Application based performance analysis of a priority flow divider valve using power hydraulic systems

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Abstract: Priority flow divider valve (PFDV) splits the flow in two different paths: Primary flow path and Secondary flow path. It can be used in twice applications: a system needs to perform dual functions against dual loads simultaneously and to obtain a stable flow by supplying a compensated flow from secondary flow line to the primary flow line using an energy storage device and a control valve. Two different hydraulic systems are identified to analyze the steady and dynamic performance of the PFDV. The first hydraulic system with PFDV is designed for automobile steering system with a load sensing control strategy in the presence of multiple actuators whereas second system is designed for wind turbine hydraulic power transmission to obtain stable power from it in the absence of generator. Both power hydraulic systems are modeled using bond graph technique and simulated in SYMBOLS SHAKTI software to analyze the PFDV performance.

Keywords: Fluid power system, Priority flow divider valve, Bond graph modeling, Dynamical system modeling, Wind turbine.

1.0 Introduction

Hydrostatic transmission system generally transmits power form one rotating element to another rotating element using a fluidic medium. It is capable to transmit higher power in a compact area. The well-known components for a hydraulic system are hydraulic pump, different control valves, accumulator and actuator or hydraulic motor. The overall efficiency of a hydraulic system is an important parameter and it is adversely affected by the leakages from its different components. The leakages are more with the increment of number of hydraulic components and hence decrease the overall efficiency of the hydraulic system. Therefore, an effective hydraulic system design may leads to the improvement of the overall efficiency of the system [1, 2]. When it is required to perform dual functions by a hydraulic power system, the engagement of the hydraulic components are generally more. Hence, the probability to effect the overall efficiency of the system is comparatively higher. To reduce the engagement of the hydraulic components or to improve the system design, a priority flow divider valve (PFDV) is a key component on a dual function deliverable hydraulic system.

PFDV performs a crucial role when two hydraulic circuits or two hydraulic actuators are functioned simultaneously. This is a very high precision hydraulic component and its primary port orifice area is adjustable as per the primary load, whereas secondary port allows excess flow to pass through it. Sometime the flow is split into predefined ratio [3, 4]. An auto regulated high precision flow divider/combined valve has been investigated by Fedoroff et al. in [5]. Authors have developed a linearized model of the valve and described its operation in detail. Generally, the flow divider valves are two types: rotary or gear type and spool-type flow divider valve. The rotary type flow divider valve provides better accuracy as compared to the spool-type flow divider valve. But the spool-type flow divider valves are more flexible regarding its design and manufacturing process. Also, it can easily integrate with other components [6]. In [7], a special split-spool type flow divider valve has been presented. The dynamic and static characteristics of the stated valve have been analyzed at various loading pressure. In [8], authors have developed an algorithm for the purpose of analyzing and optimizing the hydraulic components. The algorithm is based on Response Surface with Steepest-Descent method which provides advantages of Design Of Experiments techniques. The stated algorithm has been applied
on a two-way flow-divider spool valve. Normal flow divider valve splits the flow in two different actuators or circuits such that flow ratio does not depend on the load pressures. But in reality, it is very difficult to vary the flow ratio after manufacturing. To overcome the stated issue, a new design of flow divider valve with an adjustable flow ratio has been introduced in [9, 10]. The basic principle of the proposed valve is similar as typical flow divider valve. But the effective pressures or the forces acting on the spool ends are modified by the pilot stage. Recently, Coskun et al. designed a PFDV and investigated the input parameters by simulation using one dimensional simulation technique. Also, the simulation responses are validated experimentally [11]. In [12], an integrated hydraulic supply system is designed with the help of a PFDV. The PFDV assists to control the power supply to the power steering and braking system of a vehicle. Also, the PFDV is a critical part of the hydraulic system that is used in mining machineries. In [13], the PFDV is used in a typical power hydraulic circuit of a mining equipment to drive the twin hydro-motor simultaneously. Thereafter, the same power hydraulic system is considered for performing the fault detection, isolation, and prognosis [14]. Besides, few literatures are found regarding the use of the PFDV in wind power applications. In [15], PFDV has been used in wind turbine power transmission system for reducing the speed fluctuation of the pelton wheel. However, a similar type of proportional flow control valve has been used by Fan et al. in [16]. The stated proportional flow control valve divides the pump flow in two different paths and reduces the power fluctuation in wind turbine. This work is based on simulation, and the valve i.e. proportional flow control valve which is used to control the pump flow is quite different in working principle from PFDV. However, the purpose to use for both valves, i.e. PFDV and proportional flow control valve, are similar. Also, the manufacturing feasibility of the proportional flow control valve is highly challenged and not cost effective as per its design. Therefore, the flow divider valve can be used for both applications i.e. to perform dual functions against dual loads concurrently using a single hydraulic pump and to generate stable power in wind turbine using hydraulic power transmission system. As the stated valve is key equipment for these two identified applications, thus, it is highly important to describe the detail modeling and dynamic analysis of the proposed valve. The present study covers the detail analysis of the flow divider valve in both applications.

This study addresses the detail bond graph modeling of two simple power hydraulic systems with a PFDV. The first hydraulic system with PFDV is based on performing dual functions concurrently against dual loads whereas the second hydraulic system with PFDV is designed to obtain a stable power from a wind turbine against a variable input to the system. The systems are simulated to analyze the steady and dynamic performance of the PFDV by applying various unstable inputs using SYMBOLS SHAKTI software [17]. This study is application based and presented dual application scopes of the PFDV in signal platform. Bond graph is chosen for modeling the systems because it has unified multi-physics system modeling tool and is used for graphical representation of a physical dynamical system with power exchange among system components. Also, it is highly capable to interact multiple energy domains [18-22].

2.0 Physical model of the hydraulic system with PFDV

Power hydraulic systems transmit power using fluidic medium. It converts the mechanical power from an engine or an electric-motor to the hydraulic power by rotating the shaft of a hydraulic pump. Hydraulic pump supplies flow to the control valves that direct the same flow to the hydraulic actuators or hydraulic motor and convert the hydraulic power back to the mechanical power. In this study, two different open-loop hydraulic drive systems (refer Figs. 1 and 2) are identified to analyze the performance of the PFDV. The first hydraulic system (refer Fig. 1) is useful to perform two different functions concurrently using PFDV. Besides, another hydraulic system (refer Fig. 2) is suitable to supply constant flow whenever the input flow is of fluctuating nature. This stated system is preferable in wind turbine power transmission systems to produce a stable power.

PFDV are two types: spool type and the rotary or gear type. Spool-type PFDV is designed to deliver proportional flows from two outlets, or to deliver a priority flow from one outlet and excess flow
from another outlet. Besides, the rotary or gear type PFDV is differing from its actuation process and it has rotary spindle. The main components in a spool type PFDV are: housing with an inlet port and two outlet ports, a compression spring and an internal moveable spindle/spool. The spindle movement controls the flow rate through the primary and the secondary port of the PFDV. The cross-sectional view of the PFDV is shown in Fig. 3.

![Diagram of hydraulic system](image)

**Fig. 1:** Schematic diagram of hydraulic drive system with PFDV to perform dual functions simultaneously.

According to Fig. 1, a variable displacement hydraulic pump is used to supply a variable flow to the PFDV. The PFDV is connected with two different loads: primary load and secondary load of the steering system. The PFDV directs the pump flow towards the primary and secondary loads of the steering system. It controls the pump flow by controlling the movement of the spindle as per the loading condition of the system. Additionally, a pressure relief valve is incorporated between hydraulic pump and the PFDV to maintain the normal system pressure and hence used as a safety valve. The dynamic model of the steering system is not discussed in the present study as the focus of the study is to analyze the performance of the PFDV using steering mechanism. For more details about the dynamic model of the steering system is available in [23, 24].
Fig. 2: Schematic diagram of hydraulic drive system with PFDV and accumulator to obtain a stable speed.

The scheme to obtain a constant speed from the flywheel using a PFDV and an accumulator is shown in Fig. 2. In this scheme, a variable flow is supplied to the PFDV using a variable displacement hydraulic pump. The primary flow of the PFDV is adjusted and fixed at a threshold value as per the stable loading condition. If the supplied flow from the hydraulic pump is exceeded to maintain the stable loading, then the extra amount of flow supplies by the pump is allowed to pass through the secondary port of the tank.
PFDV. Thus, the excess of flow is stored into the accumulator in the secondary line of the system. Contrarily, when the supply flow from the pump to the PFDV is insufficient, then the primary flow is unable to supply the required flow to the hydraulic motor to maintain the constant speed of the flywheel. In such a situation, the control valve is received a signal from controller and thereafter it is enabled, and allows the accumulator to release the stored fluid to the hydraulic motor. Therefore, the stored fluid in the accumulator compensates the flow to the hydraulic motor and maintains a constant speed of the flywheel whenever the pump supply is insufficient. The stated power hydraulic system can replaces the mechanical power transmission in wind turbine application to obtain a stable power from it.

3.0 System modeling of the hydraulic systems with PFDV

3.1 Bond graph modeling of hydraulic drive system with PFDV to perform dual functions

This section presents the bond graph modeling of an open-loop hydraulic system with PFDV to perform dual function simultaneously. The bond graph model of this system (as presented in Fig. 1) is shown in Fig. 4. The developed dynamical model of the hydraulic system manages some assumptions which are as

1. Negligible fluid inertia,
2. Tank pressure is atmospheric,
3. Fluid properties does not varied with temperature,
4. Pump and motor leakages are considered only for study purpose,
5. The resistive and the capacitive effects are lumped wherever appropriate,
6. Hydraulic pump and motor specification can be changed as per the loading condition of the system.

The dynamical equation of flow through different valve ports, check valve etc. is generally a non-linear relation and it is formulated as

\[ Q_n = k_n A_n \sqrt{|\Delta P| \text{sign}(\Delta P)}, \quad (1) \]
where $A_n$ and $\Delta P$ are the port opening area and pressure difference across it, respectively; $k_n = C_d \sqrt{2\gamma \rho}$ is the flow constant and it depends upon the co-efficient of discharge ($C_d$) and the density of liquid ($\rho$) that flows through the system.

A variable speed of the electric motor helps to supply a variable flow from the hydraulic pump to the PFDV. The angular speed of the shaft of the hydraulic pump ($\omega_p$) is represented by a source of flow generalized bond graph element ($S_f_1$). The mechanical energy of the shaft is converted into the hydraulic energy by a transformer generalized bond graph element (TF) where volume displacement rate of the hydraulic pump ($D_p$) is assigned as the modulus of TF. The bulk stiffness of the working fluid ($\beta_p$) and the external leakage of the hydraulic pump ($R_{plkg}$) are represented by generalized bond graph capacitive element ($C_5$: $\beta_p$) and resistive element ($R_6$: $R_{plkg}$), respectively at $0_p$ junction. The measured pressure at the pump plenum is indicated by $P_p$ which can be computed from the definition of the bulk modulus of the flowing fluid and it is expressed as

$$\dot{P}_p = \frac{\beta_p}{\mathcal{A}_{sw}} \left\{ D_p \omega_p - \frac{P_p}{R_{plkg}} + \frac{(P_p - P_{pv})}{R_{line1}} \right\}, \quad (2)$$

where $\mathcal{A}_{sw}$ is the constant line volume which includes the average effective volume of gerotor lobes. The pipe resistance ($R_{line1}$) is represented by element $R_8$ at $1_s$ junction.

The $C_{11}$ element at $0_{pv}$ junction represents the bulk stiffness of the working fluid ($\beta_{pv}$) at PFDV plenum. The resistances of the primary port and the secondary port of the PFDV are modeled by resistive elements $R_{13}$ ($= R_{pfl}$) and $R_{16}$ ($= R_{sfl}$), respectively. Similarly, the primary load ($P_{pl}$) and the secondary load ($P_{sl}$) are expressed by source of effort elements $S_e_{18}$ and $S_e_{19}$, respectively. The PFDV spindle is travelled due to an effective force which is generated due to three different pressure forces such as pressure at PFDV plenum ($P_{pv}$), pressure due to primary load ($P_{pl}$) and pressure due to secondary load ($P_{sl}$). The transformer element (TF) which is connected between $1_s$ and $1_x$ junctions converts the pressure force into the mechanical force by assigning the area of spindle ($A_{sp}$) as the modulus of TF. The displacement of the spindle ($x_{spdl}$) is detected by a displacement sensor at $1_x$ junction. The flow resistances of the PFDV are varied as per the spindle displacement of the PFDV. The mass ($m_{sp}$) and the stiffness ($K_{sp}$) of the spindle spring are expressed by $I_{30}$ and $C_{28}$ elements at $1_x$ junction, respectively. Also, the viscous resistance of the spindle ($R_{spdl}$) is expressed by $R_{29}$ in bond graph model. The pressure at PFDV plenum ($P_{pv}$) is expressed as

$$\dot{P}_{pv} = \frac{\beta_{pv}}{\mathcal{A}_{swpv}} \left\{ \frac{(P_p - P_{pv})}{R_{line1}} - Q_{pfl} - Q_{sfl} - A_{sp}v_{spdl} - A_{sp}v_{spdl} \right\}, \quad (3)$$

where $\mathcal{A}_{swpv}$ and $v_{spdl}$ are the constant line volume and the linear velocity of the PFDV spindle, respectively. The PFDV splits the pump flow into two different paths i.e. primary and secondary path. The flow through the primary and secondary ports of the PFDV depend on the displacement of the valve.
spindle \((x_{\text{spdl}})\) which is estimated by solving the ordinary differential equation i.e. Eq. 4 which is obtained from the bond graph model (refer Fig. 2).

\[
V_{\text{spdl}} = \frac{(P_{pv} - P_{pl}) A_{sp} + P_{pv} A_{sr} - P_{pl} A_{pr} - K_{sp} x_{\text{spdl}} - R_{\text{spdl}} V_{\text{spdl}})}{m_s},
\]

where \(A_{sp}\), \(A_{sr}\) and \(A_{pr}\) are the area of the PFDV spindle, area towards secondary port and area towards primary port of the PFDV, respectively. The stiffness and the mass of the spring are denoted by \(K_{sp}\) and \(m_s\), respectively. The viscous resistance works against the movement of the spindle of the PFDV and it is denoted by \(R_{\text{spdl}}\).

In PFDV, the primary flow is adjusted as per the predefined ratio or as per the primary loading condition and the other excess flow is allowed to pass through the secondary port of the PFDV. The primary flow of the PFDV \((Q_{\text{pfl}})\) is expressed as

\[
R_{13} : Q_{\text{pfl}} = \begin{cases} 
   k_{d,\text{pfl}} A_{\text{pfl}} \sqrt{|P_{pv} - P_{pl}|} \text{sign}(P_{pv} - P_{pl}) & \text{when} \left( x_{\text{spdl}} \leq x_i \right) \\
   k_{d,\text{pfl}} A_{\text{max,pfl}} \sqrt{|P_{pv} - P_{pl}|} \text{sign}(P_{pv} - P_{pl}) & \text{when} \left( x_{\text{spdl}} > x_i \right)
\end{cases},
\]

where \(k_{d,\text{pfl}} = C_{d,\text{pfl}} \sqrt{2/\rho}\) is the flow constant. The \(C_{d,\text{pfl}}\) is the co-efficient of discharge through primary port of PFDV, \(x_i\) is the desired spindle displacement to supply the necessary flow to operate the primary circuit successfully. The \(A_{\text{pfl}}\) and \(A_{\text{max,pfl}}\) are the instantaneous opening area and maximum opening area of the primary port of the PFDV, respectively. The \(A_{\text{pfl}}\) depends on \(A_{\text{max,pfl}}\), size of the indenter i.e. diameter of the indenter \((d)\), angle of the indentation \((\alpha)\) and the displacement of the valve spindle etc [25]. It can be expressed as

\[
A_{\text{pfl}} = A_{\text{max,pfl}} \left[ \frac{\alpha}{\pi} - \frac{\sin 2\alpha}{2\pi} \right].
\]

\[
\alpha = \cos^{-1} \left( 1 - \frac{2x_{\text{spdl}}}{d_i} \right),
\]

Similarly, the secondary flow through the PFDV \((Q_{\text{sfl}})\) is expressed as

\[
R_{16} : Q_{\text{sfl}} = C_{d,\text{sfl}} A_{\text{sfl}} \sqrt{2/\rho \left| (P_{pv} - P_{sl}) \right|} \text{sign}(P_{pv} - P_{sl}) \text{ when} \left( x_{\text{spdl}} > x_i \right)
\]

where \(C_{d,\text{sfl}}\) is the coefficient of discharge through the secondary port of the PFDV. \(A_{\text{sfl}}\) is the instantaneous flow area through secondary port of the PFDV. It is the function of maximum flow area through the secondary port of the PFDV \((A_{\text{max,sfl}})\), displacement of the spindle towards secondary port \((y_s)\), size of indenter and angle of indentation [22]. The \(y_s\) is same as the excess displacement of the spindle i.e. \(y_s = x_i - x_{\text{spdl}}\). Therefore, the \(A_{\text{sfl}}\) is expressed as

\[
A_{\text{sfl}} = A_{\text{max,sfl}} \left[ \frac{\alpha}{\pi} - \frac{\sin 2\alpha}{2\pi} \right].
\]
The bond graph model derives the state-space equations of the physical hydraulic system which are given as Eq. (11–13). These are derived using a well-known computational causality method and step by step algorithm [18–22]. The storage elements having lumped single-port (C and I elements) with integral causality grants one coupled differential equation with first order. Besides in multi-port elements which possess the integral causality, the numbers of state equations are same as the number of bonds connected to it. Therefore, total number of generated differential equations is equal to the sum of integrally causalled bonds associated with the bond graph model. The C and I elements which are integrally causalled are related with generalized charge for electrical domain or generalized displacement for mechanical domain, i.e. \( q = \int f dt \) and generalized momentum, i.e. \( p = \int edt \), respectively. The \( e \) and \( f \) are the generalized effort and flow, respectively. Here, the mathematical equation is systematically and algorithmically derived by using SYMBOL SHAKTI software [17].

\[
\alpha = \cos^{-1}\left(1 - \frac{2y}{d}\right),
\]

(10)

\[
\alpha_p D_p - \frac{\dot{P}_p}{\beta_p} - \frac{P_p - P_{pv}}{R_{plkg}} = 0
\]

(11)

\[
\frac{\left(P_p - P_{pv}\right)}{R_{line}1} - Q_{pfli} - Q_{fpl} - \left(A_{br} + A_{sp}\right)v_{spdl} = 0
\]

(12)

\[
\left(P_{pv} - P_{pl}\right)A_{sp} + P_{pv}A_{br} - P_{pl}A_{pr} - K_{sp}v_{spdl} - R_{spdl}v_{spdl} - m_{spdl} = 0
\]

(13)

3.2 Bond graph modeling of the hydraulic drive system with PFDV and accumulator to obtain stable speed

The bond graph model of the physical hydraulic drive system with PFDV and accumulator (refer Fig. 2) to obtain a stable speed is shown in Fig. 5. This bond graph model is similar to the Fig. 4, except the additional modeling of the check valve, accumulator, control valve, hydraulic motor, and the flywheel.

Fig. 4: Bond graph model of the hydraulic drive system with PFDV and an accumulator, to obtain stable speed
The accumulator stores excess energy and release the same whenever the pump supply is insufficient to compensate the demand flow as per the load. The accumulator is modeled at 1_{ac} junction by the $Se_{32}$ element. It represents the final pressure ($P_2$) that is developed by the accumulator. The accumulator follows the non-linear relationship between pressure and the volume. It is assumed that the heat loss through the accumulator is negligible. Therefore, the pressure-volume relationship should be reversible adiabatic and it is expressed as

$$Se_{32} : P_2 : P_{\text{accu}} = \frac{P_1^\gamma}{(\gamma - 1) \Delta \gamma},$$

where $\gamma$ is the specific heat capacity ratio of air; $\Delta \gamma$ is the contemporary volume displacement which is obtained from integration of the flow output from the flow detector $S_{f19} (= Q_{19})$.

The control valve is modeled at 1_{cv} junction. The resistance of the control valve port is represented by a resistive element $R_{34} (= R_{cv})$. The port opening area of the control valve is controlled as per the speed of the flywheel. The control valve is in open condition only when the flywheel speed is lower than its threshold speed. This threshold speed is fixed as per the load of the system. Therefore, the flow through the control valve ($Q_{cv}$) is expressed as

$$R_{34} \cdot Q_{cv} = \Psi_0 C_{d_{cv}} a_{cv} \sqrt{\frac{2|\Delta P|}{\rho}},$$

where $C_{d_{cv}}$ and $a_{cv}$ are the co-efficient of discharge and the port opening area of the control valve, respectively; $\Psi_0$ is the On/Off controller output which checks the flywheel speed ($\omega_{fw}$) against the constant threshold speed ($\omega_{th}$). It is expressed as

$$\Psi_0 = \begin{cases} 1 & \text{if } \omega_{fw} \leq \omega_{th} \\ 0 & \text{if } \omega_{fw} > \omega_{th} \end{cases},$$

The primary flow and the secondary flow of the PFDV are combined at 0_{pl} junction. The bulk stiffness of the working fluid at the same junction is denoted by $C_{36} (= \beta_j)$, whereas the pressure ($P_j$) is measured by another sensor and it is represented by an effort detector ($D_{e48} \). The $P_j$ is computed from the definition of bulk modulus of the flowing fluid and it is expressed as

$$P_j = \frac{\beta_j}{\gamma_j} \left[ Q_{pfl} + A_p V_{spdl} + A_p V_{spdl} + Q_{cv} - \frac{(P_j - P_m)}{R_{line2}} \right],$$

where $\gamma_j$ and $\beta_j$ are the are the constant line volume and the bulk modulus of the working fluid at the junction of the primary and the secondary flow of the PFDV; The pipe resistance ($R_{line2}$) is represented by $R_{38}$ at 1_{l} junction.

The hydraulic motor is modeled at 0_{m} junction in the bond graph model. At the same junction, the external leakage of the hydraulic motor ($R_{mlkg}$) and the bulk stiffness of the working fluid at motor plenum ($\beta_m$) are represented by the resistive element ($R_{43}$) and capacitive element ($C_{40}$), respectively. The measured pressure at the motor plenum is indicated by $P_m$ and it is obtained from the definition of bulk modulus of the flowing fluid, refer Eq. (18)
\[
\dot{p}_m = \beta_m \left\{ \left( \frac{P_j - P_m}{R_{\text{line2}}} - \frac{P_m}{R_{\text{mlkg}}} - D_m \omega_{fw} \right) \right\},
\]

where \( \forall_m \) and \( \beta_m \) are the constant line volume and the bulk modulus of the working fluid at the motor plenum. The hydraulic energy of the motor is converted into the mechanical energy by a transformer element (TF), where the modulus of TF is assigned as the volume displacement rate of the hydraulic motor \( (D_m) \). The mechanical energy or the torque which is produced at the shaft of the motor helps to manage the angular rotation of the flywheel. The moment of inertia \( (J_{fw}) \) and the viscous frictional resistance \( (R_{fw}) \) of the flywheel are represented by the inertial element \( I_46 \) and the resistive element \( R_{47} \), respectively. The angular speed of the flywheel \( (\omega_{fw}) \) is measured from the speed sensor which is represented by flow dictator (Df45) in the bond graph model. Hence \( \omega_{fw} \) is expressed as

\[
\omega_{fw} = \frac{1}{J_{fw}} \left\{ P_m D_m - R_{fw} \omega_{fw} \right\}.
\]

The state equations of the bond graph model i.e. Fig. 4 are obtained with the same conditions as that of the state equations obtained from the bond graph model of Fig. 3. The state equations i.e. Eqs. 11-13 are repeated whenever the state equations are derived from the bond graph model of Fig. 4 except two parameters such \( P_{pl} \) and \( P_{sl} \) are replaced by \( P_j \) and \( P_{accu} \), respectively. The other state equations for Fig. 4 are given as

\[
Q_{pfj} + Q_{cv} + A_{pr} v_{spdl} + A_{sp} v_{spdl} - \left( \frac{P_j - P_m}{R_{\text{line2}}} - \frac{\dot{P}_j}{\beta_j} \right) = 0
\]

\[
\left( \frac{P_j - P_m}{R_{\text{line2}}} - \frac{P_m}{R_{\text{mlkg}}} - D_m \omega_{fw} - \frac{\dot{P}_m}{\beta_m} \right) = 0
\]

\[
P_m D_m - R_{fw} \omega_{fw} - J_{fw} \dot{\omega}_{fw} = 0
\]

In this study, it is considered that fluid properties do not vary with temperature. Therefore, the bulk modulus of the working fluid is constant. Hence, \( \beta_p = \beta_{fw} = \beta_j = \beta_m \). Also, the leakages through hydraulic pump and motor are considered as same i.e. \( R_{plkg} = R_{mlkg} \).

4.0 Result and Discussion

The state equations i.e. Eqs. (11-13), of the open-loop hydraulic system with PFDV and the Eqs. (11-13 and 20-22) of the hydraulic drive system with PFDV and accumulator are obtained from bond graph models Fig. 4 and Fig. 5, respectively. Thereafter, the same equations are solved in SYMBOL-SHAKTI software. This software contains different modules such as Graphical model builder, Simulation workbench, Control theoretical analysis tool, FDI model builder, etc. In the present study, the Graphical model builder module is used to generate the state equations by modelling the bond graph of the systems whereas other module, i.e. Simulation workbench is used to simulate the bond graph models. The Simulation workbench module is embedded with various integration techniques such as Runge–Kutta Gill, Runge–Kutta Fehlberg, Gear’s Predictor Corrector, etc, to solve the state equations [26]. In this present simulation study, fifth order Runge–Kutta Gill method is used to solve the differential equations. The precision of the simulation is improved by adjusting the step size in time stepping in such a way that relative truncation error should be lesser. In present simulation study the error limit is chosen as \( 5 \times 10^{-6} \).
The numerical values of the simulation parameters are presented in Table I. The values of the parameters related to PFDV are taken from experimentally validated data given in the article [13, 14] and other parameters values are used therein are considered in the present study.

**Table I: Simulation parameters**

| Parameter                                                                 | Value                        |
|---------------------------------------------------------------------------|------------------------------|
| Volume displacement rate of the hydraulic pump ($D_p$)                    | $1.91\times10^{-6}$ m$^3$/rad |
| Bulk stiffness of the working fluid ($\beta_p = \beta_{pv} = \beta_j = \beta_m$) | $8\times10^{12}$ N/m$^2$     |
| Pump and motor leakages ($R_{plkg} = R_{mlkg}$)                          | $1\times10^{-6}$ N.s/m$^3$   |
| Pipe resistance ($R_{line1}$)                                            | $1.8\times10^7$ N.s/m$^3$    |
| Constant line volume ($\mathcal{V}_{sw}$)                                | $1\times10^{-6}$ m$^3$       |
| Area of the PFDV spindle ($A_{sp}$)                                      | $5\times10^{-5}$ m$^2$       |
| Area towards primary port of the PFDV ($A_{pr}$)                         | $2\times10^{-3}$ m$^2$       |
| Area towards secondary port of the PFDV ($A_{pr}$)                       | $5\times10^{-3}$ m$^2$       |
| Mass of the spring on PFDV ($m_i$)                                       | $2\times10^{-3}$ kg          |
| Stiffness of the spring on PFDV ($K_{sp}$)                               | $1\times10^7$ N/m            |
| Viscous resistance on PFDV spindle ($R_{spll}$)                           | $2\times10^4$ N.s/m         |
| Co-efficient of discharge through primary port of PFDV ($C_{d_{pfl}}$)   | 0.6                          |
| Maximum opening area of the primary port of PFDV ($A_{max_{pfl}}$)       | $3\times10^{-6}$ m$^2$       |
| Diameter of the indenter ($d_i$)                                         | $3\times10^{-1}$ m           |
| Density of the working fluid ($\rho$)                                    | $870$ kg/m$^3$               |
| Desired spindle displacement ($x_i$)                                     | $3.5\times10^{-7}$ m         |
| Coefficient of discharge through the secondary port of the PFDV ($C_{d_{sfl}}$) | 0.6                         |
| Maximum flow area through the secondary port of the PFDV ($A_{max_{sfl}}$) | $2\times10^{-6}$ m$^2$       |
| Initial pressure of the accumulator ($P_1$)                              | $63.7\times10^6$ N/m$^2$     |
| Initial volume of the accumulator ($\mathcal{V}_1$)                      | $630\times10^{-2}$ m$^3$     |
| Heat capacity ratio of air ($\gamma$)                                    | 1.4                          |
| Co-efficient of discharge of the control valve ($C_{d_{cv}}$)            | 0.6                          |
| Port opening area of the control valve ($a_{cv}$)                         | $70.7\times10^{-5}$ m$^2$    |
| Pipe resistance ($R_{line2}$)                                            | $1.8\times10^{10}$ N.s/m$^5$|
| Constant threshold speed of the flywheel ($\omega_{th}$)                 | $150$ rad/s                  |
| Volume displacement rate of the hydraulic motor ($D_m$)                  | $1.1057\times10^{-6}$ m$^3$/rad |
| Moment of inertia of the flywheel ($J_{fw}$)                             | $1.02\times10^{-3}$ kg.m$^2$ |
| Viscous frictional resistance of the flywheel ($R_{fw}$)                 | $25\times10^{-3}$ N.m.s/rad  |

The simulation responses of the first system i.e. open-loop hydraulic drive system with PFDV are presented in Figs. 6 and 7, whereas the simulation responses for second system i.e. hydraulic drive system with PFDV and accumulator are shown in Fig. 8.
Figure 6a shows that the hydraulic pump is operated with a variable speed. The speed of the hydraulic pump ($\omega_p$) is varied from 180 rad/s to 36 rad/s. The corresponding pressure at the pump plenum is varied 5.06 MPa to 1.31 MPa (refer Fig. 6b). To operate the primary circuit, the desired displacement of the spindle ($x_i$) is taken as $3\times10^{-3}$ m (refer Fig. 6c). The spindle displacement is the major criteria whether the flow is passed through the secondary port of the PFDV or not. When the spindle displacement of the PFDV ($x_{spdl}$) is more than $x_i$, the secondary port of the PFDV is enabled and hence it allows the excess of flow to pass through it. From Fig. 6c, when time ($t$) is less than 1.001 s, the spindle displacement is more than desired displacement i.e. $3\times10^{-3}$ m as a result both ports, i.e. primary and secondary ports, are in active modes and allow flows through these (refer Fig. 6d). In subsequent time period between 1.001 s to 2.001 s, the spindle displacement ($x_{spdl}$) is lesser than its desired displacement, as a result only primary port of the PFDV is enabled and flow is passed through it. At the same time the secondary port of the PFDV is disabled and it does not allow the flow to pass through it (refer Fig. 6d).

Similarly, a sinusoidal speed is supplied to the pump as shown in Fig. 7a. The speed is varied from 20 rad/s to 180 rad/s. The corresponding minimum and maximum pressure at pump plenum are 0.94 MPa and 5.07 MPa, respectively (refer Fig. 7b). When the $x_{spdl}$ is less than $x_i$, the secondary port is closed and no flow passed through it. But when the $x_{spdl}$ is larger than $x_i$, both ports are opened and hence both primary and secondary circuits are in functioning mode (refer Figs. 7c and 7d).
The responses of the hydraulic drive system with PFDV and accumulator are presented in Fig. 8. The objective of the stated system is to obtain a stable speed from the flywheel whenever the input speed is of fluctuating manner. Figure 8a shows the speed of the hydraulic pump ($\omega_p$) is varied from 160 rad/s to 36 rad/s, whereas the output speed or the flywheel speed ($\omega_{fw}$) is obtained as stable and its numerical value is about 150 rad/s (refer Fig. 8c). This power hydraulic scheme can be useful to transmit the power from the turbine rotor to the generator in wind turbine when a stable power supply is essential against a fluctuating wind speed supply. As the $\omega_p$ is variable, the supply flow from the hydraulic pump to the PFDV is also variable nature and hence the flow through the primary and secondary port of the PFDV is also unstable nature. The flow through the primary port of the PFDV ($Q_{pfl}$) is controlled as per the loading condition of the system. The spindle displacement of the PFDV is adjustable as per the load of the system to supply a desired flow to the hydraulic motor through the primary port of the PFDV. In this system, the threshold value of the spindle displacement ($x_i$) is about $3.5 \times 10^{-3}$ m. This threshold value or the desired value of the spindle displacement is decided as per the loading condition of the system. If the instantaneous spindle displacement ($x_{spdl}$) is more than the desired spindle displacement ($x_i$) or $x_{spdl} \geq x_i$, then the excess spindle displacement helps to enable the secondary flow through the PFDV. The secondary flow is the excess flow at that corresponding load and it is stored into the accumulator in the secondary flow line. The same situation is observed in time durations $t = 0-1s$, $2-3s$ and $4-5s$ (refer Fig. 8b and Fig. 8c). In counter situation, when the pump supply is comparatively low, the spindle displacement ($x_{spdl}$) of the PFDV is also lower than its desired spindle displacement ($x_i$) i.e. $x_{spdl} \leq x_i$. As a result the pump flow is only passed through the primary port of the PFDV and hence there is no flow through the secondary port of the PFDV. The stated situation is observed whenever time durations $t = 1-2s$ and $3-4s$ (refer Fig. 8b and Fig. 8c). In this time periods $t = 1-2s$ and $3-4s$, the primary flow of the PFDV is unable to manage the desired flow supply to the hydraulic motor in order to maintain a stable speed of the flywheel. To compensate the desired flow to the hydraulic motor and to obtain a stable speed of the flywheel, the accumulator is started to discharge and hence, supplies an additional flow to the hydraulic motor.
motor through the secondary flow line (refer Fig. 8e). This secondary flow is controlled by the control valve. The operation of the control valve is controlled by a controller which receives a signal from the speed sensor of the flywheel. In order to maintain a stable speed of the flywheel, when the supplied flow to the hydraulic motor is less than its desired value, the instantaneous speed of the flywheel is decreased from its threshold speed \( \omega_{\text{th}} \) and at that instant, the control valve is enabled and allows flow through it (refer Fig. 8e). The threshold speed of the flywheel \( \omega_{\text{th}} \) depends on the loading condition of the system. In this system, the threshold speed of the flywheel is about 150 rad/s. The control valve is in opened condition during time period \( t = 1-2s \) and \( 3-4s \) (refer Fig. 8d). The pump pelumn pressure of the system is varied from 10.2 to 7 MPa whereas the motor pelumn pressure is almost constant and it is about 6.5 MPa.

![Graphs](image)

**Fig. 8** (a) Pump speed (b) Flow rate through primary and secondary port of the PFDV (c) Spindle displacement of the PFDV (d) Hydraulic pump and motor pelumn pressure (e) Time response of Accumulator flow, Control valve flow and flywheel speed for fixed threshold flywheel speed.

### 5.0 Energy and power loss through the PFDV

The responses of power and energy loss through the ports of the PFDV are presented in Fig. 9 and Fig.10 for both applications. The energy is dissipated as heat energy and it is associated with pressure head loss. To overcome the head loss, an additional power is needed to the system, and it is estimated as a product
of flow rate ($Q$) and the pressure drop across PFDV ports ($\Delta P$) the through it. Moreover, an integration of the power loss ($P_l$) provides the energy loss ($E_l$) of the system. The power and energy loss are expressed as

$$P_l = (\Delta P \cdot Q)_{\text{primary port of PFDV}} + (\Delta P \cdot Q)_{\text{secondary port of PFDV}}$$

$$E_l = \int P_l \, dt$$

![Graph](image)

**Fig. 9:** Energy and power loss in hydraulic drive system with PFDV to perform dual functions simultaneously.

**Fig. 10:** Energy and power loss in hydraulic drive system with PFDV to obtain a stable output.

The power loss through the PFDV in both applications is maximum when the system is in transition mode, means during the variation of the input of the system. The power loss through the PFDV is higher when it is connected with dual loads for performing dual functions. Similarly, the energy loss is also higher when the PFDV connected with similar systems. However, the power and energy loss through the PFDV may vary with the variation of the load capacity of the system.

### 6.0 Conclusions

This article presents a simulation based work for the performance analysis of a three ports PFDV. This valve is the key equipment in power hydraulic system and it can be used with two different power hydraulic schemes to transmit the power. Firstly, the scheme which is performed dual functions simultaneously with separate loading conditions and the second scheme which is used to obtain a stable output against a stable demand whenever the input parameter is of fluctuating nature. The prior scheme is mostly used in automobile steering systems with a load sensing control strategy in the presence of multiple actuators whereas the later scheme can be used in wind turbine application to obtain a stable power as the input wind velocity is of fluctuating nature due to cyclone, storm etc. This scheme may reduce the probability of load-shedding at the power supply point of the wind turbine by the help of a PFDV, an accumulator and a control valve.

The following findings are obtained from the present study

- The first proposed scheme (refer Fig. 1) is implemented when it is required to operate dual hydraulic circuits simultaneously with different loading conditions. The stated system successfully divides the pump flow in two different paths in a specified ratio by the use of PFDV. The flow rate ratios through these two paths are adjustable as per the loading conditions.
- The former proposed scheme (refer Fig. 2) can be suitable whenever it is required a stable output from a variable input to the system. This scheme is successfully operated by the help of a PFDV, an accumulator and a control valve.
- The power and energy loss through the PFDV is higher when it is connected with the dual loads for performing dual functions as compare to the other hydraulic drive system used in wind turbine to obtain stable output.

The second power hydraulic scheme can be executed in wind turbine application to transmit the power from turbine rotor to the generator. The present work is based on simulation analysis. Therefore, it
Compliance with ethical standards

Conflict of interest: The authors declare that they have no conflict of interest.

Declaration:

1. Availability of data and materials: The datasets generated during and/or analysed during the current study are available from the corresponding author on reasonable request.

2. Competing interests: Not applicable

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