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MODELLING AND SIMULATION OF A CLOSED LOOP PUMP HYDRAULIC CIRCUIT FOR NEUTRAL PRESSURE VARIATION STUDY

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Abstract: *Closed loop pump circuits are used in hydrostatic transmission (HST) which combines variable displacement piston pump and fixed displacement motor. This paper studies one of the most important parameters i.e. 'neutral pressure variation' which is safety critical element to prevent the unexpected vehicle movement. The HST system has been modelled using MATLAB Simulink tool with meticulous modeling efforts on the axial piston motor, relief valves, check valves, shuttle valves, charge pump circuit along with pump swash controls. The closed loop system model has been validated with test bench result. Further, neutral pressure variation as predicted by model is correlated with test data. The study of dynamical behavior of such system is very significant for accurate estimation of closed loop system performance parameters like motor starting torque, system pressure, motor speed characteristics and their dynamic fluctuations. The detailed system model proposed here enables simulation of the dynamic behavior of closed loop pump circuit, facilitating prediction of the dynamic performance of system, which can be used to improve the design parameters through parameters sensitivity analysis.*

Keywords*: Hydrostatic Transmission, Axial Piston pump, Axial Piston Motor, Neutral Pressure, Simulation*

1. Introduction

Hydrostatic transmission (HST) have been widely used in the mobile machines and off-road vehicles such as wheel loaders, fork lift, tractors, harvesters etc. Hydraulic power transmission bears several advantages such as higher efficiency, variable speed & torque control, step less speed change with easy forward/backward movement, capable to transfer over longer distance, hydrostatic braking, handling heavy loads and smooth operation [1]. Hydrostatic transmission utilizes a variable displacement axial piston pump and fixed displacement axial piston motor connected in closed loop system. Figure 1 shows the simple closed loop pump circuit of hydrostatic transmission with control. Oil is circulated by the pump to the motor and then returned directly back to the pump. A charge supply is used to supplement the closed loop system with oil as well as to avoid cavitation and the charge flow is supplied external source. The pump is driven by the prime mover, which generates flow to drive the hydraulic motor.

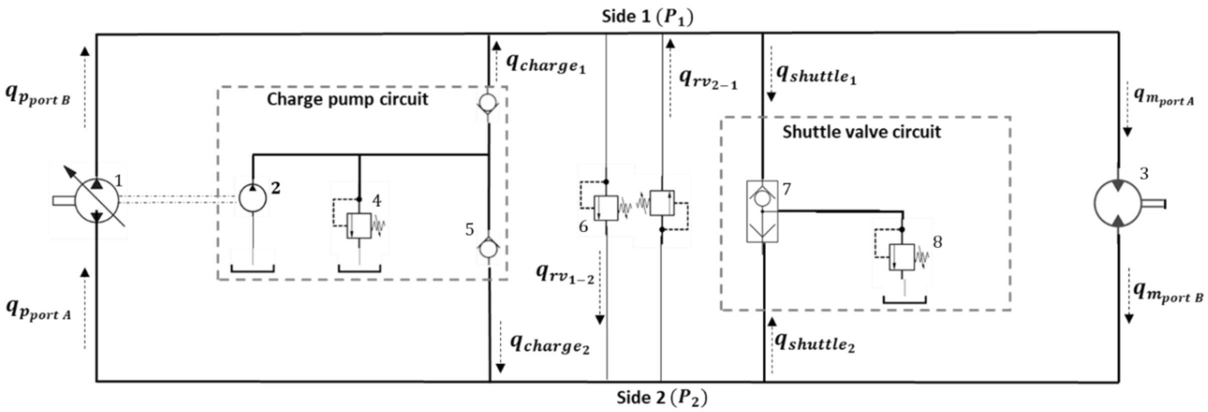


Fig. 1. Hydraulic schematic diagram of closed loop circuit in HST
1.Bidirection piston pump; 2. Charge pump; 3. Hydraulic motor; 4. charge pump relief valve; 5. Check valve; 6. IPOR relief valve; 7. Shuttle Valve; 8. Relief valve

The motor is connected to the gear box which transmits the power. Pressure is generated in the system due to the load connected to the hydraulic motor. The system pressure relief valve is used to maintain the maximum pressure in the forward and reverse direction of pump rotation. Check valves are provided to distribute the additional flow from the charge pump based on the direction of rotation of the pump. Direction and rotation of the motor depends of the output flow of the pump. Pump flow depends on the speed and displacement of the pump. The displacement of the pump is controlled by the swashplate position.

Several CTQ's can be addressed by developing a mathematical model for the closed loop pump circuit system and one of the CTQ in the current paper study is neutral pressure variation. When pump is in closed loop circuit and is brought back from full load, full stroke to no load and no stroke condition, ideally it is expected that pressure differential between pump port A and B should also fall back to zero. However, because of frictional losses and leakages, some pressure differential may occur which is called neutral pressure variation. Neutral pressure variation can be summed up as difference in pressure between two ports of closed loop circuit piston pump at no stroke and no-load condition. This is very critical when the vehicle or any machine application needs to stand at neutral i.e. without driving the external load, unexpected hazardous movement will occur when the pressure exceeds more than certain limit. Usually that limit is set to be around 30 psi. So, in any hydrostatic transmission this is very important in meeting the requirement and a mathematical model of closed loop pump circuit will help in evaluating the situation better before pump testing. Also modelling and simulating the entire closed loop circuit represent a strong and reliable approach and allow the better understanding of the baseline system, influence of design parameters on the system performance.

The aim of the paper is to show the modelling effort to develop the system level closed loop pump circuit model and to predict the neutral pressure variation CTQ. As part of this system CTQ, subsystem CTQ's like motor starting pressure, torque, torque ripple was established and verified. The model started with 49cc piston pump with 49cc fixed displacement piston motor configuration, the methodology can be scaled up to higher displacement motor/pump combination. The mathematical model was developed from first principle using MATLAB Simulink software. The system model consists of several subsystems like pump, motor, swash controls, charge pump, relief valve, check valves etc and there is complexity associated at bigger system level integration. Hence a holistic step by step approach with sub system modelling and verification of each subsystem was carried out on the flow, pressure characteristics. The model was validated with dynamic test data for the pressure, speed performance and predicted the variation of neutral pressure.

2. System Modelling of CC circuit components

This section addresses modelling aspects of various hydraulic components involved in cc circuit as shown in figure (1). Primarily, we have two main hydraulic lines, one going from pump to motor and other from motor to pump. Accordingly, hydraulic components not explicitly connected to these lines are grouped into single unit from modelling perspective, as shown in schematic diagram in figure (1) with dashed border. Thus, major components which have been modelled in following sections are bidirectional variable displacement axial piston pump with electric proportional swash control, fixed displacement axial piston motor, IPOR valves, charge pump circuit and shuttle valve circuit. Each component either supply or extract fluid from both hydraulic lines. This idea is crucial for integration of all components to generate cc circuit model.

2.1 Bidirectional Variable Displacement Axial Piston Pump Model

Mathematical modelling of axial piston pump has been addressed before in [2,3,4,5]. However, the modelling approach and conventions followed, doesn't ensures bidirectional nature of the pump model. The pump model developed in [1] has been modified to inculcate bidirectional nature in pump model. Similar approach has then been carried on for motor model as well. Axis sign convention chosen in this model is as per figure (2)**.**

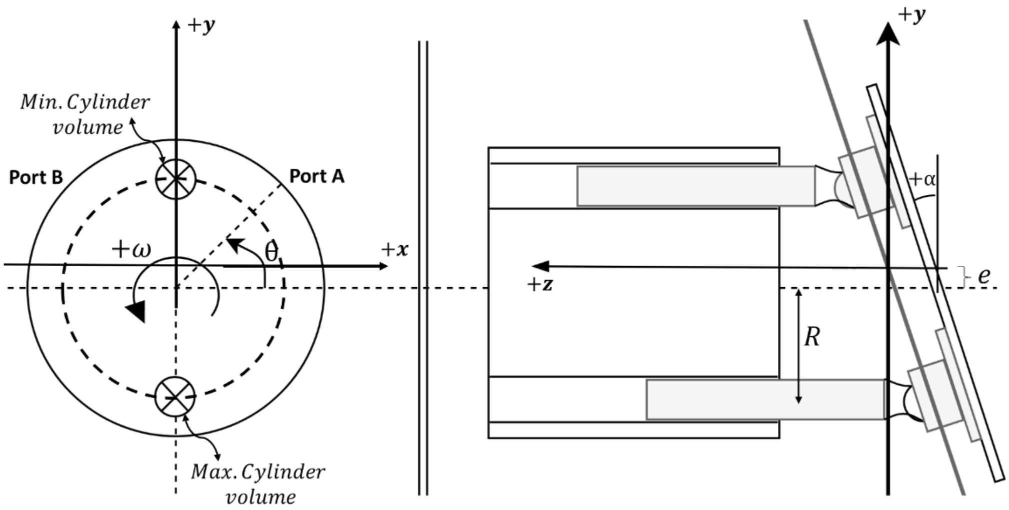


Fig. 2. Schematic diagram of pump RG kit with assigned sign convention

Port A & B side of pump as function of piston angular position, θ is defined as follow:

$$Port\ A\ Side: \theta \in \left[0, \frac{\pi}{2}\right) U \left(\frac{3\pi}{2}, 2\pi\right) \tag{1}$$

$$Port\ B\ Side: \theta \in \left[\frac{\pi}{2}, \frac{3\pi}{2}\right] \tag{2}$$

By convention, when swash angle and pump rotation are positive, port B and A are pump outlet & inlet, respectively. Bidirectional nature of pump is enabled by allowing swashplate to move in both directions.

2.1.1 Piston Kinematics & Pressure Dynamics

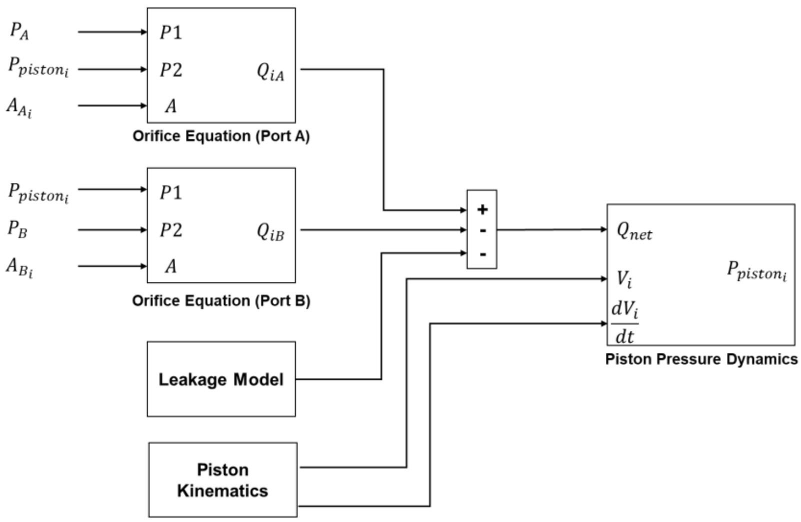


Fig. 3. Piston pressure dynamics model for pump

Flow going in and out of piston-barrel interface can be modelled using classical orifice equation based on Bernoulli's principle. [5] Thus, flow through an orifice,Q_i having pressure, P_1 and P_2 on either side and orifice area A, is given as:

$$Q_i = sign(P_1 - P_2)C_d A\sqrt{\frac{2}{\rho}|P_1 - P_2|} \tag{3}$$

At any instant, piston could be located on either port A or port B side. Depending on swash angle sign, port A & B can be inlet and outlet respectively or vice versa. Flow entering or leaving the piston through piston valve plate overlap area can be obtained using orifice equation. Figure (3) presents the schematics of piston pressure dynamics model. It's evident from geometric model that at any instant piston would be located either on port A or port B side. Depending on piston location and swash angle sign, instantaneous flow contribution of piston on either side is described in table (1)**.**

Table 1: Piston flow contribution on pump port A & B

Swash Angle Sign (α)	**Pump Port A**	**Pump Port B**	**Piston Location**	**Q_{A_i} (Sign)**	**Q_{B_i} (Sign)**
+	Inlet	Outlet	A	Q_{A_i}(+)	0
+	Inlet	Outlet	B	0	Q_{B_i}(+)
-	Outlet	Inlet	A	Q_{B_i}(-)	0
-	Outlet	Inlet	B	0	Q_{B_i}(-)

When swash angle is positive, flow entering or leaving piston as obtained from port A & B, i.e. Q_{A_i} & Q_{B_i} are either zero or positive. On contrary, for negative swash angle, they are either negative or zero. Thus, the sign change in flow contains the information of inlet and outlet switching. Q_{A_i} & Q_{B_i} can be obtained by substituting relevant information from schematic diagram shown is figure (3) into equation (3)**:**

$$Q_{A_i} = sign\left(P_A - P_{piston_i}\right)C_d A_i\sqrt{\frac{2}{\rho}\left|P_A - P_{piston_i}\right|} \tag{4}$$

$$Q_{B_i} = sign\left(P_{piston_i} - P_B\right)C_d A_i\sqrt{\frac{2}{\rho}|P_{piston_i} - P_B|} \tag{5}$$

A_i is instantaneous piston-valve plate overlap area through which fluid enters or leaves. It is a function of piston angular position θ_i. As per convention, for pressure dynamics in piston-barrel control volume, flow entering the CV is considered positive and vice versa. Multiplying port B piston flow to negative sign, ensures that net flow input to pressure dynamics block comes with correct sign irrespective of inlet and outlet change. Piston pressure, P_i is obtained by solving pressure dynamics equation for control volume shown in figure (4), obtained by applying the fundamental law of the conservation mass as discussed by Zeiger and Akers [6]:

$$\frac{dP_i}{dt} = \frac{\beta}{V_i}\left(Q_i - Q_{leak} - \frac{dV_i}{dt}\right) \tag{6}$$

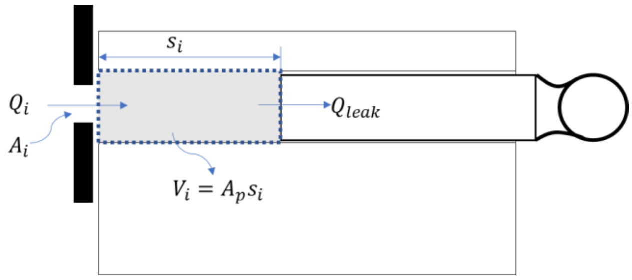


Fig. 4. Piston-barrel control volume

Control volume length, s_i is given as:

$$s_i = s_o - (Rsin\theta - e)\tan(\alpha) \tag{7}$$

Derivative of control volume dV_i/dt is obtained as:

$$\frac{dV_i}{dt} = -\frac{\pi d_p^2}{4}\left\{\left((Rsin\theta - e)\sec^2\alpha\right) * \frac{d\alpha}{dt} + Rcos\theta\tan(\alpha)\,\omega\right\} \tag{8}$$

Combining equation (6), (7) and (8), we get:

$$\frac{dP_i}{dt} = \frac{\beta\left(Q_i - Q_{leak} + \frac{\pi d_p^2}{4}\left\{\left((Rsin\theta - e)\sec^2\alpha\right) * \frac{d\alpha}{dt} + Rcos\theta\tan(\alpha)\omega\right\}\right)}{\frac{\pi d_p^2}{4}\{s_o - (Rsin\theta - e)\tan(\alpha)\}} \tag{9}$$

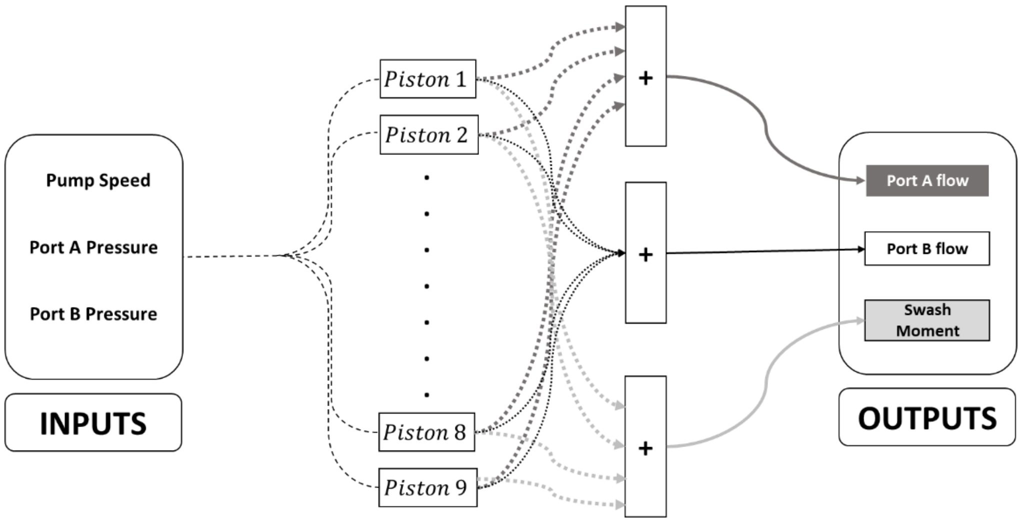


Fig. 5. Schematic overview of bidirectional pump mathematical model

Figure (5) presents an overview of bidirectional pump model. Pump speed, port A and B pressure goes as input to each piston. Each piston contributes either towards port A or port B flow which is summed up to obtain net flow through pump ports. Also, each piston produces a moment on swash plate which is discussed in next section. Likewise, each piston moment is added to obtain pump swash swivel torque.

2.1.2 Pump Swash Moment

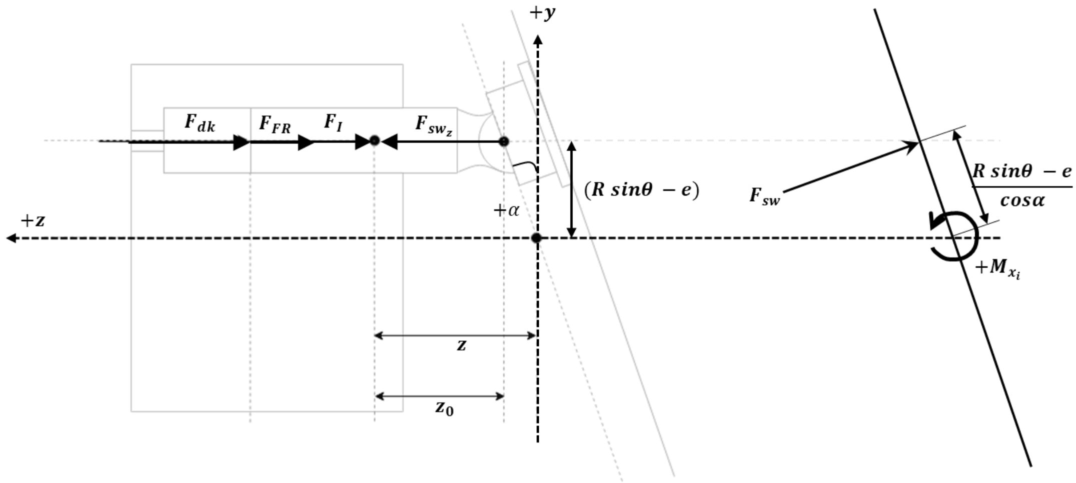


Fig. 6. Piston free body diagram for pump swash moment

Swash-piston reaction force F_{SW_i} generates a moment M_{x_i}, which produces tilting moment on swash plate. This moment needs to be countered by swash control mechanism. To obtain moment contribution of each piston, i.e. M_{x_i}, force balance on piston is done in z direction. Swash-Piston reaction force z-component, $F_{sw_{i_z}}$ is given as:

$$F_{sw_{i_z}} = F_{di} + F_{FR_i} + F_{i_I} \tag{10}$$

Where, F_{di} is pressure on piston and is described as [5]:

$$F_{di} = \frac{\pi d_k^2}{4}(P_{piston_i} - P_{case}) \tag{11}$$

Inertia force, F_{i_I} is given as:

$$F_{i_I} = -\, m_i \omega^2 R\ sin\ \theta_i\ tan\ \alpha \tag{12}$$

Modelling of friction force acting on piston barell overlap is quite complex in nature because of hydrodynamic effects involved and a huge topic of research. Friction models based on mechanical efficiency estimation [7] and empirical relation considering stribek and hydrodynamic friction as well as pressure dependant term [8] has been developed. For the sake of simplicity, friction force has been assumed to be negligible. However, suitable friction models [7,8] can be included to improve model physics. Swash reaction force, F_{sw} , perpendicular to swash plate can be obtained as:

$$F_{sw_i} = \frac{F_{sw_{i_Z}}}{\cos\alpha} \tag{13}$$

Moment arm for piston location θ_i is shown in figure (6). Considering axis sign convention, M_{x_i} is given by:

$$M_{x_i} = -\frac{(R\sin\theta_i - e)}{\cos\alpha}\{F_{dk} + F_I + F_{FR}\} \tag{14}$$

Considering 9 pistons, net pump swash moment is given as:

$$M_x = \sum_{i=1}^{9} M_{x_i} \tag{15}$$

M_x is essentially an internal disturbance which must be counteracted by swash plate control mechanism to ensure smooth operation.

2.1.3 Electric-Proportional Swash Displacement Control

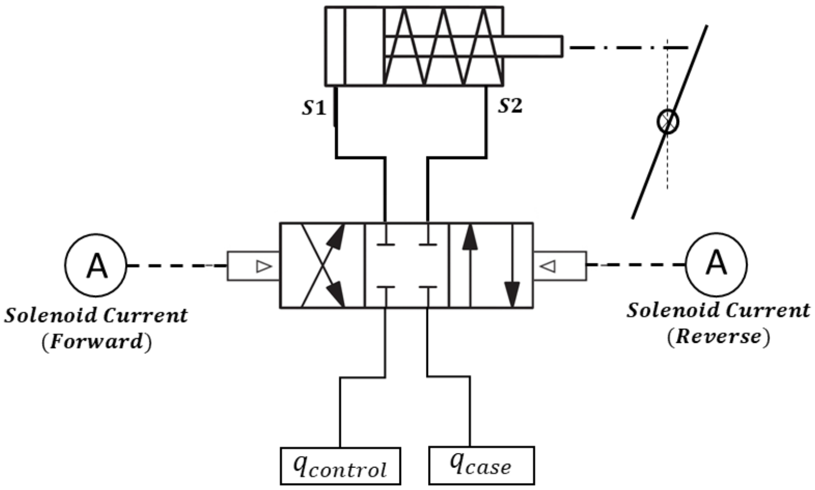


Fig. 7. Electric Proportional Pump Swash Control

Swash control circuit broadly consists of 4-way 6-position solenoid actuated directional control valve and double acting cylinder actuator which is connected to swash plate by mechanism suitable to particular pump design as shown in figure (7). Control flow, $q_{control}$ is extracted from flow supplied by charge pump to cc circuit. Under ideal circumstance, i.e. pressure drop across valve orifice is small and valve is completely shifted either of forward or reverse position, pressure on high pressure side of cylinder will nearly be equal to charge pressure. Flow drained out of control adds to case flow, q_{case}. Physical model of direction control valve has been developed following the principles discussed by Manring [9]and Merrit [10]. Forces acting on control piston can be listed as spring force, Swash reaction force, pressure force and viscous damping force. Swash swivel force acting on control pison, F_{swivel} can be obtained by multiplying swash moment M_x as described in previous section with moment arm of linkage mechanism. Pressure force is given as:

$$F_{pressure} = A_{Piston}(P_{S1} - P_{S2}) \tag{16}$$

Spring force as a function of spring displacement, x is given as:

$$F_{spring} = K(x + x_{preload}) \tag{17}$$

Likewise, viscous damping force, F_d is given as:

$$F_d = c\frac{dx}{dt} \tag{18}$$

Force balance on control piston yields:

$$M_p\frac{d^2x}{dt^2} + c\frac{dx}{dt} + K\left(x + x_{preload}\right) = F_{pressure} + F_{swivel} \tag{19}$$

At equilibrium, preload force, $Kx_{preload}$ cancels out swivel force. Pressure force is generated based on the soenoid signal input to DCV. Thus, indirectly it is control input in above dynamics. In present work, open loop control of pump is considered where predetermined forward and reverse solenoid signals are provided to DCV and accordingly control piston changes position as per dynamic equation discussed in eq. (19) thus changing swash angle.However, this can be modified to feedback controls depending on application.

2.2 Bidirectional Fixed Displacement Axial Piston Motor Model

Empirical model of axial piston motor as a part of hydrostatic transmission has been developed in [11,12] from control design perspective. Gao et. al. [13] presents physics-based model considering geometric parameters of motor from design and performance perspective. However, the bidirectional nature of motor hasn't been ensured. Thus, like bidirectional pump model, similar approach has then been carried on motor model allowing it to swing both ways. Sign convention for motor model has been adopted as per figure (8).

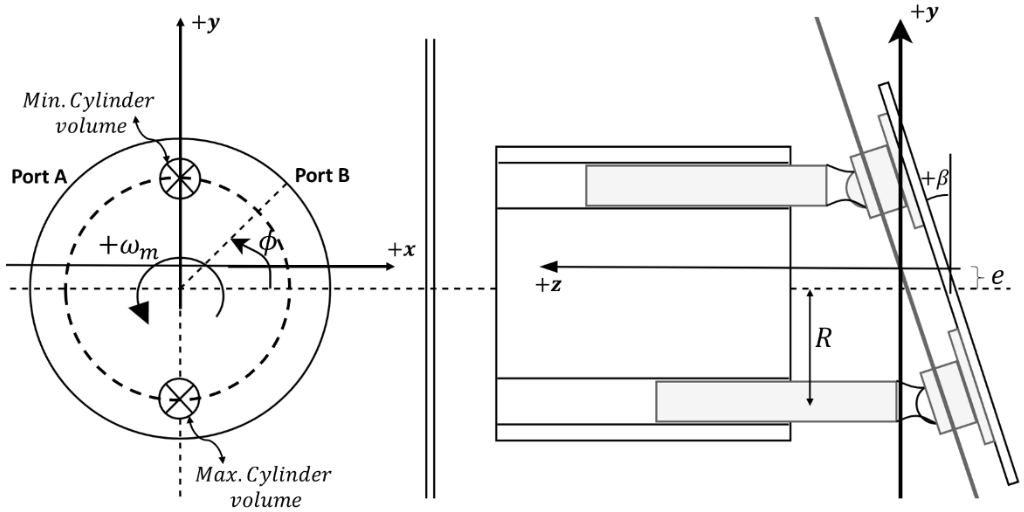


Fig. 8. Schematic diagram of motor RG kit with assigned sign convention

Port A & B side of axial piston motor as function of piston angular position, ϕ is defined as follow:

$$Port\ A\ Side: \phi \in \left[0, \frac{\pi}{2}\right) U \left(\frac{3\pi}{2}, 2\pi\right) \tag{20}$$

$$Port\ B\ Side: \phi \in \left[\frac{\pi}{2}, \frac{3\pi}{2}\right] \tag{21}$$

Being a fixed displacement motor, swash angle is fixed however motor can rotate in both senses depending on pressure at port A and B. By convention, motor would acquire positive angular velocity when port B and A are low & high-pressure side, respectively. In other word, port A will be motor inlet whereas port B will be motor outlet.

2.2.1 Piston Kinematics & Pressure Dynamics

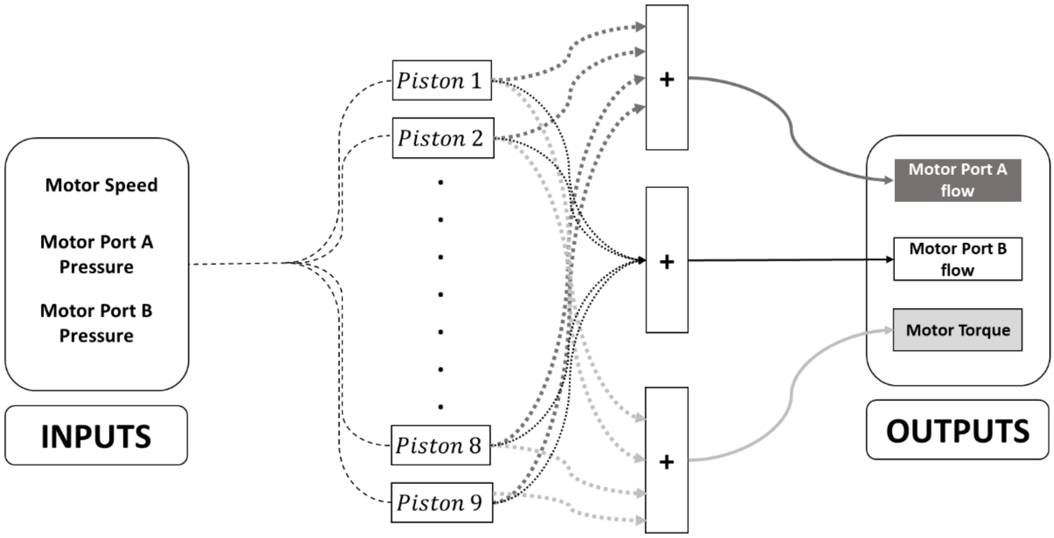


Fig. 9. Schematic overview of bidirectional motor model

Schematic of piston kinematics and dynamics for axial piston motor is like the one made for pump as shown in figure (3) with P_A & P_B being motor port pressures. Likewise, instantaneous flow contribution from each piston depending on piston location and sign of angular velocity is given below:

Table 2: Piston flow contribution on motor port A & B

Motor Angular Velocity Sign (ω)	Motor Port A	Motor Port B	Piston Location	Q_{A_i} (Sign)	Q_{B_i} (Sign)
+	Inlet	Outlet	A	Q_{Am_i}(+)	0
+	Inlet	Outlet	B	0	Q_{Bm_i}(+)
-	Outlet	Inlet	A	Q_{Bm_i}(-)	0
-	Outlet	Inlet	B	0	Q_{Bm_i}(-)

Just like in pump model, chosen sign convention for flow ensures that pressure dynamics behaves in correct fashion. Schematic overview of motor model is shown in Figure (9). Motor speed and pressure at motor port A, B goes as inputs to each piston. Port A and port B flow is determined by summing up flow output coming from each piston. Each piston will have individual contribution in net motor torque as described in following section. Motor speed is thus, determined by doing torque balance on motor shaft.

2.2.2 Motor Torque and Speed

Torque produced by motor is essentially net torque acting on the cylinder block along axis of symmetry. Since piston reaction force on cylinder contributes to cylinder torque, force balance on piston needs to be done. To obtain swash reaction force, we need pressure force, inertia force and friction force. For the sake of simplicity, friction force between piston barrel interaction has not been considered. Formula for pressure force and inertia force computation is same as eq. (11) and eq (12) respectively, with each variable bearing sense in context with motor**.** Thus component of swash reaction force in xy-plane, F_{sw_y} & F_{sw_x} as per sign convention are:

$$F_{sw_{i_y}} = -\left\{\frac{F_{dk}+F_{FR}+F_I}{\tan\beta}\right\} \tag{22}$$

$$F_{sw_{i_x}} = 0 \tag{23}$$

Slipper reaction force magnitude, F_{TG} is given by: [4]

$$F_{TG_i} = \frac{\mu\omega R}{h_g}\pi(R_g^2 - r_g^2) \tag{24}$$

R_g and r_g are slipper shoe inner and outer land radius. h_g is film thickness formed slipper shoe and swash plate. For simplicity sake, it has been assumed to be constant and of the order of 8 μm. [13] As per chosen sign convention, slipper reaction force has been resolved into x, y components, F_{TG_x} & F_{TG_y} as follow:

$$\begin{aligned} F_{TG_{i_x}} &= F_{TG}\sin\phi_i \\ F_{TG_{i_y}} &= -F_{TG}\cos\phi_i \end{aligned} \tag{25}$$

Centrifugal force, acting on piston is given as:

$$F_{\omega_i} = m_i\omega^2 R\ \frac{l_{s1}-l_{f/2}}{l_{km}} \tag{26}$$

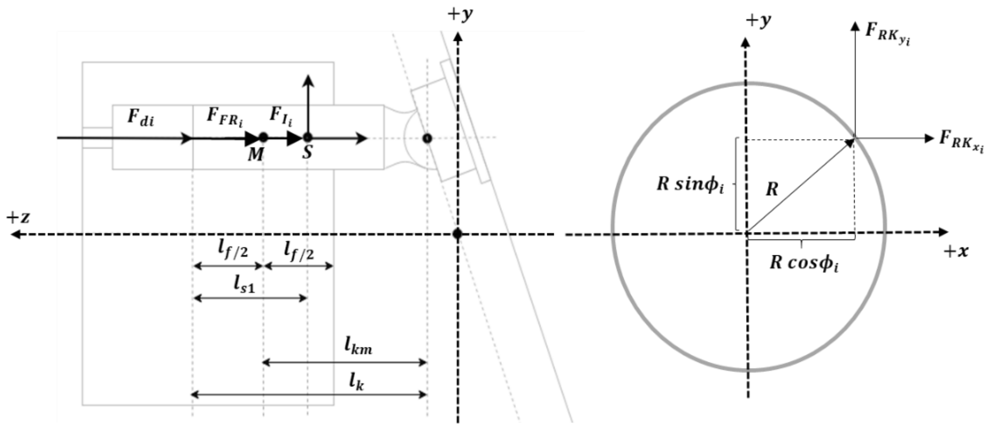


Fig. 10. Piston free body diagram for motor torque computation

ls_1 is piston's centre of mass distance measured from it's tip, as shown in figure (10). Other length, $l_{f/2}$ & l_{km} is given as:

$$\begin{gathered} l_{f/2} = (l_{fa0} + R\tan\beta(1+\sin\phi_i))/2 \\ l_{km} = l_k - l_{f/2} \end{gathered} \tag{27}$$

Resolving F_{ω_k} into x, y components,we get:

$$F_{\omega_{ix}} = F_{\omega_k} \cos\phi_i$$
$$F_{\omega_{iy}} = F_{\omega_k} \sin\phi_i \tag{28}$$

Net xy-plane piston reaction force F_{RK_i} can be resolved into $x - y$ component, $F_{RK_{i_x}}$ & $F_{RK_{i_y}}$, which are obtained in terms of forces mentioned in equations (21), (22), (23), (24), (25), (26), (28) as:

$$F_{RK_{i_x}} = F_{\omega_{k_x}} + F_{TG_x}$$
$$F_{RK_{i_y}} = F_{\omega_{k_y}} + F_{sw_y} + F_{TG_y} \tag{29}$$

Contribution to motor torque from i_{th} pistion, M_{x_i} is given as:

$$M_{z_i} = R\cos\phi * F_{RK_{iy}} - R\sin\phi * F_{RK_{ix}} \tag{30}$$

Net torque M_x, would be summation all those contribution and can be expressed as:

$$M_z = \sum_{i=1}^{9} M_{z_i} \tag{31}$$

Motor speed is obtained by newton's second law as:

$$\omega(t) = \frac{1}{J}\int_0^t (M_z - T_{load})\, dt \tag{32}$$

2. 3 Sub component modeling

The component like internal pressure override relief valves (IPOR), shut off valves was modeled based on the pressure difference logic established on the port line pressure A, B which exceeds the critical pressures. charge pump is modelled as constant flow source. All interconnecting ducts has been lumped into a single volume, thus neglecting pressure drop in pipes. Considering the complexity level, the details of the modeling is not considered much.

2.4 Integrated Closed Loop Circuit

A generic cc circuit schematic diagram has been shown in figure (1). Flow directions through each hydraulic line, as indicated in the figure correspond to case when swash angle is positive, i.e. pump is discharging flow from port B and hence, side 1 is high pressure side. Since pressure loses in Intermediate pipes have not been considered, pipes and flow ducts on each side in cc circuit, has been lumped together as one volume, V_{side_1} and V_{side_2} on each side.

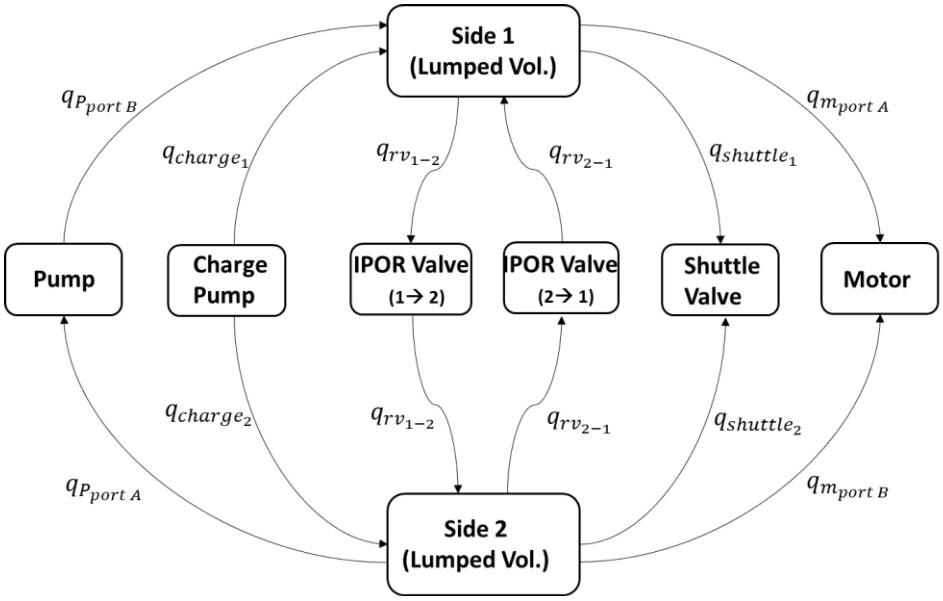


Fig. 11. Schematics of integrated CC circuit system model

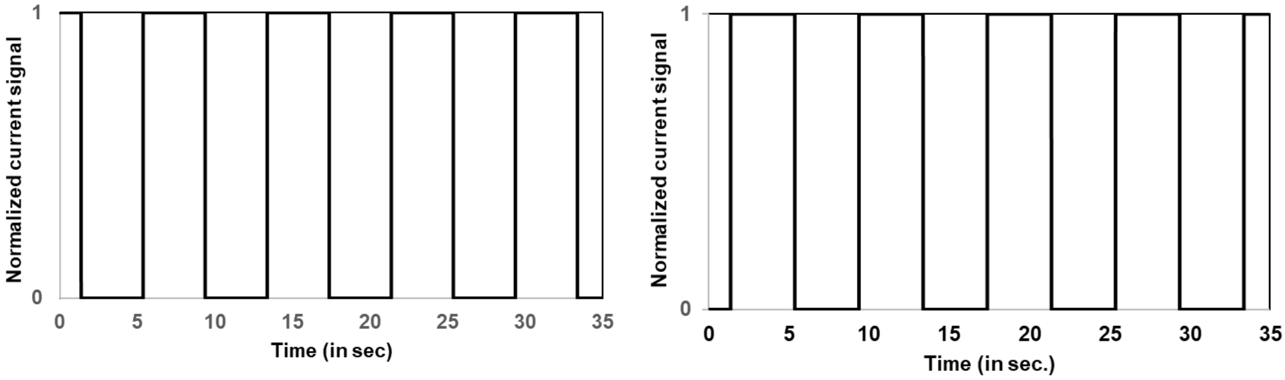
The same has been done for integrated circuit modelling as shown in figure (11). Pressure on each side is computed on solving pressure dynamics equation in each volume. Following same sign convention, flow entering the volume is considered positive and vice versa. As discussed in respective subsection, sign of flow coming from each sub-component has been carefully assigned to ensure system's bidirectionality.

3. Model Validation

Developed model has been validated using test data conducted for examining motor characteristics in closed loop circuit. 49 cc motor attached with flywheel, subjected to negligible external load torque, is connected with other cc circuit components. Motor speed is brought up to around 3000 rpm in one direction, back to zero and then, up to same speed in other direction. Predefined solenoid current input to electric proportional pump swash control is used to achieve this behavior. Pressure on each side of cc circuit, motor speed and torque are some of characteristics that has been recorded in real time test data. Major parameter pertaining to cc circuit components used in simulation has been listed in table (3).

Table 3: CC circuit parameters

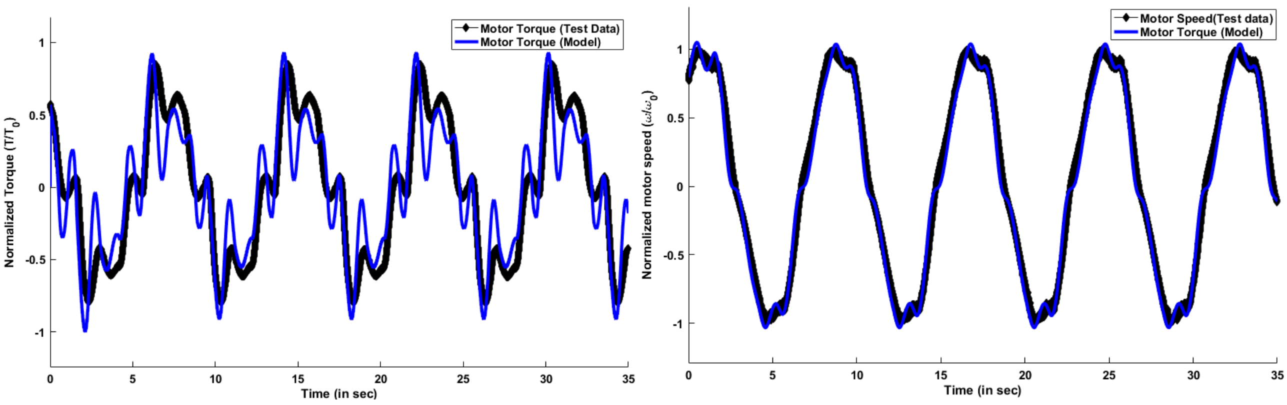
Parameters	Values
Main pump displacement	49 cc
Charge pump displacement	13.8 cc
Charge pump relief valve cracking pressure	16 bars
IPOR valve cracking pressure	320 bars
Shuttle valve cracking pressure	10 bars



(a) (b)

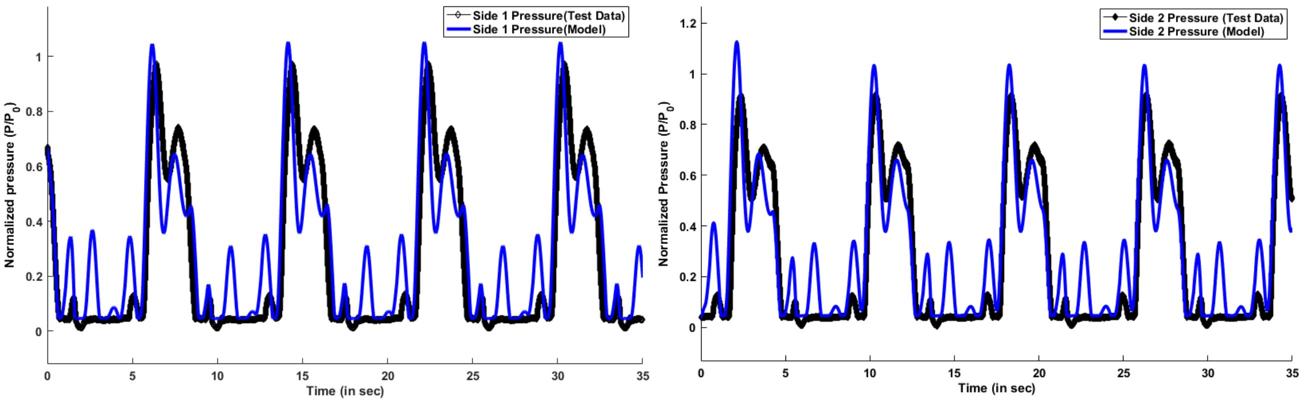
Fig. 12. Forward solenoid current signal (a) and reverse solenoid current signal (b)

Solenoid current signal input in forward and reverse as provided to pump swash control is shown in figure (12). Data has been normalized with nominal value of each parameter without any loss of trend. Figure (13) present plots of motor torque and speed in real time as predicted by model, compared with test data. Except for occasional overshoot in torque, we observe good correlation trend wise with test data. This is expected since frictional torque losses in the model has not been considered. Subsequently, same is observed for side 1 and side 2 pressure, P_1 and P_2 as shown in figure (14).



(a) (b)

Fig. 13. Normalized motor torque (a) and motor speed (b) validated with test data

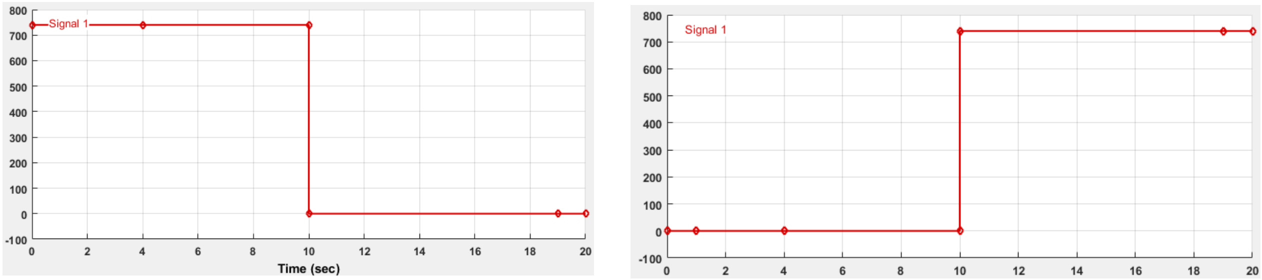


(a) (b)

Fig. 14. CC circuit side 1 pressure (a) and side 2 pressure (b) validated with test data

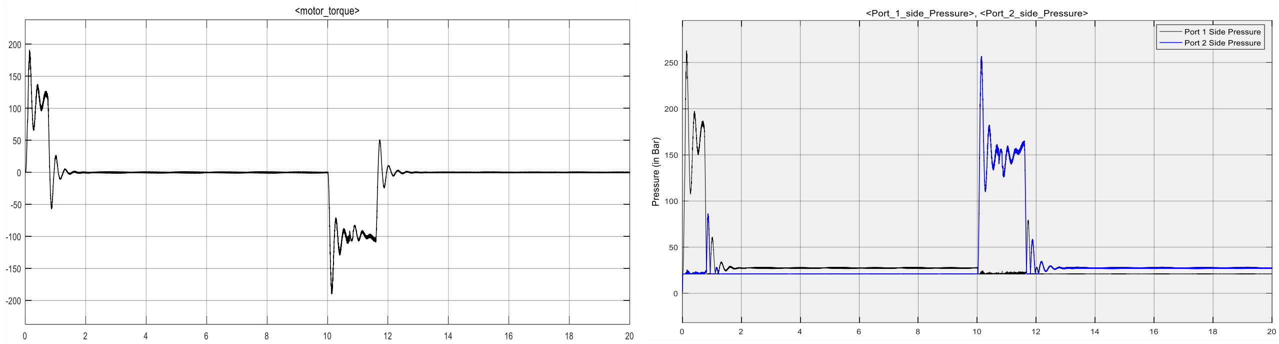
4. CTQ Prediction

To overcome load and frictional torque, motor needs high initial torque called starting or breakaway torque to start rotation. This high starting torque is physically generated when initially, pump is supplying full flow and motor is static, leading to pressure rise in high pressure line resulting in high torque produced. The same has been observed Qualitatively in figure (16). Forward and reverse solenoid signal to swash control is given in figure (15). Thus, Pump will supply flow in forward direction for some time and then flow direction will reverse. Initially motor is at rest, which leads to high pressure in port 1 side and torque as seen in figure (16a), (16b) respectively. The same is observed when pump reverses the flow, resulting in high port 2 pressure and torque in opposite sense.



(a) (b)

Fig. 15. Forward (a) and reverse (b) solenoid current



(a) (b)

Fig. 16. Motor Torque (a) and Port 1 Port 2 side pressure (b)

Pressure and torque ripple is inevitable phenomenon observed in axial piston machines. It is fluctuation in torque observed at designated operating condition. Figure (17) exhibits torque ripple as predicted by model in no load condition. This has been qualitatively verified with Ivantysyn et. al. [4].

Neutral pressure variation study has been done for the pump and is correlated with test data. During test, pump was brought to full stroke and full load condition with load pressure being around 1000 psi and then stroked back to neutral condition. This was performed in both senses i.e. forward-neutral-reverse and reverse-neutral-forward and averaged data was recorded. Simulation scenario has been set in similar fashion. Neutral pressure variation is observed while pump is going from forward to neutral as well as reverse to neutral. Two test cases have been simulated. In test case 1, pump angular speed is 710 rpm where as in test case 2, pump rotates at 1525 rpm. Figure (18a), (18b), (18c) and (18d) present model prediction of neutral pressure variation for test case 1 in forward to neutral and reverse to neutral cases respectively. Likewise, the same has been for test case 2 in figure (19). Averaged neutral pressure as obtained from model and as seen in test has been listed in table (4). The neutral pressure variation is within 30 psi for both test data as well as model prediction. Also, Model prediction accuracy can be said to be more than 95 percent given the available amount of test data. This can be further be refined if we consider a detailed and more complex motor torque loss model as given by Moslått [14].

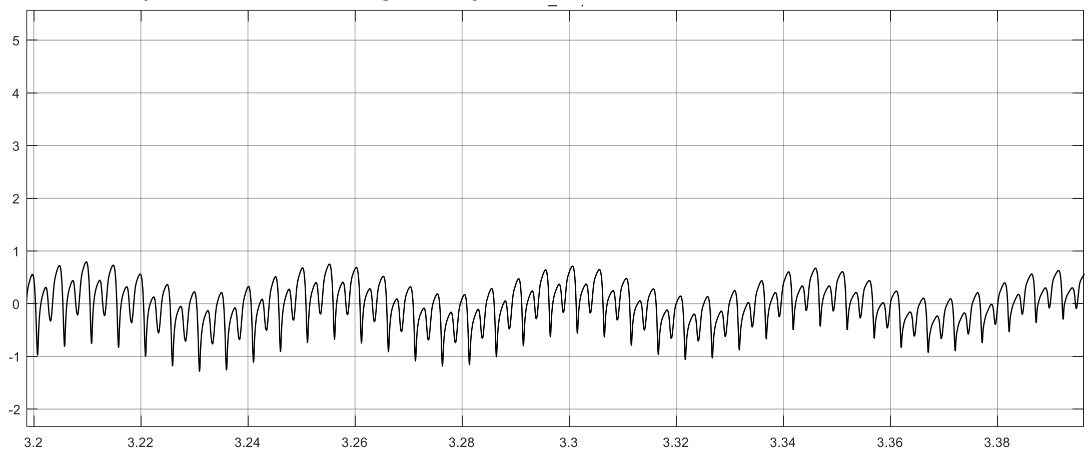


Fig. 17. Torque Ripple

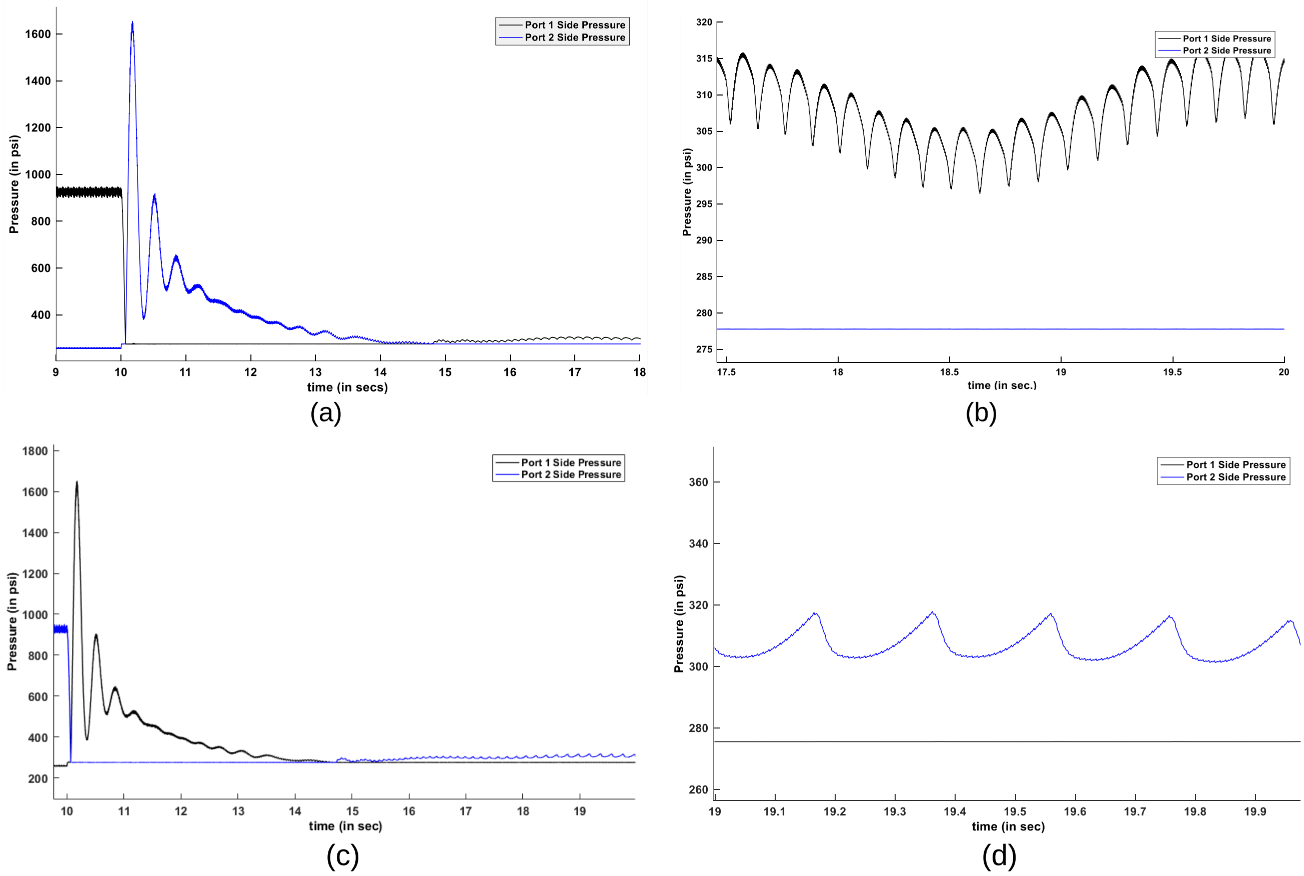


Fig. 18. Test Case 1: Model Prediction

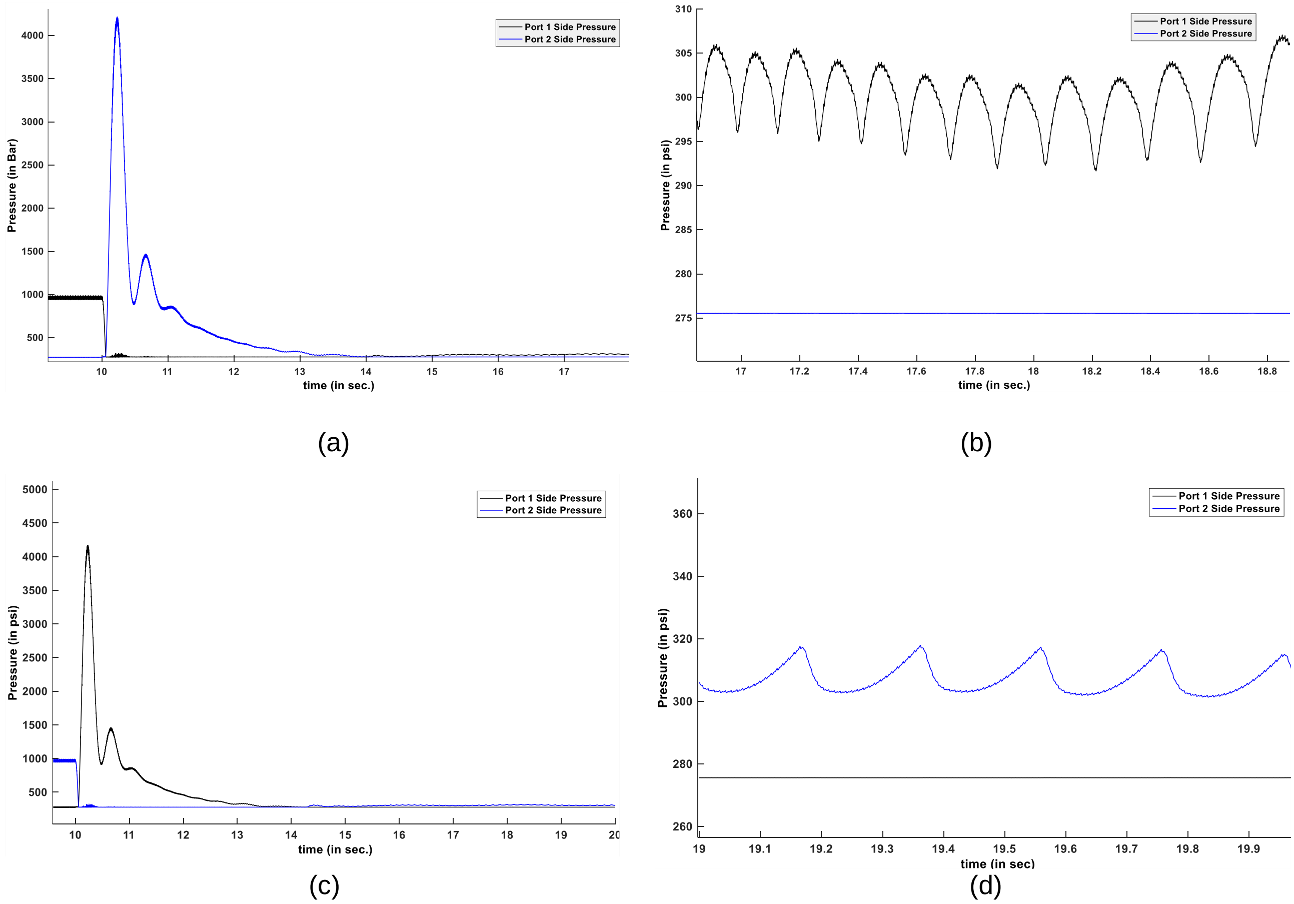


Fig. 19. Test Case 2: Model Prediction

Table 4: Neutral Pressure Correlation with Test Data

	Test Data-psi (avg.)	Model-psi (F to N)	Model-psi (R to N)
Test Case 1	28.6	28.5	29.5
Test Case 2	27.5	27.8	27.2

5. Conclusions

Current literature does not mention effects of system level interactions on closed-circuit piston pump's performance (e.g. Motor torque variation due to pump swash fluctuations), however individual details are available for few components. The major contribution of this study is the development of detailed mathematical model of closed-circuit pump with variable displacement pump, fixed displacement motor, swash controls by first principle physics knowledge and predictable accuracy on the neutral pressure variation. The experimental results are found to be consistent with the model predictions; hence the methodology developed here can be used for higher displacement machines and different applications of hydrostatic transmission. The model is useful for concept evaluation, component sizing and selection & optimization of system level performance. Future scope can be to refine the model further by inclusion of possible leakages in components, pressure drop across valves based on test data and friction dynamics for each interface.

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