

Modeling and Experimental Investigation of Tire Cavity Noise Generation Mechanisms for a Rolling Tire

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ABSTRACT

Tire cavity noise refers to the excitation of the acoustic mode of a tire cavity. The noise exhibits itself as sharp resonance-like peaks with frequencies typically in the range of 190-250Hz. For a rolling tire, the tire contact with the road moves relative to the tire. Furthermore, the load on the tire breaks the circular symmetry of the tire. Consequently, the peak frequency of the cavity noise shows dependence on the tire load and the vehicle speed. There are no models that simultaneously take these two factors into consideration. In this paper, we propose an analytical model and present experimental verifications of predictions on the noise peak frequency and its dependence on the tire load and vehicle speed. A wireless experimental measurement system is also presented which enables the measurement of tire cavity frequency for both non-rolling and rolling conditions. The results of this study show that this analytical model can be used to predict tire cavity noise frequency for a rolling tire to avoid the potential modal alignment issue for a tire/wheel assembly in the early design stage.

INTRODUCTION

The dynamics of the tire and its interaction with the road, the vehicle suspension, and the surrounding air contribute to the overall noise of a vehicle [1]. The structural vibration of a tire has been widely studied in

the literature. Recent works are focused on the dynamic effect of the tire rotation. Relentless efforts by the automotive industry to improve the noise and vibration characteristics of passenger vehicles have led to the overall reduction of the noise. In the meantime, the noise associated with the first acoustic mode of the tire air cavity has been found to be an important noise source [2-6]. Depending on the tire dimensions, the noise exhibits itself as very sharp resonance peaks with frequency typically in the range of 190-250Hz. With the gradual overall reduction of the vehicle noise, tire cavity noise becomes more annoying and efforts to mitigate this noise have been underway for some time [6]. Recent works are focused on the dynamic effect of rotation [7-11].

Tire acoustic noise has been attributed to the air cavity inside the tire. Standing waves inside the torus shaped air cavity can be excited by the road surface roughness. The frequencies of the standing waves are determined by the sound speed and the length characterizing the circumference of the air cavity. Since the fluctuation of the sound speed is very small for air at different temperatures and pressures that a tire is subjected to, the acoustic frequencies are well predicted based on the tire geometry. Moreover, the resonances exhibit very narrow bandwidth typical of systems with very low damping.

It has been shown that cavity noise can be controlled by adding damping material to the tire cavity [6]. This approach adds complexity to tire manufacturing and vehicle maintenance. It would be ideal if cavity noise can be eliminated through steps taken at the design stage. Although the resonance frequencies of the acoustic cavity have been well predicted [3-5], challenges remain in developing a theoretical model that correctly accounts for the two main factors affecting the cavity noise.

Firstly, cavity noise is affected by the tire rotation rate. More specifically, acoustic and dynamic measurements inside the vehicle show sharp peaks at frequencies which appear to depend on the vehicle speed. Secondly, the static load on the tire also affects the frequencies at which peak acceleration and sound pressure levels are recorded.

In this paper, we propose a phenomenological model for the cavity acoustic modes. The model includes the effect of the tire static load and is derived in the coordinate system fixed to the rotating tire. The resulting equations are then transformed to a stationary reference frame. The resonance frequencies and transmissibility are obtained. Measurements of the dynamic pressure inside tire cavity are obtained using a wireless sensor. Tire transmissibility is also measured under varying vehicle speed.

MODELING OF THE CAVITY ACOUSTIC MODES

The acoustic wave inside the tire cavity is governed by the wave equation. The complicated geometry of the tire cavity makes it impossible to obtain analytical solution of the wave equations. Since only the lowest acoustic modes have been identified as the contributors to the cavity noise, we represent the cavity modes using a phenomenological model. For a stationary (non-rotating) tire, the cavity modes are defined as fore/aft acoustic mode along the horizontal direction and vertical acoustic mode as shown in Figure 1. Each mode can be represented by a linear oscillator as shown below:

$$\ddot{X} + 2\zeta\omega_0\dot{X} + (\omega_0^2 + af_0)X = 0 \quad (1a)$$

$$\ddot{Z} + 2\zeta\omega_0\dot{Z} + (\omega_0^2 + bf_0)Z = 0. \quad (1b)$$

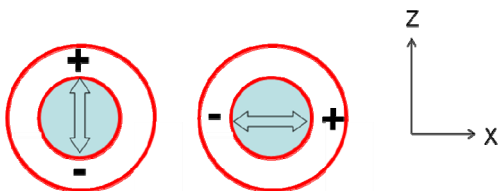


Figure 1. The two cavity modes of a stationary tire

Notice that, associated with each acoustic mode, the air molecules inside the tire move from one place to another. The dynamic variables X and Z may be visualized as the center of mass that characterize the motion of the air molecules. In the above equations,

ω_0 is the natural frequency of the acoustic cavity of the unloaded tire. It has the degeneracy of two for a perfectly symmetric tire. The coefficient ζ is the damping ratio. The tire is assumed to be subjected to a static load, f_0 , acting on the outer surface along the Z direction. The circular symmetry is broken by the tire load, causing the tire overall profile to shrink vertically and elongate horizontally. The natural frequencies of these two modes are thus affected. The coefficients a and b represent the effects on the natural frequencies of the two acoustic modes by the tire load. Experimental data show that, in agreement with our intuition, a is negative and b positive. The parameters in the phenomenological model can all be determined experimentally.

For a non-rotating tire, $\Omega = 0$, the static load f_0 causes the frequencies of the two cavity modes to split into

$$\omega_H = \sqrt{\omega_0^2 + af_0} \quad (2)$$

and

$$\omega_V = \sqrt{\omega_0^2 + bf_0}. \quad (3)$$

The frequencies ω_H and ω_V refer to the resonance frequencies of a stationary loaded tire along the fore/aft and vertical directions respectively. From the experimentally determined values of ω_H and ω_V , we obtain:

$$\frac{af_0}{\omega_0^2} = \left(\frac{\omega_H}{\omega_0}\right)^2 - 1 \quad (4)$$

and

$$\frac{bf_0}{\omega_0^2} = \left(\frac{\omega_V}{\omega_0}\right)^2 - 1. \quad (5)$$

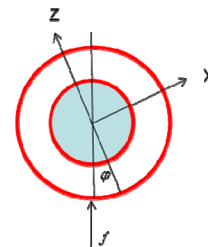


Figure 2. The cavity modes defined in a coordinate system fixed to the rotating tire. The tire load exerts a skewed force on the two modes.

When a tire rotates by an angle φ , the acoustic modes must be described by a coordinate system fixed to the tire. In this coordinate, Figure 2, the tire load is skewed. To obtain the equations of motion that correctly account for the effects of the tire load, we use the coordinate transformation:

$$x = X \cos \varphi + Z \sin \varphi$$

$$z = -X \sin \varphi + Z \cos \varphi, \quad (6b)$$

to obtain the following equations for the acoustic modes in the coordinate system fixed to the rotating tire:

$$\ddot{x} + 2\zeta\omega_0\dot{x} + (\omega_0^2 + af_0 \cos^2 \varphi + bf_0 \sin^2 \varphi)x - (a-b)f_0 \sin \varphi \cos \varphi z = cf_r \sin \varphi \quad (7a)$$

$$\ddot{z} + 2\zeta\omega_0\dot{z} + (\omega_0^2 + af_0 \sin^2 \varphi + bf_0 \cos^2 \varphi)z - (a-b)f_0 \sin \varphi \cos \varphi x = cf_r \cos \varphi \quad (7b)$$

where f_r represents the random dynamic component of the tire-road interaction force f . That is:

$$f = f_0 + f_r \quad (8)$$

The coefficient c is a scaling factor; it can be determined from experimentally obtained transmissibility.

Equations (7a) and (7b) can be put in the following form:

$$\ddot{x} + 2\zeta\omega_0\dot{x} + [\tilde{\omega}_0^2 + \frac{1}{2}(a-b)f_0 \cos 2\varphi]x - \frac{1}{2}(a-b)f_0 \sin 2\varphi z = cf_r \sin \varphi \quad (9a)$$

$$\ddot{z} + 2\zeta\omega_0\dot{z} + [\tilde{\omega}_0^2 - \frac{1}{2}(a-b)f_0 \cos 2\varphi]z - \frac{1}{2}(a-b)f_0 \sin 2\varphi x = cf_r \cos \varphi \quad (9b)$$

where

$$\tilde{\omega}_0 = \sqrt{\omega_0^2 + \frac{1}{2}(a+b)f_0} \quad (10)$$

Substituting (4) and (5) into the above we have:

$$\tilde{\omega}_0 = \sqrt{\frac{1}{2}(\omega_v^2 + \omega_H^2)} \quad (11)$$

For a tire rotating at a constant speed Ω , equations (9a) and (9b) contain terms with periodic coefficients in the form of $\cos 2\Omega t$ and $\sin 2\Omega t$. Tire rotation rate is slow compared with the natural frequencies of the lowest acoustic mode. If these terms with periodic coefficients are ignored, equations (9a) and (9b) represent two uncoupled forced oscillators whose natural frequencies are $\tilde{\omega}_0$. Even though the static load splits natural frequencies of two cavity modes, the cavity may be regarded as having two modes whose frequencies are slowly changing with time. These two slowly changing frequencies have the same average frequency at $\tilde{\omega}_0$. Therefore, a pressure sensor fixed to the tire will experience maximum sound pressure at frequency at $\tilde{\omega}_0$, independent of the tire rotation rate.

We note that the static force between the tire and the road has two effects. First, since the static tire load is skewed, the acoustic modes defined in the coordinate fixed to the rotating tire are coupled. Second, the dynamic component of the tire load acts as direct forcing on each mode and the component on each mode is again dependent on tire rotation.

We also note that the forcing terms on the right hand side of the equations are multiplied by $\sin \varphi$ and $\cos \varphi$ respectively. When the tire rotates at a constant speed Ω , the forcing frequencies will be affected by Ω . For instance, if $f_r = \cos \omega_i t$, the cavity modes are excited by the sum of two harmonic functions with frequencies $\omega_i - \Omega$ and $\omega_i + \Omega$ because of the following trigonometric identities:

$$\cos(\omega_i t) \sin(\Omega t) = \frac{1}{2} \{ \sin[(\omega_i + \Omega)t] - \sin[(\omega_i - \Omega)t] \} \quad (12)$$

$$\cos(\omega_i t) \cos(\Omega t) = \frac{1}{2} \{ \cos[(\omega_i + \Omega)t] + \cos[(\omega_i - \Omega)t] \} \quad (13)$$

Therefore, a single frequency force input from the tire-road interaction excites the two acoustic modes at two different frequencies. This complicates the study of the dynamic responses of the tire cavity mode.

An alternative set of equations can be derived that circumvent the above complications. The cavity modes illustrated in Figure 1 are associated with the pressure distribution in the tire cavity. We may now regard the variables x and z as describing the center of mass of the air inside the cavity with respect to the coordinate system fixed to the rotating tire. The motion of the "center of mass" of the air cavity generates mechanical forces that are transmitted by the tire wheel. Since the acoustic modes are defined with respect to the rotating tire, the resulting forces on the wheel when viewed in a non-rotating coordinate system change directions. Simple vector relationships lead to the following forces

$$F_x = \alpha [x \cos(\Omega t) - z \sin(\Omega t)] \quad (14a)$$

$$F_z = \alpha [x \sin(\Omega t) + z \cos(\Omega t)]. \quad (14b)$$

After we have solved equations (9a) and (9b), we may substitute the solutions to (14a) and (14b) to obtain the forces acting on the vehicle as perceived by an observed inside the vehicle.

Alternatively, instead of solving equations (9a) and (9b), we introduce the following change of variables:

$$X = x \cos(\Omega t) - z \sin(\Omega t) \quad (15a)$$

$$Z = x \sin(\Omega t) + z \cos(\Omega t) \quad (15b)$$



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