

Test Rig for Characterization of Automotive Suspension Systems

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ABSTRACT

A test rig (named RuotaVia) is presented for the in-door testing of road vehicle suspension systems. It is basically a drum (Φ 2.6 m) providing a running surface for testing the dynamic performance of a single tire or suspension system (corner). The suspension system is instrumented for the measurement of the forces and the moments acting at each joint connecting the suspension to the car body. A new 6 axis load cell was designed and manufactured for this purpose. The accelerations in various locations of the system (wheel carrier, suspension arms, ...) and the wheel centre displacements in the longitudinal and vertical directions are monitored. The effect of the dynamic interaction between the test rig and the suspension system is discussed in the paper. The direct measurement of the forces and moments at the suspension-chassis joints is still an effective way for understanding the vibration and harshness (VH) suspension performances. The proposed experimental set-up allows to perform accurate, repeatable and consistent measurements of the VH suspension behavior in the frequency range 0-70 Hz.

INTRODUCTION

The vibration of the vehicle chassis depends either on the tire, on the suspension geometry, on the springs stiffness, on the damper characteristic, on the suspension structural features. Due to these many factors influencing the chassis vibrations, it is rather difficult to model accurately the suspension VH behavior [1]. The generally unreliable responses of the suspension mathematical models are often due to inadequate modeling of the complex non-linear dynamic behavior of both the tire and the bushings. Actually many models which simulate the tire running over an obstacle have been developed [2, 3, 4, 5, 6]. The mathematical models also require the experimental identification of some parameters over a wide range of operating conditions [7, 8, 9]. Some results related to tire models performance can be found in [10]. In [11] a comparison between the measured and the predicted forces is

performed. In particular, tire models show good performance in predicting the tire loads while passing over an obstacle (cleat) but when the whole suspension (corner) model is considered the results are quite poor. The lack of accuracy is mainly due to the non linear characteristics of the elastomeric bushings which is in general not adequately modeled. In [12] both modeling and experimental tests are considered. Experimental tests are still necessary to obtain a reliable estimation of the loads acting at suspension/chassis joints for comfort and durability purposes. The system presented in the paper allows the measurement of these loads up to 70 Hz. The load sensors are positioned between the bushings and the fixed structure (which represents the car chassis). In order to reach the adequate compliance level a new 6-axis (forces and moments) sensor has been designed [13]. The analysis in the time and frequency domain of the forces transmitted to the car chassis can be performed by means of the RuotaVia test rig in order to characterize the dynamics of the tire/suspension system. A number of VH relevant performance indices, referring to a single passage over a cleat, at varying speed, have been defined, e.g. suspension system critical speeds, maximum and minimum peak values of the bushing loadings,... In this paper the RuotaVia test rig is described and some considerations on the dynamic interaction between the test rig and the suspension system are reported. The instrumentation used in tests is shown and the method for the data analysis is described.

RUOTAVIA TEST RIG

The RuotaVia test facility (Fig. 1) is located at the Laboratory for the Safety of Transport of the Politecnico di Milano [14]. Cornering, braking and passing over obstacle tests can be performed on either full vehicles or vehicle sub-systems (e.g. tires, corners). Both steady and transient state measurements can be performed. Fig. 1 shows the test rig with the tire/suspension system (corner) connected to the frame which substitutes the car body. The RuotaVia test rig is basically a steel drum (2,6 m diameter) providing a running surface for the wheel. The outer ring, made in one single piece of forged steel, is connected to the

central hub via two side disks. In order to increase the lateral stiffness of the drum, sixteen spokes are welded to the central hub and to the two side disks.



Fig. 1, The RuotaVia test rig.

The spokes are not welded to the outer ring to avoid a lobe shape deformation of the running surface due to the centrifugal force during high speed testing. The driveline is composed by a controlled electric asynchronous four poles motor (160 kVA), and by a two ratio gearbox. The rotational speed of the drum is measured by means of a 4096 points encoder. The maximum tangential speed is about 440 km/h. A concrete foundation (140 tons) supports the RuotaVia drum and the driveline. The whole structure has been designed in order to achieve a first natural frequency higher than 100 Hz. In the suspension testing a frame substitutes the car body. The frame is equipped with a sled which allows the vertical positioning of the suspension system. The sled carries a steel plate which is designed to accommodate different suspensions layouts (e.g. double wish-bone, McPherson, ...). In this way it is possible to position easily the suspension system and to give the correct static load (monitored via high precision load cell). The objective is the VH characterization of the suspension system and so the sled is locked (the low frequency modes of the car body can be neglected). It is important that the whole frame has no natural frequencies in the frequency range of interest in order to avoid parasitic vibrations.

SUSPENSION-FRAME INTERACTION

For the VH characterization of the tire/suspension system (corner), the suspension is linked to a fixed frame (Fig. 1). The dynamics of the whole supporting frame (structure + sled + 6-axis load sensors) must be similar to the chassis dynamics in order to obtain accurate data from the test rig. In fact, if the frame interacts with the suspension system, the measured loads can be a poor estimation of the real ones. A number of simple 1 DOF linear models can be used to describe such type of interaction. Two different 1 DOF mass-spring-damper models of the suspension system have been considered (one for the vertical vibrations, and one for the horizontal ones). Two different 1 DOF mass-spring models of the frame have been used to capture the dynamics of the frame at the joints in the frequency range of interest 0-70 Hz (one model has been developed for the vertical vibrations, and the other one for the horizontal ones). The parameters of the two frame models have been identified by considering the experimental accelerances [15, 16]. The identification procedure was based on the minimization of the mean square error between the measured and the computed accelerances. The accelerance is defined as the ratio between the acceleration and the applied force at a defined point in frequency domain [15, 16]. The accelerances (along three orthogonal directions) have been obtained by exciting the frame with a hammer (equipped with a load cell) and by measuring the response in terms of accelerations. The suspension and supporting frame models are coupled to describe the test rig dynamics and to show the effects of the interaction.

VERTICAL FORCES

Along the vertical direction the force is mainly transmitted to the chassis through the spring-damper group. Fig. 2 refers to the measurement of the accelerance at the upper strut mount. A 1 DOF mass-spring model has been identified (see Fig. 3 and Tab. 1) in order to reproduce the supporting frame dynamics in the frequency range 0-70 Hz (range of interest).

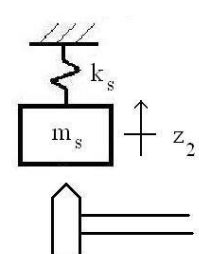
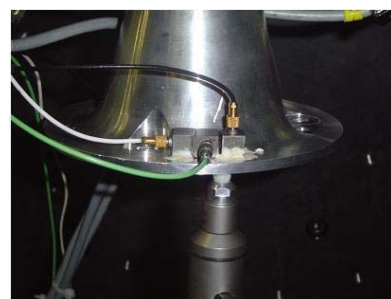


Fig. 2, Measurement of accelerance at the upper mount point. Simple 1 DOF model.

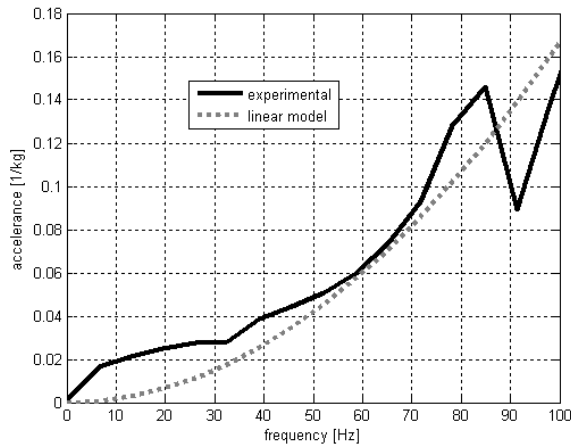


Fig. 3, Identification of the 1 DOF model of the supporting frame (vertical direction, see Fig. 2).

The 1 DOF mass-spring model of the frame is coupled to another 1 DOF simple linear model of the suspension, such a model is shown in Fig. 4 (model parameters in Tab. 2). This coupling allows to obtain the dynamic model of the suspension+frame system (= test rig) along the vertical direction (see Fig. 4).

Tab. 1 Frame Model Parameters	
m_s , frame mass associated with the vertical motion [kg]	1
k_s , frame stiffness associated with the vertical motion [N/m]	$2.4 \cdot 10^6$

Tab. 2 Suspension Model Parameters	
m , unsprung mass [kg]	40
k_p , the tire vertical stiffness [N/m]	250000
k , suspension stiffness [N/m]	30000
r , suspension damping [Ns/m]	1000

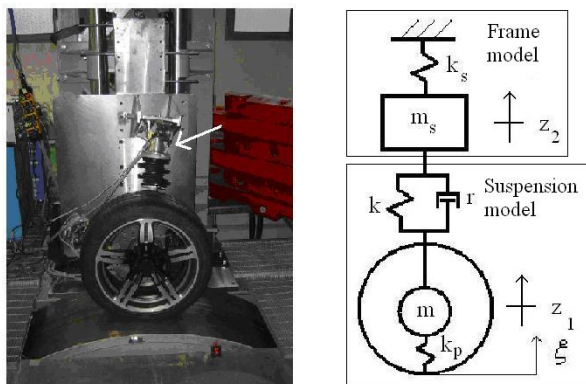


Fig. 4, Test rig and its model (vertical direction), the location of the 6-axis load cell is highlighted.

The force measured by the load cell (i.e. the force applied to mass m_s in Fig. 4) is the force transmitted by the spring-damper and in the model (Fig. 4) is represented by

$$F = r(\dot{z}_1 - \dot{z}_2) + k(z_1 - z_2) \quad (\text{eq. 1})$$

The equations of motion of the suspension system along the vertical direction can be expressed as

$$M\ddot{\vec{x}} + R\dot{\vec{x}} + K\vec{x} = \vec{F}_x \quad (\text{eq. 2})$$

where $\vec{x} = [z_1, z_2]^T$ is the two component vector of the independent variables (the position of the two masses). The mass, damping, stiffness matrixes and the forcing vector of eq. 2 can be simply computed and are not reported in the paper. By using the Fourier transform, the frequency response function (FRF) between the road input (ξ in Fig. 4) and the force F (eq. 1) can be computed.

$$FRF = \frac{F(\omega)}{\xi(\omega)} \quad (\text{eq. 3})$$

In Fig. 5 the mentioned FRF is plotted by considering two different frame stiffnesses k_s . Two frame stiffness values are considered, the identified one k_s (see Tab. 1) and $10 \cdot k_s$. As expected, along the vertical direction the force is mainly determined by the vertical dynamics of the unsprung mass. The resonant frequency is 10-15 Hz. The frame stiffness has limited influence on the response of the system in the low frequency range (in Fig. 5 the two curves are almost superimposed in the frequency range 0-70 Hz).

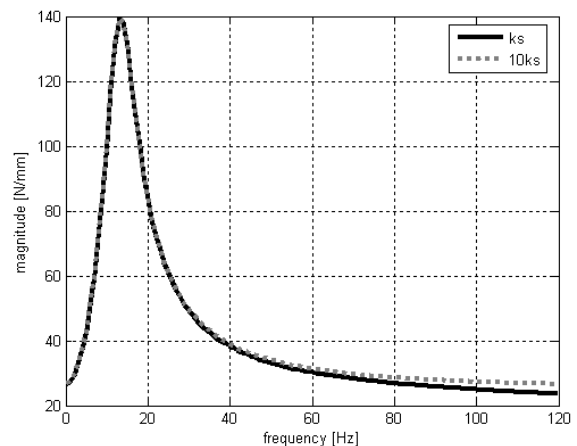


Fig. 5, FRF between the road input and the force F (eq. 1) acting in vertical direction (which represents the force measured by the load cell in Fig. 4).