



Conference Article

Tire Cavity Noise Reduction by Using Helmholtz-Based Sandwich Resonator

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Abstract

With the elimination of internal combustion engines in electric vehicles, noticeable changes have occurred in the in-cabin noise profile. The absence of engine noise has made road and tire-induced noises more prominent, leading to the emergence of passive acoustic issues such as cavity noise. Cavity noise is a distinct type of noise that negatively affects interior comfort, caused by the resonance of the enclosed air volume between the wheel and the tire.

In this study, tire-induced noise types are first classified in general terms, and then the physical basis of cavity noise is explained through the Helmholtz resonator model. Existing solutions in the literature are examined, and as an alternative, a modular and highly volume-efficient sandwich resonator design that can be integrated into all wheel types is proposed. This design differs from similar studies by maximizing the utilization of the gap volume between the wheel and the tire and being easily adaptable to different wheel geometries.

Keywords: Helmholtz resonator, sandwich resonator, cavity noise, tire–road noise

1. Introduction

In recent years, electric vehicles have played a key role in reshaping the automotive sector. With the absence of traditional engine noise, sounds resulting from tire and road contact have become more prominent within the cabin.

This trend is also supported by research in the literature comparing vehicle noise components at different speed ranges. In the research conducted by Bernhard and Wayson (2005), it was shown that as vehicle speed increases, the proportion of tire-generated noise in the overall external noise rises rapidly, as shown in Fig.1.[1]

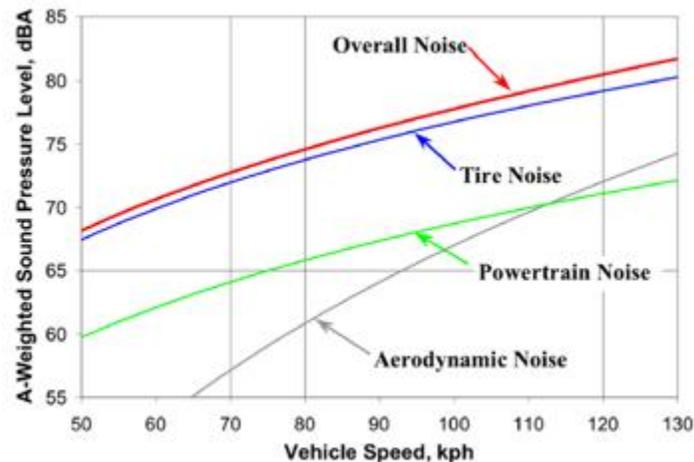


Fig.1. Contributions of various sub-sources of highway traffic noise [1]

Tire-related noises transmitted into the cabin are generally classified into two categories based on their transmission paths: structure-borne and airborne noises. Structure-borne noises occur when vibrations generated by tire-road interaction are transmitted to the passenger cabin through the wheel, axle, suspension, and body. On the other hand, airborne noises arise from sound waves travelling through the air and entering the cabin directly via windows or body cavities. Both mechanisms contribute to interior noise, which becomes more noticeable in electric vehicles due to their quiet operating nature [2].

Chang et al.(2010) [3] indicated that tire becomes the dominant noise source for interior noise when the speed is over 80 km/h , and the structure-borne noise source occurs mainly at low frequencies (below 500 Hz) while the airborne one occurs at high frequencies (500-2000 Hz) (Lopez et al., 2007 [4]). Kitahara et al.(2011)[5] reported the same results, as shown in Fig.2 [6].

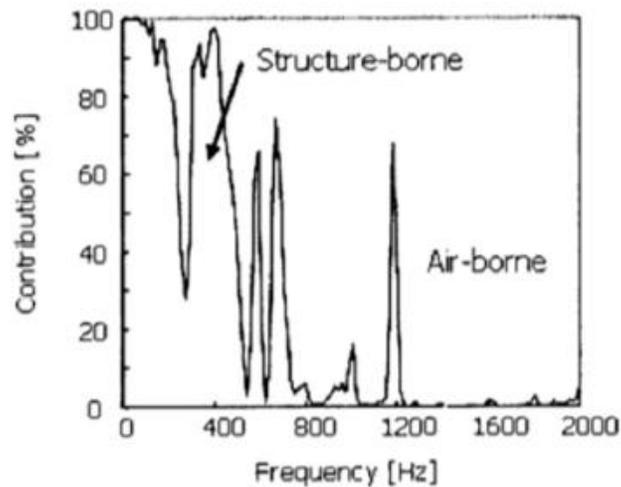


Fig.2. Contribution of tire structure-borne and airborne noise for interior noise [5].

One of the main components of interior noise is the resonance sound generated in the air cavity inside the tire or wheel. When this air volume is excited by vibrations caused by the tire's contact with the road surface, it resonates and produces sound at a specific frequency.

This phenomenon is called "cavity noise" in the literature and is a type of low-frequency and disturbing noise that is especially effective in the 180–250 Hz range [7].

The mechanism of resonance formation can be explained by the Helmholtz resonator model in acoustics. In this case, when the air column inside a closed cavity passes through a narrow neck, it oscillates and creates resonance at a specific frequency. Similarly, the air volume inside the tire or wheel behaves in the same way and generates this resonance frequency when subjected to vibration.

This type of noise reduces driving comfort and becomes much more noticeable in electric vehicles without engine sound. This sound, perceived as "thin" and "muffled," is transmitted to the interior via both structure-borne and airborne pathways [8]. Controlling cavity noise is a critical engineering challenge for improving the NVH performance of electric vehicles.

As shown in Tables 1 and 2 [5], many solutions have been proposed so far to reduce tire noise. These include optimizing tire tread patterns, using porous materials inside the tire cavity, integrating Helmholtz resonators into the wheel, and designing structures that prevent rim-cavity resonance coupling. However, most of these approaches have not seen widespread application due to factors such as geometry-specific designs, difficulties in integrating into the production process, and limited use of available cavity volume.



Table 1. Summary of Tire Noise Reduction Approaches in the Literature [5]

Category	Reference	Company	Modification	Method	Reduction Effect
Tread vibration	Iwao and Yamazaki, 1996 [9]	Nissan Motor Co., Ltd. (Japan)	Tire tread	Attachment of a rubber ring on the inside surface of the center part of the tread surface	5 dB (800-1600 Hz)
Tread vibration & cavity resonance	Saemann et al., 2011 [10]	Continental Reifen Deutschland GmbH (Germany)	Tire tread and cavity	Application of seal and foam absorber to the inside surface of tread band	7.5 dB (230-240 Hz)
Air pumping, pipe resonance	Zhou, 2013 [11]; Zhou et al., 2014 [12]	Jiangsu University (China)	Tread pattern	Reduction of fluid drag force and noise by using the bypass structure and bionic tread groove	> 10 dB (800-1500 Hz)
	Kakumu, 1990 [13]	Sumitomo Rubber Industries (Japan)	Tread pattern	Circumferential length of contact patch is substantially equal to the transverse groove pitch multiplied by an integer	5 dB
	Cusimano, 1992 [14]	Bridgestone/Firestone, Inc. (USA)	Tread pattern	Strategic placement of grooves such that the amount of groove void across the trailing and/or leading edges of the footprint is substantially uniform about the circumference of the tire	N/A
	Continental AG, 2016[15]	Continental Reifen Deutschland GmbH (Germany)	Tread pattern	(1) "Harmonic Comfort Chambers" based on the "Helmholtz resonator" positioned on the inner shoulder of the tire pattern (2) "0' dB-Eaters" uniquely shaped in-groove elements designed to split and diffuse noise waves for lower road noise	N/A
Cavity/rim coupling resonance	Fitz and Heck, 2001[16]	Epilogics Group (USA)	Rim	Lightweight steel rim to shift the modal frequency of the tire rim outside of 200-250 Hz	Ineffective (shift down 1 Hz)
	Sainty et al., 2012 [17]	RMIT University (Australia)	Tire tread	Extrusion of three strips of rubber from the tire into the cavity to shift the modal frequency of the tire cavity	Marginal (shift down 18 Hz)
	Sainty et al., 2012 [17]	RMIT University (Australia)	Rim and cavity	Attachment of elastic ring on rim with separator fins which extends into the cavity due to centrifugal forces	Effective (shift up 156 Hz)



Table 2. Summary of Tire Noise Reduction Approaches in the Literature [5]

Category	Reference	Company	Modification	Method	Reduction Effect
Cavity resonance	Molisani et al., 2003 [18], [19]	Virginia Tech & Michelin North America, Inc. (USA)	Rim	Incorporation of secondary acoustic cavities to detune and damp out the main tire cavity resonance	15 dB force transmission (230 Hz)
	Kamiyama, 2014 [20]	Honda R&D Co. Ltd. (Japan)	Rim	Assembly of separate thin, lightweight plastic resonators in the wheel well	10 dB (190-230 Hz)
	Fernandez, 2006 [21]	KTH University (Sweden)	Rim	Implementation of a Helmholtz resonator attached to the rim	Obvious (205-240 Hz)
	Sainty et al., 2012 [17]	RMIT University (Australia)	Cavity	Introduction of a sound absorption material	14 dB (225 Hz)
	Yukawa et al., 2004 [22]	Sumitomo Rubber Industries (Japan)	Cavity	Gluing of a foam layer to the inner liner beneath the tread	Obvious (interior noise)
	Pirelli, 2013 [23]	Pirelli & C. SpA (Italy)	Cavity	Pirelli Noise Cancelling System (PNCS, P ZERO™): polyurethane sponge inserted into the cavity to absorb the vibrations	2-3 dB
	Mohamed and Wang, 2015 [24]	RMIT University (Australia)	Tire	Placing a trim layer onto the inner surface of the tire tread	10 dB (225 Hz)

These limitations create the need for a more practical, adaptable concept.

In this study, a modular sandwich resonator design -originally developed within TOFAŞ design studies and used with permission- is examined to address the cavity noise problem. The design aims to make maximum use of the volume between the wheel and the tire and can be integrated into all wheel sections. It focuses on the resonance frequency from a theoretical perspective while also offering flexibility in terms of manufacturing and application.

2. Design and Methods

In this section, the modular Helmholtz-based sandwich resonator design developed to reduce tire cavity noise in electric vehicles is explained in detail. The design is intended to be applied to the space between the wheel and the inner surface of the tire and has been optimized for manufacturing simplicity, acoustic efficiency, and installation flexibility.



2.1 Structural Components

The resonator design consists of five main elements. The **resonator body** has a hollow, enclosed structure made of a rigid yet lightweight and non-metallic material, such as hard plastic or a composite polymer. Its internal volume and neck length are optimized to generate the target resonance frequency while maintaining compatibility with the tire’s inner surface.

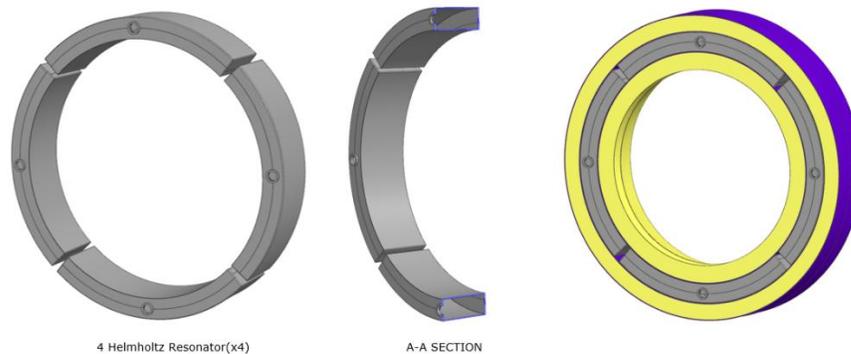


Fig.3 Patent Number: TR2021019729A2

The acoustic foam layers are positioned on the upper and lower surfaces of the resonator. These layers are made of open-cell, sound-absorbing material that reduces vibrations and dissipates unwanted acoustic energy within the cavity.

Finally, the bi-adhesive is used to bond the foam layers securely to the resonator’s upper and lower surfaces. This industrial adhesive provides strong adhesion and flexibility, resulting in a three-layer sandwich structure that combines rigidity, acoustic efficiency, and safe integration within the tire–wheel interface.

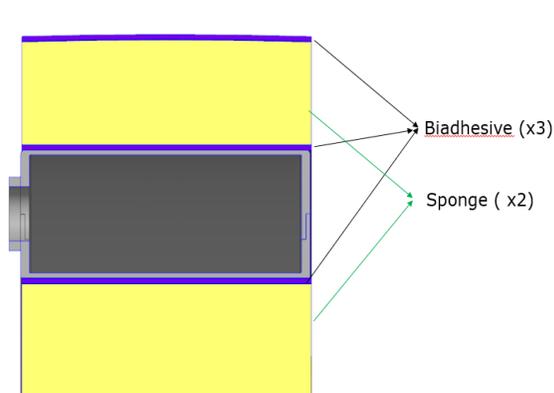


Fig.4 Patent Number: TR2021019729A2

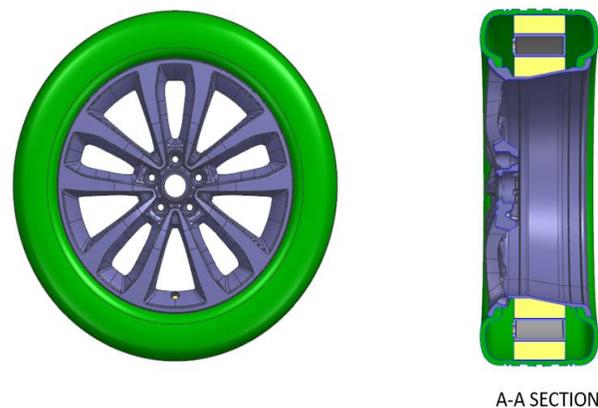


Fig.5 Patent Number: TR2021019729A2



2.2 Theory

The Helmholtz resonator can be explained using a simple mass–spring system. In this approach, the air inside the neck behaves like a moving mass, which can be written as $m = \rho(AL)$. At the same time, the compression and expansion of this air column provide an elastic restoring effect, similar to a spring. Based on classical vibration theory, the natural oscillation frequency of a mass–spring system with spring constant k is given by [25]:

$$f_r = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (1)$$

When the same concept is applied to a Helmholtz resonator, and air is assumed to behave as an ideal gas under adiabatic conditions, the resonance frequency can be expressed in terms of the resonator geometry. The resulting Helmholtz resonance frequency is given by [25]:

$$f_r = \frac{c}{2\pi} \sqrt{\frac{A}{VL_e}} \quad (2)$$

In this equation, c is the speed of sound in air (343 m/s at STP), A is the area of the opening, V is the volume of the resonator, and L_e is the effective neck length, [25]

$$L_e = L + \alpha \sqrt{A} \quad (3)$$

End Correction

In the Helmholtz resonator model, the neck length is not limited to its geometric length L . Due to the additional acoustic mass effect that occurs at the ends of the neck, it behaves as if it were effectively longer. This phenomenon is known as end correction. First analytically described by Rayleigh (1877) and later refined by researchers such as Levine & Schwinger (1948), this approach expresses the additional length at the neck's end as a coefficient proportional to the neck radius.[25]

- For the unflanged end, $\rightarrow \Delta L \approx 0.6r$
- For the flanged end, $\rightarrow \Delta L \approx 0.82r$

Combining equations (2) and (3) ;

$$f = \frac{c}{2\pi} \sqrt{\frac{A}{(L+0.8\sqrt{A}) * V}} \quad (4)$$

3. Result

This work focuses on the theoretical basis and modular geometry of the resonator design. Acoustic and structural performance evaluations are still in progress and will be shared in follow-up publications.

Table 3. Parameters of Proposed Helmholtz Resonator

Frequency	f	121.5	Hz
Speed of Sound	c	344	m/s
Length of neck	L	8	mm
Diameter of neck	D	5	mm
Cross sectional area of neck	A	20	mm^2
Base chamber inner arc radius	r_i	220	mm
Base chamber width	w	10	mm
Base chamber outer arc radius	r_o	230	mm
Arc angle	ϕ	88	deg
Base chamber height	h	100	mm
Volume of chamber	V	345,575	mm^3

This table presents the key geometric parameters considered in the evaluation of the Helmholtz resonator geometry. The listed cavity dimensions, neck geometry, and arc parameters illustrate how the chamber volume was increased and the neck characteristics were optimized to achieve a target resonance frequency of 121.5 Hz theoretically.

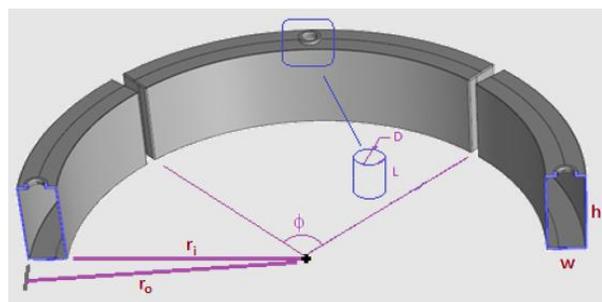


Fig.6 Patent Number: TR2021019729A2

In order to reduce the resonance frequency, it is necessary to increase L and V . It is recommended to reduce R .



The examined sandwich resonator design offers a unique engineering solution to reduce interior air cavity noise, which is becoming increasingly prominent in electric vehicles.

This design, which is modular in structure, optimized in terms of materials, and combines both resonance-dampening and sound-absorbing properties in terms of acoustic performance, goes beyond methods with limited applicability in the existing literature.

However, it should be noted that the system may also introduce certain engineering challenges in terms of long-term durability and maintenance. Therefore, future studies should focus on detailed evaluations of the design's durability and its integration into service and maintenance processes.

Such a comprehensive assessment will enhance the practical applicability of the proposed solution and help elevate the NVH performance of electric vehicles to a new level.

4. Discussion and Conclusion

This section evaluates the engineering advantages and potential challenges of the proposed design.

The main advantages of the system can be summarized as its modular structure, ease of assembly, wide adaptability, and acoustic effectiveness. The resonator is designed to be easily integrated into the tire-wheel assembly line without requiring additional robotic operations. Thanks to the flexible nature of the foam layers and the compact geometry of the resonator, it can be installed manually with minimal effort. Its modular configuration also allows the design to be adapted to wheels of different sizes and tire profiles, while the resonator's internal volume and neck length can be adjusted according to the target frequency range. From an acoustic perspective, the system not only targets cavity noise within the 180–250 Hz band but also benefits from the foam layers that help absorb mid- and high-frequency components, further improving the overall interior noise performance.

Before Assembly

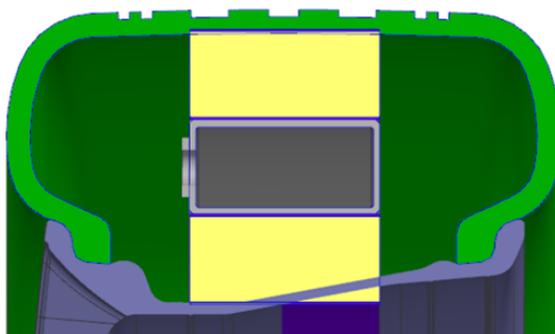


Fig.7 Patent Number: TR2021019729A2

After Assembly

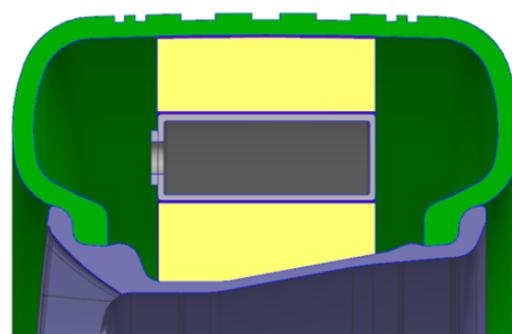


Fig.8 Patent Number: TR2021019729A2



Despite these advantages, several potential challenges must be considered. A sudden drop in tire pressure or a tire burst could place abnormal loads on the resonator, affecting its structural integrity. In addition, removal and reinstallation during tire replacement may complicate repair and maintenance operations. Thermal effects caused by brake heating and temperature fluctuations inside the wheel could also lead to long-term material fatigue or deformation. Finally, the use of extra foam and adhesive materials may increase both production cost and assembly time.

In conclusion, while the proposed system offers high engineering feasibility and strong acoustic benefits, further optimization is required for its integration into large-scale production processes.

5. Acknowledge

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