position transducers 6 and 7 that sense the swash plate position and the proportional valve spool displacement, respectively. These transducers produce voltage signals proportional to the measured variables. The output signals are fed back to the control and data acquisition system.

A control pressure supply unit is used for supplying the control pressure to the hydraulic proportional valve. In the earlier designs, the control pressure was taken as a branch from the pump delivery pressure. Recently, the need for having a separate constant control pressure supply arises to avoid the effect of the frequent change in the pump delivery pressure on the control process, particularly at reduced system loads.

The external control pressure supply unit consists of an electric motor 8 of suitable output power coupled with the control pressure pump 9 that can afford pressure up to 15 MPa. The control pressure pump draws the fluid from the main reservoir 10 via a 100 μm mesh size strainer 11. For the high sensitivity of the tested pump to the hydraulic fluid contamination, the control pressure pump is equipped with a pressure line filter 12 of a proper flow capacity and a 5 μm mesh size. The control pressure is measured using a dial pressure gauge 13. Control pressure supply line is connected in parallel to an accumulator 14 in order to absorb the possible variation of the control pressure and to keep it as a constant supply pressure. The control pressure pump is protected against overloading by a pilot operated pressure relief valve 15, which should be connected in parallel to the main supply line in the nearest possible point to the pump exit. The pressure relief valve is integrated with unloading valve 16, that is electrically actuated by push button (17), in order to remotely apply or release the control pressure.

Load disturbance unit is built to simulate different modes of change in the external load pressure. The unit consists of a 2/2 hydraulic directional loading valve 18 that initially connects the tested pump supply line to the tank via a throttle valve 19. The throttle valve is used to adjust the loading pressure to a certain value that can be measured by the dial gauge 20. When the push button 21 is down, the spool of the loading valve moves against the side spring closing the current connection of the pump with the tank line. Consequently, the tested pump is suddenly subjected to the maximum pressure adjusted by the pressure relief valve 22 simulating stepwise change in the load pressure. For testing the constant power operation of the pump, gradual change of the load pressure is needed. In this regard, the hydraulic directional loading valve 18 is kept in its initial position connecting the tested pump with the tank via the throttle valve 19. The throttle valve is then used manually to change the load pressure gradually.

Oil is kept clean by using online return filter 23 in direction of the control pressure return line. Water oil cooler 24 is connected in parallel to the main return line, after the return filter, to partially cooling the return oil and keeping its temperature within 55 to 60^{o}C as recommended. The cooler is connected to the cooling water supply and return via shutoff valves 25 and 26 respectively. The oil tank is equipped with some necessary accessories.

A thermometer 27 used for measuring the oil temperature, an air breather 28 in order to guarantee clean breathing and an oil level indicator 29 used to measure the oil quantity in the tank. The tested pump suction and return lines are connected to the oil tank via shutoff valves 30 and 31 respectively. Also, the control pressure supply and return line are connected to the oil tank via shutoff valves 32 and 33 respectively. These valves are considered in the hydraulic circuit design in order to facilitate disconnecting the different units from the tank without the need to emptying the tank.

The hydraulic test bed is interfaced with a real time control and data acquisition system. Contents of the electronic card are replaced by real time control software interfaced with the hydraulic test bed in order to facilitate controllers tuning and prototyping of the new proposed control schemes.

6 Discussion of the Results

Figure 7 shows a comparison of the theoretical and experimental results of the pump static characteristics at constant load. Results show that, in case of using either PD or fuzzy controller in double feedback loop, pump performance experiences linear static characteristics with nearly 1 to 2 % vibration in the steady state conditions. This vibration within the acceptable range and can be referred to the effect of the periodic lateral moment acting on the swash plate, dynamics of the loading unit which is not considered in the model, sensors dynamics and selected PD parameters that reduces the settling time. In case of using single feedback control loop, in spite of the nonlinear characteristics of the pump, the vibration in the steady state conditions is suppressed due to having less responsive system. Figure 8 shows a comparison of the theoretical and experimental results of the pump constant power operation. The presented results confirmed what is just previously discussed in regard to the steady state vibration and the linear characteristics of the pump. In view of the presented results, using single feedback control loop scheme shows not to fully utilize the prime mover power due to the nonlinear pump characteristics. Pump response to the stepwise increasing and decreasing of the load pressure, are shown in Fig. 9 and 10, respectively. The evident agreement between the theoretical and experimental results fairly validates the developed mathematical model when 5 ms delay time is considered in the simulation program to simulate the natural delay in the system. Results show that, in case of using double feedback control loop scheme, rise time is found equal 60 ms and 80 ms in case of using PD and fuzzy controller, respectively. In spite of having relatively faster response with PD, the fuzzy is found globally capable to replace the conventional PD one and doing nearly the same basic control action.

In case of using single feedback loop, Fig. 9c and 10c, pump performance is shown more gradual and rise time increases to 120 ms. The slowed down motion of the swash plate reduces the mechanical impact, but this is only true, when the maximum swash plate angle is commanded. On the other hand, having relatively longer settling time increases the power shock on the prime mover. Advantage of having better and gentle valve performance, as shown in Fig. 11, is added to the account of fuzzy controller and single feedback loop. No impact of the valve spool, with the physical ends of its stroke, is recorded in case of using fuzzy controller. Such valve performance increases its service life and decreases the heat generated in it. It is recorded in case of using single feedback control loop when the command signal is above 25%.

Table 1: Qualitative evaluation of the pump response features with different control schemes

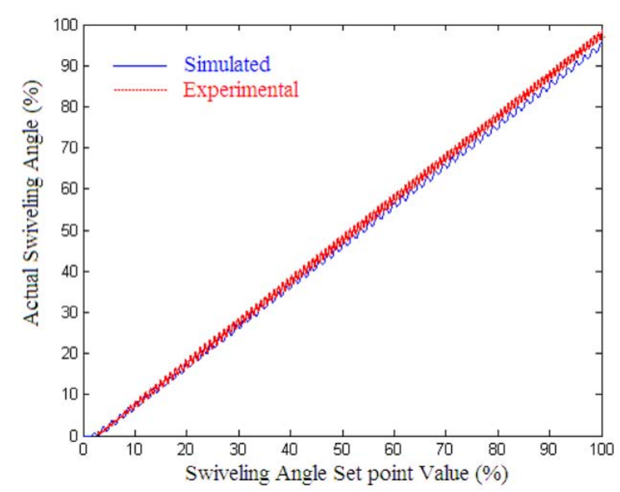
Pump Response features	**Case1**	**Case2**	**Case3**
1-Speed of response	✔	✔	×
2-Linearity	✔	✔	×
3-Convenience to constant power operation	✔	✔	×
4-Valve performance	×	✔	✔
5-Vibration in the steady state conditions	×	×	✔
6-Production cost	×	×	✔
7-Impact at the control piston end strokes	×	×	✔

Qualitative evaluation of the pump performance with the previously discussed different control schemes is concluded in Table 1. The “right” sign means relatively better. Cases 1 to 3 are assigned as follows:

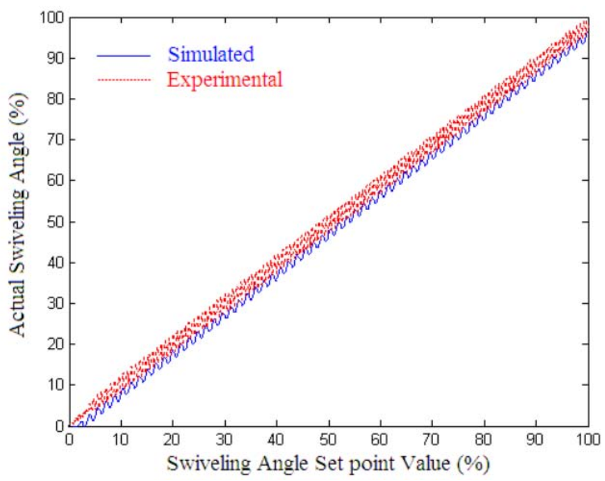
- Case1 is the case of using double feedback control loop with PD controller of customized parameters.
- Case2 is the case of using double feedback control loop with fuzzy controller.
- Case3 is the case of using single feedback control loop with PD controller of customized parameters.

7 Conclusion

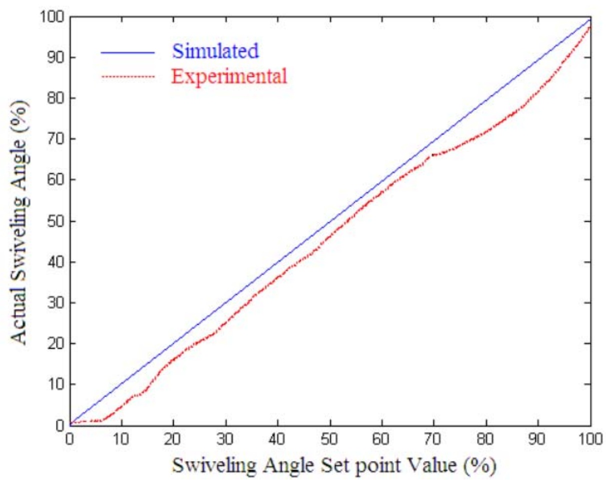
Mathematical model for the electrically controlled constant power regulated swash plate axial piston pump with conical cylinder block has been previously developed and experimentally validated. The pump currently equipped with double negative feedback control loop scheme. Simulation of the pump with the currently used control scheme experiences linear static characteristics and convenience to the constant power operation. Despite, steady state vibration of the swash plate is recorded and impact of both the valve spool and control piston with their physical stroke ends is recorded, particularly in case of using PD pump controller. In this paper, the model is used to implement pump single feedback control loop as an alternative control scheme of the double feedback loop one currently in use. Results show that using single feedback control loop reduces the speed of response and causes nonlinear static characteristics, which is inconvenient to the constant power operation of the pump. On the other hand, using such single loop suppress the steady state vibration, causes relatively good valve performance and reduces the impact on the control piston at the ends of its strokes. The main advantage of using single feedback control loop is reducing the pump production cost with having reasonable performance that might be acceptable for various commercial applications. It can be concluded that, in case of pump constant power operation, it is recommended to use the double negative feedback control loop scheme, while in other cases or if there is no high demand to protect the pump drive motor, single feed back control loop scheme is recommended.



7.a PD controller in double feedback loop

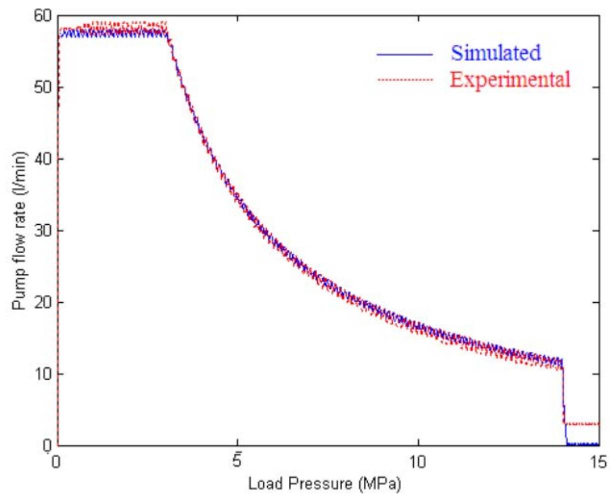


7.b Fuzzy controller in double feedback loop

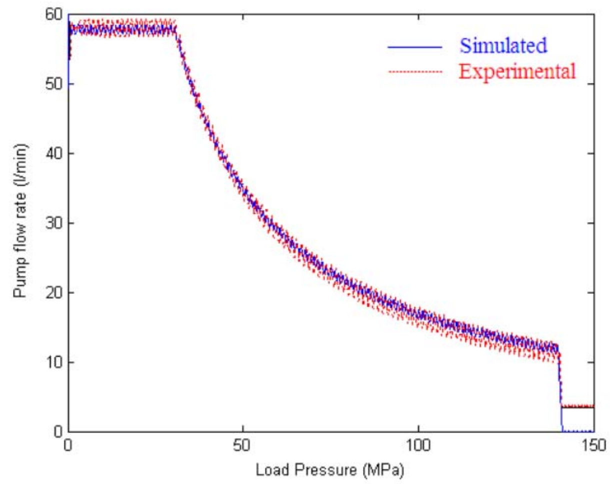


7.c PD controller in single feedback loop

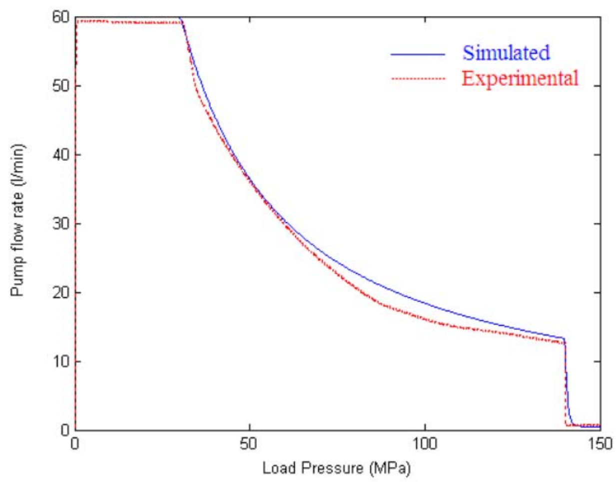
Fig. 7: *Pump static characteristics at constant load*



8.a PD controller in double feedback loop

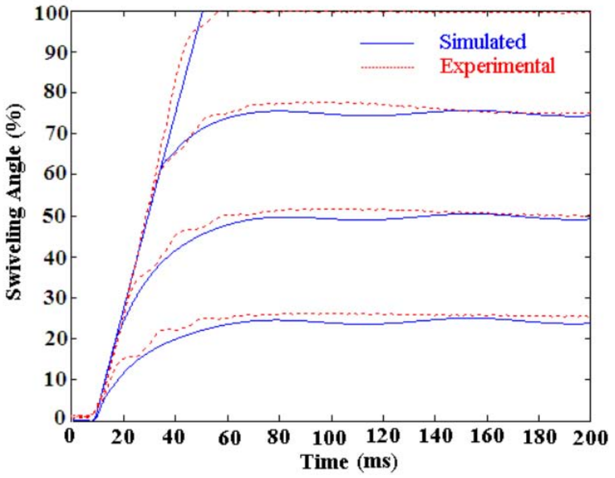


8.b Fuzzy controller in double feedback loop

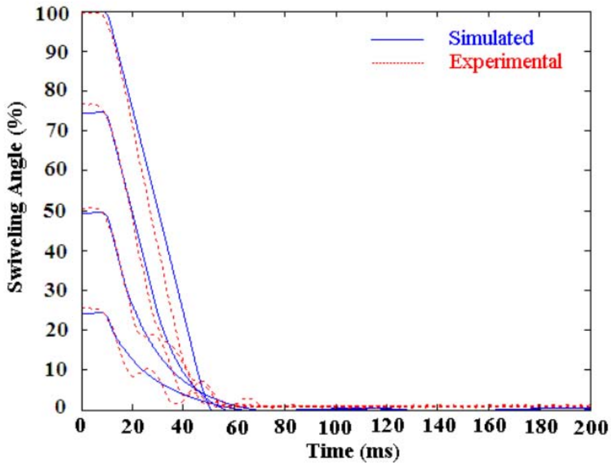


8.c PD controller in single feedback loop

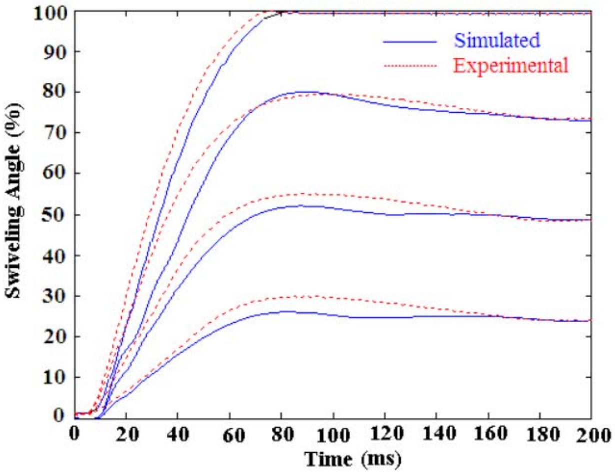
Fig. 8: *Pump constant power operation*



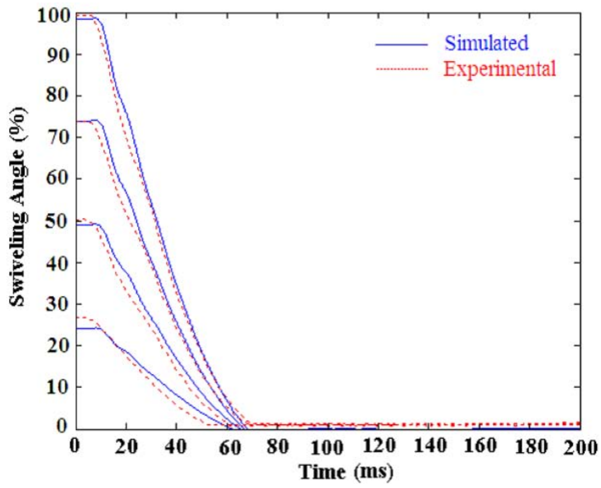
9.a PD controller in double feedback loop



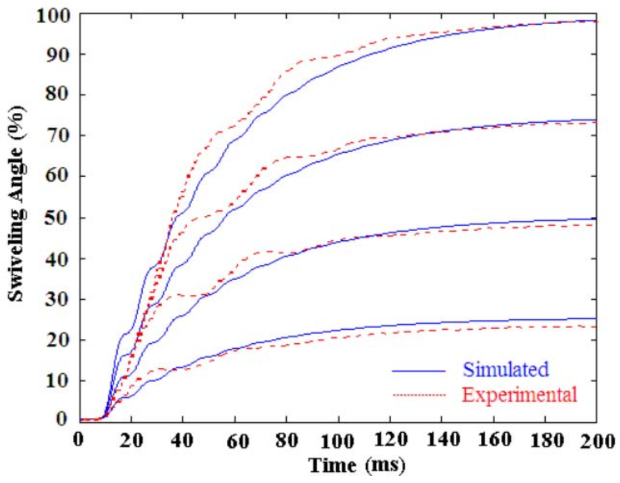
10.a PD controller in double feedback loop



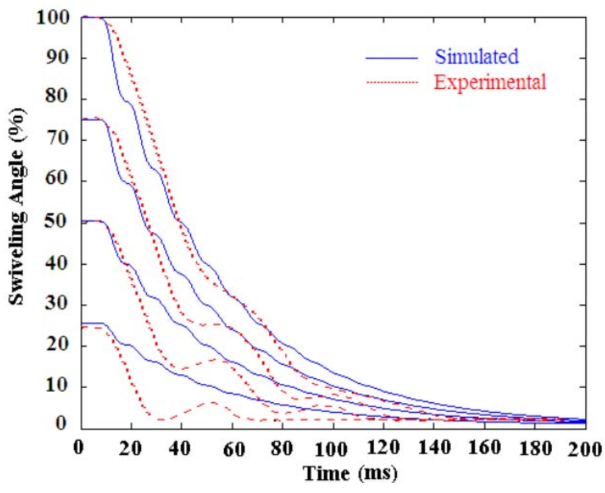
9.b Fuzzy controller in double feedback loop



10.b Fuzzy controller in double feedback loop



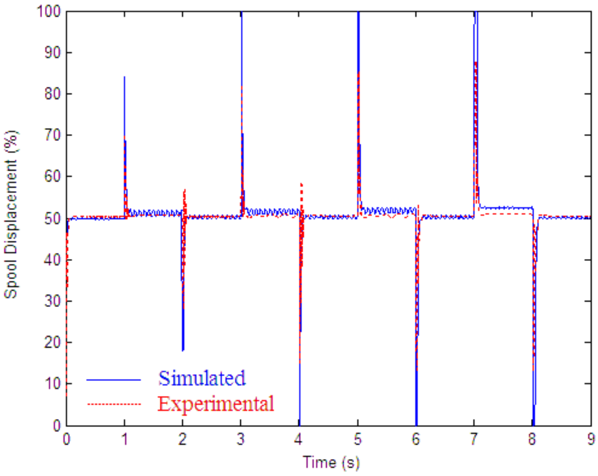
9.c PD controller in single feedback loop



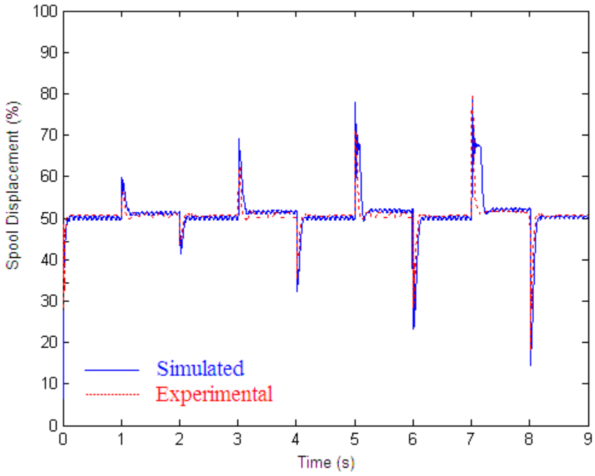
10.c PD controller in single feedback loop

Fig. 9*:* *Swash plate response to the stepwise decreasing in the load pressure*

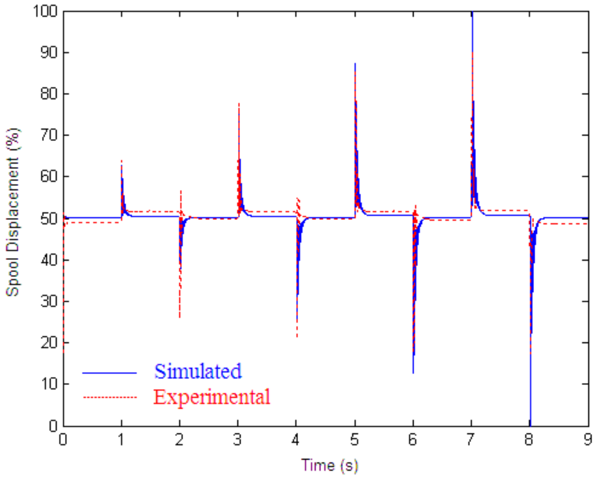
Fig. 10: *Swash plate response the stepwise increasing in the load pressure*



11.a PD controller in double feedback loop



11.b Fuzzy controller in double feedback loop



11.c PD controller in single feedback loop

Fig. 11*: Proportional valve behaviour in response to the stepwise change in the load pressure*

Nomenclature

K_p	Proportional gain	
i_v	Valve solenoid control current	[A]
K_u	Ultimate gain	
P_u	Ultimate period	
s_e	Error in the spool displacement	[m]
S_{sp}	Spool displacement set point value	[m]
s_v	Spool displacement actual valve	[m]
T_d	Derivative gain	
T_i	Integral gain	
α	Swivelling angle actual value	
α_e	Error in swivelling angle	
α_{sp}	Swivelling angle set point value	

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