Design of soundproof panels via metamaterial concept

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Abstract

THE goal of the work is to find a way to improve the sound insulation properties of different types of panels in order to meet different requirements. Inspired by the nontrivial behavior of the locally resonant acoustic metamaterials, this concept is introduced into the design of structures in order to explore the potential ways to improve the sound insulation behavior in the relevant specific frequency regions.

At relatively low frequency region when the bending wavelength is much longer than the distance between isolated resonators, which is also the interesting frequency range in the most part of the work, it may be assumed that the effects of the resonators are uniformly distributed over the entire surface. An impedance approach is hence proposed to estimate the sound transmission loss of the metamaterial panels in order to get more insights from physics. This is realized, in general, by integrating the equivalent impedance of the resonators together with the corresponding impedance of the host panel. Valuable theories are derived based on that, laying a solid foundation for effective/efficient design of metamaterial panels. This approach also provides a fast and reliable tool for the designs prior to a time-consuming and computationally expensive numerical simulation.

Based on that, a new design for locally resonant metamaterial sandwich plates is proposed to improve the sound transmission loss performance in the coincidence frequency region. A systematic method to tune the resonance frequency of local resonators is developed. This approach also supplies a method to remove the possible side-dips associated with the resonance of the resonators. The influence of the sound radiation from the resonators is further investigated with the Finite Element models. It is proposed to embed the resonators inside the core material in order to eliminate the possible influence, and also to make a smooth surface. The metamaterial sandwich panel designed in this way combines improved acoustic insulation properties with the lightweight nature of the sandwich panel.

Besides the coincidence frequency region, the ring frequency area of a cylindrical shell is another important frequency region for bad sound transmission loss. The effectiveness of locally resonant metamaterial is also investigated. Similar to the case of the flat panel, both impedance model and Finite Element model are developed for the problem of the sound transmission loss properties. The influence of the resonators is presented, and compared with the case of the flat panel. Unlike the case of the metamaterial flat panel, two side-dips around the sharp improvement cannot be avoided when applying the resonators near the ring frequency of the curved panel. The reason for that is explored by using the impedance approach. It is noticed that, while the impedance of a flat panel near the critical frequency is shifted from a mass-type impedance to stiffness-type impedance, the impedance of a cylindrical shell is shifted from a stiffness-type (tension-type) impedance to mass-type

impedance. For a traditional mass-spring type resonator, however, the equivalent impedance is always shifted from a mass-type impedance to stiffness-type impedance when the frequency crosses the resonance frequency. Therefore, when the traditional resonators are applied near the ring frequency, there are always frequencies at which the impedances cancel each other, resulting in the worsened sound transmission loss. In order to have better improvement of the sound transmission loss in this frequency region, new types of resonators have to be developed.

A locally resonant metamaterial curved double wall is proposed and studied, with the aim of addressing the mass-spring-mass resonance and ring frequency effects of the wall. The sound transmission loss properties of a curved double wall are first investigated by introducing the concept of 'apparent impedance', which expresses the properties of the entire structure in terms of the impedances of the constituting panels and air cavity. The apparent impedance derivation is validated against Finite Element models. The curved double wall is then specifically designed by adjusting the two characteristic frequencies to be close to each other in order to narrow the region associated with a poor transmission loss. This enables, subsequently, to improve the transmission loss in this region by effectively inserting tuned local resonators. The design principles are discussed, and applications for double walls consisting the same curved panels or different curved panels are both included.

Keywords: sound insulation, sound transmission loss, sound radiation, locally resonant metamaterial, sandwich panel, curved panel, double wall, curved double wall, coincidence frequency, ring frequency, mass-spring-mass resonance, flexural wave

Sammanfattning

MOTIVATIONEN av avhandlingen är att förbättra ljudisoleringsprestandan hos olika typer av paneler för att möta olika applikationsbehov. Inspirerad av det akustiska metamaterialets lokalt resonansbeteende, syftar denna forskning till att införa ett sådant begrepp i utformningen av olika typer av paneler för att utforska de potentiella sätten att förbättra ljudisoleringsbeteendet, särskilt i samband med ljudöverföringsförlusten, av dessa paneler i relevanta specifika frekvensregioner.

För detta ändamål föreslås ett impedansförfarande för att uppskatta ljudöverföringsförlusten hos metamaterialpanelerna. Ett sådant impedans-tillvägagångssätt utvecklas i allmänhet genom införande av en ekvivalent impedans associerad med resonatorerna i motsvarande impedans hos värdpanelen. Vidare kan värdefulla teorier härledas försedda med impedansanalysen. Impedansinriktningen kan inte bara tillhandahålla ett snabbt och pålitligt verktyg för konstruktionen före en tidskrävande och beräkningsmässig dyr numerisk simulering utan ocksåtydlig fysisk inblick i panelernas vibroakustiska beteende och därmed lägga en solid grund för effektiv design av metamaterialpaneler.

Påden här grunden föreslås en ny design för lokalt resonansrik metamaterial sandwichplattor för att förbättra ljudöverföringsförlustens prestanda i sammanfallningsfrekvensområdet. En systematisk metod för att ställa in resonansfrekvensen hos lokala resonatorer utvecklas för att övervinna sammanfallets fenomen. Denna metod förklarar dessutom förmågan att övervinna dipet i samband med resonansen av resonatorerna. Effekten av det utstrålade ljudet från resonatorerna undersöks ytterligare med Finite Element-modellerna. Den föreslagna nya sandwichdesignen framgår av dessa analyser och inkapslar resonatorerna inuti kärnmaterialet. Den föreslagna metamaterials sandwichpanelen kan kombinera förbättrade akustiska isoleringsegenskaper, samtidigt som sandwichpanelens lätta natur upprätthålls och dess goda mekaniska egenskaper.

Förutom den sammanfallande frekvensregionen undersöks även resonatorns påverkan påljudöverföringsförlustuppträdandet vid ringfrekvensområdet. För detta ändamål studeras ett cylindriskt skal som en representation av krökta paneler ur teoretisk och numerisk synvinkel med ett specifikt fokus påtransmissionsförlustbeteendet runt ringfrekvensområdet när skalet är monterat med lokala resonatorer. Påverkan från resonatorerna presenteras och jämförs med det för en platt platta. I stället för den extraordinära förbättringen som observerats för plattformen med metamaterial, genererar tvåkonventionella resonatorer till ringfrekvensen hos krökta paneler tvåsidospring trots en skarp förbättring vid själva ringfrekvensen. Detta fenomen förklaras av en effektiv impedanssynpunkt. Tillvägagångssättet och de slutsatser som lämnas kan därefter möjliggöra utformning av lämpliga resonatorer för att lösa ringfrekvenseffekten för krökta paneler.

En lokalt resonansk metamaterialskurvad dubbelvägg föreslås och studeras, i syfte att adressera väggens massfjäderresonans- och ringfrekvenseffekter. Egenskaperna för ljudöverföringsförlusten hos en krökt dubbelvägg undersöks först med användning av begreppet "uppenbar impedanssom uttrycker egenskaperna hos hela strukturen med avseende påimpedanserna hos de bildande panelerna och luftkaviteten. Den uppenbara impedansderivationen är validerad mot Finite Element-modeller. Den krökta dubbelväggen är dåspeciellt utformad genom att justera de tvåkarakteristiska frekvenserna för att vara nära varandra för att begränsa regionen som är förknippad med en dålig överföringsförlust. Detta möjliggör därefter att förbättra överföringsförlusten i denna region genom att effektivt infoga avstämda lokala resonatorer.

Keywords: ljudisolering, ljudöverföringstab, ljudstrålning, lokalt resonansmetamaterial, sandwichpanel, krökt panel, dubbelvägg, krökt dubbelvägg, sammanträffande frekvens, ringfrekvens, massfjäder-massresonans, böjvåg

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Zibo Liu,
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Papers included in the thesis

A Song, Y., Feng, L., **Liu, Z.**, Wen, J. and Yu, D., 2019. Suppression of the vibration and sound radiation of a sandwich plate via periodic design. *International Journal of Mechanical Sciences*, 150, pp.744-754.

B **Liu, Z.**, Rumpler, R. and Feng, L., 2018. Broadband locally resonant metamaterial sandwich plate for improved noise insulation in the coincidence region. *Composite Structures*, 200, pp.165-172.

C **Liu, Z.**, Rumpler, R. and Feng, L., 2019. Investigation on sound transmission through a locally resonant metamaterial cylindrical shell. *Accepted for publication in the Journal of Applied Physics.*

D **Liu, Z.**, Rumpler, R. and Feng, L., 2019. Locally resonant metamaterial curved double wall to improve sound insulation at the ring frequency and mass-spring-mass resonance. *Submitted to Journal of Sound and Vibration.*

Division of work between the authors

Paper A was primarily written by Yubao Song. Zibo Liu participated in the work and associated discussions, and developed the Finite Element model for the calculations. Papers B-D were written by Zibo Liu under the supervision of both supervisors.

Papers not included in the thesis

E **Liu, Z.**, Rumpler, R. and Feng, L., A finite element model for the vibroacoustic analysis of plate and sandwich structures. *BNAM 2016 - Baltic Nordic Acoustic Meeting 2016*, Stockholm, Sweden, June 2016.

F **Liu, Z.**, Feng, L. and Rumpler, R., Design of broadband acoustic metamaterials for low-frequency noise insulation. *Acoustcs'17*, Boston, Massachusetts, July 2017.

G **Liu, Z.**, Feng, L. and Rumpler, R., Investigation on the acoustic behaviour of a locally resonant metamaterial curved panel. *25th International Congress on Sound and Vibration 2018, ICSV 2018: Hiroshima Calling*, 2018, Vol.6, p.3409-3416, Hiroshima, Japan, July 2018.

Contents

Nomenclature

STL	Sound transmission loss
c_0	Speed of sound in the air
d	Thickness of the air cavity of the double wall
D	Bending stiffness of the panel
E	Young's modulus
f	Frequency
f_{co}	Coincidence frequency of the panel
f_{cr}	Critical frequency of the panel
f_{res}	Resonance frequency of the resonator
f_{ri}	Ring frequency of the curved panel/shell
f_{Sco}	Coincidence frequency of the sandwich panel
$f_{\mathrm{msm}}^{\mathrm{cd}}$	Mass-spring-mass resonance frequency of the curved double wall
$f_{\mathrm{msm}}^{\mathrm{d}}$	Mass-spring-mass resonance frequency of the double wall
i	$i = 1,\ 2$ indexing the panels that constituting a double wall
j	$\mathrm{j} = \sqrt{-1}$
k	Wavenumber $k = \omega/c_0$
m	Surface mass density of the panel
m_{r}	Surface mass density of the resonator
p_{inc}	Incident acoustic pressure
p_{trans}	Transmitted acoustic pressure
R	Radius of curvature of the curved panel/shell
t	Thickness of the panel
w	Displacement of the host panel
$\hat{W}_{\mathrm{inc}}$	Incident power
$\hat{W}_{\mathrm{trans}}$	Transmitted power
$x,\ y,\ z$	Cartesian coordinate system

Z	Impedance of the panel
Z_{a}	Specific impedance of the air
Z_{eff}	Effective impedance of the metamaterial panel
Z^{c}	Impedance of the curved panel
Z^{cd}	'Apparent impedance' of the curved double wall
Z^{d}	'Apparent impedance' of the double wall
Z^{s}	Impedance of the single-leaf panel
Z^{sa}	Impedance of the sandwich panel
$Z^{\mathrm{r}}_{\mathrm{eq}}$	Equivalent impedance of the resonators

δ	Mass ratio of the metamaterial panel
η_{r}	Damping of the resonators
θ	Elevation angle
κ	Bending wavenumber
ν	Poisson's ratio
ξ	Displacement of the resonator
ρ_0	Density of the air
σ	Surface ratio of the metamaterial panel
τ	Transmission coefficient
φ	Azimuth angle
ω	Angular frequency

Chapter 1

Introduction

1.1 Background and motivation

NOISE pollution is one of the increasingly serious problems that pose a threat to human health according to the report from the World Health Organization [1, 2]. The reduction of noise pollution is of vital importance and is a big challenge encountered by modern engineering. Generally, there are three aspects that may be taken into account in order to mitigate this severe issue, *i.e.*, reducing the level of sound sources, blocking 'sound transmission paths', or, protecting receivers. In particular, airborne sound transmission is often critical in terms of the sound transmission paths. More consideration is thus needed to be given into the technique of blocking the airborne sound transmission path. Therefore, for common structures in engineering, good airborne sound insulation performance shall be considered during the design phase.

In practice, different types of structures are required in different situations. Single-leaf and double-leaf panels are commonly adopted as a partition wall in *e.g.* a residential building, while sandwich structures, due to their good mechanical properties, are more widely used in modern engineering such as shipbuilding. In addition, the panel can be made into a curved panel as needed, which is also a common type of panel in aerospace industry.

The pioneer work in characterizing the sound insulation properties of panels may be found in references [3–5]. Based on these references, the classic mass law is proposed and the importance of the coincidence effect is recognized. The coincidence effect occurs at the frequency where the trace wavelength in the surrounding medium matches the bending wavelength in the panel such that the shielding panel is virtually acoustically transparent to the incoming sound waves, leading to a scenario of total transmission if there is no damping in the panel. These pioneer contributions, along with other early studies [6–9], laid a solid foundation for establishing an effective model to describe the sound transmission behaviour through acoustic panels. Since then, a large amount of research has contributed to this field in order to not only achieve a more physically accurate description of the phenomenon, but also improve the acoustic performance of acoustic panels by understanding these physical phenomena.

The sound insulation performance of a single-leaf panel is subjected to the mass law. Hence, it will become impractical in engineering to achieve high sound insulation in the low frequency region without adding extensive mass. As a common solution, it is prevalent to employ a double-leaf panel, a structure normally consisting two single-leaf partitions with an air cavity in between, *e.g.*, double-glazed windows, in order to obtain a high sound insulation performance. For literature on sound transmission through double-leaf panels, see [8, 10–14]. These studies have shown that improved sound insulation properties may be achieved using a lightweight double-leaf partition compared to a single-leaf panel having the same surface density. Nevertheless, a severe deterioration of sound insulation may be induced by the mass-air-mass resonance of a double-leaf panel, which may not be suppressed properly even by adding absorbent materials in the air cavity. Consequently, it is necessary to design the mass-air-mass resonance to be lower than the frequency range of interest [15].

In addition to the double-leaf partitions, the demands of modern industry drive the structure to have a higher structural rigidity and a lower weight. Sandwich panels, consisting a lightweight core and two stiff skins, are therefore widely used due to their high stiffness to mass ratio. However, these good static properties may come at the expense of the sound insulation properties, which has become an essential issue in engineering designs as the demand grows for a better acoustic environment. The bad sound insulation properties of a sandwich panel often occurs in the coincidence frequency region, which is usually much wider than for a single-leaf panel. In addition, the coincidence frequency region of sandwich panels often drops into the audible frequency range. One reason for these undesirable acoustic properties is that the bending stiffness of the sandwich plate decreases with increasing frequencies. As a result, a range of coincidence frequencies may appear instead of a typical coincidence frequency for a single-leaf panel, that is, the bending wave speed in the sandwich panel and the trace wave speed in the surrounding medium may remain close to each other over a broader frequency range. It is therefore of great interest to improve the acoustic properties of such sandwich panels. Extensive research has been conducted over the past few decades with respect to the acoustic behavior of sandwich panels [16–30]. Based on these fundamental studies on the acoustic behaviour of sandwich panels, various methods have been proposed in order to improve the sound insulation properties of sandwich panels. Kurtze *et al.* proposed a 'shear wall' of which the coincidence frequency region is designed outside the frequency range of interest by the control of the stiffness of the panel [16]. Lang *et al.* [19] and Moore *et al.* [22] improved this technique based on further understanding of the dynamic behaviour of sandwich panels, such that the coincidence frequency of a sandwich panel may be controlled in a more flexible way. Other researchers have attempted to improve the sound insulation performance in the coincidence region by added damping, which includes both the use of increased structural damping [20], and the increase in shear of the core which may lead to a high effective damping [23]. Nevertheless, due to ever-increasing acoustic demands, it remains desirable for the industry to further improve the acoustic performance while maintaining the mechanical advantages of sandwich panels. However, due to the increasing level of noise sources, the above conventional methods may no longer be able to meet industrial requirements. Therefore, it is necessary to seek for new ways in order to enhance the acoustic behaviour of sandwich panels.

In the aerospace industry, the shape of the panels used in structures typically requires a certain curvature. This type of panels is known as curved panels. In regard to the acoustic properties of curved panels, Koval made an in-depth discussion in his series of articles, with a specific focus on the phenomena of sound transmission though cylindrical shells [31–36]. Following his work, other researchers further developed the theoretical descriptions either in a more simplified way or to more general cases [37–40]. Liu *at al.* investigated the effects of ring frames and stringers on sound insulation properties of a curved panel [41] and conducted the experiments under a overpressurized condition [42]. Other related research contributions include characterizing the sound transmission properties of curved double walls [43–46] and curved sandwich panels [47–49]. Based on these aforementioned studies, in general, in addition to the coincidence frequency, there is another frequency that may cause a poor sound insulation effect for curved panels, which is the ring frequency. This frequency corresponds to the frequency at which the longitudinal wavelength in the material is equal to the circumference of the structure [50]. Traditional method such as increasing damping is not that effective for solving the acoustic performance near such problematic frequency. In reference [41], a solution was proposed in which the several acoustic insulation caused by the ring frequency may be slightly improved by adding ring frames and stringers, but at the expense of lowering sound insulation performance in higher frequencies. Other methods such as the addition of dynamic vibration absorbers or Helmholtz resonators may overall slightly improve the sound insulation [51]. However, whether the technique is effective in solving the sound insulation problem when applied to the ring frequency was not discussed. Therefore, further analysis of the acoustic properties of curved panels, in particular in the ring frequency regions, may be of great help in finding new ways to potentially address the bad sound insulation caused by the ring frequency effects.

In summary, it is necessary to develop innovative ways to improve the acoustic properties of the common acoustic panels mentioned above while maintaining the engineering advantages of these panels. And these are based on in-depth physical insights into the sound insulation behaviour of the panels. Recently, some research contributions have been devoted to the design of composite structures in order to obtain the improved sound insulation properties. These composite structures are achieved by mounting an attachment structure, *e.g.*, a heterogeneous blanket [52], the distributed vibration absorbers, or, the Helmholtz resonators [51,53] to the host structure. These studies may provide some clues for the development of innovative soundproofing acoustic panels.

Acoustic metamaterials

The concept of metamaterial has drawn tremendous attention since its first introduction by Smith *et al.* [54]. Since then, the metamaterial technology and the associated applications have grown rapidly. Liu *at al.* extended this concept to the field of acoustics, and for the first time realized acoustic metamaterials through the built-in locally resonant composite structure [55].

Acoustic metamaterials may be defined as an artificial composite, which may, at some targeted frequencies, exhibit unusual acoustic properties that are not found in nature [56,57]. With these features, acoustic waves may be manipulated in unusual ways in order to achieve functions that are difficult to achieve with traditional methods [56, 57], such as acoustic cloaking, sound focusing, extreme sound insulation, etc... The unusual acoustic properties, *e.g.*, negative mass density [58–61], negative Bulk modulus [62], or, double-negative properties [63,64], are mainly due to the local resonance or Bragg scattering. Accordingly, the metamaterial may be classified into locally resonant or Bragg type. As for the locally resonant acoustic metamaterial, it is usually realized by the periodic distribution of the local resonators, provided that the size of the period is much smaller than the acoustic wavelength. It should be noted that metamaterials do not have to be periodic. As long as the distance between the resonators is much smaller than the acoustic wavelength, the properties of the metamaterial may be displayed near the resonance frequency in a macroscopic view. The periodic arrangement is only an implementation for the locally resonant metamaterial that is easier to be analyzed and manufactured. Generally, a mass-spring system is a type of implementation of the resonators. Over the last few years, other types of resonators are also proposed such as membrane type resonators [59, 65, 66] or Helmholtz resonators [61, 67]. Although the locally resonant acoustic metamaterials have shown good acoustic properties in some cases, such properties associated with the resonance may only be found in a limited frequency band. Nevertheless, the concept of acoustic metamaterial has presented new opportunities for improving the sound insulation performance of acoustic panels. Indeed, several research contributions have been made for the metamaterial acoustic panels to achieve enhanced acoustic properties, particularly sound insulation properties [27, 28, 68–77].

1.2 Research objectives

This thesis, with the aim of improving the sound insulation performance of panels, is thus devoted to further investigating the acoustic behaviour of locally resonant metamaterials in order to find new ways to address poor sound insulation properties associated with certain frequency regions of the corresponding panels. A metamaterial panel typically consists of a host panel and resonators that are periodically mounted on the host panel, wherein this research, the local resonators are realized by periodically distributed mass-spring systems. The sound insulation performance of the panel may be improved near the resonance frequency of the resonators, *i.e.*, the working frequency band. One of the limitations of this technique is how to extend the working frequency band since the resonance occurs only in a very narrow frequency range, which has been one of the most challenging difficulties in the design of metamaterial acoustic panels. It is proposed by previous researchers to have multiple resonators with different resonance frequencies in one lattice in order to overcome the limitation [68]. However, a disadvantage of this method is that the resonance effect is weakened by the presence of multiple resonators. In addition, it is difficult to implement within a small lattice, making such approach impractical for the low frequency sound insulation. On the other hand, although in some cases the resonance may lead to an extraordinary sound insulation in certain frequency regions, in other cases these resonances may result in worse sound insulation, which needs to be avoided during the design phase. Therefore, it is significant to properly characterize the acoustic behaviour of the resonators, in order to address the deficiencies in the certain frequency regions, *i.e.*, the coincidence associated with sandwich panels, the mass-air-mass resonance associated with double-leaf panels, or, the ring frequency associated with curved panels, The investigation may further support the design of metamaterial panels with good noise insulation properties.

The objectives of the thesis may thus be derived from the previous description, as follows: *i*) Develop theoretical and numerical methods in order to study the sound transmission loss properties of metamaterial panels; *ii*) Provide reliable physical explanation of the acoustic insulation performance of metamaterial panels; *iii*) Propose potential solution or explanation in order to tackle the poor sound insulation properties associated with certain frequency regions.

1.3 Thesis outline

This thesis is organized in five chapters as follows:

- Chapter 1 is the introduction.
- Chapter 2 is a brief description of the methodology adopted in this research, including the theoretical investigation based on an impedance approach and the numerical simulation based on the Finite Element method.

- Chapter 3 introduces the sound insulation behaviour of metamaterial panels at the coincidence region. A criterion is proposed in order to address the coincidence effect. A design of the metamaterial sandwich panel is proposed with which the potential radiation from the resonators may be suppressed.
- Chapter 4 introduces the sound insulation behaviour of metamaterial curved panels at the ring region. The reasons are provided for the fact that the conventional resonators are insufficient to address the ring frequency effect. Furthermore, resonators with prospective properties are illustrated.
- Chapter 5 proposes a metamaterial design for the curved double wall which may display good sound insulaiton properties over a broad frequency range. An 'apparent impedance' approach is developed in order to estimate the sound transmission loss of curved double walls.
- Chapter 6 draws conclusions and makes recommendations for future work.

Chapter 2

Methodology

The methodology introduced herein is mainly used to study the airborne sound insulation properties of metamaterial panels. The sound insulation performance of the panel is characterized by the corresponding sound transmission loss. The research is carried out based on both theoretical investigation and numerical simulation. The theoretical investigation is based on an impedance approach developed herein, and the numerical simulation is based on the Finite Element method.

For the ease of analysis, the panel is considered to be infinitely extended for most of the cases in this research unless otherwise specified. Although this may not be a realistic model for practical situations where the size of the structure is limited, it has been demonstrated that the impedance of an infinite panel may be used to approximate the impedance of a finite-sized panel as long as the size of the panel is much larger than the acoustic wavelength [78]. This simplification may also result in omitting the effects of boundary conditions, thus allowing the research to focus on the characteristics of the impedance of the panel.

On the basis of the previous description, a representative model is established. As shown in Figure 2.1, the model may be illustrated as an unbounded metamaterial panel surrounded by two semi-infinite acoustic domains consisting of air. The metamaterial panel is excited by time-harmonic oblique incident plane waves with an elevation angle θ with respect to the normal direction of the plate and an azimuth angle φ. The time dependence is of the form $\mathrm{e}^{\mathrm{j}\omega t}$, with j the imaginary unit and $\omega = 2\pi f$ the angular frequency, and is omitted in the rest of the thesis. The area defined by the red dashed line in Figure 2.1 represents a lattice of the structure. A detailed description of the lattice is provided in Figure 2.2. As may be seen, the mass-spring resonator, acting as the locally resonant unit, is mounted on the host panel, and the host panel may be of different forms, as shown in Figure 2.3, covering the cases of interest in this research. The sound transmission loss properties of the metamaterial panels are thus investigated in the frequency range of interest (100 -

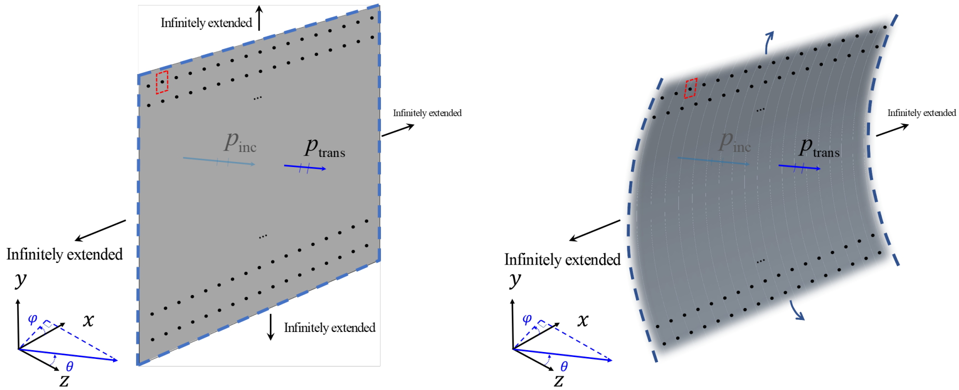


Figure 2.1: Description of sound transmission through locally resonant metamaterial panels.

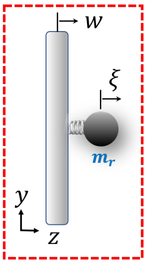


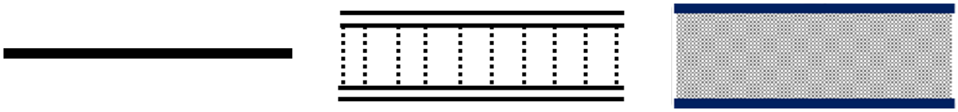
Figure 2.2: Detailed description of the lattice of the structure. The locally resonant system is realized by the mass-spring resonator.

2000 Hz in this research) based on such models.

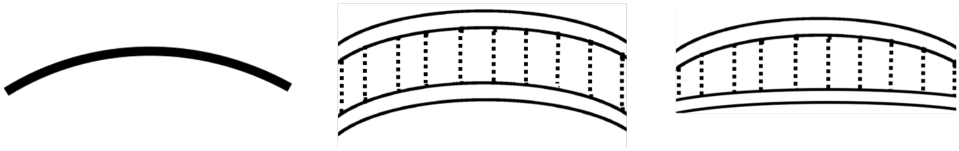
2.1 Impedance approach

The theoretical estimation based on an impedance approach is developed on the basis of the previously established model (Figures 2.1- 2.3). Such approach, in addition to being a valuable tool for effective pre-studies before more expensive simulations/experiments, may also provide reliable physical insights into the sound transmission loss behavior of metamaterial panels.

The impedance approach developed in this research is applicable under such conditions that, compared to the acoustic wavelength: *i*) The thickness of the panel is negligible, which means that the panel can be considered as a thin plate; *ii*) The distance between the resonators is much smaller.



(a) Flat single-leaf, double-wall and sandwich panel.



(b) Curved single-leaf, identical double-wall and nonidentical double-wall panel.

Figure 2.3: Depiction of the host panels of different forms, covering the cases of interest in this research.

As a starting point, the sound transmission loss of a panel is given, as

$$\mathrm{STL} = 10 \log \frac{1}{\tau}, \tag{2.1}$$

where STL stands for the sound transmission loss, τ is the transmission coefficient of the panel, and it may be expressed in the form

$$\tau = \left| 1 + \frac{Z}{2Z_\mathrm{a}} \right|^{-2}, \tag{2.2}$$

if the panel can be assumed to be infinite and homogeneous, where Z is the impedance of the panel, $Z_\mathrm{a} = \rho_0 c_0 / \cos\theta$ with ρ_0 and c_0 the density of air and the speed of sound in the air, respectively. For different forms of panels with different impedances, Z in Equation (2.2) may be replaced with the corresponding impedance of the panel as long as the above assumptions are followed. Therefore, in order to obtain the sound transmission loss of the (metamaterial) panel, it may be considered to first derive the impedance of the corresponding (metamaterial) panel.

For a metamaterial panel composed of a host panel and local resonators, an effective impedance, Z_eff, is developed in this research, expressed as a combination of the impedance of the host panel and the equivalent impedance of the resonators, Z^r_eq, such that

$$Z_\mathrm{eff} = Z + Z^\mathrm{r}_\mathrm{eq}. \tag{2.3}$$

The impedance of the host panel may be defined by the ratio of the acoustic pressure difference between the two sides of the panel to the normal component of the particle velocity if the panel can be treated as a thin plate. From the sound transmission perspective, an equivalent impedance of the resonators, as a natural extension of the impedance of the host panel, is also defined as the ratio of the pressure difference to the velocity of the resonator [79], expressed as

$$Z_{\mathrm{eq}}^{\mathrm{r}} = \mathrm{j}\omega m_{\mathrm{r}} \frac{1}{1 - f^2/{f_{\mathrm{res}}}^2}, \tag{2.4}$$

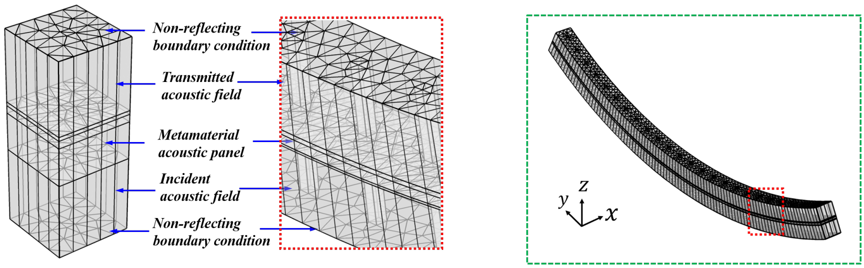
where m_{r} is the surface density of the resonator, *i.e.* the ratio of the mass of the resonator to the surface area of a lattice [27, 28, 79], and f_{res} is the resonance frequency of the resonator. Note that, the equivalent impedance is applicable only when the distance between resonators is much smaller than the acoustic wavelength. For periodic structures, the distance translates to the size of the lattice.

An 'apparent impedance' is also derived in this research for complex structures that do not conform to the thin plate assumption, such as double walls. The transmission coefficient of such structure is found to be expressed on the basis of the respective impedances of each part of the structure. By reformulating the transmission coefficient, an expression similar to Equation (2.2) may be obtained, where the emerging 'impedance' therein is defined as the 'apparent impedance' of the structure. Such 'apparent impedance' can be very convenient and efficient for investigating the sound transmission loss of double-wall structures, as detailed in [80]. For example, for the double-wall structure mounted with periodically distributed resonators, the 'impedance' may be expressed by the combination of the 'apparent impedance' and the equivalent impedance of the resonators. A metamaterial curved double wall with good sound insulation performance is thus designed in this research on the basis of the deduced 'apparent impedance' [80].

2.2 Finite Element method

In addition to the theoretical investigations introduced above, a numerical model is also established on the basis of the illustrations of sound transmission through metamaterial panels in Figures 2.1- 2.3. The numerical model in this research is based on the Finite Element method and is conducted in the commercial software COMSOL® [81]. The numerical simulations, as a more realistic representation of the practical situation, not only validate the impedance approach developed herein, but also capture more details of the vibroacoustic behavior of the metamaterial panel that is ignored in the impedance approach, such as the radiation effect of the resonators.

The Finite Element model is constructed as follows. The model consists of the incident and transmitted fields, and the intermediate partition panel. There are generally two situations in our research that need to be considered for the construction of the model, *i.e.*, the shape of the panel is flat or curved. The model of the flat panel is infinitely extended, while the model of the curved panel is constructed for a section of the panel, where the shape translates here to be infinitely long in one direction and bounded in the other. For both cases, the model is constructed for a lattice of the structure, as shown in Figure 2.4. A Floquet periodic boundary condition is implemented on the corresponding boundaries in order to simulate the infinite nature of the structure. In addition, a clamped boundary condition is selected at the boundaries in the bounded direction of the section of the curved panel.



(a) A lattice of the flat panel (left); Zoomed-in view of the lattice of the curved panel (right).

(b) A one-dimensional lattice of the section of the curved panel.

Figure 2.4: Finite Element models.

The incident and transmitted acoustic fields are simulated with the air domain. A non-reflecting boundary condition is introduced, using a plane or cylindrical wave radiation boundary condition for the flat or curved models respectively, in order to model the semi-infinite incident and transmitted acoustic fields.[1]

The partition panel in the middle is modelled with solid domain. For metamaterial panels, the resonators are modelled by soft, lightweight materials acting as springs and dense, stiff materials to represent localized masses. An aluminum panel is defined as the host panel in our study unless otherwise specified.

Structural-acoustic coupling conditions are then enforced at the interface between the acoustic and solid domains. The dimension of the mesh has been set in order to ensure the accuracy of the calculations for the shortest wavelength considered (*i.e.* highest frequency of interest).

The oblique incident sound field is modelled by a time-harmonic background pressure field with elevation angle $\theta = \pi/3$ and azimuth angle $\varphi = 0$. The sound

[1] Other methods of implementing non-reflecting boundary conditions include using a 'perfect matched layer'. However, such method is difficult to conduct since the thickness of the layer needs to be six times longer than the longest wavelength considered in order to have a satisfactory non-reflecting condition, thus making it computational costly in the low frequency range.

transmission loss may thus be estimated from the models by

$$\mathrm{STL} = 10 \log \left| \frac{\hat{W}_{\mathrm{inc}}}{\hat{W}_{\mathrm{trans}}} \right|, \tag{2.5}$$

where $\hat{W}_{\mathrm{inc}}$ and $\hat{W}_{\mathrm{trans}}$ are the incident and the transmitted powers, respectively.

In this study, the Finite Element simulations for the full models rapidly become computationally intensive, especially for the complex host panels. In order to reduce the complexity of the model, the full model is simplified by replacing the resonators with an equivalent impedance layer. The reconstructed Finite Element model is referred to as the equivalent finite element model. By using this technique, the size of the model may be significantly reduced, thus leading to much faster simulations.

Chapter 3

Solution to the coincidence effect

***Introduction to* Paper A *and* Paper B**

As a potential application for the metamaterial panels, it was shown in Paper A that the vibration and acoustic radiation suppression may be achieved with the metamaterial sandwich panel, *i.e.*, the sandwich panel with periodic distributed resonators. Indeed, a comparison to the host sandwich panel indicates that such improved behaviour may be obtained over the stop band inherited from the resonance of the resonators, expressed as the reduced vibration transmissibility and radiation efficiency respectively. In addition, these advantages observed with the metamaterial sandwich panel may not be achieved by increasing consistent distributed mass.

Nevertheless, there is a need to pay more attention to the sound transmission behaviour of metamaterial sandwich panels, in particular, in the coincidence frequency region. Moreover, it is necessary to further specify the radiation effect of only the resonators instead of the entire metamaterial panel, since it may be expected that the level of displacement of the resonator may be large at resonance. These issues are thus discussed in the following chapter, considering a metamaterial sandwich panel.

The bad sound transmission loss often occurs at the coincidence frequency range. For a sandwich panel, this frequency range may be much wider than a single-leaf panel since the bending stiffness of the sandwich decreases with increasing frequencies. It is therefore of great interest to improve the acoustic properties in this frequency range, especially for sandwich panels. This chapter introduces the use of metamaterial design to overcome the coincidence effect of sandwich panels. A systematic tuning criterion of the resonance frequency of the resonator is proposed based on an impedance approach such that the dip associated with the resonance of the resonators may be suppressed when it is tuned to the coincidence frequency region. A new design of metamaterial sandwich panels is proposed, in which the resonators are encapsulated inside the sandwich structure. In addition to overcoming the coincidence effect and limiting the noise radiation by the resonators, the proposed design allows to improve the mass ratio of the metamaterial sandwich structure. This, in turn, enables to broaden the working frequency band independently of the material adopted for the resonator. The proposed metamaterial sandwich panel thus combines improved acoustic insulation properties, while maintaining the lightweight nature of the sandwich panel and its good static properties.

3.1 Derivation of the impedance

The sound transmission loss is estimated from the impedance point of view, and the principle of designing the working frequency band of the metamaterial panels are developed, assuming a thin plate. A thin plate may be in the form of a single-leaf or sandwich panel as long as the thickness of the panel is much smaller than the acoustic wavelength in the structure. Typically, this requires that the thickness of the panel is smaller than one-sixth of the wavelength [82]. On this basis, the impedance of the single-leaf panel is first reviewed. The effective impedance of the single-leaf metamaterial panel is subsequently derived in combination with the equivalent impedance of the resonators. The same steps are followed for the sandwich metamaterial panel. The bending motion alone is considered in the sandwich panel when establishing its impedance. The effective impedance derived for the sandwich metamaterial then allows to define a criterion in order to tune the resonance frequency of the resonators.

For a single-leaf panel under acoustic excitation, the impedance of the panel Z^{s} is estimated as [9]

$$Z^{\mathrm{s}} = \mathrm{j}\omega m \left(1 - \frac{f^2}{{f_{\mathrm{cr}}}^2} \sin^4\theta\right), \tag{3.1}$$

with $f_{\mathrm{cr}} = c_0^2/2\pi\sqrt{m/D}$ the critical frequency and $f_{\mathrm{co}} = f_{\mathrm{cr}}/\sin^2\theta$ the coincidence frequency of the panel, in which, $D = Et^3/12(1-\nu^2)$ with E the Young's Modulus, t the thickness of the panel and ν the Poisson's ratio. This implies that the impedance of the panel becomes zero at the coincidence frequency.

With respect to the sandwich panel, the motion of the sandwich may be decomposed into the in-phase and out-of-phase modes [22]. In particular in the low frequency range, the dynamic behavior is dominated by the bending waves. Thus, the impedance of the sandwich panel may also be expressed in a form similar to Equation (3.1), under the thin plate assumption,

$$Z^{\mathrm{sa}} = \mathrm{j}\omega m \left(1 - \frac{f^2}{f_{\mathrm{Sco}}^2}\right), \tag{3.2}$$

where f_{Sco} is the coincidence frequency of the sandwich panel. Note that f_{Sco} is a function of frequency and may be symbolically expressed in terms of the bending wavenumber of the sandwich panel. It can be obtained based on the method introduced in PAPER B.

The effective impedance of the corresponding matematerial panel may subsequently be obtained based on Equation (2.3), as

$$Z_{\mathrm{eff}}^{\mathrm{s}} = Z^{\mathrm{s}} + Z_{\mathrm{eq}}^{\mathrm{r}}; \tag{3.3}$$

$$Z_{\mathrm{eff}}^{\mathrm{sa}} = Z^{\mathrm{sa}} + Z_{\mathrm{eq}}^{\mathrm{r}}. \tag{3.4}$$

3.2 A systematic tuning criterion

Based on the previous impedance derivation, it may be concluded that total transmission occurs when the panel impedance is zero if the damping of the panel is negligible, or, by extension, when the effective impedance of the metamaterial panel is zero. For the single-layer plate, this translates into $Z_{\mathrm{eff}}^{\mathrm{s}} = 0$, or

$$1 + \frac{\delta}{1 - f^2/{f_{\mathrm{res}}}^2} - \frac{f^2}{{f_{\mathrm{co}}}^2} = 0. \tag{3.5}$$

where $\delta = m_{\mathrm{r}}/m$ is defined as the mass ratio of the metamaterial panel. For the solution to Equation (3.5) to be real, f_{res} must satisfy

$$\left({f_{\mathrm{res}}}^2 - {f_{\mathrm{co}}}^2\right)^2 - 4{f_{\mathrm{res}}}^2{f_{\mathrm{co}}}^2\delta \geqslant 0, \tag{3.6}$$

which implies a resonance frequency such that

$$f_{\mathrm{co}}\left(\sqrt{1+\delta} - \sqrt{\delta}\right) < f_{\mathrm{res}} < f_{\mathrm{co}}\left(\sqrt{1+\delta} + \sqrt{\delta}\right). \tag{3.7}$$

Under these conditions, the effective impedance will not reach zero, which prevents the coincidence phenomenon. It is sometimes not enough just to ensure that the effective impedance of the metamaterial panel exceeds zero, since the sound transmission loss might still be less than that of the original host panel. By reconsidering the performance from a transmission perspective between the metamaterial and original host panel, a similar condition to Equation (3.7) may be established, expressed as,

$$f_{\mathrm{co}}\left(\sqrt{1+\delta/2} - \sqrt{\delta/2}\right) < f_{\mathrm{res}} < f_{\mathrm{co}}\left(\sqrt{1+\delta/2} + \sqrt{\delta/2}\right). \tag{3.8}$$

Note that the bandwidth of the working frequency range is approximately proportional to $\sqrt{\delta}$ based on our research in PAPER B, which means that it can be extended by increasing the mass ratio. Under the conditions of Equation (3.8), the sound transmission loss of the metamaterial panel is ensured to be better than that of the original host panel for any frequency close to the coincidence frequency, and the dip associated with the resonance may also be suppressed, as shown in Figure 3.1. The same conclusion may also apply to a sandwich panel, which has an analogous expression for the effective impedance, provided that the bending stiffness of a sandwich panel is a smooth function of frequency.

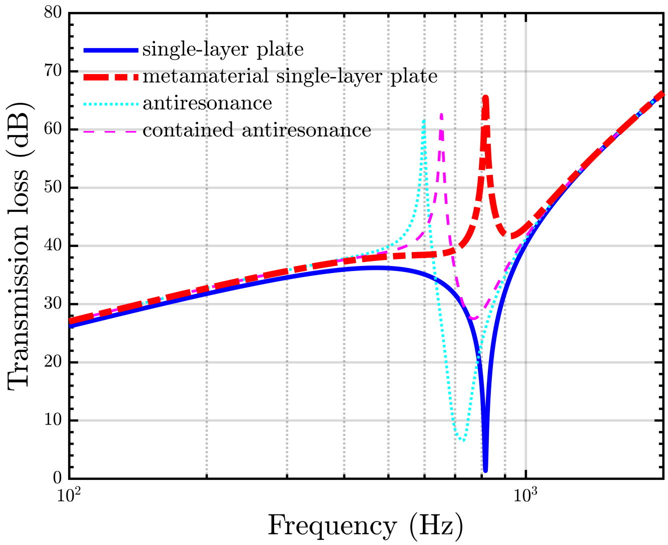


Figure 3.1: Sound transmission loss of the host panel and the metamaterial panel, with $\delta = 0.1$. Under-tuned resonance frequency: outside the suitable frequency range – Cyan-dotted line; inside the suitable frequency range – Magenta-dashed line; on the coincidence frequency – Red-dash-dotted line.

3.3 Radiation effect of resonators

The potential sound radiation from resonators may adversely affect the sound transmission loss properties of the conventionally designed metamaterial panels (By mounting the resonators on the surface of the panel). This point is considered by studying the influence of resonators with an increasing volume while keeping the mass ratio constant. The parameter σ represents the surface ratio of the resonator to the lattice. Only the sound intensity normal to the interface between the resonator and the surrounding medium contributes to the sound radiation.

In the analytical estimation, the radiation from the resonators is not included. In practice, when the resonators are mounted on the surface, the radiation from the resonators may have a substantial influence if the size of the resonator is relatively large compared to the size of the lattice, since the amplitude of the vibration velocity is large at resonance. As shown in Figure 3.2, for a set mass ratio of 0.1, as the size of the resonator increases, the radiation effect may adversely affect the sound transmission loss behavior close to the resonance frequency.

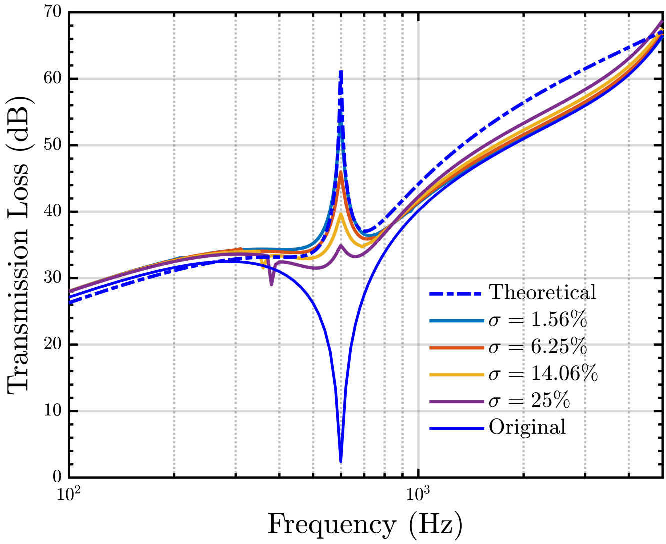


Figure 3.2: Influence from the radiation of the resonators on the sound transmission loss. σ represents the ratio of the surface area occupied by the resonator to the surface of the host panel, with $\delta = 0.1$.

3.4 Embedded design of metamaterial sandwich panel

In addition to the radiation effect of the resonators, traditional metamaterial designs have other significant limitations: *(i)* It may increase the overall mass of the structure; *(ii)* It may be impractical for most engineering applications to have a design with resonators mounted outside the sandwich (*e.g.* durability, practicality of a flat surface, etc.).

For these reasons, a design taking advantage of the sandwich configuration is proposed. In this sandwich configuration, the resonator is inserted inside the sandwich structure. The surface density is further kept constant by compensating for the mass of the resonator with the removal of a corresponding mass of the core material, as shown in Figure 3.3. Such a configuration is expected to maintain the static properties of the sandwich panel as the core material is not necessarily required to support the skin layers uniformly (*e.g.*, honeycomb sandwich plate).

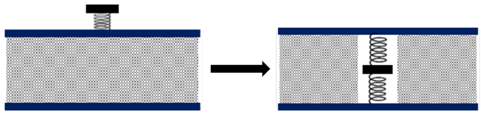


Figure 3.3: Schematic of two designs of metamaterial sandwich panel: by having resonators on the sandwich (Surface-mounted) and by embedding stepped resonators into the air inclusion of the sandwich (Embedded).

Several advantages may be expected from this configuration. First, the radiation effect can be suppressed by encapsulating the resonators inside the structure. As shown in Figure 3.4, the sound transmission loss is much better with the embedded configuration at resonance. The matching results between the theoretical

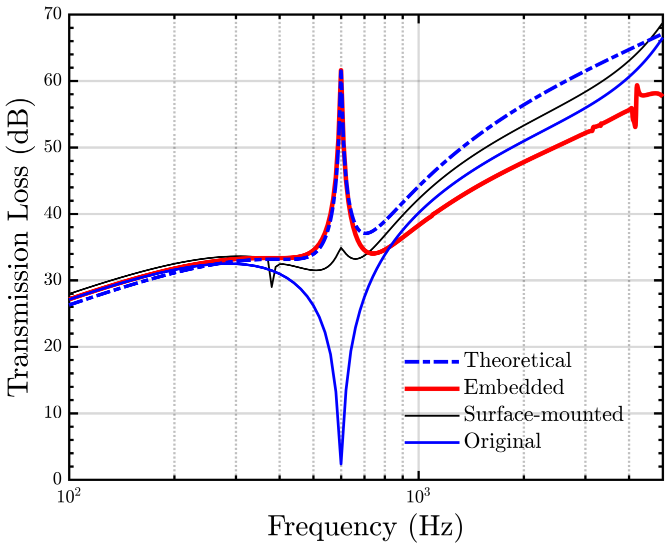


Figure 3.4: Sound transmission loss of the surface-mounted and embedded metamaterial sandwich panel.

estimation and the Finite Element model at the resonance confirm that the radiation from the resonators is suppressed with the embedded configuration. Above the coincidence range, the reduced sound transmission loss remains at a good level with the embedded configuration. This reduction of the performance may be attributed to the reduction of the effective mass. Second, as shown in Figure 3.5, it is possible to increase the mass ratio and thus extend the working frequency band while the surface density of the structure remains constant. The increased mass ratio may thus lead to an extended working frequency band, at the expense of a

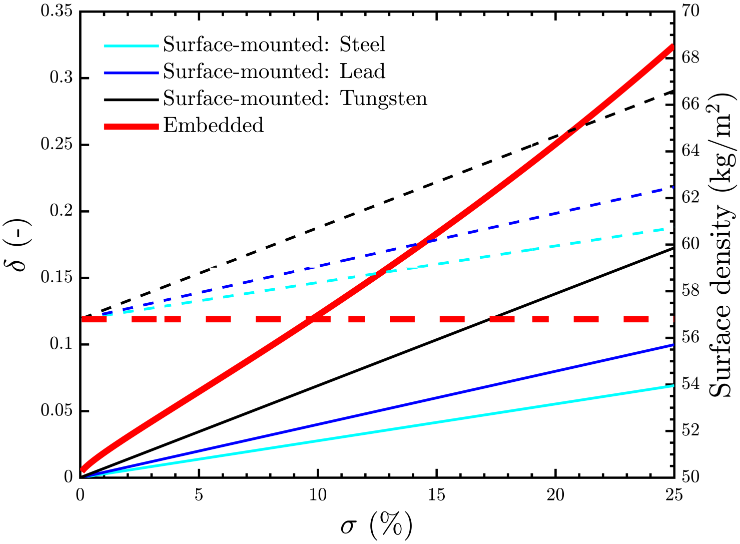


Figure 3.5: Influence and limitation of the material that is used for fabricating the stepped resonator on the mass ratio and the surface density. σ is the surface ratio occupied by resonator. Solid lines correspond to the mass ratio δ; dashed lines represent the surface density.

reduction of the sound transmission loss above the coincidence, as shown in Figure 3.6. However, the sound transmission loss for higher frequencies remains higher than for low frequencies, which leads to an overall improved performance of the sandwich with higher mass ratio. Furthermore, in agreement with Equation (3.8), this higher mass ratio implies a suitability for a broader range of frequencies for the coincidence phenomenon associated with different angles of incidence. This ensures the possibility of designing sandwich panels with both improved sound transmission loss levels and working frequency band in the low frequency range, while keeping the overall mass constant, an essential point for high performance noise reduction solutions. Third, the configuration with the flat surface is more realistic for practical applications while still benefiting from some properties of sandwich structures including low mass and high stiffness.

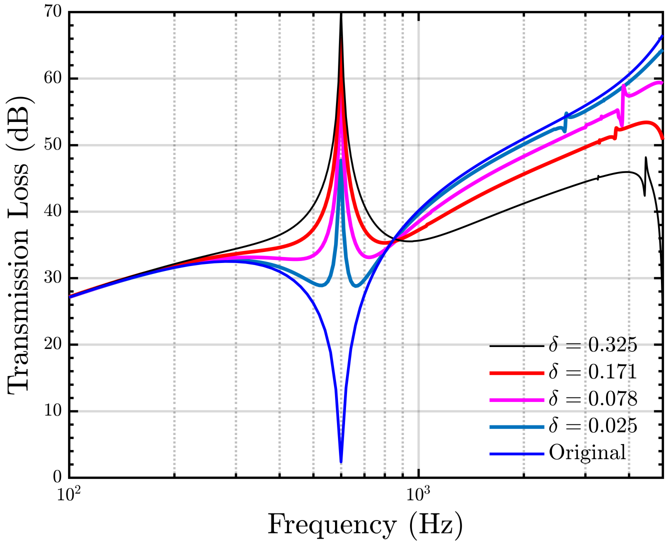


Figure 3.6: Influence of the mass ratio on sound transmission loss in the embedded design.

Chapter 4

Explanation of the ring frequency problem

***Introduction to* Paper C**

In the previous chapter, the coincidence effect was discussed, and a solution was proposed using a metamaterial design. In addition to the coincidence effect, for curved panels, the problematic ring frequency effect may also lead to a severe deterioration of the sound transmission loss. Thus, whether the metamaterial technique is also effective to address the ring frequency effect for curved panels is investigated in this chapter. For this purpose, a cylindrical shell, as a representation of curved panels, is studied from a theoretical and numerical point of view, with a specific focus on the transmission loss behaviour around the ring frequency region when the shell is mounted with local resonators. The influence from the resonators is presented, and compared with that for a flat panel. It is found that rather than the extraordinary improvement observed for the metamaterial flat panel, tuning such conventional resonators to the ring frequency of curved panels generates two side dips despite a sharp improvement at the ring frequency itself. Physical insights into this phenomenon are explained from an effective impedance point of view. The approach proposed and the conclusions provided may subsequently allow for the design of suitable resonators in order to resolve the ring frequency effect for curved panels. In addition, the interpretation derived from the present study may also be applicable to the sound insulation phenomena found in [65, 83–86] of the vibroacoustic behaviour of the resonators in connection with the appearance of dips and peak associated with the resonance.

In order to carry out the investigation, a cylindrical shell is taken as a representation of curved panels in order to study the sound transmission loss behaviour around the ring frequency region. The influence of the resonators is systematically evaluated by varying their resonance frequency to different frequency regions, *i.e.*, below, at, or above the ring frequency of the cylindrical shell. Both the theoretical estimation and the numerical simulations are conducted for the cylindrical shell. The theoretical derivation is based on an impedance approach developed for the infinite cylindrical shell. The numerical simulations are based on Finite Element models, constructed for a section of the cylindrical shell, assumed infinitely long in one direction and clamped in the other, which is a realistic representation of practical engineering situations. In order to support the explanations associated with the sound transmission loss behaviour of the metamaterial cylindrical shell, a flat panel, whose coincidence frequency is on purpose set to be about the same as the ring frequency of the cylindrical shell, is also presented. This latter test case serves as a reference in order to highlight the effect of the resonators when associated with the cylindrical shell.

4.1 Derivation of the impedance

As introduced in the first chapter, curved panels, as a common structure in aeronautical/aerospace industries, have been extensively investigated with their sound transmission loss properties. In particular, Koval developed a theory to qualitatively describe the impedance of an 'unbounded' slightly curved panel, as [31]

$$Z^{\mathrm{c}} = \mathrm{j}\omega m \left(1 - \frac{f^2}{{f_{\mathrm{cr}}}^2}\sin^4\theta - \frac{{f_{\mathrm{ri}}}^2}{f^2}\right), \tag{4.1}$$

where $f_{\mathrm{ri}} = \sqrt{Et/m(1-\upsilon^2)}/(2\pi R)$ is the ring frequency with R the radius of curvature of the curved panel, the superscript 'c' stands for 'curved'.

The effective impedance of the corresponding metamaterial panel may, again, be obtained based on Equation (2.3), as

$$Z^{\mathrm{c}}_{\mathrm{eff}} = Z^{\mathrm{c}} + Z^{\mathrm{r}}_{\mathrm{eq}}. \tag{4.2}$$

While the effective impedance of the reference metamaterial flat panel may be obtained based on Equation (3.4). The material properties of the corresponding panels are provided in Table 4.1.

Table 4.1: Material properties of the host panels.

	E(Pa)	υ(-)	ρ(kg/m^3)	t(m)	Sectional angle (rad)	R(m)	f_{co}(Hz)[1]	f_{ri}(Hz)
Flat panel	6.9e10	0.3	2700	2e-2	-	$+\infty$	816	-
Cylindrical shell	6.9e10	0.3	2700	1e-3	1.85	1	1.6e4	843[2]

[1] f_{co} is the coincidence frequency of the panel. When the incident elevation angle $\theta = \pi/3$, $f_{\mathrm{co}} = 816$ Hz.
[2] For the section of the cylindrical shell, the lowest sound transmission loss occurs at 780 Hz rather than exactly at the ring frequency, this is due to the influence of the clamped boundary condition.

4.2 Validation against the Finite Element method

The Finite Element model of a section of the curved panel is introduced in Section 2.2. In the first step, the results based on the impedance approach and the Finite Element method are here validated against each other, as shown in Figure 4.1.

Figure 4.1a shows a comparison associated with oblique incidence ($\theta = \pi/3$, $\varphi = 0$). The trends between the impedance approach and the Finite Element method are clearly the same, except for the fluctuations in the Finite Element results. These fluctuations are attributed to the eigenmodes inherent to the finite-sized nature of the model and the fact that only one oblique incident angle is considered in the Finite Element simulations.

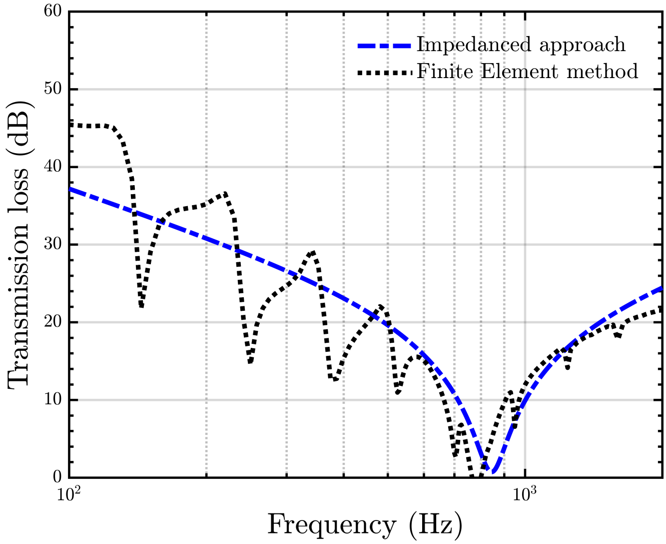
In order to further validate the impedance approach, an averaged sound transmission loss is studied with multiple angles of incidence. The averaged sound transmission loss based on the Finite Element method in 1/3-octave bands is shown in Figure 4.1b. As shown, the fluctuation due to the influence of the eigenmodes is almost entirely suppressed. In fact, the more incident angles, the closer the Finite Element results become to the impedance approach. The reason for considering only one angle of incidence here is to extract key features with reduced computational resources. The results provided by the impedance approach thus offer both satisfying accuracy as well as great efficiency for the scope of the current study.

4.3 Sound transmission loss properties around the ring frequency

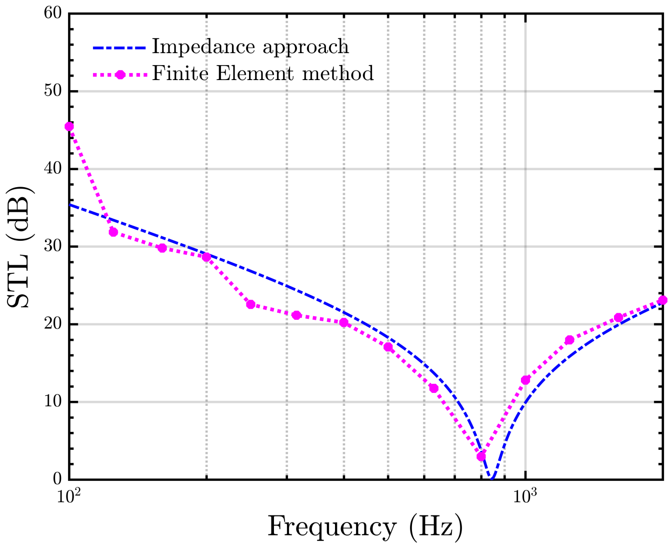
The influence of the resonators on the sound transmission loss around the ring frequency region is studied in this section. In particular, the influence of the resonators is systematically studied by tuning the resonance frequency of the resonators to be below, at, or above the specific frequencies of interest[1]. The different transmission loss behaviour is subsequently presented between the ring and coincidence frequency regions.

The results based on the impedance approach are presented in Figures 4.2. When the resonance frequency is tuned to below the specific frequencies of interest, a peak and a dip associated with the resonance are generated. For the flat panel, the peak appears first, followed by the dip, while these are inverted for the shell. The shift between the peak and the dip associated with the response of resonators may also be seen between the flat panel and the shell when the resonance frequency is tuned above the specific frequencies of interest.

[1] The frequencies of interest are referring to the resonance frequency, coincidence frequency, or the ring frequency for the resonators, the flat panel, or the cylindrical shell, respectively.



(a) One angle of incidence.



(b) Average from multiple angles of incidence.

Figure 4.1: Comparison of the sound transmission loss evaluation between the impedance approach and the Finite Element method.

These opposite effects of the resonators observed with respect to the sound transmission loss are found between the mass-controlled region and stiffness-controlled region. These opposite behaviours may be explained in terms of placing the resonance of the resonators in the mass- or stiffness-controlled regions.

As to tuning the resonance of the resonators to the specific frequencies of interest, the sound insulation behaviour of the shell, unlike for the flat panel at the coincidence frequency, results in the appearance of side dips, here denoted as 'side effects'. In fact, a similar phenomenon may also be seen in a recent experimental study on the use of resonators to act on the ring frequency effect [77]. Physically:

- For a flat panel at the coincidence frequency, the impedance is transferred from the mass-controlled region to the stiffness-controlled region,
- For a cylindrical shell at the ring frequency, the impedance is transferred from the stiffness-controlled region to the mass-controlled region.

This provides the underlying physical explanation for the ineffectiveness of tuning resonators to the ring frequency, a point which may be further explained from an impedance point of view, as detailed in Section 4.4.

The Finite Element results for a section of the metamaterial cylindrical shell, as detailed in Figure 9 in Paper C, also capture the behaviour of the resonators in agreement with the results of the impedance approach, thus validating the ability of the latter to estimate the overall sound transmission loss behaviour of the metamaterial panel.

4.4 Explanation from an impedance point of view

In order to further detail the reasons for the extraordinary behaviour associated with the coincidence effect and the 'side effects' associated with the ring frequency effect provided by the corresponding metamaterials, the impedances are studied in detail.

When the damping is small, the imaginary part of the impedance is dominant. The imaginary part of the impedance of the host panels and the equivalent impedance of the resonators are plotted in Figure 4.3a. As may be seen from the figure, regarding the host panel, the mass-controlled region corresponds to the region where the imaginary part of the impedance is greater than zero, whereas the stiffness-controlled region corresponds to the region where the imaginary part of the impedance is lower than zero. Regarding the equivalent impedance of the resonators, the imaginary part of the impedance changes from being positive below the resonance frequency to negative above the resonance frequency. These behaviours imply that the resonators have the same phase change as the impedance of the flat panel if tuned at the coincidence frequency, while they have an opposite phase change to the impedance of the cylindrical shell if tuned at the ring frequency. Therefore, the cumulative effect of the impedance of the resonators and the host panels, driving the behaviour of the metamaterial panel leads to:

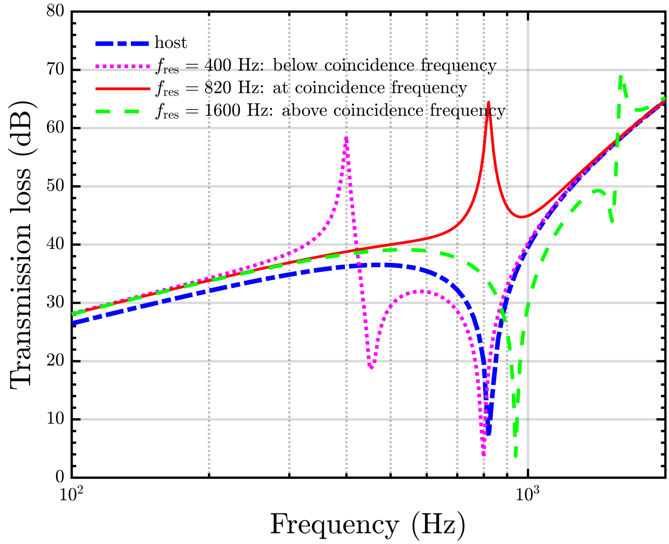
- An increase of the absolute value of the effective impedance of the metamaterial flat panel, due to the identical phase change between the resonators and the panel (see dash-dotted line in Figure 4.3b). This results in an increased transmission loss over the entire coincidence frequency region.
- The emergence of two points where the absolute value of the effective impedance of the metamaterial shell is zero, *i.e.*, where the impedance of the shell and the equivalent impedance of the resonators cancel each other, due to the opposite phase change between the resonators and the shell. This results in two dips at the frequencies where the effective impedance is cancelled out. Furthermore, the narrow improvement of the transmission loss in between these side frequencies is due to the resonance of the resonators (see dashed line in Figure 4.3b).

The above impedance-based interpretation is further summarized in Table 4.2, highlighting the cumulating or cancelling effects around the coincidence or ring frequencies, respectively. In practice, this implies the need to introduce resonators following

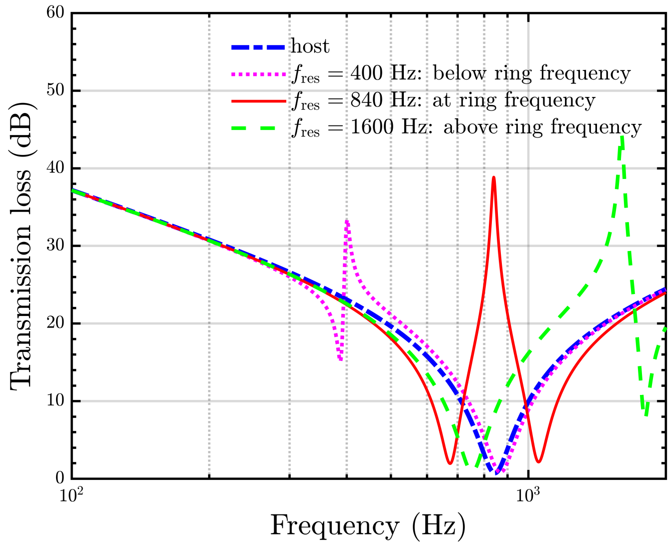
Table 4.2: Sign of impedance for different subsystems and frequency regions of interest.

Z	Resonators	Flat panel	Shell
Below freq. of interest	$+$	$+$	$-$
Above freq. of interest	$-$	$-$	$+$

the same phase change as the host panel at the specific frequencies of interest. For this purpose, the impedance approach may prove to be a powerful methodology. For the coincidence effect, this impedance approach allows to tune conventional resonators in order to suppress the dip in the transmission loss on the basis of a systematic tuning criterion introduced in Paper B. For the ring frequency, however, the impedance approach highlights that such a dip is hardly avoidable when conventional mass-spring resonators are used. In order to achieve this, specifically designed resonators may be sought, exhibiting a negative-to-positive impedance change at the resonance frequency, which cannot be achieved using conventional mass-spring resonators. The impedance approach may here open the way for the design of such unconventional resonators, *e.g.* considering active resonators, as one of the perspectives of this contribution.

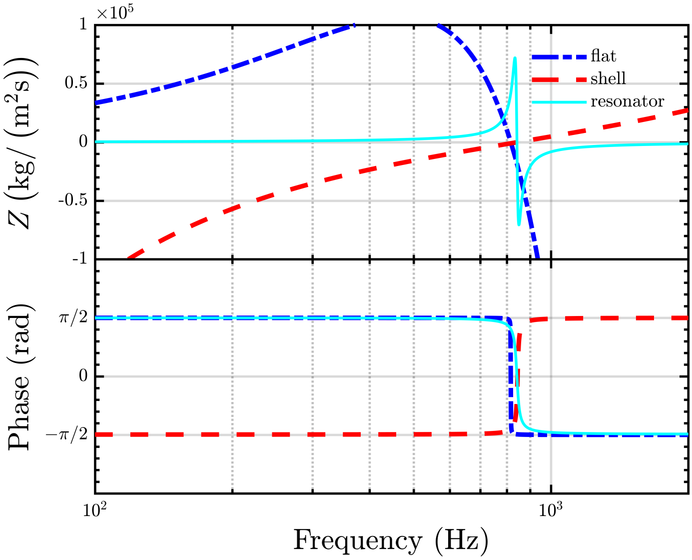


(a) Flat panel;

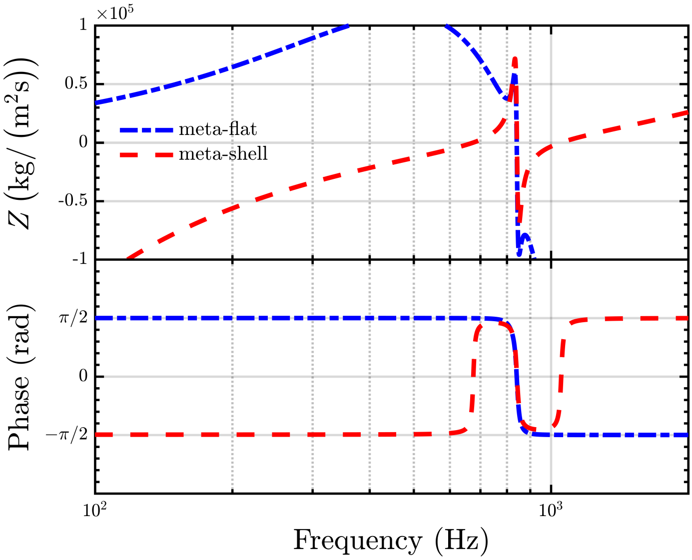


(b) Cylindrical shell.

Figure 4.2: Comparison of the influence from the resonators on the sound transmission loss between the metamaterial flat panel and the metamaterial cylindrical shell by tuning the resonance frequency of the resonators to different frequency regions.



(a) Impedance of the host panels and equivalent impedance of the resonators.



(b) Effective impedance of the metamaterial flat panel and cylindrical shell.

Figure 4.3: The impedance of the structures of interest.

Chapter 5

Design of metamaterial curved double walls

***Introduction to* Paper D**

Curved double walls or similar structures are among the commonly used structures in modern industries such as the aerospace industry. However, the combination of curved panels, and a double wall arrangement may exhibit poor acoustic insulation properties at specific frequencies. For curved acoustic panels, as interpreted in Chapter 4, the ring frequency is one of the frequencies where poor acoustic insulation properties occurs. Double-wall structures, on the other hand, may be strongly affected due to the mass-spring-mass resonance effect, which may result in a low sound transmission loss. For curved double-wall structures, however, the sound insulation performance may be even more critical since the sound transmission loss is not only affected by the ring frequency effect, but also by the mass-spring-mass resonance effect. Research with respect to the acoustic properties of such curved double-wall panels may be found in [43,45,46]. However, to the authors' knowledge, limited contributions have focused on the specific improvement of sound insulation performance of such structures around the critical frequencies mentioned above.

In this Chapter, a method for designing locally resonant metamaterial curved double walls is proposed in order to improve the bad sound insulation properties associated with the characteristic frequencies of the structure. In the previous chapter, we have shown that the sound insulation behaviour of a metamaterial panel may be strongly influenced by the resonance of the resonator, and such resonance may only occur in a very narrow frequency band, thus limiting the application potential of such metamaterial design. On the basis of these limitations, a specific design of the curved double wall is proposed, where the ring frequencies of the outer curved panels, and the mass-spring-mass resonance frequency are tuned to be close to each other, such that their bad acoustic performance may be dealt with simultaneously. In particular, the combined effect of the ring frequencies and the mass-spring-mass resonance frequency may be addressed using localized resonators. This chapter thus focuses on studying the sound insulation properties of such metamaterial curved double walls, exploring the possibility to achieve broadband acoustic insulation in the low frequency range.

The investigation conducted is based on an ‘apparent’ impedance approach and the Finite Element method as previously presented.

5.1 Derivation of the ‘apparent impedance’

In the first step, the ‘apparent impedance’ of a curved double wall is derived. The schematic description of the curved double wall may be seen in Figure 5.1. As

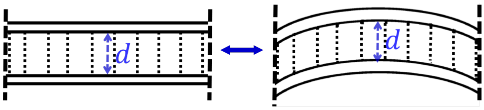


Figure 5.1: A schematic view of a double wall and a curved double wall.

shown in the figure, a double-leaf system typically consists of two panels and an internal resilient material, such as air. Provided that the system reacts locally to the acoustic excitations, the transmission coefficient for a flat double-wall system may be given, as detailed in [9], by

$$\tau = \left| 1 - \omega^2 \frac{m_1' + m_2'}{2s} + \mathrm{j}\omega \frac{m_1' + m_2'}{2Z_\mathrm{a}} \left(1 - \omega^2 \frac{m_1' m_2'}{s\,(m_1' + m_2')} + \frac{Z_\mathrm{a}^2}{s\,(m_1' + m_2')} \right) \right|^{-2}, \quad (5.1)$$

with

$$m_i' = m_i \left(1 - \frac{f^2}{f_{\mathrm{cr}\,i}^2} \sin^4\theta \right), \; i = 1, 2, \quad (5.2)$$

where m_i corresponds to the surface density of panel i.

Equation (5.1) may be reformulated in the form of Equation (2.2), as

$$\tau = \left| 1 + \frac{Z^\mathrm{d}}{2Z_\mathrm{a}} \right|^{-2}, \quad (5.3)$$

where the emerging impedance Z^d is referred to as the ‘apparent impedance’ of the double wall. With the further purpose of extending the approach to curved double walls, this ‘apparent impedance’ may be rewritten as a function of the impedances of each individual panel of the system, such that

$$Z^\mathrm{d} = Z_1 + Z_2 + \frac{\mathrm{j}\omega}{s} (Z_1 + Z_\mathrm{a})\,(Z_2 + Z_\mathrm{a})\,. \quad (5.4)$$

Z_1 and Z_2 represent the impedances of the corresponding panels, and may be expressed as

$$Z_i = \mathrm{j}\omega m_i', \, i = 1, 2. \tag{5.5}$$

In Equation (5.1) and (5.4), $s = K/d$ is the stiffness per unit area (N/m^3) of the air cavity, where d is the thickness of the air cavity. If the air cavity has absorbing materials, K is commonly expressed as $K = \rho_0 c_0^2$. For an air cavity without absorbing material, [9] recommends $K = 4\mathrm{e}5\,\mathrm{N/m^2}$, but comparison with numerical calculations shows that 5.6e5 N/m^2 may be better suited for the analyses conducted in the present contribution.

On the basis of Equation (5.4), the 'apparent impedance' for a flat double-leaf panel may be extended to the case of a curved double-leaf panel. The 'apparent impedance' of a curved double wall, Z^{cd}, may thus be obtained by substituting the impedance of the flat panel Z_i in Equation (5.4) with the impedance of the curved panel Z_i^{c} (see Equation (4.1)), such that

$$Z^{\mathrm{cd}} = Z_1^{\mathrm{c}} + Z_2^{\mathrm{c}} + \frac{\mathrm{j}\omega}{s}\left(Z_1^{\mathrm{c}} + Z_{\mathrm{a}}\right)\left(Z_2^{\mathrm{c}} + Z_{\mathrm{a}}\right). \tag{5.6}$$

It should be noted that s is assumed to be a constant here, and is obtained by averaging the thickness of the air cavity in the curved double wall. This approximation has proved to have only marginal impact on the accuracy of the results.

5.2 Characteristic frequencies of curved double walls

For a curved double wall with a small curvature, transmission loss properties collapse at the ring frequency and the mass-spring-mass resonance frequency. This reflects in the fact that the imaginary part of the impedance may become very small at these frequencies. By reconsidering the impedance from the transmission perspective, the values of the ring and the mass-spring-mass resonance frequencies of the curved double wall may be approximated.

The impedance for a curved double wall, expressed by Equation (5.6), is detailed as

$$Z^{\mathrm{cd}} = \mathrm{j}\omega\left(m'_1 + m'_2\right) + \frac{\mathrm{j}\omega}{s}\left(-\omega^2 m'_1 m'_2 + Z_{\mathrm{a}}^2 + \mathrm{j}\omega\left(m'_1 + m'_2\right) Z_{\mathrm{a}}\right), \tag{5.7}$$

where in this case

$$m_i' = m_i\left(1 - \frac{f^2}{f_{\mathrm{cr}\,i}^2}\sin^4\theta - \frac{f_{\mathrm{ri}\,i}^2}{f^2}\right), \quad i = 1,\, 2. \tag{5.8}$$

A minimum in the transmission loss occurs when the imaginary part of the panel impedance become minimum, leading to the condition

$$\omega^2 m_1' m_2' = s\left(m'_1 + m'_2\right). \tag{5.9}$$

When $f \ll f_{\mathrm{cr}\,i}$, Equation (5.9) may further be expressed as

$$4\pi^2 f^2 m_1 m_2 \left(1 - \frac{f_{\mathrm{ri}\,1}^2}{f^2}\right)\left(1 - \frac{f_{\mathrm{ri}\,2}^2}{f^2}\right) = s\left(m_1\left(1 - \frac{f_{\mathrm{ri}\,1}^2}{f^2}\right) + m_2\left(1 - \frac{f_{\mathrm{ri}\,2}^2}{f^2}\right)\right). \tag{5.10}$$

The frequency at which the sound transmission loss becomes minimum is then obtained by solving Equation (5.10). In general cases, the solutions for Equation (5.10) are complicated functions of the masses and the corresponding ring frequencies. For a special case when two curved panels have the same ring frequency f_{ri}, the solutions for Equation (5.10) are given by

$$f_{\mathrm{ri}}^{\mathrm{cd}} = f_{\mathrm{ri}}, \tag{5.11a}$$

$$f_{\mathrm{msm}}^{\mathrm{cd}} = \sqrt{{f_{\mathrm{msm}}^{\mathrm{d}}}^2 + {f_{\mathrm{ri}}}^2}, \tag{5.11b}$$

where the two characteristic frequencies $f_{\mathrm{ri}}^{\mathrm{cd}}$ and $f_{\mathrm{msm}}^{\mathrm{cd}}$ are the ring and the mass-spring-mass resonance frequencies of the curved double wall, respectively. $f_{\mathrm{msm}}^{\mathrm{d}}$ is the mass-spring-mass resonance frequency of the flat double wall with the same spacing between panels (see Figure 5.1), known to be approximated as

$$f_{\mathrm{msm}}^{\mathrm{d}} = \frac{1}{2\pi}\sqrt{s\left(\frac{1}{m_1} + \frac{1}{m_2}\right)}. \tag{5.12}$$

Note from the above expressions that for curved double walls, the mass-spring-mass resonance frequency is always higher than the ring frequency.

The sound transmission loss as well as the characteristic frequencies of a curved double wall may thus be estimated based on such an impedance approach. The result obtained from the impedance approach is here validated against the Finite Element method in Figure 5.2 for the cases `R4-2-40-2`, where `R1-2-40-2` represents a curved double wall with a radius of curvature of 1 m, a thickness of the first panel of 2 mm, an air layer of 40 mm, and a thickness of the second panel of 2 mm. As shown in Figure 5.2, the trends between the impedance approach and the Finite Element method are clearly the same, except for the fluctuations in the Finite Element results. These fluctuations are attributed to the finite-sized nature of the model which is similar to the phenomenon observed in Figure 4.1a. Nevertheless, the results provided by the impedance approach, as an efficient tool, offer satisfying accuracy for the scope of the current study. In particular, the location of the characteristic frequencies, together with the associated dips in the transmission loss at these frequencies are properly captured.

Figure 5.3 shows the sound transmission loss based on the impedance approach for curved double walls with varying parameters. As seen in Figure 5.3, the width

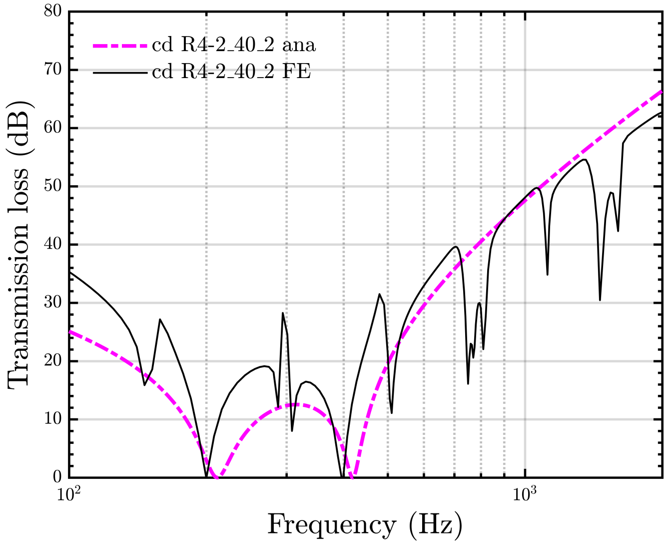


Figure 5.2: Comparison of sound transmission loss of curved double wall `R4-2-40-2` between the impedance approach and the Finite Element model.

of this frequency band may be controlled by the design parameters. This follows from Equations (5.11). For a given ring frequency of the curved double wall, f_{ri}, its mass-spring-mass resonance frequency $f_{\mathrm{msm}}^{\mathrm{cd}}$ is fully determined by the ring frequency itself, and the mass-spring-mass resonance frequency of the associated flat double-wall panel, $f_{\mathrm{msm}}^{\mathrm{d}}$ (see Figure 5.1). This implies that the frequency band of the valley may be narrowed by controlling the mass-spring-mass resonance frequency to be close to the ring frequency. Note that however close these two characteristic frequencies, it is clear from Equation (5.11) that they cannot be merged. Additionally, it is noteworthy that the closer these characteristic frequencies, the better the transmission loss properties outside of the 'valley', both for lower and higher frequencies. The potential advantage of bringing the characteristic frequencies together is to concentrate the poor transmission loss properties in a narrow band, which may subsequently be tackled by a localized design solution. Despite the potential 'valley', further comparisons in Paper D show that the curved double wall may combine the advantages of a curved panel and a flat double wall, exhibiting good sound transmission loss properties both below and above the 'valley'.

In order to improve the transmission loss over the 'valley', enabled by the narrowed valley emerging from the combined effect of the characteristic frequencies, local resonators may be well-suited in order to obtain a structure exhibiting im-

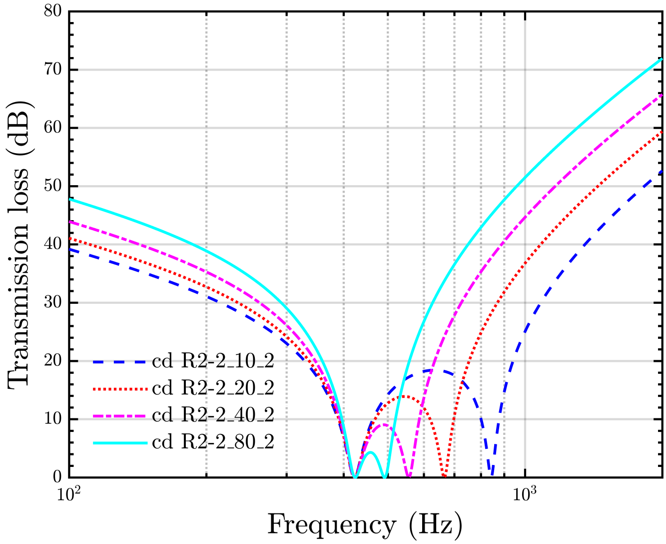


Figure 5.3: Sound transmission loss of curved double walls with different parameters based on the impedance approach.

proved broadband sound insulation properties. Overcoming this valley using locally resonant metamaterial design is therefore the focus of the following section.

5.3 Improved sound transmission loss via metamaterial design

In the metamaterial curved double wall, the mass-spring local resonators are periodically mounted on the host curved double wall. As for the impedance of the metamaterial curved double wall, on the basis of Equation (4.2), by substituting one of the individual impedances in Equation (5.6) with $Z^{\mathrm{c}}_{\mathrm{eff}}$, an effective 'apparent impedance' of the locally resonant metamaterial curved double wall may be obtained, as

$$Z^{\mathrm{cd}}_{\mathrm{eff}} = Z^{\mathrm{c}}_{\mathrm{eff}} + Z^{\mathrm{c}} + \frac{\mathrm{j}\omega}{s}\left(Z^{\mathrm{c}}_{\mathrm{eff}} + Z_c\right)\left(Z^{\mathrm{c}} + Z_c\right). \tag{5.13}$$

When both panels forming the double wall are mounted with resonators, the effective 'apparent impedance' of such metamaterial may be derived by substituting both the individual impedances with the corresponding effective impedance.

Two configurations for the metamaterial curved double wall are thus considered: *i*) Both panels of the double wall are mounted with resonators, or, *ii*) only one of the two panels of the double wall is mounted with resonators. In both cases, the total added mass is kept constant, set to be 10% of the mass of the host curved double wall. The resonance frequency of the resonator is set at the median frequency of the valley. Note that the resonators are mounted on the inner surface of the panel(s) in order to avoid potential acoustic radiation based on the explanation in PAPER B, as shown in Figure 5.4. The structural damping of the host panel, for the cases presented in the following, is set to a realistic value of $\eta_s = 0.01$, unless otherwise specified.

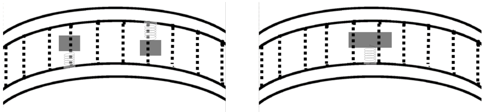
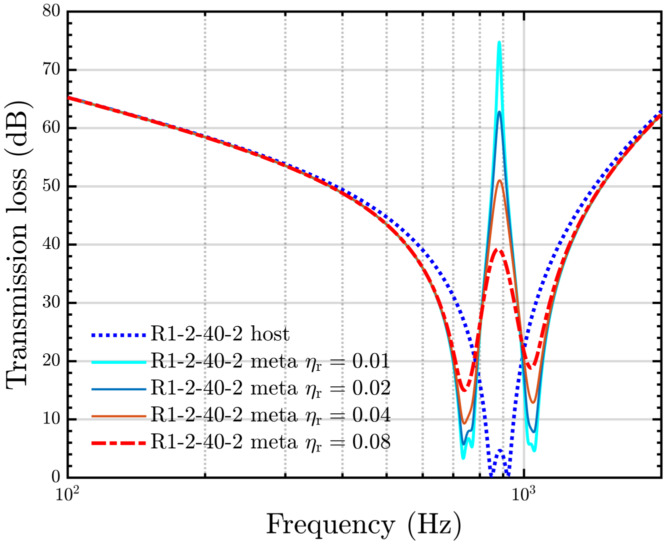


Figure 5.4: Illustration of a metamaterial curved double wall with resonators mounted on both panels (left) and a single panel (right).

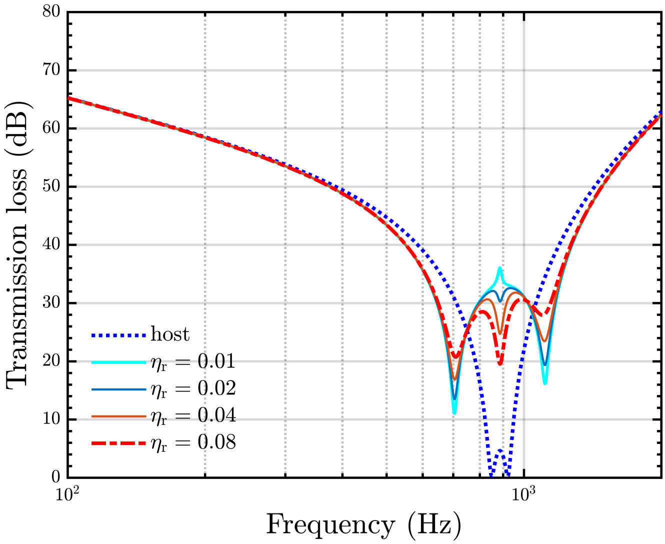
5.3.1 Identical curved double walls

In order to simplify the problem in a first step, the curved double wall studied in this section is set up with a double-wall system consisting of two identical curved panels. This type of curved double walls is referred to herein as the identical curved double wall. Curved double walls with different curved panels will be discussed in the next section.

The sound transmission loss of the proposed metamaterial is given in Figures 5.5. For the metamaterial with resonators mounted on both panels, as shown in Figure 5.5a, with little to no damping in the resonators, improvements appear very clearly in a narrow band associated with the combined characteristic frequencies, with however strong side effects in the form of side dips. However, when damping in the resonators is increased, the side dips are mitigated, leading to an arguably improved solution compared to the reference host panel. Figure 5.5b shows the sound transmission loss when the resonators are mounted on one panel only. For a set level of damping in the resonators, the results are arguably much improved compared to the case where resonators are mounted on both panels. Although the sound transmission loss at the characteristic frequencies is not as drastically improved, the interesting observation here is that the side dips are mitigated such that the overall performance of the panel is improved. For example, a minimum improvement of about 5 dB is observed for the side dips of the chosen cases, *i.e.* $\eta_r = 0.01$ (cyan solid line in Figures 5.5a and 5.5b) and $\eta_r = 0.08$ (red dash-dot line in Figures 5.5a and 5.5b). Interestingly, due to the fact that the peak appearing at the resonance frequency of the resonator is progressively converted into a dip when the damping is increased at realistic values, a trade-off against the improvement of the side dips is required. This trade-off is illustrated with a parametric study with respect to the damping of the resonator. The optimal configuration may be found when the side dips and the transmission loss at the resonance frequency of



(a) With the resonators mounted on both panels.



(b) With the resonators mounted on one of the two outer panels.

Figure 5.5: Sound transmission loss of the metamaterial curved double wall with damping η_r in the resonators.

the resonators reach similar levels. In the present situation, this configuration is met for $\eta_r \approx 0.08$. This configuration leads to a transmission loss being higher than 20 dB in the critical frequency band associated with the characteristic frequencies.

In order to provide an engineering perspective, a comparison of the transmission loss in 1/3-octave bands is plotted in Figures 5.6 and 5.7, based on the Finite Element calculations. The host panel is taken as a reference in both plots (dashed blue curves). Figure 5.6 shows that a curved double wall has the advantage of exhibiting good sound insulation properties both below and above the characteristic frequencies. Further using a metamaterial design in order to address the bad performance in this narrow frequency band allows to ensure improved sound insulation properties over the entire frequency range of interest (see Figure 5.7).

5.3.2 Nonidentical curved double walls

In practical designs, nonidentical curved double walls, consisting of two different curved panels, are more common than identical curved double walls. For nonidentical curved double walls, the differences between the curved panels may be sought in terms of material and geometric properties. An example of such nonidentical curved double wall is the side wall of an aircraft fuselage, which consists of a skin panel and a trim panel. Note that, although in most cases the difference between the skin panel and the trim panel may result in different ring frequencies, these may be tuned to the same frequency by design. Though these two cases are investigated in detial in Paper D, the results of the latter case will only be provided here.

The sound transmission loss results in 1/3-octave bands, using identically damped resonators ($\eta_r = 0.08$) attached to one of the panels, are shown in Figure 5.8. A similar acoustic behavior to the identical curved double walls previously discussed may be observed, indicating that such a configuration may respond well to a similar metamaterial design. Indeed, an improvement of more than 15 dB may be observed at the dip associated with the characteristic frequencies of the curved double wall by using suitably damped resonators. Similar conclusions may be drawn from this plot, *i.e.* an overall improved performance due to the sharp improvements of the transmission loss in the region of the characteristic frequencies (15 to 20 dB in the present case).

Therefore, using a metamaterial design in order to address the bad performance in the narrow frequency band associated with the characteristic frequencies allows the curved double wall to ensure improved sound insulation properties over the entire frequency range of interest. As illustrated with the extension to nonidentical curved double walls, the resulting design methodology to improve broadband noise insulation properties offers a valuable degree of robustness.

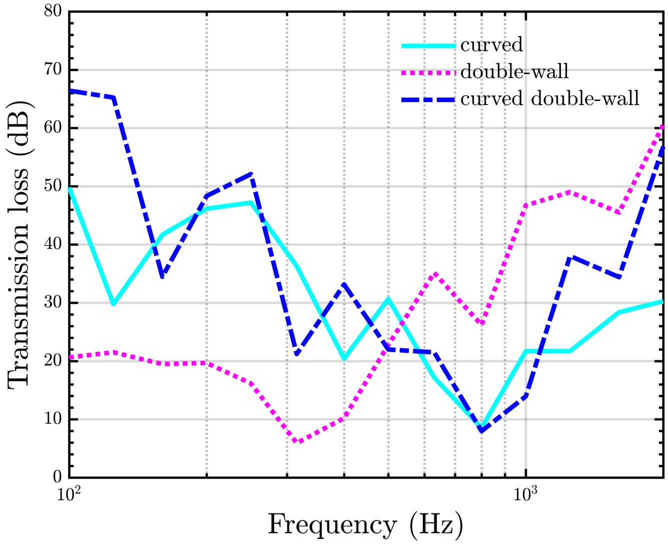


Figure 5.6: Sound transmission loss of curved panel, double wall and curved double wall in 1/3-octave bands.

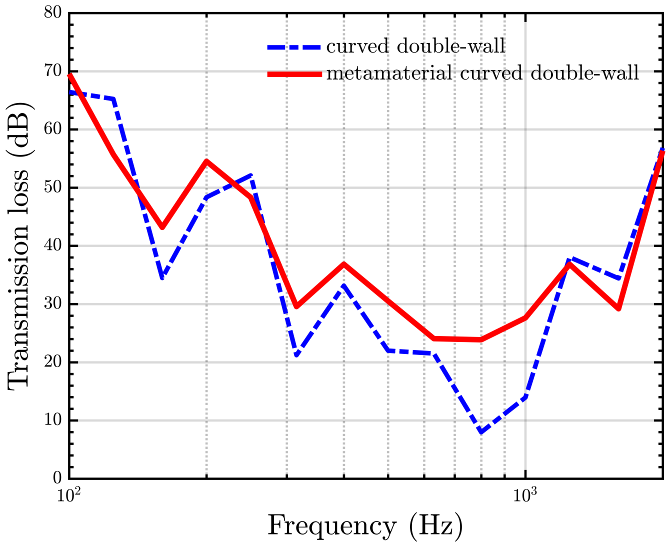


Figure 5.7: Sound transmission loss of curved double wall and the metamaterial in 1/3-octave bands.

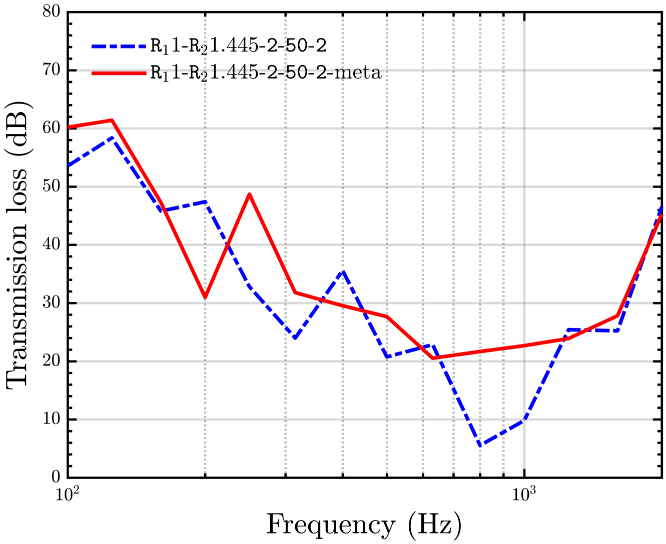


Figure 5.8: Sound transmission loss of the nonidentical curved double wall and the associated metamaterial in 1/3-octave bands.

Chapter 6

Concluding remarks and future work

6.1 Concluding remarks

The sound transmission loss behaviour of locally resonant metamaterial panels is investigated in the thesis. The new designs are proposed based on the metamaterial concept for the corresponding panels in order to address the poor sound insulation properties associated with certain frequency regions.

The acoustic behaviour of a metamaterial sandwich panel is first investigated. A new design for metamaterial sandwich panel is proposed and studied, by encapsulating stepped resonators inside the sandwich panel. This configuration with internal resonators allows to improve the sound transmission loss properties of the panel, in particular in the coincidence region. The physical insights drawn from the impedance analysis support the fine tuning of the resonators in order to suppress the coincidence phenomenon. The influence of the resonator is studied with the Finite Element models, in which the radiation from the resonators is taken into account. The results highlight that the radiation effect of the resonators may affect the sound transmission loss property, especially at the resonance. The proposed configuration is however shown to overcome the radiated sound from these resonators by embedding them in the core material. The resulting design further enables to preserve mechanical functionalities of the outer sheets as flat surfaces, thus making it a more realistic solution than most of the locally resonant metamaterial panels studied in the literature.

However, when the conventional mass-spring resonators act on the ring frequency region associated with curved panels, it may not be repeated for the good sound transmission loss behavior similar to that of the coincidence frequency region of the locally resonant metamaterial flat panel. In order to provide an explanation for this problem, a locally resonant metamaterial cylindrical shell, as a representation of curved panels, was studied. The comparison with the response of a locally resonant metamaterial flat panel using conventional mass-spring resonators allows to highlight that the efficiency of such conventional resonators is governed by the change of impedance of the host panels. The impedance approach proposed shows that the efficiency of the metamaterial structures lies in the fact that the phase change of the impedance of the resonators should be identical to that of the host plate. For the curved panel, shifting at the ring frequency from the stiffness-controlled region in the lower frequencies, to the mass-controlled region at higher frequencies, the cumulative effect with the resonators leads to the cancellation of the effective impedance at two side frequencies to the ring frequency. Despite a sharp improvement at the ring frequency itself, this side effect worsens the overall transmission loss properties of the curved panel. The proposed study further highlights the requirements for the design of suitable non-conventional resonators in order to address the ring frequency effect, based on their equivalent impedance derivation.

Locally resonant metamaterial curved double walls are proposed and their sound insulation properties investigated. In particular, a method for designing such panels is proposed in order to improve their broadband noise insulation in the low frequency range. In order to conduct the analyses associated with the development of this design methodology, an effective impedance approach, referred to as an 'apparent impedance' approach, is proposed for the estimation of the sound transmission loss of the curved double wall and its metamaterial description. In the proposed design methodology, the ring frequency and the mass-spring-mass resonance frequency of the curved double wall are designed to be very close to each other. Then, by incorporating optimally damped resonators on one of the panels of the curved double wall, the transmission drop in the region of the characteristic frequencies of the host structure may be effectively addressed. As illustrated with the extension to nonidentical curved double walls (*i.e.* where the outer panels have both different geometric and material properties), the resulting design methodology to improve broadband noise insulation properties offers a valuable degree of robustness.

6.2 Future work

Below are a few areas which the author would like to investigate futher in the future.

- While the results in most cases are shown for oblique incidence configurations with a single targeted frequency range, the extension to the diffuse field is straightforward, and multi-resonator solutions deserve more investigation.

- As for the ring frequency effect, the relative inefficiency of conventional resonators leads to an obvious goal: to develop resonators that operate in line with the properties derived from the impedance analysis in order to effectively overcome the ring frequency effect. This demands a stiffness-to-mass transition at the resonance of the resonator from a sound transmission perspective. A prospective result is virtually shown in Figure 6.1 just to illustrate the ideal situation when using the suitable resonators.

- In addition, from the author's point of view, a hypothesis may be proposed and deserves to be further tested, *i.e.* that the resonators with the suitable properties for the ring frequency effect can also be used to address the mass-spring-mass resonance of the double wall. Such hypothesis is derived based on an investigation of the sound transmission loss at the mass-spring-mass resonance. A similar sound transmission loss behaviour to the ring frequency effect is observed with a metamaterial double wall, as shown in Figure 6.2
- Last but not least, the proposed designs in this contribution are considered to be further validated with experimental studies.

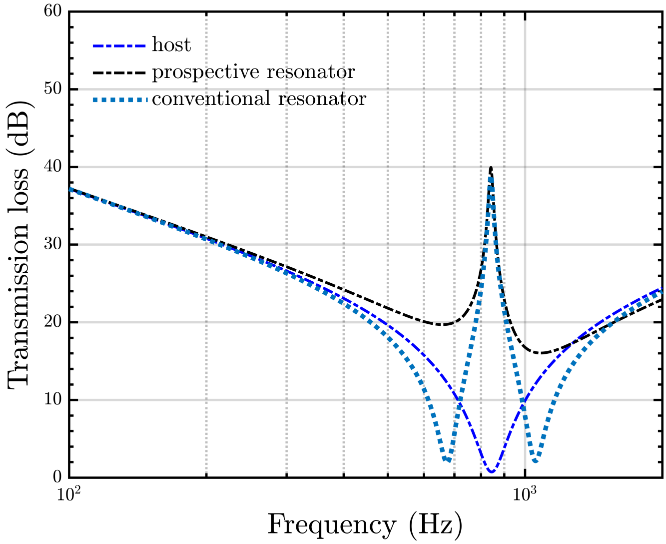


Figure 6.1: Prospective sound transmission loss behaviour associated with suitable non-conventional resonators in order to resolve the ring frequency effect.

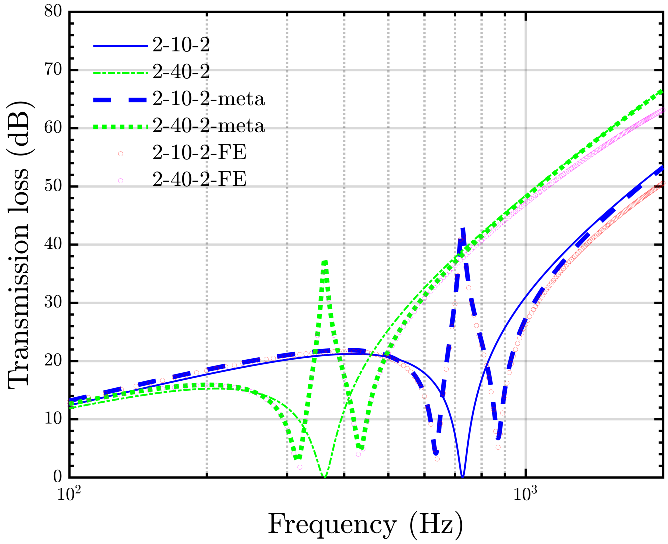


Figure 6.2: Sound transmission loss of the double walls and the associated metamaterials.

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