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Performance Modeling and Optimization of a Novel Multi-mode Hybrid Powertrain

This paper presents a systematic model for the operation simulation and optimization of a novel multi-mode hybrid powertrain. The hybrid configuration proposed in the paper features a planetary gear train for an electric CVT mode in addition to lay-shaft gears for multiple speed ratios and realizes six operation modes in a simple structure. Detailed component level models were established for the multi-mode hybrid transmission and integrated to the overall vehicle model according to the system configuration using the Simulink/Advisor platform. The vehicle control strategy was then established with the objective to optimize the overall vehicle operation and each hybrid operation mode in terms of fuel economy and emission levels. The performance of the proposed hybrid vehicle system was studied using the developed model under various operation conditions and benchmarked with a current market model with leading performance parameters. The proposed hybrid configuration shows substantial improvements over the benchmark and is validated as a viable hybrid design based on the model simulation.
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1 Introduction

Vehicles with hybrid powertrains are more fuel efficient and environment friendly compared with conventional vehicles that are powered by internal combustion engines alone. The advantages of hybrid powertrains are realized through the operation optimization of the power sources and the recovery of vehicle kinetic energy during braking and coasting. With the ever more stringent standard on fuel economy and emission levels, hybrid powertrain technologies have attracted intensive research and development interests in the automotive industry. The Toyota Prius and the Honda Insight were pioneers in the hybrid vehicle market, followed by the Ford Escape, the GMC Sierra, and other models from major automotive manufacturers. Although the market share is insignificant today due to cost constraints, it is predicted that hybrid vehicles will become more popular in the market as the related technologies continue to mature.

The system configuration and operation optimization have been the focus of research and development in hybrid vehicles in both academic community and industry. Tsai and Schultz proposed a hybrid design with multiple operation modes based on an automatic transmission platform [1,2]. The mode shift strategies and the model simulation results on the performance of the hybrid vehicle were presented in a subsequent paper [3]. Using driving pattern recognition, Jeon et al. proposed the algorithms for the operation control and optimization of a parallel hybrid electric vehicle [4]. Based on multi-objective nonlinear optimization, Won et al. proposed an energy management strategy for the torque distribution and charge sustaining control of a parallel hybrid vehicle with continuously variable transmission [5]. Using a solid oxide fuel cell as the auxiliary power unit, Gomez and Smith recently proposed a mild hybrid configuration incorporated with a continuously variable transmission for operation flexibility [6]. Rizoulis et al. proposed the control strategies for a series-parallel hybrid vehicle and improved vehicle performance in terms of acceleration and fuel economy through model simulation and experiments [7].

In the automotive industry, the research efforts have been more concentrated on developing technologies for the design and manufacturing of production hybrid models. Walters et al. presented a detailed comparison on the structure and performance of various hybrid configurations in the market and under development [8]. Technologies related with the hybrid system used in Prius have been recently improved and new generation of hybrid vehicles are being developed in Toyota Motor Corporation for further fuel economy and performance improvement [9,10]. Research and development at General Motors has led to the introduction of a hybrid full-size pickup truck in the automotive market place [11,12]. Hybrid technologies are not limited to passenger vehicles and commercial vehicles with hybrid powertrains have been recently developed and are now entering the commercial vehicle market [13].

Simulation based analysis on vehicle performance is crucial to the development of hybrid powertrain systems since design validation by hardware measurement is impractical prior to the costly prototype building. Significant amount of research efforts has recently been devoted to the modeling, simulation, and control of hybrid powertrain systems. Wipke et al. presented in detail the modeling and overall approach of the vehicle simulation platform ADVISOR and illustrated the application, capability, and reconfigurability of the modeling software tool [14]. An integrated simulation tool in Simulink was developed at the Automotive Research Center of the University of Michigan for studies on hybrid vehicle fuel economy [15]. Computer models on both component and system levels were proposed by Powell et al. for the analysis and control of the dynamic behavior of hybrid vehicles [16]. Butler et al. developed a Matlab/Simulink based modeling and simulation package for the simulation of hybrid vehicle operation under various drive cycles [17]. Rizzoni et al. proposed a system-oriented approach to facilitate the performance analysis of hybrid systems of different configurations [18]. As revealed by the literature survey, it is a standard practice to use model simulation for the design and analysis of hybrid vehicles in both component and system levels.

The Toyota Prius uses a parallel hybrid configuration, where a planetary gear train is used to split the engine torque between the drive shaft and the generator shaft. The electricity generated can be used either to charge the battery or power the motor to drive the output shaft. This design achieves an electric CVT mode for the vehicle since the planetary gear train has two degrees of free-

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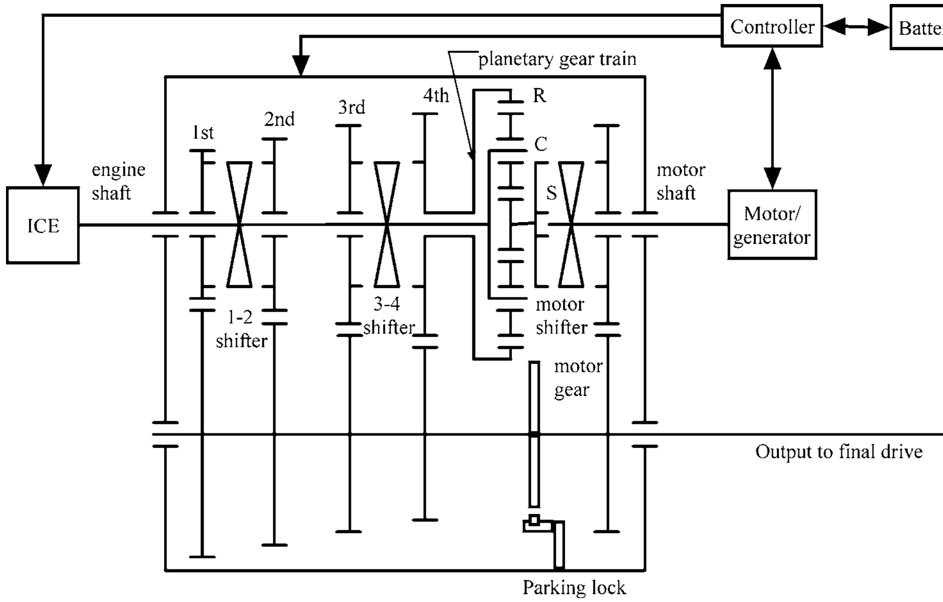

Fig. 1 Configuration of the multimode hybrid powertrain

dom. The speed of the generator can be controlled according to the vehicle driving conditions to keep the engine within the optimal operation range. Despite its advantages, the Toyota Prius has an intrinsic structural drawback in that it does not allow direct mechanical torque transmission from the engine to the drive shaft. The generator must be always loaded to provide the reaction required for the planetary gear train to transmit the engine toque to the output. Under normal driving conditions, the electricity generated is directed to power the motor and assist the drive shaft. This power flow path involves two conversions of energy forms, first by the generator and then by the motor, and each conversion results in a power loss. Consequently, the overall vehicle fuel economy performance may not be the best even though the engine operation is optimized at the most efficient status.

In this paper, a novel hybrid configuration is proposed to overcome the disadvantages of the Toyota Prius with a simple structure. This design uses one planetary gear train to provide an electric CVT mode and lay-shaft gears for other modes. Mode switching and gear shifting are actuated by step motors as in an automated manual transmission. The design uses only one electric machine, either functioning as a generator or a motor, and realizes all the functionalities required of a hybrid on an automated manual transmission platform. A detailed description on the hybrid structure and the kinematics in each operation mode are presented in Sec. 2.

The main objective of this paper is to present the novel multimode hybrid configuration mentioned above and the systematic model for the operation simulation and optimization of vehicles equipped with this hybrid system. Detailed component level models have been established for the multi-mode hybrid transmission and integrated to the overall vehicle model according to the system configuration and power flow paths on the Simulink/Advisor platform. The vehicle control strategy is then established with the objective to optimize the overall vehicle operation and each hybrid operation mode in terms of fuel economy. The performance of the proposed hybrid vehicle system is studied using the developed model under various operation conditions and benchmarked with a current market model with leading performance parameters. The model simulation has demonstrated substantial improvements over the benchmark and validated the proposed multimode hybrid configuration as a feasible vehicle powertrain with high fuel efficiency.

2 Operation Modes and Kinematics

The configuration of the proposed hybrid powertrain system is shown in Fig. 1. The design is based on the concept of automated manual transmissions. It uses a combination of lay-shaft gearing and planetary gearing for power transmission. The lay-shaft gears on the input shaft and the motor shaft free wheel unless engaged by the shifter-synchronizer assemblies. The carrier of the planetary gear train is connected with the input shaft that picks up the engine torque. The sun is connected with the motor shaft if so engaged. One motor is used either as the driving assisting unit or as the generator in charging and energy recovery operations. Mode switching and gear shifting are realized by shifters actuated by computer controlled step motors as in an automated manual transmission. The hybrid system has five operation modes for vehicle driving and one stand still mode for emergency or convenience operations. The six operation modes and the related kinematics are described in the following.

Motor Alone Mode. The vehicle is always launched in the motor alone mode. In this mode, the motor shaft is engaged by the shifter and power is transmitted to the final drive by the motor gears. Vehicle backup is realized by reversing the motor rotation. All other gears free wheel in this mode. The operation parameters for the engine and the motor in this mode are related by

$$\omega_o = \frac{\omega_m}{i_m}$$

$$T_o = i_m T_m \tag{1}$$

where ω_o and ω_m are the angular velocities of the output shaft and the motor, respectively, T_o and T_m are the output torque and the motor torque respectively, i_m is the motor gear ratio.

Combined Power Mode. The combined mode is used when high power is required in situations such as acceleration and slope climbing. In this mode the motor shaft and one of the lay-shaft pairs are engaged. One of the four available gears (as shown in Fig. 1) can be selected according to the vehicle operation condition. The motor and the engine operate at speeds that are mechanically linked by the gear ratios and drive the vehicle jointly, with the operation parameters related as follows

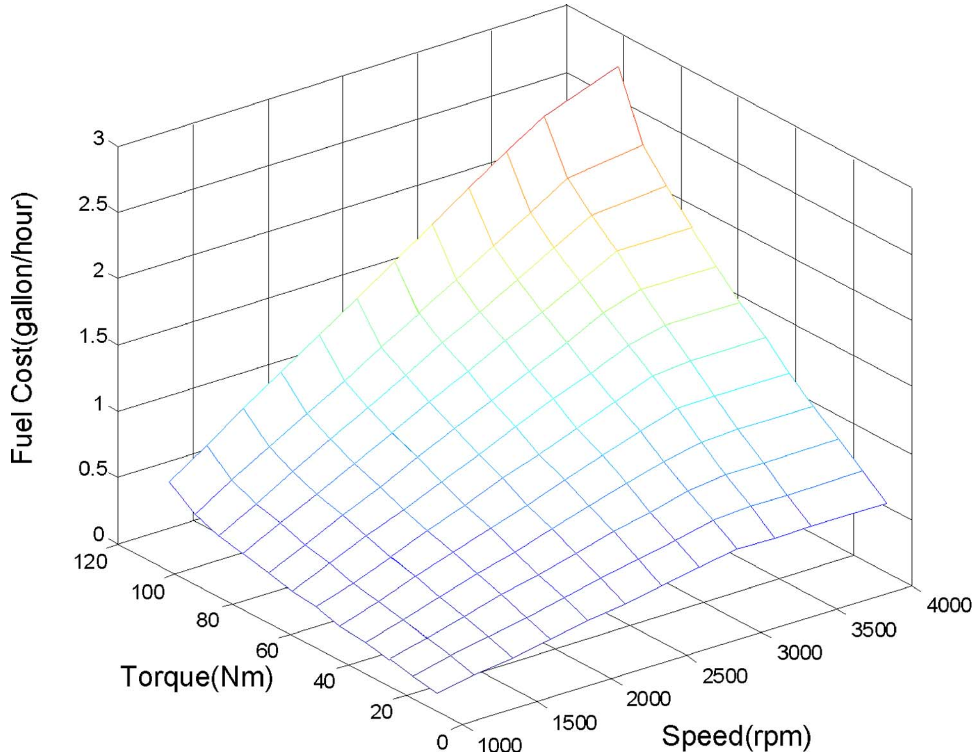

Fig. 2 Engine fuel map

$$\omega_o = \frac{\omega_e}{i_k} = \frac{\omega_m}{i_m}$$

$$T_o = i_k T_e + i_m T_m \tag{2}$$

where ω_e and T_e are the angular velocity and the torque of the engine, respectively, i_k (with $k=1, 2, 3, 4$) is the gear ratio of the lay-shaft gear pairs.

Engine Alone Mode. The engine alone mode is the most efficient mode for highway cruising. In this mode, a lay-shaft gear pair with low gear ratio is engaged to transmit the engine torque to the output shaft with the motor shaft at neutral. The vehicle runs like a conventional vehicle in this mode. The operation parameters are linked by the lay-shaft gear ratio

$$\omega_o = \frac{\omega_e}{i_k}$$

$$T_o = i_k T_e \tag{3}$$

Electric CVT Mode. The electric CVT mode provides two degrees of freedom for vehicle operation control that allows for the optimization of engine operation for best fuel economy. In this mode, the engine is the only power source that drives the vehicle and powers the generator for battery charging at the same time. The sun gear of the planetary gear train is coupled to the motor shaft by the shifter and the fourth lay-shaft gear is coupled to the ring gear. The operation parameters of the system are governed by the characteristics of the planetary gear train as follows

$$\omega_m + \beta i_4 \omega_o - (1+\beta)\omega_e = 0 \tag{4}$$

$$T_o = i_4 T_R = \frac{i_4 \beta}{1+\beta} T_e$$

$$T_m = T_s = \frac{1}{1+\beta} T_e \tag{5}$$

where β is the parameter of the planetary gear train. T_R and T_s are the ring gear torque and sun gear torque which are distributed from the engine torque at a constant proportion. The output torque T_o and the angular velocity ω_o are determined by the vehicle driving condition. The engine torque and the torque provided to the generator are determined by Eq. (5). In the electric CVT mode, the engine speed ω_e is optimized at the point for the highest efficiency corresponding to the required torque determined by Eq. (5). The generator speed ω_m is controlled to follow up the engine speed according to Eq. (4).

Energy Recovery Mode. In the energy recovery mode, the motor is coupled to the output shaft through the motor gear pair by the shifter and functions as a generator. The relations on the operation parameters are the same as that in the motor alone mode, with the power flow reversed.

Stand Still Mode. In this mode, the motor is engaged by the shifter to the sun gear and the parking locker is applied to lock the output shaft. The lay-shaft gears free wheel. This mode can be used to crank start the engine or charging a low battery at stand still by the engine. It can also be used as a generator for household electricity or other conveniences

$$\omega_e = \frac{1}{1+\beta}\omega_m$$

$$T_e = (1+\beta) T_m \tag{6}$$

It should be pointed out that this mode is not used for vehicle driving and thus not included in the powertrain system model.

Power Required on the Output Shaft. In the performance simulation of the vehicle, only the longitudinal motion of the vehicle and the associated road loads are incorporated in the system model. The output shaft torque in each driving mode must overcome the road loads, including rolling resistance, air resistance,

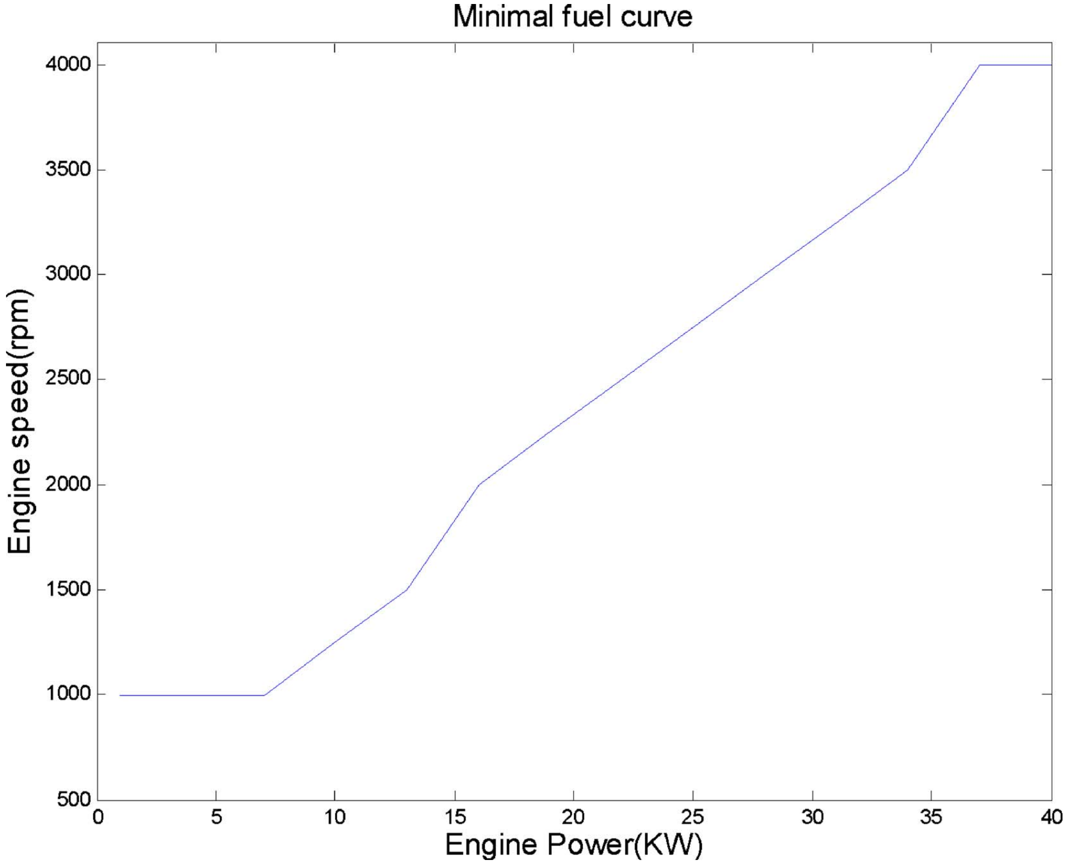

Fig. 3 Engine minimal fuel curve

and grade resistance, as well as the vehicle inertia during acceleration to follow up speed-time profiles. The equation governing the longitudinal motion of the vehicle is presented as follows

$$m\frac{dv}{dt} = \frac{i_a \eta_a T_o}{r_t} - F_B - R_R$$

$$R_R = fmg + cv^2 + Gmg$$

$$v = r_t \frac{\omega_o}{i_a} \tag{7}$$

where v is the vehicle velocity and m is the vehicle mass. i_a and η_a are the final drive ratio and efficiency. F_B and R_R are the brake force and road load. f, c, G, and r_t are the rolling coefficient, air drag factor, grade factor, and tire radius, respectively. For a given driving condition, the required output shaft torque T_o and angular velocity ω_o, and therefore the required power, are determined by Eq. (7). The vehicle controller will select the appropriate driving mode and optimize the operations of the power sources to meet the power requirement with the lowest possible fuel consumption.

3 Vehicle Control

There are two major technical issues in the operation control of vehicles equipped with the multi-mode hybrid powertrain: the selection of the five driving modes (excluding the stand still mode) and the optimization of the power sources in each operation mode. Mode selection is primarily based on vehicle power requirements under various driving conditions, with the battery state of charge as a secondary factor to be considered. It should be pointed out that the transient behaviors in mode switching processes and the related control strategy are not the main concern in the research work reported in this paper. The vehicle power requirement is decided by road loads, vehicle speed follow-up, and driver's intention reected by throttle opening, as implemented in the standard ADVISOR models [14]. The vehicle controller will first choose the appropriate operation mode from the five available driving modes based on the power requirement and the battery charge state. Once an operation mode is chosen, the vehicle controller then optimizes the engine operation status for the best fuel efficiency based on the conditions allowed by the selected mode.

3.1 Mode Selection. During vehicle operation in any drive range, the vehicle controller will select one of the five driving modes based on the conditions presented as follows:

Motor Alone Mode:

$$(P_{\text{req}} \leqslant P_{\text{eng-on}}) \cup (v_{\text{req}} \leqslant v_{\text{launch}}) \cup (\text{Soc} \geqslant \text{Soc}_{\text{eng-on}}) \tag{8}$$

Here, P_{req} is the required final drive power and $P_{\text{eng-on}}$ is the threshold to turn on the engine when the required power is above it; v_{req} is the required vehicle speed in the drive range and v_{launch} is a specified low speed value below which the engine mode is turned off; Soc is the current state of battery charge and $\text{Soc}_{\text{eng-on}}$ is the state of battery charge above which the engine power supply is turned off and the motor is turned on to avoid battery overcharge. This logical expression will guarantee that the motor alone mode will be in action when the required power is low, or when the required vehicle velocity is low.

Combined Power Mode:

$$(T_{\text{req}} \geqslant T_{\text{avl-CVT}}) \cap (\text{Soc} \geqslant \text{Soc}_{\text{low}}) \tag{9}$$

Here, T_{req} is the required final drive torque and $T_{\text{avl-CVT}}$ is the engine torque available in the electric CVT mode, which depends on the full throttle engine torque and the proportion determined by Eq. (5); Soc_{low} is the low limit battery state of charge below which the motor is not allowed to provide power and the battery must be

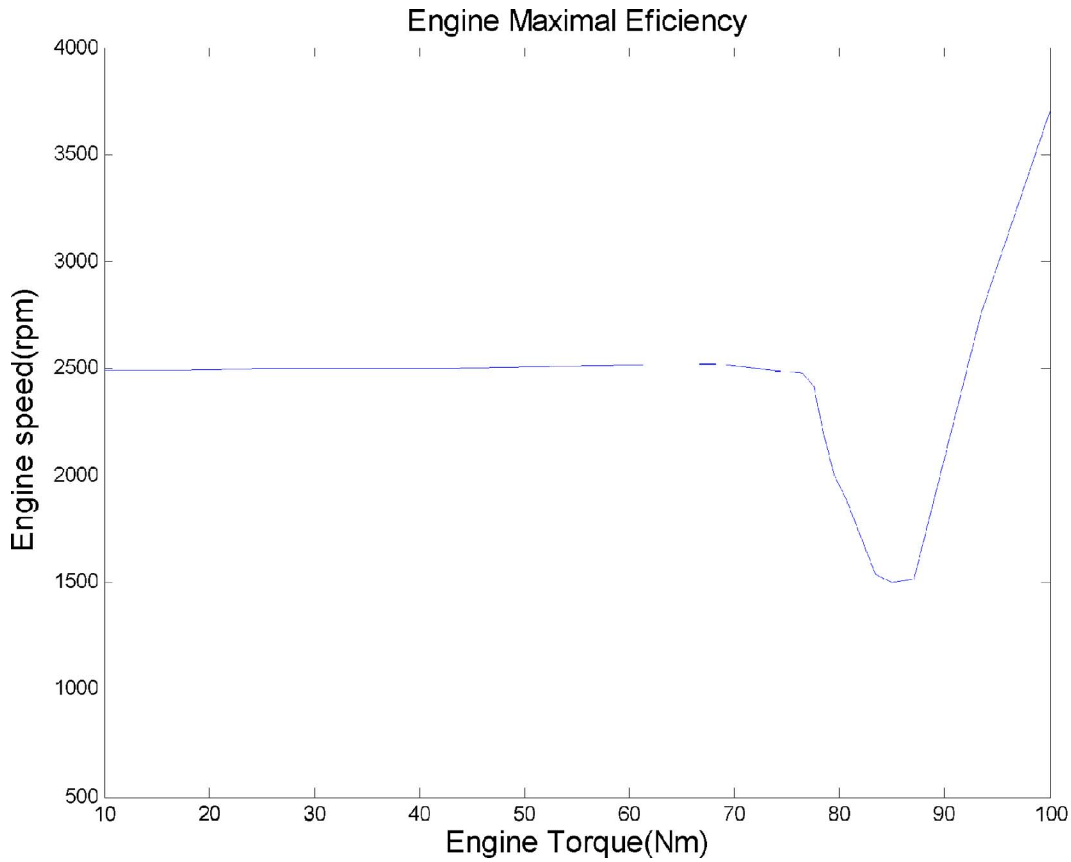

Fig. 4 Maximal engine efficiency curve

charged. This logical expression will guarantee that the combined mode will be in action when the required torque is larger than what the engine can provide alone and the battery state of charge is above the specified low limit.

Engine Alone Mode:

$$(T_{\text{req}} \geqslant T_{\text{avl-CVT}}) \cap (\text{Soc} < \text{Soc}_{\text{low}})$$

Or

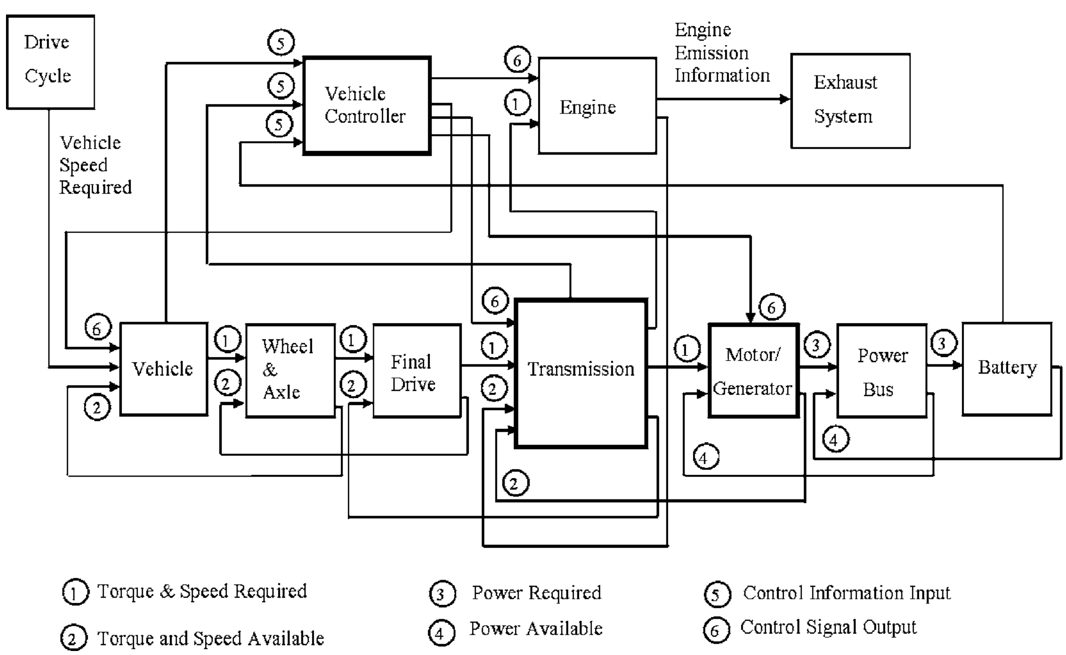

Fig. 5 Configuration of the integrated multimode hybrid vehicle model

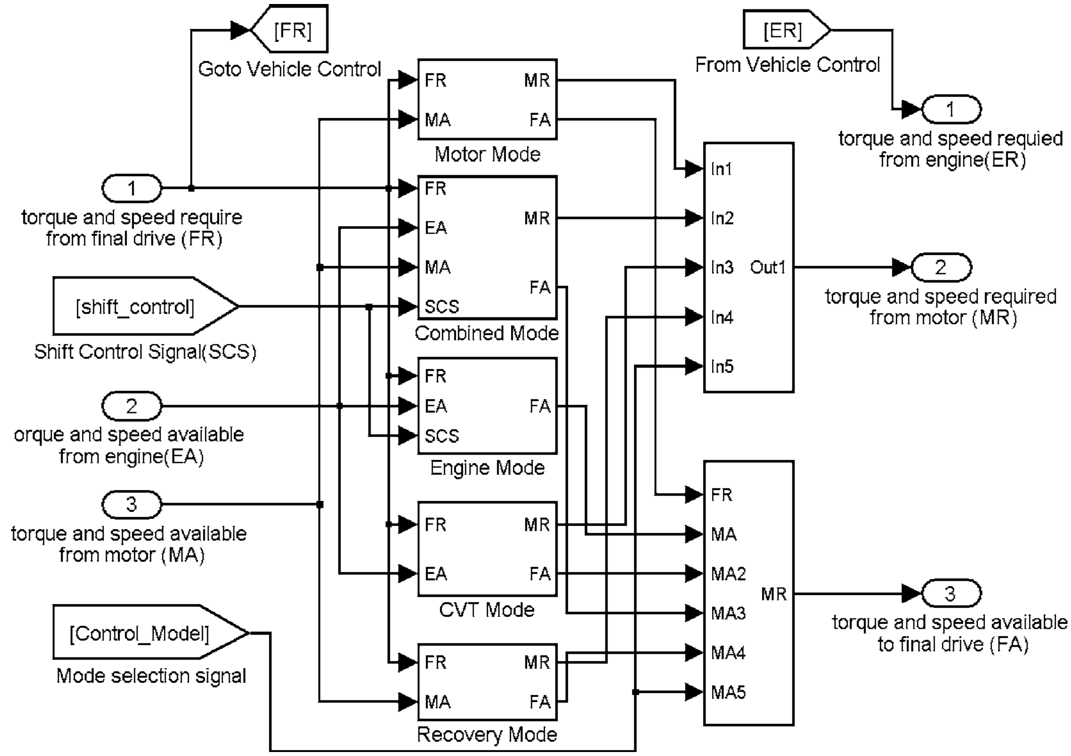

Fig. 6 Transmission model block diagram

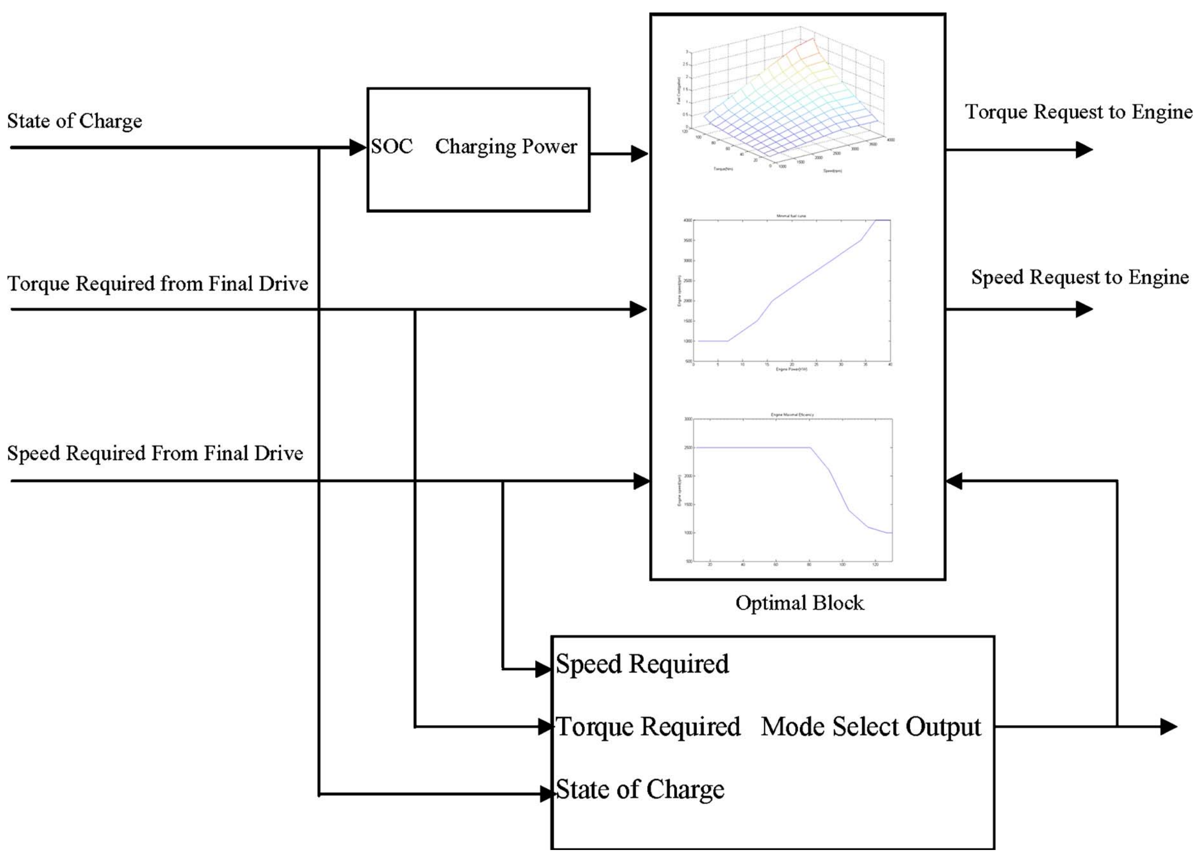

Fig. 7 Operation control block diagram

Table 1 Input data for model simulation

Vehicle data	Mass: 1368 kg, Coefficient of Aerodynamic drag:0.3
Engine data	Maximum power: 43 kW at 4000 rpm, Peak torque: 75 lb ft at 4000 rpm. Engine map range: 0–4000 rpm.
Electric motor	Maximum power: 30 kW, Torque range: from 305.0 Nm to 47.7 Nm (corresponding speed from 0 rpm to 6000 rpm), Inverter/controller efficiencies: 0.95, Motor efficiency: 0.95, Motor inertia: 0.0226 kg m^2
Transmission	Multimode hybrid powertrain: Fourth gear ratio: 0.6, Third gear ratio: 0.8, Second gear ratio: 1.2, First gear ratio: 2.4, Planet gear train parameter (β): 2.6.
Constant Electric Load	700 W
Battery	Peak power: 25 kW, Nominal voltage: 274 V

$$(\mathrm{Soc} > \mathrm{Soc}_{hc}) \cap (v_{\mathrm{req}} > v_{\mathrm{highway}})$$

Or

$$(T_{\mathrm{req}} < T_{\mathrm{avl-CVT}}) \cap (\mathrm{Soc} > \mathrm{Soc}_{\mathrm{high}}) \qquad (10)$$

Here, $\mathrm{Soc}_{\mathrm{high}}$ is the upper limit of the state of charge above which battery charging must be stopped; Soc_{hc} is a state of charge value above $\mathrm{Soc}_{\mathrm{low}}$ and the difference between the two provides a buffer zone to avoid oscillation between the engine alone mode and the electric CVT mode under highway driving conditions. The logical expression above guarantees that the vehicle will be driven on the engine alone mode in highway driving if the battery does not need charging.

Electric CVT Mode:

Table 2 Simulation results compare with Prius in some driving cycle

	Proposed multimode hybrid				Prius hybrid			
		Emissions (g/mile)				Emissions (g/mile)		
Driving Cycle	Fuel economy (mpg)	HC	CO	NO_x	Fuel economy (mpg)	HC	CO	NO_x
CYC_1015_6Prius	36.3	3.521	3.36	0.345	31.3	3.352	3.565	0.532
FTP	48.0	0.823	1.139	0.31	42.9	0.917	1.141	0.282
NYCC	20.3	7.424	7.312	0.854	18.4	6.978	7.639	1.133
ECE	30.8	12.27	11.88	1.084	24.0	11.80	13.05	1.89
UDDS	46.5	1.061	1.507	0.397	42.4	1.20	1.497	0.34
CONSTANT_65	58.6	1.027	1.659	0.463	50.7	1.03	1.741	0.519

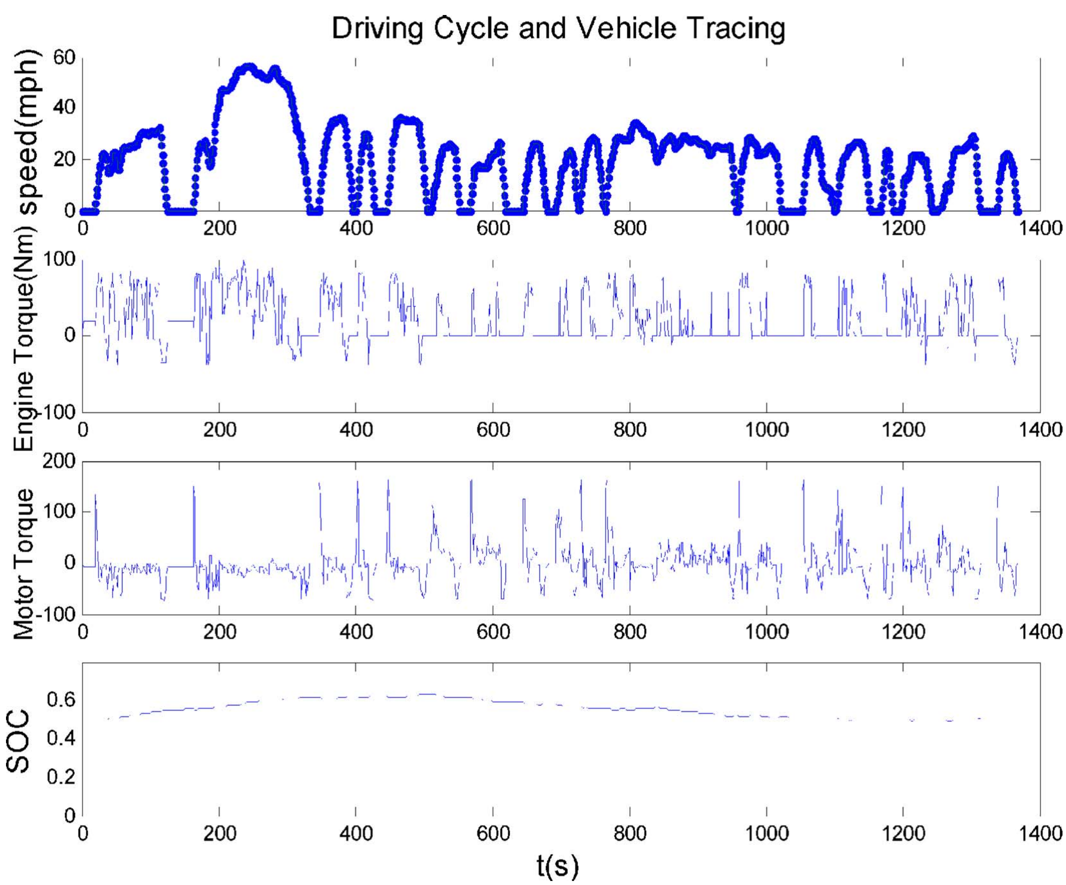

Fig. 8 Simulation results for UDDS driving cycle

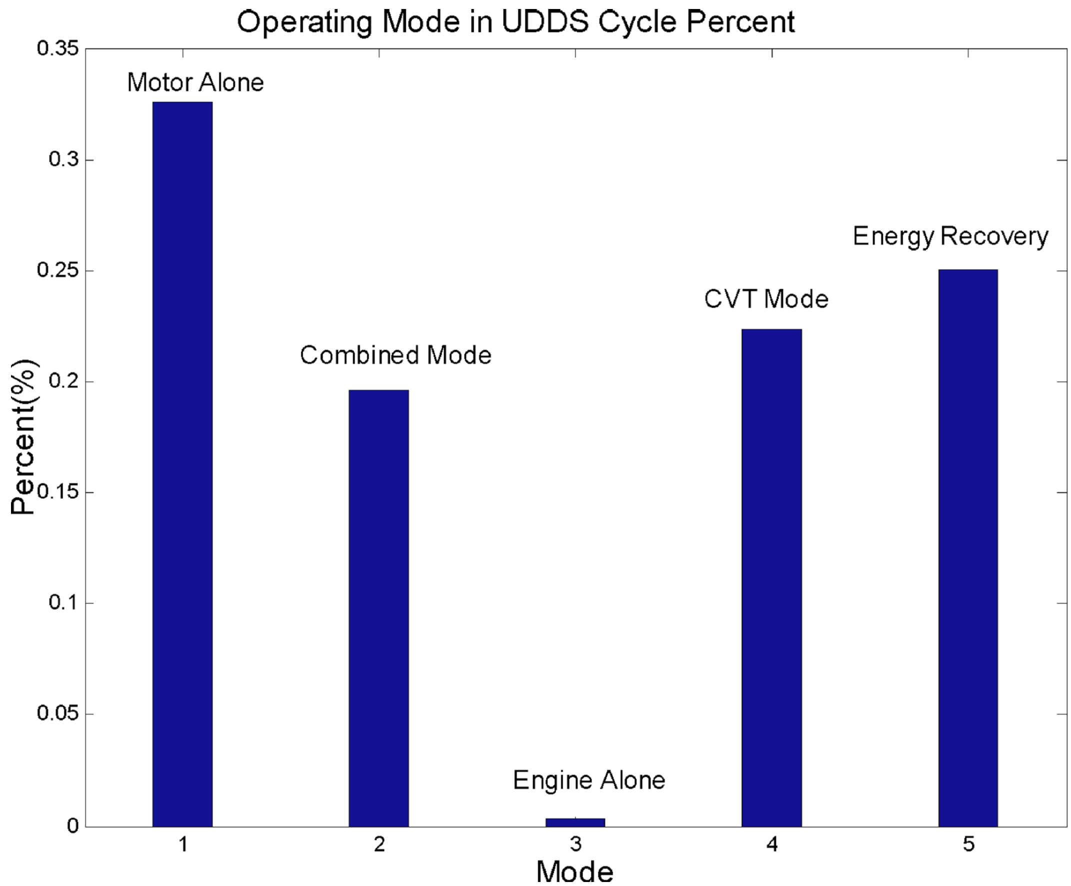

Fig. 9 Contribution of operation modes in UDDS cycle

$$(T_{\text{req}} < T_{\text{avl-CVT}}) \cap (\text{Soc} \leqslant \text{Soc}_{\text{high}})$$

Or

$$(v_{\text{req}} > v_{\text{highway}}) \cap (\text{Soc} < \text{Soc}_{hc}) \tag{11}$$

Here, v_{highway} is the speed above which the vehicle enters highway driving condition. As controlled by the above logical expression, the electric CVT mode will be in action when the required torque is smaller than what the engine can provide in CVT mode and the battery can be charged, or when the required vehicle speed is above the pre-defined highway speed and the battery needs to be charged.

Energy Recovery Mode:

$$P_{\text{req}} < 0 \tag{12}$$

The required power P_{req} becomes negative when the vehicle is being braked or when the vehicle is coasting. The energy recovery mode will be in action whenever the required power is negative, as controlled by the inequality above.

In addition to the conditions described above which are specific to the proposed multi-mode hybrid system and directly associated with the powertrain output requirements and battery status, the vehicle controller also factors in other vehicle operation conditions implemented in the driver controller of the standard ADVISOR models.

3.2 Operation Optimization. The focus of the mode operation control is to optimize the engine operations in each of the selected driving modes in terms of the fuel consumption. The engine mapping data available in the ADVISOR model for Toyota Prius is used in the research for engine operation control and for the comparative studies between the proposed hybrid system and the benchmark. The operation control strategies are constructed based on the flexibility and constraints of each of the three engine related modes, respectively, as described in the following.

Combined Power Mode: In the combined power mode, the required engine speed is coupled to the vehicle speed by the ratios of the lay-shaft gears as shown in Eq. (2). The variables available for engine operation control are the four stepped lay-shaft gear ratios and the assisting power to be provided by the motor. For a given engine, the fuel consumption F_f is a function of the engine output torque T_e and angular velocity ω_e, as shown in Fig. 2 for the Toyota Prius engine, i.e.

$$F_f = f(T_e, \omega_e) = f\left(\frac{P_e}{\omega_e}, \omega_e\right) \tag{13}$$

The function F_f is available for a given engine in terms of a two-dimensional look-up table, and if the engine power P_e is specified, it becomes a function of the engine angular velocity alone. There exists a minimal fuel consumption for a given engine power within the operation range at a certain angular velocity when the following condition is observed

$$\frac{\partial}{\partial \omega_e} f\left(\frac{P_e}{\omega_e}, \omega_e\right) = 0 \tag{14}$$

The above equation is solved numerically for each given P_e and the solutions lead to a function curve between P_e and ω_e. The curve shown in Fig. 3 is obtained using the Prius engine map data. Along this curve, the engine fuel consumption for a given power requirement is the minimal. In the combined power mode, the power to be provided by the engine is first selected based on the road load requirement. The corresponding engine angular velocity for minimal fuel consumption is then selected from Fig. 3. One of

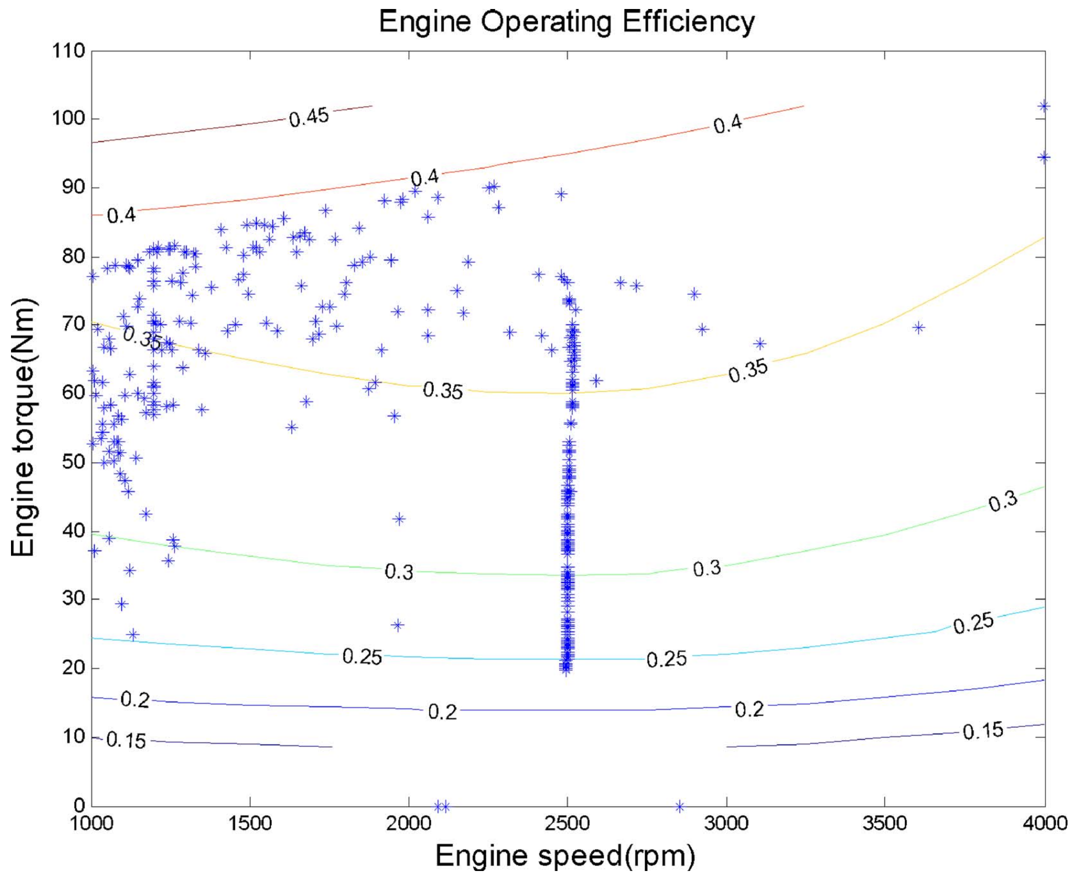

Fig. 10 Engine operation range in UDDS driving cycle

the four lay-shaft gears is chosen to provide a gear ratio that is the closest to the ratio between the angular velocities of the engine and the output shaft. The motor is controlled to operate at the angular velocity and output torque determined by Eq. (2) to provide the required torque assistance.

Engine Alone Mode: In the engine alone mode, the engine is the only power source and its output power is determined by the output shaft torque and angular velocity. For a given power required at the output shaft, the fuel consumption is the lowest if the engine operates at the lowest speed and at the highest torque that satisfy the power requirement. Therefore, the four gears are chosen in descending priority in terms of the gear ratios, with the lowest gear ratio always chosen as the first priority, so that the engine runs at the lowest angular velocity in the normal operation range. The strategy leads to the lowest fuel consumption and is similar to that adopted by a skilled driver operating a manual transmission vehicle.

Electric CVT Mode: The electric CVT mode offers the flexibility in the control of engine operation status. For a given driving condition, the engine torque and generator torque are determined by Eq. (5), but there exists an extra degree of freedom for the angular velocities as shown by Eq. (4). In the engine fuel map, there exists a maximal engine efficiency curve along which the following equations are observed

$$F_f = f(T_e, \omega_e)$$

$$\frac{\partial}{\partial \omega_e} f(T_e, \omega_e) = 0 \tag{15}$$

Here, function $f(T_e, \omega_e)$ represents the engine fuel consumption rate at a given operation status and is usually available in terms of two-dimensional lock up table. The maximal engine efficiency line is determined by numerical interpolation of the engine map data or can be directly selected from the engine fuel consumption contour plot. This line is shown in Fig. 4 for the Prius engine map. In the electric CVT mode, the torque required from the engine is calculated according to the road load. The engine angular velocity is then interpolated at this torque from the line shown in Fig. 4 for the lowest fuel consumption. The motor angular velocity is controlled to follow the engine according to Eq. (4).

4 Model Integration

The main task in the model development is focused on the transmission and vehicle controllers on the subsystem level. These models are then integrated into the vehicle system model according to the hybrid system structure, the power flows, and the control logic as shown in Fig. 5. The integrated hybrid vehicle model is established through modification and reconfiguration on the Toyota Prius model available on the ADVISOR platform. As highlighted in Fig. 5, the model blocks for the transmission, the motor/generator and the vehicle controller are modified to fit the proposed configuration. The power flow and control signals are also shown in Fig. 5. The control information input includes the vehicle speed, torque and speed required, and the battery state of charge. The control signal output includes the engine operation command, hybrid mode selection and optimization, and motor/generator operation command.

The Simulink model for the hybrid transmission is illustrated in Fig. 6. This model implements the kinematical relations for all the operation modes described in Sec. 2 and provides the interface among the transmission, the power sources, the final drive, and the vehicle controller. The power flow paths and the relations on the transmission operation parameters are imbedded in the five driving modes, respectively.

The vehicle controller contains four parts: operation controller, motor controller, brake controller, and traction controller. The

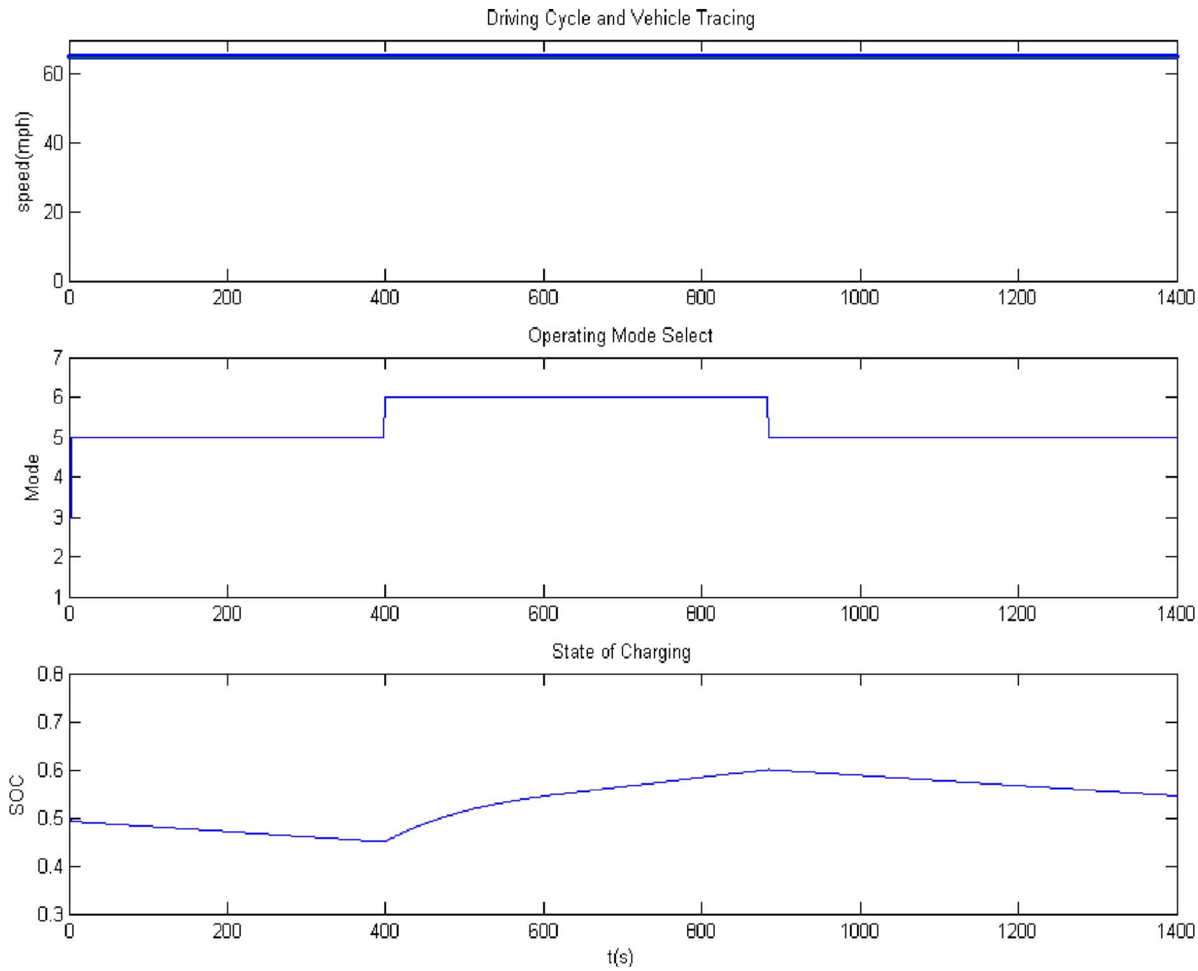

Fig. 11 Operation modes and battery status in highway cruising

brake and traction control models are the same as the standard models available on the ADVISOR platform. The motor controller of the ADVISOR Prius model is modified and adopted to the proposed hybrid configuration. The operation controller implements the strategy described in Sec. 3 and coordinates the operations of the power sources, the transmission, and other powertrain subsystems. The structure of the controller and the interactive data flows are shown in Fig. 7. As shown in the figure, the operation controller has two functions: mode selection and operation optimization for the selected mode. The optimization block contains the optimization strategies described in Sec. 3 for the combined mode, engine alone mode, and the electric CVT mode. The mode selection block implements the mode selection conditions described by the logical expressions (8)–(12). The input signals are the battery state of charge (S_{oc}), torque and speed required on the final drive shaft. Upon receiving the input signals, the controller first selects the appropriate hybrid mode and then optimizes the engine and motor operations for the best fuel economy.

5 Model Simulation and Analysis

The vehicle used for the model simulation is the same as the Toyota Prius except for the multi-mode transmission. The engine map data, vehicle dimension, and weight data are exactly the same as that for Toyota Prius. The vehicle data and the transmission design parameters are shown in Table 1. The efficiency of the powertrain is assumed to be a constant for simplicity and is the same as the ADVISOR Prius Model. The model has been applied for the performance simulation under various driving conditions, including five standard drive cycles and a constant speed highway cruising range. In the performance simulation, the vehicle operation is optimized in terms of the fuel consumption alone and the low emission levels are the results of low fuel consumption. The performance data from the model simulation are compared against that of the benchmark, the Toyota Prius, in each category and drive range under the same condition.

The integrated vehicle model has been used to simulate the performance on fuel economy and emissions for six driving cycles listed in Table 2. The driving cycle "CYC_1015_6 Prius" is designed for dyno test of Prius model according to the Japanese standard. FTP stands for Federal Test Procedure and represents a driving cycle used by the US Environmental Protect Agency (EPA) for emissions certification of passenger vehicles in the United States. NYCC is a driving cycle that mimics the traffic patterns in the city of New York. ECE stands for the driving cycle emulating urban driving in European countries and UDDS is the US EPA Urban Dynamometer Driving Schedule that simulates city driving in the United States. As an example, the simulation results for the UDDS cycle are shown in Fig. 8. The speed follow-up, the engine torque, the motor torque and battery state of charge are shown in the figure, respectively, for the whole range. The battery state of charge is assumed to be 0.5 at the start of the drive range. It fluctuates during the simulation process and remains to be about 0.5 at the end of the cycle. All of the five driving modes are actuated in the drive range and the percent of time of each mode in the range is shown in Fig. 9. As shown in the figure, the motor alone mode and energy recovery mode contribute about 57% of the total driving time because of the urban driving nature of UDDS range. The electric CVT mode and the combined mode also play an important role, contributing about 23% and 20% of the driving time, respectively. The contribution of engine alone mode is insignificant in this drive range because of the constraints on the vehicle speed.

The engine operation status in the UDDS range is shown in Fig. 10. As shown in the figure, engine operation points in the electric CVT mode have two concentrations. One is along the vertical line at 2500 rpm while the engine torque ranges from about 20 Nm to 76 Nm, the other corresponds to a torque range between 76 Nm and 92 Nm and a speed range of 1500 rpm and 2500 rpm. This result is in full agreement with the control strategy based on the maximal engine efficiency curve shown in Fig. 4. In combined mode, the operation control strategy is based on the minimal fuel consumption curve as shown in Fig. 3. The engine operation points in the combined mode can be roughly grouped into two areas, one with the torque larger than 67 Nm and engine speed higher than 1000 rpm (corresponding to engine power above the 7 kW point in Fig. 3), and the other with the engine torque below 67 Nm and speed near 1000 rpm. Since the gear pair ratios do not provide the exact match for the required engine speed, the actual engine speed is slightly different from the ideal value determined from Fig. 3. The isolated points shown in Fig. 10 near 2000 rpm indicate the status of the engine alone mode.

The engine alone mode has the highest priority for mode selection in highway cruising. Whenever the vehicle is operated, there always exists a constant battery power consumption to power the auxiliaries and thus the battery needs to be charged when the state of charge is lower than specified threshold. Figure 11 shows the alteration of the engine alone mode and the electric CVT mode when the vehicle cruises at a constant speed of 65 mph. It is clearly demonstrated that the electric CVT charging mode is needed to keep the battery state of charge above the usable level.

The simulation results on fuel economy and emissions of the multi-mode hybrid vehicle are compared with that obtained from the ADVISOR simulation model for Toyota Prius. The comparison on each of the drive ranges is shown in Table 2. In all of six driving cycles, the initial battery state of charge is assumed to be 0.5 in order to eliminate the effects of pre-stored battery energy on the fuel economy performance. The state of charge at the end of simulation remains to be about 0.5 for all of the six driving cycles, as shown in Fig. 8 for the UDDS cycle. The proposed multi-mode hybrid vehicle shows significant improvement in fuel economy for all the drive ranges, from 10% for the NYCC cycle (New York City Cycle) to 28% for the UDDS.

6 Conclusion

This paper presented a multi-mode hybrid configuration and the systematic model for the operation control and performance simulation of a vehicle equipped with this hybrid design. The hybrid configuration proposed in the paper realizes six operation modes based on an automated manual transmission platform and provides the flexibility for vehicle operation optimization under all driving conditions. Component level models for the hybrid configuration are developed in Simulink environment and integrated to the overall vehicle model using the Simulink/Advisor platform. The vehicle control strategy is established with the objective to optimize vehicle fuel economy performance. The performance of the proposed hybrid vehicle system is studied based on model simulation under various driving cycles. The simulation analysis verifies the capabilities and the advantages of the proposed hybrid system and the associated operation control. Furthermore, the multimode hybrid is benchmarked with a current hybrid vehicle using the existing Advisor model. The proposed hybrid configuration shows substantial improvements over the benchmark in fuel economy under the same conditions and is validated as a feasible hybrid design based on model simulation.

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