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A single stage spool valve for the pressure compensator of a variable displacement pump – Design, dynamic simulation and comparative study with a real pump

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ABSTRACT

The study is focused on the design of a simplified spool valve to be incorporated in the pressure compensator of a variable displacement axial piston pump in order to perform a comparative study with a commercial pump having a two stage spool valve in its compensator. The design involves evaluation of the spool size and selection of spring from static equilibrium condition to satisfy cut-in and cut-off pressure. Following the development of dynamic model of the system, a design sensitivity analysis of the spool valve has been carried out through simulation to identify the critical sizes of the parameters, which affect the pump performance. By systematic design, it is possible to have a single stage spool valve controlled pressure compensator that can produce performance of the variable displacement axial piston pump at par with the similar commercially available pump.

Keywords: Electrohydraulic Power System, Design Technique, Modeling and Simulation, Sensitivity Analysis, Experimental Verification.

1 INTRODUCTION

The variable axial piston pump has a wide range of applications in the field of aerospace to heavy industries because of their variable displacement capabilities, high working pressures, high volumetric efficiencies, and high power densities [1, 2]. Such pumps with pressure compensator have demand in systems, where fast speed of response and power saving are of utmost importance. The power saving properties of the variable displacement axial piston pump makes the hydraulic circuit energy efficient [3]. There have been numerous research works particularly on different aspects of swash plate type axial piston pump having fixed swash angle. Ye et al. [4] developed a pump model to capture the different source of noise and introducing a damping hole in the valve plate to reduce the noise emitted from the pump. Wang [5] obtained the mathematical relation between valve plate timings and pressure carryover angles which subsequently used to investigate the containment forces and moments to address the sound properties of the pump and optimize the designs. A software based investigation had been carried out to improve the pump performance and to study the cavitation characteristic including the turbulence model, cavitation model and effect of fluid compressibility by Zhao et al. [6] through CFD simulation. Wang [7] established a strong relationship between the valve plate geometry and volumetric efficiency of an axial piston pump. Flow and pressure ripples are the major drawback for any positive displacement pump. Zheng et al. [8] numerically studied the effect of piston groove pattern and groove design on the pressure

balance and flow through the groove. Kemmetmuller et al. [9] gave an idea about the mathematical model and control design strategy of a nonlinear pressure controlled self-supplied variable displacement axial piston pump, which had a single stage proportional directional control spool valve and a spring loaded actuator to control the swash angle. Wei et al. [10] used a separate pump to supply the pilot pressure to operate the spool valve and used a control cylinder with a bias spring to make the pump variable. To provide better damping and reducing the pressure gain in dynamics performance of the pump, an orifice was placed between spool valve and control cylinder by Kaliafetis et al. [11]. Zhou et al. [12] studied about dynamic lubrication between the slipper pad and swash plate experimentally for an axial piston pump. To increase the power density of a hydraulic machine Marning et al. [13] identified some significant parameters and recognized four dimensionless quantities, which helped to improve the power density of the axial piston machine. From the available previous literatures, it can be seen that there are not much works addressing the methodology to develop the control spool of the mechanism for achieving variable displacement feature of axial piston pump. In a very recent work Mondal et al. [14] formulated an innovative design procedure by developing a mathematical model for the pressure compensator of an axial piston pump. Following the same methodology, in the present work design of a rather simple single stage control spool valve has been taken up with an aim to replace the commercially used two stage valve. A parametric study has been conducted through simulation to evaluate the dynamic performance of an axial piston pump, having the designed single stage control spool valve in the compensator. The performance of the designed spool valve driven pressure compensator has been analyzed and compared both through simulation and experiment, with the performance of a commercial pump having two stage control valve. During design and development stages, analysis through simulation is preferred [15] followed by comparing with experimental results.

As shown in Fig. 1, the basic functioning of the axial piston pump is based on the reciprocation of the individual pistons housed inside a barrel and one end of it is mounted on an inclined surface known as swash plate. By the rotation of the barrel, the pistons travel over the swash plate resulting reciprocation. Each of the other ends of the pistons is exposed to barrel cylinder, which acts as the pumping chamber. The motion of the piston causes volumetric displacement of the pumping chamber, which has periodic connections with suction and delivery manifolds through a disc known as valve plate. The algebraic sum of discharge through each piston results overall delivery flow rate of the pump. For a fixed displacement pump, the volumetric displacement per rotation remains constant, while that can be varied by appropriate mechanism in a variable displacement pump. Such mechanism for an axial piston pump is known as pressure compensator.

The pressure compensator arrangement of this pump consists of a single stage control spool valve and two actuators acting in opposite manner against the swash plate at the two ends. The balancing of torque on the swash plate produced by these actuator pistons and the barrel pistons of the pump at different swash angle is the basic principle behind variable displacement operation of the pump. One of the actuators, called the stroking cylinder is connected to the delivery line of the pump through the spool valve. The other spring loaded actuator, termed as the rate cylinder, is also connected to the delivery manifold of the pump. So there is a pressure feedback to the upstream side of the rate piston and this type of arrangement is called rate feedback. The objective of this

arrangement is to go for a much lower stiffness spring on the rate cylinder. The pre-set compression of the spring in the rate cylinder determines the swash angle corresponding to which the pump has a rated pressure and discharge. Change in swash angle thereby the variable displacement feature of such a pump depends on the commencement of the displacement of a spool valve [10, 16]. The activation of the spool valve is controlled by a reference delivery pressure also known as the pilot pressure of the spool valve, which is governed by the pre-compression of a spring fitted at the end of spool valve. This reference pressure is also known as cut-in pressure of the system beyond which the spool ports begin to open so that pressurized oil comes from delivery manifold to stroking cylinder through metered opening of spool valve. The high pressure oil acts on the stroking piston to produce additional torque on the swash plate causing unbalance to the equilibrium. As a result the swash plate starts rotating in the direction of the applied additional torque. Therefore, the swash angle gets reduced causing further compression in the spring of the rate piston. The presence of high pressure oil in the rate cylinder along with compression in the spring causes the opposing torque to increase that ultimately will ensure a new equilibrium. A situation will arise, when the swash angle becomes minimum at equilibrium. The delivery pressure at that condition is known as cut-off pressure. Since the swash angle governs the swept volume of the barrel piston, minimum swash angle will correspond to minimum flow rate of the pump at maximum pressure (cut-off). This will result minimum power loss, which otherwise will be maximum if using a pressure relief valve with a fixed displacement pump.

Following the methodology proposed in an earlier work [14], the present work has been organised first by focusing on the design of a single stage spool valve for the pressure compensator of pump. The design involves evaluation of the spool size and selection of spring from static equilibrium condition to satisfy cut-in and cut-off pressure. Following the development of dynamic model of the system with subsequent implementation in MATLAB/SIMULINK environment, other parameters of the spool valve that could affect the performance of the pump have been evaluated through dynamic simulation. Results obtained from simulation for the present design has been compared with a commercial pump (A10VSO45 series pump from Rexroth) [17] having compensator with double spool in order to study the effectiveness of the simplified compensator design.

Fig. 1 Schematic of a swash-plate axial piston pump with pressure compensator

2 DESIGN OF SPOOL VALVE FROM STATIC CONDITION

The spool valve is the key element in the pressure compensator. The performance of the spool valve governs the characteristics of variable displacement axial piston pump. Following are the considerations for the design of spring and spool land of the spool valve, the schematic of which is shown in Fig. 2.

- The spool displacement starts at cut-in pressure P_{dci} , and attains its maximum displacement at the cut-off pressure P_{dco} .
- The flow force F_s , acting on the spool is neglected under static condition.
- The effect of radial clearance is neglected.

In static condition at the cut-in limit, the active pressure P_d^* acting at the left side of the spool remains at P_{dci} , whereas the back pressure P_l acting on the right side of the spool remains at the case pressure P_r . Under this situation the hydraulic force on the spool should be balanced by the spring force due to a spring pre-compression δ_{s0} leading to the relation

$$(P_{dci} - P_r)(\pi d_s^2 / 4) = k_s \delta_{s0}. \quad (1a)$$

For a realistic choice, following the previous approach of Mondal et al.[14], of both the spool land diameter, d_s and spring pre-compression, δ_{s0} , the spring stiffness k_s has been estimated. A feasible size of the spring in terms of coil diameter, wire diameter, number of coil turns has been selected using a well-known relation [18] so as to match the estimated stiffness. The equation is

$$k_s = \frac{Gd^4}{8D^3n}, \quad (1b)$$

where G is the modulus of rigidity of the spring material, d is the wire diameter, D is the mean coil diameter and n is the number of turns.

In order to accommodate the available spring, necessary modifications in terms of increase in the right side spool land diameter and the chamber length have been done. It can be observed that this diameter d_{sr} could be larger than the diameters (d_s) of the other two lands. However, this increase in size would not alter the force balance equation as long as both sides of this land experience the case pressure P_r , typically for the situation under static condition.

The maximum spool displacement, which is also the maximum spring compression, is obtained from the static force balance at cut-off condition given by

$$P_{dco}(\pi d_s^2/4) - P_r(\pi d_{sr}^2/4) = k_s(\delta_{s0} + x_{sv\max}). \quad (2)$$

Table 1 lists the parameters some of which has been chosen from the earlier experience [14], while others are obtained from the above conditions.

Fig. 2 Schematic of the single-stage control spool valve

Fig. 3 Free body diagram of the spool

Table 1 Parameter values for the designed pressure compensator

Parameter (Chosen)	Value	Parameter (obtained)	Value
d_s	$6.5 \times 10^{-3} \text{ m}$	k_s	68786 N/m
δ_{s0}	$8.6 \times 10^{-3} \text{ m}$	$x_{sv\max}$	$2.408 \times 10^{-4} \text{ m}$
G	$79.3 \times 10^9 \text{ Pa}$	d_{sr}	20.5 m
d	$4 \times 10^{-3} \text{ m}$	D	$16 \times 10^{-3} \text{ m}$
n	8		

3 DYNAMIC MODEL OF THE COMPENSATOR SYSTEM

3.1 Spool Valve Dynamics

As shown in Fig. 2, the left side of the spool valve is connected to the delivery side of the pump through an entry orifice of diameter d_{od} , due to which the pressure remains at P_d^* instead of delivery pressure P_d . The right end side of the spool is connected to the reservoir through an orifice of diameter d_{or} , due to which the pressure remains at P_l instead of reservoir pressure P_r . The chamber between the middle and right land of the spool valve is directly connected to the reservoir, thereby considered to experience the same reservoir pressure. This pressure acting on both the opposing surfaces will not result any net force acting on the spool. Under these considerations and with reference to Fig. 3, the spool displacement equation, in terms of the conditions between the net hydraulic pressure force F_p and the spring force F_{sf} , can be written as

$$\ddot{x}_{sv} = 0 \text{ for } F_p \leq k_s \delta_{s0}, \quad (3a)$$

$$\ddot{x}_{sv} = (F_p - F_d - F_s - F_{sf}) / m_{sv} \text{ for } F_p > k_s \delta_{s0}, \quad (3b)$$

where F_d is the transient flow force and F_s is the steady flow force acting on the spool expressed respectively from Merrit [19] as

$$F_d = \rho(l_d \dot{Q}_{sv} - l_r \dot{Q}_{lu}), \quad (4a)$$

$$F_s = 2C_d C_v \{A_d (P_d^* - P_{sc}) \cos \theta_{jd} + A_r (P_{sc} - P_r) \cos \theta_{jr}\}, \quad (4b)$$

where C_d and C_v are the coefficients of discharge and velocity respectively, l_d and l_r are the two damping lengths as shown in Fig. 2. The two metered flows Q_{sv} from the spool valve to the stroking cylinder and Q_{lu} to the return chamber from the stroking cylinder can be expressed in terms of the respective pressure drops as

$$Q_{sv} = C_d A_d \sqrt{2(P_d^* - P_{sc}) / \rho} \operatorname{sgn}[P_d^* - P_{sc}], \quad (4c)$$

$$\text{and } Q_{lu} = C_d A_r \sqrt{2(P_{sc} - P_r) / \rho} \operatorname{sgn}[P_{sc} - P_r], \quad (4d)$$

where θ_{jd} and θ_{jr} are respectively the jet angles of the metered flows Q_{sv} and Q_{lu} . Assuming near-zero radial clearance both the jet angles are assigned as 69° from Merrit [19]. From the geometry shown in Fig. 2, the variable opening areas A_d and A_r due to the spool displacement can be expressed as

$$A_d = n_p r_p^2 \{\theta_d - (\sin \theta_d)\} / 2, \quad (5a)$$

$$\text{and } A_r = n_p r_p^2 \{ \theta_r - (\sin \theta_r) \} / 2, \quad (5b)$$

for n_p numbers of circular port cuts of radius r_p , where as θ_d and θ_r are the angles subtended by the intersections of the spool land with the circular port cut at the delivery and return sides respectively as shown in the expanded view in Fig. 2. For a spool displacement x_{sv} , overlap o_d at the delivery side and underlap u_r at the return side, these angles can be obtained as

$$\theta_d = 2 \cos^{-1} \{ 1 - \max(x_{sv} - o_d, 0) / r_p \}, \quad (6a)$$

$$\text{and } \theta_r = 2 \cos^{-1} \{ 1 - \max(u_r - x_{sv}, 0) / r_p \}. \quad (6b)$$

Considering compressibility effect, the dynamics of the pressure P_d^* in the left chamber of the spool valve can be expressed in terms of the velocity of the spool, flow rate through the entry orifice, bulk modulus of the working fluid and dead volume of the chamber as

$$\dot{P}_d^* = \beta \{ Q_{ol} - \pi(d_s^2/4)\dot{x}_{sv} \} / \{ v_0 + \pi(d_s^2/4)x_{sv} \}, \quad (7a)$$

in which the flow rate Q_{ol} through the entry orifice of diameter d_{od} can be expressed as

$$Q_{ol} = C_d (\pi d_{od}^2 / 4) \sqrt{2(P_d - P_d^*) / \rho} \operatorname{sgn} |P_d - P_d^*|. \quad (7b)$$

In case, the pressure in the left chamber of the spool valve is considered same as the delivery pressure of the valve, the flow rate Q_{ol} can be found using continuity equation inside the chamber considering incompressible flow as

$$Q_{ol} = \pi(d_s^2/4)\dot{x}_{sv}. \quad (7c)$$

In a ploy to achieve damping in the spool motion, an orifice of diameter d_{or} is provided in the drain side of the spool valve as shown in Fig. 2, such that a back pressure P_l is developed at the return side chamber due to which a force acts on the rightmost spool land opposing the spool motion as written in Eq. (3b). In terms of the spool motion and bulk modulus β , the dynamics of this back pressure is expressed as

$$\dot{P}_l = \beta \{ \pi(d_{or}^2/4)\dot{x}_{sv} - Q_{or} \} / \{ \pi(d_{or}^2/4)(l_{sr} - x_{sv}) \}, \quad (8a)$$

where l_{sr} is the length of the rightmost spool chamber after the pre-compression that can be evaluated from the knowledge of the free length of the spring selected through static design and the amount of pre-compression and Q_{or} is the flow through this damping orifice given by

$$Q_{or} = C_d (\pi d_{or}^2 / 4) \sqrt{2(P_l - P_r) / \rho} \operatorname{sgn} |P_l - P_r|, \quad (8b)$$

In case, the back pressure is considered same as the case pressure of P_r , the flow rate Q_{or} can be found using continuity equation inside the chamber considering incompressible flow as

$$Q_{or} = \pi(d_{or}^2/4)\dot{x}_{sv}. \quad (8c)$$

Satisfying continuity across the valve, the flow Q_{sc} entering to the stroking cylinder from the spool valve that can be written from Fig. 2 as,

$$Q_{sc} = Q_{sv} - Q_{lu} - Q_c, \quad (9a)$$

where Q_c is the flow through the clearance orifice of diameter d_c of the spool valve that can be expressed as

$$Q_c = c_d (\pi d_c^2 / 4) \sqrt{2(P_{sc} - P_r) / \rho} \operatorname{sgn} |P_{sc} - P_r|, \quad (9b)$$

while, Q_{sv} and Q_{lu} have already modelled in Eqs. (4c) and (4d).

Of course, there is a direct interaction of the spool valve dynamics with the swivelling dynamics of the swash plate including the rate and stroking cylinders of the pressure

compensator. The mathematical model for the dynamics of the pressure compensator are elaborated next, with reference to Fig.4, keeping the static design of rate and stroking cylinder same as in the previous work by Mondal et al. [14].

Fig. 4 Cylinders position against swash plate

3.2 Modeling of Swash Plate and Associated Motion Dynamics

When the delivery pressure goes beyond the cut in pressure, the displacement of the spool in the valve sets in and a coupled dynamics of motion takes place amongst the stroking piston, rate piston, barrel pistons and the swash-plate with mass moment of inertia I_{sw} . Figure 4 represents such an arrangement of the swash plate in which one barrel piston (i^{th} barrel) is shown representatively. The equation of the swivelling dynamics to find out the instantaneous swash angle λ , can be derived with reference to this figure and is expressed as

$$(I_{sw} + m_s l_{st}^2 + m_r l_{rt}^2) \ddot{\lambda} = F_{sp} l_{st} - F_{rp} l_{rt} + \sum_{i=1}^9 [(F_{pi} - m_p \ddot{x}_{pi}) \{R_p \cos(\omega t + 2\pi(i-1)/9)\}] , \quad (10a)$$

where l_{st} and l_{rt} are the distances from the barrel axis to stroking cylinder and rate cylinder respectively. R_p is the pitch circle radius of the barrel and ω is the angular speed of the barrel. The initial value of the swash angle is considered as λ_{ci} .

The angular motion of the swash plate and the linear motions of the stroking and rate pistons are correlated through angular acceleration of the swash plate and the respective radii of curvatures l_{st} and l_{rt} , as

$$\ddot{x}_s = l_{st} \ddot{\lambda} , \quad (10b)$$

$$\text{and } \ddot{x}_r = l_{rt} \ddot{\lambda} . \quad (10c)$$

The acceleration $\ddot{x}_{pi}$ of the i^{th} barrel piston can be obtained from the instantaneous position x_{pi} of the piston, which is expressed in terms of the variable swash angle λ and the speed of barrel rotation ω as

$$x_{pi} = R_p [1 - \cos\{\omega t + 2\pi(i-1)/9\}] \tan \lambda . \quad (10d)$$

In Eq. (10a), F_{sp} , F_{rp} and F_{pi} are the combined forces other than inertia acting respectively on the stroking, rate and i^{th} barrel piston. In general, this force, F_{net} is composed of pressure force, f_{hyd} , viscous friction force, f_{fric} and spring force, f_{spr} if any, so as to write

$$F_{net} = f_{hyd} + f_{fric} + f_{spr} . \quad (11)$$

The force components of the above equation for the respective pistons are furnished, with reference to Fig. 4, in Table 2. The sense of any force is considered positive for the direction left to right. All the notations used in Table 2 are shown in Fig.4, which are self-explanatory.

Table 2 Composition of forces for different pistons

Piston	F_{net}	f_{hyd}	f_{fric}	f_{spr}
Stroking	F_{sp}	$(P_{sc} - P_r)(\pi d_{sp}^2 / 4)$	$-\pi \mu d_{sp}(l_s - l_{s0} - x_s)\dot{x}_s / c_{sc}$	No spring
Rate	F_{rp}	$(P_{rc} - P_r)(\pi d_{rp}^2 / 4)$	$\pi \mu d_{rp}(l_{r0} + x_r)\dot{x}_r / c_{rc}$	$k_r(\delta_0 + x_r)$
i^{th} Barrel	F_{pi}	$(P_{pi} - P_r)(\pi d_p^2 / 4)$	$-\pi \mu d_p \dot{x}_{pi}(l_{pi} - l_{p0i} - x_{pi}) / c_{bc}$	No spring

3.3 Modeling of Pressure Dynamics inside the Cylinders

The stroking, rate and barrel are the three types of cylinders in the pump that have their pistons reciprocating inside, thereby causing volumetric changes. Due to this motion, the respective pressure inside the cylinder chamber will have a dynamics that can be modelled considering compressibility of the oil. From the definition of bulk modulus, the rate of change of pressure inside a changing volume V , which is having an inlet flow Q_{in} and leakage flow from the volume Q_{leak} , can be expressed as,

$$\dot{P}_{cyl} = \beta(Q_{in} - Q_{leak} - \dot{V}) / V . \quad (12)$$

in which Q_{leak} is composed of the flow through the clearance between piston and cylinder. Hydraulic oil leakage has significant effect on the pump efficiency and the leakage depends on the pressure [20]. Only in case of the barrel there will be an additional leakage path for the supply of oil to the interface of piston shoe and slipper pad. Considering fully developed laminar flow through these passages, the respective leakage flow rates given in Table 3 can be expressed in general from [21, 22] as

$$Q_{isc} = (\pi d_{sp} c_{sc}^3)(P_{sc} - P_r) / \{12 \mu (l_s - l_{s0} - x_s)\} , \quad (13a)$$

$$Q_{lrc} = (\pi d_{rp} c_{rc}^3)(P_{rc} - P_r) / \{12 \mu (l_{r0} + x_s l_{rt} / l_{st})\} , \quad (13b)$$

$$Q_{lki} = (\pi d_p c_{bc}^3)(P_{pi} - P_r) / \{12 \mu (l_{pi} - l_{p0i} - x_{pi})\} , \quad (13c)$$

$$\text{and } Q_{lci} = (\pi d_d^4)(P_{pi} - P_r) / (128 \mu l_b) . \quad (13d)$$

The pressure terms other than the case pressure involved in Eq. (11) and Table 2 can be obtained by solving Eqs. (12) and (13), where the respective terms are elaborated in Table 3 with reference to Fig. 4.

Table 3 Cylinder wise terms involved in the respective pressure dynamics

Cylinder	P_{cyl}	Q_{in}	Q_{leak}	$\dot{V}$	V
Stroking	P_{sc}	Q_{sc}	Q_{lsc}	$(\pi/4)d_{sp}^2\dot{x}_s$	$(\pi d_{sp}^2/4)(l_s + x_s)$
Rate	P_{rc}	Q_{rc}	Q_{lrc}	$-(\pi/4)d_{rp}^2\dot{x}_r$	$(\pi d_{rp}^2/4)(l_r + l_{rl} \tan \lambda_{ci} - x_r)$
i^{th} Barrel	P_{pi}	$Q_{si} - Q_{di}$	$Q_{lci} + Q_{lki}$	$-(\pi/4)d_p^2\dot{x}_{pi}$	$(\pi d_p^2/4)(l_{pi} - l_{p0i} + x_{pi})$

In Table 3, the inlet flow Q_{sc} has already been defined in Eq. (9a), whereas Q_{rc} is the flow coming from the delivery line of the pump to the rate cylinder through an orifice of area A_0 given by

$$Q_{rc} = c_d A_0 \sqrt{2(P_{rc} - P_r)/\rho} \operatorname{sgn}[P_{rc} - P_r]. \quad (14)$$

Now Q_{di} and Q_{si} are the delivery and suction flow rate of i^{th} pump barrel. The engagement of i^{th} barrel cylinder during its reciprocation with the respective manifolds, results in delivery and suction flow rates given respectively as

$$Q_{di} = C_d A_{di} \sqrt{2|P_{pi} - P_d|/\rho} \operatorname{sgn}[P_{pi} - P_d]. \quad (15)$$

$$\text{and } Q_{si} = C_d A_{si} \sqrt{2|P_s - P_{pi}|/\rho} \operatorname{sgn}[P_s - P_{pi}]. \quad (16)$$

where the respective engaged areas A_{di} and A_{si} are at delivery and suction sides. The variation of these areas with barrel rotation are obtained following the methodology used in Mandal et al. [16, 23].

3.4 Modeling of Pressure and Flow External to the Pump

The dynamics of pressures P_d and P_s in the respective delivery and suction manifolds can be obtained by considering compressibility effect in the volume V_{pd} between delivery port and load orifice upstream and the volume V_{ps} of reservoir and suction port together. Assuming no leakage outside the pump, it is considered that the flow coming out from the load orifice Q_L re-enters the pump through the suction manifold. Hence, with reference to Fig. 1, the pressure rates can be expressed as

$$\dot{P}_d = \beta [\sum_{i=1}^n Q_{di} - Q_L - Q_{rc} - Q_{ol}] / V_{pd} \quad (17)$$

$$\text{and } \dot{P}_s = \beta [Q_L + Q_{lu} + Q_{or} - \sum_{i=1}^n Q_{si}] / V_{ps} \quad (18)$$

Now, the flow through the load orifice of area A_L can be expressed as

$$Q_L = C_d A_L \sqrt{2(P_d - P_r)/\rho} \quad (19)$$

To evaluate the dynamic performance of the pump the additional parameters used in simulation are furnished in Table 4.

4 PARAMETER SIZING THROUGH SIMULATION

Axial piston pumps are beneficial in terms of compactness and reliability [24]. There are various parameters that have significant influence in pump performance. The dynamic model of the compensator system and the pump described in the earlier section through Eqs. (3) to (19) have been coded as a Matlab/Simulink (version 8.0) model. The model is being run using ODE4 (Runge-Kutta) solver with fixed time step of 2.5×10^{-7} s. Apart

from using the parameter values listed in Table 1 and 4, the input required to run the simulation are the speed of barrel rotation ω and the loading condition of the pump represented by the area A_L of the load orifice in Eq. (19). Unless otherwise mentioned, for all the simulation and experiment exercises, the load orifice area is set initially at a value that corresponds to the near cut-in delivery pressure of 19 MPa. This area is retained for up to 30ms after which for the next 40 ms, the area is decreased linearly towards full closure and maintaining it for 40 ms targeting the cut-off condition. After that the area is increased in a same linear manner such that the pump will return back to its cut-in condition. Figure 5 shows the typical variation of the load orifice area, which has been input to the simulation model. The speed of barrel rotation is set at 1500 rpm, same as measured in the experimental setup. In this section the design analysis of single stage spool has been carried out and the pressure dynamics of the pump has been validated with the simulation performance in terms of pressure of the pump which has a compensator with double spool and the experimental pressure dynamics of the pump which also has double stage spool valve.

Table 4 Additional parameters required for dynamic simulation

Parameter	Value	Parameter	Value
A_0	$1.767 \times 10^{-6} \text{ m}^2$	l_{r0}	$10 \times 10^{-3} \text{ m}$
A_L	$8.66 \times 10^{-5} \text{ m}^2$	l_s	$48 \times 10^{-3} \text{ m}$
C_{bc}	$11 \text{ } \mu\text{m}$	l_{s0}	$5 \times 10^{-3} \text{ m}$
C_{rc}	$70 \text{ } \mu\text{m}$	l_{sr}	$41.5 \times 10^{-3} \text{ m}$
C_{sc}	$39 \text{ } \mu\text{m}$	l_{st}, l_{rt}	$50 \times 10^{-3} \text{ m}$
d_{rp}	$9 \times 10^{-3} \text{ m}$	m_p	0.0306 kg
d_{sp}	$12.85 \times 10^{-3} \text{ m}$	m_r	0.052 kg
I_s	$985.992 \times 10^{-9} \text{ kg-m}^2$	m_s	0.041 kg
l_b	$15 \times 10^{-3} \text{ m}^2$	m_{sv}	0.02 kg
l_d, l_r	$15 \times 10^{-3} \text{ m}$	o_d, u_r	$10 \text{ } \mu\text{m}, 90 \text{ } \mu\text{m}$
l_{pi}	$65.5 \times 10^{-3} \text{ m}$	R_p	$35 \times 10^{-3} \text{ m}$
l_{p0i}	$10 \times 10^{-3} \text{ m}$	V_{dp}	$1.44 \times 10^{-3} \text{ m}^3$
l_r	$35 \times 10^{-3} \text{ m}$	V_{ps}	0.714 m^3

Fig. 5 Loading pattern to the axial piston pump by changing orifice area

4.1 Effect of Considering Compressibility Model at Delivery Chamber of Spool Valve

Before the parametric study, the need of considering separate pressure node P_d^* at the delivery chamber of the spool valve has been studied. Such consideration calls for inclusion of Eqs. (7a) and (7b) in the simulation and termed as compressible model while if the delivery chamber of the valve is considered to remain same as delivery pressure P_d , of the pump, the same is termed as incompressible. In such case, Eqs. (7a) and (7b) need to be replaced by Eq. (7c) in the simulation. The study has been done in terms of observing the predictions of delivery pressure, spool displacement and pressure inside the stroking cylinder for the loading pattern mentioned above. Figure 6 shows the comparison between the two modeling approaches. It can be seen that there are hardly any differences in the predictions following the two different methods. However, in order to evaluate the size of the entry orifice, depicted in the next section, the compressible model has been used.

Fig. 6 Effect of compressible model in the extreme left chamber of the spool valve

4.2 Evaluation of the Size of Entry Orifice to the Spool Valve

Figure 7 depicts the performance of the pump for different delivery orifice at the spool valve. This orifice acts as a bridge element between the pump and the spool valve. The delivery line of the pump is connected to the spool valve through this orifice. The diameter of the orifice has been varied from zero to 1 mm and the performance of the pump has been predicted in terms of delivery pressure, swash angle and flow rate characteristics of the pump for the same loading pattern shown in Fig. 5. Of course in Fig.7 zero size orifice is indicated as 'no orifice' in which the left spool chamber is directly connected with the pump. It is clear from the figure that there is very little difference in terms of the performance of the pump with or without having an orifice at the entry of the spool valve. Hence, for the spool valve, no entry orifice has been suggested and accordingly for the rest of the study, incompressible model has been used at the delivery chamber of the spool valve for it takes less simulation runtime.

Fig. 7 Effect of entry orifice the spool valve

4.3 Evaluation of the Size of Drain Side Orifice of the Spool Valve

A similar study has been conducted by varying the drain orifice diameter from 1mm to 3mm in which Eqs. (8b) and (8c) have been invoked, while for no drain orifice case, Eq. (8c) has been used. Figure 8 illustrates the comparison of the performances in terms of delivery pressure, swash angle and delivery flow rate characteristics for load variation depicted in Fig. 5. The study reveals that direct drain connection between the spool end chamber and reservoir is the best option. Introduction of orifice creates back pressure due to which during increase in loading on the pump more amount of force was required to move the spool against the spring force. As a result the delivery pressure shoots up beyond the designed cut-off pressure, particularly for orifices of 1mm and 2mm sizes. For these orifices, the internal dynamics of the pump take longer time to stabilize the swash plate for near zero delivery flow rate due to full closure of the load orifice. Consequently, after withdrawal of the loading, it takes more time for the flow and

delivery pressure to recover to the cut-in setting. Increasing the diameter of the orifice (3mm) close to the no orifice situation shows better performance of the pump. Therefore, it is suggested that no orifice may be used at the drain side of the valve. In absence of orifice, for the rest of the study, Eq. (8c) has been used as a model of drain flow rate.

Fig. 8 Effect of drain side orifice of the spool valve

4.4 Effect of Over-Lap and Under-Lap of the Spool Valve on the Pump Performance

The lap condition of the central spool land has a great influence on the pump dynamics. In the studies conducted in the previous sections, the central spool land was considered to have overlap o_d of 10 μm at the left side and underlap u_r of 90 μm at the right side of it with respect to the metered port shown in Fig. 2. Now a set of lap combinations have

Fig. 9 Effect of over-lap and under-lap of the spool valve

been studied without altering the main configuration of the spool valve. In fact the study has been extended for reverse lap condition in which the central spool land is under

lapped at left side and overlapped at right side. Figure 9 shows the result of the study, where both are denoted with negative signs. All the performance predictions have been made against same area pattern of load orifice shown in Fig. 5. It can be observed that within a range of the lap conditions of the spool valve, the performance of the pump remains satisfactory. The study also reveals that reversed lap condition in the order of 25 μ m underlap and 105 μ m overlap may be admissible. Up to the lap condition in the order of 100 μ m overlap and 180 μ m underlap, the pump operates within the designed cut-in and cut-off condition. The degradation in the performance for larger reversed lap is due to the fact that at the beginning of the pump operation, the stroking cylinder is directly connected to the pump delivery line, which affects the cut-in pressure of the pump. On the other hand, for the larger overlap at the left side of the spool, the cut-off pressure is increased because more pressure force is required to overcome the overlap length. The corresponding dynamic characteristics of spool valve displacement, swash angle and flow rate of the pump are also shown in Fig. 9(c) and 9(d).

4.5 Evaluation of the Size of Clearance Orifice

As shown in Fig. 2, the additional orifice added between the spool valve and stroking cylinder plays an important role particularly during transition from cut-off to cut-in operating condition. It has a small influence on the cut-off pressure setting, but it is really important when the load valve comes back from its cut-off area setting to cut-in area setting and stays steadily within the cut-in area zone. This orifice acts as a bypass route of the pressurized fluid of stroking cylinder when the load of the pump is being reduced from cut-off to cut-in condition and even below that. The pre-compression of the spring of the spool valve causes closure of the metered port once the load is reduced. For depressurization of the stroking cylinder, the entrapped oil needs to be leaked to the case drain. The radial clearance and the underlap of the spool land may not be sufficient to provide the leakage necessary for the depressurization so that the swash plate can able to come back to its rated orientation. This is evident from Fig. 10, where the performance of the pump has been studied for different sizes of the clearance orifice starting from zero size orifice, equivalent to no orifice. The same loading pattern as shown in Fig. 5 has also been used for this study and apart from varying the diameter of the clearance orifice; all other parameters are kept at their nominal values. It can be seen that for a diameter of 750 μ m, the recovery of cut-in pressure from cut-off pressure occurs in the pump in a smooth manner. The behaviours of swash angle and the delivery flow rate also have been found to satisfactory. Below this size of orifice, the return of the swash angle to its rated value gets delayed as can be observed from the figure thereby the delivery pressure and flow rate of the pump get affected. Orifice size of 1mm has resulted higher pressure ripple due to oscillations in the swash dynamics. Therefore, from this study, the size of the clearance orifice has been recommended as 750 μ m. The sizing of the spool valve obtained from dynamic simulation has been furnished in Table 5.

Fig. 10 Effect of clearance orifice of the spool valve

5 PERFORMANCE ANALYSIS BETWEEN DOUBLE SPOOL COMPENSATOR WITH SINGLE SPOOL COMPENSATOR

Figure 11 presents the comparison of the dynamic performance of the present pressure compensator with another one having a two stage spool valve developed in an earlier study [14]. In both the cases apart from the spool valves all other components of the pump remain same. The pump dynamics has been studied for the same load dynamics as shown in Fig. 5. It can be observed from Fig.11(a) that a pressure overshoot appears when the orifice goes from cut-in area to cut-off area in both types of pressure compensators. However peak overshoot is more in the case of the compensator with two stage spool valve. The primary reason behind this feature may be attributed to the

combined effect of the two springs associated with the spool ends, although both the designs of pressure compensator maintain the same cut-in pressure but the cut-off pressure is high in case of compensator with double spool. Figure 11(b) illustrates that the pressure ripples in the rate cylinder at the valve transient region and the full load region is slightly more in case of single spool compensator design. The stroking cylinder pressure dynamics has been shown in the Fig.11(c). Figures 11(d) and 11(e) demonstrate the features of swash plate dynamics and the flow dynamics of the system. At the cut-off region, the flow is nearly zero for both the cases, but the swash angle is less in case of compensator with two stage spool valve than compensator with single stage spool valve.

Fig. 11 Performance comparisons between two compensators

Table 5 Parameters obtained from dynamic simulation

Parameter	Value	Parameter	Value
d_{od}	Not required	u_d	105 μm to 180 μm
d_{or}	Not required	d_c	0.75 m
O_d	-25 μm to 100 μm		

6 PERFORMANCE COMPARISON WITH A REAL PUMP

The objective of this section is to verify whether the simulated performance of the axial piston pump having the designed single spool compensator matches with a commercially available similar pump but having double spool compensator. In the earlier study made by Mondal et al. [14], the same experimental test setup had been used to verify the correctness of the simulation software and the design methodology. In Fig.12(a) the complete circuit has been shown with the symbolic representation of the components and

sensors. A standard petroleum based hydraulic oil (HLP68) is used as working fluid. The pump has been driven by a three phase induction motor controlled by a variable frequency drive (VFD) as per requirement of the speed. The variable displacement swash plate type axial piston pump and two cylinders, rate and stroking are housed in a single casing. The spool valve is fitted on the pump casing. Some components like filter and proportional valve are connected in series, while the pressure relief valve and pressure transducer are connected in parallel to the circuit. A proportional solenoid operated direction as well as flow control valve PV with an integrated control electronics ICE module mounted on it, has been used as the load orifice. The system is operated via an algorithm developed on the LabVIEW (version 2013) environment installed on the HOST PC. The algorithm generates a continuous voltage which is fed to the PV via the output module OM (NI 9263) of the NI-cRIO 9081 real time system RTS. The pump pressure is measured corresponding to this voltage feed via the pressure transducer PT. The analog signals from the PT are relayed back to the HOST PC for purposes of storage and analysis via the input module IM of the RTS. A photographic view of the setup with different components has been shown in Fig. 12(b), while the details of the hydraulic and electrical components have been given in Table 6.

The pressure compensator model which consists of a single stage spool valve has been verified with the experimental result and simulation of system with two stage spool valve subjected to same loading pattern. The loading has been obtained by varying the command signal to the PV as shown in Fig. 13(a). Figure 13(b) shows the performance comparison result in terms of the delivery pressure characteristics against the loading pattern. The spring pre-compression setting of the single spool valve has been adjusted to 0.0072 m to have a satisfactory matching with the experimental one. In case of compensator with double spool, the spring pre-compression setting of the bottom spool is kept fixed at 0.00415m, while that for the top spool is set at 0.0086m to match with the experimental result as shown in Fig.13(a). It may be mentioned that although the compensator design with single spool valve has been performed for 20 MPa cut-off pressure, it has been set as 18.5 MPa for the verification study to maintain consistency with a factory set cut-off value of the real pump.

Fig. 12 Experimental circuit diagram

Table 6 Details of the set-up components

Component (make)	Specifications
Motor, M (ABB)	3-phase; 37 KW; 2945 rpm;
Variable Frequency Drive, VFD (ABB)	30 kW
Pump, P (Rexroth)	A10VSO 45DR/31;
Pressure Transducer, PT (Kistler)	0-200 bar; Temp:-20...+85°C
Filter, F (Stauff)	Maximum Working pressure 350 bar
Proportional valve, PV (Rexroth)	4WREE series; 50 lpm at 10 bar drop
Hose Pipe (Parker)	250 bar; 16mm(5/8" BSP)
Pressure Relief valve, PRV (Yuken)	EBG-03-H-11; 210bar
Real-time system, RTS (NI systems)	1GHz, 32-bit NI-cRIO 9081
Input module, IM (NI systems)	±10 V/A/D NI-cRIO 9215
Output module, OM (NI systems)	±10V D/A NI-cRIO 9263

Fig. 13 Experimental and simulation validation for a load variation

7 Conclusion

In this present work, a simplified spool valve instead of a two stage spool valve has been incorporated in the pressure compensator of a variable displacement axial piston pump in order to perform a comparative study with a commercial pump. The design sensitivity of the spool valve has been carried out through simulation to identify the critical size of the parameters, which affects the pump performance. The study suggests no orifice at the entry of the spool valve and direct drain connection between the spool end chamber and reservoir. In particular, it has been found that introduction of drain orifice creates back pressure due to which during increase in loading on the pump more amount of force was required to move the spool against the spring force. It has been observed from the sensitivity analysis that within a range of lap conditions of the spool valve, the performance of the pump remains satisfactory. In fact the study has revealed that reversed lap condition in the order of $25\mu\text{m}$ underlap and $105\mu\text{m}$ overlap may be admissible. The role of clearance orifice has been revealed to act as a bypass route of the pressurized fluid of stroking cylinder when the load of the pump is being reduced from cut-off to cut-in condition and even below that. The size of the clearance orifice has been recommended as $750\mu\text{m}$. From the theoretical performance comparison between single stage and two stage spool valve, it has been found that although both the designs of pressure compensator maintain the same cut-in pressure, but the cut-off pressure is high in case of compensator with double spool. By systematic design, it is possible to have a single stage spool valve controlled pressure compensator that can produce performance of the variable displacement axial piston pump at par with a similar commercially available pump.

Declarations

Availability of data and materials

Not applicable.

Competing interests

The authors declare no competing financial interests.

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Authors' contributions

NM and RS contributed the theoretical, experimental work, including the built of experimental set-up and wrote the paper. DS contributed by revising the paper and took the charge of whole arrangement. All authors read and approved the final manuscript.

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Reference

- [1] Norhirni MZ, Hamdi M, Nurmaya Musa S, Saw LH, Mardi NA and Hilman N (2011) Load and stress analysis for the swash plate of an axial piston pump/motor. *Journal of dynamic systems, measurement, and control* 133(6).
- [2] Richardson D, Sadeghi F, Rateick Jr RG and Rowan S (2017) Experimental and analytical investigation of floating valve plate motion in an axial piston pump. *Tribology Transactions* 60(3): pp.537-547.
- [3] Daher N and Ivantysynova M (2014) Energy analysis of an original steering technology that saves fuel and boosts efficiency. *Energy Conversion and Management* 86: pp.1059-1068.
- [4] Ye, S. G., Zhang, J. H., & Xu, B. (2018) Noise reduction of an axial piston pump by valve plate optimization. *Chinese Journal of Mechanical Engineering*, 31(1), 1-16.
- [5] Wang S (2013) Novel Piston Pressure Carryover for Dynamic Analysis and Designs of the Axial Piston Pump. *ASME Journal of Dynamic Systems, Measurement, and Control* 135(6): 024504-1-7.
- [6] Zhao, B., Guo, W., & Quan, L. (2020) Cavitation of a Submerged Jet at the Spherical Valve Plate/Cylinder Block Interface for Axial Piston Pump. *Chinese Journal of Mechanical Engineering*, 33(1), 1-15.
- [7] Wang S (2012) Improving the volumetric efficiency of the axial piston pump. *ASME Journal of Mechanical Design* 134(11): 111001-111001-7.

- [8] Zheng N, Tong B, Yang W and Zhang G (2020) Study on pressure balance and clearance flow characteristics of composite-grooved piston–copper pair. *Journal of the Brazilian Society of Mechanical Sciences and Engineering* 42: pp.1-11.
- [9] Kemmetmüller W, Fuchshumer F and Kugi A (2010) Nonlinear pressure control of self-supplied variable displacement axial piston pumps. *Control Engineering Practice* 18(1): pp.84-93.
- [10] Wei J, Guo K, Fang J, and Tian Q (2015) Nonlinear supply pressure control for a variable displacement axial piston pump. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering* 229(7): pp.614-624.
- [11] Kaliafetis P and Costopoulos T (1995) Modelling and simulation of an axial piston variable displacement pump with pressure control. *Mechanism and Machine Theory* 30(4): pp.599-612.
- [12] Zhou, J., Zhou, J., & Jing, C. (2020). Experimental Research on the Dynamic Lubricating Performance of Slipper/Swash Plate Interface in Axial Piston Pumps. *Chinese Journal of Mechanical Engineering*, 33(1), 1-7.
- [13] Manring ND, Mehta VS, Nelson BE, Graf KJ and Kuehn JL (2013) Increasing the power density for axial-piston swash-plate type hydrostatic machines. *ASME Journal of Mechanical Design* 135(7): 071002-071002-6.
- [14] Mondal N, Saha R, Mookherjee S and Sanyal D (2019) A novel method to design pressure compensator for variable displacement axial piston pump. *Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering* 233(2): pp.314-34.
- [15] Maiti R and Narayan P (2011) An experimental investigation on swash plate control torque of a pressure compensated variable displacement inline piston pump. In *Proceedings of the 8th JFPS (Japan Fluid Power System Society) International Symposium on Fluid Power*.
- [16] Mandal NP, Saha R, Mookherjee S and Sanyal D (2014) Pressure compensator design for a swash plate axial piston pump. *ASME Journal of Dynamic Systems, Measurement, and Control* 136(2): 021001-021001-12.
- [17] Rexroth Bosch Group, – Industrial Hydraulics Catalogue. Variable Axial Piston Pump, Type - A10VSO, series–31/DR, RA92711, <http://www.pneumax.co.th/catalogue/hyd/rexroth/pump/A10VSO.pdf> /, 2004, accessed Sep20, 2020.
- [18] Bhandari VB (2011) *Design of machine elements*. 2nd ed, Tata McGraw-Hill Education: pp. 395.
- [19] Merritt HE (1967) *Hydraulic control systems*. 1st ed. John Wiley & Sons, Inc., NewYork-London-Sydney.
- [20] Özmen, Ö., Sinanoğlu, C., Caliskan, A., & Badem, H. (2020). Prediction of Leakage from an Axial Piston Pump Slipper with Circular Dimples Using Deep Neural Networks. *Chinese Journal of Mechanical Engineering*, 33, 1-11.
- [21] Ivantysyn J. and Ivantysynova M (2001) *Physical fundamentals*. In: *Hydrostatics pumps and motors*. 1st ed., Akademia Books International, New Delhi, India: pp. 32–33.
- [22] Massey BS and Ward-Smith J (2006) *Mechanics of fluids*. 8th ed., Taylor & Francis, New York: pp.201–205.

- [23] Mandal NP, Saha R and Sanyal D (2012) Effects of flow inertia modelling and valve-plate geometry on swash-plate axial-piston pump performance. Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering: 226(4), pp.451-465.
- [24] Bin, Z., Bing, X., Chunlin, X., & Huayong, Y. (2009). Modeling and Simulation on Axial Piston Pump Based on Virtual Prototype Technology [J]. Chinese Journal of Mechanical Engineering, 22(1), 84-90.

Nomenclature

A_L	load orifice area, m^2
c_d, c_v	discharge and velocity coefficient respectively
c_{rc}, c_{sc}, c_{bc}	radial clearances between rate piston and cylinder, stroking piston and cylinder and barrel piston and cylinder respectively, μm
d_d, d_{rp}, d_{sp}	diameters of the passageway through barrel piston, rate/bias piston and stroking/control piston respectively, m
d_p	barrel piston diameter, m
d_s, d_{sr}	spool land and modified spool land diameters, m
I_s	mass moment of inertia of cradle and its lever, $kg \cdot m^2$
k_r	rate cylinder spring stiffness, N/m
k_s	Stiffness of spool valve spring N/m
l_b	length of the orifice inside the each barrel piston, m
l_{rt}, l_{st}	distances from barrel axis to rate cylinder and stroking cylinder axes, m
m_r, m_s	masses of rate piston and stroking piston respectively, kg
m_p, m_{sv}	masses of barrel piston and spool respectively, kg
n	number of barrel pistons
P_{rc}, P_{sc}	pressures inside the rate cylinder and stroking cylinder respectively, Pa
P_d, P_r, P_s	delivery port, reservoir and suction port pressures respectively, Pa
P_{dci}, P_{dco}	cut-in and cut-off pressures of the pump respectively, Pa
P_{pi}	pressure inside i^{th} piston chamber, Pa
Q_L	flow rate through load orifice, m^3/s
Q_{sc}	flow rate from the spool valve to stroking cylinder, m^3/s
Q_{rc}	flow rate through the feedback orifice to the rate cylinder, m^3/s
Q_{lrc}	leakage flow rate through the radial clearance between the rate piston and cylinder, m^3/s
Q_{lsc}	leakage flow rate through the radial clearance between the stroking piston and cylinder, m^3/s
Q_{lki}	leakage flow rate through the radial clearance between the barrel piston and cylinder, m^3/s
Q_{lci}	leakage flow rate through the clearance between slipper pad and swash plate, m^3/s

R_p	pitch circle radius of piston on barrel, m
V_{pd}, V_{ps}	volumes between delivery port and load valve and between tank and suction port respectively, m ³
x_r, x_s	displacements of rate piston and stroking piston respectively, m
x_{pi}	displacement of i^{th} barrel piston from TDC, m
δ_0	pre-compressions of rate spring, m
δ_{s0}	pre-compressions of spool spring, m
β	bulk modulus of the hydraulic oil, Pa
μ	viscosity of the hydraulic oil, Pa-s
ρ	density of the hydraulic oil, kg/m ³
λ	swash angle, rad
ω	angular speed of barrel, rad/s

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