

A Methodology to Integrate a Nonlinear Shock Absorber Dynamics into a Vehicle Model for System Identification

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ABSTRACT

High fidelity mathematical vehicle models that can accurately capture the dynamics of car suspension system are critical in vehicle dynamics studies. System identification techniques can be employed to determine model type, order and parameters. Such techniques are well developed and usually used on linear models. Unfortunately, shock absorbers have nonlinear characteristics that are non-negligible, especially with regard the vehicle's vertical dynamics. In order to effectively employ system identification techniques on a vehicle, a nonlinear mathematical shock absorber model must be developed and then coupled to the linear vehicle model. Such an approach addresses the nonlinear nature of the shock absorber for system identification purposes. This paper presents an approach to integrate the nonlinear shock absorber model into the vehicle model for system identification.

Three empirical mathematical models are proposed that describe the relationship between shock velocity and shock force. These models are used to predict the force behavior of the shock absorber when the shock absorber velocities ranged from 0 to 1.5m/s, in both compression and extension direction. Empirical shock absorber models can be quickly developed using experimental data and integrated into the larger vehicle model for complete system identification. The shock velocity calculated from the vehicle model is the input to the shock absorber model and the shock force is the feedback to the vehicle model. The influence of nonlinear shock absorber models on system identification for the half

car model was analyzed. The result shows that the type of shock absorber model did not have much influence on vehicle system identification using simulated data generated from the model. Furthermore, the system identification method, integrating the nonlinear shock absorber dynamics model into the vehicle model, works well for the simulated data. Future work will focus on the use of experimental data.

INTRODUCTION

One important aspect of vehicle research is improving car ride quality and handling performance. Ride quality and handling performance are key elements to a vehicle's performance. In recent years, there have been a number of vehicle dynamics studies conducted out to improve handling and ride comfort when developing new cars. Better comfort and handling performance has been achieved through analysis and modification of vehicle dynamics, such as electronic stability control (ESC) and suspension controller design [1, 2, 3].

In general, ESC and suspension control systems operate by comparing vehicle dynamic behavior to a pre-determined dynamic mathematical model of the vehicle. Before designing the controller for such systems, high fidelity mathematical models that accurately capture the dynamics of the car suspension system and predict the vehicle behavior are critical.

In this task, it is important to know the values of different vehicle parameters that affect ride quality and handling performance, such as center of gravity, inertia, suspension

rates and travel limits, tires, track and wheelbase, un-sprung and sprung mass, aerodynamics and so on. But in general it takes too long or costs too much to measure all of these parameters. So a simple and efficient means to determine the value of these parameters is the use of system identification techniques.

Currently, most system identification methods for vehicles use linear assumptions for the development of the quarter car models, half car models, or full car models, which do not capture the nonlinear characteristic of the shock absorber [4, 5, 6, 7, 8, 9, 10, 11, 12]. The nonlinear characteristic of the shock absorber can significantly influence vehicle dynamics, especially in the vertical direction. As a result, the accuracy of the vehicle model is often greatly determined by the accuracy of the shock absorber model.

To address the need for the inclusion of nonlinear shock absorber behavior, Boggs [13, 14] integrated the shock absorber model into the vehicle model and performed the system identification. The shock absorber velocity calculated from the vehicle model was used as an input to the shock absorber model. The shock force calculated from the shock model is used as an input to the vehicle model. This technique can capture the nonlinear characteristic of shock absorber and the system identification result is the frequency response function (FRF) which describes the relationship between road & aerodynamics input and vehicle response. FRFs can also be achieved by testing the vehicle using a dynamic mechanical system in a controlled laboratory environment which often is expensive in terms of time and money.

In the process of designing a vehicle stability controller or a suspension controller, it is important to know the values of vehicle parameters that affect ride quality and handling performance not just the frequency transfer function. It is also difficult to measure these parameters and nonlinearities of the shock absorber. This paper presents a system identification method that can identify the vehicle parameters when considering the nonlinearity of the shock absorber.

In this paper, three empirical mathematical models of the shock absorber are formulated to describe the relationship between shock velocity and shock force. The models can be quickly formulated using experimental data. To validate this approach experimentally, a front shock from a Mazda CX-7 was tested on the test rig that was built using a hydraulic single-post shaker. The shock force and displacement were measured at various frequencies up to 50 Hz and shock velocities up to 1.5 m/s. After that, the empirical shock absorber models were integrated into the half car model for complete system identification. The shock velocities calculated from the half car vehicle model were the inputs to the shock absorber models and the resulting shock forces were then feedback into the vehicle model. The influence of

nonlinear shock absorber models on system identification for the half-car model was analyzed. The results show that the type of shock absorber model did not have much influence on vehicle system identification. Furthermore, the system identification method used to integrate the nonlinear shock absorber dynamics model into the vehicle model works well with simulated data.

SHOCK ABSORBER TEST

To develop a shock absorber model, the front shock absorber on the Mazda CX-7 was tested on a modified MTS single-post shaker (Figure 1). A frame with an integrated load cell was used to mount the shock absorber. This test bench is actuated by a 40kN hydraulic cylinder, with a 10 inch stroke. The system has excellent frequency response of up to 80 Hz and velocities of up to 1.5 m/s.

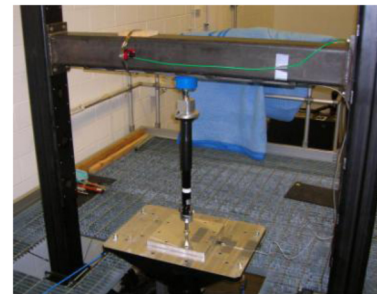


Figure 1. Mazda CX-7 Shock Absorber on a Modified MTS Vibration Table

The test bench is equipped with force, displacement and acceleration transducers. During the test, measurements of shock position, acceleration, and shock absorber force were recorded. The displacement was measured by a LVDT (Linear Variable Differential Transformer) that was installed on the single-post shaker. To address temperature effects on the characteristics of the shock absorber, the shock absorber was instrumented with a thermocouple located on the outside of the main body of the shock absorber. The shock absorber was warmed up to 30°C prior to each test.

During each test, measurements of shock position, acceleration, shock force, and temperature were recorded at 2 kHz using a SoMat eDAQ system, which is a portable stand-alone data acquisition system for testing in a wide range of environments. The eDAQ has built in signal conditioning and the capability to perform a broad range of on-board data processing. The data for all the tests were extracted from the Infield (eDAQ native) software format into LabVIEW for further analysis.

Tests on the single-post shaker were controlled by an MTS 407 controller. The controller allowed for a wide variety of inputs, including sine, triangle and square waves. In this project, sinusoidal excitations with different amplitudes and

frequencies were used to capture the behavior of shock absorber at varying shock velocities. The amplitude of the input signals changed from 5 to 50mm, and the frequency of the input signals changed from 0.1 to 60Hz. These ranges allowed the shock absorber to work in the range of 0 to 1.3m/s, which corresponded to typical velocities encountered by a vehicle suspension due to road irregularities.

The shock absorber data illustrated in Figure 2 shows the cavitation characteristics when the input amplitude is around 50mm with a frequency of 10Hz. The shock absorber data with cavitation was removed to maintain consistent shock behavior. To reduce the influence of noise on the model fit, the force-velocity data from all tests were filtered using a low-pass filter with a cutoff frequency of 50Hz. The plot of the filtered force-velocity data is shown in Figure 3.

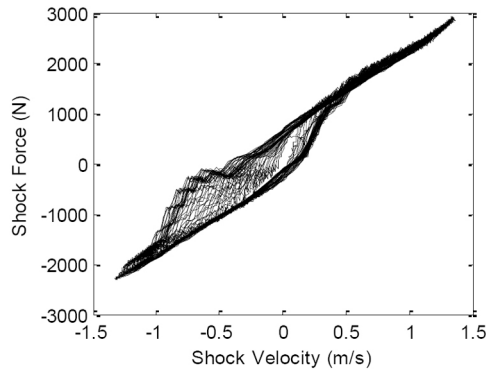


Figure 2. Force-Velocity Plot for Mazda CX-7 Front Shock with Cavitation

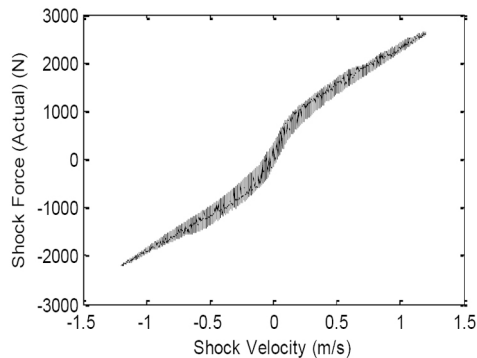


Figure 3. Force-Velocity Plot for Mazda CX-7 Front Shock

EMPIRICAL SHOCK ABSORBER MODEL

The first empirical shock absorber model considered was a linear model. This is generally the simplest model used in simulation-based analysis, as shown in equation (1).

$$F(v) = cv \quad (1)$$

Using experimental data, the linear damping coefficient was identified. The force-velocity plot for the linear shock absorber model is shown in Figure 4. Compared to the experimental data, the RMS error produced by the linear shock absorber model was 145.3N at the velocity range from 0 to 1.3m/s.

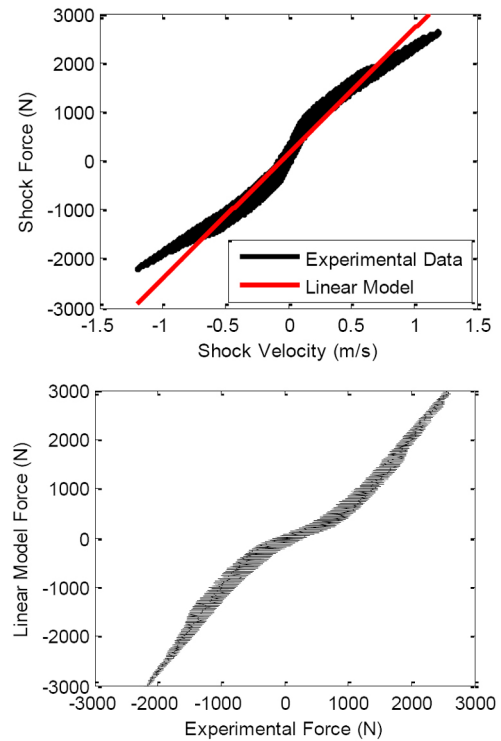


Figure 4. Linear Shock Absorber Model

The second choice for the empirical shock absorber model was a simple power function, as shown in equation (2), for rebound, and equation (3), for compression.

$$F(v) = \begin{cases} q_1 v^{d_1} + l_1, & v \geq 0 \\ q_2 (-v)^{d_2} + l_2, & v < 0 \end{cases} \quad (2)$$

$$F(v) = q_2 (-v)^{d_2} + l_2, \quad v < 0 \quad (3)$$

Compared to the experimental data, the RMS error produced by the power function model was 137.9N at the velocity range from 0 to 1.3m/s. The force-velocity plot for the power function shock absorber model is shown in Figure 5.

The third choice for the empirical shock absorber model was a segmented polynomial model. Different polynomials were