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EXPERIMENTAL STUDY FOR HIGH-FREQUENCY MODAL CHARACTERIZATION OF TIRES

Gianpiero Rocca, Bart Peeters

LMS International nv, Interleuvenlaan 68, B-3001, Leuven, Belgium
e-mail: gianpiero.rocca@lmsintl.com

Cristobal González Díaz, Jason Middelberg, Peter Kindt

Goodyear Innovation Centre – Avanue Gordon Smith, L-7750, Colmar-Berg, Luxembourg

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 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. This 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 a 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.

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 these is represented by the tire/road interaction mechanisms. Furthermore, 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.

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[1] and Lugner *et al.*[2]. 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 takes a long time 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). On the other side, 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. A novel testing method, based on Laser Doppler Vibrometry, has recently been developed to obtain the modal parameters of a rolling tire[3]. 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 Katholieke Universiteit Leuven (KUL), the Goodyear Innovation Centre Luxembourg and LMS International in order to make the recently developed testing method to characterize the rolling tire dynamic behaviour industrially applicable[4]. 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.

2. 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 400-500Hz. 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 400-500Hz[5]. 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.

2.1 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. Some experimental modal analysis has been already performed on a slick and a commercial tire[3] (both size 205/55R16). 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[3].

The response measurement could be obtained through a triaxial accelerometer located on the tire cross-section, on the outer tire surface and on the wheel rim. 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. 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. 1.a). Both the vibration velocity of the tire sidewalls and the tread surface can be acquired. 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) 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. 1.b) and low leakage errors. The shaker mainly excites the tread in radial direction (Fig. 1.a). 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. 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[6] with 15 averages. In the present case, the frequency range is 50-2048Hz with a frequency resolution of 1 Hz and no window was used (burst signals).

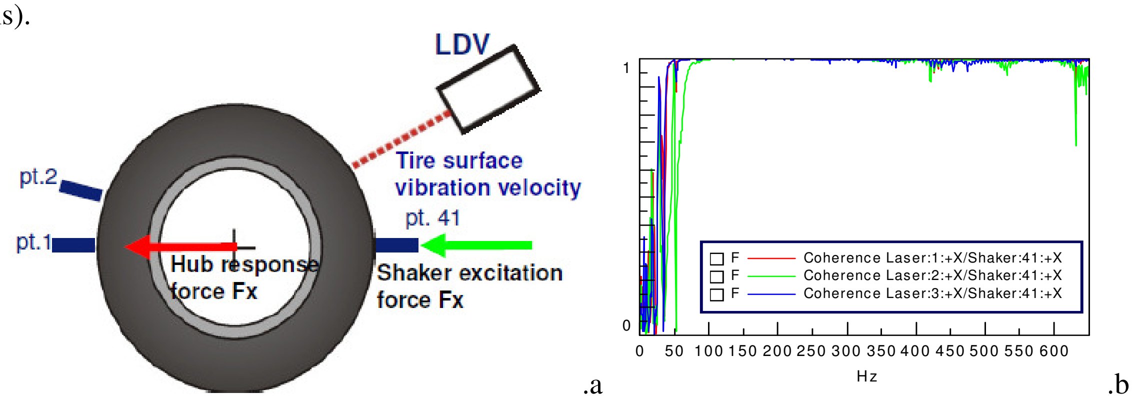


Figure 1. Innovative test setup for the modal characterization of a static tire[4]. The measurement points and the shaker excitation force location on the tire tread surface are shown for the static case. In (.b) an example of coherence function for a burst chirp excitation relative to three different measurement runs.

2.2 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. 2, on the left). The test facility is composed by a steel drum (2.5 m diameter) providing contact surface for vehicle wheels and allowing highly repetitive and 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.*[7]. 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 onto the tire surface. In this way, the testing procedure for the identification of the tire in-plane vibration modes is sped 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 Fig. 2 (on the right). At different running speeds, the excitation of the preloaded tire is provided from a rectangular section cleat (3 mm high and 25 mm wide).

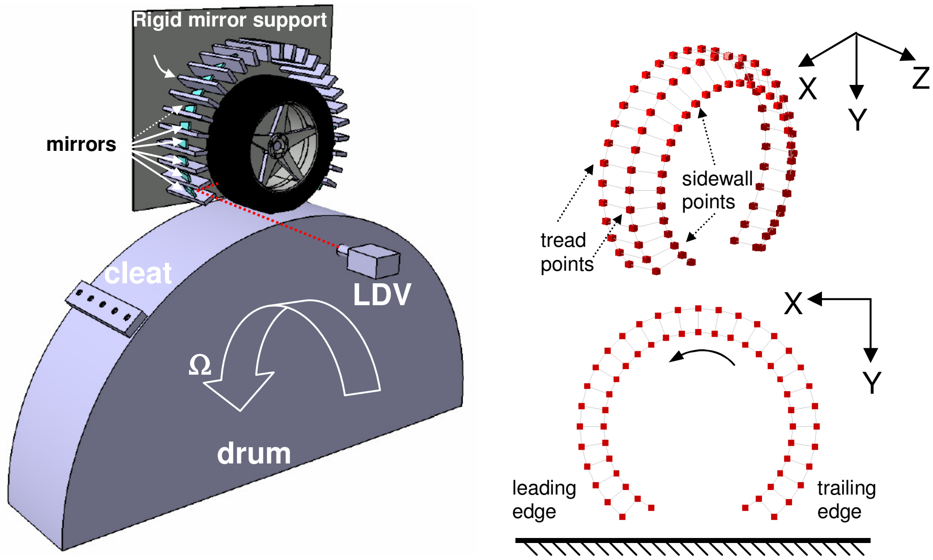


Figure 2. Innovative test setup for tire rolling vibration measurement by means of a Laser Doppler Vibrometer. On the right, a typical measurement grid is reported in an absolute reference system.

3. Vibration mode shapes. Classification

This paper applies the tire mode naming convention described by Wheeler *et al.*[8], which allows an unambiguous labelling 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. 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. Typically, an additional labelling (such as 'lateral', 'pitch', and 'torsion') is added to indicate the rigid body motion of the belt.

4. Vibration behaviour of a static tire. Boundary conditions influence

Table 1 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. 1) and by using the least squares complex frequency domain method (PolyMAX[9]) implemented in the LMS Test.Lab® software (Fig. 3, 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. For an unloaded tire (on a wheel fixed at the hub), the tire belt vibrates as rigid structure at certain resonances. The rigid belt modes consist of an axial mode (the belt translates as rigid structure along the lateral z axis, for the reference system see Fig. 2), 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. 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. 3, on the right).

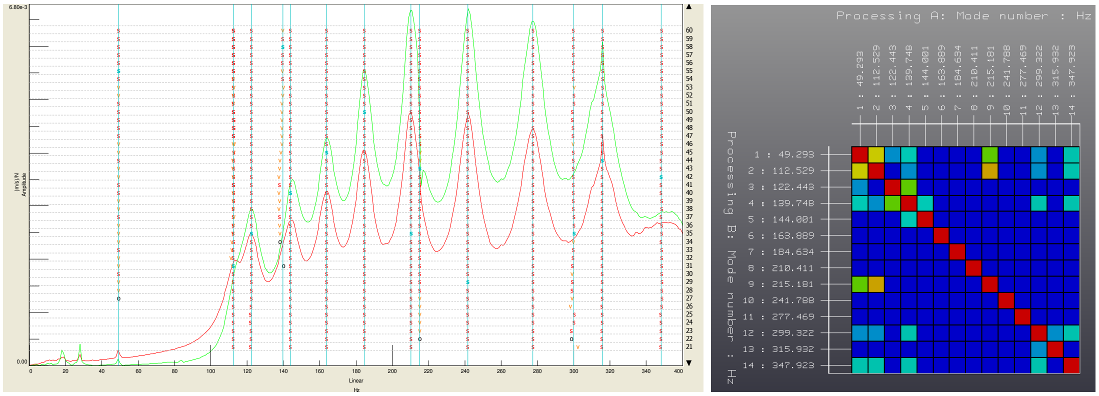


Figure 3. On the left, stabilization diagram relative to the unloaded tire measurements (LMS Polymax algorithm[9]). On the right, Modal Assurance Criterion applied to the corresponding set of modes.

Except for the axial mode, all other identified modes have double poles caused by the axi-symmetry of the tire structure (two modes have an identical natural frequency). Nevertheless, 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. In addition to the structural resonances of the tire, the resonances of the tire air cavity and the wheel rim are also identified by the measurements. 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. Thompson[10] proposed a simple analytical model for modelling 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

$$f = \frac{c}{L_c} \tag{1}$$

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[10] is able to predict the first acoustic resonance with accuracy of 1-2%. The experimental and estimated frequency values relative to the first acoustic resonance are respectively equal to 225.8 Hz and 222.6 Hz for tire 205/55R16 and 215.2 Hz and 213.9 Hz for tire 235/40R18.

Table 1. Mode shapes of the unloaded tire 235/40 R18. Wheel is fixed at a fully constrained spindle.

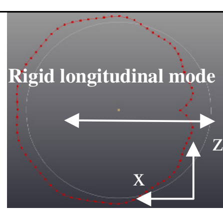
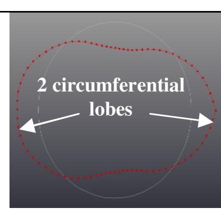
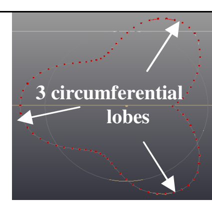
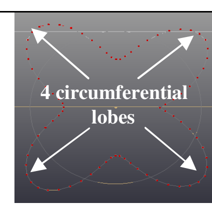
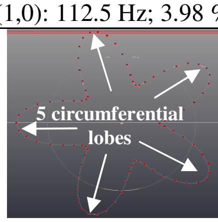
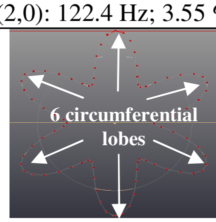
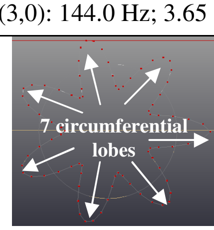
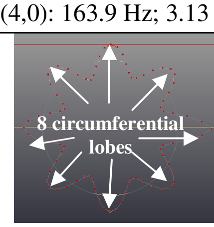
 (1,0): 112.5 Hz; 3.98 %	 (2,0): 122.4 Hz; 3.55 %	 (3,0): 144.0 Hz; 3.65 %	 (4,0): 163.9 Hz; 3.13 %
 (5,0): 184.6 Hz; 2.69 %	 (6,0): 210.4 Hz; 2.38 %	 (7,0): 241.8 Hz; 2.35 %	 (8,0): 277.5 Hz; 2.50 %

Table 2 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-axisymmetric shape of the loaded tire structure causes a split 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 fixed spindle loaded tire 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. Furthermore, the contact with the ground makes the occurrence of the *axial* and *torsional* mode impossible.

Table 2. Comparison between the modal parameters of a loaded and unloaded tire (size 205/55 R16) mounted on a steel wheel[3]. The spindle rotation and translation is constrained.

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.
...	...	...	...
1st acoustic	225.75 (0.25)	219.02 (0.69)	1st acoustic hor.
		227.35 (0.43)	1st acoustic vert.

5. Vibration behaviour of a 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. Under the assumption of white noise input[11], output spectra can be modelled in a very similar way as FRFs where the deterministic knowledge of the input is replaced by the assumption that the input is white noise. 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. Fig. 4 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). 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) 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).

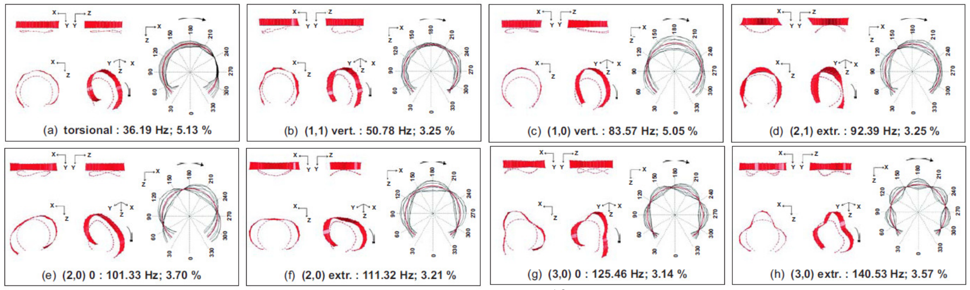


Figure 4. Operational modal parameters of a rolling tire[12] (speed 28.3 km/h, 5 mm high semicircular cleat, inflation pressure 2.2 bar). Mode shapes are shown in a fixed reference system.

5.1 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.*[7]. 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. Table 3 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.

Table 3. 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.	↓	↑	≈	≈	↓	*	↑	*	≈	≈

6. Conclusion

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 (0-500Hz) 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. 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. An innovative tire test bench able to reduce the testing time has been proposed. 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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