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Nonlinear Modeling and Analysis of Direct Acting Solenoid Valves for Clutch Control

The purpose of this paper is to develop a comprehensive nonlinear model of a typical direct acting solenoid valves utilized for clutch control in wet dual clutch transmissions. To do so, mathematical models of the integrated electrohydraulic solenoid valve and wet clutch piston assembly are developed in the SIMULINK environment of MATLAB. Through simulation the operating characteristics of the control valve are analyzed, demonstrating that the valve achieves dual functionalities of high flow and accurate pressure control depending on demands. This is realized through the designed force balancing of the valve spool. The dependency of the system to system variables on input pressure and the influence of air content on dynamic response of the valve are investigated. The resilience of output pressure is demonstrated to these variables, indicating strong system reliability. Finally, the model is then validated using in situ experimental testing on a powertrain test rig. The comparison of experimental and simulated results for steady state pressure as well as step and ramp input responses demonstrate good agreement.
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1 Introduction

Hydraulic control systems developed for clutch actuation in dual clutch transmissions require precise pressure control and rapid response to control demands. These attributes ensure high quality shift control is achieved without the loss of tractive load to the road. To overcome these issues compact and direct acting solenoids have been developed, see Ref. [1]. Such a system must be capable of providing both precise pressure control and rapid fill time, achieved through control of pressure balances on the spool, with high pressure variation increasing the flow rate. Such precise control is generally required for rapid and accurate clutch filling, alternative methods include [2], and rapid response to input signals during shift transients [3]. This is particularly important for power shifting automotive transmissions where vehicle jerk from abrupt shifting and torque hole from excessive controller delay [3] both contribute to reduced shift quality. Therefore, the purpose of this paper is to develop a detailed model of the electrohydraulic solenoid valve commonly used in dual clutch transmission powertrains for investigation of its operating characteristics.

Electrohydraulic control systems are inherently nonlinear, with strong dependency on air content in the hydraulic fluid [4], internal spool forces, including jetting through valve orifice [5] having strong influences on valve behavior. One major consideration in hydraulic power systems is the reduction of stick–slip in the valve spool as it transits between stationary and in motion, Owen and Croft [6] suggested a method of rotating the spool to reduce stick–slip issues; however, a more popular method used in industry is to integrate a dither frequency in the control signal to produce a small oscillation around a neutral point, minimizing stick–slip. For this paper it is assumed that stick–slip is effectively overcome through the use of a dither frequency and is therefore not included in the model.

There have been various alternative methods developed for the modeling and analysis of hydraulic control systems in general, and direct acting solenoid valves for clutch control. In Ref. [7], a method is proposed for the simplification of hydraulic systems with lookup tables to reduce computation demand, this method required extensive experimental analysis to implement. In Ref. [7], the same valve as is the topic of this paper is modeled using a series of transfer functions, linearizing the system to a degree. This method has the advantage of significantly reducing the computational demand of the model, but strong nonlinearities, such as the opening or closing of individual ports cannot be easily modeled [7]. This issue is overcome in Ref. [9] using separate models for different valve conditions. In such cases, it is expected that the modeling delivers a computationally compact system for control implementation; however, it is not as useful for detailed system analysis and evaluation. Nonlinear analysis in Ref. [10] is performed for pilot control relief valves, providing an extensive hydraulic model. This method develops an extensive mathematical model of the physical systems through the application of numerous well known fluid equations, i.e., orifice flow, compressibility, and jetting forces, coupling the motion of the spool valve to fluid dynamics. This method produces a strongly nonlinear model of the valve, and is generally numerically intensive to apply for transient simulations.

Modeling, through various means and techniques, of electrohydraulic valves has become an important step in the analysis and control of wet clutches [11,12], and dry clutches and synchronizers [13]. It has therefore become an important component for the evaluation and rapid development of transmission designs. The strong nonlinear dynamics of this type of valve has resulted

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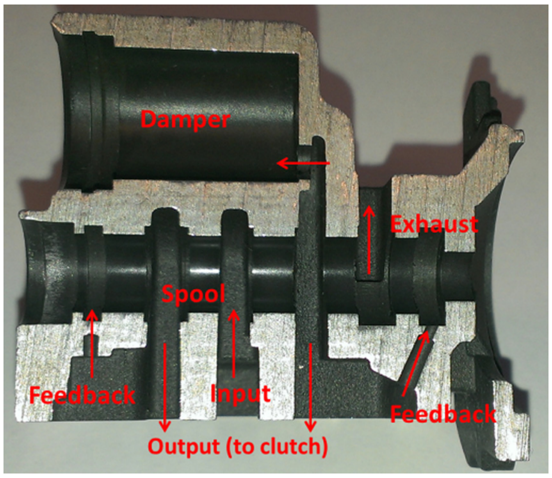

Fig. 1 Section detail of the compact direct acting solenoid valve, with flow paths illustrated

in significant variation in the detail applied to the analysis. This includes high level hydraulic models that primarily consider the piston of the actuator [13,14], through to more extensive models, such as [15], where a detailed analysis of an automatic transmission is performed.

The remainder of this paper presents a description of the solenoid valve being investigated, comprehensive modeling of the valve–clutch system, and simulation based characterization of valve operation. Then simulated analysis of the valve dependency on design and operational variables, and experimental validation of the model is conducted. Finally, concluding remarks are made.

2 Solenoid Valve Description

The direct acting clutch control valve is a normally open configuration, such that in the de-energized state the output port is open to the exhaust. The spool is designed with overlapping tolerances for port throttling during pressure control, under specific throttling conditions the output port can be closed to both input and exhaust ports. When fully energized at peak current, the output port is open to the input. A sectioned valve body is presented in Fig. 1 that is under study for this investigation. It comprises of two moving components in the damper piston and solenoid spool, the main input, output, and exhaust flow ports as well as feedback flow lines are indicated in the figure.

The valve section is represented in Fig. 2 with the spool and damper included, in the normally closed position. It also shows the springs and magnetic control force from the solenoid itself. Obviously, when the control load is placed on the spool it moves forward from the low position until the pressure on the feedback volume balances the control force. As a consequence of this specific design characteristic, the valve will be pushed to the fully open position while there is a load imbalance on the spool. Therefore, during clutch filling phase of actuation when a high volume flow is required, the spool will be completely open to maintain a high flow. Once flow reduces and pressure balancing occurs the spool will be forced to throttling flow and realize pressure control.

3 Mathematical Model Development

The mathematical modeling of hydraulic power systems is a reasonably well established field of research. There are several well know references that provide the fundamental procedures for such analyses. Typical texts include [5,16,17] provide detail on the fundamental principles of developing such models. Such methods integrate internal flow methods such as orifice plates, jet flow, and fluid compressibility with Newtonian mechanics.

3.1 Solenoid Valve.

The equation of motion for the solenoid valve spool includes spring stiffness, K_1 and K_2, viscous loss, C_D, magnetic control load from the solenoid, F_M, jet forces, F_{J1x} and F_{J2x}, and feedback pressure force, F_{P1} and F_{P2}. These are shown graphically in Fig. 3. The equation of motion is

$$M_S\ddot{x}_S = K_1(x_S + x_0) - K_2(x_S + x_0) - B\dot{x}_S + F_{J2x} - F_{J1x} - F_{P2} + F_{P1} + F_M \quad (1)$$

where M_s is spool mass, x_s and its time derivatives spool displacement, velocity, and acceleration, x_0 is preload compression of each spring, $K_\#$ is the spring stiffness, B is the damping coefficient, and F_M is the solenoid force.

Jetting forces results from a high pressure imbalance through ports, and can be calculated in general as follows [5]:

$$F_{Jx} = \frac{\rho Q^2 \cos(\theta)}{C_{C1}A_P(x)} \quad (2)$$

where Q is flow rate through the port, C_{C1} is contraction coefficient, $A_P(x)$ is the spool position dependent port opening area, ρ is density of the fluid, and θ is the inclination of the port opening. Figure 4 presents a close-up of an open port for determination of port area and inclination of opening.

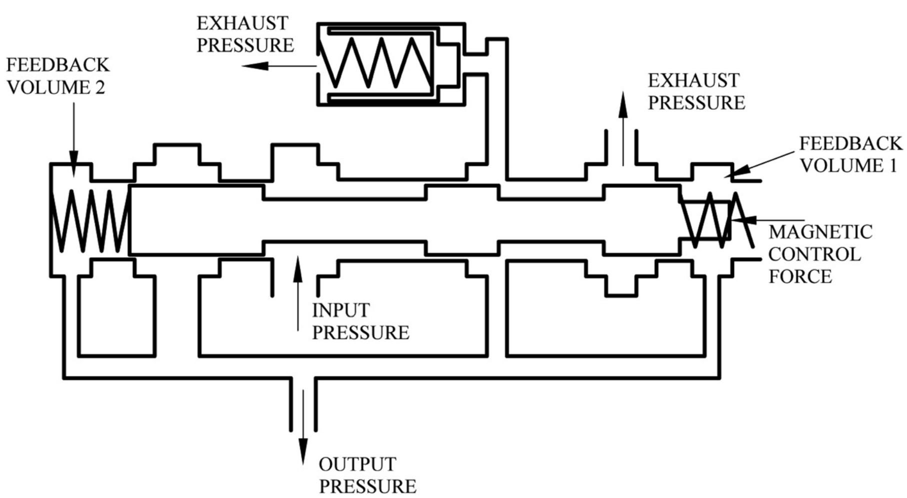

Fig. 2 Schematic of solenoid valve including damper

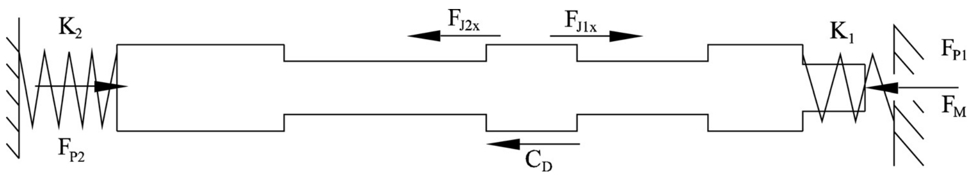

Fig. 3 Force balance of solenoid spool

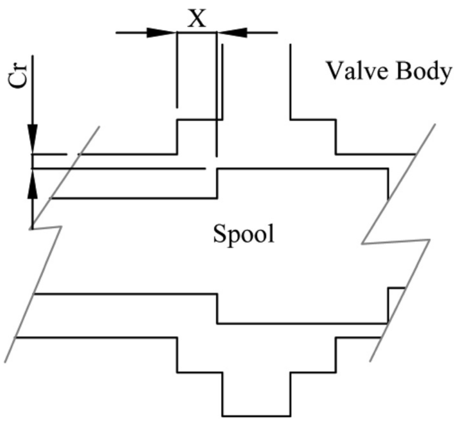

Fig. 4 Valve port detail

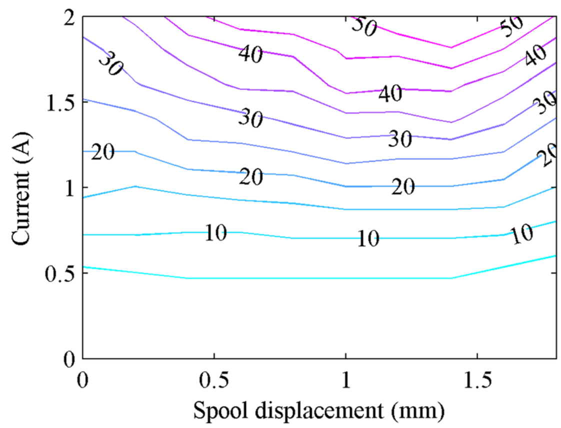

Fig. 5 Map of solenoid coil force, F_M, (N) as a function of spool position and input current

Valve port area is defined according to Fig. 4, using a simplified conical area to minimize discontinuity between leaking and open areas, as

$$A_P(x) = \begin{cases} \pi\sqrt{(c_r^2 + x^2)}D & x < 0 \\ \pi c_r D & x \geq 0 \end{cases} \tag{3}$$

where D is the spool diameter, and, inclination angle

$$\theta = \tan^{-1}(c_r/x) \tag{4}$$

The force applied from the solenoid coil to the spool is determined experimentally as a function of input current and core position over the full range of operating positions and input currents, results are shown in Fig. 5.

The feedback pressure force is naturally defined as a combination of cross-sectional area, A_V, and local pressure, P_V

$$F_P = A_V P_V \tag{5}$$

To analyze the localized pressure in the solenoid valve system, the model is refined to several smaller control volumes, for which mass conservation is applied. Figure 6 presents the various control volumes of the solenoid valve.

The pressures in control volumes for the valve or piston must also be determined. First define the control volumes for each system. Generally, these volumes are the input, output, and feedback damping volumes in valves, and for a clutch pack the control volume is the main fill volume in the piston. The flow rates into and out of control volumes are calculated using the sharp edged orifice flow equation, defined below

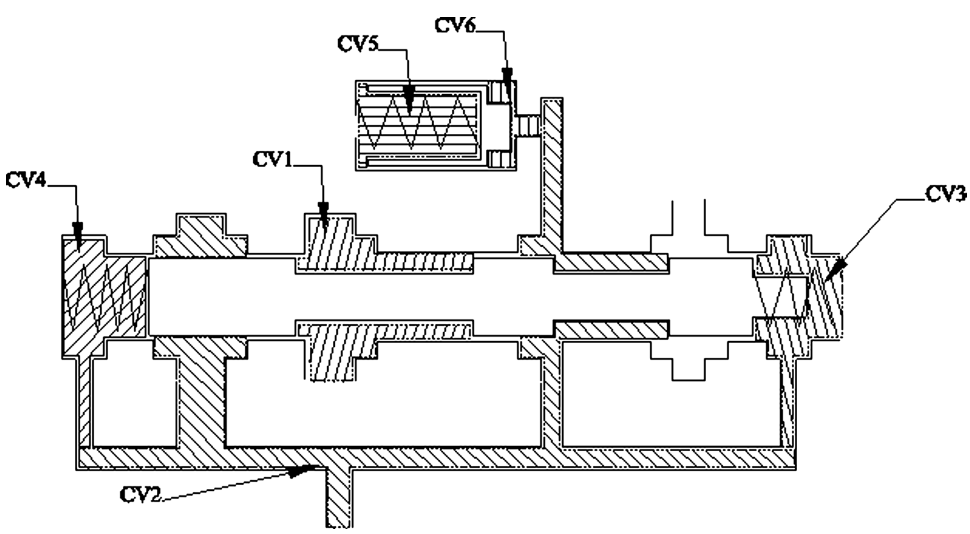

Fig. 6 Valve and damper control volumes

$$Q_\mathrm{O} = C_\mathrm{D}\pi D^2/4\sqrt{P_2 - P_1} \quad (6)$$

where Q_O is flow rate through an orifice, C_D is the modified discharge coefficient and is generally calculated per Eq. (7), P_2 is the pressure downstream of the orifice, and P_1 is the pressure upstream of the same orifice.

$$C_\mathrm{D} = C_\mathrm{d}\sqrt{2/\rho} \quad (7)$$

where C_d is the discharge coefficient, see Ref. [9] for calculation method. Very often the flow into a control volume includes leakage between the spool and housing. The leakage flow rate, Q_L, can be calculated in a similar manner to the orifice equation only the input area becomes the annular clearance between the spool and housing, and is derived thus

$$Q_\mathrm{L} = C_\mathrm{D}\pi D c_r\sqrt{P_2 - P_1} \quad (8)$$

Equation (3) provides the area for a throttling orifice for valve ports. If the port is closed the effective area is used as Eq. (8), however, for the open port the effective area is a conical surface that is made between the edge of the spool and housing for the port, see Fig. 4. To be utilized in the model the flow rate for valve port, Q_P, in Eq. (8) can be rewritten to include valve opening area

$$Q_\mathrm{P} = C_\mathrm{D}\pi D\sqrt{\left(c_r^2 + x^2\right)\left(P_2 - P_1\right)} \quad (9)$$

Particular to the feedback volumes and the clutch pack is the rate of change in volume, Q_V, with the spool or piston motion. This rate is expressed in the system as a flow rate change as fluid is forced into or out of the control volume. This is simply determined from the spool area, A_P, and instantaneous velocity of the valve.

$$Q_\mathrm{V} = A_\mathrm{P}\dot{x} \quad (10)$$

The last source of change in flow arises from the fluid compressibility; with hydraulic systems such as the one under investigation operating under pressures in the mega-Pascal range the compressibility of the fluid cannot be ignored. The bulk modulus can be significantly affected by the presence of air content in the hydraulic fluid. The flow resulting from compression of the hydraulic fluid is derived accordingly

$$Q_\mathrm{C} = \frac{V}{\beta}\frac{dP}{dt} \quad (11)$$

where Q_C is compressibility flow rate, β is bulk modulus, and V is the fluid volume. One major consideration in solenoid valves is the inclusion of air in the hydraulic fluid. This can be effectively modeled accordingly [4]

$$\beta = \frac{\beta\left(1 + 10^{-5}P\right)^{1+1/\gamma}}{\left(1 + 10^{-5}P\right)^{1+1/\gamma} + 10^{-5}R\left(1 - c_1P\right)\left(\beta/\gamma - 10^5 - P\right)} \quad (12)$$

Equations (6)–(11) are combined to provide the net flow into or out of any control volume in a hydraulic system; through mass conservation it is assumed that all the flow into and out of each control volume is zero, thus

$$Q_\mathrm{O} + Q_\mathrm{P} + Q_\mathrm{L} + Q_\mathrm{V} + Q_\mathrm{C} = 0 \quad (13)$$

Thus, for the general case of a single control volume with all applicable flows

$$\begin{aligned} &C_\mathrm{D}\pi D^2/4\sqrt{P_2 - P_1} + C_\mathrm{D}\pi D\sqrt{\left(c_r^2 + x^2\right)\left(P_2 - P_1\right)} \\ &\quad + C_\mathrm{D}\pi D c_r\sqrt{P_2 - P_1} + A_\mathrm{P}\dot{x} + \frac{V}{\beta}\frac{dP}{dt} = 0 \end{aligned} \quad (14)$$

By re-arranging Eq. (13) to determine the rate of pressure change, this is integrated numerically to determine the local pressure in specific control volumes. With the inclusion of variation in the control volume pressure is calculated as

$$\begin{aligned} P = \int \frac{\beta}{V}\Big(&A_\mathrm{P}\dot{x} + C_\mathrm{D}\pi D\sqrt{\left(c_r^2 + x^2\right)\left(P_3 - P_1\right)} \\ &+ C_\mathrm{D}\pi D c_r\sqrt{P_3 - P_1} + C_\mathrm{D}\pi D^2/4\sqrt{P_3 - P_1}\Big)dt \end{aligned} \quad (15)$$

This results in a first order differential equation that can be used to solve for the pressure in a control volume. In the development of these models the impact of pressure waves formed in opening or closing ports is assumed to be negligible. For the length of any track volume and considering the sonic velocity of the hydraulic fluid it can be expected that the resulting frequency response is of the order in kHz. This type of response is far higher than what will impact on the control of the system.

The first control volume, designated CV1, is the input port for the valve. It accepts input flow through an orifice, and expels fluid through a port that opens to the output chamber and leakage flow to the output control volume between the spool and housing and also the output port when closed. This chamber has constant volume as it is not affected by the spool position. Its equation is defined thus

$$\begin{aligned} P_\mathrm{CV1} = \int \frac{\beta}{V}\Big(&C_\mathrm{D}\pi\frac{D_\mathrm{CV1}^2}{4}\sqrt{P_\mathrm{Line} - P_\mathrm{CV1}} \\ &- C_\mathrm{D}\pi D_\mathrm{S}\sqrt{\left(x_\mathrm{S}^2 + c_r^2\right)\left(P_\mathrm{CV1} - P_\mathrm{CV2}\right)} \\ &- C_\mathrm{D}\pi D_\mathrm{S}c_r\sqrt{P_\mathrm{CV1} - P_\mathrm{CV2}}\Big)dt \end{aligned} \quad (16)$$

Control volume 2 (CV2) is the control volume associated with the output pressure. It receives input flow through the port with the CV1, and though the orifice with CV6. Output is managed as the demand from the clutch piston, and leakage flow is received from CV1 and CV4. Again this control volume is unaffected by volume changes.

$$P_\mathrm{CV2} = \int \frac{\beta}{V}\left(\begin{aligned} &C_\mathrm{D}\pi D_\mathrm{S}\sqrt{\left(X^2 + c_r^2\right)\left(P_\mathrm{CV1} - P_\mathrm{CV2}\right)} + C_\mathrm{D}\pi D_\mathrm{S}c_r\sqrt{P_\mathrm{CV1} - P_\mathrm{CV2}} + C_\mathrm{D}\pi\frac{D_\mathrm{CV5}^2}{4}\sqrt{P_\mathrm{CV5} - P_\mathrm{CV2}} \\ &- C_\mathrm{D}\pi D_\mathrm{S}c_r\sqrt{P_\mathrm{CV1} - P_\mathrm{CV4}} - C_\mathrm{D}\pi\frac{D_\mathrm{CV3}^2}{4}\sqrt{P_\mathrm{CV3} - P_\mathrm{CV2}} - C_\mathrm{D}\pi\frac{D_\mathrm{CV4}^2}{4}\sqrt{P_\mathrm{CV4} - P_\mathrm{CV2}} \\ &- C_\mathrm{D}\pi\frac{D_\mathrm{Output}^2}{4}\sqrt{P_\mathrm{CV4} - P_\mathrm{Clutch}} \end{aligned}\right)dt \quad (17)$$

The third control volume (CV3) not only receives flow through an orifice port from the CV2 but also leaks flow into the exhaust port. Additionally, this control volume is significantly affected by the displacement of the spool and hence volume change.

$$P_{\mathrm{CV3}} = \int \frac{\beta}{V_0 + dV}\left(C_{\mathrm{D}}\pi \frac{D_{\mathrm{CV3}}^2}{4}\sqrt{P_{\mathrm{CV2}} - P_{\mathrm{CV3}}} - A_{\mathrm{P}}\dot{x}_{\mathrm{S}} - C_{\mathrm{D}}\pi D_{\mathrm{S}} c_r \sqrt{P_{\mathrm{CV3}} - P_{\mathrm{Exh}}}\right) dt \tag{18}$$

Similar to the third control volume CV4 received flow from CV2 via an orifice. However, there is also leakage from CV2 into this volume, as well as the control volume modified by the displacement of the spool. Similar to the previous equation it is derived as so, noting the sign changes from Eq. (18)

$$P_{\mathrm{CV4}} = \int \frac{\beta}{V_0 + dV}\left(\begin{array}{l} C_{\mathrm{D}}\pi \frac{D_{\mathrm{CV4}}^2}{4}\sqrt{P_{\mathrm{CV2}} - P_{\mathrm{CV4}}} \\ + A_{\mathrm{P}}\dot{x}_{\mathrm{S}} - C_{\mathrm{D}}\pi D_{\mathrm{S}} c_r \sqrt{P_{\mathrm{CV3}} - P_{\mathrm{CV2}}} \end{array}\right) dt \tag{19}$$

3.2 Damper. The integral damper is used to suppress any transients arising in the system during actuation events. The damper piston is retained with a stiff spring, and is modeled as follows:

$$m_{\mathrm{D}}\ddot{x}_{\mathrm{D}} = P_{\mathrm{CV5}}A_{D1} - P_{\mathrm{CV6}}A_{D2} - C_2\dot{x}_{\mathrm{D}} - K_{\mathrm{D}}(x_{\mathrm{D}} + x_{\mathrm{D0}}) \tag{20}$$

Control volume 5 accepts feed from the pressure line and provides leak flow to CV6. It simulates the damper by absorbing shock pressure in the output line; therefore, it is affected by the displacement of its spool. The flow equation is below

$$P_{\mathrm{CV5}} = \int \frac{\beta}{V_0 + dV}\left(C_{\mathrm{D}}\pi \frac{D_{\mathrm{CV5}}^2}{4}\sqrt{P_{\mathrm{CV2}} - P_{\mathrm{CV5}}} - A_{\mathrm{P}}\dot{x}_{\mathrm{D}} + C_{\mathrm{D}}\pi D_{\mathrm{D}} c_{r\mathrm{D}} \sqrt{P_{\mathrm{CV5}} - P_{\mathrm{CV6}}}\right) dt \tag{21}$$

Finally, for CV6 inputs come from leakage from CV5 and through the orifice to CV2. The direction of flow is dependent on the instantaneous pressure in CV2, and as a consequence of the high input pressure from CV5 it is generally output flow. It also is affected by the displacement of its damper spool.

$$P_{\mathrm{CV6}} = \int \frac{\beta}{V_0 + dV}\left(C_{\mathrm{D}}\pi \frac{D_{\mathrm{CV6}}^2}{4}\sqrt{P_{\mathrm{Exh}} - P_{\mathrm{CV6}}} + A_{\mathrm{P}}\dot{x}_{\mathrm{D}} - C_{\mathrm{D}}\pi D_{\mathrm{D}} c_{r\mathrm{D}} \sqrt{P_{\mathrm{CV5}} - P_{\mathrm{CV6}}}\right) dt \tag{22}$$

3.3 Clutch. Clutch pistons have an annular design to accommodate input shafts to the transmission, a return spring, input orifice and clutch pack, shown schematically in Fig. 7. The solenoid valve drives the piston, applying load onto the clutch pack during gear change events. As such the equivalent spring system is strongly nonlinear, the return spring is reasonably soft and is used to disengage the clutch; however, when the plates are in contact the stiffness is significantly higher.

The equations of motion for the piston and control volume are as follows:

$$M_{\mathrm{Cl}}\ddot{x}_{\mathrm{Cl}} = P_{\mathrm{Clutch}}A_{\mathrm{Cl}} - B_{\mathrm{Cl}}\dot{x}_{\mathrm{Cl}} - K_{\mathrm{Cl}}(x + x_{\mathrm{Cl0}}) \tag{23}$$

The control volume, CV 7, accepts input flow from the solenoid output through an orifice. The control volume is affected by the displacement of the piston head, and its velocity provides a flow rate change for the system.

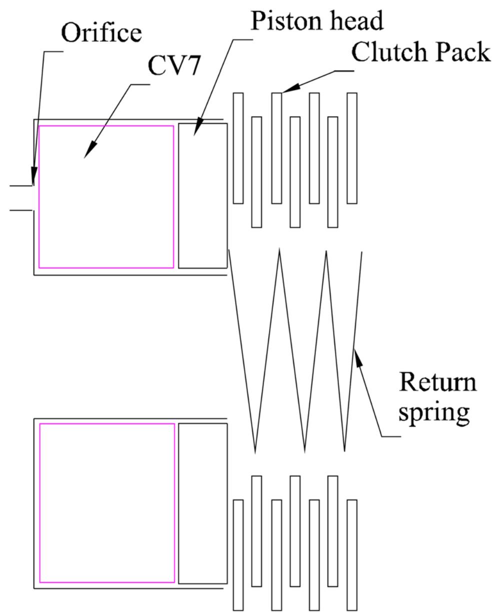

Fig. 7 Clutch piston schematic model

$$P_{\mathrm{CV9}} = \int \frac{\beta}{V_0 + dV}\left(C_{\mathrm{D}}\pi \frac{D_{\mathrm{Cl}}^2}{4}\sqrt{P_{\mathrm{CV2}} - P_{\mathrm{Clutch}}} - A_{\mathrm{Cl}}\frac{dX_{\mathrm{Cl}}}{dt}\right) dt \tag{24}$$

4 Simulations and Analysis

The purpose of this paper is to analyze and characterize the functionality of the direct acting solenoid valve, this is primarily achieved through simulations, as this is a strongly nonlinear system it will not be possible to develop any reasonable results using linear analysis. Figures 8 and 9 show the response of the valve to step inputs of 0.6 and 0.9 A. These results provide a strong demonstration of the behavior of the valve during normal operation.

Results presented in Figs. 8 and 9 demonstrate the operation of the solenoid valve. During the clutch filling period, $t < 0.1$ s, there is a strong load imbalance, see Figs. 8 and 9(*a*). Note particularly the difference between feedback pressure forces and the solenoid force during the filling period holds the spool at the limit of the open position maximizing the port opening to enhance fluid flow. However, during steady state conditions these loads are effectively balanced out such that high flow is replaced with fine pressure control. Results in (b) and (c) demonstrate the associated pressure drop as flow demand increases, and flow rapidly reducing during force balancing in the valve, leading to steady state conditions.

Influence of input pressure is an important consideration for the design of efficient transmissions, $E = Q \times P$ determines the power consumed in pressurizing the control system. Therefore, demonstrating the influence of line pressure on control of the solenoid can be used to determine how minimizing peak line pressure affects valve dynamics. Results in Fig. 10 are for the step response of the valve to a 0.9 A input, with line pressures above and below the steady state output. The results show that the input pressure limits the output pressure, below the theoretical output ($\sim$900 kPa) required by the valve, and above the theoretical output the valve operates correctly, vibration in the pressure tends to increase though.

Air content of the hydraulic fluid strongly influences response of the valve, while in simulations it can strongly influence numerical stability, as the fraction of air content to fluid volume

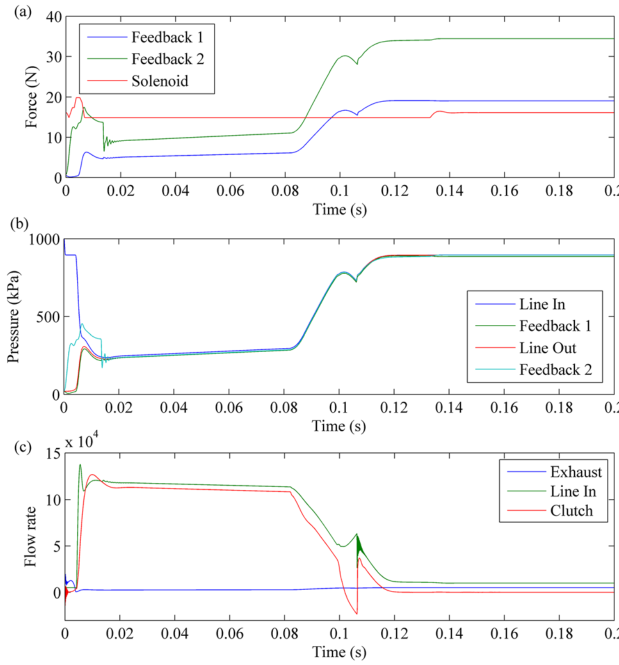

Fig. 8 Valve step response with a 0.6 A input. (*a*) Solenoid and feedback pressure forces, (*b*) pressure from input line, output line, and both feedback chambers, and (*c*) flow ranges for exhaust, input line, and to the clutch.

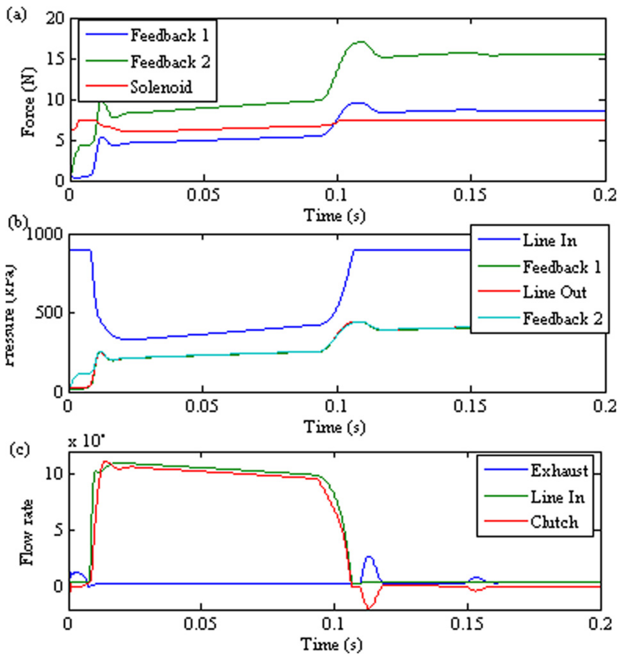

Fig. 9 Valve step response with a 0.9 A input. (*a*) solenoid and feedback pressure forces, (*b*) pressure from input line, output line, and both feedback chambers, and (*c*) flow ranges for exhaust, input line, and to the clutch.

decreases the system becomes significantly stiffer. It is well known to have the most significant impact on valve response when compared to design and manufacturing variables. The impact of entrained air content is demonstrated strongly in Fig. 11, with low air content, $<1\%$, there is minimal overshoot in the step response. However, as the air content is increased in simulations, there is a substantial increase in the overshoot

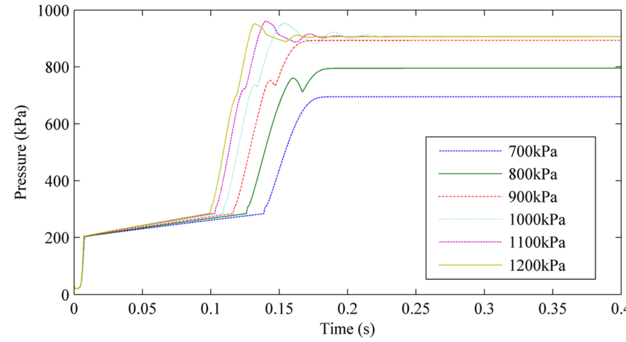

Fig. 10 Step response to 1 A input signals with different input pressures

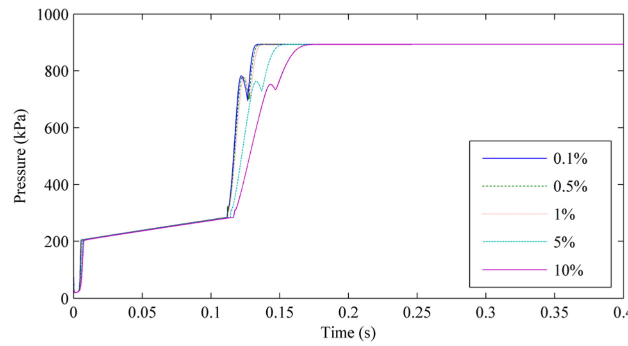

Fig. 11 Step response to 1 A input signals under different percent air contents

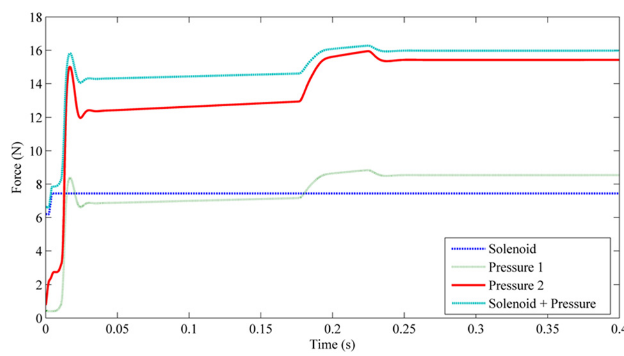

Fig. 12 Solenoid and pressure forces acting on the spool for a 0.6 step input

in pressure response. This is a consequence of the reduction in effective stiffness of the system. For example, the effective bulk modulus of the system at air content of 1% is an order of magnitude smaller than with an air content of 10%.

To demonstrate the impact of using two separate feedback pressures the solenoid force and both feedback pressure forces are captured from a 0.6 A step input. These results are shown in Fig. 12. The results demonstrate that with the designed spool diameter the feedback pressure 2 force is approximately twice that of the solenoid, thus the area would need to be reduced substantially if a force balance is to be achieved without the second feedback pressure force (denoted pressure 1). Results also demonstrate that, by combining solenoid and pressure 1 forces these reach an approximate equilibrium with the with feedback pressure 2 force, the difference resulting from the inclusion of other loads on the spool, see Eq. (1).

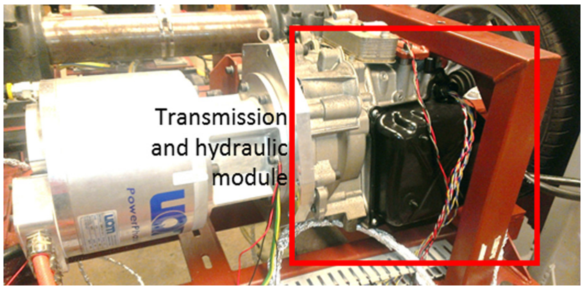

Fig. 13 Transmission and hydraulic module used for in situ testing of the solenoid valve

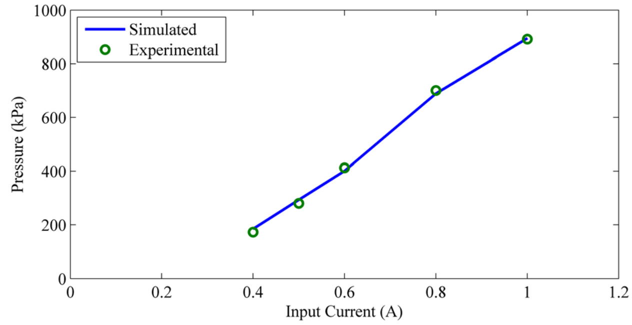

Fig. 15 Experimental and simulated steady state pressures at different input currents

5 Experimental Verification

Experimental verification of the combined clutch and valve model is conducted in situ at the University of Technology, Sydney Powertrain Systems Laboratory. The newest incarnation of this laboratory includes a two speed front wheel drive electric vehicle powertrain based on the Volkswagen DQ250 dual clutch transmission (DCT), pictured in Fig. 13. Solenoid control signals and pressure transducer sensors are integrated with dSpace MircoAutoBox II as part of the motor and transmission control system. The benefit of this type of verification is that the actuation of the valve in its normal operating environment can be studied, enabling the identification of variations in simulated and actual operating characteristics. For experimental validation herein, one control valve and pressure sensor data is recorded; results are presented in Figs. 15–18.

The experimental arrangement for solenoid valve testing is presented in Fig. 14 showing pump/motor, transmission fluid circuit, and control and data logging arrangement. Testing is conducted on the existing powertrain test system, however, in Fig. 14 only the appropriate apparatus utilized in testing is shown. These tests make use of the existing clutch, solenoid valve and pressure sensor, with the pressure sensor located on the clutch side of the output orifice. Control interface and data acquisition is achieved through MicroAutoBox II and dSpace.

Steady state pressure for the valve–clutch assembly is presented in Fig. 15 for both simulated and experimental results at different input currents. This data are collected using a series of step current inputs to the solenoid valve and taking the steady state response. This shows a strong correlation between simulated and experimental results, indicating a precise overall valve–clutch assembly model.

Two assessments are made for experimental evaluation of the valve–clutch model, response to step input, Fig. 16, and response to ramp input, Figs. 17 and 18, shown in relation to input current signal to generate hysteresis *P-I* curves for the valve. Results demonstrate a reasonable correlation between simulated and experimental results, particularly above a pressure of approximately 50 kPa, however a degree of hysteresis present in the clutch piston influences the results.

The results of two step inputs, shown Fig. 16, have a strong correlation during the majority of the transient response. Figure 16(*a*) presents the response to a 1A step input, whilst Fig. 16(*b*) is for a 0.8 A step input. Both sets of results show a strong correlation for the majority of the transient response. However, the simulated results in Fig. 16(*b*) do not correlate as well during overshoot and settling at the steady state pressure. At this stage the authors are yet to identify the source of this response. Demonstration of clutch filling occurs at 3.4 s and 4.5 s in Figs. 16(*a*) and 16(*b*), respectively, with pressure balancing occurring beyond this period.

The ramp input results presented in Figs. 17 and 18 show a reasonable correlation for the overall system. The ramp up and ramp down of input current at a slow rate is useful in establishing the degree of hysteresis in the valve–clutch assembly. These results also demonstrate a strong correlation between experimental and simulated results, however, strong hysteresis is identified in Fig. 18. This is likely to result from sticking of the clutch piston. These results further demonstrate the suitability of this modeling method in developing accurate representations of hydraulic power systems.

The noticeable pressure spikes in Fig. 17 at 0.6 and approximately 0.8 s in the simulation results are identified as resulting from interaction between the spool motions and "underlap" type port conditions. It was found from these simulations that, as the spool moved from exhaust throttling to open during ramp up, and from open to exhaust throttling during ramp down variation in spool relative position to the port generated these pressure spikes.

Hysteresis noticeable in Fig. 18 at up to 0.3 s is attributed to the motion of the valve spool. The simulation model of the spool, Eq. (1), the initial spring preload requires a small force from the solenoid to initiate motion; in this case a minimum current of approximately 0.28 A is required. Thus a step up motion of the spool occurs during initial motion. During release, as current

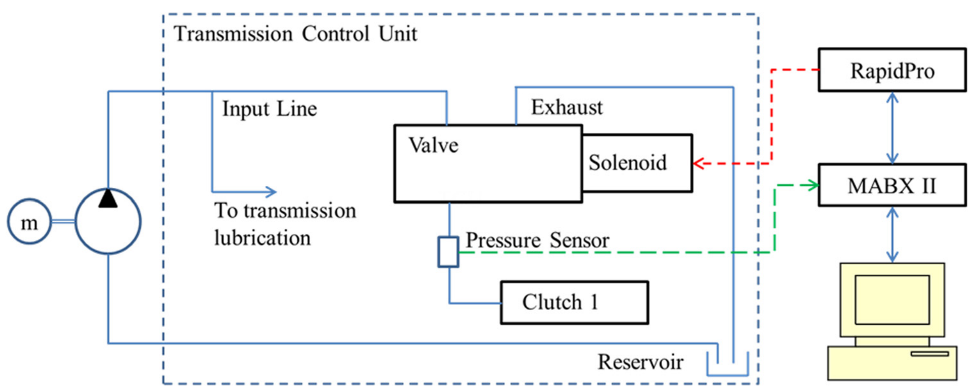

Fig. 14 Experimental setup for single clutch and solenoid assembly, (Short dash—solenoid valve control signal, Long dash—pressure sensor, blue hydraulic lines)

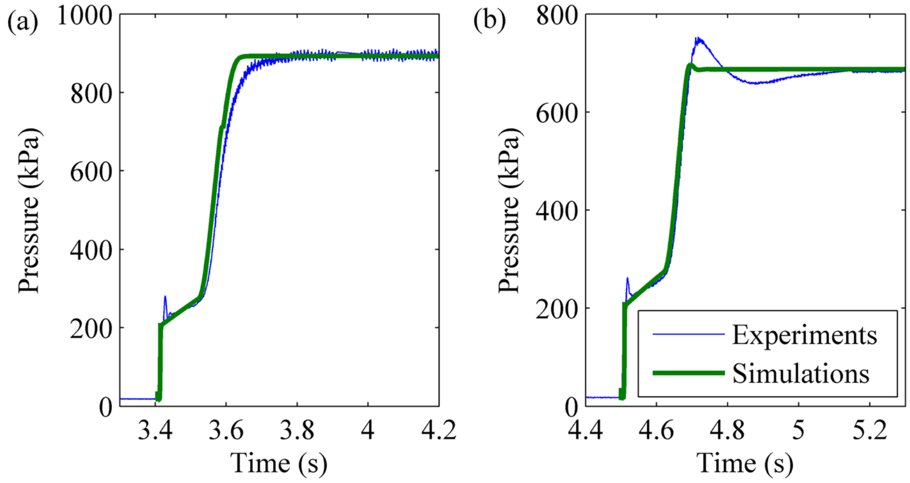

Fig. 16 Simulated and experimental responses for (*a*) 1 A and (*b*) 0.8 A step input currents

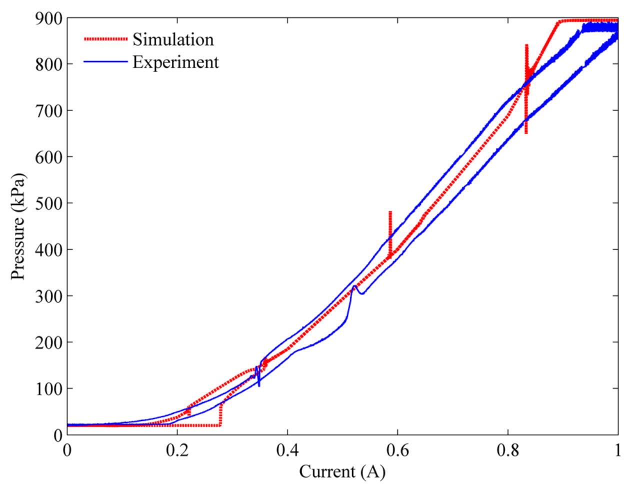

Fig. 17 Pressure versus current curve for a 0–1 A ramp input, ramp rate 0.2 A/s

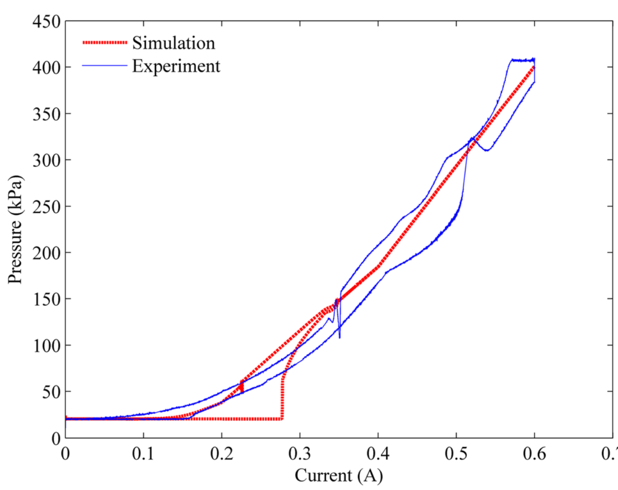

Fig. 18 Pressure versus current curve for a 0–0.6 A ramp input, ramp rate 0.2 A/s

decreases, the slowly reducing load delays the return to the zero position, influencing the drop in pressure.

6 Conclusion

The primary objective of this research has been the development of a detailed multiphysics model of a direct acting clutch control solenoid for dual clutch transmission equipped powertrains. This model, developed in the MATLAB/SIMULINK environment was utilized to evaluate the performance of the valve and identify its operating characteristics. Finally, the model was validated on an existing DCT used on a powertrain test rig. Good agreement between simulation results and experimentally obtained data were achieved. It is suggested that the expansion of the model to include the fluid pump may further enhance the obtained results.

Simulations were used to identify the operating characteristics of this particular type of valve. It is observed that there are two specific operating characteristics of the valve that have resulted in it being capable of both providing high flow when required and precise pressure control. This is realized with the balancing of the solenoid force and two fluid feedback volume forces. As a result, with high pressure difference during the clutch piston displacement, the spool is forced to the completely open position to maximize flow. As flow reduces and pressures are balanced the feedback forces rapidly orientate to pressure based control conditions. This feedback pressure balancing with the applied solenoid force, and its reactive nature enable the precise pressure control of the system.

Further study of the influence of input pressure and air content was conducted to evaluate the valve performance. It is noted that the input pressure limits the operating pressure of the valve below a fixed limit, but above this limit the output pressure is independent of the input pressure. The upper limit of the output pressure is dictated by the balancing of feedback pressures in the system with the solenoid force. Air content in the hydraulic fluid was only demonstrated to have limited effect on the control and operation of the valve, increasing rising time to some extent, suggesting that the design is reasonably robust to its variation.

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Nomenclature

A_P = the spool position dependent port opening area
A_V = cross-sectional area
B = damping coefficient and
C_{C1} = contraction coefficient
C_d = discharge coefficient
C_D = modified discharge coefficient
D = spool diameter
F_{J1x} = jetting force from input port
F_{J2x} = jetting force from exhaust port
F_M = solenoid force
F_{P1} = feedback pressure force from control volume 4
F_{P2} = feedback pressure force from control volume 3
K = spring stiffness
M_s = spool mass
P = pressure
P_V = control volume pressure
Q = flow rate
Q_L = leakage flow rate
Q_O = flow rate through an orifice
V = fluid volume
x_s = spool displacement its time derivatives are velocity and acceleration
x_0 = preload displacement for the spring
β = bulk modulus
θ = inclination of the port opening
ρ = density of the fluid

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