Shift Strategy Development for Electric Bus

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Abstract. This paper studies the shifting strategy based on an electric bus. According to the development performance requirements of the bus, a dynamic shifting strategy and an economical shifting strategy are formulated respectively. Among them, the dynamic shifting strategy is mainly based on the principle of maximum acceleration to formulate corresponding shifting strategy, and the economical shifting strategy is mainly formulated based on the test data of the motor bench and the characteristics of the vehicle speed and efficiency curve under different accelerator pedal depths. Finally, build an AVL CRUISE vehicle simulation model to simulate and analyze the shifting strategy of the electric bus and a test platform for testing. Simulation and test results show that the proposed shifting strategy is feasible.

Keywords: Shifting strategy · AVL CRUISE · Optimization

1 Introduction

The application of computer modeling and simulation technology can effectively reduce the vehicle development cycle and cost. AVL CRUISE is an advanced simulation analysis software for the study of vehicle power performance, fuel economy, emission performance and braking performance. It is a positive and modular simulation platform, which not only improves the efficiency of research and development, but also has practical significance for industry exchanges and common progress.

For an electric bus, the drive motor is its only power source. When climbing a hill or under heavy load, the motor needs to provide a large torque. In order to reduce the torque requirements for the motor, a multi-speed transmission is required. At present, multi-speed has become the development trend of electric vehicles. Formulating an appropriate gear shifting strategy has an important impact on improving the power performance and economy of the vehicle. Therefore, it is of great significance to study the shifting strategy of electric vehicles [2]. Based on an electric bus, this paper studies the dynamic shifting and economical shifting strategies respectively [3]. Among them, the formulating principle of the power shift strategy is: formulate the corresponding shifting strategy according to the principle of maximum acceleration [4]; the formulating principle of the economical shifting strategy is: according to the test map data of the motor bench, combined with different accelerator pedal depths, formulate according to the principle of optimal shift point efficiency [5]. Finally, the AVL CRUISE vehicle

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simulation model and test platform are built to verify the feasibility of the shifting strategy.

2 Powertrain Matching

There is a 12m electric bus. The parameters of the bus are shown in Table 1. The design indicators are:

(1) the maximum speed is greater than or equal to 100 km/h;
(2) the maximum grade is greater than 15% (the speed of climbing is 10 km/h), Continue to climb 7% of the vehicle speed (climb speed 15 km/h);
(3) The acceleration time of 0–50 km/h is less than 20 s.

Table 1. Vehicle parameters

Vehicle mass (kg)	18000	Windward area (m^2)	6.7
Drag coefficient	0.55	Transmission efficiency	0.92
Coefficient of Rolling Resistance	0.012	Main transmission ratio coefficient	5.7

First, determine the power of the motor. It needs to meet the requirements of the maximum speed, maximum grade and acceleration time of the bus [6]. Calculate the maximum speed demand power P_1 according to the maximum speed index, calculate the demand power P_2 according to the demand index of the maximum grade, and calculate the demand power P_3 according to the acceleration performance index, $P_{max} >$ max[P1,P2,P3].

Calculate the required power at the maximum speed according to formula (1):

$$P_1 = \frac{v_{\max}}{3600\eta_T}\left(m \cdot g \cdot f + \frac{C_D \cdot A \cdot v_{\max}^2}{21.15}\right) \tag{1}$$

Substituting the vehicle parameters into the above equation, the maximum vehicle speed and power demand curve can be obtained, as shown in Fig. 1. According to the design requirements, the maximum speed is 80 km/h. Therefore, the required power corresponding to the maximum speed of the electric vehicle is $P_1 \approx 91.88$ kw.

Calculate the required power for the maximum gradeability according to formula (2):

$$P_2 = \frac{v_i}{3600\eta_T}\left(m \cdot g \cdot f \cdot \cos\alpha_{\max} + m \cdot g \cdot \sin\alpha_{\max} + \frac{C_D \cdot A \cdot v_i^2}{21.15}\right) \tag{2}$$

Substituting each parameter into the above formula, the road gradient and power demand curve can be obtained, as shown in Fig. 2. The required power for 15% gradient and 10 km/h is $P_2 = 84.58$ kw.

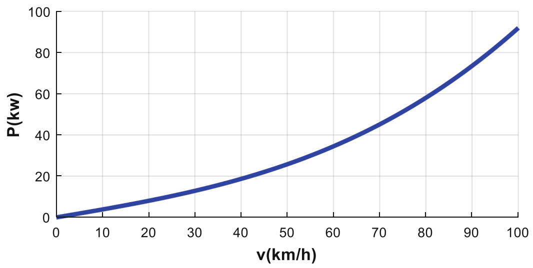

Fig. 1. Vehicle speed and power demand curve

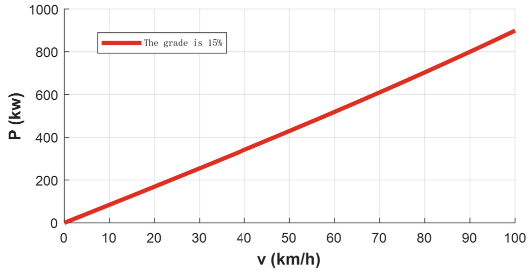

Fig. 2. Power demand curve with a slope of 15%

Calculate the required power for the maximum gradeability according to formula (3):

$$P_3 = \frac{v_m}{3600\eta_T}\left(m \cdot g \cdot f + \delta \cdot m \cdot \frac{v_m}{7.2 \cdot t_m} + \frac{C_D \cdot A \cdot v_m^2}{21.15}\right) \tag{3}$$

Substituting the parameters into the above formula, the curve of vehicle acceleration performance and power demand can be obtained, as shown in Fig. 3. According to the design index requirements: 0~50 km/h acceleration time should be less than 7s, $P_3 =$ 123.31 kw.

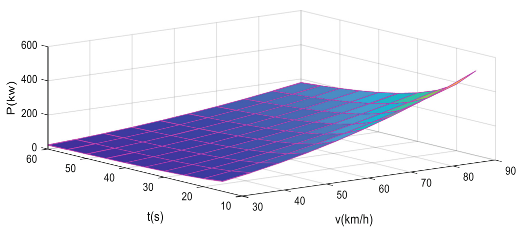

Fig. 3. Vehicle acceleration performance and power demand curve

According to the design index of the maximum speed, the maximum speed of the motor is determined by the formula (4), and $n_{max} > 2982r/min$ is calculated.

$$n = \frac{i_g \cdot i_0 \cdot v}{0.377 \cdot r} \tag{4}$$

According to the maximum climbing index and the continuous climbing index, the peak torque and rated torque of the powertrain are determined by formula (5) and formula (6). Substituting into the calculation can get M > 2737.1N.m, Me > 1346N.m.

$$M \geq \frac{\mathrm{F_t} \cdot r}{\eta_T \cdot i_0 \cdot i_g} = \frac{(mgf \cos\alpha + mg \sin\alpha) \cdot r}{\eta_T \cdot i_g \cdot i_0} \tag{5}$$

$$Me \geq \frac{\mathrm{F_t} \cdot r}{\eta_T \cdot i_0 \cdot i_g} = \frac{(mgf \cos\partial + mg \sin\partial) \cdot r}{\eta_T \cdot i_g \cdot i_0} \tag{6}$$

According to the existing powertrain products, a motor with a peak power of 150 kW, a peak torque of 1500 N.m and a maximum speed of 3200r/min was finally selected. Then, matched with a two-speed transmission, the first gear ratio was 2.5, and the second gear ratio was 1, the maximum output torque of the powertrain is: 3750 N.m. The powertrain parameter table is shown in Table 2.

Table 2. Powertrain Parameters

Transmission ratio	2.5/1	Motor peak power	150 kW
maximum speed(r/min)	3200	maximum torque(N.m)	1500

3 Shift Strategy Formulation

The electric bus is driven by an electric motor, which is different from the traditional engine and has its own characteristics. The formulation of the shifting strategy affects the power performance and economy of the vehicle [7]. The shift timing and gear selection of the automatic transmission are determined by comparing the current vehicle parameters with the parameters in the control system [8]. According to the different control parameters, it can be divided into three types: single parameter (vehicle speed), two parameters (accelerator pedal depths and vehicle speed) and three parameters (acceleration, accelerator pedal depths and vehicle speed) [9]. Since the dual-parameter shifting strategy is easier to control and implement, Since the dual-parameter shift strategy control is easier to implement, the control strategy of the shift strategy in this paper is based on the dual-parameter type of acceleration pedal and vehicle speed, and the dynamic shift strategy and the economic shift strategy are studied respectively. And optimized the original shift measurement.

3.1 Development a Dynamic Shifting Strategy

The intention of formulating a dynamic shifting strategy is to maximize the driving ability of the motor [10], meet the driving force demand of the vehicle when climbing a hill, have sufficient overtaking acceleration ability, give full play to the backup power of the motor, so that the motor is in a high load state [11]. This paper adopts the principle based on the maximum acceleration to formulate the power shifting strategy, that is, the maximum acceleration is the same before and after shifting.

According to the external characteristics of the drive motor matched with the reference vehicle model, the load curve of the drive motor under different accelerator pedal depths is drawn as shown in Fig. 4.

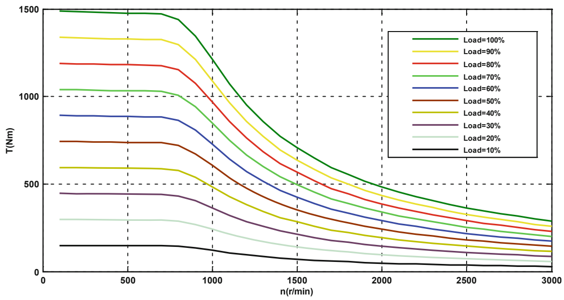

Fig. 4. Load curve of the drive motor

The driving equation of the vehicle is:

$$\frac{T_{tq} i_g i_0 \eta_T}{r} = mgf + \frac{C_d A}{21.15} u_a^2 + mgi + \delta m \frac{du}{dt} \tag{7}$$

In the formula: T_{tq} is the maximum torque; i_g is the transmission ratio of the main reducer; i_0 is the transmission ratio of the gearbox; m is the weight of the vehicle; f is the rolling resistance coefficient; C_d is the wind resistance coefficient; A is the windward area; u_a is the vehicle speed; η_T is the transmission efficiency of the transmission system; δ is the rotation mass conversion factor; r is the rolling radius of the vehicle tires; i is the maximum grade.

Considering driving on a straight road (ignoring the slope), the expression for the acceleration obtained from the driving equation of the vehicle is:

$$\frac{du}{dt} = \frac{1}{\delta m}\left(\frac{T_{tq} i_g i_0 \eta_T}{r} - Gf - \frac{C_d A}{21.15} u_a^2\right) \tag{8}$$

From the acceleration formula (8), calculate the magnitude of the acceleration under different gears and different accelerator pedal depths, and the values are shown in Fig. 5.

According to the principle of maximum acceleration, the intersection of the acceleration curves of two adjacent gears under the same accelerator pedal depths is taken as the

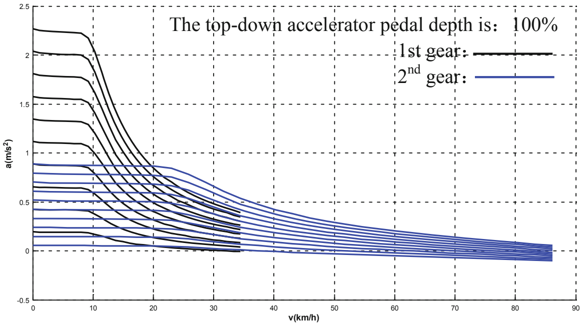

Fig. 5. Acceleration curves under different gears & different accelerator pedal depths

Table 3. Upshift speed value for dynamic shift strategy (km/h)

Accelerator pedal depth	10%	20%	30%	40%	50%
1st gear up 2nd gear	16.4847	17.6474	18.7042	19.4686	21.6764
Accelerator pedal depth	60%	70%	80%	90%	100%
1st gear up 2nd gear	22.3268	22.9663	23.8659	24.9466	25.6465

shift point, and the vehicle speed corresponding to this point is the upshift speed [13], as shown in Table 3.

Under normal circumstances, the downshift speed difference is 2–8 (km/h). In order to improve the power performance of the vehicle, a smaller value is selected for the downshift speed difference under large throttle. To avoid shifting cycles, a larger value is selected for the downshift speed difference under small throttle [14]. According to this principle, the downshift speed is obtained, as shown in Table 4.

Table 4. Downshift Speed for dynamic shift strategy (km/h)

Accelerator pedal depth	10%	20%	30%	40%	50%
2nd gear down 1st gear	14.4847	15.6474	16.6474	17.4686	19.6764
Accelerator pedal depth	60%	70%	80%	90%	100%
2nd gear down 1st gear	20.3268	20.9663	21.8659	22.9466	25.6465

3.2 Economical Shift Strategy Formulation

According to the map data of the motor bench test, combined with the torque requirements of the motor with different accelerator pedal depths, the vehicle speed and motor efficiency curves under different accelerator depths were drawn [15], as shown in Fig. 6 to Fig. 15.

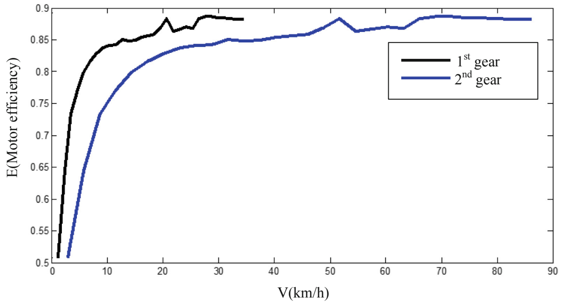

Fig. 6. When the accelerator depths is 100%, the vehicle speed-efficiency curve

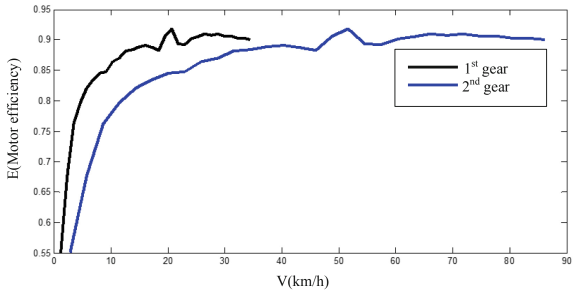

Fig. 7. When the accelerator depths is 90%, the vehicle speed-efficiency curve

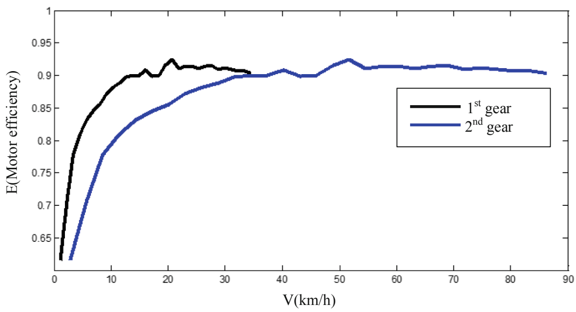

Fig. 8. When the accelerator depths is 80%, the vehicle speed-efficiency curve

To sum up the analysis of Fig. 6 to Fig. 15, the intersection of the first gear and the second gear curve is taken as the optimal economic upshift point [16]. Table 5 is the strategy of 1st gear to 2nd gear (Figs. 7, 8, 9, 10, 11, 12, 13 and 14).

Using the equal-delay shift strategy, the downshift speed difference is generally 2–8 (km/h), and the speed difference is 3 km/h [17], and the downshift point is obtained, as shown in Table 6.

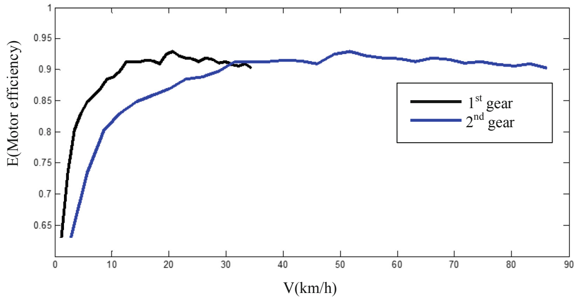

Fig. 9. The accelerator depths is 70%, vehicle speed-efficiency curve

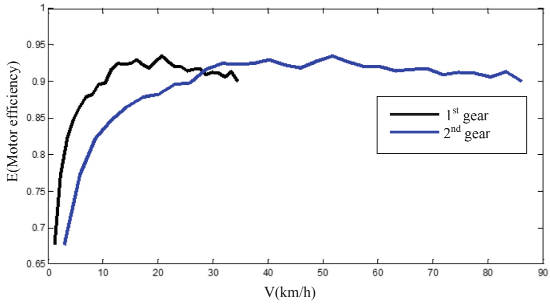

Fig. 10. The accelerator depths is 60%, vehicle speed-efficiency curve

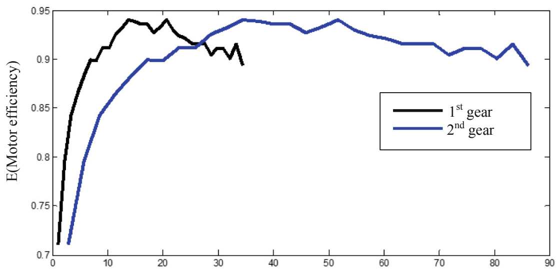

Fig. 11. When the accelerator depths is 50%, the vehicle speed-efficiency curve

3.3 Optimal Shift Strategy Formulation

Combined with the urban road conditions and the above two shift schedules, the original shifting strategy is optimized for the 12m electric bus. Due to the low efficiency of the motor in the medium and low speed region, an economical shifting strategy is adopted. In the high-speed section, the power performance of the vehicle is mainly considered, so the dynamic shifting strategy is selected. Combining the two can take into account both economical shifting and dynamic shifting. The specific shifting strategy is shown in Fig. 16.

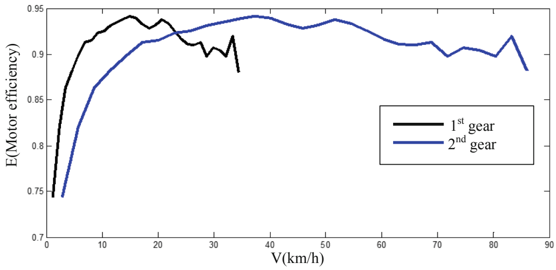

Fig. 12. The accelerator depths is 40%, vehicle speed-efficiency curve

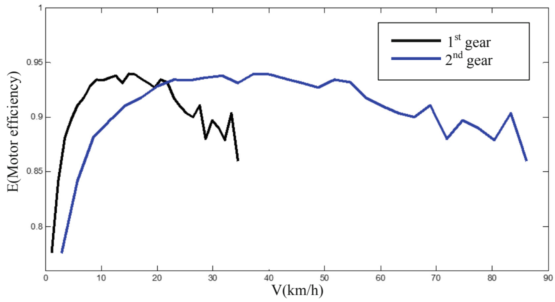

Fig. 13. The accelerator depths is 30%, vehicle speed-efficiency curve

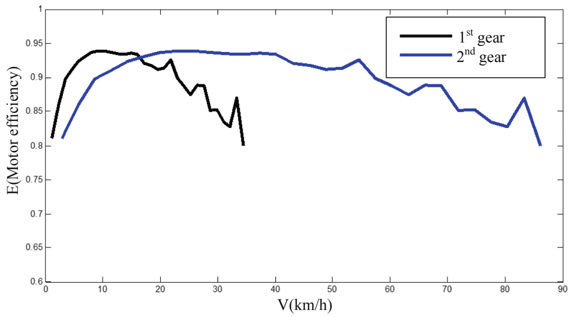

Fig. 14. The accelerator depths is 20%, and the vehicle speed-efficiency curve

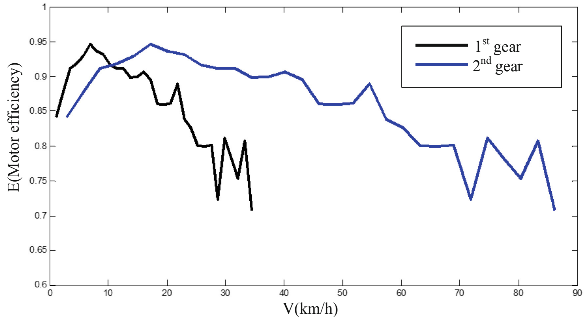

Fig. 15. The accelerator depths is 10%, vehicle speed-efficiency curve

Table 5. Upshift speed value for economical shift strategy (km/h)

Accelerator pedal depth	10%	20%	30%	40%	50%
1st gear up 2nd gear	10.4638	16.5398	21.8393	23.2600	26.7619
Accelerator pedal depth	60%	70%	80%	90%	100%
1st gear up 2nd gear	28.2059	30.873	22.9744	25.2718	26.4206

Table 6. Downshift Speed value for economical shift strategy (km/h)

Accelerator pedal depth	10%	20%	30%	40%	50%
2nd gear down 1st gear	7.4638	13.5398	18.8393	20.2600	23.7619
Accelerator pedal depth	60%	70%	80%	90%	100%
2nd gear down 1st gear	25.2059	27.873	19.9744	22.2718	23.4206

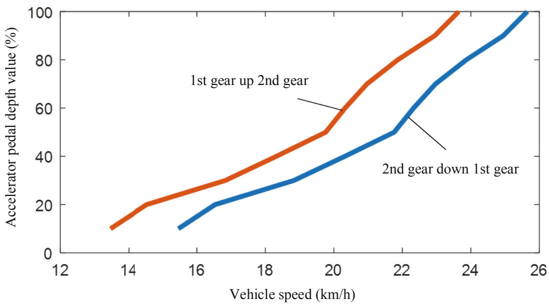

Fig. 16. Optimized shift strategy

4 Shift Strategy Simulation Analysis

Build a vehicle simulation model based on CRUISE, add the AMT Control module and the Gear Box Program module, and complete the connection of the corresponding data lines in the Gear Box Program. Figure 17 shows the built vehicle simulation model, set the Gear Box Program module as the Component Variation variable parameter, and use the Component Variation in the Calculation Center to add a Gear Box Program comparison module [18]. Import the unoptimized shift strategy into the Gear Box Program module in the original model, and set the shift strategy with the dual parameters of throttle and vehicle speed [19]; Simulation and comparison of two shifting strategies.

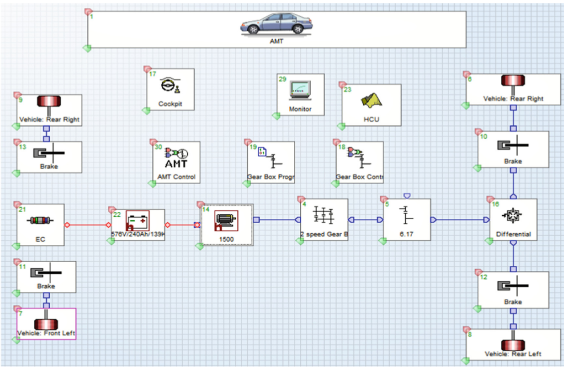

Fig. 17. Vehicle simulation model

The Cycle Run task of China's typical urban working condition (CCBC working condition) is established. The continuous driving time of China's typical urban cyclic working condition is 1314s, the whole journey is 5.83 km, the average speed is 16.10 km/h, and the maximum speed is 60 km/h [20]. The speed following curves are shown in Fig. 18 and Fig. 19. It can be seen from the figures that the two strategies are in good speed following states. In typical urban conditions in China, the actual gear is consistent with the target gear, and the shifting rules are feasible and calculated separately. The vehicle energy consumption data of the two shifting strategies shows that the power consumption per 100 km of the unoptimized shifting strategy is 76.68 kWh/100 km, and the power consumption per 100 km of the optimized shifting strategy is 73.54 kWh/100 km, which improves the economy 4.5%.

Establish a full-load acceleration task (0–50 km/h acceleration time calculation task), and compare the power performance of the two shifting strategies, as shown in Fig. 20. The acceleration time of 0–50 km with the unoptimized shifting strategy is 17.13s, and the acceleration time of 0–50 km with the optimized shifting strategy is 15.68s (Tables 7 and 8).

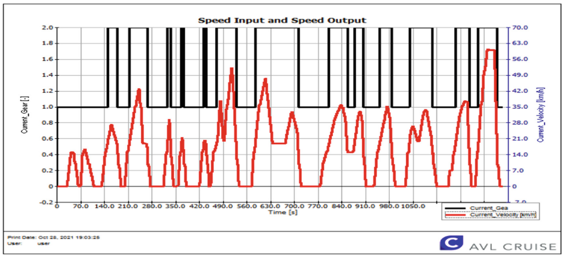

Fig. 18. Analysis of the following situation of the unoptimized shift strategy under the CCBC condition

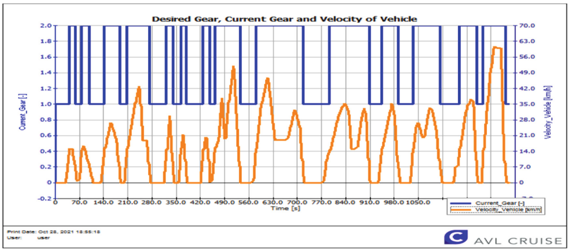

Fig. 19. Analysis of CCBC vehicle speed following and shifting situation after optimization

Table 7. Simulation of economic results

project	Unoptimized shift strategy	Optimized shift strategy
Electricity consumption per 100 km (kWh/100 km)	76.64	73.58

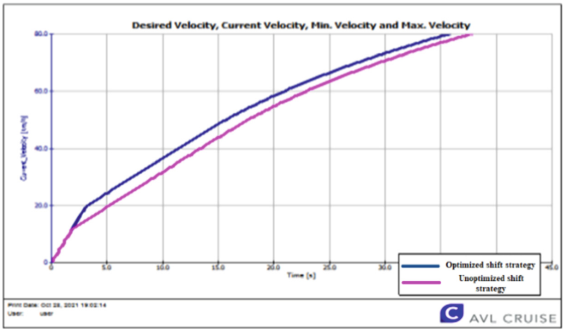

Fig. 20. Comparison of full throttle acceleration simulation results for two shifting strategies

Table 8. Simulation of 0–50km/h acceleration time results

project	Unoptimized shift strategy	Optimized shift strategy
0–50 km/h acceleration time (s)	17.13	15.68

5 Test Verfication

Based on the above research on the shifting optimization strategy, the optimized shifting strategy was applied to a 12 m electric bus, and the acceleration performance and CCBC operating conditions were tested on the hub. The test results are shown in Table 9.

Table 9. Test results of 12 m electric bus

project	Test results of automotive hub test bench
0–50 km/h acceleration time (s)	15.81
Power consumption per 100 km under CCBC cycle condition (kwh/100 km)	73.72

6 Conclusion

In this paper, the shifting strategies of the electric bus is studied, and the dynamic and economic shifting strategies are formulated respectively. Combined with the urban road conditions, the original shifting strategy is optimized for the 12 m electric bus, and AVL CRUISE is used to simulate and analyze the shifting strategies before and after optimization, and finally carry out experimental verification. Through this research on the shifting schedule, it provides a good reference for the development of the shifting strategy of electric buses. In the future, we will conduct more in-depth optimization and research on the shifting control strategy of electric buses.

References

1. 陈淑江, 秦大同, 胡明辉, 胡建军. 兼顾动力性与经济性的纯电动汽车 AMT 综合换档策略[J]. 中国机械工程, 2013,24(19)
2. 李顺波. 纯电动客车综合换挡规律研究.长春: 吉林大学硕士学位论文, 2014
3. 杨易，江青华，周兵，王继生. 纯电动汽车最佳动力性换档规律研究[J].汽车技术, 2011(3):1–5
4. 李大伟, 方锡邦, 吴哲.微型电动汽车 AMT 动力性换档规律的制定与仿真分析[J].重庆理工大学学报 (自然科学) , 2011,1(25)
5. 张煜. 纯电动客车自动变速系统及控制策略研究[D].吉林大学硕士学位论文, 2012
6. 王立国.纯电动客车动力总成控制策略研究[D].长春: 吉林大学硕士学位论文, 2009

7. 刘晓东,马建,贺伊琳,张一西,张凯.AMT驱动构型对纯电动客车综合性能影响研究[J].中国公路学报.2019(07)
8. 冉滔,方玲玲,彭思慧.纯电动汽车AMT两档换挡规律分析[J]. 汽车实用技术. 2019(13)
9. 卢代继.电动汽车用两挡AMT换挡规律的建模与仿真[J].汽车维修技师. 2021(06)
10. 刘拂晓,赵韩,江昊.纯电动汽车AMT换挡规律及仿真研究[J].合肥工业大学学报(自然科学版). 2013(11)
11. 秦大同,龙海威,胡明辉.AMT中度混合动力汽车经济性换挡规律研究[J].中国机械工程. 2013(20)
12. 何安清,孙可华,沈利芳,宋国鹏,胡远敏.基于CRUISE与ADVISOR前后向仿真软件的汽车动力性对比分析[J].客车技术与研究. 2018(02)
13. 席军强,王雷,付文清,梁万武.纯电动客车自动机械变速器换挡过程控制[J].北京理工大学学报. 2010(01)
14. LIU H, LEI Y, LI Z, et al. Gear-shift strategy for a clutchless auto-mated manual transmission in battery electric vehicles[J] . SAE In-ternational Journal of Commercial Vehicles, 2012, 5(1) : 57–62
15. 沈波.电控机械式自动变速器换档品质的研究. 吉林大学硕士学位论文, 2003
16. 刘保林. AMT 起步和换挡品质研究. 吉林大学硕士学位论文, 2003
17. 王丽芳.自动变速器换挡规律方法的研究.汽车技术[J] .1998(6)
18. 刘大权,张小东,曲金玉. 公共汽车AMT最佳动力性换挡规律的建模与仿真[J].农业装备与车辆工程. 2009(02)
19. 张兆红,刘涛,刘凯泽,王琳.纯电动客车两档变速器传动比优化仿真分析[J].汽车实用技术. 2021(05)
20. 刘勤.配置AMT的纯电动城市客车换挡策略研究.长安大学硕士学位论文.2017