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Parameters Design and Economy Study of an Electric Vehicle with Powertrain Systems in Front and Rear Axle

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ABSTRACT

To achieve higher economy of the original driving scheme with single motor and settled gear ratio, new configurations with different powertrain systems in front and rear axle were designed. Firstly, according to the power and torque required by a micro electric vehicle (mEV) in various drive cycles, the parameters of a small and high power motor were determined. Secondly, for scheme I with dual motor and one-speed gearbox, based on genetic algorithm (GA), theoptimal transmission and torque distribution were confirmed. Owing to the use of single-motor-drive mode (SMDM), the simulated energy consumption decreases by 4.01% compared with the original scheme, but the motor efficiency is still relatively low under low-speed conditions (0-1000r/min) due to the fixed transmission. Then, to solve this problem, scheme using two-speed gearbox to substitute the original one in rear axle was proposed. For this scheme, first of all, the explicit relation of energy consumption and the two-speed ratios was established based on response surface methodology (RSM); then hill climbing method was used to search the best ratios. Finally, economy performances of different schemes were discussed in simulation model and energy consumption of this scheme decreases by 7.55% compared with the original one.

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NOMENCLATURE

mEV	Micro electric vehicle	LHD	Latin hypercube design
GA	Genetic algorithm	VCU	Vehicle control unit
SMDM	Single-motor-drive mode	MCU	Motor control unit
RSM	Response surface methodology	SOC	State of charge
FRMDEV	Front-and-rear-motor-drive EV	BMS	Battery management system
PMSM	Permanent magnet synchronous motor	SM	Synchronous motor
IM	Induction motor	FT	Front transmission
SSMDM	Single-small-motor-drive mode	RT	Rear transmission

1. INTRODUCTION

Confronting the stress caused by fossil energy crisis and environmental pollution, electric vehicles, as aneffective method to reduce the use of fossil fuels and pollution, have become the strategic development direction in the world automobile industry [1]. Currently, owing to simple structure and low cost, driving schemes with single motor and settled or two-speed gearbox are most commonly used in mEV [2, 3], and great efforts have been made to improve the system efficiency [4, 5]. However, limited by the single motor configuration, higher power motor is required to maintain the dynamic performance of mEV, which causes negative impact on the mutual relation of the motor's high efficiency region and the common operating conditions.

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Configurations with single motor and multi-gearshifting transmissions or four-in-wheel motors, as high-efficiency driving schemes, have attracted a great deal of attention. However, for mEV, limited by the layout space, the former scheme is infeasible as a result of the complex powertrain system [6, 7]. For the configuration with four-in-wheel motors, as the system reliability and stability are still unsettled [8], it is still quite difficult to popularize engineering application of this technology.

Configuration with distributed powertrain systems in front and rear axle, as shown in Figure 1, shows great advantages on steering stability and system reliability, owing to the mechanical differentials in front and rear axle [9, 10]. Based on this scheme, plentiful achievements have been obtained on driving/braking dynamics, cornering and failsafe control [11-13]. Nobuyoshi Mutoh, et al. [11] have proposed torque control method, taking both slip rate and lateral stability to improve vehicle dynamic performance and running safety. However, currently, parameters design of the front-and-rear-motor-drive EV (FRMDEV), which has significant influence on vehicle economy performance, is still lack of systematic study. This paper therefore proposes mathematical methods for the powertrain system design of the FRMDEV.

2. DIFFERENTIATED DESIGN OF THE FRMDEV

As the FRMD configuration shown in Figure 1, the VCU is the control centre of the FRMDEV, and CAN bus is utilized to achieve the communications between VCU and MCU1, MCU2, BMS. By sampling the signals from the drive or brake pedal, BMS and MCU, the VCU generates the control commands of torque based on the torque distribution strategy, and then transmits them to the MCU1 and MCU2.

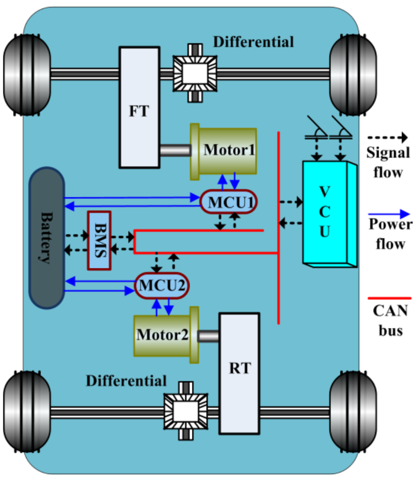


Figure 1. Topology of the FRMDEV

After receiving the control commands from VCU, the MCU1 and MCU2 control the front and rear motor to output the required torques in positive or negative turn. For the BMS, estimating SOC and then sending it to the VCU to guide torque distribution are the two key tasks.

Owing to the distributed driving configuration, there is a great deal of freedom for the differentiated design. On the side of the transmission system, the front and rear axle can be equipped with fixed or variable transmission, which will provide conveniences for the optimization of driving economy. On the other side of the motor system, various motor combinations can be achieved, which will be helpful for making full use of the combination advantages of different types of motor [14].

2. 1. Cycle Characteristics of the mEV

The characteristics of the vehicle velocity, the required driving power and torque under various cycles are important references for powertrain system design [15]. In this study, J1015, NEDC and HWFET drive cycles are chosen, which can characterize urban, suburban and high-way running conditions, respectively. Table 1 presents the detailed parameters of the original mEV, which is driven by a single-front motor with fixed transmission (the original driving scheme). Based on the parameters shown in Table 1, according Formula (1), the power and torque at wheels required by the mEV under various cycles can be confirmed, as shown in Table 2.

$$T_{r\_j}=\left(mgf+{CAV_j^2}/{21.15}+\delta m{\left(V_j-V_{j-1}\right)}/{\Delta t}\right)r$$
$$P_{r\_j}={T_{r\_j}V_j}/{3600r} \quad (1)$$

where, Δt is the sampling period, T_{r_j} and P_{r_j} are the required torqueat wheels and driving power under number j point, f is the coefficient of rolling resistance, A is the section-surface area, C is air resistance coefficient, m is the mass of mEV, δ is the rotational inertia coefficient, r is wheel radius, G is the weight of mEV,V_j and V_{j-1} are vehicle velocity under j and j-1point.

TABLE 1. Parameters of a mEV

Items	Parameters
Complete vehicle kerb mass (kg)	1000
Frontal area (m^2)	2.2
Wheel radius (m)	0.265
Rolling resistance coefficient	0.0165
Maximum velocity (km.h^{-1})	110
Maximum climbable gradient (%)	≥20
0-50km/h acceleration time (s)	≤6
Endurance mileage (km)	≥160

1) For the urban running conditions as shown in Table 2, low velocity (0-40km/h) and small driving power (0-5kW) are the two notable features. Accordingly, for the dual motor configuration, it is essential to design a small power drive-train system to match the vehicular running characteristics under urban cycle. Furthermore, the driving characteristics with low velocity and high load (torque required at wheels ＞250N.m) is another important feature of the mEV; 2) for the urban and suburban running conditions, there are two peak values of the driving power under the low and middle velocity. In addition, when the mEV operates under suburban conditions, as a result of the relatively stationary operations, the centralized location for the required torque at wheels and driving power are within 50-100N.m and 0-5kW. Therefore, for the dual motor driving scheme, reasonable powertrain system should be able to achieve single-small or single-high power motor drive mode; 3) for the high-way running conditions, the remarkable operating characteristics of the mEV are high velocity and middle load. Consequently, for the dual motor driving scheme, it is obviously important to design a reasonable high power drive-train system to ensure vehicle economy under high-way driving conditions.

2. 2. Differentiated Design Scheme Given the above analysis of the operating characteristics of the mEV under various driving cycles, a differentiated design scheme for the FRMDEV is proposed as follow:

1) For the rear powertrain system, small power PMSM and bigger ratio gearbox are preferred. First of all, a small power motor will help improve the driving efficiency under urban running conditions, where the required driving power of the mEV is relatively small. Secondly, the motor operating efficiency under the common low-velocity conditions of urban cycle will be improved, owing to the distribution feature of efficiency in PMSM, where the high-efficiency scope mainly covers low to middle speed and load. Furthermore, gearbox with bigger ratio will reduce the frequency of over-load conditions under the frequent accelerating conditions in urban cycle.

2) As for the front powertrain system, high power IM is preferred. Firstly, if the rear powertrain system is equipped with a small power motor, high power motor is indispensable to maintaining the dynamic performance of the mEV. Furthermore, high power motor used in front axle is helpful for the optimization of braking energy recovery due to the transfer of more axial load to front axle. Moreover, the high-efficiency region of IM mainly concentrates in the middle to high speed operations [16], which will be helpful for the improvement of driving efficiency under the common middle to high velocity conditions in suburban cycle.

3. SCHEME WITH DUAL MOTOR AND ONE-SPEED GEARBOX

In this section, design methods for the dual-motor-drive scheme with relatively simple structure are proposed. For this configuration, a small power PMSM and a high power IM are equipped in the rear and front powertrain system with one-speed gearboxes.

3. 1. Design of the Rear Powertrain System According to Table 2, for the small power PMSM, the reasonable nominal power and velocity should be 5-7kW and 30-40km/h. Here, the nominal velocity represents the vehicle velocity when motor operates under nominal speed.

TABLE 2. Frequency density of the power and torque required by the mEV under various cycles

		J1015			NEDC			HWFET		
		Vehicle speed ($km.h^{-1}$)			Vehicle speed ($km.h^{-1}$)			Vehicle speed ($km.h^{-1}$)		
		0-40	40-70	＞70	0-40	40-70	＞70	0-40	40-70	＞70
Power required (kW)	**0-5**	0.3956	0.1591	0.0021	0.365	0.3003	0	0.0028	0.0166	0.0798
	5-10	0.12	0.072	0.0438	0.0914	0.0199	0.0409	0.0021	0.0522	0.2923
	10-15	0.0897	0.0933	0.0069	0.0281	0.0738	0.0569	0.0065	0.0392	0.3798
	15-20	0.0017	0.0157	0	0	0	0.0218	0.002	0.0161	0.0851
	＞20	0	0	0	0	0	0.0018	0	0.0072	0.0174
Torque required at wheels (N.m)	**0-50**	0.2861	0.0224	0.0012	0.1679	0.0055	0	0.0021	0.0087	0.0548
	50-100	0.222	0.1642	0.0411	0.3141	0.2824	0.0013	0	0.0223	0.2149
	100-150	0.0249	0.025	0.0041	0.0094	0.0027	0.0718	0	0.0305	0.4008
	150-200	0.04	0.1341	0.0059	0.0037	0.028	0.0287	0	0.0316	0.1372
	＞200	0.335	0.0305	0	0.1868	0.059	0.0146	0.013	0.0382	0.0476

Note: The frequency density is the ratio of the operation points subject to the given constrains to the total operation points.

One thing should be noted that, in China, the small power motors commonly found in market are designed for low-speed mEV (maximum velocity is less than 60km/h), which are incapable for this scheme (the maximum design velocity is 110km/h).

For example, for the mEV described in Table 1, if the E1148 motor (nominal and peak speed are 3200 and 5300r/min, respectively) is chosen as the drive motor for the rear power system, the transmission of the rear axle should not greater than 4.8 to maintain the mEV working under the maximum design speed. Accordingly, the nominal velocity of the rear powertrain system is greater than 66.6km/h, which will lead the high-efficiency region of the PMSM away from the frequently low-speed operations in urban cycle. Moreover, if the transmission is small, the rear motor may frequently operate under the overload operations.

Given the above analysis, as shown in Table 3, a higher power but more suitable PMSM is chosen. To obtain a low nominal velocity, the rear transmission is set to 6.35 based on Formula (2). Accordingly, the nominal torque at rear wheels is 195N.m, which can cover the commonly operations in urban cycle shown in Table 2.

$$i \le 0.377 r n_{rm\_max} / V_{max} \tag{2}$$

where, V_{max} is the maximum design velocity of the mEV,i is the rear transmission, n_{rm_max} represents the maximum speed of the rear motor.

3. 2. Design of the Front Powertrain System

3. 2. 1. Design of the High Power Motor To maintain the dynamic performance of the mEV, the high powertrain system should meet the followings:

1) To maintain the normal operation of the mEV under the maximum design speed, according to Formula (3), the total nominal power provided by the front and rear powertrain systems should be greater than 20.2kW, the nominal power of the IM should not be less than 10.2kW.

$$P_{nom} \ge V_{max} / 3600\eta_T \left(Gf + CAV_{max}^2 / 21.15 \right) \tag{3}$$

where, P_{nom} is the total nominal power provided by the front and rear powertrain systems, η_T represents the efficiency of the transmission system.

2) To ensure the overload operation capacity of the mEV, the peak power output from the front and rear powertrain system should satisfy Formula (4). According to the parameters shown in Tables 1 and 3, the peak power of the IM should be greater than 24.18kW. Taking the redundancy design into consideration, the peak power is set within 25-30kW.

$$\begin{aligned} & P_{max} \ge max(P_{s\_max}, P_{a\_max}) \\ & P_{s\_max} = V / 3600\eta_T \left(Gfcos\alpha + Gsin\alpha + CAV^2 / 21.15 \right) \\ & P_{a\_max} = V / 3600\eta_T \left(Gf + \frac{CAV^2}{21.15} + \delta m \, dV / dt \right) \end{aligned} \tag{4}$$

where, P_{s_max} and P_{a_max} are the driving power required by the MEV on the maximum design gradient and under the maximum design acceleration respectively, α is the maximum design gradient.

3) To ensure the slope climbing ability, the pear driving torque at the front and rear wheels should meet Formula (5). Based on the given parameters, the peak torque at front wheels should be greater than 288.7N.m.

$$T_{max} \ge \left(Gfcos\alpha + Gsin\alpha + CAV^2 / 21.15 \right) r \tag{5}$$

where, T_{max} is the pear torque at the front and rear wheels.

Given the above analysis, a well-developed IM is selected as the drive motor for the front powertrain system, as shown in Table 3.

3. 2. 2. Design of the Front Transmission Figure 2 shows the optimization route for the front transmission based on GA [17, 18]. To maintain the normal operation of the mEV on the maximum design gradient and under the maximum design velocity, the range of the genetic population is constraint by 2.618-6.35. Furthermore, as the front powertrain system is mainly designed for the middle to high velocity and load conditions, according to Table 2, cycle conditions of J1015 within 45-70km/h and 10-20kW are chosen as the sampling points.

Fitness evaluation is the key factor for the optimization of the GA procedure. For the restricted operating point used in the GA procedure, the torque and speed of the number t individual are:

$$\begin{cases} n_{j\_t} = V_j i_{f\_t} / 0.377r \\ T_{j\_t} = 9550 P_{r\_j} / i_{f\_t} \eta_T n_{j\_t} \end{cases} \tag{6}$$

TABLE 3. Parameters of the dual motors

Items	Small power motor	High power motor
Nominal power	10kW	15kW
Pear power	20 kW	30 kW
Nominal speed	3200 r/min	3000 r/min
Peak speed	7000r/min	6500 r/min
Peak torque	65	95.5
Motor type	PMSM	IM

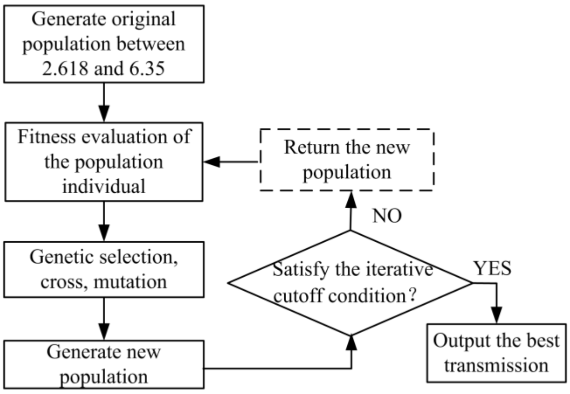


Figure 2. Route for the optimization of the front transmission

where, i_{f_t} is the number t individual in the population, n_{j_t} and T_{j_t} are the motor speed and torque of the number t individual, P_{j_t} is the battery power consumption of the number t individual under number j sampling point.

According to the measured efficiency of the IM and battery, the battery power consumption of the number t individual is as follow:

$$\begin{cases} P_{r\_j\_t}=T_{j\_t}n_{j\_t} \Big/ \eta_{m\_j\_t}\eta_{b\_j\_t}9550 & T_{r\_j\_t}\leqslant 0.9T_{b\_max} \\ P_{r\_j\_t}=\lambda_{j\_t} & T_{r\_j\_t}>0.9T_{b\_max} \end{cases} \quad (7)$$

where, T_{b_max} is the maximum torque provided by the battery, $\eta_{m\_j\_t}$ and $\eta_{b\_j\_t}$ are the measured efficiency of the motor and battery under number j sampling point, λ_{j_t} is the penalty coefficient used.

Accordingly, the fitness value of the number t individual under the chosen operating conditions is:

$$min\ \ J=\sum_{j=1}^{M}P_{r\_j\_t}\Delta t \quad (8)$$

where, M is the number of the sampling points.

Finally, based on the GA procedure, the optimum transmission (i=5.4832) is obtained after about 60 times iteration.

3. 3. Optimization of the Torque Distribution

Under any given operating condition, as the optimum torque distribution between the front and rear motor, which has significant effect on driving efficiency of the FRMDEV, is still unknown, an off-line GA optimization procedure is designed. Equation (9) presents the optimization variable, which is the ratio of torque distributed to the rear motor (T_{rm}) to the total required torque (T_{to}). Consequently, χ=0 or χ=1 means the mEV operates under the single-front or single-rear motor mode.

$$\chi=T_{rm}\Big/T_{to} \quad (9)$$

Them, to calculate the fitness value of each χ,the energy consumption of the rear motor for the number t individual under number j sampling point are obtained based on formula 10, where overload penalty coefficient (E_{rm-j-t}) is adopted to prevent the long-time overload operations of the rear motor.

$$E_{rm\_j\_t}=\phi_{j\_t}\chi_{j\_t}T_{to\_j}n_{rm\_j}\Delta t \Big/ 9550\eta_{rm\_j\_t}\eta_{b\_j\_t} \quad (10)$$

where, χ_{j_t} is the number t individual in the population,n_{rm_j} and $\eta_{rm\_j\_t}$ are the rear motor speed and efficiency, $\emptyset_{j\_t}$ is the penalty coefficient, which is:

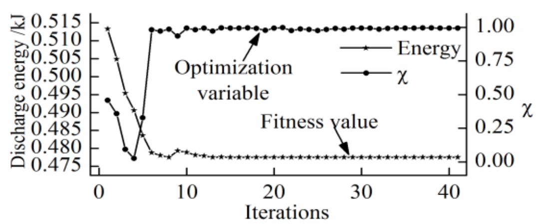
$$\begin{cases} \phi_{j\_t}=1+^{K\left(\chi_{j\_t}T_{to\_j}-T_{nom}\right)}\Big/_{T_{nom}} & \chi_{j\_t}T_{to\_j}\geqslant T_{nom} \\ \phi_{j\_t}=1 & \chi_{j\_t}T_{to\_j}<T_{nom} \end{cases} \quad (11)$$

where, K is a constant value that greater than 1.

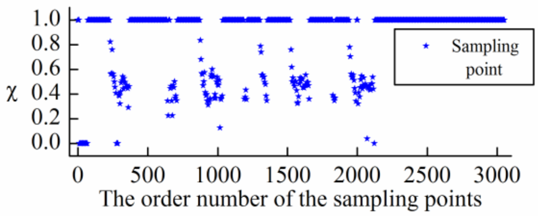
Given the above analysis, the fitness value of the number t individual is:

$$min\ J=\sum_{j=1}^{M}\left(E_{rm\_j\_t}+^{\left(1-\chi_{j\_t}\right)T_{to\_j}n_{fm\_j}\Delta t}\Big/_{9550\eta_{fm\_j\_t}\eta_{b\_j\_t}}\right) \quad (12)$$

where, n_{fm_j} and $\eta_{fm\_j\_t}$ represent the front motor speed and efficiency. The optimization results of χ based on the off-line GA are shown in Figure 3. Taking the number j sampling point (V=21.19 km/h^{-1}, T_r=199.25N.m) for example, the optimum torque distribution coefficient is obtained after about 12 times of iteration as shown in Figure 3(a). Furthermore, as optimization results shown in Figure 3(b), the single-small-PMSM-drive mode (χ=1) accounts for most of the urban driving operations (J1015 cycle), while there is only a small part of urban driving operations where the single-high-IM-drive mode (χ=0) is used.



(a) The number j sampling point



(b) The sampling points of the J1015 driving cycle

Figure 3. Optimization results of the torque distribution

Due to the common acceleration conditions of urban cycle, dual-motor-drive mode is often used to maintain the vehicle dynamics.

3. 4. Economy Analysis of this Scheme To analyze the economy performance of the mEV driven by dual motor with fixed transmission, a simulation model as shown in Figure 4 is established.

The driver-demand module is designed to calculate the torque required by the driver using PID control method. In the powertrain controller, the torque distribution between the front and rear motor is designed based on the results of the off-line GA. The command module is used to transform the actual motor torque and wheel braking force into the dimensionless control signals. In the powertrain model, the sub-models of battery, motor and vehicle are designed to calculate the states of them, which are then transferred to the powertrain controller through the simulated bus. Moreover, simulation results show that the differences between the desired velocity of the driver and the actual velocity of the powertrain model are quite small.

3. 4. 1. Advantages of this Scheme As the mEV mainly designed for the use in urban cycle, Table 4 presents the economy performance of different driving scheme under J1015 drive cycle. The original scheme is a single-front-PMSM-drive configuration with one-speed gearbox. Specifically, the nominal power of the PMSM is 25kW, the transmission is 6. Compared with the original scheme, the configuration Ⅰ driven by dual motor with one-speed gearbox shows great advantages on driving economy, as shown in Table 4.

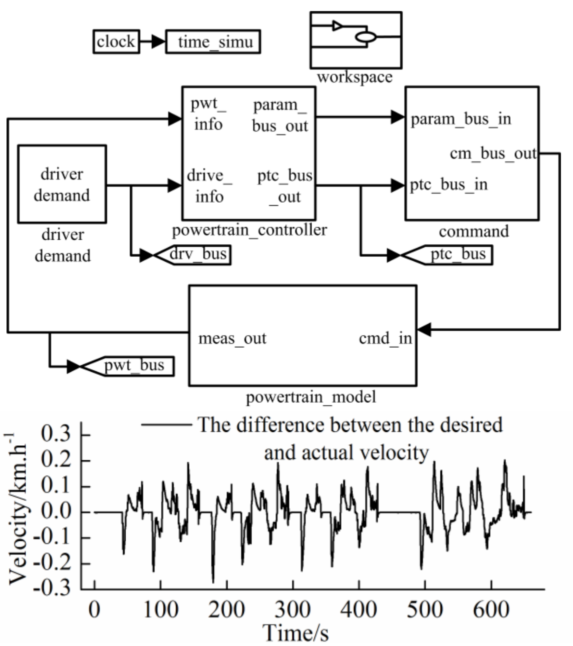


Figure 4. The simulation model for the mEV

TABLE 4. Economy performances of different schemes

Items	Average efficiency	Energy consumption/kJ
Original Scheme	0.7944	2265.31
Scheme Ⅰ	0.8183	2188.46
Scheme Ⅱ	0.8246	2173.34
Scheme Ⅲ	0.8574	2094.21

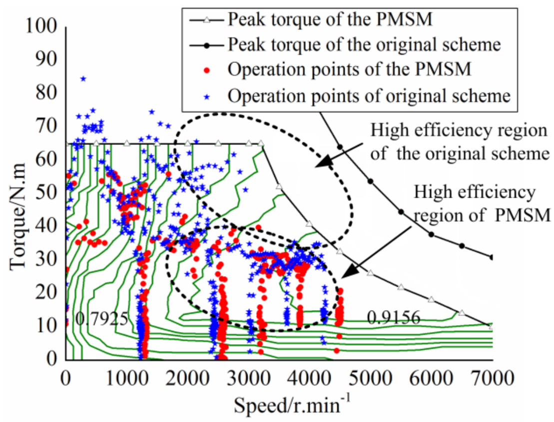


Figure 5. Motor operation points under the single and dual motor scheme with settled gear ratio

Figure 5 provides the detailed information about the distribution characteristics of the motor operations under J1015 drive cycle. In comparation with the original drive scheme, with the use of the SSMDM, the operating efficiency under urban cycle is significantly improved. As shown in Figure 5, under the same drive cycle (J1015 cycle), more motor operations are covered by the high-efficiency region of the small motor compared with the original scheme. Accordingly, the average drive efficiency of this scheme increases by 3.01% compared with the original one.

3. 4. 2. Disadvantages of this Scheme However, another thing should be noted that, as shown in Table 4, the configuration with dual motor and fixed gear ratio shows no advantage when compared with the scheme driven by single motor and two-speed transmission (scheme Ⅱ). For scheme Ⅱ, based on the original scheme, a two-speed gearbox is adopted to institute the fixed one. The optimum ratios of the first (i_1) and second shift (i_2) are 11.43 and 4.06, obtained by the mathematical method proposed in section 4.

For scheme Ⅰ and Ⅱ, Figure 6 presents the characteristics of the motor operations. Features of the motor efficiency under low speed region should be highlighted: 1) the average efficiency is relatively low; 2) under a given speed, the variation of motor torque shows a quite small impact on motor efficiency.

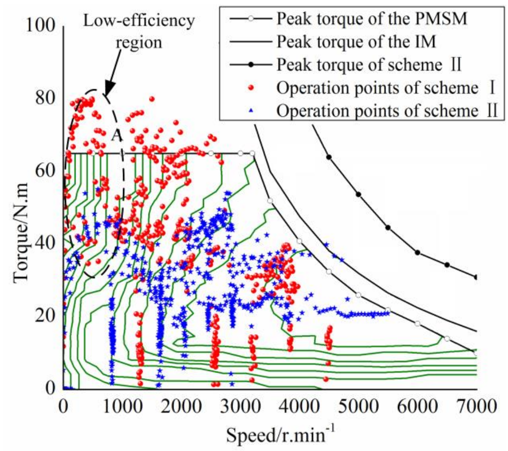


Figure 6. Motor operations under the scheme Ⅰ and Ⅱ

Influenced by the efficiency characteristics, under low speed region, the method using SSMDM to improve driving efficiency is no longer feasible, while scheme Ⅱ shows great advantages on the improvement of driving efficiency owing to the controllable transmission, and motor operations under low speed region decrease significantly owing to the use of the second shift.

4. SCHEME WITH DUAL MOTOR AND TWO-SPEED TRANSMSSION

Given the above analysis, to further optimize the driving economy of the mEV, based on the FRMD scheme, configuration with PMSM and two-speed gearbox in rear axle, IM and one-speed gearbox in front axle is proposed.

4. 1. Design of the Two-Speed Gear Ratio

4. 1. 1. Design of the Variables First of all, to ensure the dynamic performance of the mEV (operations of acceleration or slope climbing), the $i_1$is constraint by 6-15. In addition, the i_2 is constraint by 6-15 to match the maximum design velocity. As it is unpractical to obtain the effect of every possible ratio combination on driving economy, LDH method is adopted, which can use as few experiment points as possible to characterize the possible combinations of $i_1$and i_2, as shownFormula (13).

$$\vec{X} = \begin{bmatrix} i_{1st\_1} & i_{2nd\_1} \\ \vdots & \\ i_{1st\_k} & i_{2nd\_k} \\ \vdots & \\ i_{1st\_m} & i_{2nd\_m} \end{bmatrix} = \begin{bmatrix} 8.97842 & 4.44755 \\ \vdots & \\ 6.51799 & 5.43799 \\ \vdots & \\ 12.02158 & 5.08245 \end{bmatrix} \tag{13}$$

4. 1. 2. Energy Consumption Model For each LHD point, taking the number k point for example, according to Formula (14), the speed, torque and efficiency of the PMSM in first or second shift are:

$$\begin{cases} n_{k\_is} = i_{k\_is} V_j / 0.377r \\ T_{k\_is} = T_{r\_j} / \eta_T i_{k\_is} \\ \eta_{m\_k\_is} = f\left(n_{k\_is}, T_{k\_is}\right) \end{cases} \tag{14}$$

where n_{k_is}, T_{k_is} and $\eta_{m\_k\_is}$ are motor speed, torque and efficiency of the k LHD point in first or second shift.

In this study, as the small powertrain system is mainly designed to achieve higher vehicle economy under low to middle velocity and load conditions, the sampling J1015 cycle points of V_j and T_{r_j} are constraint by 0-45km/h and 0-200N.m. Aiming at the least battery energy consumption, the energy consumption of the number k LHD point can be confirmed as follows:

$$E_{j\_k} = \begin{cases} K_1 T_{r\_j} V_j \Delta t \Big/ max\left(\eta_{m\_k\_1st}, \eta_{m\_k\_2nd}\right) \eta_T \eta_{b\_j\_k} r & if: A \subset \Omega \,\&\, B \subset \Omega \\ K_1 T_{r\_j} V_j \Delta t \Big/ \eta_{m\_k\_1st} \eta_T \eta_{b\_j\_k} r & if: A \subset \Omega \,\&\, B \not\subset \Omega \\ K_1 T_{r\_j} V_j \Delta t \Big/ \eta_{m\_k\_2nd} \eta_T \eta_{b\_j\_k} r & if: A \not\subset \Omega \,\&\, B \subset \Omega \\ \Lambda_{j\_k} & if: A \not\subset \Omega \,\&\, B \not\subset \Omega \end{cases} \tag{15}$$

where E_{j_k} is the optimum energy consumption of the number k LHD point under the number j sampling point, A and B consist of n_{k_1st}, T_{k_1st}, n_{k_2nd} and T_{k_2nd}, Ω contains the peak speed and torque of the small power motor, Λ_{jk} is the penalty coefficient, K_1 is the unit conversion factor.

Given the above analysis, the response of the number k LHD under the given sampling points is:

$$E_{dis\_k} = \sum_{j=1}^{M} E_{j\_k} \tag{16}$$

4. 1. 3. Response Surface Model According to the methods proposed above, the energy consumption of the rest of the LHD points in Equation (13) can also be confirmed. Here, min-max fitting distance and stepwise regression analysis are adopted to establish themathematical model of the input factors (i_1, i_2) and the out response (E_{dis}). The six-degree polynomial of the response surface model is created as shown in Equation (17). The multiple correlation coefficient (R^2) of model is 0.998 and the studentized residuals are within -2 and +2, both indicate the model has a relatively high precision. Based on the response model, hill-climbing searching method is utilized to search the optimum transmissions. The optimal results of i_1 and i_2 are 9.905 and 4.828.

$$
\begin{aligned}
\hat{E}_{dis} = &- 1664.6817 + 697.12137 i_1 + 2135.7074 i_2 \\
&- 735.1532 i_1 i_2 - 329.6006 i_2^2 - 12.231346 i_1^3 \\
&+ 77.557368 i_1^2 i_2 + 129.0122 i_1 i_2^2 + 1.04928 i_1^4 \\
&- 1.391006 i_1^3 i_2 - 18.18846 i_1^2 i_2^2 - 0.028062 i_1^5 \\
&- 0.106105 i_1^4 i_2 + 0.60974 i_1^3 i_2^2 + 1.246804 i_1^2 i_2^3 \\
&- 1.481546 i_1 i_2^4 + 0.676546 i_2^5 + 0.0050322 i_1^5 i_2 \\
&- 0.0140566 i_1^4 i_2^2 - 0.0660401 i_1^2 i_2^4 \\
&+ 0.12317 i_1 i_2^5 - 0.0695209 i_2^6
\end{aligned}
\tag{17}
$$

4. 2. Analysis of the Proposed Schemes Based on the scheme with dual motor and variable transmission (scheme Ⅲ), GA is adopted to optimize the driving mode. Figure 7 presents the optimal motor operations under J1015. First of all, a great part of the operating points are covered by the high-efficiency of the front and rear motor. Moreover, under urban cycle, the percentages of the single-PMSM drive mode in first and second shift are 34.42 and 30.9% respectively, which indicates the designed scheme meets the differentiated design principle mentioned in section 2.

Figure 8 presents the motor operations under scheme Ⅰ and Ⅲ, which clearly shows that a great deal of motor operations under the low speed region, where driving efficiency is relatively low, are optimized to the middle speed region, owing to the use of the two-speed transmission scheme. Accordingly, for scheme Ⅲ, the driving energy consumption under J1015 cycle reduces by 4.31% compared with scheme Ⅰ.

Figure 9 presents the motor operations under scheme Ⅱ and Ⅲ. With the use of the differentiated-dual-motor scheme, under urban cycle, the high-efficiency region of the small motor covers much more cycle operations, owing to the SSMDM.In comparation with scheme Ⅱ, the driving energy consumption of scheme Ⅲ under J1015 cycle reduces by 3.64%.

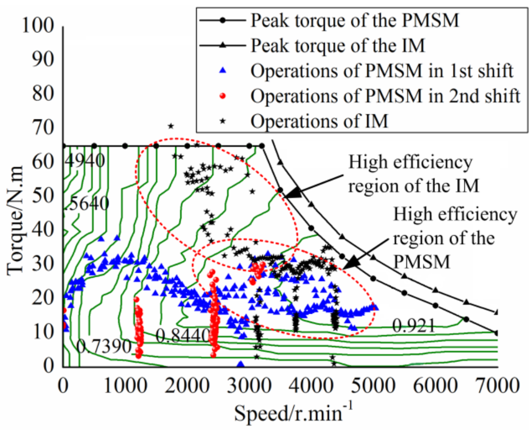


Figure 7. Simulated motor operations under scheme Ⅲ

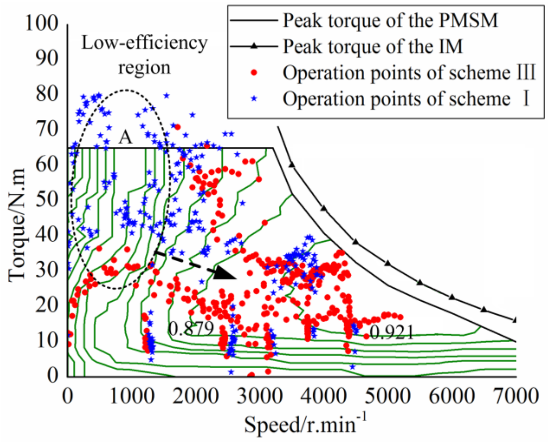


Figure 8. Motor operations under scheme Ⅰ and Ⅲ

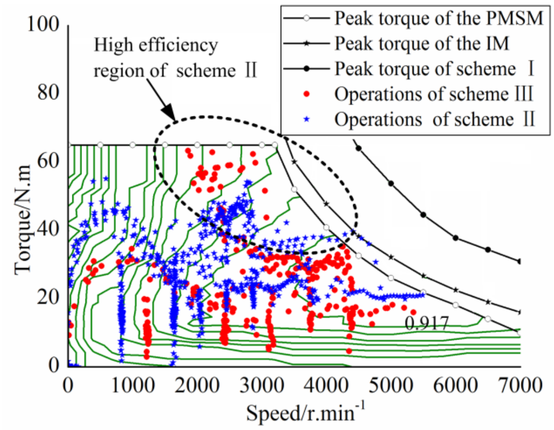


Figure 9. Motor operations under scheme Ⅱ and Ⅲ

5. CONCLUSIONS

In this paper, to develop a higher-efficiency driving scheme for the mEV with single motor and settled gear ratio, new configurations with IM and PMSM in front and rear axleare proposed. Specifically, the obtained conclusions are as follows:

1) With the use of RSM, the explicit functional relationship of energy consumption and the two-speed transmissions can be established, which is helpful for the analysis and design of gear ratios. This mathematical method can also be applied to the other schemes.

2) Compared with the scheme with single motor and settled ratio, results show that the dual-motor-and-fixed-speed configuration can improves the driving efficiency by 3.01% under urban cycle, owing to the use of the single-PMSM-drive mode under low load conditions.

3) In low-speed region, it is no longer possible to optimize driving economy by using SMDM, while the scheme with two- or multi-speed transmission is preferred owing to the controllable ratio.

4) Scheme with dual motor and variable transmission shows great advantages on the improvement of vehicle economy. In comparation with the original scheme, the driving energy consumption of this scheme reduces by 7.55%.

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Parameters Design and Economy Study ofan Electric Vehicle with Powertrain Systems in Front and Rear Axle

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چكيده

برای رسیدن به اقتصاد بالاتر، طرح رانندگی اصلی با نسبت تک موتور و دنده، تنظیمات جدید با سیستم انتقال قدرت متفاوت در جلو و محور عقب طراحی شده بود. ابتدا، با توجه به قدرت و گشتاور مورد نیاز وسیله نقلیه الکتریکی میکرو (mEV) در دوره های مختلف رانندگی، پارامترهای یک موتور قدرتی کوچک و بالا مشخص شد. در مرحله دوم، برای طرح I با موتور دوگانه و یک جعبه دنده سرعت، بر اساس الگوریتم ژنتیک(GA) ، انتقال theoptimal و توزیع گشتاور تایید شد. با توجه به استفاده از حالت رانندگی تک موتور (SMDM)، مصرف انرژی شبیه سازی شده در مقایسه با طرح اصلی ٤/٠١٪ کاهش یافت، اما راندمان موتور هنوز هم در شرایط کم سرعت (٠-١٠٠٠ دور بر دقیقه) به علت انتقال ثابت نسبتا پایین است. سپس، برای حل این مشکل، طرح با استفاده از گیربکس دو سرعته به جای یکی از گیربکس های اصلی در محور عقب پیشنهاد شده است. برای این طرح، اول از همه، رابطه صریح و روشن مصرف انرژی و نسبت دو سرعت بر اساس روش سطح پاسخ (RSM) تاسیس شد. پس از آن، روش بالا رفتن از تپه برای جستجوی بهترین نسبت استفاده شد. در نهایت، اجرای اقتصادی طرح های مختلف در مدل شبیه سازی مورد بحث قرار گرفت و مصرف انرژی این طرح در مقایسه با طرح اصلی ٧/٥٥٪ کاهش می یابد.

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