Shift schedule with dynamic three-parameter brought in angular acceleration of engine

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Abstract

In order to make automatic transmission vehicles more reasonably select shift points, ensure optimal shift schedule, shift schedule with dynamic three-parameter was proposed through the analysis of the dynamic characteristics of engine and hydrodynamic torque converter under unstable conditions and establishment of dynamic model of vehicle driveline, where parameters were vehicle speed, throttle opening and angular acceleration of engine crankshaft. Shift schedule with dynamic three-parameter was better than shift schedule with dynamic two-parameter in power performance and fuel economy which was verified by simulation test on a certain type of heavy vehicle, then power performance and fuel economy of the optimal dynamic shift schedule and the optimal fuel economy shift schedule with dynamic three-parameter were respectively compared and analyzed. The results show that angular acceleration of engine crankshaft has a great influence on the selection of shifting points, and shift schedules with dynamic three-parameter proposed have important theoretical significance and engineering application value on improving the performance of automatic transmission vehicles, improving the power performance and fuel economy, saving energy and reducing emissions.

Keywords: automatic transmission, dynamic shift schedule with three-parameter, best power performance, best fuel economy

1 Introduction

In the application of automatic transmission vehicles, shifting operation is an important aspect to improve performance of the vehicle, and perfect automatic shifting system is one of the cores and important technologies to achieve automatic shifting. The basic requirements for automatic shift system include: to ensure optimal shift schedule so that the vehicle has a satisfactory power and excellent fuel economy and reduces pollution; stationary, small impact and low noise in shifting process; shifting gears selected accurately and timely without the phenomenon of "hang the wrong gear" or "jump out gear" [1, 2]. Shift schedule and best selection of shifting point have been the hot spot of the domestic and foreign research. The traditional shift schedule according to the control parameters can be divided into single parameter, two parameters and three parameters, and the most application is selected speed and throttle opening as the control parameters in the steady state, but the starting of the vehicle and the process of shifting are completed in unsteady state [3]. After Anlin Ge presented a dynamic shift schedule with three-parameter, research institutes and experts of domestic and foreign became study shift schedule with three-parameter of automatic transmission in unsteady state. Yong Zhang solved dynamic shift schedule with three-parameter under different quality parameters of vehicle movement [4]. Xinxin Zhao amended shift schedule with three-parameter by studying the changing rate of throttle opening influence on the speed of shifting [5]. Other researches mostly chose speed, throttle opening and acceleration as shifting parameters of dynamic shift schedule [6-11]. However, if making equal points of acceleration between two adjacent gears, the acceleration is not suitable for an input parameter as intermediate variable calculated. This paper introduced angular acceleration of engine crankshaft as one of the control parameters in order to research dynamic shift schedule with three-parameter of heavy vehicle equipped hydrodynamic automatic transmission (AT) in unsteady state. On the one hand it can reflect the changing rate of throttle opening, on the other hand it can reflect load, so that it can more truly reflect dynamic process of shifting.

2 Dynamic characteristics of engine and hydrodynamic torque converter in unsteady state

2.1 CORRECTION OF ENGINE PERFORMANCE IN UNSTEADY STATE

Influenced by the driver's mode of operation and driving environment, working condition of engine is given priority in unsteady state. Accelerating condition is the most important in unsteady state, so it is necessary to research the performance of engine in accelerating condition. Accurate model of engine is based on a large number of experimental data, which is extremely complex. Taken consideration, this paper introduced angular acceleration of engine crankshaft, using experimental data of engine in steady state to simulate engine model. The torque of engine decreased in the model obtained, and there was a linear relationship with the angular acceleration of the crankshaft [12, 13]:

$$M_e^D = M_e - \frac{\partial M_e}{\partial \varepsilon_e}\frac{d\omega_e}{dt}, \tag{1}$$

where M_e^D is dynamic torque of engine, N·m; M_e is torque of engine working as external characteristic, N·m; ω_e is angular velocity of crankshaft, rad/s; $\frac{\partial M_e}{\partial \varepsilon_e}$ is the instability coefficient of engine.

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2.2 DYNAMIC CHARACTERISTICS OF HYDRODYNAMIC TORQUE CONVERTER IN UNSTEADY STATE

Dynamic torque M_B^D (N·m) on the pump shaft and dynamic torque $-M_T^D$ (N·m) on the turbine shaft were both different from its static properties when hydrodynamic torque converter worked in unsteady state [14]. Figure 1 was a simplified model of driveline, simplified rotating parts from left to right respectively represented engine, pump of hydrodynamic torque converter, liquid in the pump, liquid in the turbine, rotating parts from turbine to wheel, wheels. M_B^{HD} and M_T^{HD} in the figure are respectively hydrodynamic torque on the pump and turbine in unsteady state (N·m). M_B^H and M_T^H are respectively hydrodynamic torque on the pump and turbine in steady state (N·m). J_B and J_T are respectively inertia torque of major rotating parts such as pump and shaft of pump, turbine and shaft of turbine (kg·m²). J_{BY} and J_{TY} are respectively inertia torque of working fluid in the pump and turbine (kg·m²). J_W is inertia torque of wheels including brake drum and other parts connected with it (kg·m²).

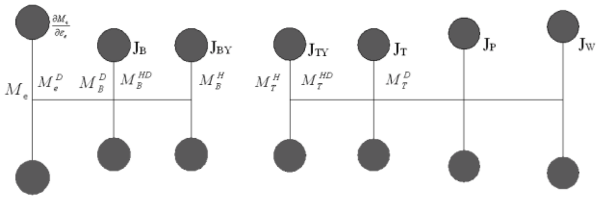

FIGURE 1 Simplified dynamic model of vehicle drive system

According to the law of conservation of momentum, dynamic torque on the pump and turbine without consideration of the change of mechanical loss was

$$\begin{cases} M_B^D = M_B^H + \rho F_{BY} \dfrac{dQ}{dt} + (J_B + J_{BY}) \dfrac{d\omega_B}{dt} \\ M_T^D = M_T^H - \rho F_{TY} \dfrac{dQ}{dt} - (J_T + J_{TY}) \dfrac{d\omega_T}{dt} \end{cases}, \quad (2)$$

where ω_B and ω_T are respectively angular velocity of the pump and turbine, rad/s; F_{BY} and F_{TY} are respectively form factor of geometric parameter between flow passes of blades of the pump and turbine; Q is circulation flow of the fluid in the working chamber of hydrodynamic torque converter, m³/s, assumed that it is consistent in steady state and in unsteady state. As the values of $\rho F_{BY} \dfrac{dQ}{dt}$ and $\rho F_{TY} \dfrac{dQ}{dt}$ are small, dynamic torque of hydrodynamic torque converter in unsteady state could be simplified as:

$$\begin{cases} M_B^D = M_B^H + (J_B + J_{BY}) \dfrac{d\omega_B}{dt} \\ M_T^D = M_T^H - (J_T + J_{TY}) \dfrac{d\omega_T}{dt} \end{cases}. \quad (3)$$

Characteristic parameters of hydrodynamic torque converter in unsteady state were: $\begin{cases} K = -\dfrac{M_T^H}{M_B^H} \\ \eta = Ki \\ \lambda_B = \dfrac{M_B^H}{\rho g D^5 n_B^2} \end{cases}$, where K is torque ratio of the hydrodynamic torque converter; η is efficiency of the hydrodynamic torque converter; i is speed ratio of the hydrodynamic torque converter; λ_B is torque coefficient of the pump, min²/(r²·m); ρ is density of the working fluid, kg/m³; D is effective diameter of the hydrodynamic torque converter, m; n_B is rotation speed of the pump, r/min.

Solving M_B^H and M_T^H in steady state through experimental data K, λ_B and η obtained by steady state, and then given angular acceleration of engine crankshaft as $\dfrac{d\omega_e}{dt} = \dfrac{d\omega_B}{dt}$, the torque of pump M_B^D in unsteady state could be obtained through taking advantage of Equation (3). Since $\dfrac{d\omega_T}{dt} = \dfrac{i_g i_0}{r_k} \dfrac{du}{dt}$, where i_g is ratio of the transmission; i_0 is the main transmission ratio; u is the speed, m/s; r_k is radius of the wheel m, then

$$M_T^D = K M_B^H - \left(J_T + J_{TY}\right) \frac{i_g i_0}{r_k} \frac{du}{dt}. \quad (4)$$

3 Dynamic shift schedule with three-parameter introduced angular acceleration of engine crankshaft

3.1 THE SHIFT SCHEDULE FOR BEST POWER PERFORMANCE

The intersections solved by curves of acceleration between two adjacent gears at the same throttle opening as shifting points to ensure best power performance. The process of solving acceleration was as follows. Driving equation for heavy vehicle with hydrodynamic torque converter was:

$$\frac{M_T^H i_g i_0 \eta_T}{r_k} = F_f + F_w + F_i + \delta m \frac{du}{dt}, \quad (5)$$

where η_T is the efficiency from turbine to wheels; F_f is rolling resistance, N, $F_f = fmg\cos\alpha$, in which f is coefficient of rolling resistance, α is road grade. F_w is air resistance, N, $F_w = \dfrac{C_D A u^2}{21.15}$, in which C_D is coefficient of air resistance, A is windward area, m². F_i is grade resistance, N, $F_i = mg\sin\alpha$; m is the quality of the whole vehicle, kg; $\dfrac{du}{dt}$ is acceleration of driving, m/s². δ is conversion factor for rotating mass of heavy vehicle, δ>1. According to the power equation of dynamics with the turbine as the source of power output, the conversion factor for rotating mass of the whole vehicle without ignoring rotational inertia from output shaft of turbine to driving pinion gear of main actuator was [15]:

$$\delta = 1 + \frac{\sum J_W}{mr_k^2} + \frac{J_P i_0^2 \eta_p}{mr_k^2} + \frac{i_g^2 i_0^2 \eta_T}{mr_k^2}\left(J_T + J_{TY}\right), \tag{6}$$

The expression of acceleration was received by substituting it into driving equation of heavy vehicle:

$$\frac{du}{dt} = \frac{\dfrac{M_T^H i_g i_0 \eta_T}{r_k} - \left(F_f + F_w + F_i\right)}{m + \dfrac{\sum J_W}{r_k^2} + \dfrac{J_P i_0^2 \eta_p}{r_k^2} + \dfrac{i_g^2 i_0^2 \eta_T}{r_k^2}\left(J_T + J_{TY}\right)}. \tag{7}$$

Acceleration curves of different gears at the same throttle opening could be obtained by using Equation (7). If there was intersection of acceleration curves between two adjacent gears, taking the intersection as shifting point; if there was not intersection, taking the maximum speed of the gear at the throttle opening as the shifting point.

According to the method above, shifting points of different angular accelerations of crankshaft at each throttle openings through changing the throttle opening and angular accelerations of engine crankshaft, so that shift schedule with throttle opening, speed and angular accelerations of engine crankshaft was got. Considering the influence of dynamic characteristics of hydrodynamic torque converter, the emphasis was on the shift schedule in condition of hydrodynamic. Figure 2 showed shift schedule from first gear to second gear in condition of hydrodynamic, and six surfaces from bottom to the top in Figure 3 were respectively shift schedule for best power performance from first gear to second gear, from second gear to third gear, from third gear to fourth gear, from fourth gear to fifth gear, from fifth gear to sixth gear, in which a is throttle opening, *v* is speed, *e* is angular acceleration of engine crankshaft.

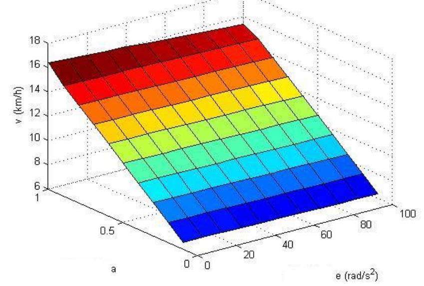

FIGURE 2 The shift schedule for best power performance from first gear to second gear

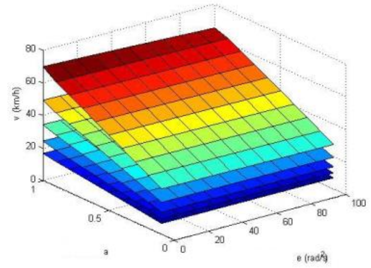

FIGURE 3 Upshifts of shift schedule for best power performance under the hydrodynamic condition

3.2 THE SHIFT SCHEDULE FOR BEST FUEL ECONOMY

Fuel consumption and emissions of harmful gas account for a dominant position when the heavy vehicle drives in unsteady state, in which working condition of acceleration is the most important [12]. This paper focused on the shift schedule for best fuel economy in accelerating condition when the position of accelerator pedal did not change. To get the shift schedule for best fuel economy, it should be made the total fuel consumption Q_s minimized, of which the vehicle continuous changed i gears from starting by the time t_s to a certain speed, namely:

$$Q_s = \sum_{n=1}^{i} \int_{u_{n-1}}^{u_n} \frac{Q_{T,n}^D \delta_n m}{F_{t,n} - \sum F} du + \int_{u_i}^{u_s} \frac{Q_{T,s}^D \delta_s m}{F_{t,s} - \sum F} du\ . \tag{8}$$

Fuel consumption of two adjacent gears was minimized when shifting, so the total fuel consumption was minimized. Fuel consumption of two adjacent gears is:

$$Q_n = \int_{u_{n-1}}^{u_n} \frac{Q_{T,n-1}^D \delta_{n-1} m}{F_{t,n-1} - \sum F} du + \int_{u_n}^{u_{n+1}} \frac{Q_{T,n}^D \delta_n m}{F_{t,n} - \sum F} du\ . \tag{9}$$

The minimum of Q_n is seeking for the extreme value of equation (9), that is $\frac{dQ_n}{du} = 0$. For two adjacent gears, there is:

$$\frac{Q_{T,n-1}^D \delta_{n-1}}{F_{t,n-1} - \sum F} = \frac{Q_{T,n}^D \delta_n}{F_{t,n} - \sum F}\ . \tag{10}$$

In the numerical solution, the fuel consumption in accelerating condition is:

$$Q_{T,n}^D = \frac{P_{e,n} g_{e,n}}{1.02 u \rho g}, \tag{11}$$

where ρg is generally preferable for 7.94-8.13N/L for diesel, ρ is the density of fuel, kg/L, g is acceleration of gravity, m/s^2; $P_{e,n}$ is the power provided by engine, kW, $P_{e,n} = \frac{v_n}{3600\eta_T}(F_f + F_w + F_i + \delta_n m \frac{du}{dt})$, where v_n is speed at n gear, η_T is mechanical efficiency of the drive train, and the other parameters is the same with the shift schedule for best power performance. Mathematical model of fuel consumption rate of engine is [16]

$$g_{e,n} = \sum_{i=0}^{S} \sum_{j=0}^{i} A_i \left[\frac{1}{2}(i+1)(i+2) - i - 1 + j\right] M_e^j n_e^{i-j}, \tag{12}$$

[g/(kw·h)], where A_i is the coefficient in the model; S is the order in the model, generally S=2. Fuel consumption fitting is shown in Figure 4, in which n is speed of engine, *Me* is torque of engine, *ge* is fuel consumption.

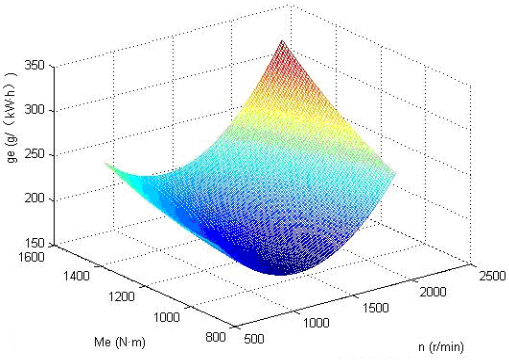

FIGURE 4 Fuel consumption

The shift schedule for best fuel economy from first gear to second gear according to Equation (10) was shown in Figure 5. Six surfaces from bottom to the top in Figure 6 were respectively shift schedule for best fuel economy from first gear to second gear, from second gear to third gear, from third gear to fourth gear, from fourth gear to fifth gear, from fifth gear to sixth gear.

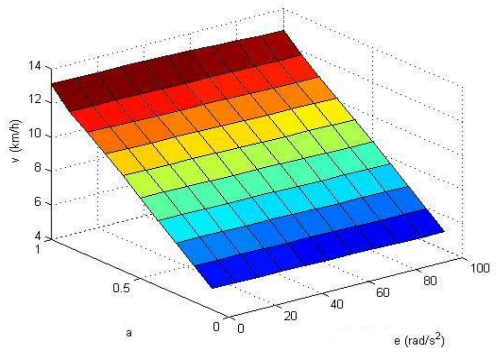

FIGURE 5 The shift schedule for best fuel economy from first gear to second gear

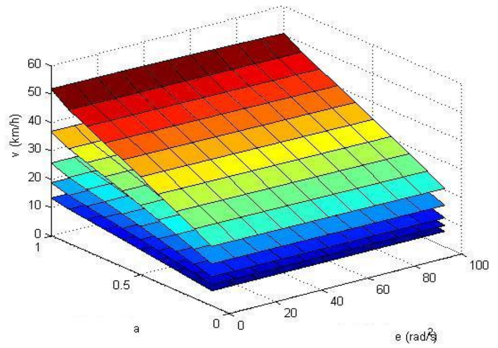

FIGURE 6 Upshifts of the shift schedule for best fuel economy under the hydrodynamic condition

4 Simulation examples and results

4.1 THE INITIAL CONDITIONS OF SIMULATION

The correctness and feasibility of theoretical research are verified though simulation test, which makes up for high-cost and long-cycle of prototype test. In the paper simulation model of the whole vehicle was established based on dynamics equations of driveline using MATLAB/SIMULINK and MATLAB/STATEFLOW, as shown in Figure 7. Then comparative analyzed the results of dynamic shift schedule with three-parameter and two-parameter. The basic parameters of the whole vehicle and powertrain were shown in Table 1. The simulation model had seven sub-modules, including: driver, engine, TC, gearbox, vehicle body, shift law, shift logic. Oscilloscope speed was used to display the speed; oscilloscope *Q* was used to display the fuel consumption per unit of time when speeding up. Due to space limitations, the internal structure of each sub-module was no longer introduced one by one.

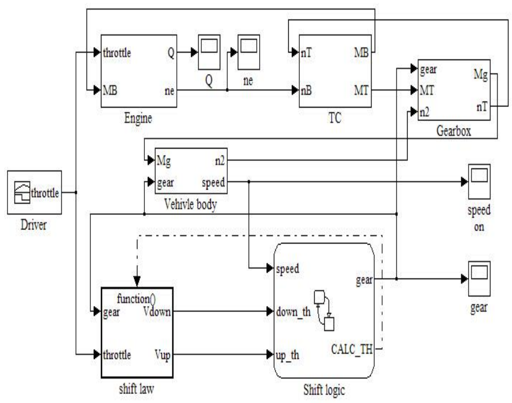

FIGURE 7 Simulation model of the whole vehicle

TABLE 1 Basic parameters of the vehicle and the driveline

N	Name of parameters	Value of parameters
1	Total quality with load	25000kg
2	Dynamic radius	0.573m
3	Coefficient of rolling resistance	0.0041+0.0000256ua
4	Coefficient of air resistance	0.9
5	Windward area	10.5m^2
6	Total efficiency of the drive train	0.85
7	Transmission ratio of reverse gear	5.08
8	Transmission ratio of first gear	3.947
9	Transmission ratio of second gear	2.659
10	Transmission ratio of third gear	2.012
11	Transmission ratio of fourth gear	1.355
12	Transmission ratio of fifth gear	1
13	Transmission ratio of sixth gear	0.6736
14	Main reduction ratio	6

Note: u_a was speed, km/h

4.2 ANALYSIS OF RESULTS

Dynamic shift schedule for best power performance with three-parameter and shift schedule for best power performance with two-parameter were respectively input into the simulation models, in order to compare acceleration time with speed from 0 to 100km/h at maximum throttle opening, and then power performance of the vehicle was evaluated [16]. Acceleration time contrasted curves of dynamic shift schedule for best power performance with three-parameter and shift schedule for best power performance with two-parameter were shown in Figure 8. We can see that it need 14 seconds as shift schedule with three-parameter, while it needs 15.64 seconds as shift schedule with two-parameter when the speed reached 100km/h. So the power performance of shift schedule with three-parameter improved 10.5% than shift schedule with two-parameter. Dynamic shift schedule for best fuel economy with three-parameter was input into the simulation model. Acceleration time contrasted curves of dynamic shift schedule for best power performance with three-parameter and shift schedule for best fuel economy with three-parameter were shown in Figure 9. We can see that it needs 16.32 seconds as shift schedule for best fuel economy when the speed reached 100km/h and decreased by 16.57% than shift schedule for best power performance.

As dynamic shift schedule for best fuel economy with three-parameter and shift schedule for best fuel economy with two-parameter, in order to fuel consumption per unit of time under different shift schedules at the moment of throttle opening maximized. Fuel consumption contrasted curves of dynamic shift schedule for best fuel economy with three-parameter and shift schedule for best fuel economy with two-parameter were shown in Figure 10. We can see that fuel consumption was 7.12 mL/s as shift schedule for best fuel economy with two-parameter at 7.32 second, while fuel consumption was 6.955 mL/s as dynamic shift schedule for best fuel economy with three-parameter and decreased 2.3% than shift schedule with two-parameter. Fuel consumption contrasted curves of dynamic shift schedule for best power performance with three-parameter and shift schedule for best fuel economy with three-parameter were shown in Figure 11. We can see that fuel consumption was 7.23 mL/s as dynamic shift schedule for best power performance with three-parameter at 6.28 second, while fuel consumption was 6.97 mL/s as dynamic shift schedule for best fuel economy with three-parameter and decreased by 3.6% than shift schedule for best power performance.

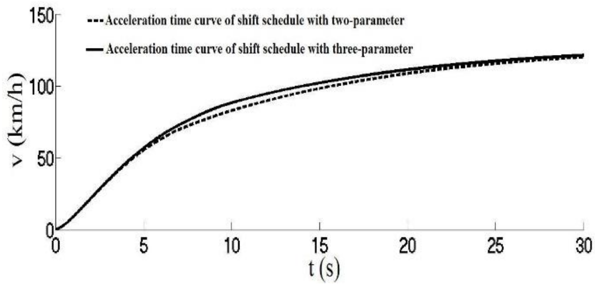

FIGURE 8 Acceleration time contrasted curves of dynamic shift schedule for best power performance with three-parameter and with two-parameter

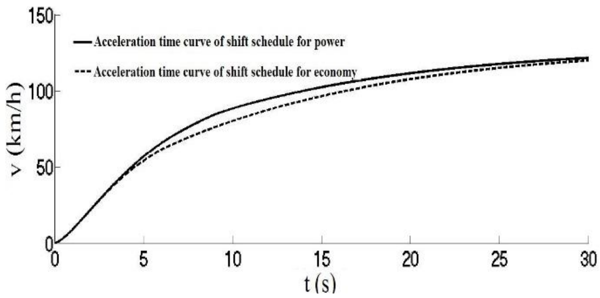

FIGURE 9 Acceleration time contrasted curves of dynamic shift schedule for best power performance with three-parameter and for best fuel economy

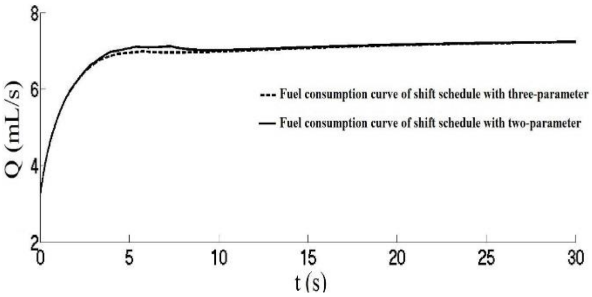

FIGURE 10 Fuel consumption contrasted curves of dynamic shift schedule for best fuel economy with three-parameter and with two-parameter

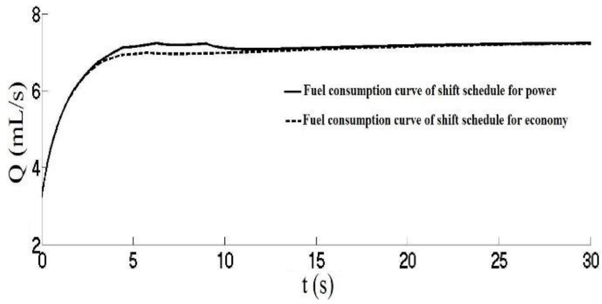

FIGURE 11 Fuel consumption contrasted curves of dynamic shift schedule f for best power performance with three-parameter and for best fuel economy

The power performance of dynamic shift schedule with three-parameter introduced angular acceleration of crankshaft was better than dynamic shift schedule with two-parameter, and fuel consumption of dynamic shift schedule for best fuel economy with three-parameter was less than dynamic shift schedule for best power performance with three-parameter.

5 Conclusions

This study proposed dynamic shift schedule with three-parameter, which making throttle opening, speed, and angular acceleration of crankshaft as control parameters based on that rate of throttle opening and external load could be reflected by angular acceleration of crankshaft, combined with the dynamic characteristics of engine and hydrodynamic torque converter in unsteady state. Angular acceleration of engine crankshaft had great influence on selection of optimal shifting points, which was verified through simulation tests of a certain heavy vehicle, and compared with dynamic shift schedule with two-parameter, the results showed that the power performance of dynamic shift schedule for best power performance with three-parameter improved 10.5% than shift schedule for best power performance with two-parameter, fuel consumption of dynamic shift schedule for best fuel economy with three-parameter was less 2.3% than dynamic shift schedule for best fuel economy with two-parameter. Meanwhile, compared dynamic shift schedule for best power performance with three-parameter with dynamic shift schedule for best fuel economy with three-parameter, the power performance of dynamic shift schedule for best fuel economy with three-parameter decreased by 16.57% than dynamic shift schedule for best power performance with three-parameter, while the fuel consumption of dynamic shift schedule for best fuel economy with three-parameter reduced 3.6% than dynamic shift schedule for best power performance with three-parameter. The shift schedule proposed had important theoretical significance and application value on improving the performance of vehicles equipped automatic transmission, improving power performance and fuel economy, saving energy and so on.

Acknowledgments

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