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EXPERIMENTAL MODAL ANALYSIS OF RADIAL TIRES UNDER DIFFERENT BOUNDARY CONDITIONS

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Abstract

Tire vibration modes as a characterization of tire dynamics are used in a broad range of tire/vehicle system simulations, ranging from low-frequency vehicle handling to higher frequency structure-borne and air-borne noise simulations. The modal parameters are used either to define a complete modal tire model or to calibrate internal parameters of a tire model to obtain the proper dynamic behaviour. This paper presents experimental modal analyses of non-rotating radial tires under different boundary conditions.

The two considered radial tires are a slick (tire without tread pattern) and a commercial tire, both size 205/55R16. The modal analysis is performed under three different boundary conditions, namely, free, unloaded wheel on a fixed hub and loaded wheel on a vehicle hub. Mode shapes, resonance frequencies and modal damping are assessed up to 300 Hz. Throughout the paper, practical testing considerations and limitations in accuracy and frequency range are discussed. These factors determine the accuracy and performance of tire/vehicle system simulations which use the experimental modal tire parameters as input data. The modal parameters of the tires under the different boundary conditions are compared and discussed. The dynamics of both rim and enclosed air cavity of the tire are also considered in the analysis.

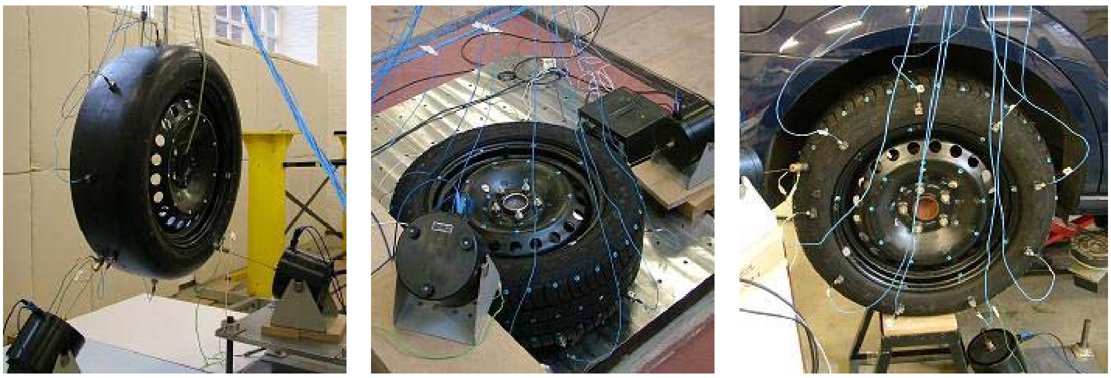
INTRODUCTION

This paper presents the experimental modal analysis of non-rotating radial tires under different boundary conditions. The experimental determination of tire modal parameters has become a major issue in a broad range of tire models. Accuracy of the estimated modal parameters and limitations in frequency range determine the overall performance of tire models. This paper applies the tire mode naming convention described by Wheeler et al. [1], which allows an unambiguous labeling of the tire modes under different boundary conditions. The estimated modal parameters of two different tires under different boundary conditions are compared in order to provide better physical understanding of tire vibration modes.

The measurements were performed at KUL Noise and Vibration Laboratory as a part of the ongoing research on tire/road noise caused by road impact excitations.

EXPERIMENTAL MODAL TESTS

Experimental modal analyses were performed on a slick and a commercial tire, both size 205/55R16 and mounted on a steel rim. A multiple input analysis was used, as this allows a better separation and identification of closely spaced or double poles. The experimental modal analyses were performed in the frequency range 0 - 512 Hz. During all the tests, the tire inflation pressure was monitored and kept constant at 2.2 bar. Three different boundary conditions were considered in the tests, namely, free, unloaded wheel on a fixed hub and loaded wheel on a vehicle hub (see figure 1). The free boundary condition was achieved by suspending the wheel on a soft spring. For the second boundary condition, the wheel was attached to the hub bearing unit of a vehicle. The hub bearing unit itself was fixed to a rigid environment, which allowed the wheel to rotate around its rotation axis. For the last boundary condition, the wheel was attached to the hub bearing unit of a vehicle and loaded (275 kg) by the partial weight of the vehicle. This boundary condition corresponds well to the operating condition of the wheel, apart from the effect of rolling.



(a) *(b)* *(c)*

Fig.1 : Boundary conditions: (a)free ; (b)unloaded on fixed hub; (c)loaded on vehicle hub.

The tire was excited by 2 electrodynamic shakers (Bruel & Kjaer 4809), connected to the tire through a stinger and a 10 mm diameter disk which was glued to the tire surface. One shaker mainly excited the tire belt in radial direction, while the other shaker mainly excited the tire sidewall in axial direction. A burst random signal, with frequency content from 0 to 512 Hz and burst time 80% of the excitation period, was amplified and delivered to the shaker. The input force and acceleration were measured by an impedance head (PCB 288D01). A singular value decomposition of the input power matrix at each frequency showed that the two input signals were sufficiently uncorrelated. The main reason to apply shaker excitation in this setup is the high controllability of the intensity and direction of the excitation.

The measurement grid used in the analysis consists of 378 points (see figure 2) defined in a cylindrical coordinate system. 324 points with a circumferential resolution of 10° are located on the tire surface and the other 54 points with a resolution of 40° are spread over the steel rim. The responses of both tire and rim were considered because under certain boundary conditions, the wheel has modes where rim and tire move or deform relative to each other. The measurement grid for the commercial tire was limited to 270 points, from which 216 points were located on the tire surface.

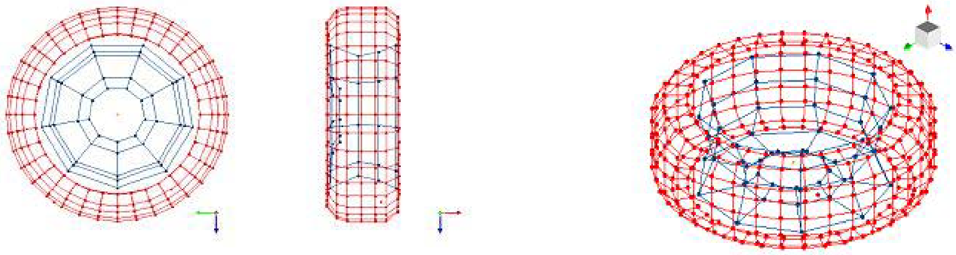


Fig. 2: Measurement grid of the wheel, consisting of tire and rim.

The radial, tangential and axial acceleration responses in each point of the grid were measured with tri-axial accelerometers (PCB 356A15; sensitivity 100 mV/g). The accelerometers were attached to the tire surface with heated beeswax. After the wax was cooled, a light and stiff connection of the accelerometer to the tire surface was achieved. To avoid mass loading of the structure, only 9 accelerometers were used simultaneously and these were distributed evenly over the circumference. The frequency response functions were obtained within the range 0 to 512 Hz with a frequency resolution of 0.5 Hz, out of 20 averages by using a Hanning window. The modal parameters were estimated by the least squares complex frequency domain method (Polymax [2]) from the LMS Test.Lab® software. This is a multiple degree of freedom method which generates a global estimation for the modal parameters. The unity modal mass scaling scheme was used to scale the mode shapes.

RESULTS

An unambiguous naming convention for the modes of an unloaded tire was described by Wheeler et al. [1]. This convention uses two indices which describe the bending order of the belt package in the two directions. The format of the notation is [c,a]. The first index 'c' represents the number of sinusoidal waves in the circumferential direction. The second index 'a' represents the number of waves in the axial direction at a circumferential location where the shape is at an extreme displacement. The above described convention is inapplicable for some of the modes where the belt acts as a rigid structure.

Unloaded slick, fixed hub boundary condition

Figures 3 to 4 show the resonance frequency, modal damping factor and 4 views of the mode shape for each mode of the unloaded slick, mounted on a fixed hub. The three measured rigid belt modes are displayed in figure 3. In the [0,0] axial mode, the belt rigidly translates in axial direction. This mode has a single root due to the axisymmetric mode shape. The next mode is the [1,1] pitch mode in which the belt is rotating around a diametric axis. The [1,0] diametric mode is characterized by a rigid translation of the belt along a diametric axis. Both pitch and diametric mode have repeated roots. The [0,0] torsion mode, which is the fourth rigid belt mode, was not identified in this test setup due to the lack of input in tangential direction.

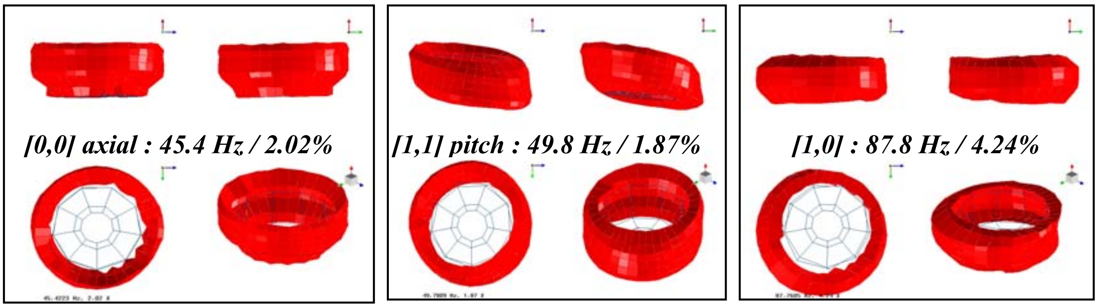


Fig. 3: Rigid belt modes of the unloaded slick mounted on a fixed hub.

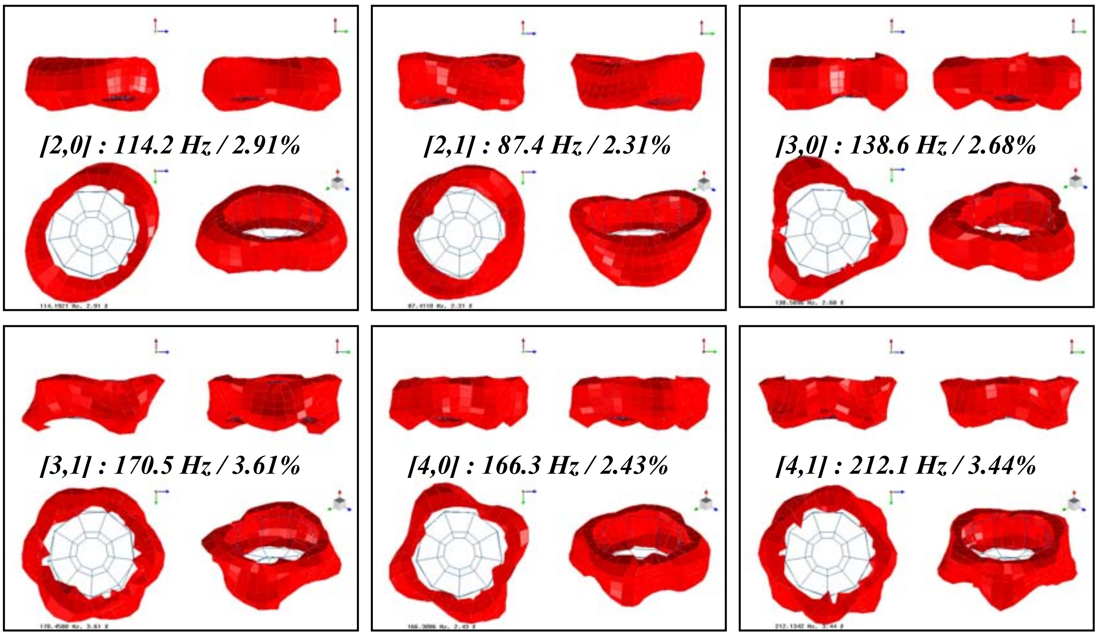


Fig. 4: First six [c,0] and [c,1] modes of the unloaded slick mounted on a fixed hub.

Figure 4 reveals that both the [c,0] and [c,1] modes involve bending of the belt in circumferential direction. Furthermore, the [c,1] modes show twisting of the belt in axial direction. Besides the structural modes of the tire, there are also modes of the tire air cavity and the rim. The [1,0] acoustic mode is the first resonance of the tire air cavity and appears at 224,8 Hz with a modal damping of 0,23%. The naming convention for the tire mode shapes can also be adopted for the acoustic mode shapes, as the acoustic pressure distribution shows a similar pattern of nodes and anti-nodes as the structural tire mode shapes. Due to the strong acoustic-structural coupling [3], the acoustic modes of the tire cavity can be observed as a deformation of the tire surface. The induced tire deformation of the cavity mode [1,0] is very similar to the mode shape of the structural tire mode [1,0], but the modal damping is considerable lower. Besides the air cavity, the rim also has resonances which contribute to the overall dynamic behavior of the wheel. The rim has a cylindrical shape which allows the same naming convention [1] to be used as the tire modes. The first two modes of the steel rim are the [2,1] and [3,1] mode which appear at respectively 186,6 Hz and 343,4 Hz.

All modes of figure 4 have repeated roots. The two resulting mode appear at the same frequency and are of equal mode shape, but have a circumferential shift of a quarter wavelength. The mode shape corresponding to a certain resonance frequency aligns itself in such a way that the excitation takes place in an anti-node. The second mode shape of the same resonance frequency will be poorly excited because the excitation takes place in one of its nodes. However, a real tire is not perfectly axisymmetric, thus causing the two mode shapes to appear at slightly different frequencies. The observed frequency differences between these modes have an average value of 0.73 Hz and never exceed 1.5 Hz in the performed analysis. The comparison of the modal participation factors [4] confirmed that the mode shape with the excitation in an anti-node is excited predominantly. This mode was used to estimate the modal parameters, which yield the most accurate estimation of the modal parameters.

Influence of boundary conditions

Figure 5 shows the natural frequencies as a function of the circumferential wave number for a commercial tire and a slick, both mounted on a fixed hub. The spatial resolution on the commercial tire was too small to identify the higher [c,1] modes. The axial and [1,0] mode of the slick appear at a lower frequency, compared to the commercial tire. As the belt has a rigid motion in those modes and the belt mass is equal for both tires, this indicates a lower sidewall stiffness of the slick.

The difference between the natural frequencies of the two tires increases with an increasing circumferential wave number. The same trend is observed in the comparison between the two tires in the free boundary condition. The circular ring supported by sidewall stiffness [5] is an appropriate analytical model to investigate this trend. The natural frequencies of the model can be expressed as a function of the wave number c, inflation coefficient C_I, radial sidewall stiffness coefficient C_R and tangential sidewall stiffness coefficient C_T :

$$f_c = \sqrt{\frac{\left(c^2-1\right)^2}{c^2+1}\frac{C_I}{4\pi^2} + \frac{c^2}{c^2+1}\frac{C_R}{4\pi^2} + \frac{1}{c^2+1}\frac{C_T}{4\pi^2}} \tag{1}$$

Equation 1 shows that the natural frequencies for lower wave numbers ($c \leq 2$) are determined by the radial and tangential sidewall stiffness. The inflation coefficient C_I is the most important factor for high wave numbers ($c \geq 4$). This indicates that the inflation coefficient C_I of the slick tire is higher compared to the commercial tire. The inflation coefficient C_I is defined as:

$$C_I \approx \frac{p}{R\, h\, \rho_b} \tag{2}$$

As the inflation pressure p is equal for both tires, the product of the ring mean radius R, belt height h and average belt density ρ_b is lower compared to the commercial tire.

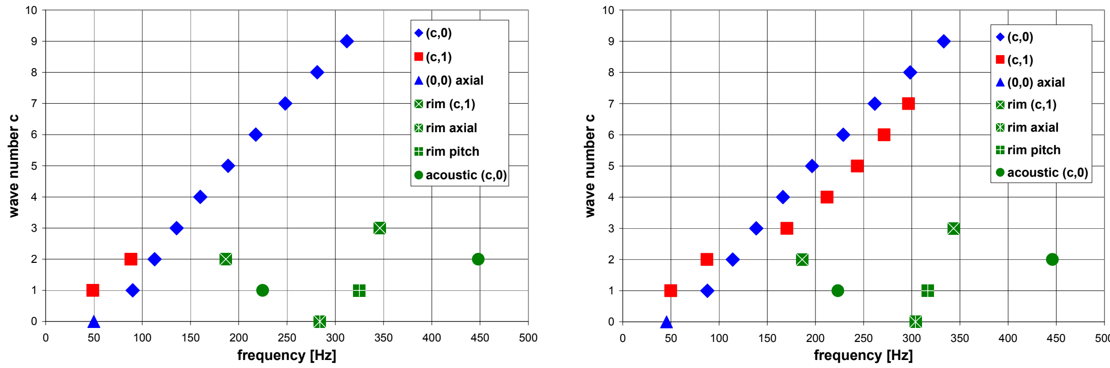


Fig. 5: Unloaded fixed hub boundary condition; commercial tire (left) and slick (right).

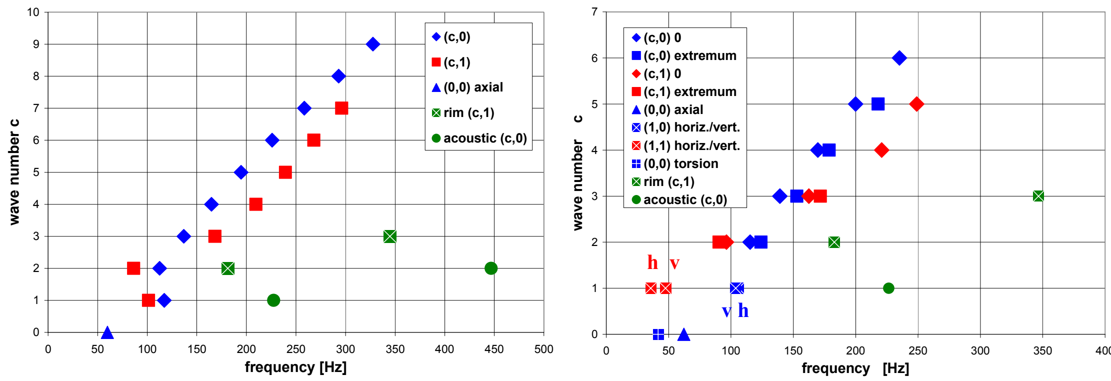
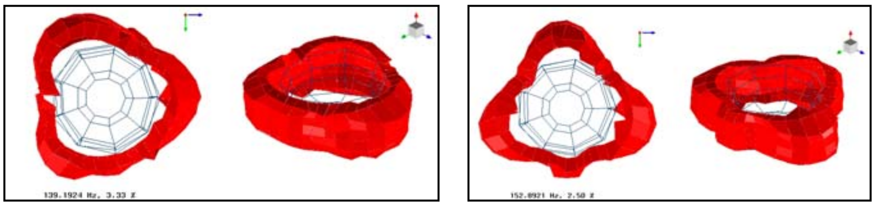


Fig. 6: Free slick (left) and loaded slick on a vehicle hub (right).

The natural frequencies of the free slick and the loaded slick on a vehicle hub are shown in figure 6. Comparison between the free (fig. 6a) and unloaded fixed hub (fig 5b) boundary condition shows that the rigid belt natural frequencies shift. For the fixed hub boundary condition, the rigid belt moves relative to the fixed rim, whereas for the free boundary condition, the rigid belt moves relative to a moving rim. The tire modes which involve bending of the belt are not influenced by the boundary condition of the rim. Figure 6b shows the natural frequencies of the loaded (275 kg) slick on a vehicle hub. This boundary condition destroys the axisymmetry of the tire, thus splitting up the repeated roots of the unloaded tire into two single roots. As both of these modes are related to the same mode of the unloaded tire, an additional distinction in labeling has to be made. A naming convention based on an observed trend in the mode shapes is introduced here. For the first mode shape, the circumferential location where the radial displacement is zero along the cross-section, is located in the middle of the tire/road contact area. This mode is labeled as "**0**". For the second mode shape, the circumferential location where the radial displacement reaches an extremum along the cross-section, is located in the middle of the tire/road contact area. This mode is from now on labeled as "**extremum**". Figure 7 illustrates the naming convention for the [3,0] modes. The mode [1,0] and [1,1] split up in a "horizontal" and "vertical" mode. The rigid belt translates along a horizontal and vertical axis for the [1,0] mode and for the [1,1] mode, the rigid belt rotates around these two axes. Also the acoustic [1,0] mode splits up into a horizontal and vertical mode, although only the vertical acoustic mode was identified in the measurements.



[3,0] "0": 139.2Hz /3.3% ***[3,0] "extr.": 152.9Hz /2.5%***

Fig. 7: [3,0] modes of the loaded slick tire mounted on a vehicle hub.

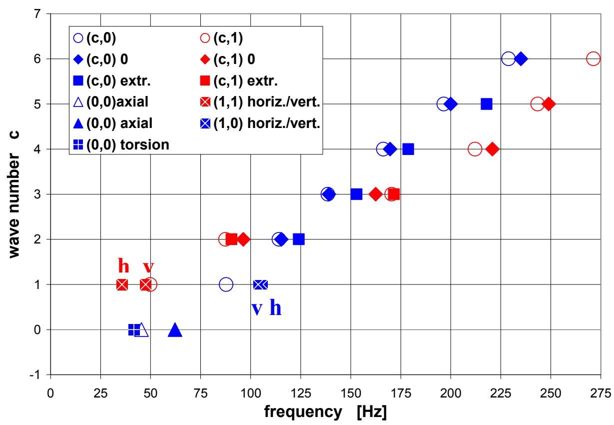


Fig. 8: Unloaded (o, Δ) and loaded (■, ◆, ▲) slick modes.

Figure 8 shows a comparison between the natural frequencies of the unloaded and loaded slick attached to the vehicle hub. Most of the loaded modes appear at higher frequencies and modal damping increases, compared to the unloaded boundary condition. The rigid belt modes are subjected to the highest frequency shift due to loading. The [1,1] mode is the only rigid belt mode which experiences no stiffening effect due to loading. The same trends are observed on the loaded commercial tire.

The unloaded [c,0] modes split up into two modes from which the [c,0] ”0” natural frequency differs little from the unloaded [c,0] natural frequency. The tire/road contact boundary condition has less influence on the [c,0] “0” mode than on the [c,0] ”extremum” mode. However, the influence increases for higher wave numbers, as the circumferential wave length becomes of the same magnitude as the tire/road contact length.

CONCLUSIONS

The ability to identify modes at higher frequencies is limited due to both the high modal density and the increasing complexity of the mode shapes. The measurements show that the [c,0] and [c,1] modes with a circumferential wave number above 9 are difficult to identify. The frequency, on which the circumferential wave number of the mode reaches 9, is largely dependent on the geometry of the tire. Around this frequency the [c,2] and [c,3] modes start to appear, which drastically increase the modal density. Wheeler and al. [1] showed that even for the unloaded boundary condition, modal densities of 2.5 modes per Hz can be expected in certain frequency regions. For the 205/55R16 tire used in this research, modes above 300 Hz were difficult to identify.

The loaded fixed hub boundary condition approaches very well the operating condition of the wheel, but the high modal density makes it difficult to obtain a complete identification of all modes in a certain frequency range.

Simple analytical models have proven to be a useful tool to investigate and explain observed trends in the experimental modal data.

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