A NOVEL ROAD DYNAMIC SIMULATION APPROACH FOR VEHICLE DRIVELINE EXPERIMENTS

Wen-Li Li *

Key Laboratory of Advanced Manufacture Technology for Automobile Parts
Ministry of Education, Chongqing University of Technology
Chongqing 400054, China
Chongqing Tsingshan Industrial
Chongqing 402761, China

Jing-Jing Wang
Chongqing Vocational Institute of Engineering
Chongqing 402260, China

Xiang-Kui Zhang
Chongqing Tsingshan Industrial
Chongqing 402761, China

Peng Yi
Chongqing Academy of Science and Technology
Chongqing 401123, China

Abstract. A dynamic simulation approach for performing emulation experiments on vehicle driveline test bench is discussed in this paper. In order to reduce costs and shorten new vehicle development cycle time, vehicle simulation on the driveline test bench is an attractive alternative at the development phase to reduce the quantity of proto vehicles. This test method moves the test site from the road to the bench without the need for real chassis parts. Dynamic emulation of mechanical loads is a Hardware-in-the-loop (HIL) procedure, which can be used as a supplement of the conventional simulations in testing of the operation of algorithms without the need for the prototypes. The combustion engine is replaced by a electric drive motor, which replicates the torque and speed signature of an actual engine. The road load resistance of the vehicle on a real test road is accurately simulated on load dynamometer motor. On the basis of analyzing and comparing the advantages and disadvantages of the inverse dynamics model and the forward model based on speed closed loop control method, in view of the high order, nonlinear and multi variable characteristics of test bench system, a load simulation method based on speed adaptive predictive control is presented. It avoids the complex algorithm of closed loop speed compensation, and reduces the influence of inaccurate model parameters on the control precision of the simulation system. The vehicle start and dynamic shift process were simulated on the test bench.

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* Corresponding author: Wen-Li Li.
1. Introduction. Bench test is an important part of modern vehicle development. A lot of time and money can be saved when using simulation models instead of real vehicle component. Vehicle driveline test bench is a platform for research, develop and testing of transmission, clutch and drive shaft. The test results were reasonable when test unit operates in a similar condition as real vehicle. With the improvement of electric and control technique, most benches use motor as dynamometer to achieve better response and dynamic performance. Research on dynamic load emulation is a hot point of test bench study. To meet the demand, static tests(tests at a constant speed or with a constant torque) and testing transmission for a dynamic test environment, as if each part are mounted and operated on a vehicle driving on road, are requisite in the development phase of automobile component [2, 3]. However, the dynamic simulation requires the precision of control system and fasten the system response speed. The coupling effect between rotation speed and torque in dynamic loading process is a key technical problem.

Compared to the classical approach, a rapid prototyping technique, called dynamic emulation of mechanical loads (DEML), was developed in order to test the performance of drives in the mechanisms. It is an interesting new aid for the design of the variable speed and torque control algorithms in mechatronics. Without the need for the prototype it enables the development of controllers for various components, additionally enabling several testing possibilities. The method is successful also in the case of rapid speed and torque changes. It enables the testing of changing inertia and highly nonlinear dynamics. Recent research, dealing with the dynamic emulation of mechanical loads [9, 4, 12, 15, 5], presented some useful approaches.

It is very convenient to carry out dynamic simulation test in the indoor laboratory by simulation technology and equipment with the condition that the simulation system is not easy to test. The research of mechanical load simulation technology provides a feasible method for the application of different drive systems. In the traditional load simulation test, the load motor is usually used for static loading test which is working at static simulation or load simulation in open loop. This method can not meet the high inertia, nonlinear (friction) and (rotational speed, torque) dynamic response of the fast operating conditions. The dynamic load simulation control method is gradually developed with the development of electronic control technology. The earlier dynamic simulation method is based on inverse dynamics model, but it has many disadvantages. The dynamic simulation method based on closed loop speed control has obtained good simulation results in literature [13], the load torque required to load the system is obtained from the three mass system simulator. However, the test bench controller was designed with linear methods. Nonlinear control methods are developed in the literature [1, 16], in order to verify the influence of nonlinear friction, the nonlinear friction compensation module is designed and the simulation results are very well. However, the speed closed loop control algorithm based on nonlinear system contains complex compensation links, and a higher requirements of nonlinear controller makes the control method complex.

In order to avoid the complexity compensation in speed closed loop control algorithm, this paper has proposed a dynamic control based on speed adaptive predictive control, which used prediction module instead of compensation module in simulation controller. It not only simplifies the control flow, but also greatly simplified the control error caused by the inaccuracy of the control flow and model parameters.
2. Dynamic emulation of mechanical loads. The mechanical load dynamic emulation control system was shown in Figure 1, set \( T_e(s) \) as drive motor electromagnetic torque, \( T_L(s) \) as load motor electromagnetic torque, \( \omega_{em}(s) \) as target speed for simulated system, \( G_{em}^{-1}(s) \) and \( G^{-1}(s) \) as the simulated object and inverse dynamic model of the simulation system, \( J_m \) and \( B_m \) as inertia and viscous friction coefficient of the load dynamic emulation system, \( J_{em} \) and \( B_{em} \) as inertia and viscous friction coefficient of the simulated system.

\[
T_e(s) = \omega_{em}(s) \quad (1)
\]

\[
T_L(s) = \omega(s) \left( G_{em}^{-1}(s) - G^{-1}(s) \right) \quad (4)
\]

The block diagram of the control algorithm is as Figure 2,
It can be concluded that in order to obtain the load motor torque set value $T_L(s)$ with formula 4, the inverse model $G^{-1}_{em}(s)$ of the simulated system must be solved. However, the inverse model of nonlinear system is difficult to be expressed by transfer function, because actual system has many nonlinear factors.

By the Newton’s law:

$$T_e - T_L = J_m \frac{d\omega}{dt} + B_m \omega \quad (5)$$

$$T_e = J_{em} \frac{d\omega}{dt} + B_{em} \omega \quad (6)$$

Combined the equation 5 and 6:

$$T_L = (J_{em} - J_m) \frac{d\omega}{dt} + (B_{em} - B_m) \omega \quad (7)$$

By the equation 7, an angular acceleration sensor must be set in the simulation process to get the load torque, otherwise, when the acceleration is obtained by differentiating the speed signal. There will be a large signal noise, leading to inaccurate estimates of acceleration.

In the aspect of dynamic load simulation control theory, the traditional control algorithm is to establish the “speed –torque” model of the target system (inverse dynamics model) and deduce the acceleration by current speed, and then get the target torque. The algorithm is simple in structure and easy to be deduced. It is especially suitable for the traditional dynamometer simulation test platform in the torque control mode. Due to the high order, nonlinear and multi variable characteristics of the dynamometer simulation system, the inverse model of nonlinear system is difficult to be expressed by transfer function, so it is difficult to obtain an accurate control model. At the same time, the control method based on inverse model must try to differentiate the speed measurement value to get the acceleration value, this process amplifies the effect of signal noise and sampling error on the system. So the dynamic load simulation control method based on inverse model is confronted with many difficulties in practical application.

2.2. The control algorithm based on speed closed loop. For dynamic simulation control algorithm, the traditional control method is to dynamically compensate the speed or position control loop and add feed forward module in the control loop to avoid solving the inverse model, using feed forward link instead of compensation module to improve the simulation results. This method has achieved satisfactory results in the nonlinear control system in literature [17, 14].

The speed closed loop control algorithm based on forward-model is shown in Figure 3, $G_t(s)$ is feedback controller, $G_d(s)$ is the transfer function between the given command and response of the load motor. The actual speed of the simulation system is followed to calculate the expected torque of the load motor. This control method avoids solving the inverse dynamics model of the simulation system, the simulation accuracy of the control system depends on the closed loop control.

The follow formula can be obtained by the control block diagram of Figure 2:

$$\frac{\omega(s)}{T_e(s)} = G_{em}(s) \frac{1/G_{em}(s) + G_t(s)}{1/G_t(s) + G_t(s)} \quad (8)$$

Z.Hakan Akpolat [10] added a feed forward compensator $G_{comp}(s)$ to the control block diagram of Figure 3 and the control block diagram was shown in figure 4-8.
In Figure 4, suppose \( G_{\text{comp}}(s) = \frac{\frac{1}{G_T(s)} + G_I(s)}{1/G_{\text{em}}(s) + G_T(s)} \), then the formula 9 can be obtained:

\[
\frac{\omega(s)}{T_e(s)} = G_{\text{em}}(s)G_{\text{comp}}(s) \frac{\left(\frac{1}{G_{\text{em}}(s)} + G_I(s)\right)}{\frac{1}{G_T(s)} + G_I(s)} = G_{\text{em}}(s)
\]  

(9)

The mechanical load dynamic model of the simulated object is maintained in the formula 9, it can achieve the goal of dynamics simulation in theory. However, the inverse dynamic model \( G_{-1}{\text{em}} \) of the simulation object need to be eliminate in the feed forward compensator. If the feed forward model is introduced on the basis of Figure 3, and then the feed forward compensator is added in Figure 4, the control block diagram shown in Figure 5.

The transfer function of the simulation system can be obtained by the control block diagram shown in Figure 5.

\[
\frac{\omega(s)}{T_e(s)} = G_{\text{em}}(s)G_{\text{comp}}(s) \frac{G(s)G_I(s)}{1 + G(s)G_T(s)}
\]  

(10)

If \( G_{\text{comp}}(s) = \frac{1 + G(s)G_I(s)}{G(s)G_T(s)} \) in equation 10, then the equation \( \frac{\omega(s)}{T_e(s)} = G_{\text{em}}(s) \) can be set up, so \( G_{\text{comp}}(s) \) was not contain \( G_{\text{em}}(s) \) anymore, it can be seen from the control block Figure 5 that the transfer function of \( T_e(s) \) to \( T_y(s) \) was one. The
inverse dynamic model is no longer needed. The control method in Figure 5 not only keeps \( G_{em}(s) \), but also avoids \( G_{\text{inv}}^{-1}(s) \), this control method meet the desired control objectives.

The speed closed loop controller contains complex compensation links, and a higher requirements of nonlinear controller makes the control method very complex. In order to avoid the complexity of this compensation algorithm in this paper, A speed adaptive predictive control module instead of compensation module in speed controller is presented. It not only simplifies the control flow, but also greatly simplified the control error caused by the inaccuracy of the control flow and the model parameters.

2.3. The control algorithm based on speed adaptive predictive control.
The goal of dynamic simulation is to establish a control strategy, make it possible to simulate the inertia much larger than that of the simulation system. In order to avoid the compensation module in the speed closed loop control algorithm, a predictive control module is used to replace the compensation in the simulation controller. The basic idea is applying the target torque which is supposed to be set in the simulated object to test bench simulation system, then set the real torque signal which fed by test bench simulation system to the mathematical model of the object which simulated, and get the speed \( \omega_{em} \). At the same time, the speed signal of the next moment can be obtained by using the estimated speed of the feedback link and the stage feedback.

The prediction method assumes that the control cycle \( \Delta t \) is small enough for the test system(ms level), it is assumed that the system is uniformly accelerated or uniformly decelerated, so equation 11 will be obtained.

\[
\alpha(k) = \frac{\omega(k+1) - \omega(k)}{\Delta t} \tag{11}
\]

The next moment speed in the simulation system can be estimated by the acceleration of the current system model and the current time.

\[
\omega(k + 1) = \omega(k) + \alpha(k) \Delta t \tag{12}
\]

The speed of the load motor is adjusted according to the feedback signal of the driving torque in the control period, it makes the actual speed of the next moment is consistent with the speed of the simulated object. The acceleration in this instruction cycle is the same as that of the simulated object, the accuracy of the simulation depends on the dynamic response time of the motor control system and the accuracy of speed prediction. The precision of dynamic load simulation of platform is affected by the estimation accuracy of speed, so the estimation accuracy of speed is an important part in the control system. In order to obtained the accuracy speed, a method of Cubic polynomial least squares fitting prediction is presented.

2.4. Cubic polynomial least squares fitting method for speed prediction.
When the speed of the simulation system is changed, the cubic polynomial least squares fitting method is used to estimate the speed of the next moment.

Suppose the cubic polynomial equation is:

\[
\omega(t) = c_0 + c_1 t + c_2 t^2 + c_3 t^3 \tag{13}
\]
If $c_0$, $c_1$, $c_2$, $c_3$ make the quadratic sum of $\omega_i$ and $\omega(t_i)$ to a minimum, that $\sum_{i=0}^{n} [\omega_i - \omega(t_i)]^2$ was minimum.

Suppose

$$H(c_0, c_1, c_2, c_3) = \sum_{i=0}^{n} [\omega_i - \omega(t_i)]^2$$

(14)

$$\frac{\partial H}{\partial c_k} = -2 \sum_{i=0}^{n} [\omega_i - \sum_{j=0}^{3} c_j t^j] t^k = 0 \quad k = 0, 1, 2, 3$$

(15)

The equation can be obtained:

$$\sum_{j=0}^{3} \left( \sum_{i=0}^{n} t_i^{j+k} \right) c_j = \sum_{i=1}^{n} (\omega_i \times t_i^{j+k}) \quad k = 0, 1, 2, 3$$

(16)

In the formula, $n$ was the number of data involved in the computation. $\omega_i$ is the measurement point in the time $t_i$. The solution of the equation is equal to a four order linear system equations. Gauss elimination method can be used to solve linear equations. Then put them into cubic polynomial, the estimated rotational speed can be obtained.

The system controller dynamically select appropriate estimation algorithm for speed estimation according to the Degree of speed change in the simulation system. The state of the simulated system acceleration is expressed by prediction coefficient $k_e$, when $k_e$ is relatively small, try to use linear fitting method, when $k_e$ is big, try to use method of quadratic curve fitting prediction. When $k_e$ is relatively large, try to use the cubic polynomial least squares fitting method. The control principle diagram of dynamic load simulation method based on speed adaptive predictive control is shown in Figure 6.

**Figure 6. Speed adaptive predictive control**

$T_{ref}$ is the input reference drive torque, $F_d(s)$ was the transfer function of the input signal from actual output signal of the drive motor. $F_T(s)$ is the transfer function of the input signal from actual output signal of the drive motor. $T_e(s)$ was electromagnetic torque of drive motor, $T_f(s)$ was compensating friction torque, $T_D(s)$ was actual torque of the drive motor measured by the sensor, $T_L(s)$ was
the actual torque measured by the sensor loaded motor, $T_F(s)$ was the disturbance torque, $J_m$ and $B_m$ were the inertia and the damping coefficient of test bench system, respectively, $J_m$ and $B_m$ were the inertia and damping coefficient of simulated object, respectively, $\omega(k + 1)$ was the speed setting signal in next moment.

2.5. **Vehicle driving resistance simulation.** The vehicle driving resistance can be divided into velocity resistance and inertia resistance. Air resistance and rolling resistance exist under any conditions when the vehicle driving and related to speed, which is called velocity resistance. Slope resistance and Acceleration resistance are called inertia resistance. Slope resistance determined by vehicle and slope and it will be a constant in certain slope. Acceleration resistance exists only under the condition of acceleration and deceleration. Acceleration resistance and deceleration resistance is the key step in the simulation of vehicle road driving conditions. There are three kinds of simulation methods for inertia simulation, including mechanical simulation, electronic-mechanical hybrid simulation and pure electric simulation. Mechanical simulation method simulate the whole vehicle inertia by mass flywheel, while the method is mature, but it need to make a lot of different inertia flywheels in order to meet the different vehicles in different conditions of inertia requirements. The electronic-mechanical hybrid simulation method uses the inertia flywheel and also makes use of the electromagnetic torque of the motor to compensate part of inertia torque. This method reduces the number of required inertia wheels and it solves the problem of inconvenient installation of mechanical inertia wheel in a certain extent. The electric simulation method is to remove the inertia flywheel part on the test platform, in addition to the moment of inertia of the platform itself, the simulation of the inertia resistance moment is completely compensated by the electromagnetic torque of the dynamometer. This method solves the problem of inconvenient installation of the mechanical inertia wheel and the limitation of the speed of the system, but it puts forward higher requirements for the control strategy of the simulation system. This paper presents a new control method of vehicle acceleration simulation based on speed adaptive predictive control. In the speed closed loop control method, a predictive module is added to replace the compensation module. It avoids the complex algorithm of closed loop speed compensation, and reduces the influence of inaccurate model parameters on the control precision of the simulation system.

The rolling resistance moment, air resistance moment and slope resistance moment can be obtained according to the vehicle longitudinal dynamic model and the real-time vehicle speed. The real time controller of simulation system send the throttle opening signal which is set by the tester and the gear-box half-shaft speed and torque which were measured in platform to vehicle dynamics model, vehicle dynamics model calculated the speed increment in current drive force according to drive statue and vehicle parameters. The speed of the next moment includes the current speed and increment. The simulation system controls the load motor speed according to the output shaft speed and the torque signal in the control cycle, makes the speed of the next time consistent with the target speed of the vehicle. Because of the small control period (ms level), this method can achieve the purpose of simulating vehicle acceleration.

The acceleration resistance can be obtained by the equation of vehicle driving resistance.

$$T_j = T_{en}i - (T_f + T_w + T_i) \tag{17}$$
A dynamic model is presented by taking Laplace transform with the equation 17.
\[
G_{em}(s) = \frac{\omega_{em}(s)}{T_e(s) - T_f(s) - T_i(s) - T_w(s)} = \frac{1}{J_{veh}(s)}
\] (18)
Where \( S \) is the Complex variable, \( \omega_{em}(s) \) is the speed of active wheels in real driving condition.

Equation 19 can be obtained by the principle of kinematics
\[
\begin{align*}
\frac{1}{2}J_B\omega^2_{ew} &= \frac{1}{2}m_{veh}\nu^2 \\
\nu &= \omega_{em}R
\end{align*}
\] (19)
The inertia of vehicle body is equivalent to the driving wheel by 19:
\[
J_B = m_{veh}R^2
\] (20)

In order to obtained the next moment speed, add the equation 17 and 20 into equation 18
\[
\omega_{em}(n+1) - \omega_{em}(n) = \frac{T_{en}(n)i - m_{veh}gf \cos \gamma(n)R - m_{veh}g \sin \gamma(n)R}{J_Ei^2 + m_{veh}R^2 + J_W} \Delta t
\]
\[- \frac{1}{2}C_D A \rho \alpha (3.6 \cdot R \cdot \omega_{em}(n))^2 R
\]
\[
\frac{J_Ei^2 + m_{veh}R^2 + J_W}
\] (21)
where \( n = 1, 2, 3, \ldots, k \) represents the cycle times, \( \Delta t \) is system sampling time, \( \omega_{ref} \) is input reference speed, \( \omega_L \) is speed of load motor, \( T_D \) is torque of drive unit, \( \omega_D \) is the speed of drive unit, \( T_L \) is torque of load unit, \( \omega_{em} \) is the speed of, \( i \) is the ratio of gear-box; \( J_E \) is inertia of engine, \( J_w \) is inertia of vehicle wheels.

The formula 21 can be simplified by ignoring air resistant.
\[
\omega_{em}(n+1) - \omega_{em}(n) = \frac{T_{en}(n)i - m_{veh}gf \cos \gamma(n)R - m_{veh}g \sin \gamma(n)R}{J_Ei^2 + m_{veh}R^2 + J_W} \Delta t
\] (22)

It can be seen from equation 21 and 22 that the simulation system speed of the next moment can be calculated by the previous moment and the current state.
Table 1. The technical data of drive motor

| Power (Kw) | Frequency (Hz) | Torque (N·m) | Speed (r/min) | Moment of inertia (kg·m²) |
|------------|----------------|--------------|---------------|--------------------------|
| 235.6      | 250            | 360          | 5000          | 0.042                    |

parameters. The wheel speed $\omega_{em}(s)$ can be simulated by controlling the speed $\omega(s)$ of load unit in platform system, and the purpose of dynamic simulation of the actual operating conditions of automotive transmission system is achieved. The control model of vehicle load simulation method based on speed adaptive predictive control was shown in Figure 8. The diagram only showed the control diagram of one load motor in the reason of the same control method between two load motor. $G_{PID}$ is engine throttle regulator, $G_{I}$ is PI control regulator.

![Figure 8. Control model of speed adaptive predictive control](image)

The maximum speed acceleration of the simulation system is determined by the power characteristics of the motor and the dynamic response of the platform. The maximum output torque of the motor is constant when the simulation speed is less than or equal to the rated motor speed. When the speed is greater than the rated speed, according to the constant power characteristics of the motor, the maximum output torque is inversely proportional to the speed. According to the power characteristics between drive motor in Figure 9 and load motor in figure 10, the ability of acceleration simulation on the test bench system is shown in Figure 11.

The maximum torque of the motor can be obtained by the following formula

$$ T_{E_{\text{max}}} = \begin{cases} 9550 \frac{P}{n} & \text{if } n \leq n_0 \\ 9550 \frac{P}{n_0} & \text{if } n > n_0 \end{cases} \quad (23) $$

Where $n_0$ is motor rated speed, $n$ is actual speed of motor.
Table 2. The technical data of load motor

| Power (Kw) | Frequency (Hz) | Torque (N·m) | Speed (r/min) | Moment of Inertia (kg·m²) |
|------------|----------------|--------------|---------------|--------------------------|
| 310        | 27.2           | 3701         | 800           | 6.3                      |

Figure 9. The characteristic curves of drive motor and engine

Figure 10. The characteristic curves of load motor
If the maximum acceleration of the simulation system is \( \alpha \), the inertia resistance moment in the process of acceleration should be supplied by drive motor can be calculated by equation 24.

\[
\begin{align*}
T_{em} &= J_{em} \alpha = \frac{J(9550P_n - M_0)}{J_{em} - J_0}, \quad n \leq n_0 \\
T_{em} &= J_{em} \alpha = \frac{J(9550P_n - M_0)}{J_{em} - J_0}, \quad n > n_0
\end{align*}
\]

(24)

Where, \( T_{em} \) is the inertia resistance moment in the process of acceleration should be supplied by drive motor, \( J_{em} \) is the simulated vehicle equivalent inertia, \( \alpha \) is the wheel angular acceleration, \( J_0 \) is the equivalent inertia of a certain gear.

The linear acceleration corresponding to the simulated inertia of the system can be obtained

\[
a_v = \frac{T_{em} \cdot R}{J_{em} \cdot 9.81}
\]

(25)

Here, \( a_v \) is the linear acceleration of vehicle.

Figure 11. Simulation range of electrical inertia on the platform system

All the torque provided by the platform motor is used for the inertia simulation when ignored the other resistance moment. According to the parameters of drive motor and load motor which shown in table 1 and table 2, the inertia simulation range of test bench for the 3th gear of the gear-box can be obtained in Figure 11. The range covered under the blue curve in the Figure is the ability of inertia simulation on the test bench. Because the load unit motor power characteristic is much larger than the drive unit motor, the range of the electric inertia simulation depends on the drive unit motor. The black line represent the acceleration curve when \( a_v = 1g \), \( a_v = 0.8g \), \( a_v = 0.5g \) and the intersected point of curves are “92.5”, “112.1”, “170.5”, it means the maximum inertia can be simulated on the test bench. It can be seen that the system can simulate the acceleration of the small car and micro vehicle with \( a_v = 1g \), it could only simulating the acceleration at \( a_v = 0.5g \) for the bigger inertia of medium-size vehicle. If air resistance, rolling resistance and slope resistance are
taken into account, the torque of simulation system should subtract the resistance torque under the current speed, the residual torque can be calculated according to the above method. The acceleration simulation on the test bench is shown in the Figure 12.

![Figure 12. Acceleration inertia simulation curve](image)

3. Experiments and results.

3.1. **Test equipment description.** Vehicle driveline test bench is shown schematically in Figure 13, a standard longitudinal Front Wheel Drive(FWD) Mechanical Transmission(MT) and clutch link the drive motor to the load motor. The combustion engine is replaced by a drive motor, which replicates the torque and speed signature of an actual engine. The road load resistance of the vehicle on a real test road is accurately simulated on test equipment by corrected road load forces. The high frequency transient performance required from the electric dynamometer.

3.2. **Vehicle start characteristic simulation.** Vehicle starting is a process that need multi system coordination, which related to the engine, clutch and driveline systems. The clutch is still in a sliding state when the vehicle at the initial stage of starting. The clutch is in the synchronous state when the clutch driven plate speed is equal to the engine speed and the torque provided by the clutch itself is greater than or equal to the desired torque.

Because the vehicle speed is very low in the starting process, the air resistance can be ignored, so the starting process of the vehicle on the horizontal road mainly includes rolling resistance and acceleration resistance. As shown in Figure 14, the drive motor simulate the engine idle state under the torque control model, the electric cylinder controls the clutch pedal to simulate the driver’s actual driving state. As shown in Figure 15, the control process was divided into $t_1$ and $t_2$ stages. In the early stage of clutch engagement at time $t_1$, the clutch pedal is in an idle state, the clutch master and slave disc has not yet begun to combine, and the shaft torque has not changed. The clutch starts to combine and the half shaft torque begins to increase at time $t_2$, when the output torque of the clutch is greater than
the resistance torque of the vehicle, the vehicle wheels start to rotate and half shaft torque begin to fluctuate. At the same time, the system calculates the acceleration according to the output torque of the clutch and the driving torque and the moment inertia of the vehicle.
3.3. Dynamic shift simulation test. As shown in Figure 16, Figure 17, Figure 18 and Figure 19 the dynamic upshift and downshift simulations are carried out on the driveline test bench by using the speed adaptive predictive control method of vehicle road simulation. In the experiment, the input speed of the gear box and the half shaft torque decreased rapidly when the clutch was separated, there is a certain oscillation in the initial stage of torque loading after the shift is completed, but it soon reached a steady state. In order to achieve higher control requirements, the more accurate parameters such as inertia and damping of the system are required to be obtained, and the control algorithm is further optimized in order to obtain faster response speed and higher control precision.
4. Conclusions. There is a very high requirements for vehicle dynamic emulation on the vehicle driveline test bench with electric drive dynamometer on mechanical properties and electrical properties, and the control strategy is more feasible and practical, it is a complex engineering problem. On the basis of analyzing and comparing the advantages and disadvantages of the inverse dynamics model and the forward model based on speed closed loop control method, in view of the high order, nonlinear and multi variable characteristics of test bench system, a load simulation method based on speed adaptive predictive control is proposed. On the test bench, the vehicle start and dynamic shift process are simulated according to the speed adaptive predictive control algorithm. It avoids the complex algorithm of closed loop speed compensation, and reduces the influence of inaccurate model parameters on the control precision of the simulation system.
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E-mail address: liwenli999@163.com
E-mail address: willie005@163.com
E-mail address: zhangxk@tsingshan.cn
E-mail address: peng.yi@cqctl.com.cn