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On the design and optimization of acoustic network resonators for tire/road noise reduction

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ABSTRACT

In this work we propose a numerical method for the calculation of resonant frequencies of network resonators and we also present an optimization method based on genetic algorithms to get targeted resonant frequencies of the network resonators. We can optimize parameters of the network structure such as junction types and end positions. Experiments are conducted on optimized wooden network resonators to validate the method. Good agreement is found between the measured and targeted resonant frequencies. Applications to tire/road noise are considered.

1. Introduction

Tire treads and road textures in the contact zones between tires and roads can be considered as acoustic network resonators. Consequently, the acoustic fields around the tire/road systems are influenced by network resonances. These network resonances in the contact zone in Fig. 1 are seen as one of the noise enhancement mechanisms in [1]. Since the network resonators in the contact zone have large influence on the acoustic fields around their resonant frequencies, the network resonators should be investigated in detail. First studies of some simple acoustic resonators are reviewed as follows.

Besides porous materials and perforated panels, narrow quarter-wave tube resonators are also widely used for the sound absorption in a wall or panel for a narrow frequency band based on the resonance of air inside the tube and the viscous shear and thermal conductivity losses on the tube walls. The model by Zwikker and Kosten [2] for wave propagation in cylindrical tubes included the viscosity and thermal conductivity. Tijdeman [3] proved that this model is complete and accurate for both narrow and wide tubes. Eerden [4] studied the influence of the viscous and thermal conductivity losses on the absorption coefficient and concluded that the viscothermal effects cannot be neglected if the resonators are used for sound absorption because they result in energy being dissipated and the effective speed of sound inside the tube can be considerably reduced. Around the resonant frequencies, we can see a maximum sound absorption. The theory and applications of quarter-wave resonators are summarized in [5].

The quarter-wave tube has an open and a closed end, but resonators with two open ends can also be used for the sound absorption, especially for the case where air needs to be transported through walls or one needs to see through the wall. Eerden [4] studied this case, and concluded that at low frequencies ($f < 2000$ Hz) the waves propagating in the resonator are not absorbed at the end but are reflected back into the resonator due to the mass reactance at the free end. For higher frequencies (2000–10,000 Hz) the waves are absorbed due to radiation into infinity. In order to create broadband sound absorption, coupled tube resonators with different cross-sectional areas and lengths can be applied. The mechanism for the broadband absorption is that the sound energy is dissipated by the viscothermal effects and the incident waves are cancelled due to the broadband resonance of air in the coupled resonators.

Helmholtz resonators (HRs) are also used to control the noise inside enclosures in many studies. Helmholtz resonators can be considered as a mass-spring system. The spring stiffness is represented by the volume of air and the mass is given by the small column of vibrating air in a perforation of the panel. The energy can be dissipated by the vibrating air and the porous material placed in the volume. See [6–10] for different applications of these Helmholtz resonators to noise reduction. More specifically T-shaped acoustic resonators can be seen in many studies for noise control in small enclosures, see for instance [11–16] for models and experimental results on these resonators.

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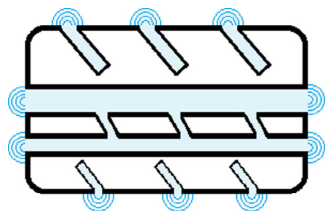

Fig. 1. Network resonators in the contact zone between a tire and a road.

If we are interested more specifically in tire noise, today the noise due to the vibrations of a rolling tire can be calculated with convincing accuracy. However, air pumping is not understood very well. In [17] Hayden described the air movement in the contact zone between a rolling tire and a road. Air is squeezed out when the treads at the entrance of the contact zone are compressed on the road surface, and flows into the voids when the treads lift up from the road surface. Daffayet et al. [18] measured the pressure in cylindrical cavities over which a smooth tire rolled. They assumed that the noise is generated by opening and closing the cavities in the contact zone. Ronneberger [19] thought that air was displaced by the changing gaps between the tire and road surfaces, because the treads are deformed by road roughness. These sources are located in the contact zone between the tire and the road and the sound is modified by the horn effect.

Horn effect is an essential noise enhancement mechanism. The tire/road system can be seen as a horn-like structure. The surfaces of the tire and the road constitute horns in front of and behind the contact zone. The noise generated in the contact zone is amplified by the multiple reflections between the tire surface and the road surface which are acoustically reflecting surfaces. The amplification of the horn effect reaches up to 10–20 dB in the results of previous studies, where the road and the tire are modeled with smooth surfaces. A first attempt at an analytical description of the horn effect was made by Ronneberger [20]. Kropp et al. [21] suggested a theoretical model based on multipole synthesis. The model can provide a reasonable prediction of noise levels at mid and high frequencies for a tire placed on a hard surface. However, it overestimates the horn amplification effect at low frequencies. Graf et al. [22,23] first investigated experimentally the horn amplification of sound generated by a simple acoustic source. The boundary element method is then shown to give predictions. The dependence of the horn-effect on different geometrical parameters is also investigated both through experiments and boundary element calculations. It shows that for the intermediate frequency range the BEM provides an excellent tool to calculate the horn effect for practical geometries. The aim of the work by Anfosso et al. [24,25] is also to predict the amplification due to horn effect. Sound pressure amplification of a 2D infinite rigid cylinder is obtained using the analytical approach based on modal decomposition of sound pressure. It gives quick and accurate results, but is limited to simple geometrical configurations and purely reflecting properties of boundaries. In [26] Fadavi et al. deal with the horn effect using a 3D cylinder tire model. The sound pressure and sound amplification are calculated in the space around the 3D tire model using the Boundary Element Method. The influence of different parameters such as the position and size of the source are studied in terms of amplification and sound pressure spectrums. All these studies are made for smooth roads and tires and do not take into account the real geometry of the tire or the road.

In this work, we want to estimate the influence of non smooth geometries on the horn effet. For this, we focus on network resonators and use several assumptions for the networks. There are only right-angled junctions in the networks. The pipes in the networks have the same cross-section. The networks could have open or closed ends. For the open ends, end corrections depend on flange shapes. So, first, methods for the calculation of end corrections will be introduced in Section 2. Next, in Section 3, a numerical method for the calculation of resonant frequencies of network resonators will be developed. Then, in Section 4 an optimization method will be proposed to get the targeted resonant frequencies. Some examples of the application of the optimization method are shown in Section 5 while comparisons with experimental measurements are given in Section 6. Last some conclusions will be given.

2. Determination of end corrections

The length is an important parameter for calculating the resonant frequencies of a network with open ends. A short distance should be added to each end of the network to get precise results. This short distance is called the end correction, which makes each straight part of the network a little longer than its physical length.

From the perspective of waves, standing waves occur during the network resonances. The sound waves are reflecting at open ends, which are not perfectly at the end sections of the network, but at small distances (end corrections) outside the network.

The end corrections of the network open ends can be obtained from the radiation impedances which have small but finite values by (1) from [27].

$$\tilde{\delta} = \mathbf{Re}\left[k^{-1}\arctan\left(\frac{-Z_r}{i\rho c}\right)\right] \tag{1}$$

The upper script $\sim$ means that it is a frequency-dependent quantity. Here only the real part of the end correction is considered, which is the most useful in the present study. To estimate the end corrections of the network, the radiation impedances of the open ends should be calculated first by the impedance transfer equation of an acoustic transmission line (2) from [27], because Z_r cannot be calculated or measured directly at the pipe end.

$$Z_r = -i\rho c\tan\left[\arctan\left(\frac{-Z_l}{i\rho c}\right) - kl\right] \tag{2}$$

Z_l is the impedance at an abscissa $x = -l$, i.e., at a distance l from the open end. It can be calculated as Dalmont did using a BEM numerical method [27].

Eq. (2) means that the radiation impedance can be obtained from the case where Z_l is an input impedance of a pipe of length l. Then the end corrections can be obtained by (1). The values of end corrections depend on flange shapes. An open pipe end with different flange shapes has different end corrections. In this section, end corrections of a network with complex flanges are calculated by Dalmont's methods or by the BEM. The flanges are a round surface and a plane surface (see Fig. 2), which will be used in Section 4. The network is shown in red. Some pipes are identified by the numbers one to three shown in green.

Only the end corrections of longitudinal pipes will be discussed. For transverse pipes they can be calculated approximately by Dalmont's fit formula (3) for rectangular flanges because the flange of transverse ends is flat.

$$\delta_{sq} = \delta_{sq\infty} + \frac{a_{sq}}{b_{sq}}(\delta_{sq0} - \delta_{sq\infty}) + 0.057\frac{a_{sq}}{b_{sq}}\left[1 - \left(\frac{a_{sq}}{b_{sq}}\right)^5\right]a_{eff} \tag{3}$$

with $\delta_{sq\infty} = 0.811a_{eff}$ and $\delta_{sq0} = 0.597a_{eff}$, in which $a_{eff} = 2a_{sq}/\sqrt{\pi}$. Here, $2b_{sq}$ is the flange width, respectively. b_{sq} is the shorter one of the two distances between the transverse end and the sides AD or BC in Fig. 2 for which the value of b_{sq} for pipe number 2 is shown. $2a_{sq} = 0.009$ m is the pipe width which is small and not shown in the figure.