

Comparison of the Performance of 7-Post and 8-Post Dynamic Shaker Rigs for Vehicle Dynamics Studies

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ABSTRACT

This paper documents a simple theoretical analysis and an experimental performance comparison of the advantages of an 8-post shaker rig relative to a conventional 7-post shaker rig. A simple static model describing the chassis roll and warp characteristics is first presented to illustrate the differences between 7-post and 8-post configurations, and the conditions where an additional aerolader provides an advantage. Using a late model NASCAR Sprint Cup car, a series of experimental tests were conducted with the 8-post shaker rig at the Virginia Institute for Performance Engineering and Research (VIPER) facility in both 7-post and 8-post configurations. Experimental results confirm the hypothesis that an 8-post configuration is able to more accurately reproduce target motions of the chassis and suspension when those motions include a chassis warp condition.

INTRODUCTION

Successful race teams look for every possible advantage over their competition to reduce lap times and win the race. While track testing is ultimately the most valid testing method available, the amount of time and resources it takes to find the ideal setup alone is prohibitive. The most common alternative for evaluating vehicle behavior is a 7-post shaker rig. Compared to track testing, rig testing is more repeatable, costs less, and can be conducted around the clock [1, 2, 3, 4].

While shaker rig testing is an attractive and common option, race teams must consider the potential problems that can have a negative impact on the fidelity of the test

results. A conventional 7-post shaker rig applies aerolader forces to the chassis at only three points in order to simulate the aerodynamic and inertial loads that excite the chassis on the track. Assuming that the chassis is rigid, three aeroloaders are sufficient to control chassis heave, pitch, and roll motions. In open-wheel racing, where 7-post rig testing originated, the chassis is much stiffer than the suspension, and this assumption is justified. Although it is often desirable for the chassis to be relatively rigid in torsion [5], this is not usually the case. When this condition is violated, and the chassis torsional stiffness is closer to the suspension stiffness, the ability of only three aeroloaders to accurately reproduce chassis and suspension motions on a 7-post rig is degraded. The use of four aeroloaders, as in an 8-post shaker rig, provides the additional degree of freedom that is required to control the rigid body modes, i.e. pitch, roll, and heave, as well as the warp mode of the chassis.

STATIC WARP MODEL

To illustrate the theoretical differences between 7-post and 8-post shaker rigs, we first present a simple static model for chassis roll and warp. This model is very similar to the model developed by Deakin et. al. [6]. The aeroloaders are designed to apply forces that represent the aerodynamic and inertial forces that are distributed across the chassis on the racetrack.

The forces generated by aeroloaders and applied to the chassis on either a 7-post or an 8-post shaker rig may be decomposed into dynamic modes as a heave force, a pitch moment, a roll moment, and a warping moment. For this analysis, we are not interested in the heave

force or pitch moment, therefore, the static roll and warp behavior may be approximated by a simple torsional spring model, as shown in Figure 1. The reason the roll and warp modes are of interest is because they are intimately coupled on a 7-post rig. In fact, a 7-post rig must induce a roll condition in order to create warp, whereas an 8-post rig can create a pure warp without any roll, or a pure roll without any warp.

The front and rear net roll stiffness, created from springs, antirollbars, tires, and suspension geometry, are represented by two torsional springs, k_F and k_R respectively. The angular deflection of these torsional springs corresponds to the front and rear chassis roll angles, θ_F and θ_R respectively, while the moment in the springs corresponds to the front and rear rolling moments.

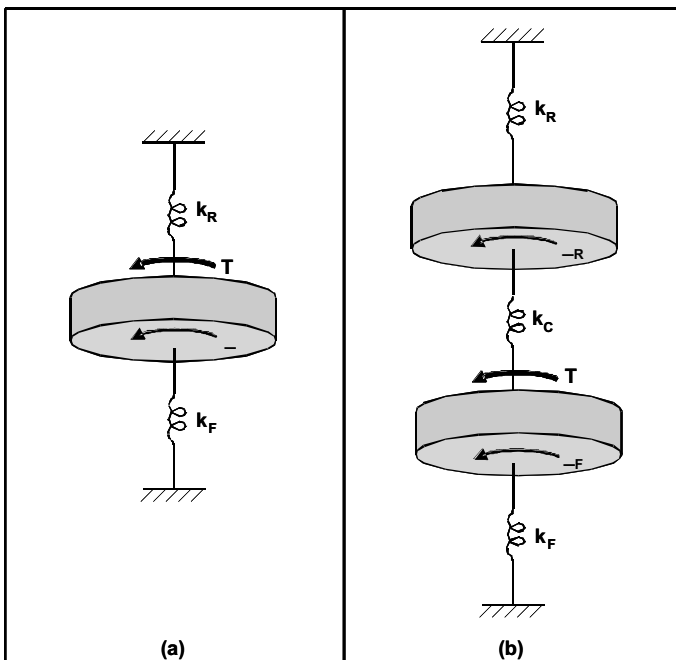


Figure 1. Static Model: (a) Rigid chassis, (b) Compliant chassis

The chassis may either be modeled as rigid or compliant. The rigid chassis model ($\theta = \theta_F = \theta_R$) is valid when the chassis stiffness (k_C) is high relative to the axle stiffness or when the forces applied to the chassis do not induce much chassis warp. The chassis warp is defined to be the relative twist from front to rear:

$$\theta_{warp} = \theta_F - \theta_R. \quad (1)$$

Figure 1a depicts the condition where the chassis is rigid. In this case, the front of the chassis rolls the same amount as the rear of the chassis when any roll moment is applied by the aeroloaders. If the chassis is rigid, a fourth aeroloader is redundant so three aeroloaders are sufficient to control three modes. Figure 1b depicts the

compliant chassis case. When the chassis is compliant, the front and rear of the chassis have independent roll angles when a torsional moment is induced by the aeroloaders or through the suspension itself.

For a 7-post rig with static excitation from two aeroloaders in front and one aeroloader in the rear, and a compliant chassis as shown in Figure 1b, the rear of the chassis will always roll less than the front of the chassis by the relationship:

$$\frac{\theta_R}{\theta_F} = \left(\frac{1}{1 + \frac{k_R}{k_C}} \right) \quad (2)$$

A 7-post configuration must produce static roll angles and rolling moments as defined by this ratio. This relationship is plotted in Figure 2.

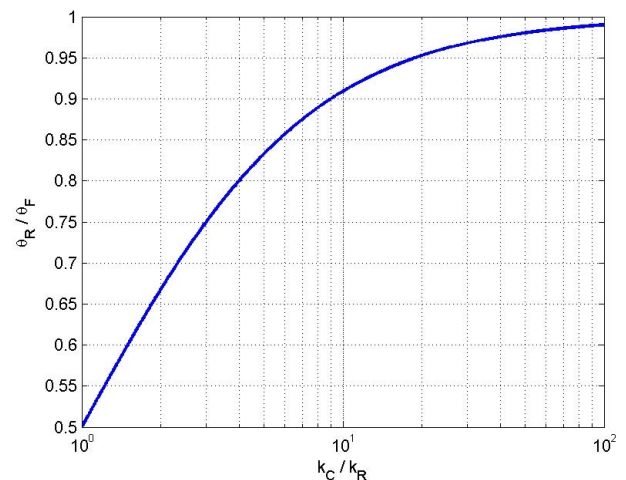


Figure 2. Static ratio of rear to front roll angle

Since the chassis compliance acts in series with the rear roll stiffness in Figure 1b, less of the rolling moment will be distributed to the rear axle than if the chassis did not warp. The rolling moment distribution observed on the rig compared to the ideal rolling moment distribution may be calculated using the relationship:

$$\frac{RMD}{RMD^*} = \frac{\left(\frac{k_C}{k_F + k_R} \right) + \left(\frac{k_R}{k_F + k_R} \right)}{\left(\frac{k_C}{k_F + k_R} \right) + \left(\frac{k_R}{k_F + k_R} \right) \left[1 - \left(\frac{k_R}{k_F + k_R} \right) \right]}, \quad (3)$$

where RMD and RMD* are the percentages of the roll moment distributed to the front axle for the compliant and rigid chassis models, respectively. This relationship is plotted in Figure 3. As this figure shows, the difference between compliant and rigid roll moment distribution decreases as the chassis stiffness increases relative to the total roll stiffness and the difference is worse as the roll stiffness distribution is biased towards the rear.