EXPERIMENTAL CHARACTERIZATION OF THE DYNAMIC BEHAVIOUR OF TIRES IN STATIC AND ROLLING CONDITIONS

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SUMMARY: The driver subjective perception of a vehicle is strongly determined by its Noise, Vibration and Harshness (NVH) behaviour. Consequently, the NVH performance has become an important design and marketing criterion for vehicle manufacturers. The low-frequency (0-500 Hz) noise and vibrations perceived by a passenger are mainly determined by the dynamic behaviour of rolling tires. Besides the increasing awareness for the problems associated with road traffic noise has lead to the demand for more quiet tires and road surfaces.

Computer aided engineering tools play an important role for improving the vehicle vibration behaviour at the earliest step of designing. In order to carry out a precise modelling of the vehicle and its subsystems, an accurate tire model is a key element in predicting the performance of the vehicle system with respect to ride comfort, NVH, durability, safety (braking manoeuvre) and tire/road interaction. To obtain the best possible performance from a tire model, a number of different measurements are required to support the tire model parameter identification process. Although tire vibration and noise behaviour has been studied for several decades, there are still some missing links in the process of accurately predicting and controlling the overall tire/road noise and vibration linked to the difficulty of performing a modal testing on a tire when rotating.

The paper deals with the main test benches, experimental activities and modal analysis techniques available nowadays for characterizing the dynamic behaviour of a static (unloaded and loaded tire on a fixed hub) and rotating radial tire under different boundary conditions. The main results arising from modal analysis of static and dynamic test on a tire are highlighted and compared. In particular the effect of rolling speed, inflation pressure, preload, temperature and excitation amplitude on the dynamic response of a rolling tire will be discussed.

KEYWORDS: Tire measurements, modal analysis, boundary condition, indoor test.

1. INTRODUCTION

The passengers of a vehicle that is crossing a road surface discontinuity, such as a joint in a concrete road surface, bumps, etc., are subjected to transient vibrations and noise that could reach high peak levels with a considerable reduction of the vehicle comfort. In order to improve the NVH characteristics of a vehicle a thorough understanding of the different noise and vibration sources in the vehicle is needed. One of this is represented by the tire/road interaction mechanisms. Additionally, the tire dynamic behaviour plays a key role in several new automotive technologies, such as, intelligent driver assistance systems, active suspension, in-wheel electric drive, intelligent tires [1] and wheels [2].

Over the last two decades, the increasing comfort, handling and durability requirements for new cars has lead to a demand for more accurate tire models for vehicle NVH simulations. A detailed investigation on the actual available tire models with a description of their capabilities and application areas has been provided by Ammon [3] and Lugner *et al.* [4], [5]. The most detailed tire models typically consist into two separate parts. The first part is represented by the *structural model* which describes the structural stiffness, damping and inertia proprieties of the tire. The second one is the *tread/road contact model* able to furnish an estimation of the contact pressure distribution and distributed friction force. Such tire models allow carrying out both transient and steady state vehicle dynamic analysis in addition to exhaustive ride and comfort studies (vehicle misuse measurement, development of the tire noise/vibration behaviour, etc.). Nevertheless, a wide number of changeable model parameters is required in order to adapt the model to the specific characteristics of the actual vehicle tire.

The parameters identification process (based on mathematical optimization algorithms) is typically complex and time long because only a large measurement database allows a correct and realistic identification and adjustment of all free tire model parameters (several tests at different running conditions and tire configurations). The final goal of this identification procedure is to obtain a generic tire model which could be used in different vehicle maneuvers simulation, at various running conditions but, at the same time, able to guarantee as soon as possible a low-cost computational time. However, the tire characteristics are strongly related to the adopted measurement data. In [6], Hüsemann and Wöhrmann pointed out as the measured tire properties could be different upon the testing procedure and the test bench adopted with following and not negligible consequences on the reliability of the vehicle dynamics simulation results. Similar investigations concerning the impact of tire measurement data on vehicle simulation model validation are provided by Sagan [7] and Klaas *et al.* [8].

Although tire/road noise and vibration phenomena have been studied for decades, there are still some missing links in the process of accurately predicting and controlling the overall tire/road noise and vibration. One of the most important missing links is represented by the effect of rolling on the dynamic behaviour of a tire. For example, it is well known that flexible rotating systems are subjected to gyroscopic effects. These effects are well understood for simple structures (e.g. rings and cylindrical shells) and can be modeled accurately. However, the gyroscopic effects for more complicated systems, such as a rotating tire in ground contact, are found to be much more complex and are not yet fully understood.

A novel testing method, based on laser Doppler vibrometry, has recently been developed to obtain the modal parameters of a rolling tire [9]. However, based on these experimental results only, it is impossible to gain full insight in the physical phenomena.

Inside the European seventh framework programme, an industry-academia partnership has been founded between the Mechanical Engineering Department of the KUL (Katholieke Universiteit Leuven), the European innovation centre of Goodyear [10] and LMS International [11], in order to make the recently developed testing method to characterize the rolling tire dynamic behaviour industrially applicable [12]. Improvements in the hardware, signal processing and modal parameters estimation procedure have been obtained in order to reduce the testing time and increase robustness and flexibility of the experimental method both for a radial tire in static and stationary rolling conditions.

In the first part of the paper, a description of the experimental approaches adopted for the tire modal parameters identification will be provided as a function of three main different boundary conditions: unloaded and loaded tire on a fixed hub and tire in rolling conditions. After a description of the data processing techniques for obtaining the tire modes, a comparison of the tire modal parameters under different boundary conditions will be performed. The main factors that influence the dynamic behaviour of a rolling tire will be finally discussed.

2. STRUCTURAL RESPONSE OF A ROLLING TIRE – MEASURING METHODS OVERVIEW

Several testing methods for studying the tire structural response have been developed in the last few years and some of these methods have been standardized. In particular, knowing the structural wave propagation and interaction characteristics of a tire in rolling conditions is fundamental for a deeper analysis of the structure-borne tire/road noise. From literature, different authors suggest *direct methods* for estimating the tire parameters. They are based on embedded sensors, such as strain gauges [13], Surface Acoustic Wave [14] or MEMS sensors [15], which allow directly measuring the tire deformations or tiring surface vibrations. However, the rotation of the tire significantly complicates the measurement of the structural response. In some case, this testing approach is also used for tire real-time monitoring purposes [1]. The privileged sensors position allows obtaining prompt information about the tire/road contact dynamics. Unfortunately, some troubles linked to the data transmission (a wireless monitoring is needed), the power supply (a battery has to be installed inside the tire for the sensor activation) and the limited operational life of the embedded sensors (due to the high stresses that characterize the contact area) have not been overcome yet. On the other side, *indirect* testing *methods* could also be exploited in order to visualize the vibration patterns of a tire surface during rolling motion by means of contactless optical measurement techniques (holographic interferometry [16] or laser doppler vibrometry [17]). No examples of the use of a scanning laser vibrometer were found in literature. In table 1 the main direct and indirect methods features of characterizing the vibration behaviour of a rolling tire are summarized.

Table 1 – Measurement methods for the structural response of a rolling tire.

Testing method	Ref. System	Structural changes	Data from contact area	Treaded tires	Remark
Embedded accelerometer	co-rotating	yes	yes	yes	Low durability of sensor fixation
Optical deformation sensor	co-rotating	yes	yes	yes	Vibrations of belt inner surface
Holographic interferometry	fixed	no	no	yes	No quantitative data
Laser vibrometry	fixed or co-rotating	no	no	no	Applicable to treaded tires

3. TIRE MODAL MEASUREMENTS

The experimental determination of tire modal parameters has become a major issue in a broad range of tire models. A tire shows modal behaviour for frequencies below 400-500 Hz. This means that structural waves, travelling in opposite direction along the tire circumference, interfere at certain frequencies and form a standing wave pattern. At higher frequencies, the increasing of the damping causes a rapid decay in amplitude of the structural waves away from the excitation area. In this case, a significant interference of waves becomes impossible, eliminating the modal behaviour above 400-500 Hz [18]. In the following, the experimental procedures typically adopted for the identification of the modal behaviour of tires in static and in rolling conditions are reported.

3.1. Unloaded and loaded tire

For the modal characterization of an unloaded/loaded tire in static conditions, experimental modal analysis (EMA) methods can be adopted. They are based on measured frequency response functions (FRF's) which describe the dynamic response of a structure to a known dynamic excitation force. In [9], experimental modal analyses performed on a slick and a commercial tire (both size 205/55R16) have been described. In this kind of test, typically the tire is excited at one location on the tread and the response is measured in several points on the tire and wheel. However, a multiple input analysis allows a better separation and identification of closely spaced or double poles (see next Section).

The response measurement could be obtained through triaxial accelerometer located on the tire cross-section, on the outer tire surface and on the wheel rim (see Fig. 1). The number of measuring points is a function of the circumferential mode number *n* of interest and the geometrical resolution that has to be sufficient in order to identify the different mode shapes. In order to reduce the effect of mass loading on the structure only few accelerometers are typically used simultaneously and distributed evenly over the tire circumference.

To overcome the loading effect making the modal characterization of the tire faster, an alternative testing method based on LDV measurements has been developed by Goodyear and KUL (Fig. 2.a). Both the vibration velocity of the tire sidewalls and the tread surface can be acquired.

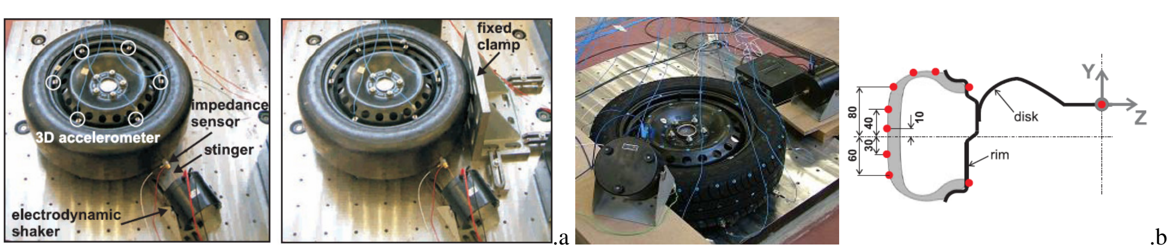
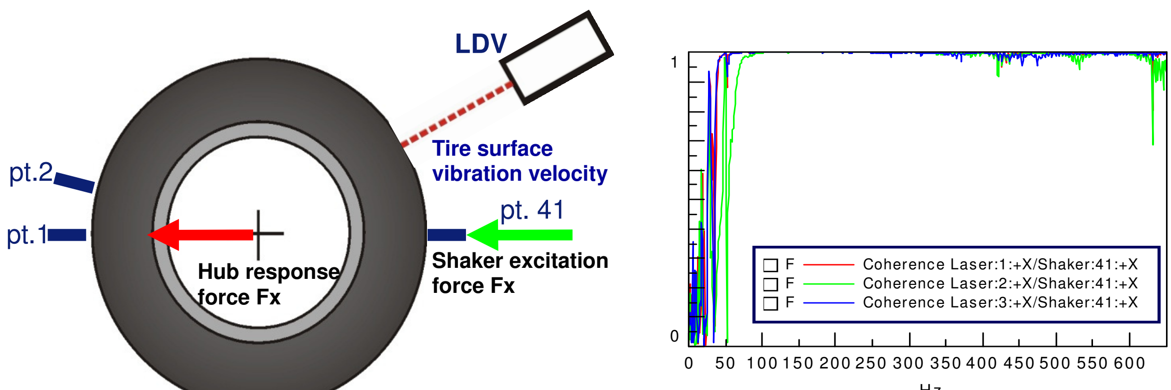


Figure 1 – Test setup for the modal analysis on an unloaded (.a, on the left) and loaded tire (.a, on the right) adopted from Kindt *et al.* [9]. Multiple input test (.b, on the left) and measurement points on the tire cross-section (.b, on the right) are highlighted.

The preliminary LDV measurements have been performed on a tire with size 235/40R18 (inflation pressure equal to 2.2 bar). The wheel rim has been rigidly clamped onto a wheel hub dynamometer (in order to measure the reaction forces at the wheel hub) and the vibration velocity response has been measured at 41 points evenly distributed over one half of the tire circumference. The tire is excited by an electrodynamic shaker connected to the tire surface through a stringer and a small diameter disk glued to the tire tread surface.
A burst random (with different burst time values, such as used in [9]) or a burst chirp signal (for instance, a sine sweep dropping to zero at the end of the excitation period) can be adopted as excitation type for the system. These excitation signals are characterized by a high coherence function values at resonance frequencies (Fig. 2.b) and low leakage errors. However, the burst signals have lower total excitation energy if compared to their periodic counterparts with a resulting lower signal to noise ratio (indicated by lower coherence values at anti-resonances where the signal is small). The shaker mainly excites the tread in radial direction (Fig. 2.a).
In [9], multiple uncorrelated inputs have been adopted for a better excitation of the modes characterized by belt displacements along the tangential and the axial direction (a better excitation energy distribution can be obtained over the entire structure with a resulting identification of the repeated roots, see Fig. 1.b, on the left). For the unloaded tire case, the choice of the circumferential location of the excitation point is arbitrary since the structure is axisymmetric. However, the loaded tire is no longer axisymmetric and thus the circumferential position of the excitation point should be chosen such that all modes of interest are well excited. A useful tool to select the excitation location is a Mode Shape Summation Plot (MSSP) [19], which shows for each response point, the contribution of the different vibration modes to the response. Since each measurement run (whose number is equal to that of the measuring points on the tire surface) counts 15 burst chirp excitation, the frequency response functions have been calculated using H_1 estimator [19] with 15 averages. In the present case, the frequency range is 0÷2048 Hz with a frequency resolution of 1 Hz and no window was used (burst signals).



.a .b

Figure 2 – Innovative test setup for the modal characterization of a static tire [12]. In the scheme the measurement points and the shaker excitation force location on the tire tread surface are shown for the static case. In (.b) an example of coherence function for a burst chirp excitation relative to three different measurement run is reported.

3.1.1 *Alternative experimental approach*

For identifying the dynamics of a loaded tire, a based transmissibility approach can also be used. As well as known, the tire transmissibility describes how an excitation applied at the contact patch results in a response at the spindle. When an uneven road excites a tire at the contact patch, the vibration energy is transmitted through the suspension towards the vehicle body (body panel and window vibrations cause noise radiation in the car cabin). Since a direct measurement of the displacement input at the contact patch of a loaded tire is hardly, the following reciprocity principle can be used. A force-out (F_1) over force-in (F_2) FRF in constrained condition corresponds with a displacement-in (X_2) over displacement-out (X_1) FRF in free-free condition [20].

$$\left|\frac{F_1}{F_2}\right|_{x1=0} = \left|\frac{x_2}{x_1}\right|_{F2=0} \tag{1}$$

Based on this approach, an experimental tire model that describes the non-rotating dynamics of a preloaded tire has been developed in [21]. With a force-platform, the response force along the three directions at the tire contact patch can be measured while force inputs are measured at the rim center by means of an impact hammer or an electromagnetic shaker (Fig. 3, on the left). By switching the force over force FRF to a displacement over displacement FRF, a tire patch to rim transfer function can be obtained. This result is valid in free-free condition that is a fundamental requirement when the tire is coupled to the vehicle model using FRF based sub-structuring (FBS) approach (Fig. 3, on the right). Nevertheless, this method is more valid for identifying inputs for a vehicle interior noise prediction than for a characterization of the tire vibration behaviour.

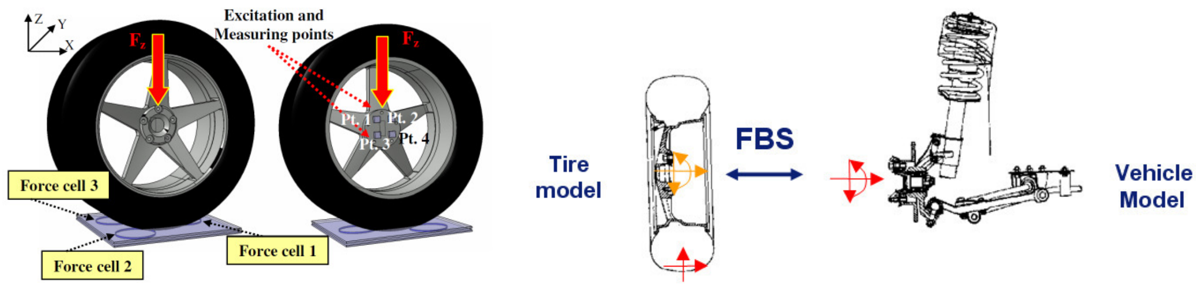


Figure 3 – FRF based sub-structuring approach for vehicle interior noise prediction.

3.2. Rolling tire. Innovative test bench

In order to characterize the vibration behaviour of a tire while running over different kind road surface (flat or uneven road, cleats with different size and shape, etc.), an innovative test bench is under development (Fig. 4, on the left). The test facility is composed basically by a steel drum (2.6 m diameter) providing contact surface for vehicle wheels and allowing highly repetitive and controllable running tests under various operating conditions. A special connection structure has been properly designed for testing different tire-wheel sizes and for clamping the wheel rim at the spindle. Typically a piezo-electric dynamometer (or a measuring hub) is adopted for measuring the three spindle forces and the three spindle moments. The test bench is quite similar to that proposed by Kindt *et al.* in [22]. The rolling tire vibrations are measured by means of a single point LDV. A circumferential pattern of plane mirrors, located around the testing tire, is used for reflecting the laser beam perpendicularly on the tire surface. In this way, the testing procedure for the identification of the tire in-plane vibration modes is speeded up. The support of each mirror can be fixed at different locations around the tire so that different circumferential measurement resolution can be obtained. The sidewall tire rolling vibrations are instead measured directly without reflecting the laser beam. An example of measurement grid is reported in figure 4, on the right.

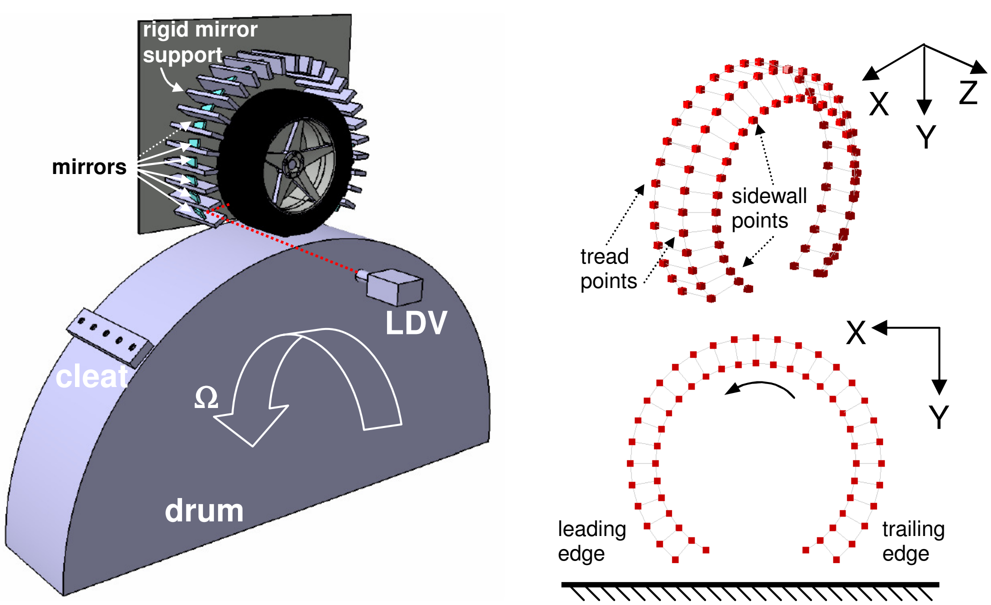


Figure 4 – Innovative test setup for tire rolling vibration measurement by means of a Laser Doppler Vibrometer. On the right, a typical measurement grid is reported in an absolute reference system.

4. MODAL ANALYSIS METHODS

This paper applies the tire mode naming convention described by Wheeler *et al.* [23], which allows an unambiguous labeling for tire modes under different boundary and operating conditions. This convention uses two indices which describe the bending order of the belt package in the two directions (Fig. 5). The format of the notation is (n,a). The first index n represents the number of circumferential bending wavelengths of the belt. The second index a represents the number of half-wavelengths in the axial direction of the belt at a circumferential location where the shape is at an extreme radial displacement. The described convention is ambiguous for some of the modes in which the belt translates or rotates as a rigid structure. For those modes, typically, an additional labeling (such as 'lateral', 'pitch', and 'torsion') is added to indicate the rigid body motion of the belt. When a tire is loaded, the structure is no longer axi-symmetric so that a split of the repeated roots of the unloaded tire into two single roots can be noted (these modes are related to the same mode of the unloaded tire). Consequently, an additional naming convention based on an observed trend in the mode shapes has been introduced in [9]. For the first mode shape, the circumferential location where the radial displacement is zero along the cross-section is found to be located in the middle of the tire/road contact area. To the name of this mode, an additional label "0" is added. For the second mode shape, the circumferential location where the radial displacement reaches an extremum along the cross-section is found to be located in the middle of the tire/road contact area (in this case, an additional label "extr" is added, see Fig. 5 on the right).

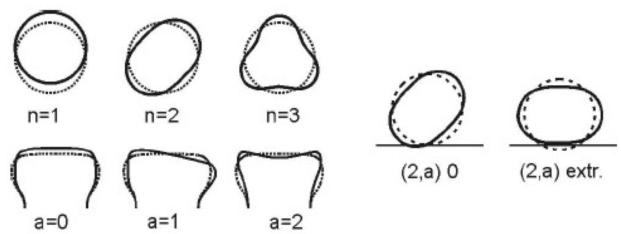


Figure 5 – On the left, naming convention for tire structural modes by Wheeler *et al.* [23]. On the right, an example of (2,a) modes for a loaded tire.

4.1. Unloaded and loaded tire, boundary conditions influence

Table 2 reports the resonance frequency, modal damping factor and the mode shape for some of the unloaded tire resonances. The modal damping ratio is expressed as a percentage of the critical damping. The modal parameters have been estimated starting from the experimental data acquired by means of the innovative test rig (Fig. 2) and by using the least squares complex frequency domain method (PolyMAX, [24]) implemented in the LMS Test.Lab® software (Fig. 6, on the left). This is a multiple degree of freedom method which generates a global estimation for the modal parameters. After the estimation of the system poles, the modes shapes are computed. The unity modal mass scaling scheme [19] has been used to scale the mode shapes. For an unloaded tire (on a wheel fixed at the hub), the tire belt vibrates as rigid structure at certain resonances. The relative modes are defined as *rigid belt modes* which consist of an *axial mode* (the belt translates as rigid structure along the lateral z axis, for the reference system see Fig. 4), a *torsion mode* (the belt rotates as a rigid structure about the lateral z axis), an (1,0) and (1,1) mode where the belt respectively translates and rotates as a rigid structure in relation to an axis in the wheel plane (x or y axis).
However, the axial and torsion mode were not identified in this test setup due to the lack of input along the axial and the tangential direction. Two resonance frequencies at 49.3 Hz probably corresponds to the rigid axial mode of the tested tire, but the associated damping value is too lower if compared to the corresponding ones obtained from [9]. This could be an index that the axial and torsion mode was not so well excited. The (2,0) mode is the first mode that involves a deformation of the belt and all higher vibration modes appear well identified up to (8,0) mode (as confirmed from the autoMAC matrix, Fig. 6, on the right).

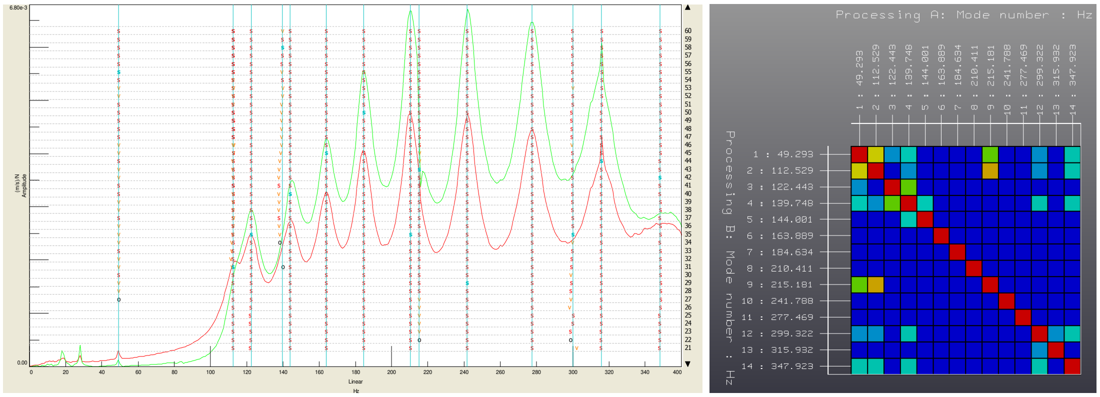
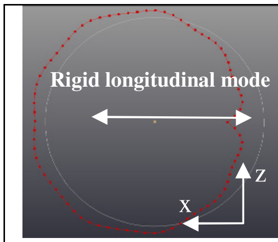
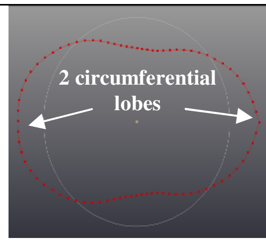
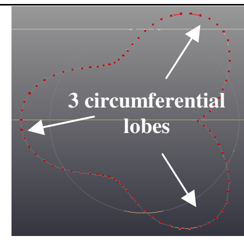
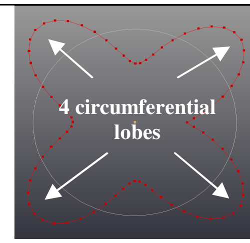
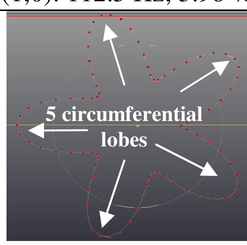
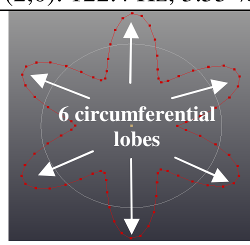
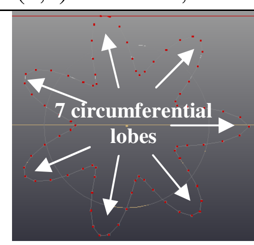
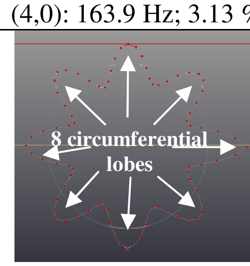


Figure 6 – On the left, stabilization diagram relative to the unloaded tire measurements and obtained by using the LMS Polymax algorithm [24]. On the right, Modal Assurance Criterion applied to the corresponding set of modes.

Table 2 – Mode shapes of the unloaded tire 235/40 R18. Wheel is fixed at a fully constrained spindle.

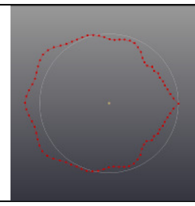
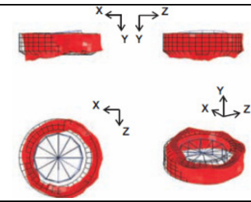
 (1,0): 112.5 Hz; 3.98 %	 (2,0): 122.4 Hz; 3.55 %	 (3,0): 144.0 Hz; 3.65 %	 (4,0): 163.9 Hz; 3.13 %
 (5,0): 184.6 Hz; 2.69 %	 (6,0): 210.4 Hz; 2.38 %	 (7,0): 241.8 Hz; 2.35 %	 (8,0): 277.5 Hz; 2.50 %

Except for the axial mode, all other identified modes have double poles caused by the axisymetry of the tire structure (two modes have an identical natural frequency). However, if the frequency separation is very small and the damping is moderate, the two separate resonance values cannot be observed. Since in the preliminary tests performed with the innovative test-bench only one excitation has been used, the double poles cannot be identified in the presented modal analysis. Besides the structural resonances of the tire, there also exist the resonances of the tire air cavity and the wheel rim. The first acoustic resonance of the air cavity appears at 215.2 Hz with a modal damping of 0.29% which is much lower than the damping ratio values associated to the structural modes of the tire. As shown in [26], a tire becomes stiffer at the air cavity resonance and high tonal noise is radiated to the exterior at this frequency range (at around 220 Hz).
In [25], Thompson proposed a simple analytical model for modeling the air cavity of an undeformed tire as an unwrapped torus. At low frequencies, the pressure waves propagating in the circumferential direction of the cavity can be considered as plane waves since their wavelength is much larger than the cavity cross-sectional dimensions. So considering the tire structure to be rigid, the first acoustic natural frequency of the air cavity can be expressed as

$$f = \frac{c}{L_c} \tag{2}$$

where L_c is the median circumferential length of the tire cavity and c is the speed sound (equal to 344 m/s). This simple model is able to predict the first acoustic resonance with accuracy of 1-2% [25]. In table 3, the experimental and estimated frequency values relative to the first acoustic resonance are reported for two different tire sizes.

Table 3 – Experimental and numerical frequency values for the 1st acoustic resonance of two different tire size. (* Results obtained and adapted from [9])

	Tire 235/40 R18	Tire 205/55 R16*
1st acoustic mode shape		
Experimental f [Hz], ξ [%]	215.2 (0.29)	225.8 (0.25)
Thompson model f [Hz]	213.9	222.6

Furthermore, looking for the table 3, at the first acoustic resonance, the belt shows a similar deformation as the (1,0) structural mode (table 2). This is due to the vibro-acoustic interaction between the air in the tire cavity and the tire structure. In literature, it is shown that the acoustic mode of the tire cavity has a noticeable influence on the structural behaviour of the tire belt but not the other way around. By means of some sound absorbing materials properly designed and introduced inside the tire, an effectively decreasing of the exterior noise and acoustic resonance energy damping out can be obtained [26].

In figure 7 (on the left), the modal parameters of two different size tires are reported as a function of the circumferential wave number. Both tires have been tested at the same boundary conditions (unloaded tire with fixed spindle). The first two (n,0) modes of the 235/40 R18 tire appear at a slightly higher frequency compared to the other one while, for high wave numbers, this trend changes and the difference between the natural frequencies of the two tires seems to increase with an increasing of the circumferential wave number. A potential explanation of this phenomenon could be found in [9].

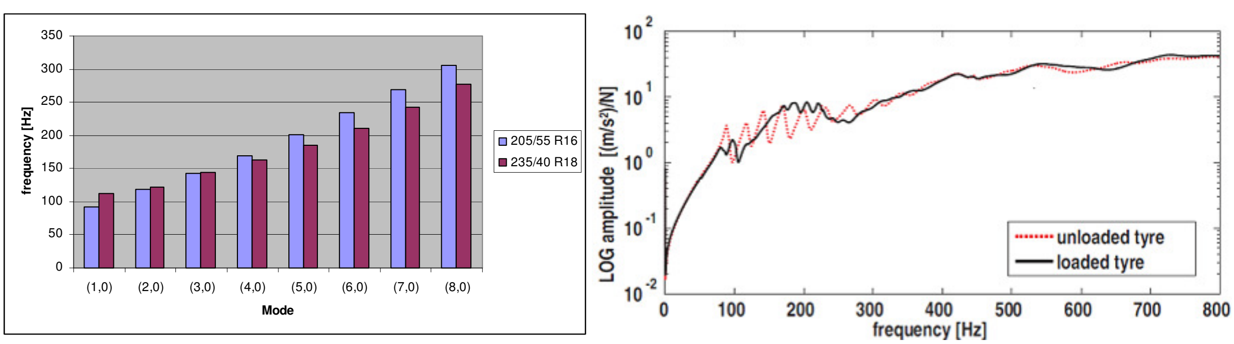


Figure 7 – On the left, circumferential modes of two different tire sizes tested at the same boundary conditions (unloaded tires with fixed hub). On the right, measured FRFs at the excitation point for a loaded and unloaded tire (adapted from [9]).

The FRFs at the excitation point for a loaded and unloaded tire are reported in figure 7, on the right. While for the unloaded tire the resonance frequencies are clearly visible below 300 Hz, for the loaded one the higher modal density (split-up of the double poles) makes difficult the resonances identification. Above 300 Hz, for both cases, no individual resonances can be individuated. Indeed, damping causes the structural waves at higher frequencies to decay rapidly in amplitude away from the excitation point (such as remarked in [18]).

Table 4 compares the modal parameters of the first modes identified in the experimental modal analysis for an unloaded and loaded tire with fixed spindle. Even if the non-axi-simmetric shape of the loaded tire structure causes a split up in the double poles of the unloaded tire into two single poles, there are still modes in which the belt mainly vibrates as a rigid structure. In particular, it has been noted that at (1,0) and (1,1) *hor.* mode, the belt of the loaded tire (with fixed spindle) mainly translates and rotates along the horizontal axis while at the (1,0) and (1,1) *vert.* mode, the belt for the loaded tire mainly translates and rotates about the vertical axis. Additionally, the contact with the ground makes the occurrence of the *axial* and *torsional* mode impossible.

Table 4 – Comparison between the modal parameters of a loaded and unloaded tire (size 205/55 R16) mounted on a steel wheel. The spindle rotation and translation is constrained. Adapted from [9].

Unloaded tire (fixed spindle)		Loaded tire (fixed spindle)	
mode	**freq. [Hz], ξ [%]**	**freq. [Hz], ξ [%]**	**mode**
axial	47.20 (1.87)	/	axial
torsional	74.25 (6.39)	/	torsional
(1,1)	56.98 (2.14)	51.85 (1.97)	(1,1) hor.
		64.75 (2.48)	(1,1) vert.
(1,0)	91.26 (4.50)	82.15 (5.42)	(1,0) hor.
		98.22 (4.09)	(1,0) vert.
(2,0)	118.55 (3.26)	117.94 (3.43)	(2,0) 0
		126.74 (3.18)	(2,0) extr.
(2,1)	89.37 (2.07)	102.10 (2.78)	(2,1) 0
		86.99 (2.46)	(2,1) extr.
(3,0)	141.97 (2.78)	142.56 (2.87)	(3,0) 0
		156.03 (2.75)	(3,0) extr.
...	...	...	...
1st acoustic	225.75 (0.25)	219.02 (0.69)	1st acoustic hor.
		227.35 (0.43)	1st acoustic vert.

4.2. Rolling tire

In order to analyze the tire/road noise generating mechanisms while the tire is running over a flat or uneven road surface, Operational Modal Analysis techniques (OMA) will be more suitable with respect to the classical modal analysis. EMA is not applicable to characterize the dynamic behaviour of a rolling tire since a direct measurement of the excitation forces caused by the surface texture cannot be performed. Frequency-domain Operational Modal Analysis methods, such as PolyMAX, require output cross-spectra as primary data. As proved in [27], under the assumption of white noise input, output spectra can be modelled in a very similar way as Frequency Response Functions (FRFs) where the deterministic knowledge of the input is replaced by the assumption that the input is white noise. This modal parameter estimation method does not directly use the raw output signals, but relies on reduced data such as cross-correlations and cross-spectra between signals measured simultaneously at different locations.

The correlation or covariance matrix $R_i \in \mathbb{R}^{lxl}$ between the measured output signals $y_k \in \mathbb{R}^l$, with l the number of outputs and k the sample index, can be firstly estimated as

$$R_i = \frac{1}{N}\sum_{k=0}^{N-1} y_{k+i} y_k^T \tag{3}$$

where N is the total number of samples and i the correlation sample index (also called time lag).
As non-parametric spectrum estimate, the so-called *weighted correlogram* can be used. It is computed as the DFT of the weighted estimated correlation matrix (3)

$$S_{yy}(\omega) = \sum_{k=-L}^{L} w_k R_k e^{-j\omega k \Delta t} \tag{4}$$

where L is the maximum number of time lags at which the correlations are estimated and w_k denotes the time window. As the correlation samples at negative time lags ($k < 0$) contain redundant information, it is sufficient to consider only the positive time lags when computing the spectra. This lead to so-called *half spectra* of which even the auto spectra have a phase different from zero

$$S_{yy}^{+}(\omega) = \frac{w_0 R_0}{2} + \sum_{k=1}^{L} w_k R_k e^{-j\omega k \Delta t} \qquad (5)$$

A more traditional non-parametric spectrum estimate is the so-called *weighted averaged periodogram* (also known as modified Welch's periodogram, [24]).
Typically, during a rolling test, each measurement run contains a different tire surface response point and the spindle forces as well. One or more forces at the wheel hub can be assumed as reference signals, since these are the only signals that are available at each run, and exponential windowing was used. In order to extract the modal parameters of the rolling tire, all runs are analyzed simultaneously by using the operational OMA method implemented in LMS Test.Lab [11]. Nevertheless, different measurement runs could cause different spectra with the same trend but with small amplitude variations due to the tire temperature variation between one run and the others (Fig. 8). In order to overcome this phenomenon, some non-classical data processing approaches have been proposed in [28]. Figure 9 reports some typical mode shapes of a tire (205/55 R16) while running over a cleat in steady state conditions (at about 30 km/h).

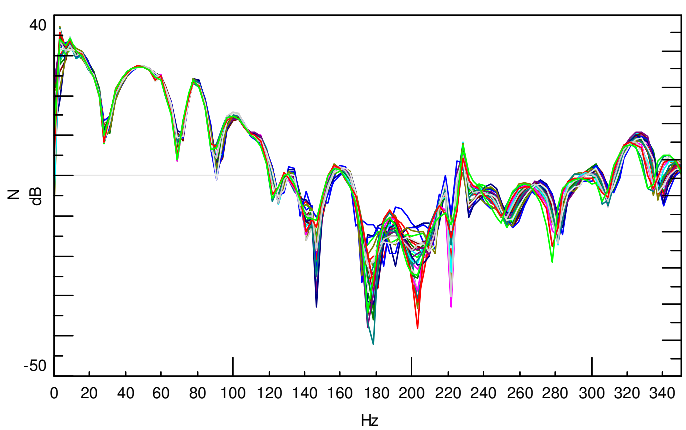


Figure 8 – Vertical spindle force power-spectra for different measurement runs.

The experimental data have been obtained using a tire-on-tire test setup developed at the Department of Mechanical Engineering of K.U. Leuven. The results coming from OMA show that no structural modes above 200 Hz have been identified (for a non-rolling tire, modes have been found up to 300 Hz – see Section 3.1) and most of them are (n,0) modes. No tilting or bending of the belt in axial direction are present (this depends on the cleat size and shape). The rolling tire resonances appear at lower frequencies compared to the static tire resonances (the frequency shift is strongly dependent on the type of mode, see next section).

4.3. Influencing factors on the rolling tire vibration behaviour

Some of the main factors that influence the dynamic response of a rolling tire are represented by the value of the rolling speed, inflation pressure, static preload, temperature and excitation amplitude. An interesting sensitivity analysis performed on a rolling tire is reported by Kindt *et al.* in [22]. Empirical observations reveal a decreasing of the rolling tire resonances compared to the non-rolling case (the (n,0) modes appear at about 10% lower natural frequency). Since this reduction in resonance frequencies already appears at low rolling speeds, not only gyroscopic effects can cause this phenomenon. Changing in operating point of the filled rubber due to the rolling condition (*Mullins effect*) or in its dynamic stiffness properties with increasing vibration amplitude (this is known as *Payne effect*) can contribute to justify the dependency of a rolling tire vibration behaviour from the running speed.

On the other side, an increasing of the tire inflation pressure (from 1.7 to 2.7 bar) causes a vertical preload rising with a generic positive frequency shift of the structural resonances. Slight variation on the structural and acoustic resonances can be noted if the tire static deflection increases due to an increasing of the static preload while as the tire temperature increases (from 25°C to 55°C) the structural and the acoustic resonances decrease and increase, respectively.
Finally, in order to investigate the non-linearity tire behaviour due to the deformation amplitude, some rolling tests have been performed with different cleat heights. Operational modal analyses results have suggested that a significant decrease can be noted only for the (1,0) structural resonance. However, from the preliminary experimental activities presented in [22], no strong explanation of this effect can be suggested and additional and deeper investigations on the deformation amplitude dependency (Payne effect) for a rolling tire are needed.

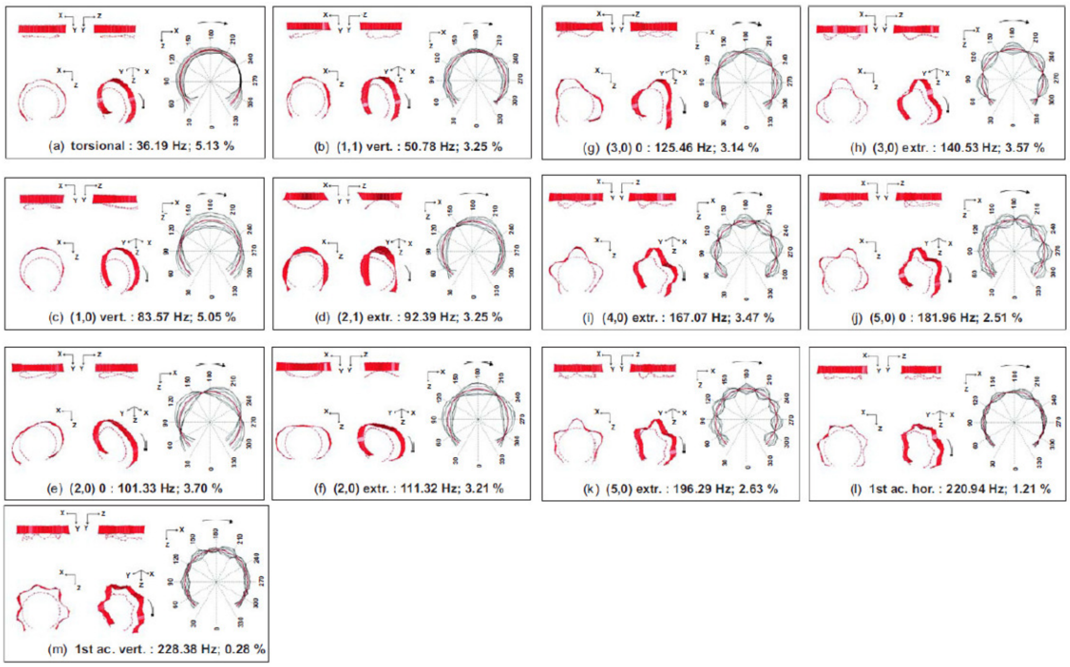


Figure 9 – Operational modal parameters of a rolling tire (running speed 28.3 km/h, 5 mm high semicircular cleat, inflation pressure 2.2 bar). Mode shapes are shown in a fixed reference system. Adopted from [28].

Table 5 shows an overview of the main effects produced on the vibration modes of a rolling tire due to an increasing of the above mentioned influencing factors. All test have been performed on a passenger car tire of size 205/55 R16 while running over a cleat [22]. The inflation pressure influence has been studied maintaining the tire deflection as constant (with a consequent static load increasing).

Table 5 – Effects on the vibration behaviour of a rolling tire due to an increasing of some influencing factors (running speed, inflation pressure, static preload, etc.) - tire size 215/55 R16.
(↑↑ high influence, ↑ medium influence, ≈ negligible influence, * influence level under studying)

Mode	Rolling speed (p=2.2bar T=30°C)		Infl. pressure (v=30km/h T=30°C)		Static preload (v=30km/h p=2.2 bar T=30°C)		Temperature (v=30km/h p=2.2bar)		Cleat height (v=30km/h p=2.2 bar T=30°C)	
	freq.	ξ	*freq.*	ξ	*freq.*	ξ	*freq.*	ξ	*freq.*	ξ
torsional	↓	↑	≈	≈	*	*	↓	*	*	*
(1,0)	↓	↑	↑↑	≈	≈	≈	↓	*	↓↓	↓
(n>1,0)	↓	≈	↑	≈	*	*	↓	*	↓	≈
(n,1)	*	*	↑	≈	*	*	↓	*	≈	≈
1st acoustic hor.	↓	↑	≈	≈	↓	*	↑	*	≈	≈
1st acoustic vert.	↑	↓	≈	≈	↑	*	↑	*	≈	≈

5. CONCLUSIONS

The increasing NVH requirements for a new road vehicle and the growing awareness of the problems associated with road traffic noise are leading to a demand for deeper analysis of the tire/road interaction phenomena. Since the low-frequency (0-500 Hz) noise and vibrations perceived by a passenger are mainly determined by the dynamic behaviour of rolling tires, more accurate tire models are required for the further optimization of the vehicle NVH features. To obtain the best possible performance from a tire model, a number of different measurements in static and rolling conditions are required to support the tire model parameter identification process.

An industry-academia partnership, coordinated by the Mechanical Engineering Department of the K.U. Leuven, and with the European innovation centre of Goodyear and LMS International as partners, has lead to make the recently developed testing method to characterize the tire dynamic behaviour industrially applicable. Starting from a brief overview of the main experimental methods available nowadays for the tire dynamics characterization, an innovative tire test bench able to reduce the testing time has been proposed. A classification and description of the main effects that influence the tire vibration modes under static and rolling conditions has been illustrated. Both experimental and operational modal analysis techniques adopted for the tire modal parameters identification have been described. The preliminary measurements performed on a static tire fixed on hub show that above 400 Hz the identification of the circumferential and out-of-plane tire modes belongs difficult due to an increasing modal density and complexity of the mode shapes.

The rolling condition causes a decreasing of the tire structural resonances and non-linear effects which are not observable on a non-rolling tire. The non-linear Payne and Mullins effects are dependent on the tire construction and occur at the lowest running speed too. Additionally many other factors influence the tire modal behaviour in rolling conditions such as the running speed, inflation pressure, excitation amplitude, etc. An accurate tire model, based on modal parameters, should take into account all these effects in order to obtain a growing detailing representation of its physical characteristics.

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