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A MOTOR-INTEGRATED PARALLEL HYBRID TRANSMISSION

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ABSTRACT

This paper presents a motor-integrated transmission mechanism for use in parallel hybrid electric vehicles. The transmission can provide five basic modes of operation that can be further classified into sixteen sub-modes: one electric motor mode, four engine modes, four engine/charge modes, three power modes, and four regenerative braking modes. Each of these sub-modes can be grouped into like clutching conditions, providing the functional appearance of a conventional 4-speed automatic transmission, with electric launch, engine-only, engine/charge, power-assist, and regeneration capability. CVT capability is provided with one of the engine/charge modes. The kinematics, static torque, and power flow relations for each mode are analyzed in detail. Finally, a notional control strategy is developed. The transmission can be incorporated not only in front-wheel drive but also in rear-wheel drive vehicles. The compactness, mechanical simplicity, and operational flexibility of the transmission make it an excellent candidate for future hybrid electric vehicles.

1 INTRODUCTION

Recently, there has been an increasing interest in the development of parallel hybrid electric vehicles (HEVs). Numerous parallel HEV configurations have been proposed [1-8]. Two parallel HEVs, the Toyota Prius and Honda Insight, have been produced. Others such as Dodges ESX3, Ford Prodigy, and GM Precept are scheduled for production in the near future.

Toyota introduced a parallel HEV called the Prius [9-11]. The Prius uses a simple planetary gear train (PGT) as an electro-mechanical transmission. The simple PGT consists of a sun gear, a ring gear, a carrier, and several planets. The carrier is connected to an engine crankshaft, the sun gear is connected to an electric generator, and the ring gear is used as the output shaft of the transmission. The output shaft is connected to a final reduction unit followed by a differential gearbox to drive the wheels. In addition, an electric motor is connected directly to the output shaft of the transmission. Such arrangement allows the engine power to be split into two paths, one through the ring gear to the output shaft and the other via the sun gear to the electric generator. Some of the electricity generated by the generator is used to drive the electric motor and the remainder is used for charging the batteries. Using the generator to regulate the speed of the engine, the transmission allows the engine to operate at optimal conditions most of the time. In this regard, the PGT functions as a continuously variable transmission (CVT). With an electronic control unit, the system can provide four modes of operation: electric motor mode, CVT mode, power mode, and regenerative braking mode. The potential disadvantages of this design include the need for two electrical motors and a constant power-split of the engine. The inability to mechanically transmit all the engine power directly to the output shaft causes the fuel economy of the vehicle on highway driving to be worse than that of combined city-highway driving.

The Honda Insight is a power-assisted hybrid in which a relatively small permanent-magnet motor is packaged between an engine and a manual transmission with a traditional clutch mechanism [12]. The motor can be used as a starter, a motor for power assist of the engine, and a generator for regenerative braking. It is also used as an engine vibration damper. This space efficient design is considered a mild hybrid (MYBRID), since it uses a relatively small motor for power-assist of the engine. While this design works well for MYBRID vehicles, it may not be appropriate for applications that require a larger motor for launching a vehicle from a standstill.

Both Chrysler and Ford employed Honda's configuration in their prototype PNGV (Partnership for the Next Generation of Vehicle).

Another promising MYBRID configuration, consisting of an automatically controlled manual transmission with a relatively small electric motor coupled directly to the lay-shaft of the transmission, is currently under development by Chrysler. In this design, the electric motor is used to assist the engine for mild hybrid operation. In addition, the motor also provides a limited amount of torque to the drive wheels while the transmission is shifting its gears. The purpose is to reduce the so-called torque hole created during gear shifting events.

This paper presents an improved design of a novel parallel hybrid transmission introduced by Tsai et al. [13, 14]. Similar to the previous design, the proposed HEV requires only one electric motor/generator and one engine. It has five basic modes of operation, called motor mode, power mode, engine mode, engine/charge mode, and regenerative braking mode. In motor mode, the electric motor alone drives the vehicle. In power mode, the transmission sums the power of the engine and the electric motor to drive the vehicle. In engine mode, the mechanism operates as a multi-speed fully automatic transmission. In engine/charge mode, it allows the engine to charge the vehicle batteries and power electrical accessories, while simultaneously powering the vehicle. One of the engine/charge modes provides CVT capability; whereby, the engine can be run at an optimal operating condition by using the electric motor to regulate the speed of the engine and to charge the batteries. During braking periods, kinetic energy of the vehicle is used to charge the batteries. Unlike the Prius, the engine power does not need to be split at all times. Further, it is fail-safe since either the motor or the engine alone can drive the vehicle.

The main improvements of the revised design are: it can be incorporated not only in front-wheel-drive but also in rear-wheel-drive vehicles and the electric motor is integrated coaxially with a compound planetary gear train. Because of the novel arrangement of the clutches, the step ratio changes of the four engine modes are closer to a constant geometric progression. Specifically, the step ratio of the first to the second gear is 2.76/1.64= 1.68, the second gear to the direct drive is 1.64/1.00=1.64, and the direct drive to overdrive is 1.00/0.64= 1.56. The original 2-DOF power mode is replaced by a one-DOF power mode.

In what follows, we first describe the kinematic structure of the hybrid transmission. Then, we analyze the kinematics, static torque and power flow relations. The various modes of operation are described in detail via a numerical example. Finally, a notional control strategy is developed.

2 KINEMATIC STRUCTURE OF THE MECHANISM

Figure 1 shows the schematic diagram of a motor-integrated hybrid transmission mechanism. The transmission is made up of two simple PGTs integrated with an electric motor that can also function as a generator. In the figure, two short parallel edges represent a gear mesh. We call the PGT on the left-hand side of the motor the *input planetary gear train* and the one on the right-hand side of the motor the *output planetary gear train*. The two sun gears are connected together by a common shaft, labeled as link 1, which is attached to the rotor of the electric motor. The input ring gear is connected to the output carrier by a shaft, labeled as link 3. Link 3 is to be connected to a final reduction unit of the transmission. For this reason, we call link 3 the output link. The engine crankshaft can be coupled to link 1 by a rotating clutch C1 or to the input carrier, link 2, by another rotating clutch C2. The output ring gear can be grounded to the casing by a band clutch B1. Lastly, the electric motor can also be held stationary by a band clutch B2.

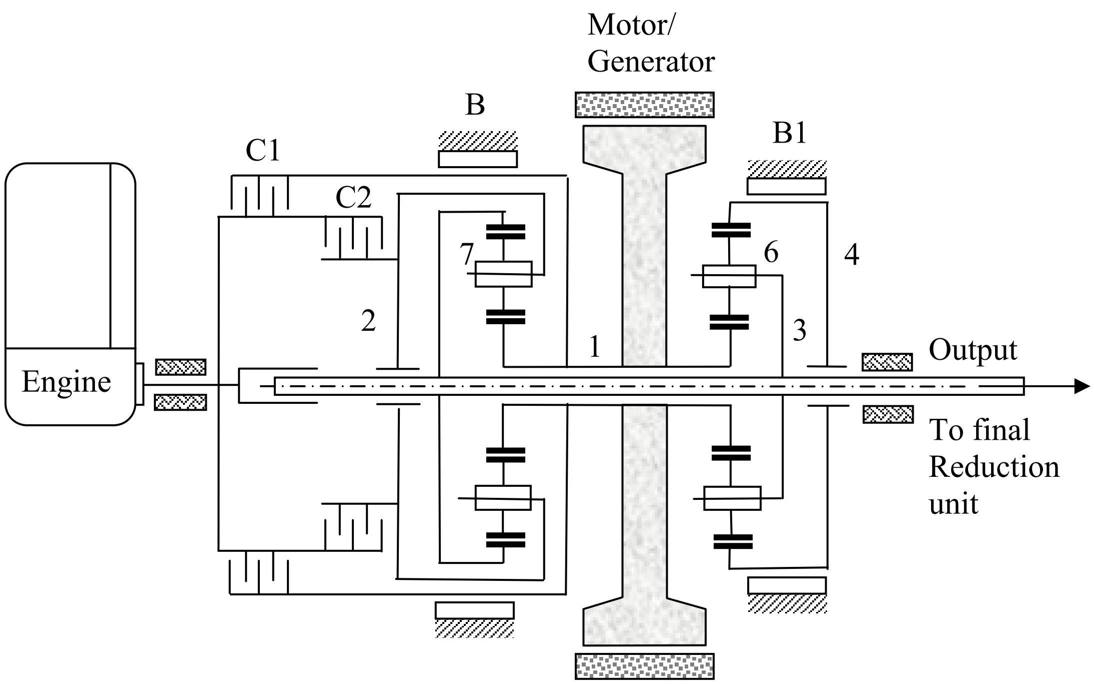

Figure 1. Motor-integrated parallel hybrid transmission.

Depending on which clutches are engaged, five basic modes of operation called *motor mode* (M), *power mode* (P), *engine mode* (E), *engine/charge mode* (EC), and *regenerative mode* (R) are possible. These five modes of operation can be further classified into sixteen sub-modes as shown in Table 1, where an “X” denotes that the corresponding clutch is engaged. Note that one of the charge modes provides CVT capability. In what follows, we derive the kinematics, static torque, and power flow relations of the transmission before we describe the five modes of operation in detail.

Table 1. Five modes of operation.

Operating Mode		**Clutching Condition**				**Motor Operating Condition**
		C1	C2	B1	B2	
Motor	M			X		Motor
Power	P1	X		X		Motor
	P2		X	X		Motor
	P3	X	X			Motor
Engine	E1	X		X		Freewheeling
	E2		X	X		Freewheeling
	E3	X	X			Freewheeling
	E4		X		X	Stationary
Engine/Charge	EC1	X		X		Generator
	EC2		X	X		Generator
	EC3	X	X			Generator
	CVT		X			Generator
Regenerative	R0			X		Generator
	R1	X		X		Generator
	R2		X	X		Generator
	R3	X	X			Generator

3 KINEMATICS

We apply the fundamental circuit equation for the kinematic analysis of the hybrid transmission [15]. Let *i* and *j* be a gear pair and *k* be the carrier. Then links *i*, *j*, and *k* form a fundamental circuit. A fundamental circuit equation can be written as

$$\omega_i - \omega_k = \pm N_{ji}(\omega_j - \omega_k), \tag{1}$$

where ω_i, ω_j, and ω_k denote the angular velocities of links *i*, *j*, and *k*, respectively, and N_{ji} represents the gear ratio between gears *j* and *i*, that is, $N_{ji} = T_j/T_i$ where T_j and T_i denote the number of teeth on gears *j* and *i*, respectively. The sign in Eq. (1) is positive or negative depending on whether the gear mesh is internal or external.

The gear train shown in Fig. 1 contains four fundamental circuits: (7, 3, 2), (7, 1, 2), (6, 4, 3), and (6, 1, 3). The fundamental circuit equations can be written as

$$\omega_7 - \omega_2 = N_{37}(\omega_3 - \omega_2), \tag{2}$$

$$\omega_7 - \omega_2 = -N_{17}(\omega_1 - \omega_2), \tag{3}$$

$$\omega_6 - \omega_3 = N_{46}(\omega_4 - \omega_3), \tag{4}$$

$$\omega_6 - \omega_3 = -N_{16}(\omega_1 - \omega_3)\,. \tag{5}$$

Equating Eq. (4) to (5) yields

$$\omega_1 + N_{41}\omega_4 - (1 + N_{41})\omega_3 = 0\,, \tag{6}$$

where $N_{41}=N_{46}/N_{16}$. Similarly, equating Eq. (2) to (3) yields

$$\omega_1 + N_{31}\omega_3 - (1 + N_{31})\omega_2 = 0\,, \tag{7}$$

where $N_{31}=N_{37}/N_{17}$. Equations (6) and (7) relate the angular velocities of the four coaxial links, 1, 2, 3, and 4. Given the angular velocities of any two links, we can solve Eqs. (6) and (7) for the other two.

4 STATIC TORQUE AND POWER FLOW RELATIONS

Numerous methods can be applied for power flow analysis of planetary gear trains. Consider the coupled planetary gear set as a system. Under steady-state operation, torques exerted on links 1, 2, 3, and 4 about the central axis of the gear set must be summed to zero; that is,

$$\tau_1 + \tau_2 + \tau_3 + \tau_4 = 0\,, \tag{8}$$

where τ_i denotes an external torque exerted on link i. Similarly, the net power flows into the system via the four coaxial links must be summed to zero; that is,

$$\tau_1\omega_1 + \tau_2\omega_2 + \tau_3\omega_3 + \tau_4\omega_4 = 0\,. \tag{9}$$

Eliminating τ_3 from Eqs. (8) and (9) and making use of Eqs. (6) and (7) yields

$$N_{41}\tau_1 + \left(\frac{N_{41}}{1 + N_{31}}\right)\tau_2 - \tau_4 = 0\,, \tag{10}$$

Similarly, eliminating τ_4 from Eqs. (8) and (9) and making use of Eqs. (6) and (7) yields

$$(1 + N_{41})\tau_1 + \left(1 + \frac{N_{41}}{1 + N_{31}}\right)\tau_2 + \tau_3 = 0\,. \tag{11}$$

Equations (10) and (11) relate torques exerted on links 1, 2, 3, and 4 in terms of the gear ratios N_{31} and N_{41}. Given torques exerted on any two links, we can find those exerted on the other two.

5 DETAILED DESCRIPTION OF THE OPERATING MODES

In this section, we describe the five basic modes of operation in detail, and demonstrate the feasibility of the parallel hybrid transmission via an example. The following data are used for the example:

(a) Input PGT: $T_1 = 42$, $T_7 = 16$, and $T_3 = 74$.

(b) Output PGT: $T_{1'} = 42$, $T_6 = 16$, and $T_4 = 74$.

(c) Final reduction ratio = 3.0.

(d) Tire radius = 0.287 m.

(e) A highway cruising speed of 100 km/hr (62 miles/hr) is desired.

Based on the data given above, we have $N_{41} = N_{31} = 1.76$, producing an output link rotational speed of $\omega_3 = 2{,}775$ rpm at the indicated highway cruising speed.

5.1 Electric Motor Mode

To launch the vehicle from a standstill and for driving in city traffic, the band clutch B1 is engaged and all the other clutches are disengaged. Using the output ring gear as a reaction member, the electric motor alone drives the vehicle through the output PGT in the forward or reverse direction, while the input PGT spins freely.

Since the output ring gear is held stationary, $\omega_4 = 0$ identically. Substituting $\omega_4 = 0$ into Eqs. (6) and (7) yields

$$\omega_1 = (1 + N_{41})\omega_3 = 2.76\,\omega_3, \tag{12}$$

$$\omega_2 = \left(1 + \frac{N_{41}}{1 + N_{31}}\right)\omega_3 = 1.64\,\omega_3. \tag{13}$$

Therefore, the speed ratio between the electric motor and the output link is 2.76. Since the input PGT spins freely, $\tau_2 = 0$ identically. Substituting $\tau_2 = 0$ into Eqs. (10) and (11), we obtain

$$\tau_4 = N_{41}\tau_1 = 1.76\,\tau_{motor}, \tag{14}$$

$$\tau_3 = -(1 + N_{41})\tau_1 = -2.76\,\tau_{motor}. \tag{15}$$

We observe that the reaction torque exerted on link 4 is 1.76 times of the motor torque and the output torque is equal to 2.76 times of the motor torque. The negative sign in Eq. (15) means that the output shaft is applying a torque on the final reduction unit.

5.2 Power Mode

When maximum acceleration is needed or during hill climbing, the transmission is shifted into a power mode. To do this, the engine must be running. Three power modes are available.

P1 Mode. In this mode, the rotating clutch C1 and the band clutch B1 are engaged whereas the other clutches are disengaged. Using the output ring gear as a reaction member, the electric motor and the engine drive the vehicle simultaneously through the output PGT at a reduction while the input PGT spins freely.

Since link 4 is grounded, the speed ratios given by Eqs. (12) and (13) remain valid. Because both the electric motor and engine are connected to a common link, they must rotate at the same speed. Equations (14) and (15) becomes

$$\tau_4 = N_{41}\tau_1 = 1.76(\tau_{motor} + \tau_{engine}), \tag{16}$$

$$\tau_3 = -(1+N_{41})\tau_1 = -2.76(\tau_{motor} + \tau_{engine}). \tag{17}$$

Hence, the reaction torque is equal to 1.76 times of the combined motor and engine torques and the output torque is equal to 2.76 times of the combined motor and engine torques.

Neglecting frictional losses, the output power is given by

$$P_{out} = -2.76(\tau_{motor} + \tau_{engine})\omega_3. \tag{18}$$

The negative sign implies that power is flowing out of the transmission. In this mode, maximum amplification of the input torques is achieved. The maximum output power, however, depends on the speeds of the motor and engine and their speed-torque characteristics.

P2 Mode. The rotating clutch C2 and the band clutch B1 are engaged and the other clutches are disengaged. Using the output ring gear as a reaction member, the electric motor and the engine drive the vehicle simultaneously at a second reduction. However, the input PTG is no longer freewheeling. The engine power transmits through the entire gear train whereas the motor power transmits through the output PGT.

The speed ratios are given by Eqs. (12) and (13). Substituting $N_{31} = N_{41} = 1.76$ into Eqs. (10) and (11) yields

$$\tau_4 = N_{41}\tau_1 + \frac{N_{41}}{1+N_{31}}\tau_2 = 1.76\tau_{motor} + 0.64\tau_{engine}, \tag{19}$$

$$\tau_3 = -(1+N_{41})\tau_1 - \left(1+\frac{N_{41}}{1+N_{31}}\right)\tau_2 = -(2.76\tau_{motor} + 1.64\tau_{engine}). \tag{20}$$

Hence, the motor torque is amplified by 2.76 times and the engine torque is amplified by 1.64 times at the output shaft. The output power is given by

$$P_{out} = -(2.76\tau_{motor} + 1.64\tau_{engine})\,\omega_3 . \tag{21}$$

Because the speeds of the electric motor and engine are related to the output shaft by Eqs. (12) and (13), the maximum output power occurs at a point different from that of the P1 mode.

P3 Mode. Both rotating clutches C1 and C2 are engaged and the other clutches are disengaged. Under this clutching condition, the gear set locks up as a rigid body. Hence, the electric motor and the engine drive the vehicle simultaneously with a one-to-one gear ratio. The output torque and output power are simply given by

$$\tau_3 = -(\tau_{motor} + \tau_{engine}), \tag{22}$$

$$P_{out} = -(\tau_{motor} + \tau_{engine})\omega_3 . \tag{23}$$

5.3 Engine Mode

When the demand for power is low and battery state-of-charge is sufficiently high to handle accessory loads, the transmission can be operated in one of the following engine modes.

E1 Mode. The E1 mode is similar to P1 mode except for the fact that the electric motor is switched off and is freewheeling. Using link 4 as a reaction member, Eqs. (12) and (13) hold for the speed ratios of the four coaxial links. Since the electric motor spins freely, $\tau_{motor} = 0$. Equations (16) and (17) reduces to

$$\tau_4 = N_{41}\tau_1 = 1.76\tau_{engine}, \tag{24}$$

$$\tau_3 = -(1 + N_{41})\tau_1 = -2.76\tau_{engine} . \tag{25}$$

Hence, the engine torque is amplified by 2.76 times at the output shaft. This corresponds to the *first gear* of a conventional automatic transmission. The output power is given by

$$P_{out} = -2.76\,\tau_{engine}\,\omega_3 . \tag{26}$$

E2 Mode. The E2 mode is similar to P2 mode except that the electric motor is switched off and is freewheeling. The output ring gear serves as a reaction member while the engine alone drives the vehicle through the entire gear train at a second *reduction*.

The speed ratios of the four coaxial links are given by Eqs. (12) and (13). Substituting $\tau_1 = 0$ into Eqs. (19) and (20) yields

$$\tau_4 = \frac{N_{41}}{1+N_{31}}\tau_2 = 0.64\,\tau_{engine}, \tag{27}$$

$$\tau_3 = -\left(1+\frac{N_{41}}{1+N_{31}}\right)\tau_2 = -1.64\,\tau_{engine}. \tag{28}$$

The output power is given by

$$P_{out} = -1.64\,\tau_{engine}\,\omega_3. \tag{29}$$

E3 Mode. At moderate vehicle speeds, the transmission can be shifted into a *direct drive* by engaging C1 and C2 clutches simultaneously. In *direct drive*, the planetary gear train locks up as a rigid body. The engine transmits its power directly to the output shaft with a one-to-one gear ratio. The output torque and power are given by

$$\tau_3 = -\tau_{engine}, \tag{30}$$

$$P_{out} = -\tau_{engine}\omega_3. \tag{31}$$

E4 Mode. At high vehicle speeds, the transmission can be shifted into an *overdrive* by engaging C2 and B2 clutches simultaneously. Using the input sun gear as a reaction member, the engine drives the vehicle through the input PGT while the output PGT spins freely.

Since the sun gear is held stationary, ω_1 =0 identically. Substituting ω_1 =0 into Eq. (7) yields

$$\omega_2 = \frac{N_{31}}{1+N_{31}}\omega_3 = 0.64\,\omega_3. \tag{32}$$

Since the output PGT spins freely, $\tau_4 = 0$ identically. Substituting $\tau_4 = 0$ into Eqs. (10) and (11) yields

$$\tau_1 = -\left(\frac{1}{1+N_{31}}\right)\tau_2 = -0.36\,\tau_{engine}, \tag{33}$$

$$\tau_3 = -(1+N_{41})\tau_1 - \left(1+\frac{N_{41}}{1+N_{31}}\right)\tau_2 = -0.64\,\tau_{engine}. \tag{34}$$

The output power is given by

$$P_{out} = -0.64\,\tau_{engine}\,\omega_3. \tag{35}$$

This is an efficient mode for highway cruising when there is no need for charging the batteries. Note that if the E4 mode is not needed, the B2 clutch can be removed from the design to reduce the cost and complexity.

5.4 Engine/Charge Mode.

EC1 Mode. As a compliment to E1 mode, with the motor functioning as a generator, the engine can be used to simultaneously power the vehicle, power electrical accessories, and charge the batteries. The kinematic, torque, and power relationships are identical to those for P1 mode, except that the applied motor torque is negative.

EC2 Mode. As a compliment to E2 mode, with the motor functioning as a generator, the engine can be used to simultaneously power the vehicle, power electrical accessories, and charge the batteries. The kinematic, torque, and power relationships are identical to those for P2 mode, except that the applied motor torque is negative.

EC3 Mode. As a compliment to E3 mode, with the motor functioning as a generator, the engine can be used to simultaneously power the vehicle, power electrical accessories, and charge the batteries. The kinematic, torque, and power relationships are identical to those for P3 mode, except that the applied motor torque is negative.

CVT Mode. At moderate and high speeds or as a compliment to E4 mode, the rotating clutch C2 is engaged whereas all the other clutches are disengaged. The electric motor is switched into a generator for charging the batteries. This clutching condition forces the input PGT to function as a power splitting, one-input and two-output device. Specifically, part of the engine power is directed to the input ring gear to drive the vehicle and the remaining part to the input sun gear to drive the generator. In this mode, it is possible to run the engine at an optimal efficiency point while regulating the speed of the vehicle by controlling the speed and load of the generator. In this regard, the transmission functions as a continuous variable transmission (CVT).

The speed ratios among the four coaxial links are given by Eqs. (6) and (7). Substituting N_{31} = 1.76 into Eq. (7) yields

$$\omega_3 = \frac{-\omega_1 + (1+N_{31})\omega_2}{N_{31}} = -0.57\,\omega_{generator} + 1.57\,\omega_{engine}. \tag{36}$$

Therefore, for a given output shaft speed, ω_3, it is possible to run the engine at an optimal operating speed by controlling the speed of the generator. For example, at a cruising speed of 100 km/hr, ω_3 = 2,775 rpm, if we wish to run the engine at ω_2 = 2,400 rpm, the electric generator should be running at ω_1 = 1742 rpm.

Since link 4 spins freely, the output PGT carries no load. Substituting τ_4 = 0 into Eqs. (10) and (11) yields

$$\tau_1 = -\frac{1}{1+N_{31}}\tau_2 = -0.36\,\tau_{engine}\,, \tag{37}$$

$$\tau_3 = -\frac{N_{31}}{1+N_{31}}\tau_2 = -0.64\,\tau_{engine}\,. \tag{38}$$

Hence, 36% of the engine torque is exerted on the input sun gear; whereas, 64% is exerted on output ring gear. Powers passing through the output shaft and the electric generator are given by

$$P_{out} = -0.64\,\tau_{engine}\;\omega_3\,, \tag{39}$$

$$P_{generator} = -0.36\tau_{engine}\,\omega_1\,. \tag{40}$$

5.5 Regenerative Braking Mode

During the braking periods, the electric motor is switched into a generator. Four regenerative modes are possible.

R0 Mode. With clutch B1 engaged and all the other clutches disengaged, the kinetic energy of the vehicle is directed through the output PGT to the generator for charging the batteries. In this regard, power flows in the opposite direction of motor mode. No engine braking is provided, leading to a maximum recovery of the vehicle kinetic energy.

R1 Mode. With C1 and B1 engaged, part of the kinetic energy of the vehicle is directed to the electric generator for charging the batteries and the remaining part to the engine. Power flows in the opposite direction of P1 mode. Hence, both the engine and electric generator provide braking efforts to the vehicle.

R2 Mode. With C2 and B1 engaged, part of the kinetic energy of the vehicle is directed to the electric generator through the output PGT for charging the batteries and the other part to the engine through the entire gear set. Power flows in the opposite direction of P2 mode. Again, both the engine and electric generator provide braking efforts to the vehicle.

R3 Mode. With both C1 and C2 engaged, the gear set locks up as a rigid body. Power flows in the opposite direction of P3 mode. Both the engine and electric generator provide braking efforts to the vehicle at a one-to-one gear ratio.

6 CONTROL STRATEGY

From Table 1 we note that shifting from M to R0 mode can be easily accomplished by changing the state of the electric motor from motor to generator and vice versa. Similarly, shifting among P1, E1, EC1 and R1 modes; P2, E2, EC2 and R2 modes; and P3, E3, EC3 and R3 modes requires only a change of the state of the electric motor from motor to freewheeling or generator. Hence, we may consider the M and R0 modes as a *motor mode*, P1, E1, EC1 and R1 modes as the *first gear*, P2, E2, EC2 and R2 modes as the *second gear*, P3, E3, EC3 and R3 modes as the *direct drive*, and the E4 mode as the *overdrive*. In this regard, we may reclassify the operating conditions into *motor mode*, *first gear*, *second gear*, *direct drive*, *overdrive*, and *CVT* modes as shown in Table 2. *CVT* may also be thought of as a charge mode compliment to *E4* mode. Each of the first gear, second gear, and direct drive consists of a power mode, an engine-only mode, an engine/charge mode and a regenerative mode.

Table 2. Alternate classification of the operating modes

Operating Mode		**Clutching Condition**				**Motor Operating Condition**
		C1	C2	B1	B2	
Motor	M			X		Motor
	R0			X		Generator
First gear	P1	X		X		Motor
	E1	X		X		Freewheeling
	EC1	X		X		Generator
	R1	X		X		Generator
Second Gear	P2		X	X		Motor
	E2		X	X		Freewheeling
	EC2		X	X		Generator
	R2		X	X		Generator
Direct Drive	P3	X	X			Motor
	E3	X	X			Freewheeling
	EC3	X	X			Generator
	R3	X	X			Generator
Overdrive	E4		X		X	Stationary
CVT	CVT		X			Generator

We may also consider the parallel hybrid transmission as having two driving schedules: a *performance schedule* and an *economy schedule*. The performance schedule provides a better performance at the expenses of less fuel-efficient than that of the economy schedule. In either case, the transmission shifts up from the motor mode to the first gear, second gear, direct drive, and then overdrive or CVT mode. The CVT mode can be applied most efficiently at moderate to high vehicle speeds for charging batteries. When the state of charge of the batteries is high, the transmission may be shifted into the overdrive to further improve the fuel economy. Within each gear, the transmission can be shifted from the power mode to the engine mode or engine/charge mode when the demand for power is reduced and to the regenerative mode when the brake is applied, and vice versa. For example, a typical performance schedule is $\mathrm{M} \rightarrow \mathrm{P1} \rightarrow \mathrm{P2} \rightarrow \mathrm{P3}$ or $\mathrm{E3} \rightarrow \mathrm{CVT}$ or $\mathrm{E4}$, and a typical economy schedule consists of $\mathrm{M} \rightarrow \mathrm{E1} \rightarrow \mathrm{E2} \rightarrow \mathrm{E3} \rightarrow \mathrm{CVT}$ or $\mathrm{E4}$. Obviously, the state of charge of the batteries should be taken consideration at all times, as well as the need to power electrical accessories. A typical engine charge schedule is $\mathrm{E1C} \rightarrow \mathrm{E2C} \rightarrow \mathrm{E3C} \rightarrow \mathrm{C}VT$.

7 SUMMARY

The hybrid transmission mechanism presented in this paper can be implemented not only for front-wheel-drive but also for rear-wheel-drive vehicles. For rear-wheel-drive vehicles, the output shaft of the transmission mechanism is connected in-line to the final drive/differential unit to drive the two rear wheels. For front-wheel drive vehicles, the output shaft is connected to either a coaxial or offset final drive/differential unit to drive the two front wheels.

Five major modes of operation that can be classified further into sixteen operating conditions are shown to be feasible. In the electric motor mode, the electric motor alone drives the vehicle at low speeds or in city traffic. In the power mode, both the electric motor and engine drive the vehicle simultaneously to achieve maximum performance, and there are three such modes. In the engine mode, the engine alone drives the vehicle with two *reductions*, one *direct drive*, and one *overdrive* similar to that of a conventional automatic transmission. In engine/charge mode, the engine powers the vehicle and charges the batteries for powering electrical accessories. In the CVT charge mode, the electric motor functions as a generator to regulate the speed of the engine and, at the same time, to charge the batteries. The CVT mode allows the engine to operate at an optimal efficiency point. In the regenerative braking mode, kinematic energy of the vehicle is used for charging the batteries. Four regenerative braking modes, some with engine assisted braking, are possible. Finally, a notional control strategy is developed.

The main advantages of using this transmission include:

- Operational flexibility. One motor mode, three power modes, four engine modes, four engine/charge modes, including one CVT mode, and four regenerative braking modes are available.
- Simplicity. One electric motor, a compound planetary gear set, and four clutches are used.
- Reliability. Use of commercially available components.

- Compactness. Electric motor is integrated with the transmission mechanism.
- Failsafe. Engine or motor alone can drive the vehicle.
- Automatic transmission quality: Two reductions, one direct drive, and one overdrive are available.
- Dual applications: Front- or rear-wheel-drive vehicles.

Finally, we note that a one-way clutch can be added in parallel to the B1 clutch to prevent link 4 from spinning backward. With the one-way clutch holding link 4 stationary, the B1 clutch does not need to be applied for the M, P1, P2, E1, and E2 modes. This will improve the quality of shift from E1 or E2 to E3 mode and from P1 or P2 to P3 mode. Hence, the B1 clutch is engaged only for the R0, R1, and R2 regenerative braking modes. Similarly, other one-way clutches can be employed to improve shift quality.

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NOMENCLATURE

Bi : band clutch i

Ci : multi-disk clutch i

N_{ij} : T_j/T_i where T_j and T_i denote the number of teeth on gears j and i, respectively

P: Power flow

τ_i : torque exerted on link i

ω_i : rotational speed of link i

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