Design and analysis on a novel electric vehicle powertrain

YU Jie[1,2]，YAO Ligang[1]，REN Chengcheng[1]，YAN Xiaolei[2]

（1. Fuzhou University, Fuzhou, Fujian 350108, 2. Fujian University of Technology, Fuzhou, Fujian 350108）

Abstract：

In order to obtain synthetically advantages from the independent multiple-motors-driven system and the centralized single-motor-driven system, a novel centralized dual-motor-coupling powertrain (DMCP) system with a single row double pinions gear train for electric vehicle (EV) is proposed. By means of changing the drive modes, the proposed powertrain can satisfy the demands of different running conditions with high energy utilization efficiency. In this paper, the analyses on the mechanism structure and operating principles of the proposed powertrain are carried out firstly. Then, for further improving the operating efficiency and providing the effective ways for optimal matching design of the novel DMCP, the parameters sensitivity analysis is studied under the certain tested driving cycle based on a systematic simulation model. The simulation result shows that the energy consumption is increased by 3% as compared to the single motor powertrain test.

Keywords：dual-motor; electric vehicle; powertrain; simulation;

1 Introduction

Since the powertrain system design has an important influence on dynamic performance and fuel economy for electric vehicle (EV) [1], the centralized dual-motor-driven coupling powertrain (DMCP) is proposed recently except the independent multiple-motors-driven system and the centralized single-motor-driven system [2]. Although the in-wheel electric motor powertrain can simplify the structure of the drivetrain, it is still difficult to overcome the shortcomings of high control complexity and manufacture level requirement [3]. On the other hand, the centralized single-motor-driven system has been applied on most modern EVs because of its reliability. However, it claims the need of ratio changing mechanism to improve motor's speed-torque characteristics, which involves more cost and shift control issues [4]. Hence, some new drivetrain with wider torque/speed characteristic and efficient operating region as well as high reliability are desirable for EV. For this reason, new powertrains based on centralized dual-motor-driven are proposed [2, 5~8]. However, some of the existing structures can only realize speed coupling function of two motors, and the others with speed and torque coupling function have complex structure. Based on the preliminary research, a novel DMCP with simple mechanical structure is designed in this paper to expend the speed-torque characteristics of motors and to reduce energy consumption. First, the mechanism structure and operating principles of the powertrain are introduced and analyzed in the remainder of this paper. Then, the parameters sensitivity is studied under certain tested driving cycle based on a systematic simulation model, which aims at offers effective ways for optimal matching design of the novel DMCP. And Multi-Island Genetic Algorithm (MIGA) is used to obtained the optimal value for parameters.

2 Structure and operating principles for the DMCP

For the understanding of novel powertrain system, the structure of DMCP is illustrated in Fig. 1, which is composed of two motors (M1and M2), a single row double pinions gear train (DEG), two clutches (CL1and CL2) and a brake (B). In this powertrain, the main-drive motor (M1) is connected to the sun gear of the DPG and could connect to the carrier when CL1 and CL2 both are engaged, while the auxiliary-drive motor (M2) could connect to the sun gear or the carrier based on the position of two clutches. As the three rotational motions (Sun, Ring, Carrier) of DPG been controlled by switching the position of CL1, Cl2 and B, the DMCP can work in different modes, which have been shown in Table I. The tractive force form two motors is coupled by the DPG unit and then exported from the ring to the final drive. As shown in Table I, the DMCP can operate in five driving modes: low-speed single motor drive (SM), low-speed dual-motor torque coupling drive (LTC), Moderate-speed single motor drive (MSM), Moderate-speed dual-motor torque coupling drive (MTC), and high-speed dual-motor speed coupling drive (HSC). By means of changing drive modes, the powertrain can satisfy the demands of different running conditions with high energy utilization efficiency.

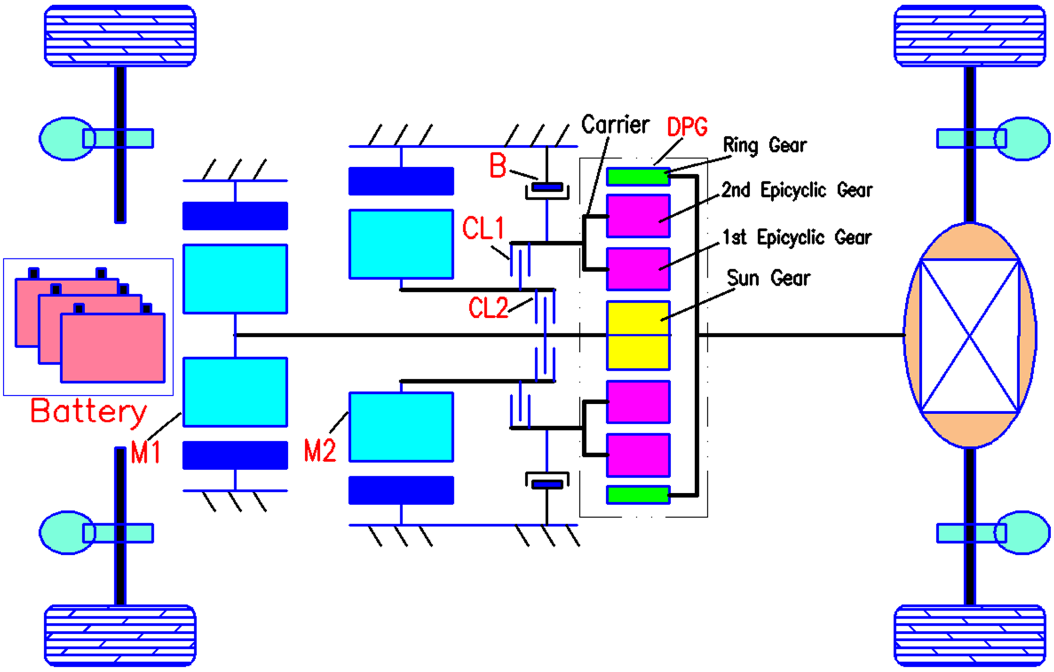


Fig. 1 Schematic of the DMCP

TABLE I

System operation schedule

Modes	M1	M2	CL1	CL2	B	Driving State
LSM	●	○	○	○	●	Low Speed Light Load
LTC	●	●	○	●	●	Low Speed Heavy Load
MSM	●	○	●	●	○	Middle Speed Light Load
MTC	●	●	●	●	○	Middle Speed Heavy Load
HSC	●	●	●	○	○	High Speed

Note: "●" Indicates that the motor is on and CL/B is engaged,

"○" Indicates that the motor is off and CL/B is disengaged.

It is worth mentioning that the model shift for DMCP is ease to realize. As the vehicle acceleration process (From low-speed to high-speed), only one of the CL1, CL2 and B change status each time to realize a new model. This means that the control for DMCP is easy and flexible to design.

To reveal the operating principles of DMCP, the kinematics equation of DPG is proposed as follows by ignoring the energy loss:

$$\begin{cases} \omega_s + (k-1)\omega_c = k\omega_r \\ T_s = \frac{1}{k-1}T_c = -\frac{1}{k}T_r \end{cases} \tag{1}$$

Where ω_s denotes the sun gear speed, ω_c is the carrier speed, ω_c is the carrier speed, k denotes the characteristic parameter of DPG, its value is Z_r/Z_s.

Then, based on the equation (1), the operating principles of the DMCP were briefly analyzed as follows:

LSM mode
$$\begin{cases} \sum I_w\dot{\omega}_w + I_{m1}\dot{\omega}_{m1}ki_0 = T_{m1}ki_0\eta_t - \sum T_{req} \\ \omega_{m1} = \omega_s = ki_0\omega_w \\ \omega_{m2} = 0 \end{cases} \tag{2}$$

LTC mode
$$\begin{cases} \sum I_w\dot{\omega}_w + (I_{m1}+I_{m2})\dot{\omega}_{m1}ki_0 = (T_{m1}+T_{m2})ki_0\eta_t - \sum T_{req} \\ \omega_{m1} = \omega_{m2} = \omega_s = ki_0\omega_w \end{cases} \tag{3}$$

MSM mode
$$\begin{cases} \sum I_w\dot{\omega}_w + I_{m1}\dot{\omega}_{m1}i_0 = T_{m1}i_0\eta_t - \sum T_{req} \\ \omega_{m1} = \omega_{m2} = \omega_s = \omega_r = i_0\omega_w \end{cases} \tag{4}$$

MTC mode
$$\begin{cases} \sum I_w\dot{\omega}_w + (I_{m1}+I_{m2})\dot{\omega}_{m1}i_0 = (T_{m1}+T_{m2})i_0\eta_t - \sum T_{req} \\ \omega_{m1} = \omega_{m2} = \omega_s = \omega_r = \omega_c = ki_0\omega_w \end{cases} \tag{5}$$

LTC mode
$$\begin{cases} \sum I_w\dot{\omega}_w + \left(I_{m1}\dot{\omega}_{m1} + \frac{1}{k-1}I_{m2}\dot{\omega}_{m2}\right)ki_0 = min\left(T_{m1}, \frac{1}{k-1}T_{m2}\right)ki_0\eta_t - \sum T_{req} \\ \omega_{m1} + (k-1)\omega_{m2} = ki_0\omega_w \end{cases} \tag{6}$$

Where $\sum I_w$ is the total wheel moment of inertia, $I_{m1}$and I_{m2}denote the moment of M1 and M2, ω_{m1} and ω_{m2} denote the speed of M1 and M2, T_{m1} and T_{m2} denote the torque of M1and M2, ω_w is the wheel speed, i_0 is the final drive gear ratio, $\sum T_{req}$ denotes the total required torque at the output axle, η_t represents the total drivetrain efficiency.

It should be noted that the torque of the ring is $k \cdot min\left(T_{m1}, \frac{1}{k-1}T_{m2}\right)$ in the equation (6) because of the limit of DPG kinematic equation.

3 Optimal matching design for the DMCP

In order to reveal the characteristics of the novel DMCP in dynamic and efficiency performance, an electric bus is selected as the target vehicle, which parameters are shown in Table II. And as shown in fig.2, the real driving cycle and the required power is test on a bus. Based on the test data analysis, the commonly used speed range of city bus is about 0~10 km/h and 20~35 km/h. And the peak speed almost at 43 km/h with required power about 150kW. Moreover, the kilowatt consumption per 1km is about 8.3 with the single motor powertrain system.

TABLE II

Vehicle Parameters

Components	parameters	values
Vehicle Body	Curb Mass (Kg)	11400
	Gross Mass (Kg)	18000
	Frontal Area（m^2）	7.8
	Drag Coefficient	0.6

	Rolling Radius（m）	0.436
	Final drive gear ratio	6.166
Battery Pack	Rated Capacity (A • H)	412AH
	Rated Voltage (V)	432V

Fig. 2 Vehicle drive cycle and power required for a city bus

The parameters needed to design for the DMCP are: the motor parameters, the characteristic parameter of DPG and the final drive gear ratio. And the powertrain parameters matching EV mainly have two principles: meeting the dynamic performance requirements (As is Shown in Table III) and high efficiency optimal matching design.

TABLE III

The dynamic performance requirements

Name	value
Maximum Speed (km/h)	≥65
Maximum slope at 10km/h vehicle speed (%)	≥30
0–50 km/h acceleration time t_{50} (s)	≤15

For the first principle, the total power must satisfy the following equation:

$$P_{1max} + P_{2max} \geq max(P_v\ , P_i\ , P_j) \tag{7}$$

$$max(T_{di}\ , P_{dj}) \leq T_{1max} + T_{2max} \leq F_\varphi \cdot r / ki_0 \tag{8}$$

$$ki_0 u_{max} / 0.377r \leq \omega_{1max} + (k-1)\omega_{2max} \tag{9}$$

$$T_{1max} = 9549 P_{1max} / \omega_{1max} \tag{10}$$

$$T_{2max} = 9549P_{2max}/\omega_{2max} \tag{11}$$

Where P_{1max} is the peak power of M1, P_{2max} is the peak power of M2, P_v is the required power at maximum speed, P_i is the required power at maximum slope, P_j is the required power for $t_{50} = 15$, T_{1max} is the maximum torque of M1, T_{2max} is the maximum torque of M2, T_{di} is the required torque for maximum slope, T_{dj} is the required torque for acceleration, F_φ is maximum force under the road adhesion condition, r is the wheel radius, u_{max} is peak speed for vehicle, ω_{1max} is the peak speed of M1, ω_{2max} is the peak speed of M2.

For the second principle, the rated power P_{1e} and speed ω_{1e} should be fit to the commonly used speed range, from which the rated speed and power of M1 can be calculated under certain driving cycle. Then, rated torque of M1 T_{1e} can be calculated as follow:

$$T_{1e} = 9549P_{1e}/\omega_{1e} \tag{12}$$

From the Eqs. (7) ~ (12), the two motors' parameters of DMCP can be obtained under the test cycle(As shown in Table IV), and the values of k and i_0 are calculated through optimal-matching method, which Multi-Island Genetic Algorithm (MIGA) is used for the multi-objective optimization problem. The weight method is used to obtain dynamic performance optimal objective function from the peak speed, 0~50 km/h acceleration time and peak slope, meanwhile the kilowatt consumption per 1km is taken as the economic optimal objective function. The optimal calculating flow is shown in Fig.3, and the optimal result is shown in fig.4. When the value of k is 2.3 and the value of i_0 is 6.21, the kilowatt consumption per 1km is about 8.0 and the dynamic performance satisfies the design requirements. And the Fig.5 shown that the current velocity can follow the desired velocity well. Besides, with the analysis on the parameters sensitivity of DMCP, it shows that the characteristic parameter k has more influence on system efficiency and peak slope, and the final drive gear ratio i_0 has more influence on the 0~50 km/h acceleration time and peak speed.

TABLE IV

The parameters of DMCP

components	Name	value
M1	Rated/peak power (KW)	70/141
	Rated/peak speed (rpm)	1500/3000
	Rated/peak torque (N · m)	30/60
M2	Rated/peak power (KW)	450/900
	Rated/peak speed (rpm)	1800/3500
	Rated/peak torque (N · m)	160/320

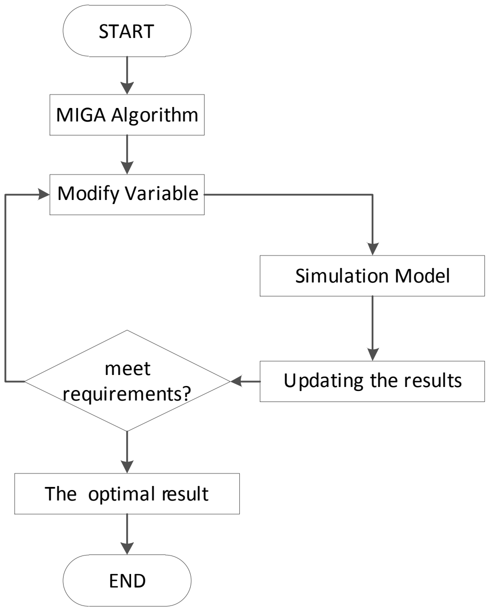


Fig. 3 The flow of MIGA optimal calculation

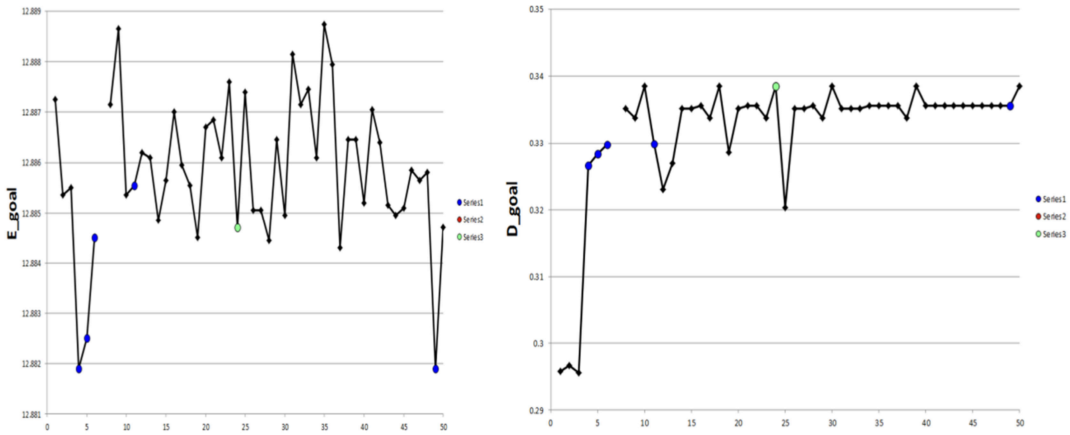
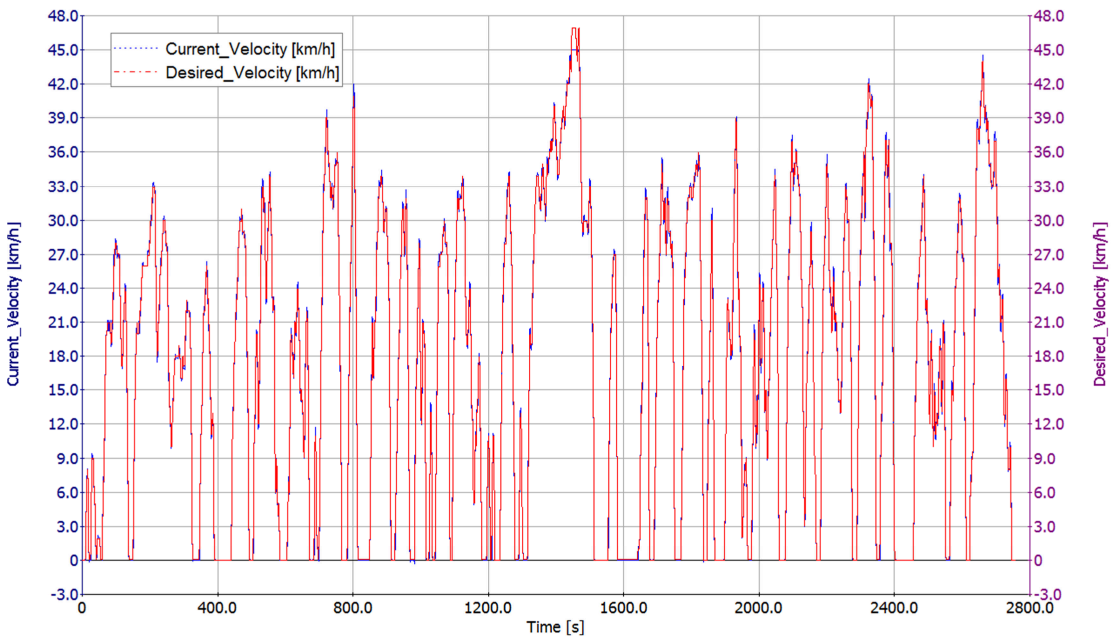


Fig. 4 The value of parameters and optimal goals with the number of iterations



5 Conclusion

In order to improve the performance on the all-electric vehicle, a dual-motor coupling powertrain system is provided with two motors, DPG, two clutches and a brake. Based on the structure and operating principles analyzed for the DMCP, the parameters are designed to follow two principles about meeting the dynamic performance requirements and high efficiency optimal matching design. The simulation results show that the energy consumption is increased by 3%, which proves the application value of the novel powertrain.

Reference：

[1] M. Ehsani, Y. Gao, A. Emadi. Modern Electric, Hybrid Electric, and Fuel Cell Vehicles-Fundamentals, Theory, and Design [M]. Second Edition. Beijing: China Machine Press, 2012: 89-104.

[2] M. Hu, J. ZENG, S. XU, et al. Efficiency Study of a Dual-Motor Coupling EV Powertrain [C]. IEEE Transactions on vehicular technology. 2015 (64): 2252-2260.

[3] H. Alipour, M. Sabahi, M. Bagher. Lateral stabilization of a four wheel independent drive electric vehicle on slippery roads. Original Research Article Mechatronics [J]. 2015(30): 275-285.

[4] Q. Ren, D. A. Crolla, A. Morris. Effect of geared transmissions on electric vehicle drivetrains [C]. IDETC/CIE. Conf., San Diego, CA, USA, 2009: 959-966.

[5] S. ZHANG, R. XIONG, C. ZHANG. Pontryagin's minimum priciple-based power management of a dual-motor-driven electric bus[J].Applied Energy 159 (2015)：370-380

[6] LiPeng ZHANG, Liang LI, BingNan QI et al. Parameters optimum matching of pure electric vehicle dual-mode coupling drive system [J]. Technological Science/Applied Energy. 2014(57)：2265-2277.

[7] G. HAN, C. ZHANG, S. ZHANG, et al. Control strategy of the dual motors coupling propulsion for battery electric buses [J]. Applied Mechanics and Materials. 2014 (528): 364-370.

[8] X. WU, S. HAN, H. SUN. Efficiency analysis of planetary coupling drive system with dual motors on electric bus [C]. Proceedings of 2015 IEEE International Conference on Mechatronics and Automation: 2547-2552.