Human Vehicle Dynamics Coupling of Miniature Electric Vehicle Roll Stability: Effects of Driver Weight, Wheel Base and Track Width

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Abstract. As a continuation of the previous research, influences of human vehicle dynamics coupling (HVDC) on miniature electric vehicle (MEV) roll stability are studied by varying driver weight, wheel base and track width. A multi-body dynamics model is developed for the human-vehicle system by ADAMS/Car software considering driver’s dynamics and is validated theoretically. Simulation results show that the MEV roll stability is decreased by the HVDC; the influences become greater with an increase in the proportion of the driver weight to curb weight of the MEV in both time and frequency domain; with the decrease of wheel base, the differences caused by the HVDC exhibit an increasing trend in the maximum value and overshoot; and the differences in the steady stable roll angle and phase lag at 0.6 Hz indicate an increasing tendency with the decrease of track width.

Keywords: Miniature electric vehicle; Roll stability; Driver weight; Wheel base; Track width

1. Introduction
In recent years, numerous miniature electric vehicles (MEVs) have been widely manufactured, such as Renault Twizy weighted 450 kg, X Electrical Vehicle LSEV weighted 450 kg and CHJAutomotive SEV (Smart Electric Vehicle) less than 350 kg. These MEVs can ease many problems, e.g. traffic congestion, resource shortage, and difficulty for parking, derived from the growth of automotive production and sales. This indicates the MEVs which are light weight and small in size will be one of the development trends in future automobiles [1, 2]. However, roll stability of the MEVs is susceptible, as a result of decreases in longitudinal and lateral dimensions and maintenance of vertical size after vehicular miniaturization [3].

When the moving MEV changes the direction of travel, the driver who is elastically supported on and constrained to the vehicle must counteract lateral, roll and yaw accelerations. Vibrations and rocking motions of the driver body will in turn affect vehicle dynamics stability as the weight of the driver is close to that of the MEV. The process indicating the interaction of motions and forces between the driver and the vehicle here is called human vehicle dynamics coupling (HVDC).

It has been noticed in the previous study [4] that the MEV shows greater roll angle in time domain and lower resonance frequency of roll motion on account of the HVDC, and the influences go higher with a decrease in vehicle mass. Further in-depth understanding of the influences of the HVDC on the MEV roll stability needs to be gained under various driver weight, wheel base and track width.

Literatures have discussed the influences of vehicle parameters on vehicle roll stability. Zhang et al. developed a 9 degree of freedom (DOF) model and indicated that the vehicle with increased suspension stiffness displayed better untripped rollover resistance capacity [5]. Wei et al. analysed the effects of clearance in the kinematic pair of steering system on vehicle handling and stability [6].
Wang et al. discussed the effects of track width and roll stiffness on vehicle roll stability [7]. Mereno et al. analysed the influences of overweight and the distribution of the centre of gravity (CG) on driving stability and rollover risk of long combination vehicles [8-9]. Shim and Velusamy identified and adjusted critical suspension design parameters to improve vehicle roll stability [10]. Zhang et al. discussed hard point coordinates and bushing stiffness of vehicle suspension, and provide a multi-objective optimization method to enhance the handling stability [11].

The influences of the HVDC on the MEV roll stability are studied under different conditions of driver weight, wheel base and track width in the following paper. In section 2, a multi-body dynamics model is built for the human-vehicle system by the software ADAMS (Automatic Dynamic Analysis of Mechanical System). In section 3, the virtual prototype model is validated by the mathematical model presented in previous study. In section 4, the effect of driver weight, wheel base and track width are discussed. Conclusions are drawn in section 5.

2. Multi-body Dynamics of Human-vehicle System

2.1. MEV Model

Main parameters of the MEV are presented in table 1, which shows great differences in vehicle curb weight, wheel base, and track width between the MEV and traditional large-sized vehicles. Based on these parameters, a multi-body dynamics model of the MEV which is comprised of front and rear suspension system, tires, steering system, powertrain, body system and braking system is built. The stiffness and damping coefficients of the suspensions are calculated by the method in Wang [12]. The moments of inertia of the MEV are estimated according to the empirical formula in Yin et al. [13].

| Project       | Parameter     | Project       | Parameter     |
|---------------|---------------|---------------|---------------|
| Length        | 2337 mm       | Width         | 1191 mm       |
| Wheel base    | 1686 mm       | Height        | 1461 mm       |
| Track width   | 1094/1080 mm  | Rated passengers | 2             |
| Curb weight   | 450 kg        | Max speed     | 80 km/h       |

2.2. Seated Driver Model

The seated driver is modelled as two rigid segments: the lower body segment denotes the hip, the buttocks, the legs and the feet; the upper one includes the truck, the arms and the head. The human body dimensions of adults are referred to Chinese Standards GB/T 10000-1988 [14] and GB/T 13547-1992 [15]. The parameters of the 90th percentile male aged from 18 to 25 are chosen in modelling the seated driver.

2.3. Contact Analysis of Seated Human Body and Vehicle

For a realistic driver, the contact with a vehicle locates at the seat cushion, backrest, safety belt, steering wheel and the floor. The interrelation and interaction at the contact surface are strongly complicated, including linear and nonlinear deformations, coupled with static and sliding frictions. However, the contact relation between the vehicle and the seated human body can still be expressed by the linear elasticity theory, on condition that the intention is only to investigate the impact of the motion of the seated human body subjected to large-displacement stimuli on the dynamics responses of the vehicle, and motion state of the seated human body at every moment are not necessary to be accurately calculated. Linear springs and dampers are thus employed to represent the flexibilities and frictions.

The seated human body is flexibly constrained to a vehicle mainly through the vehicle seat which undergoes most of the forces and moments from the seated body. Therefore, the interconnections between the seated human body and the vehicle are simplified to the interconnections between the
seated body and the seat. Tri-axis translational springs and dampers are adopted to represent the elastic forces and frictions at the human-seat interface. Under this circumstance, at least three non-collinear contact points are essentially required to place the springs and dampers to simulate the interactive moments due to roll, pitch and yaw motion of the seated human body, as shown in figure 1(a). It should be pointed out that the number of the spring and damping coefficients for modelling the seated human body and computation for parameter identification increase with the number of contact points. Therefore, the three contact points mentioned above are decreased to a single one for simplicity by introducing tri-axis translational and rotational springs and dampers, as shown in figure 1(b). Theoretically, it can be located anywhere as there is only one contact point. However, placing the contact point at the CG of the seated body can omit the moments produced by the gravity in differential equations of motion.

\[ \begin{align*}
(k_{1x}, k_{1y}, k_{1z}) \\
(k_{2x}, k_{2y}, k_{2z}) \\
(k_{3x}, k_{3y}, k_{3z})
\end{align*} \]

\[ \begin{align*}
(k_{8x}, k_{8y}, k_{8z}) \\
(c_{10}, c_{11}, c_{12}) \\
(k_{12x}, k_{12y}, k_{12z}) \\
(c_{20}, c_{21}, c_{22})
\end{align*} \]

**Figure 1.** Distribution of contact points: (a) Three or more points; (b) Only one point.

### 2.4. Integrated Model

A multi-body dynamics model of human-vehicle system is built, and is employed to conduct the research in the following section. The flexible constraints of the driver to the MEV are simplified into tri-axis translational and rotational springs and dampers between the seated human body and the vehicle seat, at the CGs of the lower and upper body respectively. Additionally, the two body segments are connected by a tri-axis translational and rotational springs and dampers at the waist. Thus, a simulation model considering the HVDC is constructed, as shown in figure 2. The spring and damping parameters at the human-seat interface are referred to the literature [16]. A comparison model neglecting the HVDC is built by assigning fixed joints between the driver and the MEV.

\[ \text{Roll, Yaw} \]

**Figure 2.** Multi-body dynamics model of human-vehicle.

### 3. Model Validation

Luo et al. [17] suggested a 6-DOF mathematical model, including lateral motions, yaw motions and roll motions of both the MEV and the driver, and a 3-DOF contrast model for the human-vehicle
system. In this section, the multi-body dynamics models are validated theoretically by the mathematical models.

In the validation, the MEV is driven on fine road at a constant speed of 10 m/s; the maximum steering wheel angle is 60°, and the corresponding angle of front tires is 3° in the sine manoeuvre. It is driven in a straight line at the beginning of the simulation for 1s; it then negotiates a sine turn with duration of 2s and finishes with a straight-line event at the last 2s.

The simulation results of the multi-body models and mathematical models are compared, as shown in figure 3. The responses derived from the ADAMS models shows a good agreement with the data obtained by the theoretical models, except small mismatches in the roll angle. The conformity indicates the validity of the ADAMS models.

![Comparison curves](image)

**Figure 3.** Comparison curves.

### 4. Simulation and Results

The influences of the HVDC on the MEV roll stability are only studied in this section, as little influences are caused on the yaw and lateral response [17].

#### 4.1. Effect of Driver Weight

The load of the MEV is mainly the driver. Therefore, fishhook tests with the speed of 45km/h and the steering wheel angle of 70° are carried out under different driver weight. The results are shown in figure 4. The peak roll angle, steady stable value (SSV) and the transient overshoot increase when the HVDC is considered. The SSVs increase by 0.09, 0.12°, 0.17° and 0.21° for the MEV with the driver of 60 kg, 70kg, 80 kg and 90 kg, respectively; and the relative increments are 3%, 3.9%, 5.4% and 6.5%. The differences between the results neglecting the HVDC and those with the HVDC are 7.39%, 8.9%, 11% and 12.88% in the overshoots. The increased roll angle and overshoot indicate that the tendency to rollover of the MEV increases on account of the HVDC. The influences go higher with increasing driver weight.
Sine sweep steers are carried out under the speed of 40 km/h and the peak steering wheel angle of 70°. The results are shown in figure 5. The steady state gain (SSG), amplitude ratio and phase lag of the roll angle become larger while the resonance frequency decreases after the HVDC considered. With the increase of the driver weight, the differences caused by the HVDC are increasingly remarkable in the SSG, resonance frequency and the phase lag angle at 0.6 Hz.

Combined with the results in the previous study [4], it can be concluded that the MEV roll stability is decreased by the HVDC, and the influences become greater with an increase in the proportion of the driver weight to the total of the MEV in both time and frequency domain.

4.2. Effect of Wheel Base and Track Width
Fishhook tests are conducted under different wheel base and track width. Only 3 group of simulation results are shown in figure 6 to avoid dense curves. The roll angle of the MEV increases with a decrease in both wheel base and track width, and a greater change in the roll angle occurs with a variation in track width. With the decrease of wheel base, the differences between the roll angles considering and neglecting the HVDC exhibit an increasing trend in the maximum value and overshoot. The peak increases by 0.29°, 0.31°, 0.38°, 0.46° and 0.49° respectively, coupled with the overshoot individually by 5.18%, 6.67%, 8.26%, 9.38% and 10.15% while the wheel base is 1.25, 1.2, 1.1, 1.0 and 0.9 times of the original one. With the decrease of track width, the steady stable roll angle indicate an increasing tendency after the HVDC considered. The SSV increases by 0.08°, 0.09°, 0.11°, 0.13° and 0.16° while the track width is 1.25, 1.2, 1.1, 1.0 and 0.9 times of the original one.
Figure 6. Effect of wheel base and track width in time domain: (a) Wheel base; (b) Track width.

The results of sine sweep steers are shown in figure 7. The difference at 0.6 Hz between the phase lag of the roll angles considering and neglecting the HVDC increases with the decrease of track width, which increases by 6.34°, 6.43°, 6.61°, 6.98° when the track width is 1.2, 1.1, 1.0 and 0.9 times of the original one. The differences caused by the HVDC in the SSG, resonance frequency, amplitude ratio and phase lag at 0.1 Hz do not show a clear regularity with the variation of wheel base and track width.

Figure 7. Effect of wheel base and track width in frequency domain: (a) Wheel base; (b) Track width.

5. Conclusions
A multi-body dynamics model of human-vehicle system is built by ADAMS/Car software considering the driver’s dynamics and is validated theoretically. The model is employed to conduct the simulations of the influences of the HVDC on the MEV roll stability under different driver weight, wheel base and track width. Results show that the MEV roll stability is decreased by the HVDC; the influences become greater with an increase in the proportion of the driver weight to curb weight of the MEV in both time and frequency domain; with the decrease of wheel base, the differences caused by the
HVDC exhibit an increasing trend in the peak and overshoot; with the decrease of track width, the differences in the steady stable roll angle and phase lag at 0.6 Hz indicate an increasing tendency.

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References
[1] Bonills D, Schmitz K E and Akisawa A 2012 Demand for mini cars and large cars; decay effects, and gasoline demand in Japan Energy Policy 50: 217-27.
[2] Liu H, Wang H, Luo Q, et al. 2007 Energy, environment and economic assessment of life cycle for electric vehicle Technol. Econ. Areas Commun. 9: 45-8.
[3] Chen Q P, Li X Q and Zhou, X J 2017 Analysis and research on the man-machine dynamics behaviour coupling mechanism of the super miniature electric vehicle Int. J. Veh. Saf. 9: 352-68.
[4] Luo S, Shu H Y and Chen X B 2016 Human vehicle dynamics coupling effect of miniature vehicle roll characteristic Int. J. Elec. Hyb. Veh. 8: 97-108.
[5] Zhang N, Dong G M and Du H P 2008 Investigation into untripped rollover of light vehicles in the modified fishhook and the sine manoeuvres. Part I: Vehicle modelling, roll and yaw instability Veh. Syst. Dyn. 46: 271-93.
[6] Wei D G, Wang Z H, Zhang Y T, et al. 2014 A study on the influence of the clearance in steering system on vehicle handling and stability Automobile Engineering 36: 139–44.
[7] Wang R, Li X S, Ren Y Y, et al. 2013 Roll stability analysis of passenger car based on lateral-load transfer rate Journal of Human University (Natural Science) 40: 49-54.
[8] Moreno G G, Barreto R L P, Vieira R. S, et al. 2016 Three-dimensional analysis of vehicle stability using graph theory Graph-Based Modelling in Engineering: 117–29.
[9] Moreno G G, Manenti V, Vieira R S, et al. 2018 Rollover of long combination vehicles: effect of overweight Multibody Mechatronic Systems 54: 497-505.
[10] Shim T and Velusamy P C 2011 Improvement of vehicle roll stability by varying suspension properties Vehicle System Dynamics 49: 129-52.
[11] Zhang L X, Liu J Q, Pan F, et al. 2020 Multi-objective optimization study of vehicle suspension based on minimum time handling stability Proc. IMechE Part D: J Automobile Engineering 234: 2355-63.
[12] Wang W Y 2004 Vehicle Design Beijing China Machine Press.
[13] Yin L, Lv C and Li L 2016 Analysis and improvement on vehicle body yaw based on Adams/Car Bus Technology and Research 6: 22-5+49.
[14] GB/T 10000-1988 1988 Human Dimensions of Chinese Adults.
[15] GB/T 13547-92 1992 Human Dimensions in Workspaces.
[16] Luo S, Shu H Y and Chen X B 2019 Experiment and modeling of seated human body exposed to large-magnitude vibration in six single directions J. Mech. Med. Biol. 19 1950037.
[17] Luo S, Shu H Y, Yao Z J, et al. 2018 Effects of human-vehicle dynamics interaction on the handling stability of miniature automobile J. Harbin Inst. Technol. 50: 146-53.