Available online www.jsaer.com

Journal of Scientific and Engineering Research, 2023, 10(2):55-62

Research Article

ISSN: 2394-2630
CODEN(USA): JSERBR

Power Matching Simulation Optimization of Pure Electric Car based on MATLAB

ZHANG yu-xue*, ZHAO xu, ZHAO pei-dong, ZHANG yao-wen

*College of Transportation and Vehicle Engineering, Shandong University of Technology, China

Abstract In order to protect the environment, save energy and reduce emissions, major car companies have focused on the development of new energy vehicles. However, some key technologies of pure electric vehicles have not yet achieved phased breakthroughs, so the reasonable matching of the power system has become an important way to improve the power performance of the vehicle. This paper adopts the method of joint simulation of MATLAB and Cruise. Cruise establishes the vehicle model, and MATLAB establishes the control strategy model. Finally, the simulation results are analyzed and the vehicle model is optimized by changing the transmission ratio.

Keywords Pure electric vehicles, Power matching, Joint simulation

1. Introduction

In recent years, with the breakthrough of lithium battery technology, the cost of lithium battery as power battery has been greatly reduced, which indirectly leads to the rise of pure electric vehicle industry [1-3]. Pure electric vehicles in various countries are also increasing rapidly. For new users who buy vehicles, more and more people tend to buy pure electric vehicles and new energy vehicles [4]. Although global sales of pure electric vehicles have increased dramatically in recent years, they still account for only 3 % of total vehicle sales [5]. However, in the current situation, pure electric vehicles have great development potential. Countries are increasing their promotion policies for electric vehicles [6], and have issued and implemented new electric vehicle development strategies to further promote the development of electric vehicles.

For the design and optimization of power system parameters of pure electric vehicles, there are studies at home and abroad. Among them, Qian Juan [7] proposed a multi-objective optimization of automobile transmission system parameters based on particle swarm optimization algorithm. The proposed particle swarm optimization algorithm can effectively solve the problem of transmission system parameter optimization. The main optimization method is to take the shortest acceleration time of 0 ~ 100km / h and the lowest fuel consumption of 100 km as the optimization objectives of power performance and fuel economy respectively. The accurate simulation of ADVISOR is combined with the efficient optimization of PSO algorithm to obtain the optimal solution. Wang Fei et al. [8] used Simulink to establish the model of hybrid electric vehicle, and used genetic algorithm to optimize the parameters. The results show that the optimized vehicle performance has been greatly improved compared with that before optimization.

2. Dynamics analysis and parameter matching

The main difference between pure electric vehicles and traditional cars is that one is based on the motor as the power source, and the other is based on the engine as the power source, but its mechanical properties are not fundamentally different from traditional cars. Therefore, we can use the analysis of the force of the transmission car to carry out mechanical analysis. At present, there are main evaluation indexes for the main performance of electric vehicles, which can be used to judge the performance of electric vehicles.

2.1 Force Analysis of Pure Electric Vehicle

In order to better match the power, the longitudinal force analysis of the car is essential. The detailed longitudinal dynamic analysis can determine the parameter limit under the design goal, so as to select the appropriate parts model in the power system. The expression of vehicle force equation is:

$$F_t = F_w + F_f + F_i + F_j \quad (1)$$

Fig.1 is the driving force diagram of the car during climbing ; ft represents the driving force of the car ; f_w represents air resistance ; ff represents rolling resistance ; fi represents climbing resistance ;

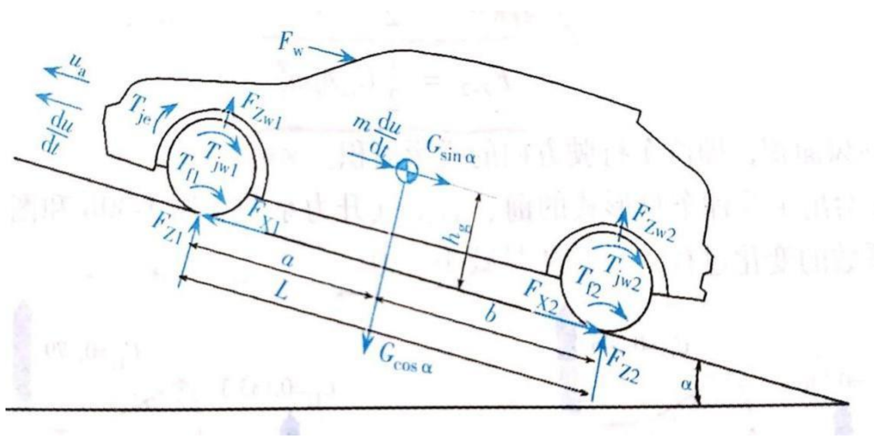


Figure 1: Vehicle driving force diagram

2.2 Theoretical calculation of drive motor

For the selection of motor power, if the selected power is large, it will increase the backup power of the vehicle and improve the dynamic performance, but too large will keep the motor in a low power state and low utilization rate. Moreover, the selection of high-power electric opportunities will increase the volume and mass of the motor, increase the power consumption, and reduce the mileage. If the power is too small, it will not be able to meet the basic driving conditions. Therefore, we need to select the most suitable motor through theoretical calculation. The peak power calculation formula at the maximum speed:

$$p_{u\max} \geq \frac{1}{\eta_t}\left(\frac{mgf}{3600}u_{\max} + \frac{C_D A}{76140}u_{\max}^2\right) \quad (2)$$

Among them, is the transmission efficiency, the value of 0.9; g is the acceleration of gravity, with a value of 9.8; to design the maximum speed; the air resistance coefficient; a is the windward area; the rolling resistance coefficient. The peak power calculation formula under the maximum climbing degree:

$$p_{\alpha\max} \geq \frac{u_{\min}}{3600\eta_t}\left(mgf\cos\alpha_{\max} + mg\sin\alpha_{\max} + \frac{C_D A u_{\min}^2}{21.15}\right) \quad (3)$$

Among them, the stable speed of ramp driving is 15 km / h ; to design the maximum climbing degree ; the power calculation formula required to meet the acceleration time condition :

$$p_{js} \geq \frac{u}{3600\eta_t}\left(mgf + \frac{\delta m du}{dt} + \frac{C_D A u^2}{21.15}\right) \quad (4)$$

Among them, to accelerate the end speed; rotating mass conversion coefficient, value 1.1; du / dt is the vehicle acceleration. In order to make the design calculation more simple and convenient, the above formula can be modified according to the empirical formula:

$$v = v_m\left(t/t_m\right)^{0.5} \quad (5)$$

Among them, for the car to accelerate the speed; to accelerate time
Substitute (5) into (4) to obtain

$$p_{js} \geq \frac{1}{3600 t_m \eta_t}(\delta m \frac{u_m^2}{2} + mgf \frac{u_m}{1.5} t_m + \frac{C_D A u_m^3}{21.15 \times 2.5} t_m) \quad (6)$$

According to the above formula, the vehicle parameters are substituted into the calculation to obtain the motor power requirements that meet the major conditions.

2.3 Theoretical calculation of motor speed

The pure electric vehicle has only one power source of the motor, and its maximum speed is closely related to the peak speed of the motor. Therefore, we can calculate the maximum speed of the motor according to the set maximum speed. The calculation formula is

$$n_{\max} = \frac{u_{\max} i_0}{0.377 r_w} \quad (7)$$

Among them, the highest speed; the maximum speed; for the transmission ratio of the transmission system; for the radius of the tire.

The speed of pure electric vehicles is closely related to the speed of the drive motor, so the rated speed of the motor is best to select the speed corresponding to the speed of the vehicle at most times. The formula is:

$$n_b = \frac{3.6 i u_n}{2\pi / 60 r_w} \quad (8)$$

According to the force analysis of the car in the climbing process, the formula is:

$$T_{\max} = \frac{mgf \cos(arctg\phi_{\max}) r_w}{i\eta_t} + \frac{mg \sin(arctg\phi_{\max}) r_w}{i\eta_t} \quad (9)$$

Among $T_{\max}$ is the peak torque.

2.4 Transmission system matching calculation

The maximum transmission speed ratio is determined from the direction of force, and the maximum transmission speed ratio should meet the requirements of maximum climbing degree. The calculation formula is:

$$i_{\max} \geq \frac{(mgf \cos\alpha_{\max} + mg \sin\alpha_{\max}) r}{T_{\max} \eta_t} \quad (10)$$

Among them, the maximum climbing degree; maximum torque of the motor.

The minimum transmission ratio is determined from the speed direction, and the minimum transmission ratio should meet the requirements between the maximum speed and the maximum speed of the motor. The calculation formula is:

$$i_{\max} \leq \frac{0.377 n_{\max} r}{v_{\max}} \quad (11)$$

where r is the wheel radius; maximum motor speed; maximum speed.

3. Co-simulation of MATLAB and Cruise

In this paper, the establishment of the model is mainly realized by the joint simulation of MATLAB and Cruise. The control strategy is established by MATLAB, and the modules of each component are established by Cruise.

3.1 Crusie builds vehicle model

Select the corresponding automotive components from the component library, place them in the vehicle modeling work window, and establish connections for each component. The connection is mainly divided into three parts, mechanical connection, power connection and signal connection. In the following vehicle model Figure 2, the blue line represents the mechanical connection, the red represents the power connection, the red arrow at the rectangular angle of each module represents the signal output, and the green represents the signal input.

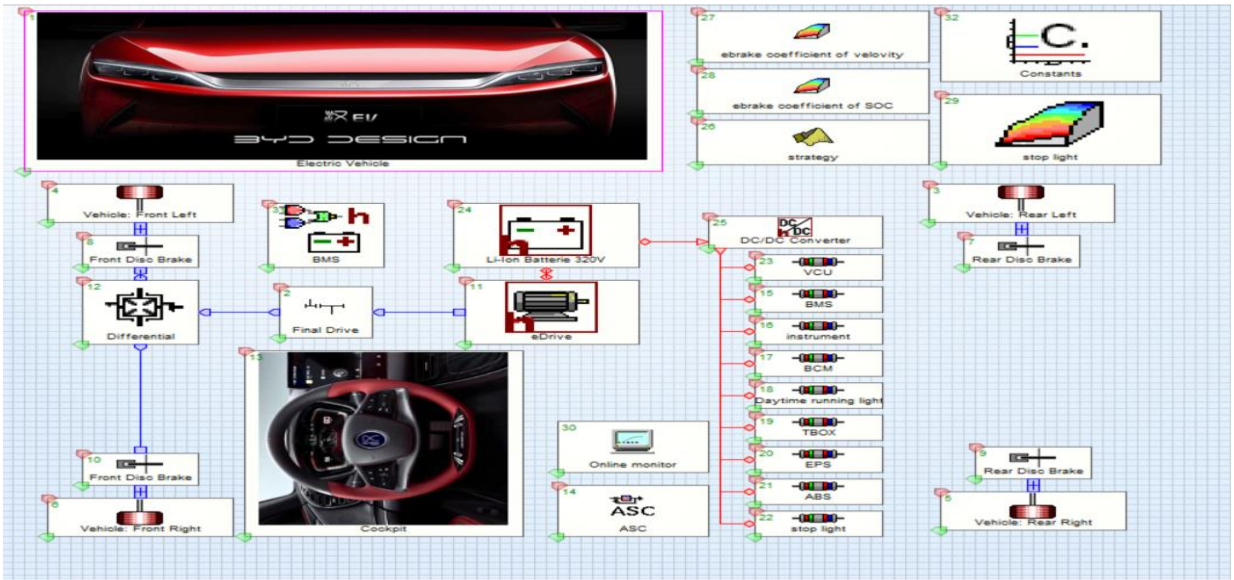


Figure 2: Vehicle model diagram

3.2 Establishment of control strategy

The main goal of the control strategy is energy recovery. The main control idea is to calculate the braking force of the wheel end to the motor end when the vehicle is in the braking state, and to judge whether the motor meets the braking force demand. If it is satisfied, the electric braking is adopted. If it is not satisfied, the residual braking force demand of the electric braking is given priority to be supplemented by the mechanical braking. When the vehicle is not in the braking state, the motor is in the driving mode, and the motor load is equal to the accelerator pedal opening * motor external characteristics, as shown in Figure 3.

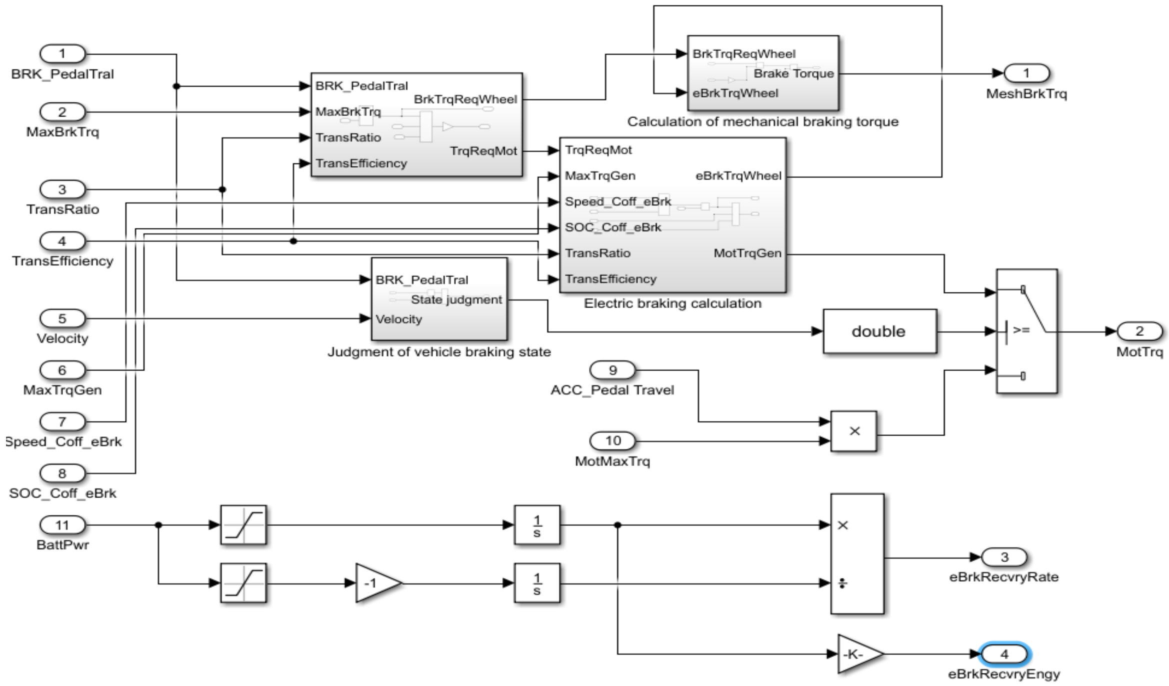


Figure 3: Control strategy

3.3 Co-simulation

In the Cruise vehicle model, the MATLAB DLL module is added to specify the location of the DLL file, and the input and output parameters are set up. Set up as shown in Figure 4 below, and then connect the MATLAB DLL module signal.

Description of Data Bus

hannel	Description	Unit	Connection	Decouple	Invert
	Brake_Pedal_Travel	-	required		unchanged
	Maximum Brake Torque	Nm	required		unchanged
	Transmission Ratio of Single Ratio Transmission	-	required		unchanged
	Transition efficiency	-	required		unchanged
	velocity	km/h	required		unchanged
	Maximum Torque - Generator	Nm	required		unchanged
	Speed_Coff_Ebrake	-	required		unchanged
	SOC_Coff_Ebrake	-	required		unchanged
	ACC Pedal Travel	-	required		unchanged
	MotMaxTrq	Nm	required		unchanged
	BattPwr	kW	required		unchanged
	MotTrq	Nm	optional		unchanged
	Brake Torque	Nm	optional		unchanged
	eBrkRecvryRate	-	optional		unchanged
	eBrkRecvryEngy	kWh	optional		unchanged

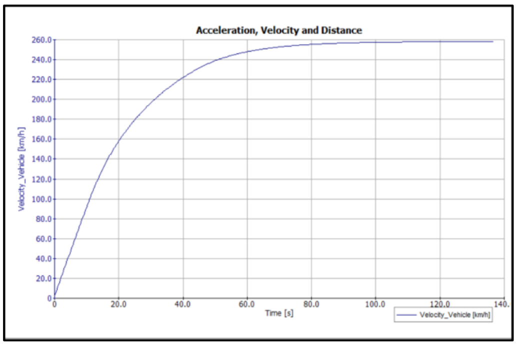
Figure 4: MATLAB DLL module input and output parameters

4. Simulation results and optimization

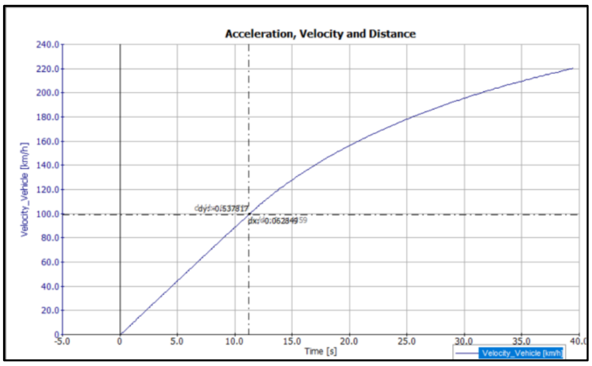
The simulation analysis is carried out by Cruise, and the results are compared with the design objectives. Then the transmission ratio is optimized. Finally, the optimized results are simulated.

4.1 Dynamic simulation results

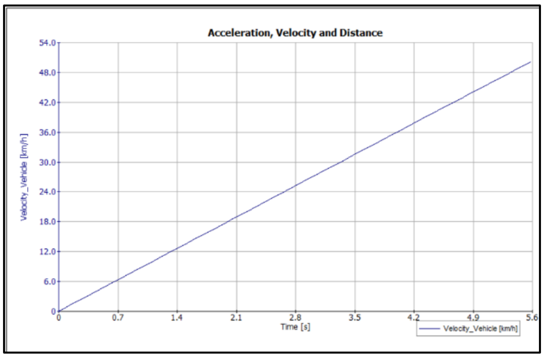
The maximum speed is the main index representing the power performance of the car. It shows that the car can drive to the highest speed and can maintain a speed of more than 1km. It is mainly determined by the power of the motor. In figure 5, it can be seen that the maximum speed reaches 258km / h. The acceleration time of 100 km is 11.33km / h, the acceleration time of starting acceleration (0 ~ 50km / h) is 5.56s, the acceleration time of medium speed section (50 ~ 80km / h) is 3.39s, and the acceleration time of high speed section (80 ~ 120) is 4.65s. For the determination of the highest climbing degree, the maximum climbing degree of the uphill stable speed at 10km / h is 29 %. The speed is too high, and the transmission ratio needs to be optimized in the later period.



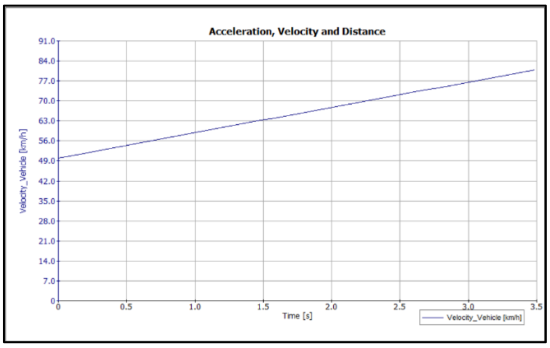
(a) Maximum speed curve



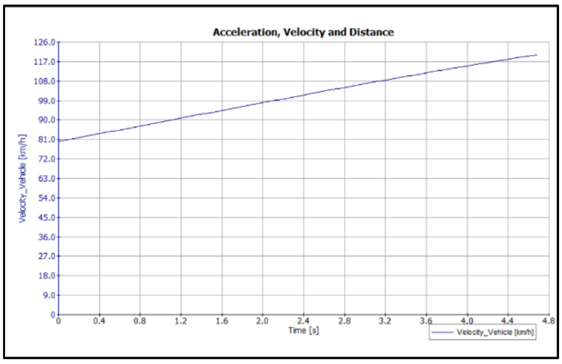
(b) Hundred kilometers acceleration time



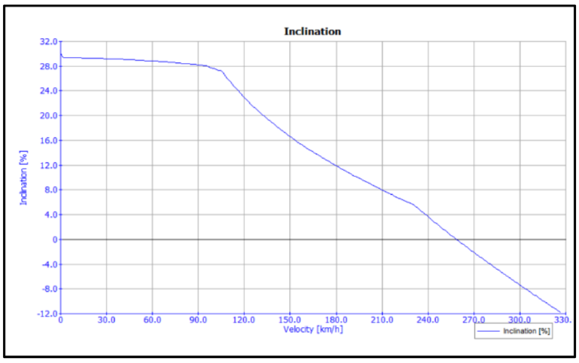
(c) starting-acceleration time



(d) Medium speed acceleration time



(e) High speed acceleration time

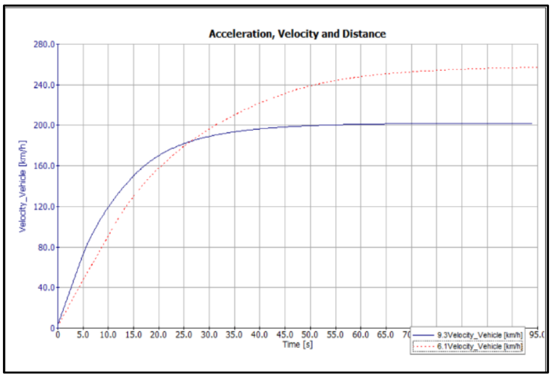


(f) Climbing curve

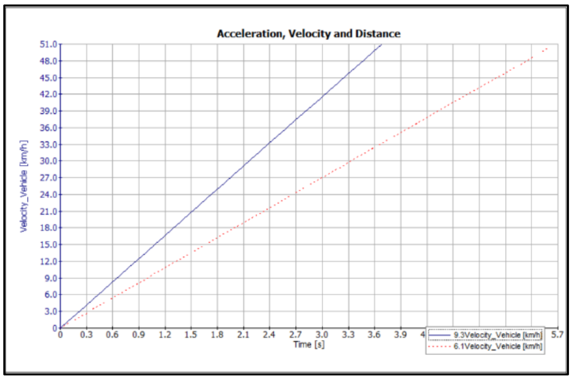
Figure 5: Vehicle dynamic simulation results diagram

4.2 Simulation optimization

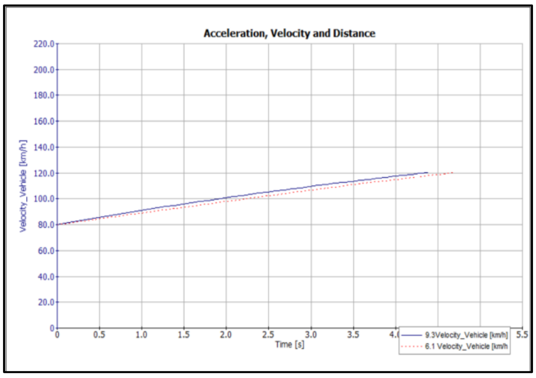
It can be seen from Fig. 6 (a) that the acceleration time of 100 kilometers is obviously lower than that before optimization. The optimized acceleration time is 7.89 s, which also meets the design requirements. The acceleration time of each speed segment is compared below. As shown in Figure 6 (b), (c) and (d), the acceleration time of each speed section is shorter than that before optimization. It can be seen in the figure that the start acceleration (0 ~ 50km/h) time is 3.61 s, the medium speed section acceleration (50~80km/ h) time is 2.28 s, and the high speed section acceleration (80 ~ 120km / h) time is 4.34 s. As shown in Figure 6 (e), it can be clearly seen that the optimized climbing performance has been greatly improved. When the vehicle speed is maintained at 10km / h, the maximum climbing degree is 48 %.



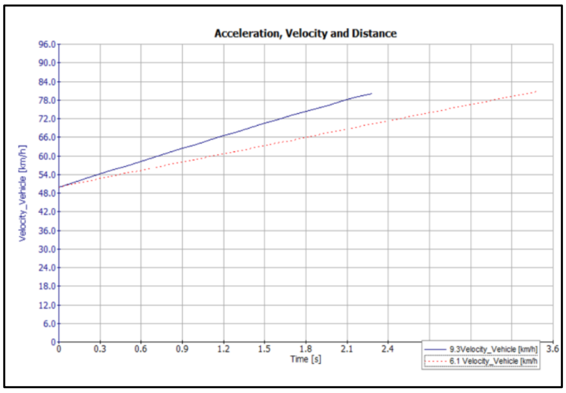
(a) Comparison of maximum speed after optimization



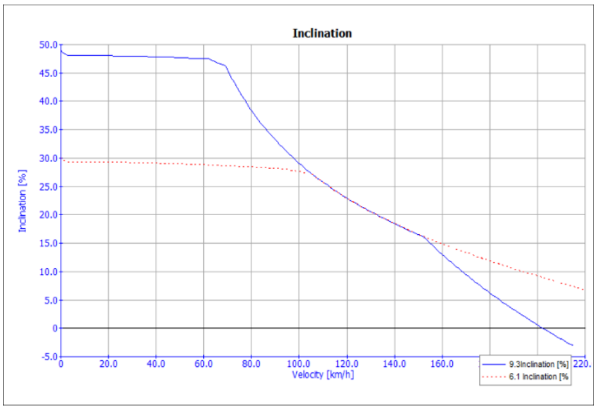
(b) Start acceleration time comparison



(c) Comparison of medium speed section



(d) Comparison of high speed section



(e) Comparison of maximum climbing degree

Figure 6: Comparison of simulation results after optimization

5. Conclusion

Through the joint simulation of MATLAB and Cruise, the preliminary simulation results are obtained. It is found that the maximum speed is much higher than the design value, but the maximum climbing degree and the 100 km acceleration time do not meet the design requirements. Therefore, the transmission speed ratio is also optimized. After optimization, the comparison before and after optimization shows that the 100 km acceleration time is reduced, the maximum climbing degree is increased, and the endurance mileage is reduced. However, each performance index is within the design index, and the final result of power matching is obtained.

References

[1]. Aden Seaman, John Mcphee. Symbolic Math-based Battery Modeling for Electric Vehicle Simulation [C]. Proceedings of the ASME 2010 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference, August 15-18, 2010, Canada, DETC 2010-28814:1-9.

[2]. Jiaqi Wang, Yanfang Liu, Qiang Liu, Xiangyang Xu. Power-based Shift Schedule for Pure Electric Vehicle with a Two-speed Automatic Transmission [J]. IOP Conference Series: Materials Science and Engineering, 2016, 157(1).

[3]. S S Bhurse, A A Bhole. A Review of Regenerative Braking in Electric Vehicles [C], 2018 International Conference on Computation of Power, Energy, Information and Communication (ICCPEIC), 2018. 363-367.

[4]. Y Xu, L Jiang, B Wei, et al. An Optimal Torque Distribution Strategy for Four-Motorized-Wheel Electric Vehicle Considering Energy Conversation [C], in IEEE Access, 2020, 8: 135975-135988.

[5]. Zhang J, Yang Y, Zhou Y, et al. Parameter closed-loop optimization for pure electric vehicles: unified design of power system and control parameters [J]. J Braz. SOC. Mech. Sci. Eng. 2020, 42: 229.

[6]. M Y Karelina, O N Didmanidze, V A Rakov. Methodological Approaches to Estimation of the Braking Energy Recovery Properties [J], Systems of Signals Generating and Processing in the Field of on Board Communications, 2021. 1-4.

[7]. Qian Juan, Wang Dongfang, Miao Xiaodong. Multi-objective optimization of automotive powertrain parameters based on particle swarm optimization [J]. Machinery manufacturing and automation, 2018,47 (03): 168-170.

[8]. Wang Fei, Li Leilei. Research on parameter matching and optimization of pure electric vehicle power system [J]. Agricultural equipment and vehicle engineering, 2020, 58 (12): 93-97.