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Effect of Dynamic Swiveling Torque and Eccentricity on the Design of Compensator Cylinders for a Variable Displacement Axial Piston Pump–Modelling & Simulation

Ishita De [a], Subhasish Sarkar [b], Shouvik Chaudhuri [c], Nitesh Mondal[d*], Niraj Kumar[d]

[a] *Department of Mechanical Engineering, Institute of Engineering & Management, Salt Lake-Kolkata, India,700091.*
[b]*Department of Mechanical Engineering, Jadavpur University, Kolkata, India, 700032.*
[c] *Centre for Industrial Mechanics, University of Southern Denmark, Sønderborg, Denmark, 6400.*
[d] *Department of Mechanical Engineering, Ghani Khan Choudhury Institute of Engineering & Technology, Malda, India,732141.*

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Abstract

Axial piston pumps are an integral part of hydraulic systems due to their compactness, simple design and quick response to variable loading conditions when provided with a pressure compensator unit. The work comprises a simple method of designing the rate and stroking cylinders of the pressure compensator on the basis of dynamic swiveling torque on the swash plate at cut-in and cut-off conditions, respectively. A key section of this research is the effect of eccentricity on the pump size and performance. A mathematical model of a fixed displacement pump has been adopted to analyze the components of the swash plate swiveling torque, and to determine the torque values at maximum and minimum flow conditions and design the compensator cylinders. Then, the eccentricity is varied and the performances of the fixed and variable displacement pump are recorded through dynamic simulation modelling. The outcome of the study on eccentricity is the reduction in size of the pressure compensator cylinders and thus, the overall pump size.

Keywords: Dynamic, Swivelling Torque, Eccentricity, Pressure compensator, Pump size.

Nomenclature

ao	Instantaneous linear acceleration of barrel piston,m/s2
cb	Barrel piston-bore clearance, m
cr	Clearance between rate piston and cylinder, m
cs	Clearance between stroking piston and cylinder, m
db,dr,ds	Diameter of barrel piston, rate cylinder and stroking cylinder, respectively, m
dd	Diameter of flow passage to slipper, m
DR ,dR	Mean diameter and coil diameter of the rate spring, respectively, m
e	Eccentricity, m
fo	Instantaneous frictional force on barrel piston due to leakages, N
F1-9	Flow force on respective barrel pistons
Fb	Pressure force by barrel piston, N
Fr,Fs	Pressure force by rate and stroking cylinder, respectively, N
FR	Spring force by rate spring, N
fr,fs	Frictional force on rate piston and stroking piston due to leakages, respectively, N
G	Rigidity modulus of the rate spring material, Pa
k_R	Rate spring stiffness, N/m
l_{bi}	Length of i^{th} barrel piston, m
l_o	Instantaneous moment arm of barrel piston, m
l_r	Rate piston moment arm (distance between the rate cylinder axis and barrel axis), m
l_s	Stroking piston moment arm (distance between the stroking cylinder axis and barrel axis), m
m_b	Mass of barrel piston, kg
N_R	Number of active turns of the rate spring
N	Speed of the pump, rpm
P_b	Instantaneous pressure due to fluid flow, Pa
P_d	Delivery pressure, Pa
P_l	Back pressure at return side chamber, Pa
P_r	Reservoir pressure, Pa
P_i,P_o	Average delivery pressure at cut-in and cut-off condition, respectively, Pa
Q_b	Leakage flow through barrel piston-bore clearance, m^3/s
Q_l	Leakage flow to slipper, m^3/s
Q_{lr},Q_{ls}	Leakage flow through rate and stroking piston-cylinder clearance, respectively, m^3/s
Q_p	Pilot flow to rate piston, m^3/s
Q_s,Q_d	Supply and delivery flow to barrel piston, respectively,m^3/s
Q_v	Flow from spool valve to stroking piston, m^3/s
R_p	Pitch circle radius, m
T_b	Average swiveling torque offered by the barrel pistons, Nm
T_r	Rate piston torque, Nm
T_s	Stroking piston torque, Nm
T_{bi} ,T_{bo}	Maximum swiveling torque at the cut-in and cut-off condition, respectively, Nm
x_{bi}	Displacement of i^{th} barrel piston from TDC, m
δ_R	Rate spring pre-compression, m
λ	Swash angle, degree
θ_o	Instantaneous position of the barrel pistons, degree
Ω	Angular velocity of the pump, rad/s

* Corresponding author e-mail: niteshju@yahoo.com.

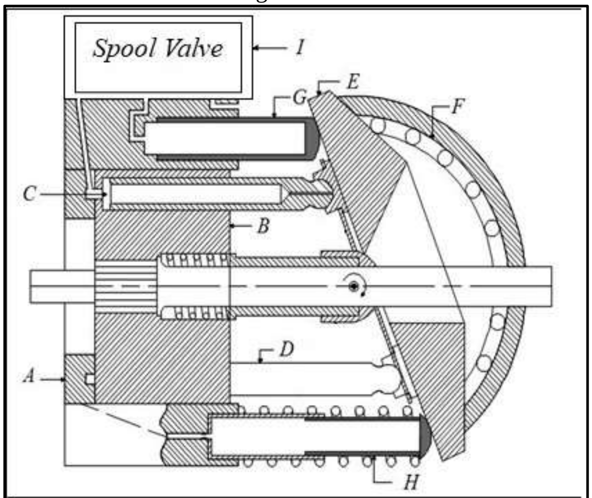
1. Introduction

Swash-plate type axial piston pumps are a type of positive displacement pump that have been used extensively as a part of electro-hydraulic circuits for the purpose of power supply and flow generation. The various fields utilizing these pumps or the pump controlled system include aerospace, air conditioning, agriculture, artillery, construction, earth moving machines, mining, regenerative braking systems [1-4]. The major advantages of these hydraulic units lie in their simple design, quick response to load variations, compactness and ability to work with high pressures with good control [5]. Energy saving is an important and prospective approach to install a pump in an industry [6].The original US patent by Cooper and Hampton [7], was designed to utilize the working fluid's pressure to reciprocate the barrel pistons and thus, improve the energy efficiency. But the discrete numbers of barrel pistons that reciprocate in a sinusoidal manner produce a lot of fluctuations in the delivery pressure and flow, which causes vibrations and generates noise. This tampers with the efficient functioning of the pump. A lot of work has been done to rectify these issues by making design modifications to the pump and to improve its performance. Valve plate designs have been studied and altered to tackle the problems of cavitation and pressure ripples [8-9]. Piston design was also found to affect pump performance [10]. Detailed mathematical models of the pump have been developed for steady state [11] as well as in multi-body simulation environments [12]. Several studies have been carried out to determine flow and pressure dynamics for noise reduction [11,13-15], role of leakage flow in the pump performance [16] and operational stability [14].The problem of control of axial piston pumps has been investigated by Green and Crossley[17] and Yamaguchi and Ishikawa [18]. Gao et al. [19]studied the characteristics of variable displacement pumps and the variable- displacement output was implemented by controlling the swash plate angle with a feedback control unit. The pump can also be made to work under variable loading conditions by integrating it with a pressure compensator unit[20]. The pressure compensator unit provides enhanced efficiency and quicker response than valve-controlled systems [21]. The pressure or flow of the pump is controlled by adjusting the swash plate angle, which was identified as a major geometrical feature by Zeiger and Akers [22]. The stroking volume of the barrel pistons directly depend on the swash plate angle and thus, any changes to the swash angle would lead to changes in the flow output. In the pressure compensator unit, a part of the pump's delivery flow is feedback to the control piston through a spool valve and directly to the rate piston. The purpose of this unit is to alter the swash angle by creating a swiveling torque on the swash plate [8,23]. At a certain set pressure, the swash angle approaches zero, which leads to reduction in the stroking volume of the barrel piston and hence, the output flow becomes negligible. Therefore, knowledge of the dynamics of the swash plate and its optimization is of key importance. Research has been carried out on the same by various authors [22-26]. Also, any oscillations in the swash plate would cause output pressure and flow ripples [23, 26-29]also studied the effect of radial clearance between the piston and cylinder of the pressure compensator elements.The pressure compensator ensures reduced oscillations of the swash plate and retains the efficiency of the pump over a range of varying pressures. Thus, it is essential to design this unit with utmost care. Another important design parameter is the distance between the barrel axis and swash plate pivot, also known as eccentricity. Zeiger and Akers [30]clearly showed the effect of eccentricity on the swiveling torque experienced by the swash plate. An increase in the positive eccentricity would lead to higher torque offered by each barrel piston and thus, higher average torque on the swash plate. Although the value of eccentricity has been included in the swash plate torque analysis in prior studies [30-31] its effect on the size of the fixed displacement pump and performance of the variable displacement pump, has not been investigated to the best of our knowledge.Computerized Fluid Dynamic (CFD) [32-33] is a good option to analyze the dynamics of fluid inside the pump. There are very few literatures which have described the design procedure of pressure compensator components based on the static swiveling torque acting on the swash plate [23] and produce the concept of design sensitivity or role of various components theoretically [34-35] and experimentally [36].

Considering the above facts and prior literature, the primary objective of the present work is to design the pressure compensator cylinders on the basis of dynamic swiveling torque acting on the swash plate at maximum and minimum swash conditions for a variable displacement pump. Also, the effect of eccentricity on the performance of the pressure compensator cylinders and the resulting pump size is analyzed, since no prior study of this sort seems to be present in literature, which in turn, constitutes the novelty of this work. The swash plate axial piston pump with pressure compensator has been described in detail in Section 2. The design methodology for the pressure compensator cylinders and the modeling of the pump with eccentricity considerations is discussed in Section 3. This section explains the swiveling torque components in detail and considers maximum swiveling torque values, at the cut-in and cut-off pressure limits for maximum and minimum swash angles, respectively, in order to design the cylinders. The dynamic simulation and the validation of the model has been discussed at section 4. Section 5 discusses the simulation results of the pump performance for variation of cylinder diameters of the pressure compensator and the effect of eccentricity on the size of the pump and its performance. The pump operation is simulated in MATLAB/Simulink for this study using a previously developed mathematical model of the pump [4, 9, 23]. A brief conclusion to this work is drawn in Section 6.

2. System Description

The features of the pump include a motor-driven splined shaft on which a rotating barrel is mounted. The barrel houses nine pistons in individual bores, separated by equal distances from one another. In Fig. 1, the instantaneous positions of only two pistons are shown for reference. One end of the barrel pistons is linked to a swash plate with the help of slippers and a retainer plate. The swash plate is a simple, flat plate which works like a cam. The swash plate is usually inclined at an angle from the perpendicular to the barrel axis. This angle, termed as swash angle, is one of the key features that determine the discharge of the pump. The flat end of the barrel has a valve plate attached to it by means of hydrodynamic force produced by the working fluid. The valve plate has two kidney-shaped ports called the suction and discharge port through which the working fluid enters and leaves the pump respectively. The two ports are separated by two bridges. A common circle, known as the pitch circle, passes over the two ports. The barrel pistons are arranged such that they rotate along the pitch circle. During normal operation of the fixed displacement pump, the swash plate is set at a fixed angle and power is supplied to the pump. The splined shaft and along with it, the barrel starts to rotate, causing the piston assembly to rotate as well. The reciprocation of the pistons is made possible by the inclination of the swash angle.



A: Valve Plate, *B*: Barrel, *C*: Barrel Bore, *D*: Barrel Piston, *E*: Swash Plate, *F*: Bearing, *G*: Control Piston, *H*: Rate Piston, *I*: Spool Valve

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

A pressure compensator is introduced to make the pump handle variable loading conditions. This unit consists of a rate piston, spool valve and a stroking piston. The rate piston and stroking piston are placed diametrically opposite to one another from the barrel axis. The spool valve is directly linked to the stroking piston by means of a circumferential port. The valve is used to create or block the path of delivery flow to the stroking piston. Rate piston and spool valve are directly connected to the pump delivery line. The spool valve has a spring attached to the other end. For delivery pressures below a certain pressure value set by the spring force of the spool valve (known as cut-in pressure), the spool remains stationary. Thus, the pathway of the delivery flow to the stroking cylinder remains blocked and so, the dynamics of the stroking cylinder remains unaffected for pressure values below the cut-in pressure. However, the delivery flow reaches the rate piston and causes it to remain completely extended. This enables the rate piston to produce enough force to hold the swash plate at its maximum angle. The pre-compression of the rate spring, which coils around the rate piston, determines the maximum swash angle for the particular preset cut-in pressure. The barrel pistons cover maximum swept volume, and thus, maximum discharge of working fluid is obtained at this condition.

Beyond the cut-in condition, the pressure force begins to overcome the spool spring force and slowly displaces the spool valve.The flow of discharged fluid to the stroking piston is initiated and the pressure force actuates the piston. This causes the stroking piston to produce a torque on the swash plate. As this torque increases, it tends to overcome the initial torque produced by the rate piston and so, the swash angle reduces. The reduction leads to a decrease in the swept volume of the barrel pistons, which causes a decrease in the output flow of the pump. As the delivery pressure gradually increases, there comes a point where the stroking piston gets completely extended. This extension produces high enough torque to reduce the swash angle to its minimum value and thus, the net output flow of the pump becomes negligible. The corresponding pressure is termed as cut-off pressure.

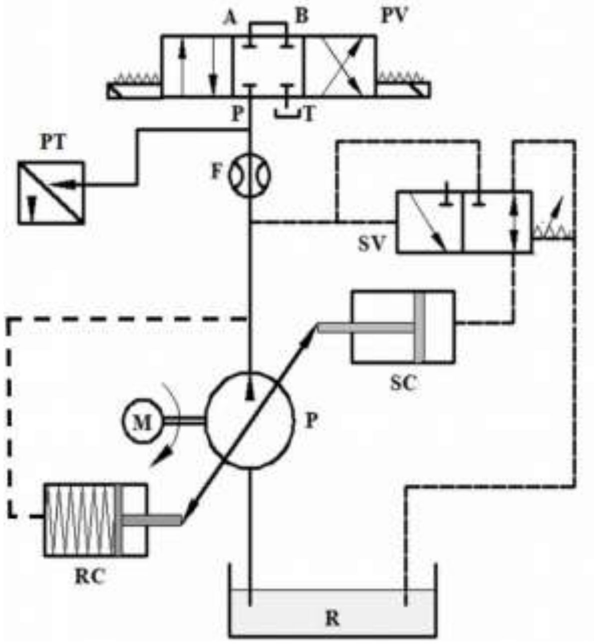


Figure 2. Hydraulic circuit diagram of a swash plate axial piston pump with pressure compensator.

Figure2 shows the hydraulic circuit of the variable displacement pump. The integral parts of the pump have been highlighted. The motor**(M)** initiates the barrel rotation and the pump**(P)**draw in the fluid from the reservoir**(R)**. The connection of the swash plate with the pressure compensator cylinders determines the swash angle and thus, output flow of the pump. The proportional valve**(PV)** acts as the load orifice, the opening area of which determines the load requirement on the pump. Depending on the load orifice area the pressure has been developed in the outlet of the pump. The corresponding delivery pressure is measured by the pressure transducer**(PT)**. The change in pressure leads to displacement of the spool valve **(SV)**and the pressure compensator cylinders act in accordance with the respective changes to vary the output flow rate. The flow rate of the pump is measured using a flow meter **(F)**. There is also a connection of the spool valve case with the reservoir in order for the working fluid to flow out of the valve.

3. Design Methodology

The design methodology can be separated into two sections, one concerned with the design of the cylinders in the pressure compensator and the next one, with the modelling of the pump considering its eccentricity.

3.1. Design of Pressure Compensator Cylinders

The pressure compensator cylinders design is primarily based on the torque balance offered by three main elements, viz. the barrel-cylinder unit, the rate cylinder and the stroking cylinder. When the swash plate is at any static equilibrium position other than maximum swash angle, the net torque on the swash plate about the swiveling axis is given by:

$$T_b + T_r + T_s = 0 \tag{1}$$

Where T_b is the total swiveling torque offered by the barrel cylinders, T_r is the rate cylinder torque and T_s is the stroking cylinder torque on the swash plate, respectively.

In Fig. 3, the engagement of the pressure compensator cylinders and barrel pistons is shown. The torque offered by these units depends on the pressure force of the working fluid on the barrel cylinders, as well as on the moment arm of each unit. Due to continuous rotation of the barrel pistons, the moment arm of the pistons continuously changes and is a function of the position of the barrel pistons on the pitch circle ($R_p sin\theta_o$). On the other hand, the moment arm of the rate and stroking cylinders (l_r, l_s) is fixed and is taken as the distance of the respective cylinders from the barrel axis.

The design of the pressure compensator cylinders is based on a fixed displacement pump model, which has been adopted from earlier works[4,7,19].The maximum dynamic swiveling torque on cut-in and cut-off pressure is measured at maximum and minimum swash angle. The only unit that contributes to the swiveling torque in a fixed displacement pump is the set of nine barrel pistons. The evaluation is carried out by identifying the various forces acting on a single barrel piston, as shown in Fig. 4a. The three major forces on the piston include:

1. Pressure force due to suction and discharge of the working fluid
2. Viscous force due to radial leakage flow through the clearance between the barrel bore and piston
3. Inertia force due to piston reciprocation.

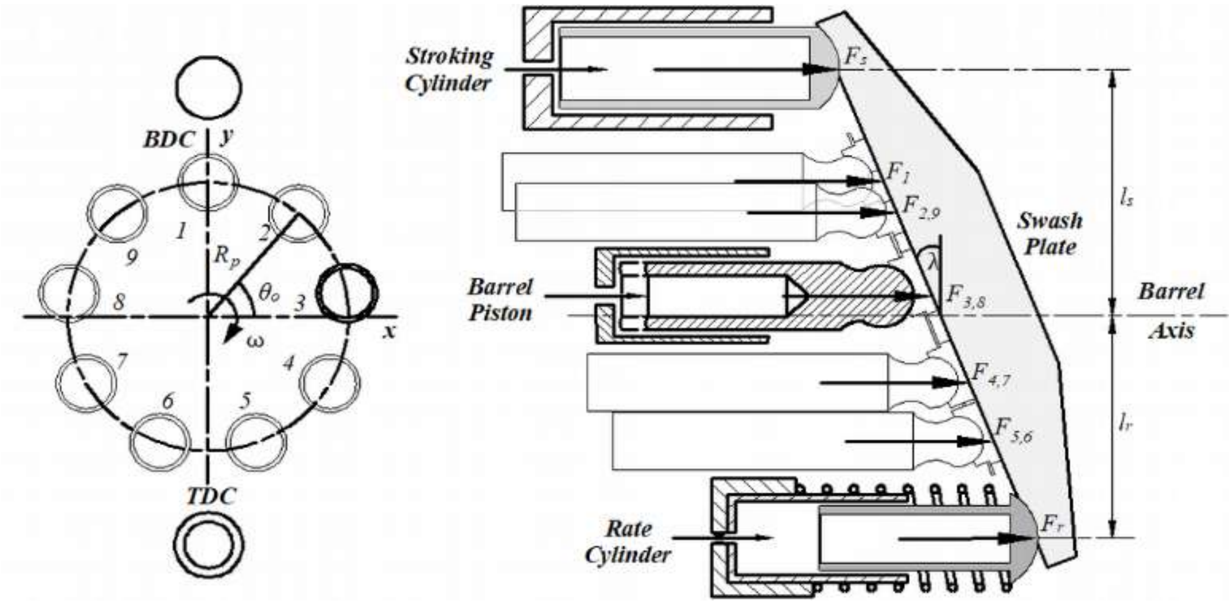


Figure 3. Engagement of rate and stroking cylinders and barrel pistons with the swash plate.

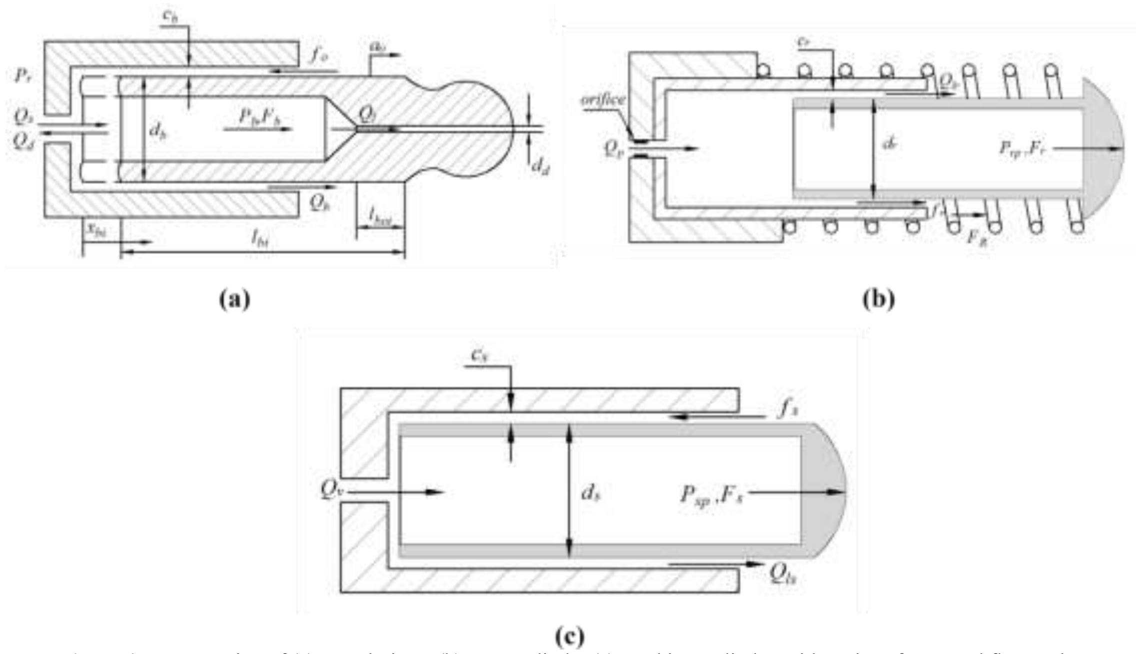


Figure 4. Cross-section of (a) Barrel Piston (b) Rate Cylinder (c) Stroking Cylinder, with various forces and flow paths.

The sum of these forces acts over the moment arm, which is the distance between the barrel axis and piston at a particular instant. The total swiveling torque is given by summing up the torque offered by all nine barrel pistons and can be expressed as,

$$T_b = \sum_{i=1}^{9} \{(P_b - P_r)\frac{\pi}{4} d_b^2 - f_o + m_b a_o\} l_i \quad (2)$$

where the pressure force is given by the product of the area of barrel piston (of diameter d_b) and the difference between the instantaneous pressure due to fluid flow (P_b) and reservoir pressure (P_r); inertia force as the product of mass of piston (m_b) and instantaneous linear acceleration(a_o); and (f_o) is the instantaneous viscous force due to leakages. l_i denotes the instantaneous moment arm.

The instantaneous viscous force due to leakages through the clearances is in turn given by,

$$f_o = -\pi \mu d_b \dot{x}_{bi} \left(l_{bi} - l_{boi} - x_{bi}\right) / c_b \quad (3)$$

wherex_{bi} is the displacement of i^{th} barrel piston from TDC, l_{bi} is the length of the piston. The negative sign indicates that the direction of the force is opposite to that of piston reciprocation.

3.1.1. Design of Rate Cylinder

A part of the main flow is directed towards the rate piston with the help of a connecting line. A cross-sectional view of the rate cylinder along with all the forces and flow paths is shown in Fig.4(b). The flow of the working fluid towards the rate piston provides the necessary pressure force to hold the rate piston in position. Thus, the net pressure force experienced by the rate piston is given by the difference between the delivery pressure and drain pressure. A rate spring coils around the rate piston, the main purpose of which is to provide additional support to the rate piston to hold the swash plate at its maximum angle. For all values of the pump delivery pressure below the set cut-in pressure, the spool valve remains closed. When the delivery pressure reaches the cut-in pressure, the pressure inside the rate piston is equal to the cut-in pressure. Due to this, a torque is created on the swash plate, which helps retain the maximum swash angle by balancing the swiveling torque offered by the barrel pistons. Since the pressure inside the stroking piston is a bare minimum, the stroking piston torque is negligible. Hence, the torque given by the rate cylinder on the swashplate is equal to swivelling torque offered by the barrel piston.Considering that the rate spring has no pre-compression and the rate piston has zero displacement, presented as

$$T_{bi} = (P_{rp} - P_r)\frac{\pi}{4} d_r^2 l_r \quad (4)$$

The stiffness of the rate spring determines the maximum swash angle taken up by the swash plate. Therefore, the design of the rate spring is essential. The mean diameter of the spring (D_R) must be a little higher than the rate piston diameter in order to accommodate the rate piston on it. Thus, appropriate values for the spring dimensions are chosen and the stiffness of the spring (k_R) is calculated as,

$$k_R = G d_R^4 / 8 D_R^3 N_R \quad (5)$$

Where$G$is the rigidity modulus of the spring material, d_R is the coil diameter of the spring andN_R is the number of active turns [37].

3.1.2. Design of Stroking Cylinder

As the delivery pressure goes beyond the cut-in pressure value, the pressure force overcomes the spool spring force and the spool valve slowly starts displacing. A direct result of this is the entry of the fluid to the stroking cylinder through the circumferential port on the spool valve. The stroking piston gets displaced and produces a torque on the swash plate in order to reduce the swash angle. Once the pressure increases to the cut-off value, the spool valve opens completely and the stroking piston gets displaced to its maximum extent. This causes the swash plate to attain minimum swash angle as a result of which the swept volumes of the rotating pistons become minimum. Due to this, the corresponding output flow becomes negligible. Hence, the stroking piston is designed at the cut-off conditionas shown in Fig.4(c).

For the cut-off condition, the pre-compression of the rate spring is considered as the rate piston unit is also actively engaged in the torque balance on the swash plate. The swiveling torque by the barrel pistons and the stroking piston torque act in the same direction, while the rate piston torque tries to counter them. Thus, Eqn. 1 is rearranged and the torque components of the pressure cylinders are substituted to get the following equation:

$$l_s\{(P_{spo} - P_r)\pi d_s^2 / 4\} + T_{bo} = \{(P_{rco} - P_r)\pi d_r^2 / 4 + k_R \delta_R\} l_r \quad (6)$$

where l_s is the moment arm or the distance between the axis of the stroking piston and barrel, P_{spo} and P_{rco} are the average delivery pressure at cut-off condition inside the stroking and rate cylinder, d_s is the diameter of the stroking cylinder, T_{bo} is the maximum swiveling torque at cut-off condition, k_R is the rate spring stiffness and δ_R is the pre compression of the rate spring.

Table 1. Values of Design Parameters

Design Parameters	Values
Rate spring stiffness (G)	79×10^9 Pa
Reservoir pressure (P_r)	2×10^5 Pa
Cut-in average delivery pressure (P_{rp})	190×10^5 Pa
Cut-off average delivery pressure (P_{spo})	200×10^5 Pa
Distance between stroking piston axis and barrel axis (l_s)	50×10^{-3} m
Distance between rate piston axis and barrel axis (l_r)	50×10^{-3} m
Barrel piston diameter (d_b)	7.5×10^{-3} m
Mass of barrel piston (m_b)	
Pitch circle radius (R_p)	35×10^{-3} m

4. Dynamics Simulation

The entire work has been done with the assumption that the properties of the working fluid remain the same. The reservoir pressure is maintained at a constant value of $2\text{x}10^5$ Pa. The barrel speed of the pump is set to 1500 rpm and the angular velocity is 50π radian/s. Additionally, since the clearance between the pressure compensator cylinders and pistons are very small compared to the diameter of the pistons, the leakage flow through these clearances is negligible and therefore, not considered for these piston-cylinder units in this design. The remaining pump parameters are obtained from previous works. The work is based on an adopted model of a fixed displacement pump. The simulations are carried out in MATLAB/Simulink and the equations are solved using the 4th order Runge-Kutta solver with a fixed step size of 2.5×10^{-7}s.

4.1. Diameter of Pressure Compensator Cylinders

The corresponding setting for cut-in condition has been done by fixing the swash plate at an angle of 22^0, which is the maximum angle the swash plate can possess.The flow and pressure dynamics greatly depend on the swash angle as increase in swash angle leads to increase in stroking volume of the barrel pistons and thus, increase in delivery flow and delivery pressure. By setting the load orifice area corresponding to cut-in opening area, the average delivery pressure is maintained at 190×10^5 Pa (Fig. 5(a)), which is considered as the pre-set cut-in pressure value. Figure 5(b) shows the simulation result of the average delivery flow at this condition, which corresponds to the maximum flow that the pump can produce. The spool valve keeps the path of the delivery flow to the stroking cylinder blocked. Hence, the swash plate only experiences torque by the barrel pistons and the rate cylinder, which is completely extended.The fluctuating nature of the flow causes fluctuations in the torque experienced by the swash plate and these variations are as shown in Fig. 5(c). For the design of the rate cylinder, the value of maximum swiveling torque at cut-in condition is considered for sturdy design. From the simulation results, this value is found to be -63.04 Nm. Plugging in the remaining design parameters in Eqn. 4, the optimum value of the rate cylinder diameter is 9.25×10^{-3} m.

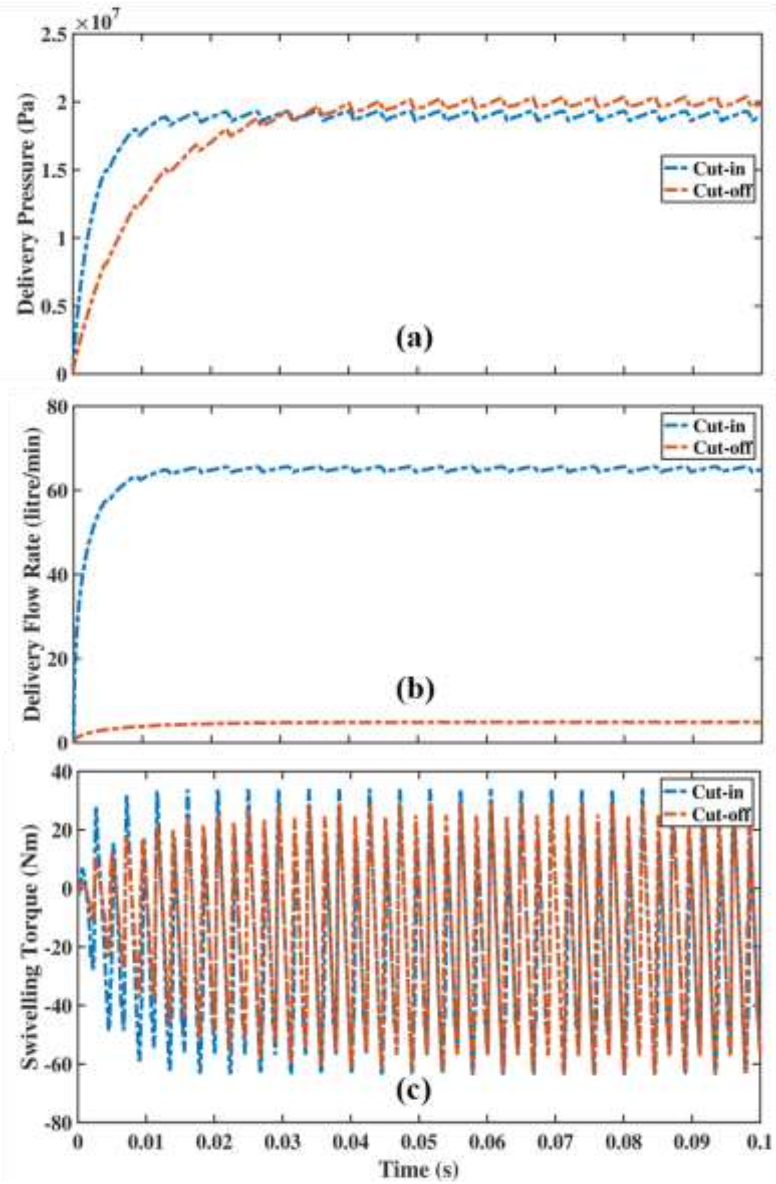


Figure 5. Simulation results for (a) Delivery pressure; (b) Delivery flow rate; (c) Swiveling torque with respect to time for cut-in and cut-off conditions.

The rate spring is designed on the basis of the rate cylinder diameter. The coil diameter of the spring should be a little higher than the rate cylinder diameter and thus, is taken as 10.5×10^{-3} m. From the standards table for spring design, the spring index is chosen as 11.5 and the number of active turns as 11. For the considered material of the spring, the spring stiffness is found to be 538.94 N/m (Eqn. 5), which lies well within the estimated range of required spring stiffness for robust design.

As the stroking cylinder design is carried out at the cut-off condition, the swash angle is set to 0^0, which is the ideal minimum angle the swash plate can be held at. However, this cannot be attained practically due to internal leakages within the pump. This angle is achieved when the delivery flow reaches the stroking cylinder and actuates it to its maximum extent, which rotates the swash plate to attain minimum angle. The load orifice area opening is set corresponding to cut-off condition and the average delivery pressure becomes equal to the pre-set cut-off pressure value of 200×10^5 Pa (Fig. 5(a)). The corresponding delivery flow for this condition is also recorded and displayed in Fig. 5(b). It is seen that the delivery flow reduces to the order of 2 litre/min. This flow compensates the internal leakages within the pump and so, the net flow out of the pump becomes zero.This situation corresponds to minimum swash angle, which drops the flow pressure. Once again, the rate cylinder tries to extend, thus increasing the flow but then it is opposed by the stroking cylinder. This results in oscillation of the swash plate close to zero angle and the net flow out of the pump is equal to the internal leakage of the pump. From the simulation result of the swiveling torque on the swash plate (Fig. 5(c)), the maximum torque value obtained is -62.26 Nm. The remaining parameters are substituted in Eqn. 6 and the optimum value of the stroking cylinder diameter is found to be 12.87×10^{-3} m.

Thus, by setting the swash angle at the cut-in and cut-off conditions, the average delivery pressure is seen to correspond with these conditions. The delivery flow rate changes with the swash angle, as mentioned in the theory related to the pump. Therefore, the dynamic simulation validates the design methodology used. From the simulation runs, the respective maximum swiveling torque values for the two conditions are found and used to design the pressure compensator cylinders.

4.2. Validation of Model

Figure 6. shows the distribution of pressure dynamics according to the load valve dynamics from cut-in to cut-off. Mondal et al. [23] developed the compensator actuator based on the static torque balance but in the current work is based on the maximum and minimum dynamic torque acting on the swash plate. The tendency of the pressure distribution is very similar. The frequency of the pressure ripples is high in case of static torque balance model [23] than the present work. The high frequency pressure ripples leads the vibration in the system. Hence, from this section it has been established that the dynamic torque balance concept can reduce the vibration of the system.

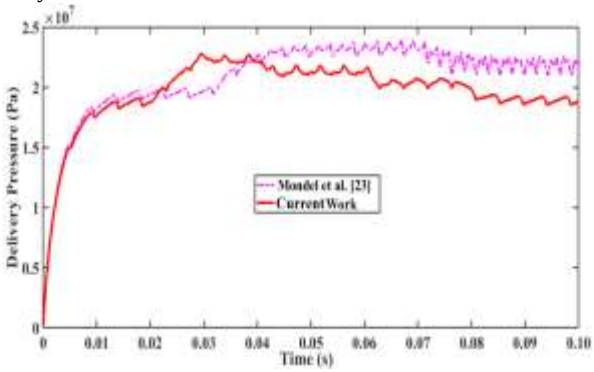


Figure 6. Distribution of pressure dynamics

5. Modeling of Pump considering Eccentricity

Eccentricity, which is the distance between the barrel axis and the point about which the swash plate swivels, has been found to be another important design parameter. The instantaneous moment arm of a barrel piston is given by the distance between the piston axis and the swash plate pivot. When eccentricity is provided to the pump, the distance between the swash plate pivot and the piston axis either increases or decreases, depending on whether the eccentricity value is positive or negative. The moment arm in Eqn. 2 is therefore modelled as,

$$l_o = R_p \sin\theta_o + e \tag{7}$$

Where R_p is the pitch circle radius, θ_o is the instantaneous position of the barrel piston and e is the eccentricity.

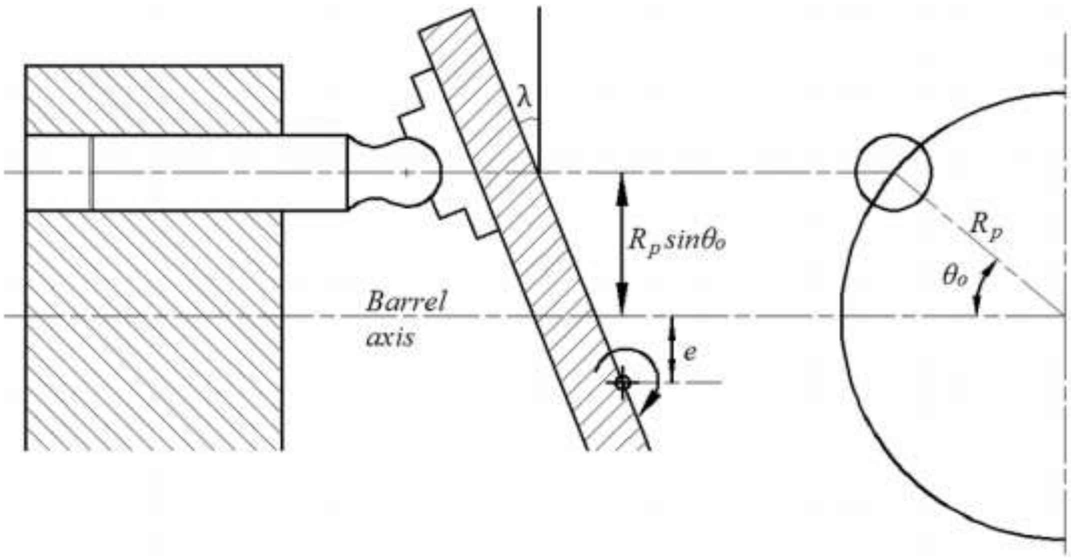


Figure 7. Pump configuration with positive eccentricity

Due to this, the swiveling torque by the barrel piston also changes. For the initial design of the pressure compensator cylinders, the eccentricity value has been considered as zero, that is, the swash plate pivot is made to coincide with the barrel axis. In Fig. 7, the positive eccentricity is presented. In this case, the swash plate pivot is positioned closer to the rate piston side. This leads to a decrease in the distance between the rate piston and swash plate pivot and an increase in the distance between the stroking piston and swash plate pivot. In other words, the rate piston moment arm decreases and thus, the rate piston torque decreases.

$$T_r = F_r\left(l_r - e\right) \tag{8}$$

Consequently, the stroking piston moment arm and thus, the stroking piston torque would increase.

$$T_s = F_s\left(l_s + e\right) \tag{9}$$

Referring to Eqns. 4 and 6, it is evident that there would be some changes in the diameters of the rate and stroking piston due to changes in each torque component. It is therefore clear that a change in configuration due to eccentricity would lead to changes in the pump size. Thus, effects of eccentricity variations are carefully analyzed and the results are discussed in the following sections.

5.1. Effect of Eccentricity on Pump Size

For this section, the simulations of the fixed displacement pump model are made to run for various positive and negative eccentricity values. The pressure compensator cylinders are designed using the same steps as discussed in the previous section. For each case, the maximum swiveling torque at cut-in condition is used for calculating the rate cylinder diameter, while the stroking cylinder diameter is calculated by considering the maximum swiveling torque at cut-off condition. All other pump parameters are maintained as in the cases above. Using Eqns. 4 and 6, the diameters of the rate and stroking cylinders are found. The results are tabulated and shown in Table 2. From the table, it can be seen that the diameters of the rate and stroking cylinders decrease as the positive eccentricity is increased and vice-versa. Thus, by configuring the pump with appropriate positive eccentricity value, the size of the pressure compensator cylinders can be reduced and as a result, the pump size can also be reduced.

5.2. Effect of Rate Cylinder Diameter on Pump Performance

The performance of a variable displacement pump is assessed by considering various rate cylinder diameters. In the study, all other design parameters are maintained same as the previous cases. Table 3 shows the design considerations made in this section.

It can be noticed from Fig. 8 (a) that as the rate cylinder diameter is decreased **(Design 2)**, the cut-off swash angle is closer to the ideal condition of 0^0. This is because the swash plate does not receive much resistive torque from a smaller rate piston. Therefore, it can easily attain a lower swash angle due to the stroking piston torque at the cut-off condition. With an increased rate cylinder diameter **(Design 3)**, the cut-off swash angle follows an oblique pattern, with higher values than acceptable. It deviates from the ideal no-flow condition and thus, lowers pump performance.

As the rate cylinder diameter is reduced, a slight reduction in delivery pressure is observed corresponding to the cut-off condition (Fig. 8(b)). It is almost equivalent to the delivery pressure variation observed in the standard design **(Design 1)**. However, as rate cylinder diameter is increased, a high delivery pressure is produced at the cut-off condition **(Design 3)**. This is a result of higher cut-off swash angle, which allows more fluid flow and therefore, larger delivery pressure. Such high pressures may damage the pump and therefore, should be prevented.

Table 2. Pressure compensator diameters corresponding to various eccentricity values

S.No.	Eccentricity (e), m	Cut-in Torque (T_{bi}), Nm	Rate Cylinder Diameter (d_r), m	Cut-off Torque (T_{bo}), Nm	Stroking Cylinder Diameter (d_s), m
1	-0.0015	-83.95	10.507×10^{-3}	-83.88	15.12×10^{-3}
2	-0.0011	-78.48	10.20×10^{-3}	-78.10	14.54×10^{-3}
3	-0.0010	-77.00	10.11×10^{-3}	-76.66	14.39×10^{-3}
4	0	-63.30	9.25×10^{-3}	-62.26	12.87×10^{-3}
5	0.001	-49.66	8.285×10^{-3}	-47.86	11.34×10^{-3}
6	0.0011	-48.39	8.187×10^{-3}	-46.42	11.07×10^{-3}
7	0.0015	-42.94	7.743×10^{-3}	-40.73	10.36×10^{-3}

Table 3. Rate cylinder diameter variations

Design No.	Rate cylinder diameter (d_r), m	Stroking cylinder diameter (d_s), m
Design 1	9.25×10^{-3}	12.87×10^{-3}
Design 2	7.25×10^{-3}	12.87×10^{-3}
Design 3	11.25×10^{-3}	12.87×10^{-3}

The delivery flow rate overlaps for all three design considerations (Fig. 8(c)). However, a slightly higher flow is observed for Design 3 during the gradual change from cut-in to cut-off condition and vice-versa. Although this may improve pump performance, the negative effects of other parameters have greater impact. Therefore, a lower rate piston diameter enhances the pump performance.

5.3. Effect of Stroking Cylinder Diameter on Pump Performance

The stroking cylinder diameter is varied to assess its impact on the variable displacement pump's performance. Table 4 contains the various design considerations for this study.

Table 4. Stroking cylinder diameter variations

Design No.	Rate cylinder diameter (d_r), m	Stroking cylinder diameter (d_s), m
Design 1	9.25×10^{-3}	12.87×10^{-3}
Design 2	9.25×10^{-3}	10.87×10^{-3}
Design 3	9.25×10^{-3}	14.87×10^{-3}

From Fig. 9(a), it is observed that cut-off swash angle is closer to ideal condition for higher stroking cylinder diameter **(Design 3)** than the corresponding standard design **(Design 1)**. This is a result of higher stroking torque generated by the bigger stroking cylinder. This pushes the swash plate to achieve a lower swash angle at the cut-off condition. For lower stroking cylinder diameter **(Design 2)**, the cut-off swash angle is close to cut-in angle and therefore, shows a large deviation from the ideal no-flow condition. Fig. 9(b) shows the variation in delivery pressure for the design considerations. As the stroking cylinder diameter is reduced **(Design 2)**, the delivery pressure is higher than the corresponding standard design **(Design 1)**. This is due to the higher swash angle at the cut-off condition, which increases the delivery pressure. However, delivery pressure corresponding to larger stroking cylinder diameter **(Design 3)** is equivalent to standard design **(Design 1)**. It is observed from Fig. 9(c) that the delivery flow overlaps for all three designs, with a slight increase for lower stroking cylinder diameter **(Design 3)** during the gradual change in swash angle. Thus, a higher stroking cylinder diameter enhances pump performance.

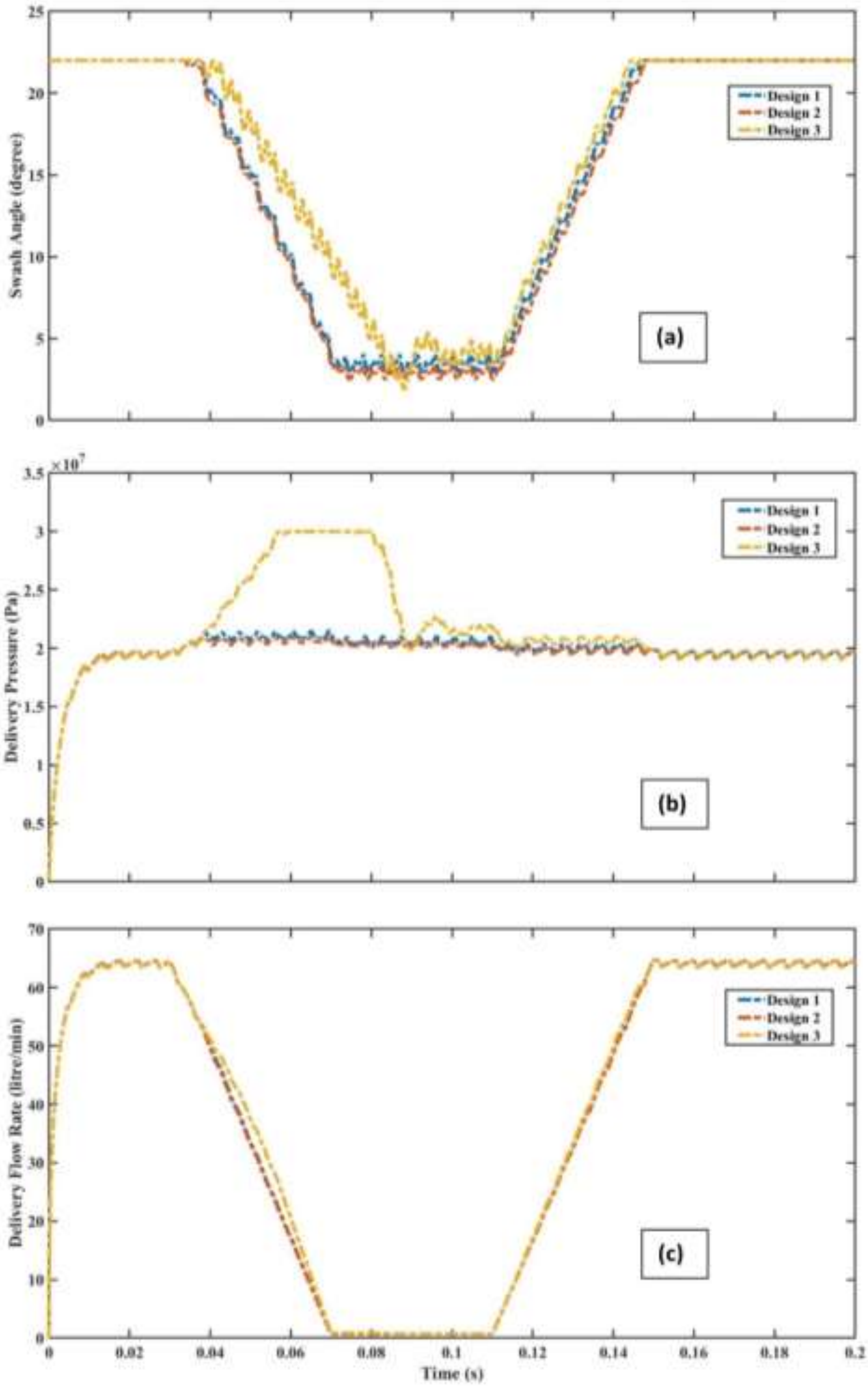


Figure 8. Simulation results for (a) Swash angle; (b) Delivery pressure; (c) Delivery flow rate with respect to time for various rate cylinder design considerations.

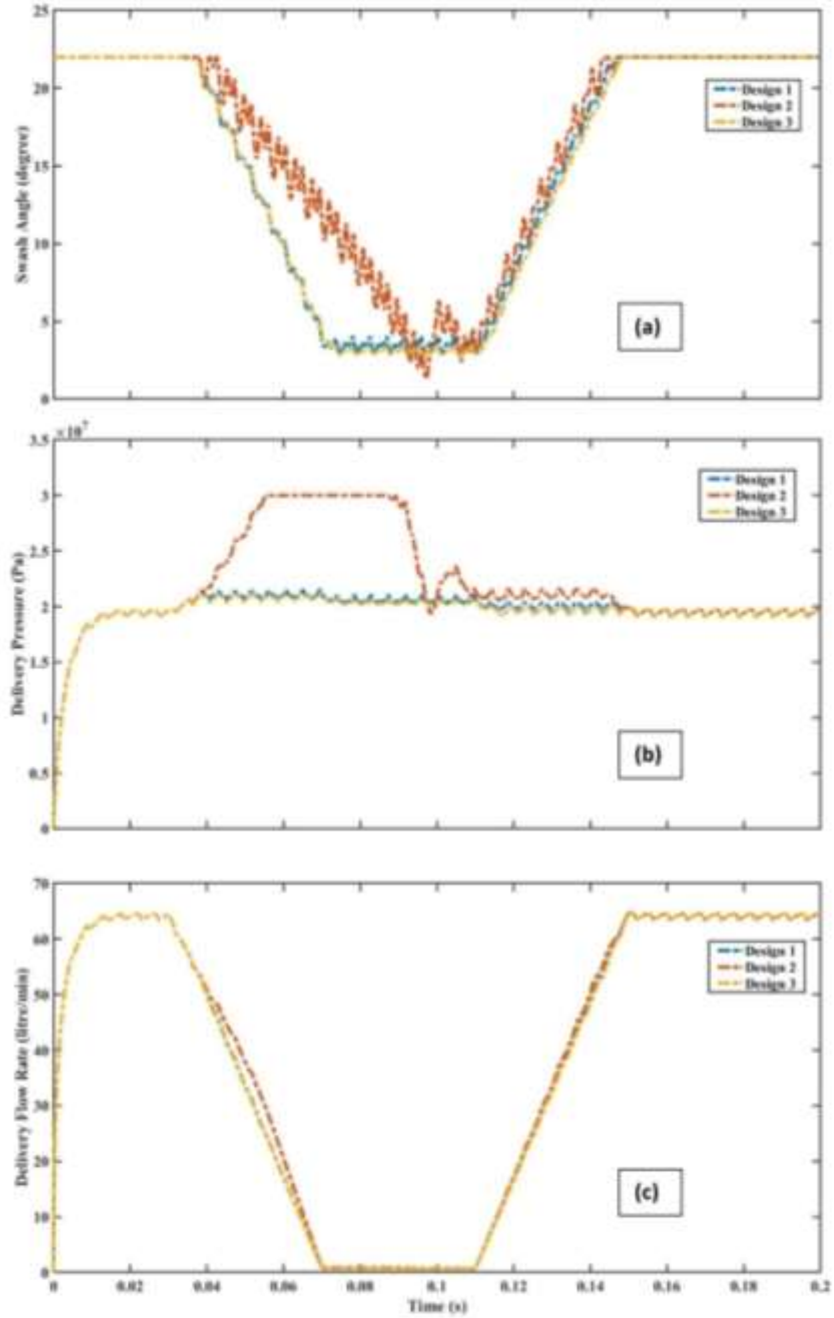


Figure 9. Simulation results for (a) Swash angle; (b) Delivery pressure; (c) Delivery flow rate with respect to time for various stroking cylinder design considerations.

5.4. Effect of Positive Eccentricity on Pump Performance

Positive eccentricity corresponds to the condition in which the swash plate pivot is closer to the rate cylinder (Fig. 7). In this study, the eccentricity is varied, as provided in Table 5, to study its effect on pump performance. Furthermore, the rate and stroking cylinder diameters are maintained at 9.25x10−3 m. and 12.87x10−3 m, respectively. The simulations period is considered as 0.1s, within which the working of the pump reaches stability. No variations in the swash angle at the cut-in condition are observed from Fig. 10(a). Variations are noticed during transitions (cut-in to cut-off condition and vice-versa) and throughout the cut-off section. Higher positive eccentricity lowers the cut-off swash angle. This shows that the swash angle closest to ideal cut-off swash angle is achieved by increasing positive eccentricity **(Design 3)**. Variations are observed in the delivery pressure during the transition and at the cut-off section (Fig. 10(b)). The delivery pressure undergoes a slight reduction with increased eccentricity. However, the delivery flow rate overlaps for all considered values of positive eccentricity (Fig. 10(c)). Thus, higher positive eccentricity enhances pump performance.

Table 5. Positive eccentricity considerations

Design No.	Eccentricity (*e*), m
Design 1	0
Design 2	0.001
Design 3	0.0015

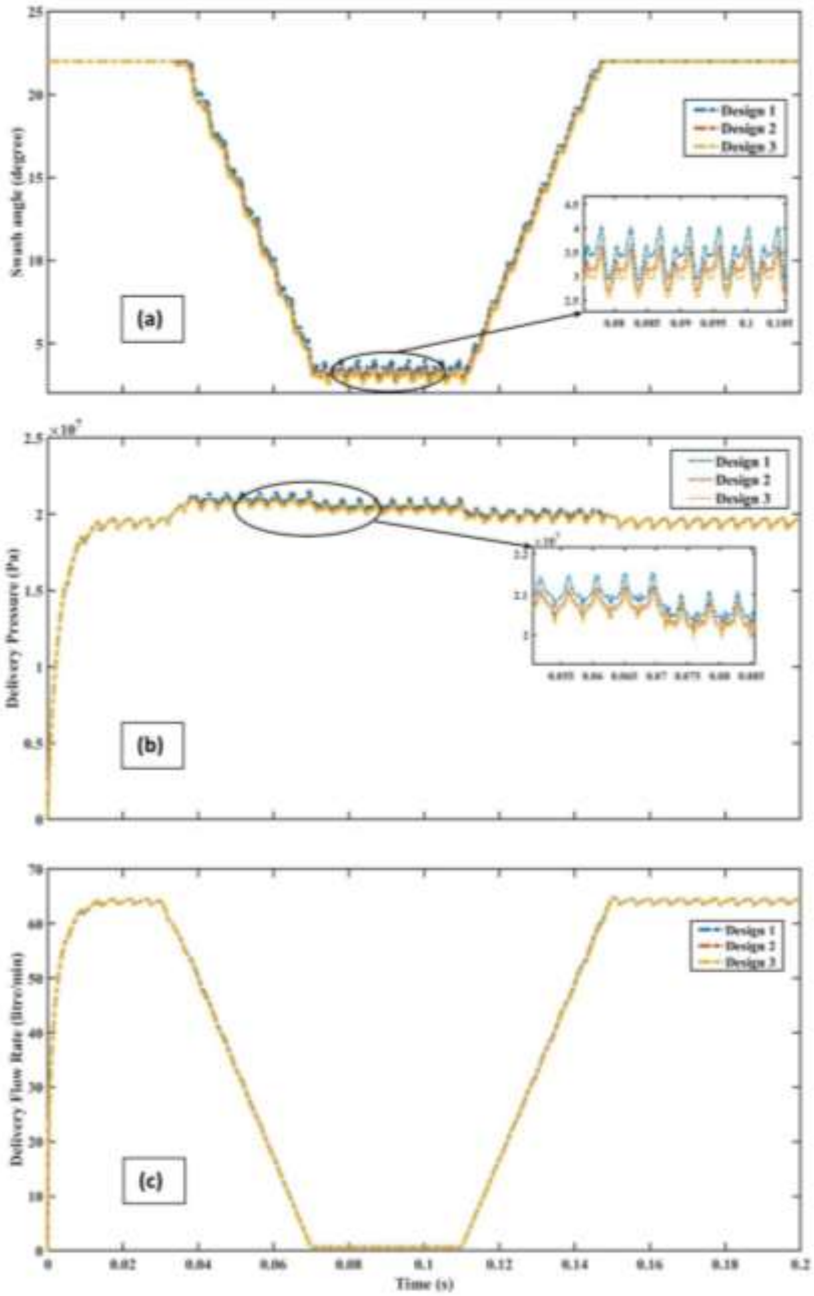


Figure 10. Simulation results for (a) Swash angle; (b) Delivery pressure; (c) Delivery flow rate with respect to time for various positive eccentricity design considerations.

5.5. Effect of Negative Eccentricity on Pump Performance

Negative eccentricity corresponds to the condition in which the swash plate pivot is closer to the stroking cylinder (Fig. 7). For this study, the design considerations are provided in Table 6. Rate and stroking cylinder diameters are maintained at the values specified for the positive eccentricity scenario and the simulation period is also considered as 0.1s.

Table 6. Negative eccentricity considerations

Design No.	Eccentricity (*e*), m
Design 1	0
Design 2	-0.001
Design 3	-0.0015

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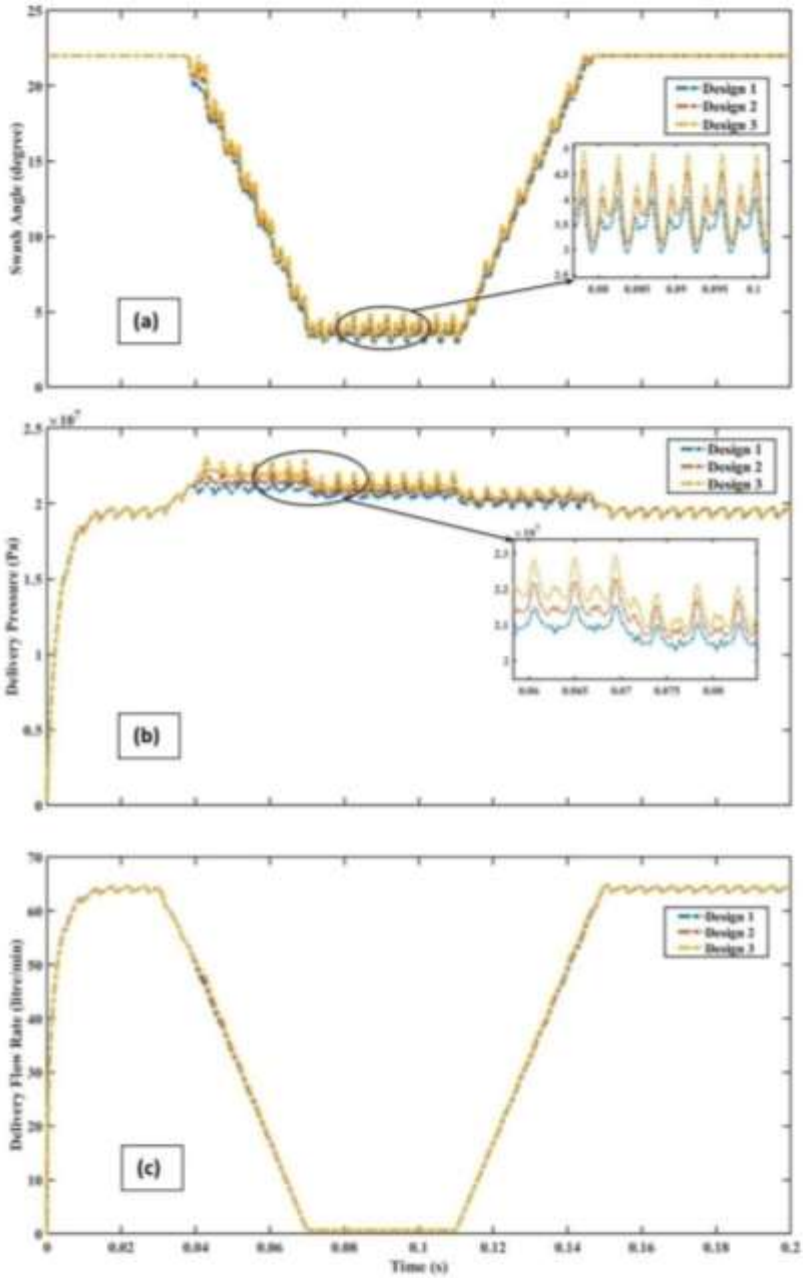

Figure 11. Simulation results for (a) Swash angle; (b) Delivery pressure; (c) Delivery flow rate with respect to time for various negative eccentricity design considerations.

It is observed from Fig. 11 (a) that a higher negative eccentricity **(Design 3)** increases the cut-off swash angle from the ideal condition. Moreover, a slightly higher delivery pressure is produced during the cut-off condition corresponding to this design (Fig. 11(b)). However, delivery flow rate overlaps with the other design considerations (Fig. 11(c)). Therefore, higher negative eccentricity lowers the pump performance through deviation from ideal no flow condition.

Thus, no noticeable changes are observed in the delivery flow. But the delivery pressure reduces with increased positive eccentricity. However, higher negative eccentricity causes the pump performance to slightly deteriorate due to deviated cut-off swash angle. Therefore, provision for an optimum value of positive eccentricity within the range of 0.001 m to 0.0015 m is recommended.

6. Conclusion

The dynamic swiveling torque has been evaluated at maximum and minimum swash conditions through MATLAB/Simulink environment for a fixed displacement pump. The diameters of the rate and stroking cylinders are evaluated using the estimated torque. A low stiffness spring has been designed based on the rate cylinder diameter to hold the swash plate mechanically at maximum swash angle for a variable displacement pump. The developed design procedure based on the dynamic swiveling torque balance on the swash plate helps the designer to design a variable displacement pump from a fixed displacement pump without changing the main pump configuration.

The eccentricity simulations show that the diameters of the pressure compensator cylinders decrease with higher positive eccentricity values without any changes in the performance of a fixed displacement pump. This major finding can thus be used to reducethe size of the pump. Therefore, manufacturingcost and material cost can be reduced. For a variable displacement pump, lower rate cylinderdiameter and higher stroking cylinder diameter, enhancesits performance. Also, the delivery flow remains the same for various positive and negative eccentricity values with a minute reduction in thedelivery pressure for higher positive eccentricity. Moreover, a lower swash angle is achieved with increased positive eccentricity. Hence, the pump operates closer to ideal no flow condition during the cut-off operating cycle, which in turn improves the pump performance.

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