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Modelling and Co-simulation Based on AMESim and Simulink for Light Passenger Car with Dual State CVT

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Abstract

Co-simulation platform was constructed using MATLAB/Simulink and AMESim for light passenger car with dual state CVT. The vehicle dynamic model, hydraulic system and controller model were established based on the co-simulation platform. Through simulation analysis of typical working conditions, it was validated that the co-simulation platform was effective and practical for R&D of passenger car with CVT.

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Keywords: Modelling; Co-simulation; Light Passenger Car; Dual State CVT

1. Introduction

Dual state CVT is a combination of optimal matching of CVT and torque converter. Vehicle with dual state CVT has good performance to start up, low speed crawling at the stage of low speed moving, and good driving and economy performance at the stage of high speed moving. Furthermore, engine efficiency can be improved, and vehicle's power and economy performance can be increased to great degree.

2. Co-simulation Platform Structure

Modeling and simulation using MATLAB/Simulink tool for dynamic analysis has a very high level of accuracy, but also has the conflict between model simplification and precision for complex nonlinear system. LMS Imagine.Lab AMESim software provides graphical modeling approach, which eliminates the tedious mathematical modeling, code programming and has ensures high modeling efficiency, but has the weakness of complex control system modeling. In this paper, co-simulation platform based on

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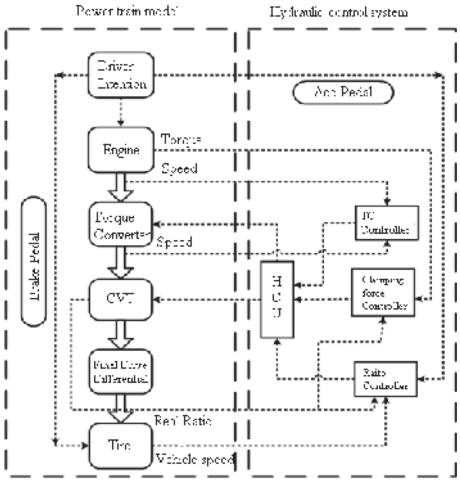
Simulink and AMESim was established combining the advantages of the both. The block diagram is shown in Fig.1 (a).

3. Dynamic Model of Light Passenger Car with Dual State CVT

3.1. Engine Model

The dynamic characteristics of the engine are simplified as a first order lag inertia considering engine output torque is in non-steady-state conditions in most cases[1]. The non-steady-state torque value can be obtained through correcting the steady-state values by experience factor. Fig.1 (b) shows the numerical model of steady-state.



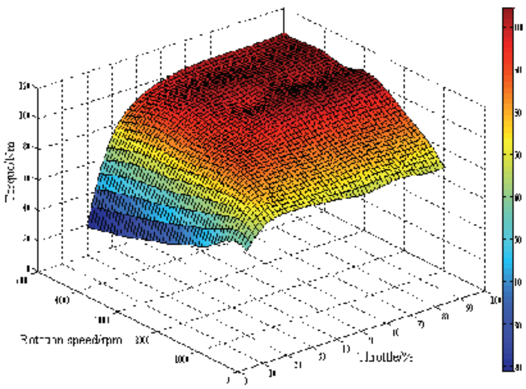


Fig.1 (a) Co-simulation Platform Based on AMESim and Simulink (b) Steady-state numerical model of Engine output torque

With a first order lag inertia, the dynamic characteristics of the engine are expressed as:

$$T_e = e^{-\tau s} \cdot T_s\left(\alpha, \omega_e\right) / \left(k_\tau s + 1\right) \tag{1}$$

Where, T_e is engine torque of non-steady-state; T_s is output torque of steady-state; α is the throttle opening; ω_e is engine speed; τ is lag time; k_τ is dynamic fit coefficient.

3.2. Torque Converter Model

The input and output characteristics of torque converter are expressed as follows:

$$K = \omega_i / \sqrt{T_i} \tag{2}$$

$$SR = \omega_i / \omega_t \tag{3}$$

$$TR = T_t / T_i \tag{4}$$

Where, K is the capacity factor; TR is torque ratio; SR is speed ratio; ω_i is impeller speed; ω_t is turbine speed; T_i is impeller torque; T_t is turbine torque.

3.3. CVT Model

The torque balance equation of input and output shaft of CVT[2-5] is:

$$\left(T_e - I_e \cdot d\omega_e / dt\right) \cdot i_{CVT} \cdot \eta - T_d = I_d \cdot d\omega_d / dt \tag{5}$$

Where, I_e is engine inertia; T_d is the equivalent inertia of CVT output shaft converted from vehicle driving resistance; I_e is the equivalent rotation inertia of CVT input shaft converted from flywheel and primary pulley; I_d is the equivalent inertia of CVT output shaft converted from secondary pulley and final drive; ω_d is the angular velocity of CVT output shaft; i_{CVT} is CVT transmission ratio; η is transmission efficiency.

3.4. Tire Model

Considering tire rotation inertia, *HBPacejka* is selected as the tire model, which can be expressed as follows:

$$F_x = D\sin(C\arctan(Bs - E(Bs - \arctan(Bs)))) \tag{6}$$

Where, F_x is the tire longitudinal force; s is slip ratio; D is the peak factor; B is the stiffness factor; E is the curve shape factor; C is the shape characteristic factor.

3.5. Hydraulic Control System Model

3.5.1. Hydraulic System

By simplifying the hydraulic circuit appropriately, retaining the main control valves, such as the speed ratio control valve, clamping force control valve[6], the hydraulic control unit was established which was shown in Fig.2(a).

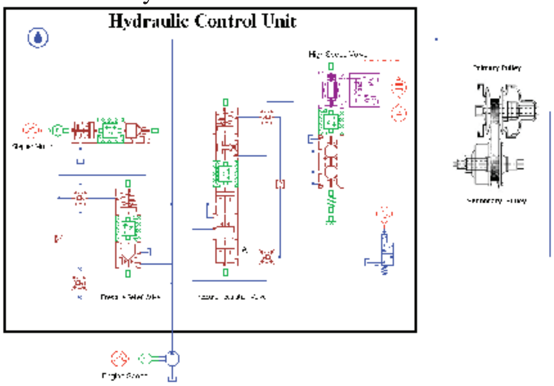
3.5.2. Controller Model

Torque converter controller only considered the lock/unlock state control. In order to take into account the transmission efficiency and lock-up comfort requirements and avoid excessive torque impact, torque converter controller selected the coupling point as the lock-up control point. When the torque converter speed ratio increased to the coupling point, the controller would send lock-up command. When the vehicle speed was lower than the threshold value which was preconfigured (such as 15km/h), the controller would send unlock command.

Clamping force control can guarantee the safety of torque delivery, which can improve transmission efficiency and the key components' life. The clamping force controller considers both the transfer of engine torque and the current ratio simultaneously. The controller receives speed, torque and pressure sensor signals in real time to calculate the value of current engine torque and transmission ratio at this time, then determines the target clamping force control command to clamping force control valve.

Ratio controller according to the selected driving mode and the driver's intention (the throttle opening and the brake pedal position and other parameters) to adjust the actual ratio to track the target ratio changing [7-9]. Taking into account the complex operating conditions and good adaptation to nonlinear systems, a fuzzy ratio controller was designed. Input signals are the difference between the target ratio and the actual ratio, and deviation of the rate of change, the control signal is the duty cycle signal to control ratio control valve. Fig.2 (b) is the fuzzy controller output surface.

The vehicle dynamic model based on co-simulation platform is illustrated as Fig.3 (a).



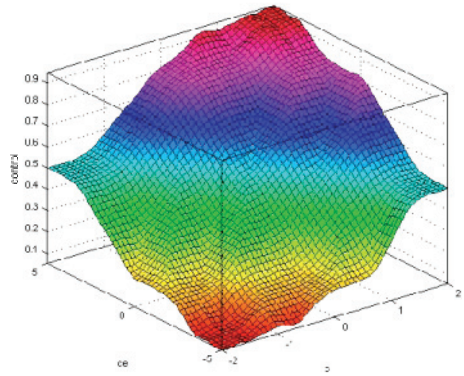
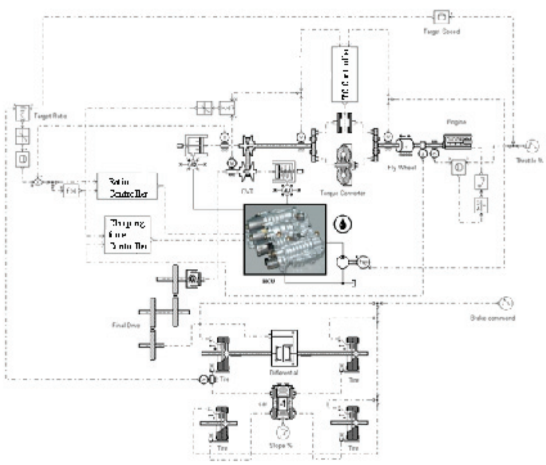


Fig.2 (a)AMESim Model of CVT Hydraulic Control Unit (b) Fuzzy Controller output surface



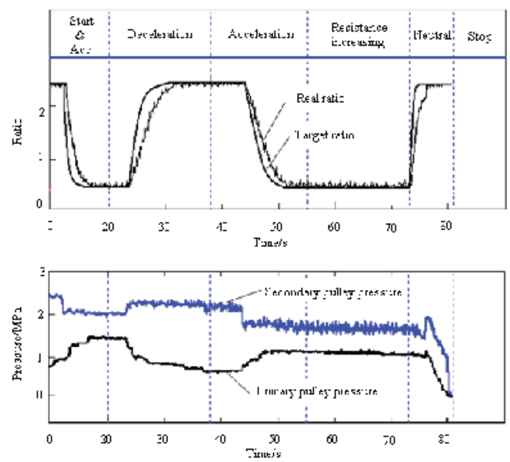


Fig.3 (a)Vehicle Model based on co-simulation platform (b) Simulation analysis results under cycle operating condition

4. Simulation Analysis under Typical Operating Conditions

To validate the dynamic characteristics and control effect of vehicle co-simulation model, typical cycle conditions were selected, which included passenger car's start up, acceleration, deceleration, increasing in driving resistance, neutral gear position and stop. The simulation results were shown as Fig.3 (b).

As shown in Fig.3 (b), when simulation began, the car started up and increased the velocity. As the speed increased, the CVT transmission ratio decreased from maximum to minimum. After maintaining a constant speed, then reduced the throttle opening, which made the car slow down, then the primary pulley cylinder drained and pressure down. The actual ratio tracked the target speed to increase. Then acceleration again, with the speed increasing, pressure of primary cylinder increased at the same time. When the driving resistance increased, the load torque of transmission increased as well as the pressure of secondary pulley cylinder to ensure effective torque transferring. Subsequently, the transmission lever was set to neutral gear, the pressure of primary and secondary pulley cylinders drained at the same time, and the speed ratio of CVT automatically went back to the maximum value to ensure start up again. At last the gear lever was set in the park and the car stopped [10].

From the simulation results, it was validated that the vehicle dynamic model based on co-simulation platform can well adapt to the typical operating conditions. The hydraulic control unit can accord with the driving conditions and driver's intention to control the hydraulic actuators to realize corresponding driving behavior, such as start up, acceleration, deceleration, neutral gear position and stop.

5. Conclusion

A modeling and co-simulation method for light passenger car with dual state CVT was introduced. The co-simulation platform was based on Simulink and AMESim. The vehicle power-train dynamic model and hydraulic system model were constructed by AMESim, and the controller models were built by MATLAB/Simulink. Through simulation analysis results, it was validated that the co-simulation platform was effective and practicable for R&D of dual state CVT.

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Appendix A.

A.1. Basic Parameters of the Vehicle Model

Parameter	Symbol	Unit	Value
Vehicle Mass	M	Kg	1560
Engine Max. Power	P_{emax}	Kw	56
Engine Max. Torque	T_{emax}	Nm	115
Frontal Area	A	m^2	1.68
Wind Resistance Coefficient	C_D		0.32
Rolling Resistance Coefficient	f		0.012
Wheel Rolling Radius	r_w	m	0.27
CVT Ratio Range	i_{CVT}		0.442~2.45
Center Distance Between Primary and Secondary Pulley	A_p	m	0.15
Final Drive Ratio	i_0		5.249
Min. Radius of Primary Pulley	r_p	m	0.03057
Min. radius of Secondary Pulley	r_s	m	0.03231