Experimental Study of PD Controller for Engagement Control of an Electro-Mechanical Friction Clutch (EMFC) System

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Abstract: Nowadays, most of the clutches used in continuously variable transmission (CVT) applications are based on electro-hydraulic and electro-magnetic actuations. The designs of these clutches cause some energy losses due to the actuated clutches need continuous power to supply force to engage the clutches. This continuous energy consumption becomes one of the major losses in CVT’s clutches system as it can reduce the transmission efficiency. Therefore, this paper attempts to introduce a novel electro-mechanical friction clutch (EMFC) system as a viable solution for reducing the power loss in CVT applications. This EMFC consists of mechanical actuator, a standard dry friction clutch, clutch linkages and a DC motor. The DC motor is used to power the mechanical actuator via power screw mechanism. This paper proposes proportional differential (PD) controller to control the DC motor such that the mechanical actuator satisfy the desired engagement and disengagement processes of EMFC system. The initial values of the PD gains are derived using Astrom and Hagglund tuning method and Ziegler-Nichols formula. Then, these gains are manually fine tuned to improve the performance of the clutch engagement control until satisfying the predetermined optimal global criterion. Experimental results are presented to demonstrate the effectiveness of the proposed controller and discussed with considering different operating conditions and the dynamic behaviours of the clutch.

Key-Words: Electromechanical Friction Clutch, Clutch Engagement Control, PD Controller.

1 Introduction

With the current trend for the theorised threat of global warming and pollution control, it is important for car manufactures to minimise fuel consumption and exhaust emissions. Parallel with the demand for driving comfort which has become a priority at all levels in the market and increasingly competition from other manufactures a way to reduce costs is highly needed. Thus, the trend towards more highly automated transmissions will play important role in future automotive systems.

Different approaches, such as automatic transmission (AT), automated manual transmission (AMT) and continuously variable transmission (CVT) are exploiting the availability of reliable drive-by-wire technologies. These transmissions represent the key elements for the improvement of vehicle safety, comfort, reliability and driving performances as well as the reduction of fuel consumption and emissions [1]. For example, vehicle with AMT is generally constituted by a dry friction clutch-by-wire system as means of easing the driver’s task and thus, enhancing driving satisfaction [2]. Clutch-by-wire gives the possibility to handle high torques form the engine without needing too high pedal force at the clutch pedal. With respect to manual transmission, the AMT allows to improve driving comfort and shift quality by controlling the dry clutch engagement process. The engagement of dry friction clutch is a very important process both to ensure small friction wear and good power-train performance [3-5].

Currently, most of the clutches used in CVT applications are based on hydraulic and electro-magnetic actuations. These clutches are selected because they can be controlled electronically [5]. Relating to this technology, a novel electro-mechanical friction clutch (EMFC) system for a novel electro-mechanical dual acting pulley (EMDAP) CVT applications have been developed by Vehicle Powertrain Research Group (VPRG) in Universiti Teknologi Malaysia (UTM) as the future generation transmission. The EMFC consists of mechanical actuator, a typical dry friction clutch, clutch linkages and a DC motor. The DC motor system is used as an actuator to actuate power screw mechanism for
engaging and disengaging the clutch. This novel EMFC enables the clutch to be operated electronically so that a suitable closed loop control strategy can be applied in order to satisfy clutch engagement control objectives such as smooth engagement process with minimum engagement time [4].

This research proposes a PD controller for the clutch engagement process of EMFC system. The controller is designed using Ziegler-Nichols method to investigate EMFC’s performance.

Basically in EMDAP CVT applications, its pulley ratio can be varied continuously and clutch engagement and disengagement are not required during CVT ratio changes. However, standing start and stopping still require smooth clutch engagement and disengagement to prevent engine stalling. This has always been the greatest challenge to transmission engineers because this unavoidable process dissipates quite a large amount of energy [2] especially for the clutch engagement process during standing start.

A Matlab/Simulink® software is also used to develop computer-based control system in order to search for an appropriate algorithm that will be used to control the plant of EMFC system and to record the experimental data. Computer simulation and experimental results are presented in this paper to demonstrate the effectiveness of the proposed control scheme for the closed loop system.

2 System Modelling
This section describes the mathematical models of the EMFC system. The model covers a DC motor in conjunction with reduction gearing system, a power screw mechanism, clutch linkages and the typical dry friction clutch as shown in Fig. 1.

Fig. 1 Simplified model of designed EMFC

2.1 Modelling of DC motor
The equations describing the dynamic behaviour of the DC motor are given by the following equations:

\[ V_a = L_a \frac{di_a}{dt} + R_a i_a + e_b \]  

\[ T_m = K_f i_a (t) \]  

\[ T_m - T_l = J_m \frac{d^2 \theta(t)}{dt^2} + B_m \frac{d \theta(t)}{dt} \]  

\[ e_b = e_b(t) = K_b \frac{d \theta(t)}{dt} \]

where \( V_a \), \( L_a \), \( i_a \) and \( R_a \) represent the motor voltage, inductance, current and resistance respectively. \( K_b \) represents back emf constant, \( e_b \) represents back emf, \( \theta \) represents angular displacement, and \( T_m \), \( K_f \), \( T_l \) and \( J_m \), \( B_m \) represents the motor torque, torque constant, load torque and motor inertia respectively. Meanwhile \( B_m \) denotes viscous friction coefficient.

2.2 Modelling of Gear Reducer and Power Screw Mechanism
A gear reducer serves as a speed reducer and torque multiplier. The gear reducer has ratio 5:1 and it is coupled with the DC motor shaft. The gear reducer output is connected to the power screw mechanism to produce a linear movement. The linear movement is then transmitted to actuate the shift arm either to engage or disengage the clutch system. The power screw mechanism is converted every 360° of rotation into 2 millimetres of linear movement.

Fig. 2 Force diaphragms for the power screw to disengage (A) and engage (B) the clutch

The torque required to move the power screw axially from engaged to disengage is given by:

\[ T_{\text{lift}} = \left( \mu \pi \frac{d_m}{2} + l \right) \left( \pi \frac{d_m}{2} + \mu l \right) xD_x \]  

\[ T_{\text{lower}} = \left( \mu \pi \frac{d_m}{2} + l \right) \left( \pi \frac{d_m}{2} + \mu l \right) xD_x \]

where, \( T_{\text{lift}} \) and \( T_{\text{lower}} \) represent the torques for lifting and lowering the load. \( P_k \) and \( P \) denote the forces required to raising and lowering the load. The normal, axial and friction forces are represented by \( D_x \), \( N \) and \( fN \).
respectively. Meanwhile \( d_{oa}, l, \mu \) and \( \lambda \) are the dimension of the power screw which correspond to the mean diameter, pitch, friction and lead angle.

### 2.3 Modelling of Clutch Linkage

It is shown in Fig. 3 that the clutch linkage equations of \( S_1 = 2X_1, S_2 = S_2, \theta_2 = S_2 / R_2, \theta_3 = \theta_3, S_3 = \theta_3 x R_3, \)
\( S_3 = S_4, \theta_4 = S_4 / R_4, \theta_5 = \theta_5, S_5 = \theta_5 x R_5 \)  
(7)
where:
\( X_1 \) : Number of revolution of outer power screw
\( S_1 \) : Displacement of inner power screw
\( S_2 \) : Angular Displacement of clutch fork arm (external)
\( \theta_2 \) : Angle of Displacement of clutch fork (external)
\( S_3 \) : Angular Displacement of clutch fork arm (internal)
\( \theta_3 \) : Angle of Displacement of clutch fork (internal)
\( S_4 \) : Angular Displacement of diaphragm spring A
\( \theta_4 \) : Angle of Displacement of diaphragm spring A
\( S_5 \) : Angular Displacement of diaphragm spring B
\( \theta_5 \) : Angle of Displacement of diaphragm spring B

### 2.4 Modelling of Dry Friction Clutch

Mathematical models of dry friction clutch engagement have been proposed in literatures using many different approaches [1-4, 6-10]. The electro-mechanical friction clutch (EMFC) uses a typical dry clutch system. This clutch consists of three basic parts which are the engine flywheel, a dry friction disc and a pressure plate. The flywheel and pressure plate are the driving members. They are attached to and rotate with the engine crankshaft. The dry friction disc is the driven member. In order to investigate the system and to design a controller through analytical procedures, a simplified mathematical model of the typical dry friction clutch was developed based on [4] and [10]. It can be thought as a system consisting of two spinning disc models. Fig. 3 shows the load-deflection curve for the diaphragm spring of the EMFC plotted based on the experimental data.

For numerical application, this nonlinear function was approximated by a 4th order polynomial [11] and expressed in terms of the displacement as \( x \).

\[
F_a (x) = -4E10x^4 + 1E9x^3 - 3E7x^2 + 33126x - 16.54 \quad (8)
\]

### 3 Proposed Controller

This paper proposes PD controllers for the DC motor to control the clutch engagement process. The parallel form of the PD controller can be written as:

\[
G_{pd} = K_p (1 + T_d s) \quad (9)
\]

The PD controller is quite easy to be achieved and can make the system having excellent performance through the optimization of the P and D parameters [12]. Significant efforts have been devoted to providing of PD controller parameters tuning. A method for automatically tuning of simple regulators was introduced in Astrom and Hagglund [13]. The idea was to determine the critical period of waveform oscillation (\( T_c \)) and the critical gain (\( K_c \)) from a simple relay feedback experiment and also to use the Ziegler-Nichols method type of control design method to determine the suitable value of three parameters, namely \( K_p, K_i \) and \( K_d \) to satisfy certain control specifications.

The main objective of this design controllers is to ensure that two fundamental conditions; no-kill conditions and no-lurch conditions have to be satisfied as in [8] and [9]. The no-kill condition states that the engine stall must be avoided, whereas the no-lurch condition assumes that the unwanted oscillations induced in the power-train should be reduced in order to allow the driver comfort. However these requirements are in conflict with the minimum duration of the engagement time, such as oscillations induced in the power-train system due to the sudden change of torque within limited time during the clutch engagement process. Furthermore, an engine could be stalled if an excessively fast engagement process occurs. The general closed-loop scheme of the controlled system is reported in Fig. 4.
3.1 Experimental Setup
The laboratory test-bench consists of dry friction clutch and electro-mechanical clutch actuator, engine as power source, water-brake dynamometer as a variable load, torque-speed sensors for measuring engine and output speeds and torques, and data logger systems for recording the data during experiments as shown in Fig. 5.

![Fig. 5 Experimental bench-test](image)

Experimental studies of the proposed PD controller are carried out in order to investigate its effectiveness in clutch control. Control performance is determined based on percent overshoot (POS), settling time $T_s$, and steady state error $E_{ss}$ of the system as shown in Fig. 6. A closed loop control application incorporating the Astrom-Hagglund method based on a relay feedback controller was carried out to attain the critical period of waveform oscillation ($T_c$) and the critical gain ($K_c$) which are then used as initial gains of the PID controller.

A step input corresponding to the displacements of the inner power screw of the EMFC’s actuator positions is used as input for relay feedback experiment. The amplitude of the relay controller is set to 10 since the input voltage in the range of -5 to +5 volts are needed to drive the DC motor system to actuate the inner power screw.

![Fig. 6 Transient response](image)

3.2 Initial Parameters of PD Controller
Fig. 6 shows the results of relay feedback experiment of the DC motor to actuate the inner power screw for full clutch engagement process. The $V_{ed}$ acts as reference input for inner power screw position, the $V_{ea}$ represents the actual inner power screw position and $V_{rl}$ is the relay controller output. According to Fig. 9, the values of $T_c$ is 0.351s, $a$ and $d$ are 0.93V and 10V respectively. By using equation (15), the critical gain ($K_c$) was found to be 13.69. Once $T_c$ and $K_c$ values are obtained, the PID parameters ($K_p$, $T_i$, and $T_d$) can be specified using Ziegler-Nichols formula. The $K_p$, $K_i$, and $K_d$ of the clutch engagement PID controller variations are shown in the Table 1.

**Table 1 PID controller variations**

<table>
<thead>
<tr>
<th>Tuning Method</th>
<th>Controller Type</th>
<th>$K_p$</th>
<th>$K_i$</th>
<th>$K_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ziegler Nichols</td>
<td>P</td>
<td>6.8450</td>
<td>0.0000</td>
<td>0.0000</td>
</tr>
<tr>
<td></td>
<td>PI</td>
<td>6.1610</td>
<td>20.6502</td>
<td>0.0000</td>
</tr>
<tr>
<td></td>
<td>PID</td>
<td>8.2140</td>
<td>46.8034</td>
<td>0.3604</td>
</tr>
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</table>

From Table 1, it can be seen that the value of the integral gain ($K_i$) is much bigger compared to other gains. Based on the system behaviour performed during the relay feedback experiment, a small tolerable steady state error has occurred; therefore the integral gain is not used for controlling this kind of system because the use of big integral gain makes the system unstable as shown in Fig. 7 for PI and PID controller. The P controller makes the system oscillates around the set point in a decaying sinusoid. It can be observed that the PD controller can be considered has a good performance in terms of percent overshoot, settling time and steady state error. Thus, the PD controller gives better result with minimum error and less overshoot.

![Fig. 7 Response curve for PID controller variations of Ziegler Nichols parameter tuning](image)

3.3 Fine Tuning of PD Controller
The initial parameter values obtained from relay feedback experiment needs to be fine tuned for the clutch engagement process. The tuning process is conducted by examining the output responses of engagement by inner power screw position sensor when...
the position reference is shifted up from 0 mm to 14 mm. The tuning process will only fine tune the differential part of PD controller manually and leave the proportional part unchanged. However, the proportional part also needs to be fine tuned if the tuning process of the differential part does not satisfy the control performance for the PD controller in terms of percent overshoot (POS), settling time $T_s$ and steady state error $E_{ss}$. The output responses of the system during fine tuning process for the engagement PD controller based on Ziegler Nichols tuning method is shown in Fig. 11. The fine tuned values of proportional and differential parts of PD controller derived from Fig. 8 are $K_p = 0.8$ and $K_d = 0.02$.

By performing a closed loop control application, the smoother engagement process has been achieved, where the percent overshoot of the clutch output torque for the closed loop system (about 5%) is smaller than that of the open loop system (about 90%) during transition from slip to full engagement as shown in Fig. 9. It can be observed that the PD closed loop controller slightly reduces the unwanted oscillations induced in the power-train during slipping phase of the engagement process with minimum engagement time is about 1.45 seconds.

Fig. 10 and Fig. 11 show the characterizations of the EMFC system in terms of time response of the EMFC’s actuator inner power screw (a), engagement time and clutch input and output speed behaviours (b) and clutch torque behaviours (c). Based on the experiment results, when the input speed from the engine increases, the time taken by the output clutch also increases because the output clutch needs more time to slip (at slipping phase period) in order to achieve the same speed with the input clutch for the constant applied torque during the engagement process. The engagement time is shown in Table 2.

![Fig. 8 Response curves of the different PD controller parameter tuning](image)

### 4 Performance Evaluation

The dynamic behaviour of the clutch was conducted with constant engine speed of 2000 rpm with applied torque of 10 Nm in order to avoid the engine stall during standing start scenario. The clutch was controlled to be initially disengaged, slipped, engaged, slipped again and finally fully disengaged. During this process, the data was taken by data logger system and recorded by computer. The results are given in Fig. 12. Fig. 12 (a) shows the response curves for the closed loop of EMFC system while Fig. 9 (b) shows the response curves for the open loop system.

![Fig. 9 Response curves at 2000 rpm of the engine speed](image)

**Table 2 Clutch engagement time**

<table>
<thead>
<tr>
<th>Initial Engine Speed (rpm)</th>
<th>Engagement Time (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>1.20</td>
</tr>
<tr>
<td>1500</td>
<td>1.30</td>
</tr>
<tr>
<td>2000</td>
<td>1.45</td>
</tr>
</tbody>
</table>
5 Conclusion

Simplified static and dynamic models of a novel electro-mechanical friction clutch (EMFC) have been developed and presented. The engagement process during standing start scenario for the EMFC has been considered. A Matlab/Simulink® simulation scheme has been built and a proposed PD controller scheme has been implemented and tested within the experimental environment of the EMFC system. A bench test of the electro-mechanical friction clutch (EMFC) system to specifically investigate the dry clutch engagement behaviours with the proposed PD controller for the processes has been carried out. The proposed controller with a tuning method from the simulation studies has been tested and optimized by designing and tuning the proposed controller on an EMFC bench-test. The results of this work show that the application of Astrom-Hagglund method and Ziegler-Nichols formula are capable of providing a practical solution for obtaining initial parameters of the PD controllers for the EMFC engagement processes. Some experimental results of the control engagement process have been evaluated in terms of position control application and it has shown adequate improvement which is achievable by means of the proposed controller.

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