Design and Modeling of Energy Efficient Dual Clutch Transmission With Ball-Ramp Self-Energizing Mechanism

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***Abstract*—Dual clutch transmission (DCT) which can simultaneously improve acceleration performance and fuel efficiency compared to automatic transmissions (AT) and manual transmissions (MT) is one of the most noted studies in recent powertrain applications. However, much of energy consumption of clutch actuator reduces fuel efficiency of DCT-equipped vehicles. In order to reduce energy consumption of these actuators, a method of modifying mechanism and a method of improving it through control have been studied. However, the results of each study still use more energy or make the actuator less controllable. Thus, in this manuscript, we propose a ball-ramp DCT (BR-DCT) with a clutch actuator that changes mechanism to reduce power consumed by engagement and improve controllability. DCT is designed using the proposed structure and design parameter constraint selection method, and energy efficiency improvement effect of BR-DCT is verified through simulation. We also demonstrate effect of BR-DCT prototype on actuator energy saving through test bench experiment. As a result, clutch actuator of BR-DCT uses 84% less energy than conventional DCT and can improve vehicle fuel efficiency by 0.5%. In addition, since it has same characteristics as conventional DCT, it verifies that it can fully utilize controller developed previously.**

***Index Terms*—Dual clutch transmission, clutch actuator, transmission, self-energizing, actuation energy.**

I. Introduction

THE demand for an efficient power train has always been with the history of automobile development. Recent research on energy-efficient transmissions is essential because the transmission has a direct impact on the energy efficiency of the vehicle [1], [2]. As a result of these studies, dual clutch transmission (DCT) is presented as a new solution to address the limitations of manual transmission (MT) and automatic transmission (AT).

MT is the most fuel-efficient among transmissions, because it directly cuts off or transmits engine power in a mechanical method through the clutch, but there is inconvenience that the driver must operate the clutch continuously for gear shifting and the shift time of the transmission is slow [3], [4]. AT does not require manual operation for shifting, but it has a disadvantage that energy efficiency is lower than MT due to the torque converter transmitting power indirectly through fluid and not blocking engine power completely. Indeed, torque converters of the AT have low energy transmission efficiency [5], [6].

To overcome these drawbacks, DCT is researched actively [7]–[10]. By using double clutches, DCT transmits or blocks power from the engine directly like a manual transmission. Therefore, DCT can also imporve fuel efficiency by minimizing power transmission loss. In addition, DCT enables quick shifting and minimizes shifting shock by using clutch actuator and engine control properly. For these reasons, commercial DCTs have been reported to increase fuel efficiency by up to 15% compared to AT [11]. Also, since the DCT operates alternately with double clutches, the gear actuator can pre-engage the next gear before shifting. The shifting shock can be minimized because the gear change is made in advance [12]. Therefore, it is possible to achieve a smooth automatic shift operation as AT. However, since a large engaging force of the clutch is required, the required energy of the clutch actuator becomes large. That is, the consumed energy, size and the weight of the clutch actuator are increased, thereby reducing the overall energy efficiency of the vehicle. This is the same in DCT applied to electric vehicles as well as internal combustion engine vehicles. The DCT actuator applied to an electric car has a direct effect on the electric efficiency because it is supplied with energy from the battery.

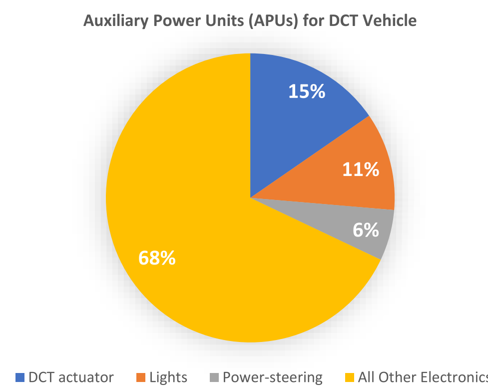

Fig. 1. The amount of auxiliary power units used in vehicles equipped with DCT.

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In addition, DCT has limitations in size and structure. It is larger in size and complicated in structure than MT, because it uses double clutches. Particularly, when an electric motor is used as a clutch actuator, a large amount of electric energy is required. This requires a large capacity battery, size and load torque on the alternator for electrical generation. Therefore, electric energy consumption is increased compared to AT and MT vehicles. For these reason, the portion of DCT to total electric energy consuming of automobiles is the largest among all energy consumption parts [13]–[17]. Also, DCTs using hydraulic actuators consume significant amounts of energy to drive line pressure pumps and solenoid valves. Therefore, a hydraulic actuator DCT-equipped vehicle consumes more energy than an AT or MT vehicle. Therefore, if the energy consumption of DCT clutch actuator can be reduced, overall energy efficiency of vehicle can be improved. To minimize the energy consumption of DCT, researches on the improvement of clutch actuation using a mechanical method and a hydrostatic clutch actuator (HCA) have been conducted [18]–[20].

First, research on the improvement of the energy consumption of a mechanical clutch actuator by using a rack-pinion self-energizing DCT (RP-DCT) are studied [18]–[20]. In this research, the self-energizing principle of the rack-and-pinion mechanism provides additional engaging force. However, backlash of gears and nonlinearity caused by inevitable friction reduce clutch torque control performance and energy efficiency as well.

Second, hydrostatic clutch actuators (HCAs) have been studied [21]. HCA, which is constructed with an electro-hydraulic actuator to perform force base control through a low level controller can reduce unnecessary energy consumption during engage compared to position control based clutch actuator. However, because of electric motor and hydraulic parts, it has large size and expensive. There is also a limitation that it does not reduce energy required to engage the clutch.

Therefore, this manuscript proposes a ball-lamp dual clutch transmission (BR-DCT) to address for the drawbacks of RP-DCT and HCAs. Specifically, it aims at increasing energy efficiency by using ball-ramp self-energizing mechanism as same as RP-DCT. In order to increase energy efficiency of vehicles using DCT, there is a method to increase power transmission efficiency or to reduce energy consumption of clutch actuators. However, DCT is designed based on MT with the highest power transmission efficiency, so it has same efficiency as MT. Therefore BR-DCT, which improves efficiency by reducing clutch actuator energy consumption, can overcome limitations of conventional DCT. In addition, it can reduce price and weight of batteries, alternators and motors by using less capacity. Furthermore, proposed system improves the controllability by increasing clutch torque control performance than RP-DCT. Additionally, it reducing the volume and weight by using optimized design of ball-ramp mechanism and lever. In section II, the mechanical design of the BR-DCT and its advantages obtaining from dynamic model are discussed. In section III, simulation is performed to verify how energy consumption of the clutch actuators are reduced and how the BR-DCT affects energy efficiency of the vehicle. In section IV, verifies dynamic model and energy efficiency that derived in Section II with the powertrain test bench of the BR-DCT by comparing conventional DCT.

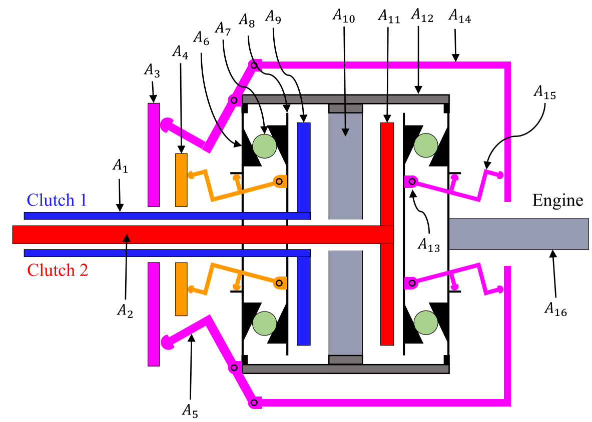

Fig. 2. A schematic diagram showing the structure of the proposed BR-DCT.

TABLE I
NAME OF EACH PARTS OF BR-DCT

No.	Name	No.	Name
A1	Vehicle side hollow shaft	A9	Clutch 1
A2	Vehicle side core shaft	A10	Engine side middle plate
A3	Outer release bearing	A11	Clutch 2
A4	Inner release bearing	A12	Bridge
A5	Outer side lever	A13	Hinge of lever
A6	Engine side plate	A14	Outer lever hinge plate
A7	Ball	A15	Inner side lever
A8	Clutch side plate	A16	Engine side shaft

II. MECHANICAL DESIGN AND MODELLING FOR BR-DCT

The BR-DCT proposed in this manuscript is designed to reduce clutch actuator energy consumption by using a self-energizing mechanism to address for the disadvantages of conventional DCT. In addition, it aims to reduce the nonlinearity due to backlash and friction of gear, which is a disadvantage of RP-DCT, and reduce it to a size and weight applicable to a vehicle.

The proposed DCT uses a ball-ramp mechanism for the self-energizing effect. This mechanism is similar to RP-DCT, but differs in that it consists of ball and spiral ramps rather than rack and pinions. This difference improves the controllability of the self-energizing DCT by eliminating nonlinear friction and backlash effects in gears. Specifically, since the ball and ramp always maintain point contact, amount of nonlinear friction is small and backlash effect of gears can be eliminated. In addition, the ball-ramp self-energizing mechanism has a major advantage in that clutch torque is transmitted to the ball and ramp to create an additional normal force to enhance the clutch engagement force without additional energy consumption compared to conventional DCT.

Fig. 2 and Table I show the structure of the BR-DCT proposed by this manuscript and the name of each part. To use the self-energizing effect, clutch side plate (A8) moves relative to engine side plate (A6) and transmits friction force between engine side middle plate (A10) and clutch 1 (A9). Engine side middle plate and engine side plate are connected by a bridge

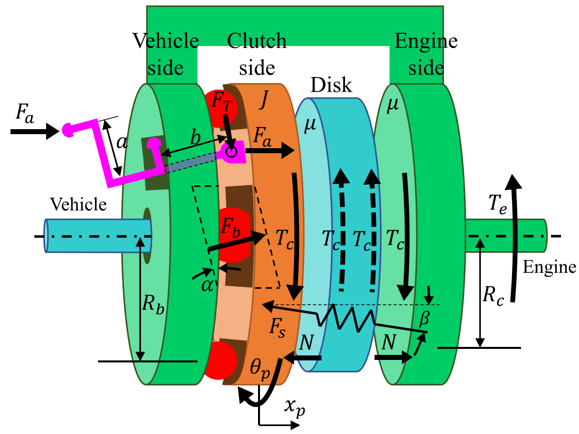

Fig. 3. Free body diagram when a positive friction torque is transmitted to the BR-DCT clutch disk. The green parts are connected to engine, the orange part has limited relative movement to the engine, and the blue part is connected to vehicle.

(A12). Since engine side plate and clutch side plate have same ramp angle, ball (A7) always maintains point contact between two plates. Therefore, ball acts as a medium to transmit force. Since RP-DCT uses rack-pinion instead of ball-ramp in this structure, it makes line contact instead of point contact, which increases friction nonlinearity and backlash.

The method of transmitting engagement force on clutch side plate is an important feature in designing the BR-DCT. Unlike the RP-DCT, lever (A5, A15) was used to transmit the translational force. The lever is connected to clutch side plate by a hinge (A13), and the other side contacts engine side plate with orthogonal to radial direction. Therefore, force transmitted from release bearing (A3, A4) causes the translational-rotational motion of lever with the contact point to engine side plate as supporting point. The translational force corresponds to the normal force generating clutch torque, and the rotational force induces relative movement between engine side plate and clutch side plate to produce a self-energizing effect.

Because of levers, the diaphragm spring used in return of clutch disk in conventional DCT could not be used in BR-DCT. Therefore, a compression return spring, which allows an angle change between clutch side plate and engine side middle plate, induces clutch normally disengage at outer diameter of clutch pack.

A. Dynamic Model of BR-DCT

Fig. 3 shows the free body diagram of the BR-DCT. The force and torque balance equation that can be derived from this figure is as follows.

$$J\ddot{\theta}_p = F_T R_a + T_c - F_b R_b \sin\alpha - F_s R_s \sin\beta \quad (1)$$

$$F_T = \frac{a}{b} F_a \quad (2)$$

$$m\ddot{x}_p = F_a + F_b \cos\alpha - F_s \cos\beta - N \quad (3)$$

$$T_c = \mu N R_c \quad (4)$$

TABLE II
DESCRIPTION OF EACH NOTATIONS

Symbol	Name	Symbol	Name
F_a	Clutch actuator force	μ	Clutch disk friction coefficient
a, b	Lever actuation length	N	Clutch disk Normal force
F_T	Clutch actuator rotational force	R_a	Lever actuation radius
F_b	Ball reaction force	θ_p	Clutch side plate rotation angle
R_b	Ball actuation radius	x_p	Clutch side plate displacement
α	Ramp angle	F_s	Return spring force
T_c	Clutch friction torque	β	Return spring angle
R_c	Clutch disk effective radius	R_s	Return spring actuation radius
J	Clutch side plate inertia	m	Clutch side plate mass

Equations (1), (2), (3) and (4) are torque, lever, force balance and clutch friction torque equations, respectively. Where, the description of each notations is shown in Table II.

Where, substituting (2) into (1) is equivalent to (5).

$$J\ddot{\theta}_p = \frac{a}{b} F_a R_a + T_c - F_b R_b \sin\alpha - F_s R_s \sin\beta \quad (5)$$

Assuming the steady state conditions $\ddot{\theta}_p = 0$ and $\ddot{x}_p = 0$, (3) and (5) are summarized as follows.

$$T_c = F_b R_b \sin\alpha - \frac{a}{b} F_a R_a + F_s R_s \sin\beta \quad (6)$$

$$N = F_a + F_b \cos\alpha - F_s \cos\beta \quad (7)$$

(8) is substituted for (7) in (4) and can be expressed as (9) by summarizing it as F_b.

$$T_c = \mu R_c (F_a + F_b \cos\alpha - F_s \cos\beta) \quad (8)$$

$$F_b = \frac{T_c - \mu R_c F_a + \mu R_c F_s \cos\beta}{\mu R_c \cos\alpha} \quad (9)$$

(9) is substituted for (6), equation is summarized as (10).

$$T_c = \frac{\mu R_c (R_b \tan\alpha + \frac{a}{b} R_a)}{R_b \tan\alpha - \mu R_c} F_a - \frac{\mu R_c (R_s \sin\beta + R_b \tan\alpha \cos\beta)}{R_b \tan\alpha - \mu R_c} F_s \quad (10)$$

$$G = \frac{R_b \tan\alpha + \frac{a}{b} R_a}{R_b \tan\alpha - \mu R_c},$$

$$G_s = \frac{R_s \sin\beta + R_b \tan\alpha \cos\beta}{R_b \tan\alpha - \mu R_c} \quad (11)$$

In (10), the coefficient of $\mu R_c F_a$ is defined as self-energizing gain (G), and the coefficient of $\mu R_c F_s$ is defined as return spring gain (G_s) as (11).

B. Constraints for Design Parameters

By increasing G and decreasing G_s, we can obtain large T_c with small F_a as shown in (10). However, the following constraints are required in order to increase both energy efficiency of the BR-DCT and robustness of clutch torque controller.

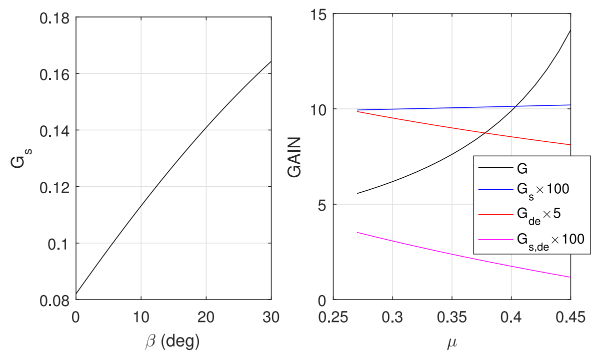

Fig. 4. Left: β-dependent variation of G_s. Right: μ-dependent variation of G, G_s, G_{de} and $G_{s,de}$.

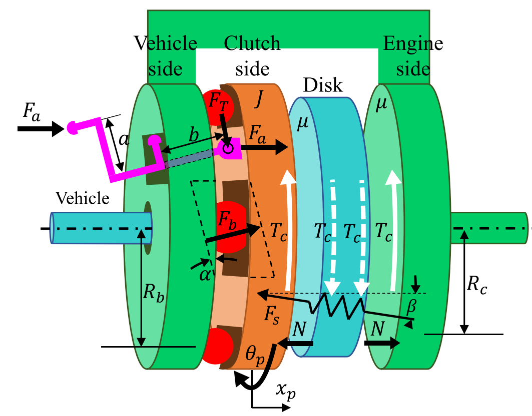

Fig. 5. Free body diagram when negative friction torque is transmitted to the clutch disk of BR-DCT.

Since R_c, R_a, R_b and a/b are determined by packaging and design specifications, G varies with α, which is not related to the design specification directly. If G is negative, self-locking occurs. That is, negative engage force is required to achieve target clutch torque. Therefore, engagement force must be given in the opposite direction. The physical phenomenon means that the clutch is remained engagement without engaging force in clutch actuator. That is, only on-off control is possible, which makes it impossible to control when clutch torque control is required during shifting. Therefore, clutch torque control is possible when the self-energizing gain is positive.

If G is positive, it should be able to generate a larger clutch torque at the same engage force compared to conventional DCT. In case of conventional DCT, (3) can be modified as (12).

$$m\ddot{x}_p = F_a - F_s - N \tag{12}$$

Therefore, assuming $\ddot{x}_p = 0$, which is a steady state condition, (4) is summarized as (13).

$$T_c = \mu R_c (F_a - F_s) = \mu R_c F_a - \mu R_c F_s \tag{13}$$

Based on the above equation, both G and G_s of conventional DCT can be defined as 1. Therefore, constraint that the gain (G) of the BR-DCT clutch actuator should be greater than 1 in order to consume less energy than the conventional DCT can be selected. However, if G is too large, it is difficult to control clutch torque because sensitivity of clutch actuator increases. Therefore, it is necessary to select value of G properly.

Unlike conventional DCT, BR-DCT can also change the value of G_s, which indicates effect of return spring on clutch torque. When clutch is engaged, the less F_s interfere with T_c, the lower energy consumption of clutch actuator. That is, the closer the G_s is to 0, the more energy efficient. Assuming that R_c, R_a and R_b are defined by the design specification, G_s defined in (11) can be considered to depend on α and β. Since α depends on G, G_s is defined as a function of β. Therefore, left side of the Fig. 4 shows the relationship between G_s and β. This figure shows that G_s and β are proportional. In the range where β is positive, G_s can be minimized by setting β close to zero. If the return spring angle β is less than 0, direction of spring deformation is reversed, so it can not act as a compression spring. Additionally, because of the space constraint, tension spring can not be used. Therefore, β should be selected close to zero.

Another constraint is based on the negative clutch torque. As an example, clutch torque acts in the opposite direction on clutch side plate when engine brake is activated. (Fig. 5) At this time, since the direction of T_c is opposite in (1), (10) is changed as in (14).

$$T_c = \frac{\mu R_c (R_b \tan\alpha + \frac{a}{b} R_a)}{R_b \tan\alpha + \mu R_c} F_a - \frac{\mu R_c (R_s \sin\beta + R_b \tan\alpha \cos\beta)}{R_b \tan\alpha + \mu R_c} F_s \tag{14}$$

In (14), the coefficient of $\mu R_c F_a$ is defined as self-de-energizing gain (G_{de}) and the coefficient of $\mu R_c F_s$ are defined as self-de-energizing return spring gain ($G_{s,de}$). Since G_{de} has a larger denominator than G, G_{de} has a smaller value than G. For same reason, $G_{s,de}$ has a smaller value than G_s. If G_{de} is too small, the maximum force of selected actuator through G can not maintain clutch engagement. For additional reasons, G_{de} must have a value greater than 1 to reduce energy consumption compared to conventional DCT like G. Therefore, G_{de} must be greater than 1.

Assuming that maximum engine brake torque is equal to maximum engine torque multiplied by a constant, it can be expressed as (15).

$$T_{\max.eng.brake} = \eta T_{e.\max} \tag{15}$$

Where, $T_{\max.eng.brake}$ and $T_{e.\max}$ are maximum engine brake torque and maximum engine torque respectively. η is assumed to be 0.1 in this manuscript because it has a value of 0.05 to 0.15 for a typical passenger car. If the constraints of G and G_s are satisfied, G_s is close to zero, so maximum engine torque that can be transferred to maximum force of clutch actuator can be expressed by modifying (10) as in (16).

$$T_{e.\max} = 2T_c = 2\mu R_c G F_{a.\max} \tag{16}$$

Based on the above equations, the constraint for maintaining engagement when negative clutch torque acts can be expressed as (17).

$$\eta T_{e.\max} < 2\mu R_c G_{de} F_{a.\max} \tag{17}$$

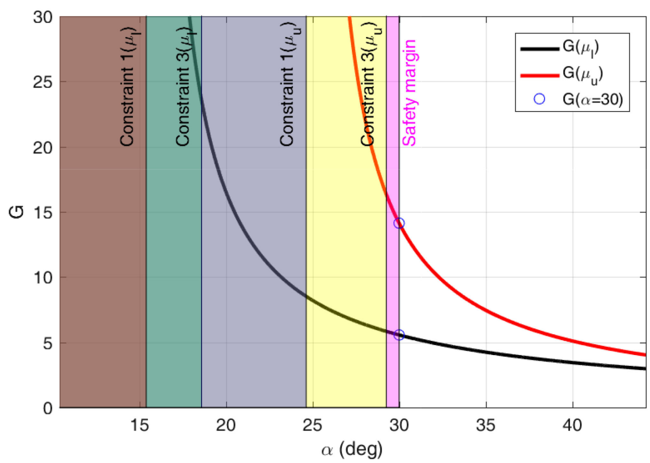

Fig. 6. Selection of self-energizing gain using constraint of proposed BR-DCT. Each constraint is represented by a shaded area of a different color.

Substituting (16) into (17) derives to the inequality (18).

$$\eta < \frac{G_{de}}{G} = \frac{R_b \tan\alpha - \mu R_c}{R_b \tan\alpha + \mu R_c} \tag{18}$$

In conclusion, the BR-DCT satisfies the inequality (18) to maintain engaging force even under the condition of the engine brake.

G is greater than G_{de} means that the BR-DCT has a similar characteristic to the one-way clutch because the maximum friction torque varies depending on the direction of the transmission torque. Therefore, in the torque phase where two clutches come in contact simultaneously (tie-up), the maximum friction torque of the clutch to disengage (off-going clutch) is reduced. Therefore, mechanically, the torque transmitted to the off-going clutch is reduced, which reduces the jerk of the output torque caused by the clutch tie-up. Using this advantage to control output torque, the same result can be achieved with less control effort.

The design constraint of the proposed BR-DCT can be summarized as follows.

1) $\tan\alpha > \mu\frac{R_c}{R_b}$: To avoid self-locking and achieve energy efficiency. (from $G > 0$)
2) $\beta > 0, \beta \to 0 (G_s \to 0)$: To achieve energy efficiency and conservation for compress spring.
3) $\tan\alpha < \mu\frac{R_c(1+\eta)}{R_b(1+\eta)}$: To hold the clutch engaging when engine brake (negative clutch torque) applied.

Therefore, in order to increase the energy efficiency and controllability of the BR-DCT, α satisfying the constraints of 1) and 3) should be selected. Since G and α are inversely proportional (see Fig. 6), smallest α can be selected to maximize G. From this, we could choose G that satisfies all conditions. The constraint of β is reflected in the design of the BR-DCT prototype.

C. Design Parameter Selection

The design parameters of the BR-DCT targeting the passenger vehicle were selected as shown in Table III through the constraint selection method of the proposed BR-DCT design parameters.

TABLE III
SELECTED BR-DCT DESIGN PARAMETERS

Symbol	Value ($\mu = 0.3$)
α	30°
β	5.6°
G	6.19
G_{de}	1.90
G_s	0.10
$G_{s,de}$	0.09

Since the target vehicle is a general passenger car, and the BR-DCT of this vehicle is aimed at transmitting the maximum engine torque ($T_{e.\max} = 250$ Nm). Where, μ is assumed to be in the range of $\mu_l = 0.27$ and $\mu_u = 0.45$, which is clutch disk friction coefficient range of a typical passenger car. The changes of G and G_s within this range are shown in right side of the Fig. 4. In this figure, G and G_s increase with μ, and G_{de} and $G_{s,de}$ decrease. Among them, G varies greatly according to the change of μ. Based on this phenomenon, it is easy to estimate μ using clutch torque and actuator force. Also, R_c, R_a, R_b and a/b were chosen to maximize G considering packaging. As shown in Fig. 6, α is selected as minimum value satisfying the four constraints according to the range of μ, considering safety margin, and corresponding G is also determined.

The BR-DCT prototype is designed as shown in Fig. 7 based on the selected design parameters. The prototype is almost same size as the commercial DCT clutch pack. In addition, actuation lever which shape is similar to the diaphragm spring was designed to save space and implement a self-energizing mechanism. The number of ball-ramp sets and the ball diameter, which are additional design parameters, were selected in total 16 sets and 8 mm considering the maximum transmission torque without increasing the size of the packaging. As a result, energy consumption of clutch actuators is expected to be improved from a minimum of 5.56 to a maximum of 14.13 times compared to the conventional DCT through the design parameter selection of the proposed BR-DCT.

III. ENERGY EFFICIENCY SIMULATION

Simulations were conducted to investigate the effect of energy consumption of clutch actuator on the proposed BR-DCT. The simulation environment used SimDriveline from MATLAB/Simulink. For comparison, clutch actuators modeled with conventional DCT and BR-DCT were constructed on same two vehicles. In particular, clutch actuator of the BR-DCT is modeled to have a different self-energizing gain depending on the sign of clutch torque. Clutch actuator used in this simulation was an electric motor. Power calculation for modeling energy consumption of a clutch actuator using an electric motor is expressed by following equation.

$$P_{act.in} = Vi = Fv + i^2R \tag{19}$$

Where, V is motor supply voltage, i is motor current, R is motor resistance, F is actuator force, and v is actuator velocity. In other words, it is a power balance equation that generates mechanical energy from electrical energy and losses due to

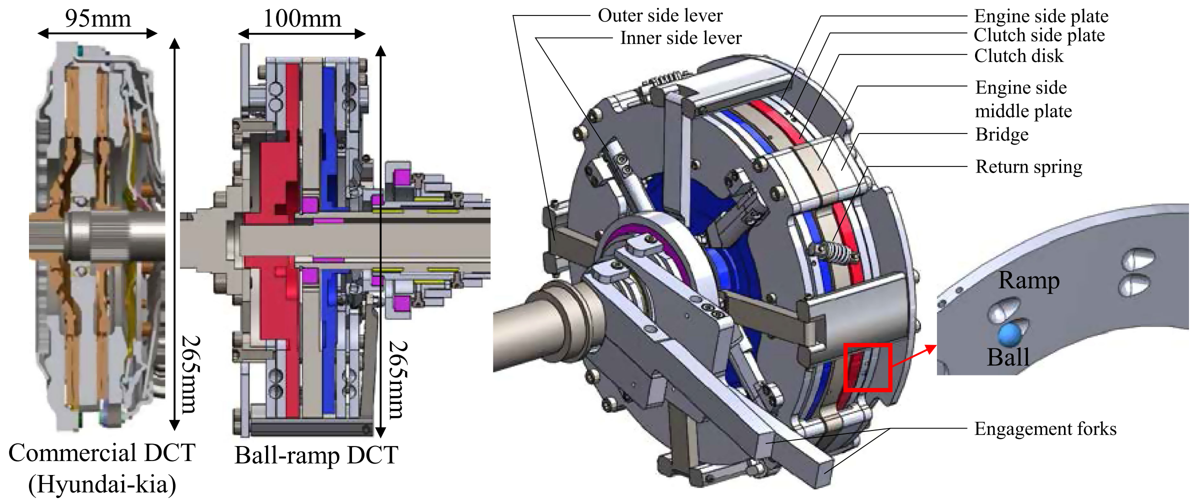

Fig. 7. Left: Comparison of cross-section between BR-DCT and commercial DCT (Hyundai-kia [22]). The sizes of the two DCT packages are almost same. Right: The name of each part of BR-DCT and ball and ramp structure.

resistance. Therefore, energy consumed by clutch actuator is sum of mechanical energy and resistance loss.

The modeling of alternator to calculate fuel consumption according to electric energy consumption was performed by multiplying electric power conversion efficiency by mechanical power. The electric power produced by alternator is used not only as a clutch actuator but also as an energy source for vehicle electrical components. Therefore, consumption power of auxiliary parts is expressed as P_{aux}, and relation is expressed as (20).

$$P_{alt} = (\eta T \omega)_{alt} = P_{act.in} + P_{aux} \tag{20}$$

Where, T is alternator torque, ω is alternator angular velocity, and η is electric energy conversion efficiency. In addition, value of G was selected as the minimum value of 5. And the difference in current consumption is assumed to be five times. All powertrain, vehicle, transmission controller except clutch actuator were selected equally.

The simulation was based on the scenario that driver model follows the driving cycle. At this time, in order to see energy consumption of clutch actuator according to frequent shift, New European Driving Cycle (NEDC) was used as representative driving cycle as shown in Fig. 8(a).

In the simulation results of Fig. 9, vehicle equipped with BR-DCT has less current consumption of clutch actuator and smaller alternator negative torque than vehicle with conventional DCT. In other words, it means that electrical energy of vehicle is small in the same driving situation and torque of alternator applied to the engine is reduced. In conclusion, BR-DCT improves fuel efficiency compared to conventional DCT in the same driving condition.

Table IV shows total amount of electric energy used by clutch actuators and total fuel consumption as a result of simulation. Using the proposed BR-DCT under same driving conditions, current consumption is reduced and energy consumed by clutch actuator is reduced by 84%. The clutch actuator energy saving effect affects fuel consumption, which means that a vehicle with BRDCT consumes 0.5% less fuel than a conventional DCT equipped vehicle. This result implies that fuel consumption can be improved by reducing energy consumption of clutch actuator.

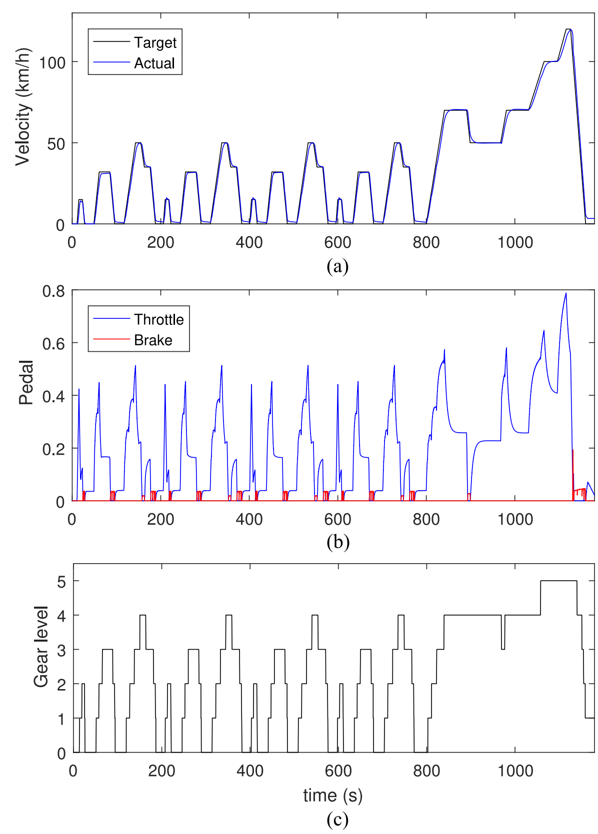

Fig. 8. (a) Comparing the target and actual speed of vehicle with the driving cycle. (b) The amount of accelerator pedal and brake pedal used by driver model. (c) The gear level of vehicle that receives driver command and is controlled by transmission controller.

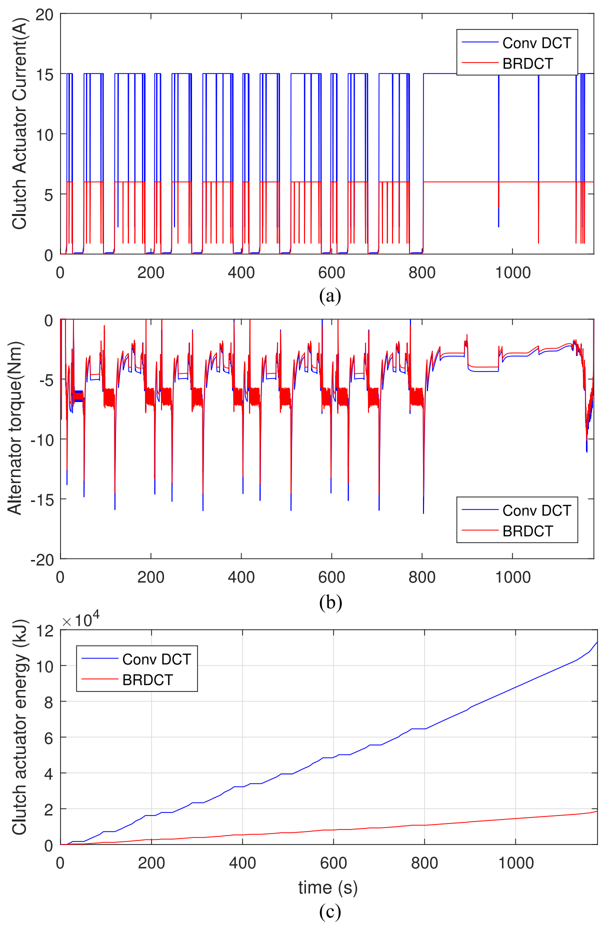

Fig. 9. (a) The current used by clutch actuator. (b) Negative alternator torque to generate auxiliary power and clutch actuator power. (c) The amount of electrical energy supplied to clutch actuators, including mechanical power and resistance losses.

TABLE IV
FUEL CONSUMPTION SIMULATION RESULT

	Conv. DCT	BR-DCT
Clutch actuator total energy consumption (kJ)	113,070	18,526
Vehicle total fuel consumption (g)	7,709	7,671

IV. EXPERIMENTAL ANALYSIS OF BR-DCT

A. Experimental Setup

Experiments were conducted to verify energy-efficiency improvement of BR-DCT clutch actuators. Therefore, a test bench simulating the powertrain of a vehicle with DCT was constructed as a module structure and a conventional DCT module designed for comparison with the proposed BR-DCT module was constructed.

The test bench consists of engine, DCT, vehicle, and load modules to simulate powertrain (Fig. 10). Each module can be disassembled or assembled. Therefore, it is possible to change target vehicle or replace the transmission. Engine module uses Permanent Magnet Synchronous Motor (PMSM) which can output torque up to 170 Nm and inertia disk for inertia modeling of internal combustion engine is connected. PMSM operates to simulate an engine that is compatible with the DCT's specifications. The target engine has a displacement of 1,600cc, a fuel of gasoline, a maximum power of 200 hp and a maximum torque of 270 Nm. The DCT module after the engine module consists of clutch pack & actuator, gear, torque sensor, and final gear. The clutch pack & actuator can be replaced with BR-DCT and conventional DCT as shown in Fig. 11. The DCT module is fixed to the first and second gears of each clutch for installation of the torque sensor.

Both DCTs use a linear actuator using a brushless DC (BLDC) motor and a ball screw as a clutch actuator. An incremental encoder is used to measure the displacement of clutch. A force sensor is mounted on the BR-DCT to measure engagement force. For conventional DCT module, 6-speed DCT clutch pack and actuator of Hyundai-kia were used. For comparison, BR-DCT and conventional DCT were same clutch disk. In addition, since the map between current and force was measured in advance for force measurement of conventional DCT clutch actuator, current measurement value can be converted to force in the experiment. Thus, energy consumption of the BR-DCT and the conventional DCT can be measured by integrating actuator power equation (19). Therefore, since torque capacity and friction coefficient of clutch disk are same, energy efficiency can be evaluated according to the change of clutch actuator.

The vehicle module consists of a shaft for oscillation implementation of the output shaft and an inertia plate and reduction gear for mass implementation. The vehicle module used in experiment modeled weight of a typical passenger car of 1500 kg. The load module is implemented to generate negative torque using the electro-hydraulic brake system.

B. Self-Energizing Gain Validation

The self-energizing gain mentioned in Section II is the most important factor that can represent energy efficiency of the BR-DCT. In (10), G can be measured in relation to $T_c - G_s F_s$ and F_a. Therefore, $T_c, G_s F_s$ and F_a should be measured for G measurements. In (4), where T_c is the friction torque acting on clutch disk, μ depends on the friction state. Specifically, in case of static friction, it means limit value of the friction coefficient. In case of kinetic friction, it is expressed as a constant. Therefore, a constant friction coefficient can be assumed if the torque sensor measures T_c in clutch slip state where kinetic friction is maintained (clutch and engine speeds are different). Also, $G_s F_s$ changes with displacement of clutch disk, so it can be measured through the actuator encoder. Therefore, in the experimental scenario, clutch slips and kinetic friction torque is transmitted to vehicle by holding a constant force to clutch actuator and temporarily increasing torque of engine when vehicle speed is low. G can be measured by measuring T_c (torque sensor), F_a (force sensor) and $G_s F_s$ (encoder) at this time. Where μ is assumed to be nominal value of 0.3. In this case, the self-energizing gain of the BR-DCT is expected to be 6.19 and the self-de-energizing gain to be 1.9. This experimental scenario is carried out in both clutch 1 with first gear and clutch 2 with second gear.

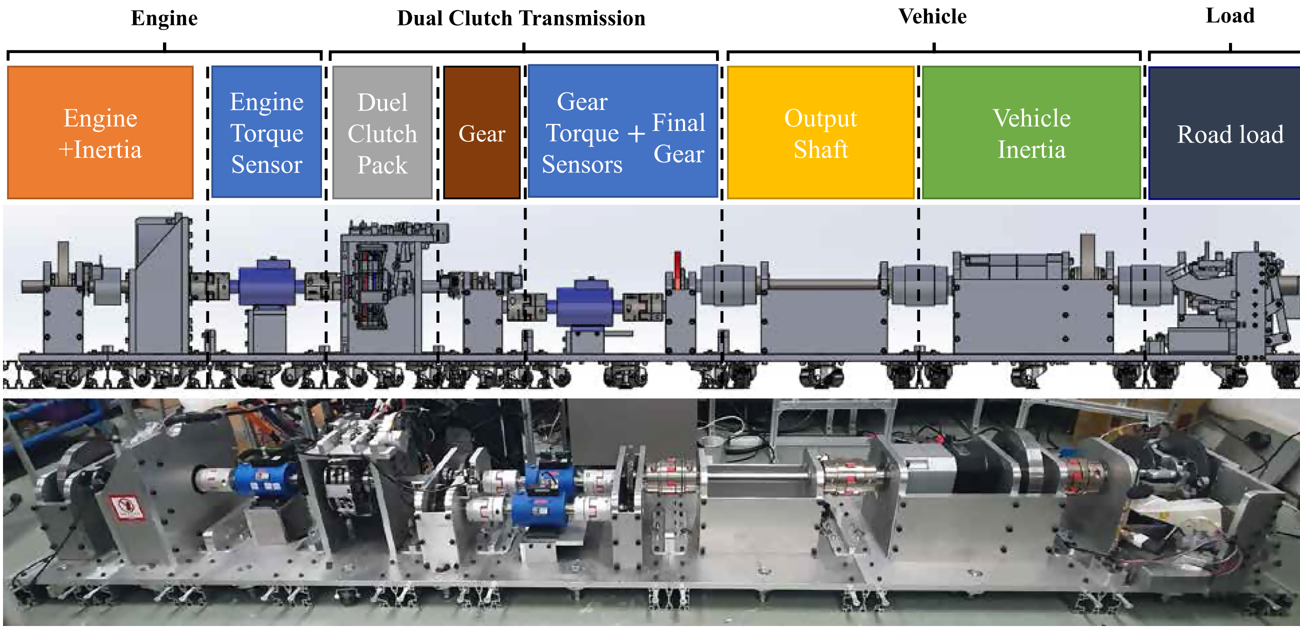

Fig. 10. Powertrain test bench for experimental analysis. Each powertrain component is interchangeable and modular, so it can be replaced with other modules.

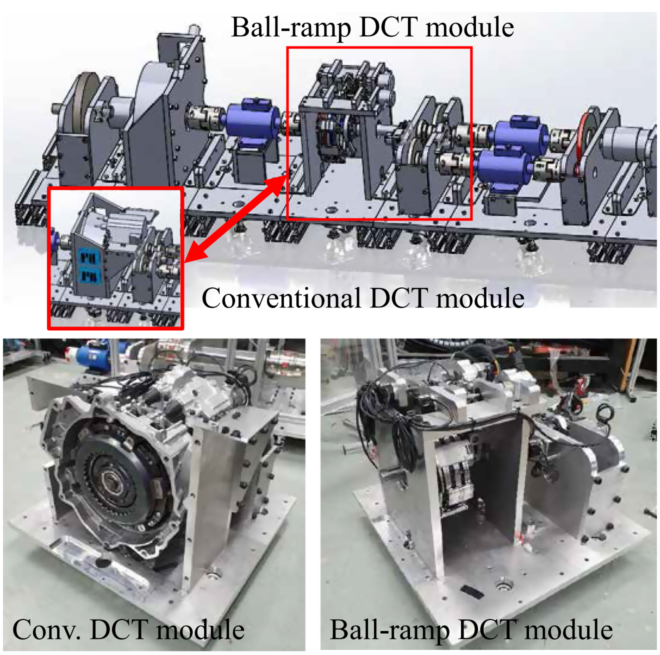

Fig. 11. Exchangeable BR-DCT module and conventional DCT module. Conventional DCT uses 6-speed DCT of Hyundai kia.

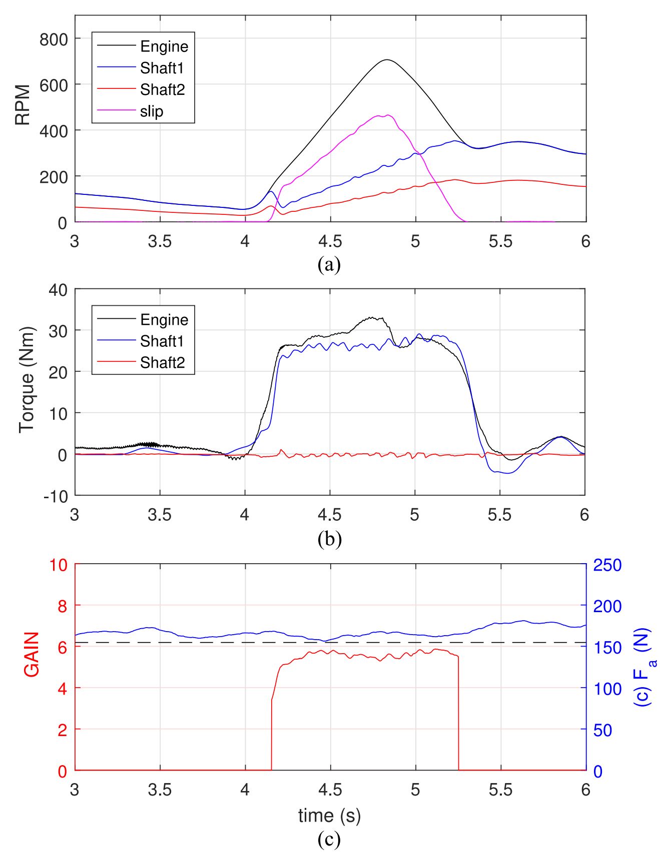

Fig. 12. Self-energizing gain validation of clutch 1.

Fig. 12(c) shows the self-energizing gain when engine torque is transmitted in the slip state by engaging clutch 1 with the first gear. As can be seen in Fig. 12(a) , at 4.15 seconds clutch slip occurred by increasing engine torque while vehicle speed was low and clutch actuator was holding a constant force. It can be seen that G maintains a constant value in slip occurrence interval (Fig. 12(c), 4.15–5.25 seconds). Also, G is larger than 1 and slightly smaller than the design value 6.19 (dotted line). Since static friction clutch torque is generated when clutch slip does not occur, it does not correspond to the maximum value of G. Therefore, G did not plot when slip did not occur. This experiment was repeated several times after changing F_a. As a result, average value of $G = 5.29$ was measured.

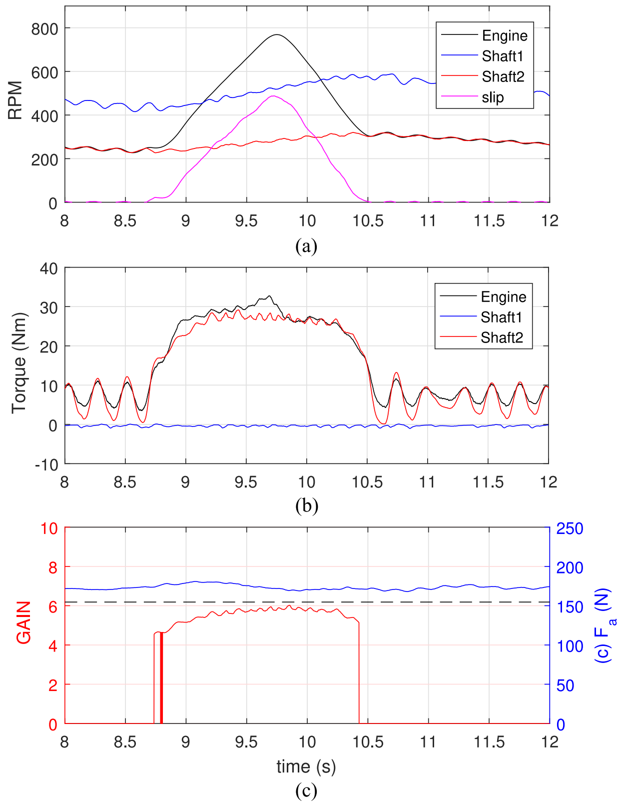

Fig. 13. Self-energizing gain validation of clutch 2.

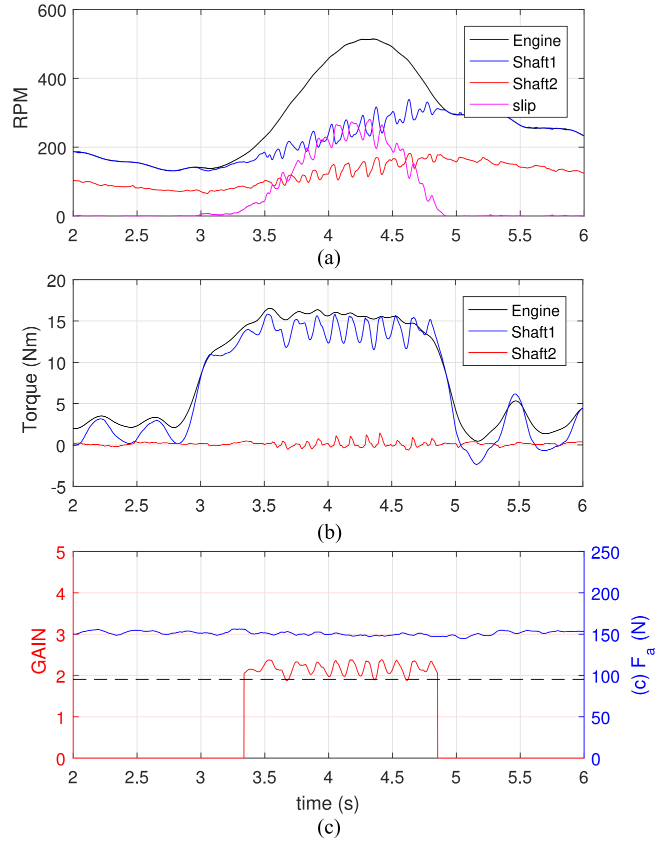

Fig. 14. Self-de-energizing gain validation of clutch 1.

Fig. 13 shows the result of same experiment in Fig. 12 in clutch 2. A slip occurred at 8.7 seconds, and G remained relatively constant at this interval. Also, G is greater than 1 and slightly smaller than the design value of 6.19. In the same way as clutch 1 experiment, a average value of $G = 5.70$ was measured.

In both experiments, we can see that G is measured slightly lower than the design value, because F_a which has a force measurement error due to friction of the ball-screw and other parts is measured based on force sensor of clutch actuator. Experimental results show that the self-energizing mechanism consumes less energy than the conventional DCT because it can reduce engaging force of clutch actuator. Also, since G has a constant value, proportional control is possible instead of on-off by using clutch actuator force when controlling clutch torque.

As mentioned in (14), when the sign of clutch torque changes, the self-de-energizing effect occurs and G changes to G_{de}. Therefore, it is necessary to check G_{de} when changing direction of clutch torque to see if clutch engagement can be maintained well even under engine brake. Figs. 14 and 15 show the experiment using same experimental scenarios as in Figs. 12 and 13, with engine torque reversed. The clutch slip occurred in 3.2 seconds, and G_{de} is greater than 1 and slightly larger than the design value of 1.9. In (c) of both figures, $T_c - G_{s,de}F_s$ according to F_a change is shown, and slopes of each clutch obtained with least square are 2.08 and 1.96. As in the case of the G measurement experiment, it appears that there is a difference between the design value and measured value due to the F_a measurement error caused by friction between the parts.

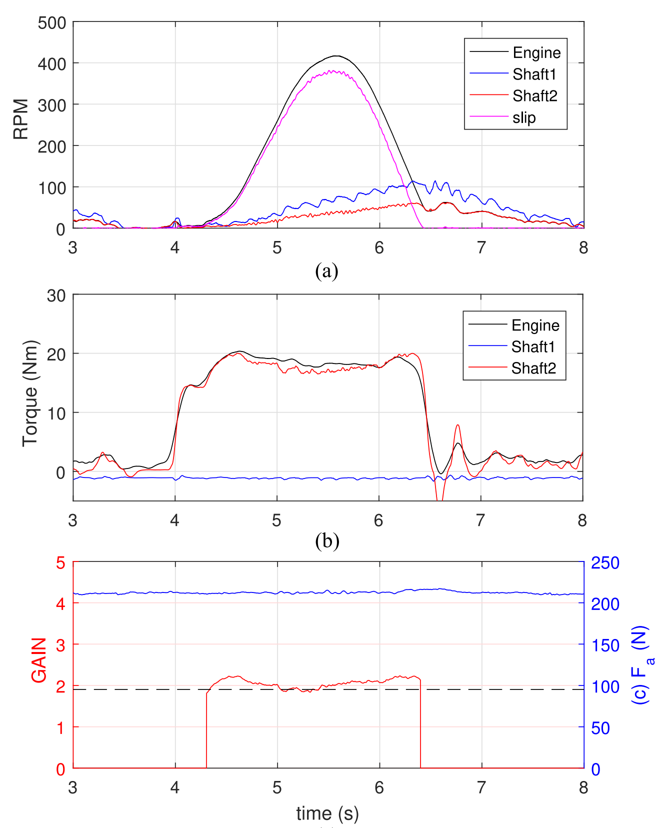

Fig. 15. Self-de-energizing gain validation of clutch 2.

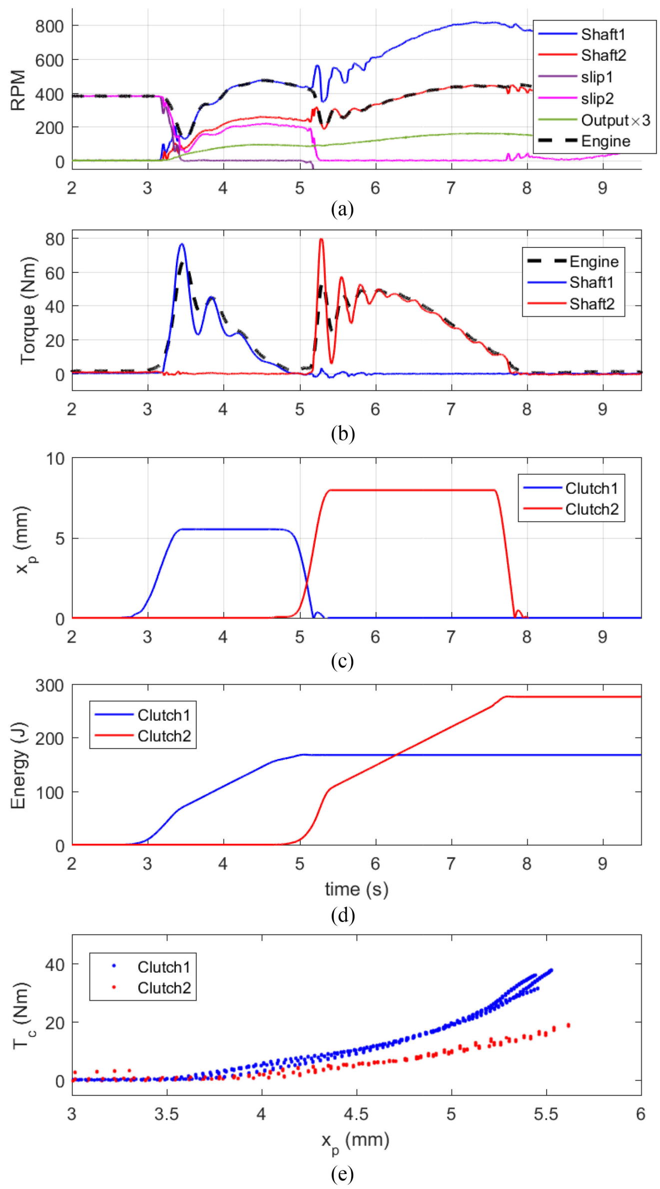

Fig. 16. Conventional DCT transmission experiment.

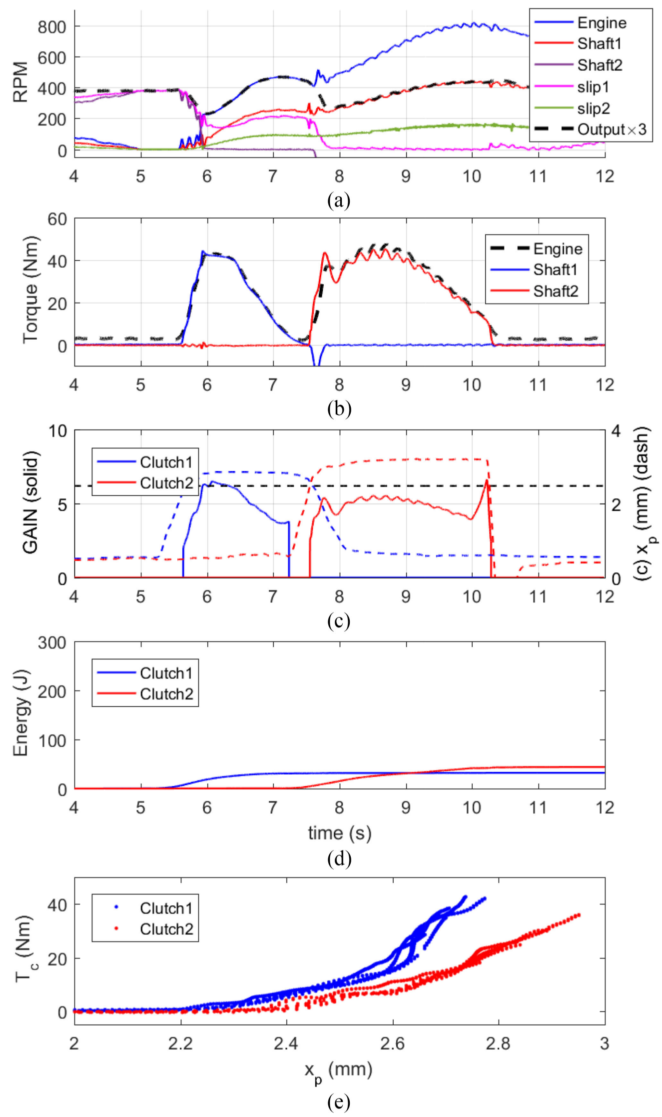

Fig. 17. BR-DCT transmission experiment.

In this experiment, G_{de} is greater than 1 and maintains a constant value, so clutch torque control is possible even under engine brake conditions. Also, because it is measured at 2.08 and 1.96, it consumes less energy than conventional DCT to maintain clutch engagement.

C. Transmission Experiment

The greatest difference between the conventional DCT and the BR-DCT is energy consumption of clutch actuator when transmitting same clutch torque. Therefore, energy consumption of actuator can be compared through a shift experiment that each DCT transmits same clutch torque.

Section IV-B showed the self-energizing gain is relatively constant, which means that clutch torque can be controlled through force control of actuator when clutch slip occurs. However, clutch torque is proportional not only to actuator force, but also to position in conventional DCT. In addition, some DCT controller studies perform clutch torque control based on clutch actuator position control. [23]–[25] Therefore, in order to control clutch torque by using both actuator force and position as in the conventional DCT, the relationship between position and clutch torque needs to be clarified. Therefore, T_c and x_p should be measured when a clutch slip occurs.

For this reason, a shift scenario which has a lot of actuator movements was selected to accurately identify position and torque. The engine controls to maintain constant speed, shifting to first gear connected to clutch 1, and then shifting to second gear connected to clutch 2. Where, input of clutch actuator was created as an open loop. Also, for a reasonable comparison of energy consumption of actuators, we selected the scenario that both vehicle reach same speed within same time. Therefore, the magnitude of input of both DCT actuators was modified to have same slip time. As a result, both vehicles will shift at same slip time, so final speed is same. Therefore, the total torque

TABLE V
FUEL CONSUMPTION SIMULATION RESULT

Actuator energy consumption(J)	Conv. DCT	BR-DCT	Energy saving
Simulation (first transmission)	385.6	68	82%
Experiment	443.4	75.68	83%

transmitted during shifting becomes equal. Figs. 16 and 17 show the experimental results based on these experimental scenarios.

In Figs. 16 and 17(a) , engine maintains 400 rpm and controller engages clutch 1 to deliver torque to vehicle between 3.2 seconds and 5.7 seconds. After that, clutch 1 with first gear is disengaged and clutch 2 with second gear is engaged. In addition, both conventional and BR-DCT have a slip time similar to 0.3 seconds. Also, rpm of output shaft in both figures accelerate to 50 rpm for 3.7 seconds. Referring to (c) of the Fig. 17, the G of the BR-DCT reaches about 5.4 while clutch 1 slip occurs(at 5.9 seconds). This is same as the experimental result Section IV-B. For comparison, the actuator energy consumption of each DCT and its differences are shown in Table V along with the simulation results in Section III.

Fig. 16(e) shows clutch torque according to actuator position of the conventional DCT. It can be seen that both clutch 1 and clutch 2 of the conventional DCT transmit same T_c at the same position. This means that clutch torque can be controlled by position control. Fig. 17(e) shows clutch torque according to actuator position of BR-DCT, which is similar to conventional DCT.

In other words, BR-DCT has characteristics similar to conventional DCT and means that clutch torque can be controlled by applying same position control method. In (d) of the two graphs and Table V, energy consumed by clutch actuator can be compared. The energy consumed by clutch actuators of the BR-DCT is about five times less than energy consumed by actuators of the conventional DCT. In conclusion, the BR-DCT has same features and functions as the conventional DCT, but clutch actuator energy consumption is reduced upto 83%.

V. CONCLUSION AND FUTURE WORK

Conventional DCT has many advantages compared to AT and MT, but it has a disadvantage of high energy consumption of clutch actuator. However, using clutch actuator mechanism of the proposed BR-DCT, self-energizing effect can be used to reduce energy consumption of clutch actuator. In Section IV-C, energy consumption of the BR-DCT is about 5 times less than that of the conventional DCT. In addition, the BR-DCT have same characteristics as the conventional DCT, which is suitable for use in vehicle transmissions. Specifically, the actuator force and clutch torque should be linear when the clutch slips and the clutch torque can be controlled by the position control of the actuator. This characteristic proved by the self-energizing gain validation experiment of Section IV-B that clutch torque is constantly proportional to force of actuator when clutch slip occurs like conventional DCT. In other words, self-energizing gain was constant when clutch slip occurred and value was measured to be close to the design value. In addition, clutch torque control according to position control is possible because actuator position and clutch torque have one to one relationship in the shift test(Section IV-C). Therefore, the ball-ramp DCT can be applied to control methods that use the actuator position as a control input used in conventional DCT, and is suitable for use as a vehicle transmission.

However, since right side of the Fig. 4 shows that G varies with change of μ, μ must be accurately estimated to determine clutch torque. Therefore, it is necessary to propose a μ estimation algorithm using change of G. To this end, the design of a BR-DCT model-based μ and clutch torque observer will be proposed in a future study. In Fig. 16(b), the self-energizing gain does not reach its maximum immediately when clutch slip occurs. It can be seen that the gain is not constant in transition situation where actuator force begins to be transmitted to clutch. However, in Fig. 16(e), clutch torque is determined by actuator position. That is, if the gain does not reach maximum value, clutch torque can be controlled through actuator position, and if the gain is stabilized, clutch torque can be controlled through actuator force. Therefore, future research will provide an index to detect the degree of self-energizing effect, and switch control methods of actuator position and force control.

REFERENCES

[1] A. Irimescu, L. Mihon, and G. Pădure, “Automotive transmission efficiency measurement using a chassis dynamometer,” *Int. J. Automot. Technol.*, vol. 12, no. 4, pp. 555–559, 2011.

[2] A. Turner, K. Ramsay, R. Clark, and D. Howe, “Direct-drive rotary-linear electromechanical actuation system for control of gearshifts in automated transmissions,” in *Proc. IEEE Vehicle Power Propulsion Conf.*, 2007, pp. 267–272.

[3] J. Franco, M. A. Franchek, and K. Grigoriadis, “Real-time brake torque estimation for internal combustion engines,” *Mech. Syst. Signal Process.*, vol. 22, no. 2, pp. 338–361, 2008.

[4] H. Kuroiwa, N. Ozaki, T. Okada, and M. Yamasaki, “Next-generation fuel-efficient automated manual transmission,” *Hitachi Rev.*, vol. 53, no. 4, pp. 205–209, 2004.

[5] M. Kulkarni, T. Shim, and Y. Zhang, “Shift dynamics and control of dual-clutch transmissions,” *Mechanism Mach. Theory*, vol. 42, no. 2, pp. 168–182, 2007.

[6] M. Schuster, “Utilizing an energy-in-energy-out approach to simplify heavy-duty transmission efficiency testing,” SAE Technical Paper, Tech. Rep. 2000-01-3483, 2000.

[7] B. Matthes, “Dual clutch transmissions–lessons learned and future potential,” *SAE Trans.*, vol. 144, pp. 941–952, 2005.

[8] U. Wagner, R. Berger, M. Ehrlich, and M. Homm, “Electromotoric actuators for double clutch transmissions,” in *Proc. 8th LuK Symp.*, 2006, pp. 137–153.

[9] C.-Y. Cho, J.-H. Kam, H.-K. Hong, and C. Lövenich, “More efficiency with the dry seven-speed dual-clutch transmission by Hyundai,” *ATZ Worldwide*, vol. 118, no. 6, pp. 38–41, 2016.

[10] W. Grosspietsch and J. Sudau, “Dual clutch for powershift transmissions-a traditional engaging element with new future?” *VDI BERICHTE*, vol. 1565, pp. 259–274, 2000.

[11] U. Wagner and A. Wagner, “Electrical shift gearbox (ESG)-consistent development of the dual clutch transmission to a mild hybrid system,” SAE Technical Paper, Tech. Rep., 2005.

[12] S. Bai, R. L. Moses, T. Schanz, and M. J. Gorman, “Development of a new clutch-to-clutch shift control technology,” *SAE Trans.*, vol. 111, pp. 1663–1672, 2002.

[13] W. Chen, S. Round, and R. Duke, “Design of an auxiliary power distribution network for an electric vehicle,” in *Proc. 1st IEEE Int. Workshop Electron. Des., Test Appl.*, 2002, pp. 257–261.

[14] I. Evtimov, R. Ivanov, and M. Sapundjiev, “Energy consumption of auxiliary systems of electric cars,” in *Proc. MATEC Web Conf.*, vol. 133. EDP Sciences, 2017, Art. no. 06002.
[15] T. Salameh, “SOFC auxiliary power units (APUs) for vehicles,” in *Proc. TRRF05 Fuel Cell Technol.*, Dec. 2008, pp. 1–8.
[16] G. Leen and D. Heffernan, “Expanding automotive electronic systems,” *Computer*, vol. 35, no. 1, pp. 88–93, 2002.
[17] J. G. Kassakian, H.-C. Wolf, J. M. Miller, and C. J. Hurton, “Automotive electrical systems circa 2005,” *IEEE Spectr.*, vol. 33, no. 8, pp. 22–27, Aug. 1996.
[18] J. J. Oh, J. Kim, and S. B. Choi, “Design of self-energizing clutch actuator for dual-clutch transmission,” *IEEE/ASME Trans. Mechatronics*, vol. 21, no. 2, pp. 795–805, Apr. 2016.
[19] J. Kim and S. B. Choi, “Design and modeling of a clutch actuator system with self-energizing mechanism,” *IEEE/ASME Trans. Mechatronics*, vol. 16, no. 5, pp. 953–966, Oct. 2011.
[20] J. Kim and S. B. Cho, “Self-energizing clutch actuator system: Basic concept and design,” in *Proc. FISITA World Automot. Congr.*, 2010, pp. 1–8.
[21] B. Mller, M. Kneissler, M. hias Gramann, N. Esly, R. Daikeler, and I. Agner, “Smaller, smoother, smarter advance development components for double clutch transmissions,” Schaeffler Kolloquium, Baden-Baden, Tech. Rep. 9, 2010, pp. 164–177.
[22] C.-Y. Cho, J.-H. Kam, H.-K. Hong, and C. Lövenich, “More efficiency with the dry seven-speed dual-clutch transmission by Hyundai,” *ATZ Worldwide*, vol. 118, no. 6, pp. 38–41, 2016.
[23] J. Park, S. Choi, J. Oh, and J. Eo, “Adaptive torque tracking control during slip engagement of a dry clutch in vehicle powertrain,” *Mechanism Mach. Theory*, vol. 134, pp. 249–266, 2019.
[24] H.-S. Jeong and K.-I. Lee, “Friction coefficient, torque estimation, smooth shift control law for an automatic power transmission,” *KSME Int. J.*, vol. 14, no. 5, pp. 508–517, 2000.
[25] H.-S. Jeong and K.-I. Lee, “Shift characteristics analysis and smooth shift for an automatic power transmission,” *KSME Int. J.*, vol. 14, no. 5, pp. 499–507, 2000.

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