Modeling and Simulation of a PC-Based Driving Simulator*

Rongwen HUANG** and Wangshuan LEE**

This paper describes the simulation and control of a PC-based driving simulator that is constructed by five hydraulic cylinders and can perform five degree-of-freedom (DOF) motions. The mathematical equations of vehicle dynamics are derived from the 2-DOF bicycle model and incorporated with the tire, steering and suspension subsystems. These equations of motion are then programmed by MATLAB, transferred into C++ code in MIDEVA environment, and further developed into motion platform control program by C++ Builder. In addition, virtual reality environment is projected onto a 72" screen to present the street scene and simulate the sixth DOF motion of a vehicle, and can be employed either in off-line analysis and simulations or on-line simulations. In the future, a road surface model and other more complicated vehicle dynamics models will be incorporated, and a pressure vest will be installed to stimulate the driver to feel the longitudinal translation.

Key Words: Driving Simulator, Vehicle Dynamics, Virtual Reality

1. Introduction

The advanced development in computer technology has made the application of PCs in several engineering fields, such as computer graphics and virtual reality (VR) technology, come true. Consequently, various types of simulators have been developed. Among them, the flight simulator is the most successful development, followed by the driving simulator for road vehicles.

The purpose of this study is aimed at the simulation and control of a PC-based road-vehicle driving simulator. The governing equations of a vehicle incorporated with the tire, steering and suspension systems are described. These vehicle dynamics equations are then programmed by MATLAB, transferred into C++ code in MIDEVA environment, and further developed into motion platform control programs in Borland C++ Builder 5.

To perform the simulation functions, the authors utilize interface technology to acquire the voltages of an angular potentiometer (steering wheel angle) and two linear motion potentiometers (throttle valve and brake positions). The signals are sent to the vehicle dynamics analysis program along with the longitudinal, lateral and vertical velocities/displacements of the vehicle's mass center. The roll, pitch and yaw rates/angles are also carried out, converted into analog signals, sent to cylinders to actuate the motion platform and perform 5-DOF motions. Meanwhile, driver's operation signals acquired are also sent to Superscape VR program and the control codes are written into the objects to process these signals and synchronize the VR environment with the real world. The virtual reality environment is then projected onto a 72" screen to present the street scene and to simulate the sixth DOF motion of a vehicle.

2. Vehicle Dynamics Analysis

The vehicle subsystems that affect vehicle dynamics include the tires, the steering system, the suspension system, and the acceleration/deceleration systems. Other external factors, such as the driving

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** Department of Mechanical Engineering, National Changhua University of Education, 2, Shita Rd., Changhua City, Changhua 500 Taiwan, R.O.C.
E-mail: rongwen@cc.ncue.edu.tw

inputs of the driver, the grade of the road, vehicle load and its distribution, wind speed and its direction, as well as its interaction with the vehicle body and under-hood components will also have an effect on vehicle dynamics as a whole. As a result, vehicle dynamics is so complex that analyzing the influences of all vehicle subsystems and external factors simultaneously is virtually impossible. Thus, in order to look closely at the details of each influencing factor, we can analyze their characteristics one at a time, and then integrate several of them to study their interactions.

It is convenient to treat the vehicle mass (body) as a rigid body with six degrees of freedom (an example of a fixed bodied coordinate system is shown in Fig. 1^{(1),(2)}). The three linear movements of the vehicle body to the ground are longitudinal, lateral, and vertical velocities along the x , y and z directions and are defined as u , v and w , respectively. In addition, the roll, pitch, and yaw rates (denoted as p , q , and r) are the angular velocities of the body about x , y and z axes⁽¹⁾⁻⁽³⁾. With some specific constrains for different interests, the degrees of freedom of the vehicle body can be further reduced. For instance, a 2-DOF model with two tires (Fig. 2) can be used as a simple vehicle model in some analysis cases.

In Fig. 2, if the tire cornering force F_y is modeled as in proportion to the slip angle α (assumed constant all the time), i.e. $F_y = C\alpha$, it is called a linear non-dynamic tire model. In 2-DOF bicycle model we have one front tire and one rear tire, each with its own F_y , C and α . Therefore, in linear non-dynamic tire model, the tire lateral forces are^{(1),(4),(5)}

$$F_{yf} = C_f \alpha_f \quad (1)$$

Fig. 1 SAE vehicle axis system⁽¹⁾

Fig. 2 Simple vehicle model represented as a bicycle

$$F_{yr} = C_r \alpha_r \quad (2)$$

In the equations above, C_f and C_r are the cornering stiffness of the front and rear tires (N/rad), and α_f and α_r are the slip angles of the front and rear tires (rad). Let δ be the input steering angle on the tire (rad). From kinetic analysis, the tire slip angles can be expressed as:

$$\alpha_f = \delta - \frac{v + ar}{u} \quad (3)$$

$$\alpha_r = \frac{v - ar}{u} \quad (4)$$

As shown in Fig. 3, the vehicle body while cornering tends to roll out of turn due to non-rigidity of the suspension system. Let the suspension system be represented by two springs at each axle, then the roll angle ϕ is a function of the lateral separation between the springs, spring rate, lateral acceleration a_y ($=u^2/gR$), vehicle weight, as well as the position of the roll center.

Each suspension has a suspension roll center, defined as the point in the transverse vertical plane through the wheel centers at which lateral forces may be applied to the sprung mass without producing suspension roll^{(1),(6)-(9)}. It's derived from the fact that all suspensions possess a roll axis, which is the instantaneous axis about which the unsprung mass rotates

Fig. 3 Rolling of a simple vehicle in cornering⁽¹⁾

with respect to the sprung mass when a pure couple is applied to the unsprung mass. The roll center is the intersection of the suspension roll axis with the vertical plane through the centers of the two wheels, as Fig. 4 shows. The vehicle roll axis is the connection of the front and rear suspension roll centers⁽¹⁾.

For the vehicle shown in Figs. 3 and 4, assume bounce w and its time rate w' , pitch q and its time rate q' be zero. According to kinematics, the acceleration of any point, which can be chosen as the vehicle center of gravity, in the vehicle body is^{(10),(11)}:

$$v_y' = (v - pz + rx)' = v' + p'z + pz' + r'x + rx' \quad (5)$$

where v is the vehicle lateral velocity; p and r , vehicle roll and yaw rate; x and z , the x and z direction distances of the point to the roll axis. Since $w = 0$, we have $z = \text{constant}$ and the equations of motion of a vehicle with three rigid body DOF (v , p , and r) can be expressed as^{(12)–(19)}

$$\sum F_y = 2(F_{yf} + F_{yr}) = m(v' + ur) + m_s h p' \quad (6)$$

$$\sum M_z = 2aF_{yf} - 2bF_{yr} = I_z r' - I_{xz} p' \quad (7)$$

$$\sum M_x = I_x p' - I_{xz} r' + m_s h (v' + ur) = m_s g h \phi - \left[\frac{\partial L}{\partial \phi} \right]_f + \left[\frac{\partial L}{\partial \phi} \right]_r \phi - \left[\frac{\partial L}{\partial p} \right]_f + \left[\frac{\partial L}{\partial p} \right]_r p \quad (8)$$

where

- u : vehicle's longitudinal (forward) velocity (m/s),
- v : vehicle's lateral (side) velocity (m/s),
- p : vehicle's roll rate (rad/s),
- r : vehicle's yaw rate (rad/s),
- ϕ : vehicle's roll angle (rad),
- m : total mass of the vehicle (kg),
- m_s : sprung mass of the vehicle (kg),
- I_z : vehicle's mass moment of inertia about z -axis (kg-m-s²),
- I_{xz} : vehicle's product moment of inertia about xz plane (kg-m-s²),
- I_x : vehicle's mass moment of inertia about x -

Fig. 4 Force analysis for roll⁽¹⁾

axis (kg-m-s²),

h : height of the vehicle's center of gravity from ground (m),

a, b : distances between vehicle's center of gravity and front and rear wheels, respectively (m),

g : gravity constant = 9.81 (m/s²),

F_{yf}, F_{yr} : lateral forces generated at front and rear tires (N),

$\partial L / \partial \phi|_f, \partial L / \partial \phi|_r$: roll stiffness of the vehicle's front and rear suspension systems (N-m/rad),

L : vehicle's roll moment about x -axis (N-m), and

$\partial L / \partial p|_f, \partial L / \partial p|_r$: roll damping of the vehicle's front and rear suspension systems (N-m-s/rad).

Then, the lateral forces F_{yf} and F_{yr} generated on tires can be further expressed as:

$$F_{yf} = C_f \alpha_f + \frac{\partial F_y}{\partial \phi} \frac{\partial \phi}{\partial \phi} |_f \phi = C_f \left(\delta - \frac{v + ar}{u} + \varepsilon_f \phi \right) + \frac{\partial F_y}{\partial \phi} \frac{\partial \phi}{\partial \phi} |_f \phi \quad (9)$$

$$F_{yr} = C_r \alpha_r + \frac{\partial F_y}{\partial \phi} \frac{\partial \phi}{\partial \phi} |_r \phi = C_r \left(-\frac{v - br}{u} + \varepsilon_r \phi \right) + \frac{\partial F_y}{\partial \phi} \frac{\partial \phi}{\partial \phi} |_r \phi \quad (10)$$

where

$\varepsilon_f, \varepsilon_r$: roll steer coefficients of front and rear axles = radian of tire steering angle/radian of roll (≈ 0 , neglected),

$\partial F_y / \partial \phi|_{f,r}$: lateral forces of front and rear tires due to camber (N/rad),

ϕ : camber angle generated on the vehicle tire due to roll effect (rad), and

$\partial \phi / \partial \phi|_{f,r}$: tire generated camber angle/roll angle of front and rear axles (rad/rad).

Referring to Fig. 5 (shown in a bicycle model), the additional DOF in longitudinal direction can be modeled as⁽¹²⁾⁽¹³⁾⁽²⁰⁾:

$$\sum F_x = m_e (u' - vr) \quad (11)$$

where

m_e : vehicle mass = m_{vehicle}^* ratio accounting for the rotational components of the drive train (kg),

u' : time rate of vehicle's forward speed u (m/s²),

$\sum F_x$: the sum of longitudinal forces, including the x -component of tire lateral force F_y , braking

Fig. 5 Force components in x and y directions when cornering

force F_b , traction force F_t from both front tires and rear tires, aerodynamic drag force F_a , and tire rolling resistance force F_r , i.e.,

$$\sum F_x = -2F_y \sin \delta - F_a - F_{tf} - F_{tr} - F_{bf} - F_{br} - F_{rf} - F_{rr} \quad (12)$$

Moreover, if the steering system is modeled as a second order dynamic system as shown in Fig. 6, its governing equation can be written as^{(7),(8),(21),(22)}:

$$I_{tz}\delta'' + C_{st}\delta' + K_{st}\left(\delta - \frac{\delta_{st}}{n}\right) = -\sum (\text{restoring torque}) \quad (13)$$

where

I_{tz} : moment of inertia of the tire about z axis ($\text{kg}\cdot\text{m}^2$),

C_{st} and K_{st} : damping and stiffness of the steering system ($\text{N}\cdot\text{m}\cdot\text{s}/\text{rad}$ and $\text{N}\cdot\text{m}/\text{rad}$),

δ_{st} : the steering wheel input angle (rad), and

n : the steering system gearbox ratio.

The restoring torque is the sum of cornering force times both the caster trail L_1 (due to the geometry of the steering system) and the pneumatic trail (tire characteristics).

Therefore, there are four geometric DOFs for the current model (v , r , p in Eqs. (6) – (8) and u in Eq.

(a)

(b)

Fig. 6 Steering system and restoring torque

(11)), while δ in Eq.(13) is an additional integration variable. Due to the limitation of space, linear dynamic tire model (which will add one more differential equation for each tire) and non-linear tire model (which needs tedious iteration to obtain tire forces)^{(23)–(26)} are not detailed here.

3. Construction, Simulation and Control of Driving Simulator

Due to the increased commercial demands and research interests, various types of simulators have been developed^{(27)–(35)}. To represent a vehicle in the driving simulator, a motion platform that can produce 6-DOF motions is required. Currently, Stewart platform is the most popular six-axis motion platform^{(36)–(39)}. In reality, the vehicle longitudinal motion can hardly be simulated. Therefore, in this study, the authors utilize only five cylinders to construct a motion platform that can produce 5-DOF motions (Fig. 7). Incorporated with virtual reality technology to analogy the longitudinal translation^{(40),(41)}, the designed driving simulator can simulate the 6-DOF motions of a vehicle and can be used specifically in road vehicle dynamics analysis and simulation.

The constructed driving simulator consists of a 5-axis motion platform mentioned above, a vehicle dynamics analysis and motion control computer, a virtual reality computer, three potentiometers and three data acquisition cards. The graphical representation is shown in Fig. 8. Also, for the safety of the driver (operator), a capsule is welded to the upper plate in Fig. 7, and a driver's seat used exclusively for racing cars and a steering wheel taken apart from a real vehicle are installed into it.

A Pentium 866, equipped with an AX5611C and an A626 data acquisition cards, executes the numerical analysis of vehicle dynamics and the control of motion platform. The mathematical equations of vehicle

Fig. 7 Designed motion platform

dynamics are programmed in MATLAB^{(42),(43)}, translated into C++ codes by MIDIVA⁽⁴⁴⁾, and integrated by C++ Builder into a control code. The steering wheel angle, accelerator and brake pedal positions are encoded into analog voltages by transformers, acquired by AX5611C, converted into digital signals, and are sent to vehicle dynamics analysis programs. The vehicle's longitudinal, lateral and vertical displacements, as well as roll, pitch and yaw angles, are carried out and transferred into required actuation voltages, converted into analogue signals and sent to cylinders by A626.

A Pentium 650, also equipped with an AX5611C data acquisition card, is used to perform virtual reality for the audio/video presentations and x -direction simulation. Driver's operation signals are acquired and sent to Superscape VR program through dynamic data exchange functions. Using Superscape Control Language (SCL), the control codes are written into the objects to process these signals and synchronize the VR environment with the real world. When the motion platform is not actuated (i.e., Pentium 650 working alone), this can be used exclusively as a driving simulator to test the driving skill of the driver if a gear-shifting device is added. On the other hand, by using network technology, the calculated results of the vehicle motions in Pentium 866 can be sent to the VR program in Pentium 650 to synchronize the real world. In this case, Pentium 650 does not need to

perform data acquisition task thus its burden would be reduced and the VR environment frame rate can be increased.

4. Results and Discussions

The major results of this paper are:

1. A full vehicle dynamics simulation program is developed using MATLAB.
2. A 5-axis motion platform is constructed and controlled to perform 5-DOF motions similar to the motions of a road vehicle.
3. A virtual reality environment that is used to simulate the x -dir motion and to aid the video/audio presentation is constructed and can be employed either in off-line analysis and simulations or on-line simulations.
4. By using network technology, the calculated signals of the vehicle motions in one computer can be sent to the VR program in the other computer to synchronize the real world.

The constructed driving simulator, as shown in Fig. 9, utilizes only five cylinders to reduce the cost. It is aimed at simulating the road vehicle. Because numerical calculations (especially the iteration procedures) consume much CPU time, control program only acquires operation signals every tenth of a second. The requirement of 10 frames per second (fps) visual display is fulfilled and the VR environment synchronizes the real world as Figs. 10 and 11 show.

Fig. 8 Graphical representation of the driving simulator

Fig. 9 The constructed driving simulator

Fig. 10 VR environment of the driving simulator⁽¹⁾ Fig. 11 VR environment of the driving simulator⁽²⁾

Fig. 12 User interface of the control program (Borland C++ Builder)

The graphical user interface (GUI) created by C++ Builder displays not only the numerical values of the vehicle motion variables, but also the trajectory of the vehicle center of gravity, as shown in Fig. 12.

5. Conclusions and Future Works

This paper describes the construction of a 5-axis driving simulator based on vehicle dynamics analysis and system integration. MATLAB software provides many built-in *m*-files for mathematical calculations and is adopted to ease the tedious burden of writing numerical analysis codes if C++ is used instead. On the other hand, C++ Builder makes available of many GUI design functions superior to MATLAB. There-

fore, MIDEVA is used to translate *m*-files into C++ codes, and the data acquisition and control codes are then integrated in the C++ Builder environment to provide a more user-friendly man-machine interfaces.

The constructed driving simulator can perform lateral and vertical displacements and roll, pitch and yaw motions similar to the 5-DOF motions of a road vehicle while the longitudinal translation is simulated by VR technology. The virtual reality environment that is used to simulate the *x*-dir motion can also be employed either in off-line analysis and simulations or on-line simulations.

In the future, more powerful computers can be used to increase the number of fps and to speed up the

numerical calculations. Thus all programs can run on a single computer and more complex vehicle models can be adopted such as the including of powertrain. Also, a road surface model and more complicated visual displays will be incorporated to enhance the reality of the simulator. Furthermore, a pressure vest can be added to produce the feeling of acceleration/deceleration to imitate longitudinal translation. By adding these accessories, a complete six degrees of freedom driving simulator will be complete.

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