Effects of rotation on the tyre dynamic behaviour under different operating conditions

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Summary
In order to increase the understanding of the effects of rotation on the tyre dynamic behaviour and to establish the missing links in the process of accurately predicting and controlling the overall tyre/road noise and vibrations, a European industry-academia partnership project, named TIRE-DYN, has been founded between Goodyear, KU Leuven and LMS International. By means of experimental and numerical analyses, the effects of rotation on the tyre dynamic behaviour are quantified.

This paper presents the results of vibration simulations on a rotating tyre. Modal parameters of the rolling tyre are estimated from a modal analysis. In addition, the dispersion curves, which give detailed insight in the wave propagation behaviour of a structure, are analysed for the rolling tyre. The main objective of these analyses is to obtain a thorough understanding on the influence of rolling on the tyre dynamic behaviour.

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1. Introduction

Despite the best efforts of engineers over the last decades to deepen the understanding of tyre/road noise and vibration phenomena, there are still some missing links in the process to accurately predict the overall tyre/road noise. A significant missing link is represented by the influence of rolling on the tyre dynamic behaviour. For example, it is well known that flexible rotating structures are subjected to gyroscopic effects. These effects are understood for simple structures such as rings and cylindrical shells and can be predicted or modelled correctly [1]. However, the gyroscopic effects for more complicated structures, such as a loaded rotating tyre in ground contact, are found to be more difficult and are not yet fully understood [2]. Already in 1890, Bryan [3] described the rotating modes of a rotating bell or cylinder. After him, many authors have contributed to improve the understanding of the behaviour of rotating structures [4-7]. They range from analytical models [8] to extremely sophisticated numerical models [9, 10]. However, no validated highly detailed model, which includes all the complex tyre/road noise generation phenomena and effect of rotation on the tyre dynamic behaviour, has been proposed.
Moreover, tyre/road noise is the main contributor to road traffic noise at speeds above 40 km/h for passenger cars and above 70 km/h for heavy trucks. Consequently, inside the European seventh framework program, an industry-academia partnership project, named Tire-Dyn [11], has been founded between KU Leuven [12], Goodyear [13] and LMS International [14] to quantify the effects of rotation on the tyre dynamic behaviour [15, 16].
This paper presents a numerical analysis that provides more understanding of the types of waves that propagate in an unloaded (not in contact with the ground) tyre of size 205/55R16. Results are shown for the non-rotating and rotating tyre at different speeds. The next section presents in detail a description of the numerical model which has been used in this paper. Section 3 presents the dispersion diagram of the non-rotating and rotating tyre. In addition, the modal parameters of the tyre are estimated with a Modal Analysis (MA).

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2. Numerical procedure – Finite Element Analysis (FEM).

Based on the commercially available finite element methods to solve gyroscopic eigenvalue problems, the complex eigen-modes of a stationary rolling tyre are shown in this paper. This calculation is performed on an advanced highly detailed (construction and material data) finite element model of a smooth or slick tyre with size 205/55R16. The tyre has a full tread without grooves (pattern). The tyre model includes the air cavity inside the tyre and a rigid rim is assumed. The Finite Element Method (FEM) software used is Abaqus 6.11.1. The model (unloaded and non-rotating tyre) is validated by comparing the measured natural frequencies to the natural frequencies from the Abaqus frequency extraction (*Frequency). The tyre is discretised in 180 segments of approximately 11 cm in the circumferential direction. The tyre cross-section is discretised in around 100 sectors. The mesh size of the model in circumferential and cross-sectional direction is a good compromise between the frequency range of interest and the CPU time. The 3D model is obtained by fully revolving a 2D cross-section. The detailed material data is taken from the Goodyear database. The model takes into account the viscoelastic nature of the tyre components and the vibro-acoustic coupling effects. The boundary conditions used in this model are the following: the wheel centre is free to rotate around its axis and all the other directions are constrained. The Frequency Response Functions (FRF's) and the dynamic response due to a harmonic radial excitation force (applied at a node) are obtained from a direct steady-state harmonic analysis (Abaqus command *Steady State Dynamic, Direct). The direct method makes the analysis significantly more expensive in terms of calculation time, since it computes the steady-state harmonic response directly in terms of the physical degrees of freedom of the model. However, this approach offers the possibility to use visco-elastic material properties of the different compounds as a function of frequency. In the performed simulation the tyre is excited in the radial direction with a point force at the centre of the tread-band and the vibration velocity is calculated at 180 equally spaced points along the tread-band centre line in an Eulerian coordinate system (the nodes of the mesh stay fixed in space while the materials flow through the mesh). Consequently, 180 mobility Frequency Response Functions (ratio between vibration velocity and input force) are obtained from the simulation. The frequency resolution and frequency range chosen for the simulation are 1 Hz and 1-1000 Hz, respectively.

3. Dispersion diagrams and Modal Analysis

The outputs of the simulations are analysed in terms of dispersion curves or frequency-wavenumber spectra and modal parameters estimated from a Modal Analysis.

In this paper the tyre naming proposed by Wheeler *et al.* [16] which uses two indices (n,a) is used; the first index; n represents the number of circumferential bending wavelengths of the belt and it is also known as circumferential mode number (which is equal to wavenumber multiplied by tyre's radius) and the second index; a represents the waves in the axial direction of the belt at a circumferential location where the shape is at an extreme radial displacement.

Based on the 180 simulated FRF's for different tyre rotation speeds, an operational modal analysis is performed to extract the modal parameters: natural frequencies, model damping (Table I) and mode shapes (Figure 2). In addition, the natural frequencies are directly extracted from the dispersion curves. The dispersion curves give the relation between the wavenumber and the frequency of the waves which propagate in the tyre. These dispersion curves are obtained by applying at each frequency a Fast Fourier Transform (FFT) to the 180 mobility FRF's along the complete circumference. By doing that, the spatial domain is converted to the wavenumber domain. It should be noted, that since the excitation force is radial, only waves which respond to this excitation are manifested in the dispersion curves studied in this paper. Hence, in the modal analysis only the modes having significant radial response along the centre of the tread-band are identified, that is: [n,0], [n,2], [n,4], [n,6],… In addition to the resonance frequencies, the modal damping ratios are also available from the modal analysis. The modal parameters have been estimated by using one of the state-of-the-art modal parameters estimators; the Polyreference least-squares complex frequency-domain method (PolyMax [17]) which is implemented in the LMS Test.Lab® software.

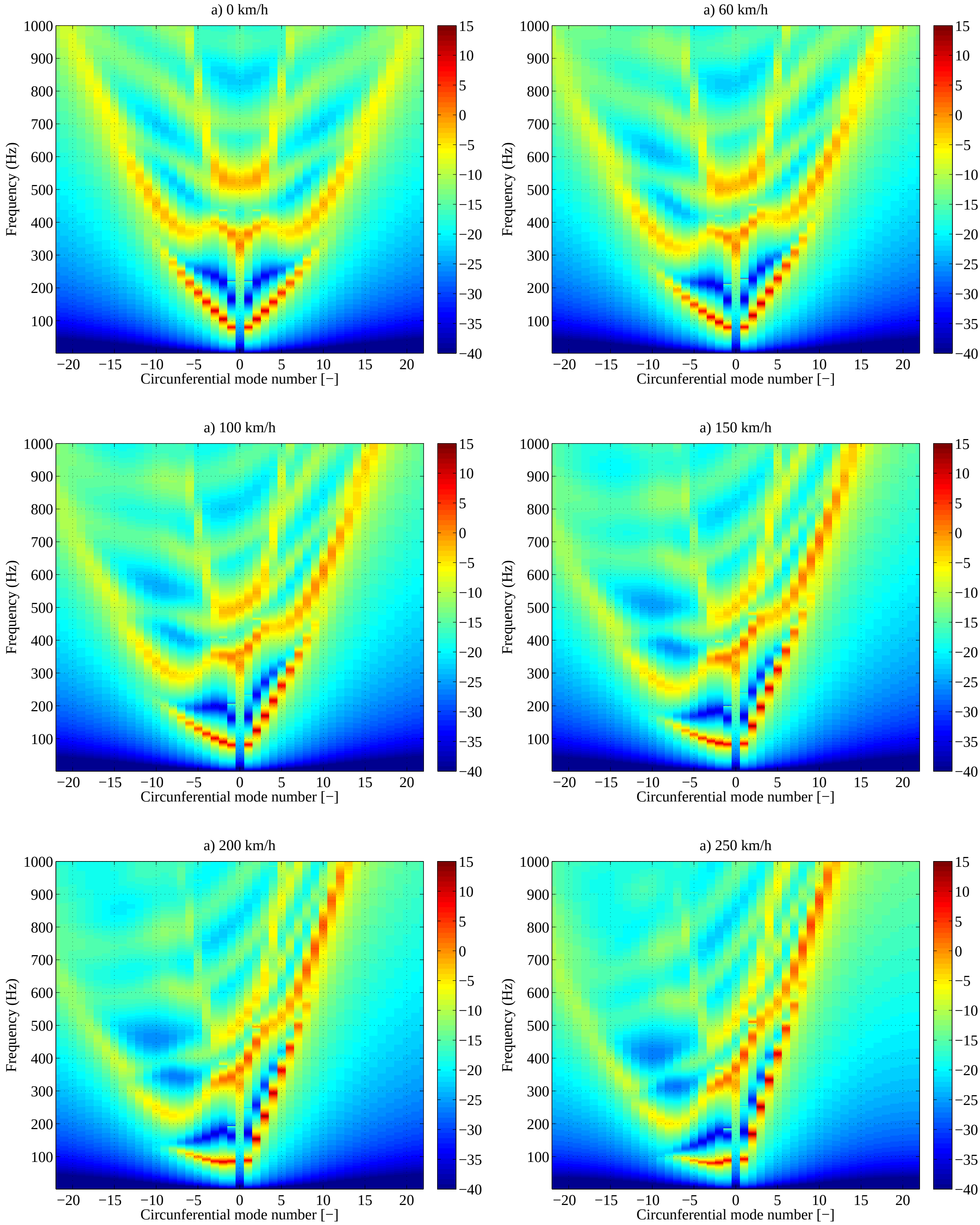

Figure 1. Frequency-wavenumber plots(dB) for different speeds (0km/h, 60km/h, 100km/h, 150km/h, 200km/h, 250km/h). Positive wavenumbers correpond to waves travelling in the tyre rotation direction and negative wavenumbers to waves travelling in the direction opposite to the tyre rotation.

Due to the fact that the unloaded and non-rotating tyre (Figure 1a, Table I and Figure 2 at 0 km/h) is axisymmetric, the poles of the system are double and thus two modes appear at the same frequency. However, that is not the case for the loaded non-rotating tyre (no longer axisymmetric), where a split of the unloaded double modes into two single modes happens. In this paper, only results from an unloaded tyre are presented.

Figure 1 shows the frequency-wavenumber spectra in dB for different rotation speeds; 0 km/h (stationary tyre), 60 km/h, 100 km/h, 200 km/h and 250 km/h. At 0 km/h, the dispersion curve (Figure 1a) shows that the positive- and negative-travelling waves with the same wavenumber have the same frequency ($f_i(k) = f_i(-k)$). At the resulting resonance a standing wave vibration pattern can be observed. Since a tyre can be considered as a ring-like structure, the resonance modes are found at the frequencies where an integer number of wavelengths equals the tyre circumference. When the tyre is rotating, the tyre is subjected to Coriolis acceleration, hence the wave speed of the positive- and negative-going wave diverge from each other. The phase speed of the positive-going wave increases with the rotation speed, while the phase speed of the negative-going wave decreases by the same amount. Consequently, the resonance mode associated with the positive-going wave appears at a different frequency compared to the resonance mode associated with the negative-going wave ($f_i(k) \neq f_i(-k)$) when the tyre is rotating. Thus, each double mode of the stationary tyre is associated with one positive- and one negative-going wave when the tyre rotates. At the resonance frequencies of the rotating tyre, a travelling wave vibration pattern (complex mode shape) can be observed.

Table I. Natural frequencies of the (-) and (+) going waves (Dis=dispersion wave extraction) and associated modal damping ratios for different speeds and modes.

	0 kph			**60 kph**			**100 kph**		
Mode	***Dis(Hz)***	***MA (Hz)***	***ξ(%)***	***Dis(Hz)***	***MA(Hz)***	***ξ(%)***	***Dis(Hz)***	***MA(Hz)***	***ξ(%)***
[1,0]	81	81.1	3.56	81	81.7	3.52	81		
				82			83	82.6	3.46
[2,0]	106	105.9	2.84	96	96.2	3.01	91	90.9	3.05
				117	117.0	2.62	125	125.5	2.45
[3,0]	131	131.3	2.62	112	112.3	2.91	102	101.9	3.04
				153	153.4	2.32	170	17.4	2.10
[4,0]	158	158.1	2.68	131	130.6	3.07	116	115.7	3.24
				191	190.8	2.34	216	215.8	2.05
[5,0]	186	186.4	2.91	151	151.0	3.4	131	131.6	2.60
				229	229.1	2.50	261	261.7	2.17
[6,0]	216	216.4	3.2	173	173.4	3.87	149	149.5	4.14
				268	267.7	2.78	307	308.3	2.42
1stAcous.	220	220.7	0.02	212	212.3	0.03	206	206.6	0.03
				229	229.0	0.03	234	234.6	0.03
2nd Acou.	438	437.6	0.32	420	420.5	0.41	409	409.3	0.49
				454	454.2	0.38	466	465.7	0.42

The values of the natural frequencies and modal damping are summarised in table I and plotted in figure 3, where one can see that the branches of the dispersion curves are tilted due to the rotation of the tyre. The natural frequencies have been extracted in two different ways; from the dispersion curves and by means of a Modal Analysis. The results suggest that both methods to extract the resonance frequencies are in good agreement.

Figure 2 shows that in the stationary tyre, the positive- and negative-going waves interfere with each other at a single natural frequency to produce a mode shape with standing wave pattern. However, when the tyre is rotating, the standing wave pattern will not occur. The positive- and negative-going waves with the same wavenumber appear at different frequencies and thus no interference between the two waves will occur (which is needed to develop a standing wave vibration pattern). For each circumferential mode number n, two resonances are observed at different frequencies. One resonance corresponds to a travelling wave in the positive direction and