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Assessing the Dynamic Performance and Energy Efficiency of Pure Electric Car with Optimal Gear Shifting

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Abstract: Traditional pure electric cars generally adopt single-speed transmission for cost consideration. However, with the renewal and iteration of technology, small electric cars are all developed in the direction of power performance and environmental protection. Gear shifting makes it possible for the motor to work in a more efficient range, which possibly improves the performance of the entire powertrain. In this paper, a small electric car is designed, its power parameters are matched, and the energy-saving space and effect brought by adding multiple-gear shifting transmissions are discussed. To begin, the power-matching design was carried out, and then the transmission ratio was determined by particle swarm optimization. Finally, the power performance and fuel economy of this designed car equipped with different types of transmissions were analyzed and compared through simulation experiments. The results show that the electric car equipped with two-speed transmission has improvements in most important indicators, among which the acceleration time of 0 to 100 km/h is decreased by 17.7%, and the power consumption is reduced by 1.8%. To sum up, the feasibility of applying multiple-gear shifting to small electric cars is verified, and the experimental results provide a valuable reference for the development of electric cars.

Keywords: pure electric car; dynamic performance; energy efficiency; optimal gear shifting; particle swarm optimization

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1. Introduction

With the increase in global energy consumption and the aggravation of natural environment pollution, the adoption of new energy vehicles to replace traditional fuel vehicles has become a global consensus and is being gradually implemented. Pure electric vehicles, as the mainstream products of new energy vehicles, are undergoing continuous technological evolution. Apart from the research on improving battery technology, realizing better energy saving is also a hot issue in the research and development of electric vehicles. Energy saving means a higher mileage range under the same condition. The works on energy saving include an energy-efficient control allocation scheme for dual-actuation electric motors (driving or regenerative braking dual modes) [1], energy-efficiency optimization allocation based on a motor efficiency map to reduce motor power losses and obtain energy recovery [2], predictive driving control strategy based on optimal control theory and traffic preview information [3], energy-efficient optimal control based on dynamic traffic information flow [4], slide mode control based chassis energy efficiency and driving performance comprehensive control strategy [5]. Besides the energy-saving aspect, the dynamic performance of electric vehicles, especially small pure electric vehicles, is also another issue that users concern about. Better dynamic performance and energy efficiency have been the desired indicators for small pure electric vehicles. Apart from the above-mentioned energy-efficient control methods, optimizing the powertrain, e.g., power matching and multi-gear transmission, etc., is an additional approach to improve energy saving as well as dynamic performance. The related research is as follows. Ning et al. put forward a comprehensive optimization matching method for the powertrain using traditional longitudinal dynamics to match the electric power system for the drive system and adopting cut-and-try approach for the energy storage system [6]. Gao et al. proposed a new type of 2-speed inverse automated manual transmission (I-AMT) [7], and they further studied the shift control after optimizing the gear ratio using dynamic programming methods and realized the smooth shift process without a torque hole. Fang et al. proposed a new two-speed uninterrupted mechanical transmission (UMT), which can realize seamless switching between two gears [8]. The control system based on a fuzzy logic controller (FLC) detects the driver's intention according to the speed and gas pedal position signals. Fu proposed a continuous variable transmission (CVT) configuration based on an electric oil pump (EOP) for electric vehicles and constructed a dynamic model of electric vehicles equipped with CVT [9]. Liu et al. designed a kind of two-speed AMT without a clutch and synchronizer, established the shift dynamics model and the shift-motor model, and controlled the motor and shift time, which reduced the power consumption [10]. He et al. established a hybrid electric vehicle model and proposed control strategies for the electric motor and engine to achieve clutchless shift control, which effectively improves the shift quality of the vehicle [11]. Liu et al. proposed a coordinated control strategy for two-speed clutchless AMT based on model predictive control (MPC), which effectively improved shift smoothness [12]. Because the motor has a good speed-torque characteristic in a large range compared with the engine, as well as the reasons of cost and operating expense, the general pure electric vehicle is only equipped with the transmission with sngle fixed ratio transmission at present. Although it can meet the common requirements of starting, low speed, high speed and other working conditions, there is still much room for improvement. It has become a research direction to improve vehicle performance by adding multi-speed transmission.

Hu et al. proposed a two-speed automatic mechanical transmission for pure electric vehicles, and the test on the two-speed gearbox test bench showed that the ride comfort of the proposed shift-control strategy reached the bus standard [13]. Qin et al. analyzed the shifting process of a pure electric vehicle equipped with a two-speed automatic mechanical transmission without a clutch and proposed a control strategy that can achieve smooth, reliable and fast shifting for electric vehicles [14]. In addition, Jaehoon et al. proposed an optimized design method for a lightweight two-speed transmission for electric vehicles [15], which can improve transmission efficiency through the optimal design of gear train. Angeles et al. designed a new multi-speed transmissions (MSTs) shift control scheme and proposed a two-stage control algorithm to make the electric vehicle shift more smoothly [16]. Eckert et al. proposed a multi-objective optimization method for design variables such as gear ratio, number of gears, differential ratio, tire size and shift control of an automatic transmission, which improved its economic efficiency and power performance [17]. Hu et al. proposed dynamic programming(DP)-based optimization method of gear shift schedule for electric buses equipped with 4-AMT to improve the energy economy [18]. Liang et al. proposed a gear-shifting control strategy for pure electric vehicles with inverse automated manual transmission (I-AMT) to improve the dynamic performance of pure electric vehicles [19].

To summarize, most of the current research focuses on the design of transmission structure, the shifting smoothness and the mechanical efficiency improvement, whereas the matching design method of an electric vehicle equipped with multiple-gear transmission and its improvement in performance and efficiency of an electric vehicle is less studied and verified. The motivation of this work is to investigate the dynamic performance and energy efficiency when the two-speed gear shifting is optimized to the best status and to assess the application value of two-speed transmission in pure electric vehicles. In this paper, the power-matching design of a small electric car equipped with two-speed gear shifting is carried out based on the application background of a small electric car, and the optimization is done for the purpose of economy and dynamic performance. Particle swarm optimization has been proven to be effective and applied to solve various kinds of optimization problems [20]. For the above optimization, particle swarm optimization is employed for its higher precision or easier realization compared with GA, DE and other meta-heuristic algorithms [21]. The model of this electric car is established, and the efficiency and performance improvement effect of the electric car to be equipped with optimal gear shifting is investigated based on the current typical electric vehicle test cycles. The contribution of this work is that it presents a detailed evaluation of the possible maximal dynamic performance and energy efficiency improvement of a pure electric car with multiple shifting gears by optimization, and the resultant data and conclusions are of some references for the multiple-gear electric car design.

2. Power Matching of Pure Electric Car

The pure electric car developed in this paper is a small passenger car that is positioned for short-distance driving within cities, towns, or local areas. Based on this demand, the multiple-gear shifting automatic transmission and other parameters are matched and designed for the whole car. Because the permanent magnet synchronous motor (PMSM) has the characteristics of small space volume, flexible and simple structure, and stable and convenient control, it is also the mainstream driving scheme in the present pure electric car market, so the permanent magnet synchronous motor is selected as the driving motor type in this paper.

If the parameters of the pure electric car powertrain are reasonably matched, the advantages of each component can be fully utilized so that the whole car can overcome the resistances such as wind resistance, rolling resistance, air resistance, and acceleration resistance and simultaneously meet the design indexes of power performance and economy. For the parameter matching of the powertrain of the pure electric car, the parameters of battery, motor, and main reducer should be calculated mainly according to the design indexes such as maximum speed, maximum gradability, acceleration time, and mileage range of the car. The design requirements for the electric car in this paper are shown in Table 1.

Table 1. Electric car design requirements.

Parameters	Description	Value
$U_{\max}$	Maximum speed	120 km/h
U_a	Urban normal speed	30 km/h
t_a	Acceleration time	≤6 s (0 to 50 km/h)
$\alpha_{\max}$	Maximum climbing grade	≥14° (at 30 km/h)
$L_{mileage}$	Mileage range	≥300 km

Electrically controlled automatic mechanical transmission has a low cost, simple structure, less failure rate, and cheap maintenance advantages, so the two-speed gear-shifting automatic mechanical transmission scheme is adopted, and the specific powertrain system structure of the electric car is shown in Figure 1, which consists of three pairs of gears and two shafts, in which gear pair 1 and 2 are for low-speed shift, and gear pair 3 and 4 are for high-speed shift. Gear pairs 5 and 6 form a reverse shift, and the reverse movement is realized by reversing the rotation direction of the motor. Due to the diversity of operating conditions of cars, the motor applied to pure electric cars has the following characteristics: it can adapt to frequent start, stop, acceleration and deceleration, torque dynamic change, and other operating conditions, and it can also provide large torque at a low-speed case and high rotation at high speed case. The main calculation parameters for motor selection include maximum torque, rated power, maximum power, and maximum speed.

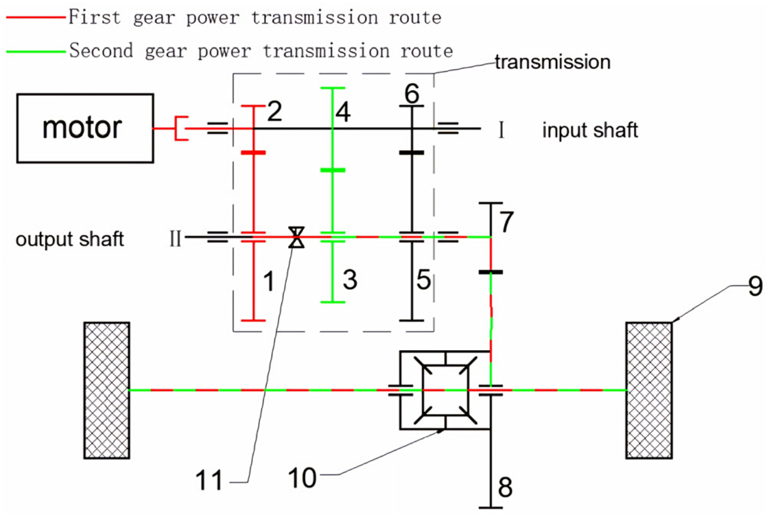

Figure 1. Structure diagram of electric car powertrain. 1, 2: lower-speed gears; 3, 4, 5, 6: high-speed gears; 7: reducer input gear; 8: reducer output gear; 9: wheel; 10: differential; 11: synchronizer.

Firstly, according to the driving conditions of the car, the driving force demand of the pure electric car is calculated as

$$F_t = Gf\cos\alpha + G\sin\alpha + \frac{C_D A U_a^2}{21.15} + \delta m \frac{dU_a}{dt} \tag{1}$$

where F_t is the driving force, G is the car gravity, f is the rolling resistance coefficient, α is the climbing angle, C_D is the air resistance coefficient, A is the windward area, U_a is the driving speed, δ is the rotating mass conversion coefficient, m is the car mass, and t is time.

The motor power is determined based on the maximal designed speed. When the car is running at the maximum speed, the road is level without gradient, so only the resistance generated by air and tires needs to be calculated. The power demand is given by

$$P_{m1} = \frac{U_{\max}}{3600\eta_t}\left(\frac{C_D A U_{\max}^2}{21.15} + mgf\right) \tag{2}$$

where η_t is the transmission efficiency, P_{m1} is the power required, g is the acceleration of gravity, and the meanings of other parameters are the same as above.

When the electric car is running on the road with a gradient, the acceleration is 0, the acceleration resistance is also 0, and the power demand P_{m2} is given by

$$P_{m2} = \frac{U_a}{3600\eta_t}\left(\frac{C_D A U_a^2}{21.15} + mgf\cos\alpha + mg\sin\alpha\right) \tag{3}$$

When under acceleration conditions, the car power balance equation is given by

$$P_{m3} = \frac{1}{3600\eta_t}\left(\frac{C_D A v_m^3}{21.15 \times 2.5} + mgf\frac{v_m}{1.5} + \delta m \frac{v_m^2}{7.2t_a}\right) \tag{4}$$

where P_{m3} is the power required, and v_m is the post-acceleration speed.

The minimal transmission ratio (corresponding to Gear II) should ensure that the car can reach the expected maximal speed, and the power output of the motor should be able to overcome the driving resistance when running at the maximum speed. The corresponding inequalities are given by

$$\min\left(\prod_k i_k\right) \leq \frac{0.377 n_{\max}}{U_{\max}} R \tag{5}$$

$$min\left(T_t \eta_t \prod_k i_k\right) \geq \left(mgf + \frac{C_D A U_{\max}^2}{21.15}\right) R \tag{6}$$

where $\min(\prod_k i_k)$ is the minimal value of the total transmission ratio of the electric car, k takes the the values of 1 and 2 which denote the transmission and the main reducer respectively, namely i_1 denoting the gear ritio of the tramsmission and i_2 denoting the gear ritio of the reducer, $n_{\max}$ is the maximum speed of the motor, R is the wheel radius, and T_t is the peak torque of the motor.

According to Equations (5) and (6), the range of the minimum transmission ratio is $4 \leq \min(\prod_k i_k) \leq 6.81$.

The maximal transmission ratio (corresponding to Gear I) is determined by considering its gradability and low-speed performance, and its calculation formula is given by

$$\begin{aligned} &\max\left(\prod_k i_k\right) \geq \tfrac{F_{\max} \cdot R}{\eta_t \cdot T_{\max}} \\ &F_{\max} = F_f + F_i + F_w \\ &F_f = mgf \cdot \cos a \\ &F_i = mg \cdot \sin a \\ &F_W = \tfrac{C_D A U_a^2}{21.15} \end{aligned} \tag{7}$$

where $\max\left(\prod_k i_k\right)$ is the maximal value of the total transmission ratio of the electric car, and the meaning of subscript k is same as the above. $F_{\max}$ is the running resistance, $T_{\max}$ is the peak torque, F_f is the rolling resistance, F_i is the slope resistance, and F_W is the air resistance.

According to the above equations, the initial matching of the transmission ratio is completed in Table 2.

Table 2. Initial matching parameters of transmission ratio.

Parameter	Value
Transmission Gear I (low gear) ratio	5
Transmission Gear II (high gear) ratio	2.5

According to the above calculation results, a certain model of PMSM is selected as the driving motor, the specific parameters is shown in Table 3, and its efficiency map is presented in Figure 2. The motor efficiency data revealed by Figure 2 are stored in a two-dimensional table for further calling in the simulation experiment section.

Table 3. Motor parameter.

Parameters	Description	Value	Unit
V_r	Rated voltage	350	V
n_r	Rated speed	3000	rad/s
T_r	Rated torque	160	N.m
P_r	Rated power	45	kW
$n_{\max}$	Maximum speed	7000	rad/s

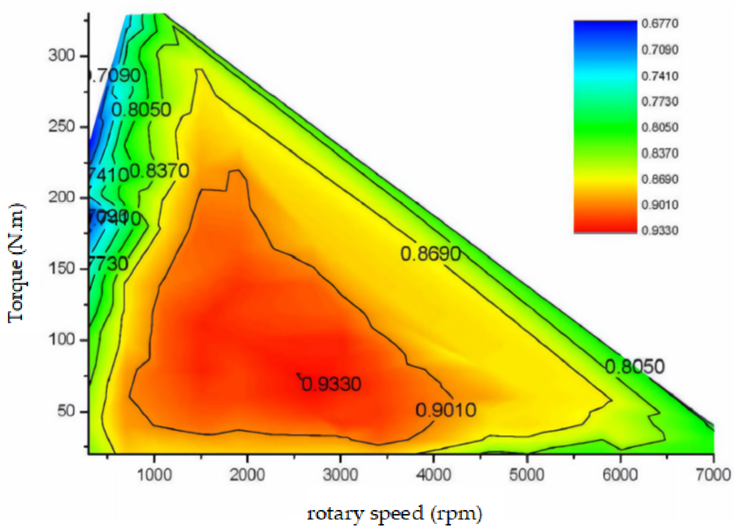

Figure 2. Motor efficiency map.

Compared with other power batteries, lithium batteries have a higher energy density, can be quickly charged, have a long cycle life, and have a high safety factor. In this paper, the lithium battery is selected, and the high-voltage platform is selected. The energy required for the mileage range S of the pure electric car can be calculated by the constant speed method and the working condition method. In the preliminary design, the constant speed method is used for the theoretical calculation of the endurance range, so the resistance power and the energy consumption of the whole car when the pure electric car runs at the constant speed v_a are given by

$$P_a = \frac{v_a}{3600\eta_t}(mgf + \frac{C_D A v_a^2}{21.15}) \tag{8}$$

$$W_r = \frac{P_a S}{\eta v_a} = \frac{(mgf + \frac{C_D A v_a^2}{21.15})L}{3600\eta} \tag{9}$$

where P_a is the power required for constant speed driving, L is the driving mileage, W_r is the energy required for the driving mileage L, and v_a is the travel speed.

Assuming that the effective capacitance coefficient of the power battery pack is 0.9, the energy of the power battery E_b meets the following conditions: $E_b \geq W_r/0.9$. Substituting the W_r value determined by Equation (9) to this inequality obtains $E_b \geq 36.2\text{kW}\cdot\text{h}$. Because the power battery of pure electric cars in the market all use standard boxes, this paper uses a single battery series to form a standard box and selects several standard boxes in parallel to form the power battery. According to the mileage range and motor voltage requirements, the ternary lithium power battery parameters are selected, as shown in Table 4.

Table 4. Power battery parameters.

Parameters	Description	Value	Unit
V_c	Cell voltage	3.6	V
n_b	Number of batterys	100	-
C_p	Battery pack energy	4	kW.h
V_t	Total voltage	360	V
n_c	Number of standard containers	10	-
C_t	Total battery energy	40	kW.h
m_b	Total battery weight	157	kg

The system block diagram and the simulation model are further established as shown in Figures 3 and 4, respectively. It is noted that in this paper the simulation model part is established in Simulink, and the subsequent algorithm part is realized with Matlab programing language in m file form to ensure the two parts into a seamless joint.

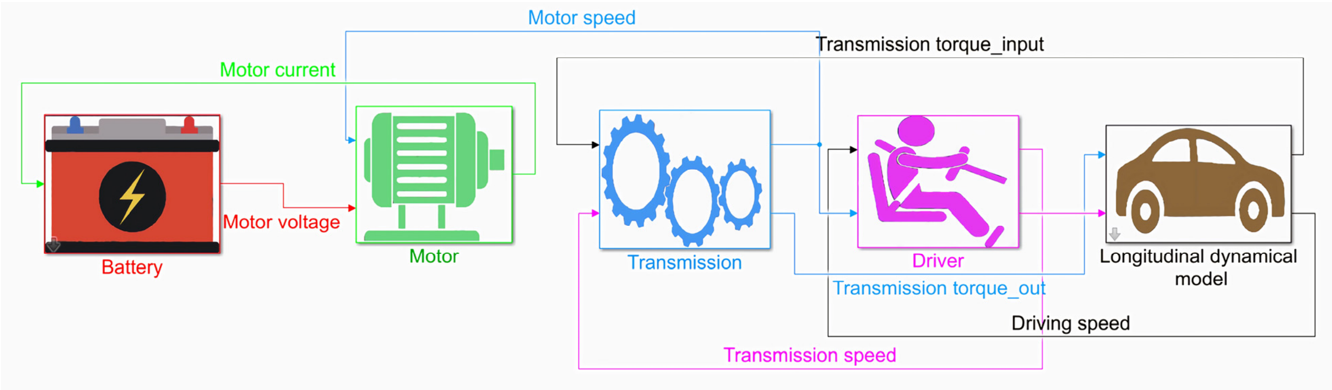

Figure 3. Block diagram of the entire car.

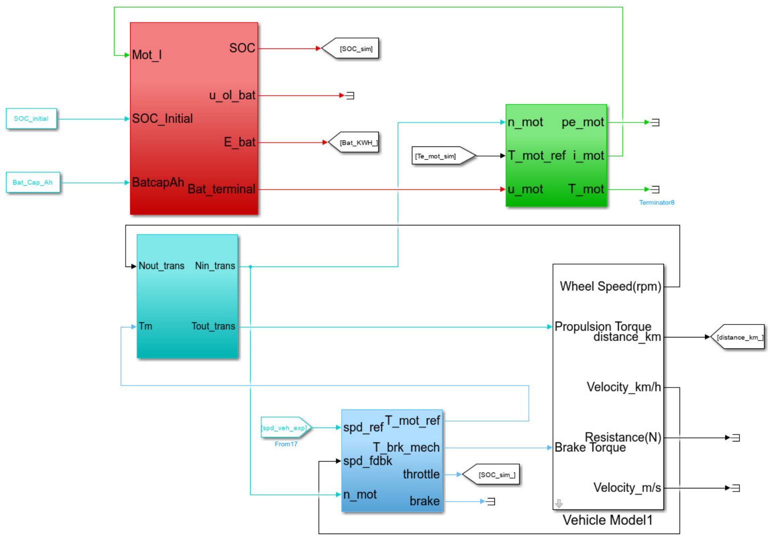

Figure 4. Simulation model of the entire car.

3. Energy-Saving Optimization

Considering that the aforementioned Table 2 is the result of a preliminary calculation and only meets the design requirements, it is necessary to further optimize to obtain an optimal transmission ratio meanwhile seeing that the designed electric car shall be optimal under typical comprehensive working conditions and the particle swarm optimization algorithm is adopted to realize automatic optimization of the whole range. To improve the optimization precision, the particle swarm optimization algorithm is further improved by making two learning factors to decrease nonlinearly in the iteration process. These two learning factors are in the two terms of the velocity update formula, characterizing the individual cognition and swarm cognition weights, respectively.

3.1. Optimal Shifting Based on Particle Swarm Optimization (PSO) Algorithm

3.1.1. Optimization Model

In the optimization, the transmission ratios of the two gears (Gear I and Gear II) are taken as the optimization parameters, namely the optimization variable $X = \{r\_gear1, r\_gear2\}$, where r_gear1 is the total gear ratio of the low-speed gear transmission, and r_gear2 is the total gear ratio of the high-speed gear transmission both without including the gear ratio of main reducer. The optimization objective is to minimize the power consumption of the whole car under NEDC working conditions when the first and second gear transmission ratios are within their respective constraint ranges. Therefor the specific optimization model is as follow.

(1) Objective function:

$$\min F(x) = \min(power\_comsuption) \tag{10}$$

(2) Constraints:

$$\begin{array}{l} 6.31 < r\_gear1 \times r_0 \\ 4 \leq r\_gear2 \times r_0 \leq 6.8 \end{array}$$

where r_0 is the gear ratio of is the main reducer.

3.1.2. Overview of Particle Swarm Optimization

The particle swarm optimization (PSO) algorithm was proposed by Eberhart and Kennedy in 1995, and its idea comes from the swarm intelligence embodied by birds in the process of food foraging [20]. This algorithm can find the global optimal solution with high probability and efficient, and is also easy to implement. The updating rules for each particle in the algorithm are as follows.

The velocity updating rule is given by

$$v_{id}^{k+1} = wv_{id}^{k} + c_1 r_1 (p_{id,pbest}^{k} - x_{id}^{k}) + c_2 r_2 (p_{d,gbest}^{k} - x_{id}^{k}) \tag{11}$$

where w is the inertia weight; d is the particle dimension number; i is the particle number; c_1 is the individual learning factor and c_2 is the swarm learning factor; r_1 and r_2 are both random numbers in the interval [0, 1]; k is the number of iterations; v_{id}^{k} represents the velocity component of particle i in the dth dimension in the kth iteration; $p_{id,pbest}^{k}$ represents the historical optimal position component of the particle i; and $p_{d,gbest}^{k}$ represents the historical optimal position component of the whole swarm in the dth dimension in the kth iteration, which is the optimal solution in the entire swarm after the kth iteration.

The position updating rule is given by

$$x_{id}^{k+1} = x_{id}^{k} + v_{id}^{k+1} \tag{12}$$

where x_{id}^{k} denotes the position vector component of in the dth dimension of particle i after the kth iteration and v_{id}^{k+1} denotes the corresponding velocity vector component.

3.1.3. Improved Particle Swarm Optimization Algorithm

The above standard PSO has a fast convergence speed in the early stage; however, its search precision in the later stage is not so satisfactory. One reason leading to low precision is that the step sizes of the particle are still large relative to the accuracy of solving the objective function. To overcome this drawback, nonlinear learning factors are proposed in this paper, namely the original fix-valued learning factors (namely fixed c1 and c2) are changed to nonlinear learning factors as

$$c_1 = c_{1\_start} * (1/(e^{k/iterMax})^6) + c_{1\_end} \tag{13}$$

$$c_2 = c_{2\_start} * (1/(e^{k/iterMax})^6) + c_{2\_end} \tag{14}$$

where c_{1_start} and c_{2_start} are the initial values of c_1 and c_2, respectively, c_{1_end} and c_{2_end} are the corresponding final values, respectively, $iterMax$ is total iteration times.

The above nonlinear learning factors (Equations (13) and (14)), as newly added updating rules, are executed before Equation (11) in each iteration. The other parts of PSO remain unchanged. The introduction of nonlinear learning factors formed the improved PSO algorithm (hereinafter referred to as the improved PSO). To verify the performance of the improved PSO, a comparative test on minimizing sphere function $f(x) = x_1^2 + x_2^2$, $-10 \leq x_1, x_1 \leq 10$ is completed; and PSO and improved PSO are with the same parameters setting: swarm size is 10 and *iterMax* = 30 except the difference of $c_1 = c_2 = 2$ in PSO, and $c_{1\_start} = c_{2\_start} = 2$ and $c_{1\_end} = c_{2\_end} = 0.5$ in improved PSO. The convergence process of the two algorithms are demonstrated in Figure 5.

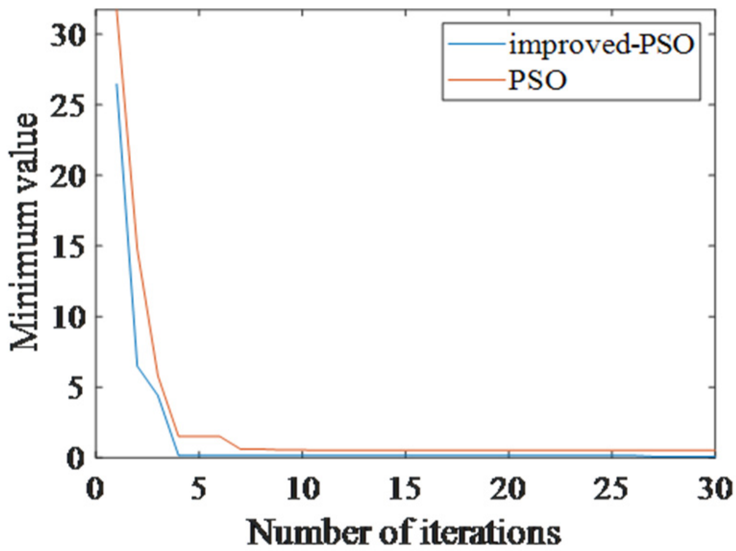

Figure 5. Iteration curves of the two algorithms to optimize sphere function.

The statistics results of the 100 thousands runs are presented in Table 5. From this table, it can be concluded that the improved PSO obviously outperforms PSO.

Table 5. Statistics result comparison of PSO and improved PSO.

Algorithm	Mean	Standard Deviation	Best Value	Worst Value
PSO	11.5632	10.5354	2.8664×10^{-4}	80.7517
Improved PSO	1.8131	3.2205	3.8030×10^{-7}	54.6974

3.1.4. Energy-Saving Optimization Based on Improved PSO Algorithm

Before optimizing, a parameter analysis is completed based on the value range of the design variables, as shown in Figure 6. From this figure, the optimal value ranges of the two variables are around 4~7.5 and 2.5~3.5, respectively. Based on this range, the PSO algorithm is used to find the best parameter values. The flowchart of solving the optimization model using particle swarm optimization algorithm is shown in Figure 7.

To check the performance of improved PSO and ensure to obtain the best transmission ratios, PSO, differential evolution (DE) and genetic algorithms (GA) are also run as the auxiliary comparison algorithms. Figure 8 present the detailed particle distribution, average and optimum values in each iteration of PSO to vefify preliminarily verify the correctness of PSO program. The converging process of all the four algorithms are shown in Figure 9. Table 6 presents the optimal results obtained by the improved PSO.

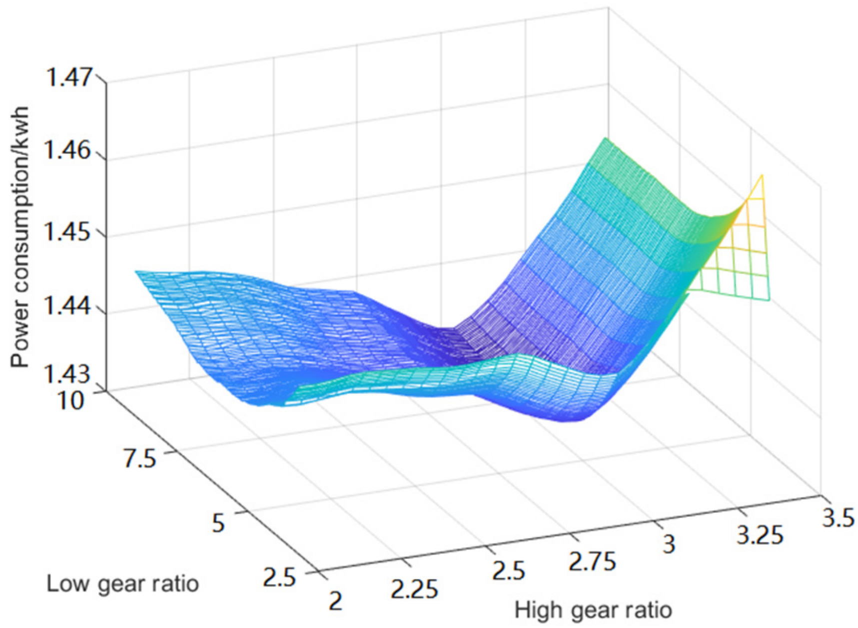

Figure 6. Relation of gear ratios and power consumption.

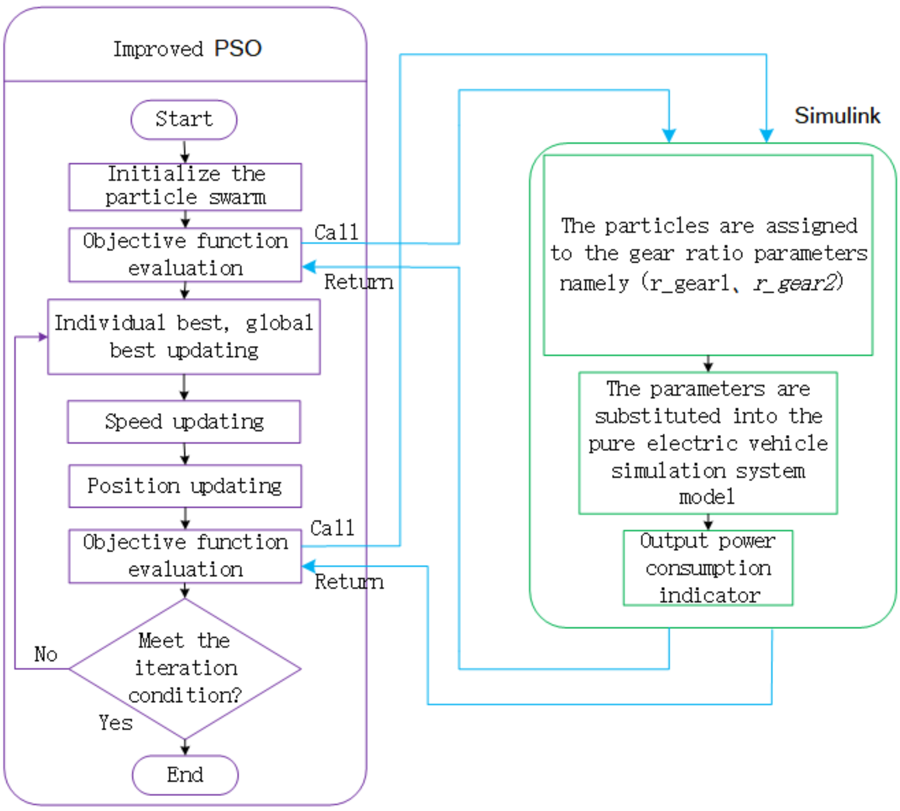

Figure 7. Optimization flowchart.

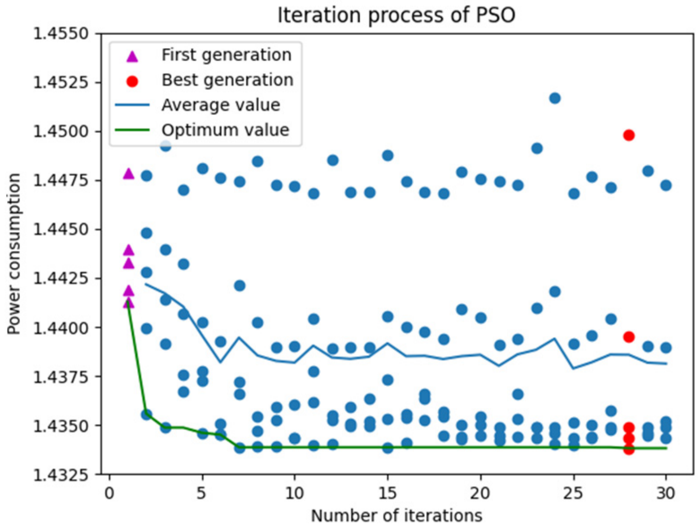

Figure 8. Iteration process of PSO algorithm.

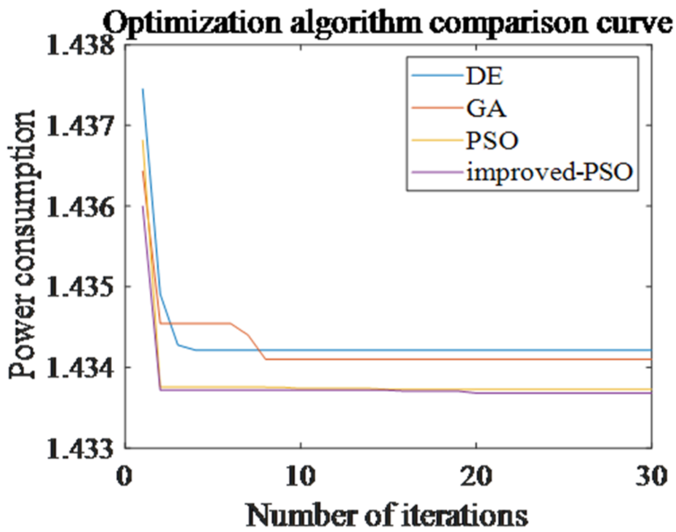

Figure 9. Comparison of four algorithms.

Table 6. Parameter values of transmission ratio after optimization.

Parameter	Value
Transmission gear I (low gear) ratio	6.35
Transmission gear II (high gear) ratio	2.95

3.2. Simulation Experiments

3.2.1. Shift Simulation Based on Maximal Motor Efficiency

The purpose of shift rules based on motor efficiency is to make automatic transmission shift gear with high motor efficiency so as to reduce energy consumption. The idea is to compare the motor efficiency of each gear under the same conditions and choose the gear corresponding to the higher motor efficiency, which uses the follow rule:

$$r\_gear = \begin{cases} r\_gear1 & if(eff1 > eff2) \\ r\_gear2 & if(eff1 < eff2) \end{cases} \tag{15}$$

where r_gear1 is the first gear transmission ratio, r_gear2 is the second gear transmission ratio, $eff1$ and $eff2$ are the motor efficiencies when choosing the first gear transmission ratio and the second gear transmission ratio, respectively.

Power consumption will also be reduced due to the high efficiency of the motor. The car shift frequency simulation results under NEDC conditions are shown in Figure 10.

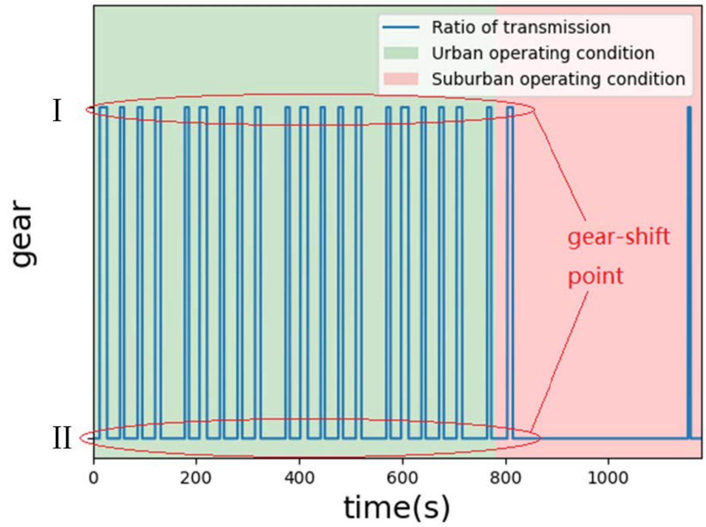

Figure 10. **Gear** shifting curve under different conditions.

3.2.2. Shift Simulation Based on Maximal Economy

The purpose of the economical shift rule is to make automatic transmission shifts at the most economical shift point so as to reduce energy consumption. The idea is to compare whether the first gear or the second gear has lower energy consumption under the same accelerator pedal opening, the energy consumption is determined by the efficiency of the motor during operation, and the efficiency of the motor is obtained by looking up the efficiency map (Figure 2). Taking the 50% opening of the accelerator pedal as an example, the relationship between the efficiency of the driving motor and the car speed is shown in Figure 11.

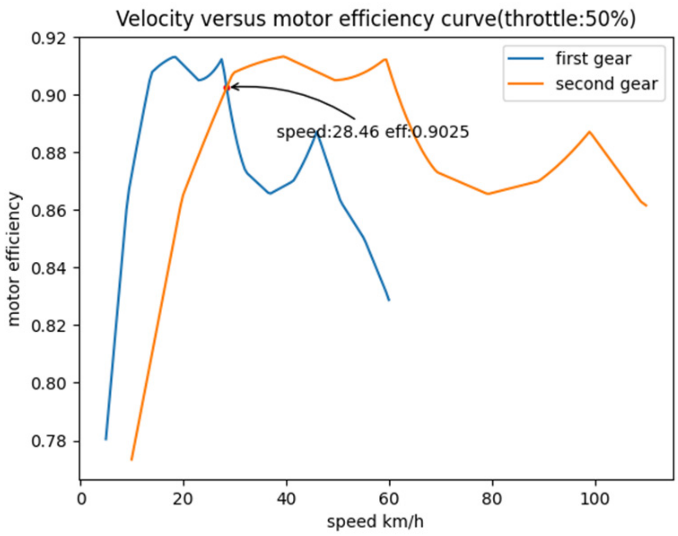

Figure 11. Motor efficiency and car speed relation curve.

It can be seen that when the accelerator pedal opening value is 50%, the car speed is within the range of 0~28.46 km/h, and gear I is engaged, and the efficiency of the driving motor is high. At speeds above 28.46 km/h, the motor efficiency is high when gear II is engaged. The shift point is, therefore, 28.46 km/h at this accelerator pedal opening value of 50%:

$$i = \begin{cases} r\_gear1 & if(r\_gear2 \text{ Conditions not valid}) \\ r\_gear2 & if(0.4 < throttle < 0.5 \ \& \ speed > 28.46) \end{cases} \tag{16}$$

where *throttle* is the accelerator pedal opening, and *speed* is the driving speed.

The same method can also be used to obtain the shift points at other accelerator pedal openings, as shown in Table 7. Connecting these shift points results in a red upshift curve in Figure 12. Similarly, by selecting a downshift speed difference, the blue downshift curve is obtained; thus the entire economy shift rule curves are completed as shown in Figure 12. It is noted that the shift rule with the accelerator pedal opening between 40% and 50% is shown in Equation (16), and the other accelerator pedal opening cases can be formulated similar to Equation (16). The resultant shift curves according to the economy shift rule curve is shown in Figure 13.

Table 7. Shift points of economy shift rules.

Valve Opening	Shift Point (km/h)	Valve Opening	Shift Point (km/h)
10%	29.39	60%	30.42
20%	28.32	70%	31.42
30%	22.5	80%	32.26
40%	25.56	90%	34.06
50%	28.46	100%	39.77

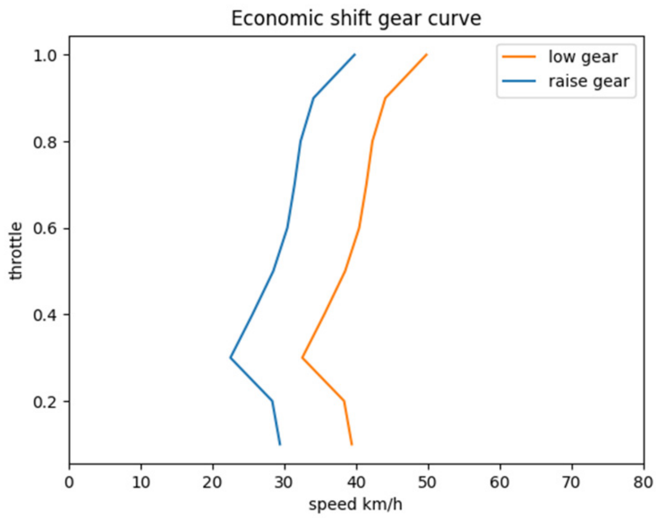

Figure 12. Economy shift schedule curve.

3.2.3. Final Selection of Shift Rules

The shift optimization based on the motor efficiency and the economic shift optimization is adopted to improve the motor efficiency as much as possible when the car runs in different working conditions so as to improve the economy. In fact, the efficiency range through normal driving conditions to reach the peak efficiency zone is between 65% and 95% [22]. In electric vehicles, peak efficiency is sacrificed in order to achieve a better performance curve over a wider speed range. An efficiency of 75% is considered a good quality factor for small variable speed motors [23]. In this design, the simulation experiment is conducted on motor efficiency under NEDC operating conditions with different shift strategies, and the results are shown in Figure 14.

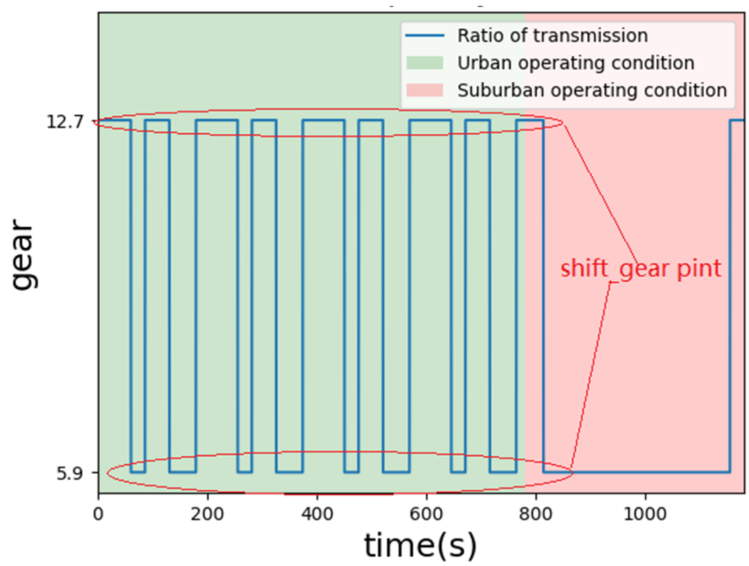

Figure 13. Shift curve. with maximal economy.

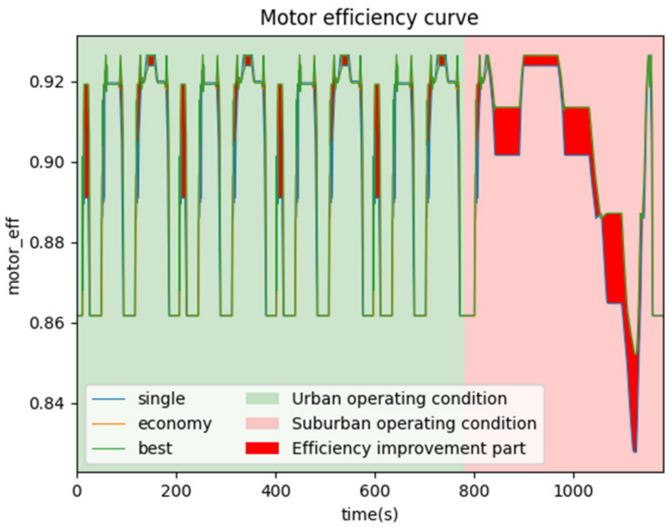

Figure 14. Motor efficiency curve.

It can be seen from Figure 14 that the shift times of the economic shift optimization are much less than that of the optimal shift optimization, but the motor efficiency is almost the same in the actual operating conditions, and it is better with two-gear shifting than that of the electric car without the transmission for most of the time, especially in the late stage of the operating conditions of high speed.

Because the optimal shifting principle only relies on a single parameter of motor efficiency, frequent gear shifting may occur during actual operation under complex working conditions. Frequent gear shifting in actual operation produces a more serious impact on the driving experience and is not suitable for practical use. The economic shift considers the motor efficiency and the throttle opening and gives the referable shift schedule curve. The shifting frequency under NEDC conditions is greatly reduced using the economic shift principle compared with the motor efficiency shift principle. Therefore, this paper finally chooses the economic shift principle, and the subsequent tests are all adopt the economic shift principle.

4. Dynamic Performance Test

4.1. Accelerated Performance Test

Compared to the sinle fixed ratio transmission of electric car, this paper proposes a new two-speed transmission, optimizes its gear ratios and determine the shifting principle. To test the dynamic performance and meanwhile check impact of using the economic shift principle on the power performance, the acceleration performance simulation experiments of 0 to 50 km/h and 0 to 100 km/h are carried out respectively in this section, and the experimental results are shown in Figures 15 and 16. Table 8 shows the performance comparison data.

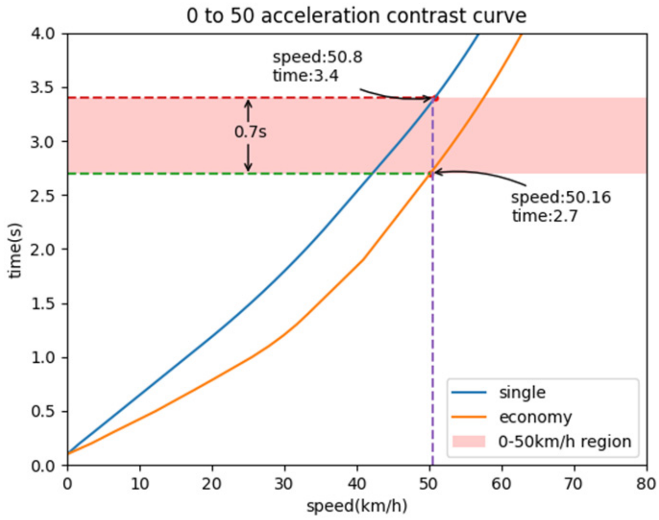

Figure 15. Acceleration curve of 0 to 50 km/h.

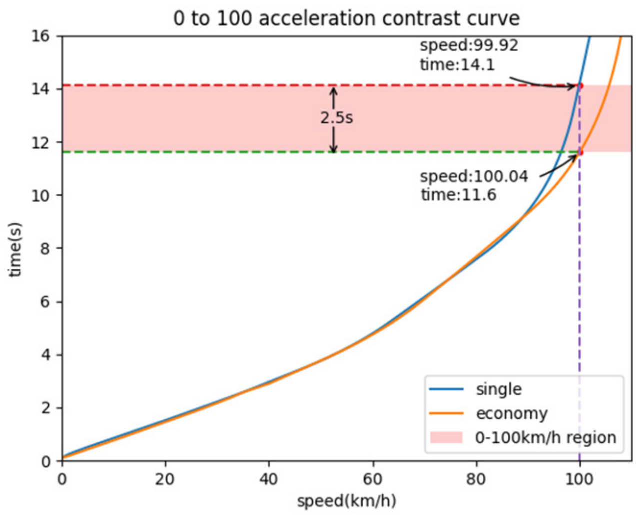

Figure 16. Acceleration curve of 0 to 100 km/h.

Table 8. Acceleration performance index comparison.

Performance Index	Acceleration Time of 0 to 50 km/h		Acceleration Time of 0 to 100 km/h	
	Before Optimization	After Optimization	Before Optimization	After Optimization
Acceleration time	2.7 s	3.4 s	11.6 s	14.1 s
Performance improvement	20.6%		17.7%	

According to the figures obtained from the simulation experiment, it can be seen that in the 0 to 50 km/h acceleration case, although the maximum transmission ratio of the two-gear transmission is larger than that of the single fixed ratio transmission, the acceleration time is about 0.7 s faster than that of the electric car with the single fixed ratio transmission, and even under the influence of the shift strategy, the acceleration time is improved by about 20.6%. for the acceleration case from 0 to 100 km/h, the acceleration time of two-gear shift electric car is about 2.5 s or about 17.7% (relative percentage) less than that of the sinlge fixed ratio transmission electric car.

4.2. High-Speed Performance Test

Since the two-gear transmission has a smaller ratio, it will have better performance under the high-speed working condition. Therefore, the simulation experiment of the maximum speed that can be achieved by different schemes under the acceleration duration of 150 s is designed, and the results are shown in Figure 17.

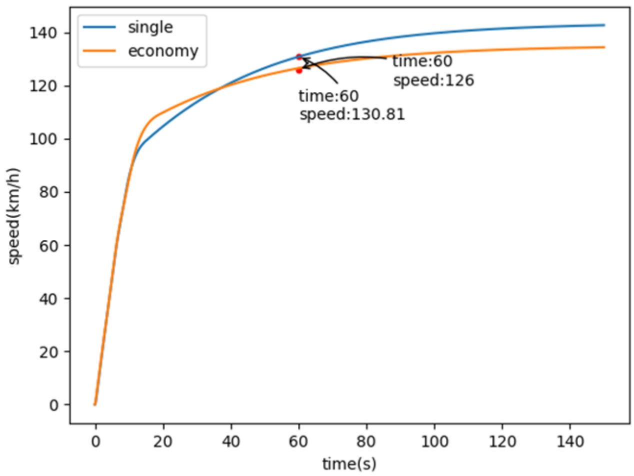

Figure 17. Maximum speed simulation.

The simulation results show that the acceleration curves of the two schemes tend to be flat after 60 s of continuous acceleration. The maximum speed of the two-gear transmission is 126 km/h in a short time, while the maximum speed of the single main reducer electric car is 130.81 km/h in a short time, which is reduced by about 3%. The reason for this is that the single-speed transmission has a larger gear ratio than the two-speed transmission, and when the motor torque is constant, the transmission provides a larger torque, so the maximal car speeds have a slight difference.

5. Energy-Efficiency Test

5.1. Comparison of Power Consumption

The NEDC and CLTC test cycles are both adopted for comparison. The NEDC (abbreviation of the New European Driving Cycle) consists of the urban driving cycle and the suburban driving cycle. The urban driving cycle consists of four cycle units, with a maximum speed of 50 km/h, an average speed of 19 km/h during the test, each cycle running time of 195 s, and a total driving distance of 4.052 km. There is one cycle unit in the suburban driving test, with an average speed of 62.6 km/h, an effective driving time of 400 s, and a total driving distance of 6.955 km. This operating cycle is shown in Figure 18. CLTC is the abbreviation of the "China Light Vehicle Test Cycle", which stands for the driving conditions of light-duty vehicles in China. It includes urban, suburban, and high-speed driving conditions, with accumulated mileage of 14.48 km, a maximum speed of 114 km/h, an average speed of 28.96 km/h, and a cycle time of 1800 s. This operating cycle is shown in Figure 19.

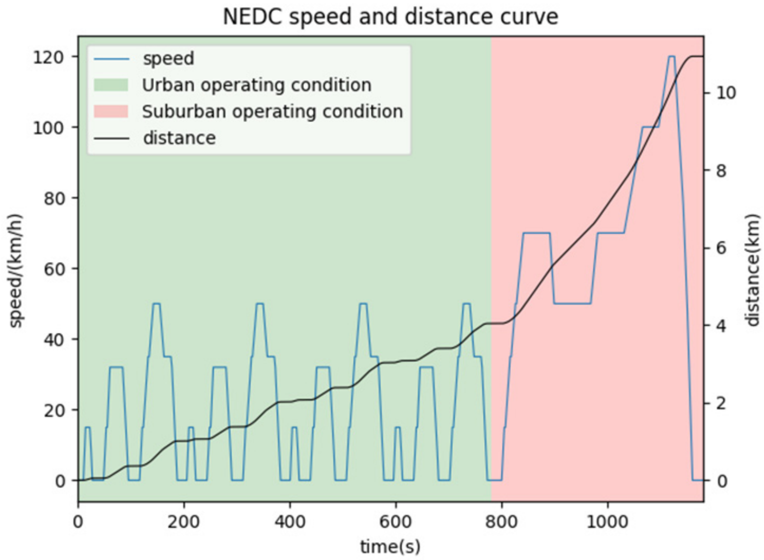

Figure 18. NEDC cycles.

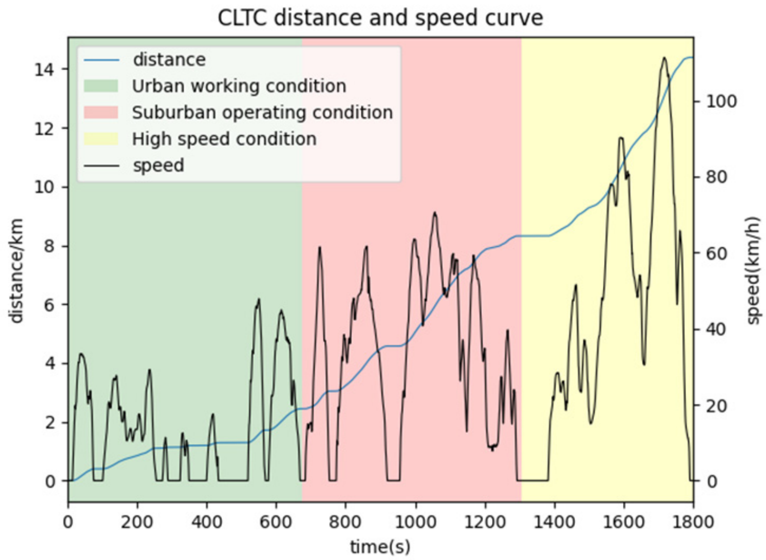

Figure 19. CLTC cycles.

The power consumption of this pure electric car equipped with the two-gear shifting and the pure electric car equipped with the single fixed ratio transmission is compared, and the comparison results under NEDC and CLTC conditions are shown in Figures 20 and 21. Detailed data are shown in Table 9.

Table 9. Comparison of power consumption data.

Type	NEDC (kWh)	CLTC (kwh)
Single fixed ratio transmission	1.4615	1.7486
Two-speed transmission	1.4350	1.7295

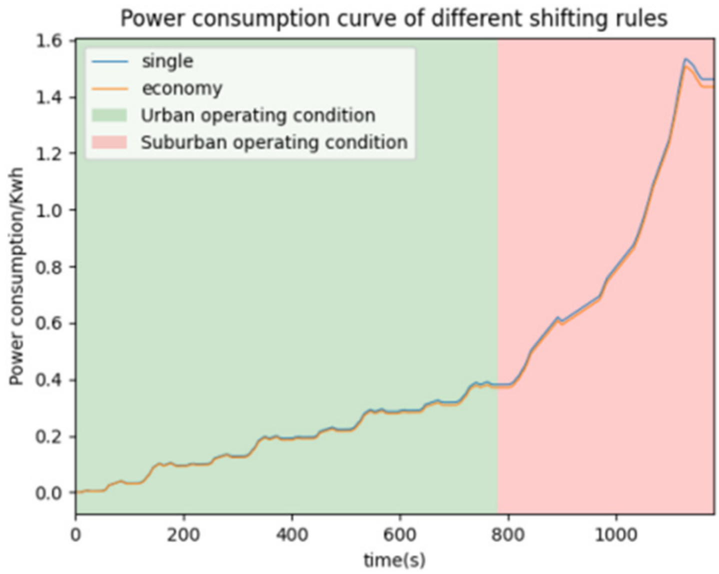

Figure 20. NEDC power consumption.

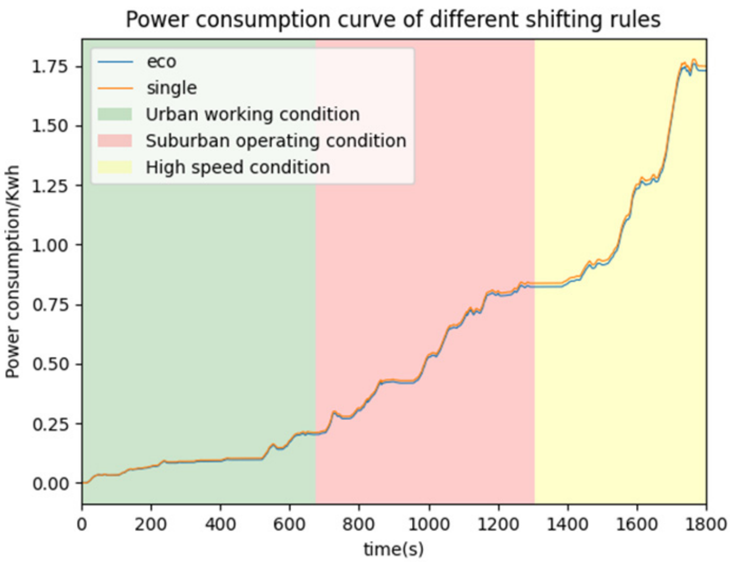

Figure 21. CLTC power consumption.

According to the above simulation results, we can conclude that the power consumption of the pure electric car with the two-speed transmission is much lower than that of the pure electric car with single fixed ratio transmission in a NEDC working condition. The specific value is that the power consumption of a single-stage final drive is 1.4615 kWh, and the power consumption of a pure electric car with the two-speed transmission is 1.4350 kWh, which is reduced by 0.0265 Wh, which is nearly 1.8% reduction compared with that of a pure electric car with the two-speed transmission. The power consumption of the single fixed-ratio transmission is 1.7486 kWh under CLTC working conditions, and the power consumption of pure electric cars with two-gear transmission is 1.7295 kWh, which is reduced by 0.0191 Wh and 1.1% reduction. However, since the transmission gearbox is added, the curb weight of the car will increase, and its economy will also decrease. Therefore, the simulation experiment on the influence of increased weight on energy consumption is conducted, and the experiment result is shown in Figure 22. It can be seen from the test results that the influence of weight increase on energy consumption is approximately proportional. When the weight of the two-speed transmission is 40 kg, the power consumption will increase by about 0.0258 kWh under NEDC conditions. It follows that there is still room for optimization of a two-speed transmission compared to a single fixed-ratio transmission when the weight added to the reducer is less than 40 kg. It can also be seen that the optimization space is small under CLTC conditions.9

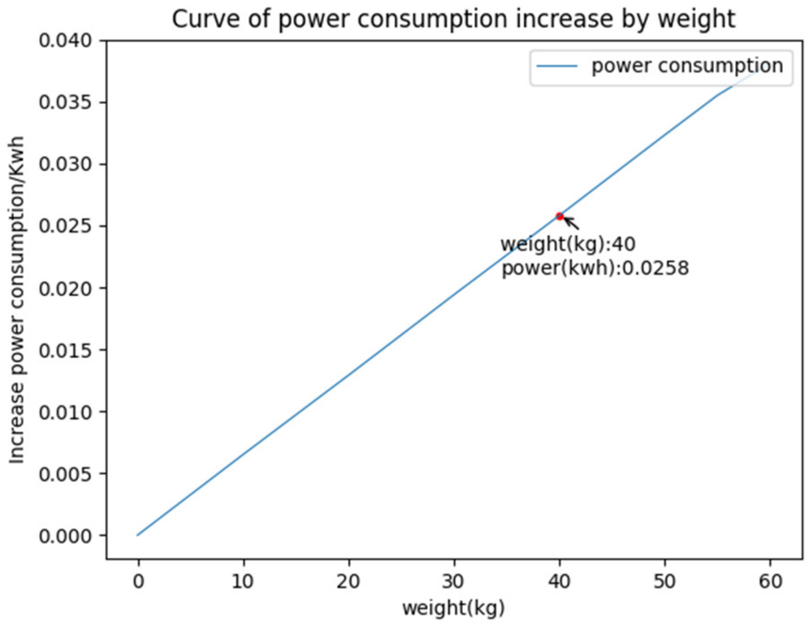

Figure 22. Effect of transmission weight on power consumption.

5.2. Mileage Range Comparison

According to the power consumption data made in the previous part, the driving range of different schemes under NEDC working conditions can be calculated. The total battery capacity adopted in this paper is 40 kWh, the whole journey time under NEDC working conditions is 1180 s, the driving range is about 11 km, the whole journey time under CLTC working conditions is 1800 s, and the driving range is 14.48 km. The range data thus calculated are shown in Table 10 below.

Table 10. Mileage range data.

Protocol	Total NEDC Range (km)	CLTC Total Range (km)
Single fixed ration transmission	301	331
Two-speed transmission	307	334

According to the above data analysis, under NEDC conditions, the mileage range of the electric car with two-speed transmission is 6 km more than that of the electric car with a single fixed-ratio transmission, which is about 2% higher. Under CLTC conditions, the mileage range of the former is 3 km longer than that of the latter, with an increase of about 1%.

6. Discussion

To summarize the numerical results of Sections 4 and 5, under the optimal shifting, the electric car equipped with two-speed transmission has improvements in both dynamic performance and energy-saving indicators. The acceleration time of 0 to 100 km/h is decreased by 17.7%, and the power consumption is reduced by 1.8%. The dynamic performance has great improvement, which can bring a better driving experience; meanwhile, the energy efficiency also has a little improvement, and because of the weight increase of introducing transmission gearbox, the improvement is not so large. However, it does not cause efficiency reduction and ensures the feasibility of a practical application.

To sum up, the feasibility of applying multiple-gear shifting to small electric cars is verified, and the experimental results provide a valuable reference for the development of electric cars.

7. Conclusions

This paper focuses on investigating the performance indicators improvement of electric cars to be added multi-gear transmission. The following works are completed: the power system of the EV was first matched, and the parameter matching optimization model is established. For two-gear transmission, the improved particle swarm optimization algorithm was proposed to solve the appropriate transmission ratio of each gear, and the shift strategy of two-gear transmission is designed according to the determined transmission ratio. According to the indexes of the designed car, the simulation comparison of two different transmissions is carried out.

The simulation experiment results show that the designed car achieves all the performance indexes requirement, and the electric car equipped with the two-gear transmission is better than the electric car with only single fixed-ratio transmission including 17.7% and 1.8% improvements in acceleration performance and energy efficiency respectively. Therefore adding a two-gear transmission is feasible and promising. In the future, the structural design and shifting control of the two-gear shifting mechanism will be further studied.

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References

1. Chen, Y.; Wang, J. Energy-efficient control allocation for over-actuated systems with electric vehicle applications. *Dyn. Syst. Control Conf.* **2011**, *1*, 37–44. [CrossRef]
2. Hua, M.; Chen, G.; Zhang, B.; Huang, Y. A hierarchical energy efficiency optimization control strategy for distributed drive electric vehicles. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2019**, *233*, 605–621. [CrossRef]
3. Zheng, C.; Xu, G.; Cha, S.W.; Liang, Q. A Predictive driving control strategy of electric vehicles for energy saving. *Int. J. Precis. Eng. Manuf.* **2015**, *16*, 197–202. [CrossRef]
4. Hu, J.; Xiao, F.; Mei, B.; Lin, Z.; Fu, C. Optimal energy efficient control of pure electric vehicle power system based on dynamic traffic information flow. *IEEE Trans. Transp. Electrif.* **2021**, *8*, 510–526. [CrossRef]
5. Chen, G.; Hua, M.; Zong, C.; Zhang, B.; Huang, Y. Comprehensive chassis control strategy of FWIC-EV based on sliding mode control. *IET Intell. Transp. Syst.* **2019**, *13*, 703–713. [CrossRef]
6. Ning, G.; Xiong, L.; Zhang, L.; Yu, Z. Method of electric powertrain matching for battery-powered electric cars. *Chin. J. Mech. Eng.* **2013**, *26*, 483–491. [CrossRef]
7. Gao, B.; Liang, Q.; Xiang, Y.; Guo, L.; Chen, H. Gear ratio optimization and shift control of 2-speed I-AMT in electric vehicle. *Mech. Syst. Signal Process.* **2015**, *51*, 615–663. [CrossRef]
8. Fang, S.; Song, J.; Song, H.; Tai, Y.; Li, F. Design and control of a novel two-speed uninterrupted mechanical transmission for electric vehicles. *Mech. Syst. Signal Process.* **2016**, *75*, 473–493. [CrossRef]
9. Fu, B. Influencing factors of electric vehicle economy based on continuously variable transmission. *Int. J. Automot. Technol.* **2022**, *23*, 717–728. [CrossRef]
10. Liu, W.; Bi, S.; Su, Z. Two-speed pure electric vehicle AMT transmission. In *Emerging Trends in Intelligent and Interactive Systems and Applications. IISA 2020. Advances in Intelligent Systems and Computing*; Tavana, M., Nedjah, N., Alhajj, R., Eds.; Springer: Cham, Switzerland, 2021; Volume 1304. [CrossRef]
11. He, H.; Liu, Z.; Zhu, L.; Liu, X. Dynamic coordinated shifting control of automated mechanical transmissions without a clutch in a plug-in hybrid electric vehicle. *Energies* **2012**, *5*, 3094–3109. [CrossRef]

12. Liu, T.; Zeng, X.; Song, D. MPC-based coordinated control of gear shifting process for a power-split hybrid electric bus with a clutchless AMT. *Chin. J. Mech. Eng.* **2022**, *35*, 144–157. [CrossRef]
13. Hu, J.; Ran, H.; Pang, T.; Zhang, Y. Parameter design and performance analysis of shift actuator for a two-speed automatic mechanical transmission for pure electric vehicles. *Adv. Mech. Eng.* **2016**, *8*, 1–15. [CrossRef]
14. Qin, D.; Yao, M.; Chen, S.; Lyu, S. Shifting process control for two-speed automated mechanical transmission of pure electric vehicles. *Int. J. Precis. Eng. Manuf.* **2016**, *17*, 623–629. [CrossRef]
15. Jaehoon, C.; Suh, J.; Nogill, P. Optimal design of lightweight two speed transmission of electric vehicles. *Tribol. Lubr.* **2020**, *36*, 96–104. [CrossRef]
16. Angeles, J.; Roozegar, M. A two-phase control algorithm for gear-shifting in a novel multi-speed transmission for electric vehicles. *Mech. Syst. Signal Process.* **2018**, *104*, 145–154. [CrossRef]
17. Eckert, J.J.; Santiciolli, F.M.; Silva, L.C.A.; Dedini, F.G. Vehicle drivetrain design multi-objective optimization. *Mech. Mach. Theory* **2021**, *156*, 104–123. [CrossRef]
18. Hu, Y.; Liu, C.; Xiong, G.; Xi, J. Optimization of gear shift schedule for electric buses equipped with 4-AMT using dynamic programming. In Proceedings of the 2015 IEEE Intelligent Vehicles Symposium (IV), Seoul, Republic of Korea, 28 June–1 July 2015; pp. 431–436. [CrossRef]
19. Liang, Q.; Gao, B.Z.; Chen, H. Gear shifting control for pure electric vehicle with inverse-AMT. *Appl. Mech. Mater.* **2012**, *190–191*, 1286–1289. [CrossRef]
20. Kennedy, J. Particle swarm optimization. In Proceedings of the ICNN'95—International Conference on Neural Networks 2010, Perth, Australia, 27 November–1 December 1995; Volume 4, pp. 760–766. [CrossRef]
21. Wang, M.; Lu, G. A modified sine cosine algorithm for solving optimization problems. *IEEE Access* **2021**, *9*, 27434–27450. [CrossRef]
22. Li, Y.; Liu, M.; Lau, J.; Zhang, B. A novel method to determine the motor efficiency under variable speed operations and partial load conditions. *Appl. Energy* **2015**, *144*, 234–240. [CrossRef]
23. de Santiago, J.; Bernhoff, H.; Ekergård, B.; Eriksson, S.; Ferhatovic, S.; Waters, R.; Leijon, M. Electrical motor drivelines in commercial all-electric vehicles: A review. *IEEE Trans. Veh. Technol.* **2012**, *61*, 475–484. [CrossRef]