Research of Optimizing Dynamic Performance Shift Schedule for Three Gears Transmission-By-Wire in Electric Vehicle

Chen Qi, Pan Zhang, Zhengbin Guo, Nana Lv, Jinyu Qu

***Abstract*— The shift schedule of automatic transmission plays an important influence on 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 theoretical model for the optimizing dynamic performance shift schedule was analyzed and the shift point speed is gotten under different accelerator pedal opening. The curve of optimizing dynamic performance shift schedule is developed which could improve the performance of electric vehicle, so as to achieve the purpose of saving energy and increasing the driving range.**

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

I. INTRODUCTION

Shift schedule refers to the regular pattern that transmission shifts according to the driving condition of electric vehicle [1-3]. According to the number of shift control parameters, the shift schedule can be divided into single parameter shift schedule, double parameters shift schedule and three parameters shift schedule. In this paper the double parameter shift schedule is adopted and the control parameters are the speed and the opening of accelerator pedal. According to the difference of shifting speed, the double parameters shift schedule can be divided into equal delay, convergent, divergent and combination.

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 TBW in electric vehicle is researched. The shift schedule is theoretically analyzed, the shift point speed is gotten under different accelerator pedal opening and analyze the shift schedule curve. The 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 [4].

II. VEHICLE PARAMETERS

With economy development and continuous development of new energy, electric vehicles' development will have more and more space to become the main force of automobile market [5]. The parameters matching and the dynamics shift schedule are studied based on a pattern electric vehicle. The basic parameters of the vehicle are shown in Table 1.

III. MOTOR'S OPERATING CHARACTERISTICS

Motor as the only power source of electric vehicle has a direct impact on the performance of the fuel economy and dynamic performance [6]. The motor used is a 15kW permanent magnet synchronous motor. And its technical parameters are shown in Table 2. The motor 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.

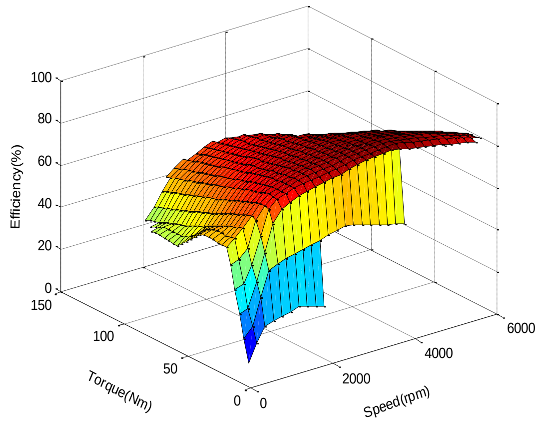

Fig.1 The motor speed, torque and efficiency of the three-dimensional MAP

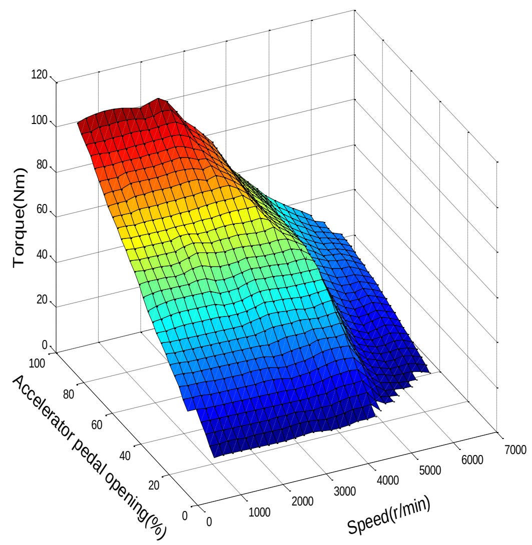

Fig.2 Torque variation at different accelerator opening

Table 1The basic parameters of vehicle

Parameter	Numerical	Unit
L * W * H	3797/1510/1820	mm
Train efficiency	0.96	--
Curb Weight	1030	Kg
Full quality	1310	Kg
Radius of the wheel	0.2805	M
Air resistance coefficient	0.6	--
Frontal area	2.27	m^2
Wheelbase	2500	mm
Main reduction ratio	3.182	--
T r1	1.7	--
T r2	1.3	--
T r3	1.0	--

Table 2 Technical parameters of permanent magnet synchronous motor

parameter	symbol	unit	numerical
Rated voltage	U	V	108
Rated power	P_e	kW	15
Peak power	$P_{\max}$	kW	28
Rated torque	T_e	Nm	46
Peak torque	$T_{\max}$	Nm	170
Rated speed	$n_{\max}$	r/min	3000
Peak speed	n_e	r/min	6800

From Fig.2 we can see that, 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. *Analysis of Shift Point*

To ensure the optimizing dynamic characteristics of electric vehicle, 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]:

$$\frac{d\mu}{dt}_{n} = \frac{d\mu}{dt}_{(n+1)} \tag{1}$$

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

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

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]:

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

Electric vehicle runs with certain acceleration drove by motor and the transmission system, and it overcomes rolling resistance, air resistance, grade resistance and acceleration resistance. The electric vehicle'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:

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

Kinematic differential equation is:

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

This can also be written as:

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

Where: δ is the correction coefficient of rotating mass; m is the full mass; μ_a 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: $f = 0.0076 + 0.000056\mu_a$.

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

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

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[8]:

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

When the preliminary calculation of power is done, if we do not know the exact value of I_f and $\sum I_w$, we can also use the relationship between the rolling resistance coefficient and the total driveline gear ratios ($i_g i_0$) to determine the approximate value of δ. This article selects δ_1 =1.40, δ_2 =1.18, δ_3 =1.10.

The relationship between n gears 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:

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

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:

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

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

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

This can also be written as:

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

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):

$$\mu_a = \frac{-B_0 \pm \sqrt{B_0^2 - 4A_0C_0}}{2A_0} \quad (17)$$

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

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≤α≤1.

For example, the opening of the accelerator pedal is 100%, we can get the 1-2 gear acceleration curve which is shown in Fig.3, and the 2-3 gear acceleration curve is shown in Fig.4.

According to the steps above, 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	25.37	36.89
0.2	30.14	47.23
0.3	40.37	53.17
0.4	41.28	62.75
0.5	42.26	69.68
0.6	42.74	70.27
0.7	44.73	75.16
0.8	49.63	79.44
0.9	51.06	85.27
1.0	51.74	86.16

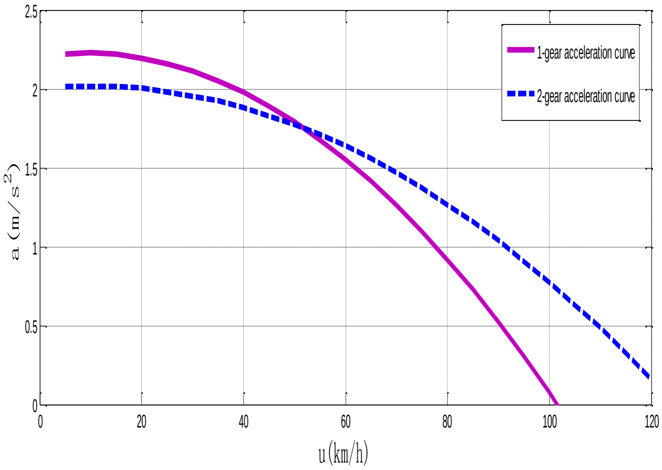

Fig.3 The 1-2 gear acceleration curve at 100% throttle opening

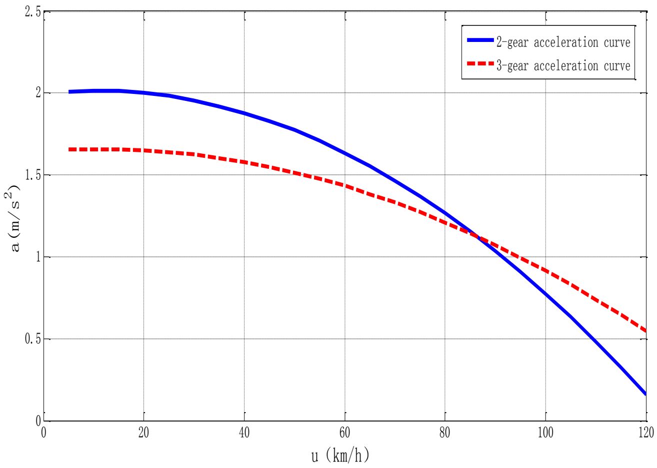

Fig.4 The 2-3 gear acceleration curve at 100% throttle opening

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 [10]. 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 [11]:

(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 [12], the optimizing dynamic performance shift schedule of EV is developed as shown in Fig.5.

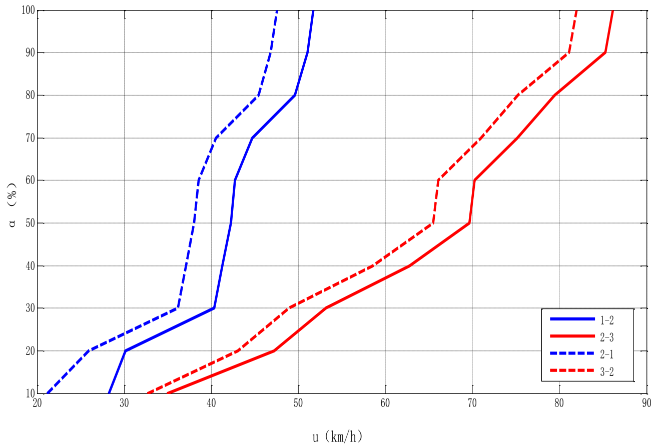

Fig.5 The optimizing dynamic performance shift schedule

IV Conclusion

In this paper, the shift points' vehicle speeds are calculated through the theoretical analysis and research of the three-geared optimizing dynamic performance shift schedule for TBW. 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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