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Control Strategy for a Two Mode Hybrid Electric Vehicle Using EVT Mode and Fixed Gear Mode

Jeongmin Kim[1], Jaeyoung Kang[2], Yongha Kim[3], Hyunsoo Kim[4]
[1]*School of Mechanical Engineering, Sungkyunkwan University, Suwon, Korea, pcpppp@empal.com*
[2]*School of Mechanical Engineering, Sungkyunkwan University, Suwon, Korea, kdance00@skku.edu*
[3]*School of Mechanical Engineering, Sungkyunkwan University, Suwon, Korea, yongha5@hanmail.net*
[4]*School of Mechanical Engineering, Sungkyunkwan University, Suwon, Korea, hskim@me.skku.ac.kr*

Abstract

In this paper, a control strategy is proposed for a power split type two mode HEV which consists of two(2) electrically variable transmission modes and four(4) fixed gear modes. The optimal operation mode is selected from the view point of the maximum available output torque with respect to the driver's demand and vehicle velocity. And the optimized shift map is constructed by the optimization process, which minimizes the fuel consumption. It is found from the simulation results that the control strategy for the power split two mode HEV shows satisfactory performance in driving and regenerative braking, which provides the demanded wheel torque with minimum fuel consumption.

Keywords: Hybrid electric vehicle, Power split hybrid, Planetary gear, Lever analysis, Two mode HEV

1 Introduction

Various hybrid architectures have been proposed and series hybrid, parallel hybrid and power split hybrid configurations are most typical ones in open literatures. Series hybrid is conceptually the simplest HEV which has no direct mechanical link between the engine and wheels. The engine power only can be delivered by generator and motor(by two motor-generators), in other words, it has only electrical path between the engine and wheels. So even the engine speed is decoupled from the vehicle speed, it shows poor efficiency due to the energy conversion and large electric machine size limits its applicability to specific vehicle types. Parallel hybrid, which uses one electric motor, has been used in many passenger car applications since the electric machines can be sized only for the desired functions rather than the full engine power and drivetrain efficiency is relatively high. Since parallel hybrid has direct mechanical link between the engine and wheels when the engine is on, it requires some kinds of transmission. In these HEVs, the motor is considered as an auxiliary power unit which assists the engine[1]. Therefore, relatively small size motor is used between the engine and the motor. This architecture has some advantages such as simple control strategy and low extra cost. However, small motor size can not allow ZEV(zero emission vehicle) mode operation and limits the recuperation energy in the regenerative braking. And in order to realize the EV(electric vehicle) mode, the clutch should be installed between the engine and the motor, and relatively large motor is required to propel the vehicle.

The power split configuration has both mechanical and electrical path combining the planetary gear set and two motors. In one path (mechanical path), the power from the gasoline engine is directly transmitted to the vehicle's wheels. In the other path (electrical path), the power from the engine is converted into the electricity by a generator to drive the electric motor or to charge the battery[2]. This configuration has many advantages, such as continuously variable transmission(CVT) operation, electric-only capability, and seamless capability to decouple the road load demand from the engine in a manner similar to that in a series hybrid[3]. Since the engine is decoupled from the wheels, the engine can be operated on its best efficiency region and the mechanical path from the engine to the wheels provides better fuel economy compared with the series hybrid. In addition, the electric-only drive can be implemented owing to its relatively large size motors. Moreover, it can start and stop the engine while the vehicle is moving. These characteristics enable improvements to fuel consumption and reductions in emissions[4].

However, in spite of the many advantages of the power split hybrid, the followings have been frequently mentioned as major drawbacks such as energy loss in some speed ratio range due to power circulation and high cost that comes from using two motors. The energy loss is due to the fact that a portion of the engine output power may be routed along the closed path through a mechanical to electrical or from the electrical to the mechanical, in other words, double energy mode conversion in supplying driven-axle power from the engine[5]. This process is so called “energy recirculation” and decreases the transmission efficiency[6]. In particular, in the case of an HEV concept that is geared to a low fuel consumption, the high power loss of the transmission can render a design useless[7]. And in the power split HEV, since one of the two motors is used to achieve the engine optimal operation on the best fuel efficient region while the other motor needs to be operated to propel the vehicle or to charge the battery depending on the driver’s demand and the vehicle condition, it causes relatively large motor size. Recently, power split hybrid configuration is extended to two mode hybrid configuration with a fixed gear mode to overcome these drawbacks by combining parallel and power split HEV concepts, which can be seen in GM two mode power split hybrid, 'Tahoe'. It can has characteristics of parallel and power split HEV having many merits of power split configuration explained above and relatively small motor. However, this type of hybrid configuration requires complicated control strategy due to the increased gear sets, added clutches and brakes to implement the two modes.

In this paper, control strategy for a power split type two mode HEV with fixed gear is proposed considering the wheel torque demand while minimizing the fuel consumption and performance of the control strategy is evaluated by simulation.

2 Dynamic equations of two mode hybrid electric vehicle

Figure 1 shows GM two mode power split structure investigated in this study[8]. The two mode HEV consists of three planetary gears, two electric machines, two brakes and two clutches. The two mode HEV can provide two EVT(electrically variable transmission) modes and four fixed gear modes by operating brakes and clutches as shown in Table 1.

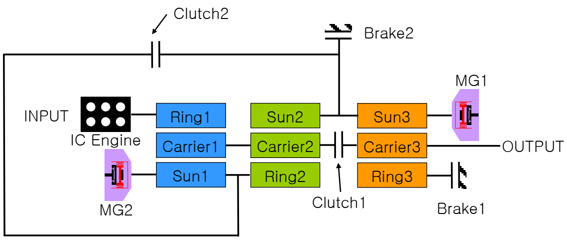

Figure 1: Structure of GM two mode HEV[8]

Table 1: Clutches and brakes operation for two mode HEV

Mode	Brake1	Clutch1	Brake2	Clutch2
EVT(Input split)	O			
Fixed gear 1	O			O
Fixed gear 2	O	O		
EVT (Compound split)		O		
Fixed gear 3		O		O
Fixed gear 4		O	O	

Figure 2 shows lever models of the two mode HEV for each operation mode[9]. From the lever model of Figure 2a, the dynamic equations of the input split mode can be obtained as

$$J_e \dot{\omega}_e = T_e + \frac{Z_{r2}Z_{r1}/Z_{s2} + Z_{r1}}{Z_{r2}Z_{r1}/Z_{s2} - Z_{s1}} T_{MG2} \tag{1}$$

$$T_{OUT} = (1+\frac{Z_{r3}}{Z_{s3}})((1-\frac{Z_{r2}Z_{r1}/Z_{s2} + Z_{r1}}{Z_{r2}Z_{r1}/Z_{s2} - Z_{s1}})T_{MG2} + T_{MG1}) \tag{2}$$

$$\omega_{MG1} = (1+\frac{Z_{r3}}{Z_{s3}})\omega_{OUT} \tag{3}$$

$$\omega_{MG2} = (1-\frac{Z_{r2}Z_{r1}/Z_{s2} + Z_{r1}}{Z_{r2}Z_{r1}/Z_{s2} - Z_{s1}})(1+\frac{Z_{r3}}{Z_{s3}})\omega_{OUT} + \frac{Z_{r2}Z_{r1}/Z_{s2} + Z_{r1}}{Z_{r2}Z_{r1}/Z_{s2} - Z_{s1}}\omega_e \tag{4}$$

where r is the ring gear, s is the sun gear, c is the carrier, Z_r is the teeth number of the ring gear, Z_s is the teeth number of the ring gear, 1 means the first planetary gear, 2 means the second planetary gear, 3 means the third planetary gear, J_e is the inertia of the engine, ω_e is the engine speed, T_e is the engine torque, ω_{MG1} is the MG1 speed, T_{MG1} is the MG1 torque, ω_{MG2} is the MG2 speed, T_{MG2} is the MG2 torque, ω_{OUT} is the output speed and T_{OUT} is the output torque. And the vehicle dynamic equation can be obtained as

$$T_{OUT} = \frac{r_t}{N_f}(M\dot{V} + F_{load}) \tag{5}$$

$$\omega_{OUT} = \frac{V}{r_t}N_f \tag{6}$$

where r_t is the tire radius, N_f is the final reduction gear ratio, M is the vehicle mass, V is vehicle velocity and F_{load} is the road load. From Eq. (1), (2) and (5), it is noted that engine speed ω_e is not affected by the MG1 torque, T_{MG1} meanwhile the vehicle velocity V depends on T_{MG1}. This is because the motor MG1 is not connected with the output shaft. Dynamic equations for the compound split mode can be obtained in a similar manner from the lever model of Figure 2b.

$$J_e\dot{\omega}_e = T_e - \frac{Z_{r1}}{Z_{s1}}T_{MG2} + \frac{Z_{r1}Z_{r2}}{Z_{s1}Z_{s2}}T_{MG1} \tag{7}$$

$$T_{OUT} = (1+\frac{Z_{r1}}{Z_{s1}})T_{MG2} + (1-\frac{Z_{r1}Z_{r2}}{Z_{s1}Z_{s2}})T_{MG1} \tag{8}$$

$$\omega_{MG1} = (1-\frac{Z_{r1}Z_{r2}}{Z_{s1}Z_{s2}})\omega_{OUT} + \frac{Z_{r1}Z_{r2}}{Z_{s1}Z_{s2}}\omega_e \tag{9}$$

$$\omega_{MG2} = (1+\frac{Z_{r1}}{Z_{s1}})\omega_{OUT} - \frac{Z_{r1}}{Z_{s1}}\omega_e \tag{10}$$

It is seen from Eq. (7) and (8) that the engine speed and the output shaft torque(vehicle velocity) depend on the motor torque T_{MG1} and T_{MG2}, which means that T_{MG1} and T_{MG2} should be controlled considering the engine speed and the vehicle velocity. From the lever model of Figure 2c~Figure 2f, dynamic equations for the fixed gear modes can be obtained as

$$\omega_e = N_e \times \omega_{OUT} \tag{11}$$

$$\omega_{MG1} = N_{MG1} \times \omega_{OUT} \tag{12}$$

$$\omega_{MG2} = N_{MG2} \times \omega_{OUT} \tag{13}$$

$$T_{OUT} = N_e \times T_e + N_{MG1} \times T_{MG1} + N_{MG2} \times T_{MG2} \tag{14}$$

where N_e is the equivalent gear ratio between engine and output, N_{MG1} is the equivalent gear ratio between MG1 and output and N_{MG2} is the equivalent gear ratio between MG2 and output. The N_e, N_{MG1} and N_{MG2} can be obtained as Table 2.

Table 2: Gear ratios of GM two mode HEV for fixed gear modes

	N_e	N_{MG1}	N_{MG2}
Fixed Gear 1	$1+\frac{Z_{r3}}{Z_{s3}}$	$1+\frac{Z_{r3}}{Z_{s3}}$	$1+\frac{Z_{r3}}{Z_{s3}}$
Fixed Gear 2	$1+\frac{Z_{s1}Z_{s2}Z_{r3}}{Z_{r1}Z_{r2}Z_{s3}}$	$1+\frac{Z_{r3}}{Z_{s3}}$	$1-\frac{Z_{s2}Z_{r3}}{Z_{r2}Z_{s3}}$
Fixed Gear 3	1	1	1
Fixed Gear 4	$1-\frac{Z_{s1}Z_{s2}}{Z_{r1}Z_{r2}}$	0	$1+\frac{Z_{s2}}{Z_{r2}}$

3 Control strategy

In this study, a control strategy for the power split type two mode HEV with fixed gears is proposed considering the following design specifications;

- is able to provide the demanded wheel torque by the driver.
- minimizes the fuel consumption while satisfying the driver's demand.

The control strategy is developed by the optimization process, which selects the optimal operation mode with respect to the driver's demand and the vehicle velocity.

3.1 Maximum output torque

First, the operation mode is derived from the viewpoint of the wheel torque demand. The objective function is set up to maximize the output torque $|T_{out}(V,T_e,T_{MG1},T_{MG2})|$ under the constraint of the battery power.

$$Obj: \quad Max\,|T_{out}(V, T_e, T_{MG1}, T_{MG2})| \tag{15}$$

$$Subject\ to: \ P_b \le P_{b\_\max} \tag{16}$$

$$0 \le V \le 180 \tag{17}$$

$$0 \le T_e \le T_{e\_\max} \tag{18}$$

$$-T_{MG1\_\max} \le T_{MG1} \le T_{MG1\_\max} \tag{19}$$

$$-T_{MG2\_\max} \le T_{MG2} \le T_{MG2\_\max} \tag{20}$$

$$0 \le \omega_e \le \omega_{e\_\max} \tag{21}$$

$$-\omega_{MG1\_\max} \le \omega_{MG1} \le \omega_{MG1\_\max} \tag{22}$$

$$-\omega_{MG2\_\max} \le \omega_{MG2} \le \omega_{MG2\_\max} \tag{23}$$

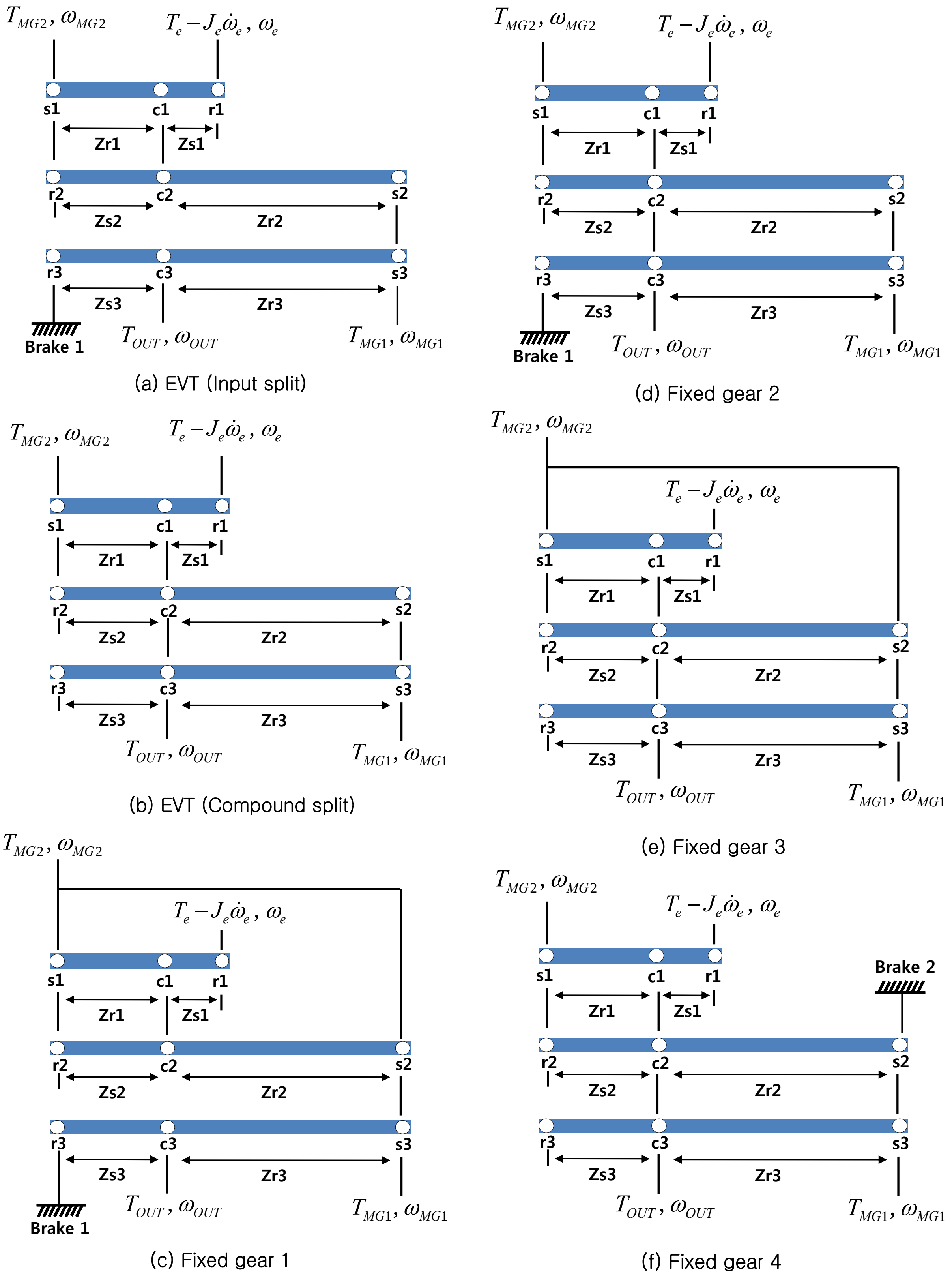

Figure 2: Lever model of two mode HEV

where P_b is the battery power and *max* means maximum value. In the optimization process, the vehicle velocity V, the engine torque, T_e, the two motor torque T_{MG1}, T_{MG2} are used as the design parameters. The objective function is set up to maximize the output torque $|T_{out}(V,T_e,T_{MG1},T_{MG2})|$ under the constraint of $P_b < P_{b\_max}$. The optimization procedure is as follows : first, for the given ω_e, the engine torque T_e is obtained as the maximum torque from the engine characteristics curve(Figure 3). T_{MG1} and T_{MG2} are determined as the maximum torque from the motor characteristic curve(Figure 4) using ω_{MG1} and ω_{MG2} that can be calculated by the lever relationship, Eq (3), (4), (9), (10), (12) and (13). T_e, T_{MG1} and T_{MG2} are selected within the physical operating range between the minimum and the maximum. This calculation process is repeated for the vehicle velocity, $V=0 \sim 180kph$. The vehicle parameters used in the simulations are shown in Table 3. The maximum output torque plot for each operation mode is shown in Figure 5.

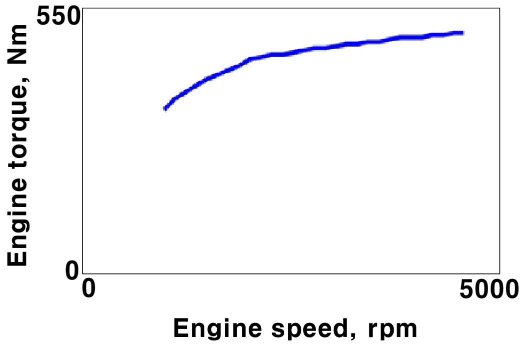

Figure 3: Engine characteristic curve

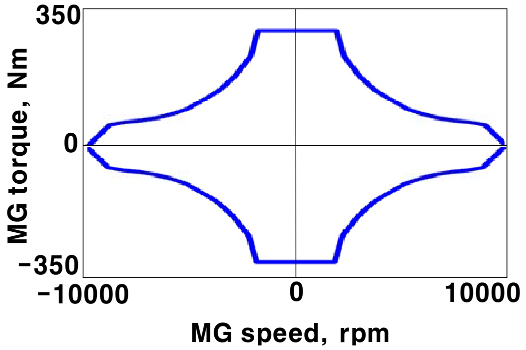

Figure 4: MG characteristic curve

Table 3: Vehicle parameters

Parameters	
All planetary gear ratio , Z_r / Z_s	2
Ring gear teeth number, Z_r	78
Final reduction gear ratio, N_f	3.08
Tire radius, r_t	0.4 m
Engine inertia, J_e	0.7 kg·m^2
Vehicle mass, M	2680 kg
Engine	250kW 1,000 RPM base speed
MG1, MG2	60 kW, 300 Nm max. torque 10,000 RPM max. speed
Battery	Ni-MH, 30kW 300 Volt

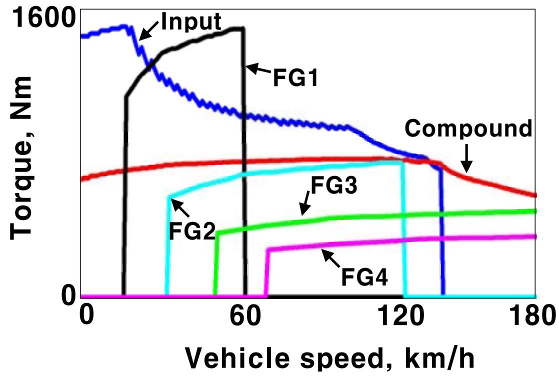

Figure 5: Maximum output torque for each operation mode

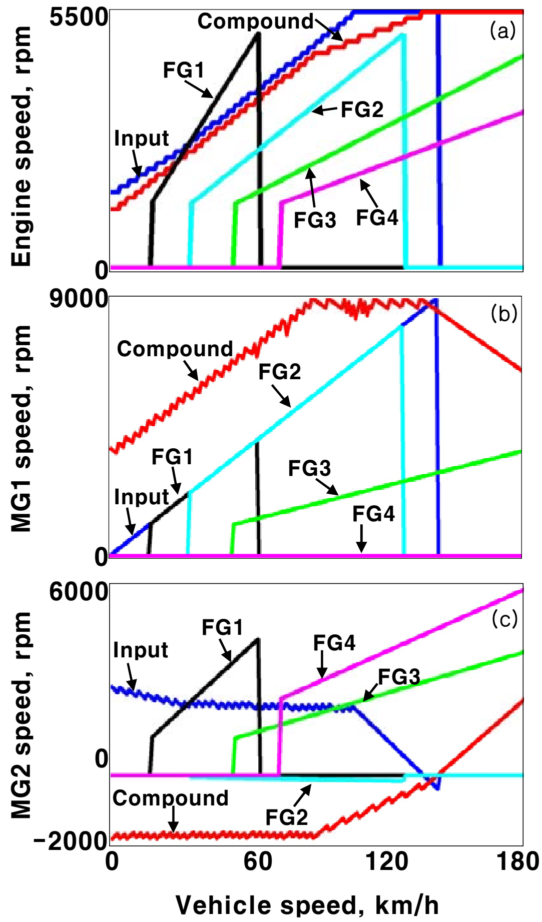

Figure 6: Speed characteristics for maximum torque

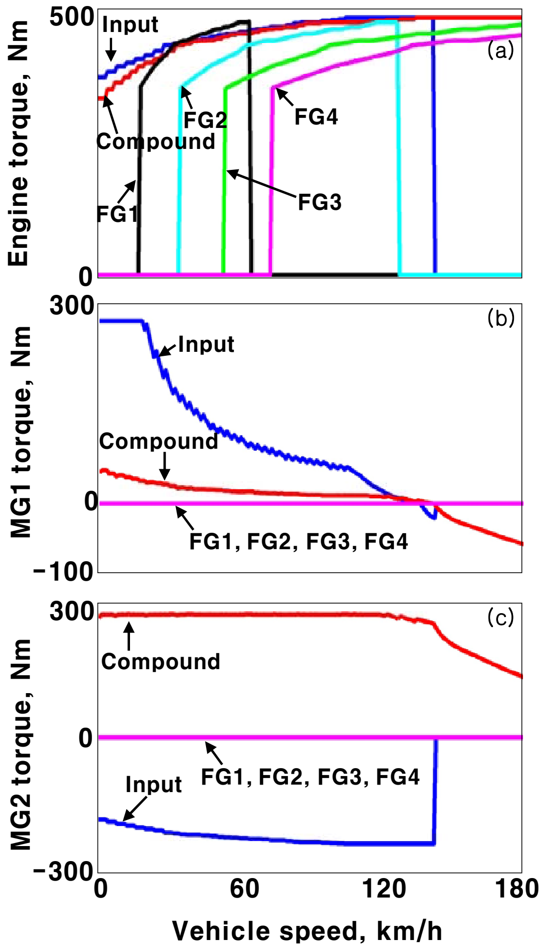

Figure 7: Torque characteristics for maximum output torque

It is seen from Figure 5 that the input split mode shows relatively high torque for V=0~140kph the third planetary gear(Figure 1) plays like the reduction gear with the gear ratio, $1+Zr3/Zs3$. However, it can not provide the torque above V=140kph due to the MG1 speed limitation. The compound split mode shows almost constant torque for most of the velocity range. For the fixed gear mode, the output torque is determined according to the gear ratio. It is noted that the fixed gear FG1 can not deliver the output torque since the engine can not work below V=20kph that is equivalent to the engine speed ω_e=1,000rpm. However, the EVT mode such as the input or compound split mode is able to generate the output torque in this range since the engine speed can be maintained above 1,000rpm by the MG1 and MG2 control.

Figure 6 shows the engine speed(a), MG1 speed(b) and MG2 speed(c) for each mode. For the input split mode, the engine speed increases with the increasing vehicle velocity up to V=90kph and remains constant at ω_e=5,500rpm in the high velocity range. However, the engine cannot be operated above V=140kph due to the MG1 speed limitation. For the compound split mode, the engine speed increases with the vehicle velocity and reaches the maximum value ω_e=5,500rpm. For the fixed gear mode, the engine speed increases in the vehicle velocity range, which is determined depending on the gear ratio.

Figure 7 shows the maximum available torque of the engine, MG1 and MG2 for each mode. For the fixed gear mode, the MG1 and MG2 torque remain zero since it is assumed that there is no charge or discharge of the battery. However, for the EVT mode, the MG1 and MG2 torque are generated by the lever relationship.

3.2 Optimized shifting map

Now, we need to select the operating mode which minimizes the fuel consumption while satisfying the demanded wheel torque. The optimal operating mode selection for the given driver's demand and vehicle velocity can be defined as the following optimization problem.

$$Obj: \quad Min\ |FC(\text{Mode})| \tag{24}$$

$$Subject\ to: \quad T_{OUT}(\text{Mode}) \leq T_{OUT\_max}(\text{Mode}) \tag{25}$$

where FC(mode) is the fuel consumption for the given operating mode, $T_{OUT\_\max}$(mode) is the maximum available output torque of the given mode (Figure 5).

The optimization process is as follows : first, the fuel consumption of the given operating mode is calculated with respect to the engine speed and torque that can be determined from Eqs. (1)~(14). At this moment, the demanded wheel torque is compared with the maximum available torque of each mode. Among the candidates, the operating mode which shows the lowest fuel consumption is selected as the optimal operation mode.

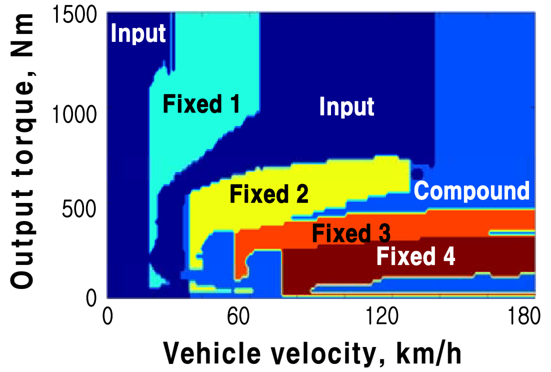

Figure 8: Optimized shift map of power split type two mode HEV

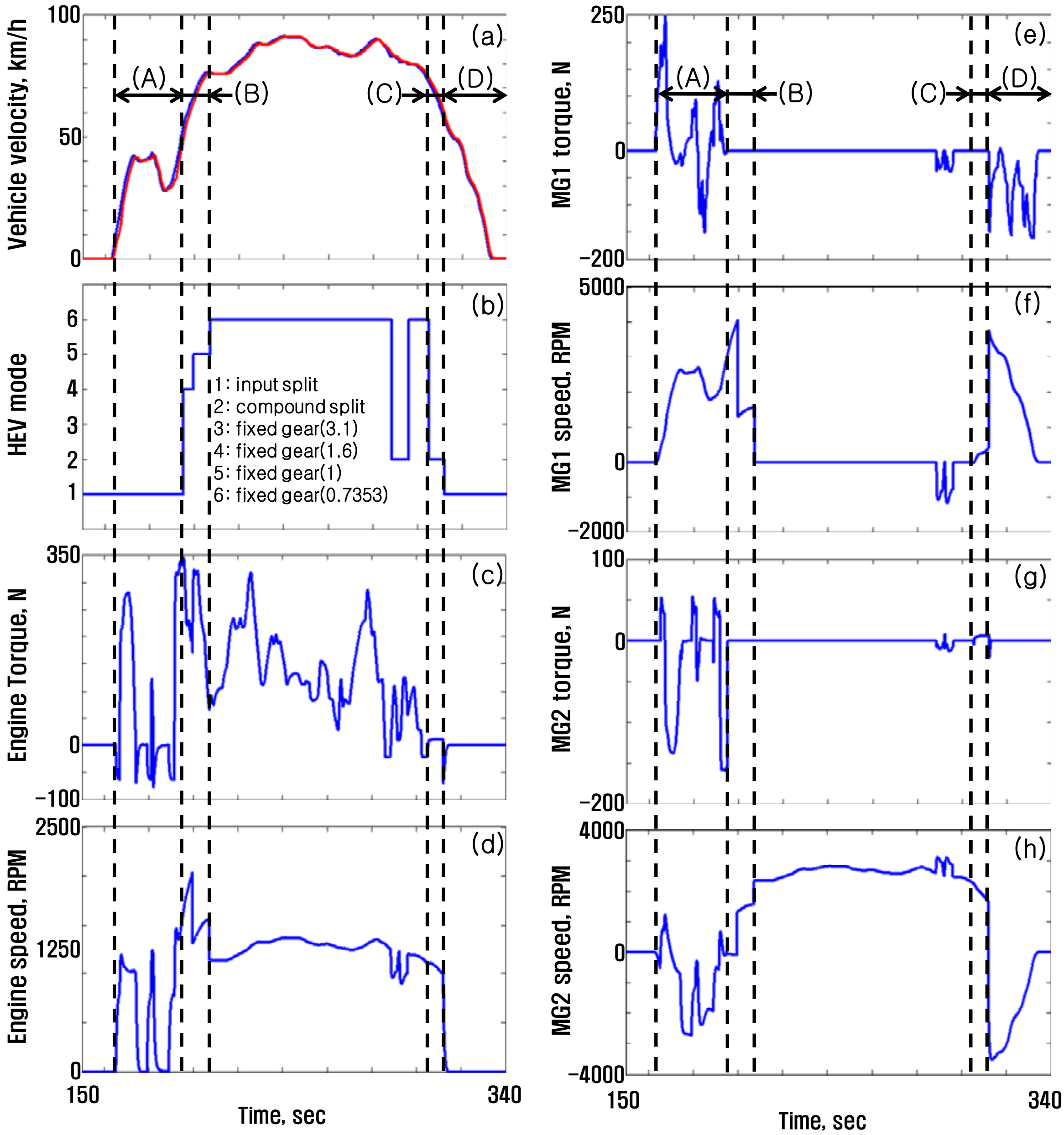

Figure 9: Simulation results of power split type two mode HEV

In Figure 8, the optimized shift map(case of P_b is zero) is shown, which is obtained from the optimization process.

4 Simulation results

In Figure 9, simulation results are shown for the 150~340 seconds of FUDS(federal urban driving schedule). The vehicle velocity follows the target velocity closely by the six operation modes(Figure 9a and Figure 9b). For V=0~40kph(A), the vehicle is operated by the input split mode. When the vehicle requires acceleration performance(B), the vehicle is operated by the fixed gear mode 4, 5 according to the shift map(Figure 8). For the high vehicle velocity over V=70kph, the vehicle is operated by the fixed gear mode 6. If the vehicle needs deceleration at relatively high vehicle velocity, the compound split mode(C) is used for the regenerative braking. And the vehicle is operated by switching the compound split mode to the input split mode for the regenerative braking at the vehicle velocity V=40~0kph. The input split mode is used for the vehicle velocity under V=40kph(D). Figure 9c and Figure 9d show the engine torque and speed. The engine speed is maintained around

1250 rpm for the best fuel economy. During the input split mode(A), the MG1 and MG2 torque shows almost mirror image to satisfy the EVT system torque equilibrium shown in Figure 7b and Figure 7c. In this paper the MG1 and MG2 are controlled to generate no torque. However, the MG1 and MG2 can be used to generate torque to perform the battery power assistant(B).

5 Conclusion

In this paper, a control strategy for a power split type two mode HEV is proposed. First, dynamic equations of the power split HEV are derived, which consists of two(2) electrically variable transmission modes and four(4) fixed gear modes. Using the dynamic equations, the optimal operation mode is selected from the view point of the maximum available output torque with respect to the driver's demand and vehicle velocity. Next, the optimized shift map is constructed by the optimization process, which minimizes the fuel consumption while satisfying the demanded wheel torque. It is found from the simulation results that the control strategy for the power split two mode HEV shows satisfactory performance in driving and regenerative braking, which provides the demanded wheel torque with minimum fuel consumption.

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Authors

Jeongmin Kim received B.S. and M.S. degrees in mechanical engineering from Sungkyunkwan University, Suwon, Korea, in 2003 and 2005, respectively, where he has been working towards the Ph.D. degree.
His research interests are modelling, design and control of the powertrain system for a hybrid electric vehicle.

Yongha Kim received B.S. degree in mechanical engineering from Sungkyunkwan University, Suwon, Korea, in 2009, where he has been working towards the M.S. degree.
His research topic covers the dual mode power split HEV system.

Jaeyoung Kang received B.S. degree in mechanical engineering from Sungkyunkwan University, Suwon, Korea, in 2009, where he has been working towards the M.S. degree.
He is interested in design and control of the HEV power split system.

Hyunsoo Kim received a Ph.D. degree in mechanical engineering from The University of Texas at Austin, USA, in 1986.
Since 1986, he has worked as a Professor, Chairman, and Dean of the College of Engineering at Sungkyunkwan University. His main research interests focuses on the powertrain system of the HEV, plug-in HEV and In-wheel electric vehicles. Recently, he is working on development of the power split-dual mode HEV system and regenerative braking system.