

A Method for Vibration and Harshness Analysis Based on Indoor Testing of Automotive Suspension Systems

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ABSTRACT

The paper presents a method for the indoor testing of road vehicle suspension systems. A suspension is positioned on a rotating drum which is located in the Laboratory for the Safety of Transport at Politecnico di Milano. Special six-axis load cells have been designed and used for measuring the forces/moments acting at each suspension-chassis joints. The forces/moments, wheel accelerations, displacements are measured up to 100 Hz. Two different types of test can be performed. The tire/wheel unbalance effect on the suspension system behavior (Vibration and Harshness, VH) has been analyzed by testing the suspension system from zero to the vehicle maximum speed on a flat surface and by monitoring the forces transmitted to the chassis. In the second kind of test, the suspension system has been excited as the wheel passes over different cleats fixed on the drum. By analyzing the forces and the acceleration measured on the suspension system, a set of performance indexes has been derived from the analysis of the data both in time and frequency domain. A simple mathematical model of the tire/suspension system has been derived and validated to analyze the relative effect of the road irregularity and of the tire/wheel unbalance on an actual suspension dynamic behavior. It has been shown, that the direct measurement of the forces and moments at the suspension-chassis joints is an effective way for understanding the vibration and harshness suspension performances.

INTRODUCTION

The vibration of the vehicle chassis depends on the tire, on the suspension geometry, on the stiffness of the springs, on

the damper setting, on the suspension structural features and on the geometric tolerances of the suspension components. Due to these many factors influencing the chassis vibrations and harshness (VH), it is rather difficult to model the suspension vibrational behavior [17]. Experimental measurements are necessary to obtain a reliable estimation of the loads acting at suspension/chassis joining points for comfort and durability purposes. The main research activity described in the paper is related to the measurement of VH performances up to 100 Hz in order to include the tire dynamics and the effect of the tire/wheel unbalance. A car suspension is positioned on a rotating drum which is located in the Laboratory for the Safety of Transport at Politecnico di Milano. Special six-axis load cells have been designed and used for measuring the forces/moments acting at each suspension-chassis joints. 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 (force and moments) sensor has been designed because no commercially available sensors could be used [4]. The analysis in the time and frequency domains of the measured forces transmitted to the car chassis can be performed by the RuotaVia test rig in order to characterize the dynamics of the tire/ suspension system.

Two different types of test have been performed. The tire/wheel unbalance effect on the suspension system behavior has been analyzed by testing the suspension system from zero to the vehicle maximum speed on a flat surface and by monitoring the forces transmitted to the chassis, in particular the harmonic components at 1x and 2x the rotational speed of the wheel. In the second type of test, the suspension system

has been excited as the wheel passes over different cleats fixed on the drum. Transient wheel motions have been recorded. By analyzing the forces and the acceleration measured on the suspension system, a set of performance indices has been derived from the analysis of the data both in time and frequency domain. The influence of the running conditions and of the different suspension settings on the relevant performance indexes related to the vibration behavior of the tire/suspension system have been assessed by introducing a simple model of the tire, suspension system. In the paper a number of comfort indexes have been derived to be considered for the suspension design process (useful for comparing different design solutions). The paper is organized as follows. The first section presents the main technical specifications of the RuotaVia test rig while the numerical and experimental analysis of the test rig compliance are reported in the second one. A complete description of the instrumentation properties and of the measurement method of the VH suspension characteristics is presented. In particular, two type of tests and analysis that can be performed on the suspension system are described. In the last section, a simple validated mathematical model of the tire/suspension system has been used to show the relative effect of the road irregularity and of the tire/wheel unbalance on the suspension dynamic behavior.

TEST RIG

The RuotaVia test facility (Fig. 1) is located at the Laboratory for the Safety of Transport of the Politecnico di Milano [1, 2, 3]. Both steady and transient state measurements on either full vehicles or vehicle sub-systems (single suspension system or single tire) can be performed. Fig. 1.a shows a suspension system under test connected to the carrying frame above the drum. A brief description of the main technical features of the RuotaVia test rig follows. The test facility is composed basically by a steel drum (2,6 m diameter) providing a running contact surface for vehicle wheels. The drum components have been studied through optimization techniques in order to obtain high stiffness properties along the radial and lateral direction with a limited mass. The central hub of the drum is connected via sixteen spokes and two side disks to the outer ring. The radial spokes are welded both to the hub and to the two side disks, but not to the outer ring (see Fig. 1.b). The outer ring is made from forged steel in one single piece. The sixteen radial plates were introduced to increase the lateral stiffness of the drum. This allows to avoid the undulation of the ring (working surface) during high speed testing (see Sect. FLAT SURFACE TEST). By means of two dial test indicators, the maximum radial tolerance of the drum has been measured at the centre of the running surface. This value is smaller than 0.04 mm. A limited value of the surface unevenness is fundamental for the characterization of the tire/suspension vibration and harshness behavior (see Sect. FLAT SURFACE TEST). The driveline is composed by a controlled electric asynchronous

four poles motor (160 kVA), and by a two ratio gearbox. They are constrained to a rigid frame (Fig. 1.c). The rotational speed of the drum is measured by means of a 4096 points encoder. The minimum eigenfrequency of the drum (relevant for dynamic measurements) is 96 Hz and the maximum tangential speed is about 440 km/h. The whole structure has been designed in order to achieve a first natural frequency higher than 100 Hz.

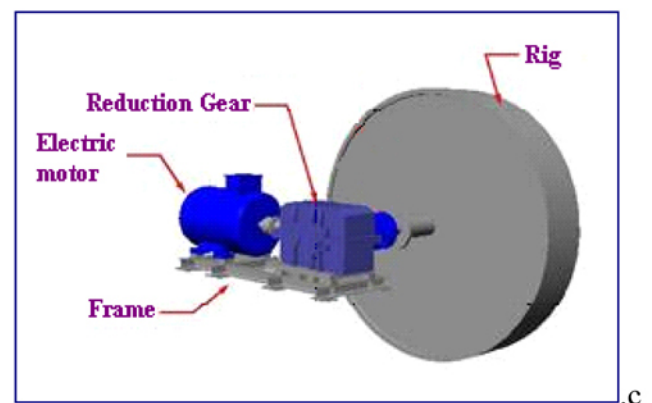
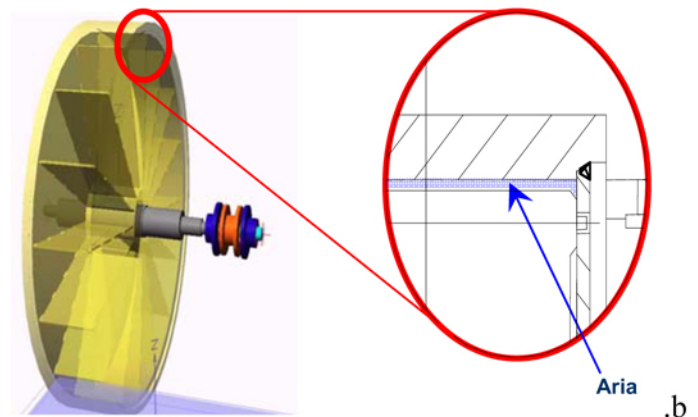
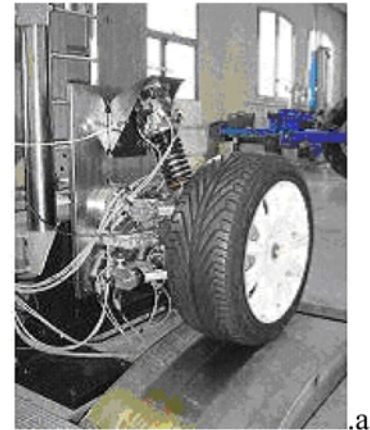


Fig. 1. RuotaVia test rig. A tire/suspension system on the rotating drum (.a) and the main components of the drum (.b) and the test rig driveline (.c)

A special connection structure has been designed (Fig. 2) for testing vehicle suspensions. The superstructure carries a rigid steel plate free to move in the vertical direction. The suspension under test is connected to the plate superstructure as shown in Fig. 1.a. The steel plate is designed to accommodate different suspensions layouts (e.g. double wish-bone, M-link, ...) and it is possible to position easily the suspension system and to apply the correct static load (monitored via high precision load cell). When the suspension is properly settled in the static position, the vertical movement of the superstructure sled is locked.

SUSPENSION FRAME COMPLIANCE

The connection structure have to be designed in order to avoid parasitic vibrations due to the structure natural frequencies which affect the measurement. Moreover, the dynamics of the whole supporting frame (structure + rigid plate + 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 [11]. In the early stage, a FE model has been used to predict the dynamic performance of the connection structure (Fig. 2.a). Then a validation of the theoretical results has been conducted by measuring the acceleration (or inertance) of the supporting frame at each suspension connection joint (Fig. 2.b). The acceleration is defined as the ratio between the acceleration $a(\omega)$ and the applied force $F(\omega)$ at a defined point in the frequency domain [14, 15].

$$A(\omega) = \frac{a(\omega)}{F(\omega)} = -\omega^2 \alpha(\omega) = 4\pi^2 \frac{f^2}{k} \quad (eq.1)$$

with

$$x(t) = Xe^{i\omega t} \Rightarrow a(t) = -\omega^2 Xe^{i\omega t}$$

$x(t)$ is the sinusoidal input (displacement) of the system (with amplitude equal to X), ω is its natural frequency and $\alpha(t)$ is the ratio X/F . The accelerances (along three orthogonal directions) have been obtained by exciting the supporting frame/cell system with a hammer (equipped with a load cell) and by measuring the response (Fig. 3).

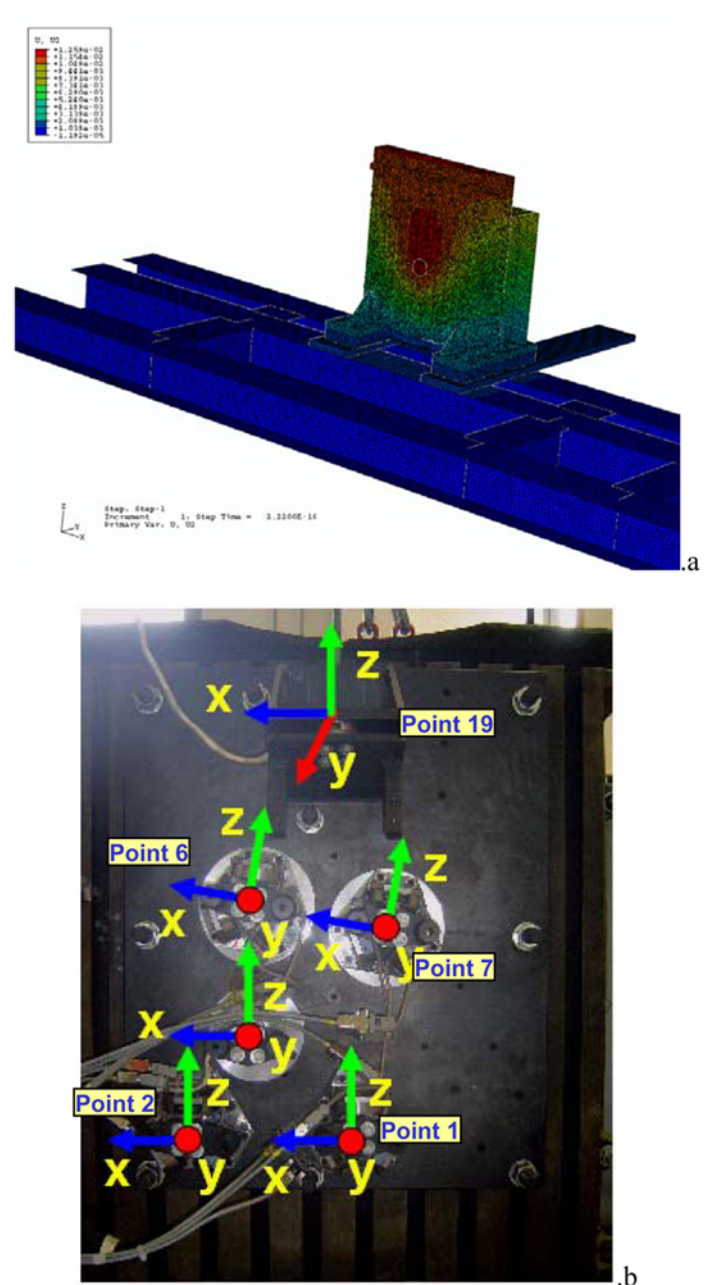


Fig. 2. RuotaVia connection structure. Example of FEM simulation (.a) and measurement of accelerance at the each suspension point (.b).

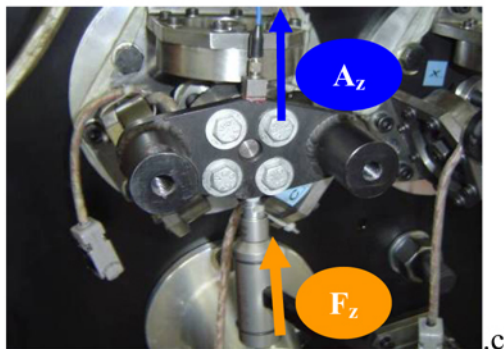
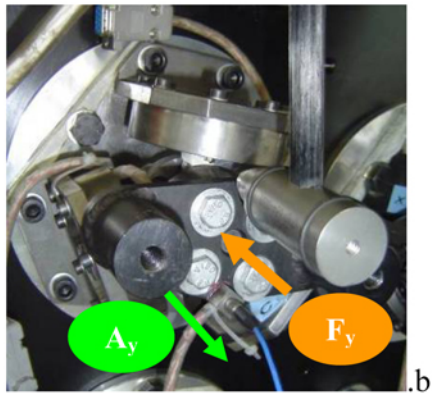
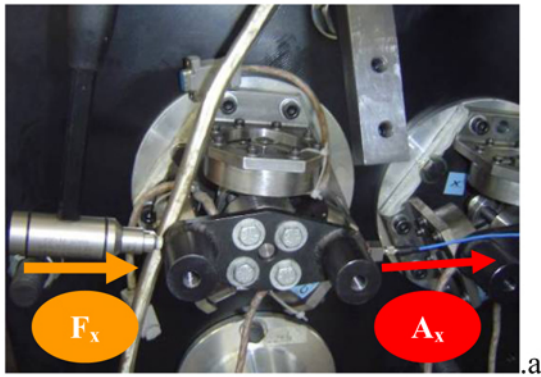


Fig. 3. Measurement of the acceleration of the supporting frame at a suspension connection joint along the local x (.a.), y (.b) and z (.c) direction.

An equivalent stiffness can be used to calculate the acceleration properties of the frame at the joints in the frequency range of interest (one model for each local direction). The parameters of the whole supporting frame model has been identified by considering the experimental accelerances [14, 15]. The identification procedure was based on the minimization of the mean square error between the measured and the computed accelerances. The curve described by the [equation 1](#) is superimposed to the experimental acceleration and the stiffness k is identified for each local direction at each suspension connection joint (i.e. in [Fig. 4](#) the two curves are reported).

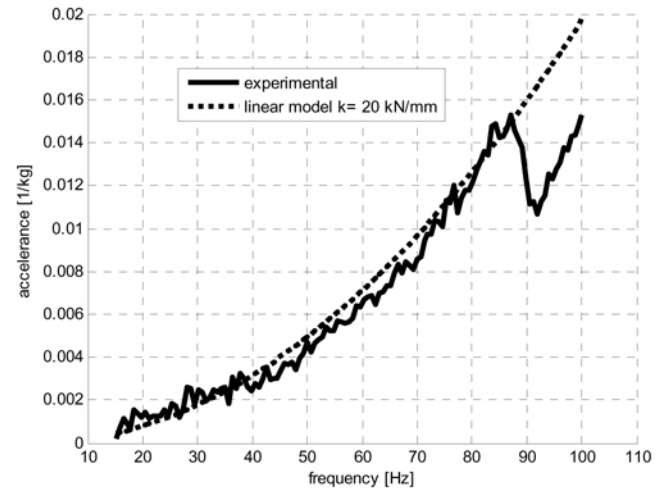


Fig. 4. Identification of the equivalent stiffness for the local z direction of the supporting frame at point 1 (see [Fig. 2](#)).

INSTRUMENTATION

The main objective of the experimental study is measuring the forces and moments applied at the suspension-chassis joints as the wheel passes over a flat surface or over a cleat at different vehicle speeds. The tested suspension is connected to the chassis via two elastic bushings at each wishbone arm (upper and lower) and at the top end attachment of the suspension strut. Each connection point has been equipped with a special high precision 6-axis load cell as shown in [Fig. 2.b](#). In order to measure the resultant force and moment which are applied to the superstructure (i.e. vehicle chassis) the steering link can also be instrumented.

The 6-axis load cells have been properly designed, modeled, calibrated and patented [4] ([Fig. 5.a](#)). Many commercially available multi axis load cells are able to measure the steady or dynamic forces and moments acting on a structure [5, 6, 6, 8, 9]. Nevertheless their use is not widespread in the research and development activities for their high cost. The 6-axis load cell has been conceived in order to have a simple device able to measure the three components of a force and the three components of a moment acting on the load cell itself but with good performances in terms of natural frequencies, limited mass and cost, high resolution and low cross-talk (for the technical specification, see [Tab. 1](#)). A comprehensive description of the six-axis load cells is reported in [12].

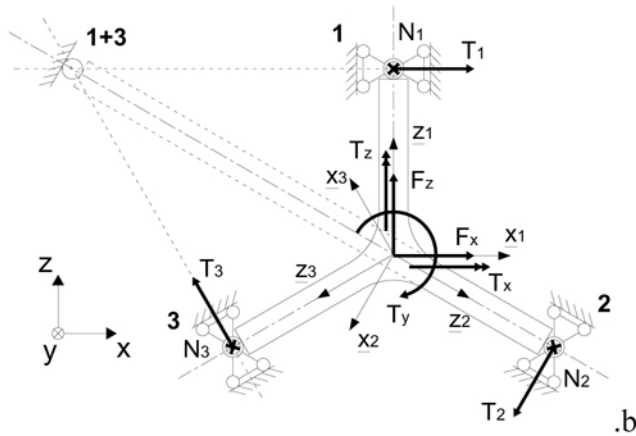


Fig. 5. A six-axis load cell prototype can be noticed and the characteristic three spoke structure is highlighted (.b). Concept design of the six-axis load cell suitable to measure the 3 forces (F_x , F_y , F_z) and 3 moments (T_x , T_y , T_z) acting on the cell itself (.a).

Tab. 1. Six-axis load cell. Technical specifications. The y direction is along sensor axis.

Technical specification			
Maximum force	Fy	10	kN
	Fx	5	kN
	Fz	5	kN
Resolution		± 5	N
Maximum torque	Mx	0.5	kNm
	My	0.5	kNm
	Mz	0.5	kNm
Resolution		± 0.5	Nm
Total mass		4.9	kg
1st nat. frequency		>650	Hz
Cross-talk		$< \pm 1.0\%$	
Overload capacity		+30%	

CALIBRATION

The calibration method used for the six-axis load cell is described in the following. Known forces and moments were applied and measured by means of a single axis precision load cell. Two different calibration rigs have been used. The first calibration rig has been employed to apply pure axial loads (Fig. 6.a). The load cell was placed horizontally so as the loads acted along the sensor axis y. A second calibration rig has been used to apply forces and moments in the directions F_x , F_z , T_x , T_y , T_z (Fig. 6.b). The orientation of the reference system axes is reported in Fig. 5.b. According to [10], by applying more than 150 load combinations, via a minimization of the square errors between the applied loads and the load cell measures, the identification of the components of the calibration matrix (M_i eq. 2) has been performed. This procedure has been used to estimate the cross-talk of the device. The vector of the 3 forces and 3 moments acting at the centre of the sensor can be calculated simply by multiplying the calibration matrix M_i by the vector of the tensions ΔV measured at the six half bridges [12]: