EMPIRICAL INVESTIGATION ON THE INFLUENCE OF BENT AXIAL PISTON PUMPS’ CONTROL SYSTEMS’ GAINS ON HYDRAULIC PIPES STABILITY

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ABSTRACT

Due to their superior performance, high specific power, and flexible operating adaptability, axial piston pumps are widely used in both industrial and mobile applications. In this type of pumps, the pump flow is designed to meet load requirements by the aid of control unit. A pressure transducer senses the load and feeds this information to the control system’s arithmetic unit which computes the required piston stroke to generate the required flow rate. The existing pump’s control system consists of double negative feedback control structure. The inner loop controls the proportional hydraulic valve position; while the outer loop controls the swash plate angle. In order to reduce the pump production cost, simplify its design, and improve its performance; the control system can be redesigned and re-constructed to meet the desirable performance. The objective of this paper is to propose a novel single PID controller and investigate the pump overall performance including the hydraulic system smoothness (pipe vibration) and improving its response to load requirement. The results are simulated and then validated experimentally, which show promising results, and the new strategy can be implemented on the axial piston pumps successfully.
Key words: Axial Piston Pump, PID Controller, PD Controller, Vibration Signature, Pipes Stability.

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1. INTRODUCTION

In the past thirty years, extensive studies were carried out on axial piston pumps. The early models were named barrel cylinder block (conventional design), as the cylinder block was parallel to its axis. Akers and Lin [1] modeled the pump to regulate the pump pressure. The pump had a combination of an axial piston pump and single-stage electro hydraulic valve that controlled the pump flow rate. They applied a step input and investigated the transient response by applying optimal control theory for the system open loop. They found that the system performance was improved. Another study was done by Kaliftas and Costopoulos [2] (modeled and studied the static and dynamic characteristics of the pump that was equipped with pressure regulator). The results were obtained numerically and compared with the manufacturer’s dynamic operating curves. In [3], Manring and Johnson modeled the barrel pump and studied the effect of some parameters on the pump performance. Recently, bent axial pump was introduced (cylinder axis and the pump rotating shaft have an inclination angle between them called the conical angle). The advantages of this arrangement are: increased pump specific power, reduced detachment force, and increased piston stroke. This design was extensively studied by many researchers, for example [4, 5, 6], and [7]. Khalil et al [4] modeled and simulated the pump kinematics, and in [5], they proposed a pump controlled by a single PD feedback loop and compared it with different control strategies. However, they did not investigate the pump performance during the delivery and they did not investigate the operation smoothness. Chikhalsouk and Bhat [7] derived new simplified expressions for the piston kinematics and cylinder pressure and flow rate (by using the pump geometry). The expressions were compared with those derived by Khalil [6] (by using frame coordinate approach), and the results showed good agreement between the two approaches. Although variable pressure or pressure compensated systems can be used to improve efficiency, loss across the flow valve or control valve still exists. This has led to the development of a different “load sensing system”, which is commonly found in mobile hydraulics as a “driving concept” with high running efficiency [Backe, 1991], [9]. Load sensing systems use a load-sensing valve to sense the load pressure which is then fed back to a pump compensator. By means of a compensator control valve, the displacement of the pump is adjusted to deliver the required flow and maintain a pressure 70-140 kPa higher than the load pressure. This desired constant pressure difference across the flow metering valve is set by the compensator. Thus the pump pressure follows changes in the load pressure, while the pump provides only the flow demanded by the metering valve. Another problem with load sensing systems is the risk of instability which can occur through the pressure feedback line. To make the load sensing system more stable, different kinds of hydraulic “signal filters may be used in the load sensing line. However, in many cases, this kind of filtering slows down the system dynamic response. Many studies
have attempted to improve the dynamics of load sensing systems, such as using electric hydraulic load sensing systems [Backe, 1991, 1993; Luomaranta, 1999], [10]. In their studies, the load pressure was measured using a pressure transducer on the load sensing valve; in addition the pump was equipped with an electro-hydraulic directional valve to control the displacement of the pump. The load sensing line was replaced by an electric signal line including a pressure transducer, electrical controller/filter and an electrically controlled load sensing pump. With this electric load sensing line, different control strategies would be implemented. With the help of an electronic filter and controller, any oscillation in the load sensing signal would be attenuated; thus, it was possible to design a load sensing system that was stable but still demonstrated fast response. The overall efficiency for a very simple pump-controlled hydraulic system under ideal operating conditions is about 70% [Cundiff, 2002], [11]. The total efficiency of a pump/motor combination is much less when the system operates in a low rotational speed range. If hydraulic control valves are included to control the actuators in hydraulic drive systems, the overall efficiency can be substantially reduced under certain loading conditions. The fluid power industry is conventionally classified into industrial and mobile hydraulics. In industrial hydraulics, the power supply can be set to be sufficient to satisfy the system requirement regarding both flow and pressure, [12]. The model of a hydraulic pump which was similar to that used in this study had been verified by Kavanagh [1987], [13]. The parameters of the pump model were experimentally measured over a wide range of the pressure. Hence, it was believed that the model of the pump was correct and that the error in model predictions was possibly due to the model of the DC motor rather than the pump. Further, it was believed that the major problem was the prediction of the electrical time constant which was mainly related to the inductance of a DC motor. One problem with load sensing systems for multi load applications is stability which can arise from load interactions through the feedback line. To minimize these interactions, pressure compensated (PC) control valves are often used. Although they are not more efficient than the traditional load sensing systems, they can be used to minimize interactions. [14] Matching the pump flow to a varying demand load flow can improve the power efficiency due to the elimination of the loss across the relief valve. Normally, a variable displacement pump is used. A study by Mansouri et al. [2001, [15]] gives another approach. A latching valve, which switches the on/off position extremely rapidly (750μs), but remains latched in the closed or open position using residual magnetism, was used to control the flow output in order to achieve a variable flow supply. When the latching valve is in the closed position, pump flow is directed to the “hydraulic rail” and compressed to high pressure fluid. In the open position, flow is “shorted” back to the inlet of the pump (at low pressure). By applying switched-mode control to change the state of the control valve, the flow could be modulated with minimal losses; further a variable pump with excellent transient response characteristics can be emulated. Energy can be saved with this approach compared to conventional variable displacement pumps, particularly at partial pump load conditions. If only an approximate performance is required. A similar study has been done by Martin [1992], [16], in which a model for a Moog773 servo valve was developed and then simplified by (a) neglecting factors which had minimal effects on the performance of the system and (b) by linearizing the model. Martin found that there are two important model equations for the servo valve. One is the electrical model of the torque motor that relates the current through the coils of the torque motor to the voltage across the coils. The other equation is the hydraulic model that relates the flow rate through the valve to the current in the coils. The flow rate
across the hydraulic motor is affected by leakage and fluid compression. The leakage term in Equation D.4 is proportional to the pressure drop across the leakage path. Leakage in the hydraulic motor is also known to be the function of motor rotational speed [Merritt, 1967] but for this model and for the initial controller design, the simplified model of leakage in Equation D.4 was used. For the feasibility study, the effects of the lines between the pump and motor are considered negligible. Compressibility effects due to the volume of fluid in the connecting lines are simply lumped into the volume of motor piston chambers. Using Newton’s second law, the torque equation of the motor is [Merritt, 1967], [17]. In the present paper, a new control strategy is proposed to control the bent axial piston pump. This strategy has a single feedback PID controller. The expected advantages are: simple design, low cost, improved responsive performance to the requirements, and smooth operation that reduces vibration/noise levels and increases its service life. The paper is divided into the following sections: A short introduction with brief literature review about the axial piston pumps’ control system strategies, the axil piston pump’s structure and the control units and axial piston pumps strategies are explained in items 2 and 3. The axial piston pump mathematical model and the two different control strategies are developed in section 4. The experimental set up, the results are presented in section 5 and 6. Finally, the work is summarized in a brief conclusion.

2. AXIAL PISTON PUMPS’ STRUCTURE

Figure 1 shows the major components of bent axial piston pump. The pump consists of finite number of pistons (9) nested in their block, in circular array and bent about its axis with an angle (conical angle). The cylinder block is held tightly against a valve plate using the force of the compressed cylinder-block spring and a less obvious pressure force within the cylinder block itself. A thin film of oil separates the valve plate from the cylinder block which, under normal operating conditions, forms a hydrodynamic bearing between the two parts. A ball-and-socket joint connects the base of each piston to a slipper. The slippers themselves are kept in reasonable contact with the swash plate by a retainer mechanism and a hydrodynamic bearing surface separates the slippers from the swash plate. The swash-plate swiveling angle, α, is generally controlled by an external control mechanism.

![Figure 1 Components of a Bent Axial Piston Pump](image-url)
3. CONTROL UNITS AND CONTROL STRATEGIES FOR AXIAL PUMPS

3.1. Problem Statement
In the earlier pump designs, the pump was introduced without control unit and was inappropriate for variable loads. To manage the excess flow, a discharge valve was used to manage the excessive flow. A relief valve solved the excessive flow problem; however, a fluid overheating was encountered. The overheating deteriorated oil properties and damaged the pump parts. The loss of the pump power and the overheating were major challenges that engineers encountered and a control system was required to elucidate these difficulties.

The later design equipped the pump with a double negative feedback control unit to control both valve spool and swiveling angle to match load requirements by controlling the swash plate inclination. The pump power matched completely the load, and the oil retained its properties. This solution was costly. More parts were needed, and exposed a kind of sluggish response. In [5], a single PD feedback control loop was suggested and compared with several control strategies. The objective was to simplify the control structure and the related electronics, which reduced the pump’s production cost. However, the piping system smoothness was not investigated, which is considered as defective components (if the vibration level exceeds certain levels) in hydraulic systems.

In the coming sub-item, the control unit components, the pump mathematical model, and the single feedback control diagram illustration will be described.

3.2. Control Unit Components
The control unit (shown characteristically in Figure 2) is enclosed of an open center proportional directional valve and symmetric hydraulic control cylinder. When the valve solenoid gains a control signal above zero in an open loop, a proportional electromagnetic force continues on the valve spool and powers it to transfer against its return spring. A simple second order differential equation is used to define dynamic of the valve spool displacement and control system. Control piston of the symmetric hydraulic control cylinder is modeled as physical integrator. Also, second order differential equation is used to represent the angular momentum of the swash plate swiveling motion considering the spring, damping and force effects of the attached control piston. Pressure difference across the control piston is occasioned as a significance of the movement of the valve spool. Control piston is attached mechanically to the swash plate. The model is further developed to include dynamic of the swash plate considering all the moments acting on it that would be overcome by the control unit; inner feedback control loop is used for valve spool accurate positioning using displacement sensor and PID controller. Control pressure across the control piston, due to movements of the proportional valve spool, can be initiated by solving the continuity equation for the variable control volumes of the control piston side chambers. Outer feedback control loop is implemented to control the swiveling angle using displacement sensor and PD controller. System pressure is permanently monitored, using pressure transducer, and fed back to an arithmetic unit. The arithmetic unit provides the angle set point built on the pump requirements (constant power operation). Power, maximum/maximum pressure and flow that should be followed by the control unit are fed to the control card electronically. Pump and valve controllers are shown in Figure 2.
Figure 2 A diagram symbol of the control unit/pumps, and the hydraulic flows, [7].

4. PUMP MATHEMATICAL MODEL
The swash plate is modeled as a single degree-of-freedom system (SDOF) with reference of the swiveling angle of the swash plate about the y axis i.e. $\alpha$. The swash plate mass moment of inertia, piston spring stiffness, and damper coefficient are denoted as $J_{eq}$, $K_{eq}$, $C_{eq}$, respectively. Moreover, such the hydraulic and its damper coefficients are.

The equation of motion is expressed as:

$$J_{eq} \alpha + C_{eq} \alpha + K_{eq} \alpha = M(t) \quad (1)$$

The transfer function for the swash plate can be expressed in S-domain as

$$T = \frac{\alpha(s)}{M(s)} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \quad (2)$$

Where
- $\alpha^-$ = angular acceleration
- $\alpha'$ = angular velocity
- $\alpha$ = angular displacement
- $M$ = Applied moment on the swash plate
- $\omega_n$ = Natural frequency rad/sec
- $\zeta$ = Damping ratio
- $S$ = Laplace variable

The proper controllers three gains (proportional, integral, and derivative) govern the hydraulic system overall performance (load responsiveness and stability), which are leading to superior pump performance; and are the top primacy for the pump designers and manufacturers.

4.1. Double Negative Feedback Loop
Figure 3 demonstrates the diagram depiction of an axial piston pump with double negative feedback control strategy. The inner loop controls the location of the spool by means of LVDT position transducer that detects the proportional valve spool movement (PID controller). While, the outer loop controls the pump inclination angle by using LVDT position transducer that detects the swash plate location (PD controller).
4.2. Single Feedback Loop

In [5], the spool dynamic performance and its linearity were tested. It indicated that the valve open loop static characteristic basically experiences non-linearity of maximum 7%; which could be denoted mostly to the electro-magnetic features of the proportional solenoid. Hence, it is suitable to advance in accepting the single strategy, which is obtainable in Figure 4.

This effort will be founded mostly on applying a single loop and matching the results with the work offered earlier in [5]. They implemented a single PD controller ($K_p=5.3$ and $K_d=0.02$) and it exhibited satisfactory results and applicability of the single loop strategy. However, they did not include extra examinations to contain for example the pipe smoothness, which is one of this study goals.

The “Ultimate Sensitivity” method presented by Ziegler–Nichols is adopted to parameterize the suggested PID controller. Using this method, the proportional gain is increased until the system started to be marginally steady and constant oscillation appears with amplitude limited by the proportional valve capacity. The equivalent gain is defined as $K_u$ (Ultimate Gain) and the oscillation period is $P_u$ (Ultimate Period). Lastly, outlining the three gains according to the method are calculated. The PID controller is more adjusted to suite the operational conditions in the planned single control loop. The ideal performance of the pump is found at $K_p=9.5$, $K_i=8.4$, $K_d=0.01$. Computer runs are carried out to simulate the dynamic performance of the pump, under the matching loading settings, with the proposed single feedback loop. Systematic results are tested experimentally. Outcomes are presented and discussed in section 6.

**Figure 3** Axial piston pump with double feedback control loop demonstration

**Figure 4** Block diagram, (single PID controller)
5. INVESTIGATIONAL SET UP

The complete set up was presented in former study paper [7]; therefore, this part will cover only the vibration level measurements. In order to record the pipe vibration levels, a piezoelectric accelerometer is sited at the midpoint of the pipe and on the pump (Figure 5). The detailed signal is instantaneously fed to amplifier-filter to increase the feature of the recorded signal and decrease the noise. To transform the analog into digital signal, software based on LABVIEW® is used to attain the data. The data acquisition card model is DAQ 6062 E (from National Instruments Inc.). The Set Up diagram illustration is showed in Figure 6. The sampling time is selected to be 5 m sec to obtain consistent results with total number of points is 2000 points for every run, which is enough to cover and characterize the final profile of the signal. After filtering the signals, they are passed through oscilloscope (Agilent – 54624 A) and then recorded. The pump flow rate is 1 L/Sec, and the fluid velocity through the pipe is 2.94 m/sec. The fluid velocity cannot generate instability (the flutter velocity is calculated to be 94.5 m/sec). The pipe is selected long enough to reflect any instability in the hydraulic unit and the length is selected to be 10 m. Two sets of pump set up are arranged to be furnished as follows:

1. Single PD feedback control loop (as proposed by Khalil [5]) demonstration to simulate the pump performance under the PD control strategy.
2. Single PID feedback control loop to simulate the pump performance under the novel proposed control structure.

The physical electrical control arrangement is substituted by real time control software to help in the prototyping of the control system. The pump is equipped with single PID controller with expedient regulated gains. And the evaluation with the present double strategy performance is conducted.

Figure 7 demonstrates the theoretical/investigational comparison of the pump static characteristics at predetermined operating condition with evaluating the vibration levels for the PD/ PID proposed single strategies.
6. RESULTS AND DISCUSSION

Figures 7-1 and 7-2 illustrate proportional valve behavior when subjected to stepwise change in the load pressure with the PD and PID controllers, respectively. In the PD case, the spool impacts rise/drop is more than 25% of its maximum value. However (PID), the spool impacts rise/drop is about 15% of its maximum value. Figures 7-3 and 7-4 indicate the swash plate response to the stepwise reducing in the load pressure. The PID strategy displays faster response to the change in the load requirements in comparison to the PD controller. For example at the full load, rise time in the PID single strategy equals to 20 sec, which is the half the recorded rise time in the PD controller. Figures 7-5 and 7-6 illustrate the pump response to ramp input. In this direction, the PID single feedback overweighs the PD controller as there is a predictable response, which is not the case of the PD controller. Moreover, Figs. 7-7 and 7-8 compares the step response of the proportional valve at several percentages 25, 50, 75, and 100%. Accordingly, in Figure 7-7, the results displays little overshooting (less than 5%) as the valve moves to a higher percentage with continuous oscillation in the valve response that can be observed during the process, which would lead to pressure oscillation. The results are illustrated in Figure 7-8. Finally, in order to guarantee the fine superiority of the proposed PID controller, another criterion is selected to be benchmarked, which is the process smoothness. The vibration signatures are recorded for the two conditions (PD and PID single feedback control strategies). The signatures are presented in Figures 7-9 and 7-10. From the figures, it is clear that the process smoothness is virtually lost with the PD control strategy and there is a kind of excessive/undesirable instability. However, the system with the proposed PID controller inclines to exhibit more stability and less oscillation, which is regular in hydraulic applications. In the hydraulic applications, it is sensible to have least instability to extend the system lifespan.
Figure 7-1 PID controller in single feedback loop: Proportional valve behavior in response to the stepwise change in the load pressure

Figure 7-2 PD controller in single feedback loop: Proportional valve behavior in response to stepwise change in the load pressure [5]

Figure 7-3 PID controller in single feedback loop: Swash plate response to the stepwise decreasing in the load pressure

Figure 7-4 PD controller in single feedback loop: Swash plate response to the stepwise decreasing in the load pressure [5]
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Figure 7-5 PID controller in single feedback loop: Pump static characteristics at constant load

Figure 7-6 PD controller in single feedback loop: Pump static characteristics at constant load [5]

Figure 7-7 PID controller in single feedback loop: Step response of the proportional valve

Figure 7-8 PD controller in single feedback loop Step response of the proportional valve [5]
7. CONCLUSIONS

The existing design of the axial piston pump is equipped with a double negative feedback control loop structure. This design is complicated and puts extra cost that makes the pump less competitive. It also has a large rise time. It was proposed earlier that the pump can complete correctly by employing a single negative feedback control loop system by eliminating the return electrical line for the inner loop. That structure was equipped with a PD controller and the results were good for certain applications but the levels of the noise and vibration were not inspected. In the present paper, a control strategy that keeps the pump with low production costs and that was accomplished by implementing a single feedback control loop to control the swash plate inclination angel. The investigational results display the speed of the swash plate response was dramatically improved (dropped from 40 to 20 ms). Additional investigational runs were steered to link the vibration levels between the single feedback control schemes with PD and PID controllers. The results displayed by the PID controller reduced the vibrations to minor levels, in comparison with those with PD controller. The proposed control strategy could be applied in the pump that necessitates high response and low vibration levels.
REFERENCES


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