

Application of System Identification for Efficient Suspension Tuning in High-Performance Vehicles: Full-Car Model Study

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ABSTRACT

One popular complement to track testing that successful race teams use to better understand their vehicle's behavior is dynamic shaker rig testing, such as 7-post and 8-post testing. Compared to track testing, rig testing is more repeatable, costs less, and can be conducted around the clock. While rig testing certainly is an attractive option, an extensive number of tests may be required to find the best setup. To make better use of rig test time, more efficient testing methods are needed. One method to expedite rig testing is to use rig test data to perform system identification and generate a model of the experiment, which may then be applied to identify potential gains for further rig study.

This study develops a system identification method for use in rig testing, using data generated from a known physical model. The results show that this method can be used to accurately predict sensor response during an 8-post test for different shock selections. Future work will apply the model developed in this study to a car tested on an 8-post rig and to perform shock selection.

INTRODUCTION

During a 7-post or 8-post rig test, the tires are supported by 4 actuated wheel platforms, or wheelloaders, that simulate inputs from the track surface. Three or four actuators, or aeroloaders, attach to the vehicle's chassis to provide forces which simulate the effect of inertial and aerodynamic forces present during a track test. The vehicle is shaken on the rig to simulate how the car

would respond at a particular track or to characterize the vehicle response to more general waveforms such as sine waves or broadband signals. An example car on an 8-post rig is shown in Figure 1.

The performance of the different setups is compared by calculating a series of performance metrics. These metrics are typically used during rig tests to help summarize how the shock selection affects various aspects of vehicle behavior including mechanical grip, aerodynamics, suspension travel, and suspension loading.



Figure 1. Synergy Racing car on 8-post rig

This sensitivity information is one of the main products of the rig test. The team will use this information to help select or eliminate potential shock packages to try on the race track, and will use the sensitivity information to help make adjustments based on how the vehicle performs on the track.

The main goal of this research is to accurately predict this sensitivity information using simulations, which can augment knowledge gained from rig testing. To create an accurate model for these simulations, we will use rig test data to perform system identification, which will produce a vehicle model.

This paper develops the ID methods that may then be applied to rig test data. The first section develops the equations of motion for a linear 7 DOF vehicle model. This model treats the nonlinear shock model force as an input to a linear dynamic system, allowing us to apply linear system identification techniques. This linear vehicle model may then be coupled with nonlinear dynamic shock models to create simulated rig data sets generated using a known system model.

The methods required to perform the system ID are then discussed and applied to the simulated test data. The results of the ID are compared with the theoretical response to verify that the ID process is working properly.

To verify that the method will produce a model that will accurately simulate the response of the original model, results from the original model and the identified model are compared for multiple shock selections.

BACKGROUND

Before developing the system identification method, background information is provided on 7-post/8-post testing and the use of experimental data to improve simulation studies.

7-POST/8-POST TESTING - Due to the highly competitive nature of racing, very little of the work done in 7-post and 8-post testing has been published. Kelly et al. discusses the use of linear FRF estimates to develop track drivefiles [1]. Kowalczyk uses a linear 7 degree of freedom vehicle model to help identify tradeoffs and determine realistic limitations of tuning [2]. Kasprzak used a half-car model to help find potential gains for further exploration on a 4-post rig with soft springs attached to the chassis to simulate downforce [3]. Each of these papers successfully used linear models to aid in rig testing.

EXPERIMENTAL DATA FOR SIMULATION - To evaluate a given model's accuracy in predicting experimental results, it is common practice to use experimental data to validate the model accuracy. If desired, the model can also be updated to better match the experimental data. This process of model validation and updating can be performed on the component level, the system level, or both depending on the application. Some relevant examples showing trends applying

experimental data to model validation and updating are shown below.

In the automotive industry, vehicle models are quite commonly used early in the development cycle when vehicle data is not available. This makes model updating using vehicle system data impractical, so the focus is often on assembling vehicle models from validated component models. Vilela correlated proving ground data collected for a selection of different proving ground tests with a detailed ADAMS model assembled using validated component models [4]. The correlation was determined to be acceptable, indicating that the model could have been used before the vehicle was manufactured.

Often, vehicle-level test data for an existing vehicle is used to refine a vehicle model, which will be used in future studies. The experimental data is often assumed to be the ideal response for the model to try to achieve, although experimental errors can make this a poor assumption. One basic method to use test data to update models is to first perform a parametric study to determine the sensitivity of modeling error to model parameters. The model parameters can then be adjusted using the sensitivity information to reduce modeling error [5].

A more systematic method for model updating is to apply optimization methods to find parameter values that minimize some measure of the model error. Hu performed model validation and updating on a half-car ride model using proving ground data [6]. The initial guess at the parameters was defined using component tests, which were then updated to better match the proving ground data using optimization of modeling error.

While many studies use proving ground data for model development, there has been an increasing trend to use laboratory test data since the experiments are run in a more controlled environment with higher repeatability [7]. For example, Cheli and Sabbioni performed 4-post testing on a van to identify the unknown stiffness, damping, and inertial parameters [8]. The parameters were optimized to best fit the experimental data using nonlinear optimization. Further experiments were performed to show that there was acceptable correlation between the model and data. The ability of the model to predict the effect of changing parameters was not discussed.

Most model updating studies attempt to match the displacements and/or accelerations of the sprung and unsprung masses. One additional sensor measurement that is important for a rig testing model to predict is the tire normal force. Ziegenmeyer optimized the parameters of a linear quarter-car model to minimize the

prediction error for quarter-car rig test [9]. The model was then used to predict the acceleration of the sprung mass, the unsprung mass and tire normal force with reasonable accuracy.

The above studies all use structural dynamic models to model vehicle behavior, where the models have been constructed using components that have a specific physical meaning, such as masses, springs, dampers, and their geometrical configuration. Many applications do not require accurate estimation of the structural parameters, only a model of the input-output behavior. If this is the case, a more generic dynamic model structure can be used, such as a state-space or transfer function model. These modeling structures have more modeling freedom, allowing them to account for more complex behavior, such as chassis flexible body modes. These models can be identified using generic system identification tools. By using a generic input-output model structure, there is no need for prior knowledge of the geometry of how components in the physical system are arranged or the boundary conditions for their connections. This makes them useful for quickly generating models from test data with minimal model development effort.

This paper will focus on developing a generic linear dynamic model from simulated rig test data that will allow us to predict the sensor response for different drivefiles and shock selections. Prior research by the authors has shown this method to successfully predict the response of a laboratory quarter-car rig [10].

Extending this method from the ideal simulation environment in this work to 8-post rig test data will present nonlinearities which will degrade the performance of this method. Potential sources of nonlinearity include tire separation, coil bind, coil separation, progressive spring rates, bump stops, nonlinear motion ratios, tire and joint friction, and play in the suspension.

Despite these potential nonlinearities, there are several reasons to suggest that the linear model developed here may be a useful approximation of an 8-post rig test. First, linear modeling has already been used to model the response of a rig test with positive results [1-3]. Second, the weight of the vehicle and aerolader downforce will act to preload the tires, reducing the amount of time that the tires will separate from the wheelloaders. Third, suspension joint friction and slop will often be minimal to provide good handling. Finally, the RMS-based response measures commonly used for analysis are gross dynamic measures of the response, so even our model does not match the true response exactly, it is still possible that our RMS trends will match.

Future work will apply this method to actual 8-post rig data to determine if this model can be successfully used to predict the results of 8-post rig tests.

FULL-VEHICLE MODEL

This section develops the equations of motion for a linear full-vehicle model, where the nonlinear shock model force is treated as a feedback input to a linear dynamic vehicle model. This allows us to decouple the nonlinear shock models from the linear vehicle model and apply linear analysis to better understand the effect of shock selection on vehicle performance.

STATE-SPACE MODEL - A diagram for the 7 degree of freedom model used for this study is shown in Figure 2. The coordinate system is chosen about the center of gravity, which lies a distance a behind the front axle. We assume the center of gravity to lie along the vehicle centerline and the front and rear track widths to be equal. The influence of external forces applied to the 3 DOF chassis model by the aeroloaders may be represented as a heave force, roll moment, and pitch moment. At each corner, there is a suspension, which consists of a linear spring and an arbitrary shock force. The tire is modeled as a point mass, linear tire stiffness, and a single point tire input.

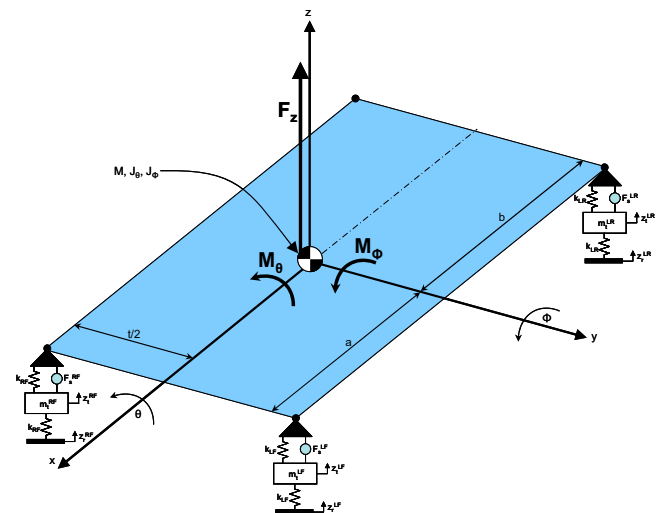


Figure 2. 7 degree of freedom vehicle model

The displacement of the four corners of the chassis, z_{corners} , may be expressed in terms of the chassis heave, roll, and pitch (Φ) by the transformation

$$\begin{bmatrix} z_{corner}^{LF} \\ z_{corner}^{RF} \\ z_{corner}^{LR} \\ z_{corner}^{RR} \end{bmatrix} = \begin{bmatrix} 1 & \frac{t}{2} & -a \\ 1 & -\frac{t}{2} & -a \\ 1 & \frac{t}{2} & b \\ 1 & -\frac{t}{2} & b \end{bmatrix} \begin{bmatrix} z \\ \theta \\ \phi \end{bmatrix} \quad (1)$$

$$z_{corners} = T\Phi$$

Where a and b define the position of the front and rear axles in relation to the chassis center of gravity, while t defines the track. Similarly, the suspension forces applied to the four corners of the chassis, $F_{suspension}^{\Phi}$, may be expressed as an applied heave force, roll moment, and pitch moment ($F_{suspension}^{\Phi}$) using the transformation

$$\begin{bmatrix} F_{suspension}^z \\ M_{suspension}^{\theta} \\ M_{suspension}^{\phi} \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 \\ \frac{t}{2} & -\frac{t}{2} & \frac{t}{2} & -\frac{t}{2} \\ -a & -a & b & b \end{bmatrix} \begin{bmatrix} F_{LF} \\ F_{RF} \\ F_{LR} \\ F_{RR} \end{bmatrix} \quad (2)$$

$$F_{suspension}^{\Phi} = T^T F_{suspension}$$

Summation of forces and moments on the chassis yields

$$J_C \ddot{\Phi} = F_{suspension}^{\Phi} + F_A \quad (3)$$

where I_C is a 3x3 diagonal matrix containing the chassis inertial properties, while F_A is a 3x1 vector of the forces and moments applied to the chassis to simulate aerodynamic and inertial loading. Summation of forces on the four tires yields

$$M_t \ddot{z}_t = -F_{suspension} + F_{tires} \quad (4)$$

where M_t is a 4x4 diagonal matrix containing the four tire masses and z_t is the tire vertical displacement. The suspension and tire forces, $F_{suspension}$ and F_{tires} , are defined to be positive in compression and may be written as

$$F_{suspension} = K_s (z_t - z_{corners}) + F_s \quad (5)$$

$$F_{tires} = K_t (z_r - z_t) \quad (6)$$

Substituting Equations 1-4 into Equations 5-6 yields the equations of motion

$$\begin{bmatrix} J_C & 0_{3 \times 4} \\ 0_{4 \times 3} & M_t \end{bmatrix} \begin{bmatrix} \ddot{\Phi} \\ \ddot{z}_t \end{bmatrix} + \begin{bmatrix} T^T K_s T & -T^T K_s \\ -K_s T & K_s + K_t \end{bmatrix} \begin{bmatrix} \Phi \\ z_t \end{bmatrix} = \begin{bmatrix} I_3 & 0_{3 \times 4} & T^T \\ 0_{4 \times 3} & K_t & -I_4 \end{bmatrix} \begin{bmatrix} F_A \\ z_r \\ F_s \end{bmatrix} \quad (7)$$

$$M \ddot{Z} + KZ = Lu$$

The equations of motion can be rewritten in state-space form as

$$\begin{aligned} x_1 &= Z, \quad x_2 = \dot{Z} \\ \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} &= \begin{bmatrix} 0_{7 \times 7} & I \\ -M^{-1}K & 0_{7 \times 7} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ M^{-1}L \end{bmatrix} u \\ \dot{x} &= Ax + Bu \end{aligned} \quad (8)$$

In addition to defining the dynamic equations, sensor response may be defined by an output equation of the form

$$y = Cx + Du \quad (9)$$

Output equations are formed for shock velocity, tire force, ride height, suspension deflection, chassis acceleration, wheel acceleration, and spring forces.

The model parameters used are shown in Table 1. The sprung and unsprung masses correspond to a 3000 lb car with 85% sprung mass, and 45% of the unsprung mass in the front. The roll and pitch inertia are 40% and 60% of the track and wheelbase, respectively. The suspension and tire stiffnesses are within the range of typical values. While these parameters do not necessarily reflect those of a NASCAR Cup car, they do describe a vehicle whose behavior is qualitatively similar, making it sufficient for ID method development.

Table 1. Model Parameters

Track Width	70 in
CG-Front Axle	60 in
CG-Rear Axle	50 in
Sprung Mass	2550 lbm
Roll Inertia	1.5E+06 lbm-in ²
Pitch Inertia	5.5E+06 lbm-in ²
Front Unsprung Mass	101 lbm
Rear Unsprung Mass	124 lbm
Front Spring Rate	1280 lbf/in
Rear Spring Rate	588 lbf/in
Tire Stiffness	2000 lbf/in
Linear Damper	30 lbf/(in/s)

FREQUENCY RESPONSE - The general behavior for the linear portion of the vehicle model may be summarized by the frequency response. These plots provide insight on system behavior, which can be applied during the identification process. The frequency response may be calculated by taking the Laplace transformation of Equation 8 and substituting it into Equation 9

$$\begin{aligned} y(s) &= [C(sI - A)^{-1}B + D]u(s) \\ &= H(s)u(s) \\ &= H_{rig}(s)u_{rig}(s) + H_{shocks}(s)u_{shocks}(s) \end{aligned} \quad (10)$$

where the input vector was partitioned between inputs due to the rig actuators and inputs due to the shock forces in the last step.

For brevity we only consider the shock velocity outputs, but the trends observed here may be applied to the remaining input-output pairs. The frequency response for all four shock velocities and for three different inputs is shown in Figure 3.

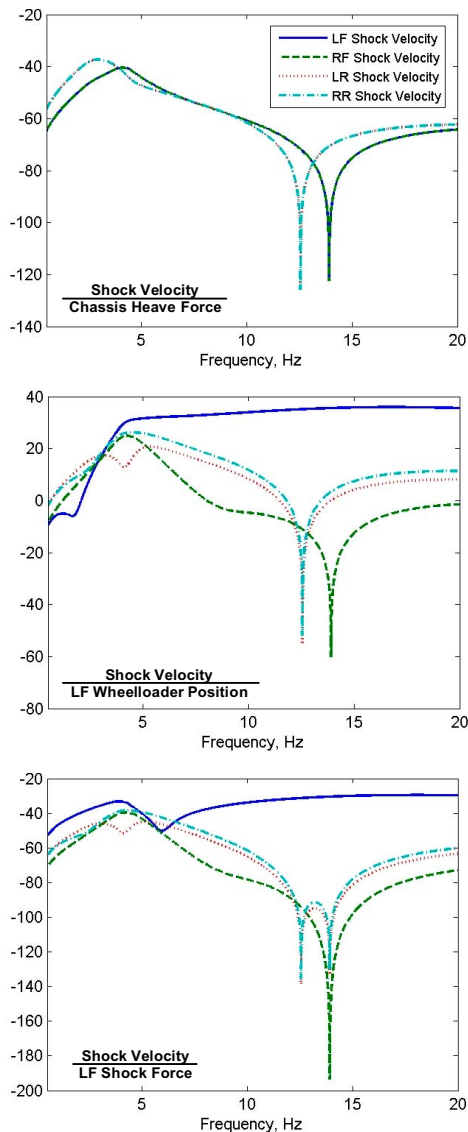


Figure 3. Shock velocity FRFs

For the chassis force input, 1000 lbs chassis heave force gives 10 in/s near resonance, and 1 in/s below 1 Hz and above 9 Hz. Further chassis velocity can be attained by aerolader force by applying roll and pitch moments.

The left front wheelloader input creates about 1 in/s shock velocity near DC and 30 in/s shock velocity near

resonance for a 1 inch left front wheelloader input. The left front shock velocity frequency response maintains a high level at higher frequencies due to the proximity of the wheelloader to the shock. All other shock velocities drop about 20 dB above 15 Hz due to the filtering effect of the chassis.

The left front external shock force produces about 1 in/s left front shock velocity for every 100 lbs. The other shock velocities have similar levels at below 8Hz, but the level drops 40 dB at higher frequencies due to the filtering effects of the chassis.

These observations can be used to assist our identification efforts. To successfully perform frequency response estimation using experimental data, we need to have significant output energy at all frequencies of interest relative to the noise level. This will provide a good signal-to-noise ratio, or coherence near 1 for a linear system. To design excitation to provide good output signal levels, the system behavior should be considered.

Notice that the chassis heave force is effective for exciting response below 5 Hz, but the response drops off at higher frequencies. This trend is also true for chassis roll and pitch moments. Conversely, the wheelloaders are ineffective at exciting outputs below 3 Hz but are useful for higher frequency excitation. Also, the most effective excitation at higher frequency will be the wheelloader closest to the sensor of interest. This suggests the need for excitation from both the wheelloader(s) closest to the sensor and the aeroloaders if we want a good signal-to-noise ratios from 1 to 20 Hz.

While it is ideal to get good signal-to-noise ratios in all output signals at all frequencies, this is often impractical due to the system's behavior, physical input limitations, and vehicle response limitations. If the output level of a particular sensor due to a particular input will always be low over a range of frequencies for the desired excitation signals to be used with the model, the poor frequency response estimation for that input-output pair in that frequency range may be replaced by a small or zero frequency response estimation with minimal reduction in model accuracy from the ideal model. Therefore, it is not critical to accurately estimate the FRF in frequency ranges that are known to have low response levels.

Notice that accurate frequency response estimation for aerolader inputs is only critical at lower frequencies. At higher frequencies, the response is attenuated, so a poor frequency response estimate could be replaced by a small or zero frequency response to reduce modeling error.