Automotive Transmission Clutch Fill Optimal Control: An Experimental Investigation

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Abstract—Clutch to clutch shift control technology, which is the key enabler for a compact and low cost automotive transmission design, is important for both automatic and hybrid transmissions. To ensure a smooth clutch to clutch shift, precise synchronization between the on-coming and off-going clutches is critical. This further requires the on-coming clutch to be filled and ready for engagement at the predetermined time. To optimize this process, the clutch fill was formulated as an optimization problem in our previous work and a customized dynamic programming method was proposed as a solution. Following this idea, this paper presents the clutch fill experimental setup and the optimal control implementation.

I. INTRODUCTION

The automotive transmission, as a key component of the vehicle propulsion system, is designed to transfer the engine torque to the vehicle driveline with desired ratio smoothly and efficiently. The main components of a typical automatic transmission include the torque converter, the planetary gear set and the clutch system. The power transfer and gearshift is in fact realized by engaging and disengaging the clutch systems [1-3]. For a smooth clutch to clutch shift, precise synchronization of the on-coming clutch and off-going clutch is critical, which otherwise will cause undesirable torque interruption and oscillation [4]. To ensure precise synchronization, before clutch engagement, it is necessary to actuate the oncoming clutch to a position where the clutch packs are in contact. This process is called clutch fill and plays an important role for the clutch to clutch shift technology.

The commonly used clutch actuation devices in transmissions are electro-hydraulically actuated clutches. A schematic diagram of a typical transmission clutch actuation system is shown in Figure 1. Once the clutch is to be engaged, pressurized fluid flows into the clutch chamber and pushes the clutch piston towards the clutch packs until they are in contact (clutch fill). At the end of the clutch fill process, the input pressure \( P_s \) is further increased, which then squeezes the clutch piston to the clutch packs to transfer the engine torque to the vehicle driveline. It is imperative to control the clutch piston to reach the clutch packs within a specified clutch fill time because an improper clutch fill process can result in either an under-fill or an over-fill [4], both of which can cause the failure of the clutch shift synchronization and therefore negatively affect the clutch shift quality. Therefore, designing the clutch fill pressure command is critical to achieve a fast and precise clutch fill and a smooth start to the clutch engagement process. Moreover, it is desirable to synthesize the clutch fill pressure to reduce the peak flow demand during the clutch fill process. This feature would enable a smaller transmission pump to improve vehicle fuel economy [5]. However, there are two main challenges associated with the clutch fill control design. First, even small errors in calculating the clutch fill pressure and fill time could lead to an over-fill or an under-fill, which will adversely impact the shift quality. Second, currently there is no pressure sensor inside the clutch chamber, and consequently a pressure feedback control loop could not be formed. Therefore, it is necessary to design an open loop pressure control profile, which should be optimal in the sense of peak flow demand and also robust in terms of clutch fill time. Clearly, the traditional approach based on manual calibration is not effective to achieve the above objectives. In this paper, we will present a systematic approach to solve this problem.
experimental investigation of the optimal clutch fill control. Due to the compact design of the clutch pack, it is necessary to select the proper sensors that can fit the tight space constraint. This requirement has been addressed when designing the experimental setup. In addition, to enable a systematic and model based control design, a precise dynamic model of the clutch fill process is necessary. Previous works [7-8] on clutch control mainly focused on dynamic models during the clutch engagement process. The clutch engagement is typically operated at a much higher pressure than the clutch fill, and the existence of the large clutch engaging reaction force make it reasonable to neglect several types of forces with smaller magnitude. However, due to the different operating condition, it is necessary to construct a physically sounded model capturing the key dynamics of the clutch fill. Through a series of experiments, the dynamic model parameters are identified. To this end, the optimal clutch fill input pressure designed by the customized dynamic parameters are identified. To this end, the optimal clutch fill control problem as an optimization problem. Section 3 presents the system model and formulates the clutch fill control problem as an optimization problem. Section 3 presents the system model and formulates the clutch fill control problem as an optimization problem. Section 4. Conclusion is provided in section 5.

II. PROBLEM DESCRIPTION

A. System Modeling

A simplified schematic diagram of the transmission clutch system is shown in Figure 1. \( p_p \) is the supply pressure command and also the control input to the system, \( p_p \) is the pressure inside the clutch chamber, and \( x_p \) is the clutch piston displacement. The pressurized fluid flows into the clutch chamber and pushes the clutch piston to the right, and finally contacts the clutch pack. We call this process clutch fill. The dynamics associated with clutch fill could be modeled as:

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{1}{M_p} \left[ A_p (x_3 + P_c - P_{atm}) - D_p x_2 - F_{drag} (x_3 + P_c, x_2) - K_p (x_1 + x_{po}) \right] \\
\dot{x}_3 &= \frac{\beta}{V_0 + A_p x_1} \left[ \text{sign} (u - x_p) C_d A_{orifice} \sqrt{\frac{2}{\rho} \frac{u - x_p}{x_p} - A_p x_1} \right] \\
\end{align*}
\]

where \( x_1 \) is the clutch piston displacement, \( x_2 \) is the clutch piston velocity, \( x_3 \) is the clutch chamber pressure, \( u \) is the supply pressure control input, \( M_p \) is the effective mass of the piston, \( A_p \) is the piston surface area, \( D_p \) is the clutch damping coefficient, \( P_{atm} \) is the atmospheric pressure, \( K_p \) is the return spring constant, \( x_{po} \) is the return spring preload, \( \beta \) is the fluid bulk modulus, \( V_0 \) is the chamber volume, \( C_d \) is the discharge coefficient, \( A_{orifice} \) is the orifice area, and \( \rho \) is the fluid density. \( F_{drag} \) is the piston seal drag force, which is dependent on the piston motion. It is modeled as:

\[
F_{drag} = \begin{cases} k_m (x_3 + P_c) + c_m \frac{x_2}{\alpha} & (x_2 \neq 0) \\
F_{stick} & (x_2 = 0) \end{cases}
\]

where \( k_m \) and \( c_m \) are constant, \( \alpha \) is the piston seal damping coefficient, and \( F_{stick} \) is the static stick friction force from the Kanopp’s stick-slip model [9]. The stick friction is often neglected in the clutch dynamic models for engagement [7-8], as it is relatively small comparing with the clutch engagement force. But due to the lower operating pressure during the clutch fill, the stick friction force becomes critical. For numerical stability, it is assumed that the drag force is \( F_{stick} \) when the piston velocity \( x_2 \) is within a small interval around zero [9], which is called the stick region as shown in Figure 2. When the velocity is in this region, the value of \( F_{stick} \) is to balance the net force and the piston is assumed to be static. Moreover, there is a maximum value constraint for the stick friction. If the net force exceeds the maximum stick friction, the piston will move. The maximum stick friction, which is noted as \( F_{static} \), is proportional to the clutch chamber pressure and can be modeled as:

\[
F_{static} = k_s (x_3 + P_c) + c_s
\]

where \( k_s \) and \( c_s \) are constant.

In addition, \( P_c \) is the centrifugal force induced pressure generated from the rotation of the clutch assembly [6].

B. Formulation of the Clutch Fill Control Problem

To enable a fast and precise clutch fill, the clutch piston must travel exactly the distance \( d \), which is required for the piston to contact the clutch pack, in the desired clutch fill time \( T \). Also, at time \( T \), the piston velocity \( x_2 \) should be zero, and the pressure force inside the chamber must be equal to the spring force in order to keep the piston in contact with the
clutch pack. These requirements can translate into a set of final conditions that the system must satisfy:

\[
\begin{align*}
  x_i(T) &= d, \\
  x_j(T) &= 0 \quad \text{and} \\
  x_k(T) &= \frac{K_p(d + x_m) + P_{\text{ave}} - P_c}{A_p}
\end{align*}
\]  

(6)

where \( d \) is the desired clutch stroke, and \( x_i(T), x_j(T) \) and \( x_k(T) \) are the final states.

Among the controls that can bring the clutch from initial states to the final states (6), we would like to take the one that has minimum peak flow demand. This will enable a smaller displacement transmission pump, which in turn will improve the vehicle fuel economy and reduce cost [5]. To reduce the peak flow demand, we need to minimize the peak value of the piston velocity \( x_2 \) during the clutch fill process since the clutch fill flow is proportional to the piston velocity. Therefore, \( x_2 \) should quickly approach to the average velocity, and stay at the average velocity as long as possible, and at the end goes to the final state conditions as shown in Figure 3. Note that the total area enclosed by the \( x_2 \) trajectory should be the piston displacement \( d \).

![Piston velocity versus time](image)

Figure 3. Desired trajectory of \( x_2 \)

In addition, the piston velocity can not increase too fast at the beginning of the clutch fill. Before the clutch starts moving, the clutch chamber pressure increases and therefore the stick friction on the clutch piston also increases. The stick friction will reach its maximum value \( F_{\text{static}} \) and then the clutch piston will move. Once moving, the piston drag force will switch from stick friction to drag force in motion, which is smaller than the \( F_{\text{static}} \). This friction force transient is non-smooth and nonlinear [9], which makes it difficult to track and control a fast changing piston velocity profile given the fact that there is currently no feedback sensor for the clutch fill process. Therefore, in Figure 4, the initial velocity of the clutch fill is designed to be small for a short duration and then go up quickly to its peak value. This profile will enable the piston to start slipping slowly at the beginning of clutch fill and therefore avoid the sticking behavior while quickly increasing the velocity.

However, the above considerations have not taken system robustness into account. In particular, the solenoid valve, which is used to generate the input pressure command \( u \), has time delay and subsequently results in the shift of \( x_2 \) trajectory as shown in Figure 4. Note that the final time \( T \) is fixed, so the piston could not travel to the desired distance \( d \) due to the shift and the difference between the desired trajectory and the shifted one will be

\[
\Delta d = \int_{T-dT}^{T} x_2(t)dt
\]  

(7)

Therefore, to minimize \( \Delta d \), the value of \( x_2(t) \) between time \( T-dT \) and \( T \) should be as small as possible as shown in Figure 4, and we can claim that the unique shape of \( x_2 \) trajectory with a tail will enhance robustness of the system for time delay. In addition, from (6), we can see that the clutch fill final states are determined by the spring stiffness \( K_p \), the piston area \( A_p \), the spring preload \( x_m \), and the centrifugal pressure \( P_c \). \( K_p, A_p, x_m \) and \( x_m \) can be measured accurately and will not change much with the environment. \( P_c \) can also be accurately determined based on the transmission rotational velocity, and it in fact has less influence on the final condition due to its small magnitude comparing with the other forces. Therefore the final states that the clutch fill system can reach are typically robust. Furthermore, from (2) and (3), \( K_p, A_p, \) and \( x_m \) are also the main parameters that determine the clutch dynamics at low velocity, therefore the clutch fill dynamic model and control will be more robust in the low velocity region than they are in the high velocity region. Therefore even if the clutch fill could not follow the desired trajectory due to the uncertainties of other model parameters in the transient and high velocity region, the control at the low velocity region (from \( T_2 \) to \( T \) in Figure 4) will be robust and continue pushing the clutch piston to approach the desired final position. However, we notice that the longer the tail is, the higher the peak value would be for \( x_2 \). This represents the trade-off between peak flow demand and system robustness.

![Shifted trajectory of \( x_2 \)](image)

Figure 4. Shifted trajectory of \( x_2 \)

Now we are ready to formulate the clutch fill control as an optimization problem. The cost function of the optimization problem is:

\[
g = \int_{0}^{\frac{x_i(T) - d}{T}} (x_i(t) - \frac{d}{T}) dt + \lambda_i \int_{\frac{x_i(T) - d}{T}}^{\frac{x_i(T) - d}{T}} (x_i(t) - \frac{d}{T}) dt + \lambda_i \int_{\frac{x_i(T) - d}{T}}^{T} (x_i(t) - \frac{d}{T}) dt + \lambda_i \int_{\frac{x_i(T) - d}{T}}^{T} (x_i(t) - \frac{d}{T}) dt + \lambda_i \int_{\frac{x_i(T) - d}{T}}^{T} \left( \frac{K_p(d + x_m) + P_{\text{ave}} - P_c}{A_p} \right) dt
\]  

(8)

In particular, the first term of the cost function ensures the piston to start with a low velocity \( v_m \). And the second term
ensures that the velocity $x_2$ remains close to the average velocity $d/T$, which will minimize the peak value of $x_2$ and therefore the peak flow demand. The third term ensures $x_2$ to be as small as possible (close to zero) from time $T_1$ to final time $T$, which will enhance system robustness. The last three terms ensures that the system will reach the specified final conditions in the required time $T$. $\lambda_1$, $\lambda_2$, $\lambda_3$, $\lambda_4$, and $\lambda_5$ are the weighting factors.

III. OPTIMAL CONTROL DESIGN

A systematic solution to the above optimization problem can be determined recursively via the customized dynamic programming method introduced in [6]. In brief, the optimization process can be described as follows:

**Step 1.** Define $X(k) = [x_1(k), x_2(k), x_3(k)]^T$. The system model (1) is first discretized into $X(k+1) = f(X(k), u(k))$. Due to its unique structure, the discrete model can be formulated into the following reverse model, which is explained in detail in [6].

$$x_1(k) = x_1(k+1) - \Delta x_1(k) = R_1(k)$$

$$x_2(k) = R_2(k)$$

$$u(k) = \text{sign}(W)(\frac{W}{C_d A_{\text{inlet}}} + \frac{p}{2} + x_3(k)) = R_3(k)$$

where

$$W = (x_1(k+1) - x_1(k)) \times \left( \frac{V_0 + A_p x_1(k)}{\Delta t \beta} + A_p x_2(k) \right)$$

The equation $R_2(k)$ determines the in-chamber pressure $x_3(k)$ in the step $k$ and involves the drag force $F_{\text{drag}}$ term, which has different models depending on the piston velocity $x_2$. When $x_2(k)$ is not zero, $F_{\text{drag}}$ can be obtained from equation (4) and then $R_2(k)$ is

$$R_2(k) = \frac{1}{A_p - k_m \times \text{tanh} \left( \frac{x_2(k)}{\alpha} \right) \times \left( x_1(k+1) - x_2(k) \right) \times \frac{M_p}{\Delta t} - A_p \times k_j \times \text{tanh} \left( \frac{x_2(k)}{\alpha} \right) \times \left( x_1(k+1) - x_2(k) \right) \times \frac{M_p}{\Delta t}}$$

$$= A_p \times (P_r - P_{\text{atm}}) + D_p x_2(k) + K_p (x_1(k) + x_{\text{ro}})$$

$$+ (c_m + k_m P_s) \times \text{tanh} \left( \frac{x_2(k)}{\alpha} \right)$$

When $x_2(k)$ is zero but $x_3(k+1)$ is nonzero, which means that the piston starts moving, the $F_{\text{drag}}$ is assumed to be the maximum static friction $F_{\text{static}}$ in (5) and $R_2(k)$ becomes

$$R_2(k) = \frac{1}{A_p - k_j \times \text{tanh} \left( \frac{x_2(k)}{\alpha} \right) \times \left( x_1(k+1) - x_2(k) \right) \times \frac{M_p}{\Delta t} - A_p \times k_j \times \text{tanh} \left( \frac{x_2(k)}{\alpha} \right) \times \left( x_1(k+1) - x_2(k) \right) \times \frac{M_p}{\Delta t}}$$

$$= A_p \times (P_r - P_{\text{atm}}) + D_p x_2(k) + K_p (x_1(k) + x_{\text{ro}})$$

$$+ (c_j + k_j P_s) \times \text{tanh} \left( \frac{x_2(k)}{\alpha} \right)$$

When both $x_2(k)$ and $x_3(k+1)$ are zero, which means that the piston stays static, the chamber pressure $x_3(k)$ variation is assumed to be small and then $R_3(k)$ becomes

$$R_3(k) = x_1(k+1)$$

For notation simplicity, we can denote Eq. (9), (10) and (11) as $[x_1(k), x_2(k), u(k)] = R(x_1(k+1), x_2(k+1), x_3(k+1), x_3(k))$. **Step 2.** The dynamic programming is modified to reduce the computational complexity by taking advantage of the unique structure of the system model. The proposed customized Dynamic Programming algorithm is explained in [6].

IV. SIMULATION AND EXPERIMENTAL RESULTS

A. Experimental Setup

To test the proposed optimal clutch fill control, a transmission clutch control test bed is designed and built. The main parts include a servo motor, an automotive transmission pump, a pilot-operated proportional relief valve, a proportional reducing/relieving valve, two pressure sensors, a clutch mounting device/fixtures, a displacement sensor, a power supply unit with servo amplifier and an XPC-target real time control system.

The clutch assembly with associated sensor instrumentation is shown in Figure 5. The clutch piston displacement is measured by a micro gauging differential variable reluctance transducer (MGDVRT). Given the compact structure of the clutch system and the small travel distance (0.58 mm) of the clutch piston, the displacement sensor must have a small size but high resolution (1.5 µm for the selected one). The selected MGDVRT sensor measures the differential reluctance when the core of the sensor moves in the magnetically shielded coil. To avoid the magnetic influence from the clutch shell, a sensor mount made of an acrylic rod, which is a non conductive material, is fabricated and attached to the sensor body to provide additional magnetic shielding. The sensor mount also has a rear outer thread to allow the rod to be rotated forward or backward and therefore can be used to adjust the sensor position. Besides, a very accurately machined parallel sets are used as spacers between the clutch mounting plate and the sensor mount to ensure that the sensor is perpendicular to the surface of the clutch piston. In addition, the clutch system input pressure is measured using an Omega pressure sensor with measurement range from 0 to 30 psi.
B. System Identification

To implement the dynamic programming and optimal control, the parameters in the clutch system model (1-3) need to be identified. The effective mass of the piston \( M_p \), the piston surface area \( A_p \), the return spring constant \( K_p \), the preload distance of the return spring \( x_{ps} \), the stick friction peak value \( F_{static} \), the orifice area \( A_{orifice} \), the fluid density \( \rho \), the discharge coefficient \( C_d \), the bulk modulus \( \beta \) and the clutch chamber volume \( V_0 \) can be measured directly. As the clutch is not rotating with the lab experimental setup, the centrifugal pressure \( P_c \) is zero.

The \( F_{static} \), which is the maximum drag force when the clutch stays static, is measured by recording the clutch chamber pressure at the start of the piston motion. In the experiment, the clutch piston is kept static at the specific lift and then moves to the next position as shown in Figure 6. The calibrated \( F_{static} \) friction force at different pressures can be finally calculated and then fitted as a linear equation (5) as shown in Figure 7.

\[ \text{Static friction maximum (Newton)} = \alpha \times \text{Pressure (Pa)} \]

\[ \alpha = 4.1054 \times 10^{-6} \text{ (m/s)} \]

\[ F_{static} = x \times 10^{5} \text{ (N)} \]

The remaining model parameters, the damping coefficient \( D_p \), the piston seal damping coefficient \( \alpha \), and the piston seal drag force \( F_{drag} \) while the piston is moving, are identified using the least square estimation approach [10]. The identification result is exhibited in Figure 8, which shows the matching between the identified simulation results and the experimental data. The measured and identified parameter values are presented in Table 1.

C. Clutch Fill Simulation and Experimental Results

An optimal input pressure is derived to achieve the desired clutch fill velocity profile using the customized dynamic programming method. The desired final state conditions are \( x(T)=d=0.00058 \text{ (m)}, x(T)=0 \text{ (m/s)}, x(T)=1.875 \times 10^5 \text{ (Pa)} \). Figure 9 (dashed red line) shows the optimal control input obtained using the proposed customized Dynamic Programming. The resulting optimal control input pressure is then generated in the experiment to verify its performance as shown Figure 9 (solid blue line). The experimental clutch piston velocity profile using the design optimal control input is shown in Figure 10.

**Figure 8. Comparison between experimental data and the simulated output for piston displacement**

**Table 1. Parameter Values of the Dynamic Model**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( M_p )</td>
<td>0.4 (kg)</td>
</tr>
<tr>
<td>( x_{ps} )</td>
<td>1.485 (mm)</td>
</tr>
<tr>
<td>( x_{ps} )</td>
<td>0.000375 (s)</td>
</tr>
<tr>
<td>( K_p )</td>
<td>242650 (N/m)</td>
</tr>
<tr>
<td>( D_p )</td>
<td>135.4 (N/m/s)</td>
</tr>
<tr>
<td>( A_p )</td>
<td>0.006 (m²)</td>
</tr>
<tr>
<td>( \alpha )</td>
<td>4.1054 \times 10^{-6} \text{ (m/s)}</td>
</tr>
<tr>
<td>( \beta )</td>
<td>17000 (bar)</td>
</tr>
<tr>
<td>( V_0 )</td>
<td>7.8e-5 (m³)</td>
</tr>
<tr>
<td>( \kappa_0 )</td>
<td>0.00106 (m²)</td>
</tr>
<tr>
<td>( \zeta_0 )</td>
<td>47.52 (N/m)</td>
</tr>
</tbody>
</table>

**Figure 7. Correlation of stick friction and pressure data**

**Figure 6. Experiments for measuring the stick friction**

**Figure 9. Optimal input pressure and the experimental tracking results**
This paper presents an experimental investigation of the optimal transmission clutch fill control, following the previous work on the clutch fill control design using a customized Dynamic Programming method [6]. The objective of the clutch fill control is to enable a fast and precise clutch fill and reduce the peak flow demand. We formulated this problem into a constrained optimization problem. To validate the optimal control, a transmission clutch control test bed is designed and built for experimental investigation. A dynamic model which captures the key clutch fill dynamics is constructed and identified with the experimental data. Finally, simulation and experimental results show the effectiveness of the proposed control method.

VI. ACKNOWLEDGMENTS

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VII. REFERENCES


