Research on vehicle performance test based on DirectInput data acquisition

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Abstract: Aiming at the high cost of high-performance driving simulator at present, a low-cost vehicle simulation system with universal applicability was developed based on Microsoft Visual C++6.0 software platform. A data acquisition interface program based on DirectInput components and the real-time data acquisition program of driving operation was designed. Based on the dynamic model of vehicle simulation, a data output conversion model was established for the driver operation input collected in real time. For real-time vehicle dynamics simulation model, the real-time simulation control process was realized, the simulation description and decoding quality of the ground, hanging, no suspension quality coordinate system and the real-time simulation for buses with nine degrees of freedom dynamic model were established. Finally, the real-time and reliability of the simulation system were verified through the simulation test. The experiment results show that the average response time of the simulation system is 11.465ms. Emergency braking time is 4.40s with an error of 4.762%. The braking distance is 70.145m and the error is 5.959%.

1. Introduction
The vehicle simulation technology based on the real-time collection and calculation of driver's operation data is mainly applied to the research and development of various types of vehicle driving simulators. In 2010, Daimler Benz Corp developed a 7 degree of freedom driving simulator driven by electric power, which was mainly used in the field of vehicle safety and automotive product development [1]. In 2015, Germany implemented the structure of wheel self-propelled simulator proposed by BMW Company in 2002. The simulator can simulate the pitching, vertical and lateral movement of vehicles. And the driving wheel can complete the advance and steering function to achieve the platform's yaw, lateral and vertical movement [2]. Compared with foreign countries, in 70s of twentieth Century, China gradually began to independently develop low-level driving simulator. In 1996, the first 6 degree of freedom driving simulator in China was successfully developed by Jilin University [3], which was mainly used in the pilot hardware in the loop test. Moreover, the dynamic model of the driving simulator was further updated in 2010, the degree of freedom of the driving simulator was expanded, and the motion accuracy and ability were further enhanced [4]. In 2009, the Traffic Management Science Research Institute of the Ministry of public security developed an advanced driving simulator with five channels of vision. The edge fusion and geometric correction technology have greatly improved the driving simulator's immersion [5][6][7].

Aiming at the problem of high cost of high-performance driving simulator at present stage[8], a data output transformation model is built based on DirectInput technology to collect the driver's operation
input amount in real time. At the same time, a vehicle dynamic model is established, which accords with the reality of the simulation object, and also it meets the real-time and accuracy requirements. Furthermore, According to the operation process of the driver under different conditions, the control process of vehicle dynamic real-time simulation is designed. The parallel vehicle simulation system based on DirectInput data acquisition is developed on the Microsoft Visual C++6.0 software platform and the reliability of the system is analyzed according to the simulation test.

2. Virtual performance test model for vehicle performance

Figure 1 is the technology roadmap of this research. Firstly, the data acquisition technology of driving simulator based on DirectInput component is studied, and the real-time data acquisition of driving simulator is realized, and the data transformation model of driving operation is established, and the input parameters of the dynamic model, such as the front wheel angle, engine torque, braking force moment and transmission ratio, are obtained. Then a nine degree of freedom bus dynamics model is established to calculate vehicle motion state parameters and vehicle attitude parameters in the simulation process. Finally, parallel vehicle simulation system is developed, and the real-time performance and reliability of the simulation system are verified through simulation tests.

2.1. Operational data output transformation model

(1) Torque output transformation model of Transmission system

The parallel vehicle driving simulation system simplifies the transmission shaft and the driving shaft into purely rigid objects, that is, their only rigid transmission torque \( T \). Therefore, this part mainly studies the output conversion model of clutch, transmission and main reducer. Figure 2 shows the mathematical model of clutch transfer torque and clutch pedal stroke.
Figure 2. Mathematical model of clutch transmission torque and clutch pedal travel

Where, the CE segment is the stage of resistance, $T_c \leq T_f$, $w_c = 0$.

$$I_e \frac{dw_c}{dt} = T_e - T_c$$

(1)

$T_c$ is friction torque for clutch transmission, $T_f$ is vehicle resistance equivalent to the input shaft of the transmission, $w_c$ is the clutch drive plate angular velocity (equal to engine crankshaft), $w_e$ is the angular velocity of clutch driven plate, $I_e$ is the moment of inertia of the engine flywheel.

The BC segment is a vehicle acceleration phase.

$$I_e \frac{dw_c}{dt} = T_e - T_c$$

(2)

$$I_v \frac{dw_v}{dt} = T_v - T_f$$

(3)

$I_e$ is equivalent to the translational motion and moment of inertia of the vehicle equivalent to the input shaft of the transmission.

Section AB: $T_c < T_{c_{\text{max}}}$, $I_e = T_c$, $w_c = w_e$

$$(I_v + I_e) \frac{dw_c}{dt} = T_e - T_f$$

(4)

In the model, $T_c$ is regarded as static friction torque $T_{cs}$, The specific formula is as follows:

$$T_c = \text{sign}(w_e - w_c)T_{cs} = \text{sign}(w_e - w_c)\mu_s F_{n} R_c = \text{sign}(w_e - w_c)\mu_s K_{n} L_c R_c$$

(5)

Where, $\mu_s$ is the static friction coefficient of the clutch; $F_n$ is the positive pressure of the diaphragm spring on the pressure plate, $N$, $K_n$ is the spring stiffness, $\frac{N}{m}$, $L_c$ is the effective stroke of the clutch at this moment, $mm$, $R_c$ is the equivalent radius of the clutch;

2.2 vehicle real-time simulation dynamic model

(1) Kinematics equation of passenger car body

The bus quality is divided into two parts, namely the suspended mass and the non hanging mass. Based on this, three coordinate systems are set up, as shown in Figure 3.
Fig 3. Establishment of relative coordinate system

By using the ground coordinate system OFXFYFZF, the non-hanging mass coordinate system \( O_dX_dY_dZ_d \), the hanging mass coordinate system \( O_bX_bY_bZ_b \). The relative displacement of the non-hanging mass coordinate system \( O_dX_dY_dZ_d \) and the ground coordinate system \( O_FX_FY_FZF \) is expressed as \( r_d \). It is shown that the relative displacement of the hanging mass coordinate system \( O_bX_bY_bZ_b \) and the non-hanging mass coordinate system \( O_dX_dY_dZ_d \) is expressed as \( r_{bd} \). It is indicated that the angular velocity of the non hanging mass coordinate system \( O_dX_dY_dZ_d \) relative to the ground coordinate system OFXFYFZF is expressed as \( \omega_{O_d} \). There are:

\[
\omega_{O_d} = \begin{bmatrix} 0 & 0 & \psi \end{bmatrix}^T \tag{6}
\]

\[
\dot{r}_d = \begin{bmatrix} v_x & v_y & 0 \end{bmatrix}^T \tag{7}
\]

\[
r_{bd} = \begin{bmatrix} 0 & -h \sin \phi & 0 \end{bmatrix}^T \tag{8}
\]

Where, \( v_x \) represents the longitudinal speed of the vehicle, \( v_y \) represents lateral velocity of vehicle, \( h \) represents the height of the vehicle's center of gravity from the ground, \( \phi \) represents the side slope of a vehicle.

It can be further obtained:

\[
\ddot{r}_d = \frac{d\dot{r}_d}{dt} = \frac{d\dot{r}_d}{dt} + \omega_{O_d} \times \dot{r}_d = \begin{bmatrix} \ddot{v}_x - v_y \psi & \ddot{v}_y + v_x \psi & 0 \end{bmatrix}^T \tag{9}
\]

\[
\ddot{r}_{bd} = \frac{d^2}{dt^2} (r_{bd}) + \dot{\omega}_{O_d} \times \dot{r}_{bd} + \omega_{O_d} \times (\omega_{O_d} \times \dot{r}_{bd}) + 2 \omega_{O_d} \times \frac{d}{dt} (r_{bd})
\]

\[
= \begin{bmatrix} h(\sin \phi \psi + 2 \cos \phi \psi) \\ h(\sin \phi \psi^2 - \cos \phi \psi + \sin \phi \psi^2) \\ 0 \end{bmatrix} \tag{10}
\]

Where, \( \ddot{r}_d \) is the acceleration of the vehicle's center of gravity, \( \dot{\psi} \) is the vehicle yaw rate.

Therefore:

\[
\ddot{r}_b = \ddot{r}_d + \ddot{r}_{bd} = \begin{bmatrix} \ddot{v}_x - v_y \psi + h(\sin \phi \psi + 2 \sin \phi \psi) \\ \ddot{v}_y + v_x \psi + h(\sin \phi \psi^2 - \cos \phi \psi + \sin \phi \psi^2) \\ 0 \end{bmatrix} \tag{11}
\]
Where, $\ddot{r}_b$ is the lateral and longitudinal acceleration of the vehicle centroid relative to the ground coordinate system under the action of external force.

The angular acceleration of the hanging mass relative to the ground coordinate system $O_{XF}Y_{ZF}$ is further obtained.

$$\alpha = \begin{bmatrix}
I_{xx} \ddot{\theta} - \left(I_{yy} - I_{zz}\right) \sin \phi \cos \varphi \ddot{\varphi}^2 \\
I_{xx} \varphi \ddot{\psi} + \left(I_{yy} - I_{zz}\right) \left(\cos(2\varphi) \varphi \ddot{\varphi} + \sin \phi \cos \varphi \dddot{\varphi}\right) \\
\left(I_{yy} \sin^2 \varphi + I_{zz} \cos^2 \varphi\right) \ddot{\psi} + 2 \left(I_{yy} - I_{zz}\right) \sin \phi \cos \varphi \dddot{\varphi}
\end{bmatrix}$$

(12)

(2) Mechanics analysis of bus chassis

In Figure 4, if the effect of wind on the vehicle is not considered, the external force on the longitudinal and lateral direction of the vehicle is mainly determined by the tire, the resultant force in the longitudinal direction $F_x$ and the resultant force in the transverse direction $F_y$ is shown as follows:

$$F_x = (F_{x1} + F_{x2}) \cos \delta - (F_{y1} + F_{y2}) \sin \delta + F_{x3} + F_{x4}$$  \hspace{1cm} (13)$$

$$F_y = (F_{x1} + F_{x2}) \sin \delta + (F_{y1} + F_{y2}) \cos \delta + F_{y3} + F_{y4}$$ \hspace{1cm} (14)

Where, $F_{x1}, F_{y1}\ (i = 1, 2, 3, 4)$ represent the longitudinal and transverse forces of a tire, in which tire 1 and tire 2 are steering wheels, and mainly related to tire friction coefficient and vertical force, $\delta$ is the front wheel angle of the vehicle.

Figure 4. Stress analysis of chassis

Without considering the ideal conditions of wind load, the external force moment of the vehicle in the roll direction and the yaw direction are all derived from the tires. In this paper, the vehicle quality is divided into suspended and non hanging mass. As the non hanging mass is smaller than the total mass ratio of the vehicle, assuming that the centroid of the suspension mass is the centroid of the vehicle, the elastic moment produced by the suspension spring must be considered when considering the suspension force $M_s$. Damping torque generated by shock absorbers $M_d$ is the role of hanging mass in the roll direction.

$$M_x = F_xh \cos \phi + mgh \sin \phi - M_s - M_d$$ \hspace{1cm} (15)$$

$$M_y = \left(\left(F_{x2} \cos \delta - F_{y2} \sin \delta + F_{x4}\right) - \left(F_{x1} \cos \delta - F_{y1} \sin \delta + F_{x3}\right)\right) \left(B/2\right)$$

$$+ \left(F_{x1} + F_{x2}\right) \sin \delta + \left(F_{y1} + F_{y2}\right) \cos \delta) l_y - \left(F_{y3} + F_{y4}\right) l_y - F_x h \sin \phi$$ \hspace{1cm} (16)

(3) Dynamics equation of passenger car with four degree of freedom bus body

According to equation 8, equation 12, equation 13 and equation 14, equation 15, equation 16, the equation of dynamic equation of passenger car with four degree of freedom bus body can be deduced as follows:
Vertical:
\[
m(\dot{v}_x + \dot{v}_y \psi + h(\sin \varphi \psi + 2 \cos \varphi \psi)) = (F_{x_1} + F_{x_2}) \cos \delta - (F_{y_1} + F_{y_2}) \sin \delta + F_{x_3} + F_{x_4} \tag{17}
\]

Horizontal:
\[
m(\dot{v}_y + \dot{v}_y \psi + h(\sin \varphi \psi^2 - \cos \varphi \psi + \sin \varphi \psi^2)) = (F_{x_1} + F_{x_2}) \sin \delta + (F_{y_1} + F_{y_2}) \cos \delta + F_{y_3} + F_{y_4} \tag{18}
\]

Rollover:
\[
I_{x_1} \ddot{\varphi} - (I_{y_1} - I_{z_2}) \sin \varphi \cos \varphi \psi^2 = F_{h} h \cos \varphi \cos \varphi + mgh \sin \varphi - M_s - M_d \\
= ((F_{x_1} + F_{x_2}) \sin \delta + (F_{y_1} + F_{y_2}) \cos \delta + F_{x_3} + F_{x_4}) h \cos \varphi + mgh \sin \varphi - M_s - M_d \tag{19}
\]

Yaw:
\[
(I_{yy} \sin^2 \varphi + I_{zz} \cos^2 \varphi) \dot{\psi} + 2(I_{yy} - I_{zz}) \sin \varphi \cos \varphi \psi \dot{\psi} = ((F_{x_1} \cos \delta - F_{y_2} \sin \delta + F_{x_4}) - (F_{x_1} \cos \delta - F_{y_1} \sin \delta + F_{x_3})) (T/2) + ((F_{x_1} + F_{x_2}) \sin \delta + (F_{y_1} + F_{y_2}) \cos \delta) l_{f} - (F_{y_3} + F_{y_4}) l_{r} - F_{x_4} h \sin \varphi \tag{20}
\]

3. Example analysis
In order to verify the accuracy of the simulation mathematical model established in this paper, the simulation of the vehicle under emergency braking is carried out, and the reliability of the simulation system is verified by comparing the actual data [10]. Figure 5 shows the Trucksim simulation curves of real time speed, braking distance with braking time obtained by emergency braking of a bus on dry road at 80km/h, 100km/h and 120km/h respectively. When the initial braking speed is 100km/h, the total braking time is 4.2s when the braking speed is reduced to 0 km/h, and the braking distance is 66.2m. The basic parameters of the simulation vehicle set up by the simulation system are set up according to the references. Therefore, the reliability of the simulation system can be verified by comparing the simulation data between them.

![Fig. 5 Relationship between emergency braking speed, emergency braking distance and time](image-url)
The simulation vehicle starts to accelerate gradually after it starts, and the speed gradually rises to 100km/h through shifting gears. After that, the speed of the vehicle is stabilized at 100 km/h by controlling the accelerator pedal, then the brake pedal is pressed to reduce the speed to 0km/h, and the emergency braking is carried out at 166.80s time with a speed of 99.696 km/h. At 171.20s, the speed dropped to 0 km/h. Compared with the actual braking time, the braking time is 4.40s, the error is 4.762%, the braking distance is 70.145m, and the error is 5.959%.

4. Conclusion
(1) The DirectInput module is used to design the data acquisition interface program for driving simulator, and a data output conversion model is established for the real-time driver's operation data, including the torque output transformation model of the transmission system based on the research of the dynamic model of the simulation vehicle.

(2) The vehicle dynamics model for real-time simulation is established. The ground, suspended mass and non-hanging mass coordinate system suitable for simulation description and calculation are established. On the basis of simplifying the dynamics model of bus chassis, a real-time dynamic simulation model of passenger car with nine degrees of freedom is established, including the bus body dynamics model.

(3) The simulation data obtained from TruckSim simulation are compared with the experimental data of 100km/h-0km/h emergency braking. The parallel vehicle simulation system developed under the same conditions has an emergency braking time of 4.40s and the error is 4.762%, the braking distance is 70.145m and the error is 5.959%, which verifies the reliability of the simulation system.

(4) The vehicle model established in this paper is a bus dynamics model which cannot be well applied to other models. Further improvements can be made to improve the universality of the vehicle by developing the dynamic models of different models.

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