

Vehicle Dynamics Simulation for Predicting Steering Power-Off Limit Performance

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ABSTRACT

A simulation tool has been developed for predicting steering effort of a vehicle during steering power-assist system failure. The vehicle system is modeled with the inclusion of a system-level vehicle model and a steering system model that are linked together through the steering moment at the kingpin and front road wheel angle. A driver model has also been designed to provide closed-loop steering angular input to make the car follow a certain target path. The simulation model is correlated well with actual vehicle tests under various steering input and lateral acceleration conditions. Also illustrated are some examples of comparison between measured and simulated sensitivity study for selected factors.

INTRODUCTION

Power-assisted steering systems have been invented with the intention of providing an energy source (steering force) that augments the muscular energy of the driver to make a car easier to steer. The most popular types of the energy source include hydraulic power steering and electric power steering. However, the vehicle needs to be properly designed to help prevent extremely heavy steering and holding efforts that might make it difficult to steer in case the power-assisting system fails (referred to as steering power-off hereafter).

Steering power-off performance is more important for low to medium speed steering maneuvers, such as, city block turning, driving through an intersection, cornering a curve, and decelerated turns when exiting highways. A goal of the vehicle design for steering power-off performance is for the vehicle to be able to be manipulated without major deviations from the road or the lane you are driving on, or to be safely stopped to avoid such deviations.

Similar to other aspects of vehicle dynamic performance, steering power-off performance is affected by many vehicle design parameters, such as steering ratio, suspension kinematics and compliance, tire characteristics, as well as vehicle weight and distribution. Most of these parameters are usually fixed

at the early stage of vehicle design and development. Design change to those parameters at a later stage would significantly cost resources and delay the vehicle development project. It is important to develop a prediction tool to help the pre-determining of vehicle parameters and guide improvement activities.

The purpose of this research is to establish a simulation tool for predicting steering effort, a key parameter of vehicle maneuverability, under steering power-off conditions. A simulation model is created by integrating a detailed steering system model into a system-level vehicle model. A PID-controller type of driver model is designed for the closed-loop directional control of the car during simulation study. During simulation, the power-assisting system is turned off. The simulation is then validated against vehicle measurement for various conditions, such as steady-state and transient-state steering input under low and middle speed as well as low and high lateral accelerations. The simulation tool is further used to predict the effect of factor changes. Predicted and measured trends are presented.

VEHICLE AND DRIVER MODEL

The functional diagram of the simulation model is shown in Fig 1. The model includes a vehicle model which calculates vehicle response at a system level, a kingpin kinematics model for calculating steering moment around the kingpin, a driver model for calculating steering wheel correction angle and a steering model involving the mechanism and compliance of the steering system. An external file is used to control the interfacing of those four parts as well as the overall simulation.

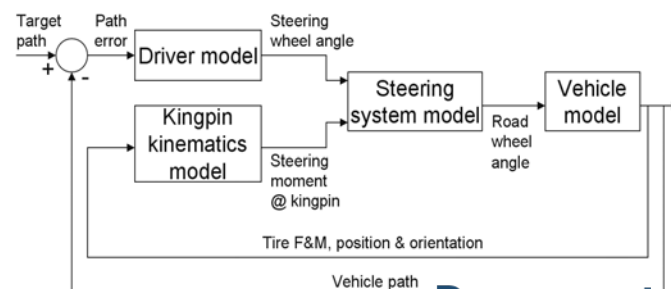


Figure 1 Functional diagram of the simulation model

VEHICLE MODEL - Right-handed coordinate system is used for the vehicle, with X axis pointing forward, Y pointing to the left-hand side of the vehicle and Z axis pointing up, as shown in Fig. 2.

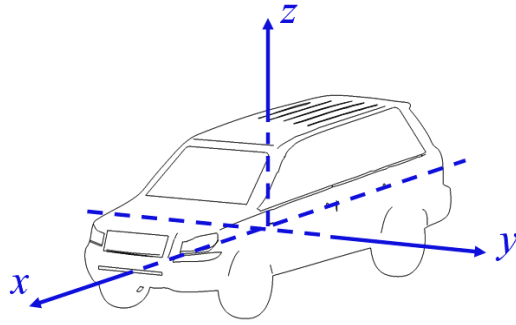


Figure 2 Vehicle coordinate system

The vehicle model is built on a system-level commercial software tool for simulating the dynamic behavior of cars. Since a separate steering model with much more details is used, as described in the preceding section, the steering system of the vehicle model is turned off. The vehicle model is validated with actual vehicle on-road measurement in frequency response of vehicle yaw-rate to steering angle input.

The steering moment around the kingpin can be directly obtained from the vehicle model. However, for higher accuracy, the steering moment resulted from the kingpin kinematics model should be used. The effect of certain factors, such as wheel bump, steer angle, tire slip angle or rate, and even the gyroscopic effect of the spinning wheels can be included. The basic formulas for the steering moment calculation are given in the Appendix.

STEERING SYSTEM MODEL - A steering system usually includes a steering mechanical mechanism and a power-assisting system. In the mechanism of a rack-and-pinion type of steering system, the steering wheel and upper column shaft are connected to a lower column shaft through a universal joint and a rubber coupler. The lower end of the lower column shaft is connected to an input shaft of a control valve through another universal joint. With a torsion bar, the input shaft is allowed to have angular displacement relative to a pinion. The input shaft and the pinion are co-axially assembled to become the key portion which provides feedback of assist demand to the power-assisting system. The pinion is then connected with the steering rack through the rack-pinion gear. The rack is supported by a rack housing which is mounted to the vehicle body through rubber mounts. The mechanism can be represented by a four degree-of-freedom system. The motion of the steering wheel, the lower column shaft, the rack and the rack housing are governed by the following four equations of motion [1, 3].

$$I_{sw} \ddot{\delta}_{sw} = T_{sw} - B_{cc} (\dot{\delta}_{sw} - \dot{\delta}_{ls}) - K_{cc} (\delta_{sw} - \delta_{ls}) \quad (1)$$

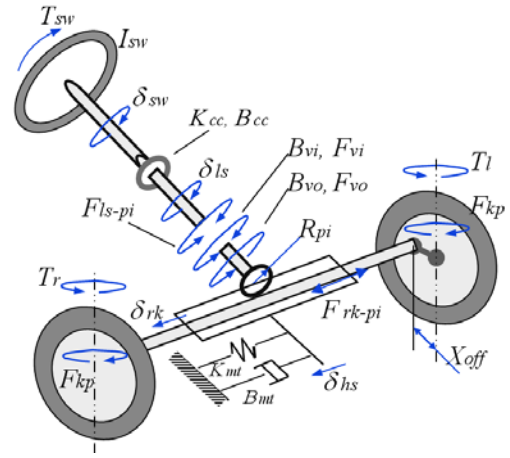


Figure 3 Diagram of a steering system

$$I_{ls} \ddot{\delta}_{ls} = B_{cc} (\dot{\delta}_{sw} - \dot{\delta}_{ls}) + K_{cc} (\delta_{sw} - \delta_{ls}) - F_{ls-pi} - B_{vi} \dot{\delta}_{ls} - F_{vi} \text{sign}(\dot{\delta}_{ls}) \quad (2)$$

$$M'_{rk} \ddot{\delta}_{rk} = -F_{rk-hs} + (I_{pi} / R_{pi}^2) \ddot{\delta}_{hs} + [F_{ls-pi} - B_{vo} \dot{\delta}_{pi} - F_{vo} \text{sign}(\dot{\delta}_{pi})] / R_{pi} + [T_r + T_l - 2F_{kp} \text{sign}(\dot{\delta}_{rk})] / X_{off} \quad (3)$$

$$M'_{hs} \ddot{\delta}_{hs} = F_{rk-hs} + (I_{pi} / R_{pi}^2) \ddot{\delta}_{rk} - [F_{ls-pi} - B_{vo} \dot{\delta}_{pi} - F_{vo} \text{sign}(\dot{\delta}_{pi})] / R_{pi} - (B_{mt} \dot{\delta}_{hs} + K_{mt} \delta_{hs}) \quad (4)$$

where T_{sw} denotes steering wheel torque, δ_{sw} steering wheel angle, I_{sw} the inertia of steering wheel and upper column shaft, δ_{ls} lower column shaft angular displacement, and K_{cc} and B_{cc} represent the spring constant and damping coefficient of steering column rubber coupler, respectively. I_{ls} is the inertia of the lower column shaft. F_{ls-pi} is the action and reaction force between the lower column shaft and the pinion which is a function of torsion bar stiffness, damping and friction between the input shaft and the pinion [1]. For a hydraulic power steering system, the damping coefficient B_{vi} and the friction F_{vi} of the seal between the control valve input shaft and the valve housing should also be considered.

In equation (3), δ_{rk} is the displacement of the rack, δ_{hs} the displacement of the rack housing, δ_{pi} the angular displacement of the pinion, T_r and T_l the right and left steering moment around the kingpin, F_{kp} the pivot friction around the kingpin. M'_{rk} is the equivalent rack mass which includes the effect of the mass of the rack, the tire mass and inertia around the radial axis of the tire, pinion inertia I_{pi} , tire mass, center offset, pinion radius R_{pi} and steering arm X_{off} , i.e., the distance from rack centerline to the kingpin. F_{rk-hs} is the action and reaction force between the rack and the rack housing [1]. For a hydraulic power steering system, the damping coefficient