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Technical Note

Pressure compensator design, simulation and performance evaluation of a variable displacement swash plate type axial piston pump

Nitesh MONDAL[1,*], Rana SAHA[2], Dipankar SANYAL[3]

[1]*Department of Aerospace Engineering & Applied Mechanics, Indian Institute of Engineering Science and Technology, Shibpur, India*
[2]*Department of Mechanical Engineering, Jadavpur University, Kolkata, India*
[3]*Department of Mechanical Engineering, Jadavpur University, Kolkata, India*

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ABSTRACT

This work presents a simple design procedure of a pressure compensator of a swash plate type variable displacement axial piston pump (VDAPP). The route of the work mainly focuses on static design through balancing the torque given by the pump pistons, rate cylinder and stroking cylinder on the swash plate during cut-in (maximum flow) and cut-off (minimum flow) pressure condition of the system with an objective of minimizing the output pressure ripples. The outcome in terms of pressure from the dynamic simulation of the designed compensator with pump has been compare with experimental result obtained from a reference commercial pump has compensator with duel spool. The model has been used for performance prediction for wide variations of the load valve area settings.

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INTRODUCTION

VDAPP are used as a power supply of energy efficient hydraulic systems [1–3]. Some such very common application areas are aerospace, agricultural, automotive, construction, mining, and transport equipment. The activation mechanism of variable displacement feature of such pump is totally dependent on the displacement of a spool valve [1–4]. To provide better damping and reducing the pressure gain an orifice is placed between spool valve and control cylinder [2]. The main focuses of this work is design a compensator on the base of balancing of torque due to the various components on swash plate at static maximum and minimum flow condition with an objective of minimizing the output pressure ripples. Manring [6–7] analyzed the cause of flow ripples and capitation in an axial piston pump. Indeed, in a modern research work [8], it had been exposed

*Corresponding author.
*E-mail address: niteshju@yahoo.com

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that the oscillations of swash plate is responsible for efficiency, flow pulsation and noise. Figure 1 represents the pressure compensator arrangement in such a pump consists of a single stage control spool valve and two actuators acting in opposite manner against the swash plate at the two ends. One of the actuators, recognized as stroking cylinder or control cylinder is connected to the delivery port of the spool valve. The other actuator known as the rate cylinder is usually spring-loaded for conventional design and linked to the delivery manifold of the pump through an orifice. The present work involves design of the pressure compensator for a specified pump, developing the dynamic model of the system and MATLAB/SIMULINK based dynamic simulation. Simulation results for the present design have been compared with a commercial pump (Rexroth-A10VSO).

DESIGN METHODOLOGY

The aim of the static torque balance designed to meet the acceptable performance as Rexroth A10VSO45 series pump at 1500 rpm. Following up the literature [10], the rated flow, torque and power have been premeditated at 1500 rpm.

Estimation of Maximum Swiveling Torque

According to the Figure 2, the force inside each pump piston due to the pressure F_{pi} on the swash plate. All these forces merge to generate a torque to rotate the swash plate about x axis. The figure expose that the instantaneous positions of pump pistons 6 to 9 on the delivery side manifold.

The separation portion of the valve plate is known as bridge. When the pump piston is at bridge at BDC; the piston pressure is the maximum for that moment and the pressure create adverse swiveling torque. During the period of rotation between leaving the manifold by two consecutive pistons, this torque keeps on increasing. To determine the maximum torque these are adopted. All other piston namely 1 to 5 are in the suction side and the pressure on these pistons are suction pressure P_s that considered equivalent to the tank pressure P_r and 6 to 9 are under delivery pressure P_d. Therefore, for the static design of the pressure compensator, the maximum swiveling torque is expressed in terms of the barrel or pump piston diameter d_p and pitch circle radius of barrel R_p as

$$T_s = \sum_{i=1}^{9}\left[\left(P_{pi} - P_r\right)\left(\frac{\pi d_p^2}{4}\right)R_p \cos\left\{\theta + \frac{2\pi(i-1)}{9}\right\}\right] \quad (1)$$

Equation 1 is the most important equation to for this numerical design concept. The function of equation (1) in designing the to actuating cylinders (rate and stroking) for a particular referrence cut-in and cut-off pressures for the pump are described subsequently.

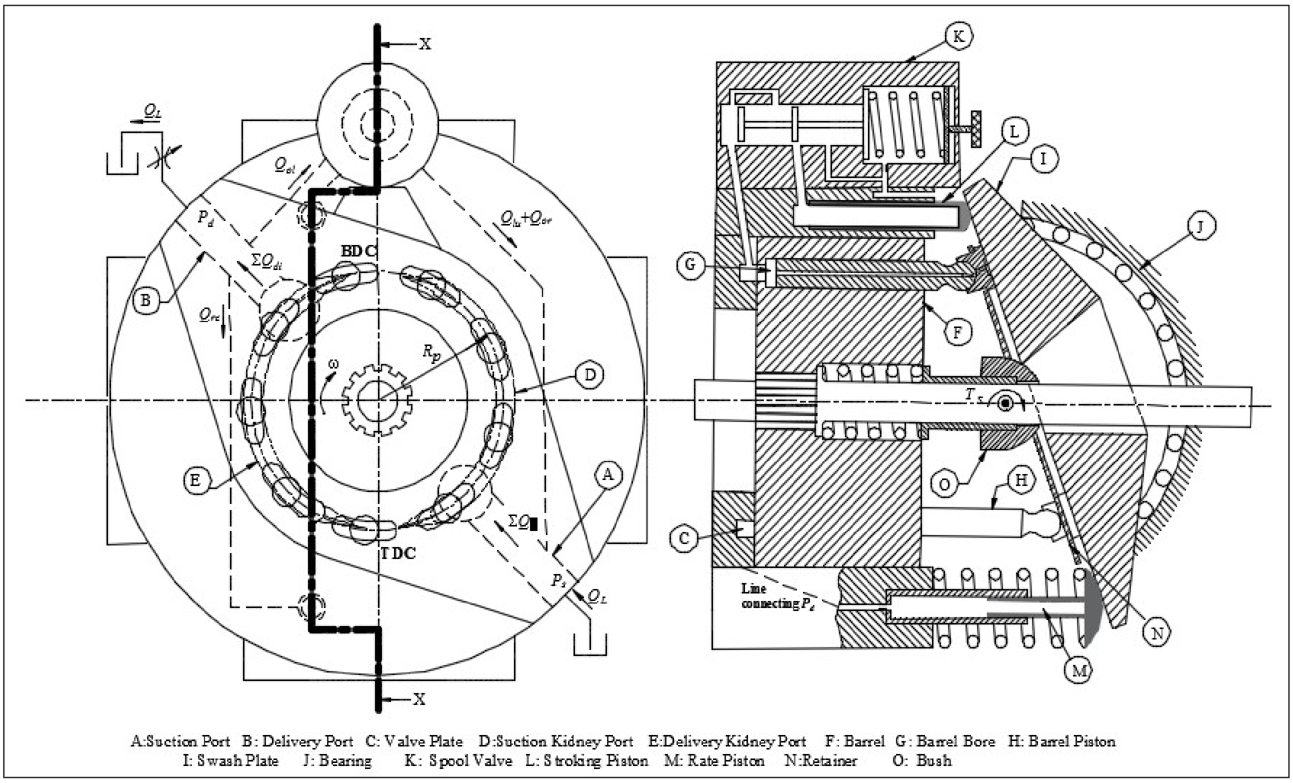

Figure 1. Schematic of a swash-plate axial piston pump with pressure compensator.

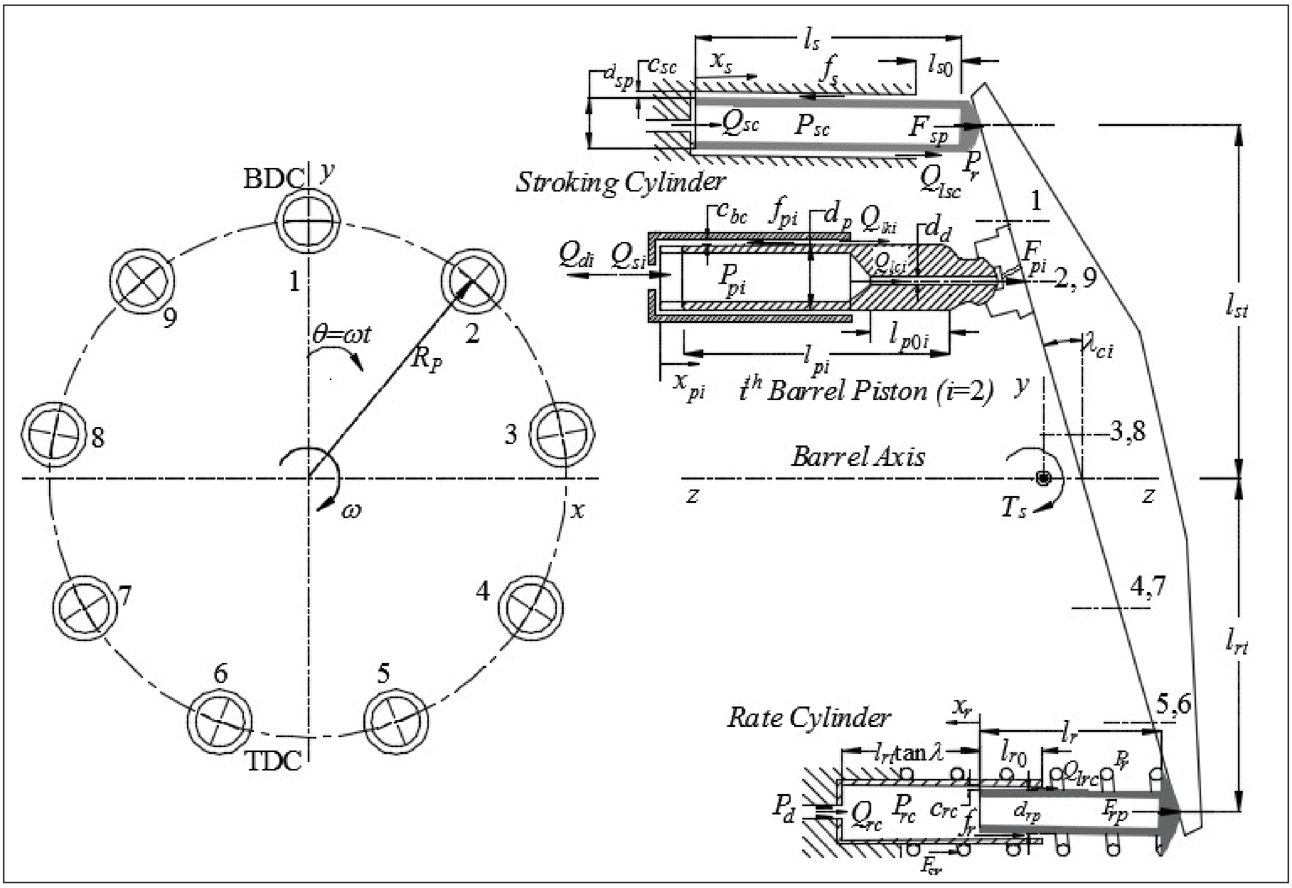

Figure 2. Geometric and dynamic variables with pump piston 1 at BDC and swash plate at cut-in angle.

Design of Rate and Stroking Piston

When the VDAPP operates at a set reference pressure (cut-in delivery) with P_d shown in Fig. 2 becoming equal to P_{dci} the maximum swiveling torque T_{sci} on the swash plate from the barrel pistons should be counterbalanced by the torque due to the pressure force and the force due to the spring attached with the rate cylinder. Considering no pre-compression at the cut-in situation, the torque balance is written as

$$T_{sci} = \left(P_{dci} - \Delta P_o\right)\left(\frac{\pi d_{rp}^2}{4}\right) l_{rt} \tag{2}$$

Where l_{rt} is the distance between the rate piston axis from the barrel axis and the pressure drop $\Delta P_o = P_{dci} - P_{rc}$ across the orifice involving the rate cylinder with the manifold of delivery gives rise to the orifice flow

$$Q_{rc} = C_d \sqrt{\frac{2\Delta P_o}{\rho}} \tag{3}$$

For a choice of ΔP_o and l_{rt} in equation (2), the diameter of rate piston d_{rp} is received. As mentioned earlier, this orifice does not allow the swash plate, rate and stroking piston assembly to change their speed at a rate too high [**5**]. Since the rate piston should remain immobile up to the cut-in reference pressure, this flow would leak through the small radial gap between piston and cylinder. The expression of flow can be

$$Q_{lrc} = \frac{\pi d_{rp} c_{rc}^3 \left(P_{rc} - P_r\right)}{\left\{12\mu\left(l_{ro} + x_r\right)\right\}} \tag{4}$$

where x_r is the displacement of the rate piston, which is zero at cut-in condition. For a known orifice area A_o, initial engaged length l_{ro} of the rate piston inside the cylinder and pressure drop ΔP_o, the radial clearance c_{rc} can be obtained by solving equations (3) and (4).

When the delivery pressure assumes the cut-off value P_{dco}, the corresponding cut-off swiveling torque T_{sco} together with the rate piston torque are opposed by the stroking piston torque to maintain minimum swash angle. Assuming zero swash condition at the cut-off corresponding to rate piston displacement equal to (l_{rt} tan λ_{ci}), λ_{ci} being the cut-in swash plate angle, the torque balance is expressed as

$$\left(P_{dco} - \Delta P_v\right)\left(\frac{\pi d_{sp}^2}{4}\right) l_{st} = -T_{sco} + \left\{\left(P_{dco} - \Delta P_o\right)\left(\frac{\pi d_{rp}^2}{4}\right) + k_r l_{rt} tan\lambda_{ci}\right\} l_{rt} \tag{5}$$

where $\Delta P_v = P_{dco} - P_{sc}$ is the overall pressure drop across the spool valve labeled as K in Figure 1, P_{sc} is the pressure in the stroking cylinder and exploded in Figure 3, l_{st} is the distance between stroking piston axis and barrel axis and k_r is the stiffness of the spring inserted outside the rate cylinder. The spring diameter is preferred rooted in the rate piston diameter. For a chosen material, outer diameter D the stiffness is found from the standard relation

$$k_r = \frac{G_r d_r^4}{8D^3 n_r} \tag{6}$$

where G_r is the modulus of rigidity of the spring material, d_r is the wire diameter and n_r is the number of active turns of the spring. These values are obtained from online commercial spring design software [11]. For a choice of l_{st} and ΔP_v, the diameter d_{sp} of the stroking piston is evaluated from equation (5). At the cut-off condition, the spool becomes fully open at a metering orifice, as shown by the dotted lines in Figure 3. This opening connects the stroking cylinder with the delivery pressure line through for directing part Q_{sc} of the valve flow Q_{sv} towards the stroking cylinder. The other portion, termed as underlap leakage Q_{lu}, enters the other valve chamber through to the underlap u_r at the right side of the metering spool land and exits through the return port. Thus, the flow relation at this metering port can be expressed as

$$Q_{sv} - Q_{lu} = Q_{sc} \tag{7}$$

where invoking the orifice flow model, it can be written with reference to Figure 3 that

$$Q_{sv} = C_d A_d \sqrt{\frac{2\left(P_{dco} - P_{sc}\right)}{\rho}} \tag{8a}$$

$$\text{and } Q_{lu} = C_d A_r \sqrt{\frac{2\left(P_{sc} - P_r\right)}{\rho}} \tag{8b}$$

$$\text{where } A_d = \frac{n_p r_p^2 \left\{\theta_d - \left(\sin\theta_d\right)\right\}}{2} \tag{9a}$$

$$\text{and } A_r = \frac{n_p r_p^2 \left\{\theta_r - \left(\sin\theta_r\right)\right\}}{2} \tag{9b}$$

for n_p numbers of circular port cuts of radius r_p, whereas 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. 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

$$\theta_d = 2\cos^{-1}\left\{1 - \frac{\max\left(x_{sv} - o_d, 0\right)}{r_p}\right\} \tag{10a}$$

$$\text{and } \theta_r = 2\cos^{-1}\left\{1 - \frac{max\left(u_r - x_{sv}, 0\right)}{r_p}\right\} \tag{10b}$$

As long as the system remains below or at cut-in pressure, the valve flow Q_{sv} remains essentially blocked by the overlap design of the spool land. The spool displacement compensates the overlap at the cut-in condition and becomes steady at its maximum at the cut-off condition. At this displacement, the underlap leakage Q_{lu}, reduces to zero as the right-side metered port opening gets closed. Beyond the cut in, the stroking cylinder receives the flow Q_{sc} through the delivery side of the valve port. This flow is partly utilized to push the swash plate towards the cut-off setting and partly leaks to the casing through the radial clearance C_{sc} around the stroking piston. Similar to equation (4), this leakage can be modeled [10] as

$$Q_{lsc} = \frac{\pi d_s c_{sc}^3 \left(P_{sc} - P_r\right)}{\left\{12\mu\left(l_s - x_s\right)\right\}} \tag{11}$$

where l_s is the length of the stroking cylinder and x_s is the stroking piston displacement that is zero at cut-in and maximum (l_{rt} tanλ_{ci}) at the cut-off condition. Equating Q_{lsc} with Q_{sc} along with substitution of equations (8a) and (11) for cut-off condition in equation (7) with Q_{lu} as zero yields

$$C_d A_d \sqrt{\frac{2\left(P_{dco} - P_{sc}\right)}{\rho}} = \frac{\pi d_{sp} c_{sc}^3 \left(P_{sc} - P_r\right)}{\left\{12\mu\left(l_s - l_{st}\, tan\lambda_{ci}\right)\right\}} \tag{12}$$

from which the radial clearance c_{sc} can be determined. Leakages through this clearance and the return-side underlap at the spool port allow depressurization of the stroking cylinder during recovery of the swash angle from near-zero position, as the delivery pressure of the pump decreases from cut-off pressure to the cut in or even below. With reduction of the delivery pressure, the spring force on the spool causes closure and opening respectively at the delivery and return sides of the metered port. Consequently, the stroking cylinder gets disconnected from the delivery and reconnected to the casing. This reconnection aids the swash plate to rotate back towards its maximum angle

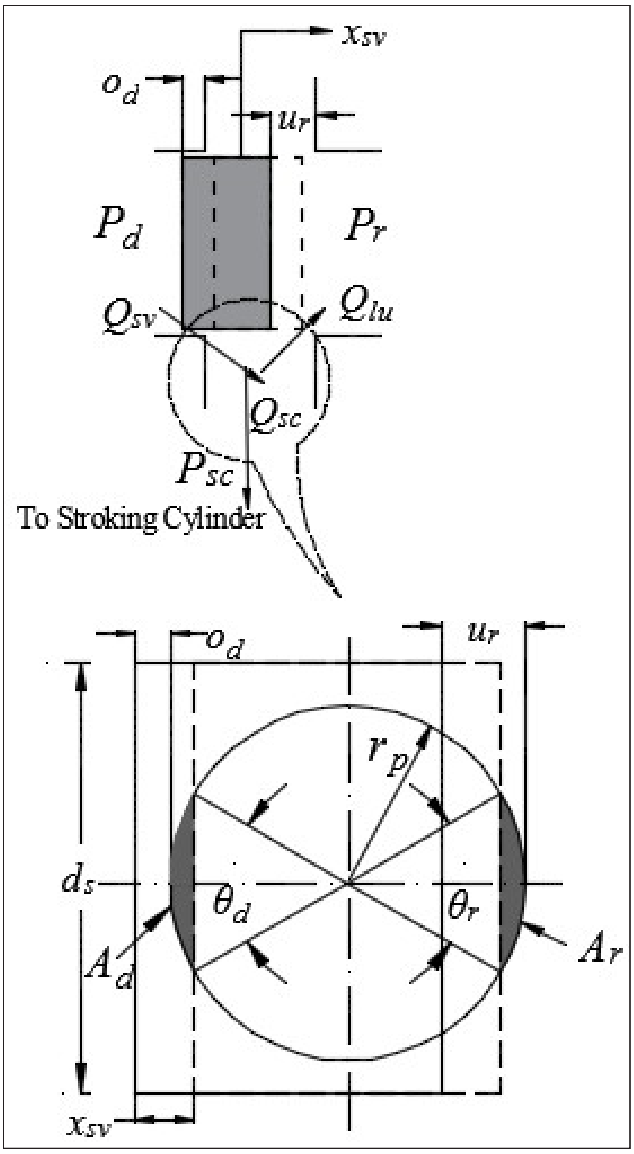

Figure 3. Compensator valve port configuration.

by providing an oil path to the casing through the valve together with that always exists through the radial clearance around the cylinder. Of course, a wider reconnection underlap causes a faster recovery of the swash angle. Another important aspect of the compensator valve design is its spring.

Compensator Spring Design

The design of the spring of the spool valve shown in Figure 3 presumes a small but sensible choice of the spool land diameter d_s. 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 opposite side 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

$$\left(P_{dci} - P_r\right)\left(\frac{\pi d_s^2}{4}\right) = k_s \delta_{s0} \tag{13}$$

For a chosen, the spring stiffness has been estimated. From the above equation by neglecting the force due to case drain pressure P_r. A realistic size of the spring in relations of coil and wire diameter, number of active turns of the coil and free length has been designated through commercial spring design software [11] so as to competition the predictable stiffness. To pile up the accessible spring in the spool housing some modifications of the spool diameter and length of the return chamber is required.

The maximum compensator spool shift, which is also the maximum spring compression, is achieved from the force stability at cut-off state given by

$$\frac{\pi\left(P_{dco} d_s^2 - P_r d_{sr}^2\right)}{4} = k_s\left(\delta_{s0} + x_{sv\,max}\right) \tag{14}$$

where d_{sr} is the modified diameter of the return side spool land. The obtained maximum spring compression has been verified from [11] for safe travel of the spring.

EXPERIMENTAL VALIDATION

The experimental validation has been done on the system cut-off at 14.9 Mpa and this cut-off valve has been established by the adjusting the pre-compression of the spool valve spring. And there also a good match of delivery pressure at full open of the load orifice. The cut-off value of the delivery pressure can be altered by the pre-compression of the spring has been studied theoretically. For a particular pre compression (0.00535m) of the spring of compensator spool is enough to catch the pressure dynamics with the variation of load valve area. The cut-off pressure indicated by B is high due to the increment of spring pre-compression at the full closed condition of load valve, similarly Figure 4D shows that the pressure dynamics for the decrement of pre-compression of the same spring. At the full opening condition, the pressure is same for all pre-compression because there is no role of pressure compensator at this stage, the pump acts as a fixed displacement pump and flow is passed through the load valve only. The magnitude of pressure ripples of the simulation model is same as the experiment. So there is no chance of noise due to the pressure ripples.

PERFORMANCE ANALYSIS OF THE SYSTEM AT DYNAMIC CONDITION

Adapting the dynamics model from [5, 12] the performance of the pump has been studied on

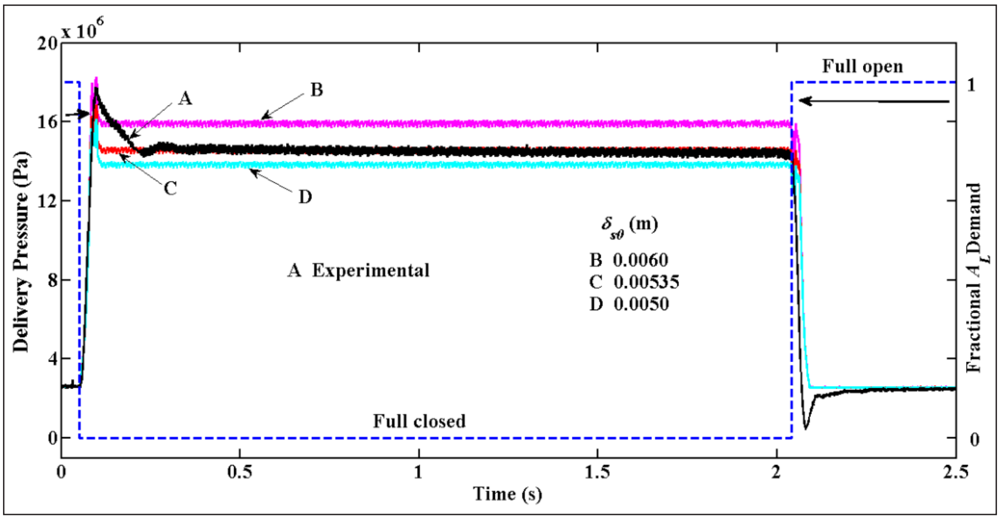

Figure 4. Experimental validation of pressure *dynamics.*

MATLAB-Simulink type 8.0, version R2011a and resolved by using Dormand-Prince ODE solver with fixed or constant time step of 25 X10^{-8}s. The performance of the pump has been studied in terms of delivery pressure, flow rate, dynamics of swash angle and spool displacement for different loading condition of the pumping system. The load area varies as first 30 ms the load area fixed at cut-in pressure then 40 ms the load area decreasing upto 99% of cut-in value further next 30 ms the load area maintain 1% opening of load area which maintain the cut-off reference pressure value. At 100% opening of the load valve-pressure is minimum, flow is maximum, swash angle is maximum and the spool displacement is zero. This is the initial condition of pump dynamics. Now, the variations of pressure to spool displacement have been started due to the load variation from full to 1% opening. The delivery pressure is increasing due to the change of load area towards closing. The flow dynamics totally depends on the swash plate dynamics and the swash plate dynamics depends on the spool valve opening. When the spool valve opening is more than more fluid comes to the stroking cylinder which generates high pressure force on the swash plate as a result the swash plate moves from maximum angle to minimum.

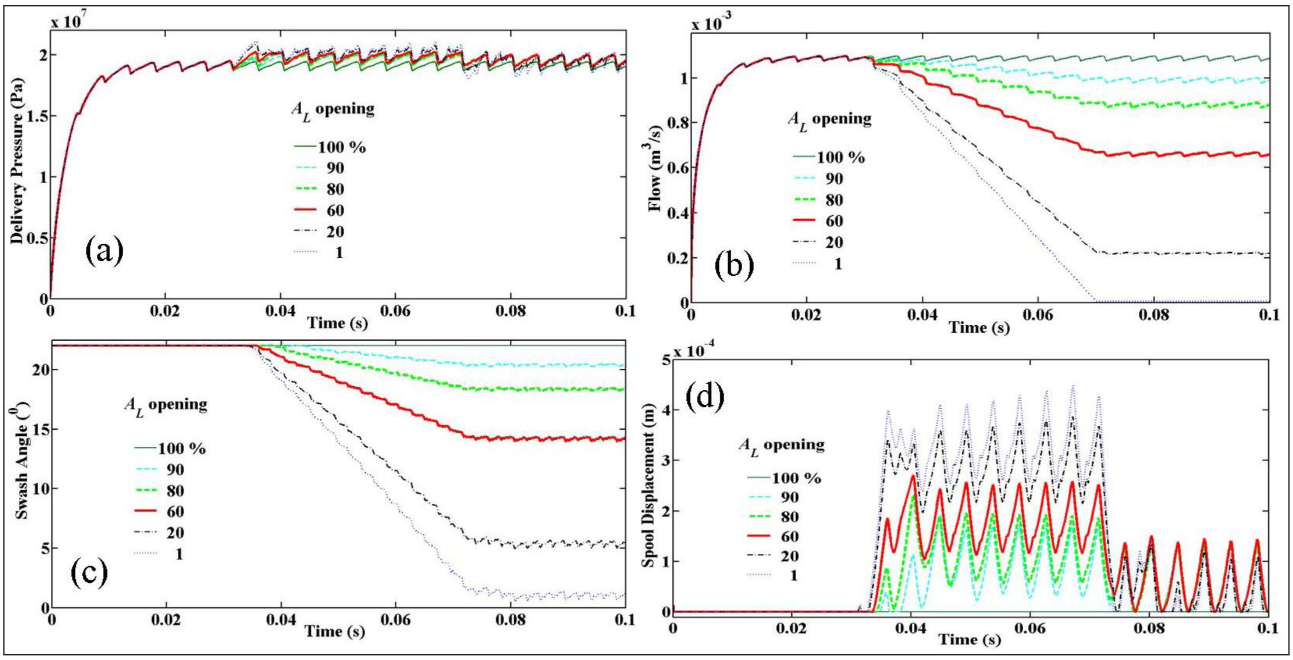

Figure 5. Dynamic performance at different loading condition.

When the pressure of the system maintains its cut-off valve, the swash angle, flow rate and spool displacement becomes minimum at the respective load valve area opening. It is clear from the figure that swash plate has great role on the dynamics of variable displacement pump [13]. The pressure ripple of this system is same for every opening area and the flow ripples is decreasing with the decreasing of load valve opening area. So this balance design has no source of noise due to the ripples.

CONCLUSION

The mathematical model of the Pressure Compensated VDAPP with rate piston pressure feedback at different loaded condition has been developed. The three components namely Spool valve, stroking piston and rate piston have been designed based on the static torque balance on the swash plate at different conditions. The modeldynamics interconnected among spool valve, stroking piston, rate piston and an axial piston pump. It is noticeable that the dynamics performance of the pump is acceptable for different load conditions and dynamic simulation follows the experimental result as well. The propose model can be used to minimise the pressure ripples in the delivery pressure that is the cause of minimization of noise. so, it is clear from the result of different loading condition and experimentally that the model of the rate feedback pressure compensator design is one of the satisfactory design to minimize the pressure ripples which one of the main source of noise.

NOMENCLATURE

A_L	Load orifice area, m^2
P_l	pressure at the right side of spool chamber, Pa
P_{rc}, P_{sc}	Rate piston and Stroking piston pressure respectively, Pa
P_{dci}, P_{dco}	Cut-in and cut-off pressures of the pump, Pa
P_{pi}	Pressure inside i^{th} piston barrel piston chamber, Pa
Q_{sc}	Flow rate from the spool valve to stroking piston, m^3/s
Q_{rc}	Flow rate through the feedback orifice to the rate cylinder, m^3/s
Q_{lrc}	Leakage flow between the rate piston and cylinder, m^3/s
Q_{lsc}	Leakage flow rate between the stroking piston and cylinder, m^3/s
Q_{lki}	Leakage flow rate between the barrel piston and cylinder, m^3/s
Q_{lci}	Leakage flow rate through the clearance between slipper and swash plate, m^3/s
R_p	Pitch circle radius of barrel, m
x_{pi}	Displacement of ith piston from TDC, m
δ_0, δ_{s0}	Pre-compressions of rate spring and spool spring respectively, m
μ,ρ	Viscosity, Pa-s and density, kg/m^3 of oil respectively
ω	Angular speed of barrel, rad/s

AUTHORSHIP CONTRIBUTIONS

Concept: N.M., R.S.; Design: N.M., R.S.; Supervision: D.S.; Materials: N.M., R.S.; Data: N.M., R.S.; Analysis: N.M., R.S.; Literature search: N.M., R.S.; Writing: N.M., R.S.; Critical revision: D.S.

DATA AVAILABILITY STATEMENT

The published publication includes all graphics and data collected or developed during the study.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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