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DEVELOPMENT OF AUTOMATIC CLUTCH ACTUATOR FOR AUTOMATED MANUAL TRANSMISSIONS

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ABSTRACT−With the growing traffic density and increasing comfort requirements, the automation of the drive train has gained importance in vehicles. The automatic clutch actuation relieves a driver especially in urban and stop-and-go traffic environments. In this paper, an electro-mechanical actuator for clutch-by-wire (CBW) system is implemented as the first stage for the development of automated manual transmissions. The prototype of the automatic clutch actuator is designed systematically, which is composed of the electric motor, worm and worm wheel, and crank mechanism. A test rig is developed to perform the basic function test for the automatic clutch actuation. The developed prototype is validated by the experimental results performed on the test rig.

KEY WORDS : Automated manual transmission (AMT), Clutch-by-wire (CBW), Automatic clutch actuator, Test rig

1. INTRODUCTION

Drivers are becoming more fatigued and uncomfortable as the traffic density increases, which can lead to slower reaction time. They then face the danger of traffic accidents due to the inability to cope with frequent shifting. To reduce this risk some drivers prefer automatic transmissions to manual transmissions. However, the automatic transmission comes with both higher fuel consumption and costs. For these reasons, the attention given to the automated manual transmission (AMT) is increasing, that has the advantages such as high efficiency, low cost and easy manufacturing (Nordgard, 1995; Fischer, 1998).

By the 1960s, automotive manufacturers began to offer automated clutch operating systems designed to simplify vehicle operation. But the early systems were functionally inadequate, maintenance-intensive and prone to frequent repairs. These disadvantages can now be eliminated with use of modern vehicle electronics. Most European clutch manufacturers, for example, Fitchel & Sachs, Luk and Valeo, have been intended on developing an electronically controlled clutch. There had been research into both the design and control of the clutch automation systems. Improvements have been shown in shift feeling and cost reductions (Valeo, 2002).

The development process of the AMT is divided into two, clutch automation and shift automation. Clutch automation is considered with the driver's desire to gearshift and operate the clutch automatically. In the manual transmission vehicle, the clutch cuts power during shifting and stopping, while transmitting power during driving. The torsional vibration of the vehicle powertrain is very sensitive to clutch operation and has an influence on ride comfort. Therefore, it is an important part of clutch control at both launching and shifting to minimize the torque fluctuations during engaging and disengaging the clutch (Szadkowski, 1992; Szadkowski, 1992; Rha, 2003). To overcome this problem, it is necessary to develop adequate actuators, sensors and control algorithms.

In this paper, the prototype of an electro-mechanical type automatic clutch actuator is implemented using a DC motor and crank mechanism. The prototype, with the advantages of low cost and simple construction can fit into a small vehicle. The parameters of the crank mechanism, DC motor and gear ratio are systematically selected and designed using load analysis. The test rig is developed to perform the basic function test for this automatic clutch. The developed automatic clutch prototype is validated by experimental data employing the test rig.

2. DEVELOPMENT OF AUTOMATIC CLUTCH

2.1. Conceptual Design of CBW System

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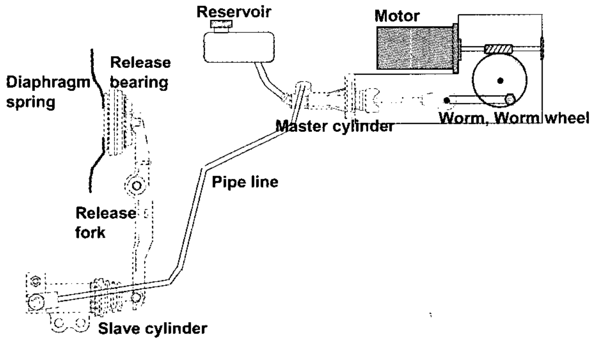


Figure 1. Schematic diagram of electro-mechanical type CBW system.

The conventional hydraulic clutch system consists of master cylinder, slave cylinder, reservoir, release fork and release bearing (Shaver, 1997). In this paper, an electro-mechanical type CBW system is proposed, which replaces the mechanical clutch control mechanism between master cylinder and pedal with an electro-mechanical actuator. The driver's desire to disengage a clutch is converted into the velocity of a DC motor and that changes into the hydraulic oil pressure. The pressurized oil passes through the pipe and is delivered to the slave cylinder. The displacement of slave cylinder induces the stroke of release bearing through the release fork. This causes the release bearing to pull the diaphragm spring, at which point the clutch is finally disengaged. Figure 1 shows the schematic diagram of the CBW system considered in this paper.

2.2. Design of Crank Mechanism

For a design of automatic clutch actuator, the desired maximum force of master cylinder should be induced by the clamp load of the diaphragm spring. The clutch system for a sport utility vehicle is studied in this paper. Figure 2 shows the experimental data from the clutch manufacturer.

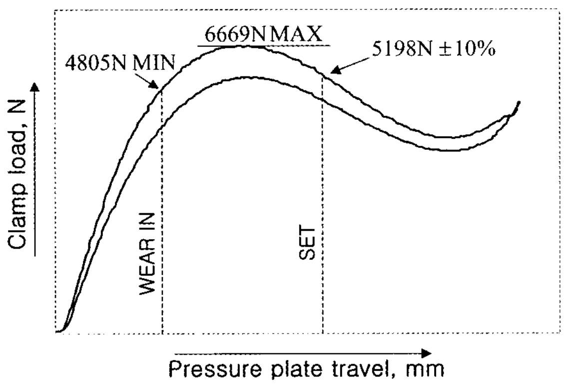


Figure 2. Load-deflection characteristics of clutch.

The clutch clamp load is initially 5,198 N and the release load considered cushion spring is 1,700 N. However, the clamp load increases to 6669N as a result of clutch abrasion caused by long distance driving. A release load of 2,200 N is needed to disengage the clutch. The conventional hydraulic clutch system has an efficiency of 70%. Therefore, the force to disengage the clutch is calculated as 1,026N max. from Equation (1).

$$F_P = \frac{F_{RB} A_{MC}}{C_{RF} A_{SC}} \eta \tag{1}$$

where, F_P is the force of master cylinder, F_{RB} is the force of release bearing, C_{RF} is the ratio of release fork, A_{MC} is the area of master cylinder, A_{SC} is the area of slave cylinder, η is the efficiency of clutch mechanism.

Various clutch actuator mechanisms are employable, these include: worm gear, lead-screw or lever-mechanism types (Pollak, 2002). In this paper, the worm gear and crank mechanism are selected for advantageously allowing small space and large torque. Figure 3 shows the schematic diagram of the offset crank mechanism.

To determine the design parameters with crank arm r, link length l and offset e as examples, mechanism analysis is performed using Equations (2), (3) (Cho, 1985).

$$S = \sqrt{(r+l)^2 - e^2} - (r\cos\theta + \sqrt{l^2 - (r\cos\theta - e)}) \tag{2}$$

$$T = r\left(\sin\theta + \frac{\cos\theta(r\sin\theta - e)}{\sqrt{l^2 - (r\sin\theta - e)^2}}\right)F_p \tag{3}$$

Worm wheel reaction torque and master cylinder stroke change according to the parameter variations when the force of the master cylinder is given 1026 N as the previously calculated.

Figure 4 shows the simulation results for reaction torque and stroke, with variations in crank arm radius, link length and offset. The length of the crank arm has a large amount of influence over reaction torque, that both the piston stroke and rotating angle of worm wheel must be taken into consideration to determine this parameter. On the other hand, the length of link and offset are only concerned with space because they have little effect on

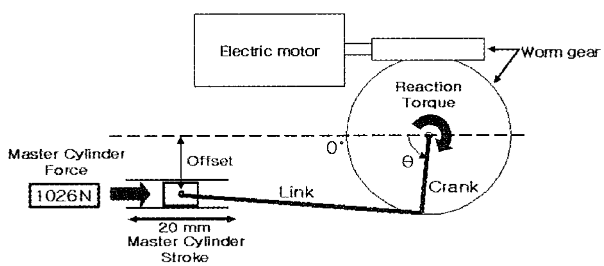
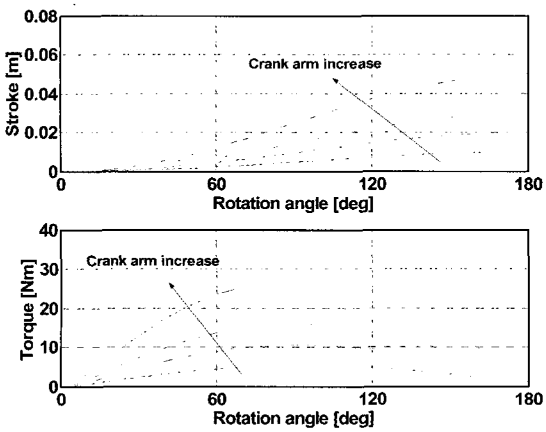
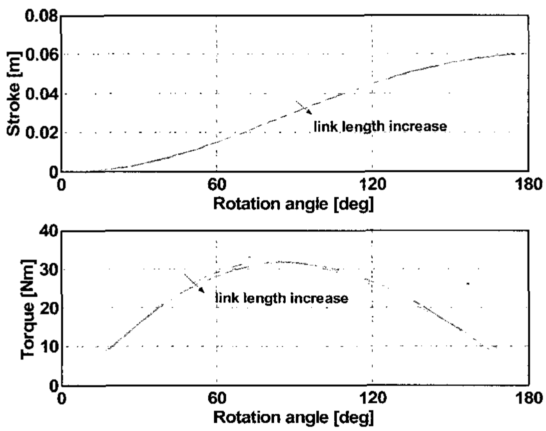


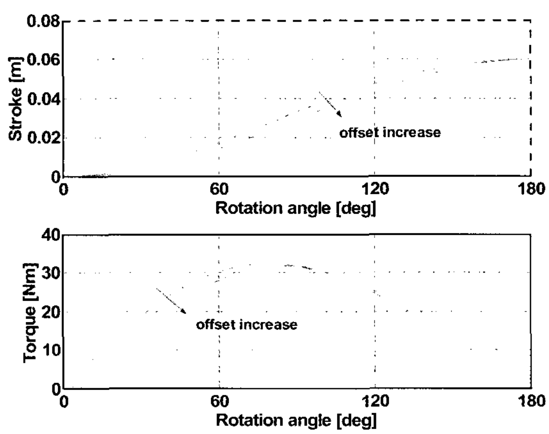
Figure 3. Schematic diagram of offset crank mechanism.



(a) Variation of crank arm radius



(b) Variation of link length



(c) Variation of offset

Figure 4. Simulation results for reaction torque and stroke with variation of crank arm radis, link length and offset.

the reaction torque. From previous load analysis, the selected specifications are as shown in Table 1. The calculated maximum reaction torque of the worm wheel

Table 1. Specifications of electro-mechanical type actuator.

Crank	15 mm
Link	150 mm
Offset	15 mm
Worm wheel reaction torque	15.4 Nm (at 89.2°)
Piston stroke	20 mm (5 mm~25 mm)
Worm wheel angle	83.66° (50.76°~134.42°)
Gear ratio	63 : 1
Worm wheel torque	20.2 Nm (margin 1.3)
Disengagement time	0.46 sec (83.66°/0.46 sec)

is 15.4 Nm and the piston stroke is 20 mm. A wiper DC motor is selected as an actuator and its output torque is 0.32 Nm. The worm gear ratio is selected based on a desired allowance in wiper motor torque and angular speed of the worm wheel. We choose a worm gear ratio of 63:1 based on the torque margin of 1.3 and clutch actuator's target speed of 20 mm/500 msec.

2.3. Implementation of Automatic Clutch Actuator

The CBW actuator is composed of a DC motor, worm gear, master cylinder and sensor, its prototype is shown in Figure 5. The TPS (Throttle Position Sensor) is used for a feedback sensor and attached to the worm wheel. The CBW actuator prototype was developed with the ability to adjust the crank arm and/or change the motor if desired, hence it is somewhat large. A compact, more practical version could also be developed for vehicle applications.

2.4. Test Rig

The CBW test rig was developed for the purpose of performance testing of an automatic clutch actuator, and

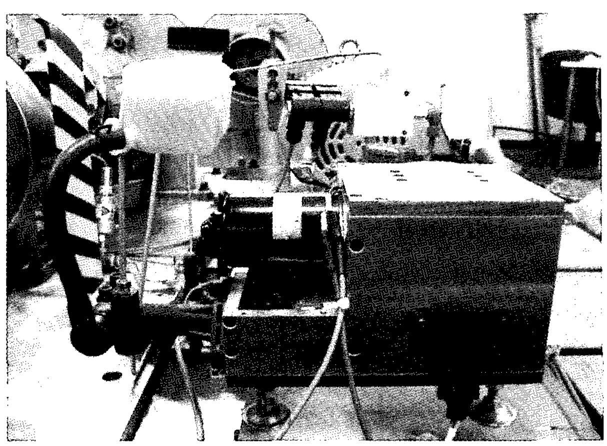
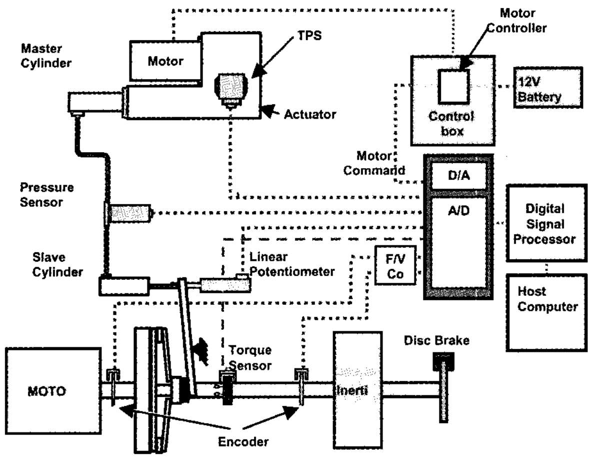
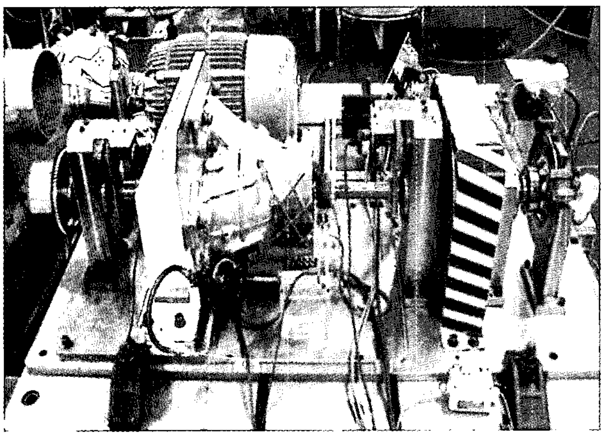


Figure 5. Prototype of electro-mechanical type actuator.



(a) Schematic diagram of test rig



(b) Experimental apparatus

Figure 6. CBW test rig.

shown in Figure 6. An induction motor and inverter controller simulate an automotive engine in various driving conditions. And many sensors installed on the test rig, such as encoders for the measurement of input and output drive shaft speeds, linear potentiometers and TPS for the displacements of master and slave cylinders, and a torque transducer for load torque measurements of drive shaft. The controller for a test rig consists of signal amplifiers, F/V converters, AD/DA boards, digital signal processor and a PC for acquiring data and signal processing.

3. EXPERIMENTAL RESULTS

3.1. Control Characteristics of CBW Prototype

The step responses for an automatic clutch prototype are shown in Figure 7. Displacement of the master cylinder follows the reference input by using a PID controller and its maximum stroke is 20 mm. From Figure 7, the disengagement time is measured up to 350 msec and engagement time up to 240 msec. The step response of slave cylinder is similar to the master cylinder, except that the slave cylinder has a time delay of 46 msec. This is caused by the ineffective stroke of master cylinder during the disengagement process.

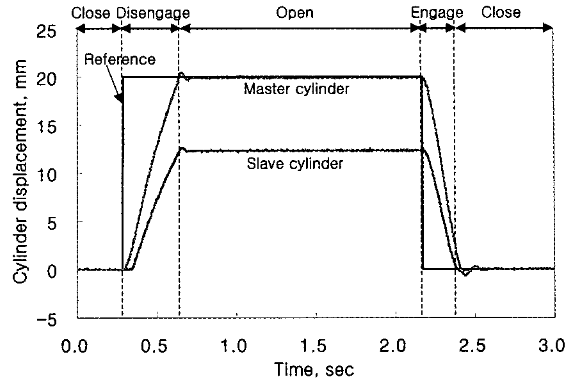


Figure 7. Step response of CBW actuator.

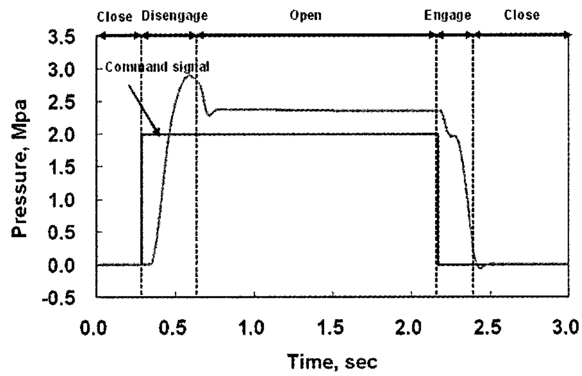


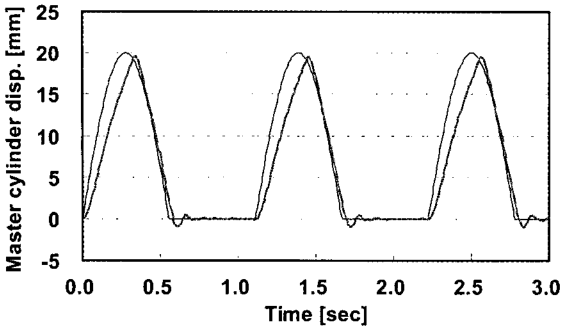
Figure 8. Experimental results for hydraulic line pressure.

Figure 8 shows the measured hydraulic line pressure. The hydraulic pressure increases to a maximum of 2.9 MPa and shows somewhat of an overshoot, caused by the release load characteristics of the diaphragm spring. The pressure converges to 2.35 MPa at steady state and the slave cylinder force is calculated to be 1100 N, which is equal to the target force.

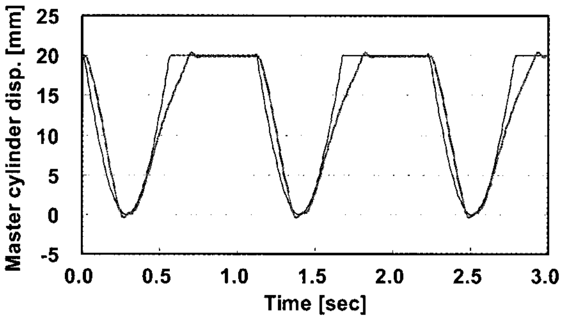
Figure 9 illustrates the time responses of the master cylinder with a half sinusoidal input of 0.9 Hz. The experiments are performed in two ways; from disengagement to engagement and from engagement to disengagement. They show that the responses have nonlinear characteristics.

3.2. Clutch Control Profile

Reduction in the fluctuation of transmitted torque to the clutch is the objective of the clutch actuator during engagement time conditions. Since the torque fluctuation

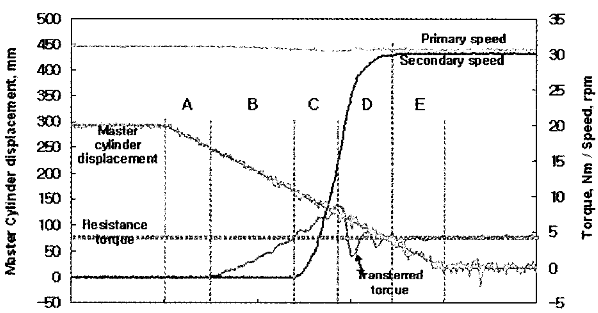


(a) From disengagement to engagement

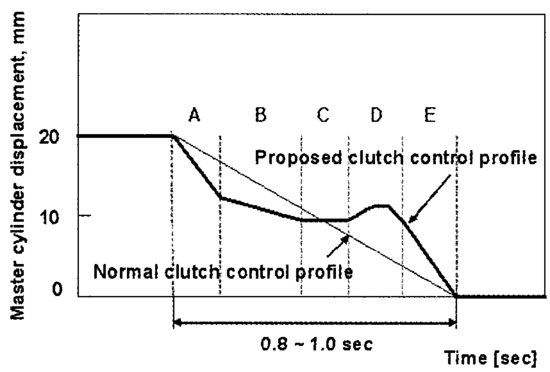


(b) From engagement to disengagement

Figure 9. Dynamic responses for 0.9 Hz half sinusoidal input.

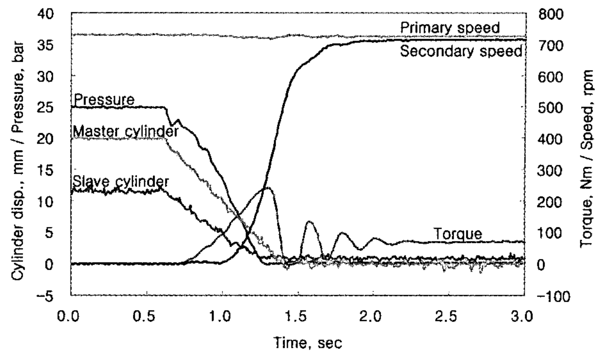


(a) Engagement process

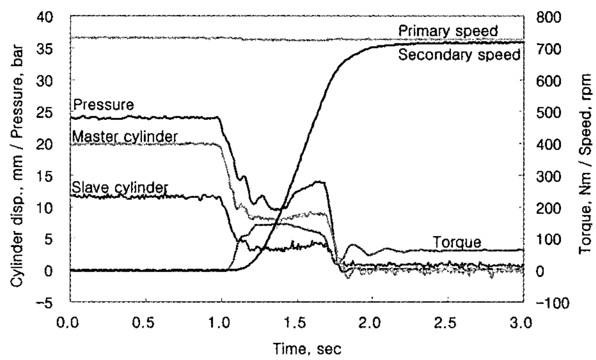


(b) Clutch control profile

Figure 10. Clutch engagement process and control profile.



(a) Experimental results with normal clutch control profile



(b) Experimental results with proposed clutch control profile

Figure 11. Launching test.

relates to ride comfort while the engagement time relates to the life span of clutch pressure plates, this paper proposes the clutch control profile at the given engagement time. There are 5 regions of interest during the clutch engagement launching process. The drive shaft velocity and transmitted torque are shown at launching in Figure 10(a). Normally the time of 2 seconds and over is required for drivers to engage the clutch at launching. However, for the purpose of fast maneuverability, this paper proposes the clutch control profile at given engagement time of 1 second as shown in Figure 10(b).

3.3. Launching Test

The experimental results with and without control algorithm at launching condition are as shown in Figure 11. Figure 11(a) shows test results when engaging during 0.8 sec with normal clutch control profile. The transmitted torque has the fluctuation due to inertia and load corresponding to rolling resistance of a vehicle. After that the speed of friction disc is synchronized with a flywheel when the transmitted torque is greater than rolling resistance. The transmitted torque leads to a large fluctuation but the speed of flywheel generates a small oscillation under the influence of inverter controller. The fluctuation and settling time of torque can be reduced by controlling the displacement of master cylinder with proposed profile as shown in Figure 11(b). These results imply that the automatic clutch actuator prototype with CBW control algorithm proposed in this paper is able to control the clutch automatically with less torque fluctuation at given engaging time.

4. CONCLUSIONS

This paper describes the systematic development of the clutch actuator for the CBW system and its prototype implementation. Employing the test rig, the performance of the developed prototype was evaluated through both launching and shifting tests. Performance results show it has a response time of 350 msec for disengagement and 240 msec for engagement, which are suitable for a CBW actuator when considering the target speed of 20 mm/500 msec. The developed CBW actuator, with the advantages of simple construction is expected to fit into a small vehicle.

Further research will be conducted on an engagement algorithm concerned with transmitted torque. Experiments of worn clutches and research surrounding a compact design for real vehicles will also be pursued.

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