Experimental vehicle model considering suspension coupling effect for handling analysis

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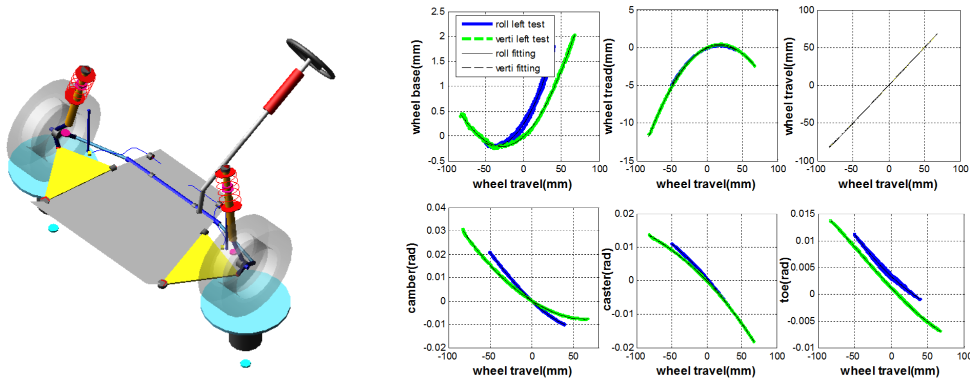
Abstract
In spite of increasing computational power of modern processors, an efficient vehicle model in multi-body vehicle simulation remains an important issue. There are many vehicle models built for vehicle dynamics specifically for the ride and handling [1-2]. To avoid the computational complexity associated with bushings or kinematic linkages, a semi-empirical vehicle model was developed. In this model, all suspension characteristics are lumped into unsprung mass at each corner, which includes kinematic and compliance characteristics between wheel and body from the SPMD(Suspension Parameter Measurement Device) test[3-4]. In this study, this test is carried out in ADAMS/CAR[5] as shown Figure 1(a). This can measure the relative displacement data by moving the wheel along the vertical direction. The results are defined as relative vehicle movements between wheel and body. Figure 1(b) shows test results from the SPMD. The ride test is obtained with parallel wheel travels in right-and-left wheels. The roll test is obtained from an opposite wheel travel test. Even for independent suspensions, the wheel alignment characteristics are different since the coupling effect might be occurred by wheel travel position of right and left side.



(a) SPMD test environment (b) Results from ride and roll tests

Figure 1: SPMD test environment and results.

In order to express this characteristic, the relative displacement will be introduced. In the modeling, the equations of motion are calculated according to the generalized degrees of freedom by the vehicle and wheels center of gravity (CoG). All degrees of freedom are calculated in the vehicle coordinate system, and any representation of the velocities or positions in the global reference system are carried out by coordinate transformation. By using the equation (1), the velocities of the chassis and wheels are written.

$$\dot{y} = B\dot{q}$$
$$B = \begin{bmatrix} I & & & & & \\ D_{fl} & Ri_{fl}, Ro_{fr} & Ro_{fr} & & & STR \\ D_{fr} & Ro_{fl} & Ri_{fr}, Ro_{fl} & & & STR \\ D_{rl} & & & Ri_{rl}, Ro_{rr} & Ro_{rr} & \\ D_{rr} & & & Ro_{rl} & Ri_{rr}, Ro_{rl} & \\ D_{str} & & & & & \end{bmatrix} \quad (1)$$

where B is the velocity transformation matrix, D, Ri, Ro and STR are the transform matrices to define relative relation between wheel and body. And and are global velocity and generalized velocity, respectively. Using the displacement z of wheels and steering displacement δ, the Ri, Ro and STR can be parameterized.
By using the equation (1), the equation of motion for the vehicle model can be written as;

$$\bar{M}\ddot{q} = \bar{Q} \tag{1}$$

where $\bar{M} = B^T MB$ is the mass matrix, $\bar{Q} = B^T \left(Q - MH\right)$ is applied force, Coriolis and centrifugal force.
In order to verify the model, the handling test simulation was performed. In the vehicle simulation, the MF-Tire model was employed. Comparing multi-body model, the dynamic performance of the proposed model can be enhanced.

Acknowledgments

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