Research and Simulation Analysis of Optimizing Dynamic Performance Shift Schedule for Transmission-By-Wire in Electric Vehicle

Shenchao Zhu, Jinyu Qu, Yunchao Yin, Pan Zhang

***Abstract*—The shift schedule of automatic transmission is the core technology of control system, the selection of control parameters directly influences the dynamic performance, fuel economy and the driving range of the vehicle. In this paper we studied the optimizing dynamic performance shift schedule for the three-geared transmission-by-wire. The developing method for the optimizing dynamic performance shift schedule was analyzed, and the shifting process model is set up in Matlab/Simulink. The optimizing dynamic performance shift schedule curve is developed for three-geared transmission-by-wire which could improve the performance of electric vehicle, so as to achieve the purpose of saving energy and increasing the driving range.**

***Index Terms*—Electric Vehicle; Transmission-By-Wire; Optimizing Dynamic Performance; Shift Schedule; Simulink**

I. INTRODUCTION

Shift schedule refers to the regular pattern that transmission shifts according to the traveling condition of electric vehicle (EV). The transmission is determined to upshift, downshift or hold the current gear, according to the variation of the control parameters [1-2]. Shift schedule is the core technology of the automatic transmission's control theory, which determines the performance degree of powertrain; thereby the EV's driving range and acceleration performance are affected. Shift schedule includes dynamic performance shift schedule and economy shift schedule, in which, the dynamic performance shift schedule promotes the dynamic performance of EV and economy shift schedule improve the fuel economy of the vehicle.

The mechanical and hydraulic systems are replaced in X-By-Wire technology by a series of operation that is based on wires and electronic controller. Driver's steering movements are transmitted to electrical signals through the sensors, and electronic control unit generates control signals for driving actuators according to the input signals. The optimizing dynamic performance shift schedule for the three-geared transmission-by-wire (TBW) in EV is researched. Firstly, the shift schedule is theoretically analyzed, and then the shifting process model is established in MATLAB/Simulink to simulate and analyze the shift schedule curve. The simulation result shows that the developed optimizing dynamic performance shift schedule could improve the vehicle's performance and realize the purpose of energy saving and driving range increased.

II. MATCHING OF VEHICLE PARAMETERS

With energy consumption and economy development, and continuously development of new energy, EVs' development will have more and more space to become the main force of automobile market [3]. The parameters matching and the dynamics shift simulation are studied based on a certain pattern Chery EV. The basic parameters of the vehicle are shown in Table 1.

III. MOTOR'S OPERATING CHARACTERISTICS

Motor as the only power source of EV has a direct impact on the performance of the fuel economy and dynamic performance [4]. The motor used is a 15kW permanent magnet brushless synchronous motor. And its technical parameters are shown in Table 2. The rotating speed, torque and efficiency of the three-dimensional MAP are shown in Fig.1. The torque changing at different accelerator opening is shown as Fig.2.

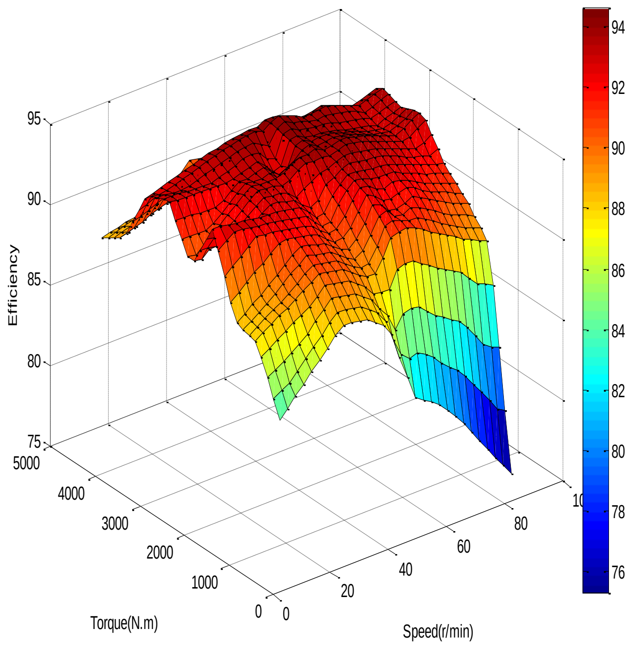

Fig.1 The motor speed, torque, and efficiency of the three-dimensional MAP

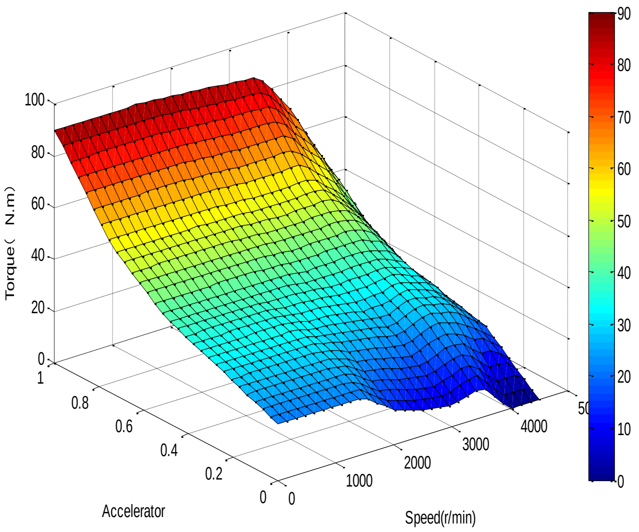

Fig.2 Torque variation at different accelerator opening

Table 1The structural parameters of vehicle

parameter	Numerical	unit
L * W * H	4560/1822 /1630	mm
Train efficiency	0.9	--
Curb Weight	1200	Kg
Full quality	1480	Kg
Radius of the wheel	0.215	M
Rolling resistance coefficient	0.0166	--
Air resistance coefficient	0.3	--
Frontal area	1.38	m^2
Wheelbase	2340	mm
Main reduction ratio	4.612	--
T r1	1.7	--
T r2	1	--
T r3	0.7	--

Table 2 Technical parameters of permanent magnet brushless motor

parameter	symbol	unit	numerical
rated power	P_e	kW	15
Peak power	P_{max}	kW	40
Rated torque	T_e	Nm	40.1
Peak torque	T_{max}	Nm	90
Peak speed	n_e	r/min	4500
Rated speed	n_{max}	r/min	3500

As Fig.2 shows, when the motor speed is constant, the greater the degree of throttle opening is the larger output torque from driving motor is; when the throttle opening is constant, the greater the rotating speed of the driving motor is the smaller output torque from driving motor is. In high degree throttle opening, the big torque output meets the requirement of dynamic performance; in small degree throttle opening, the small torque output and high rotating speed meet the requirement of fuel economy performance.

IV. THE DEVELOPMENT OF THE OPTIMIZING DYNAMIC PERFORMANCE SHIFT SCHEDULE

A. The Characteristics Analysis of Optimizing Dynamic Shifting

To ensure the optimizing dynamic characteristics of EV, the intersection of two adjacent gear acceleration curves under the same accelerator opening is used as the optimizing dynamic performance shifting point, which is[5]:

$$\frac{dv}{dt_n} = \frac{dv}{dt_{(n+1)}} \tag{1}$$

Connect the intersections under different accelerator opening, the optimizing dynamic shifting curve will be obtained. The relationship between rotating speed and torque on same drive motor operating point is:

$$T = \begin{cases} T_p & (n \le n_m) \\ \dfrac{9550 P_p}{n} & (n > n_m) \end{cases} \tag{2}$$

Where: T_p is the maximum torque at a certain accelerator opening of drive motor (Nm); P_p is the maximum power at a certain accelerator opening of drive motor (kW).

From the formula (2), it can be seen that the motor's partial load characteristic below the rated speed shows constant torque characteristic, so there is no intersection of adjacent gear acceleration curve below the base speed. Only the motor torque characteristic above the rated speed is considered when calculates the optimum dynamic shift schedule, which is[6]:

$$T = \frac{9550P_p}{n} \quad (n > n_m) \tag{3}$$

B. Theoretical Analysis of Optimizing Dynamic Shifting Schedule

EV runs with certain acceleration drove by motor and the transmission system, and it overcomes rolling resistance, air resistance, grade resistance and acceleration resistance. The EV's running equation is :

$$F_t = F_f + F_w + F_i + F_j \tag{4}$$

Where, F_t is the driving force; F_f is rolling resistance; F_w is the air resistance; F_i is the grade resistance; F_j is the acceleration resistance.

This can also be written as:

$$\frac{T_m i_g i_0 \eta_T}{r} = mgf + \frac{C_D A v^2}{21.15} + mgi + \delta m \frac{du}{dt} \tag{5}$$

Kinematic differential equation is:

$$\delta m \frac{du}{dt} = \frac{T_m i_g i_0 \eta_T}{r} - mgf - \frac{C_D A v^2}{21.15} - mgi \tag{6}$$

This can also be written as:

$$\delta m \frac{dv}{dt} = \frac{T_m i_g i_0 \eta_T}{r} - \frac{C_D A v^2}{21.15} - mg\psi \tag{7}$$

Where: δ is the correction coefficient of rotating mass; m is the full mass; v is driving speed; T_m is motor dynamic output torque; i_g is the transmission ratio; i_0 is the main reducer gear ratio; η_T is the transmission efficiency; r is wheel rolling radius; g is the acceleration of gravity; f is rolling drag coefficient; C_D is air resistance coefficient; A as windward area; i is the road gradient; ψ is road resistance coefficient ($\psi = f + i$)[7].

The rolling resistance coefficient can be approximately calculated by the empirical formula: $f = 0.0076 + 0.000056v$.

Simultaneous (1) and (5), we have:

$$\frac{\frac{T_m i_g^n i_0 \eta_T}{r} - \frac{C_D A v^2}{21.15} - mg\psi}{\delta_n} = \frac{\frac{T_m i_g^{n+1} i_0 \eta_T}{r} - \frac{C_D A v^2}{21.15} - mg\psi}{\delta_{n+1}} \quad (8)$$

Where: δ_n and δ_{n+1} are rolling resistance coefficients of transmission n gear and $n+1$ gear.

The rolling resistance coefficient δ is mainly affected by the inertia of the flywheel (I_f), the inertia of wheel (I_w) and the driveline gear ratios, the relationship is:

$$\delta = 1 + \left(\frac{1}{m}\right)\frac{\sum I_w}{r^2} + \left(\frac{1}{m}\right)\frac{I_f i_g^2 i_0^2 \eta_T}{r^2} \quad (9)$$

The relationship between n gear driving force F_t^n 、 and $n+1$ gear driving force F_t^{n+1} is:

$$F_t^n = \frac{i_g^n}{i_g^{n+1}} F_t^{n+1} = qF_t^{n+1} \quad (10)$$

Where: i_g^n and i_g^{n+1} are respectively n 、 $n+1$ gear ratios; q is ratio of n gears and $n+1$ gear.

The relationship between the output torque and its rotating speed under the same accelerator opening can be fitted to a quadratic curve, the fitting relation is:

$$T_m = An^2 + Bn + C \quad (11)$$

The relationship between motor speed and EV travelling speed is:

$$v = 0.377\frac{rn}{i_g^n i_0} \quad (12)$$

Relationship between each gear's driving force F_t^n and the motor output torque is:

$$F_t^n = \frac{T_m i_g^n i_0 \eta_T}{r} \quad (13)$$

The expression for each gear's driving force F_t^n is:

$$F_t^n = A_n v^2 + B_n v + C_n \quad (14)$$

From (8) and (12), we can get:

$$i_g^{n+1}(A_n v^2 + B_n v + C_n) = i_g^n(A_{n+1} v^2 + B_{n+1} v + C_{n+1}) \quad (15)$$

This can also be written as:

$$A_0 v^2 + B_0 v + C_0 = 0 \quad (16)$$

Where,

$$A_0 = \frac{(q\delta_{n+1} - \delta_n)A_n - (\delta_{n+1} - \delta_n)}{21.15}$$

$$B_0 = (\delta_{n+1} - \delta_n)B_n - 0.000056mg$$

$$C_0 = (\delta_{n+1} - \delta_n)[C_n - (i_g^n i_0 + 0.0076)mg]$$

The optimizing shift points can be solved from equation (14):

$$v = \frac{-B_0 \pm \sqrt{B_0^2 - 4A_0 C_0}}{2A_0} \quad (17)$$

Compared the two roots with the gear's maximum speed $v_{n\max}$ and the minimum speed of the next gear $v_{(n+1)\min}$, if $v_{(n+1)\min} < v_n < v_{n\max}$, v_n is the optimizing dynamic shifting point speed.

V. SIMULATION AND ANALYSIS

A. Analysis of Shifting Point

In conjunction with Figure 2, deeming the throttle opening, motors' torque and rotating speed characteristics as linear, the linear interpolation function of torque, speed and accelerator opening according to the formula (3) and (10) can be expressed as:

$$T_{tq} = f_0(\alpha, n) \quad (18)$$

Where, α is throttle opening, $0 \le \alpha \le 1$.

Taking the complexity of the model into account, the slope's impact on shifting is ignored [8]. According to the analysis of the shift point and mathematical model, the optimizing dynamic performance shift point model in different gears is established in MATLAB/Simulink, it is shown as Fig.3. The 1-2 gear acceleration curve at 100% throttle opening is shown in Fig.4. The 2-3 gear acceleration curve at 100% throttle opening is shown in Fig.5.

According to the optimizing dynamic performance shift point model, the corresponding up-shift point vehicle speed under different throttle opening are summarized as shown in Table 3.

Table 3 Summary of optimizing dynamic up-shifting points

Throttle opening (%)	U-V(1-2) (km/h)	U-V (2-3) (km/h)
0.1	27.80	36.82
0.2	43.34	49.54
0.3	44.66	71.25
0.4	44.62	71.34
0.5	44.59	71.29
0.6	44.69	71.33
0.7	44.85	71.42
0.8	44.93	71.46
0.9	44.88	71.47
1.0	44.84	71.44

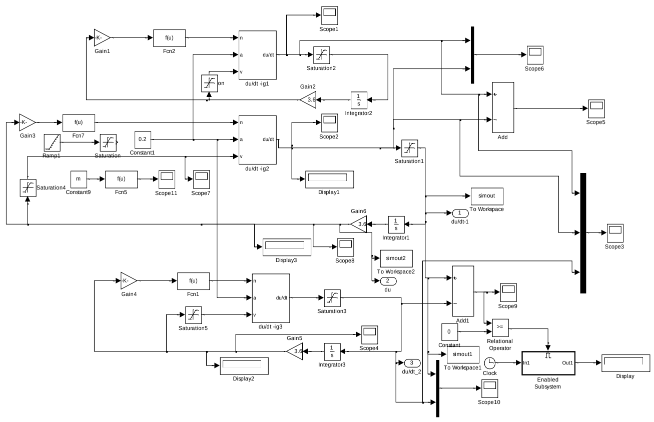

Fig.3 The optimizing dynamic performance shift point model

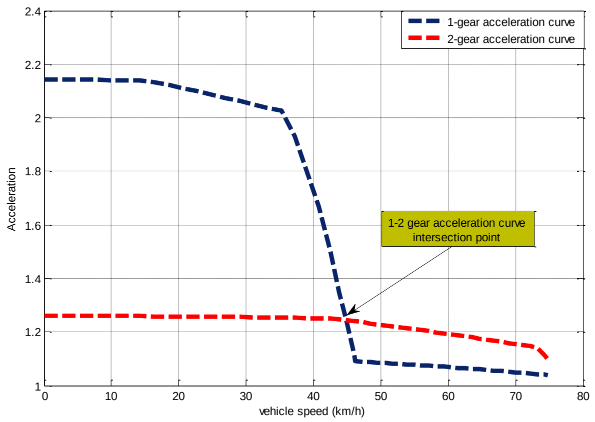

Fig.4 The 1-2 gear acceleration curve at 100% throttle opening

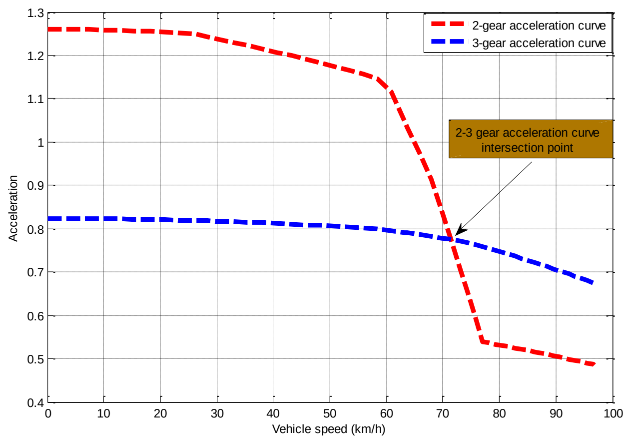

Fig.5 The 2-3 gear acceleration curve at 100% throttle opening

B. *The Development of Shift Schedule Curve*

The adjacent gears acceleration intersection are selected as shift points[9], the optimizing dynamic performance shift schedule is developed through polynomial fitting the corresponding adjacent gear shift points at different accelerator opening. To simplify the design, the front and rear of the vehicle shift state assumptions are:

(1)Shift time is short, the normal automatic transmission shift time is 500-800ms; it can be assumed that the vehicle speed is substantially unchanged before and after the shift;

(2) Road environmental factors before and after the shift remain unchanged, namely rolling resistance is unchanged;

(3) Due to the short shift time, the accelerator opening degree is the same before and after the shift.

(4) Without considering the dynamic changes during shifting driving train;

(5) Vehicle is traveling on a good level road.

Due to the power interruption, in the actual condition the vehicle speed will be reduced 1-2 (km/h). The deceleration difference is about 2-8 (km/h) according to traditional experience [5]. To avoid TBW cycle shift or misuse shift, the deceleration difference is introduced. The deceleration difference selection rules are shown as follow, according to theoretical analysis and traditional experience:

(1)After the downshift, the drive motor speed and torque should work within the normal range;

(2)Under the same gear, the big throttle opening degree with little deceleration difference improves vehicle's dynamic performance; the big throttle opening degree with big deceleration difference improves vehicle's economy performance.

(3)At the same throttle opening, high gear has greater deceleration difference than the low-speed gear;

(4)Avoid overdriving motor speed after shifting to improve the life of the drive motor.

Based on the above deceleration difference rules, through reasonably formulating deceleration difference, the optimizing dynamic performance shift schedule of EV is developed as shown in Fig.6 and Fig.7.

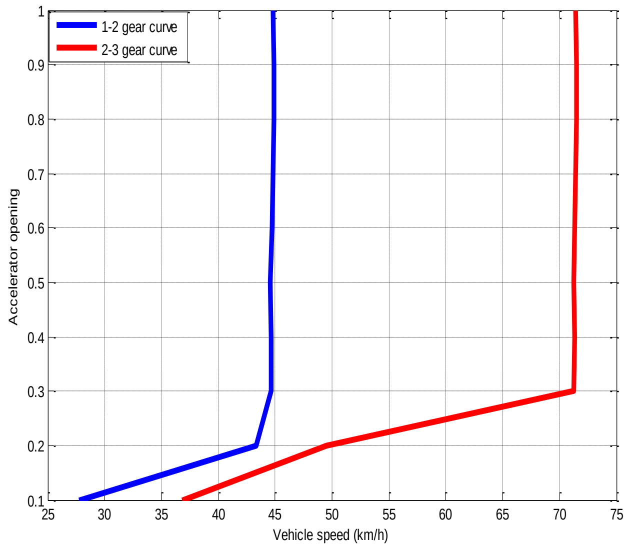

Fig.6 The optimizing dynamic performance upshift schedule

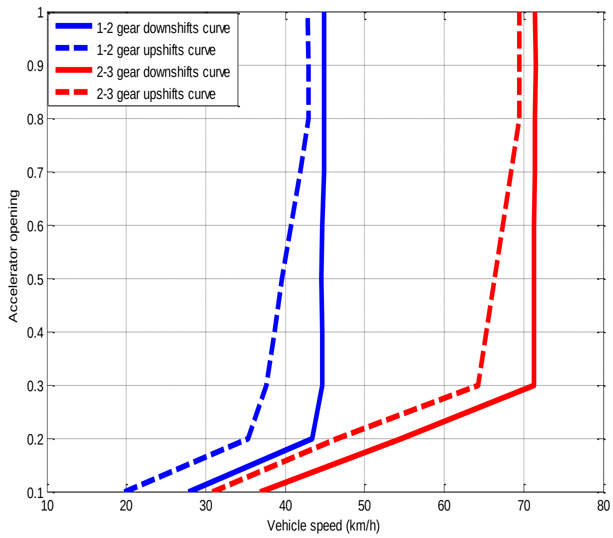

Fig.7 The optimizing dynamic performance shift schedule

VI. CONCLUSION

Firstly, the shift points' vehicle speeds are calculated through the theoretical analysis and research of the three-geared optimizing dynamic performance shift schedule for TBW. Then, according to the simulation and optimization analysis of the shifting process model established in MATLAB/Simulink, the optimizing dynamic performance shift schedule for the three-geared TBW is developed. The shift schedule could improve the vehicle performance, realize energy saving and increasing driving range. This research laid foundation for the latter part of the real vehicle tests.

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