

Parallel Hybrid Car Power System Design and Optimization

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Abstract: Structure and working principle of parallel hybrid car power system were introduced. Design method of power system parts, such as engine, motor, gearbox and battery pack were proposed. A parallel hybrid car power system design was introduced as an example. Power system performance simulation model based on ADVISOR was established. Accelerating ability, gradeability and driving range of the car were simulated. Simulation results show that power system performance indexes meet design requirements. The design method is feasible. Optimization design of SOC working range of battery pack were finished. Research results show when lower limit of SOC is 0.4 and upper limit of SOC is 0.79, battery pack has optimal comprehensive performance.

Introduction

A lot of fuel car using was important cause of urban smog. Compared with fuel cars, gas-electric hybrid car had obvious advantages in reducing emissions and saving fuel consumption. As the government support for new energy cars purchase strength increasing, public awareness of energy conservation and emissions reduction, gas-electric hybrid car were used in the city gradually. Power system performance was an important index to hybrid car. Design method of power system are important research content of hybrid car [1,2]. Power system parameters matching test design was a reliable method with characters of long cycle and high cost. To get power system working performance with simulation had become a trend of power system design research [3,4,5].

According to design requirement of hybrid car's accelerating ability, gradeability, driving range in hybrid drive model and pure electricity drive model, power system parts design method, such as engine, motor, gearbox and battery pack, were proposed. Power system performance simulation model based on ADVISOR was established to validate design method. At the same time, based on power system performance simulation model, optimization design of SOC working range of battery pack were carried out.

Parallel hybrid car power system structure researched in this paper was shown in Fig. 1, engine, ISG motor were connected with gearbox through velocity torque sensor, battery pack was connected with ISG motor. The power system could work in hybrid drive model and pure electric drive mode. In hybrid model, load was driven by engine and ISG motor simultaneously. In pure electric drive mode, load was driven by ISG motor alone, and engine was in a stall condition.

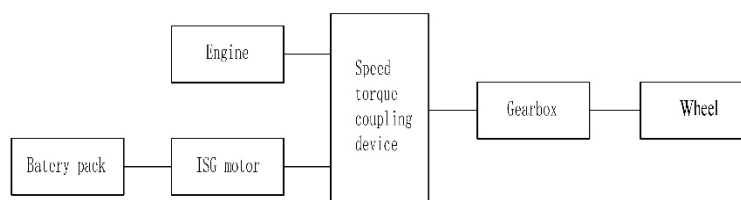


Fig.1 Parallel hybrid car power system structure

Hybrid power system design method

According to parallel hybrid power system structure shown in Fig 1, hybrid power system design principles were determined as follows. Power system output power must great than total load power

during the whole driving process. Considering power system had two working models, hybrid power system design were divided into 5 steps.

Step 1. To calculate ISG motor parameters, battery pack capacity and power according to hybrid car dynamic performance indexes in pure electricity drive model.

Step 2. To calculate engine parameters according to hybrid car dynamic performance indexes in hybrid drive model.

Step 3. To calculate gearbox parameters according to hybrid car dynamic performance indexes.

Step 4. To validate power system parts design results with simulation method.

Step 5. If design results didn't meet design requirement, return to step 1 and adjust parts parameters again.

In this paper, hybrid car dynamic performance indexes including maximum driving velocity, acceleration time of 0~100km/h, equivalent fuel consumption per hundred kilometers, maximum gradeability in hybrid drive model; maximum driving velocity, acceleration time of 0~60km/h, maximum gradeability, maximum driving range in pure electric drive model.

ISG motor parameters design method

When car was in pure electric drive model and driving at maximum velocity v_m , the required power P_1 of the car is following

$$P_1 = \frac{v_m}{3600h} (mgf + \frac{C_d A}{21.15} v_m^2) \quad (1)$$

While, v_m is maximum velocity of car in pure electric drive model, unit is m/s , m is car mass, unit is kg , g is acceleration of gravity, unit is m/s^2 , f is rolling resistance coefficient, C_d is Air resistance coefficient, h is gearbox efficiency, A is windward area, unit is m^2 .

When car was in pure electric drive model, driving with velocity v_p on slope with maximum gradeability a_m , the required power P_2 of the car is following

$$P_2 = \frac{v_p}{3600h} (mgf \cos a_m + mg \sin a_m) \quad (2)$$

While, v_p is climbing velocity, unit is m/s , a_m is maximum gradient in pure electric drive model.

When car was in pure electric drive model, velocity is increased from 0 to v_a , the required power P_3 of the car is following

$$P_3 = \frac{1}{3600ht_a} (\frac{t_a mgf v_a}{1.5} + \frac{C_d A t_a}{52.875} v_a^3 + \frac{mdv_a^2}{7.2}) \quad (3)$$

While, v_a is acceleration end velocity, unit is m/s , d is car mass conversion factor, t_a is time of acceleration, unit is s .

ISG motor maximum power P_m is following

$$P_m = \max(P_1, P_2, P_3) \quad (4)$$

ISG motor rated power P_e is following

$$P_e = \max(P_1, P_2) \quad (5)$$

ISG motor rated velocity n_e is following

$$n_e = \frac{n_{max}}{b} \quad (6)$$

While, n_{max} is ISG motor maximum velocity, b is constant power coefficient.

ISG motor rated torque T_e is following

$$T_e = 9550 \frac{P_e}{n_e} \quad (7)$$

Engine parameters design method

When car is in hybrid drive model and driving at maximum velocity v_{mh} , calculation method of car required power P_{1h} is following

$$P_{1h} = \frac{v_{mh}}{3600h} (mgf + \frac{C_D A}{21.15} v_{mh}^2) \quad (8)$$

While, v_{mh} is maximum velocity of car in hybrid drive model, unit is m/s .

When car is in hybrid drive model, driving with velocity v_{ph} on slope with maximum gradeability a_{mh} , calculation method of car required power P_{2h} is following

$$P_{2h} = \frac{v_{ph}}{3600h} (mgf \cos a_{mh} + mgsin a_{mh}) \quad (9)$$

While, v_{ph} is climbing velocity, unit is m/s , a_{mh} is maximum gradient in hybrid drive model.

When car is in hybrid drive model, velocity is increased from 0 to v_{ah} , calculation method of car required power P_{3h} is following

$$P_{3h} = \frac{1}{3600ht_{ah}} (\frac{t_{ah} mgf v_{ah}}{1.5} + \frac{C_D A t_{ah}}{52.875} v_{ah}^3 + \frac{mdv_{ah}^2}{7.2}) \quad (10)$$

While, v_{ah} is acceleration end velocity, unit is m/s , t_{ah} is time of acceleration, unit is s .

Calculation method of engine maximum power P_{dm} is following

$$P_{dm} = \max(P_{1h}, P_{2h}, P_{3h}) \quad (11)$$

Calculation method of engine rated power P_{de} is following

$$P_{de} = \max(P_{1h}, P_{2h}) - P_e \quad (12)$$

Gearbox parameters design

Velocity range of ISG motor is great than engine in hybrid power system. Gearbox is needed to match ISG motor velocity and engine velocity. Gearbox ratio i is a main parameter, which is mainly depend on car maximum driving velocity and engine maximum rotation velocity. Minimum gearbox ratio i_{min} is needed in hybrid drive mode. i_{max} is decided by engine maximum driving velocity.

Calculation method of i_{min} is following

$$i_{min} \geq 0.377r \frac{n_{dmax}}{v_{mh}} \quad (13)$$

While, r is car wheel radius, unit is m , n_{dmax} is engine maximum rotation velocity, unit is r/min .

Maximum gearbox ratio i_{max} is needed in pure electric drive mode, and car is in a state of crawling slope with maximum gradient. Calculation method of i_{max} is following

$$i_{max} \leq \frac{1}{Th} [mgr(f \cos a_m + \sin a_m) + \frac{CAv_p^2}{21.15}] \quad (14)$$

Battery pack parameters design method

Main design parameters of battery pack are power and energy. The requirement of power and energy are different in pure electric drive model and hybrid drive model. In pure electric drive model, battery pack power need to meet the requirements maximum load power, and battery pack energy need to meet the requirements of driving range. In hybrid mode, battery pack power need to great than the difference value between maximum load power and engine output power, and battery pack power energy need to meet the requirements of car surplus energy efficient recycling.

Calculation method of P_b is following

$$P_b = P_e + P_a \quad (15)$$

While, P_a is car attachment required power, unit is KW.

When car driving range is L with constant velocity v_e , Calculation method of required power W is following

$$W = \frac{LP_1}{v_e} \quad (16)$$

While, L is rated driving range, unit is km.

Calculation method of number of single cell N is following

$$N = \frac{h_e EC}{1000W} \quad (17)$$

While, E is working voltage of single cell, unit is V, C is single cell battery capacity, unit is Ah, h_e is depth of discharge of battery.

Power system design example

Power system parts design of a hybrid car were introduced as design example. Hybrid car basic parameters were shown in Table.1, corresponding hybrid car dynamic performance indexes were shown in Table.2.

Table.1 Hybrid car basic parameters

| parameters (unit) | value | parameters (unit) | value |
|--------------------------------|-------|---------------------------------|-------|
| car mass (kg) | 1200 | gearbox efficiency | 0.91 |
| rated load (kg) | 160 | wheel radius (m) | 0.305 |
| air drag coefficient | 0.30 | windward area (m ²) | 2.02 |
| height of mass center(m) | 0.40 | wheelbase (m) | 2.65 |
| rolling resistance coefficient | 0.02 | mass conversion coefficient | 1.05 |

Table.2 Dynamic performance indexes

| Hybrid drive | | Pure electric drive | |
|-----------------------------------|-------|----------------------------------|-------|
| parameters (unit) | value | parameters (unit) | value |
| maximum driving velocity (km/h) | >160 | maximum driving velocity (km/h) | >120 |
| maximum gradeability(%) 0~100km/h | >25 | maximum gradeability(%) 0~60km/h | >20 |
| acceleration time (s) | <9 | acceleration time (s) | <6 |
| fuel consumption of 100km (L) | <6 | range (km) | >50 |

In this paper, $n_{\max}=5000r/m$, $n_{d\max}=5000r/m$, $v_p=40km/h$, $v_{ph}=60km/h$, $L=50km$, $P_a=3KW$, gears of gearbox are 5 shifts. Combined with parameters shown in Table.1 and Table.2, power system parts were calculated with Eq.1~Eq.17. Power system parts calculation results were shown in Table.3.

Table.3 Calculation results

| Name | parameters | |
|--------------|-------------------------------|-------------------------|
| engine | maximum output power (KW) | 62 |
| | maximum output velocity (rpm) | 5000 |
| | maximum output power (KW) | 42 |
| ISG motor | rated output power (KW) | 17 |
| | rated velocity (rpm) | 1250 |
| battery pack | battery capacity (Ah) | 45 |
| | battery cell numbers | 25 |
| gearbox | number of gears | 5 |
| | gear ratio ($i_1 \sim i_5$) | 4.1, 4.8, 5.6, 6.6, 7.7 |

Power system performance simulation

Simulation research was carried out to validate the design results. The simulation software was ADVISOR2002, which was developed by USA renewable energy laboratory. As a mature simulation software of car power system design, ADVISOR had been widely used in car power system design[6,7]. In hybrid drive mode, simulation content covers maximum driving velocity of car, acceleration time of 0 ~ 100 km/h and maximum gradability at velocity of 40 km/h. In pure electric drive mode, simulation content covers maximum driving velocity of car, acceleration time of 0 to 60 km/h, maximum gradability at velocity of 30 km/h, maximum driving range.

In ADVISOR, simulation parameters setting covers car mass, driving cycles, gradability performance testing and acceleration performance testing. Car parameters setting covers car, ISG motor, battery pack, gearbox and wheel/axle. There were three typical cycles in ADVISOR software. CYC_UDDS cycles were used in this paper considering generality, velocity curve of CYC_UDDS cycles was

shown in Fig.2, driving time was 1369s, driving distance was 11.99km, maximum driving velocity was 91.25km/h, average driving velocity was 31.51km/h.

Hybrid power system performance simulation were carried out based on the simulation model. Vehicle dynamic performance simulation result in hybrid drive model are as follows: Gearbox gear working curve is shown in Fig.3, which shows that gear are changed with car driving velocity changing. It shows gearbox working status is normal. Car acceleration time of 0~100km/h is 8.4s, equivalent fuel consumption of 100km is 4.3L, maximum gradeability is 35% , maximum driving velocity is 200km/h.

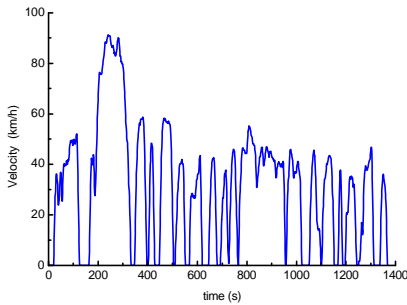


Fig.2 Velocity curve

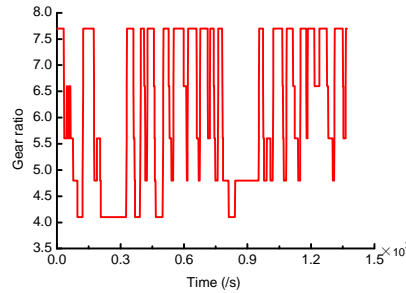


Fig.3 Gearbox shift curve

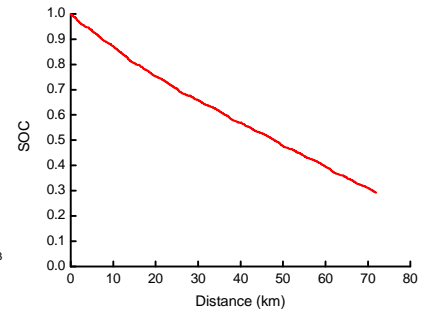


Fig. 4 Driving range simulation curve

During simulation, in order to protect battery pack life, lower limit of battery pack SOC is set as 0.3. Hybrid car dynamic performance simulation results in pure electric drive model are as follows: acceleration time of 0~60km/h is 4.4s, maximum gradeability is 25%, maximum driving velocity is 150km/h. Hybrid car driving range curve is shown in Fig.4, it shows that driving range of the hybrid car is about 70km, which is great than design indexes.

Contrast with dynamic performance indexes in Table.2, it shows that hybrid car power system parts design results can meet design requirement. It means power system parts design method proposed in this paper are reasonable.

SOC working range optimization

SOC working range was a important parameter to battery pack. To increase SOC working range was benefit to decrease engine fuel consumption, but was harmful to battery pack working life. So, it was very important for battery pack to set reasonable SOC working range.

Simulation research on SOC working range with engine fuel consumption and battery pack working life were carried out. In simulation model, lower limit of SOC was defined as SOC_lo, upper limit of SOC was defined as SOC_hi. Considering battery pack working requirement, SOC_lo was set as 0.4~0.6, SOC_hi was set as 0.65~0.85. the relationship between engine fuel consumption with SOC_lo and SOC_hi were simulated.

The relationship between engine fuel consumption with SOC_lo was shown in Fig 5, which shows that engine fuel consumption is decreased with SOC_lo increasing. The relationship between engine fuel consumption with SOC_hi was shown in Fig 6, which shows that engine fuel consumption is increased with SOC_hi increasing.

In order to obtain mathematic relationship between engine fuel consumption with SOC_lo and SOC_hi. Curves of Fig. 5 and Fig. 6 were fitted. Define x_1 as SOC_lo, x_2 as SOC_hi, y_1 as fuel consumption corresponding with x_1 , y_2 as fuel consumption corresponding with x_2 . Curve fitting results of Fig.5 is as follows

$$y_1 = -1.1299 \cdot x_1^2 - 1.2538 \cdot x_1 + 5.2908$$

Curve fitting results of Fig. 6 is as follows

$$y_2 = 29.5448 \cdot x_2^2 - 39.0158 \cdot x_2 + 17.0105$$

In order to obtain optimization value of SOC working range, optimization index of y was defined as follows

$$y = k \cdot y_1 + (1-k) \cdot y_2$$

While, k was coefficient. In this paper, according to battery pack using experiences, k was set as 0.4. So, optimization object was to obtain maximum value of y .

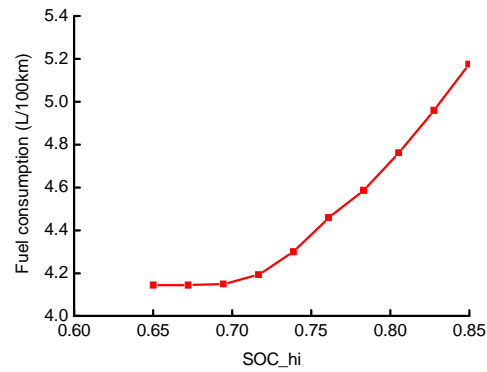
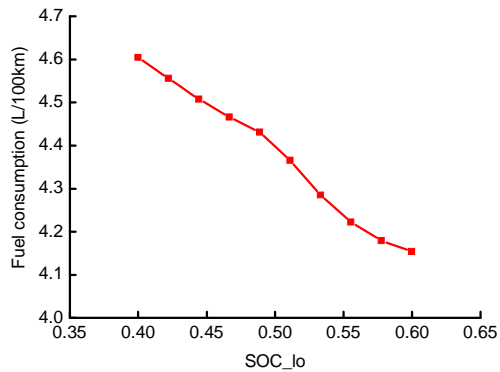


Fig. 5 Fuel consumption with SOC_lo Fig. 6 Fuel consumption with SOC_hi

Maximum value of y was obtained with parameter optimization method. Optimization results are as follows: maximum value of y is 1.1104, when $x_1=0.4$ and $x_2=0.79$. so, optimal SOC working range of battery pack is SOC_lo=0.4, SOC_hi=0.79.

Conclusions

A kind of hybrid power system design method was proposed, simulation results shows that the method is suitable for parallel hybrid power system parts design.

Optimization design of SOC working range of battery pack were finished. research results shows when SOC_lo=0.4 and SOC_hi=0.79, battery pack has optimal comprehensive performance.

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