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On the Design of a Novel Series-parallel Hybrid Transmission with a Compound Planetary Gear Train

Huu-Tich Ngo**, Kuen-Bao Sheu* and Hong-Sen Yan**

Keywords：hybrid electric vehicle, hybrid transmission, series-parallel hybrid, transmission efficiency.

ABSTRACT

This paper presents a systematic process to design a novel series-parallel hybrid transmission. Firstly, the general characteristics of a desired hybrid transmission system are concluded based on a study of numerous existing designs of hybrid transmission systems and the characteristics of provided input/output powers. Then, a systematic design procedure, including four main steps, is presented to synthesize a novel hybrid transmission system. The compound planetary gear trains (PGTs) are used to synthesize clutchless hybrid transmission by assigning input/output powers in the second step. Thirdly, clutches are added to control the operation modes of the synthesized systems. And, system parameters are addressed in the fourth step to get the most suitable series-parallel hybrid transmission that satisfies the design specifications. Finally, the designed system is analyzed for transmission efficiencies, kinematics and power flows in each operation mode, and vehicle performances. It is shown that the designed system satisfies all the design requirements and specifications.

INTRODUCTION

Figure 1 shows a typical series-parallel hybrid transmission system. It includes three power sources, an internal combustion engine (IE) and two electric motor/generators, coupled to a hybrid transmission, which is a gear train or a planetary gear train. One of the electric motor/generators, normally a bigger one,

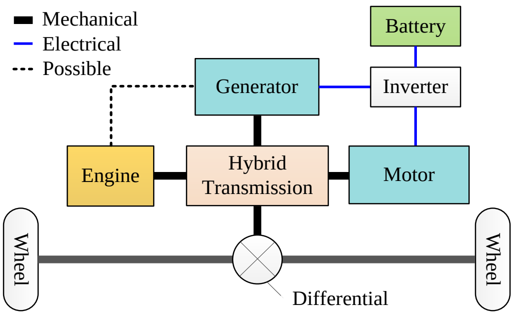

Fig. 1 A typical series-parallel hybrid system

driving the vehicle at low speed as in city traffic condition, is named as electric motor (M). The other one is named as generator (G) since it mostly worked as a generator. In some systems, the generator is also used for starting the IE, so it is called integrated starter generator (ISG). The series-parallel system can be classified into two types, power-split hybrid electric vehicle (HEV) and multi-mode HEV, depending on the used hybrid transmission (Zhuang W. et al. 2020). In this configuration system, the IE and the electric motor /generator (M/G) are allowed either or all to provide power to drive the vehicle. The motor and generator are switched between the motor mode and the generator mode to drive the vehicle or to charge the batteries, respectively, during vehicle operation. In addition, the transmission can split the IE power, part for driving the vehicle and part for driving the generator (or motor in the generator mode) to charge the batteries. The IE is mostly operated at the highest efficiency condition and the vehicle speed is controlled by one of the two electric motors, thus it is also referred as an electronic continuously variable transmission (E-CVT).

Numerous configurations of series-parallel hybrid transmissions were developed in the past few decades (Liu et al. 2008, Zhang et al. 2016, Ngo and Yan 2016, Kim et al. 2016, Pei et al. 2018, Holmes and Schmidt 2001, Chung et al. 2020). Several series-parallel HEVs have been launched to the market, such as: Toyota Prius and Camry, Chevy Volt, Ford Fusion, etc. Toyota was the most successful company to launch a commercial series-parallel HEV, the Prius, into the world market

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***Department of Mechanical Engineering, National Cheng Kung University, 1 University Road, Tainan 70101, Taiwan, ROC.*

**Department of Vehicle Engineering, National Formosa University, 64 Wunhua Road, Huwei, Yunlin, 63208, Taiwan, ROC.*

in 2000s [9, 10]. Toyota's hybrid system namely Hybrid Synergy Drive (HSD), is installed in many Toyota HEV models (Vasilash et al. 2005, Hata et al. 2006, Wakura et al. 2007, Nagamatsu et al. 2012, Iwanaka et al. 2012). The first generation of the HSD system is used in the Prius, Figure 2(a). A simple planetary gear train (PGT) is used as a split device transmission. The motor is connected directly to the output shaft of the transmission. In the second generation of HSD, the motor is connected to the output shaft by a reduction ratio, another simple PGT with one link being fixed is employed, Figures 2(c) and (d) (Nagamatsu et al. 2012). There is no clutch in these systems, thus the transmissions are simpler than the traditional transmission systems. However, controlling the system becomes much more challenging. And, when the motor drives the vehicle, it also spins the generator or engine shaft leading to the spinning losses. Furthermore, the engine power is always split while driving the vehicle, causing power loss in transmission. Figure 2(b) shows the Chevy Volt's multi-mode series-parallel hybrid system, using a simple PGT with two clutches and a brake to control (Conlon et al. 2011, Duhon et al. 2015). The two clutches allow the engine and generator engaged or disengaged from the transmission, thus the system can be operated as a series HEV. However, the engine can only drive the vehicle in a CVT power mode, in which the motor is running to regulate the engine and the output. In searching for suitable configurations from numerous possible hybrid transmissions, some systematic methodologies were prosed (Gupta et al. 2013, Wang et al. 2014, Payri et al. 2014). Most of the researches try to figure out all possible configurations, including unfeasible ones, and then analyze all of them to search for the feasible systems. Numerous hybrid transmission systems are utilizing various PGTs with different number of clutches and brakes added to control the system operation modes (Liu and Peng 2010). Zhang et al. analyzed many possible hybrid systems and showed that the fuel consumption of a system is reduced by adding clutches to control its operation modes and using a proper designed control algorithm. However, the fuel consumption is not improved if the clutches are added to incorrect positions (Zhang et al. 2012). This paper employed a systematic design approach for synthesizing a novel design of series-parallel hybrid transmission using a compound PGT (Ngo and Yan 2016). The advantages of using this approach is that the concluded specifications and constraints are used in each design step to reduce much calculation load, and no complex analysis processes are required for synthesizing the desired systems.

The following sections of this paper is organized as follows. Section 2 concludes design specifications of the desired hybrid system. Section 3 focuses on the systematic design process, subject to

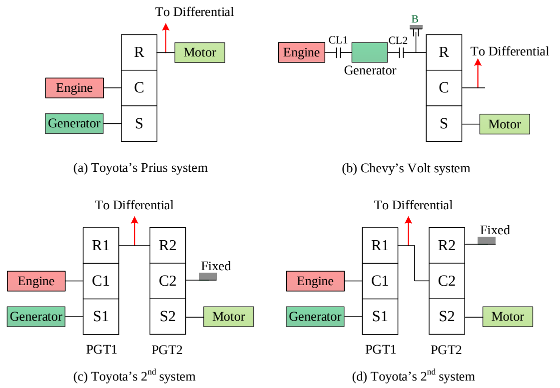


Fig. 2 Existing hybrid transmissions

the concluded design specifications along with the system parameters to synthesize desired clutchless and clutched hybrid transmission systems. Section 4 analyzes the new hybrid transmission system including transmission efficiencies, kinematics and power flows, and drivability of the vehicle. And, conclusions are presented in Section 5.

SPECIFICATIONS

The target of this paper is designing a novel series-parallel hybrid transmission system focused on 2PG system as listed in Table 1. The required performances of the vehicle include vehicle maximum speed up to 220 km/h, and the time for acceleration from 0 to 100 km/h is less than 8 seconds. Based on a study of the characteristics of a general series-parallel hybrid system, numerous existing series-parallel systems and the input powers, the characteristics of the desired series-parallel hybrid transmission system are concluded as follows:

Table 1 Specifications of input powers

Input	Speed (rpm)	Highest efficient speed (rpm)	Max torque (Nm)	Max power (KW)
ISG (G)	0~6,000	3,000	150.2	46
Motor (M)	0~12,000	3,500	297.9	105.9
Engine (E)	0~5,600	3,000	280.4	135

1. The system contains three input powers (an engine, a motor, and an integrated starter generator) and one output to be able to perform as a series-parallel hybrid system. The required specification characteristics of the input powers are listed in Table 1. The engine and the generator have the similar characteristic of the highest efficient speed at 3,000 rpm.
2. The system must be able to operate as a series, a parallel or a series-parallel system. In the series

operation mode, one motor drives the vehicle while the engine drives the generator to generate electricity to charge the batteries.

3. In the parallel operation mode, the engine and the motor can both provide the power to drive the vehicle.
4. In the series-parallel operation mode, the motor and the engine drive the vehicle while part of the engine power drives the generator to charge the batteries or transmit directly to the motor.
5. The system must be able to operate as an e-CVT system. In that case, the engine is controlled to perform at the highest efficiency by the ISG, so that the mechanism used must be a two-degree of freedom (2-DoF) design, such as planetary gear trains.
6. The system must be able to regenerate kinetic power during braking and it also can recharge the batteries at standstill by using the engine to drive the ISG.

CONFIGURATION SYNTHESIS

To design a novel series-parallel hybrid transmission for passenger vehicles with the specified requirements, a systematic design process is utilized, Figure 3 (Ngo and Yan 2016). The process consists of four main steps: planetary gear trains, clutchless hybrid transmissions, clutched hybrid transmissions, and desired hybrid transmission. Each step is achieved by carrying out the corresponding procedure: power arrangement, clutch arrangement, and parameter design. These steps are presented in the following sections.

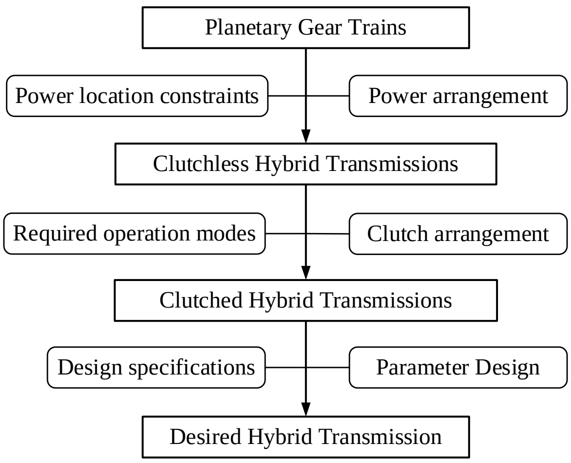


Fig. 3 Procedure of novel hybrid transmission design

Planetary Gear Trains (PGTs)

Simple and compound planetary gear trains are addressed as follow.

Simple PGT

A simple PGT includes a ring gear (R), a sun gear (S), and a carrier (C) that are adjacent to a ground link (frame). It is a two-degree-of-freedom (2-DoF) mechanism. The basic speed ratio of the simple PGT, R_0, is defined as the ratio of the angular velocity of the sun gear to that of the ring gear with the carrier being fixed, i.e.,

$$R_0 = \omega_S / \omega_R = -N_R / N_S \tag{1}$$

where ω_S and ω_R are the angular velocities of the sun and ring gears, respectively. N_R and N_S are the numbers of teeth on the ring and sun gears, respectively. The relationship of the speeds of the three members (R, S, C) is:

$$(R_0 - 1)\omega_C = R_0\omega_R - \omega_S \tag{2}$$

where ω_C is the angular velocities of the carrier.

Under steady-state operation, torques applied on the ring gear, sun gear and carrier about the central axis of the simple PGT must sum to zero, i.e.,

$$T_R + T_S + T_C = 0 \tag{3}$$

where T_R, T_S, and T_C denote the external torques applied on ring gear, sun gear and carrier, respectively, and positive indicating the counter-clockwise direction. The power flows into the simple PGT via the three coaxial links must sum to zero, i.e.,

$$T_S\omega_S + T_R\omega_R + T_C\omega_C = 0 \tag{4}$$

Eliminating ω_R and T_R from Eq. (4) by using Eqs. (2) and (3), yields

$$T_S = -\frac{1}{1 - R_0} T_C \tag{5}$$

Eliminating ω_S and T_S from Eq. (4) by using Eqs. (2) and (3), yields

$$T_R = -\frac{R_0}{R_0 - 1} T_C \tag{6}$$

The simple PGT becomes 1-DoF when one member is fixed to the ground link or engaged to one another member, Figure 4. When two members are engaged, the PGT becomes a rigid block. Table 2 lists the reduction ratio of the simple PGT when one of the three members is fixed to the ground link. Let R denote the speed ratio of the simple PGT when the input and output are assigned and one link being fixed, i.e.,

$$R = \omega_{out} / \omega_{in} \tag{7}$$

where ω_{out} and ω_{in} are the angular velocities of the output shaft and input shaft, respectively. The values of the reduction ratio and the corresponding of transmission efficiency in each circumstance are also listed in Table 2 (Sheu 2007).

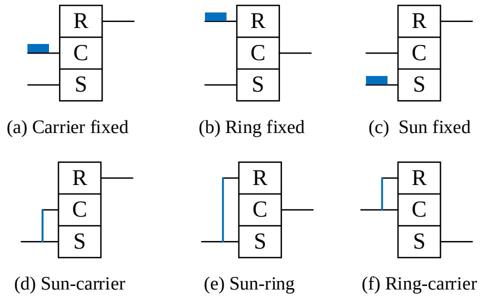
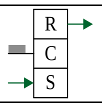
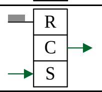
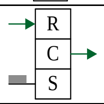
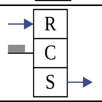
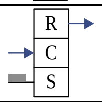
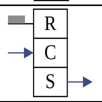


Fig. 4 1-DoF simple PGTs

Table 2 Gear ratios of the simple PGT with one link fixed

	Diagram	Reduction ratio (R)	Range of R	Transmission efficiency (η_{tm})
1		$1/R_0$	$-1<R<0$	η_0
2		$1/(1-R_0)$	$0<R<0.5$	$\dfrac{1-R_0\eta_0}{1-R_0}$
3		$R_0/(R_0-1)$	$0.5<R<1$	$\dfrac{\eta_0-R_0}{1-R_0}$
4		R_0	$R<-1$	η_0
5		$(R_0-1)/R_0$	$1<R<2$	$\dfrac{\eta_0(1-R_0)}{1-\eta_0 R_0}$
6		$1-R_0$	$R>2$	$\dfrac{\eta_0(R_0-1)}{R_0-\eta_0}$

η_0 is the basic transmission efficiency of the simple PGT

Compound PGTs

Compound PGTs are planetary gear trains having two or more simple PGTs in its structures. The most common compound PGTs includes seven members (two simple PGTs) or 9 members (three simple PGTs), which employed in various automatic and hybrid transmissions.

Kinematic analysis is an important step to determine the speed reduction ratios of the PGTs when input(s) and output are assigned, and various approaches have been proposed (Wilkinson 1960, Freudenstein and Yang 1971, Hsieh and Yan 1992). Hsieh and Tsai developed a methodology for calculating the overall speed ratio of a PGT by using fundamental gear entities (FGEs) and fundamental circuits (Hsieh 1996). A compound PGT is decomposed into many lowest level subsystems that contain only one FGE, and then the overall speed ratio of planetary gear mechanism can be expressed in terms of the speed ratio of its FGE. Tsai proposed a method using block diagrams for the kinematic analysis of planetary gear systems (Tsai et al. 2010). This method employed control techniques to analyze kinematic relationship via the block diagrams for planetary gear systems. Level analogy method is useful for analyzing the kinematics of compound PGTs, and it is used in many research papers to analyze complex compound PGTs (Zhang et al. 2015, Zhuang et al. 2015).

This research focuses only on the compound PGTs that contain two simple PGTs. Figure 5 shows two types of compound PGTs with seven members. The first type is the compound PGT with one common link to connect the two simple PGTs, and one member of a simple PGT is fixed. The second type is the compound PGT with two common links to connect the two simple PGTs. In the previous research, all possible compound PGTs were generated, including 18 compound PGTs of the first type and 12 compound PGTs of the second type (Ngo and Yan 2016). For the simpler design structures, the first type of compound PGT is focused to design a series-parallel hybrid transmission system. Figure 6 shows 18 compound PGTs that contain two simple PGT and one common link.

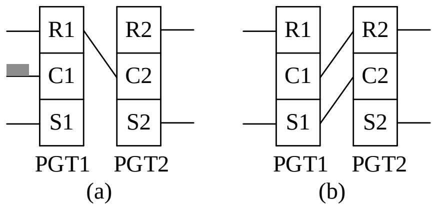
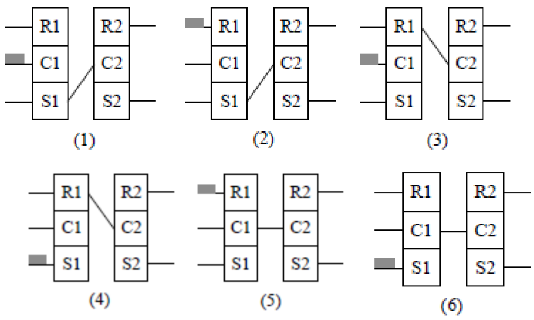


Fig. 5 Two types of compound PGTs



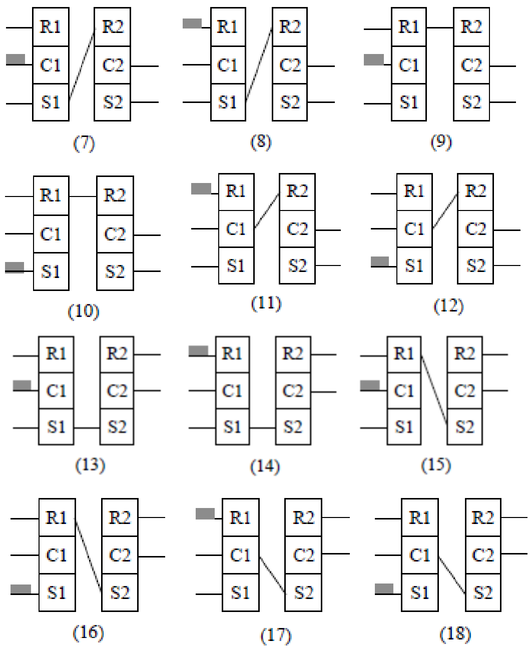


Fig. 6 Atlas of compound PGTs [Ngo and Yan 2016]

Clutchless hybrid transmission systems

Clutchless transmission systems are the obtained systems after assigning the inputs and output into a planetary gear train by utilizing power arrangement technique subject to power location constraints.

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Power location constraints

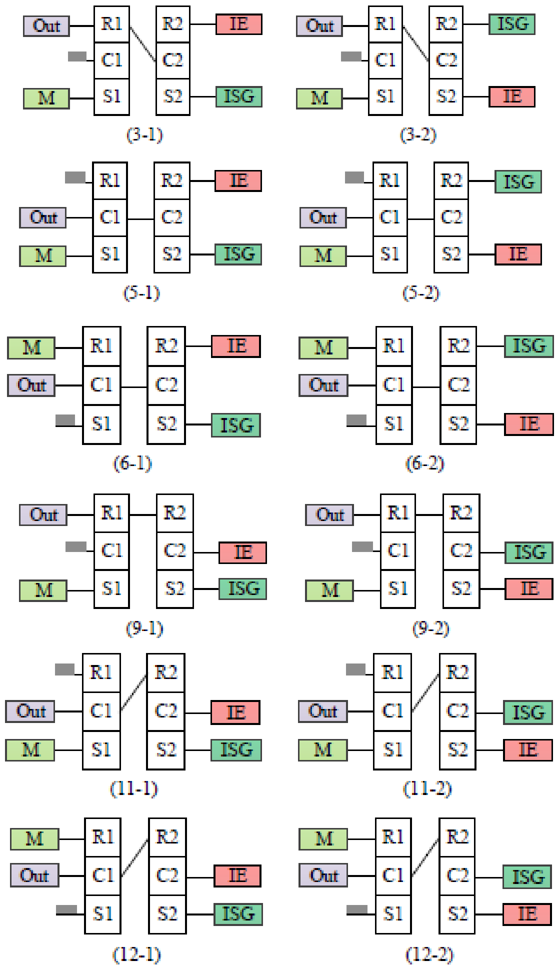
Based on the study of existing designs and characteristics of the output/input powers, including the output link (Out), the motor (M), the engine (IE) and the generator (ISG), constrains for the locations of the input powers in the system are as follows:

1. Output link (Out): There must be an output link, and it must be on a common link. The output link must be connected to the ring gear (R) or the carrier (C) of the 1-DoF planetary gear train (PGT1) in order to achieved the reduction ratio from the inputs.
2. Motor (M): There must be a motor, and the speed of the motor to the output (Out) must be a reduction ratio. Thus, the motor must not be connected to the carrier of the 1-DoF PGT in order to achieve the speed reduction at the output (Out).
3. Generator (ISG): There must be a generator, and it must not be connected to the common link of the 2-DoF PGT2 in order to be able to control the first of the two inputs of the PGT2.
4. Engine (IE): there must be an engine, and it must not be connected to the common link of the 2-DoF PGT2 and the link connected to the generator in order to control the second of the two inputs of the PGT2.

Power arrangement process

Power arrangement process is carried out by following steps:

1. Assign the output (Out): Since the output must be connected to the ring gear or the carrier of the 1-DoF PGT1, 12 compound PGTs, numbers (3), (4), (5), (6), (9), (10), (11), (12), (15), (16), (17) and (18), satisfies the constrain.
2. Assign the motor (M): Since the motor must be connected to the sun gear or ring gear of the PGT1, 9 compound PGTs, numbers (3), (5), (6), (9), (11), (12), (15), (17) and (18), satisfies the constrain.
3. Assign the engine (IE) and generator (ISG): Since the engine and the generator must not be connected to the common link, they are connected to two separated links. The assignment generates 18 results as shown in Figure 7.



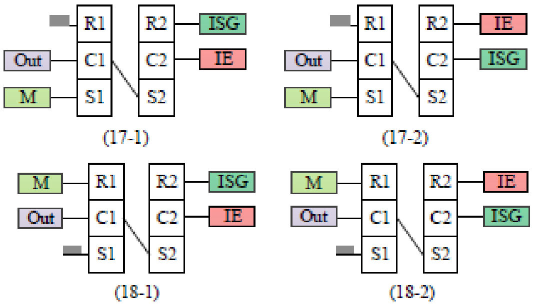


Fig. 7 Atlas of clutchless hybrid transmissions

Clutched hybrid transmission systems

A hybrid transmission system having operation modes controlled by clutched and brakes is called a clutched hybrid transmission system. The system is obtained by assigning clutches and brakes into a clutchless system obtained above subject to the predefined operation modes.

Required operation modes of a new system

A hybrid system operation mode represents the operating conditions of the system at that mode. From the operation mode, the working conditions of the inputs and output of the system are described. A series-parallel hybrid transmission typically consists of six primary operation modes (Mi et al. 2011). Based on the requirements of the new series-parallel hybrid transmission system and the primary operation modes, seven modes are concluded as follows:

1. *EV mode (Motor alone mode)*: Only the motor drives the vehicle, and only the motor is connected to the output.
2. *HEV series mode*: Only the motor drives the vehicles, while the engine drives the ISG working as the generator to generate electricity for the motor or for charging the batteries. The engine and the ISG run at the same speed for highest efficiency.
3. *Combined power mode*: The motor, engine and (or) ISG are connected to the output to provide power to drive the vehicle.
4. *Engine alone mode*: The engine must be connected to the output to drive the vehicle alone, and the ISG is fixed or free rotating.
5. *Power split mode*: The power from the engine is split into two parts, one for driving the vehicle and the other for driving the generator to charge the batteries or to provide electric to the motor directly.
6. *Regenerative braking mode*: During the vehicle braking process, the motor or the ISG is turned into the generator mode to regenerate electricity to charge the batteries.
7. *Stationary charging mode*: When the vehicle is at a standstill, the engine drives the ISG to charge the batteries for later use.

Clutch arrangement process

To control a clutchless system to perform the concluded seven operation modes, clutches are added to the system by using clutch arrangement techniques. Since there are many clutchless systems, the system number (3-1) in Figure 7 is taken arbitrarily to express the clutch arrangement process subject to the concluded seven operation modes. The steps of the process are as follows:

1. *EV mode*: Since the motor is connected to the output only, clutch CL1 is assigned to the common link to disconnect the engine and the ISG from the output, Figure 8.
2. *HEV series mode*: Since the motor alone driving the vehicle and the engine driving the ISG to generate electricity at the same speed, clutch CL2 is added to lockdown the PGT2. Since there are three positions that CL2 can be located, three systems are obtained, Figure 8.
3. *Combined power mode*: In this mode, the motor, the engine and the ISG must be connected to the output. With the two assigned clutches, this mode can be fulfilled, thus no more clutch is added.
4. *Engine alone mode*: Since the engine alone drives the vehicle, the PGT2 must be with 1-DoF, and it is done with clutch CL2 engaged. The ISG is rotated freely.
5. *Power split mode*: In this mode, the engine drives the vehicle and one of the motors turns into the generator mode. This mode is fulfilled with clutch CL1 engaged.
6. *Regenerative braking mode*: This mode is a reverse of one of the other modes, thus no more clutch is added.
7. *Stationary charging mode*: This mode is similar to the series mode; thus, no more clutch is needed.

Based on the concluded required operation modes, three clutched hybrid transmission systems are generated from one corresponding clutchless hybrid transmission system by using the mentioned techniques. The only difference among the three hybrid transmission systems, with the same system characteristics, is the location of clutch (CL2). In what follows, only one system is represented for the three possible locations of the clutch CL2. By applying the techniques for the other clutchless systems, 18 feasible clutched hybrid transmission systems are obtained as shown in Figure 9.

For the new synthesized hybrid transmissions, they all can performs the following same seven operation modes, and some modes are divided into sub-modes, as listed in Table 3.

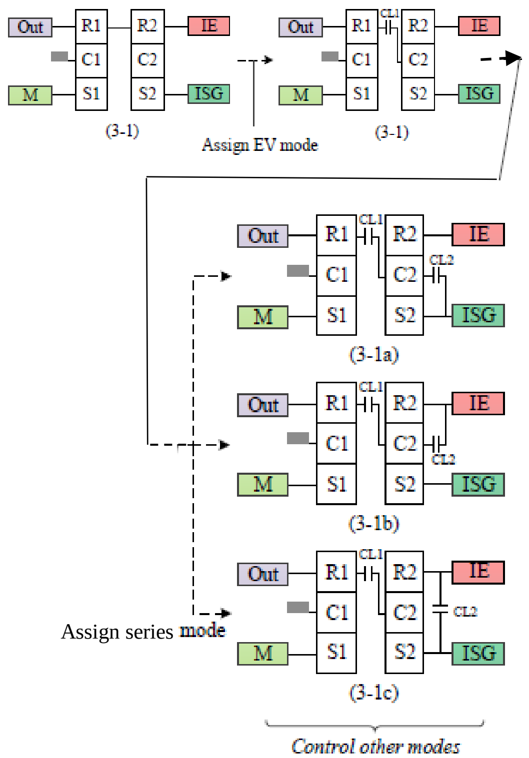
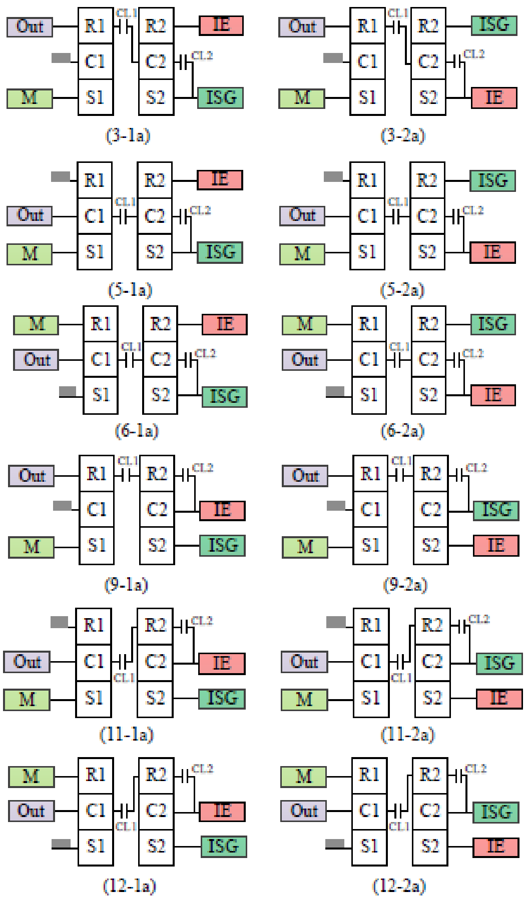


Fig. 8 Clutch arrangement process



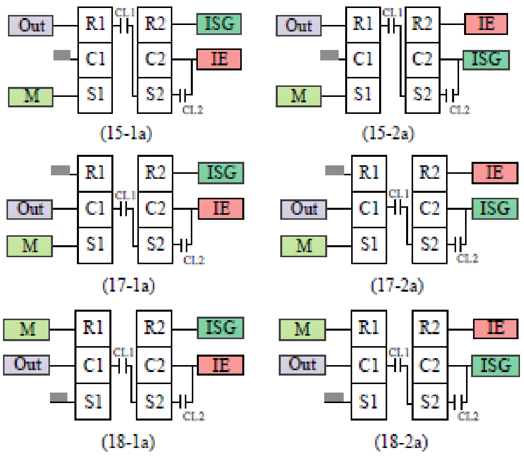


Fig. 9 Atlas of clutched hybrid transmissions

Table 3 Possible operation modes of the new hybrid transmissions

Operation mode	Clutch engaged		Power status		
	CL1	CL2	Motor	ISG	IE
EV		(x)	M	off	off
HEV series		x	M	G	on
Power mode 1	x		M	G	on
Power mode 2	x	x	M	M	on
CVT charging	x		free (G)	G	on
Engine drive	x	x	free	free	on
Regenerative braking 1		(x)	G	(free)	(on)
Regenerative braking 2	x		G	G	on
Stationary charging		x	off	G	on

M – Motor mode; G – Generator mode; x – clutch engaged; () possible

1. *EV mode (no clutch is engaged)*: Only the motor (M) is powered by the batteries to drive the vehicle. This is also called launch and back up mode, using in city drive, low speed, and low power demand.
2. *HEV series (only clutch CL2 is engaged)*: When the vehicle needs low power demand and the batteries is running out, the ISG starts the engine (IE). Then, the engine drives the ISG for generating electricity to charge the batteries or to provide directly the electricity to the motor.
3. *Power mode 1 or e-CVT 1 (only clutch CL1 is engaged)*: When the vehicle needs higher power demand, the engine power is provided to the output with the motor to drive the vehicle. The engine is controlled by the ISG to operate at its highest efficiency condition.

4. *Power mode 2* (*both clutches CL1 and CL2 are engaged*). When the vehicle at high speed with high power demand, clutch CL2 is engaged to lock the PGT2. The engine and the ISG run at the same speed and along with the motor to drive the vehicle. Three power sources can drive the vehicle.
5. *CVT charging* or *e-CVT 2* (*only clutch CL1 is engaged*): During highway cruising, the power from the engine is over vehicle requirement, part of the power is converted into electric power by the ISG to charge the batteries. The motor also can be a generator if needed.
6. *Engine drive (both clutches CL1 and CL2 are engaged):* When vehicle speed satisfies the high efficiency speed range of the engine, the engine alone dives the vehicle in a direct drive mode, generally in high way cruising condition.
7. *Regenerative braking 1 (no clutch is engaged)*: This mode is the reversion of the EV mode while the engine is not connected to the output. The motor is turned into the generator mode to regenerate electricity from the output kinetic energy.
8. *Regenerative braking 2 (only clutch CL1 is engaged)*: This mode is activated when the engine is still running and connected to the output. The motor and the ISG are turned into generator mode to convert the kinetic energy from the output into the electricity to charge the batteries.
9. *Stationary charging (only CL2 is engaged)*: The vehicle is stopped and the batteries are low, the engine drives the ISG to generate the electricity to charge the batteries for later use.

System parameter

Design and selection of system parameters are addressed as follows.

Design Specifications

The required hybrid transmission system is utilized in a vehicle with the following characteristics. The vehicle mass is m = 2,000 kg, the tire specification is 235/55/R18, and the tire dimensions are calculated as listed in Table 4. The engine can reach maximum power at 5,200 rpm. The vehicle is desired to run up to 220 km/h.

Table 4 Tire dimensions

Size	sidewall height	OD overall diameter (D_w)	Ride height overall radius (R_w)
235/55/R18	129.25mm	715.7mm	357.85mm

The 18 synthesized clutched hybrid transmissions is illustrated generally as shown in Figure 10 with the motor connected to the first

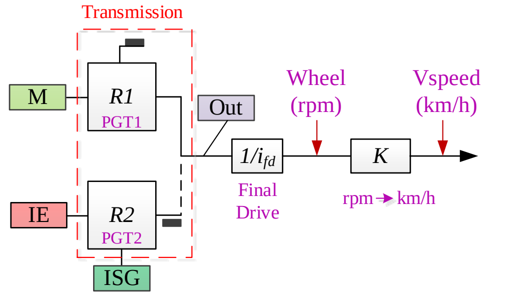


Fig. 10 Block diagram of the hybrid transmission system

planetary gear PGT1 having reduction ratio *R1*. The engine and the ISG are connected to the second PGT2 having reduction ratio *R2*. The power from transmission is passed through a final drive gear ration i_{fd} to drive the wheel shaft. And, K is the conversion coefficient to convert rpm to km/h, K = $36R_w\pi/300$.

EV and HEV series mode

When the motor drives the vehicle alone in the EV or HEV series mode, the vehicle is driven by the motor and the vehicle speed (*Vspeed*) is:

$$Vspeed = \frac{\omega_M R_1}{i_{fd}} K \quad \text{(km/h)} \tag{9}$$

where, ω_M is the speed of the electric motor (M).
Thus, the relationship between the final drive ratio (i_{fd}) and the PGT1 reduction ratio is:

$$i_{fd} = \frac{\omega_M R_1}{Vspeed} K = \frac{\omega_{output}}{Vspeed} K \tag{10}$$

where, ω_{output} is the speed of the output link of the transmission.

In this mode, the engine drives the generator to generate electricity for the motor or for charging the batteries. The PGT2 is locked, thus the engine and the ISG are operated as the same optimum speed conditions (3,000 rpm).

HEV CVT mode

The relationship of the sun gear, ring gear, and carrier of the PGT2 is:

$$\omega_{S2} - \omega_{R2} R_{02} = (1 - R_{02})\omega_{C2} \tag{11}$$

where, R_{02} is the basic speed ratio of the simple PGT2.

Figure 11 shows different speed relationship when the engine and the ISG are assigned to the PGT2. When the engine and the ISG are assigned to different location of the PGT2, ring gear R2, carrier C2 or sun gear S2, six configurations are generated as no. (1) to (6). The engine is operated at 5,200rpm for maximum power, red horizontal line at 5,200rpm, and it is located at three different locations, the ring gear, carrier and sun gear of the PGT2. The ISG is operated from zero to maximum speed 6,000rpm. The arrows show the output speed when the engine and ISG operated at 5,200rpm and 6,000rpm, respectively at different configurations. As shown in Figure 11, cases (2) and (3) are more feasible since the output speed is more stable when the ISG speed changed. Here, case (3) is selected for next calculation process. The engine is connected the carrier and the ISG is connected to the sun gear of the PGT2.

The relationship between the output speed and the speed of the inputs is:

$$\omega_{output} = \frac{(1-R_{02})\omega_{IE} - \omega_{ISG}}{-R_{02}} \qquad (12)$$

where, ω_E and ω_{ISG} are the speeds of the engine and the ISG, respectively.

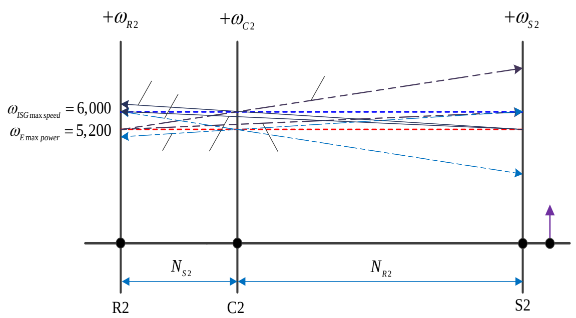


Fig. 11 Speed relationship in the PGT2

Based on the analysis of the vehicle maximum speed requirement, vehicle operation conditions, and planetary gear ratio design, the most feasible configuration for the transmission is selected, type (11-2a) in Figure 9. Some gear teeth are designed and selected for the new hybrid transmission, as listed in Table 5. Based on the vehicle speed at 3,300rpm of the motor speed, R_{01} = -2.071 (basic speed ratio of the simple PGT1), R_{02} = -2.5 and i_{fd} = 2.41 (53/22 teeth) are selected for the desired new hybrid transmission. And, the gear teeth numbers of the PGT1 and PGT2 are shown in Figure. 12.

ANALYSIS OF THE NEW SERIES-PARALLEL HYBRID SYSTEM

In this section, the novel hybrid transmission is fully analyzed for its mechanical transmission efficiencies, kinematics and drivability of the vehicle.

Table 5 Reduction gear ratios of the hybrid transmission system (Mspeed max 12,000 rpm)

	R_{01}	R_{02}	i_{fd}	Vspeed max km/h	Mspeed rpm	Vspeed km/h
1	-2.071	-2.5	1.99	264	3,300	72
2	-2.071	-2.5	2.41	219	3,300	**60**
3	-2.071	-2.5	2.89	182	3,300	**50**
4	-2.5	-2.5	1.75	264	3,300	72
5	-2.5	-2.5	2.11	219	3,300	**60**
6	-2.5	-2.5	2.53	182	3,300	**50**

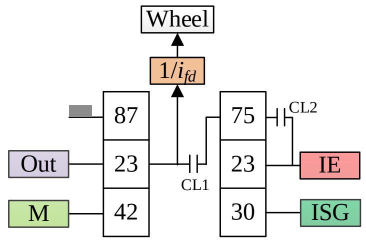


Fig. 12 Novel series-parallel hybrid transmission with gear teeth.

ANALYSIS OF THE NEW SERIES-PARALLEL HYBRID SYSTEM

In this section, the novel hybrid transmission is fully analyzed for its mechanical transmission efficiencies, kinematics and drivability of the vehicle.

Efficiency of the EV mode

Figure 13 shows the two modes (1-DoF & 2-DoF) of the new system. The efficiency of the new system is divided into two main parts corresponding to the two modes, EV mode and CVT mode. To calculate the efficiency of each mode, the basic efficiency of the PGTs need to be calculated in advanced.

Basic mechanical efficiency of a PGT is defined as the efficiency of the power transferred from the sun gear to the ring gear while the carrier fixed. It is calculated by multiplying the efficiencies of two gear pairs, sun gear with planet gear and planet gear with ring gear.

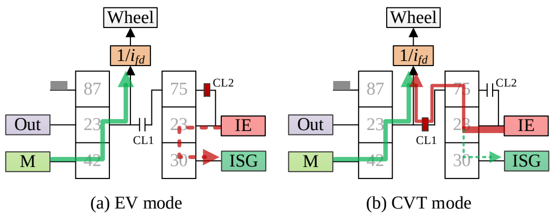


Fig. 13 New transmission at EV and CVT modes

The basic mechanical efficiency of the first PGT1 (η_{01}) is calculated as follows:

$$\eta_{01} \approx \left(1-\frac{1}{5}(\frac{1}{42}+\frac{1}{23})\right)\times\left(1-\frac{1}{5}(\frac{1}{23}-\frac{1}{87})\right) \approx 0.98 \quad (13)$$

Similarly, basic mechanical efficiency of the second PGT2 (η_{02}) is calculated as follows:

$$\eta_{02} \approx \left(1-\frac{1}{5}(\frac{1}{30}+\frac{1}{23})\right)\times\left(1-\frac{1}{5}(\frac{1}{23}-\frac{1}{75})\right) \approx 0.979 \quad (14)$$

Mechanical efficiency of the final gear ratio (η_{fd}) is:

$$\eta_{fd} \approx \left(1-\frac{1}{5}(\frac{1}{22}+\frac{1}{53})\right) \approx 0.987 \quad (15)$$

When the motor drives the output from the sun gear to the carrier of the PGT1, the transmission efficiency ($\eta_{M/O}$) is calculated from Table 2 as:

$$\eta_{M/O} = \frac{1-R_{01}\eta_{01}}{1-R_{01}} = 0.987 \quad (16)$$

In the HEV series mode, the engine drives the ISG for generating electricity to provide power to the motor. Assumed that the efficiencies of the motor and the ISG in operating is $\eta_{ISG} = \eta_M = 0.95$ for both the motor and generator modes, the transmission efficiency of the system (η_{EV}) is,

$$\eta_{EV} = P_{out}/P_{in} = \eta_{IE/ISG}\eta_{ISG}\eta_M\eta_{M/O} = 1\times 0.95\times 0.95\times 0.987 = 0.89 \quad (17)$$

where, P_{out} and P_{in} are the powers on the output shaft and the input shaft, respectively. $\eta_{IE/ISG}$ is the transmission efficiency of the power from the IE to the ISG. In this case the PGT2 is locked, thus $\eta_{IE/ISG} = 1$.

Efficiency of the PGT2 at CVT mode

Transmission efficiency from the engine to the ring gear ($\eta_{IE/O}$) is:

$$\eta_{IE/O} = \frac{\eta_{02}(1-R_{02})}{1-\eta_{02}R_{02}} = 0.994 \quad (18)$$

Transmission efficiency from the engine to the sun gear ($\eta_{IE/ISG}$) is:

$$\eta_{IE/ISG} = \frac{\eta_{02}(R_{02}-1)}{R_{02}-\eta_{02}} = 0.985 \quad (19)$$

And, transmission efficiency of the power from the engine to the motor to drive the vehicle through the generator ($\eta_{IE/M}$) is,

$$\eta_{IE/M} = \eta_{IE/ISG}\eta_{ISG}\eta_M\eta_{M/O} = 0.985\times 0.95\times 0.95\times 0.987 = 0.877 \quad (20)$$

Kinematics and power flows of the new hybrid transmission

Based on the operation modes and transmission efficiency in each mode, the power flow in each operation mode of the hybrid transmission is shown in Figure 14, and the kinematics and power flows are calculated as follows:

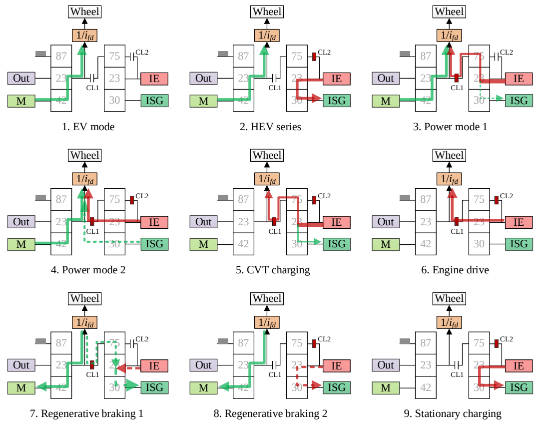


Fig. 14 Power flow in each mode

EV mode (no clutch is engaged)

In this mode, only the motor (M) is powered by the batteries to drive vehicle. This is also called launch and back up mode, using in city drive, low speed, and low power demand. The engine and the generator are off.

The speed/torque/power relationships of the input and output are:

$$\omega_{output} = \frac{1}{(1-R_{01})}\omega_M = 0.326\omega_M \quad (21)$$

$$T_{output} = -\eta_{M/O}(1-R_{01})T_M = -0.987\times 3.071\times T_M = -3.03T_M \quad (22)$$

where ω_{output} and ω_M denote the angular velocities of the output link and the motor, T_{output} and T_M denote the torques on the output link and the motor, respectively.

The output power is given by

$$P_{output} = -\eta_{M/O} P_M = -0.987 P_M \quad (23)$$

where P_{output} and P_M denote the power on the output link and the motor shaft, respectively.

HEV series or e-CVT (only clutch CL2 is engaged)

When the vehicle needs low power demand and the batteries is running out, the ISG starts the engine (IE). Then, the engine drives the ISG generating electricity to charge the batteries or to provide directly electric to the motor.

The speed/torque/power relationships of the input and output are:

$$\omega_{output} = \frac{1}{(1-R_{01})}\omega_M = 0.326\omega_M \quad (24)$$

$$\omega_{ISG} = \omega_{IE}\ ;\ T_{ISG} = -T_{IE} \quad (25)$$

$$\begin{aligned} T_{output} &= -\eta_{M/O}(1-R_{01})T_M \\ &= -0.987 \times 3.071 \times T_M = -3.03T_M \end{aligned} \quad (26)$$

where ω_{ISG} and ω_{IE} denote the angular velocities of the ISG and the IE, T_{ISG} and T_{IE} denote the torques on the ISG and the IE, respectively.

Assumed that the power of the motor to drive the vehicle is from the engine only, thus:

$$P_M = \eta_{EV} P_{IE} = 0.89 P_{IE} \quad (27)$$

$$P_{output} = -\eta_{M/O} P_M = -0.88 P_{IE} \quad (28)$$

where P_{IE} denotes the power on the IE shaft.

Power mode 1 (only clutch CL1 is engaged)

When the vehicle needs higher power demand, the engine power is provided to the output with the motor to drive the vehicle. The engine is controlled by the ISG to operate at its highest efficiency condition.

The speed/torque/power relationships of the input and output are:

$$R_{02}\omega_{output} = (R_{02}-1)\omega_{IE} + \omega_{ISG} \quad (29)$$

$$2.5\omega_{output} = 3.5\omega_{IE} - \omega_{ISG} \quad (30)$$

The power from the engine is divided into two paths.

$$P_{IE} = -P_{IE/ISG} - P_{IE/O} \quad (31)$$

where $P_{IE/ISG}$ and $P_{IE/O}$ denote the powers of the IE to the ISG and the IE to the output link, respectively. The motor provides the extra power with the engine to drive the vehicle, and

$$\begin{aligned} T_{output} &= -\eta_{M/O}(1-R_{01})T_M - \eta_{IE/O}\left(\frac{R_{02}}{R_{02}-1}\right)T_{IE} \\ &= -3.03T_M - 0.71T_{IE} \end{aligned} \quad (32)$$

The torque on the ISG transmitted from the engine is:

$$T_{ISG} = -\eta_{IE/ISG}\left(\frac{1}{1-R_{02}}\right)T_{IE} = -0.281T_{IE} \quad (33)$$

The power generated by the ISG is:

$$P_{ISG} = T_{ISG}\omega_{ISG} = -\eta_{IE/ISG}P_{IE/ISG} = -0.985P_{IE/ISG} \quad (34)$$

If the power is used for the motor to drive the vehicle, the power for the output link is:

$$P_{ISG/O} = \eta_{IE/M} P_{IE/ISG} = 0.877 P_{IE/ISG} \quad (35)$$

Power mode 2 (both clutches CL1 and CL2 are engaged)

When the vehicle at high speed with high power demand, clutch CL2 is engaged to lock the PGT2. The engine and the ISG run at the same speed and along with the motor to drive the vehicle. Three power sources can drive the vehicle.

The speed/torque/power relationships of the input and output are:

$$\omega_{output} = \omega_{IE} = \omega_{ISG} = 0.326\omega_M \quad (36)$$

$$\begin{aligned} T_{output} &= -\eta_{M/O}(1-R_{01})T_M - T_{IE} - T_{ISG} \\ &= -3.03T_M - T_{IE} - T_{ISG} \end{aligned} \quad (37)$$

The output power is given by

$$P_{output} = -0.987P_M - P_{IE} - P_{ISG} \quad (38)$$

where P_{ISG} denotes the power on the ISG shaft.

CVT charging (only clutch CL1 is engaged)

During highway cruising, the power from the engine is over vehicle requirement, and part of the power is converted into electric power by the ISG to charge the batteries. The motor also can be a generator if needed.

The speed/torque/power relationships of the input and output are:

$$2.5\omega_{output} = 3.5\omega_{IE} - \omega_{ISG} \quad (39)$$

The power from the engine is divided into two paths, and

$$P_{IE} = -P_{IE/ISG} - P_{IE/O} \quad (40)$$

The torque of the output shaft is:

$$T_{output} = -\eta_{IE/O}\left(\frac{R_{02}}{R_{02}-1}\right)T_{IE} = -0.71T_{IE} \tag{41}$$

The torque of the engine transmitted to the ISG is:

$$T_{ISG} = -\eta_{IE/ISG}\left(\frac{1}{1-R_{02}}\right)T_{IE} = -0.281T_{IE} \tag{42}$$

The power generated by the ISG is:

$$\begin{aligned}P_{ISG} &= T_{ISG}\omega_{ISG} = -\eta_{IE/ISG}P_{IE/ISG}\\ &= -0.985P_{IE/ISG}\end{aligned} \tag{43}$$

And, the power for driving the vehicle is:

$$P_{output} = \eta_{IE/O}P_{IE/O} = 0.994P_{IE/O} \tag{44}$$

Engine drive (both clutches CL1 and CL2 are engaged)

When vehicle speed satisfies the high efficiency speed range of the engine, the engine alone dives the vehicle in a direct drive mode, generally in high way cruising condition.

The speed/torque/power relationships of the input and output are:

$$\omega_{output} = \omega_{IE} \tag{45}$$

$$T_{output} = -T_{IE} \tag{46}$$

And, the output power is given by

$$P_{output} = -P_{IE} \tag{47}$$

Regenerative braking 1 (no clutch is engaged)

This mode is the reversion of the EV mode while the engine is not connected to the output. The motor is turned into the generator mode to regenerate electricity from the output kinetic energy.

The speed/torque/power relationships of the input and output are:

$$\omega_M = (1-R_{01})\omega_{output} = 3.071\omega_{output} \tag{48}$$

All kinetic power from the output is converted into the electricity by the motor, working as a generator. The power of the motor is:

$$P_M = -\frac{\eta_{01}(R_{01}-1)}{R_{01}-\eta_{01}}P_{output} = -0.986P_{output} \tag{49}$$

The power on the batteries is:

$$P_{battery} = \eta_M P_M = -0.937P_{output} \tag{50}$$

Regenerative braking 2 (only clutch CL1 is engaged)

This mode is activated when the engine is still running and connected to the output. The motor and the ISG are turned into generator mode to convert kinetic energy from the output into the electricity to charge the batteries.

The speed/torque/power relationships of the input and output are:

$$\omega_M = (1-R_{01})\omega_{output} = 3.071\omega_{output} \tag{51}$$

$$2.5\omega_{output} = 3.5\omega_{IE} - \omega_{ISG} \tag{52}$$

If the ISG is running freely, all the kinetic power from the output is converted into the electricity by the motor, working as a generator. The power of the motor is:

$$P_M = -\frac{\eta_{01}(R_{01}-1)}{R_{01}-\eta_{01}}P_{output} = -0.986P_{output} \tag{53}$$

The power on the batteries is:

$$P_{battery} = \eta_M P_M = -0.937P_{output} \tag{54}$$

If the ISG is operated as a generator, the power from the output is divided into two paths, one for driving the motor and the other for driving the ISG. Thus,

$$P_{output} = P_{O/M} + P_{O/ISG} \tag{55}$$

$$P_M = -\frac{\eta_{01}(R_{01}-1)}{R_{01}-\eta_{01}}P_{O/M} = -0.986P_{O/M} \tag{56}$$

$$P_{ISG} = -\eta_{02}P_{O/ISG} = -0.979P_{O/ISG} \tag{57}$$

where $P_{O/M}$ and $P_{O/ISG}$ denote the powers of the output to the PGT1 and PGT2, respectively.

The power on the batteries is:

$$\begin{aligned}P_{battery} &= \eta_M P_M + \eta_{ISG}P_{ISG}\\ &= -0.937P_{O/M} - 0.93P_{O/ISG}\end{aligned} \tag{58}$$

Stationary charging (only CL2 is engaged)

In this mode, the vehicle is stopped and the batteries are low. The engine drives the ISG to generate the electricity to charge the batteries for later use. Thus,

The speed/torque/power relationships of the input and output are:

$$\omega_{ISG} = \omega_{IE} \tag{59}$$

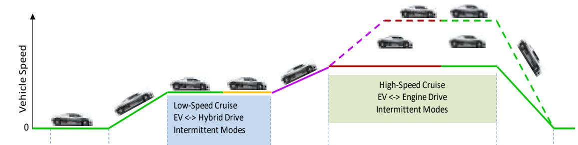
$$T_{ISG} = -T_{IE} \tag{60}$$

$$P_{ISG} = -P_{IE} \tag{61}$$

The power on the batteries is:

$$P_{battery} = \eta_{ISG} P_{ISG} = -0.95 P_{IE} \tag{62}$$

It is flexible in controlling the designed hybrid transmission during vehicle operation by switching among the nine operation modes. Figure 15 shows an example of the vehicle operating conditions, its corresponding operation modes, and its inputs/output and clutching conditions for controlling reference.



Motor	Stopped	Running	Running	Running	Running	Running	Free	Free	Regenerating
ISG	Stopped	Stopped	Stopped	Generating	Generating	Running	Free	Generating	Free
IE	Stopped	Stopped	Stopped	Running	Running	Running	Running	Running	Running
Battery	Discharging	Discharging	Discharging	Charging	Discharging	Discharging	Idle	Charging	Charging
Mode	Stationary charging	EV	EV	HEV Series	Power 1	Power 2	IE drive	CVT charging	Regenerative braking 2
CL1					X	X	X	X	
CL2	(X)	(X)	(X)	X		X	X		X
Remark					CVT	Direct drive	Direct drive	CVT	

Fig. 15 Possible mode switching in the new designed system

Drivability Analysis (Performance Analysis)

The drivability performance of a hybrid vehicle generally includes the maximum speed, acceleration, and grade ability. Given vehicle system components, it is necessary to calculate the vehicle maximum speed, acceleration, and grade ability under different driving conditions.

Maximum speed

When the motor operated at its maximum speed 12,000 rpm, the vehicle speed is 219 km/h, and the speed of the output shaft is 3,906 rpm.

Theoretically, the maximum speed of vehicle can be determined, when the engine and the generator operated at the speed for maximum power, as:

$$\omega_{output} = \frac{3.5}{2.5}\omega_{IE} - \frac{1}{2.5}\omega_{ISG} = 4880 \text{ (rpm)} \tag{63}$$

$$Vspeed = \frac{36 R_w \pi}{300 i_{fd}} \omega_{output} = \frac{36 \times 0.357 \times \pi}{300 \times 2.41} 4880 = 272 \text{ km/h} \tag{64}$$

Therefore, the maximum speed of vehicle is limited by the speed of the motor and the powers of the inputs. Since the maximum speed of the motor is 12,000 rpm, the maximum speed of the vehicle can be up to 219 km/h.

Acceleration

To calculate the maximum acceleration of a hybrid vehicle, it is necessary to consider the road condition, the wind condition, and the altitude at which the vehicle operates. These environmental factors can be found in Society of Automotive Engineers (SAE) standard J2188. For a given vehicle, the acceleration time t_{acc}(s) of the vehicle from an initial speed v_i to the final speed v_f is calculated based on the following expression:

$$v_f = v_i + \int_0^{t_{acc}} a(t)dt = v_i + \frac{3.6}{m}\int_0^{t_{acc}} [F_{\text{traction}} - (F_{\text{rolling}} + F_{\text{aero}} + F_{\text{grade}})]dt \tag{65}$$

where, a is vehicle's acceleration (m/s^2), $F_{traction}$ is traction force on the wheel from the drivetrain (N), v is vehicle speed (km/h), $F_{rolling}$ is rolling resistance force (N), F_{aero} is aerodrag resistance force (N), and F_{grade} is grade weigh forces (N).

The forces are calculated as:

$$F_{traction} = \eta_{fd}\left[\eta_{M/O}(1-R_{01})T_M + \eta_{IE/O}\frac{R_{02}}{R_{02}-1}T_{IE}\right]\frac{1}{i_{fd}R_w} \tag{66}$$

$$F_{rolling} = k_{rrc}k_{sc}mg \tag{67}$$

$$F_{aero} = k_{aero}v^2 \tag{68}$$

$$k_{aero} = \frac{1}{2}\rho C_d A_F \tag{69}$$

$$F_{grade} = mg\sin(\alpha) \tag{70}$$

Where k_{rrc} is rolling resistance coefficient (=0.03 for tar or asphalt), k_{sc} is road surface coefficient (~0.7 dry road), m is vehicle mass (2,000 kg), k_{aero} is aerodrag factor, ρ is air mass density (1.225 kg/m^3), C_d is aerodynamic drag coefficient (N.s^2/kg.m), A_f is vehicle frontal area (m^2), and α is road incline angle (rad).

The vehicle frontal area is calculated as (Rajesh 2012):

$$A_f = 1.6 + 0.00056(m - 765) = 2.29 \text{ (m}^2\text{)} \tag{71}$$

When the vehicle starts launching, the engine can also work at 1,714 rpm to provide extra power with the motor drive the vehicle. $T_{IE} \approx 250; T_M \approx 297.9$ (Nm) and vehicle mass = 2,100 kg (including one man and fuel), the traction force is:

$$F_{traction} = \eta_{fd}\left[\eta_{M/O}(1-R_{01})T_M + \eta_{IE/O}\frac{R_{02}}{R_{02}-1}T_{IE}\right]\frac{1}{i_{fd}R_w} \qquad (72)$$
$$\approx 9278.74 \ \ \text{(N)}$$

The rolling resistance force is:

$$F_{rolling} = k_{rrc}k_{sc}mg \qquad (73)$$
$$= 0.7\times 0.03\times 2100\times 10 = 441 \ \ \text{(N)}$$

Assume that the aerodrag resistance force and grade weigh forces are zeroes. Then the vehicle speed after 6.6 sec is:

$$v_f = \frac{3.6}{m}\int_0^{6.6}(F_{\text{traction}} - F_{\text{rolling}})dt \qquad (74)$$
$$= \frac{3.6}{2100}(9278.74-441)\times 6.6 \approx 100 \ \ \text{(km/h)}$$

This means that vehicle needs ~6.6 sec to reach 100 km/h from zero with the assumed conditions.

Grade ability calculation

The percent grade ability versus vehicle speed is defined as

$$Pct = 100\tan\left[\sin^{-1}\left(\frac{P_{fd}-(P_R+P_A)}{mgv_{spd}}\right)\right] \qquad (75)$$

The power on the final drive shaft (W), P_{fd}, is:

$$P_{fd} = F_{traction}v_{spd} \qquad (76)$$

The required power to overcome rolling resistance (W), P_R, is:

$$P_R = F_{rolling}v_{spd} = k_{rrc}k_{sc}mgv_{spd} \qquad (77)$$

And, the required power to overcome air drag (W), P_A, is:

$$P_A = F_{aero}.v_{spd} = (k_{aero}v_{spd}^2)v_{spd} = k_{aero}v_{spd}^3 \qquad (78)$$

where, v_{spd} is vehicle operating speed (m/s)

When the vehicle is launched from zero speed, P_A is zero. With $T_{IE}\approx 240; T_M \approx 297.9$ (Nm); and vehicle mass = 2100 kg (including one man and fuel), the percent grade ability of the vehicle speed is:

$$Pct = 100\tan\left[\sin^{-1}\left(\frac{F_{\text{traction}}-F_{\text{rolling}}}{mg}\right)\right] \qquad (79)$$

or road incline angle is 24.7º.

CONCLUSIONS

A systematic design process is presented in this work to synthesize a novel parallel hybrid transmission system subject to design specifications and vehicle performances. There are 18 compound planetary gear trains chosen for the synthesizing process. And, 18 clutchless hybrid transmission systems are obtained by assigning the input/output powers using the power arrangement techniques subject to power location constraints. By adding two clutches to control the systems subject to the required operation modes, 54 feasible clutched hybrid transmission systems are obtained.

The synthesized systems can perform nine clutching conditions, which are grouped in seven operation modes. The EV mode is activated to start and launch the vehicle at low and middle speed. The HEV series mode is used when the batteries are running out to a predetermined level. The power modes are needed for high power demand conditions. The CVT charging mode is for driving and charging simultaneously. The engine drive mode is activated when cruising. The regenerative braking modes are switched during braking process. And, the stationary charring mode is used for charging the batteries at standstill. In addition, the nine clutching conditions are flexibly controlled during vehicle running.

Based on the specifications of the input/output powers and requirements of vehicle performances, gear teeth number of all the gears in the system are calculated and selected. Furthermore, the new parameterized systems are analyzed for transmission efficiencies, kinematics and power flows in each operation mode, and the drivability of the vehicle are calculated. The results show that the vehicle can run the maximum speed at 220 km/h, accelerate from 0 to 100 km/h in 6.8 seconds, and achieve the gradeability of 24.7 degrees.

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NOMENCLATURE

B	brake
C, R, S	carrier, ring gear and sun gear of planetary gear train
CL	clutch
CVT	continuously variable transmission
e-CVT	electric continuously variable transmission
EV	electric vehicle
G	generator
HEV	hybrid electric vehicle
HSD	hybrid Synergy Drive
ISG	integrated starter generator
i_{fd}	final drive ratio
Out	speed reduction at the output
M	electric motor
P_{**}	power acting on the axis of the subscript
R	basic speed ratio of the simple PGT
R_{01}, R_{02}	basic speed ratio of the PGT1 and PGT2
T_{**}	torque applied on the element of the subscript
ω_{**}	angular velocity of the element of the subscript
η_{01}	efficiency of the first PGT1
η_{02}	efficiency of the second PGT2
η_{**}	efficiency of the element of the subscript

新型複式行星齒輪系串並聯混合動力系統的設計

吳友績 顏鴻森
國立成功大學機械工程系

許坤寶
國立虎尾科技大學車輛工程系

摘要

本文提出一種新型串並聯混合動力系統的設計方法。首先歸納現有混合動力系統之動力輸入與輸出的特性，提出一個包括四個主要步驟的系統設計方法來合成新型的混合動力系統。本文主要以複式行星齒輪系經由動力輸入與輸出的分配合成出各種沒有離合器的動力系統，然後適當的加入離合器以得到各種運作模式的混合動力系統，最後根據車輛動力特性需求來獲得滿足設計規範的串並聯混合動力系統。本文在各種運作模式下，分析所設計的混合動力系統之車輛動力性能與傳動效率，結果顯示所設計的系統可滿足設計需求和規範。