

Vehicle System Simulator: Development and Validation

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ABSTRACT

A graphical user interface (GUI) toolbox called Vehicle System Simulator (VSS) is developed in MATLAB to ease the use of this vehicle model and hopefully encourage its widespread application in the future. This toolbox uses the inherent MATLAB discrete-time solvers and is mainly based on Level-2 s-function design. This paper describes its built-in vehicle dynamics model based on multibody dynamics approach and nonlinear tire models, and traction/braking control systems including Cruise Control and Differential Braking systems. The built-in dynamics model is validated against CarSim 8 and the simulation results prove its accuracy. This paper illustrates the application of this new MATLAB toolbox called Vehicle System Simulator and discusses its development process, limitations, and future improvements.

INTRODUCTION

Vehicles, derived from the Latin word “vehiculum” are nonliving means of transport. Transportation can be considered as a means of conveyance from one point or place to another. Most vehicles represent interesting and complex dynamics systems that require careful analysis and design to make sure they perform properly and hold stability. The stability aspect of vehicle motion has to do with assuring that the vehicle does not deviate spontaneously from a desired path and is still under control. The control aspect of vehicle stability has to do with the ability of a human operator or an automated system to guide the vehicle along the desired path. In either case, the dynamic parameters of such vehicles should be properly tailored to allow controllability with reasonable ease and precision. The inherent complexity of actual vehicle systems makes an exact analysis of the system impossible; however, simplifying assumptions may reduce

the system complexity level in such a way that the behavior or performance approximates that of the real system. This process is called “modeling”, and the simplified version of the real system is called the mathematical model of the system.

Throughout this paper, the development process of a vehicle dynamics simulation toolbox called Vehicle System Simulator (VSS) is explained along with its inherent vehicle dynamics model and traction/braking control systems. The built-in nonlinear vehicle model is developed base on multibody dynamics approach and nonlinear tire models, both physical and semi-empirical. To control the vehicle speed Cruise Control and Differential Braking systems are also described in this paper. The cruise control is mainly based on PID control concept while the differential braking system includes Active Slip Control (ASC), Differential Brake Distributor, and Antilock Braking System (ABS) to distribute the braking torques on the wheels.

HISTORY OF VEHICLE DYNAMICS MODELING

While the first self-propelled steam road vehicle was successfully tested in 1767 by Cugnot, the pioneer work in vehicle engineering occurred in 1903 when the Wright brothers successfully built their first airplane. A few years later, G. H. Bryan started his pioneering work on a mathematical theory of airplane stability [1]. Development of scale models and wind tunnels evolved the understanding of aerodynamics and how it affects the stability of an aircraft. Yet lack of understanding of the role of the tire mechanics in the stability of an automobile had slowed down the development of an automotive stability theory.

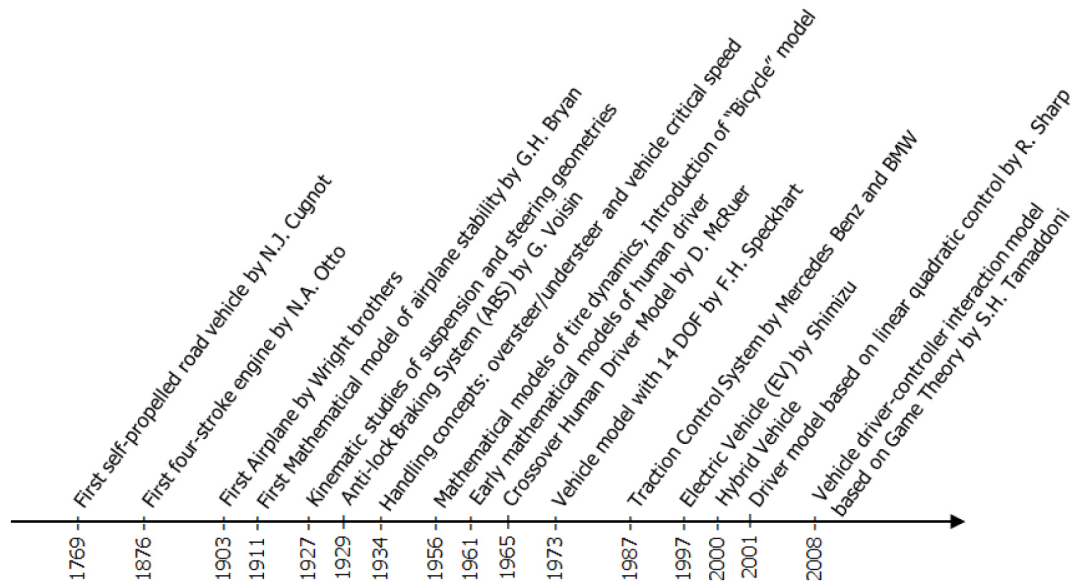


Figure 1. History of vehicle study

During 1900 and 1930, the emphasis on improving the ability of the driver to control the vehicle led to kinematic studies of suspension and steering geometries resulting in improved designs including Ackermann steering geometry. In that period, vehicles were plagued with the phenomenon of steering shimmy. To explain this phenomenon, Broulheit introduced the basic concepts of sideslip and slip angle for the first time [2]. Following Broulheit's work, Becker, Fromm, and Maruhn published a text on the role of the tire in steering system vibrations and further developed the field of tire mechanics [3].

In 1930's, the Cadillac Suspension Group of General Motors (GM) developed the first independent front suspension system. Olley and his co-researchers in GM found that certain steering geometries led to a condition called roll oversteer which was later recognized unsafe at high speeds [4]. In 1934 an unpublished report by Olley stated the proposition of oversteer/understeer and the idea of critical speed [5]. In 1935 Goodyear Tire and Rubber Company began rolling drum tests to determine tire characteristics. The results of lateral tire characteristics including data on cornering force and self-aligning torque were published in [6].

During 1940's very little progress was made due to World War II. In 1950 Lind Walker introduced the concept of the neutral steer line and the stability margin [7]. Although these concepts had already been established in aeronautical circles, Walker for the first time suggested them as criteria for steady state directional stability in automobiles.

By the middle of the 1950s a basic understanding of the tire enabled the creation of reasonably accurate mathematical tire models. In 1956 William F. Milliken, David W. Whitcomb, and Leonard Segel of the Cornell Aeronautical Laboratory,

published the first major quantitative and theoretical analysis of vehicle handling in a series of papers [8,9,10].

Milliken [8] stated that the major effort in handling research to date has been in the recognition of individual effects, their isolation, and examination as separate entities. There were no universally accepted set of reference axes and measured tire data of the period were typically confined to two or three of the possible six force/moments. This made translation of the data from one set of axes to another difficult if not impossible. Milliken also emphasized the need to concentrate on the objective analysis of car stability and control; therefore, he made the following distinctions between stability and control, performance and ride. In general, an automobile has six-degrees-of-motion freedom, and stability and control may be thought of as those lateral motions out of the plane of symmetry involving rolling, yawing and side slipping. ("Performance", by way of distinction, is concerned with fore-and-aft motions in the plane of symmetry, such as acceleration, speed, and braking, while "ride" is composed of the vertical and pitching motions in the plane of symmetry.)

Leonard Segel's work [9] was devoted to derivation of a set of nondimensionalized linearized three degree of freedom equations for lateral and directional motion. Segel ignored the bounce and pitch degrees of freedom of the chassis and used a fixed longitudinal roll axis parallel to the ground. The other simplifying assumptions included constant forward velocity, fixed driving thrust divided equally between the rear wheels, decoupled tire lateral properties from longitudinal properties, and a single non-rolling lumped mass as the unsprung mass. An experimental validation of the model was performed by a 1953 Buick Super four door sedan at 32 mph in a series of frequency response curves with the results showing good correlation.

D. W. Whitcomb's paper [10] introduced the concept of the “static margin” and discussed automobile stability and control using a two degree of freedom model including yaw and side-slip which is today referred as bicycle model.

In 1961 Martin Goland and Frederick Jindra added the roll degree of freedom as a quasi-coordinate to the two degree of freedom (yaw and sideslip) vehicle model of Segel in order to calculate the vertical load on the tires. The effects of load transfer and the variation of the cornering performance of the tires with vertical loading were thus presented in [11].

In 1967 R. Thomas Bundorf of General Motors published the relation of vehicle design parameters to the characteristic speed and to understeer [12]. Bundorf argued that under most normal driving conditions, which he characterized as having lateral accelerations below $1/3 g$, a vehicle can be accurately modeled by a linear model.

In 1973 Frank H. Speckhart presented a vehicle model containing fourteen degrees of freedom [13]. Six degrees of freedom were assigned to the sprung mass, four degrees of freedom were associated with the suspension movement at the four corners of the vehicle, and four rotational degrees of freedom were assigned to the wheels. He used a Lagrangian approach in deriving his equations.

During 1970's several time-saving methods were published such as Bernard's digital simulation code for Highway Safety Department [14], Hybrid Computer Vehicle Handling Program [15], and Ford Motor Company's model containing detailed three degree of freedom model of the front suspension [16].

In 1987 twenty-five different suspension types were tested to understand the effects of suspension design on the stability of vehicles [17]. A typical three degree of freedom lateral dynamics model was used with the addition of a quasi-static pitch degree of freedom. The roll axis of sprung mass was a function of body roll. This set of tests helped Nalecz to understand that the assumption of a fixed roll axis could not be justified for certain types of suspensions such as double wishbone and MacPherson strut.

In 1992 another model with eight degrees of freedom, consisted of a three degree of freedom lateral dynamics model coupled to a five degree of freedom planar rollover model, was developed [18]. The lateral dynamics model was derived in the same manner as Segel's original model. The rollover model consisted of sprung and unsprung masses connected through the various elements of the suspension system. The effects of aerodynamic forces and moments and lateral and longitudinal weight transfer were also included in this model.

In 1996 Michael R. Petersen and John M. Starkey described a relatively detailed straight-line acceleration vehicle model for predicting vehicle performance [19]. The model included longitudinal weight transfer effects, tire slip, aerodynamic drag, aerodynamic lift, transmission and driveline losses and rotational inertias of the wheels, engine and driveline components. The model used a manual transmission with fully clutch engagement in such a way that shifts were simulated by disengaging the clutch completely, assuming that the engine torque is zero during the shift, changing the gear ratio, and then reapplying the full torque of the motor. This model enabled the authors to conduct sensitivity analyses in order to determine which design parameters most strongly affected vehicle performance.

VEHICLE SYSTEM SIMULATOR

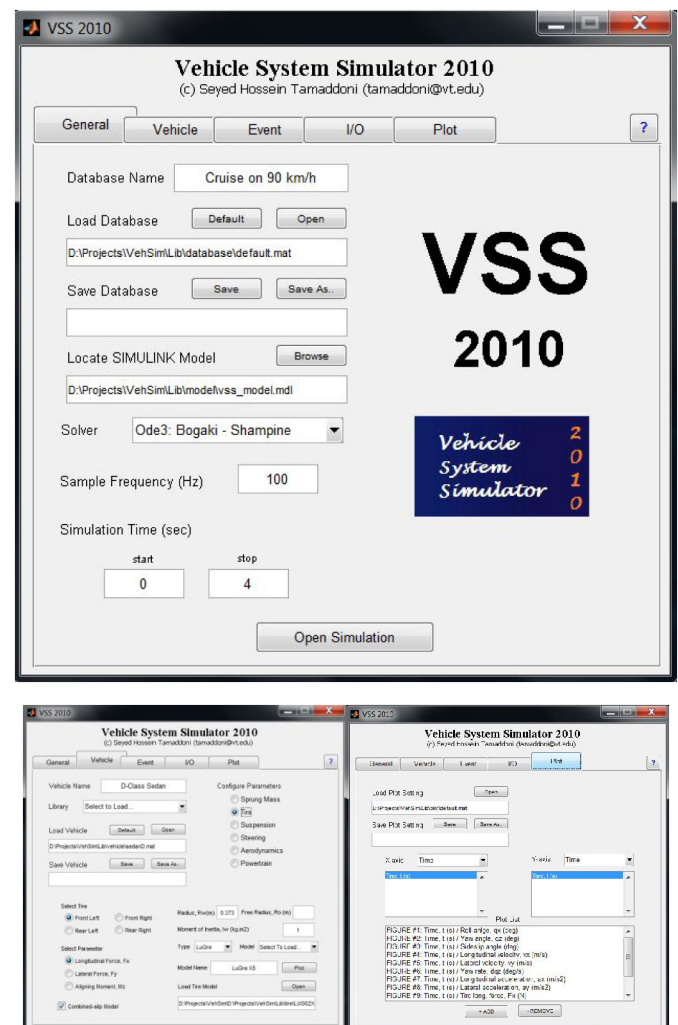


Figure 2. MATLAB GUI of Vehicle System Simulator

Vehicle System Simulator (or VSS) [20] is a user-friendly toolbox in MATLAB [21] developed in Center for Vehicle Systems and Safety (CVeSS) at Virginia Tech. This toolbox uses the inherent MATLAB discrete-time solvers and is

mainly based on Level-2 s-function design. The graphical user interface (GUI) part of this toolbox takes user inputs including vehicle parameters, event specifications, and plot configurations, while the Simulink model contains the mathematical model of the designated vehicle dynamics. The screenshots of Vehicle System Simulator are shown in [Figure 2](#).

DYNAMICS MODEL

Vehicle System Simulator (VSS) has a complex built-in nonlinear model of a vehicle system which includes longitudinal and lateral translational motions, and roll and yaw motions with rotational dynamics of each of the four wheels. The following sections briefly describe the governing dynamics of a road vehicle system.

BODY DYNAMICS

[Figure 3](#) depicts the sprung and unsprung body coordinates of a four-wheel vehicle. The unsprung mass is connected to the sprung mass with four suspension linkage shown by hashed lines. The body coordinates U and S are intermediate axis systems for the unsprung and sprung masses, respectively.

The presented vehicle handling model has five degrees of freedom: longitudinal velocity v_x , lateral velocity v_y , roll angle θ_x , pitch angle θ_y , and yaw angle θ_z . The nonlinear vehicle model is derived by writing the translational and rotational equations in the vehicle fixed coordinate frame.

Assuming that the sprung mass can only rotate along the X and Y axes of the unsprung coordinates, the sprung body will perform the pitch and roll motion. It is also assumed that the unsprung mass performs yaw motion and there is no relative yaw angle between the sprung and unsprung bodies.

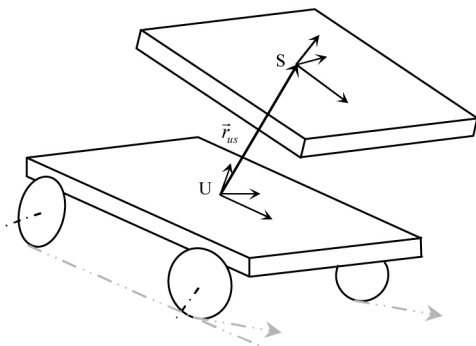


Figure 3. Vehicle schematics

The rotation matrices of roll and pitch motions of the sprung coordinate S along the X and Y axes of the unsprung coordinate U are

$${}^S_R_U = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \theta_x & -\sin \theta_x \\ 0 & \sin \theta_x & \cos \theta_x \end{bmatrix}, \quad {}^S_R_y = \begin{bmatrix} \cos \theta_y & 0 & \sin \theta_y \\ 0 & 1 & 0 \\ -\sin \theta_y & 0 & \cos \theta_y \end{bmatrix} \quad (1)$$

The location of the sprung coordinate relative to the unsprung coordinate is given by

$${}^S\vec{r}_{us} = {}^U_R_y \cdot {}^U_R_x \cdot \begin{bmatrix} 0 \\ 0 \\ -h \end{bmatrix} \quad (2)$$

The angular velocities and accelerations are obtained as

$${}^U\vec{\omega}_u = \begin{bmatrix} 0 \\ 0 \\ \dot{\theta}_z \end{bmatrix}, \quad {}^S\vec{\omega}_s = {}^U\vec{\omega}_u + \begin{bmatrix} \dot{\theta}_x \\ \dot{\theta}_y \\ 0 \end{bmatrix} \quad (3)$$

$${}^U\vec{\alpha}_u = \begin{bmatrix} 0 \\ 0 \\ \ddot{\theta}_z \end{bmatrix}, \quad {}^S\vec{\alpha}_s = {}^U\vec{\alpha}_u + \begin{bmatrix} \ddot{\theta}_x \\ \ddot{\theta}_y \\ 0 \end{bmatrix} + {}^U\vec{\omega}_u \times {}^S\vec{\omega}_s \quad (4)$$

The translational velocities and accelerations are

$${}^U\vec{V}_u = \begin{bmatrix} v_x \\ v_y \\ 0 \end{bmatrix}, \quad {}^U\vec{a}_u = \begin{bmatrix} \dot{v}_x \\ \dot{v}_y \\ 0 \end{bmatrix} + {}^U\vec{\omega}_u \times {}^U\vec{V}_u \quad (5)$$

$${}^U\vec{V}_s = {}^U\vec{V}_u + {}^S\vec{\omega}_s \times {}^S\vec{r}_{us} \quad (6)$$

$${}^U\vec{a}_s = {}^U\vec{a}_u + {}^S\vec{\omega}_s \times {}^S\vec{\omega}_s \times {}^S\vec{r}_{us} + {}^U\vec{\alpha}_s \times {}^S\vec{r}_{us} \quad (7)$$

The sprung-mass moment of inertia matrix is also transformed to the unsprung body coordinate system as

$${}^U\vec{I}_s = {}^S_R_y \cdot ({}^S_R_x \cdot {}^S\vec{I}_s \cdot {}^S_R_x^T) \cdot {}^S_R_y^T \quad (8)$$

The equations of motion can be written as

$$\begin{cases} (m - m_s) \cdot {}^U \ddot{a}_u + m_s \cdot {}^U \ddot{a}_s = \sum_i {}^U \vec{f}_i \\ {}^U \dot{\vec{H}}_u + {}^U \dot{\vec{H}}_s + {}^U \vec{r}_{us} \times (m_s \cdot {}^U \ddot{a}_s) = \sum_i {}^U \vec{\tau}_i \end{cases} \quad (9)$$

where

$$\begin{aligned} {}^U \dot{\vec{H}}_u &= {}^U \vec{I}_u \cdot {}^U \ddot{\alpha}_u + {}^U \vec{\omega}_u \times ({}^U \vec{I}_u \cdot {}^U \vec{\omega}_u) \\ {}^U \dot{\vec{H}}_s &= {}^U \vec{I}_s \cdot {}^U \ddot{\alpha}_s + {}^U \vec{\omega}_u \times ({}^U \vec{I}_s \cdot {}^U \vec{\omega}_s) \end{aligned} \quad (10)$$

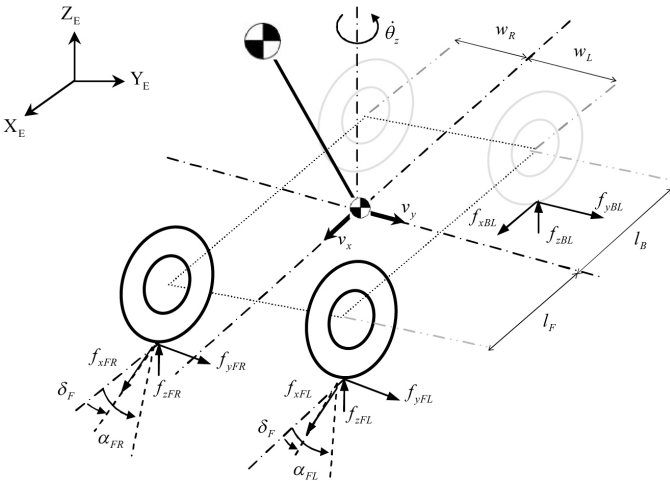


Figure 3. Vehicle's lower-body schematics

The final equations are in form of

$$\mathbf{A}(q) \ddot{q} + \mathbf{B}(q, \dot{q}) = \eta \quad (11)$$

where the generalized forces are given by

$$\eta_x = \frac{MU^2}{L} (\sum f_x \cos \delta - \sum f_y \sin \delta) \quad (12)$$

$$\eta_y = \frac{MU^2}{L} (\sum f_x \sin \delta + \sum f_y \cos \delta) \quad (13)$$

$$\eta_{\theta_x} = MU^2 (-k_x \theta_x - c_x \dot{\theta}_x + gh_s (\sin \theta_x \cos \theta_y - \cos \theta_x \sin \theta_y)) \quad (14)$$

$$\eta_{\theta_y} = MU^2 (-k_y \theta_y - c_y \dot{\theta}_y + gh_s (\cos \theta_x \sin \theta_y + \sin \theta_x \cos \theta_y)) \quad (15)$$

$$\eta_{\theta_z} = MU^2 (\sum l'_x \sin \delta - \sum l'_y \cos \delta + \sum w'_x \cos \delta - \sum w'_y \sin \delta) \quad (16)$$

where

$$\begin{aligned} l' &= [l_F \quad l_F \quad -l_B \quad -l_B]^T, \\ w' &= [w_R \quad -w_R \quad w_L \quad -w_L]^T, \\ \delta &= [\delta_F \quad \delta_F \quad 0 \quad 0]^T \end{aligned} \quad (17)$$

For a simple case in which there is no pitch motion, i.e., $\forall t: \theta_y=0$, and relatively small roll motion, i.e., $\forall t: \theta_x \ll 1$, the simplified equations of motion are given by

$$m(\dot{v}_x - v_y \dot{\theta}_z) - m_s h_s (\theta_x \ddot{\theta}_z + 2\dot{\theta}_x \dot{\theta}_z) = \sum f_x \cos \delta - \sum f_y \sin \delta \quad (18)$$

$$m(\dot{v}_y + v_x \dot{\theta}_z) - m_s h_s (\theta_x \dot{\theta}_z^2 - \ddot{\theta}_x) = \sum f_x \sin \delta + \sum f_y \cos \delta \quad (19)$$

$$\begin{aligned} m_s h_s (\dot{v}_y + v_x \dot{\theta}_z) + (i_{xx} + m_s h_s^2) \ddot{\theta}_x + (i_{zz} - i_{yy} - m_s h_s^2) \theta_x \dot{\theta}_z^2 \\ = -k_x \theta_x - c_x \dot{\theta}_x + m_s h_s g \sin \theta_x \end{aligned} \quad (20)$$

$$\begin{aligned} -m_s h_s \theta_x (\dot{v}_x - v_y \dot{\theta}_z) + (i_{xz} + i_{sz}) \ddot{\theta}_z \\ = \sum l'_x \sin \delta - \sum l'_y \cos \delta + \sum w'_x \cos \delta - \sum w'_y \sin \delta \end{aligned} \quad (21)$$

where l' and w' are defined in (17).

WHEEL DYNAMICS

In addition to the rigid body vehicle degrees of freedom, one rotational degree of freedom was considered for each wheel. Wheel dynamics will be of importance when studying the brake dynamics of the vehicle. Figure 4 shows a free body diagram of a wheel.

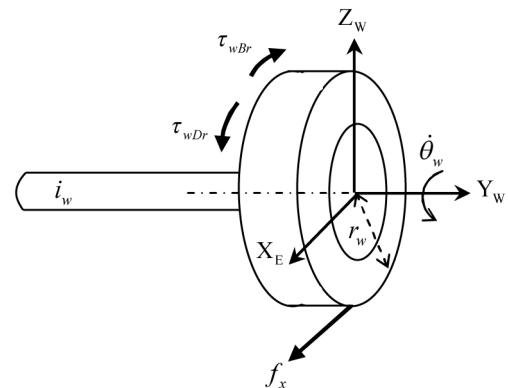


Figure 4. Rotational degree of freedom of wheels

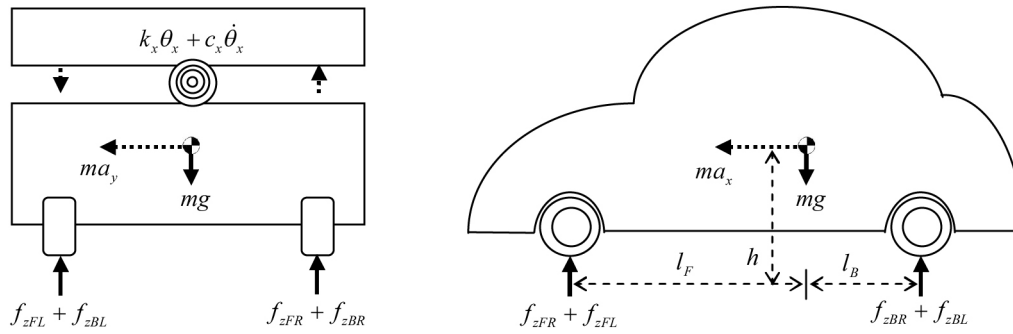


Figure 5. Vertical force schematics

The equation of motion for each wheel is given by:

$$i_w \ddot{\theta}_w = -f_x r_w + \tau_{wDr} - \tau_{wBr} \quad (22)$$

where i_w is the wheel y-axis moment of inertia, f_x is the longitudinal tire force applied on each wheel, $\dot{\theta}_w$ is the wheel rotational velocity, r_w is the effective radius of the wheel, and, τ_{wDr} , τ_{wBr} represent the non-dimensional applied driving and braking moments on each wheel, respectively.

TIRE DYNAMICS

For vehicle dynamics modeling, proper representation of tire dynamics is essential since the functional characteristics of the vehicle (e.g. handling, ride) are significantly influenced by the tire forces. As shown in Equations (18),(19),(20),(21), (22), tire forces and moments are the dominant factors in calculating the generalized forces and play important roles in wheel dynamics. Therefore, analysis of tire dynamics is the basis for the analysis of vehicle dynamics.

Tire models can be categorized as physical, empirical, or a combination of the two, i.e., semi-empirical. Physical tire models focus on representing the interaction of the tire compound and structure mathematically. Empirical tire models curve-fit test data of existing tires to predict how the tire will respond. Semi-empirical models combine the attributes of a physical and empirical model. Most tire models that are currently available are only sufficient for modeling steady-state conditions. Many of these models lack the capabilities of modeling transient handling maneuvers, especially the response of tire friction forces. The steady-state tire models often have computational benefits, but lack the capability to model evasive or high-speed maneuvers, where the tire transient forces and moments become dominant.

Tire longitudinal force f_x and lateral force f_y depend on the vertical load on the tire such that they increase with increasing vertical tire force in pure slip conditions. The significance of tire force estimation in the modeling vehicle dynamics necessitates an accurate approximation of the

vertical load on each tire. Figure 5 shows the vertical forces acting on a vehicle.

Total tire vertical load on each wheel is the summation of static and dynamic weight distributions and are given by:

$$\begin{aligned} f_{zFR} &= \frac{m}{2} \left(\left(\frac{l_B}{l_F + l_B} \right) g - \left(\frac{h_s}{l_F + l_B} \right) a_x - \left(\frac{h_s}{w_R + w_L} \right) a_y - \left(\frac{h_s}{w_R + w_L} \right) \left(\frac{k_s \theta_s + c_s \dot{\theta}_s}{m_s} \right) \right) \\ f_{zFL} &= \frac{m}{2} \left(\left(\frac{l_B}{l_F + l_B} \right) g - \left(\frac{h_s}{l_F + l_B} \right) a_x + \left(\frac{h_s}{w_R + w_L} \right) a_y + \left(\frac{h_s}{w_R + w_L} \right) \left(\frac{k_s \theta_s + c_s \dot{\theta}_s}{m_s} \right) \right) \\ f_{zBR} &= \frac{m}{2} \left(\left(\frac{l_F}{l_F + l_B} \right) g + \left(\frac{h_s}{l_F + l_B} \right) a_x - \left(\frac{h_s}{w_R + w_L} \right) a_y - \left(\frac{h_s}{w_R + w_L} \right) \left(\frac{k_s \theta_s + c_s \dot{\theta}_s}{m_s} \right) \right) \\ f_{zBL} &= \frac{m}{2} \left(\left(\frac{l_F}{l_F + l_B} \right) g + \left(\frac{h_s}{l_F + l_B} \right) a_x + \left(\frac{h_s}{w_R + w_L} \right) a_y + \left(\frac{h_s}{w_R + w_L} \right) \left(\frac{k_s \theta_s + c_s \dot{\theta}_s}{m_s} \right) \right) \end{aligned} \quad (23)$$

The first term in Equation (23) is the component of static weight due to vehicle center of gravity location; the second and third terms are the effects of longitudinal and lateral acceleration, respectively; and the fourth term comes from the suspension dynamics.

Calculation of other significant tire forces, i.e., lateral and longitudinal, depends on adequate knowledge of tire properties and contact patch properties. There are some semi-empirical and/or simplified models that provide tire forces as functions of slip angle, longitudinal slip, and normal load. Before the tire models are introduced, important variables such as longitudinal slip and sideslip angle are defined.

The longitudinal slip κ is defined as the difference between the tire tangential speed and the speed of the axle relative to the road, and is given by

$$\kappa = \begin{cases} 0 & , v_{w,x} = r_w \dot{\theta}_w = 0 \\ \frac{r_w \dot{\theta}_w - v_{w,x}}{\max(r_w \dot{\theta}_w, v_{w,x})} \cdot \text{sgn}(|r_w \dot{\theta}_w| - |v_{w,x}|) & , \text{otherwise.} \end{cases} \quad (24)$$

where the tire velocity for each wheel is approximated as