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On the Design of Novel Parallel Hybrid Transmissions Based on a Seven-Link Ravigneaux Gear Train

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Keywords hybrid transmission, Ravigneaux gear train, automatic transmission, operation mode.

ABSTRACT

This paper presents four novel parallel hybrid transmissions based on the structures of three and four-speed automatic transmissions using seven-link Ravigneaux gear train. The proposed designs have some advantages while using conventional automatic transmission structures such as simple mechanical layout, compact design, and flexible operation. They are generally able to perform five primary operation modes; electric motor mode, power mode, power split mode, engine mode, and regenerative braking mode. By using an electronic power control unit, the switching between operation modes is controlled based on vehicle operating conditions. They can either operate as parallel hybrid transmissions or conventional three or four-speed automatic transmissions in the engine mode. One of the new hybrid transmissions is presented as a numerical example to illustrate the working principle, kinematics, static torque and power flow of each operation mode. Furthermore, a feasible clutching logic for control strategy is developed. And, the results show that the proposed design is feasible.

INTRODUCTION

A typical structure of parallel hybrid transmission system consists of two power sources, an electric motor/generator (*M/G*) and an internal combustion engine (*IE*), and a hybrid transmission, as shown in Fig. 1. The hybrid transmission is a

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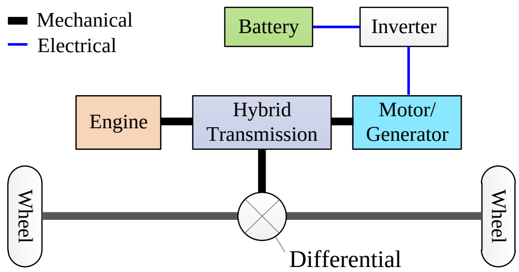


Fig. 1. A typical parallel hybrid system

non-conventional transmission mechanism that allows the *IE* and the *M/G* to either or both connect to the final drive to provide power to drive the vehicle. The *M/G* can be switched between motor (*M*) mode and generator (*G*) mode depending on vehicle operating conditions. When the *M/G* operates as a motor, its uses the energy stored in the batteries to drive the vehicle. And in reverse, it is operated as a generator taking power from the *IE* or the vehicle to charge the batteries. During braking process, it is turned into a generator mode to convert the mechanical energy from the vehicle to charge the batteries for later use. By using an electronic control unit system, the *IE* is generally regulated to operate at its optimized working conditions, thus reduce fuel consumption.

Along with series-parallel hybrid systems that consist of two *M/Gs* and one *IE* in the transmission systems, numerous configurations of parallel system have been developed [1-7]. Tsai and his co-authors proposed several novel configurations of parallel hybrid transmission for passenger vehicles in the 2000s [1-3]. The proposed designs adapted existing four-speed Simpson gear train automatic transmissions to develop the corresponding parallel hybrid transmissions with four automatically controlled clutches to regulate five primary operation modes. Thirteen sub-modes of these systems are available. This kind of configurations can be found in some other research [4, 6, 7]. To avoid using rotating clutches for simpler structure and control, Esmail used only brakes and one-way clutches (OWC) in his proposed parallel hybrid transmissions [5]. And, three commonly used transmission gear sets are adapted, namely Simpson, Ravigneaux, and Type-6206 gear sets.

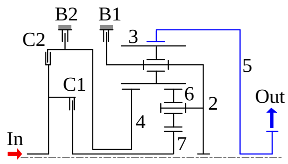
Various planetary gear trains (PGTs) can be developed as three and four speed automatic transmissions [8]. Generally, they are seven or nine-link PGTs that consist of two degrees of freedom (2-DoF). Since the *IE* is the only power source in an automatic transmission, one of the links in the gear train must be clutched to the gear box or to another link making the mechanism become 1-DoF device. Hence, numerous parallel hybrid transmissions with two power sources (*IE* and *M/G*) can be developed from such automatic transmission mechanisms [9]. However, in the most developed designs, the *IE* alone cannot drive the vehicle in the backward direction as in an automatic transmission vehicle.

This paper proposes four configurations of novel parallel hybrid transmissions that can either operate as automatic transmissions or a hybrid type base on the mechanical structures of the existing Ravigneaux-type automatic transmissions. The systems can perform five primary operation modes that can be further sub-divided into sub-modes. In addition, the *IE* can also be able to drive the vehicle in reverse direction leading to more safety operation. In what follows, we firstly introduce the existing Ravigneaux-type automatic transmissions. Then, kinematics and power flows of the seven-link Ravigneaux gear train are analyzed. Afterward, the mechanical structures and the primary operation modes of the systems are presented. Finally, one of the new systems is selected arbitrarily as a numerical example to demonstrate the feasibility of the proposed designs.

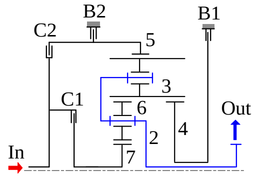
RAVIGNEAUX-TYPE AUTOMATIC TRANSMISSIONS

Figures 2(a) and (b) show two three-speed automatic transmissions [8]. Both of them consist of a seven-link Ravigneaux gear train, two clutches (C1 & C2), and two brakes (B1 and B2). The Ravigneaux gear train consists of gear box 1, carrier 2, planet gear 3, sun gear 4, ring gear 5, planet gear 6, and sun gear 7. Generally, sun gear 7 is smaller than sun gear 4. The ring gear and the carrier are selected as the output links as shown in Figs. 2(a) and (b), respectively. These two concepts can provide three forward speeds and one reverse speed.

Figures 2(c) and (d) show the tables of corresponding clutching sequence of the transmissions shown in Figs. 2(a) and (b), respectively, where an “x” presents the corresponding clutch or brake is engaged. The brake can be replaced by a multi-disc in order to smooth the clutch shifting [3]. By regulating the engagement of the two clutches and two brakes, the vehicle can achieve three forward speeds and one reverse speed. Speed ratio of the tran-



(a)



(b)

Range	Clutches Engaged				Comment
	C1	C2	B1	B2	
First	x		x		UD
Second	x			x	UD
Third	x	x			DD
Reverse		x	x		RD

(c)

Range	Clutches Engaged				Comment
	C1	C2	B1	B2	
First	x		x		UD
Second		x	x		UD
Third	x	x			DD
Reverse	x			x	RD

(d)

Fig. 2. Thee-speed automatic transmissions

-smission is defined as the ratio of the angular velocity of the input link to the angular velocity of the output link. The speed ratios greater than one at the first and second speeds are called under drive (UD). It is equal to the one at the third speed called direct drive (DD). And, it is a negative number at the reverse speed called reverse drive (RD). Some transmissions also have the speed ratio less than one called over drive (OD). In addition, the gear train mechanism in an automatic transmission always has one degree of freedom (1-DoF).

KINEMATICS AND POWER FLOWS OF RAVIGNEAUX GEAR TRAIN

For kinematic and power flow analyses of planetary gear trains (PGTs), numerous method and approach have been presented [10-14]. A compound PGT can be analyzed by using the method of fundamental gear entities (FGE) proposed by Hsie and Tsai [14]. If a compound PGT consists of *m* FGEs, then *m* kinematic equations can be obtained.

Kinematic Analysis

For a simple PGT with only one FGE, denoted as ring gear - *R*, planet gear - *P*, carrier - *C* and sun gear - *S*), the kinematic equation is formulated as

$$\omega_S - K_i\omega_R + (K_i - 1)\omega_C = 0 \qquad (1)$$

where, ω_S, ω_R, ω_C are the angular velocities of the sun gear, ring gear, and carrier of the simple PGT, respectively. K_i denotes the basic gear ratio of a simple PGT and is defined as $K_i = (-1)^n N_R/N_S$, where N_R and N_S are the number of teeth of the ring gear and sun gear, respectively, and n is the number of the planet gears between the ring gear and the sun gear.

For the Revigneaux gear train with seven-link shown in Fig. 2(a) or (b), there are three FGEs of ring gear, planet gear, carrier, and sun gear, (5, 3, 2, 4), (5, 3-6, 2, 7), and (4, 3-6, 2, 7). However, only two of the kinematic equations of the FGEs are independent, such as

$$\omega_4 - K_1\omega_5 + (K_1 - 1)\omega_2 = 0 \tag{2}$$

$$\omega_7 - K_2\omega_5 + (K_2 - 1)\omega_2 = 0 \tag{3}$$

Eliminating ω_5 from Eq. (2) by using Eq. (3) yields

$$\omega_7 - K_3\omega_4 + (K_3 - 1)\omega_2 = 0 \tag{4}$$

where $K_1 = -N_5/N_4$, $K_2 = N_5/N_7$, and $K_3 = -N_4/N_7$. Referring to the geometry of the Ravigneaux gear train, it is shown that $K_1 < -1$, $K_2 > 1$, and $K_3 < -1$. In the automatic transmissions, the ring gear 5 and the carrier 2 are selected as the output links to obtain the sets of desired speed ratios for UD, DD, (OD), and RD. An automatic transmission consists only one input power, thus the mechanism is with 1-DoF. Then one of the three coaxial links that is not the output link must be fixed or two of them are engaged together. Let ω_i, ω_o, and ω_g are the angular speeds of input link, output link, and the reference link (ground link), respectively, then the speed ratio is defined as the ratio of the angular velocity of the input link to the angular velocity of the output link as

$$R^g_{i,o} = \frac{\omega_i - \omega_g}{\omega_o - \omega_g} \tag{5}$$

The speed ratios of the gear train with input link, output link, and grounded link assigned are listed in Table 1, and the speed ratio ranges are classified into three groups, $R^g_{i,o} > 1$ (UD), $0 < R^g_{i,o} < 1$ (OD), $R^g_{i,o} < 0$ (RD).

Power Flow Analysis

For the power flow analysis of planetary gear train, various methods have been proposed [2, 10-12]. With the Ravigneaux gear train as a system, under steady-state operation and neglecting frictional losses, torques applied on links 2, 4, 5, and 7 about the central axis of the gear train must sum to zero; that is

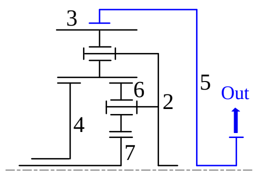
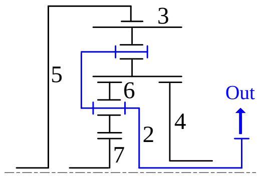
$$T_2 + T_4 + T_5 + T_7 = 0 \tag{6}$$

Where T_i denotes the external torque applied on member i, positive in the counter-clockwise direction.

Similarly, power flows into the gear train via four coaxial links must sum to zero; that is,

$$T_2\omega_2 + T_4\omega_4 + T_5\omega_5 + T_7\omega_7 = 0 \tag{7}$$

Table 1. Speed ratio equations with the output assigned

	
Speed ratio	Speed ratio
$R^2_{4,5} = K_1 < 0$	$1 < R^7_{4,2} = \frac{K_3 - 1}{K_3}$
$1 < R^7_{4,5} = \frac{K_1(K_3 - 1)}{K_2 - 1}$	$1 < R^5_{4,2} = 1 - K_1$
$1 < R^2_{7,5} = K_2$	$1 < R^4_{7,2} = 1 - K_3$
$1 < R^4_{7,5} = \frac{K_2(1 - K_3)}{K_2 - K_3}$	$R^5_{7,2} = 1 - K_2 < 0$
$0 < R^4_{2,5} = \frac{K_1}{K_1 - 1} < 1$	$1 < R^4_{5,2} = \frac{K_1 - 1}{K_1}$
$1 < R^7_{2,5} = \frac{K_2}{K_2 - 1}$	$0 < R^7_{5,2} = \frac{K_2 - 1}{K_2} < 1$

Eliminating T_5 from Eqs. (6) and (7), and making use of Eqs. (2) and (3), yields

$$T_2 = -R^5_{4,2}T_4 - R^5_{7,2}T_7 \tag{8}$$

Eliminating T_2 from Eqs. (6) and (7), and making use of Eqs. (2) and (3), yields

$$T_5 = (R^5_{4,2} - 1)T_4 + (R^5_{7,2} - 1)T_7 \tag{9}$$

Based on Eqs. (8) and (9), if two of the four torques exerted on two links are known, then the other two can be calculated in terms of the velocity ratios.

PROPOSED PARALLEL HYBRID TRANSMISSIONS

Based on the structures of the existing automatic transmissions using seven-link Ravigneaux gear train, four parallel hybrid transmissions are proposed as shown in Fig. 3.

Kinematic Structures

Figures 3(a) and (b) show two novel parallel hybrid transmissions based on the modification of existing three-speed and Borg-Warner four-speed aut-

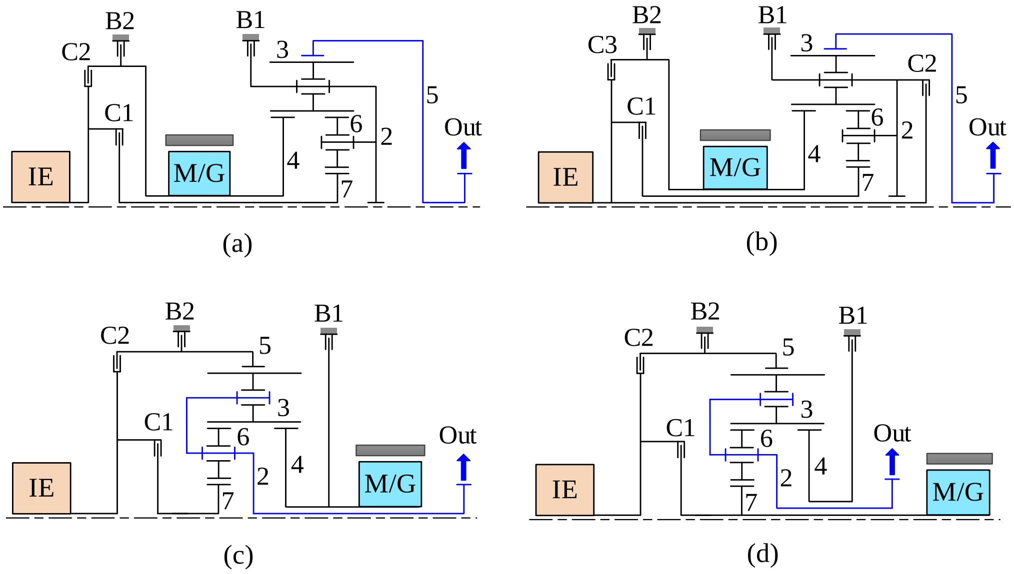


Fig. 3. Proposed parallel hybrid transmissions

-tomatic transmissions [8]. Ring gear 5 is selected as the output link and the connections of the *IE* to the transmissions are the same as in their original automatic transmissions. The *M/G* is coupled to sun gear 4. Figs. 3(c) and (d) show two other novel parallel hybrid transmissions with the carriers as the output links. The arrangement of the *IE*, clutches and brakes are the same as in the original corresponding automatic transmission. There are two positions that the *M/G* be coupled, sun gear 4 and sun gear 7 as shown in Figs. 3(c) and (d), respectively. Different *M/G* positions lead to different of working principle in the two systems. The advantage of the structures is that we can modify few elements in the existing automatic transmissions to obtain the desired hybrid transmissions.

Operation Modes

The proposed hybrid transmissions can generally perform five primary operation modes that can be further subdivided into sub-modes, including electric motor mode, power mode, power split mode, engine mode, and regenerative braking mode as follows:

1. *Motor alone mode*: The *M/G* operating as a motor using energy from the batteries to drive the vehicle in forward and backward direction. It operates as an electric vehicle (EV). The *IE* is standstill in this mode.
2. *Power mode*: The *IE* is turned on when the vehicle demands high power for hard acceleration or hill climbing. The hybrid transmission allows the *IE* and the *M/G* both provide power to drive the vehicle.
3. *Power split mode*: When the power provided by the *IE* is greater than that of vehicle requires, the *M/G* is turned into the generator mode to convert the excessive to store into the batteries.
4. *Engine mode*: When the power demand is moderately high or near the *IE* optimized working conditions, the transmission can operate as an automatic transmission with the *IE* as the only input power to drive the output shaft.
5. *Regenerative braking mode*: The *M/G* is turned into a generator mode to convert energy from the vehicle's braking process to charge the batteries.

In addition, the stationary charging mode is a selective mode. When the vehicle is standstill and the batteries are insufficient, the *IE* can drive the *M/G* operated as a generator to charge the batteries. These five operation modes are fundamental for any of the parallel hybrid transmission [15].

NUMERICAL EXAMPLE OF THE PARALLEL HYBRID TRANSMISSION

To demonstrate the feasibility of the proposed parallel hybrid transmissions, the system shown in Fig. 3(c) is selected arbitrarily as a numerical example for kinematic and power flow analyses.

Let the number of teeth on the gear of the transmission mechanism be N_3 = 24, N_4 = 36, N_5 = 84, N_6 = 20, and N_7 = 30, then K_1 = -2.3333, K_2 = 2.8, and K_3 = -1.2 Table 2 shows some numerical values of the velocity ratios of the gear train. There are fourteen possible clutching conditions that can be grouped in five operation modes as listed in Table 3, where "x" denotes the corresponding clutch or brake engaged. The visual kinematic and power flow of the system in some operation modes are shown in Fig. 4.

Table 2 Numerical values of the velocity ratios

$R^{7}_{4,2} = 1.8333$	$R^{4}_{7,2} = 2.2$	$R^{4}_{5,2} = 1.4286$
$R^{5}_{4,2} = 3.3333$	$R^{5}_{7,2} = -1.8$	$R^{7}_{5,2} = 0.6428$

Table 3 Fourteen possible clutching conditions of the new parallel hybrid transmission

Operating Mode	Clutches Engaged				Remark
	C1	C2	B1	B2	
1. Electric motor mode				x	*M*
2. Power mode 1 (CVT1)	x				*M/G*
3. Power mode 2 (CVT2)		x			*M/G*
4. Power mode 3 (DD)	x	x			*M*
5. Power mode 4 (RD)	x			x	*M*
6. Power split mode	x	x			*G*
7. Eng. mode 1 (UD)	x		x		*M* stop
8. Eng. mode 2 (UD)		x	x		*M* stop
9. Eng. mode 3 (DD)	x	x			*M* free
10. Eng. mode 4 (RD)	x			x	*M* free
11. Reg. braking mode 1				x	*G*
12. Reg. braking mode 2	x				*G*
13. Reg. braking mode 3		x			*G*
14. Reg. braking mode 4	x	x			*G*

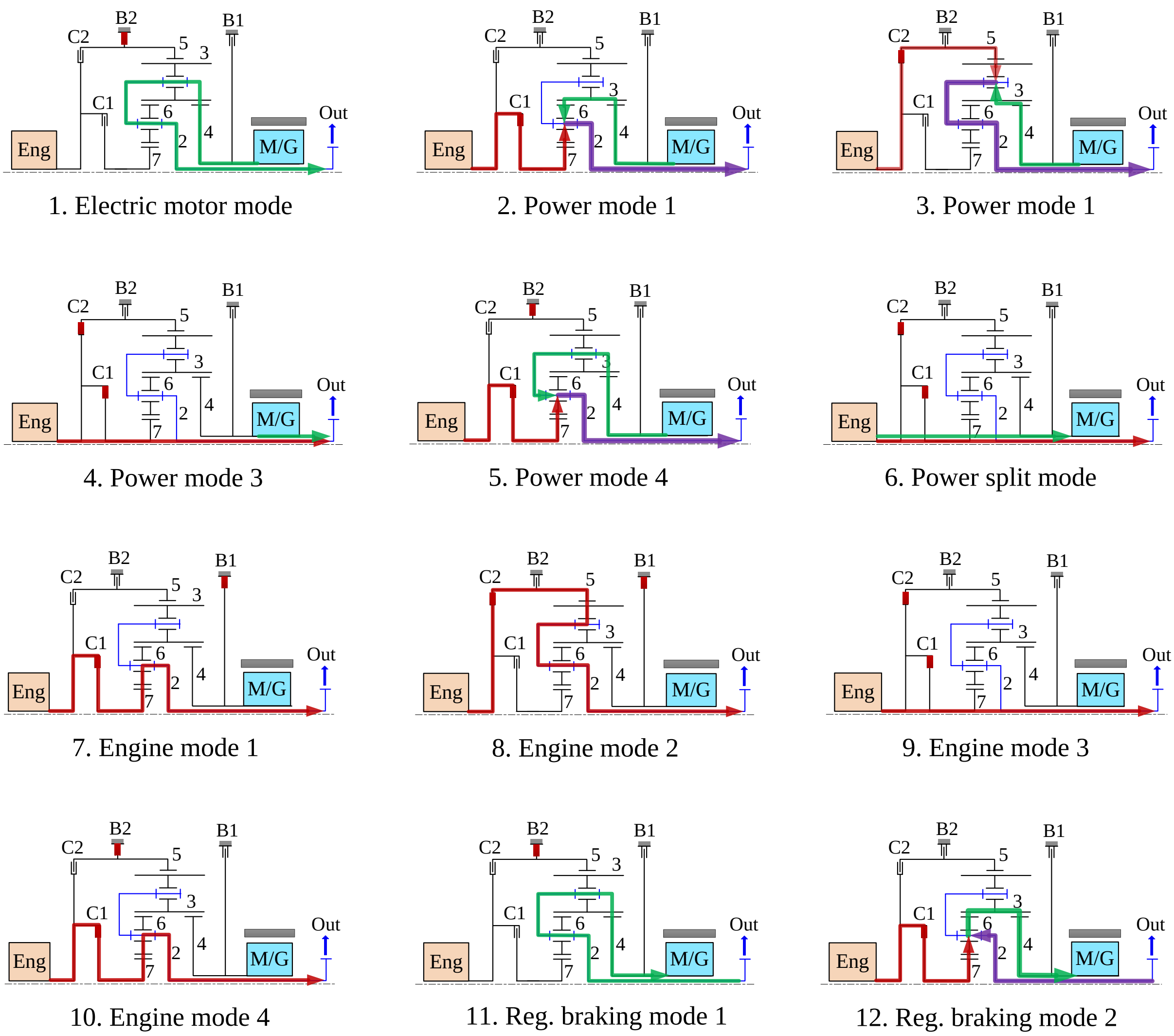


Fig. 4. Power flow of the new parallel hybrid transmission

Electric Motor Mode

When the vehicle starts launching or driving at low speed in city traffic, the *M/G* is the only power source using energy from the batteries to drive the vehicle in the forward and backward directions. The mechanism is with 1-DoF, and only brake B2 is engaged to fix ring gear 5, as shown in Fig. 4(1). And, the *IE* is still off due to its poor working efficiency at the low speed conditions.

Brake B2 is engaged, hence $\omega_5 = 0$. Substituting $\omega_5 = 0$ and the value of K1 = -2.3333 into Eq. (2) yields

$$\omega_{M/G} = 3.3333\omega_2 \tag{10}$$

Thus, the speed ratio of the *M/G* and the output link is 3.3333 and they rotate in the same direction.

Since sun gear 7 is free rotating, $T_7 = 0$. Substituting $T_7 = 0$ into Eqs. (8) and (9) yields

$$T_2 = -3.3333T_{M/G} \tag{11}$$

$$T_5 = 2.3333T_{M/G} \tag{12}$$

Then, neglecting all losses, the output power is given by

$$P_{output} = T_2\omega_2 = -3.3333T_{M/G}\omega_2 \tag{13}$$

The negative value shows that the *M/G* is providing power to drive the vehicle.

Power Modes

When the vehicle needs high power for hard acceleration or hill climbing, the *IE* is turned on to provide extra power with the *M/G* to the output shaft. Four power modes are available.

Power Mode 1 (CVT1)

In this mode, brake B2 is disengaged to release ring gear 5 for free rotating, and clutch C1 is engaged to connect the *IE* shaft with the sun gear 7. Two input powers are coupled to the transmission at the same time, and the mechanism is with 2-DoF, as shown in Fig. 4(2). Substituting the value of K_3 = -1.2 into Eq. (4) yields

$$\omega_2 = 0.4545\omega_{IE} + 0.5455\omega_{M/G} \tag{14}$$

Eq. (14) shows that the *IE* speed can be any value at any vehicle speed by regulating the *M/G* speed. Thus, the *IE* is generally regulated to operate at its optimized working conditions for the highest efficiency.

Since ring gear 5 is free rotating, $T_5 = 0$. Substituting $T_5 = 0$ and the values of $R^5_{4,2} = 3.3333$ and $R^5_{7,2} = -1.8$ into Eqs. (9), then (8) yields

$$T_{M/G} = 1.2T_{IE} \tag{15}$$

$$T_2 = -2.2T_{IE} \tag{16}$$

The output power is given by

$$P_{output} = T_2\omega_2 = -2.2T_{IE}\omega_2 \tag{17}$$

Power Mode 2 (CVT2)

This mode is similar to *Power Mode 2*, brake B2 and clutch C1 are disengaged and clutch C2 is engaged to connect the *IE* shaft with ring gear 5. Sun gear 7 is free rotating. Two input powers are coupled to the transmission, and the mechanism is with 2-DoF, as shown in Fig. 4(3). Substituting the value of K_1 = -2.3333 into Eq. (2) yields

$$\omega_2 = 0.7\omega_{IE} + 0.3\omega_{M/G} \tag{18}$$

Since sun gear 7 is free rotating, $T_7 = 0$. Substituting $T_7 = 0$ and the values of $R^5_{4,2} = 3.3333$ and $R^5_{7,2} = -1.8$ into Eqs. (9) and (8) yields

$$T_{M/G} = 0.4286T_{IE} \tag{19}$$

$$T_2 = -1.4286T_{IE} \tag{20}$$

Eqs. (16) and (20) shows that with the same torque value of the IE, the output torque is greater in *Power Mode 1*. And, the output power is given by

$$P_{output} = T_2\omega_2 = -1.4286T_{IE}\omega_2 \tag{21}$$

Power Mode 3 (DD)

When the vehicle is at high speed, clutches C1 and C2 are both engaged to lock up the gear train as a rigid body. The transmission is under DD condition, as shown in Fig. 4(4). The speed and torque relationship of the transmission are

$$\omega_2 = \omega_{IE} = \omega_{M/G} \tag{22}$$

$$T_2 = -T_{IE} - T_{M/G} \tag{23}$$

The output power is given by

$$P_{output} = T_2\omega_2 = -(T_{IE} + T_{M/G})\omega_2 \tag{24}$$

Power Mode 4 (RD)

In this mode, brake B2 is engaged to fix ring gear 5, and clutch C1 is engaged to connect the *IE* shaft with the sun gear 7. The *IE* and the *M/G* both provide power to drive the vehicle in the backward direction, as shown in Fig. 4(5).

Brake B2 is engaged, hence $\omega_5 = 0$. Substituting $\omega_5 = 0$ and the value of $K_2 = 2.8$ into Eq. (3) yields

$$\omega_{IE} = -1.8\omega_2 \tag{25}$$

$$\omega_{M/G} = -1.85\omega_{IE} \tag{26}$$

Eq. (26) shows that the *M/G* in this mode is rotating in the opposite direction with the *IE* for the vehicle reverse speed. Substituting the values of $R^5_{4,2} = 3.3333$ and $R^5_{7,2} = -1.8$ into Eq. (8) yields

$$T_2 = -3.3333T_{M/G} + 1.8T_{IE} \tag{27}$$

The output power is given by

$$P_{output} = T_2\omega_2 = (-3.3333T_{M/G} + 1.8T_{IE})\omega_2 \tag{28}$$

Power Split Mode

This mode is similar to *Power Mode 3* with both clutches C1 and C2 engaged to lock up the gear train as a rigid body. The power provided by the *IE* is greater than that of vehicle requires. Thus, the *M/G* is turned into a generator mode converting part of the engine power to charge the batteries, as shown in Fig. 4(6). The speed and torque relationship of the transmission are

$$\omega_2 = \omega_{IE} = \omega_{M/G} \tag{29}$$

$$T_2 = -T_{IE} - T_G \tag{30}$$

The output power is given by

$$P_{output} = T_2\omega_2 = (-T_{IE} - T_G)\omega_2 \tag{31}$$

Engine Modes

When the *IE* drives the vehicle alone, the transmission can be regulated as a three-speed automatic transmission with the M/G is free rotating or fixed.

Engine Mode 1 (UD)

In this mode, clutch C1 is engaged to connect the *IE* shaft to sun gear 7, and brake B1 is engaged to fix the *M/G* shaft, as shown in Fig. 4(7). The speed and torque relationship of the transmission are

$$\omega_{IE} = 2.2\omega_2 \tag{32}$$

$$T_2 = -2.2T_{IE} \tag{33}$$

The output power is given by

$$P_{output} = -2.2T_{IE}\omega_2 \tag{34}$$

Engine Mode 2 (UD)

In this mode, clutch C2 is engaged to connect the *IE* shaft to ring gear 5, and brake B1 is engaged to fix the *M/G* shaft, as shown in Fig. 4(8). The speed and torque relationship of the transmission are

$$\omega_{IE} = 1.4286\omega_2 \tag{35}$$

$$T_2 = -1.4286T_{IE} \tag{36}$$

The output power is given by

$$P_{output} = -1.4286T_{IE}\omega_2 \tag{37}$$

Engine Mode 3 (DD)

In this mode, both clutches C1 and C2 are engaged to lock up the gear train as a rigid body. The *M/G* is set to be free rotating. The transmission is under DD condition for vehicle high speed operation, as shown in Fig. 4(9). The speed and torque relationship of the transmission are

$$\omega_{IE} = \omega_2 \tag{38}$$

$$T_2 = -T_{IE} \tag{39}$$

The output power is given by

$$P_{output} = -T_{IE}\omega_2 \tag{40}$$

Engine Mode 4 (RD)

This mode is used to drive the vehicle in the backward direction. Brake B2 is engaged to fix ring gear 5, and clutch C1 is engaged to connect the *IE* shaft with the sun gear 7. The M/G is free rotating, as shown in Fig. 4(10). The speed and torque relationship of the transmission are

$$\omega_{IE} = -1.8\omega_2 \tag{41}$$

$$T_2 = 1.8T_{IE} \tag{42}$$

The output power is given by

$$P_{output} = 1.8T_{IE}\omega_2 \tag{43}$$

Regenerative Braking Modes

One of the advantages of a hybrid vehicle when compared with a conventional one is that the hybrid transmission allows the *M/G* to operate as the generator to convert kinematic energy into electric for charging the batteries during braking process. Four regenerative braking modes corresponding with four operating modes are available.

Regenerative Braking Mode 1

This mode is the corresponding mode of the *Electric Motor Mode* with only brake B1 engaged. Thus, the relationships of kinematic and power flower follow that of *Electric Motor Mode*, but in the reverse direction, as shown in Fig. 4(11).

Regenerative Braking Mode 2

This is the corresponding mode of the *Power Mode 1* with only clutch C1 engaged. Thus, the relationships of kinematic and power flower follow that of *Power Mode 1*, but in the reverse direction, as shown in Fig. 4(12). Both the *IE* and the *M/G* provide braking efforts to stop the vehicle.

Regenerative Braking Mode 3

This is the corresponding mode of the *Power Mode 2* with only clutch C2 engaged. Thus, the relationships of kinematic and power flower follow that of *Power Mode 2*, but in the reverse direction. Both the *IE* and the *M/G* provide braking efforts to stop the vehicle. With the same value of the output torque, the torque applied on the M/G shaft is smaller than that of in *Regenerative Braking Mode 2*.

Regenerative Braking Mode 4

With clutches C1 and C2 engaged, the transmission is locked up as a rigid body. This is the corresponding mode of the *Power Mode* 3 for vehicle high speed braking. Thus, the relationships of kinematic and power flow follow that of *Power Mode 3*, but in the reverse direction. Both the *IE* and the

M/G provide braking efforts to stop the vehicle.

FEASIBLE CLTUCHING LOGIC

As shown in the previous section, fourteen clutching conditions that can be grouped into five primary modes are available in the presented systems. However, the system is not required to carry out all of them. Based on the system working conditions and driving options, the most suitable mode is selected. For the system control strategy, there is a need to develop a shifting strategy between the operation modes. In what follows, the shifting preference is developed in a matrix form. Let the value (i, j) of the matrix be assigned the number of change of state. Here, a change of state is defined as the change of a clutch or brake engagement status from "engaged" to "disengaged" or the change of the *M/G* status from "*M* mode" to "*G* mode" or vice versa. Note that the change of the *M/G* state from "*M* stop" to "*M* free" and vice versa is not considered as a change of state. And, if more than two changes of state are required, no value is assigned. The transition from Power split mode (Ps) to the Regenerative braking mode 4 (R4) needs no change of state, thus the value of 0 is assigned.

In this regard, the value of 1 is the most feasible, 2 is a feasible, and no value is an impractical shift. Analysis of fourteen clutching conditions listed in Table 3 yields the corresponding clutching logic listed in Table 4. For instance, the value of (i = 1, j = 2) equals 2 since there are two changes of state of clutch C1 and brake B2 when the system switches from Electric motor mode to Power mode 1. It can be noted that the value of 2 in the Table 4 is equivalent to a single clutch to clutch shift in an automatic transmission.

Based on the feasible clutching logic, an alternative classification of the operation modes for control strategy can be developed for further study.

CONCLUSIONS

Four novel parallel hybrid transmissions have been proposed in this work based on the adaption of the three and four-speed automatic transmissions. These hybrid transmissions can generally perform five primary operation modes, including electric motor mode, power mode, power split mode, engine mode, and regenerative braking mode. A numerical is presented to show the working principle and demonstrate the feasibility of the proposed designs. It can perform fourteen clutching conditions that can be grouped into five primary modes. Furthermore, a feasible clutching logic for control strategy is developed.

Since adapting the conventional automatic transmission structures and using only one *M/G*, some potential advantages of the proposed parallel hybrid transmissions are as follows:

- Compact and simplicity. Ravigneaux gear train is widely used in three and four-speed automatic transmissions. The proposed designs do not need a torque converter, and only one *M/G*, two clutches and two brakes are used.
- Flexibility in operation. Five primary operation modes that can be further sub-divided into sub-modes are available in the proposed designs provide a number of control options in a single hardware. Continuously variable transmission (CVT) is also available in a power mode or a power split mode.
- Three and four-speed automatic transmission capability. The parallel hybrid transmissions are able to operate as hybrids or automatic transmission systems to further improve the overall efficiency of the vehicles. In the engine mode, the *IE* alone can drive the vehicle with

Table 4 Feasible clutching logic

		1	2	3	4	5	6	7	8	9	10	11	12	13	14
		M	P1	P2	P3	P4	Ps	E1	E2	E3	E4	R1	R2	R3	R4
1	M	*	2	2		1					2	1			
2	P1	2	*	2	1	1	1	2		2	2		1		2
3	P2	2	2	*	1		2		2	2				1	2
4	P3		1	1	*	2	1			1			2	2	1
5	P4	1	1		2	*					1	2	2		
6	Ps		1	2	1		*			1			1	1	0
7	E1		2					*	2	2	2		2		
8	E2			2				2	*	2				2	
9	E3		2	2	1		1	2	2	*	2		2	2	1
10	E4	2	2			1		2		2	*	2	2		
11	R1	1				2					2	*	2	2	
12	R2		1		2	2	1	2		2	2	2	*	2	1
13	R3			1	2		1		2	2		2	2	*	1
14	R4		2	2	1		0			1			1	1	*

two under drive (UD), one direct drive, (one over drive), and one reverse drive similar to the corresponding automatic transmissions.

- Operational safety. Either the *IE* or *M/G* alone can drive the vehicle in the forward and backward directions, thus enhances the safety of the system.

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創新型並聯式混合傳動統使用七桿拉維娜式行星齒輪之設計

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摘 要

本文基於七桿拉維娜（Ravigneaux）行星齒輪三速與四速自動變速箱的架構，提出並聯式混合傳動系統之新設計。由於採用傳統自動變速箱的架構，此新設計有簡單的機械架構、設計緊湊、操作靈活等優點。所提出的設計能夠執行為馬達驅動模式、動力模式、動力分流模式、引擎模式及制動模式五個基本的操作模式。通過使用一個電子功率控制單元， 基於車輛運作條件控制操作模式之間的切換。此外，此設計可以作為並聯式混合動力傳動裝置或在引擎操作模式中常規自動變速器操作。為了證明本研究所提出的新設計之可行性，一種新型混合動力傳動機構被任意挑選來詳細說明每個操作模式之工作原理、運動分析、力矩與功流分析。此外，為控制策略發展一種可行的離合邏輯。結果表明所提出的設計是可行的。