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Research Article

Dynamic Analysis and Control of the Clutch Filling Process in Clutch-to-Clutch Transmissions

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Clutch fill control in clutch-to-clutch transmissions influences shift quality considerably. An oncoming clutch should be applied synchronously with the release of an offgoing clutch to shift gear smoothly; therefore, the gap between the piston and clutch plates should be eliminated when the torque capacity is near zero at the end of the clutch fill phase. Open-loop control is typically implemented for the clutch fill because of the cost of pressure sensor. Low control precision causes underfill or overfill to occur, deteriorating shift quality. In this paper, a mathematical model of an electrohydraulic clutch shift control system is presented. Special dynamic characteristic parameters for optimal clutch fill control are subsequently proposed. An automatic method for predicting initial fill control parameters is proposed to eliminate distinct discrepancies among transmissions caused by manufacturing or assembling errors. To prevent underfill and overfill, a fuzzy adaptive control method is proposed, in which clutch fill control parameters are adjusted self-adaptively and continually. Road vehicle test results proved that applying the fuzzy adaptive method ensures the consistency of shift quality even after the transmission's status is changed.

1. Introduction

Automatic transmissions are used to transfer the power of an engine smoothly and effectively to vehicle wheels at optimal transmission ratios according to performance requirements and economic demand. Figure 1 shows a two-dimensional structural diagram of an 8-speed automatic transmission with five shift elements, namely, a brake $B1$ and four clutches $C1$–$C4$, which are engaged or separated by the electrohydraulic control system of transmissions. To shift from one gear to another in an automatic transmission, one clutch must be released and another must be applied synchronously, which is called a clutch-to-clutch shift [1, 2]. With the development of six-speed transmissions or even more speeds nowadays, considerable time and effort has been made to study the clutch-to-clutch shift control technology [1, 3, 4]. Asynchronous clutch control would cause power interruption or overconstraint [5]. Therefore, the gap between the piston and clutch plates for the oncoming clutch should be eliminated when the clutch torque capacity is near zero at the end of the clutch fill phase [6, 7]. Nowadays most clutch torque models do not take hydraulic dynamic characteristics into account [8]. To optimize the engagement of clutches, clutch fill is usually formulated as an optimization problem. Open-loop clutch pressure control was proposed as a solution by means of dynamic programming algorithm for cost reasons [6]. However, this control method requires precision tracking of the input pressure. Relative experiments were proposed and the experimental results were used for optimal control of clutch fill process [9]. Low control precision causes underfill or overfill to occur, deteriorating shift quality.

Most automatic transmissions use the electrohydraulic driving pattern, in which the shifting elements are controlled separately. Figure 2 shows a partial electrohydraulic control system for a single clutch and a torque convertor. The main line pressure $P_{CV,P}$ is provided by an oil pump, which is mechanically connected with the engine output shaft. A flow regulation valve (indicated by number 1) is used to discharge redundant fluid flow, especially at a high rotation speed, and to prevent the pressure and fluid flow in the main hydraulic circuit from exceeding the limitation. A pressure-regulating valve (indicated by number 5) is used to regulate the pressure

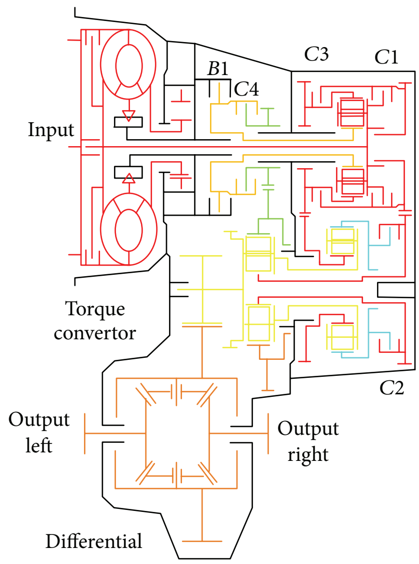
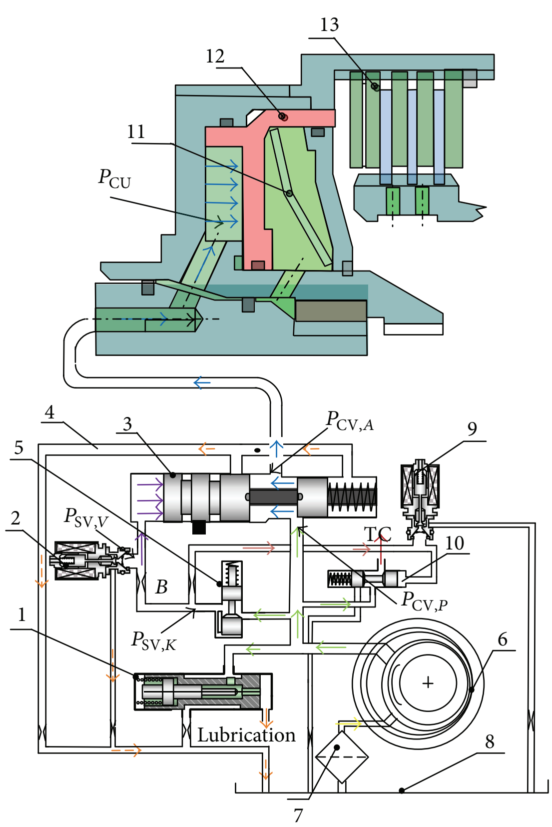


Figure 1: Schematic diagram of an 8-speed automatic transmission.



(1) Flow control valve
(2) Solenoid valve
(3) Shift valve
(4) Oil duct
(5) Pressure-regulating valve
(6) Pump
(7) Filter
(8) Tank
(9) Solenoid valve
(10) Control value
(11) Disc spring
(12) Piston
(13) Clutch friction disc

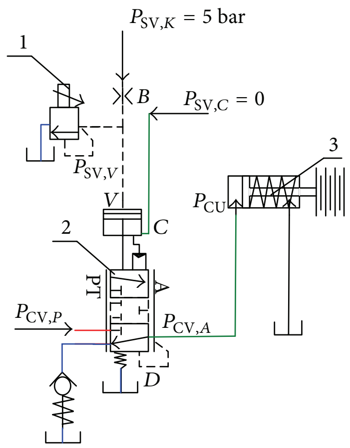
Figure 2: A partial electrohydraulic control system for a single clutch.

in the main hydraulic circuit to satisfy the pressure demands $P_{\mathrm{SV},K}$ for all shift valves' control; here 5 bar is preferred which is realized by the dimensions of the pressure-regulating valve. The clutch piston chamber pressures are jointly controlled by a solenoid valve (indicated by number 2) and a shift valve (indicated by number 3). By controlling the electric current of the solenoid valve, the pilot pressure $P_{\mathrm{SV},V}$ of the shift valve can be regulated. The pilot pressure determines the position of the shift valve spool and thereby determines the pressure and fluid flow in the clutch piston chamber. When the clutch piston chamber pressure increases, once the frictional resistance and the return spring force are overcome, the clutch piston starts to move rightward until the gap is eliminated and then force is applied. To prevent overfill or underfill, the fill phase is controlled to end before a preset time. To reduce the cost of transmission sensors, no pressure sensors are assembled for clutches; consequently, obtaining the feedback pressure for closed-loop control is difficult. Moreover, because of the strong nonlinear pressure characteristics of solenoid valves and the response delay, obtaining precise target pressure by controlling the current is difficult. Furthermore, the clutch pressure response differs based on the status of transmissions, such as the tolerance and wear of parts. Therefore, building an open-loop clutch pressure control model for clutch fill phase based on the dynamic pressure characteristics of electrohydraulic clutch control systems is necessary to satisfy the requirements of clutch fill time and maximal fluid flow. Previous models have primarily depended on car test results, which cannot be adapted to complex working environments. After transmissions run for an extensive period, wear inevitably occurs. Therefore, a fuzzy adaptive fill control method, which was proven to be capable of improving the shift quality effectively, is proposed in this paper.

2. Model of Hydraulic Clutch Shift Control Unit

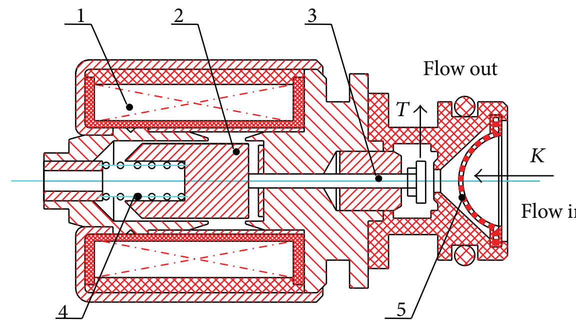
Figure 3 shows a clutch hydraulic control unit, in which the clutch piston chamber pressure is controlled by a solenoid valve and a shift control valve mainly. The solenoid valve controls the pilot pressure $P_{\mathrm{SV},V}$ at port V of the shift valve to control the position of the valve spool and change the opening area of port A, thereby regulating the clutch piston chamber pressure P_{CU}. Port P is connected to the main line pressure $P_{\mathrm{CV},P}$ of this system, and port T is used for discharging the clutch cylinder. The pressure $P_{\mathrm{CV},C}$ at port C is controlled by a limp home valve, and there is no pressure in normal modes. Pressure response in failure modes is not discussed here. Orifice B is used to control the total fluid flow into the solenoid valve and the shift valve. Here some symbols and subscripts are appointed in advance for computational convenience; see Nomenclature.

2.1. Dynamic Model of a Solenoid Valve. Figure 4 shows the structural diagram of a normally open high-speed proportional solenoid valve [7], mainly composed of a coil, armature, return spring, and valve spool. The coil produces



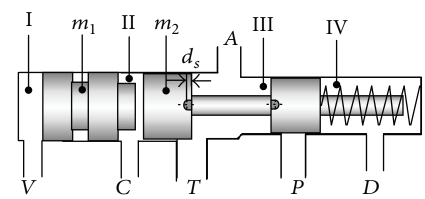
(1) Clutch control solenoid valve
(2) Shift valve
(3) Clutch

Figure 3: Schematic diagram of a hydraulic clutch control unit.



(1) Coil
(2) Armature
(3) Valve spool
(4) Spring
(5) Strainer

Figure 4: Structural diagram of a variable-force solenoid valve.



P: Inlet
V: Pilot port
C: Limp home port
A: Outlet
D: Pressure regulation port
T: Discharging port
I–III: Chambers

Figure 5: Structure of the shift valve.

electromagnetic force and overcomes the resistance of the return spring, and the armature drives the valve spool to move in the axial direction of the spool, thereby regulating the opening area of the valve and, subsequently, the pressure.

According to the laws established by Newton, the dynamic equation of the valve spool can be described as

$$m_{SV}\ddot{x}_{SV} + b_{SV}\dot{x}_{SV} + k_{SV}\left(x_{SV} + x0_{SV}\right) = G_{SV}. \tag{1}$$

The electromagnetic subsystem of the solenoid valve can be simplified into a series connection of resistive and inductive components [10]. The electromagnetic force is calculated using

$$G_{SV} = E_f i_{SV}^2, \tag{2}$$

where the dynamic equation of the electric circuit is written as

$$i_{SV} = \frac{\left(e_{SV} - R_{SV}i_{SV} - E_v i_{SV}\dot{x}_{SV}\right)}{L_{SV}}. \tag{3}$$

During the movement of the valve spool, the dynamic equation for the hydraulic subsystem is expressed as

$$Q_{SV,T} = \mathrm{Cd}_{SV}A_{SV,T}\left(x_{SV}\right) \times \sqrt{\frac{2}{\rho}P_{SV,V}}. \tag{4}$$

2.2. Dynamic Model of a Shift Valve. Figure 5 shows the structural diagram of a shift control valve with double spools. For convenience, the symbol m_1 represents the pilot spool and m_2 the main valve spool. When the electric control system fails to provide pilot pressure at port V, the safety pressure at port C still can control the main spool m_2 to ensure the clutch pressure which can be supplied from port A. There are sectional area differences in chambers II and III.

According to the laws established by Newton, the dynamic equations of the valve spools are described as

$$m_{CV,m_1}\ddot{x}_{CV,m_1} = P_{SV,V}S_{CV,I} - P_{CV,C}S_{CV,II_l} - F_{m_2m_1}, \tag{5}$$

$$\begin{aligned} &m_{CV,m_2}\ddot{x}_{CV,m_2} + b_{CV}\dot{x}_{CV,m_2} + k_{CV}\left(x_{CV,m_2} + x0_{CV,m_2}\right) \\ &\quad = P_{CV,C}S_{CV,II\_r} + F_{m_2m_1} - P_{CV,A}\Delta S_{CV,III} - P_{CV,D}S_{CV,IV}, \end{aligned} \tag{6}$$

where $\Delta S_{CV,III} = S_{CV,III\_l} - S_{CV,III\_r}$ is the sectional area difference of chamber III.

Because of the movement of the main spool, the relationship between the pressure and fluid flow of the shift control valve at all ports can be expressed as

$$\begin{aligned} Q_{CV,A} = {} & \mathrm{sign}\left(P_{CV,P} - P_{CV,A}\right)\mathrm{Cd}_{CV} \\ & \times A_{CV,A}\left(x_{CV,m_2}\right)\sqrt{\frac{2}{\rho}\left|P_{CV,P} - P_{CV,A}\right|}, \end{aligned}$$

$$Q_{CV,T} = \mathrm{Cd}_{CV}A_{CV,T}\left(x_{CV,m_2}\right)\sqrt{\frac{2}{\rho}P_{CV,A}},$$

$$Q_{\mathrm{CV},V} = \operatorname{sign}\left(P_{\mathrm{SV},K} - P_{\mathrm{SV},V}\right) \mathrm{Cd}_B \times A_B \sqrt{\frac{2}{\rho}\left|P_{\mathrm{SV},K} - P_{\mathrm{SV},V}\right|} - Q_{\mathrm{SV},T}, \tag{7}$$

$$\dot{P}_{\mathrm{SV},V} = \frac{\beta_{\mathrm{CV}}}{V0_{\mathrm{CV,I}} + x_{\mathrm{CV},m_1} S_{\mathrm{CV,I}}} \left(Q_{\mathrm{CV},V} - \dot{x}_{\mathrm{CV},m_1} S_{\mathrm{CV,I}}\right), \tag{8}$$

$$\dot{P}_{\mathrm{CV},D} = \frac{\beta_{\mathrm{CV}}}{V0_{\mathrm{CV,IV}} - x_{\mathrm{CV},m_2} S_{\mathrm{CV,IV}}} \left(Q_{\mathrm{CV},D} + \dot{x}_{\mathrm{CV},m_2} S_{\mathrm{CV,IV}}\right). \tag{9}$$

2.3. Dynamic Model of the Clutch Fill Phase. Automatic transmission fluid (ATF) flows from port A of the shift valve into the clutch chamber through pipes in gear structure of the transmission. For convenience, the symbols l and r are the equivalent length and radius of these, respectively; P_c is the equivalent centrifugal pressure in the piston chamber caused by rotational speed of the clutch, calculated in [11]. With the function of the increasing inlet pressure, after the return spring force and friction force of the clutch are overcome, the piston starts to move rightward until the gap is eliminated and then force is applied.

According to the laws established by Newton, the dynamic equations of the clutch piston are described as

$$\begin{aligned} m_{\mathrm{CU}} \ddot{x}_{\mathrm{CU}} &= S_{\mathrm{CU}}\left(P_{\mathrm{CU}} + P_c\right) - F_{\mathrm{seal}} - k_{\mathrm{CU}}\left(x_{\mathrm{CU}} + x0_{\mathrm{CU}}\right) \\ &\quad - b_{\mathrm{CU}} \dot{x}_{\mathrm{CU}} \quad \left(x_{\mathrm{CU}} < d\right), \\ F_{\mathrm{APP}} &= S_{\mathrm{CU}}\left(P_{\mathrm{CU}} + P_c\right) - F_{\mathrm{seal}} \\ &\quad - k_{\mathrm{CU}}\left(d + x0_{\mathrm{CU}}\right) \quad \left(x_{\mathrm{CU}} = d\right). \end{aligned} \tag{10}$$

The fluid flow into the clutch piston chamber through pipes and the piston chamber pressure can be expressed as

$$Q_{\mathrm{CU}} = \frac{\pi r^4}{8\eta l}\left(P_{\mathrm{CV},A} - P_{\mathrm{CU}}\right) + Q_{\mathrm{CU}}^{\mathrm{leak}}, \tag{11}$$

$$\dot{P}_{\mathrm{CU}} = \frac{\beta_{\mathrm{CU}}}{V0_{\mathrm{CU}} + x_{\mathrm{CU}} S_{\mathrm{CU}}}\left(Q_{\mathrm{CU}} - \dot{x}_{\mathrm{CU}} S_{\mathrm{CU}}\right). \tag{12}$$

The sealing force F_{seal} is defined as

$$F_{\mathrm{seal}} = \begin{cases} \left(k_{\mathrm{seal}} \cdot \left(P_{\mathrm{CU}} + P_c\right) + c_m\right) \tanh\left(\dfrac{\dot{x}_P}{\alpha}\right) & \left(\left|\dot{x}_{\mathrm{CU}}\right| > 0.1\ \mathrm{mm/s}\right) \\ k_s \cdot \left(P_{\mathrm{CU}} + P_c\right) + c_s & \left(\left|\dot{x}_{\mathrm{CU}}\right| \leq 0.1\ \mathrm{mm/s}\right). \end{cases} \tag{13}$$

The static viscous friction F_{stick} can be computed using the Kanopp stick-slip model [12]. This viscous friction is generally ignored in dynamic models [13]. However, in the clutch fill phase, the pressure of ATF fluid is low and, thus, a large proportion of viscous friction exists. From the perspective of numerical computation, the friction of the O seal ring is assumed to be equal to F_{stick} when the velocity of the clutch piston is under 0.1 mm/s. When this occurs, F_{stick} is the force used to balance the piston, and its acceleration is 0. Therefore, this viscous friction is the maximal limit value. Once the friction exceeds this limitation, the piston starts to move.

2.4. Clutch Pressure Characteristics. By combining (1)~(12), the clutch piston chamber pressure response can be simulated. Figure 6(a) shows the pressure response when applying the step current command. The simulation results indicated that the clutch piston chamber pressure is initially 310 mA. Therefore, a calculated pilot pressure of at least 0.6 bar is required to overcome the resistance of the shift valve spool. When the small step control is applied, the clutch pressure increases with step to approximately 0.28 bar. If the pressure is not sufficiently high to overcome the resistance of the shift valve spool, then the clutch pressure does not respond to the current step, the so-called loss of step. When large step current control is applied, the clutch pressure overshoots or oscillates, but it rapidly stabilizes because of system damping (II in Figure 6(a)); clearly, the stabilizing time of the shift valve is shorter than that of the solenoid valve. Based on Figure 6(a), the steady characteristics of the clutch piston chamber pressure, exhibited when the control current increases, can be obtained, as illustrated in Figure 6(b), which shows that hysteresis exists.

3. Optimization of Clutch Fill Control Parameters

3.1. Selection of Control Parameters. The objective of clutch fill control is to stably eliminate the gap between the piston and clutch plates within a preset time. The ideal piston velocity in the clutch fill phase is shown in Figure 7. In the early period from $T0$ to $T2$, the piston velocity is nearly zero, and the dead volume of the clutch cylinders is first filled with ATF fluid. Consequently, the piston velocity is low within a short period from $T1$ to $T2$, rapidly increases to the maximum within the period from $T2$ to $T3$, is maintained from $T3$ to $T4$, and finally decreases from $T4$ to $T5$, and the clutch is engaged.

Figure 8 shows two methods for clutch fill pressure control based on the piston motion shown in Figure 7: triangle fill and square fill. P_{pre}, P_{FP}, and P_{FTP} are the prefill pressure, fast fill pressure, and stable fill torque pressure, respectively. P_{FTP} is the pressure when the clutch transfers a small torque. In practice, based on characteristic curves, such as pressure versus current curves, the target pressure can be transformed into the control current of the solenoid valve. Therefore, analysis of the target pressure and the real pressure is necessary for optimizing clutch fill control. According to Figures 7 and 8, the clutch fill phase is divided into three stages.

(1) Prefill Stage. The prefill stage shown in Figure 8 ensures that the dead volume in the clutch cylinder (Figure 2) is filled with ATF fluid and an initial amount of pressure. At this stage, because of the sealing force, the piston velocity is extremely low.

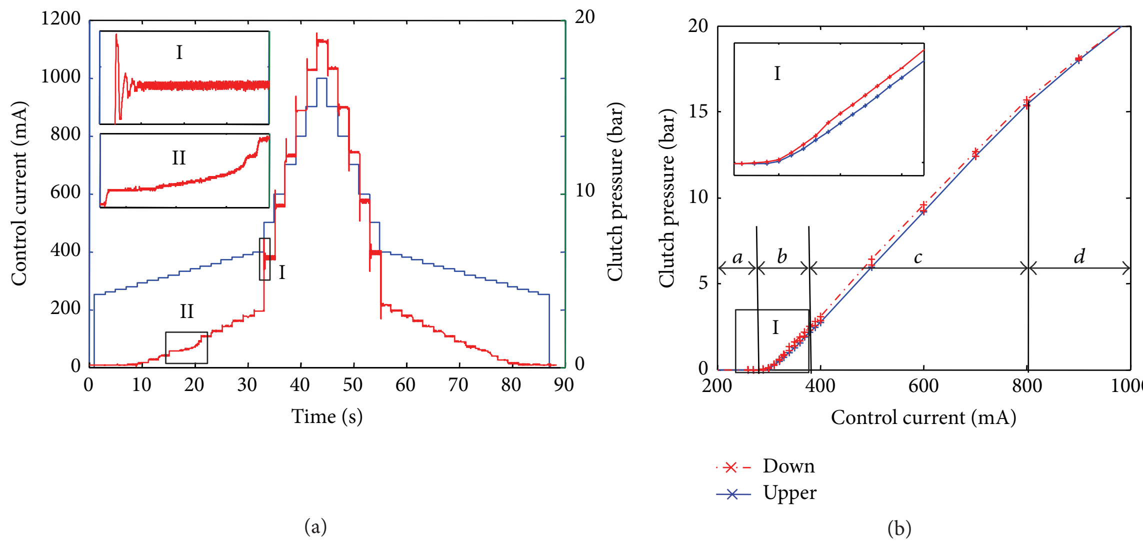


Figure 6: Simulation results for the step response of the system.

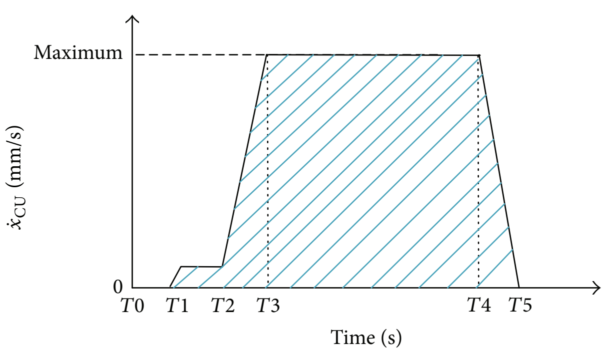


Figure 7: Ideal piston velocity.

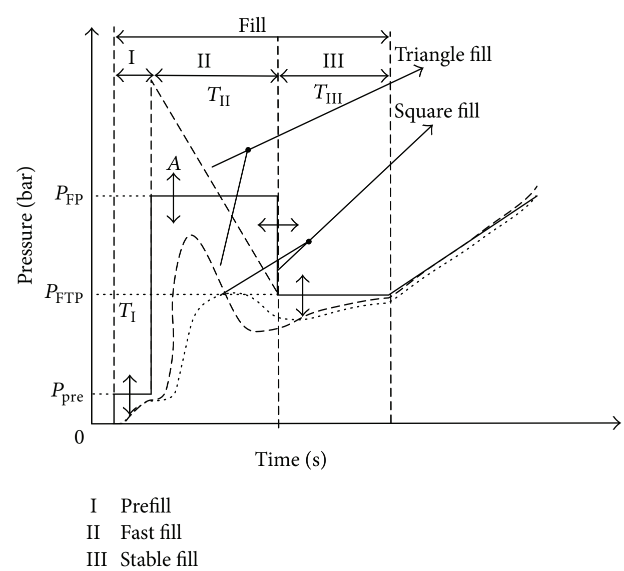


Figure 8: Design of clutch fill control pressure.

(2) Fast Fill Stage. This stage is the most crucial and difficult stage, corresponding to the period from $T2$ to $T3$ in Figure 7. The triangle fill method can be used to increase the pressure peak substantially and enhance the piston velocity. However, applying this method also increases the instability of the fill phase, and an excessively high pressure peak increases the demand for systemic fluid flow, thereby affecting the main pressure supply. Regarding the square fill method, the pressure response is relatively slow but stabler than that produced when applying the triangle fill method.

(3) Stable Fill Stage. At this stage, the piston is driven stably to eliminate the gap between clutch plates, corresponding to the period from $T3$ to $T5$ in Figure 7. The control current is generally unvarying in this process. As the piston moves, the elasticity of the wave spring gradually increases and, thus, the pressure gradually increases until the piston stops moving when the gap is completely eliminated.

Four parameters listed in Figure 8 were selected as clutch fill control parameters: fast fill pressure P_{FP}, rapid fill time T_{II}, stable fill pressure P_{FTP}, and prefill pressure P_{pre}. These parameters were used to determine the variation in the clutch fill pressure and the movement of the piston; thus, the clutch fill control was optimized by optimizing these parameters.

3.2. Influence of Clutch System Parameters on Clutch Filling. The aforementioned equations indicate that numerous parameters affect the clutch fill process. The consistency of several parameters, such as the number of coils in the solenoid valve, the magnetic resistance, and the sectional size of the valve core, is maintained during assembly. However, the consistency of other parameters, such as the spring stiffness and preload force, main supply pressure, ATF fluid temperature, and sealing force, is impossible to maintain, which causes differences in the oil fill results. Therefore, the effects of these parameters on the clutch fill phase should be analyzed and controlled to optimize the open-loop clutch fill control.

Based on the aforementioned (5)–(12), the influence of these parameters on filling the clutch under the same target oil fill pressure was simulated, as shown in Figure 9.

(1) ATF Fluid Temperature ϑ. Figure 9(a) shows the clutch piston chamber pressures at various temperatures. The temperature of the ATF fluid directly affects its viscosity; thus, the resistance of motion increases at low temperatures. In addition, the contraction of components contributes to changes in the tolerance clearance. These factors slow clutch fill. As shown in Figure 9(a), when the temperature is below -20°C, the pressure response is extremely slow at the fast oil fill stage and stable oil fill stage. As the temperature increases, the speed of the oil fill response and the peak pressure clearly increase. At high temperatures, such as 120°C, the clutch piston chamber pressure rapidly reaches the kiss-point pressure, but this process may cause overfill and shifting impact to occur. Therefore, clutch fill control parameters should be adjusted based on temperature to meet the shift quality requirements at various temperatures.

(2) Main Oil Pressure $P_{\mathrm{CV},P}$. Equation (8) indicates that the main supply pressure affects the flow of the system and that it is directly related to the rotation speed of the engine; thus, the main supply pressure considerably influences the clutch fill performance. Figure 9(b) shows the simulated clutch piston chamber pressure under various main supply pressures. Underfill clearly occurs between 5 and 10 bar, whereas overfill occurs at 20 bar. Therefore, the influence of the main supply pressure should be considered; in other words, the clutch fill intensity should be reduced to prevent overfill at high pressures and increased to prevent underfill at low pressures.

(3) Spring Stiffness k_{CU} *and Preload* $x0_{\mathrm{CU}}$. Tolerances exist because of both manufacture and assembly of return springs in clutches, which are mainly reflected in the stiffness and preload of the return springs. These two parameters determine the spring resistance that occurs when the piston moves. A strong resistance causes the clutch to reach the kiss point under high pressure and over a long time. Although this time can be shortened when the resistance is weak, shortening the time may cause the piston to move too fast and thereby affect the shift quality. Figures 9(c) and 9(d) show the simulation results for clutch fill pressure at various spring stiffness and preload values. Clearly, increased spring stiffness and preload result in a high maximal fill pressure and a low pressure change rate during the rapid fill stage.

(4) Static Frictional Resistance Coefficient k_s. The static frictional resistance coefficient affects the maximal static friction that must be overcome when the piston is static or moving at a low speed. This coefficient and the preload $x0_{\mathrm{CU}}$ jointly affect the pressure when the piston starts to move. The larger the k_s is, the higher the peak pressure is during the rapid fill stage (Figure 9(e)). However, if k_p and $x0_{\mathrm{CU}}$ are extremely small, then the acceleration of the clutch piston becomes extremely high, easily causing a piston movement bump to occur. If the piston speed reduces to a certain level, then double-peak fill occurs because of the static friction (Figure 9(e)), resulting in underfill; therefore, the clutch cannot reach the kiss point.

(5) Sealing Resistance Coefficient k_{seal}. Figure 9(f) shows the effects of the sealing resistance coefficient k_{seal} of the clutch cylinder. The influence of the tolerance of k_{seal} on clutch fill pressure is relatively small compared with that of other parameters. The resistance caused by k_{seal} influences the acceleration of the piston. Thus, a high speed clearly affects resistance. Therefore, k_{seal} can magnify the effects of temperature, piston preload, piston stiffness, and main supply pressure on the speed of the piston.

3.3. Optimization of Clutch Fill Control Parameters. Since the manufacturing tolerance, assembly tolerance, and working conditions also affect clutch fill pressure, clutch fill control should cover all types of qualified tolerance and normal working conditions.

3.3.1. Prefill Pressure. Prefill pressure P_{pre} is included to ensure that the pipes in the gear structure of the transmissions between the shift valve and clutch piston are filled with fluid and is not high to the extent that the resistance required to move the piston is overcome. Therefore, P_{pre} should not be too high to overfill when using clutches with low resistance. Moreover, the minimum prefill pressure should avoid the dropping downward of ATF oil of the rotating clutch. The oil flow must be completed in the prefill time T_{I}. Furthermore the prefill time is generally limited to the minimum shift time. The constraint condition on P_{pre} and T_{I} can be defined as

$$\begin{gathered}
\int_0^{T_{\mathrm{I_max}}} \left(Q_{\mathrm{CV},A} - Q_{\mathrm{CU}}^{\mathrm{leak}}\right) dt = V0_{\mathrm{CU}} + \pi r^2 l, \\
T_{\mathrm{I_max}} = 0.1 \times T_{\mathrm{shift}}, \\
T_{\mathrm{I}} = \left(1 - \lambda_{T_{\mathrm{I}}}\right) T_{\mathrm{I_min}} + \lambda_{T_{\mathrm{I}}} T_{\mathrm{I_max}}, \\
P_{\mathrm{pre_max}} S_{\mathrm{CU}} = \left(F_{\mathrm{stick}}\right)_{\min} + \left(k_{\mathrm{CU}} x0_{\mathrm{CU}}\right)_{\min}, \\
P_{\mathrm{pre_min}} = \rho g h_c, \\
P_{\mathrm{pre}} = \left(1 - \lambda_{P_{\mathrm{pre}}}\right) \cdot P_{\mathrm{pre_min}} + \lambda_{P_{\mathrm{pre}}} \cdot P_{\mathrm{pre_max}}.
\end{gathered} \tag{14}$$

3.3.2. Fast Fill Pressure and Time. The fast fill stage is the period when the piston accelerates after overcoming all types of resistance. The fast fill pressure and time affect the fill speed and stable fill pressure status. Figure 10(a) shows the simulation results for varying target pressures. Increasing the fast fill pressure enhances the clutch pressure response speed and increases the fill pressure at the initial stage. The time for the clutch pressure to reach the target value is shortened with increasing the fill pressure. However, pressure fluctuation occurs during the third and fourth fill process because the piston speed is still extremely high when the gap between clutch plates is eliminated. The ATF fluid cannot absorb all of the remaining energy instantly; therefore, the pressure fluctuation occurs inevitably.

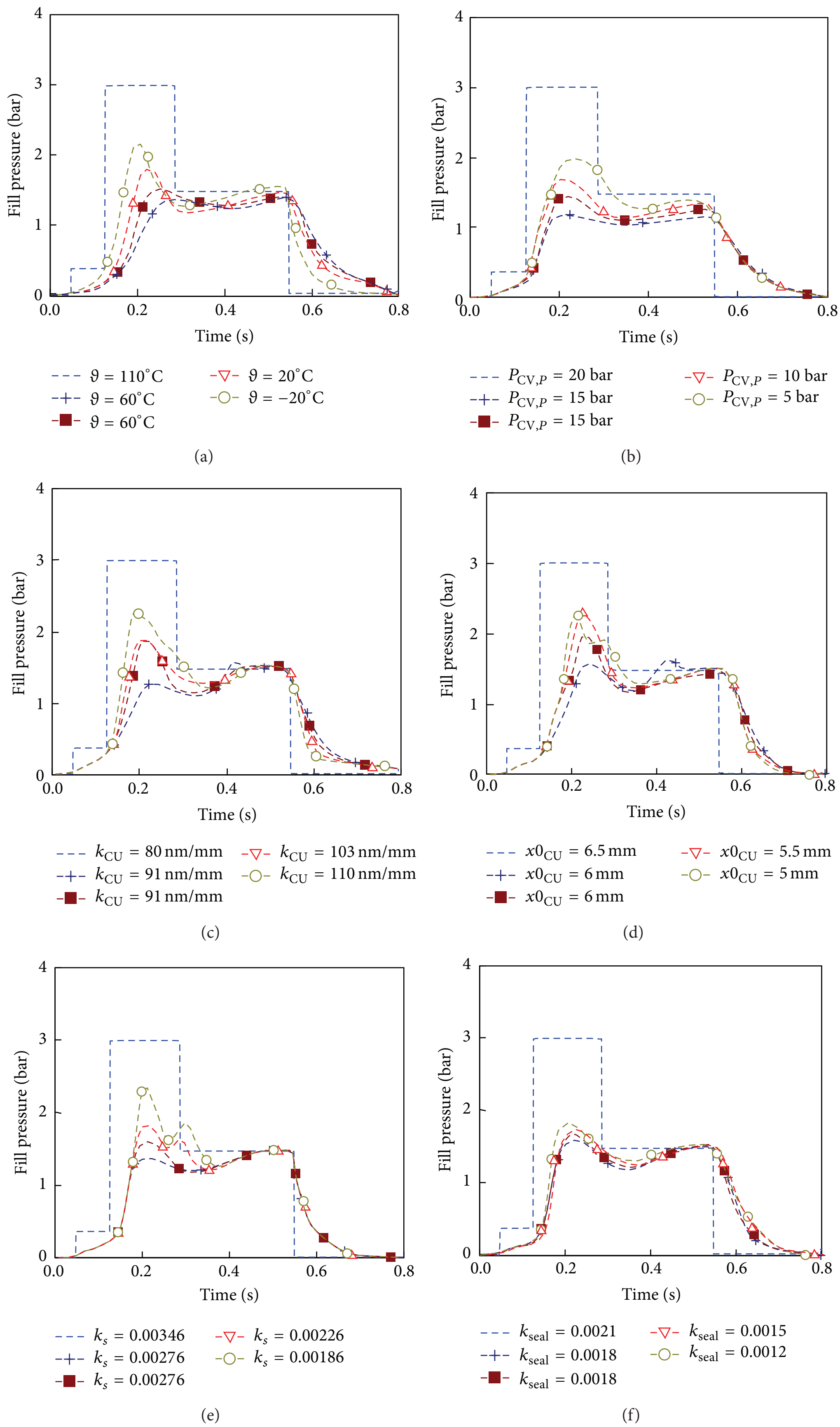


Figure 9: Oil fill results for various clutch parameters.

Once the shift valve suddenly opens wide, a large amount of ATF oil flows to the piston cylinder, which increases the flow and reduces the main line pressure (Figure 10(a)). So the main line pressure and the system flow can be used to evaluate the effects of fast fill pressure. Because the variation in main line pressure also affects other clutch pressures, it should not be reduced excessively.

The real fill pressure at the stable fill stage increases with the fast fill time (Figure 10(b)). The reason is that the filled volume and the maximal clutch piston speed increase at the fast fill stage. Subsequently, the clutch piston eliminates the gap and reaches the kiss point rapidly. So if a long fast fill time is required, overfill may occur and the shift quality may be affected.

The clutch fill is affected by the fast fill time T_{II}, the fast fill pressure P_{FP}, the stable fill time T_{III}, and the stable fill pressure P_{FTP} combined. P_{FTP} affects the transferred torque after engagement, which has priority over clutch fill control. So P_{FTP} is treated as the input for optimizing other parameters.

The fast fill time T_{II} affects clutch fill significantly (Figure 9(b)), while the stable fill time T_{III} affects it slightly. The difference of the control effect of T_{III} for different transmissions could be compensated by adjusting the fast fill time T_{II}. So the preferred strategy is to adjust T_{II} while T_{III} is usually set to be a constant, which is determined by the velocity and displacement requirements of the clutch piston. The stable fill time T_{III} is formulated as

$$T_{\mathrm{III}} = \sqrt{\frac{d \times \left[P_{\mathrm{FTP}} S_{\mathrm{CU}} - F_{\mathrm{seal}} - k_{\mathrm{CU}} \cdot \left(0.75d + x0_{\mathrm{CU}}\right)\right]}{2m_{\mathrm{CU}}}}. \quad (15)$$

In order to avoid bad shifting performance, the percentage of the clutch fill time in the whole shift time is limited strictly, such as within 47% used in this paper. So the fast fill time T_{II} is constrained by

$$\begin{aligned} T_{\mathrm{II}} &\leq T_{\mathrm{II\_max}} \\ &= 0.47 T_{\mathrm{shift}} \\ &\quad - \sqrt{\frac{d \times \left[P_{\mathrm{FTP}} S_{\mathrm{CU}} - F_{\mathrm{seal}} - k_{\mathrm{CU}} \cdot \left(0.75d + x0_{\mathrm{CU}}\right)\right]}{2m_{\mathrm{CU}}}}. \end{aligned} \quad (16)$$

Meanwhile, the fast fill pressure P_{FP} should be applicable for different conditions such as different temperatures of ATF fluid and main line pressures. Here the fast fill time T_{II} is again limited as $T_{\mathrm{II}} = 0.7T_{\mathrm{II\_max}}$. According to the control requirement, the gap between clutch plates should be eliminated with the clutch fill time, which is formulated as

$$\begin{aligned} d_{\mathrm{FP}}(t) + d_{\mathrm{FTP}}(t) = &\int_{T_I}^{0.7T_{\mathrm{II\_max}}} \dot{x}_{\mathrm{CU}}\left(P_{\mathrm{FP}}\right) dt \\ &+ \int_{0.7T_{\mathrm{II\_max}}}^{T_{\mathrm{III}}} \dot{x}_{\mathrm{CU}}\left(P_{\mathrm{FTP}}\right) dt = d. \end{aligned} \quad (17)$$

In order to engage clutches smoothly, the energy of the clutch piston in the end should be smaller than the extrusion energy W_s of ATF fluid between clutch plates; otherwise there is an impact. So the clutch fill pressures are constrained by

$$\begin{aligned} 0 \leq\ & \varsigma_{\mathrm{FP}} \cdot P_{\mathrm{FP}} \cdot S_{\mathrm{CU}} \cdot d_{\mathrm{FP}} + \varsigma_{\mathrm{FTP}} \cdot P_{\mathrm{FTP}} \cdot S_{\mathrm{CU}} \cdot d_{\mathrm{FTP}} \\ & - \frac{1}{2} k_{\mathrm{CU}} d^2 - F_{\mathrm{seal}} \cdot d \leq W_s, \end{aligned} \quad (18)$$

where ς_{FP} and ς_{FTP} are equivalent factors of the target pressure and real pressure at fast fill phase and stable fill phase, respectively.

In order to avoid that the main line pressure decreases below the safe level in fast fill phase, a safety fluid flow of ATF fluid is required. So the maximum velocity of the piston is usually limited by

$$\dot{x}_{\mathrm{CU}}\left(P_{\mathrm{FP}}\right) S_{\mathrm{CU}}\big|_{\max} \leq \lambda_c Q_{\mathrm{pump}}, \quad (19)$$

where Q_{pump} is the flow of pump; λ_c is the safety factor for flow loss.

So the fast fill pressure can be controlled as

$$P_{\mathrm{FP}} = \lambda_{\mathrm{FP}} \min\left(P_{\mathrm{FP\_max\_1}}, P_{\mathrm{FP\_max\_2}}\right) + \left(1 - \lambda_{\mathrm{FP}}\right) P_{\mathrm{FP\_min}}. \quad (20)$$

Oversized λ_{FP} would cause unstable clutch fill process; here an optimal value $\sqrt{2}/3$ is tested to be effective.

3.3.3. Stable Fill Pressure. The stable fill pressure mainly affects the final stage of clutch engagement. At this stage, because of increased spring resistance, the piston decelerates until the gap is eliminated. The simulation results for various stable fill pressures are shown in Figure 11. During period A in Figure 11, a reduced amount of ATF fluid is fed into the piston chamber because of a sudden current reduction. However, the piston continues to move rapidly, causing the fill pressure to decrease according to (12). During period B in Figure 11, when the piston speed decreases to a certain level, namely, $Q_{\mathrm{CU}} > \dot{x}_{\mathrm{CU}} S_{\mathrm{CU}}$, the fill pressure increases to the target level gradually. In an ideal situation, the clutch pressure should be only P_{KP} and the piston speed should equal zero when the fill is nearly completed. However, numerous factors affect fill pressure; consequently, realizing the ideal situation is nearly impossible. Nevertheless, the robustness of the clutch fill control can be improved by reducing the fast fill pressure and increasing the stable fill pressure. Clearly, reducing the fast fill pressure reduces the maximal piston speed; increasing the stable fill pressure ensures that the fill pressure reaches the kiss-point pressure before the fill process is completed. An appropriate stable fill pressure P_{FTP} value can be calculated by

$$P_{\mathrm{FTP}} = P_{\mathrm{KP}} + \frac{10 \cdot k_{T2P}}{\left(e^{\kappa - 1}\right)^2}. \quad (21)$$

4. Automatic Test Method for Determining Initial Clutch Fill Parameters

Because of distinct inconsistencies in the mass production of transmissions and in the open-loop control characteristics

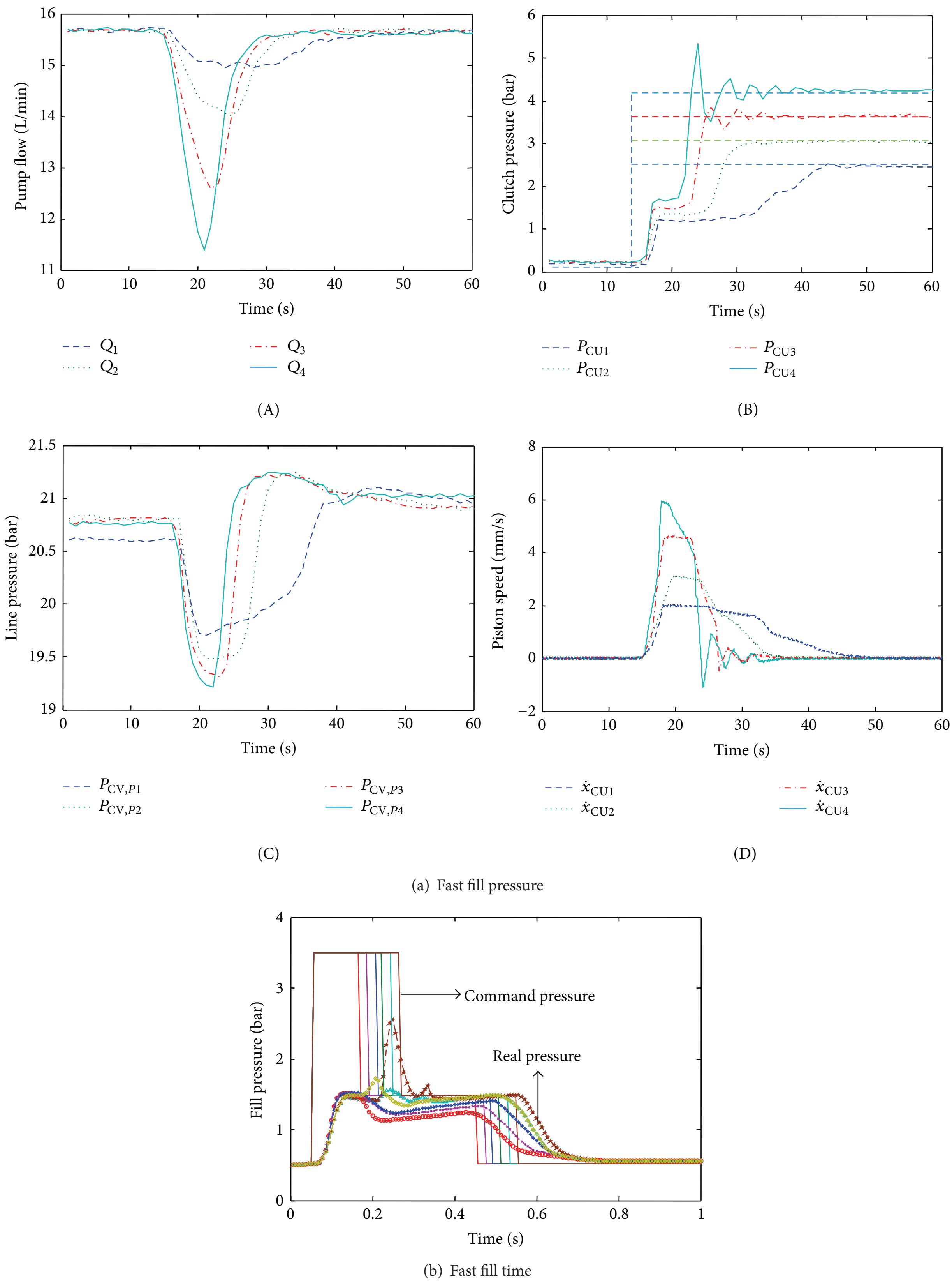


FIGURE 10: Pressure response at various target pressures and fast fill times.

of clutch fill pressure, obtaining the initial fill parameters is crucial to improving shift quality. The direct method is to calculate fill parameters based on obtained clutch pressure signals by sensors from the transmission end-of-line (EOL) test according to previos equations.

In the EOL test, because the allowed testing time is short, only one key control parameter can be tested while the other control parameters remain unchanged. By analyzing the effects of the aforementioned parameters on fill pressure, the fast fill time can be adjusted to alter the characteristic parameters, thus enabling the clutch pressure to reach the kiss-point pressure P_{KP} within the preset time. The clutch fill testing scheme is formulated as shown in Figure 12. During testing, the fast fill time gradually increases and whether the fill requirement is satisfied can be determined by observing three conditions denoted as $G1$, $G2$, and $G3$, respectively, in Figure 12. $G2$ is defined from T_{start2} to T_{end2}. $G1$ is defined from T_{start1} to T_{end1}. The condition $G1$ requires that, immediately after the fast fill stage ends, the pressure is both below P_{KP} and within the range of P_{L1} to P_{H1}, ensuring that the piston moves smoothly. The condition $G2$ requires that, after the fill stage ends, the clutch pressure is exceeding P_{KP}, ensuring that the piston reaches the final engaged point within the preset time. The condition $G3$ requires that, during the fast fill stage, the maximal fill pressure cannot exceed P_{H3}. The ranges for $G1$, $G2$, and $G3$ are related to the mass tolerance ranges of the transmission components.

In the fill phase, the clutch pressure within the range of $G1$ and $G2$ is associated with the fast fill time. This pressure can be increased by increasing the fast fill time; therefore, the optimal fast fill time can be obtained by applying the learning rules of the EOL test shown in Table 1. Figure 13 shows the automatically measured EOL results. Beginning from 120 ms, the fast fill time increases in increments of 10 ms. As the fast fill time increases, the pressure at $G1$ and $G2$ increases until it reaches the required level at 170 ms. Moreover, during each fast fill stage, the main line pressure of the system decreases suddenly and causes flow deficiency when a large amount of ATF fluid instantly enters the piston cylinder.

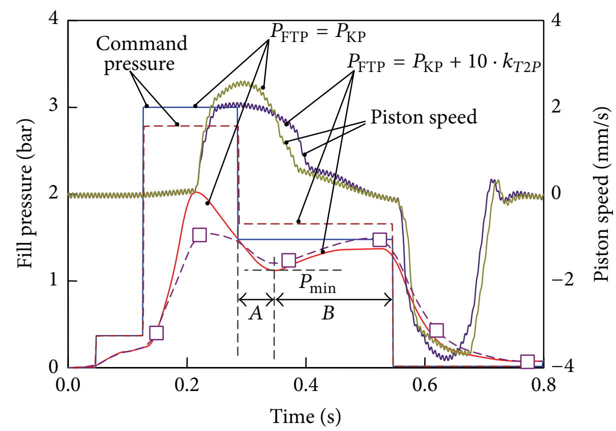


Figure 11: Fill results at various stable fill pressures.

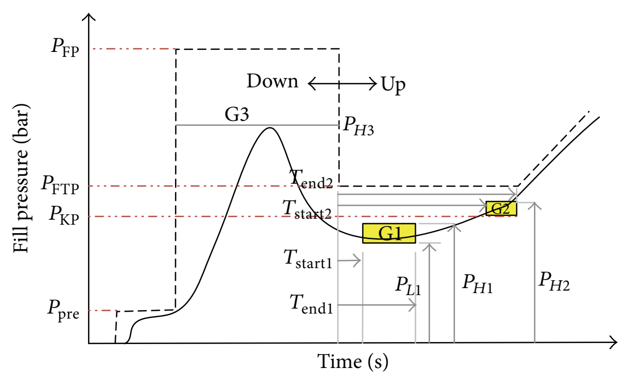


Figure 12: Fill end-of-line test method based on the pressure feedback.

Table 1: Learning rules of the fill end-of-line test.

G1	G2	G3	Fill learning rule
$P < P_{L1}$	$P < P_{KP}$	$P < P_{H3}$	Up
$P_{L1} \leq P \leq P_{H1}$	$P < P_{KP}$	$P < P_{H3}$	Up
$P > P_{L1}$	*	$P < P_{H3}$	Down
$P_{L1} \leq P \leq P_{H1}$	$P > P_{H2}$	$P < P_{H3}$	Down
$P_{L1} \leq P \leq P_{H1}$	$P_{KP} \leq P \leq P_{H2}$	$P < P_{H3}$	Correct
Other conditions			Fail

5. Effects of Overfill and Underfill on Shift Quality

Shift quality optimization is applied to ensure that the torque from the offgoing clutch is transferred smoothly to the oncoming clutch without flares or tie-up.

5.1. Effects of Underfill. Because numerous factors affect clutch fill and an open-loop is used for pressure control, the piston cannot be ensured to reach the kiss point exactly when the clutch fill phase ends. Figure 14 shows how underfill influences clutch shifting processes. When clutch fill is completed, the clutch pressure does not reach the kiss point at the beginning of the torque exchange stage. However, because of underfill, the pressure of the oncoming clutch cannot remain equal to the target pressure at the initial stage of torque exchange, causing the engine load to decrease and, thus, engine flare to occur. If engine flare continues for a long time at a high rotational speed, the clutch friction plates generate a high amount of heat. Therefore, underfill would shorten the life of clutches. Moreover, engine flare aggravates the slipping of clutches and causes the line pressure to increase based on the PI (proportional-integral) control of the offgoing clutch. Although clutch slipping can be reduced, it causes fluctuation in the rotational speed (Figure 14).

5.2. Effects of Overfill. Overfill means that a certain amount of torque is transferred by clutches during the clutch fill phase. Overfill occurs when the fill pressure is too high. Figure 15 shows the pressure response of the clutch when overfill occurs. At the fill stage, after the fast fill period ends, the pressure is still higher than P_{FTP}; consequently, the oncoming

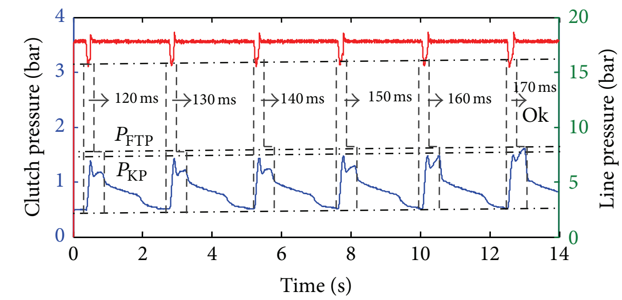


Figure 13: Automatically measured fill results.

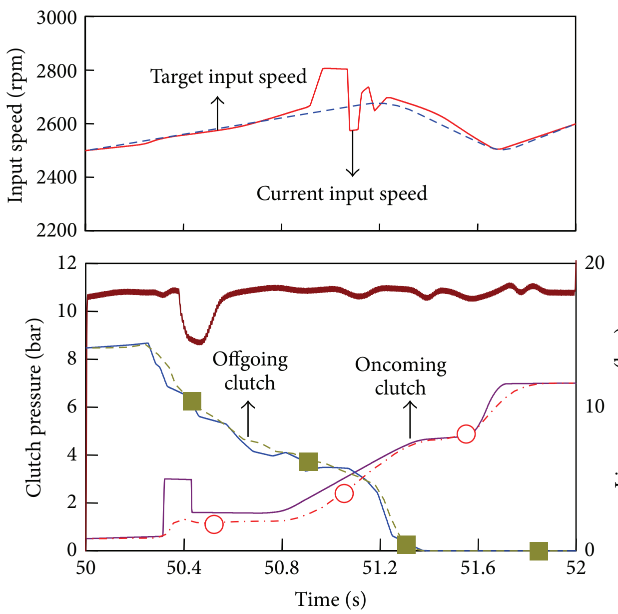


Figure 14: Effects of underfill.

clutch can transfer a small amount of torque. This inevitably causes a sudden increase in engine load, causing the rotation speed of the transmission input shaft to decrease rapidly, resulting in negative slip of the offgoing clutch. Typically the pressure is reduced through PI control to compensate for the negative slip. Negative slip that still exists before the torque phase considerably affects the torque exchange phase because, at this moment, the slow decrease in the torque of the offgoing clutch and the rapid increase in the torque of the oncoming clutch cause a substantial shift impact. If a small amount of positive slip exists before the torque phase begins, then a buffer zones is used for the torque exchange control; thus, the shift impact can be avoided. In conclusion, overfill causes transmission shift impact to occur.

6. Fuzzy-Adaption-Based Correction Method

Since certain components, especially those in constant motion such as the clutch return spring and the seal ring, exhibit performance decay after being used for a long time, the characteristics of these components change as the working time increases. The EOL test data represent only the initial

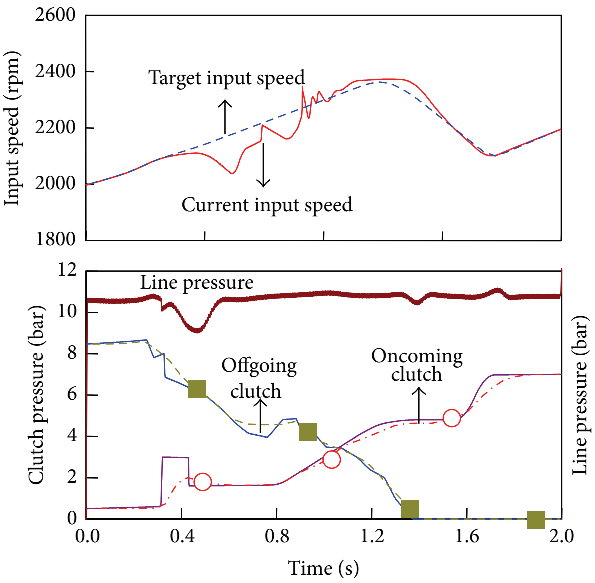


Figure 15: Effects of overfill.

characteristics of transmissions. Therefore, achieving clutch fill control by using the adaption method is the prerequisite for ensuring shift quality throughout the entire life cycle of a transmission. Figure 16 shows the proposed adaption control strategy for the clutch fill process. The current clutch slip is used to evaluate the current shift quality during the shifting process and to optimize the fill control parameters. Because the fill control affects only the shift quality at the fill and torque exchange stages, the fill control parameters T_{II} and P_{FTP} can be optimized by monitoring the clutch slip based on $W1$ and $W2$, shown in Figure 16.

According to the input speed n_i, the output speed n_o, and the ratio of current gear, the clutch slip n_s can be calculated by

$$n_s = n_i - n_c = n_i - n_o \cdot i_c. \quad (22)$$

Both the fast fill time T_{II} and stable fill pressure P_{FTP} affect clutch fill results. The stable fill pressure P_{FTP} exerts a more evident influence on the speed adjustment required for the clutch to reach the kiss-point pressure than T_{II} does but easily leads to excessively high pressure and so affects shift quality. The fast fill time T_{II} determines the movement speed of the piston after the fast fill stage. The correction coefficients of the control parameters P_{FTP} and T_{II} can be obtained by using an empirical fuzzy adaption control method, which is described in Figure 17. The input variable of fuzzy adaptive control is the clutch slip n_s. The clutch slip is processed fuzzily and then fuzzy subsets $F1$ and $F2$ are obtained, respectively. The correction factors K_P and K_T for fast fill pressure and fill time are then calculated. First, the current clutch slip domain in region $W1$ is defined as $W1 = \{k_1, k_2, k_3, -20\,\text{rpm}\}$ and the fuzzy subset of which is {NB, NM, NS, ZO}. The current clutch slip domain in region $W2$ is defined as $W2 = \{k_4, k_5, -20\,\text{rpm}, 40\,\text{rpm}, k_6, k_7, k_8\}$, the fuzzy subset of which is {NB, NS, ZO, PS, PM, PB}. Based on the following observations, the fuzzy control rules shown in Tables 2 and 3 are used.

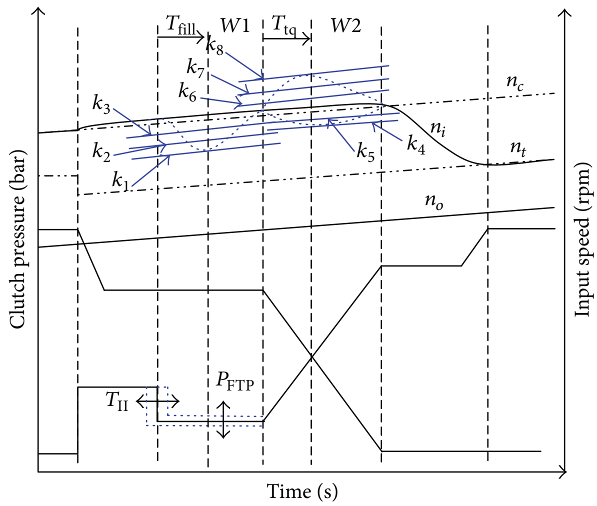


Figure 16: Adaption control strategy for the clutch fill process.

Table 2: Empirical fuzzy control rules for fast fill time.

K_T	W2 NB	NS	ZO	PS	PM	PB
$W1$						
NB	NB	NB	NB	NB	NB	NB
NM	NB	NB	NM	NS	NS	NS
NS	NM	NS	NS	ZO	ZO	PS
ZO	ZO	ZO	ZO	PS	PM	PB

Table 3: Empirical fuzzy control rules for stable fill pressure.

K_P	W2 NB	NS	ZO	PS	PM	PB
$W1$						
NB	NM	NM	NM	NM	NS	NS
NM	NM	NM	NS	NS	NS	NS
NS	NM	NS	NS	ZO	ZO	PS
ZO	NS	NS	ZO	PS	PS	PM

(1) The control priority level for adjusting fill control parameters is higher when negative slip occurs in $W1$ and $W2$ than when positive slip occurs.

(2) In $W1$ and $W2$, the fill control parameters are not adjusted within the slip range of −20 to 40 rpm.

(3) The priority for adjusting fast fill time T_{II} is higher than that for adjusting the stable fill pressure P_{FTP}.

(4) When a negative slip occurs in $W2$, only the stable fill pressure P_{FTP} must be adjusted.

Based on the rules shown in Tables 2 and 3, the correction coefficients for fast fill time T_{II} and stable fill pressure P_{FTP} can be obtained using

$$\begin{aligned} T_{II}(n) &= T_{II}(n-1) + 20 \cdot K_T, \\ P_{FTP}(n) &= P_{FTP}(n-1) + 0.2 \cdot K_P. \end{aligned} \tag{23}$$

Figures 18 and 19 show the adaption result for the fast fill time T_{II} and stable fill pressure P_{FTP} of clutch $C4$ during the 200,000 km transmission durability test. The fill control parameters should be corrected when the transmission is being used to enable the fast fill time and stable fill pressure to meet the shift quality requirements. Figure 18 shows that a high main line pressure results in a short fast fill time and a high clutch torque coefficient results in a low stable fill pressure. In Figure 19, the horizontal axis is the clutch torque coefficient, namely, the ratio coefficient of the clutch torque to the input shaft torque of the transmission, which is related to the mechanical structure of the transmission. The clutch torque coefficient varies when using different gears. The test results indicate that the transmission characteristics change as the mileage increases especially the first 8,000 km.

7. Conclusions

(1) A mathematical model of an electrohydraulic clutch shift control system in an automatic transmission was presented.

(2) By analyzing the effects of key model parameters on clutch filling process, four special dynamic characteristic parameters were chosen for optimal clutch fill control.

(3) An automatic method was proposed for testing initial clutch fill parameters.

(4) In order to prevent the underfill or overfill in clutch fill phase, a fuzzy adaption control method was proposed. 200,000 km road vehicle test results verified that this method can effectively prevent the clutch shift quality from declining through the natural decay of the performance of components during the life cycle of the transmission.

Nomenclature

SV, CV, CU, SP:	Solenoid valve, shift control valve, clutch, and return spring, respectively
$m_{\text{sub1,sub2 (optional)}}$:	Mass of the spool sub2 of the part sub1, and only one subscript is used if there is only one spool
$Q_{\text{sub1,sub2}}$:	Fluid flow at port sub2 of the part sub1
$Q^{\text{leak}}_{\text{sub1}}$:	Leakage of the part sub1
$P_{\text{sub1,sub2}}$:	Pressure at port sub2 of the part sub1
$G_{\text{sub1,sub2 (optional)}}$:	Electromagnetic force exerted on the spool sub2 in the part sub1, and only one subscript is used if there is only one spool
$F_{\text{sub1,sub2 (optional)}}$:	Spring force exerted on the spool sub2 in the part sub1, and only one subscript is used if there is only one spool
F_{sub1sub2}:	Force exerted on the mass sub1 from the mass sub2
k_{sub1}:	Stiffness of the return spring of the part sub1
b_{sub1}:	Stamping coefficient of the part sub1

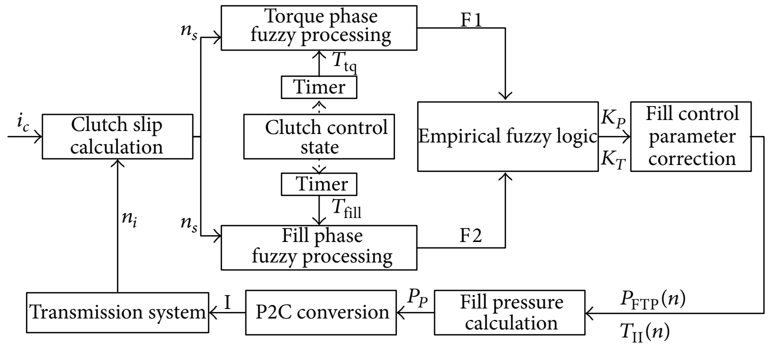


FIGURE 17: Fill fuzzy adaptive control system diagram.

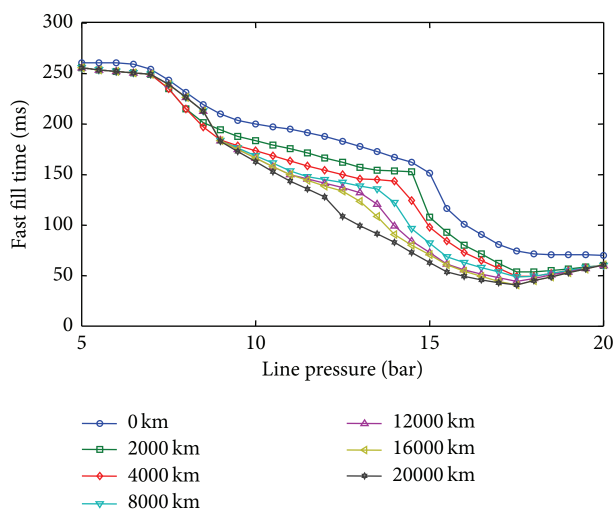


FIGURE 18: Adapted fast fill time during the entire life cycle of the transmission.

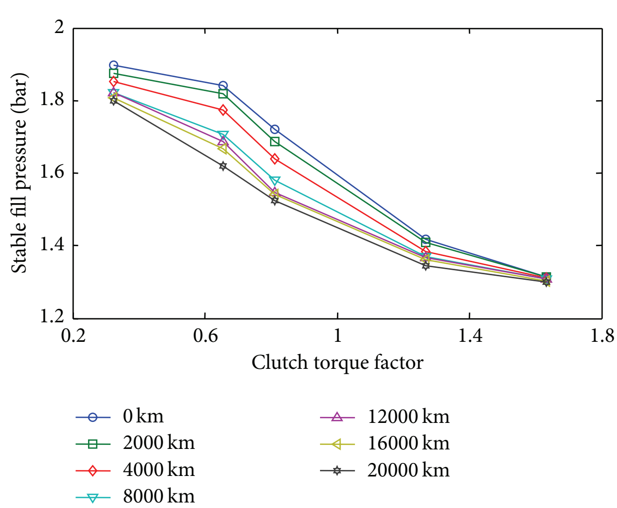


FIGURE 19: Adapted stable fill pressure during the entire life cycle of the transmission.

$x_{\text{sub1,sub2 (optional)}}$: Displacement of the mass sub2 of the part sub1
$x0_{\text{sub1,sub2 (optional)}}$: Preload of the mass sub2 of the part sub1
$\dot{x}_{\text{sub1,sub2 (optional)}}$: Velocity of the mass sub2 of the part sub1
$\ddot{x}_{\text{sub1,sub2 (optional)}}$: Acceleration of the mass sub2 of the part sub1
$S_{\text{sub1,sub2 (optional)}}$: Sectional area of the chamber sub2 of the part sub1
$V0_{\text{sub1,sub2 (optional)}}$: Initial volume in the chamber sub2 of the part sub1
$A_{\text{sub1,sub2}}$: Opening area of the port sub2 of the part sub1
e_{sub1}: Voltage applied in the coil of the part sub1
L_{sub1}: Inductance in the part sub1
R_{sub1}: Electric resistance of the part sub1
i_{sub1}: Electric current in the coil of the part sub1
Ev_{sub1}: Back EMF coefficient of the part sub1
Ef_{sub1}: Magnetic force coefficient of the part sub1
β_{sub1}: Bulk modulus of the part sub1
Cd_{sub1}: Unitless discharge coefficient of the part sub1
ρ: Density of the automatic transmission fluid (ATF)
k_{seal}: Sealing force of the clutch piston
k_s: Static frictional resistance coefficient of the clutch piston
F_{stick}: Static viscous friction of the clutch piston
P_{pre}: Prefill pressure of the clutch piston
P_{FP}: Fast fill pressure of the clutch piston
P_{FTP}: Stable fill pressure of the clutch piston
P_{KP}: Kiss-point pressure when the clutch starts to transfer torque
F_{APP}: Cylinder force exerted on clutch plates
k_{T2P}: Torque-pressure characteristic coefficient of the clutch
κ: Relative torque coefficient of the input shaft in the current gear
T_{I}: Prefill time of the clutch piston
T_{II}: Fast fill time of the clutch piston
T_{III}: Stable fill time of the clutch piston

T_{shift}: Required shift time
ϑ: Temperature of ATF fluid
λ_{var1}: Weight factors of the variable var1
h_c: Vertical height from port A of the shift value to the center of clutch piston chamber
d: Maximum stroke of the clutch piston
d_{per}: Stroke of the clutch piston in clutch fill phase per
α: Damping coefficient of O-type seal ring.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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