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Microperforated Acoustic Metasurfaces for Low-Frequency Broadband Noise Reduction

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Declaration

I declare that this thesis has not been submitted as an exercise for a degree at this or any other university and it is entirely my own work. I confirm that the work has been completed in full compliance with relevant university policies, including the Academic Integrity policy, the Trinity Policy on Good Research Practice, and the guidance on AI and GenAI in Teaching, Learning, Assessment and Research.

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Jiayu Wang, August 2025

*To my parents, **Guoming Wang** and **Xiaoqin Wang**, with love, gratitude, and pride.*

Abstract

This thesis focuses on the development of microperforated acoustic metasurfaces to address the challenges of achieving low-frequency, broadband noise reduction in compact configurations, with specific applications in aeroengine noise control. The work is motivated by the stringent space and weight constraints in aeroengine and duct systems, where efficient sound absorption is crucial.

A multi-chamber microperforated panel absorber (MC-MPPA) was proposed to enable multiple acoustic resonances within a limited volume, thereby broadening the absorption bandwidth. A two-point impedance method (TpIM), grounded in acoustic network theory, was developed to model and optimise the absorber's impedance characteristics. Experimental validation showed that MC-MPPA samples achieve absorption coefficients exceeding 0.8 across two distinct frequency bands: 397–1000 Hz and 698–1895 Hz, corresponding to sub-wavelength ratios of 1/18.2 and 1/10.5, respectively.

To further enhance low-frequency absorption, particularly below 500 Hz, a new class of microperforated acoustic metasurfaces incorporating extended-neck Helmholtz resonators was introduced. Three configurations were designed and 3D-printed, exhibiting remarkable subwavelength performance. For instance, a deep subwavelength configuration achieved over 97% absorption at 150 Hz, corresponding to a subwavelength ratio of 1/48. A broadband-optimised configuration demonstrated an average absorption coefficient exceeding 0.74 in the 300–500 Hz range with a thickness of only 27.5 mm, and over 0.88 in the 543–945 Hz range with a thickness of just 26.2 mm.

In addition, internal perforated partitions enhance the acoustic performance of the optimised MC-MPPA at higher frequencies and lower porosities (e.g., 0.5%–1.0% within 20–3000 Hz), enabling superior absorption with fewer surface perforations. This effectively reduces aerodynamic drag under grazing flow, making the design well-suited for aero-engine liners.

To further address the practical requirements of aeroengine liner applications, the acoustic performance of the proposed designs was extended to grazing flow conditions through an impedance-based optimisation framework. A Mach-number-dependent impedance model was incorporated into the TpIM, enabling accurate prediction of impedance under flow. A Cremer-impedance-inspired cost function was employed to guide the optimisation, resulting in high-performance liners that were tested at Mach numbers up to 0.25 and sound pressure levels (SPLs) up to 140 dB. The 32 mm-thick MC-MPPA liner achieved up to 25 dB of transmission loss in the 600–2000 Hz range at an incident SPL of 140 dB and a grazing flow Mach number of 0.25. Under the same conditions, the 32 mm-thick CMHA liner achieved up to 15 dB of transmission loss below 1000 Hz. Additionally, in the absence of flow, the CMHA liner exhibited broadband transmission loss of 10–23 dB below 1000 Hz.

All proposed designs and theoretical modeling methods were experimentally validated using impedance tubes and grazing flow test rigs. The strong agreement between predictions and measurements confirms the reliability of the theoretical frameworks and demonstrates the practical feasibility of 3D-printed acoustic metamaterials for real-world noise control applications.

Overall, this thesis advances the state of the art in low-frequency broadband microperforated acoustic metasurfaces, supported by rigorous theoretical modeling and experimental validation. The developed solutions are compact, lightweight, and tunable, and are well suited for noise control in aeroengines and duct systems.

List of Publications

Publications related to this thesis include:

Journal Papers:

1. **J. Wang**, G. J. Bennett, Multi-chamber micro-perforated panel absorbers optimised for high amplitude broadband absorption using a two-point impedance method, *Journal of Sound and Vibration*, 547 (2023) 11527. <https://doi.org/10.1016/j.jsv.2022.117527>
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3. **J. Wang**, G. J. Bennett, Deep sub-wavelength broadband metasurface with micro-perforated panels and extended-neck Helmholtz resonators, *Applied Physics Letters*, 126 (2025) 141703. <https://doi.org/10.1063/5.0258251>
4. **J. Wang**, G. J. Bennett, Incorporating extended neck Helmholtz resonators into an advanced multi-degree of freedom acoustic metamaterial for low-frequency, broadband acoustic absorption, *Applied Acoustics*. 238 (2025) 110750. <https://doi.org/10.1016/j.apacoust.2025.110750>.
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6. J. Killeen, I. Davis, **J. Wang**, G. J. Bennett, Fan-noise reduction of data centre telecommunications' server racks, with an acoustic metamaterial broadband, low-frequency sound-absorbing liner, *Applied Acoustics*. 203 (2023) 109229. <https://doi.org/10.1016/j.apacoust.2023.109229>.

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Time flies. The four years of my PhD journey are drawing to a close. During these years, both the world and I have undergone profound changes.

Now, standing at the threshold of completing my doctorate—just one step away from the degree—I ask myself: if I could go back four years and choose again, would I still resolutely pursue a PhD? Would I still dedicate the prime years of my life to academia and research? My answer is a resounding yes. These past four years have been the most focused, pure, and intellectually fulfilling period of my life. Although I am currently going through a difficult phase in my personal life, I believe this thesis—and all the days and nights of hard work behind it—will shine as a beacon in my future.

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Nomenclature

Acronyms

AC	Air Cavity
CMHA	Compact MHA
DDOF	Double-Degree-of-Freedom
DE	Dielectric Elastomer
DLR	German Aerospace Center
ECM	Equivalent Circuit Model
HVAC	Heating, Ventilation, and Air Conditioning
IFAR	International Forum for Aviation Research
IPP	Interior Perforated Thin Panel
JAXA	Japan Aerospace Exploration Agency
JCA	Johnson-Champoux-Allard
KTH	Royal Institute of Technology
LAUM	Laboratoire d'Acoustique de l'Université du Mans
MC-MPPA	Multi-Chamber Micro-Perforated Panel Absorbers
MDOF	Multi-Degree-of-Freedom
MHA	Micro-Perforated Panel Coupled with an Extended-Neck Helmholtz Resonator Absorber

MPP	Micro-Perforated Panel
mSLA	Masked Stereolithography
NASA	National Aeronautics and Space Administration
POHC	Perforated Plate Over Honeycomb
PPETs	Parallel-Arranged Perforated Plates with Extended Tubes
PSAM	Porous Sound-Absorptive Material
PVDF	Polyvinylidene Fluoride
PZT	Piezoelectric Ceramic
SAC	Sound Absorption Coefficient
SDOF	Single-Degree-of-Freedom
SeMSA	Segmented Membrane Sound Absorber
SPLs	Sound Pressure Levels
TL	Transmission Loss
TpIM	Two-Point Impedance Method
ZML	Zero Mass Flow Liner
Greek Symbols	
α	Sound Absorption Coefficient
α_{∞}	Tortuosity
γ	Ratio of Specific Heats
λ	Wavelength
μ	Dynamic Viscosity
ω	Angular Frequency ($\omega = 2\pi f$)
ϕ	Perforation Porosity (Ratio of Micro-Perforated Area to Total Area)
xxx	

ρ_0	Air Density
τ	Panel Thickness
Roman Letters	
Λ'	Thermal Characteristic Length
Λ''	Viscous Characteristic Length
Sh	Shear Number
σ	Flow Resistivity
τ_i	Thickness of the Internal Panels
τ_t	Thickness of the Top Panels
c_0	Speed of Sound
c_α	Average Sound Absorption Coefficient
C_p	Specific Heat Capacity at Constant Pressure
D	Depth of Sub-Chamber
d	Perforation Diameter
f	Frequency (Measured in Hertz)
H	Total Thickness of Absorber
H_{duct}	Height of the Duct
J_m	Bessel Function of the First Kind of Order m
K	Thermal Conductivity
k	Wavenumber
k_x	Axial Wavenumber Component
k_y	Transverse Wavenumber Component
k_{ef}	Effective Wave Number of the Equivalent Fluid

L	Overall Length of Absorber
M	Mach Number of the Grazing Flow
n	Number of Perforations
r	Reflection Coefficient
W	Overall Width of Absorber
y_{ij}	Admittance Between Nodes i and j
Z_{ef}	Effective Characteristic Impedance of the Equivalent Fluid
Z_{ij}	Acoustic Impedance Between Nodes i and j

Terminology

Term ¹	Definition
Metasurfaces	In this thesis, metasurfaces primarily refer to structures engineered through artificial design or geometric optimization to achieve sound absorption performance unattainable by conventional materials. Such structures are collectively referred to as metasurfaces or metamaterials in this thesis, without distinction between the two terms.
Sub-wavelength	Refers to a structural thickness smaller than the sound wavelength. In this thesis, it typically denotes a thickness between one-thirtieth and one-tenth of the wavelength ($\lambda/30$ to $\lambda/10$).
Deep sub-wavelength	Refers to a structural thickness much smaller than the sound wavelength. In this thesis, it typically means a thickness below $\lambda/30$, aiming at effective absorption below 200 Hz.
Low-frequency	In this thesis, unless otherwise specified, refers to sound frequencies below 1000 Hz.
Broadband	In this thesis, an absorber is considered broadband if it is effective over a frequency range of at least 1000 Hz, typically targeting effective absorption above 500 Hz. For absorption effective below 500 Hz, a frequency range of at least 200 Hz is considered broadband.
Ultra-broadband	In this thesis, an absorber is considered ultra-broadband if it is effective over a frequency range of at least 2000 Hz.

¹Although some definitions may differ slightly from conventional usage, they are applied consistently to maintain clarity and rigour throughout this thesis.

Term	Definition
Sound Absorption Coefficient	In this thesis, the sound absorption coefficient specifically refers to the value measured under normal acoustic incidence conditions in an impedance tube. The coefficient is calculated by first determining the reflection coefficient from the material's acoustic impedance and the characteristic impedance of air, then computing the absorption coefficient as one minus the squared magnitude of the reflection coefficient. This coefficient quantifies the material's ability to absorb sound, ranging from 0 (no absorption) to 1 (total absorption). In this thesis, the sound absorption coefficient is abbreviated as SAC and denoted by α .
Average Sound Absorption Coefficient	In this thesis, the average absorption coefficient refers to the sound absorption coefficient averaged over a specific frequency range, weighted by the power spectral density of the targeted noise source. This measure reflects the absorber's effectiveness with respect to the actual spectral content of the noise. When the targeted noise is assumed to be white noise, the average absorption coefficient simplifies to the arithmetic mean of the absorption coefficient over the specified frequency band. In this thesis, it is abbreviated as c_α .
60% absorption	Refers to a sound absorption coefficient of 0.6.

Chapter 1

Introduction

1.1 Research Background and Motivation

Mitigating low-frequency noise remains a persistent challenge in acoustic engineering. Effective sound absorption at low frequencies typically requires material depths of at least one-quarter of a wavelength ($\lambda/4$), resulting in substantial material thicknesses. This constraint not only increases the size and weight of acoustic treatments but also limits their application in compact or weight-sensitive environments, such as aerospace, land-based transportation, or urban infrastructure.

Aircraft noise has long been a major concern for airlines, aircraft manufacturers, and communities living near airports. Among the dominant noise sources, fan noise from high-bypass-ratio turbofan engines is particularly prominent during take-off and landing. The increasing bypass ratio restricts the available space for nacelle acoustic liners, further complicating low-frequency noise control due to the longer wavelengths involved. Developing compact yet efficient liner designs capable of low-frequency sound attenuation remains a critical challenge.

Micro-perforated panel absorbers, which employ panels with small perforations backed by an air cavity to absorb sound through viscous and thermal dissipation, are gaining increasing attention due to their robustness and cleanliness, making them well-suited for noise control applications. However, conventional MPPAs often suffer from limited low-frequency absorption and narrow bandwidth, particularly in shallow configurations (≤ 50 mm in depth), posing challenges for achieving broadband low-frequency attenuation.

Despite recent progress, achieving efficient low-frequency and broadband absorption in a lightweight, compact structure —particularly one suitable for aircraft nacelle liners —remains an ongoing challenge. Addressing this issue is critical to improving noise reduction solutions for modern aeroengine systems.

1.2 Research Scope and Objectives

This thesis investigates the development of microperforated acoustic metasurfaces designed to address the challenges of achieving low-frequency, broadband noise control in compact configurations, particularly for aeroengine noise control applications.

The research addresses the following key areas:

1. **Design and optimisation of multi-chamber microperforated panel absorbers (MC-MPPAs):** Designing MC-MPPA structures to achieve multiple acoustic resonances within a limited volume for broadband absorption, and modelling and optimising their impedance characteristics using the two-point impedance method (TpIM).
2. **Enhancement of low-frequency absorption performance:** Introducing new microperforated acoustic metasurfaces integrated with extended-neck Helmholtz resonators to enhance low-frequency sound absorption, particularly below 500 Hz—and even approaching 150 Hz.
3. **Optimisation design and application validation under grazing flow conditions:** Constructing Mach-number-dependent impedance models to accurately predict impedance characteristics under flow conditions and guiding the optimisation using a Cremer-impedance-inspired cost function. Testing the liner under grazing flow conditions with Mach numbers up to 0.25 and sound pressure levels up to 140 dB to validate the practical application potential of the proposed design.
4. **Experimental validation and fabrication:** Experimentally validating all proposed designs and theoretical modeling using impedance tubes and grazing flow test rigs. Strong agreement between predictions and measurements confirms the reliability of the theoretical frameworks. All test samples are fabricated by 3D printing, demonstrating the practical feasibility of 3D-printed acoustic metasurfaces for real-world noise control applications.

This study aims to provide a lightweight, compact, and efficient noise control solution for environments where space and weight constraints are critical. The outcomes are expected to contribute to advancements in both fundamental acoustics research and practical engineering applications, particularly in the field of aircraft engine noise reduction.

1.3 Structure of the Thesis

The remainder of this thesis will be organised as follows:

- **Chapter 2** presents a comprehensive literature review, identifying the key challenges in turbofan aeroengine noise reduction and examining state-of-the-art research on MPPAs as a promising approach to address these issues.

- **Chapter 3** introduces the concept and design of an optimised multi-chamber micro-perforated panel absorber with micro-perforated adjoining panels.
- **Chapter 4** details the development of an enhanced MC-MPPA featuring varying sub-chamber depths, which achieves ultra-broadband low-frequency sound absorption.
- **Chapter 5** investigates the design and optimisation of an MC-MPPA for broadband noise attenuation under grazing flow conditions. The fabricated liner samples were tested in a grazing flow facility to evaluate their performance.
- **Chapter 6** presents a metasurface composed of micro-perforated panels coupled with extended-neck Helmholtz resonators, forming a unit absorber referred to as an MHA. This design aims to further enhance low-frequency absorption, particularly below 500 Hz.
- **Chapter 7** develops, fabricates, and characterizes a deep sub-wavelength broadband MHA for low-frequency noise absorption. To broaden the bandwidth and reduce thickness while keeping the total area constant, a compact three-unit configuration is proposed and its transmission loss under grazing flow is experimentally evaluated.
- **Chapter 8** summarizes the conclusions of this thesis and provides an outlook for future work.

Chapter 2

Literature Review

2.1 Introduction

Turbofan engines are the primary propulsion systems for modern commercial aircraft, offering high efficiency while also generating substantial noise. Among the various noise sources, low-frequency noise, particularly from components like the fan, presents a significant challenge. These low-frequency sounds are capable of propagating over long distances, negatively impacting both passenger comfort and the surrounding environment. Traditional noise control methods often struggle to effectively mitigate low-frequency noise due to its long wavelengths and complex propagation characteristics. As aviation noise regulations become increasingly stringent, finding effective solutions for low-frequency noise reduction is becoming a critical issue for the industry.

Given the challenges posed by low-frequency noise, the demand for innovative noise control materials is increasingly urgent. One promising solution is the micro-perforated panel absorber, which is recognized for its simple structure, lightweight properties, and the ability to be made from environmentally friendly materials. MPPA's unique configuration of micro-sized perforations and panel designs enables efficient sound absorption.

This chapter aims to review the current state and challenges of turbofan engine noise reduction and explore state-of-the-art research on micro-perforated panel absorbers as a promising solution.

2.2 Turbofan Aeroengine Noise Reduction

2.2.1 Turbofan aeroengine noise challenges

The advent of the jet age marked a turning point in aviation history, bringing about remarkable technological advancements and enabling faster, more efficient air travel. However, it also introduced new challenges, with aircraft noise emerging as a significant issue [1]. This problem has far-reaching implications, affecting not only airlines and aircraft manufacturers but also, and perhaps most critically, communities living in proximity to airports. For example, the European Aviation Environmental Report 2022 [2] estimates that approximately 4 million people in Europe are highly annoyed by aircraft noise, while 1.7 million experience severe sleep disturbance—an increase of 24% and 31%, respectively, compared to 2005. As an example, consider Figure 2.1 and Table 2.1, adapted from [3], which illustrate the noise contours of Dublin Airport and the population and area affected by Dublin airport noise in 2023. The data indicate that annually, over 73,300 people and 171 km² of land are exposed to

noise levels of 45 dB L_{den} ¹, while more than 93,352 people and 165 km² of land are affected by noise levels of 40 dB L_{night} ². Prolonged exposure to high noise levels has been linked to

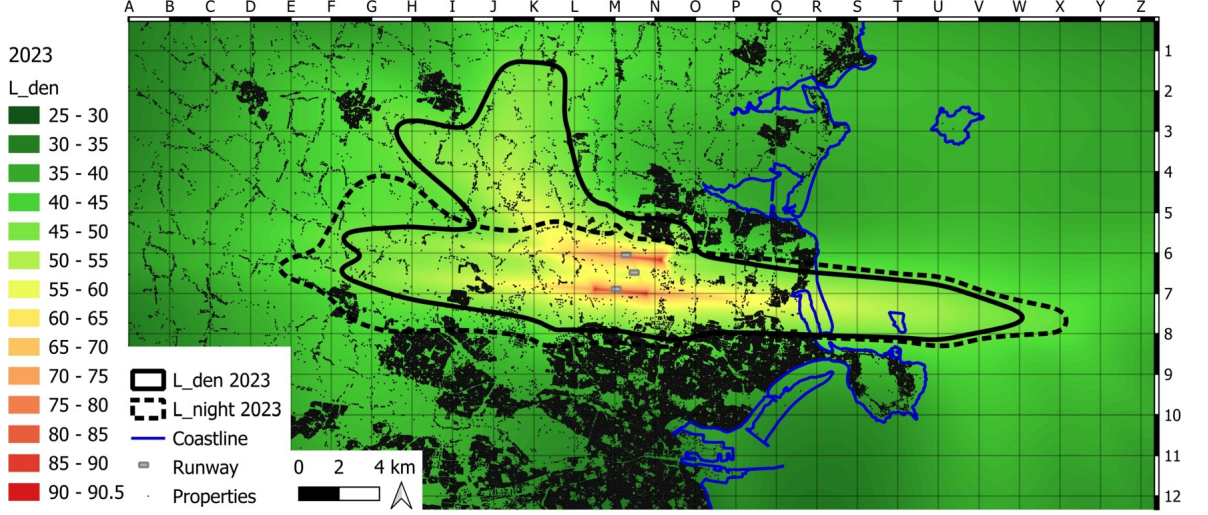


Figure 2.1: Noise contours around Dublin Airport in 2023 [3].

Table 2.1: Population and area exposure to Dublin airport noise in 2023 [3].

Exposure	L_{den} (45 dB)	L_{night} (40 dB)	L_{den} (55 dB)	L_{night} (50 dB)
Area (km ²)	171.48	165.66	29.35	29.65
Number of houses	30334	38083	1296	2555
Number of people	77377	93352	2771	6012

cardiovascular problems, sleep disturbances, and cognitive impairments, especially in children [4–8]. These health risks have underscored the urgency of mitigating aircraft noise and have driven regulatory bodies to impose stringent noise control standards on aviation operations. The World Health Organization Environmental Noise Guidelines for the European Region recommend limiting aircraft noise levels to 45 dB L_{den} and 40 dB L_{night} [2]. The European Commission’s Flight Path 2050 vision aims to reduce perceived aircraft noise emissions by 65% by 2050, relative to the performance of typical new aircraft introduced in 2000 [9].

Addressing aircraft noise is inherently complex due to its multifaceted nature [10]. Noise is generated by various components of an aircraft, including engines, airframe structures, and aerodynamic interactions [11]. Among these, engine noise is the most significant contributor during critical phases of flight, such as take-off and landing, particularly for modern high-bypass-ratio turbofan engines [12].

¹ L_{den} is the sound pressure level averaged over the year for the day, evening and night time periods, with a +5 dB penalty for the evening and +10 dB for the night.

² L_{night} is the sound pressure level averaged over the year for the night time period only.

2.2.2 Acoustic liners

Perforated plate over honeycomb (POHC) liners

One of the primary methods for mitigating engine noise involves the use of acoustic liners installed in the nacelle intake and bypass ducts. These liners typically rely on perforated faceplates backed by honeycomb structures. The faceplates feature millimeter-scale circular perforations, which attenuate noise through mechanisms such as thermoviscous effects and vortex shedding [8]. As shown in Figure 2.2, traditional POHC liners are commonly used for turbofan engine noise reduction, with two primary configurations: Single-Degree-of-Freedom (SDOF) and Double-Degree-of-Freedom (DDOF) liners [13].

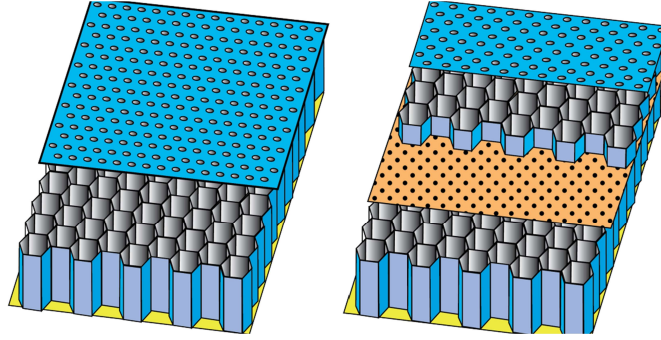


Figure 2.2: Schematic illustration of a traditional SDOF liner (left) and a traditional DDOF liner (right) [13].

The SDOF liner typically comprises a honeycomb-like cavity structure, a solid back plate, and a perforated facesheet. However, this liner design yields a relatively narrow sound absorption spectrum. The DDOF liner, on the other hand, is essentially an SDOF arrangement with an embedded porous septum or perforated plate. While the DDOF liner does expand the sound absorption bandwidth, its structure becomes more complex and its weight increases. Additionally, the DDOF liner generally necessitates more space compared to the SDOF liner.

Moreover, their performance in reducing low-frequency noise—characterized by longer wavelengths—remains a challenge due to spatial constraints imposed by the increasing bypass ratios [14–20]. Larger air cavities are typically required for low-frequency attenuation, but the available space for liner installation is becoming more limited as nacelle designs prioritize aerodynamic efficiency and weight reduction [16, 18]. This constraint has fueled ongoing research into advanced acoustic liner designs capable of delivering efficient noise reduction within limited spatial envelopes. Efforts to address these challenges have led to innovations

in liner designs, such as folded cavity structures [12, 21, 22], flexible damping materials [16, 23, 24], foam-metal composites [25–27], and other advanced configurations [16, 28–32].

The stringent standards for modern aviation noise emissions have increased the demands on the acoustic performance of aircraft engine liners [14–17]. To address this, the International Forum for Aviation Research (IFAR) launched a benchmarking initiative to evaluate acoustic liner testing methods and performance prediction models. Led by the National Aeronautics and Space Administration (NASA) [33, 34], the project has involved collaboration with universities and research institutions such as the Royal Institute of Technology (KTH) [18], the German Aerospace Center (DLR) [19, 20, 35] and the Japan Aerospace Exploration Agency (JAXA) [36]. Its goal is to standardize comparisons across various liner flow-duct facilities globally and to ensure consistency in post-processing techniques and prediction methodologies. Ongoing efforts continue to focus on improving liner performance.

Folded cavity liners

Sugimoto et al. [12] introduced the concept of folded cavity liners, as shown in Figure 2.3(a). Their study investigated the effectiveness of these liners in attenuating both mid-to-high-frequency noise and low-frequency noise. The performance of folded cavity liners can be optimised across a broad frequency spectrum by adjusting the geometry and properties of the resistive layers. These liners were experimentally tested in the grazing flow facility at the National Aerospace Laboratory in the Netherlands. Palani et al. [21, 22, 37] introduced

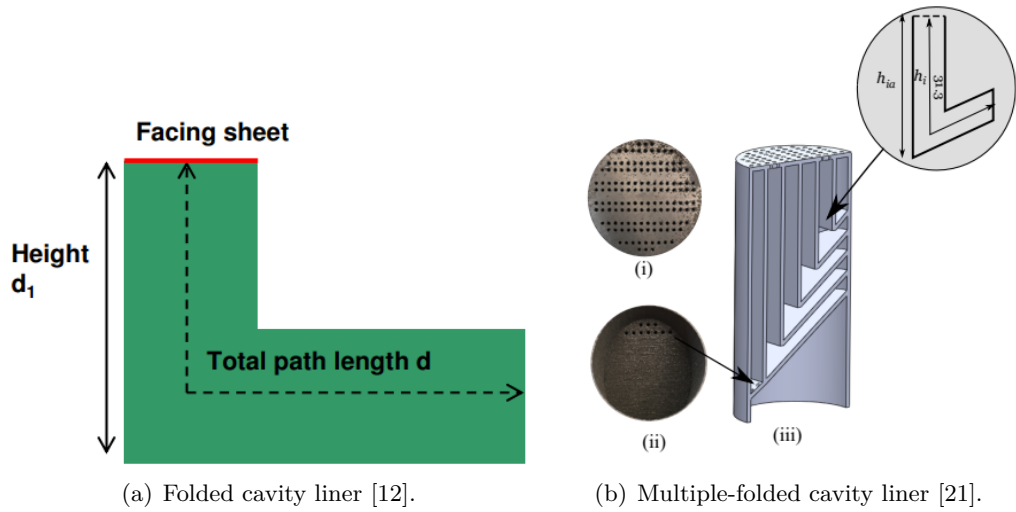


Figure 2.3: Folded cavity liners.

the concept of a multiple-folded cavity liner, as shown in Figure 2.3(b), which demonstrated

significant improvements in broadband sound absorption compared to conventional designs. Their results showed that multiple-folded cavity liners achieved substantial sound absorption under normal acoustic incidence within specific frequency ranges. Furthermore, the multiple-folded cavity liner was experimentally tested in the grazing flow facility at the National Aerospace Laboratory in the Netherlands, where the results demonstrated enhanced broadband attenuation for downstream sound propagation. The measurements closely matched the theoretical predictions, indicating excellent agreement.

Flexible plate liners

Kohlenberg et al. [24, 38] introduced a Multi-Degree-of-Freedom (MDOF) Helmholtz resonator liner, which featured a flexible plate dividing the back cavity, as shown in Figure 2.4(a). This innovative design combined the acoustic damping of the Helmholtz resonator with the material damping of the flexible plate. The concept was supported by a theoretical waveguide theory model and was experimentally validated using the grazing flow facility at DLR in Berlin, demonstrating effective dissipation of low-frequency noise. Similarly, Knobloch et al. [16, 39] introduced the flexible structure liner design incorporating flexible cavity walls, as shown in Figure 2.4(b). This design intentionally promoted acoustic-structure interaction at the cavity walls, enhancing the liner's damping characteristics by incorporating additional energy dissipation mechanisms. Neubauer et al. [23] conducted experimental and numerical investigations on a curved liner featuring a perforated face sheet combined with flexible walls. This design integrated the high acoustic dissipation provided by the apertures with the low-frequency geometric resonance of a flexible plate, thereby enhancing low-frequency and broadband acoustic attenuation. The design is illustrated in Figure 2.4(c). Additionally, their study addressed the manufacturing challenges associated with curved liners. Murata et al. [40] proposed a liner with a gap between the fine perforated film (FPF) and the surface of the acoustic liner panel to mitigate the effect of flow entering the liner openings, as shown in Figure 2.4(d). Their study suggested that the fine perforated film with a gap structure has potential for designing high-efficiency liners in grazing flow conditions.

Foam-metal liners

Porous metals can serve as acoustic liners capable of functioning under extreme conditions, including high temperatures and pressures. Tuasikal et al. [27] investigated two well-known types of stainless-steel porous materials—sintered fiber and cellular foam—with varying

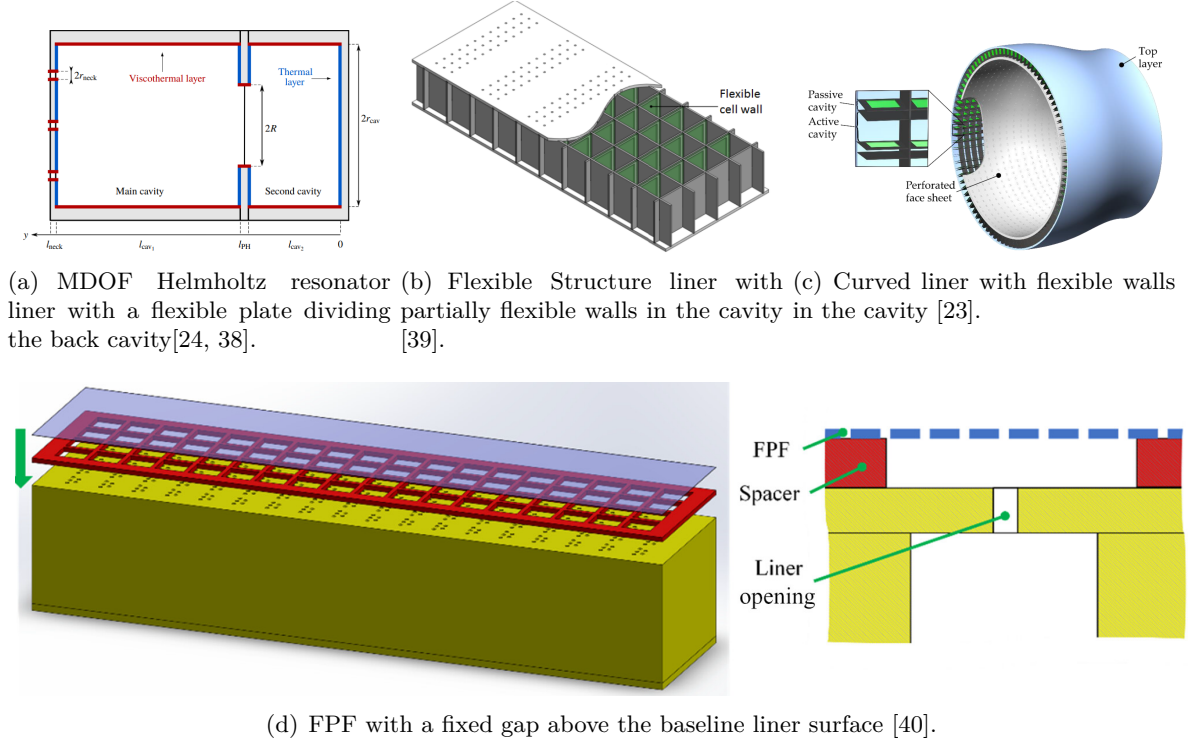
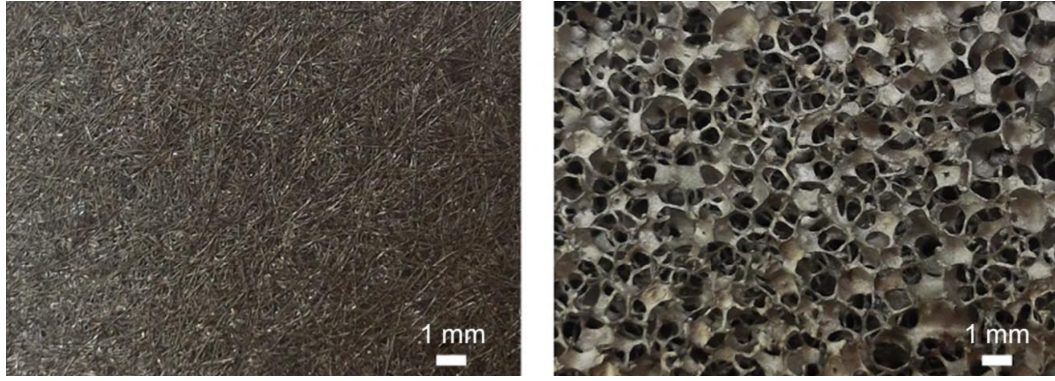


Figure 2.4: Flexible plate liners

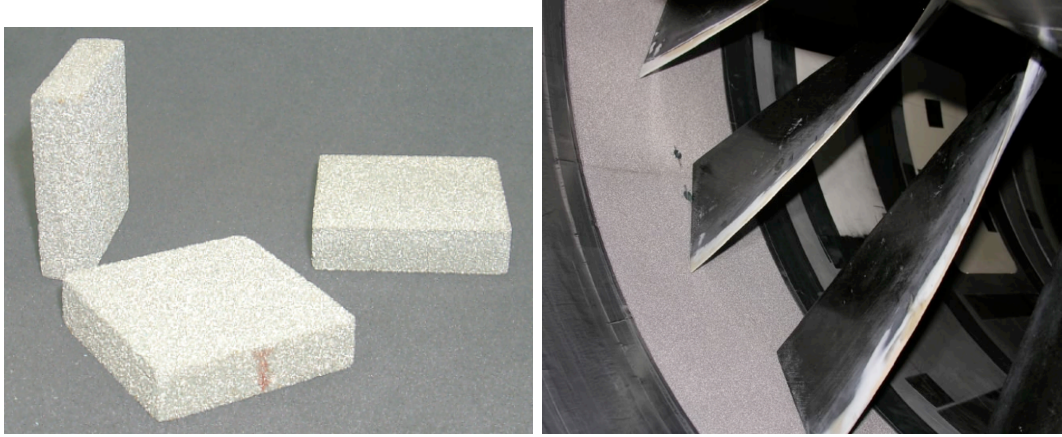
porosities, to assess their acoustic performance as liners under grazing flow conditions, as shown in Figure 2.5(a). They found that the impact of grazing flow on the impedance of these porous metals depended on the pore structure and porosity of the samples. In fact, as early as 2008, NASA [25, 26] developed and tested a foam-metal liner for low-speed fan noise reduction. As shown in Figure 2.5(b), this foam-metal liner was made from a cobalt alloy due to its consistent acoustic properties throughout the material. When it was placed near the rotor, where it disrupted the noise-generation mechanism, the liner offered improved attenuation. This design not only reduced noise by up to 4 dB but also decreased the number of engine components and weight by replacing the fan rub strip, thereby enhancing overall engine performance.

Active-control liners

DLR [16, 28–30] proposed the Zero Mass Flow Liner (ZML) concept, as shown in Figure 2.6(a). In this design, the perforated face sheet was connected to a single large volume, where a periodic bias flow was introduced through the liner’s perforated faceplate to control impedance. This periodic bias flow was actuated by a secondary acoustic source positioned at the back of the air cavity. By adjusting the flow’s characteristics, such as frequency



(a) Sintered fiber (left) and cellular foam (right) [27].



(b) Foam-metal structure made from a cobalt alloy (left) and foam-metal liner installed over a rotor (right) [25, 26].

Figure 2.5: Foam-metal liners

and amplitude, the liner's acoustic properties could be tuned to suit various grazing flow velocity conditions. Subsequently, the hybrid ZML concept was introduced, combining a conventional honeycomb structure with an excited ZML cavity, as shown in Figure 2.6(b) [16, 41]. For aero-engine applications, a jet-edge actuator powered by bleed air might have replaced loudspeakers as the excitation source [16, 29]. Experimental results demonstrated that both ZML and hybrid ZML effectively provided broadband dissipation of high peak levels across a range of grazing flow Mach numbers. Dodge et al. [31], Solano and Cattafesta [32] introduced an embedded dielectric elastomer (DE) membrane acoustic liner, which differed from traditional SDOF acoustic liners. In this design, a DE membrane was placed between the perforated face sheet and the rigid backing, as shown in Figure 2.6(c). DE membranes are electroactive polymers that deform when an electric field is applied. The membrane was pre-stressed to create an in-plane tensile force. Upon applying the electric field, this stress

was reduced, increasing the membrane's compliance. As a result, the resonance frequency of the acoustic liner decreased, enabling dynamic tuning of its acoustic impedance.

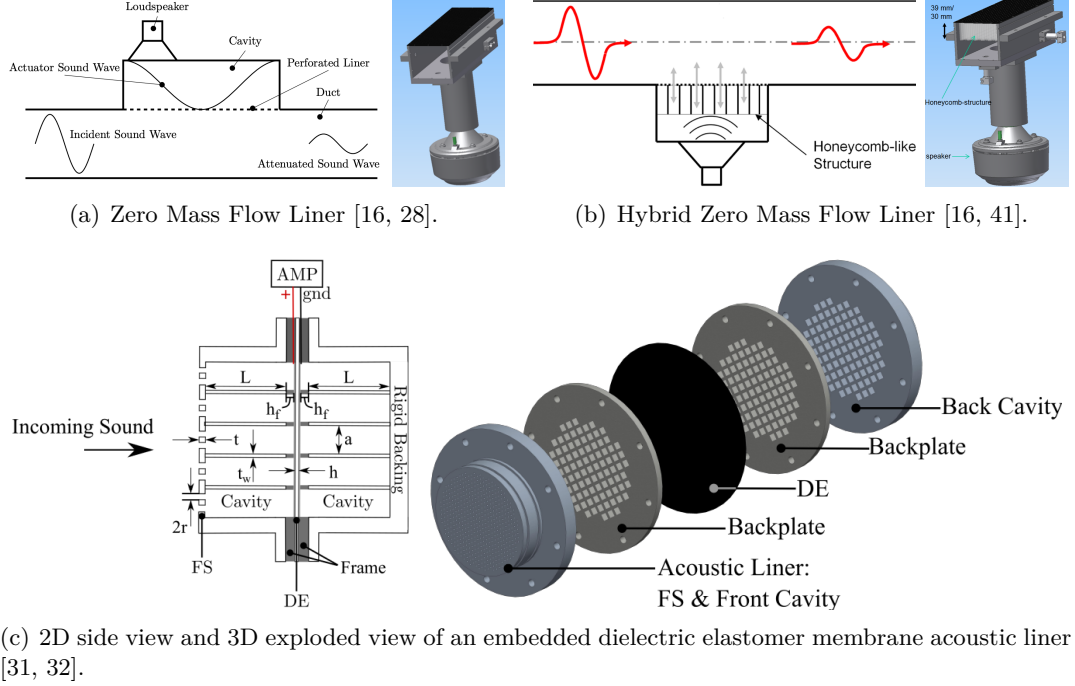


Figure 2.6: Active-control liners.

2.3 Micro-Perforated Panel Absorber

2.3.1 MPPA basic configuration

A micro-perforated panel absorber is a simple and highly versatile acoustic absorber [42–58]. Building on the theories of Rayleigh [59] and Crandall [60], Maa [42–44] first proposed a general theory and design of the MPPA as a useful noise absorbing technology in 1975. The traditional MPPA consists of a front micro-perforated panel backed by an air cavity. When sound waves pass through the micro-perforations, viscous and thermal losses occur, effectively dissipating acoustic energy. The rigid backing seals the structure, forming enclosed resonant cavities. The MPPA is more simple in structure compared to porous material-based metamaterials [61, 62], and compared to membrane type acoustic metamaterials [63–66], they are more robust and less vulnerable to high temperatures. However, the sound absorption bandwidth of traditional micro-perforated panel structures is limited. Thus, effective low-frequency and broadband sound absorption in a light, small form-factor micro-perforated panel structure is still a considerable challenge.

Passive MPPA

Extensive research has been conducted on MPPAs to improve their sound absorption properties. Min and Guo [67] proposed a novel MPPA design, in which arrays of sub-cavities with varying depths were arranged in parallel behind a micro-perforated panel, as shown in Figure 2.7(a). This design theoretically achieved an average absorption coefficient of more than 0.5 and a maximum absorption coefficient of 0.98 in the frequency range from 450 to 3500 Hz, although the maximum cavity depth reached 100 mm. As early as 2011, Wang and Huang [45] examined the sound absorption of a compound MPPA array comprising three parallel MPPAs with varying cavity depths. Their results indicated that the MPPA array required lower acoustic resistance for effective sound absorption compared to a single MPPA. Bucciarelli et al. [68] proposed a multi-layer MPPA design, where multiple micro-perforated panel layers were arranged in parallel and separated by multiple air gaps, as shown in Figure 2.7(b). Both five-layer and six-layer MPPA designs theoretically and experimentally demonstrated over 90% sound absorption in the frequency range from 400 Hz to 2000 Hz. However, the entire thickness of these MPPA designs was over 110 mm (five-layer MPPA) and over 132 mm (six-layer MPPA), respectively. Carbajo et al. [69] proposed the design of an MPPA with micro-perforated partitions, as shown in Figure 2.7(c). This design achieved a broader bandwidth of sound absorption than traditional hard wall partitioned MPPAs. Wang and Li [70] proposed an MPPA design of hybrid-arranged perforated panels with perforated partitions, as shown in Figure 2.7(d). Similar to Refs. [67, 68], this design improved the sound absorption of the MPPA but required a deep air back cavity.

Several studies have proposed various MPPAs incorporating trapezoidal cavities, corrugated, and wavy micro-perforated panels, which significantly improved the sound absorption bandwidth or enhanced low-frequency noise control capabilities. Wang et al. [47] introduced an MPPA incorporating a trapezoidal cavity, as shown in Figure 2.8(a). The incorporation of a trapezoidal cavity into the MPPA significantly altered the sound absorption mechanisms and the frequency distribution of its absorption coefficients, compared to conventional MPPAs with a uniform air gap. Specifically, the MPPA with a trapezoidal cavity generated additional spectral peaks and enhanced absorption performance at the absorption curve's troughs. This characteristic can be further exploited to optimise the effective absorption range by adjusting the cavity's shape and dimensions. Wang and Liu [71] investigated the sound absorption properties of incorporating a corrugated micro-perforated panel, as shown

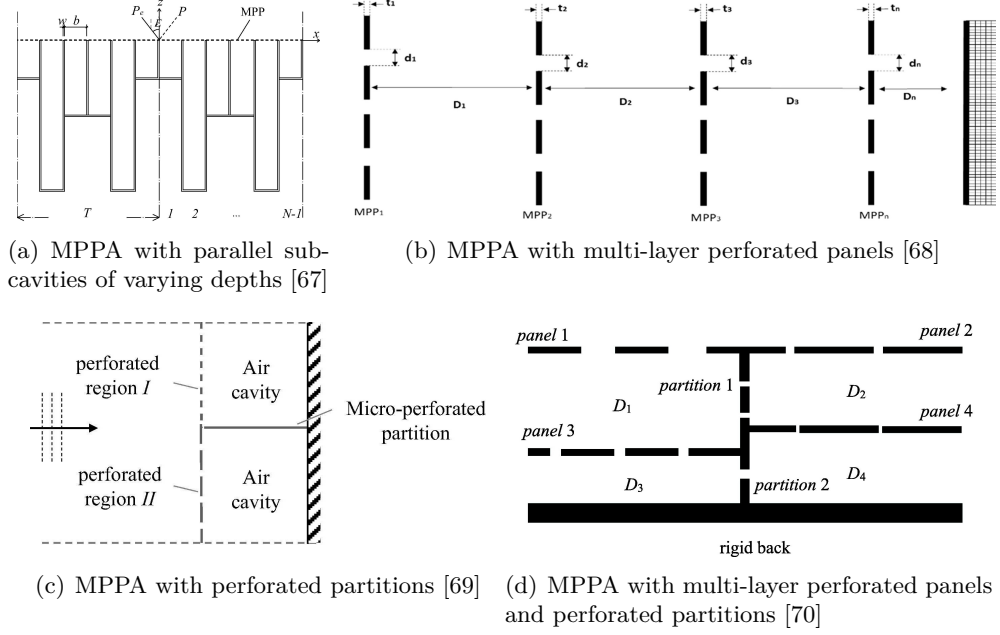


Figure 2.7: MPPA designs with varying cavity depths, layers, or partitions.

in Figure 2.8(b). The experimental results demonstrated that, compared to a flat micro-perforated panel, the corrugated MPPA significantly enhanced sound absorption at the dips of the absorption curve. This improvement was attributed to the excitation of multiple acoustical modes at both the peak and dip frequencies. Yang et al. [72] conducted a study on the sound absorption capabilities of an MPPA incorporating a wavy micro-perforated panel, as shown in Figure 2.8(c). The normal sound absorption curve of the MPPA with the wavy micro-perforated panel shifted to a lower frequency range compared to those of the MPPA with a flat micro-perforated panel and the MPPA with a corrugated micro-perforated panel, which held promise for low-frequency noise control.

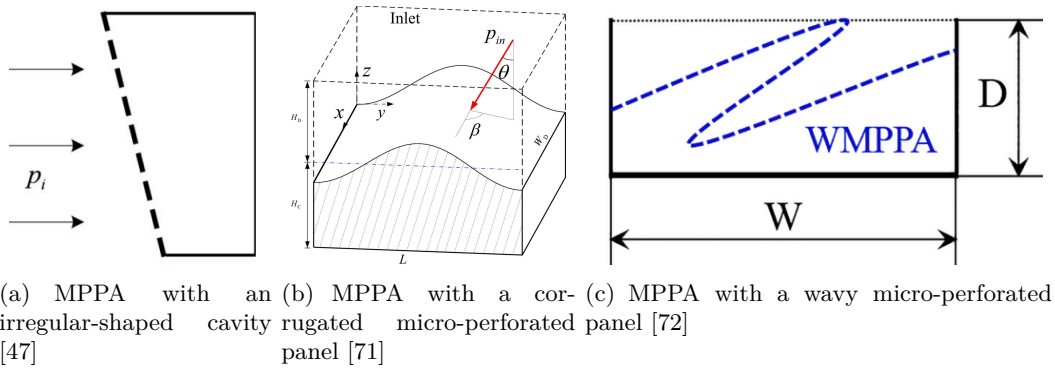


Figure 2.8: MPPA designs with trapezoidal cavities, corrugated, and wavy micro-perforated panels.

Active MPPA

Recently, there has also been interest in changing the material of the micro-perforated panel and in adding active control. Kong et al. [73] investigated an active MPPA that incorporated a micro-perforated panel made from polyvinylidene fluoride (PVDF) film, as shown in Figure 2.9(a). By applying specific voltages and oscillation frequencies, the sound absorption coefficients of the absorber were significantly enhanced at the driving frequencies. This design was compact and provided adjustable acoustic properties, potentially offering a novel approach to intelligent noise control. Liu et al. [46] proposed an active MPPA designed to enhance low-frequency sound absorption by integrating a conventional MPP with a surface-mounted piezoelectric ceramic (PZT), which was shunted to an electrical circuit, as shown in Figure 2.9(b). A 3D finite element model was employed to examine the coupling effects between the MPP and the shunted PZT, revealing three key energy dissipation mechanisms: mechanical damping, electrical shunt damping, and Helmholtz resonance. The study investigated several multimode shunt configurations to optimise panel vibrations and, consequently, sound absorption. The results demonstrated that a low-frequency broadband sound absorber can be achieved by exploiting the strong coupling among the acoustic, electrical, and mechanical domains, with absorption peaks occurring below the Helmholtz resonance frequency of the MPP. Ma et al. [74, 75, 76] employed one or two point sources positioned near the backing wall of the cavity to actively enhance the sound absorption of the MPPA, as shown in Figure 2.9(c). The absorption performance was significantly improved across the entire low-frequency range, particularly below 200 Hz [74].

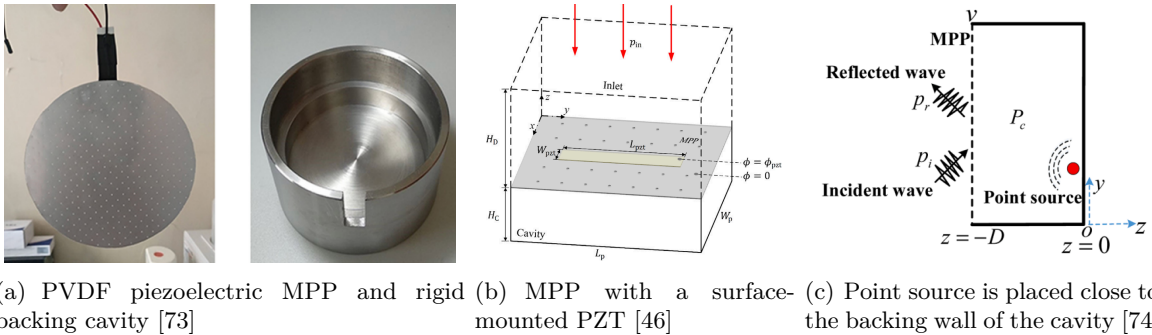


Figure 2.9: Active MPPA designs

Compared to passive MPPAs, active MPPAs offer tunability of the absorption performance. However, they also come with several drawbacks, including increased system com-

plexity, higher cost, reliance on external power and control algorithms, reduced long-term stability, and greater maintenance requirements. Therefore, their application requires a careful trade-off between performance gains and practical implementation challenges.

MPPA manufacturing

Research progress in micro-perforated sound absorbers is greatly influenced by the evolution of process and manufacturing technologies [77, 78]. The advancement in these technologies can result in the development of more advanced and efficient MPPAs [77, 79–81]. Despite the substantial advancements achieved in the past two decades, there remain limitations in their scope and nature. As early as 1998, Maa [44] postulated that the attainment of broadband sound absorption in a single-layer micro-perforated sound absorber could be realized through a reduction of the perforation diameter (below 0.1 mm) and an increase in the perforation ratio [55, 82]. However, the creation of perforations of such diminutive size, particularly in instances where the perforation ratio is substantial, presents a huge challenge and results in subpar economic efficiency for traditional processing techniques, such as machine punching, and for additive manufacturing technologies, such as 3D printing. Therefore, the design and fabrication of an MPPA with excellent acoustic performance within the confines of current common processing and manufacturing technologies constitutes a noteworthy undertaking, with far-reaching practical applications in industry and significant economic value.

MPPA acoustic modeling

With the development of MPPA, accurate analysis of acoustic systems becomes more and more critical. The equivalent circuit model (ECM) method [69, 83] uses electrical circuits to characterise the operational parameters of an acoustic system. The ECM usually uses $\Delta - Y$ and $Y - \Delta$ transforms to simplify the circuit to estimate acoustic impedance. However, as the complexity of the acoustic system increases, the ECM becomes unmanageable. For this reason, Davis et al. [84] proposed a graph-theory approach, which is essentially the two-point impedance theory proposed by Tzeng and Wu [85], but used instead to estimate acoustic impedance. The sound absorption coefficient results obtained by this graph-theory-based approach were in good agreement with those obtained by numerical simulations. This initial study provided evidence of the great potential of the two-point impedance theory for complex acoustic analysis. In the current work, a two-point impedance method is developed and employed to further the state-of-the-art in MPPA research.

2.3.2 MPPA integrated with other acoustic materials/structures

To overcome the limitations of MPPA—including insufficient low-frequency absorption, narrow absorption bandwidth, and sensitivity to complex acoustic or flow environments—researchers have increasingly integrated MPPAs with other types of acoustic materials or structures, such as porous materials, Helmholtz resonators, membranes, and coiled cavities. These hybrid systems combine the strengths of different absorptive mechanisms, enabling broader frequency coverage, enhanced low-frequency performance, and improved overall efficiency and adaptability in practical applications.

MPPA integrated with the membrane

The integration of MPPA with membranes offers a promising approach to enhance acoustic absorption performance. McKay et al. [63] introduced an innovative sound absorber, named the Segmented Membrane Sound Absorber (SeMSA), as shown in Figure 2.10(a), which leveraged the exceptional visco-thermal loss properties of MPP and broadened its frequency response through the integration of decorated membrane resonators. Experimental results demonstrated that this compact SeMSA exhibited effective low-frequency broadband sound absorption, achieved at a deep sub-wavelength thickness. Killeen et al. [86] effectively evaluated the performance of SeMSA in reducing axial fan noise in an IT server rack application. The experimental setup for measuring fan sound power is illustrated in Figure 2.10(b). The SeMSA deflector plate liner achieved a notable overall noise reduction of 2.5 dB, corresponding to a 43.9% decrease in total sound emissions. This improvement was achieved with a material thickness of only 15 mm. Notably, a 3 dB reduction was observed at the blade passing frequency of 620 Hz. To improve the sound absorption performance of a single-layer MPP, Gai et al. [87, 88] proposed a composite MPPA design consisting of a single-layer MPP integrated with membrane cells and mass blocks, as shown in Figure 2.10(c). Their results indicated that the sound absorption performance increased with the addition of membranes and mass. By adjusting the size of the membrane cells as well as the number, size, and position of the mass blocks, it was possible to achieve a broader absorption bandwidth and higher absorption peaks than those of a conventional single-layer MPP.

Integrating a membrane with a MPPA offers a promising approach to achieving compact, broadband, and low-frequency acoustic absorption. However, this hybrid configuration introduces additional complexity in terms of design, fabrication, and long-term reliability.

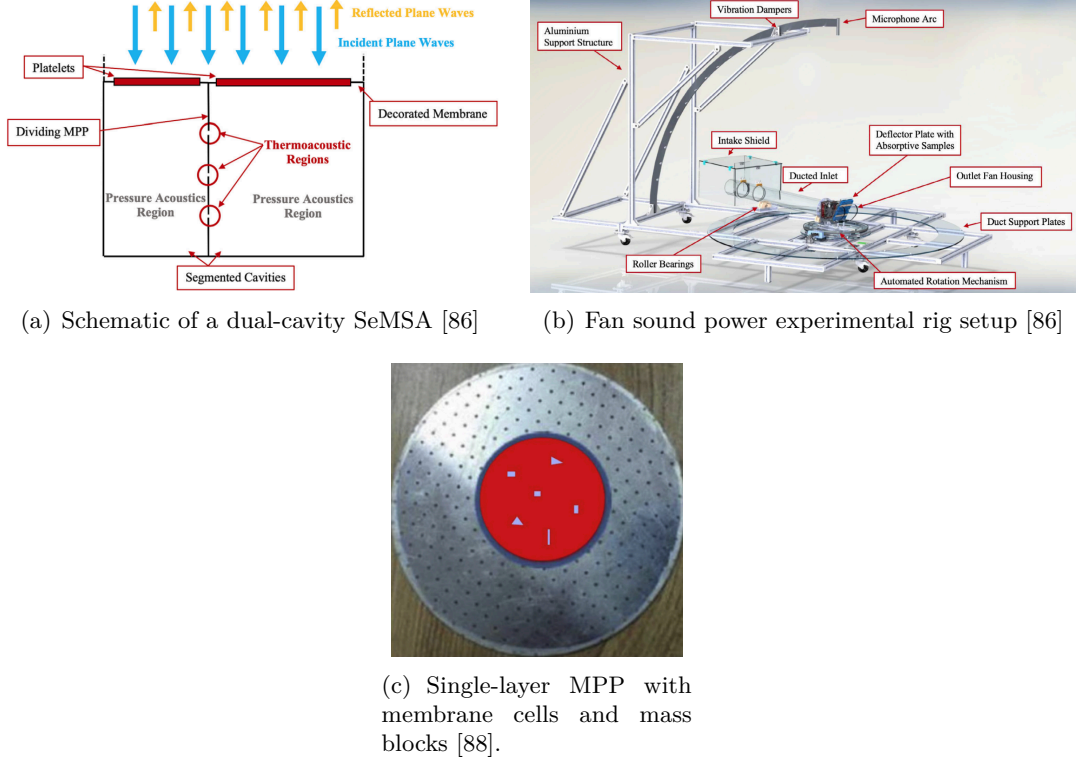


Figure 2.10: MPPA integrated with membranes.

As a result, it is not suitable for operation in extreme environments, such as in aero-engine nacelle liners, where high-speed airflow, and intense mechanical vibrations may compromise the structural integrity and performance stability of the absorber.

MPPA integrated with the porous materials

Another promising method to improve the sound absorption performance of MPPA involves combining the MPP with porous materials. Li et al. [89] proposed a three-layer MPPA design comprising a micro-perforated panel layer, an air cavity, and a porous material layer to enhance the sound absorption bandwidth. The study specifically investigated how the position of the porous material layer affected the acoustic performance of the MPPA, as shown in Figure 2.11(a). The results indicated that placing the porous material in front of the MPP with an intervening air layer significantly improved sound absorption compared to positioning it behind the MPP. Similar to Li et al. [89], Khosravani and Reinicke [90] also proposed a three-layer MPPA design consisting of an MPP layer, a porous material layer, and an air-back cavity, as shown in Figure 2.11(b). Their study utilized advanced manufacturing techniques for fabricating MPPs, and the results indicated that the perforation porosity and

air gap significantly affected the resonant frequency corresponding to the maximum sound absorption coefficient. Furthermore, the performance of the 3D-printed MPPs highlighted the ability of 3D printing to fabricate high-performance sound-absorbing panels. Li et al. [91] proposed a compound MPPA consisting of an MPP, parallel-arranged perforated plates with extended tubes (PPETs), and porous sound-absorptive material (PSAM), as shown in Figure 2.11(c). This design aimed to enhance sound absorption in the frequency range from 100 Hz to 1600 Hz. Experimental results showed that this design achieved a sound absorption coefficient greater than 0.5 from 225 Hz to 1100 Hz. Kim and Yoon [92] proposed an MPPA configuration that incorporated a perforated panel and a porous separating partition, as shown in Figure 2.11(d), to extend the sound absorption bandwidth. Numerical simulations demonstrated that, after optimisation, the design effectively broadened the absorption range.

However, the inclusion of porous materials inevitably increased the absorber's weight and, as with all porous materials, made it susceptible to moisture and dust accumulation.

MPPA integrated with Helmholtz resonators

However, membrane and porous materials are often sensitive to environmental factors such as temperature, humidity, and pressure, which limits their effectiveness in harsh operating conditions. Helmholtz resonator [93–98] is being re-examined recently and is playing an increasingly important role in the field of sound absorption. The Helmholtz resonator, characterized by its enclosed air cavity and a small aperture linking it to the external environment, demonstrates resonance at a distinct frequency. This resonant frequency is largely dependent on the cavity's internal volume and the physical properties of the aperture. Compared to the MPPA for the same depth, Helmholtz resonators can function at lower frequencies, but their absorption bandwidth is considerably narrower. As a result, they are primarily used for attenuating tonal noise in the low-frequency range. So far, the singular application of traditional acoustic materials such as MPPAs, or Helmholtz resonators seems to be somewhat inadequate for achieving perfect low-frequency and broadband absorption in the deep sub-wavelength regime. The integration of MPPA with Helmholtz resonators presents a promising avenue for research into broadband, low-frequency sound absorption in the deep sub-wavelength regime [97, 99].

Park [95] proposed a design that integrated an MPPA with a backing Helmholtz resonator, as shown in Figure 2.12(a). This configuration significantly enhanced sound absorption, particularly at low frequencies. To reduce the overall thickness of the absorber, the neck of the

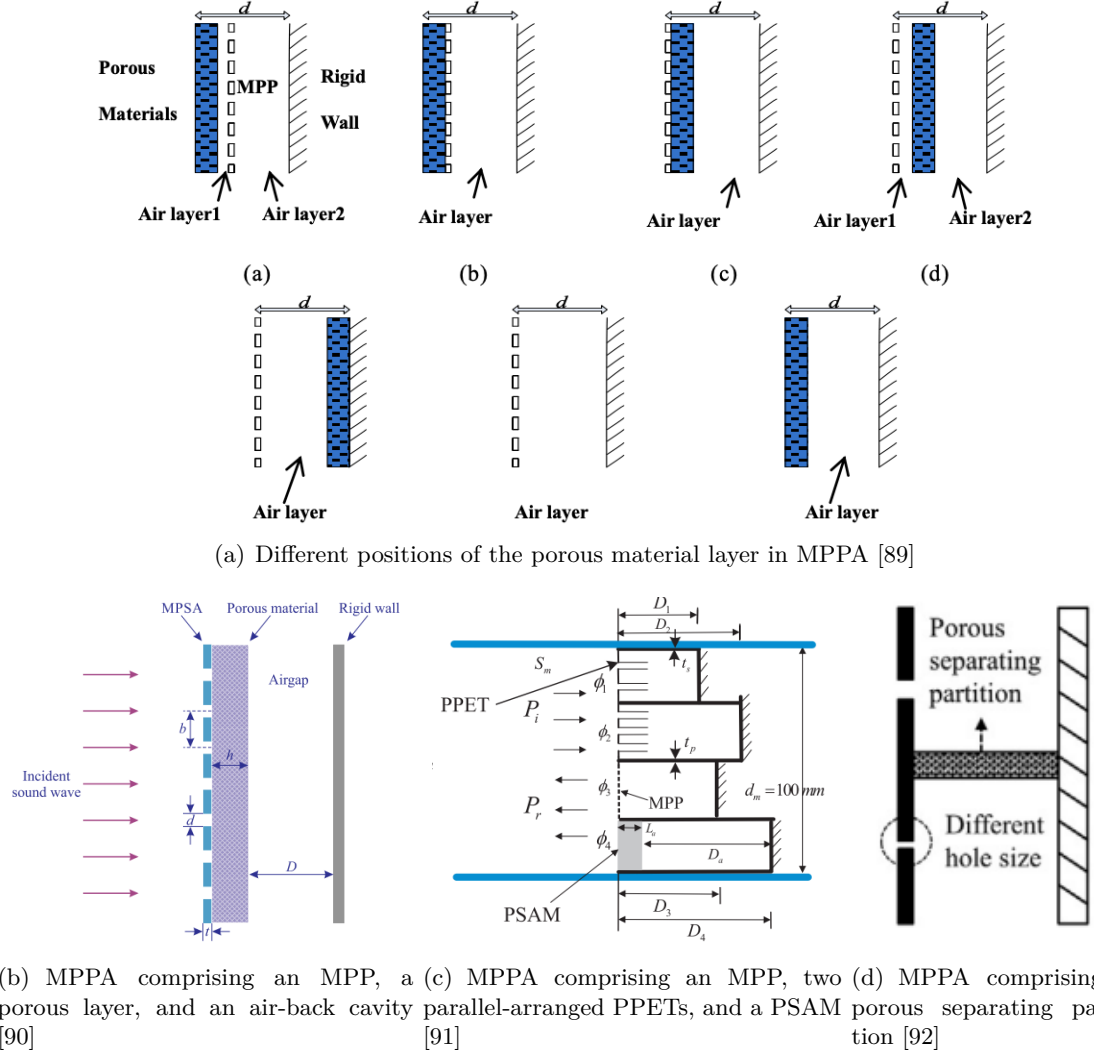
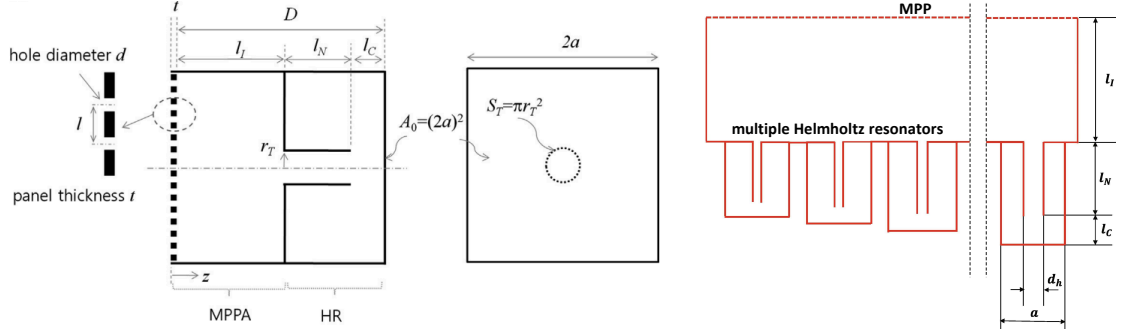
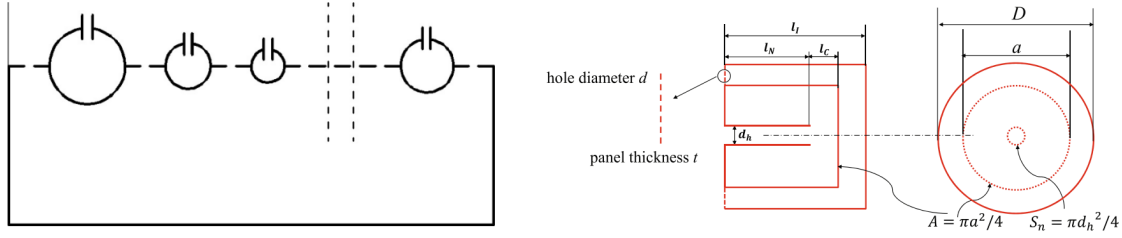


Figure 2.11: MPPA integrated with porous materials.

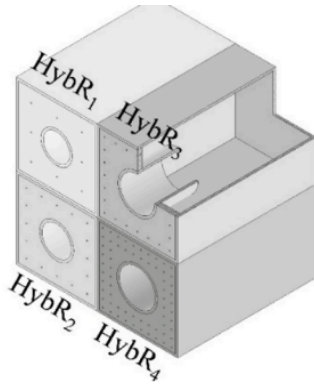
Helmholtz resonator was extended into its cavity. The resulting structure exhibited two absorption peaks, occurring slightly below and above the respective resonant frequencies of the uncoupled Helmholtz resonator and MPPA. Subsequently, Mahesh and Mini [99] conducted a theoretical investigation into the sound absorption performance of an MPPA backed by multiple extended-neck Helmholtz resonators, as shown in Figure 2.12(b). This configuration resulted in multiple absorption peaks in the low-frequency range, thereby broadening the effective absorption bandwidth. Beck et al. [103, 104] proposed, as early as 2015, a dual-resonance liner that combined Helmholtz resonators with an MPPA, enhancing low-frequency absorption by introducing a second low-frequency resonance. At the designed Helmholtz resonance frequency, the normal incidence absorption coefficient was over ten times higher than that of a conventional POHC liner, while maintaining similar performance at higher frequen-



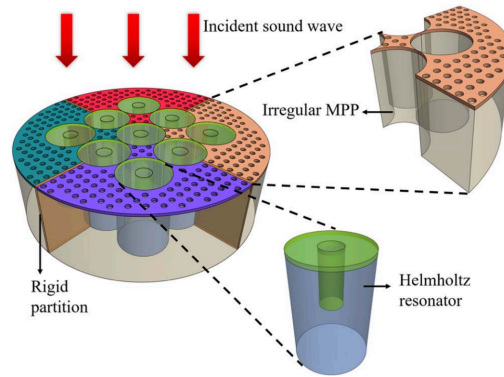
(a) MPPA backed by an extended-neck Helmholtz resonator [95] (b) MPPA backed by multiple extended-neck Helmholtz resonators [99]



(c) MPPA with parallel conventional Helmholtz resonators and a single MPP [100] (d) MPPA with parallel extended-neck Helmholtz resonators and a single MPP [99]



(e) MPPA with multiple MPPs and extended-neck Helmholtz resonators [101]



(f) MPPA with parallel extended-neck Helmholtz resonators and MPPs [102]

Figure 2.12: MPPA integrated with Helmholtz resonators.

cies. Similarly, Gai et al. [100] introduced a design that integrated an MPPA with conventional Helmholtz resonators, as shown in Figure 2.12(c). Results demonstrated two absorption peaks and one anti-resonant frequency. The Helmholtz resonators predominantly influenced the low-frequency peak, while the high-frequency peak closely corresponded to that of a standalone single-layer MPP. Notably, employing a parallel arrangement of multiple Helmholtz resonators with the MPP structure enhanced sound absorption performance, particularly in the low-frequency range, by introducing additional absorption peaks. Similarly, Mahesh and

Mini [99] proposed an MPPA design that integrated an MPP with extended-neck Helmholtz resonators using a parallel arrangement, as shown in Figure 2.12(d). It was observed that this configuration, involving both the MPP and multiple Helmholtz resonators in parallel, significantly reduced the overall thickness of the absorber. Ryoo et al. [101] also proposed an MPPA design featuring a parallel arrangement of an extended-neck Helmholtz resonator and an MPP within a single unit cell. To further broaden the absorption bandwidth, a supercell configuration comprising multiple single unit cells was explored, as shown in Figure 2.12(e). Experimental results demonstrated that these modified designs can attain an absorption coefficient exceeding 0.9 within the frequency range of 472 to 739 Hz, despite having a subwavelength thickness of $\lambda/12$. Li et al. [102] proposed an MPPA design consisting of parallel arrays of extended-neck Helmholtz resonators and MPPs, as shown in Figure 2.12(f). The Helmholtz resonator cavities and the backing air cavities of the MPPs were independent. Experimental results demonstrated that a 40 mm-thick structure provided effective broadband sound absorption at low frequencies, covering the range from 375 to 1600 Hz. Other studies [105–107] have also made efforts to integrate MPPA with Helmholtz resonators. In general, one notable advantage of this approach is its ability to achieve efficient sound absorption in the low-frequency range as well as broadband sound absorption in the low-to-medium frequency range. However, a potential challenge associated with this combination is the increase in absorber thickness, which can pose limitations to its practical application in various industries.

MPPA integrated with coiled-up channels cavities

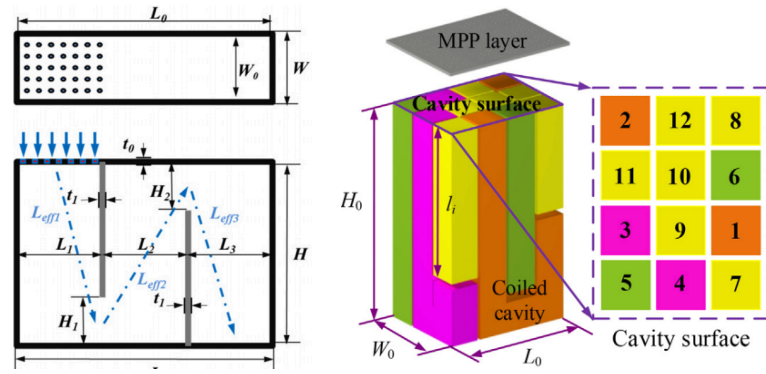
MPPA structures are typically limited in their low-frequency performance due to the limited cavity depth. To overcome this limitation, coiled-up channel cavities [108–110] have been introduced to extend the effective acoustic path length without increasing the overall dimensions of the structure. By directing sound waves through a folded or meandering path, these channels effectively lower the system's resonant frequency, thereby improving low-frequency absorption within a compact design.

Wu et al. [111] developed an MPPA design by integrating an MPP with coiled Fabry–Pérot (FP) channel cavities, as shown in Figure 2.13(a). Theoretical, numerical, and experimental results demonstrated that acoustic absorption exceeding 99% can be achieved at a resonant frequency below 500 Hz, with a subwavelength thickness of only $\lambda/30$. Liu et al. [112] further developed an MPPA design consisting of 12 units, as shown in Figure 2.13(b), and the overall

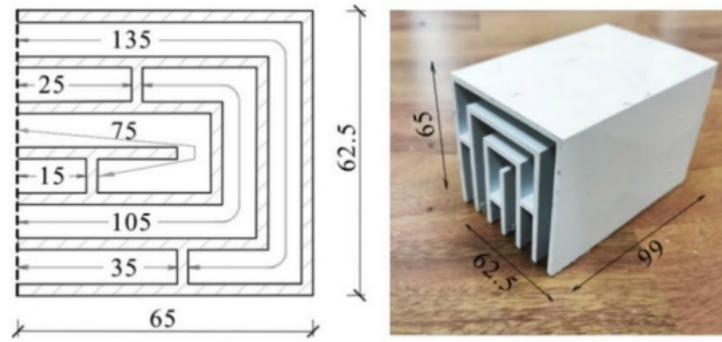
structure was characterized by multi-order peaks in the low-to-mid frequency range. Each unit of this structure comprised an MPP backed by a coiled cavity. This MPPA design exploited the cavity's multi-order resonance and the MPP's broadband absorption capacity. As a result, it not only secured high-order peaks of exceptional magnitude but also ensured a broader bandwidth for each peak. This structure achieved a broadband absorption in the frequency range of 380 Hz–3600 Hz with a total thickness of 72 mm. Min et al. [113] presented an MPPA design by integrating an MPP with parallel coiled-up cavities of different depths, as shown in Figure 2.13(c). The study evaluated the effective depths of the coiled-up cavities. It is shown that, for evaluating the effective depth of the coiled-up cavities, the diagonal lines of the subchannels in cavities with a coiled-up angle of 180° can accurately represent the effective depths, while the combination of the subchannel centerlines and a quarter arc within the coiled-up area was more suitable for those with a coiled-up angle of 90° . This design was subsequently applied to noise reduction in traffic tunnels [114]. Results showed that the implementation of an absorber on one sidewall of the tunnel reduced the reverberation time by 69%, improved the speech transmission index from 0.11 to 0.32, and enhanced noise attenuation by 17 dB at a distance of 100 m, approaching semi-free field conditions [114].

However, integrating MPPAs with coiled-up channel cavities also introduces some drawbacks. First, the design of coiled channels can reduce thickness but may increase both the absorber's volume and structural complexity. This could complicate integration in space-constrained applications, such as aircraft or other high-density environments. Second, coiled channels may increase flow resistance, particularly under high-speed or turbulent airflow conditions, which could impact the system's aerodynamic performance and potentially lead to unstable flow phenomena.

Table 2.3.2 summarises and compares the advantages and disadvantages of the acoustic materials and structures integrated with MPPA discussed in this section, highlighting their performance trade-offs.



(a) MPPA comprising an MPP backed by a coiled-up channel [111]
 (b) MPPA consisting of 12 units, each MPP comprising an MPP and a coiled cavity. [112]



(c) MPPA comprising an MPP backed by parallel coiled-up cavities of different depths [113]

Figure 2.13: MPPA integrated with coiled-up channels cavities

Table 2.2: Comparison of characteristics of different acoustic materials/structures integrated with MPPA

Acoustic materials/structures	References	Advantages	Disadvantages
Membrane	• McKay et al. [63] • Killeen et al. [86] • Gai et al. [87] • Gai et al. [88]	• Enhanced low-frequency absorption • Fibre-free configuration	• Susceptible to temperature and pressure variations • Low structural robustness • Complex design
Porous materials	• Li et al. [89] • Khosravani and Reinicke [90] • Li et al. [91] • Kim and Yoon [92]	• Broadband sound absorption • Relatively easy fabrication	• Material degradation over time • Susceptible to moisture and dust
Helmholtz resonators	• Park [95] • Mahesh and Mini [99] • Beck et al. [103] • Beck [104] • Gai et al. [100] • Ryoo et al. [101] • Li et al. [102] • Zhao et al. [105] • Aurégan and Farooqui [106] • Li et al. [107]	• Enhanced low-frequency absorption • Fibre-free configuration	• Increased structural thickness
Coiled-up cavity channels	• Wu et al. [111] • Liu et al. [112] • Min et al. [113] • Min et al. [114]	• Enhanced low-frequency absorption • Space-efficient design • Fibre-free configuration	• Complex design and fabrication • Potential flow resistance and airflow issues

2.4 Key Contributions of the Thesis

This study investigates microperforated acoustic metasurfaces for low-frequency broadband noise reduction. Within the scope of this work, it provides insights into their acoustic modelling and explores practical design strategies for potential applications in aircraft engine noise control. The key contributions of this study are as follows:

First, an optimised multi-chamber micro-perforated panel absorber with micro-perforated adjoining panels is proposed. By using the TpIM based on graph theory, a model for multi-chamber MPPAs which are coupled with each other in two dimensions through perforated side panels can be developed, an onerous task for standard equivalent circuit analysis due to the generation of non-planar circuits. The resulting model lends itself to optimisation and thus allows the differing geometric parameters of a tessellated network of chambers to be combined to maximise the sound absorption in a frequency range for a particular depth. An average absorption coefficient of 0.83 can be achieved experimentally in the frequency range of 660 Hz to 2 kHz with an MC-MPPA of only 20 mm thick. In addition, when the air cavity depth is slightly increased to 50 mm, an average absorption coefficient of greater than 0.82 can be achieved in the 281 Hz to 1 kHz frequency range. Experimentally, an absorption coefficient of approximately 0.82 is achieved at 340 Hz at a depth of 50 mm, which is a depth-to-wavelength ratio of 20, making the MC-MPPA a deeply subwavelength absorber.

Second, to achieve ultra-broadband absorption at low frequencies, this study introduces an MC-MPPA with varying sub-chamber depths. Notably, the MC-MPPA test pieces experimentally achieved sound absorption coefficients exceeding 0.8 across the frequency bands from 397 Hz to 1000 Hz and from 698 Hz to 1895 Hz. The absorber's thickness is a mere 47 mm, equivalent to approximately $1/18.2$ and $1/10.5$ of the sound wavelength at the minimum operational frequency, respectively. Theoretically, with a maximum sub-chamber depth of just 20mm, average absorption coefficient values of 0.68 and 0.65 were observed over the frequency intervals of 200 Hz to 3000 Hz and 200 Hz to 4000 Hz, respectively. The optimisation algorithm permits the definition of practical geometric parameters, promising substantial industrial benefits.

Third, to extend the noise reduction capabilities of MC-MPPAs to grazing flow environments, this study designs and optimises MC-MPPAs for broadband attenuation in grazing flow acoustic incidence conditions. Mach number dependent analytical equations for impedance are input into a graph-theory-based TpIM so that the total impedance of

these highly complex multi-degree of freedom absorbers can be modeled and calculated. A novel Cremer impedance-based cost-function is presented which attempts to optimise the impedance of the absorber within a broadband target frequency range of interest. The optimisation routine converges on an optimal impedance and also provides the geometric parameters of the absorber, such as individual sub-chamber depths, hole diameters and porosities per uniquely perforated chamber top, or internal side-wall. This allows the absorber to be 3D printed and numerically modeled using COMSOL. Liner samples 200 mm in length were 3D printed and tested in a grazing flow facility as a function of Mach number - up to $M=0.25$, sound pressure level - up to 140 dB, and where the acoustic source can be located upstream or downstream of the liner. Experimental results compare extremely well with those predicted validating the analytical design and optimisation methodology. Transmission loss (TL) levels of 25 dB were achieved at 140 dB at $M=0.25$.

Fourth, to improve low-frequency broadband noise absorption, this study presents an acoustic metasurface composed of a micro-perforated panel coupled with an extended-neck Helmholtz resonator, forming a unit absorber referred to as an MHA. Individual sub-chambers can have unique properties such as depth, porosity, hole diameter, extended neck length, and neck internal diameter. An optimisation algorithm is implemented to ensure that broadband absorption is maximised in a low-frequency range of interest. The Helmholtz resonators couple with the response of the micro-perforated panels so that the combined response is both more broadband and of lower frequency than could be attained by the elements individually. Low frequencies below 500 Hz are of interest in this section, and the area exposed to the sound field is limited to 0.04 m^2 , so the technology is considered to be a locally reacting absorber. Experimental results indicate that perfect sound absorption of over 99.9% at 329.6 Hz with a 51 mm thickness (sub-wavelength ratio = $1/20.4$) can be achieved with one configuration, which emphasises maximum absorption. A second configuration, which emphasises broadband absorption, achieves an average sound absorption coefficient of 0.8 in the 254 Hz to 500 Hz range. Experimental results outperform some results found in the literature.

Finally, a deep sub-wavelength broadband MHA is developed, fabricated, and characterized for noise absorption. Experimental results demonstrate that a 48 mm deep MHA achieves near-perfect sound absorption ($>97\%$) at 150 Hz. This performance is realized with a sub-wavelength thickness of only $1/48$ of the operating wavelength and a volume-normalized

wavelength ratio of $1/51$. Additionally, the MHA exhibits a half-absorption bandwidth of 48 Hz. To broaden the sound absorption bandwidth while keeping the total area constant and to further reduce the total thickness, a novel configuration integrating three Compact MHA (CMHA) units is proposed. Experimental results demonstrate that the CMHA achieves an average sound absorption coefficient (SAC) exceeding 0.74 in the 300–500 Hz range with an overall thickness of only 27.5 mm, and an $\text{SAC} > 0.88$ in the 543–945 Hz range with a reduced thickness of 26.2 mm. Experimental results compare well with theoretical and numerical predictions, highlighting the potential of the proposed design for practical noise control applications. Furthermore, the TL performance of the CMHA liners under grazing flow is experimentally evaluated. Despite its compact size (32 mm thickness), the CMHA achieved broadband transmission loss (10–23 dB) below 1000 Hz in the absence of flow. Moreover, its transmission loss performance remained effective under grazing flow. In the presence of downstream sources and under the highest SPL and Mach number conditions, the TL reached approximately 15 dB below 1000 Hz, further extending the effective noise reduction range.

Each of these contributions has undergone rigorous validation through theoretical analysis, numerical simulations, and experimental testing, highlighting their strong potential for practical noise control applications in engineering environments.

2.5 Summary

This chapter has reviewed the current challenges and developments in turbofan aeroengine acoustic liner technologies. The review has emphasized the persistent difficulties associated with improving liner performance in reducing low-frequency noise—characterized by longer wavelengths—due to spatial constraints imposed by increasingly high bypass ratios. The development of efficient acoustic liner designs thus remains an ongoing challenge.

To address these issues, various liner configurations have been examined, ranging from conventional perforated plate over honeycomb designs to more advanced concepts such as folded cavity liners, flexible plate systems, foam-metal hybrids, and active-control liners. These technologies exhibit varying degrees of effectiveness, each presenting trade-offs in terms of frequency range, structural complexity, and suitability for integration into real-world engine nacelles.

The chapter then turned to micro-perforated panel absorbers, highlighting their advantages, including simple structure, low weight, tunable acoustic performance, and compatibil-

ity with sustainable materials. Both passive and active MPPA configurations were analyzed, along with typical manufacturing approaches and acoustic modeling techniques. In addition, a detailed review was provided on hybrid structures that combine MPPAs with other acoustic elements—such as membranes, porous materials, Helmholtz resonators, and coiled-up cavities—to enhance low-frequency and broadband performance.

In summary, MPPA-based technologies represent a versatile and increasingly promising approach to addressing the stringent requirements of modern aeroacoustic applications. Nevertheless, challenges remain in achieving deep-subwavelength broadband absorption, maintaining performance under grazing flow, and enabling practical integration within the confined spaces of engine ducts. These limitations form the basis for the research presented in the following chapters, which investigates microperforated acoustic metasurfaces for low-frequency broadband noise reduction. This research contributes to advancements in both the fundamental understanding and engineering implementation of such metasurfaces, particularly in the context of aircraft engine noise control.

Chapter 3

MC-MPPA Optimised for Broadband Absorption via the TpIM

3.1 Introduction

MPPAs possess a structurally simpler configuration and demonstrate superior mechanical robustness and thermal resistance compared to porous material-based acoustic metamaterials [61, 62] and membrane-type counterparts [63–66]. Despite these advantages, traditional MPPA designs are inherently constrained by a limited sound absorption bandwidth. As such, realizing efficient low-frequency and broadband acoustic absorption within a lightweight and compact MPPA framework remains a critical and ongoing research challenge.

In this Chapter, a novel, deeply sub-wavelength, low-frequency, broadband MPPA is presented. The main objectives of this Chapter are as follows: (1) to introduce the novel acoustic sound absorber, *viz.*, the multi-chamber, micro-perforated panel absorber with micro-perforated internal panels; (2) to establish a graph-theory-based predictive theoretical method, TpIM, for optimising acoustic treatments; (3) to develop an optimised MC-MPPA with high amplitude, low-frequency and broadband sound absorption; (4) to validate both the proposed predictive theoretical method and optimised MC-MPPA designs numerically and experimentally.

3.2 Multi-Chamber MPPA Design

In this section, a rectangular MC-MPPA is proposed, which is a complex acoustic system consisting of numerous micro-perforated panels and air cavities (as shown in Figure 3.1). This rectangular MPPA can be subdivided into $n_{\text{rows}} \times n_{\text{cols}}$ chambers where n_{rows} and n_{cols} are the numbers of chambers in the x -direction and y -direction, respectively. Appendix. A.1 provides a visual representation of the numbering convention used for the top plate and the internal walls of the (3×3) MC-MPPA.

Figure 3.1 shows the geometric design of the (1×1) , (1×2) , (1×3) , (2×2) , (2×3) and (3×3) chamber MC-MPPAs. One or more MPPs seal the top surface, and the remaining external surfaces of the MPPA are rigid unperforated panels. This thesis refers to the top MPPs as $\text{MPP}_{t(1)}$, $\text{MPP}_{t(2)}$, $\dots$, $\text{MPP}_{t(n_{\text{rows}} \times n_{\text{cols}})}$, respectively. Each top MPP has an air-back cavity corresponding to it. These air cavities are referred to as $\text{AC}_{(1)}$, $\text{AC}_{(1)}$, $\dots$, $\text{AC}_{(n_{\text{rows}} \times n_{\text{cols}})}$, respectively. In this Chapter, each air cavity has the same depth, D .

Regardless of the number of chambers inside the MC-MPPA, the overall shape of the MC-MPPA is rectangular, allowing it to be easily tessellated for practical industrial applications. The overall length and width of the absorber is set to be L and W respectively. The height,

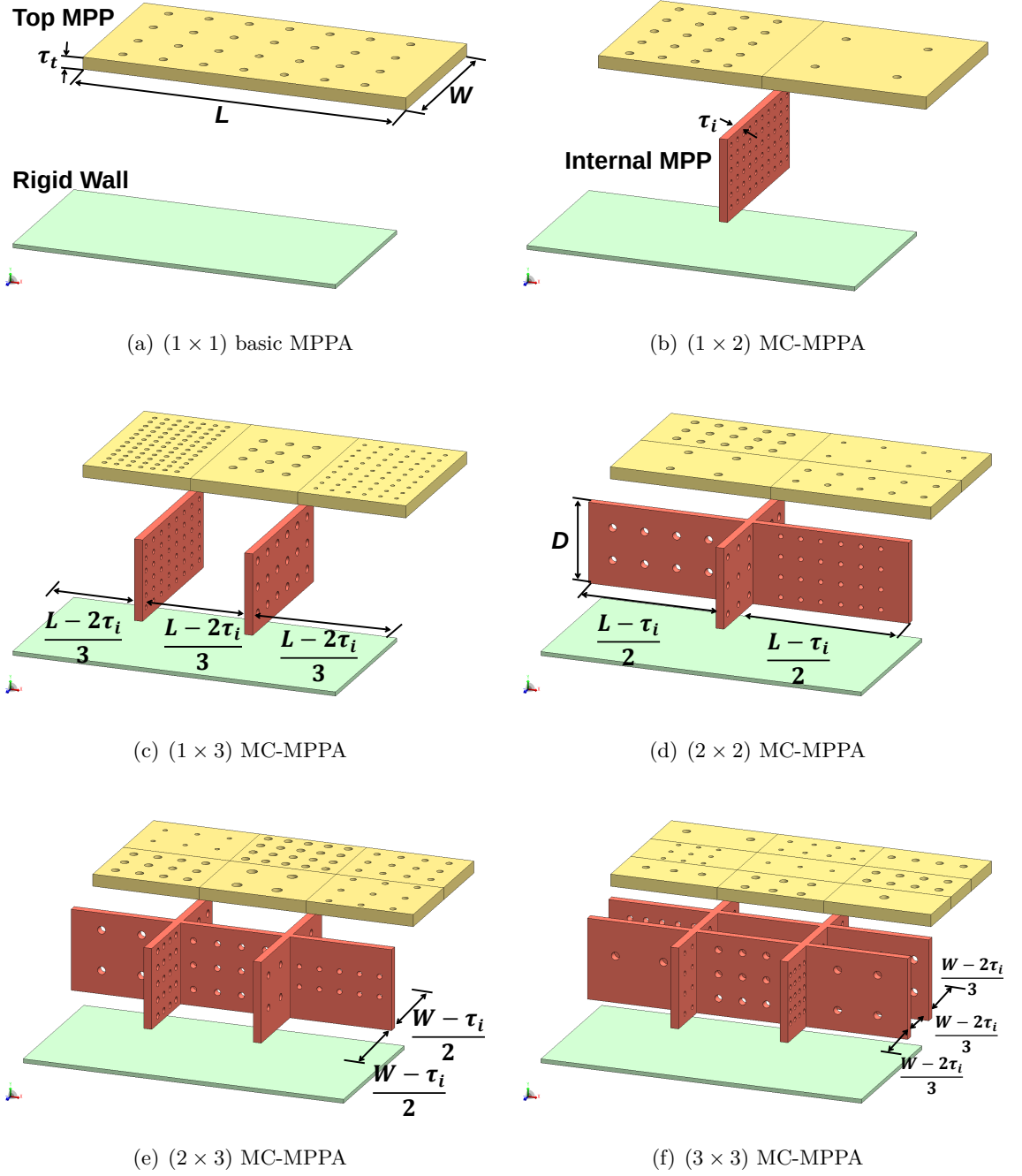


Figure 3.1: Schematic representations of the MC-MPPAs examined.

H , of the MPPA is obtained by adding the thickness of the top MPP τ_t to the maximum depth, D , of the AC.

Inside the MPPA, adjoining chambers are separated by internal side-wall MPPs. To facilitate their mathematical treatment in the optimisation process, each internal MPP is

CHAPTER 3. MC-MPPA OPTIMISED FOR BROADBAND ABSORPTION VIA THE TPIM

labelled as $\text{MPP}_i(1), \text{MPP}_i(2), \dots, \text{MPP}_{i((n_{\text{rows}}-1)n_{\text{cols}}+(n_{\text{cols}}-1)n_{\text{rows}})}$, following a serpentine naming method.

Table 3.1 presents the constitutive parameters of the $(n_{\text{rows}} \times n_{\text{cols}})$ MC-MPPA under study with the four controlling parameters for MPPs defined as follows:

- panel thickness τ ;
- perforation diameter d ;
- perforation porosity ϕ (ratio of micro-perforated area to total area).
- perforation number n

It is worth noting that in this study, all the top MPPs have the same panel thickness, τ_t , as each other. Similarly, all the internal MPPs have the same panel thickness τ_i as each other, the value of which may be the same or different to that of τ_t . Allowing these parameters to vary relative to each other would certainly increase the capacity for optimisation but would significantly increase the complexity of manufacturing. In addition, if the top MPPs were to vary in thickness independently of the cavity depths the result would be an uneven surface which would cause difficulty in applications with flow [115–117].

Table 3.1: Geometric and structural parameters of the MC-MPPA.

Location	Symbol	τ (mm)	d (mm)	ϕ	n	D (mm)
Top	$\text{MPP}_{t(1)}$	τ_t	$d_{t(1)}$	$\phi_{t(1)}$	$n_{t(1)}$	$D_{(1)}$
	$\text{MPP}_{t(2)}$		$d_{t(2)}$	$\phi_{t(2)}$	$n_{t(2)}$	$D_{(2)}$
	$\text{MPP}_{t(3)}$		$d_{t(3)}$	$\phi_{t(3)}$	$n_{t(3)}$	$D_{(3)}$
	$\vdots$		$\vdots$	$\vdots$	$\vdots$	$\vdots$
	$\text{MPP}_{t(n_{\text{rows}}n_{\text{cols}})}$		$d_{t(n_{\text{rows}}n_{\text{cols}})}$	$\phi_{t(n_{\text{rows}}n_{\text{cols}})}$	$n_{t(n_{\text{rows}}n_{\text{cols}})}$	$D_{(n_{\text{rows}}n_{\text{cols}})}$
Internal	$\text{MPP}_{i(1)}$	τ_i	$d_{i(1)}$	$\phi_{i(1)}$	$n_{i(1)}$	—
	$\text{MPP}_{i(2)}$		$d_{i(2)}$	$\phi_{i(2)}$	$n_{i(2)}$	—
	$\text{MPP}_{i(3)}$		$d_{i(3)}$	$\phi_{i(3)}$	$n_{i(3)}$	—
	$\vdots$		$\vdots$	$\vdots$	$\vdots$	—
	$\text{MPP}_{i(G)}$		$d_{i(G)}$	$\phi_{i(G)}$	$n_{i(G)}$	—

n_{rows} : Number of sub-chambers in each row; n_{cols} : Number in each column. There are $n_{\text{rows}} \times n_{\text{cols}}$ MPPs on the top surface.

$G = (n_{\text{rows}} - 1)n_{\text{cols}} + (n_{\text{cols}} - 1)n_{\text{rows}}$: Total number of MPPs embedded in the internal walls, corresponding to partitions between adjacent sub-chambers.

“—” indicates that no cavity is associated with the internal MPPs.

The acoustic impedance Z_{mpp} of each MPP can be calculated by Eq. (3.1). In this Chapter, COMSOL's interior perforated thin panel boundary condition to calculate the MPP impedance [63, 84] is used. In a previous numerical study [63], this has been shown to agree closely with fully resolved perforated holes in a COMSOL model.

$$\frac{Z_{\text{mpp}}}{\rho_0 c_0} = - \left(\frac{i\omega}{c_0 \phi} \frac{\tau + 8d\Lambda(\phi)/3\pi}{Y_v} \right) \quad (3.1)$$

where i is the imaginary unit, ρ_0 is the air density and c_0 is the sound speed in air. The angular frequency is defined as $\omega = 2\pi f$ where f is the frequency in Hertz. The term Y_v is given by

$$Y_v = \frac{J_2(k_v d/2)}{J_0(k_v d/2)}, \quad k_v^2 = -i\omega \frac{\rho_0}{\mu} \quad (3.2)$$

where J_n is the n th order Bessel function and μ is the dynamic viscosity of air.

The hole-to-hole interaction, if any, is determined by the Fok function, $\Lambda(\phi)$, which is expressed by

$$\Lambda(\phi) = \sum_{n=0}^8 a_n (\sqrt{\phi})^n \quad (3.3)$$

where the following first eight coefficients [63] are used in this study: $a_0 = 1$, $a_1 = -1.4092$, $a_2 = 0$, $a_3 = 0.33818$, $a_4 = 0$, $a_5 = 0.06793$, $a_6 = -0.02287$, $a_7 = 0.063015$ and $a_8 = -0.01614$.

Regarding the air cavities within the MPPA, the acoustic impedance, Z_{ac} , of each AC can be calculated by Eq. (3.4).

$$Z_{\text{ac}} = -i\rho_0 c_0 \cot\left(\frac{\omega D}{c_0}\right) \quad (3.4)$$

3.3 Methods

3.3.1 Equivalent circuit model of the MC-MPPA

Analogies have been drawn between electronics and acoustics for many years which allow equivalent circuit models to be developed and for analysis and simplification methods to be transferred between the domains. Figure 3.2 shows ECMs of the (1×2) and (1×3) chamber MC-MPPAs. The standard approach to obtain the impedance between the input and output

terminals of these *planar* circuits is to follow a series of steps to simplify the circuit using $\Delta - Y$ transforms, for example.

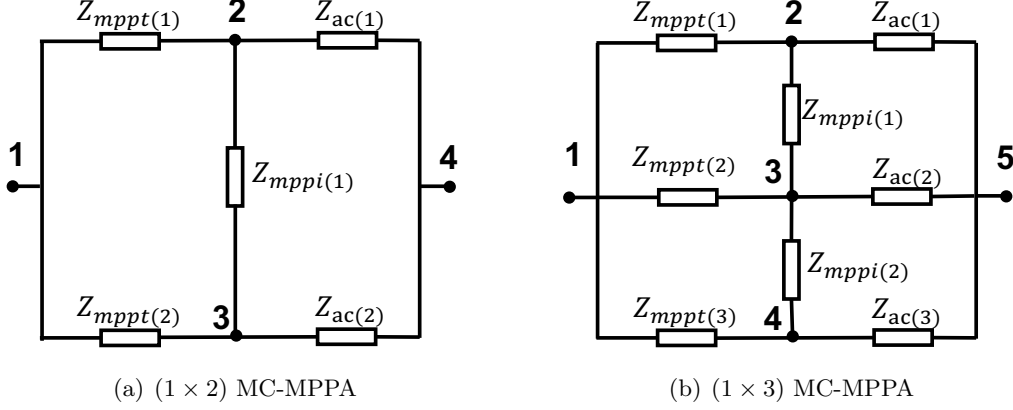


Figure 3.2: MC-MPPA equivalent circuit models.

As an illustration, the (1×2) MC-MPPA as seen in Figure 3.2 (a), can initially be simplified as

$$\begin{aligned}
 Z_A &= \frac{Z_{24}Z_{34}}{Z_{24} + Z_{34} + Z_{23}} \\
 Z_B &= \frac{Z_{24}Z_{23}}{Z_{24} + Z_{34} + Z_{23}} \\
 Z_C &= \frac{Z_{34}Z_{23}}{Z_{24} + Z_{34} + Z_{23}}
 \end{aligned} \tag{3.5}$$

where Z_{ij} in these equations is the acoustic impedance between nodes i and j . For more detail of the steps taken in this and the following simplification processes please refer to the work of Carbajo et al. [69] and McKay et al. [63]. Following this initial step, the acoustic impedance between nodes 1 and 4 in Figure 3.2 (a) can be calculated to be

$$Z_{14} = \frac{(Z_{12} + Z_B)(Z_{13} + Z_C)}{Z_{12} + Z_{13} + Z_B + Z_C} + Z_A \tag{3.6}$$

Given this value, the specific impedance z_{14} is given by

$$z_{14} = Z_{14}S \tag{3.7}$$

where S is the area of normal incidence of the sound wave.

The reflection coefficient r , which is a function of the impedance, is given by the expression

$$r = \frac{z_{14} - \rho_0 c_0}{z_{14} + \rho_0 c_0} \tag{3.8}$$

and from this the normal incidence sound absorption coefficient can be calculated according to

$$\alpha = 1 - |r|^2 \quad (3.9)$$

For the second example in Figure 3.2, the (1×3) MC-MPPA, the acoustic impedance between nodes 1 and 5 can be simplified to be

$$\begin{aligned} Z_A &= \frac{Z_{12}Z_{13}}{Z_{12} + Z_{13} + Z_{23}} \\ Z_B &= \frac{Z_{12}Z_{23}}{Z_{12} + Z_{13} + Z_{23}} \\ Z_C &= \frac{Z_{13}Z_{23}}{Z_{12} + Z_{13} + Z_{23}} \end{aligned} \quad (3.10)$$

$$\begin{aligned} Z_{A1} &= \frac{(Z_B + Z_{25})Z_C}{(Z_B + Z_{25}) + Z_C + Z_{35}} \\ Z_{B1} &= \frac{Z_C Z_{35}}{(Z_B + Z_{25}) + Z_C + Z_{35}} \\ Z_{C1} &= \frac{(Z_B + Z_{25})Z_{35}}{(Z_B + Z_{25}) + Z_C + Z_{35}} \end{aligned} \quad (3.11)$$

$$\begin{aligned} Z_{A2} &= \frac{Z_{C1}Z_{45}}{(Z_{B1} + Z_{34}) + Z_{C1} + Z_{45}} \\ Z_{B2} &= \frac{Z_{C1}(Z_{B1} + Z_{34})}{(Z_B + Z_{25}) + Z_C + Z_{35}} \\ Z_{C2} &= \frac{(Z_{B1} + Z_{34})Z_{45}}{(Z_B + Z_{25}) + Z_C + Z_{35}} \end{aligned} \quad (3.12)$$

In Figure 3.2 (b), the acoustic impedance between nodes 1 and 5 is given by

$$Z_{15} = \left(\frac{(Z_{14} + Z_{C2})(Z_A + Z_{A1} + Z_{B2})}{Z_{14} + Z_{C2} + Z_A + Z_{A1} + Z_{B2}} + Z_{A2} \right) \quad (3.13)$$

and the specific impedance z_{15} is given by Eq. (3.7). Then normal incidence sound absorption coefficient is given by Eqs. (3.8) and (3.9).

3.3.2 Two-point impedance method applied to the MC-MPPA

Figure 3.3 shows the MC-MPPAs from Figure 3.1 represented schematically as bi-pyramid molecular structures where the yellow branches represent the acoustic impedances of the top MPPs, the red branches represent the acoustic impedances of internal MPPs and the green branches represent the acoustic impedances of the air cavities themselves.

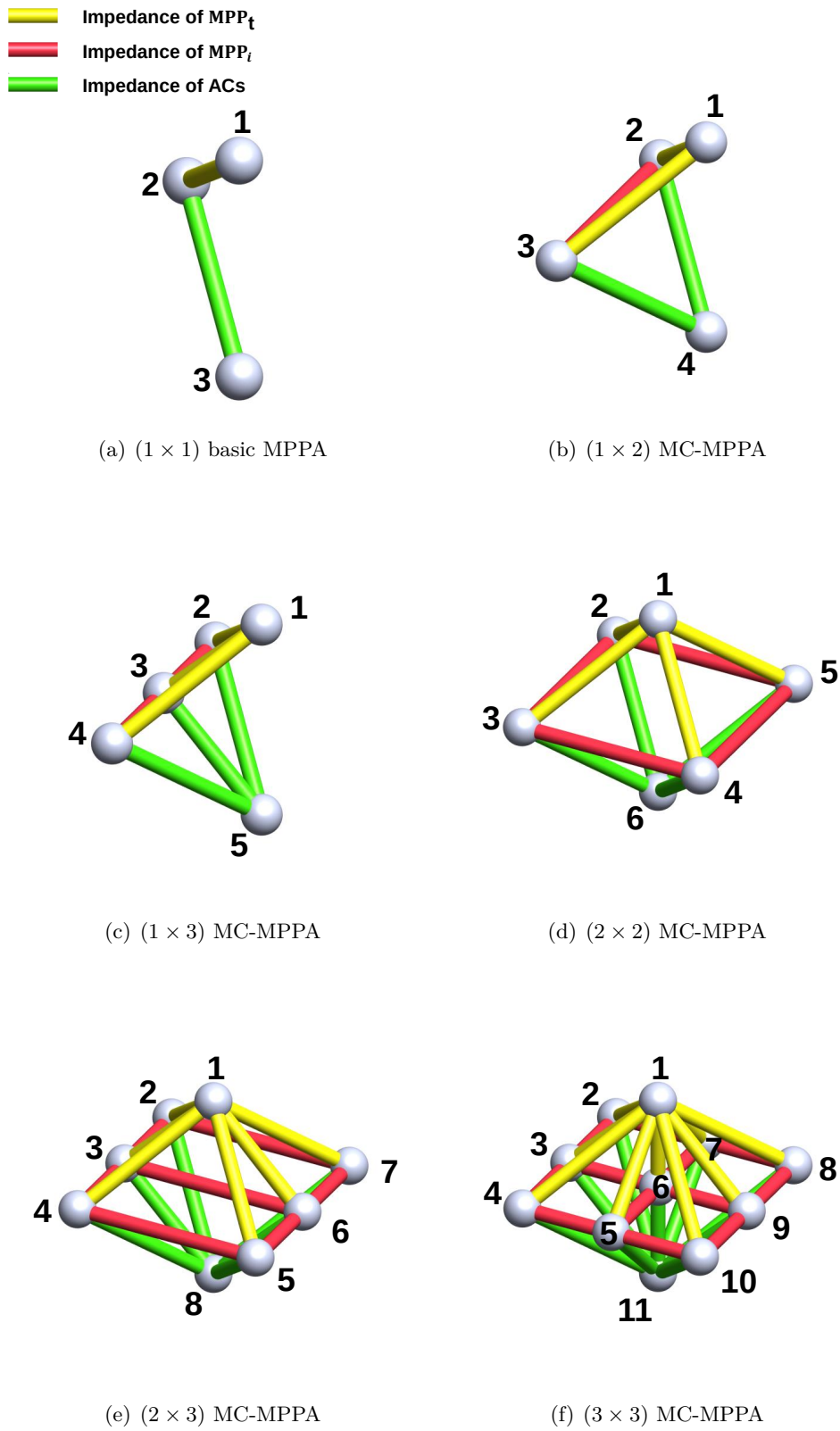


Figure 3.3: MC-MPPA molecular structures.

The two-point impedance method can be used to determine the impedance between any two nodes in a complex impedance network [85, 118] whether the circuit is planar or not. Whilst the ECM methods in Section 3.3.1 are possible to apply to simple circuits, they become unmanageable for more complex non-planar circuits required by the higher order MC-MPPAs, some of which are illustrated in Figures 3.3 (d), (e) and (f).

Initially, the TpIM method was used in electrical engineering to analyse complex circuits. In the paper of Davis et al. [84], it was applied to determine the acoustic impedance between any two nodes of a complex acoustic system. In this study, the two-point impedance method is presented to estimate the acoustic impedance of the multi-chamber MPPA complex acoustic system without the requirement of equivalent circuit simplification. The great advantage of the TpIM is that being matrix based, it makes the acoustic system intuitive. The TpIM is able to accurately estimate the impedance between any two nodes, regardless of the dimensionality of the circuit.

From the given molecular structure, the subsequent Laplace matrix, denoted as $\mathbf{L}$, can be formulated as follows:

$$\mathbf{L} = \begin{pmatrix} y_{1,1} & -y_{1,2} & \cdots & -y_{1,\mathcal{N}} \\ -y_{2,1} & y_{2,2} & \cdots & -y_{2,\mathcal{N}} \\ \vdots & \vdots & \ddots & \vdots \\ -y_{\mathcal{N},1} & -y_{\mathcal{N},2} & \cdots & y_{\mathcal{N},\mathcal{N}} \end{pmatrix} \quad (3.14)$$

where, $\mathcal{N} = (\text{n}_{\text{rows}} \times \text{n}_{\text{cols}}) + 2$ represents the number of nodes in the system. The terms $y_{i,j}$ ($i \neq j$) are the admittances between nodes i and j , with $y_{i,j} = y_{j,i} = 1/Z_{i,j}$. The inverse of the impedance's required to populate this matrix are calculated from the coloured branches in the molecular structures. The yellow branches are the top MPP impedances, the red branches are the internal MPP impedances and the green branches are the cavity impedances. The terms $y_{i,j}$ in the matrix, where there are no branches connecting nodes, i and j , are set to zero. Clearly, therefore, the impedance between many nodes is known, *a priori*, according to the given models. However, the strength of the TpIM is that it allows the impedance between any two nodes, such as between 1 and 11 in the 3×3 MC-MPPA configuration, which is the objective, to be calculated.

The diagonal components of the Laplace matrix $\mathbf{L}$, $y_{i,i}$, describe the total admittance connected to each node in the network, and are expressed by:

$$y_{i,i} \equiv \sum_{j=1}^{\mathcal{N}} y_{i,j} \quad i \neq j \quad (3.15)$$

Employing the Laplace matrix $\mathbf{L}$ enables the construction of the Hermitian matrix $\mathbf{L}^\dagger \mathbf{L}$, capable of being diagonalized, where $\dagger$ symbolizes Hermitian conjugation. In order to calculate the impedance of the network represented by the Laplace matrix the following eigenvalue equation is solved:

$$\mathbf{L}^\dagger \mathbf{L} \psi_\beta = \eta_\beta \psi_\beta, \quad \eta_\beta \geq 0, \quad \beta = 1, 2, \dots, \mathcal{N} \quad (3.16)$$

Here, η_β and ψ_β represent the eigenvalues and eigenvectors of $\mathbf{L}^\dagger \mathbf{L}$, respectively. Making use of the following,

$$\lambda_\beta = \psi_\beta^\dagger \mathbf{L} \psi_\beta \quad (3.17)$$

the acoustic impedance between any two nodes, represented as i and j , within the network can be determined by the equation:

$$Z_{i,j} = \sum_{\beta=2}^{\mathcal{N}} \frac{1}{\lambda_\beta} (\psi_\beta(i) - \psi_\beta(j))^2 \quad (3.18)$$

and thus, the specific impedance, $z_{i,j}$, can be calculated from

$$z_{i,j} = Z_{i,j} S \quad (3.19)$$

where, S is the projected area normal to the incident sound wave. The specific impedance of the (3×3) MC-MPPA, denoted as $z_{1,11}$, refers to the specific impedance between nodes 1 and 11. Given this value, the normal incidence sound absorption coefficient, can be calculated from Eqs. (3.8) and (3.9).

3.3.3 Finite element model

Initially, the ECM method will be used to validate the TpIM but this can only be done for some simple circuits. In addition, a finite element analysis using COMSOL Multiphysics 6.0. was conducted in order to validate both the TpIM and ECM results for simple circuits and the TpIM for more complex circuits. Both of these latter methods, once developed, are quick to execute which is to their benefit but they both depend on a lumped element assumption. As the objective of this research is to develop noise-absorbing technologies which are thin

and for them to be efficient at low frequencies, this assumption should not be violated but it is still necessary to verify their accuracy. The complex and non-planar (2×3) MC-MPPA, as shown in Figure 3.4, illustrates one of the test-set ups and the boundary conditions. As is standard, an impedance tube style set-up was implemented numerically with a normally incident plane wave pressure field being generated upstream of the test section. The pressure acoustics module was implemented to model the air domain both upstream and in the air cavities. For the MPPs, COMSOL’s interior perforated thin panel (IPP) boundary condition, Eq. (3.1), was implemented. In previous work by McKay et al. [63] for a similar set-up, it was demonstrated that the IPP boundary condition provides results as accurate as a fully resolved model of the holes themselves with much less computational cost. The details of the global parameters used in the simulations are listed in Table 3.2.

Table 3.2: Details of FEM parameters settings.

Name	Value
Temperature	293.15 K
Air density	$1.204 \text{ kg} \cdot \text{m}^{-3}$
Speed of sound	$343.1 \text{ m} \cdot \text{s}^{-1}$
Air dynamic viscosity	$1.825 \times 10^{-5} \text{ kg} \cdot (\text{m} \cdot \text{s})^{-1}$
Tube length	0.2 m

3.3.4 Experimental setup

A rectangular impedance tube, as shown in Figure B.1, with internal cross-sectional dimensions of $80 \text{ mm} \times 50 \text{ mm}$ was used to experimentally measure the normal acoustic incidence sound absorption coefficients of the MC-MPPAs. More details about the Impedance Tube Measurements can be found in Appendix B.1.

The MC-MPPAs were 3D printed. Technologies such as these are difficult and expensive to manufacture using traditional fabrication techniques and it is due to the advances in 3D printing methods that such great strides have been possible in acoustic metamaterials development [119–122]. The additive manufacturing 3D printer used in this Chapter was an Elegoo Saturn S masked stereolithography (mSLA) printer. Stereolithography is a form of additive manufacturing often used for fast prototyping, particularly when the prototype involves small features, and thus, is very suitable for the small micro-perforations of the MC-MPPA. It uses a liquid resin photopolymer that can be cured when exposed to UV light. The cured resin forms very thin solid layers which stack to form the object. Stereolithography is

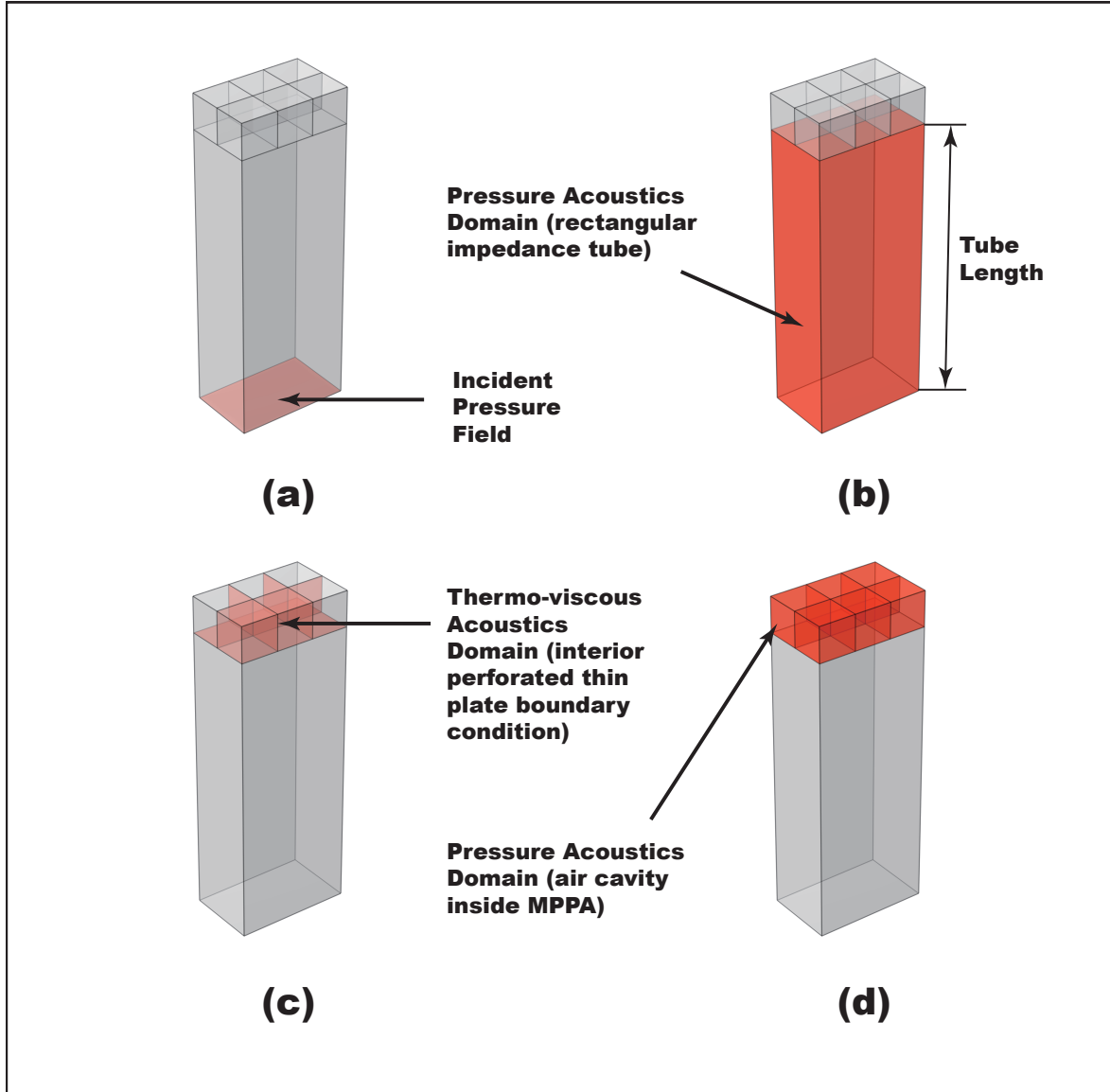
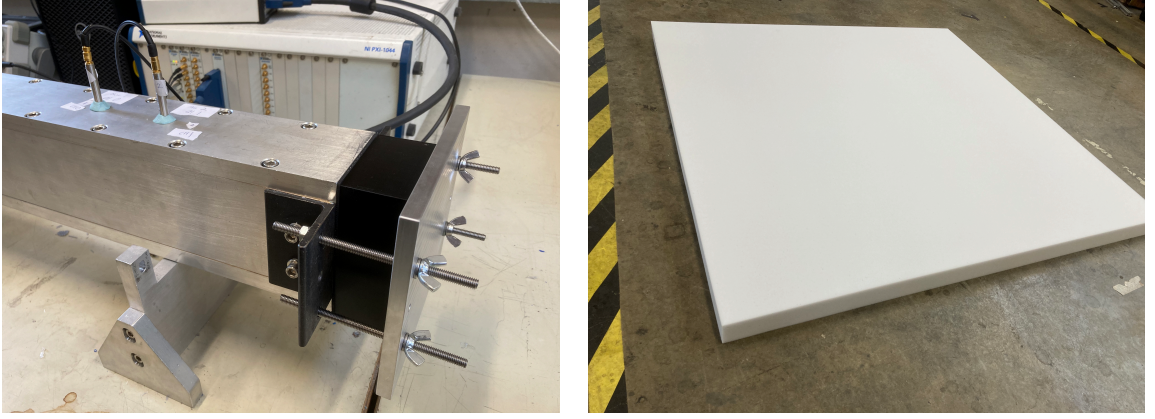


Figure 3.4: COMSOL Multiphysics FEM model of (2×3) MC-MPPA.

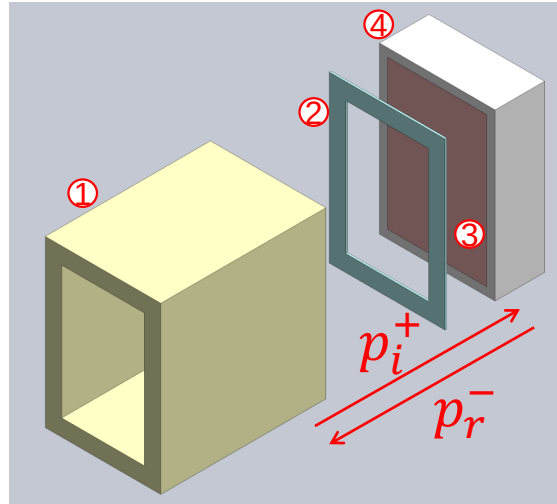
often used in prototyping as it is able to achieve high levels of precision and accuracy. The Elegoo Saturn S has a 9.1" LCD screen with a $48 \mu\text{m}$ xy resolution. It can print at layer heights from 0.01 mm up to 0.15 mm, and has a z-axis resolution of $1.25 \mu\text{m}$.

In order to validate the performance experimentally of the MC-MPPA, it was compared to a single-layer of Basotect G+ (BASF, Germany) of the same dimensions. Basotect G+ is a melamine resin foam with a high sound absorption capacity and is an industry-standard [123]. The same impedance tube set-up was used to measure its absorption coefficient as a function of its thickness. Figure 3.5 shows the test set-up. The Basotect G+ block, manually cut to the required thickness, was positioned within a solid holder backed by a thick aluminum

plate. A rubber gasket was installed between the holder and the aluminum impedance tube to prevent any air leakage.



(a) Black sample holder with solid aluminium backing. (b) Sample sheet of Basotect G+ melamine resin foam.



(c) 1: Impedance tube, 2: Rubber Gasket, 3: Block of Basotect G+, 4: Sample Holder.

Figure 3.5: Experimental impedance tube test set-up for benchmarking against Basotect G+ melamine resin foam.

3.4 Initial Validation of the TpIM

In order to test the validity of the proposed TpIM approach, the sound absorption of the (1×2) and (1×3) MC-MPPAs from 10 Hz to 3000 Hz was examined using the three different methods (ECM, TpIM and FEM) described in Section 3.3. The geometrical parameters of the perforated panels studied can be found in Table 3.3. In addition, a comparison was also

made with the results of Carbajo et al. [69] from the literature. WebPlotDigitizer [124] was used to obtain the data from the Carbajo reference.

Figure 3.6 (a) shows the results for the (1×2) MC-MPPA. The width and length were set to be 10mm and the depth to be 20 mm in order to agree with the dimensions of the Carbajo et al. [69] set-up. Figure 3.6 (b) shows the results for a (1×3) MC-MPPA. In order to investigate the performance with larger dimensions, the width, length and depth were chosen to be 40 mm, 40 mm and 25 mm respectively.

It can be observed that from both figures that the TpIM method produces the same results as the Kirchhoff law-based ECM method, showing that this method can be used as an alternative approach complete with its advantages. It can also be observed that in both sets of figures that both models are validated by the finite element method of COMSOL. Finally, the results reproduced in this Chapter for the (1×2) MC-MPPA agree perfectly with those published in the literature by Carbajo et al. [69].

Comparing the two figures from the point of view of performance, It can be observed that both provide a high absorption coefficient close to unity with the frequency range of the deeper MC-MPPA decreasing, as would be expected, but with a narrower bandwidth.

Table 3.3: Geometrical parameters of (1×2) and (1×3) MC-MPPAs as used in Section 3.4.

Type	Element	τ (mm)	d (mm)	ϕ (%)
1×2 chambers	$MPP_{t(1)}$	2	1	1
	$MPP_{t(2)}$	2	2	4
	$MPP_{i(1)}$	1	0.1	1
1×3 chambers	$MPP_{t(1)}$	2	1	1
	$MPP_{t(2)}$	2	1	1
	$MPP_{t(3)}$	2	1	1
	$MPP_{i(1)}$	1	0.1	0.8
	$MPP_{i(2)}$	1	0.1	0.8

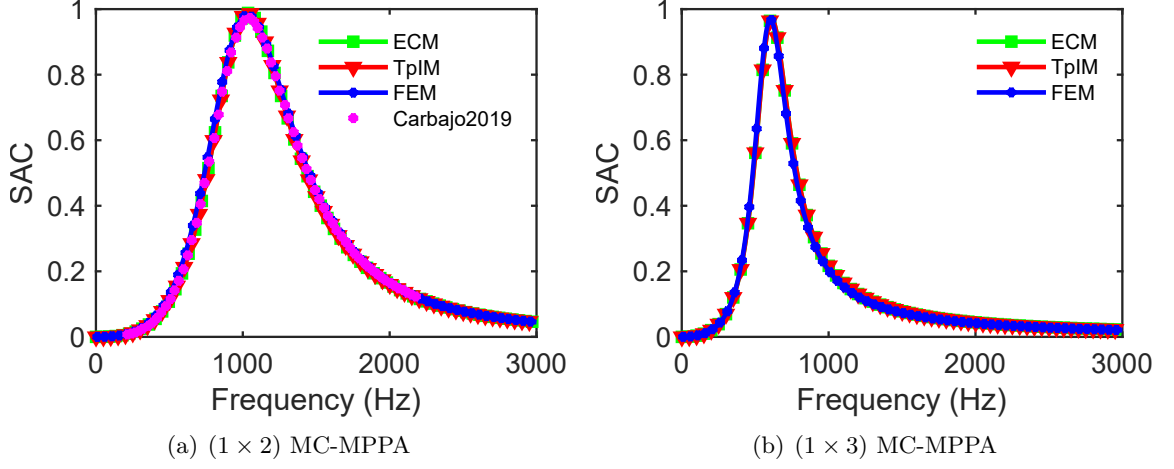


Figure 3.6: Comparison of sound absorption coefficients.

3.5 Optimisation

The absorption coefficients of the MC-MPPAs in Figure 3.6 are seen to be high for these relatively arbitrarily chosen parameters but the frequency responses are quite narrow, and typically become more so with decreasing frequency range. One of the main objectives of applying the TpIM to the MPPA technology is to provide an theoretical model and with that model optimise the absorbers so that they are as efficient as they can be for a fixed set of parameters.

In the work of McKay et al. [63], the performance of the membrane-based acoustic absorber, SeMSA, was quantified by the linear averaging of the absorption coefficient spectrum in a defined frequency range to give an averaged absorption coefficient. However, this approach is really only suitable when the target noise spectrum to be absorbed is white. In this Chapter, and in the work of Davis et al. [84], arbitrary target spectra, for example, those produced by industrial fans or aeroengines, which might contain a mixture of tones and broadband noise, are also of interest.

Considering a target sound spectrum described by an auto-spectral density function $G_{xx}(\omega)$, the sound absorption that a sound absorber with an absorption spectrum $\alpha(\omega)$ would achieve in Decibels can be written as:

$$C_\alpha = 10 \log_{10}(1 - c_\alpha) \quad (3.20)$$

where c_α is the average absorption coefficient weighted by the sound spectrum of the targeted noise source:

$$c_\alpha = \frac{(\sum_{\omega_1}^{\omega_2} \alpha(\omega) G_{xx}(\omega))}{(\sum_{\omega_1}^{\omega_2} G_{xx}(\omega))} \quad (3.21)$$

and where $\omega_1 = 2\pi f_1$ and $\omega_2 = 2\pi f_2$, with f_1 and f_2 being the lower and upper limits of the frequency range of interest. The absorption coefficient, $\alpha(\omega)$, is calculated from Eq. (3.9) using the TpIM.

In the current study, to examine the potential optimal performance of the MC-MPPA, two target sound sources, $G_{xx}(\omega)$, are defined. Figure 3.7 shows their sound spectra. For white noise, $G_{xx}(\omega) \equiv 1$, whereas the fan noise signal used here, simulated by Matlab, contains a fundamental tone plus two harmonics: with peaks at 600 Hz, 1200 Hz and 1800 Hz respectively.

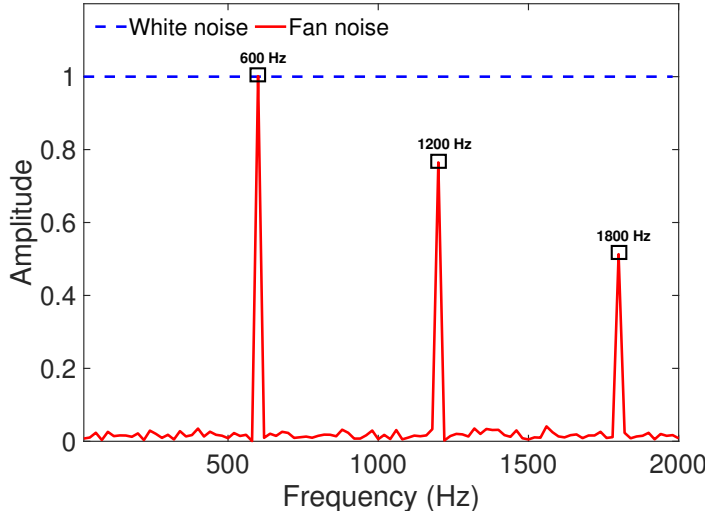


Figure 3.7: Targeted fan noise spectra.

The sound absorption factor, C_α , provides a suitable cost function that can be minimised in an optimisation routine in order to generate the parameter set for an optimal MC-MPPA. C_α can be expressed as a function of the geometric parameter set chosen for the optimisation algorithm.

An optimisation method based on Sequential Quadratic Programming (SQP) was used to minimise the cost function. SQP is a classical iterative method for constrained nonlinear optimisation which applies to optimisation issues where the objective function and the con-

straints are twice continuously divisible. For more detail about SQP, readers are referred to Refs. [125] and [126].

Here, Matlab's built-in fmincon SQP algorithm is used to complete the optimisation:

$$\begin{aligned} & \text{Minimize } C_\alpha \text{ in Eq.(3.20)} \\ & \text{subject to } \begin{cases} d_t(i) \in [d_{\min} \ d_{\max}], \quad i = 1, 2, \dots, n_{\text{rows}}n_{\text{cols}} \\ d_i(j) \in [d_{\min} \ d_{\max}], \quad j = 1, 2, \dots, (n_{\text{rows}} - 1)n_{\text{cols}} + (n_{\text{cols}} - 1)n_{\text{rows}} \\ \phi_t(i) \in [\phi_{\min} \ \phi_{\max}], \quad i = 1, 2, \dots, n_{\text{rows}}n_{\text{cols}} \\ \phi_i(j) \in [\phi_{\min} \ \phi_{\max}], \quad j = 1, 2, \dots, (n_{\text{rows}} - 1)n_{\text{cols}} + (n_{\text{cols}} - 1)n_{\text{rows}} \end{cases} \end{aligned} \quad (3.22)$$

Algorithm (3.22) does not apply to the basic (1×1) chamber MPPA which has no internal perforated sidewalls and so is treated more simply as:

$$\begin{aligned} & \text{Minimize } C_\alpha \text{ in Eq.(3.20)} \\ & \text{subject to } \begin{cases} d_t(1) \in [d_{\min} \ d_{\max}] \\ \phi_t(1) \in [\phi_{\min} \ \phi_{\max}] \end{cases} \end{aligned} \quad (3.23)$$

The parameter settings for the optimisation are listed in Table 3.4.

Table 3.4: Parameter settings for optimisation. Two ranges of hole diameters were examined.

Parameters	Value
$[d_{\min} \ d_{\max}]$	$[0.05 - 0.3] \text{ mm}, [0.3 - 1] \text{ mm}$
$[\phi_{\min} \ \phi_{\max}]$	$[0 - 20]\%$
$[D_{\min} \ D_{\max}]$	$[10 - 40] \text{ mm}$
$[f_1 \ f_2]$	$[20 - 2000] \text{ Hz}$
$[L, W]$	$[50 \text{ mm}, 40 \text{ mm}]$
$[\tau_t, \tau_i]$	$[2 \text{ mm}, 1 \text{ mm}]$

3.6 Results

In order to evaluate the performance of the MC-MPPA four test cases were examined:

- Case 1. Target spectrum: White noise; Perforation diameter range: $[0.05-0.3] \text{ mm}$.
- Case 2. Target spectrum: White noise; Perforation diameter range: $[0.3-1.0] \text{ mm}$.
- Case 3. Target spectrum: Fan noise; Perforation diameter range: $[0.05-0.3] \text{ mm}$.

- Case 4. Target spectrum: Fan noise; Perforation diameter range: [0.3-1.0] mm.

As has been discussed already, the aim is to assess how well the multi-chamber MPPA will absorb sound, and also how well the optimisation algorithm will perform, depending on the type of noise source present. Therefore, both white noise and tonal noise have been examined.

Further, a 0.3 mm perforation is approximately the smallest hole than can be reliably 3D printed in large numbers by a desktop 3D printer currently and is also about the size of the smallest drill bit that good workshops would be able to use in a CNC milling machine with thin panels. Therefore, two hole size ranges have been examined, each of which corresponds to the hole size ranges used in the optimisation routine. The [0.3 - 1] mm range is a manageable sub-millimetre range whether using 3D printing or traditional fabrication methods, whereas the [0.05 - 0.3] mm range examines towards the micron level with 0.05 mm (50 micron) being a common lower level examined in numerical studies of MPPAs to be found in the literature.

The absorption coefficient as a function of frequency, where the frequency range is from 20 Hz to 2 kHz, is plotted for each of the four cases in Figures 3.8 - 3.11. To evaluate the impact of increasing the number of chambers in one and two dimensions and to compare them to the standard single chamber resonator, each of the six chamber configurations, as illustrated in Figure 3.1, are examined.

Figures 3.12 - 3.15 plot the average absorption coefficients of Cases 1 - 4 and Basotect G+ in the frequency range from 200 Hz to 2000 Hz, weighted by the sound spectrum of the target source from Eq. (3.21). As previously discussed, the optimised theoretical frequency range chosen for the MC-MPPAs is [20 2000] Hz. However, in Figures 3.12 - 3.15 the [200 2000] Hz range is chosen to allow for comparison with the experimental results of the Basotech G+ benchmarking porous material. It should also be noted that the thickness of the air cavity of the MC-MPPAs is increased from 10 mm to 40 mm in 5 mm increments, and as the thickness of the MC-MPPAs' top MPP is 2 mm, the overall thickness of the MC-MPPAs is actually increased from 12 mm to 42 mm in 5 mm increments.

The specific values of c_α in the [200 2000] Hz frequency range, corresponding to Figures 3.12 - 3.15 can be found in Tables 3.5 - 3.8.

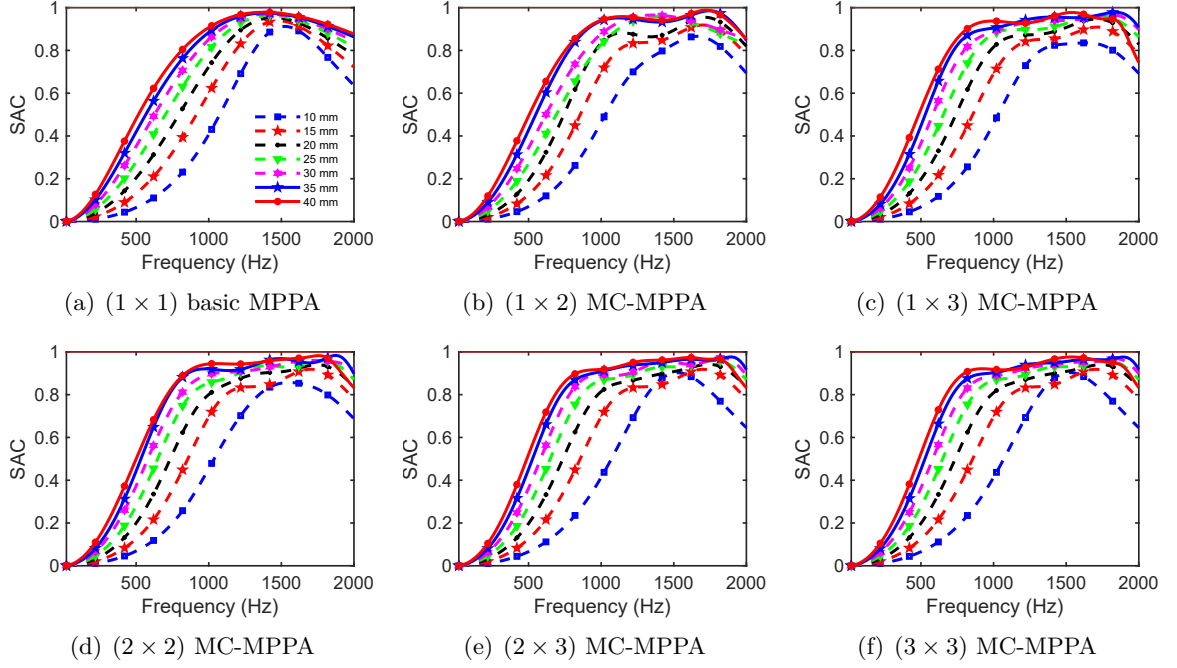


Figure 3.8: Sound absorption coefficients for Case 1, air cavity depth range: $[10 \ 40]$ mm, frequency range: $[20 \ 2000]$ Hz, target spectrum: white noise and perforation diameter range: $[0.05 \ 0.3]$ mm, $L = 50$ mm, $W = 40$ mm.

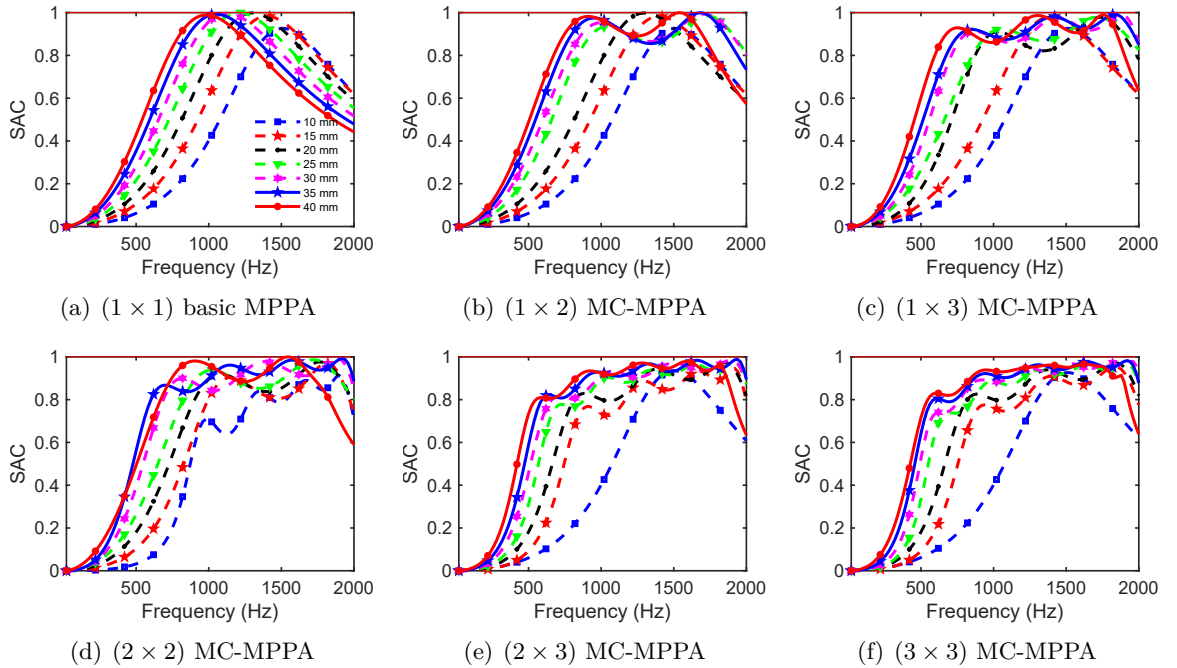


Figure 3.9: Sound absorption coefficients for Case 2, air cavity depth range: $[10 \ 40]$ mm, frequency range: $[20 \ 2000]$ Hz, target spectrum: white noise and perforation diameter range: $[0.3 \ 1.0]$ mm, $L = 50$ mm, $W = 40$ mm.

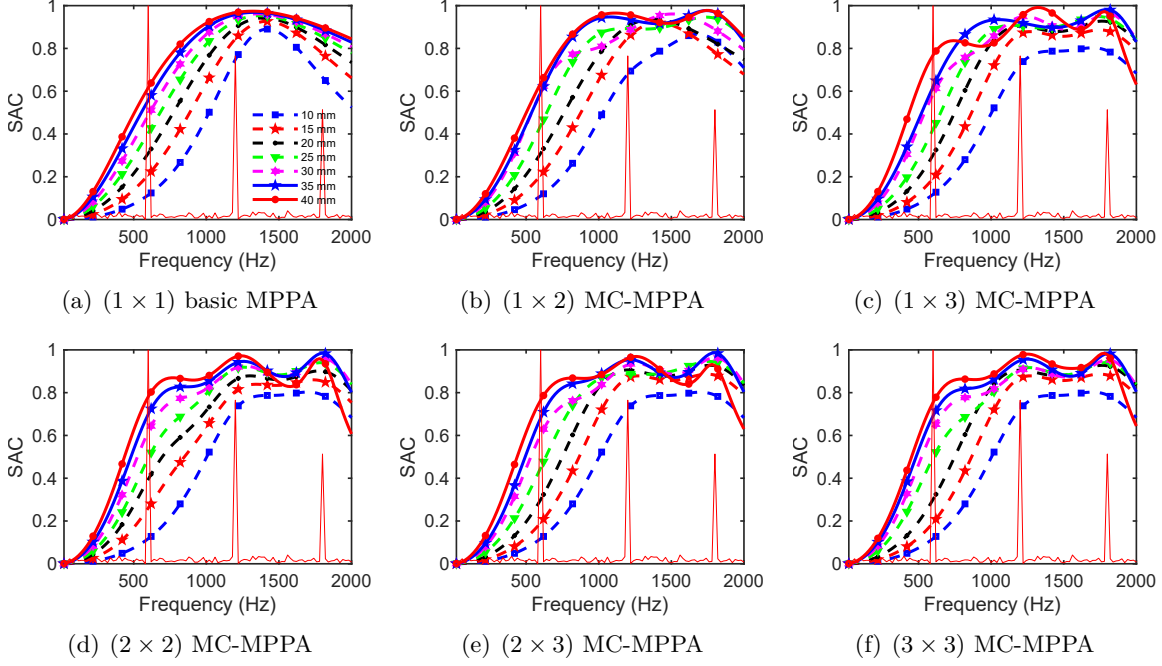


Figure 3.10: Sound absorption coefficients for Case 3, air cavity depth range: $[10 \ 40]$ mm, frequency range: $[20 \ 2000]$ Hz, target spectrum: fan noise and perforation diameter range: $[0.05 \ 0.3]$ mm, $L = 50$ mm, $W = 40$ mm.

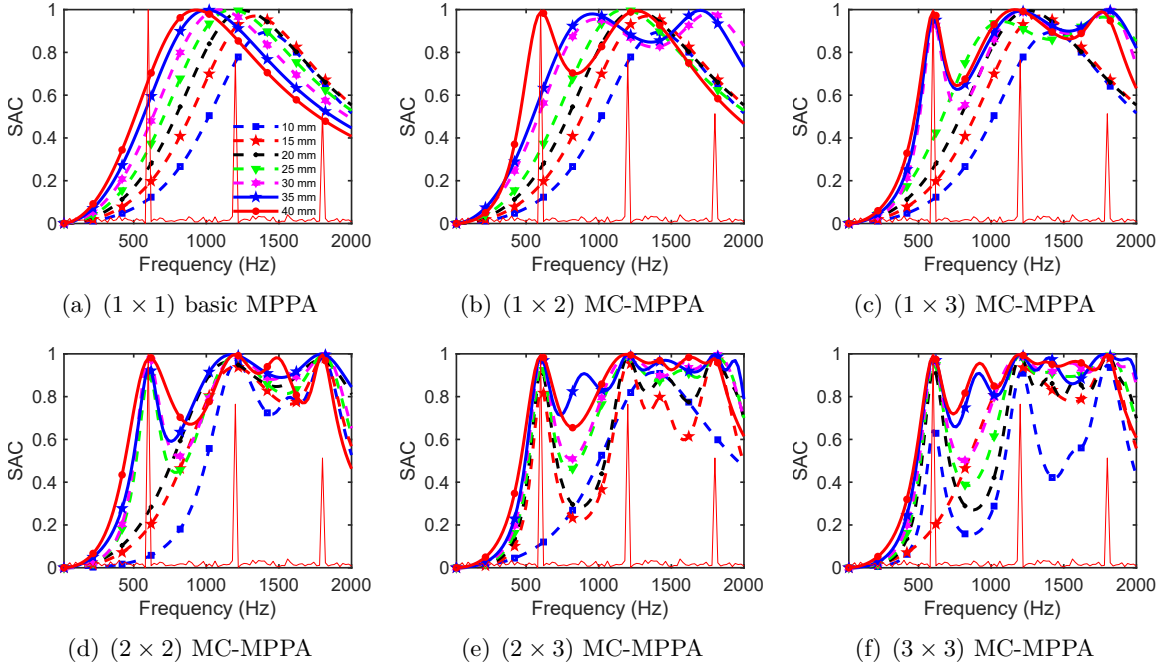


Figure 3.11: Sound absorption coefficients for Case 4, air cavity depth range: $[10 \ 40]$ mm, frequency range: $[20 \ 2000]$ Hz, target spectrum: fan noise and perforation diameter range: $[0.3 \ 1]$ mm, $L = 50$ mm, $W = 40$ mm.

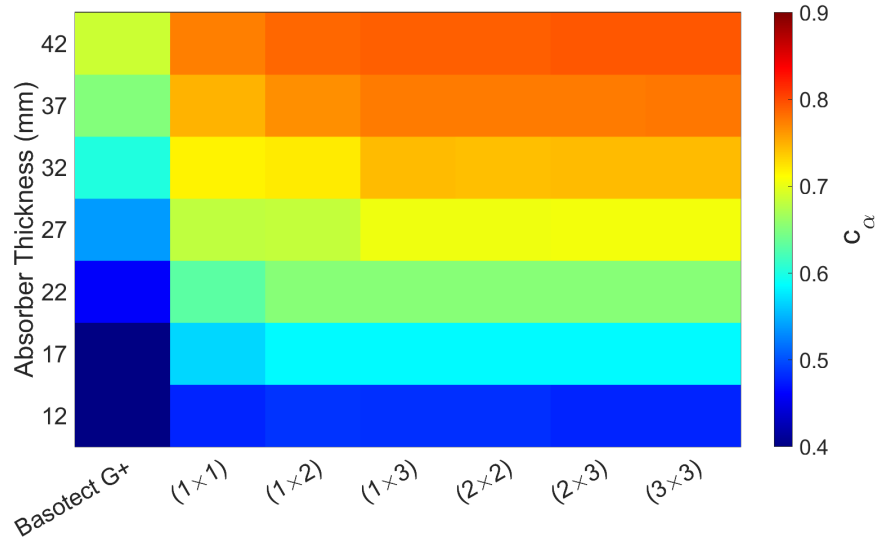


Figure 3.12: Colourmap for Case 1 and Basotect G+ of average sound absorption coefficient c_α , overall thickness [12 42] mm, frequency range: [200 2000] Hz. Numerical values can be found in Table 3.5

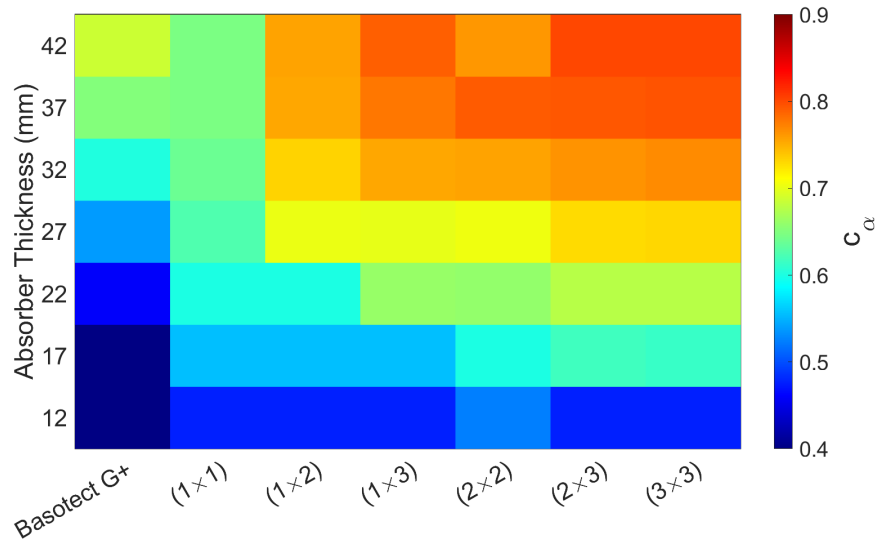


Figure 3.13: Colourmap for Case 2 and Basotect G+ of average sound absorption coefficient c_α , overall thickness [12 42] mm, frequency range: [200 2000] Hz. Numerical values can be found in Table 3.6

3.6.1 Numerical validation

A second series of numerical simulations were carried out to prove the validity of the two-point impedance method. After having shown that the TpIM approach can provide equivalent results to the ECM method, now the *optimised* TpIM results are examined and also for the

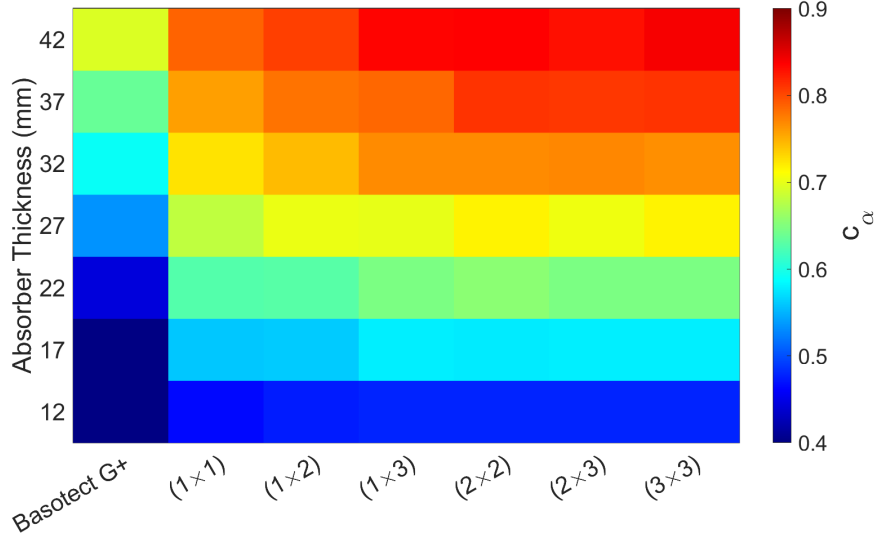


Figure 3.14: Colourmap for Case 3 and Basotect G+ of average sound absorption coefficient c_α , overall thickness [12 42] mm, frequency range: [200 2000] Hz. Numerical values can be found in Table 3.7

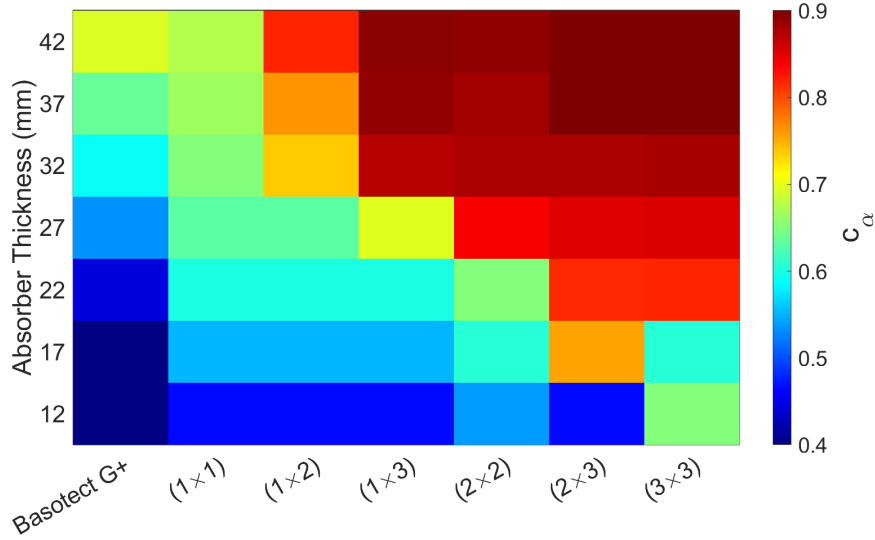


Figure 3.15: Colourmap for Case 4 and Basotect G+ of average sound absorption coefficient c_α , overall thickness [12 42] mm, frequency range: [200 2000] Hz. Numerical values can be found in Table 3.8

full test set of chambers from Figure 3.1. Case 1 is chosen for the numerical validation, where white noise is chosen as the target spectrum and a very shallow depth of only 25 mm is examined to test the efficacy of the technology. Results are found in Figures 3.16.

Table 3.5: Average sound absorption coefficient c_α for Case 1, absorber thickness: [12 42] mm, air-back cavity depth: [10 40] mm, frequency range: [200 2000] Hz (For the reader's comparison, the maximum average absorption coefficient is bolded for each air back cavity depth).

Absorber Thickness (mm)	(1×1)	(1×2)	(1×3)	(2×2)	(2×3)	(3×3)	Basotect G+
12	0.4782	0.4862	0.4856	0.4850	0.4786	0.4787	0.2992
17	0.5675	0.5846	0.5845	0.5847	0.5845	0.5845	0.4003
22	0.6313	0.6523	0.6526	0.6531	0.6533	0.6533	0.4605
27	0.6797	0.6829	0.7032	0.7046	0.7047	0.7048	0.5371
32	0.7178	0.7212	0.7441	0.7420	0.7443	0.7448	0.6023
37	0.7485	0.7664	0.7760	0.7751	0.7768	0.7770	0.6515
42	0.7736	0.7858	0.7890	0.7904	0.7936	0.7936	0.6869

Table 3.6: Average sound absorption coefficient c_α in Case 2, absorber thickness: [12 42] mm, air-back cavity depth: [10 40] mm, frequency range: [200 2000] Hz (For the reader's comparison, the maximum average absorption coefficient is bolded for each air back cavity depth).

Absorber Thickness (mm)	(1×1)	(1×2)	(1×3)	(2×2)	(2×3)	(3×3)	Basotect G+
12	0.4778	0.4778	0.4778	0.5242	0.4781	0.4778	0.2992
17	0.5558	0.5558	0.5558	0.5995	0.6168	0.6142	0.4003
22	0.6000	0.6000	0.6610	0.6590	0.6770	0.6772	0.4605
27	0.6254	0.7008	0.7004	0.7036	0.7291	0.7311	0.5371
32	0.6395	0.7340	0.7538	0.7555	0.7648	0.7690	0.6023
37	0.6462	0.7539	0.7787	0.7916	0.7929	0.7957	0.6515
42	0.6476	0.7555	0.7896	0.7628	0.8014	0.8022	0.6869

3.6.2 Experimental validation

Two (3×3) MC-MPPA test samples, named Test sample 50 and Test sample 20 respectively, were fabricated using an mSLA 3D-printing process using 405 nm Photopolymer Resin (colour: grey):

- (1) **Test sample 50** (as shown in Figure 3.17). Target spectrum: White noise; Perforation diameter range: [0.5-1.0] mm; Perforation porosity range: [0-5] %; Working frequency range: [200 1000] Hz and air cavity depth: 50 mm.
- (2) **Test sample 20** (as shown in Figure 3.18). Target spectrum: White noise; Perforation diameter range: [0.5-1.0] mm; Perforation porosity range: [0-5] %; Working frequency range: [200 2000] Hz and air cavity depth: 20 mm.

CHAPTER 3. MC-MPPA OPTIMISED FOR BROADBAND ABSORPTION VIA THE TPIM

Table 3.7: Average sound absorption coefficient c_α in Case 3, absorber thickness: [12 42] mm, air-back cavity depth: [10 40] mm, frequency range: [200 2000] Hz (For the reader's comparison, the maximum average absorption coefficient is bolded for each air back cavity depth).

Absorber Thickness (mm)	(1 × 1)	(1 × 2)	(1 × 3)	(2 × 2)	(2 × 3)	(3 × 3)	Basotect G+
12	0.4646	0.4750	0.4788	0.4788	0.4787	0.4788	0.2857
17	0.5591	0.5611	0.5786	0.5771	0.5785	0.5786	0.3893
22	0.6277	0.6301	0.6464	0.6551	0.6463	0.6464	0.4431
27	0.6812	0.7016	0.7000	0.7182	0.7032	0.7182	0.5335
32	0.7242	0.7448	0.7687	0.7688	0.7694	0.7669	0.5898
37	0.7592	0.7806	0.7862	0.8106	0.8095	0.8110	0.6365
42	0.7878	0.8045	0.8338	0.8374	0.8280	0.8401	0.6942

Table 3.8: Average sound absorption coefficient c_α in Case 4, absorber thickness: [12 42] mm, air-back cavity depth: [10 40] mm, frequency range: [200 2000] Hz (For the reader's comparison, the maximum average absorption coefficient is bolded for each air back cavity depth).

Absorber Thickness (mm)	(1 × 1)	(1 × 2)	(1 × 3)	(2 × 2)	(2 × 3)	(3 × 3)	Basotect G+
12	0.4646	0.4646	0.4646	0.5374	0.4660	0.6510	0.2857
17	0.5511	0.5511	0.5511	0.6054	0.7561	0.6055	0.3893
22	0.6008	0.6008	0.6008	0.6517	0.8169	0.8192	0.4431
27	0.6307	0.6307	0.6966	0.8405	0.8530	0.8538	0.5335
32	0.6501	0.7374	0.8708	0.8766	0.8778	0.8793	0.5898
37	0.6640	0.7643	0.8900	0.8807	0.9020	0.8991	0.6365
42	0.6741	0.8199	0.8930	0.8907	0.8986	0.9056	0.6942

Figures 3.17 and 3.18 show CAD images and photos of the two experimental samples. Two depths were chosen for these samples. 50 mm (2") is a typical depth for an aeroengine acoustic liner in industry and 20 mm was chosen to test the abilities of this technology at a very small depth. The porosity range was decreased in order to reduce the optimum number of holes to reasonable quantities and, as with the Basotect analysis, the samples were optimised for frequencies above 200 Hz due to low-frequency limitations in the test rig. The full set of geometric parameters of these test samples can be found in Tables E.1 and E.2.

The Dino-Lite Premier AD7013MT digital desk-top microscope was used to assess the printing accuracy of the 3D-printed holes. Figure 3.19 shows images taken by a microscope of four randomly selected 0.5 mm-diameter holes from test samples. Figure 3.20 shows images taken by a microscope of four randomly selected 0.9mm-diameter holes from test samples. For these samples, the averages for the diameters are 0.43 mm and 0.22 mm both of which are

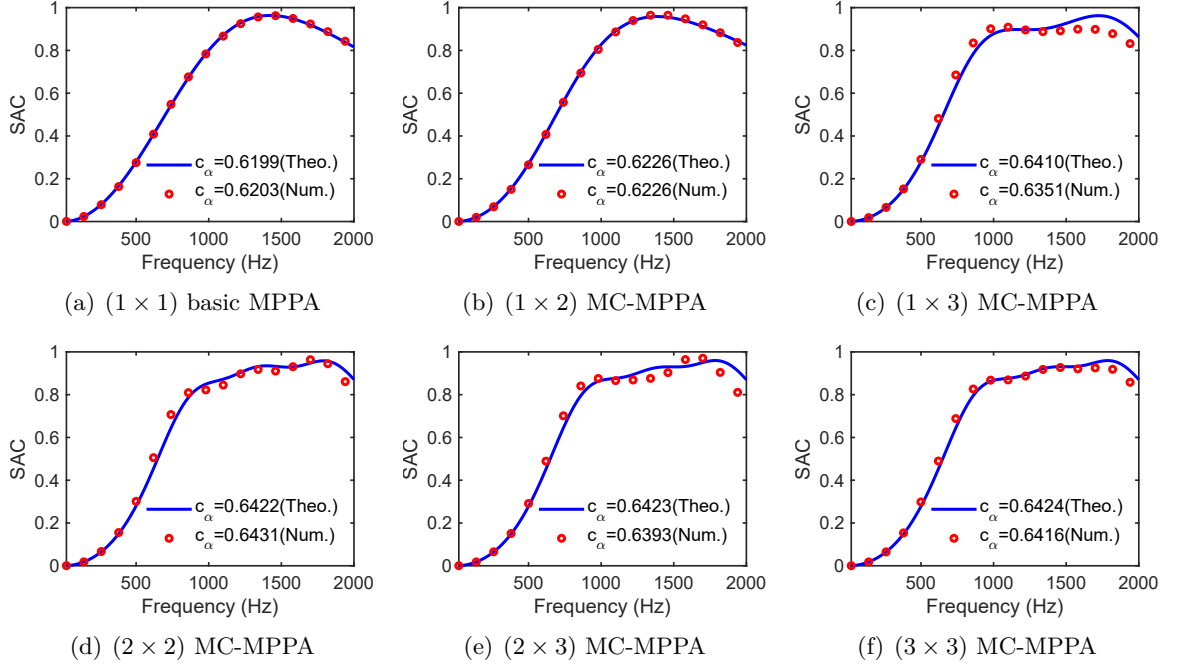
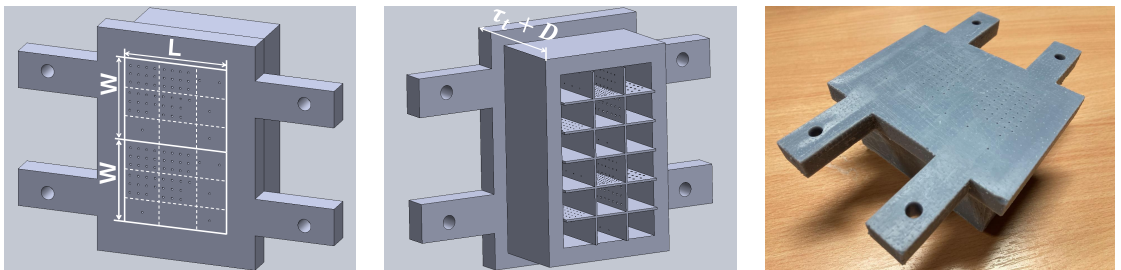


Figure 3.16: Theoretical (TpIM) and numerical sound absorption coefficients. $D = 25$ mm.

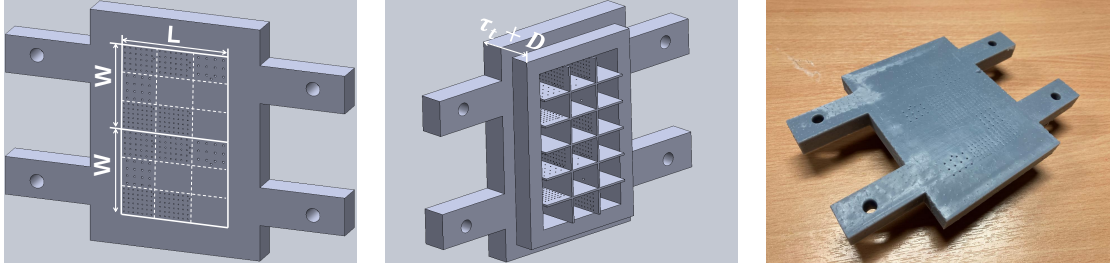
less than as designed. Kennedy et al. [81] discussed how deviations in surface roughness and dimensional fidelity can have a comparable impact on the experimentally measured values of the absorption coefficient. In further tests, it may be possible to calibrate the actual hole diameter printed to that designed.

The experimentally measured sound absorption curves for Test sample 50 and Test sample 20, and for the identically sized Basotect G+, are shown in Figures 3.21 and 3.22. In addition, the theoretical, numerical, and experimental values of c_α for Test Sample 50 and Test Sample 20 across different frequency ranges are listed in Tables 3.9 and 3.10.



(a) Top view of the MC-MPPA (b) View of the internal perforated (c) Photograph of MC-MPPA with showing the perforations in the panels. Hole number and poros- rig attachment flanges. MPPT. This is the side exposed ity can vary from panel to panel. to the acoustic plane waves in the A solid backing is attached to this impedance tube. side.

Figure 3.17: CAD and photo images of Test Sample 50 MC-MPPA.



(a) Top view of the MC-MPPA showing the perforations in the panels. (b) View of the internal perforated MPPT. (c) Photograph of MC-MPPA with poros-rig attachment flanges. This is the side exposed to the acoustic plane waves in the impedance tube. The solid backing is attached to this side.

Figure 3.18: CAD and photo images of Test Sample 20 MC-MPPA.

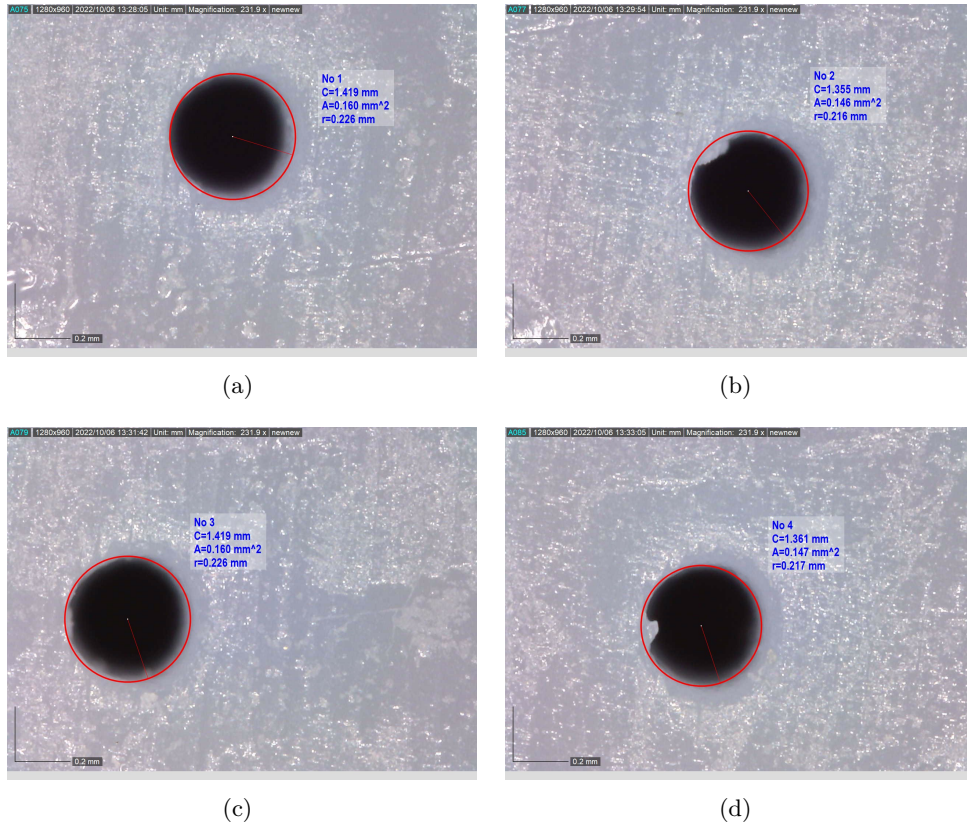


Figure 3.19: Microscope images of four randomly chosen 0.5 mm-diameter 3D-printed holes from test samples, where r represents the radius of the hole, A represents the area of the hole, and C represents the perimeter of the hole.

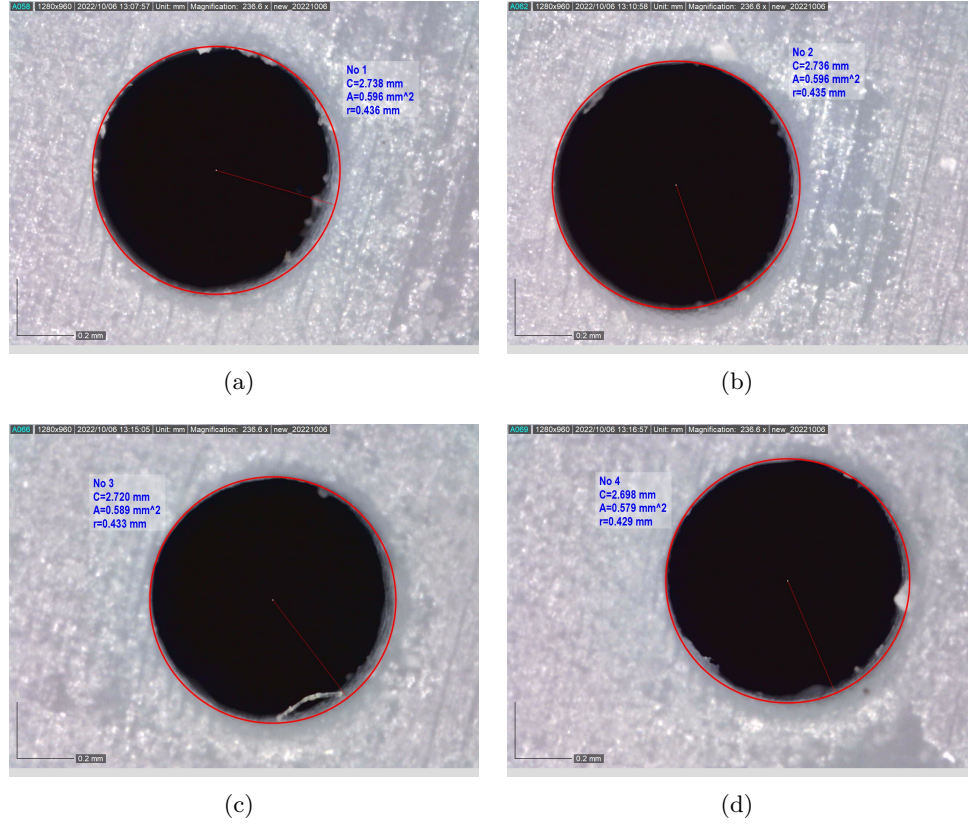


Figure 3.20: Microscope images of four randomly chosen 0.9 mm-diameter 3D-printed holes from test samples, where r represents the radius of the hole, A represents the area of the hole, and C represents the perimeter of the hole.

Table 3.9: Theoretical, numerical and experimental c_α of the Test sample 50 in different frequency ranges (data correspond to the SACs curves in Figure 3.21).

	c_α in [200 1000] Hz	c_α in [281 1000] Hz
Theo.	0.8071	0.8692
Num.	0.7843	0.8444
Exp.	0.7720	0.8225
Basotect G+	0.6177	0.6496

Table 3.10: Theoretical, numerical and experimental c_α of the Test sample 20 in different frequency ranges (data correspond to the SACs curves in Figure 3.22).

	c_α in [200 2000] Hz	c_α in [660 2000] Hz
Theo.	0.6746	0.8586
Num.	0.6686	0.8428
Exp.	0.6703	0.8273
Basotect G+	0.4605	0.5470

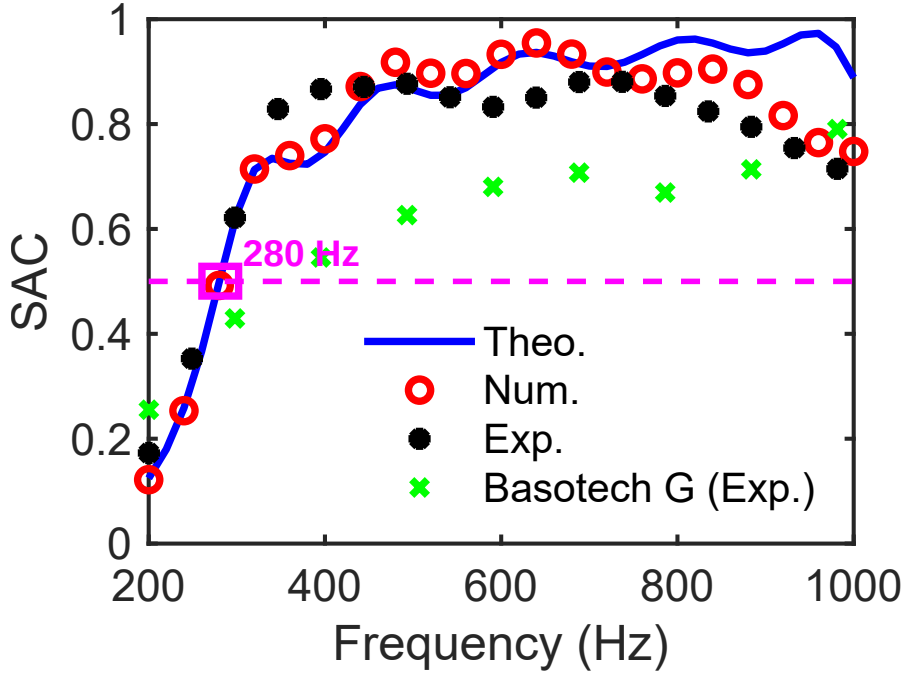


Figure 3.21: Test sample 50's theoretical (TpIM), numerical and experimental sound absorption coefficients. $D = 50$ mm. Target spectrum: white noise; Perforation porosity range: [0-5] %; frequency range: [200 1000] Hz. Basotect G+ (52 mm thickness) result shown for comparison

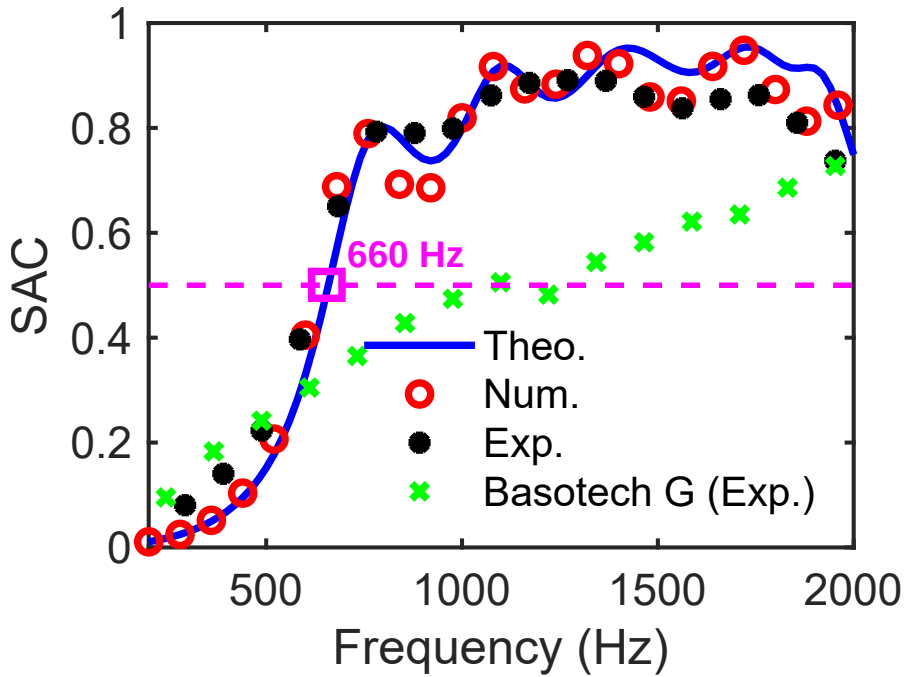


Figure 3.22: Test sample 20's theoretical (TpIM), numerical and experimental sound absorption coefficients. $D = 20$ mm. Target spectrum: white noise; Perforation porosity range: [0-5] %; frequency range: [200 2000] Hz. Basotect G+ (22 mm thickness) result shown for comparison

3.7 Discussion

In order to understand the effects of the different parameters they are each examined in turn.

- **The effect of optimisation**

A general observation can be made initially by comparing the spectra in Figure 3.6 to those in Figures 3.8 and 3.9. The spectral shapes for MPPAs as commonly seen in the literature are similar to those in Figure 3.6, i.e. relatively narrowband with a bandwidth which decreases with frequency. However, while different parameters and configurations have been examined by researchers with a view to improving the absorption at the maximum peak or the effective frequency range, for example, there has been little optimisation conducted. It is clearly seen here that the capacity of the MPPA is significant once the parameters are optimised, with very significant gains to be made in bandwidth and magnitude upon a judicious choice of parameters. This applies to both the basic MPPA, as can be seen in Figure 3.8(a), as well as to the MC-MPPA technology. Indeed, the application of optimisation to even the basic MPPA, resulting in greater bandwidth, is in itself a significant outcome of this Chapter. Referring to Tables 3.5 - 3.8, It can be observed that the maximum average absorption coefficient for the basic MPPA, for a thickness of 42 mm, is 0.79 in the 200 Hz - 2 kHz frequency range. In addition, at this same depth, all of the optimised MC-MPPAs have an absorption value greater than this, up to a maximum value of 0.91 for Case 4, making the TpIM optimisation process an extremely effective tool in absorber design.

- **The effect of perforation size**

White Noise Source. When the target sound source is white noise, a comparison between Case 1 (as shown in Figure 3.8) and Case 2 (as shown in Figure 3.9) as well as the values in Tables 3.5 - 3.6 can be made to highlight the effect of pore diameter.

Focusing initially on the (1×1) basic MPPA, for the larger hole diameter range (Case 2), the bandwidth is larger compared to that seen in Figure 3.6, which has arbitrary values as opposed to optimised ones, and its maximum absorption coefficient is close to unity. As is typical with increasing depth, the frequency of the peak decreases. In addition, it is interesting to note that the average absorption coefficient, c_α , actually increases somewhat from 0.48 at 12 mm absorber thickness to 0.65 at 42 mm absorber thickness. The behaviour in Case 1 for the basic MPPA with the smaller hole diameter

range is somewhat different. With increasing depth, the magnitude tends towards unity and its bandwidth significantly increases in both frequency directions with the c_α values increasing from 0.48 for the 12 mm absorber thickness to 0.77 for the 42 mm absorber thickness. The increase in bandwidth with decreasing pore size is in agreement with the theory of Maa [43].

For the MC-MPPAs, the main difference due to the pore size range is that, interestingly, the larger pore size range is advantageous, as it allows the lower frequency tone to be attenuated.

For the MC-MPPAs, from observation of Case 2, it can be observed that the absorption curves contain a number of perturbations in magnitude as a function of frequency which seem to be related to the number of chambers. These perturbations, however, seem to be damped by the smaller pore diameters of Case 1. The other key difference is that for the higher pore diameter range, lower frequencies can be attenuated.

Fan Noise Source. When the target sound source is fan noise as opposed to white noise, a comparison between Case 3 (as shown in Figure 3.10) and Case 4 (as shown in Figure 3.11), as well as the values in Tables 3.7 - 3.8, can be made to highlight the effect of pore diameter.

For the (1×1) basic MPPA, the behaviour and values are almost unchanged between the white noise and fan noise cases i.e. Figure 3.10(a) vs Figure 3.8(a) and Figure 3.11(a) vs Figure 3.9(a). This indicates that there is no scope for the optimisation algorithm to target the attenuation of tones when the technology is the basic MPPA.

In contrast with the (1×1) basic MPPA, referring now to the multi-chamber MPPAs, the behaviour between the small and large pore diameter range for the fan source is quite different. With increasing depth and chamber number, the smaller pore diameter range MC-MPPAs are able to form 2-3 peaks, the higher frequency two performing excellently at matching the frequency of the associated tones, whereas, the lower frequency peak in the absorption curve is unable to descend to the fundamental frequency to mask it fully.

For the larger pore diameter range, however, all tones can be frequency matched successfully, the more so with increasing depth and chamber number.

- **The effect of multiple chambers**

Referring to Tables 3.5 - 3.8, and paying attention to the numbers in bold, it can be observed that the value of the average absorption coefficient, c_α , increases in value with an increasing number of chambers with the (3×3) MC-MPPA generally performing best with c_α values of between 0.79 and 0.91 for the (3×3) MC-MPPA at 42 mm absorber thickness versus 0.65 and 0.79 for the (1×1) basic MPPA.

Referring to the frequency plot of Figure 3.9, it can be observed that by increasing the number of chambers and the air cavity depth, the response approaches a square wave with high amplitude absorption spreading over an increasingly broad frequency range.

Referring to the frequency plot of Figure 3.11, it can be observed that by increasing the number of chambers, the absorption response is ever more capable of masking the tonal peaks in addition to filling the intervals with increasing air cavity depth.

Finally, by referring to Figures 3.12 - 3.15, it is particularly stark how the MC-MPPAs perform better than the (1×1) basic MPPA with regards to c_α value for the larger pore diameter range, i.e. Figures 3.13 and 3.15, whereas whilst the MC-MPPAs are also significantly better than the (1×1) basic MPPA for the smaller pore diameter range, the improvement is not quite as great.

Figures 3.12 - 3.15 also provide further insight into the effect of the number of chambers. It is important to note that while the (3×3) configuration does yield higher c_α values than the (3×2) arrangement, the improvement is marginal. This diminishing return illustrates a critical design trade-off: although increasing the number of chambers enhances broadband absorption, it also significantly increases system complexity, fabrication difficulty, and potentially cost.

A further increase in the number of chambers beyond the (3×3) configuration appears to contribute little additional benefit to the absorption performance, while introducing substantial structural complexity. Therefore, based on the results obtained, the (3×3) MC-MPPA is considered to offer an optimal balance between acoustic performance and design feasibility. As such, the subsequent chapters will focus on this configuration as a representative case for further experimental and theoretical investigation.

- **Comparison with the benchmark porous material Basotect G+**

Figures 3.12 - 3.15 and Tables 3.5 - 3.8 allow us to compare, in addition to the bench-

mark (1×1) basic MPPA, the performance of the MC-MPPA with the industry standard Basotect G+. As expected, the performance of Basotect G+ improves with increasing depth, but for any particular depth, its absorption coefficient is always significantly less than that of the MC-MPPA.

Comparing Basotect with the (1×1) basic MPPA, it generally performs worse, as the Basotect material does not lend itself to optimisation using the TpIM routine. However, it is most likely that Basotect performs better than many unoptimised examples of the basic (1×1) MPPA to be found in the literature. An exception to this observation with the current data is that Basotect is better than the (1×1) MPPA for the deepest examples for the larger pore diameter range where the (1×1) MPPA underperforms.

Again, as the Basotect material does not lend itself to optimisation using the TpIM routine, it cannot be optimised to attenuate tones, and therefore when tones exist at low frequencies, the only approach with Basotect is to dramatically increase its depth. This underscores the significant advantage of the optimised MC-MPPA over basic porous materials. Comparing the results for the (3×3) MC-MPPA to Basotect in Figure 3.15 is stark.

- **Numerical Validation**

Referring to Figure 3.16, it can be observed that a comparison of the numerical results versus the theoretical TpIM for the six configurations in Figure 3.1 for a white noise case. Overall the agreement is very satisfactory and is sufficiently close to allow us to state, in addition to the comparisons made in Figure 3.6, that the TpIM approach is validated for both the optimised conditions and for higher-order chamber configurations. Allowing ourselves to inspect a bit more closely, it can be observed that there is a slight deterioration with increasing frequency in (1×3) , (2×3) and (3×3) cases. It is not quite as straightforward to say that the disagreement is a function of the number of chambers as the (3×3) case seems to be more accurate than the (1×3) case. As the depth of this test case is only 25 mm, the slight error is very unlikely to be attributable to a violation of the lumped element application but perhaps the porosity of the holes is such that hole-to-hole spacing is not negligible at higher frequencies. Further investigation will be conducted to try and understand the small discrepancy.

- **Experimental Validation**

Referring to Figure 3.21, the absorption curve for the experimental results is nearly identical to those of the numerical results. Comparing these results with the theoretical results, there is a slight deviation above 700 Hz, which may be caused by a violation of the lumped element approximation as the air cavity depth is 50 mm in this Test sample 50. However, the close agreement of the experimental results to both the theoretical and numerical results demonstrates the viability of the technology.

For Test sample 50, the average sound absorption coefficient c_α in the 200 Hz to 1000 Hz range for the theoretical, numerical and experimental analyses, are 0.8071, 0.7843 and 0.7720, respectively. When the frequency is greater than 281 Hz, which is the frequency at which the absorption coefficient increases above 0.5, Test sample 50 performs particularly well (theoretical c_α : 0.8692 numerical c_α : 0.8444 and experimental c_α : 0.8225) the response of the MC-MPPA being essentially continuously broadband. The identically-sized Basotech G+'s measured absorption coefficient, on the other hand, is only 0.6177 between 200 Hz and 2000 Hz and 0.6496 between 281 Hz and 2000 Hz.

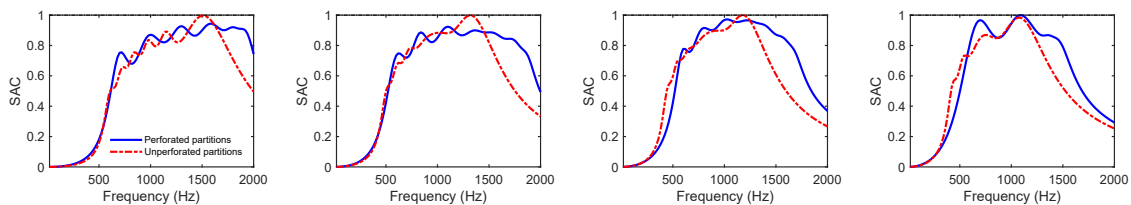
Referring to Figure 3.22, in the 200 Hz to 2000 Hz frequency range examined for the very thin Test Sample 20, the average sound absorption coefficient c_α from the theoretical, numerical and experimental analyses are 0.6746, 0.6686 and 0.6703, respectively. When the frequency is greater than 660 Hz, Test sample 20 performs particularly well (theoretical c_α : 0.8586, numerical c_α : 0.8428 and experimental c_α : 0.8273). On the other hand, for the same thickness of the absorber, Basotech G+'s measured absorption coefficient is only 0.4605 between 200 Hz and 2000Hz and 0.5470 between 660 Hz and 2000Hz. In addition, there is a good agreement for this sample between all of the theoretical, numerical and experimental results. As the sample is so thin, there appears to be no issue with the lumped element assumption at the higher frequency end.

- **Comparison of the sound absorption properties of the (3 × 3) MC-MPPA with and without perforated internal panels.**

An additional set of tests was performed to evaluate the benefit of the perforated internal walls. Figures 3.23 and 3.24 show the results of the comparison for both the white noise and fan noise cases for the (3 x 3) MC-MPPA which has the most internal walls of the configurations examined. An initial observation is that when there are no perforations in the internal panels the results are quite good. This clearly emphasises the

fact that the optimised TpIM method is a powerful tool which can be used to maximise the potential of such technologies. When internal panel perforations are included in the model and optimisation routine, Figure 3.23 shows that these perforations serve to broaden the response when white noise is the source. In addition, when tonal noise is the source, the perforations allow additional peaks to be attenuated.

In Appendix C, to further investigate the influence of internal perforated partitions on the acoustic performance of the optimised MC-MPPA, a theoretical analysis was conducted by comparing two representative configurations introduced in Chapter 5: MC-MPPA-II and MC-MPPA-III. Both configurations feature variable sub-cavity depths and identical outer geometry, with the key distinction being the presence of internal perforated partitions—MC-MPPA-II has no internal partitions, while MC-MPPA-III incorporates them. The results suggest that internal perforated partitions can achieve equivalent or even improved acoustic performance with fewer perforations in the top plate. This reduction in the number of holes could lead to lower aerodynamic drag, making the design particularly beneficial for applications under grazing flow conditions, such as aero-engine acoustic liners.



(a) air cavity depth: 20 mm, overall absorber thickness: 22 mm, perforated partitions: $c_\alpha = 0.6166$, unperforated partitions: $c_\alpha = 0.5726$ (b) air cavity depth: 25 mm, overall absorber thickness: 27 mm, perforated partitions: $c_\alpha = 0.6327$, unperforated partitions: $c_\alpha = 0.5741$ (c) air cavity depth: 30 mm, overall absorber thickness: 32 mm, perforated partitions: $c_\alpha = 0.6265$, unperforated partitions: $c_\alpha = 0.5602$ (d) air cavity depth: 35 mm, overall absorber thickness: 37 mm, perforated partitions: $c_\alpha = 0.5919$, unperforated partitions: $c_\alpha = 0.5434$

Figure 3.23: Sound absorption coefficients of optimised (3×3) MC-MPPA with and without perforated internal panels, frequency range: [20 2000] Hz, target spectrum: white noise, perforations diameter range: [0.5 1.0] mm and perforations porosity range: [0 5]%, $L = 50$ mm, $W = 40$ mm.

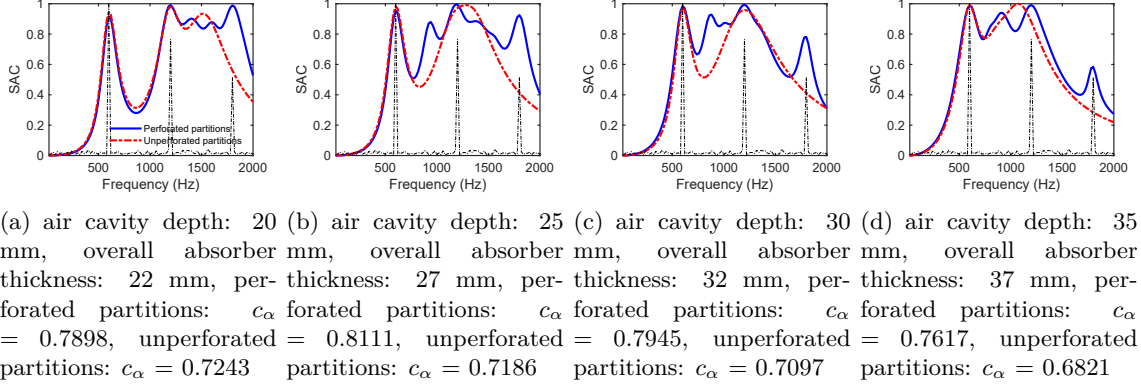


Figure 3.24: Sound absorption coefficients of optimised (3×3) MC-MPPA with and without perforated internal panels, frequency range: $[20 \ 2000]$ Hz, target spectrum: fan noise, perforations diameter range: $[0.5 \ 1.0]$ mm and perforations porosity range: $[0 \ 5]\%$, $L = 50$ mm, $W = 40$ mm.

3.8 Summary

1. The TpIM approach allows MPPA absorbers to be modelled theoretically and optimised

In this Chapter, an alternative to the equivalent circuit model method based on electro-acoustic analogies is presented. This alternative, based on graph theory, is called the Two-Point Impedance Method and has been transferred from the electronics domain to the acoustics domain in this Chapter and applied to the MPPA technology. This improved method allows single (1×1) MPPAs, or a group of six (1×1) MPPAs (which is the case of the (3×3) MC-MPPA without internal perforations, to be optimised, resulting in a broader band response than is typically found in the literature. In addition, MC-MPPAs have been studied theoretically. The development of the TpIM approach facilitates the examination of single chamber MPPAs tessellated alongside one another and separated by perforated walls in two dimensions, which is unmanageable using the ECM method, and allows for an optimisation of the system parameters, such as hole diameter and porosity, to be performed.

The TpIM has been successfully validated against the ECM method, numerical results and results to be found in the literature for (1×2) and (1×3) chambers and also against numerical results for (1×1) , (1×2) , (1×3) , (2×2) , (2×3) and (3×3) MC-MPPAs. The TpIM has also been successfully validated against experimental data for the (3×3) MC-MPPA with both white noise and tonal noise sources.

2. In addition to white noise, tonal noise can be attenuated with the MC-MPPA

In this Chapter, the capacity of the TpIM to be applied to target noise sources containing multiple tones, which is an important scientific challenge, has also been successfully demonstrated. By increasing the number of chambers, the MC-MPPA is increasingly able to attenuate additional tones, with, for the Case 4 example given in this Chapter, the majority of fan noise being attenuated by the (3×3) MC-MPPA for air cavity depths greater than 20 mm.

In addition to the benefit arising from an increase in chambers allowing multiple tones to be more efficiently suppressed, increasing the number of chambers improves absorption in general, with the value of the average absorption coefficient, c_α , being better than the (1×1) MPPA for almost all of the configurations examined with the greatest improvement being 0.91 (3×3) versus 0.67 (1×1) for an overall thickness of 42 mm for Case 4.

3. Pore diameter can have an effect on the sound absorption performance of an MPPA for different target sources and different numbers of cavities.

The literature reports pore diameters to a micron level numerically, whereas there are still challenges to manufacturing such small sizes practically. As a result of this, this research examines two principal pore size ranges: $[0.05 - 0.3]$ mm and $[0.3 - 1]$ mm. In general, for a white noise source, a decrease in the pore size results in an increase in the absorption coefficient and a broadening of the frequency range for the basic (1×1) MPPA, which is in agreement with Maa [42]. For the MC-MPPA, an increase in pore diameter allows slightly lower frequencies to be attenuated and results in perturbations in the response. When the noise source is tonal, there is no change in the response of the (1×1) MPPA, demonstrating that it is unable to target tones. For the MC-MPPAs, an increase in pore diameter benefits the attenuation of tones. The interesting conclusion from this Chapter is that, for the MC-MPPA with both white noise and tonal noise sources, there is no need to manufacture micron-level holes, as it appears that the perforated internal panels and the optimisation are sufficient to maximise performance.

4. Perforated internal panels allow tones to be absorbed and increase frequency bandwidth response.

The MC-MPPA is capable of attenuating specific acoustic tones whereas the (1×1) MPPA is not. In addition, for tessellated MPPAs, perforated panels are more capable of attenuating tones than solid internal panels and they increase absorption bandwidth. This offers new approaches for the attenuation of industrial fan noise, domestic appliances and aeroengines [127].

5. The MC-MPPA can provide broadband, low-frequency (below 1000 Hz), broadband noise attenuation

Effective low-frequency, broadband noise absorption is an intractable challenge. The optimised (3×3) MC-MPPA, for an air cavity depth of as little as 50 mm, achieves an absorption coefficient of 0.77 between 200 Hz and 1000 Hz, or 0.83 between 281 Hz and 1000 Hz. This result has been achieved experimentally.

6. The MC-MPPA performs significantly better than benchmark alternatives.

The MC-MPPA significantly outperforms both the basic (1×1) MPPA and Basotect G+, which is an industry-standard melamine resin foam.

7. The MC-MPPA is a deeply sub-wavelength broadband noise sound absorber.

Theoretically, perfect attenuation at 600 Hz is achieved at a depth of 42mm, which is a depth-to-wavelength ratio of 14. Experimentally, an absorption coefficient of approximately 0.82 is achieved at 340 Hz at a depth of 52 mm, which is a depth-to-wavelength ratio of 20.

Chapter 4

Ultra-Broadband Sound Absorption in a Compact MC-MPPA

4.1 Introduction

The development of MPPA technologies is closely linked to advances in manufacturing technologies. In Chapter 3, the impact of perforation diameter and perforation porosity on the sound absorption characteristics of an MC-MPPA was investigated. The theoretical results indicated that when the noise of interest is white noise, there has been a strong agreement between the theory put forth by Maa [44] and the observed increase in the sound absorption bandwidth with decreasing perforation diameter. Despite significant progress, the fabrication of perforations with extremely small diameters—particularly in combination with high perforation ratios—remains a major challenge. It often results in low economic efficiency when using conventional techniques such as mechanical punching, as well as additive manufacturing methods like 3D printing. Therefore, designing and fabricating MPPAs with excellent acoustic performance within the limitations of current manufacturing technologies represents a notable technical challenge, yet holds great promise for industrial applications and offers considerable economic potential.

In this Chapter, a robust absorber with sub-chambers of varying depth and porosity is presented. Its design and optimisation is facilitated by the introduction and original application of the TpIM which enables the modeling of a technology of such complexity. The introduction of the TpIM to acoustic metamaterial modeling is a milestone as the ubiquitous ECM based on electro-acoustic analogies can only represent very basic circuits and so has been an obstacle to innovative development. The current work has the following objectives: (1) to develop and optimise a multi-chamber MPPA where each sub-chamber can have unique porosity both on the faceplate and on internal partitions as well as unique depth. (2) to achieve ultra-broadband sound absorption with this novel MC-MPPA. (3) to identify which parameters are most critical to control during production to maintain optimal absorption characteristics through a sensitivity analysis. (4) to design and fabricate a compact MC-MPPA whose geometric detail is achievable with reasonable current processing and manufacturing technologies. (5) to validate the compact MC-MPPA design numerically and experimentally.

4.2 Design

The MC-MPPA investigated in this chapter is characterized by adjacent panels with different porosities and varying depths of its sub-chambers, as shown in Figure 4.1. This configuration

will be referred to as MC-MPPA-III in subsequent chapters. The key parameters of the investigated MC-MPPA are outlined in Table 4.1. The MC-MPPA structure is divided into nine chambers arranged in a 3×3 configuration, where each chamber can have unique depths and is separated by micro-perforated adjacent panels. TpIM is applied to determine the impedance of the MC-MPPA. Figure 3.3(f) in Chapter 3 shows the MC-MPPA from Figure 4.1, represented schematically as a bi-pyramidal molecular structure. As in Chapter 3, the objective here is to calculate the impedance of the overall system using a matrix approach. The optimisation from Chapter 3 will then be used to maximize the total SAC within a specified frequency range to obtain the optimised MC-MPPA design. In this Chapter, white noise is the target noise.

Table 4.1: Parameters used to characterize the acoustic behavior of the MC-MPPA-III.

MC-MPPA type	MPPs Location	MPPs Symbol	MPPs thickness	Perforation diameter	Perforation porosity	Perforation number	Sub-chamber depth
III	Top	$MPP_{t(1)}$	τ_t	$d_{t(1)}$	$\phi_{t(1)}$	$n_{t(1)}$	$D_{(1)}$
		$MPP_{t(2)}$		$d_{t(2)}$	$\phi_{t(2)}$	$n_{t(2)}$	$D_{(2)}$
		$\vdots$		$\vdots$	$\vdots$	$\vdots$	$\vdots$
		$MPP_{t(9)}$		$d_{t(9)}$	$\phi_{t(9)}$	$n_{t(9)}$	$D_{(9)}$
	Internal	$MPP_{i(1)}$	τ_i	$d_{i(1)}$	$\phi_{i(1)}$	$n_{i(1)}$	
		$MPP_{i(2)}$		$d_{i(2)}$	$\phi_{i(2)}$	$n_{i(2)}$	
		$\vdots$		$\vdots$	$\vdots$	$\vdots$	
		$MPP_{i(12)}$		$d_{i(12)}$	$\phi_{i(12)}$	$n_{i(12)}$	

Two MC-MPPA experimental samples were fabricated, each optimised for a specific target frequency range: [20 1000] Hz and [20 2000] Hz. These were respectively designated as Test Piece 1000 and Test Piece 2000, according to their optimisation frequency bands. A detailed illustration of the latter is provided in Figure 4.2. Both the core (air cavities and perforated adjoining panels) and the top MPPs were 3D printed with an mSLA printer. The maximum depth of the sub-chamber air-cavity in both samples was 45 mm and the total thickness of each technology was 47 mm including the thickness of τ_t . The optimised geometric parameters of the experimental Test Pieces 1000 & 2000 can be found in Tables E.3 and E.4.

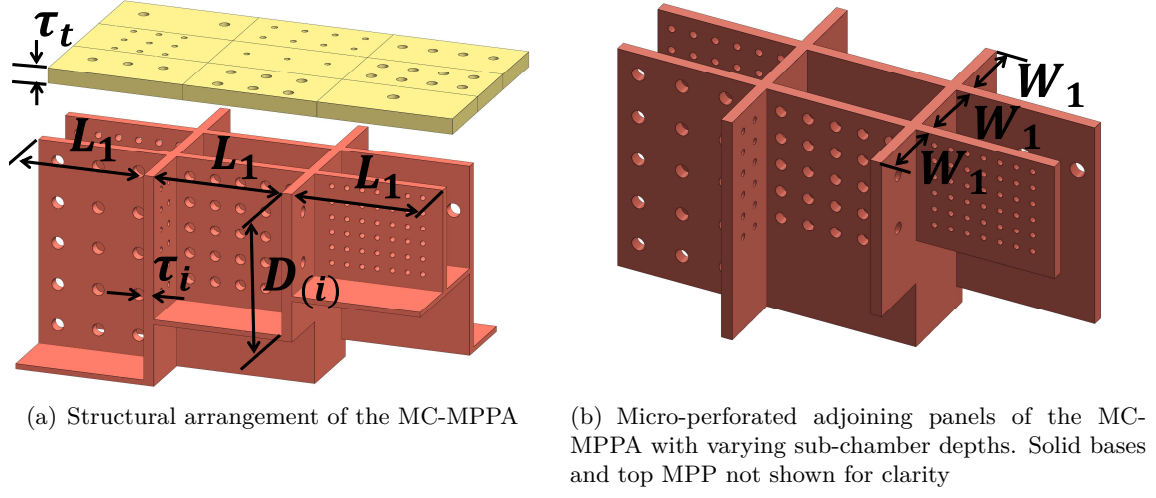


Figure 4.1: Schematic of the enhanced MC-MPPA.

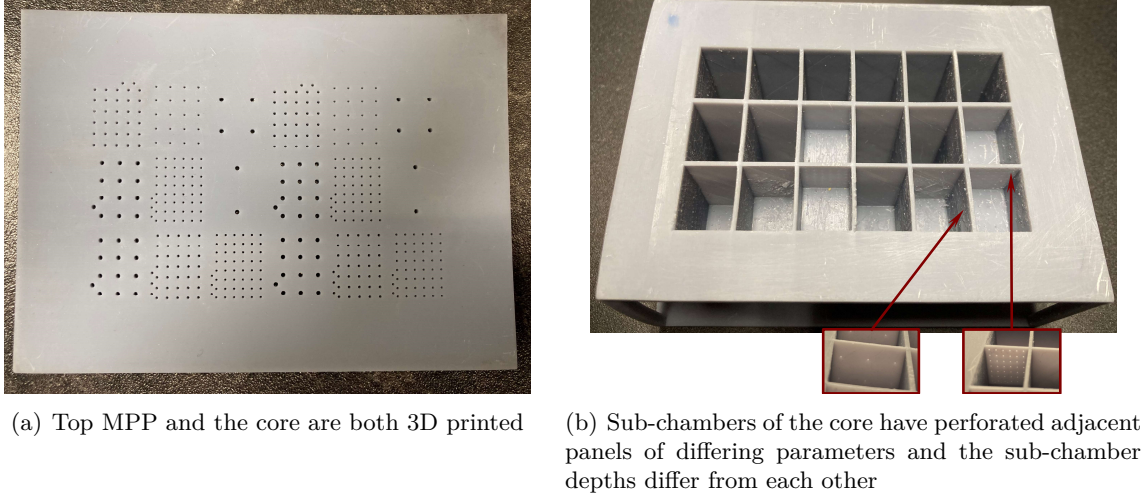


Figure 4.2: Photograph of the MC-MPPA test samples.

4.3 Results

Figure 4.3 displays the theoretical SAC curves of MC-MPPAs that have been optimised for four different frequency ranges: $[20\ 1000]$ Hz, $[20\ 2000]$ Hz, $[20\ 3000]$ Hz, and $[20\ 4000]$ Hz. The geometric parameters of the MC-MPPA that correspond to each of the absorption curves in Figure 4.3 are provided in Tables E.5 - E.8 in Appendix E.3. Furthermore, Table 4.2 provides the specific values of c_α for Figure 4.3 in the four different frequency ranges as a function of maximum sub-chamber depth $D_{\max}$. It is clear that the c_α value is dependent on the frequency range chosen for evaluation and that values of above 0.8 and even 0.9 over

a wide frequency range can be attained through a judicious choice of the frequency limits of optimisation.

Figure 4.5 compares the theoretical to the numerical results, which were calculated using COMSOL, in the same frequency ranges, of the MC-MPPAs with varying overall absorber thicknesses across different frequency ranges. These figures were generated to investigate the relationship between absorber thickness and sound absorption performance of the MC-MPPA for various frequencies ranges. The numerical results presented in these figures serve to validate the theoretical results depicted in Figure 4.3 and increase confidence in the novel TpIM approach to theoretical modeling and the optimisation algorithm.

Investigating the sensitivity of broadband absorption performance to various types of parameters becomes crucial when considering uncertainties in manufacturing parameters. Through the analysis of the theoretical model, it is possible to determine which parameters significantly impact the broadband absorption performance, and which have a relatively minor effect. Figure 4.4 presents an analysis of the sensitivity of the MC-MPPA's geometric parameters. This analysis aims to derive general principles from the theoretical model that can inform optimisation and practical manufacturing processes, thus ensuring the stability and reliability of the broadband absorption performance.

In order to provide experimental validation, Figures 4.6(a) and 4.6(c) depict the theoretical, numerical, and experimental SACs of the Test Piece 1000 and Test Piece 2000 MC-MPPA samples. These figures provide a comprehensive comparison between the expected and actual SACs of the MC-MPPA in these frequency ranges. The acoustic resistance and reactance of Test Piece 1000 and 2000 MC-MPPA samples, both numerical and experimental, are plotted in Figures 4.6(b) and 4.6(d).

To emphasize the compactness of the MC-MPPA design, illustrated by the ratio of the structure's thickness to its minimum operational wavelength, Figure 4.7 presents a comparative study. This Chapter underscores the MC-MPPA's effective performance, especially regarding its absorption bandwidth and efficiency at deep-subwavelength scales.

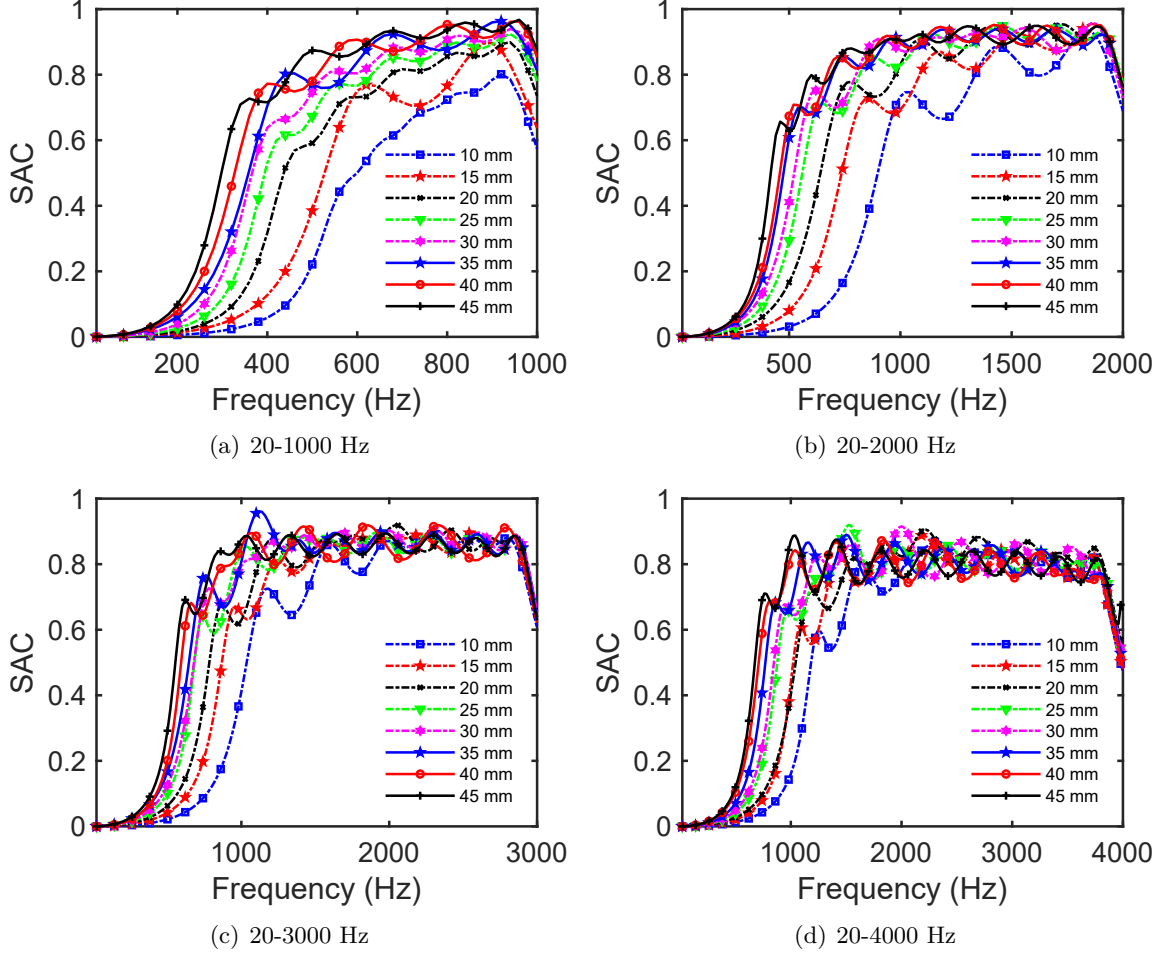


Figure 4.3: Theoretical SACs of the optimised MC-MPPA as a function of core depth and optimised frequency range.

Table 4.2: Average SACs of the MC-MPPA in four different frequency ranges (Data corresponds to the SACs curves in Figure 4.3).

[200 1000] Hz		[200 2000] Hz		[200 3000] Hz		[200 4000] Hz	
$D_{\max}$ (mm)	c_{α}	$D_{\max}$ (mm)	c_{α}	$D_{\max}$ (mm)	c_{α}	$D_{\max}$ (mm)	c_{α}
10	0.4097	10	0.5200	10	0.5905	10	0.5881
15	0.5000	15	0.6153	15	0.6479	15	0.6249
20	0.5818	20	0.6772	20	0.6780	20	0.6483
25	0.6393	25	0.7138	25	0.7008	25	0.6625
30	0.6855	30	0.7340	30	0.7158	30	0.6727
35	0.7163	35	0.7541	35	0.7261	35	0.6821
40	0.7527	40	0.7665	40	0.7305	40	0.6874
45	0.7834	45	0.7812	45	0.7424	45	0.6984

4.4 Discussions

4.4.1 Absorption response of the MC-MPPA

Regarding the acoustic response of the MC-MPPA, results demonstrate ultra-broadband sound absorption characteristics when all parameters of each chamber are allowed to vary independently. Figure 4.6(a), shows that an average SAC over 0.8 can be achieved experimentally by the MC-MPPA at a maximum sub-chamber depth of only 45 mm in the frequency range of 276 Hz to 1000 Hz, while Figure 4.6(c), demonstrates that the same SAC value can be achieved at a maximum sub-chamber depth of 45 mm in the frequency range of 325 Hz to 2000 Hz. Additionally, for higher frequencies, as presented in Figure 4.3 and Table 4.2, the average SAC of MC-MPPA at a maximum sub-chamber depth of 45 mm is 0.7424 and 0.6984 at frequency ranges of [200 3000] Hz and [200 4000] Hz, respectively. Notably, even at a maximum sub-chamber depth of only 20mm, MC-MPPA achieves an average SAC of 0.6780 and 0.6483 at frequency ranges of [200 3000] Hz and [200 4000] Hz, respectively.

Regarding the "compact" acoustic performance of the MC-MPPA, as illustrated in Figure 4.6(a) and 4.6(c), the experimental SACs of Test Piece 1000 exceed 0.8 within the frequency range of 397 to 1000 Hz. Remarkably, the absorber's thickness is a mere 47 mm, equivalent to $1/18.2$ of the sound wavelength at the minimum operational frequency. Similarly, the SACs of Test Piece 2000 surpass 0.8 in the 698 to 2000 Hz frequency range, with the absorber maintaining the same thickness, equating to $1/10.5$ of the sound wavelength at its minimum operational frequency.

The MC-MPPA's acoustic absorption capabilities can also be explained by its acoustic reactance and resistance, as shown in Figures 4.6(b) and 4.6(c). An ideal impedance match requires the acoustic reactance to be zero and the acoustic resistance to be one, achieved simultaneously. Analysis of numerical and experimental acoustic reactance and resistance from Figures 4.6(b) and 4.6(c) reveals that the numerical and experimental results align closely. Significantly, within the frequency range where SAC exceeds 0.8, Figures 4.6(a) and 4.6(b) indicate that the acoustic reactance approaches zero, and the acoustic resistance nears one. This pattern is also observed in Figures 4.6(c) and 4.6(d). Specifically, for Test Piece 2000, the experimental data at approximately 1400 Hz show acoustic reactance approximately zero and acoustic resistance close to one, aligning with its experimental absorption curve at the same frequency, where SAC nearly reaches one.

4.4.2 Impact of the air cavity depth in sub-chambers

As can be seen in Figure 4.3 and Table 4.2, an increase in the *maximum* sub-chamber depth, $D_{\max}$, corresponds to an enhancement in the SAC of the MC-MPPA. In this research, it has been observed that fixing the sub-chamber depths to a common value limits the operating frequency of the MC-MPPA. However, as illustrated in Figure 4.3 and Table 4.2, allowing the depths to vary from one sub-chamber to another effectively broadens the operating frequency range from [20 2000] Hz to [20 4000] Hz. Upon further investigation, as detailed in the Tables E.5 - E.8 in Appendix E.3, it is observed that when the optimised frequency is confined to 20-1000 Hz, the depths of individual sub-cavities tend to remain consistent. However, as the upper limit of the optimised frequency increases, the difference in depth among sub-cavities within the same MC-MPPA becomes more pronounced. This observation further corroborates the conclusion from an alternative perspective.

4.4.3 Sensitivity analysis of MC-MPPA's geometric parameters

Given the precision required in MC-MPPA manufacturing, even slight deviations in geometric parameters can significantly impact performance. A sensitivity analysis is essential to identify which parameters are most critical to control during production to maintain optimal absorption characteristics.

The sensitivity of seven key geometric parameters is analyzed: chamber depth, top MPP's perforation diameters, top MPP's perforation porosity, top MPP's perforation thickness, internal MPP's perforation diameters, internal MPP's perforation porosity and internal MPP's perforation thickness. The results are plotted in Figure 4.4. Each of these parameters plays a crucial role in determining the acoustic impedance and, consequently, the absorption performance of the MC-MPPA. In this analysis, the ranges of parameters have been chosen at the current common processing and manufacturing technologies. Details of the sensitivity analysis of MC-MPPA's geometric parameters can be found in Appendix D.

The analysis demonstrates that the broadband absorption performance exhibits significant sensitivity to the chamber depth, and the porosity of the top and internal MPPs, while showing moderate sensitivity to the panel thickness. Conversely, alterations in the diameters of the top and internal MPPs' perforations have a negligible impact. This evidence indicates that stricter regulation of chamber depth, perforation porosity, and MPP thickness during production can lead to substantial enhancements in performance consistency. These findings

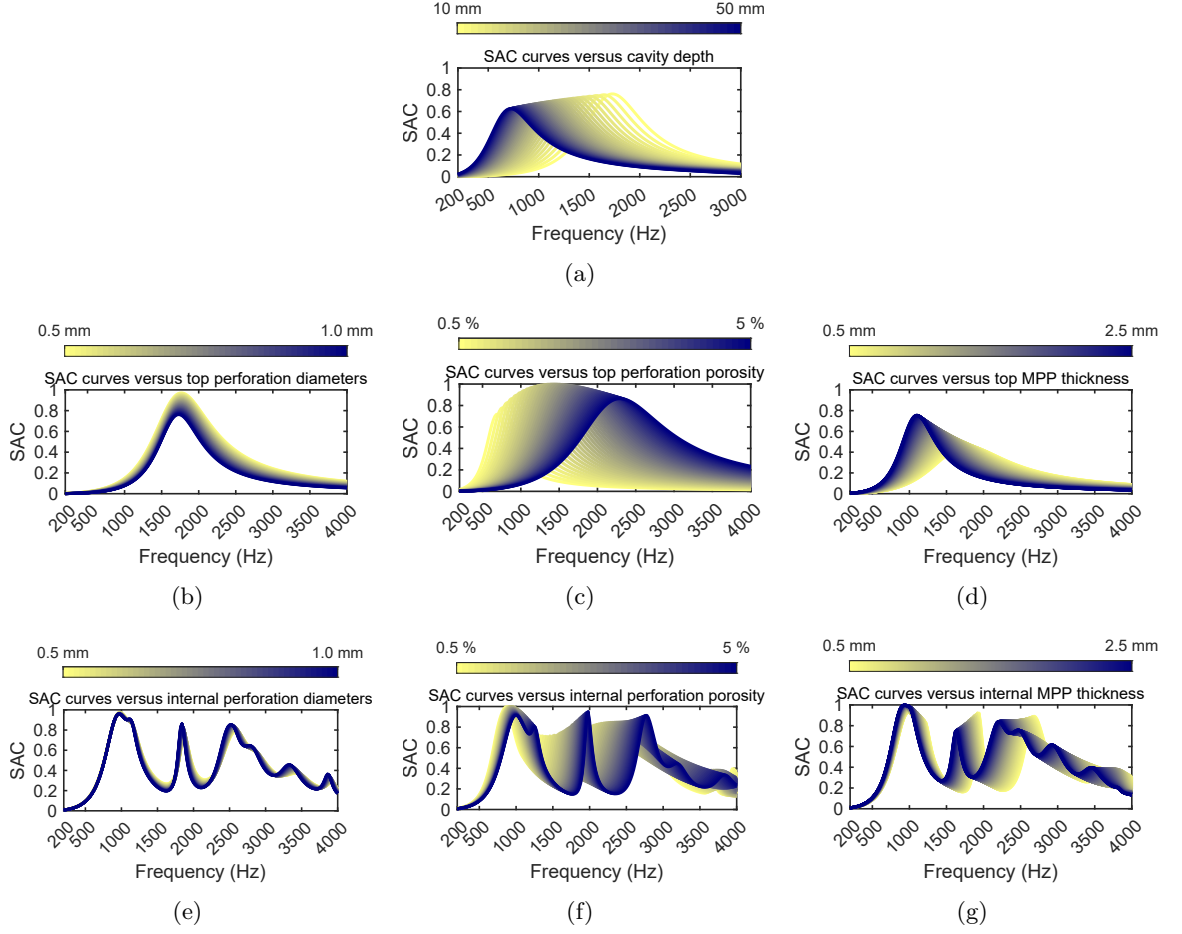


Figure 4.4: Sensitivity analysis of MC-MPPA's geometric parameters. (a) Curve's color gets darker as its depth increases from 10 mm to 50 mm. (b) and (e) Curve's color gets darker as its perforation diameter increases from 0.5 mm to 1.0 mm. (c) and (f) Curve's color gets darker as its perforation porosity increases from 0.5 % to 5.0 %. (d) and (g) Curve's color gets darker as its thickness increases from 0.5 mm to 2.5 mm.

offer critical guidance for optimising the MC-MPPA and its manufacturing process in future research and can explain differences between experimental and theoretical/numerical results.

4.4.4 Numerical and experimental validation

Regarding the numerical validation, Figure 4.5 presents a comparison between the theoretical and numerical SACs of MC-MPPAs for varying overall absorber thicknesses across four distinct frequency ranges. Generally, the numerical simulations agree well with the theoretical results. Upon closer examination, minor discrepancies are observed with increasing depth and frequency. These marginal discrepancies may be explained by a violation of the lumped element assumption and the possibility that the spacing between the perforations is not negligible. A more exhaustive exploration of these phenomena will ensue in subsequent studies.

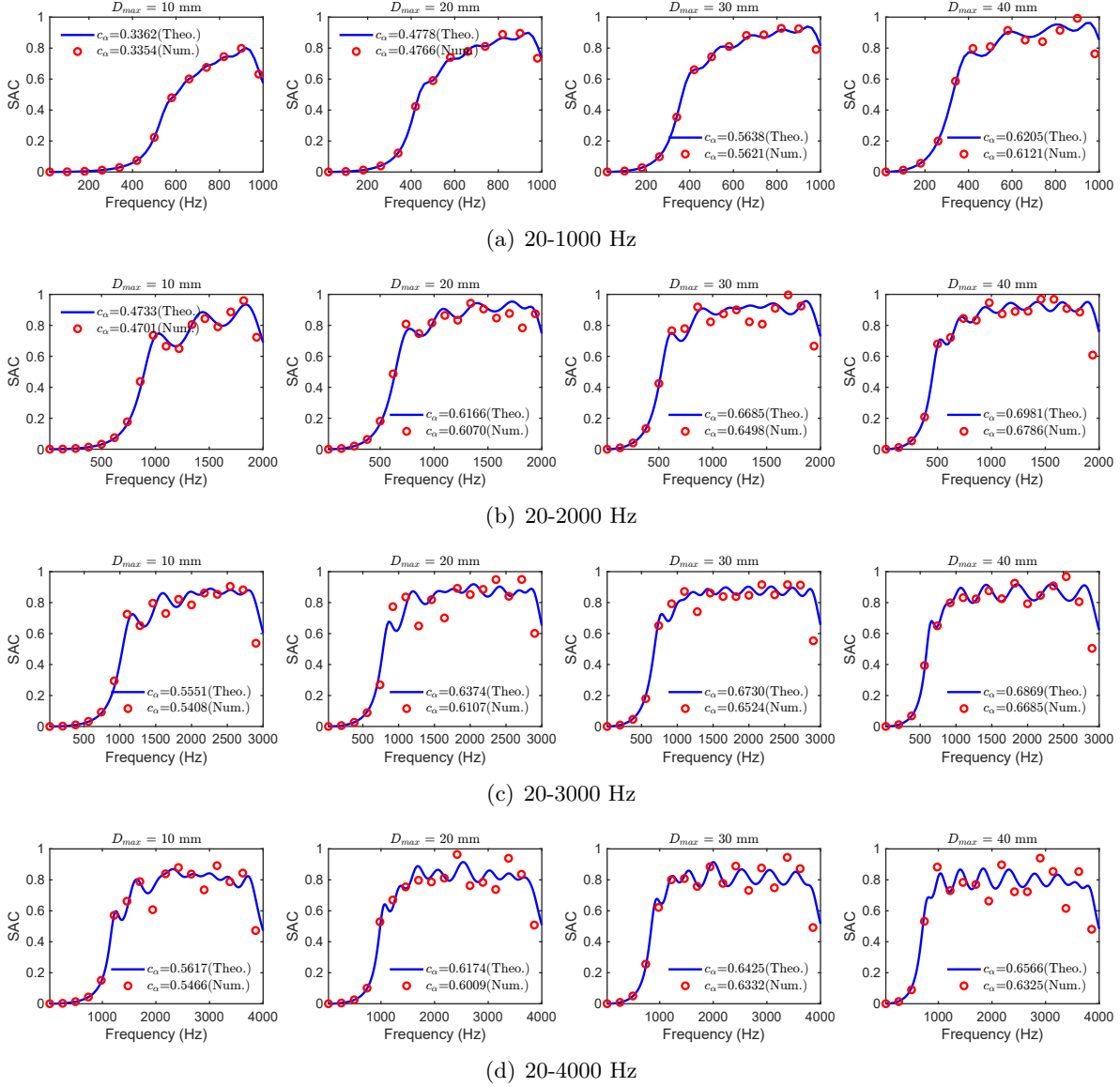


Figure 4.5: Theoretical and numerical SACs of MC-MPPA with different overall absorber thicknesses in different frequency ranges.

Regarding the experimental validation, as illustrated in Figure 4.6, the experimental results demonstrate a high degree of agreement with both the theoretical values and the numerical simulations and demonstrate that the design is more than just conceptual but is realizable.

4.4.5 Analysis of the processing and manufacturing challenges associated with the MC-MPPA under investigation

Generating numerous micro-holes (with diameters smaller than 0.3 mm) in a thin sheet presents a substantial challenge for both 3D printers and CNC milling machines. Even when successfully produced, the process requires significant time and specialized equipment costs.

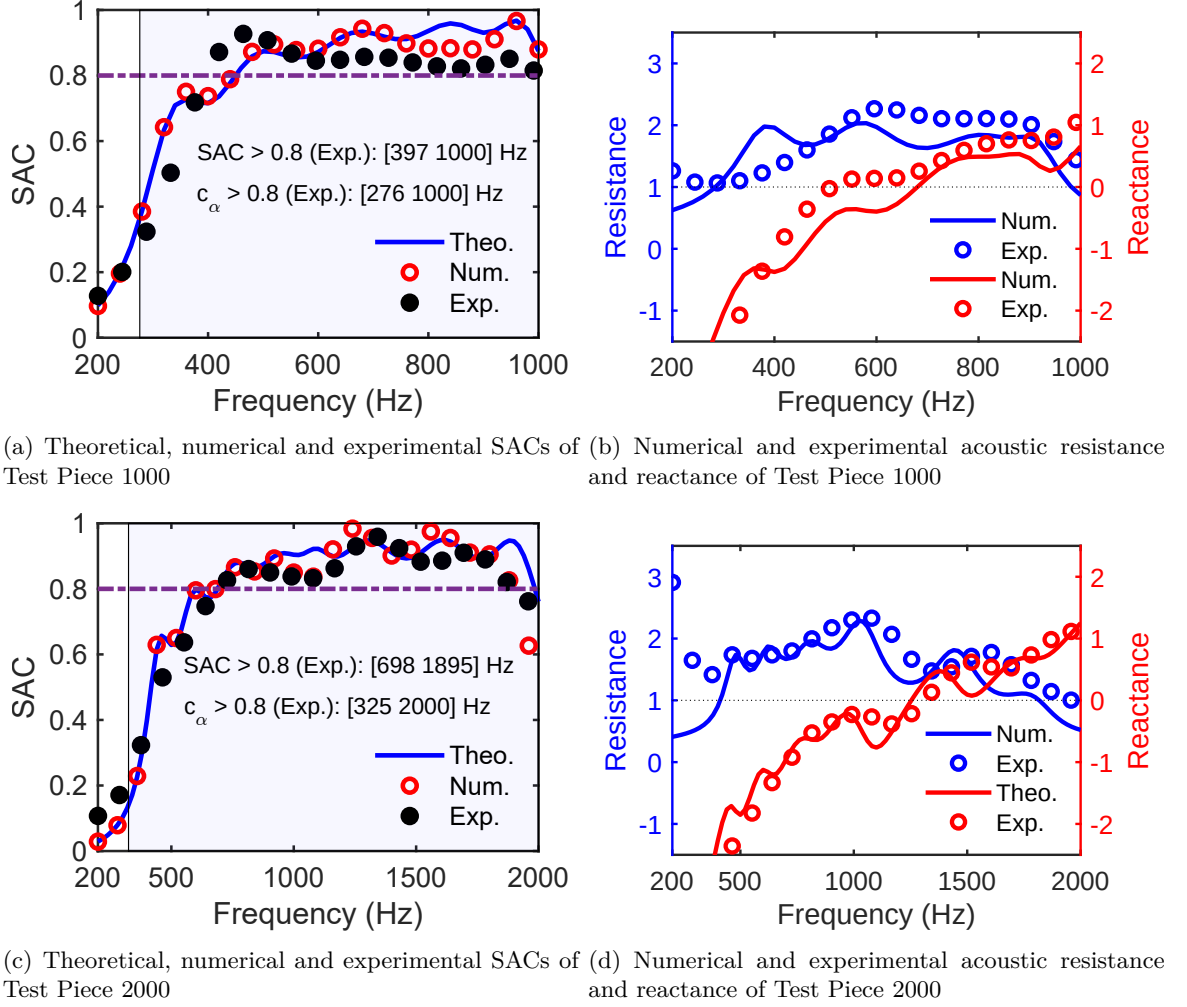


Figure 4.6: Theoretical, numerical, and experimental SACs, and numerical and experimental acoustic resistance and reactance for Test Pieces 1000 and 2000.

Consequently, large-scale engineering applications employing micro-perforated plates with diameters under 0.3 mm are limited by their economic efficiency. In principal, an increase in diameter and a decrease in the number of micro-holes machined in a thin plate result in reduced requirements for the machining and manufacturing process, and therefore decreased production costs, and enhanced potential for large-scale industrial applications.

In this Chapter, the SAC performance of the MC-MPPA is considered, along with the processing and manufacturing challenges associated with the designed MC-MPPA. Consequently, the diameter of the micro-holes is controlled to be within the range of 0.5 mm to 1.0 mm, and the porosity is set between 0% and 5%. In addition, the plate thicknesses are fixed at 1 mm and 2 mm. This design can undoubtedly be easily printed or machined by the vast majority of 3D printers and CNC machines. Examining the geometric parameters (listed in

Tables E.3 and E.4) of the two MC-MPPA test pieces, Test Piece 1000 and Test Piece 2000, reveals that the smallest micro-hole diameter is 0.5 mm, and the largest number of holes is 183, which are reasonable quantities.

4.4.6 Comparative analysis

Figure 4.7 presents a comparison of the sound absorption bandwidth (SAC > 0.8) between the MC-MPPA and other compact broadband MPP-type sound absorbers. In this figure, the red bars represent the MC-MPPAs under investigation, namely Test Piece 1000 (TP1000) and Test Piece 2000 (TP2000), with data sourced from Figures 4.6(a) and 4.6(c). The green bar illustrates the hybrid sound absorber described by Wu et al. [111], which combines an MPP and coiled-up Fabry–Pérot channels. The yellow bar depicts the MPPA with an integrated Helmholtz resonator, as explored in Li et al. [107]. The magenta bar represents the single-layer MPPA from Li et al. [107], while the cyan bar corresponds to the double-layer MPPA, also from Li et al. [107]. It is important to note that all the data compared are experimental. The sound acoustic performance of the MC-MPPA is notably high, particularly in aspects of absorption bandwidth and efficiency at deep-subwavelength levels. This efficiency is distinctly marked by the proportion of the structure’s thickness relative to its minimum operational wavelength.

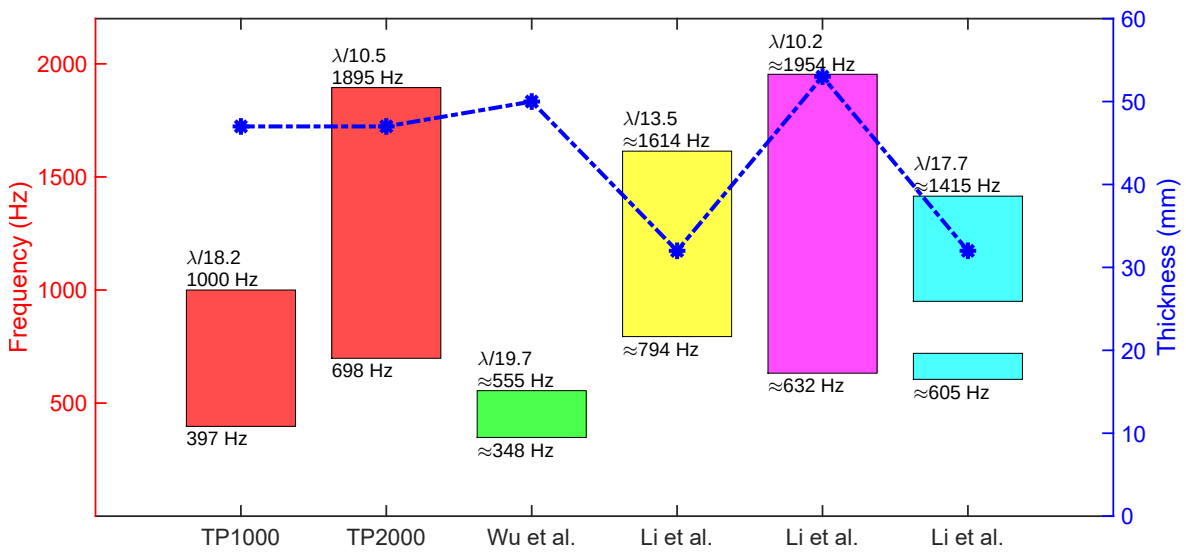


Figure 4.7: Comparison of sound absorption bandwidth (SAC > 0.8) among MC-MPPAs and other compact broadband MPP-type sound absorbers.

4.5 Summary

This Chapter concentrates on exploring the acoustic properties of the MC-MPPA while varying the depths of sub-chambers. The acoustic properties of the MC-MPPA have been evaluated, taking into account the limitations of current processing and manufacturing technologies. The investigation led to the following findings:

1. A compact MC-MPPA can demonstrate ultra-broadband sound absorption capabilities. Experimental results from MC-MPPAs, with a thickness of only 47 mm, exhibit an average SAC exceeding 0.8 within the frequency ranges of [276 1000] Hz and [325 2000] Hz. Theoretically, even with a maximum sub-chamber depth of just 20 mm, the MC-MPPA attains average SACs of 0.6780 and 0.6483 in the frequency ranges of [200 3000] Hz and [200 4000] Hz, respectively.
2. The experimental SACs of Test Piece 1000 exceed 0.8 within the whole frequency range of 397 to 1000 Hz. Remarkably, the absorber's thickness is a mere 47 mm, equivalent to $1/18.2$ of the sound wavelength at the minimum operational frequency.
3. The variable sub-chamber depth of the MC-MPPA considerably broadens the sound absorption bandwidth. Varying sub-chamber depths effectively extends the operating frequency range from [20 2000] Hz to [20 4000] Hz and simultaneously improves performance.
4. Sensitivity analyses show that the absorption performance of MC-MPPA is predominantly influenced by several key factors: the chamber depth, the porosity of the top and internal microperforated panels (MPPs), and the thickness of their perforations. Consequently, these parameters warrant meticulous regulation throughout the design and fabrication process to guarantee uniform performance.
5. The design and fabrication of a compact MC-MPPA exhibiting exceptional acoustic performance is achievable through current common processing and manufacturing technologies, providing substantial practical applications and economic value within the industry.
6. Whilst a 3×3 chamber MC-MPPA was examined, other geometries can be modeled. Similarly, the depths and frequency ranges were chosen as examples only of what is possible.
7. As every optimised curve has a frequency dependent ramp-up and ramp-down in absorption coefficient, designs can be made such that the high level absorption plateau

spans the frequency range of interest. This will result in even high average absorption values.

Chapter 5

Broadband Sound Transmission Loss of MC-MPPAs in a Grazing Flow

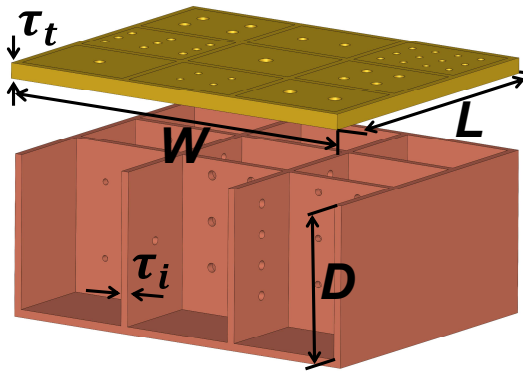
5.1 Introduction

The development of multi-chamber MPPAs represents a major improvement in acoustic design, allowing for enhanced sound absorption across a wider frequency range. By optimising the geometric features, the multi-chamber structure maximizes sound absorption over a broader frequency range without adding extra thickness or weight to the system. In Chapter 3 and Chapter 4, research shows that by fine-tuning the dimensions of the chambers and perforation parameters, multi-chamber MPPAs can be particularly effective for applications requiring wideband noise control. To date, designing high-efficiency liners for variable grazing flow conditions still remains a significant challenge. Undoubtedly, broadening the application of multi-chamber MPPAs as acoustic liners for reducing aeroengine fan noise would have far-reaching significance.

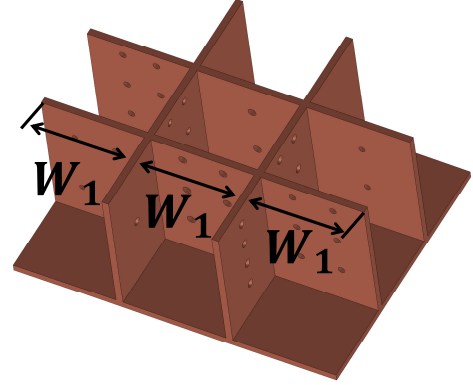
This Chapter advances micro-perforated panel absorber technology by developing three sub-wavelength MC-MPPA liners designed to achieve broadband sound transmission loss under various grazing flow conditions. The system was modeled using graph-theory-based TpIM and Cremer impedance methods, optimising the MC-MPPA for maximum sound transmission loss within a selected frequency range. Experiments at the Laboratoire d'Acoustique de l'Université du Mans (LAUM) grazing flow facility investigated the effects of flow and flow speed on the acoustic performance of the MC-MPPA. Nonlinear effects were examined by varying acoustic excitation amplitude, and the influence of acoustic source position was studied.

5.2 Designs

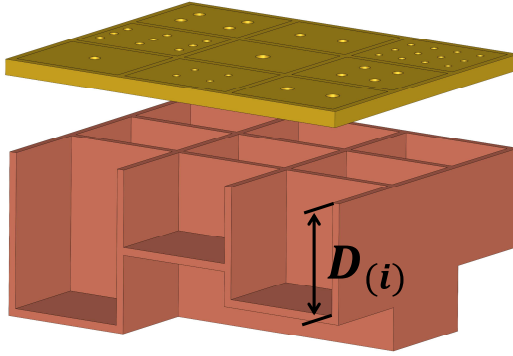
In this Chapter, three variations of the MC-MPPA are investigated, as shown in Figure 5.1. MC-MPPA-I and MC-MPPA-III have already been examined for normal acoustic incidence in Chapters 3 and 4. Here, a third variant, MC-MPPA-II, is added to the examination for completeness, as the three variations are capable of performing in different ways. As each would present different challenges with respect to large scale manufacturing processes, it was believed useful to evaluate the three so that informed choices can be made. All three are rectangular and lend themselves well to tessellation. In Chapter 3, different numbers of sub-chambers, 1×3 , 2×2 etc. were examined, but in this Chapter the focus will be on the 3×3 unit cell as it was shown to be the most efficient although this can depend on the outer dimensions with respect to the frequencies of interest to be examined.



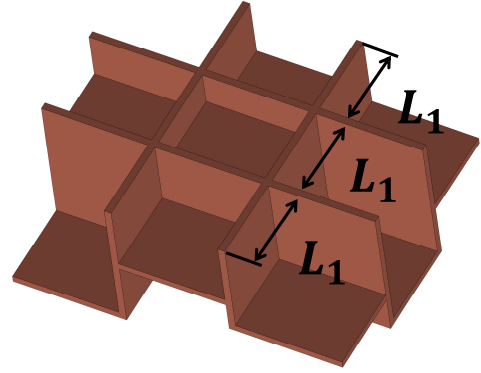
(a) MC-MPPA-I, varying sub-chamber faceplate porosity, constant sub-chamber depth.



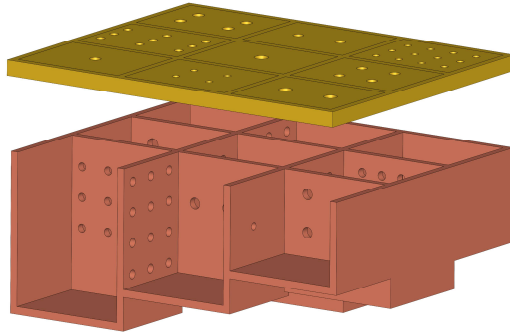
(b) Varying micro-perforated internal panels of MC-MPPA-I



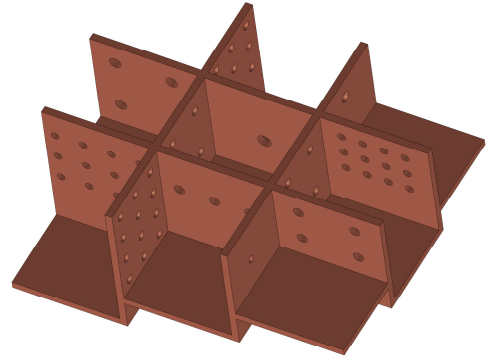
(c) MC-MPPA-II, varying sub-chamber faceplate porosity, varying sub-chamber depth.



(d) Non-perforated internal panels of MC-MPPA-II



(e) MC-MPPA-III, varying sub-chamber faceplate porosity, varying sub-chamber depth.



(f) Varying micro-perforated internal panels of MC-MPPA-III

Figure 5.1: Geometric layout of the MC-MPPA-I, MC-MPPA-II and MC-MPPA-III structures.

- MC-MPPA-I, as depicted in Figure 5.1(a) and 5.1(b), is characterized by its internal micro-perforated panels and a consistent depth across sub-chambers. The sub-chambers themselves are non-locally reacting as they are interconnected through the

internal wall micro-perforations. However, as the outer boundary wall of the 3×3 unit cell is non-perforated, it can be considered a locally reacting liner at the unit cell length scale. Chapter 3 highlighted its potential as an advanced deep subwavelength absorber, demonstrating its capability for high amplitude broadband sound absorption in impedance tube assessments.

- MC-MPPA-II, as shown in Figure 5.1(c) and 5.1(d), has no internal perforations but the depths of each sub-chamber are allowed to vary from each other. This may be considered to be an easier option to manufacture.
- MC-MPPA-III, Figure 5.1(e) and 5.1(f), merges the two concepts with micro-perforated internal panels with variable sub-chamber depths. This model, an advanced version of MC-MPPA-I, allows further control to broaden the bandwidth response. Unlike traditional POHC SDOF liners, where each sub-chamber is of unique depth and whose faceplate has a uniform porosity over each duct, this design allows the perforation characteristics (hole count and diameter) to be unique for every perforated surface and for the depth to vary per sub-chamber. Chapter 4 confirmed that this lightweight, compact design is not only viable in terms of design and fabrication but also achieves ultra-broadband low-frequency absorption, with its geometrical details feasible with mSLA 3D printing.

The constitutive parameters of the models MC-MPPA-I and MC-MPPA-II are detailed in Table 5.1. The constitutive parameters of the model MC-MPPA-III are detailed in Table 4.1. For these models, the dimensions of length (L) and width (W) are specified, along with the thicknesses of the top and internal micro-/non-perforated panels, denoted by τ_t and τ_i , respectively. The dimensions of each sub-chamber, L_1 and W_1 , are calculated as $\frac{L-4\tau_i}{3}$ and $\frac{W-4\tau_i}{3}$, thereby enabling a rectangular configuration that is optimal for tessellation in future industrial settings. The specific dimensions used in this Chapter are given in Appendix E.4 and were chosen to suit the grazing flow rig in the experimental test campaign described in Section 5.6.1. The total height is determined by adding the thickness of the top panel to the depth of the deepest chamber. The top micro-perforated panels are labeled from $MPP_{t(1)}$ to $MPP_{t(9)}$. In the cases of MC-MPPA-I and MC-MPPA-III, the internal MPPs are labeled from $MPP_{i(1)}$ to $MPP_{i(9)}$. The depths of the air chambers vary from $D_{(1)}$ to $D_{(9)}$, with the deepest chamber being termed $D_{\max}$. Specifically for MC-MPPA-I, each chamber possesses

an identical depth denoted as D . Additionally, each top/internal MPP is characterized by unique parameters for micro-perforation: the perforation diameter (d) and the porosity (ϕ), the latter being defined as the ratio of the micro-perforated area to the total area.

Table 5.1: Parameters used to characterize the acoustic behavior of the MC-MPPA-I and MC-MPPA-II.

MC-MPPA type	MPPs Location	MPPs Symbol	MPPs thickness	Perforation diameter	Perforation porosity	Perforation number	Sub-chamber depth
I	Top	$MPP_{t(1)}$	τ_t	$d_{t(1)}$	$\phi_{t(1)}$	$n_{t(1)}$	D
		$MPP_{t(2)}$		$d_{t(2)}$	$\phi_{t(2)}$	$n_{t(2)}$	
		$\vdots$		$\vdots$	$\vdots$	$\vdots$	
		$MPP_{t(9)}$		$d_{t(9)}$	$\phi_{t(9)}$	$n_{t(9)}$	
	Internal	$MPP_{i(1)}$	τ_i	$d_{i(1)}$	$\phi_{i(1)}$	$n_{i(1)}$	
		$MPP_{i(2)}$		$d_{i(2)}$	$\phi_{i(2)}$	$n_{i(2)}$	
		$\vdots$		$\vdots$	$\vdots$	$\vdots$	
		$MPP_{i(12)}$		$d_{i(12)}$	$\phi_{i(12)}$	$n_{i(12)}$	
II	Top	$MPP_{t(1)}$	τ_t	$d_{t(1)}$	$\phi_{t(1)}$	$n_{t(1)}$	$D_{(1)}$
		$MPP_{t(2)}$		$d_{t(2)}$	$\phi_{t(2)}$	$n_{t(2)}$	$D_{(2)}$
		$\vdots$		$\vdots$	$\vdots$	$\vdots$	$\vdots$
		$MPP_{t(9)}$		$d_{t(9)}$	$\phi_{t(9)}$	$n_{t(9)}$	$D_{(9)}$

5.3 Analytical Methods

As was mentioned, MC-MPPA-I and MC-MPPA-III have already been examined for normal acoustic incidence [55, 57]. In this Chapter, the state of the art is advanced by examining all three variations in grazing flow acoustic incidence. In order to achieve this, more advanced analytical models for the faceplate which take flow into account and which are Mach number dependent are used. The TpIM is employed, as before, using these models which allows the surface impedance to be calculated. In this Chapter, a Cremer impedance model is also incorporated into the optimisation algorithms. This sets the objective impedance to be the theoretical optimal impedance as defined by Cremer [128] and subsequent research advances. In addition, as the Cremer model defines an optimal impedance for a single frequency only, this Chapter presents a novel cost function which proposes a kind of “least-squares” approach with attempts to maximum absorption over a broad frequency range.

5.3.1 Impedance model of an MPP under grazing flow conditions

Grazing flows can significantly complicate the prediction of acoustic behaviors of MPPs due to the complex interplay between the acoustic waves and the flow field within and in close proximity to the perforation holes. Åbom et al. [129–132] investigated the effects of grazing flow on the impedance of MPPs. They also proposed analytical models for MPPs with circular and slit-shaped perforation holes under grazing flow conditions and successfully verified their accuracy through experimental validation.

The acoustic impedance of an MPP, Z_{mpp} , consisting of a real part, resistance ($\Re$) and an imaginary part, the reactance (χ), can be expressed as:

$$Z_{\text{mpp}} = \Re + i\chi \quad (5.1)$$

where i denotes the imaginary unit. Åbom et al. [129–132] have proposed a formula for the resistance and reactance of MPPs when the effects of grazing flow are taken into consideration:

$$\Re = \text{Re} \left[\frac{i\omega\tau}{c_0\phi} \left[1 - \frac{2}{\text{Sh}\sqrt{-i}} \frac{J_1(\text{Sh}\sqrt{-i})}{J_0(\text{Sh}\sqrt{-i})} \right]^{-1} \right] + \frac{2\alpha'R_S}{\rho_0 c_0 \phi} + \beta \frac{|M|}{\phi} \quad (5.2)$$

$$\chi = \text{Im} \left[\frac{i\omega\tau}{c_0\phi} \left[1 - \frac{2}{\text{Sh}\sqrt{-i}} \frac{J_1(\text{Sh}\sqrt{-i})}{J_0(\text{Sh}\sqrt{-i})} \right]^{-1} \right] + \frac{\delta\omega dF}{2c_0\phi} \quad (5.3)$$

where ρ_0 is the air density, and c_0 the speed of sound in air. The angular frequency, ω , is formulated as $\omega = 2\pi f$, with f signifying the frequency, measured in Hertz. J_n represents the Bessel function of n th order and μ symbolizes the dynamic viscosity of air. The Shear number is given by $\text{Sh} = d\sqrt{\omega\rho_0/(4\mu)}$ is . The resistive end-correction expression is $\frac{2\alpha'R_S}{\rho_0 c_0 \phi}$, where R_S is the surface resistance on one side of the plate, which is calculated by $R_S = 0.5\sqrt{2\mu\rho_0\omega}$. Moreover, the non-dimensional resistive and reactive end-correction coefficients α' and δ for the perforations with sharp edges are defined in Eqs. (5.4) and (5.5) as by Temiz et al. [133],

$$\alpha' = 5.08\text{Sh}^{-1.45} + 1.70 - 0.002/(\tau/d) \quad (5.4)$$

$$\delta = 0.97 \exp(-0.20\text{Sh}) + 1.54 - 0.003/(\tau/d) \quad (5.5)$$

The term $\beta \frac{|M|}{\phi}$ accounts for the linear grazing flow effect on the MPP's resistance which is a function of only Mach number and is independent of the frequency. The dimensionless

coefficient β , empirically defined by Åbom et al. [129–132], is given by Eq. (5.6).

$$\beta = 0.15 \pm 0.0125(\text{std}) \quad (5.6)$$

The term F represents the linear effect of the grazing flow on the MPP's reactance, [129, 131, 132], according to Eq. (5.7) .

$$F = \left(1 + (12.6 \cdot M)^3\right)^{-1} \quad (5.7)$$

Concerning the air cavities present within the analyzed MC-MPPA, the acoustic impedance, designated as Z_{ac} , for each cavity is calculated by Eq. (3.4),

The impedance model for the internal MPPs which are not exposed to flow are calculated using Eq. (3.1), as discussed in Chapters 3 and 4. This model is the same as that used by COMSOL when applying the IPP boundary condition. Therefore, that model will be used in the TpIM when modeling the impedance of the internal walls as it is equivalent to the COMSOL boundary condition which will be needed for the numerical work in Section 5.5. The IPP boundary condition has been shown to be sufficient as previous numerical studies [63] have validated that this IPP method closely aligns with a fully resolved representation of perforated holes in a COMSOL model.

5.3.2 Acoustic modeling of the MC-MPPA using TpIM

Figure 5.2 shows the investigated MC-MPPAs from Figure 5.1 represented schematically as bi-pyramid molecular structures. In this Chapter, TpIM is used to determine the impedance of the MC-MPPA.

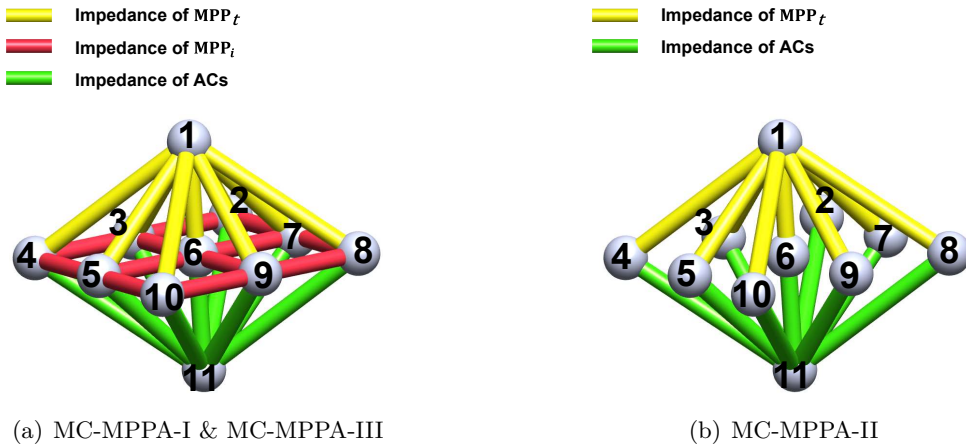


Figure 5.2: Molecular structure of impedances of the investigated MC-MPPA-I, MC-MPPA-II and MC-MPPA-III.

5.3.3 Exact Cremer impedance in rectangular lined ducts based on the classical boundary condition

The Cremer Impedance method published by, L. Cremer [128], can provide optimum propagational damping for a particular set of duct parameters. There is an assumption of local reaction i.e. a pure impedance boundary and so the frequency range has an upper limit related to the plan view geometric length scales of the MC-MPPA. For MC-MPPA-II, which does not have internal perforations, will have a higher critical frequency than MC-MPPA-I and MC-MPPA-III. The Cremer Impedance is commonly used in duct acoustic applications, such as for the design of mufflers for internal combustion engines [134–137]. Although it was originally formulated for rectangular ducts without flow, it has been further developed and successfully applied to situations with flow by Tester [138], for example.

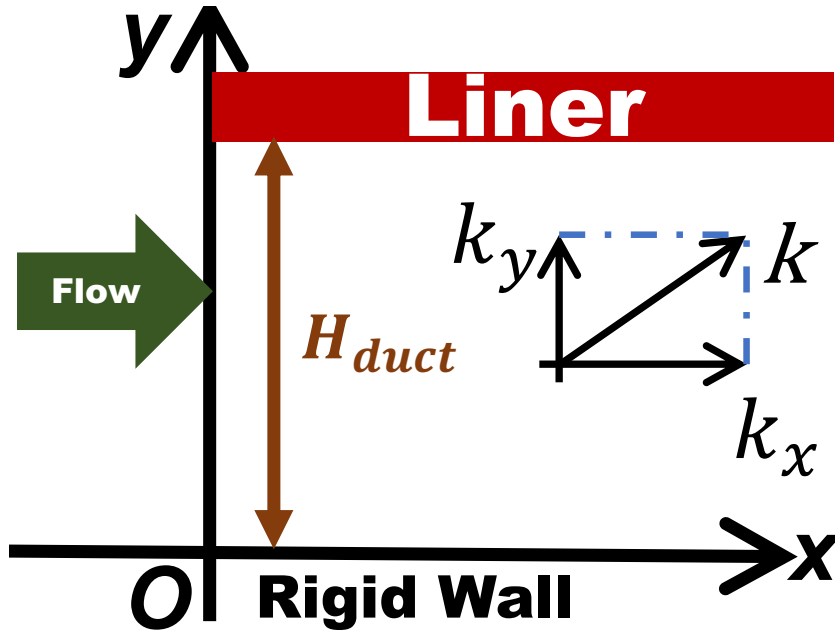


Figure 5.3: A schematic representation of a liner in a rectangular duct with a grazing flow.

Figure 5.3 illustrates the location of an acoustic liner of impedance Z on a single side of a 2-D infinitely-long rectangular duct, with a rigid wall on the opposite side of the duct. The height of the duct is H_{duct} . The orthogonal components of the wavenumber k are axial wavenumber k_x and transverse wavenumber k_y respectively. The grazing flow within the duct remains at a constant Mach number, M . The Cremer impedance was initially proposed by Cremer [128] for no-flow conditions and for rectangular ducts which would suit low-flow speed Heating, ventilation, and air conditioning (HVAC) analysis. Tester [138], motivated by

aeroengine applications, extended the theory for flow in circular ducts with a focus on high frequencies and well-cut on modes. The Cremer impedance can offer a reasonable estimate of the optimal impedance for a particular set of duct parameters (M , k_x , k_y and H_{duct}) [134–137]. The relationship between k_x and k_y can be expressed as :

$$k_x/k = -\frac{M}{1-M^2} \pm \frac{\sqrt{1 - (k_y/k)^2 (1-M^2)}}{1-M^2} \quad (5.8)$$

where $k = 2\pi f/c_0$ is the wavenumber [136, 139, 140]. It is necessary to select branch ($\pm$) of the root to ensure that the modes are damped in the positive x direction. The corresponding dispersion equation for the modes in a rectangular duct [136, 137, 139, 140] can be expressed as follows,

$$\left(1 - M \frac{k_x H_{\text{duct}}}{k H_{\text{duct}}}\right)^{-n} \cdot k_y H_{\text{duct}} \cdot \tan(k_y H_{\text{duct}}) = -ik H_{\text{duct}} \frac{\rho_0 c_0}{Z} \quad (5.9)$$

When the exponent $n = 1$ in Eq. (5.9) this corresponds to the classical boundary condition (Classical B.C.) [136, 139]. Alternatively, when $n = 2$, it corresponds to the Ingard-Myer boundary condition (IM B.C.) [136, 139]. The IM B.C. [137, 140–142], which postulates the continuity of normal acoustic displacement across a boundary layer of infinitesimal thickness is a widely used boundary condition in grazing flow rigs. However, this results in a negative real part of the impedance in the upstream direction at low frequencies, rendering the solution invalid. Åbom and Jacob [136, 139] noted that by replacing the IM B.C. with the Classical B.C., i.e., continuity of pressure and normal velocity can eliminate the aforementioned issue. Even if the IM B.C. is valid, the Classical B.C. is a more appropriate choice when the Helmholtz number kH is less than 1 [139]. Thus the Classical B.C. ($n = 1$) is adopted as the boundary layer condition in this current study.

When considering a normalized liner impedance Z , of Helmholtz number (kH), and Mach number M , the axial and transverse Helmholtz numbers $k_x H$ and $k_y H$ can be obtained by solving Eq. (5.9). According to Cremer [128] and Tester [138], to obtain an exceptional point where mode merging occurs, the first derivative of the dispersion equation must be zero at the root:

$$\frac{\partial}{\partial(k_y H_{\text{duct}})} \left[\left(1 - M \frac{k_x H_{\text{duct}}}{k H_{\text{duct}}}\right)^{-1} \cdot k_y H_{\text{duct}} \cdot \tan(k_y H_{\text{duct}}) \right] = 0 \quad (5.10)$$

Equation (5.8) and Eq. (5.10) are solved using the “fsolve” solver of Matlab with an asymptotic initial value $k_y H_{\text{duct}} = 2.11 + 1.13i$, as suggested by Zhang et al. [140], to obtain both the axial and transverse wavenumbers. The “exact” Cremer impedance to the complete

branch point equation is derived with the introduction of a mean flow related term, using Eq. (5.11),

$$Z_{\text{cr}}|_M = \frac{ik(1 - Mk_x/k)}{k_y \tan(k_y H_{\text{duct}})}. \quad (5.11)$$

This solution is the plane wave solution based on the Classical B.C. where the normalized Cremer impedance, which was normalized to the characteristic impedance $\rho_0 c_0$ in air, symbolized by $Z_{\text{cr}}|_M$, is the optimum propagational damping of a rectangular duct (see Figure 5.3) in the presence of a uniform, grazing flow at a certain Mach number. Figure 5.4 illustrates the theoretically predicted normalized exact Cremer impedance Z_{cr} for an infinitely long rectangular duct of height, H_{duct} , 40 mm as a function of Mach number, where the real component is denoted by "Re" and the imaginary component by "Im", based on the classical boundary condition. It should be indicated that, under conventional boundary conditions, the real component of the Cremer impedance remains positive even at low frequencies. These results are in agreement with the findings previously reported in Åbom and Jacob [136, 139].

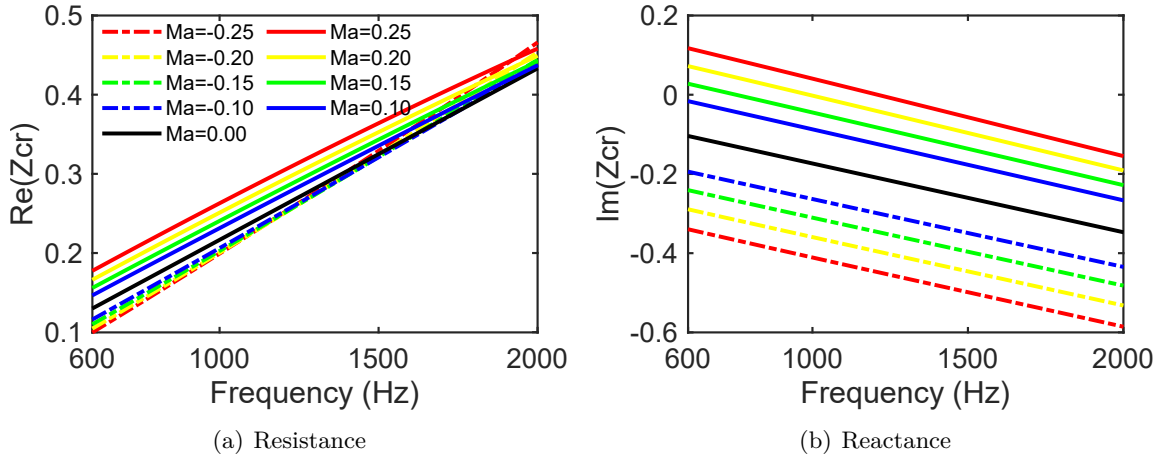


Figure 5.4: Normalized exact Cremer impedance Z_{cr} for an infinitely long rectangular duct of height $H_{\text{duct}} = 40$ mm as a function of Mach number.

5.4 Optimisation Strategy

In order to obtain the best possible performance from the MC-MPPA designs, i.e., a high absorption, broadband response, an optimisation routine was designed and conducted. However, the optimisation of the three configurations face three significant challenges: firstly, the structure of the MC-MPPA in general cannot be modeled acoustically using conventional

Equivalent Circuit Model techniques, secondly, the impedance of the faceplate MPP varies as a function of Mach number when a grazing flow is present, thirdly, optimising a liner for broadband absorption in a grazing flow duct poses substantial difficulties.

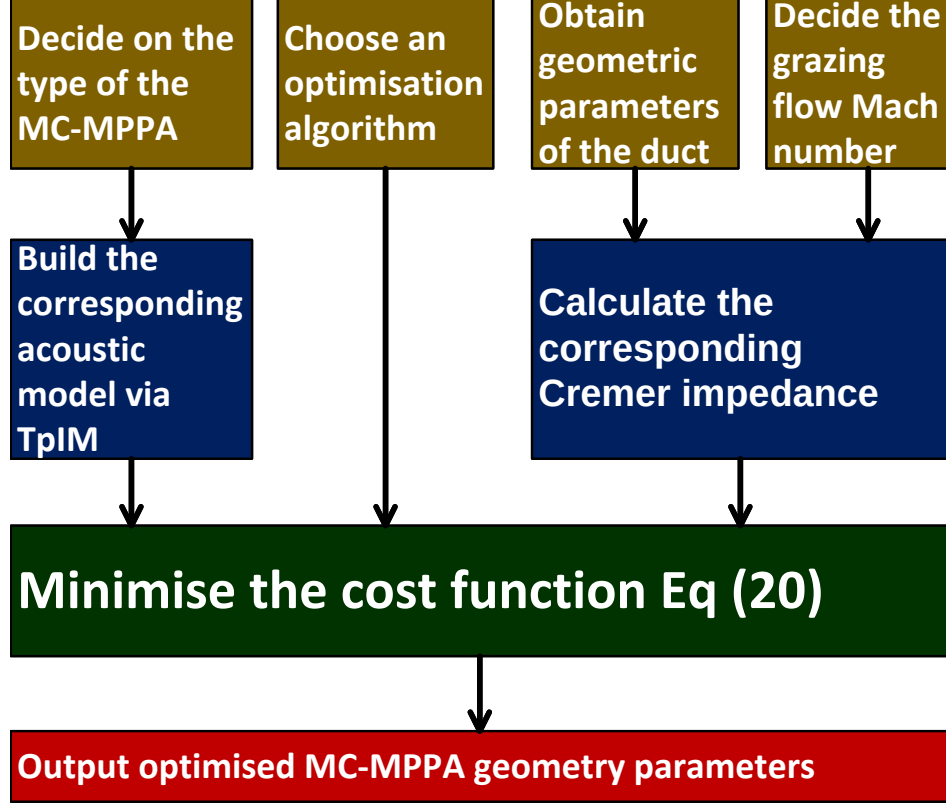


Figure 5.5: Flowchart of the optimisation strategy.

In view of the above challenges, the optimisation strategy shown in Figure 5.5 is proposed here. In the first step, the type of MC-MPPA to be studied is chosen and then the TpIM is applied using the analytical models provided in Section 5.3.1 to determine the overall impedance. In the second step, the duct geometry and the grazing flow Mach number are defined, such that the exact Cremer impedance can be calculated as described in Section 5.3.3. In the third step, the objective is to optimise the MC-MPPA such that its acoustic impedance matches the Cremer impedance for a specific frequency, as defined by Eq. (5.11), i.e.,

$$Z_{\text{MC-MPPA}} = Z_{\text{Cr}}. \quad (5.12)$$

While Eq. (5.12) is useful for determining the optimal design of the MC-MPPA for a specific target frequency, it may not be adequate to obtain a globally optimal solution across the entire frequency range of interest. To overcome this limitation, a new approach is proposed,

which is a kind of “least-squares” modification of Eq. (5.12) in the form of Eq. (5.13), to serve as a cost function for optimisation:

$$J = \frac{1}{\omega_2 - \omega_1} \int_{\omega_1}^{\omega_2} \sqrt{[\operatorname{Re}(Z_{\text{Cr}}) - \operatorname{Re}(Z_{\text{MC-MPPA}})]^2 + [\operatorname{Im}(Z_{\text{Cr}}) - \operatorname{Im}(Z_{\text{MC-MPPA}})]^2} d\omega \quad (5.13)$$

where $\omega_1 = 2\pi f_1$, $\omega_2 = 2\pi f_2$, while f_1 and f_2 are the lower and upper limits of the target frequency range respectively. Eq. (5.13) can be employed to ensure that the acoustic impedance of the MC-MPPA matches the Cremer impedance to the greatest extent possible across the frequency range $[f_1 \ f_2]$.

PlatEMO[143, 144] was chosen as an optimisation development platform. PlatEMO is an open-source environment developed for use with MATLAB for solving complex optimisation problems and provides a variety of built-in algorithms. After a period of evaluation, the Genetic Algorithm (GA) [145] was employed to minimize the cost function Eq. (5.13) and to determine the optimised geometric parameters for the MC-MPPAs. The geometric parameters considered by the optimisation routine are shown in Table 5.1. Furthermore, the search ranges and settings for these parameters during the optimisation process are listed in Table 5.2.

The output from the optimisation routine is the set of geometric parameters which physically describes the absorber, which, for a pre-defined maximum depth and Mach number, best absorbs the grazing flow duct noise in the frequency range of interest. Knowing the geometry, allows the MC-MPPA absorbers to be 3D printed and tested experimentally, as will be discussed in Section 5.6. In addition, the geometric specification also allows a COMSOL model representation of the liner to be assessed for performance and to calculate the transmission loss as will be discussed in Section 5.5.

Table 5.2: Parameter settings for optimisation.

Parameters	Value
$[d_{\min} \ d_{\max}]$	[0.5 - 1.0] mm
$[\phi_{\min} \ \phi_{\max}]$	[0 - 5]%
$[f_1 \ f_2]$	[600 - 2000] Hz
$[L, \ W]$	[50 mm, 40 mm]
$[\tau_t, \ \tau_i]$	[2 mm, 1 mm]
$D_{\max}$	10 mm and 30 mm
M	0.05, 0.10, 0.15, 0.20 and 0.25

5.5 Numerical Prediction Model

In the present investigation, a representative acoustic grazing rig was numerically modeled using the Acoustics Module of the COMSOL Multiphysics software framework. Flow effects are incorporated into the analysis by implementing the Mach number dependent impedance model, outlined in Section 5.3.1, for the faceplate MPPs which would be exposed to the flow in a grazing flow duct. Regarding the internal MPPs of the MC-MPPA, the COMSOL interior perforated thin panel boundary condition is used. A straight, rectangular duct featuring a cross-sectional dimension of 50 mm in width and 40 mm in height is examined, as depicted in Figure 5.6. This geometry is chosen to match the dimensions of the LAUM grazing flow duct that is used for the experiments described in Section 5.6. The lined section has a streamwise length of 200 mm. Five MC-MPPAs (of dimension $W = 50$ mm and $L = 40$ mm) are arranged in series along the duct in the liner location, resulting in a treatment that is 200 mm long and 50 mm wide, matching the dimensions of the duct's experimental test-section. This rig setup involves the introduction of a plane acoustic wave into the system at its upstream end defined as $p_0 e^{-ikx}$. To simulate this, a pressure wave with an amplitude, p_0 , of 1 Pa, is added as a Background Acoustics Field characteristic in a restricted domain, backed by a perfectly matched layer (PML). This facilitates the exit of reflected waves from the computational domain. To further prevent reflection and ensure the free exit of waves, a PML is also integrated at the downstream outlet. This setting is identical to that in Ref. [146]. In Figure 5.6, the direction of the incident wave is denoted by the red arrow. The orange domain in the figure illustrates the background acoustic field, whereas the yellow domain indicates the perfectly match layer. The transmission loss in the system is defined as

$$TL = 20 \log_{10} \left(\left| \frac{1 \text{ Pa}}{p_{out}} \right| \right) \quad (5.14)$$

where p_{out} is the integrated pressure at the outlet.

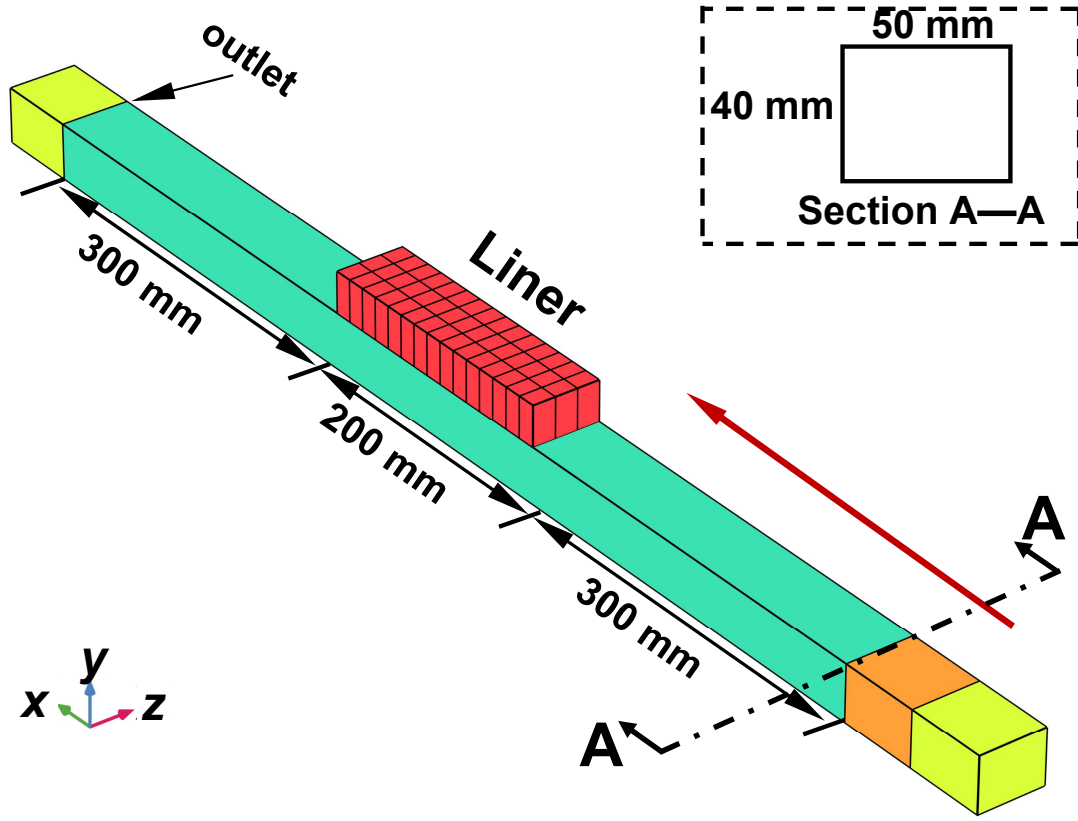


Figure 5.6: Sketch of the grazing acoustic incidence tube modeled in COMSOL.

5.6 Experimental Setup

5.6.1 Grazing flow facility at LAUM

A grazing flow test duct in the acoustics laboratory of Le Mans University (LAUM) was used for experimental validation of the analytical and numerical models, (see Figure 5.7) [147–150]. This rectangular duct has an internal cross-sectional area of 50 mm wide $\times$ 40 mm high. This Chapter focused on frequencies lower than the rig’s first cut-off frequency (around 3440 Hz), considering only plane acoustic waves that propagate to the test section. The optimised MC-MPPA was placed in the wall of a 200 mm-long test section.

Flow rates equivalent to mean velocities of up to Mach 0.25 were generated by a compressor located upstream of the test section. Incident acoustic waves with amplitudes up to 140 dB were generated by acoustic drivers installed either upstream or downstream of the test section. This allows the impedance to be examined for cases where the plane acoustic waves are traveling either with or against the direction of the flow. The excitation signal was a sinusoidal sweep from 200 Hz to 3500 Hz with a 10 Hz step, and the incident wave amplitude

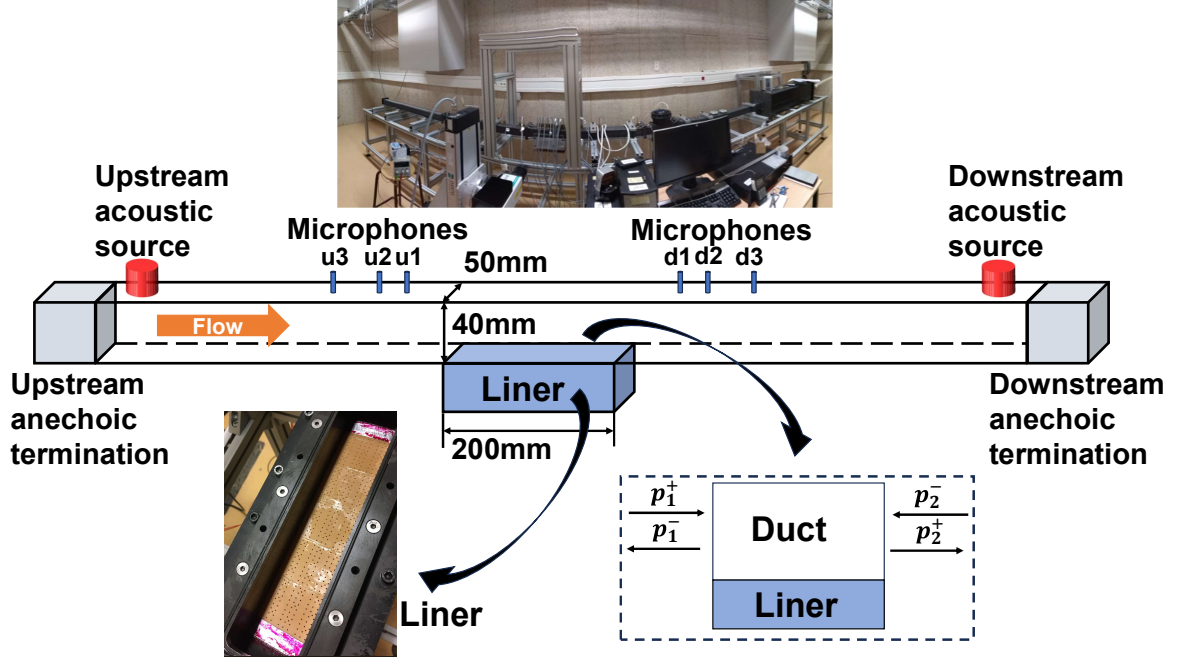


Figure 5.7: Schematic of the grazing flow facility at LAUM.

was kept constant at the selected level for the entire frequency range. For a specific MC-MPPA configuration examined, results were recorded as a function of Mach number, acoustic amplitude and acoustic wave direction.

Measurements were recorded from two sets of three flush-mounted microphones located both upstream and downstream of the test section. At each frequency of the sine sweep, the acoustic pressure at each microphone was determined by averaging the pressure readings over 200 cycles without flow and 1000 cycles with flow. From these acoustic data were then computed the transmission and reflection coefficients ($T^\pm$ and $r^\pm$), which constitute the scattering matrix $\mathbf{S}$ defined by,

$$\begin{pmatrix} p_d^+ \\ p_u^- \end{pmatrix} = \mathbf{S} \begin{pmatrix} p_u^+ \\ p_d^- \end{pmatrix} = \begin{bmatrix} T^+ & r^- \\ r^+ & T^- \end{bmatrix} \begin{pmatrix} p_u^+ \\ p_d^- \end{pmatrix} \quad (5.15)$$

where subscripts u and d represent upstream and downstream of the test-section respectively. The Transmission Loss (TL) in decibels (dB) is also calculated as $-20 \log |T^\pm|$. Finally, a local optimisation, which aims at minimizing the difference between the experimental scattering

matrix and results from multimodal computations, was performed [151, 152]. Then, the impedance that is found to minimize the mismatch between these quantities is considered as the effective sample impedance.

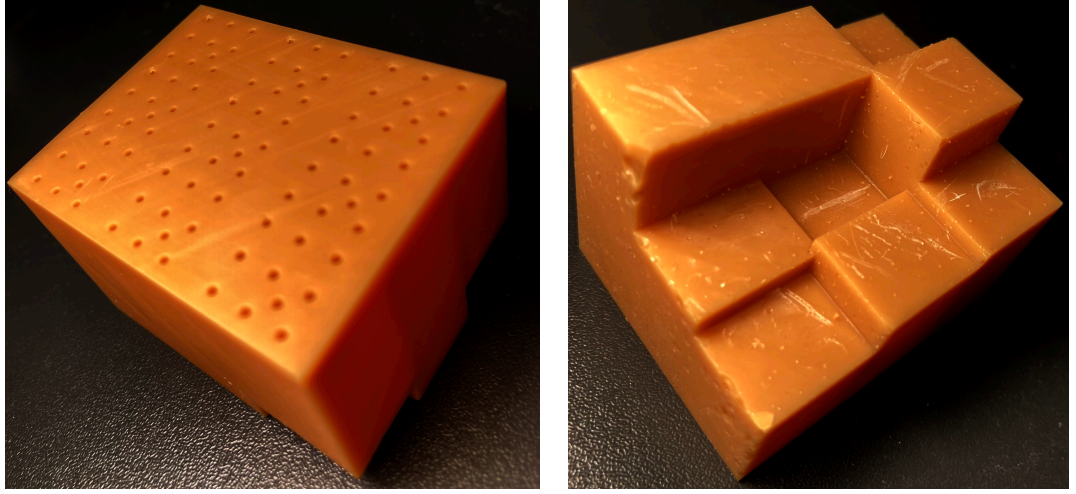
5.6.2 MC-MPPA liner samples

As discussed, five 3D printed MC-MPPAs are required for the 200 mm length liner test section. As each Mach number, maximum depth and frequency range requires a unique design, a rationalisation was required in order to reduced the number of tests and manufactured samples to a manageable number. As it is possible for MC-MPPA-II to achieve maximum cavity depth of all sub-chambers, if dictated by the optimisation, it was decided to remove MC-MPPA-I from the experimental test matrix. It was also decided to design samples only for the no flow case, $M = 0$, and the maximum velocity case, $M = 0.25$, which are considered to be the most interesting. In addition, the maximum depth was set to be 32 mm, including the thickness of the faceplate. Table 5.3 lists the four liner designs chosen for examination. The geometric parameters of tested MC-MPPA liner samples can be found in Tables E.9, E.10, E.11 and E.12. As each one requires five 3D printed blocks, that resulted in the requirement for twenty good quality prints in total. An example of one of the 3D printed blocks is shown in Figure 5.8.

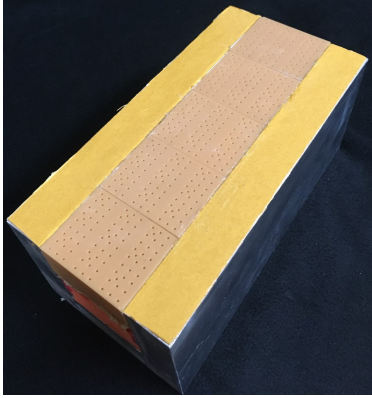
Table 5.3: Details of MC-MPPA liner samples chosen to be manufactured by 3D printing. $D_{\max} = 32$ mm.

Liner	MC-MPPA type	Target frequency range	Target Mach number	Geometric parameters	Predicted TL curve
II-00	MC-MPPA-II	[600 2000] Hz	$M=0$	Table E.9	red solid line in Figure 5.15(e)
II-25	MC-MPPA-II	[600 2000] Hz	$M=0.25$	Table E.10	green dashed line in Figure 5.15(e)
III-00	MC-MPPA-III	[600 2000] Hz	$M=0$	Table E.11	red solid line in Figure 5.15(f)
III-25	MC-MPPA-III	[600 2000] Hz	$M=0.25$	Table E.12	green dashed line in Figure 5.15(f)

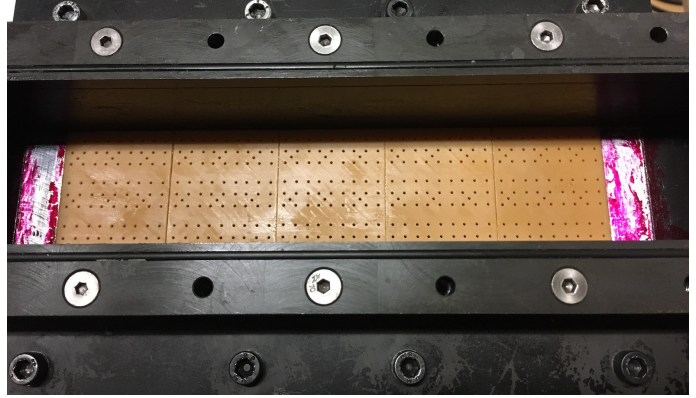
The 3D printing was conducted using an mSLA 3D printing technique with a standard photopolymer resin activated by a 405 nm wavelength light source. The mSLA 3D printer used in this Chapter was the Original Prusa SL1S SPEED. In order to ensure an airtight resonator other than airflow through the designed perforations, it was possible to print the entire MC-MPPA block as a single component with the faceplate integrated into the print.



(a) Isometric view (top) of MC-MPPA component with integrated faceplate within the II-00 liner (b) Isometric view (bottom) of MC-MPPA component with integrated faceplate within the II-00 liner.



(c) Photo of II-00 liner



(d) II-00 liner installed in the Grazing Flow Facility

Figure 5.8: Photographs of tested liners.

As part of the manufacturing calibration and quality control process, a number of initial blocks were dissected to inspect for trapped uncured resin. In addition, using a high powered microscope, the interior hole diameters and wall thicknesses were measured so that the 3D printed geometries could be compared to those designed.

In this Chapter, the search range for the optimisation algorithm of the hole diameters was set to be between 0.5 mm and 1.0 mm. To further assess the quality of the 3D printing, 3D printed holes were randomly selected for examination. The design diameters from the optimisation routine in these particular examples were 0.55 mm, 0.74 mm, and 1.00 mm. These holes were examined under the Dino-Lite Premier AD7013MT digital desktop micro-

CHAPTER 5. BROADBAND SOUND TRANSMISSION LOSS OF MC-MPPAS IN A GRAZING FLOW

scope to evaluate the printing accuracy. Although quite accurate, discrepancies are a reason for differences between the predicted and experimentally measured results.

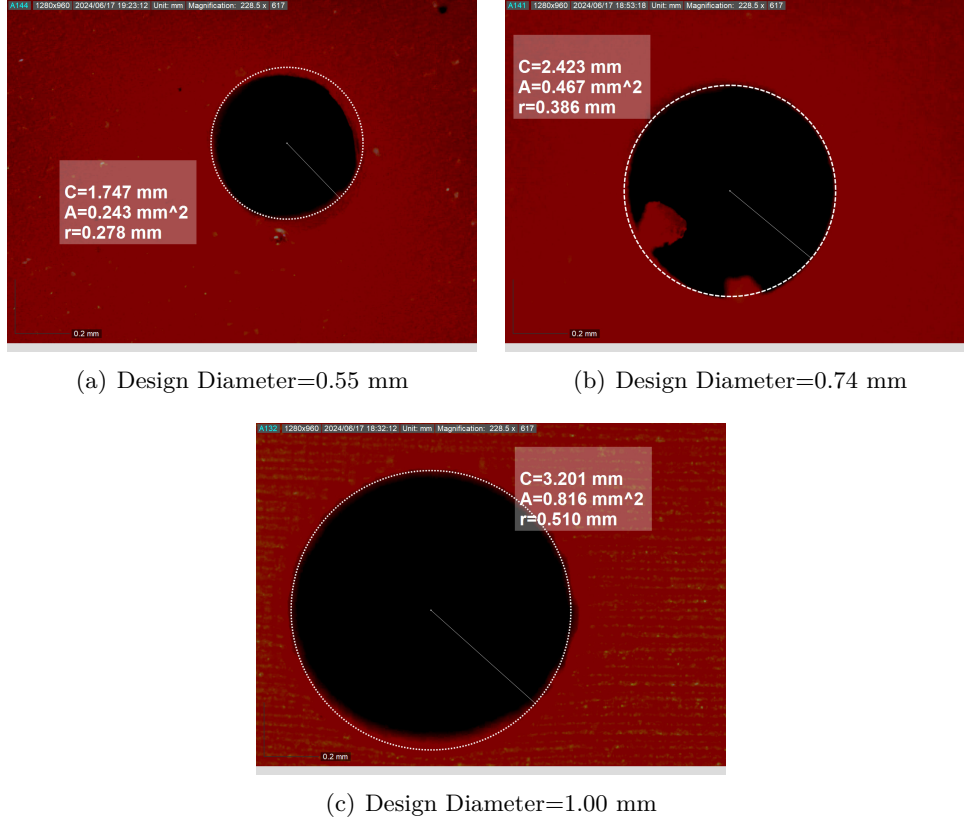


Figure 5.9: Microscope images of 0.55 mm, 0.74 mm, and 1.00 mm diameter holes from the 3D-printed test samples, where r represents the radius of the hole, A represents the area of the hole, and C represents the perimeter of the hole.

Additional quality control tests were conducted by measuring the performance of the MC-MPPA samples in normal incidence. Noise absorption coefficient tests were performed on each MC-MPPA specimen using a normal incidence impedance tube. The testing equipment and procedures adhered to the ISO 10534-2 standard [153], and the results are presented in Figures 5.10 - 5.13. The theoretical sound absorption coefficients were calculated using the two-point impedance method based on the geometric parameters of each MC-MPPA. As shown in Figures 5.10 - 5.13, the experimentally measured sound absorption coefficient curves align with the theoretical curves, indicating that the 3D printing quality is satisfactory and that the fabricated MC-MPPA samples meet the requirement. Although the emphasis of this chapter is on grazing flow, some observations on these results are still worth noting. First of all, it can be seen that there is very good repeatability between the test results for a specific

design, showing that the 3D printing process can provide consistent samples. Secondly, for the $M = 0$ absorption curves, effectively perfect absorption can be achieved over a range of 500 Hz which clearly demonstrates that this technology is superior to SDOF or DDOF solutions.

The 3D printed samples were assembled in a block of 5 and inserted into the Le Mans Grazing flow rig as shown in Figure 5.8. The experimental test matrix is presented in Table 5.4. Even though the absorbers were designed for optimal performance at either $M = 0$ or $M = 0.25$, all samples were tested at both of these Mach numbers as well as at $M = 0.15$ in order to see how they would perform outside of their design Mach number. In addition, for the $M=0$ case, the performance was recorded at incident acoustic wave amplitudes of 100 dB, 120 dB, 130 dB, and 140 dB. However, for $M = 0.15$ and $M = 0.25$, only sound levels of 130 dB and 140 dB were tested due to excessive flow noise, which would result in a poor signal-to-noise ratio at 100 dB and 120 dB.

To ensure locally reacting behavior of the liner, the backing cavity was divided into sub-cavities with length and width equal to approximately $\lambda/14$ (12 mm) and $\lambda/11$ (15.3 mm) respectively, at the highest testing frequency of 2000 Hz (where $\lambda = 171.5$ mm) [140].

Table 5.4: Grazing flow facility test matrix (Check marks indicate tests conducted at corresponding Mach numbers and sound pressure levels).

Mach number	without flow				$M=0.15$		$M=0.25$	
Sound pressure level	100 dB	120 dB	130 dB	140 dB	130 dB	140 dB	130 dB	140 dB
II-00 liner	✓	✓	✓	✓	✓	✓	✓	✓
II-25 liner	✓	✓	✓	✓	✓	✓	✓	✓
III-00 liner	✓	✓	✓	✓	✓	✓	✓	✓
III-25 liner	✓	✓	✓	✓	✓	✓	✓	✓

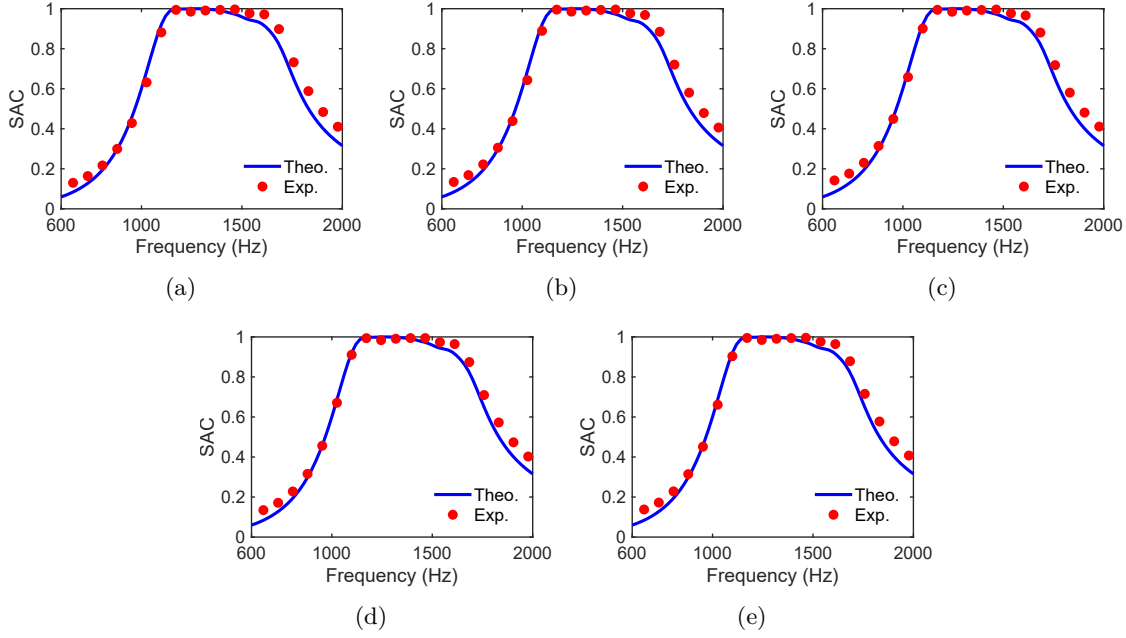


Figure 5.10: Sound absorption coefficients of the five individual MC-MPPA units constituting the II-00 liner under normal acoustic incidence.

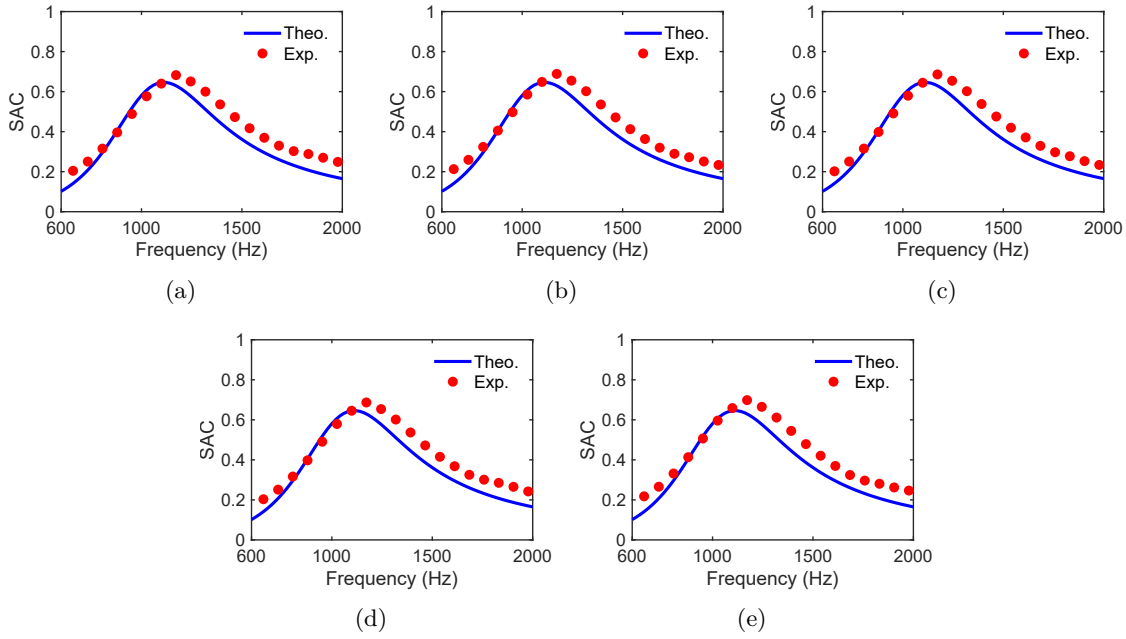


Figure 5.11: Sound absorption coefficients of the five individual MC-MPPA units constituting the II-25 liner under normal acoustic incidence.

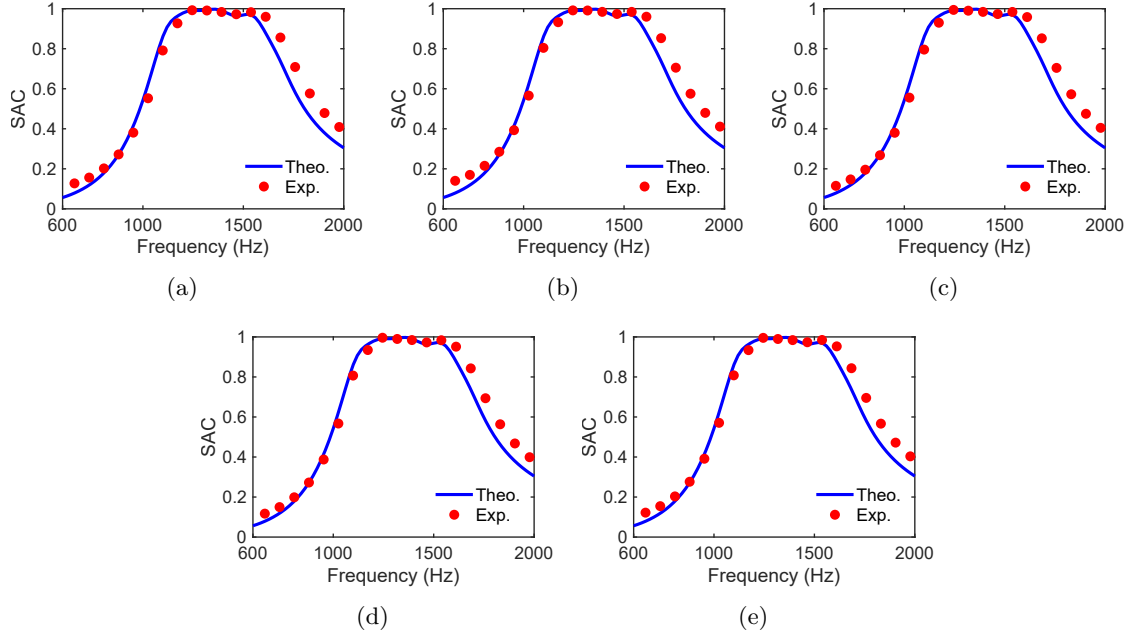


Figure 5.12: Sound absorption coefficients of the five individual MC-MPPA units constituting the III-00 liner under normal acoustic incidence.

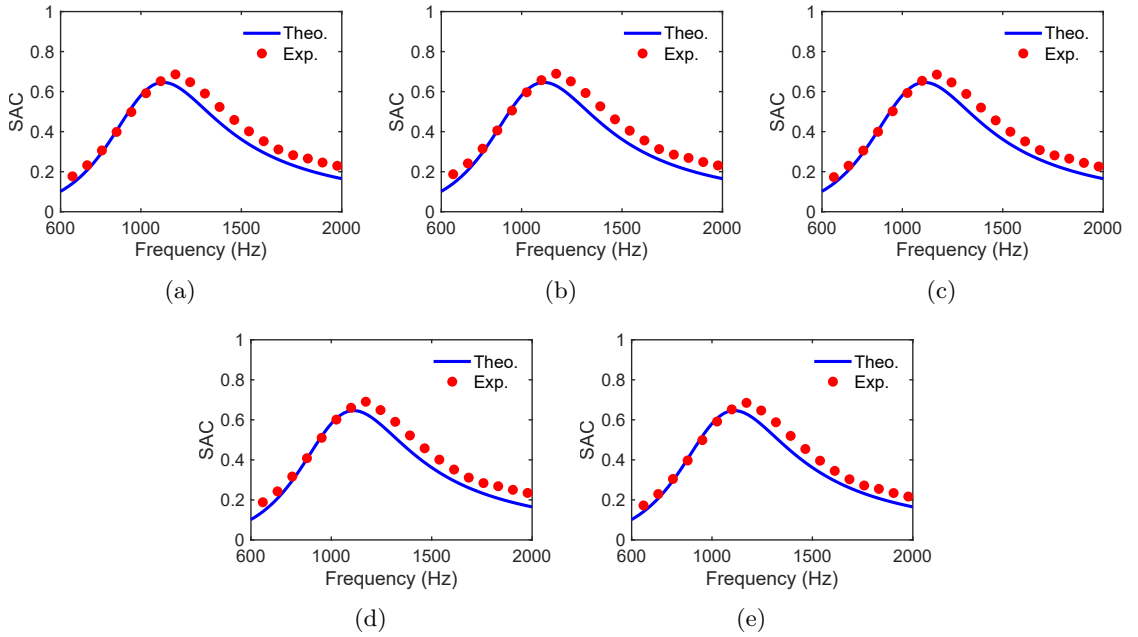


Figure 5.13: Sound absorption coefficients of the five individual MC-MPPA units constituting the III-25 liner under normal acoustic incidence.

5.7 Results and Discussion

5.7.1 Numerical results

The predicted Transmission Loss results are presented in Figure 5.15. As discussed, to arrive at these results, the analytical equations for impedance, Eq. (5.2) and Eq. (5.3), for a specific Mach number, were entered into the TpIM formulation, and the Cremer impedance-based cost function, at the same Mach number, was used in the optimisation routine in order to maximise the attenuation in the duct over the frequency range of interest for that Mach number. The overall impedance of MC-MPPA liner is attained once the optimisation routine has converged. The resistance and reactance components of this impedance is plotted in Figures 5.21 - 5.23 and are compared to the experimental results. In order to obtain a value for Transmission Loss so that the methodology can be validated against experimental results, a few approaches can be taken. In this Chapter, it was decided to create a COMSOL numerical model of the duct in 3D with the same geometry as the experimental grazing flow tube in Le Mans, and to calculate the Transmission Loss in this way. The outputs from the optimisation routine are the sub-cavity depths, the hole count and hole diameter per perforated surface in the MC-MPPA in question which satisfy the objective criteria. These parameters were then used on COMSOL in order to calculate the Transmission Loss across the liner. The subcavity depths were geometrically represented in the COMSOL model using Pressure Acoustics, whereas, due to the very large number of perforations of very small diameter, the porosity detail was applied to the perforated surfaces in the COMSOL model as boundary conditions. The top MPPs were modeled by entering equations Eq. (5.2) and Eq. (5.3) directly manually into COMSOL, whereas COMSOL's IPP boundary condition was used for the internal MPPs which were not exposed to the flow. COMSOL was then used to calculate the transmission loss across the liner. At this stage, a different Mach number can be used with these equations in COMSOL. This allows the performance of a liner designed for a specific Mach number to be assessed at a different Mach number as was conducted experimentally, see Table 5.4. Two examples of the COMSOL solution are given in Figure 5.14.

Figure 5.14 provides two examples of the solution from COMSOL. The actual geometry of the grazing flow duct and of the individual sub-chamber cavities, which are seen to have different depths, are modeled. Impedance boundary conditions are applied to the perforated surfaces.

The example in Figure 5.14(a) corresponds to the III-00 liner, whose design is optimised for quiescent flow and which is examined in the COMSOL model for $M = 0$. The corresponding transmission loss plot is shown in Figure 5.21(g), where the numerical solution predicts a maximum TL attenuation of approximately 25 dB at 1350 kHz. Figure 5.14(b) shows the same liner but which is instead examined in the COMSOL model at $M = 0.25$. The TL plot for this case is provided in Figure 5.23(g), where the numerical solution predicts a maximum TL attenuation of approximately 13 dB at 1550 kHz. In both cases, the downstream sound pressure levels are attenuated at the respective resonance frequencies.

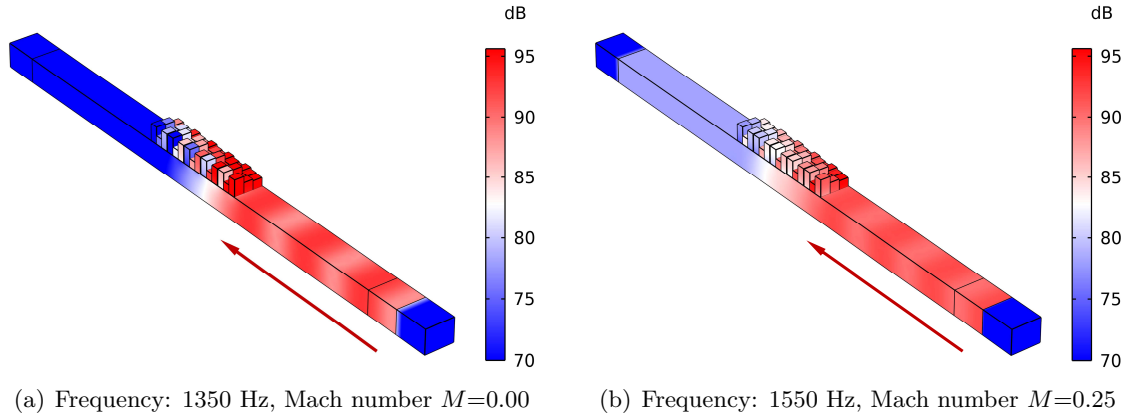


Figure 5.14: COMSOL numerical solutions showing the sound pressure levels in the III-00 liner and in the grazing flow duct. The direction of the incident wave is indicated by the red arrow.

Figure 5.15 presents the results from a liner composed of five blocks, similar to that illustrated in Figure 5.14, for all three MC-MPPA variants for two $D_{\max}$ depths and for a range of Mach numbers between 0 and 0.25. In this figure, the Mach numbers used at the COMSOL stage are the same as the design Mach number. The frequency range examined was between 600 Hz and 2000 Hz. The general observation is that, as would be expected, an increase in Mach number results in a decrease in TL. For the shallower design, $D_{\max} = 10$ mm, an increase in Mach number tends to also narrow and shift the response to higher frequencies, whereas for the deeper absorber, $D_{\max} = 30$ mm, the frequency response also narrows but remains centered or perhaps decreases slightly. Despite this, max/min TL values of 20 dB/10 dB are predicted for the absorber with a subchamber depth of only 10 mm, which increases to max/min TL values of 27 dB/15 dB when the $D_{\max}$ is increased to 30 mm. The exception to these trends are the results of MC-MPPA-I at $D_{\max} = 30$ mm. In that case, Mach numbers of $M = 0$ and $M = 0.05$ result in a narrowing of response and in a very

high TL of almost 80 dB for $M = 0$ to be achieved. As the optimisation seeks to broaden the response, it seems clear that, for this variant, the different response is due to the fixed cavity depth demonstrating the benefit of the variability in the other two designs. However, it is promising that such a very high TL “might” be possible with such a shallow depth in MC-MPPA-I. The term “might” is used as unfortunately this configuration was not assessed experimentally but might be further investigated in future work.

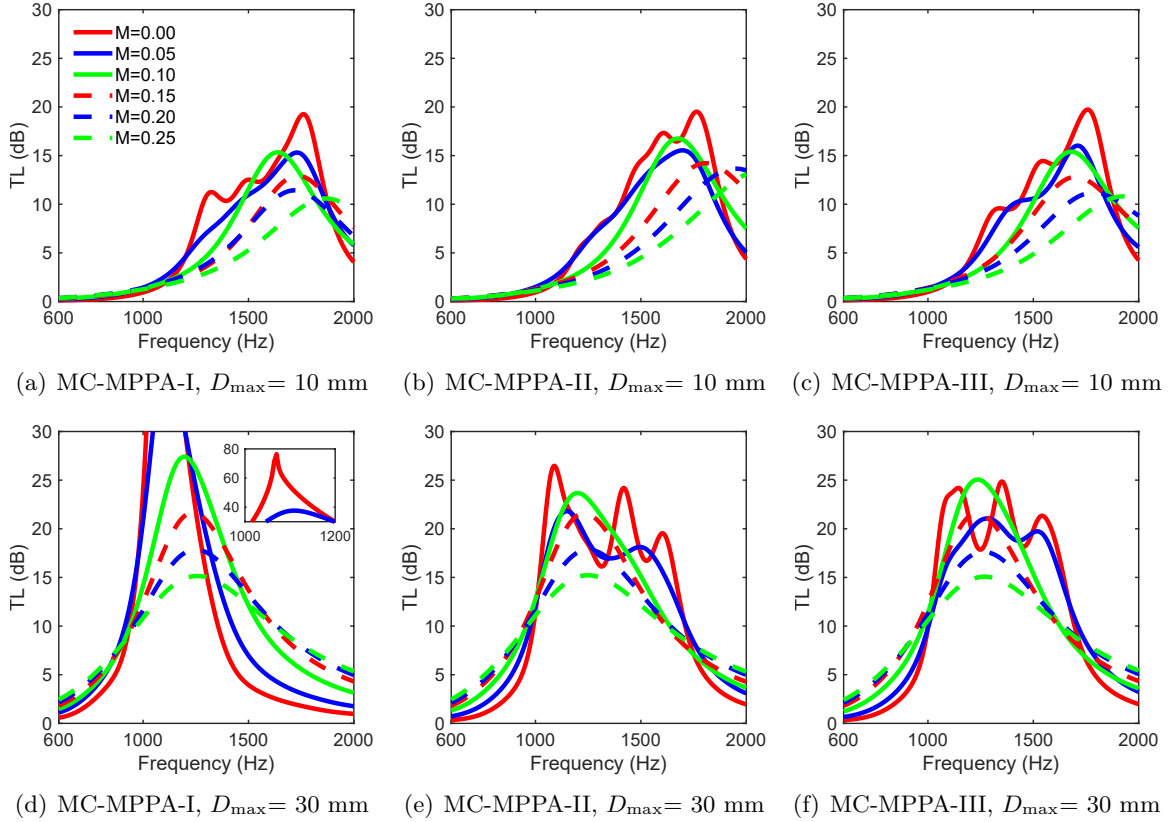


Figure 5.15: Predicted sound transmission loss (in dB) of MC-MPPA-I, MC-MPPA-II, and MC-MPPA-III with a maximum sub-chamber depth of 10 mm and 30 mm in the presence of grazing flow at varying Mach numbers.

5.7.2 Experimental characterisation of MC-MPPA-II and MC-MPPA-III. $M = 0$, transmission loss and impedance measurements. Non-linear effects.

Figure 5.16 presents the experimentally measured TL results of the four liners tested, see Table 5.4, in the grazing flow rig but in quiescent flow conditions $M = 0$. Figures 5.16(a) and 5.16(c) are the two liners designed for no flow whereas Figures 5.16(b) and 5.16(d) are liners designed for $M = 0.25$ but applied to the no flow case. They are therefore outside their design conditions. In all cases, the TL changes with the incident acoustic level such that an

increase in the latter results in a lower transmission loss. For the two no-flow designs, the responses are quite similar to each other, with TL values ranging between 17 dB and 27 dB. They are also more broadband compared to the liners operating outside their design Mach number. However, although more narrowband, the transmission loss responses at quiescent flow of the liners designed for $M = 0.25$ reach very high values, above 60 dB for liner III-25. This can be seen as a very beneficial consequence.

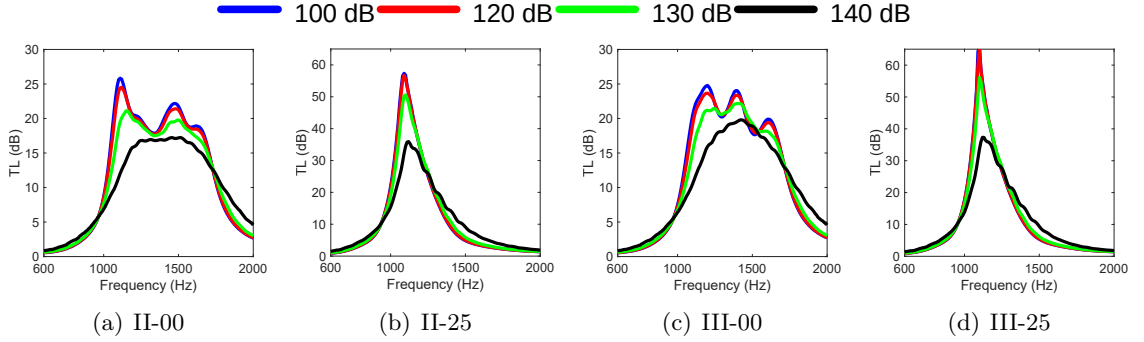


Figure 5.16: Experimental results. $M = 0$. TL as a function of SPL. Liners II-00, II-25, III-00, and III-25.

Figure 5.17 presents the experimentally measured impedance results for the same liners, also for the no-flow case. Here it can be observed that the resistance seems to be much more sensitive to SPL compared to the reactance which seems to vary very weakly. The increase of resistance due to the increase in SPL corresponds to the fact that the air oscillating in the holes takes, for high acoustic velocities, the form of a jet flow and vortex rings. These structures quickly convert acoustic energy into hydrodynamic energy through viscosity, creating an additional nonlinear dissipative effect [154]. It can be observed that for the liners designed for the no flow case, their resistance is approximately equal to one in the frequency range where the transmission loss is a maximum. On the other side, the reactances tend to flatten out around zero at that frequency range, explaining the broadband behavior of these liners. Looking at the reactance values for the other two sample, the sharp peak in the transmission loss is found to appear when the reactance is zero, and its great value is associated to a low resistance value, probably close to the optimal impedance at this particular frequency.

Influence of Mach number, acoustic source location and of non-linear effects. Transmission loss and impedance measurements.

Figure 5.18 provides the experimentally determined transmission loss for several Mach number values. As was stated, flow measurements were only taken at SPL values of 130 dB and 140

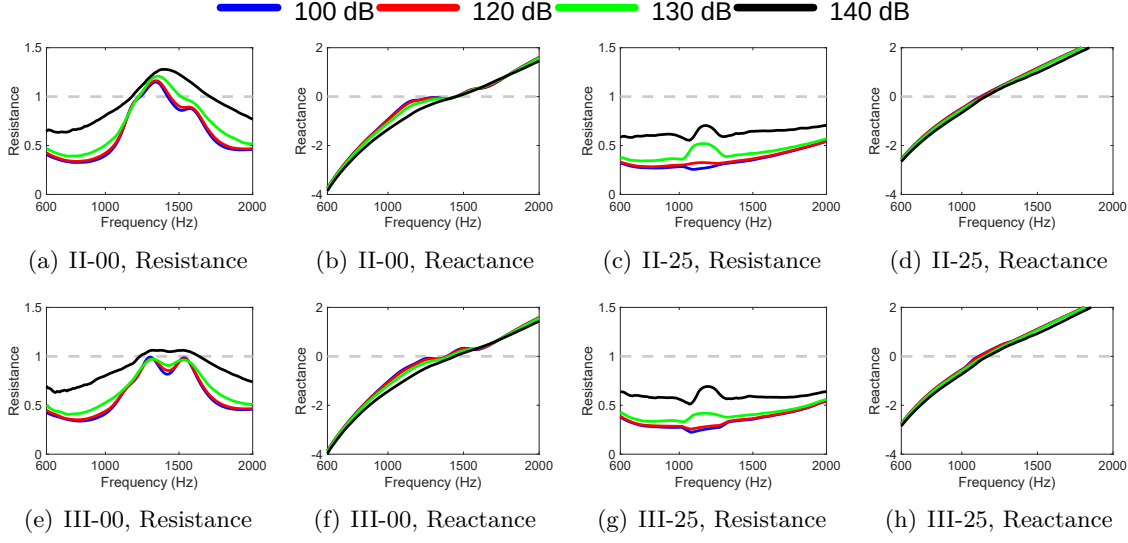


Figure 5.17: Experimental results. $M = 0$. Educed normalized impedance as a function of SPL. Liners II-00, II-25, III-00, and III-25.

dB in order to ensure a sufficient signal-to-noise ratio. In the figure, the quiescent results at these SPLs are also plotted for reference. On the one hand, measurements were achieved when the acoustic source was upstream of the liner, meaning that the acoustic waves were traveling in the same direction as the flow. In addition, a second set of tests were performed when the acoustic source was located downstream of the liner, with its corresponding acoustic waves traveling upstream against the flow. When there is flow, i.e. for $M = 0.15$ and $M = 0.25$, it can be seen that there is little difference in the transmission loss between 130 and 140 dB at a specific Mach number. This observation is made irrespective of the location of the acoustic source, and agrees with previous studies where flow effects have been shown to overcome the nonlinear effects associated to high acoustic velocities [155].

When the acoustic source is upstream, an increase in Mach number tends to decrease the transmission loss and broadens the response, albeit at a lower amplitude, and shifts the frequency response to higher frequencies. When the acoustic source is located downstream of the liner, the results are quite different. Indeed, it can be seen in Figure 5.18 that, for the II-00 and III-00 liners, an increase in flow speed does not induce a transmission loss decrease. In fact, at 140 dB, an increase is even noticed. A broadening effect also results in an increase in TL at the lower and higher frequency ranges. For the II-25 and III-25 liners, there is a decrease in transmission loss with increasing Mach number, compared to the very high values measured with no flow, but the peak levels are higher than those of the II-00 and III-00 samples. In addition, when the acoustic source is downstream, there is not the

same shift in frequency range compared to the upstream case. With respect to TL levels, at maximum SPL and Mach number, a value of approximately 10 dB is achieved when the acoustic source is upstream. In contrast, when the acoustic source is located downstream, the TL level ranges between 20 and 25 dB, which can be considered a significant attenuation at these difficult conditions.

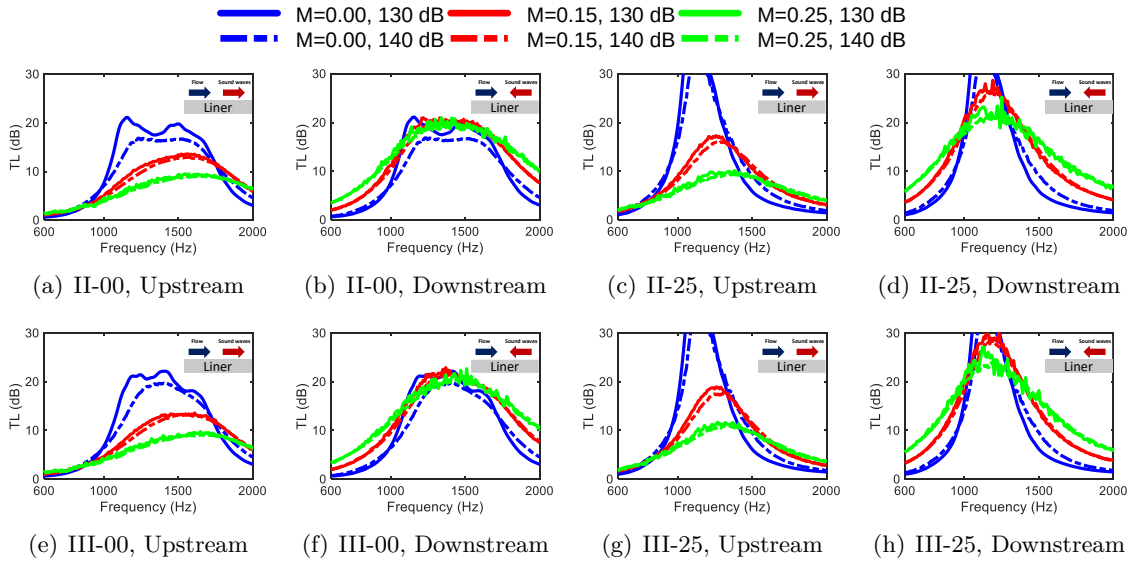


Figure 5.18: Experimental results. $M = 0$, $M = 0.15$, $M = 0.25$. TL as a function of Mach number and of SPL. Liners II-00, II-25, III-00, and III-25. Upstream: acoustic waves in the direction of flow, Downstream: acoustic waves against flow.

Figures 5.19 and 5.20 provide the corresponding resistance and reactance values. For the resistance values, it can be seen that, once more, there is only a little difference in value between the 130 dB and 140 dB results in presence of flow. In both cases, irrespective of the location of the acoustic source, an increase in Mach number results in an increase in resistance. However, the values are higher when the acoustic source is located downstream. Also to be noted is that an increase in Mach number tends to flatten out the resistance as a function of frequency, because of the vanishing of the nonlinear effects at the liner resonance. For the reactance values, an increase in Mach number tends to decrease the values slightly, and once more the II-00 and III-00 values tend to flatten slightly at zero in the frequency range corresponding to maximum TL.

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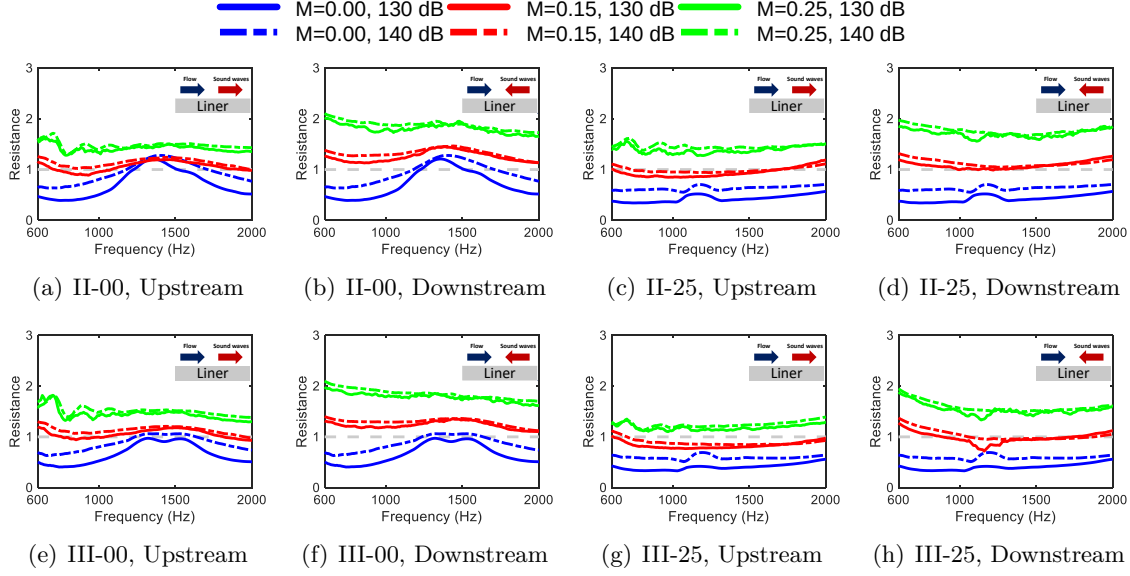


Figure 5.19: Experimental results. $M = 0$, $M = 0.15$, $M = 0.25$. Resistance as a function of Mach number and of SPL. Liners II-00, II-25, III-00, and III-25. Upstream: acoustic waves in the direction of flow, Downstream: acoustic waves against flow.

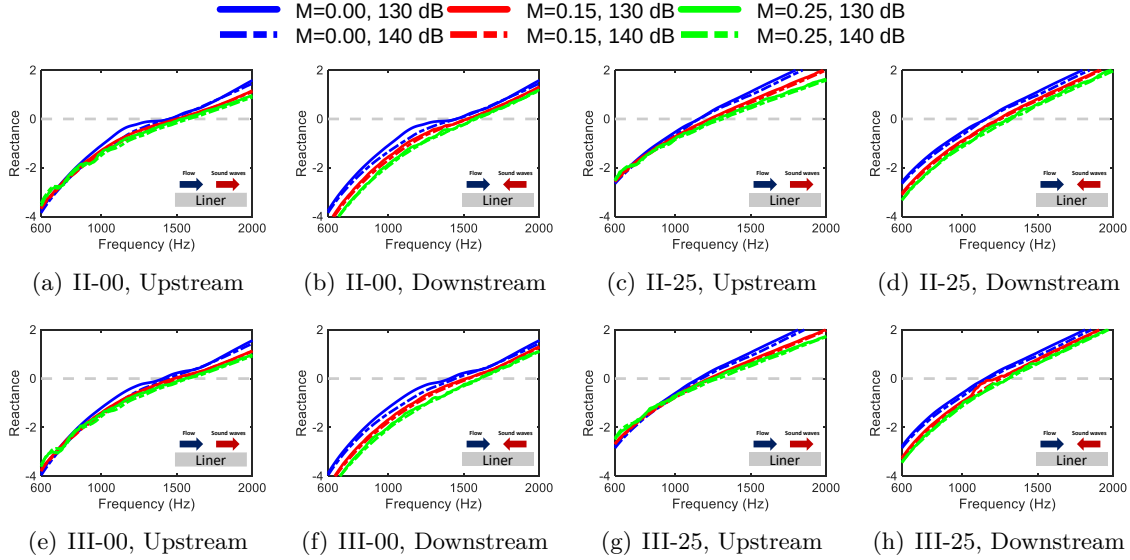


Figure 5.20: Experimental results. $M = 0$, $M = 0.15$, $M = 0.25$. Reactance as a function of Mach number and of SPL. Liners II-00, II-25, III-00, and III-25. Upstream: acoustic waves in the direction of flow, Downstream: acoustic waves against flow.

5.7.3 Experimental validation of the analytical model-optimisation method-COMSOL model methodology.

The liners tested in the experimental campaign were based on MC-MPPA technologies which were designed using analytical grazing flow models, a two-point impedance method and a novel Cremer impedance optimisation routine made to achieve a broadband response. Fig-

ures 5.21 - 5.23 provide plots comparing predicted and experimental results with the aim of validating the design methodology.

In Figure 5.21 are presented the results for the no-flow case $M = 0$. At 100 dB, it can be observed that in the left column a remarkably good agreement between the predicted and the experimental TL results. For the II-00 and III-00 liners, which are operating at their design Mach number, the amplitude match is effectively perfect and the broadband range is well captured, with only very small differences in peak frequency locations. For the II-25 and III-25 liner results, once again the agreement is extremely good. When looking at the 140 dB results, it can be observed that, as in Figure 5.16, the experimental result has a lower TL value than the predicted result. This is as expected, as no attempt is made to model any nonlinear behavior in the analytical models for the MC-MPPA. This is however an area for future investigation. The agreement at 100 dB also, therefore, implies that 100 dB is in the linear operation range for this absorber technology.

Figure 5.21 also displays a comparison between predicted and experimental results for impedance. Once more, It can be observed that the predicted results agree well, even if the resistances deviate slightly from the measured ones at the lowest and highest frequencies. There is a better agreement in the reactance results, and this quantity is also less sensitive to the SPL, as expected.

In Figures 5.22 and 5.23 are shown the validation plots when there is a flow, at $M = 0.15$ and $M = 0.25$. As no results are available at 100 dB for the flow cases, no “linear” result exists against which a comparison can be made. However, It can be observed that the 140 dB experimental result for TL is broadly similar to the predicted case with an amplitude drop similar to that seen when comparing the 100 dB and 140 dB results in Figure 5.16. Therefore, the main explanation for the discrepancy in TL is the nonlinear behavior of the absorber at high SPL. With regards to the impedance values, the agreement between the predicted and measured values at 140 dB is again remarkably good with discrepancies mainly at the lower frequencies. It should be noted that predicted results are only compared to the experimental results when the acoustic source is located upstream of the liner, as there is no accommodation in the analytical model for a counter-direction acoustic field.

The liners tested in the experimental campaign were based on MC-MPPA technologies which were designed using analytical grazing flow models, a two-point impedance method and a novel Cremer impedance optimisation routine for a broadband response.

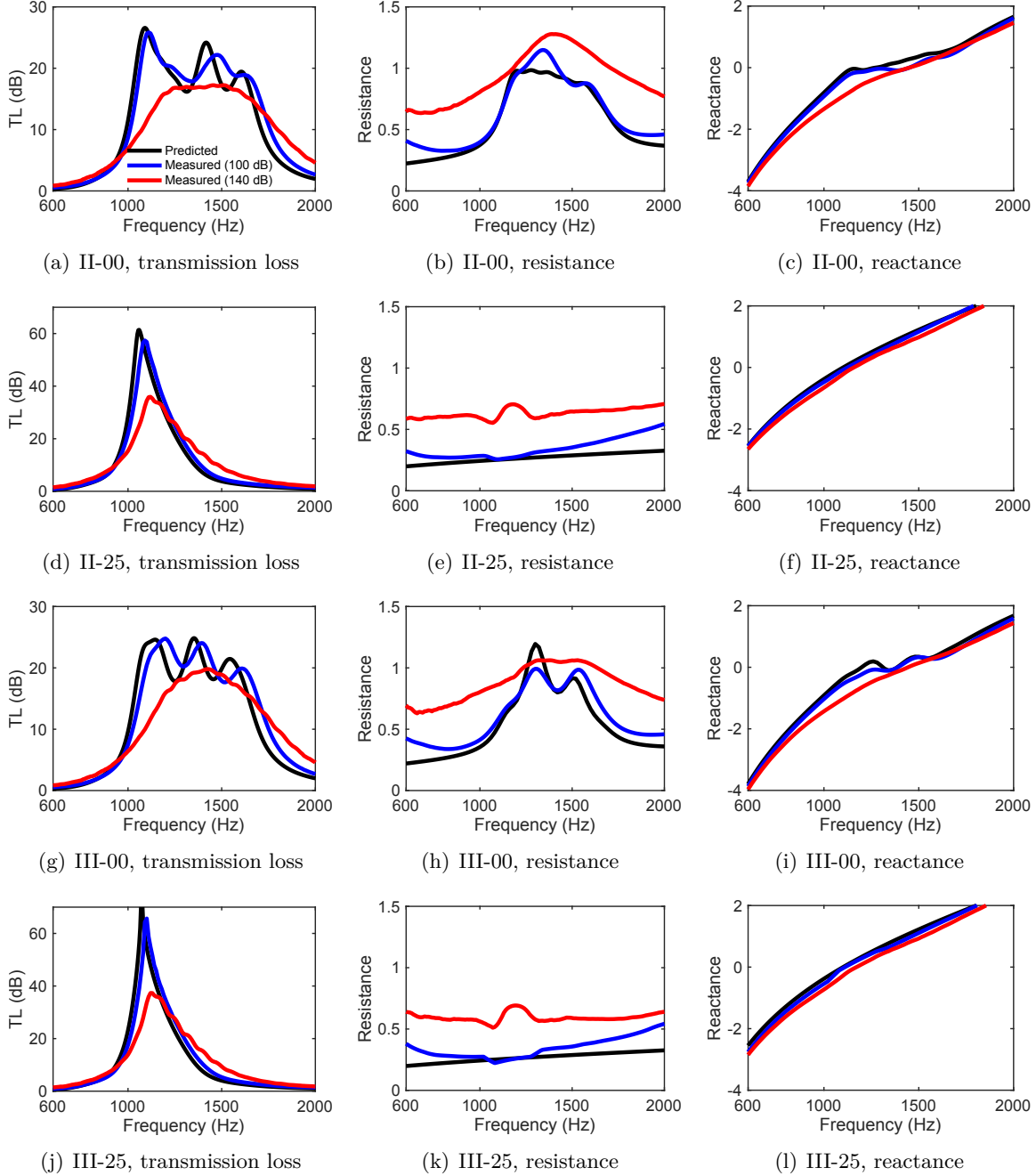


Figure 5.21: Validation results. $M = 0$. Comparison between predicted and experimental results for TL and impedance.

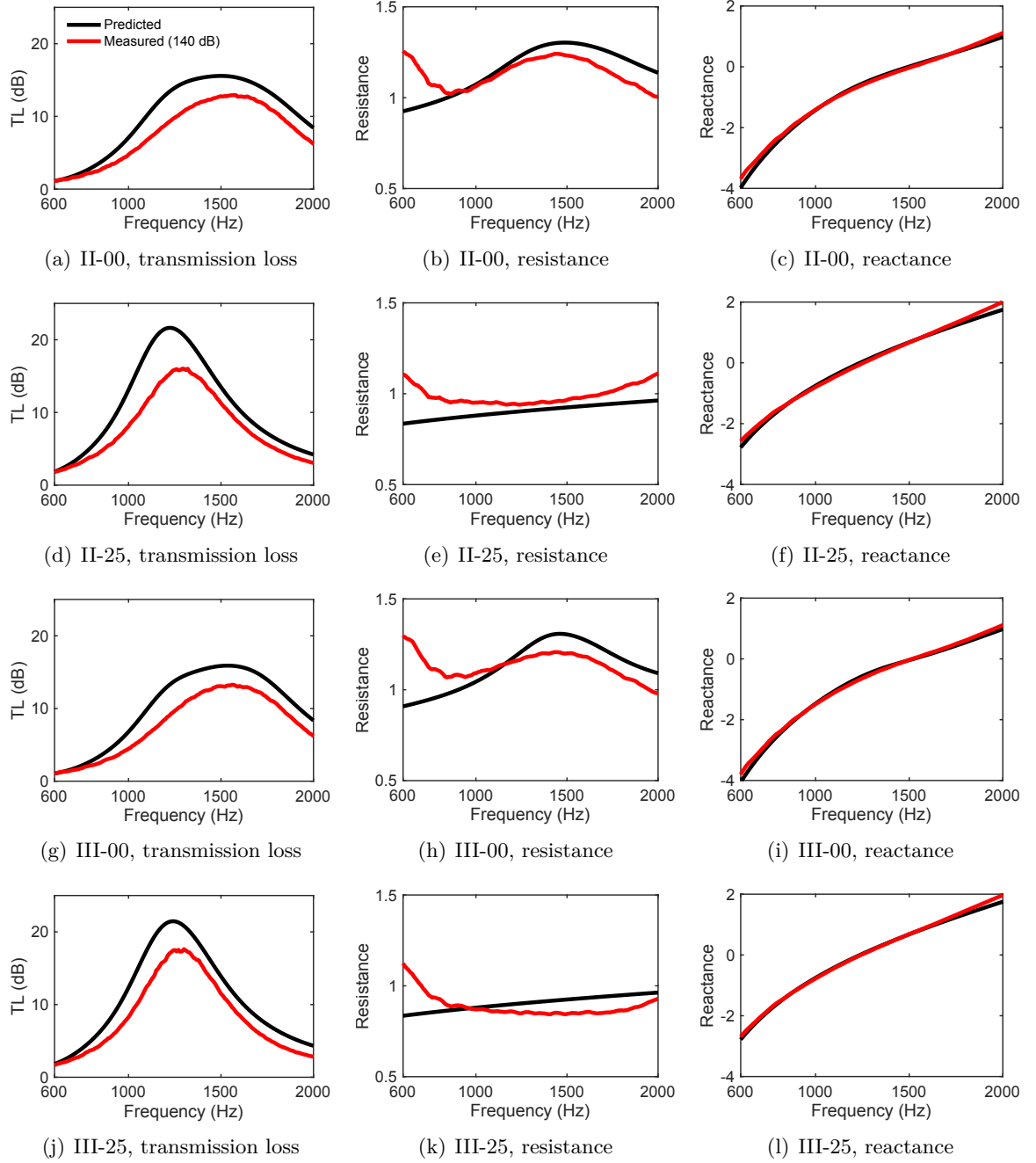


Figure 5.22: Validation results. $M = 0.15$. Comparison between predicted and experimental results for TL and impedance.

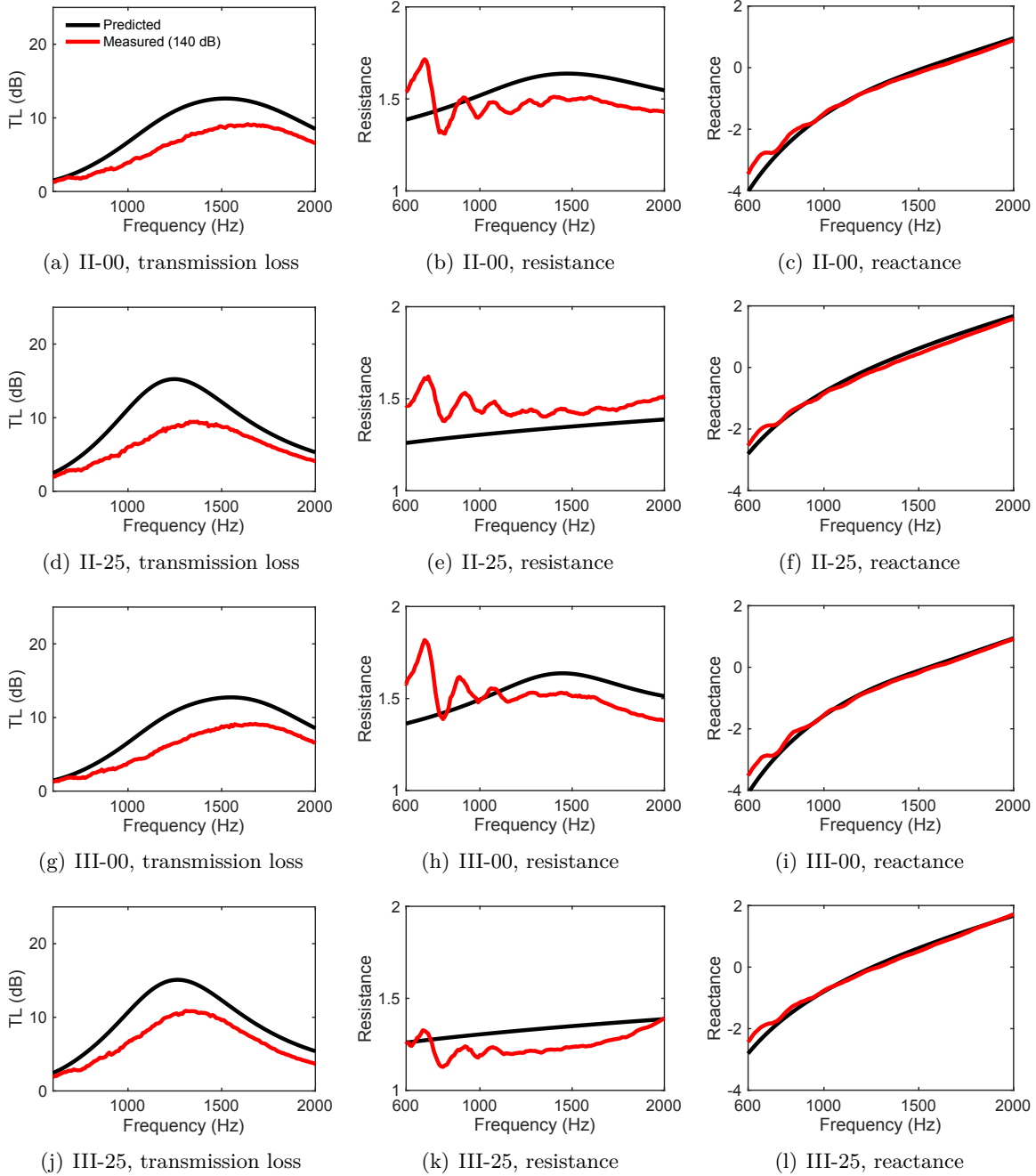


Figure 5.23: Validation results. $M = 0.25$. Comparison between predicted and experimental results for TL and impedance.

5.8 Summary

This Chapter introduces a significant advancement in micro-perforated panel absorber technology for broadband sound transmission loss under grazing flow conditions. MC-MPPA variations are designed using analytical grazing flow models, a graph-theory-based two-point impedance method and a novel Cremer impedance optimisation routine, to enable optimisation of the MC-MPPA for maximum sound transmission loss within a selected broadband frequency range under grazing flow conditions. Experiments were conducted at the LAUM grazing flow facility in Le Mans to investigate the effects of grazing flow up to $M = 0.25$ on the acoustic performance of the MC-MPPA. Furthermore, nonlinear effects were examined by varying the acoustic excitation amplitude (up to 140 dB), and the influence of the acoustic source location relative to the liner was also studied. The main conclusions of this Chapter are as follows:

- (1) The integration of the two-point impedance method, an analytical grazing flow model for an MPP and the Cremer impedance enables the design and optimisation of an MC-MPPA for grazing flow acoustic conditions. Additionally, a novel “least-squares” type cost function is developed for optimisation under grazing flow or grazing acoustic incidence conditions, facilitating the identification of an optimum MC-MPPA design across a broad frequency range. This methodology was validated with experimental results.
- (2) High accuracy manufacture of the liner test samples was achieved with the mSLA 3D printing. The complex components with submillimeter holes, even in internal walls, were successfully printed as a single part with an integrated face-plate. Destructive measurement techniques showed accurate geometries, no trapped uncured resin and normal incidence measurements verified accurate performance.
- (3) The experimental results demonstrate that the three variations behave either similarly or quite differently from each other depending on the test conditions. As different manufacturing techniques are required for them all, this allows the most appropriate variation to be chosen to suit the application and resources.
- (4) When the liner is operating at its design Mach number, TL levels of between 17 dB - 27 dB are obtained in a grazing (no-flow) acoustic incidence condition over approximately

a 700 Hz bandwidth below 2 kHz at 100 dB. Higher SPLs reduce the TL demonstrating a nonlinear behavior. However, TL levels of 15 dB over approximately a 500 Hz bandwidth below 2 kHz are achievable at 140 dB.

- (5) When the liner is operating outside its design condition, in the no-flow case, the TL response is narrower but at much higher levels: 37 dB at 140 dB and up to 65 dB at 100 dB which is a very advantageous consequence.
- (6) In the grazing flow case, TL levels of at least 10 dB are achieved at 140 dB, even when operating outside its design flow conditions when the acoustic source is upstream of the liner. When a liner designed to operate at $M = 0.25$ is exposed to $M = 0.15$, its performance greatly improves to values of up to 15 dB, which is another beneficial consequence.
- (7) When the acoustic source is downstream of the liner, results are even better when there is flow. TL levels of up to 25 dB can be achieved at 140 dB, indicating the present concept to be a suitable inlet liner for an aeroengine where the sound propagates against the flow.
- (8) Resistance values of the impedance are seen to increase and flatten with frequency with increasing Mach number irrespective of the location of the acoustic source, whereas the reactance tends to decrease marginally.
- (9) Experimental results match the TL and impedance values extremely closely validating the hybrid-analytical/numerical methodology developed in this research.
- (10) The TL results discussed in the conclusions section are for a maximum liner depth, including the face-plate thickness, of 32 mm, making it a significantly sub-wavelength liner.

Future work will seek to implement nonlinear terms into the analytical model in order to try to model the change in behavior at high SPLs.

Chapter 6

Incorporating Extended Neck Helmholtz Resonators into an MPPA

6.1 Introduction

In previous Chapters, MC-MPPAs have demonstrated good low-frequency broadband sound absorption performance. However, their effectiveness remains limited in the lower frequency range, especially within the even lower frequency band below 500 Hz, which is critical for many practical applications.

To address this limitation, integrating MC-MPPAs with other types of acoustic materials or structures presents a promising strategy. Among the potential candidates, the Helmholtz resonator is a classical low-frequency absorber known for its simple geometry and tunable resonance characteristics. It typically consists of a cavity connected to the external environment through a narrow neck, resembling the shape of a bottle. When subjected to incident sound waves, the air in the neck oscillates against the compressible air inside the cavity, producing resonance that allows efficient sound energy absorption at a specific frequency. The resonant frequency is determined by the neck's length and cross-sectional area, as well as the cavity volume, and is given by:

$$f = \frac{c_0}{2\pi} \sqrt{\frac{A}{VL_{\text{eff}}}} \quad (6.1)$$

where c_0 is the speed of sound in air, A is the neck cross-sectional area, V is the cavity volume, and L_{eff} is the effective neck length including end corrections. Achieving resonance at low frequencies typically requires a large cavity volume or an extended neck, which significantly increases the overall structural size. This makes traditional Helmholtz resonators unsuitable for space-limited environments such as aircraft fuselages or compact ducts.

Based on previous developments of MC-MPPAs with varying cavity depths, the present study proposes a hybrid configuration that incorporates an embedded extended-neck Helmholtz resonator into the MC-MPPA structure. This design aims to enhance broadband sound absorption below 500 Hz while maintaining a reduced form factor. The resulting structure holds promise as a clean and space-efficient acoustic liner for noise control in aerospace, architectural, and industrial applications.

This Chapter first introduces a single unit referred to as MHA. In this acronym, "M" stands for Micro-perforated panel, "H" for Helmholtz resonator, and "A" for Absorber. Subsequently, for the same plan area, 40 mm × 40 mm, a number of unit cells are combined with each other and with MPPAs without Helmholtz resonators in order to broaden the sound absorption bandwidth at low frequencies. This latter "Hybrid" configuration is termed the

HMHA, which consists of nine chambers. Conceptually, the HMHA integrates four MHAs and four micro-perforated panel absorbers into a compact configuration. This design enhances acoustic resonance at low frequencies and promotes coupling among its components. These designs are investigated theoretically, numerically, and experimentally.

6.2 MHA: An MPPA Coupled with an Embedded Extended Neck Helmholtz Resonator

6.2.1 Lumped element model

Figures 6.1(a) and 6.1(b) illustrate novel design aimed at minimizing the thickness of the acoustic absorber while optimising space utilization: a perforated panel with holes of diameter d and porosity ϕ , backed with an air cavity (AC1) and coupled with an embedded extended neck Helmholtz resonator (HR), hereafter referred to as MHA. The resonator's neck, of internal diameter d_n and length L_n , is designed to extend into an adjoining cavity (AC2). The micro-perforated top panel has a thickness of τ and dimensions of L_{ac1} and W . Similarly, the non-perforated top panel, also with a thickness of τ , has dimensions of L_{ac2} and W . The depth of the MPPA air cavity (AC1) is denoted by D_{ac1} . The depth of the HR cavity (AC2), denoted by D_{ac2} , is measured with respect to an incident plane wave. The axis of the extended neck is positioned at half of D_{ac2} , provided that $D_{ac1} \geq D_{ac2}$. If this condition cannot be met, the axis of the extended neck is positioned at half of D_{ac1} . The wall thickness of the Helmholtz resonator's neck is τ' and is set to be equal to the thickness of the panel separating the two cavities. The total thickness/depth, H , of the MHA is calculated by adding τ to the larger of D_{ac1} or D_{ac2} .

MPP specific acoustic impedance (z_{mpp})

When a plane acoustic wave is incident normal to the MPP, the specific acoustic impedance of the MPP, denoted as z_{mpp} and defined in Eq. (6.2), can be treated as the ratio of the sound pressure difference (Δp) across the MPP to the product of the average particle velocity ($\bar{u}$) over the perforation area and the MPP porosity (ϕ).

$$z_{mpp} = \frac{\Delta p}{\phi \bar{u}} \quad (6.2)$$

According to Maa [44], the calculation of the specific acoustic impedance z_{mpp} for an MPP with circular perforations can be conducted using Eq. (6.3) where the dependence

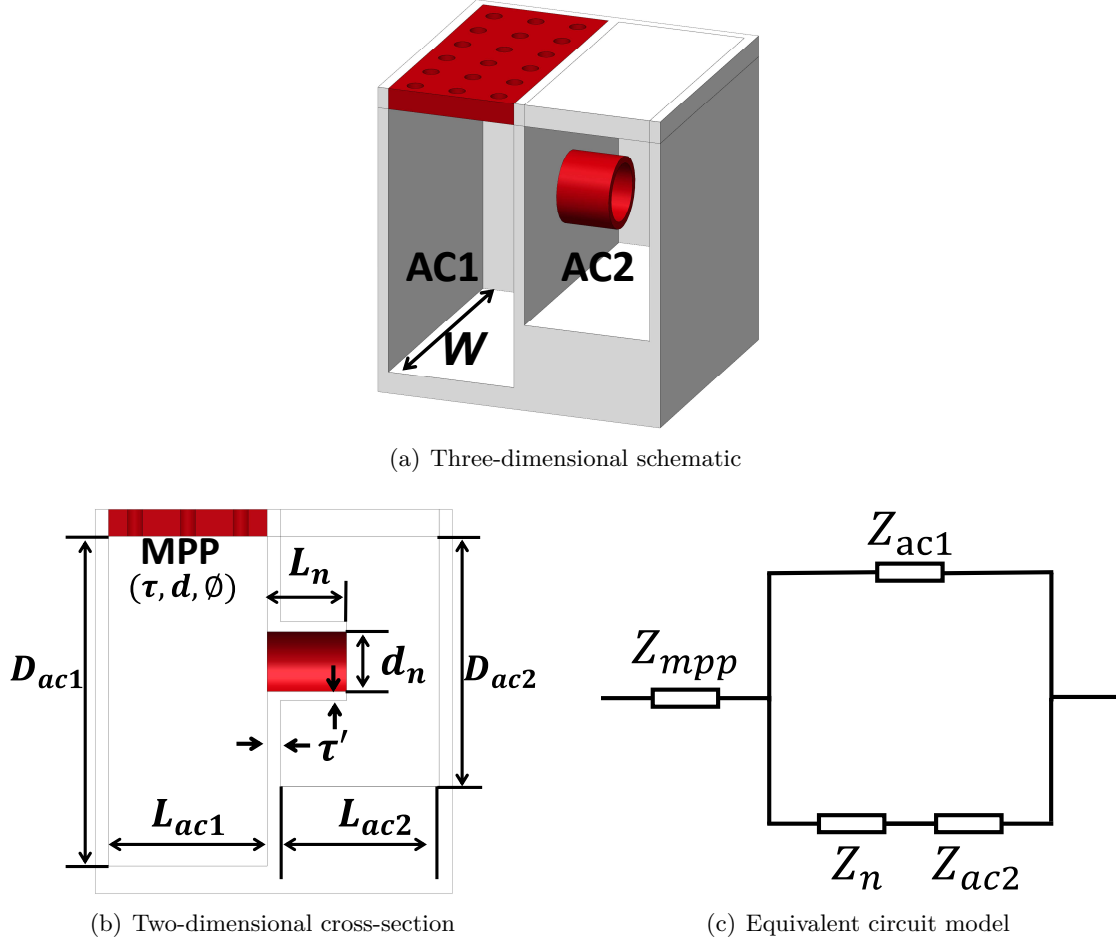


Figure 6.1: MHA design.

of the variables on angular frequency ω in this equation and in most others is omitted for brevity:

$$z_{\text{mpp}} = \frac{i\omega\tau}{c_0\phi} \left[1 - \frac{2}{\text{Sh} \sqrt{-i}} \frac{J_1(\text{Sh} \sqrt{-i})}{J_0(\text{Sh} \sqrt{-i})} \right]^{-1} + \frac{2\alpha_s R_S}{\rho_0 c_0 \phi} + i\delta_s \omega \frac{d}{2c_0 \phi}, \quad (6.3)$$

The $\exp(i\omega t)$ convention is used in this Chapter. The corresponding air properties used can be found in Table 6.1. The current study does not consider the non-linear effects of the MPP. In Eq. (6.3), $\frac{2\alpha_s R_S}{\rho_0 c_0 \phi}$ represents the resistive end-correction, details can be found in Section 5.3.1.

HR neck specific acoustic impedance (z_n)

The Helmholtz resonator, as illustrated in Figure 6.1, studied in this Chapter is characterised by a neck extending into the cavity. The specific acoustic impedance of the Helmholtz res-

6.2. MHA: AN MPPA COUPLED WITH AN EMBEDDED EXTENDED NECK HELMHOLTZ RESONATOR

onator's neck can be calculated using Eq. (6.4):

$$z_n = \frac{1}{\phi_n} \left(z_N + 2\sqrt{2\omega\rho_0\mu} + i\omega\rho_0\delta \right), \quad (6.4)$$

where $\phi_n = A_n/A_{hr}$ represents the ratio of the inner cross-sectional area ($A_n = \pi d_n^2/4$) of the Helmholtz resonator neck to the contact area ($A_{hr} = W \times \min[D_{ac1}, D_{ac2}]$) between the two cavities.

In Eq. (6.4), the term

$$z_N = -\rho_0 c_0 \frac{2i \sin\left(\frac{k_{c,n} L_n}{2}\right)}{\sqrt{(\gamma - (\gamma - 1)Y_h)Y_v}} \quad (6.5)$$

symbolizes the specific acoustic impedance of the HR neck. Due to the considerable length of the Helmholtz neck, it is necessary to account for both viscous and thermal losses. γ is the specific heat ratio of air.

The complex wavenumber, $k_{c,n}$ in the cylindrical HR neck, which accounts for both viscous and thermal losses, can be expressed as follows:

$$k_{c,n} = k_0 \sqrt{\left(\frac{\gamma - (\gamma - 1)Y_{h,n}}{Y_{v,n}} \right)} \quad (6.6)$$

The viscous function of the cylindrical HR neck, $Y_{v,n}$ and the thermal function of the cylindrical HR neck, $Y_{h,n}$, are shown in Eq. (6.7).

$$\begin{aligned} Y_{v,n} &= -\frac{J_2(k_v d_n/2)}{J_0(k_v d_n/2)} \\ Y_{h,n} &= -\frac{J_2(k_h d_n/2)}{J_0(k_h d_n/2)} \end{aligned} \quad (6.7)$$

where the acoustic wave number k_0 , viscous wave number k_v and the thermal wave number k_h are defined as:

$$k_0 = \frac{\omega}{c}, \quad k_v^2 = -i\omega \frac{\rho_0}{\mu}, \quad k_h^2 = -i\omega \frac{\rho_0 C_p}{K} \quad (6.8)$$

and where C_p is the specific heat at constant pressure and K is the fluid thermal conductivity.

In Eq. (6.4), the term $2\sqrt{2\omega\rho_0\mu}$ represents the end-correction for the acoustic resistance, accounting for frictional losses attributed to the internal boundaries of the HR neck. Similarly, the term $i\omega\rho_0\delta$ represents the end-correction for the acoustic reactance due to acoustic wave radiation. The total length end-correction in the neck, $\delta = \delta_1 + \delta_2$, is the sum of the length corrections at both ends of the neck. The transition from the MPPA cavity (AC1) to the

neck approximates a flanged pipe termination; thus, the corresponding length end-correction δ_1 is given by [156]

$$\delta_1 = \delta_\infty + \frac{d_n/2}{b} (\delta_0 - \delta_\infty) + 0.057 \frac{d_n/2}{b} \left[1 - \left(\frac{d_n/2}{b} \right)^5 \right] d_n/2 \quad (6.9)$$

where $\delta_\infty = 0.8216(d_n/2)$ is the length end-correction of an infinitely flanged pipe and $\delta_0 = 0.6133(d_n/2)$ is the length end-correction of an ideal unflanged pipe. The parameter b is an 'effective' value for a rectangular flange; In this Chapter, b is calculated as $b = \sqrt{WD_{ac1}/\pi}$. The transition from the neck to the HR cavity (AC2) approximates an unflanged pipe termination with a thin wall thickness, τ' , so the length end correction, δ_2 , should be [156]

$$\delta_2 = \delta_\infty + \frac{d_n/2}{d_n/2 + \tau'} (\delta_0 - \delta_\infty) + 0.057 \frac{d_n/2}{d_n/2 + \tau'} \left[1 - \left(\frac{d_n/2}{d_n/2 + \tau'} \right)^5 \right] d_n/2 \quad (6.10)$$

Air cavity specific acoustic impedance (z_{ac})

In this Chapter, under the condition $\lambda \gg H$, uniform pressure can be assumed within the air cavity. The specific acoustic impedances of AC1 and AC2 can be defined as:

$$\begin{aligned} z_{ac1} &= -i \frac{\rho_{ac} c_{ac}^2}{\omega V_{ac1}} A_{ac1}, & A_{ac1} &= L_{ac1} W \\ z_{ac2} &= -i \frac{\rho_{ac} c_{ac}^2}{\omega V_{ac2}} A_{ac2}, & A_{ac2} &= D_{ac2} W \end{aligned} \quad (6.11)$$

where $V_{ac1} = L_{ac1} \times W \times D_{ac1}$ is the volume of AC1, $V_{ac2} = L_{ac2} \times W \times D_{ac2} - \pi(d_n/2 + \tau')^2 \times (L_n - \tau')$ is the volume of AC2 minus the volume occupied by the HR neck and

$$\rho_{ac} = \frac{\rho_0}{Y_{v,ac}}, \quad c_{ac} = \frac{\omega}{k_{c,ac}} \quad (6.12)$$

where the complex wavenumber, $k_{c,ac}$, in the rectangular air cavity, which accounts for both viscous and thermal losses, can be expressed as follows:

$$k_{c,ac} = k_0 \sqrt{\left(\frac{\gamma - (\gamma - 1) Y_{h,ac}}{Y_{v,ac}} \right)} \quad (6.13)$$

The viscous function, $Y_{v,ac}$ and the thermal function, $Y_{h,ac}$ of the rectangular air cavity with cross-sectional dimensions A and B are shown in Eq. (6.14). [157]

$$\begin{aligned}
 Y_{v,ac} &= k_v^2 \sum_{m=0}^{\infty} \left[(\alpha_{m,v} m')^{-2} \left(1 - \frac{\tan(\alpha_{m,v} A/2)}{\alpha_{m,v} A/2} \right) + (\beta_{m,v} m')^{-2} \left(1 - \frac{\tan(\beta_{m,v} B/2)}{\beta_{m,v} B/2} \right) \right] \\
 Y_{h,ac} &= k_h^2 \sum_{m=0}^{\infty} \left[(\alpha_{m,h} m')^{-2} \left(1 - \frac{\tan(\alpha_{m,h} A/2)}{\alpha_{m,h} A/2} \right) + (\beta_{m,h} m')^{-2} \left(1 - \frac{\tan(\beta_{m,h} B/2)}{\beta_{m,h} B/2} \right) \right]
 \end{aligned} \tag{6.14}$$

where $m' = (m + 1/2)\pi$ and

$$\begin{aligned}
 \alpha_{m,v} &= \sqrt{k_v^2 - \left(\frac{2m'}{A} \right)^2}, & \beta_{m,v} &= \sqrt{k_v^2 - \left(\frac{2m'}{B} \right)^2} \\
 \alpha_{m,h} &= \sqrt{k_h^2 - \left(\frac{2m'}{A} \right)^2}, & \beta_{m,h} &= \sqrt{k_h^2 - \left(\frac{2m'}{B} \right)^2}
 \end{aligned} \tag{6.15}$$

For AC1, let $A = L_{ac1}$ and $B = W$. For AC2, let $A = D_{ac2}$ and $B = W$.

The total specific impedance of the MHA, as derived from the equivalent circuit model, depicted in Figure 6.1(c), can be calculated using the following equation:

$$z_{mha} = (Z_{mpp} + \frac{1}{1/Z_{ac1} + 1/(Z_n + Z_{ac2})}) \times A_{surface} \tag{6.16}$$

It should be noted that each specific acoustic impedance in the circuit must be normalized by the cross-sectional surface area of the acoustic element [63, 158]. Consequently, the acoustic impedances are calculated as follows: $Z_{mpp} = z_{mpp}/A_{mpp}$, $Z_{ac1} = z_{ac1}/A_{ac1}$, $Z_n = z_n/A_n$ and $Z_{ac2} = z_{ac2}/A_{ac2}$. As illustrated in Figure 6.1(a) and (b), the cross-sectional areas of the MPP and the AC are equal, given by $A_{mpp} = L_{ac1} \times W$, and the surface area of the elementary cell is expressed as $A_{surface} = (L_{ac1} + L_{ac2} + \tau') \times (W)$. Furthermore, the normal incidence sound absorption coefficient (SAC), α , can be determined by

$$\alpha = 1 - |r|^2 \tag{6.17}$$

where reflection coefficient of r is given by

$$r = \frac{z_{mha} - \rho_0 c_0}{z_{mha} + \rho_0 c_0} \tag{6.18}$$

6.2.2 Numerical modelling

A numerical model representing a three-dimensional, two-microphone impedance tube, compliant with ISO 10534-2 [153], is modelled using COMSOL Multiphysics 6.0. It is used to calculate the numerical SAC of the tested specimens. The tube's cross-sectional area is de-

Table 6.1: Air properties involved in the current study.

Air density	ρ_0	1.204kg/m ³
Speed of sound	c_0	343m/s
Dynamic viscosity	μ	1.81×10^{-5} kg/(ms)
Ratio of specific heats	γ	1.4
Specific heat capacity	C_p	1005 J/(kg · K)
Thermal conductivity	K	0.0257 W/(m · K)

terminated by the dimensions of the target test piece. In accordance with standard practices, a numerical simulation of an impedance tube setup is conducted, wherein a normally incident plane wave pressure field is generated upstream of the test section. The interfaces between the air and solid components are modeled as acoustically hard boundaries. The pressure acoustics module is used to simulate the air domain within the upstream domain of the tube and the air-cavity domains of the specimens, excluding the neck of the Helmholtz resonator. For the MPPs, the condition of the internal perforated thin panel boundary, as represented by Eq. (6.3) in COMSOL, is applied.

One approach to model the domain inside the resonator neck would be to use COMSOL's thermo-viscous acoustics module requiring detailed geometric definition and meshing of this subdomain and terminations. Similar to the use of the IPP boundary condition, a model of the thermo-viscous behaviour can be used as an alternative to save time and computational resources. Consequently, the Johnson-Champoux-Allard (JCA) model is chosen to approximate the sound propagation in the resonator's neck [99]. In the JCA model, the domain inside the resonator neck is modelled as an equivalent fluid using a complex effective density $\tilde{\rho}(\omega)$ and an effective bulk modulus $\tilde{K}(\omega)$. $\tilde{\rho}(\omega)$ and $\tilde{K}(\omega)$ can be expressed as: [159, 160]

$$\tilde{\rho}(\omega) = \frac{\alpha_\infty \rho_0}{\phi} \left[1 + \frac{\sigma \phi}{i \omega \rho_0 \alpha_\infty} \sqrt{1 + i \frac{4 \alpha_\infty^2 \mu \rho_0 \omega}{\sigma^2 \Lambda''^2 \phi^2}} \right] \quad (6.19)$$

$$\tilde{K}(\omega) = \frac{\gamma P_0 / \phi}{\gamma - (\gamma - 1) \left[1 - i \frac{8K}{\Lambda'^2 C_p \rho_0 \omega} \sqrt{1 + i \frac{\Lambda'^2 C_p \rho_0 \omega}{16K}} \right]^{-1}} \quad (6.20)$$

The JCA model utilizes five parameters—porosity (ϕ), flow resistivity (σ), tortuosity (α_∞), viscous characteristic length (Λ''), and thermal characteristic length (Λ')—to characterize sound wave propagation in a specific medium. Using these, the characteristic impedance Z_{ef} and the wave number k_{ef} of the modeled equivalent fluid inside the resonator neck can be expressed as

$$Z_{\text{ef}} = \sqrt{\tilde{K}(\omega)\tilde{\rho}(\omega)}, \quad k_{\text{ef}} = \omega\sqrt{\tilde{\rho}(\omega)/\tilde{K}(\omega)} \quad (6.21)$$

For sound waves travelling through air within the cylindrical cross-section of the Helmholtz resonator's neck, the parameters of the JCA model are specified as follows [99, 161]:

- porosity ϕ : 1;
- flow resistivity σ : $\frac{8\mu}{(d_n/2)^2}$;
- tortuosity α_∞ : 1;
- viscous characteristic length Λ'' : $d_n/2$;
- thermal characteristic length Λ' : $d_n/2$.

6.2.3 Initial validation

To assess the accuracy of the theoretical and numerical models of the Helmholtz resonator, in particular for designs incorporating an extended resonator neck, a validation is conducted by comparing to previously published experimental results: Huang et al. [162]. A 3D-printed version of their resonator is shown in Figure 6.2(a), and its geometric details are detailed in Table 6.2. The reference model is simply an extended neck into a closed cylindrical volume of diameter 29 mm. There are no micro-perforations. Their experimental sound absorption coefficient for normal acoustic incidence is shown in Figure 6.2(b) using WebPlotDigitizer to capture their data. Superimposed onto the plot are the theoretical and numerical absorption coefficients calculated following the procedures described in Section 6.2.1 and 6.2.2 and using the geometric data from Table 6.2. The results compare well and thus the theoretical models and numerical approach are validated.

Table 6.2: Geometric details of the 3D printed neck-extended Helmholtz resonator developed by Huang et al. [162].

HR neck's internal diameter	r_A	r_c	l_a	$L_u + L_b$	$b, (nominal)$	b_t
3.5 mm	14.5 mm	13.3 mm	40 mm	50 mm	1 mm	1 mm

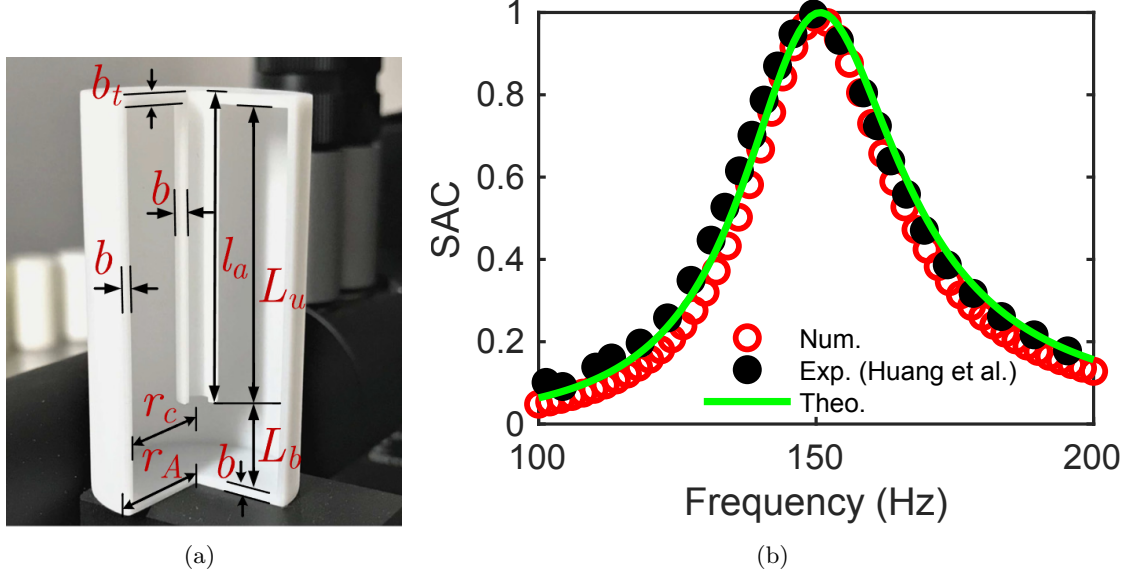


Figure 6.2: Photo of the 3D-printed neck-extended Helmholtz resonator developed by Huang et al. [162], along with comparison results against the experimental data presented in Huang et al. [162].

6.3 Sensitivity Analysis and Optimisation

6.3.1 Sensitivity analysis

A parametric sensitivity study is conducted in order to evaluate how different geometric parameters affect the response of the MHA (as shown in Figure 6.1). It is acknowledged that a superposition of the response of individual parameters in isolation will not necessarily represent the behaviour as a whole, especially when the optimisation is carried out on all parameters simultaneously, but it is still possible to gain useful insight. The sensitivity analysis is conducted on seven critical geometric parameters: d , ϕ , τ , D_{ac1} , d_n , L_n , and D_{ac2} . The following parameters remain constant: $L_{ac1} = L_{ac2} = 19.5$ mm, $W = 20$ mm and $\tau' = 1$ mm.

Figure 6.3 illustrates the impact of the parameters on the acoustic absorption performance of the MHA, within the frequency range of [100, 1000] Hz. Each subplot delineates the relationship between the examined geometric parameters and the SAC across the specified frequency range. The values for each parameter analyzed in these sensitivity studies are provided in Table 6.3.

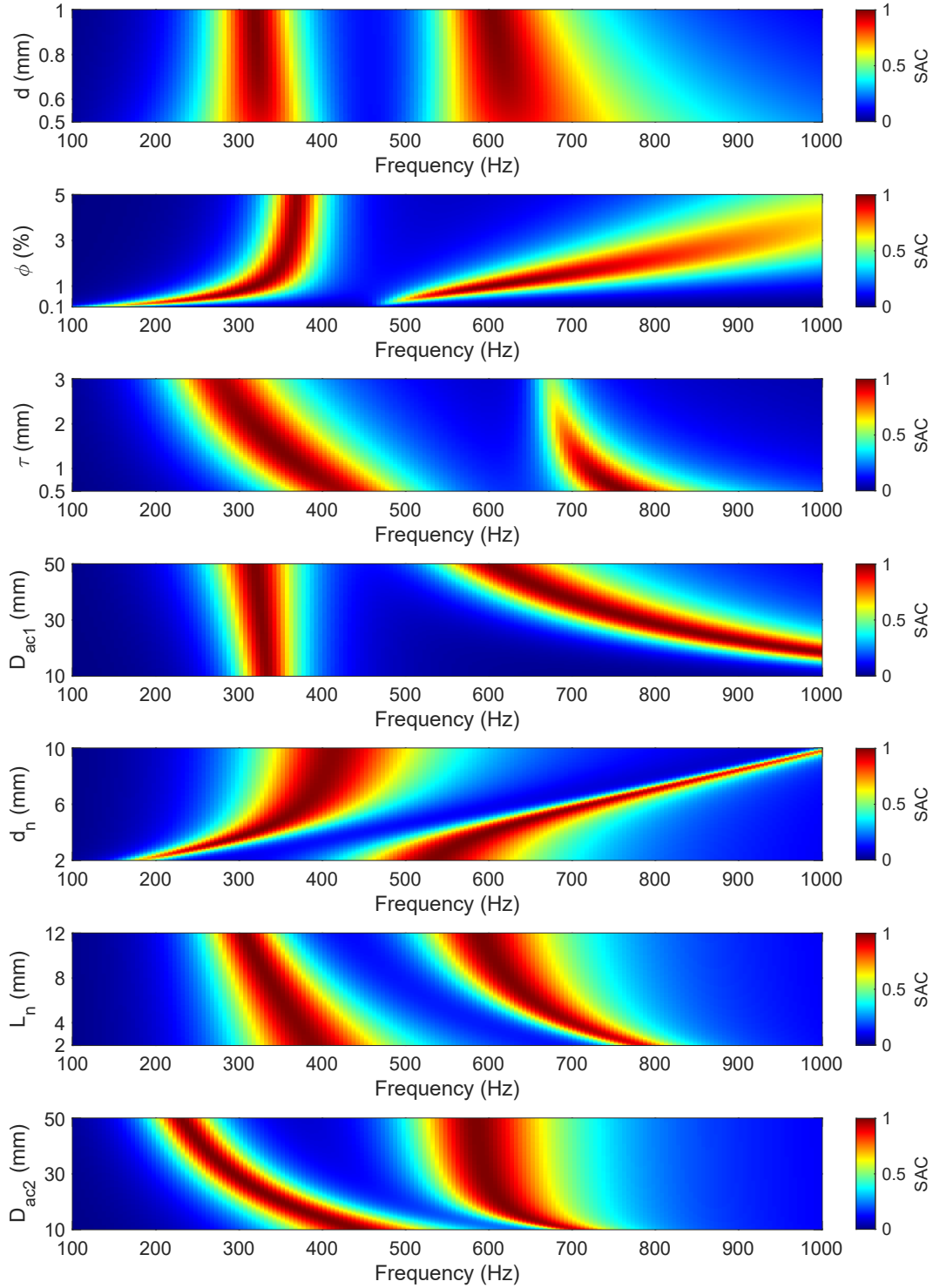


Figure 6.3: Geometrical parameter studies of MHA in the frequency range [100, 1000] Hz. From top to bottom: sensitivity analysis of d , ϕ , τ , D_{ac1} , d_n , L_n , and D_{ac2} .

The depicted graphs each feature two prominent absorption peaks: the lower frequency peak is caused by the Helmholtz resonator, while the higher frequency peak is attributed to the interaction between the MPP and its air-backed cavity.

The analysis reveals that variations in the parameters d and D_{ac1} , as to be expected, primarily influence the acoustic properties of the MPPA, exerting relatively little impact on the sound absorption efficiency of the Helmholtz resonator in the parameter range examined. However it can be seen that as the MPP hole diameter decreases, $d < 0.6$, the absorption from the Helmholtz resonator begins to decrease also and presumably would continue to do so with increasing blockage. The principle observation is that an increase in D_{ac1} reduces the frequency of the MPPA peak while widening the absorption bandwidth.

Geometric parameters τ , L_n , d_n and D_{ac2} significantly influence both the Helmholtz resonator's acoustic performance and the acoustic properties of the MPP and its air-backed cavity. Reducing τ causes both the first and second absorption spectra of the MHA to shift upward in frequency. This modification expands the bandwidth of the second absorption spectrum but has a marginal impact on the bandwidth of the first. A value for τ of 1 mm or less is optimal with a very significant reduction in absorption coefficient for the MPPA with increasing values of τ . Reducing L_n also results in an upward frequency shift for both absorption spectra. However, this change increases the bandwidth of the first absorption spectrum while reducing that of the second. Decreasing d_n shifts the first absorption spectrum of the MHA to lower frequencies but narrows the absorption bandwidth. This adjustment also lowers the frequency of the second absorption spectrum but widens its bandwidth. An increase in D_{ac2} shifts both the first and second absorption spectra of the MHA towards lower frequencies. While this increase reduces the first absorption bandwidth, it expands the second absorption bandwidth.

The influence of the geometric parameter ϕ on the acoustic performance of the MHA is subtly complex. When ϕ exceeds 2%, changes primarily affect the acoustic properties of the MPPA. Conversely, for ϕ values between 0.1 and 2, reducing ϕ causes a downward shift in both the first and second absorption spectra, narrowing the absorption bandwidth and reducing the peak absorption.

An initial observation from the sensitivity analysis is that it is difficult to determine the optimal value of individual parameters. In this chapter, both broadband and low-frequency absorption are required. However, as shown by many of the variables, these two objectives often conflict with each other. That is to say, for a particular parameter, changing it may broaden the absorption coefficient, which is advantageous, but may also increase its frequency,

which is disadvantageous. Therefore, an optimisation routine is required to find an optimum set of values in the solution space.

A second comment is that, although the frequency range of interest in this Chapter, as stated, is less than 500 Hz, it was necessary for the frequency range of the sensitivity study to extend up to 1000 Hz. This is because it was noticed that only by increasing the frequency range could the two peaks be examined independently from one another, thereby allowing the sensitivity analysis to be conducted. As will be shown, when the frequency range is compressed to between 200 Hz and 500 Hz, both peaks coalesce due to coupling. This is an advantageous and desirable attribute of the technology; however, a larger frequency range was needed to examine the effect of the individual geometric parameters.

6.3.2 Optimisation of MHA's geometrical parameters

Optimising the geometric parameters of the MHA under study is crucial to achieving broadband and efficient sound absorption across the specified frequency range. To this end, Eq. (3.20) and Eq. (3.21) have been formulated as a cost function to quantify the total SAC within the specified frequency range. The objective of this Chapter is to address the challenge of low-frequency noise absorption, particularly for frequencies below 500 Hz. Consequently, the frequency range for optimisation has been established from 200 Hz to 500 Hz. White noise, $G_{xx}(\omega) \equiv 1$, is the target noise in the current study. PlatEMO [143, 144], an open-source MATLAB-based platform designed for complex optimisation problems, has been selected as the optimisation framework. It offers an extensive suite of built-in algorithms. The Genetic Algorithm (GA) [145] is utilized to minimize Eq. (3.20) within the designated frequency range. The geometric parameters under optimisation are detailed in Table 6.4. The following parameters remain constant: $L_{ac1} = L_{ac2} = 19.5$ mm, $W = 40$ mm and $\tau' = 1$ mm. Note that the number of holes, n , in the MPP is used rather than porosity to facilitate more manageable and accurate manufacturing of the MPP. In this optimisation, the number of holes is considered to range from 0 to 10 for the convenience of either 3-D printing or machining. Given the limited number of holes, and therefore the space between them, the interaction between the holes can be considered to be negligible.

Figure 6.4(a) displays the optimised theoretical SAC curves for the MHA across the frequency range of 200 to 500 Hz and for total thicknesses varying from 11.2 mm to 51 mm. Figure 6.4(b) presents a colormap representing these coefficients, highlighting the variations over the specified frequency and thickness ranges. The final optimised geometric parameters

CHAPTER 6. INCORPORATING EXTENDED NECK HELMHOLTZ RESONATORS INTO AN MPPA

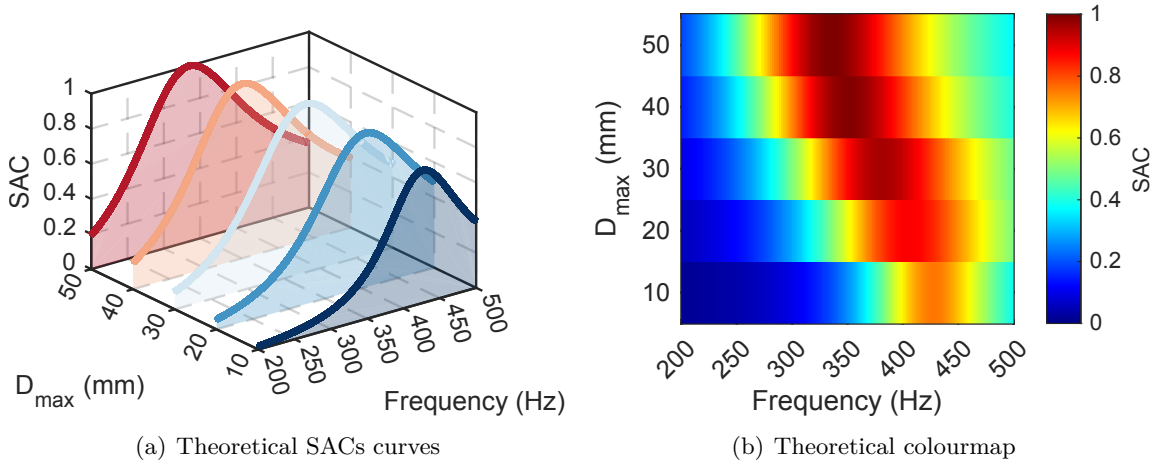
Table 6.3: Settings of geometric parameters corresponding to each sensitivity study of geometric parameters. The following parameters remain constant: $L_{ac1} = L_{ac2} = 19.5$ mm, $W = 20$ mm and $\tau' = 1$ mm.

Sensitivity analysis of d						
$d(\text{mm})$	$\phi(\%)$	$\tau(\text{mm})$	$D_{ac1}(\text{mm})$	$d_n(\text{mm})$	$L_n(\text{mm})$	$D_{ac2}(\text{mm})$
[0.5 1.0]	1	1.0	50	4	10	25
Sensitivity analysis of ϕ						
$d(\text{mm})$	$\phi(\%)$	$\tau(\text{mm})$	$D_{ac1}(\text{mm})$	$d_n(\text{mm})$	$L_n(\text{mm})$	$D_{ac2}(\text{mm})$
1.0	[0.1 5]	1.0	50	4	10	25
Sensitivity analysis of τ						
$d(\text{mm})$	$\phi(\%)$	$\tau(\text{mm})$	$D_{ac1}(\text{mm})$	$d_n(\text{mm})$	$L_n(\text{mm})$	$D_{ac2}(\text{mm})$
1.0	1	[1.0 3.0]	50	4	10	25
Sensitivity analysis of D_{ac1}						
$d(\text{mm})$	$\phi(\%)$	$\tau(\text{mm})$	$D_{ac1}(\text{mm})$	$d_n(\text{mm})$	$L_n(\text{mm})$	$D_{ac2}(\text{mm})$
1.0	1	1.0	[10 50]	4	10	25
Sensitivity analysis of d_n						
$d(\text{mm})$	$\phi(\%)$	$\tau(\text{mm})$	$D_{ac1}(\text{mm})$	$d_n(\text{mm})$	$L_n(\text{mm})$	$D_{ac2}(\text{mm})$
1.0	1	1.0	50	[2 10]	10	25
Sensitivity analysis of L_n						
$d(\text{mm})$	$\phi(\%)$	$\tau(\text{mm})$	$D_{ac1}(\text{mm})$	$d_n(\text{mm})$	$L_n(\text{mm})$	$D_{ac2}(\text{mm})$
1.0	1	1.0	50	4	[2 12]	25
Sensitivity analysis of D_{ac2}						
$d(\text{mm})$	$\phi(\%)$	$\tau(\text{mm})$	$D_{ac1}(\text{mm})$	$d_n(\text{mm})$	$L_n(\text{mm})$	$D_{ac2}(\text{mm})$
1.0	1	1.0	50	4	10	[10 50]

of the MHA corresponding to Figure 6.4 can be found in the appendix in Table E.13. Table 6.5 presents the frequencies at which absorption peaks occur for various absorber thicknesses, along with the frequency ranges (bandwidths) over which the absorber achieves 50% and 70% absorption, as indicated in Figure 6.4. Theoretically, an MHA can achieve varying degrees of sound absorption depending on its total thickness and the target frequency of sound. At a frequency of 426 Hz, the MHA demonstrates approximately 76% sound absorption with

Table 6.4: Parameter settings for optimisation of MHA. The following parameters remain constant: $L_{ac1} = L_{ac2} = 19.5$ mm, $W = 40$ mm and $\tau' = 1$ mm.

Parameters	Lower bound	Upper bound
d	0.5 mm	1.0 mm
n	0	10
τ	1 mm	3 mm
D_{ac1}	10 mm	50 mm
d_n	2 mm	10 mm
L_n	2 mm	12 mm
D_{ac2}	10 mm	50 mm
f	200 Hz	500 Hz

**Figure 6.4:** Theoretical SAC curves and colormap for optimised MHA in [200 500] Hz with D_{max} : [10 50] mm. Corresponding geometric parameters are in Table E.13.

a total thickness of 11.2 mm, which is about $1/72$ of the wavelength of the peak frequency. Similarly, at 404 Hz, it can absorb about 88% of the noise with a thickness of 21.6 mm, approximately $1/39$ of the wavelength. Near-perfect absorption of 96% is achieved at 380 Hz with a 31.0 mm thick absorber, roughly $1/29$ of the wavelength. Moreover, the MHA exhibits over 99% absorption at frequencies of 349 Hz and 334 Hz, with thicknesses of 41.0 mm and 51.0 mm, corresponding to $1/24$ and $1/20$ of their respective wavelengths. In other words, the MHA can be considered to be tunable, where its optimal frequency range can be achieved by adjusting its thickness.

More importantly, and in contrast to the analysis of individual parameters, as the total thickness of the MHA increases, its peak absorption shifts continuously towards lower frequencies and its response becomes more broadband concurrently, i.e., the frequency ranges at which the absorber achieves 50% and 70% absorption also shift to lower frequencies. Ulti-

mately, the MHA achieves perfect sound absorption of over 99.9% with a central frequency of 334 Hz, with a thickness of only 51 mm and achieves 70% sound absorption in the frequency range of 279 Hz to 408 Hz, and 50% sound absorption from 255 Hz to 454 Hz. This finding is attractive and will be validated in subsequent experiments using the 51-mm thick MHA case.

Table 6.5: Frequencies at which MHA’s absorption peaks occur for various absorber thicknesses, along with the frequency ranges over which the absorber achieves 50% and 70% absorption, as indicated in Figure 6.4.

D_{max} ($H = \tau + D_{max}$)	SAC peak (Sub-wavelength ratio)	Frequency range (bandwidth)	
		SAC >0.7	SAC >0.5
10 mm (11.2 mm)	0.76 at 426 Hz ($\lambda/72$)	[408-446] Hz (13%)	[383-478] Hz (32%)
20 mm (21.6 mm)	0.88 at 404 Hz ($\lambda/39$)	[361-456] Hz (32%)	[334-498] Hz (55%)
30 mm (31.0 mm)	0.96 at 380 Hz ($\lambda/29$)	[325-448] Hz (41%)	[299-495] Hz (65%)
40 mm (41.0 mm)	>0.99 at 349 Hz ($\lambda/24$)	[294-422] Hz (43%)	[270-468] Hz (66%)
50 mm (51.0 mm)	>0.99 at 334 Hz ($\lambda/20$)	[279-408] Hz (43%)	[255-454] Hz (66%)

6.3.3 Experimental validation of the optimised MHA

An experimental validation of the MHA design was conducted for the case of maximum cavity depth equal to 50 mm (overall thickness is 51 mm). Figure 6.5 displays photographs of the fabricated MHA sample. As depicted in Figure 6.5(a), the core is made from Polylactic Acid (PLA) filament through the Fused Deposition Modeling (FDM) 3D printing method. Figure 6.5(b) shows the facesheet which is constructed using a photopolymer resin that cures under UV light at a wavelength of 405 nm, employing an mSLA 3D printing method. The mSLA method is chosen here as it is more suitable for producing accurate hole geometries. The sample is placed at the end of the same impedance tube (40 mm $\times$ 40 mm), as shown in Figure 6.5(c), and measurements are performed using the two-microphone method. Details of the experimental setup are provided in B.1. The geometric parameters of the fabricated HMHA are presented in Table 6.6.

The theoretical, numerical, and experimental sound absorption curves of the optimised MHA are shown in Figure 6.6, demonstrating excellent agreement. Experimental results indicate that the MHA achieves perfect sound absorption of over 99.9% at the central frequency of 329.6 Hz, with a thickness of only 51 mm—approximately 1/20.4 the wavelength of the sound at this frequency. Furthermore, it achieves 70% sound absorption in the frequency range of 273 Hz to 400 Hz, and 50% sound absorption from 249 Hz to 447 Hz. These results

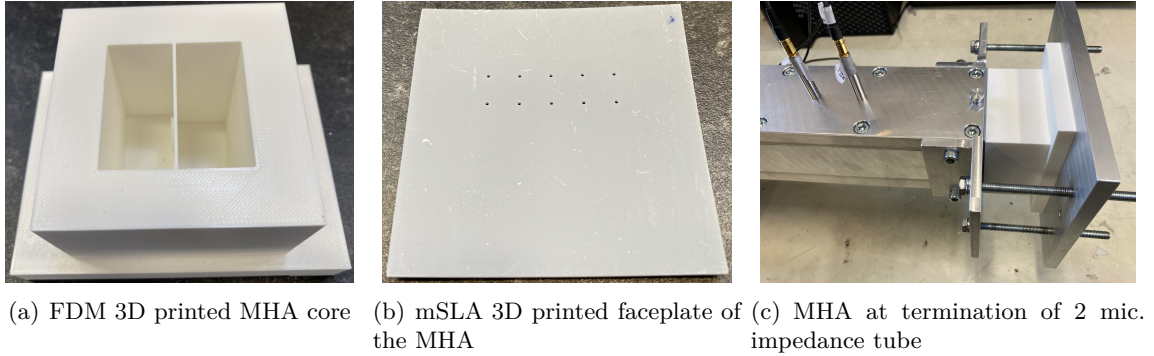


Figure 6.5: Experimental testing of the MHA sample.

show a very slight deviation from the theoretical data in Table 6.5, which is likely due to manufacturing or testing errors in the test pieces.

The acoustic absorption capabilities of the MHA can be attributed to its acoustic reactance and resistance, as illustrated in Figure 6.6(b). For an ideal impedance match, it is necessary for the acoustic reactance to equal zero and the acoustic resistance to equal one, a condition that must be met concurrently. Analysis of the theoretical and experimental data on acoustic reactance and resistance, as presented in Figures 6.6 (a) and (b), indicates that at a frequency of approximately 329.6 Hz, where the SAC approaches unity, the acoustic reactance approaches zero, and the acoustic resistance approaches one.

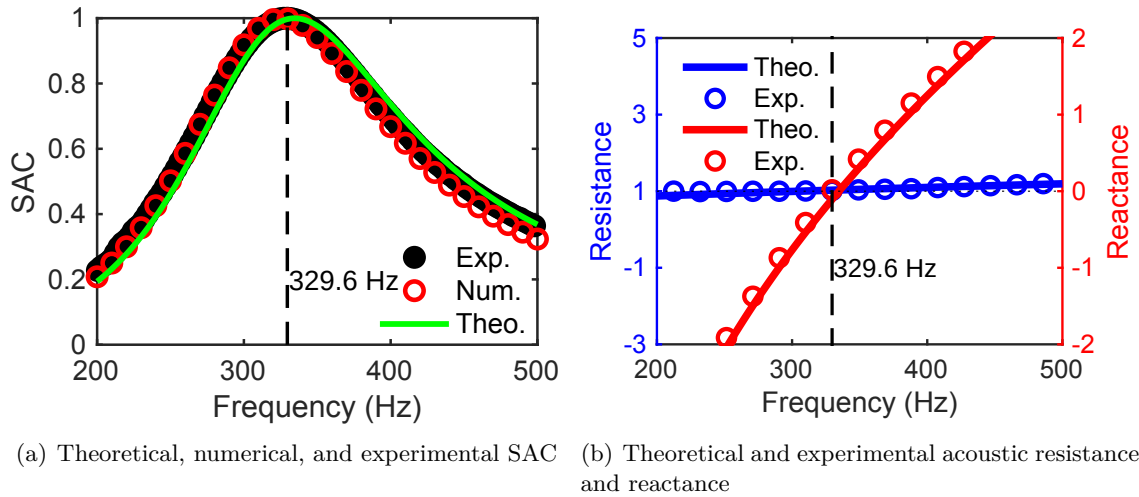


Figure 6.6: Theoretical, numerical, and experimental SAC curves, and the corresponding acoustic resistance and reactance for the fabricated MHA sample (51 mm thickness). Vertical dashed line corresponds to the experimental maximum.

Table 6.6: Geometric parameters of the fabricated MHA.

d	n	τ	D_{ac1}	d_n	L_n	D_{ac2}	τ'	L_{ac1}, L_{ac2}	W
0.89 mm	10	1 mm	50 mm	10 mm	2 mm	43 mm	1 mm	19.5 mm	40 mm

6.4 HMHA: A Hybrid, Low Frequency, Broadband Sound Absorber Based on MHA Unit Cells

6.4.1 Design and optimisation

To broaden the sound absorption bandwidth of the MHA while keeping the total area constant, Figure 6.7 presents a novel variation, termed the Hybrid MHA (HMHA), which consists of nine chambers. The HMHA can be conceptualized as an integration of four MHAs and four micro-perforated panel absorbers into a more compact configuration, thereby enhancing acoustic resonance within the low-frequency spectrum and facilitating a coupling effect among its components. Each chamber within this design can have unique depths, denoted as $D_{(i)}$, where i is the chamber number. The dimensions of each chamber are denoted by length L_1 and width W_1 . Similar to the description in Section 6.2.1, the thickness of the *top* plate in the HMHA is symbolized as τ , whereas the thickness of the *internal* adjacent panels is represented by τ' . Once again, the wall thickness of the Helmholtz resonator's neck is τ' . The total thickness, H , of the HMHA is obtained by summing τ and the maximum sub-chamber depth, $D_{\max}$. As depicted in Figure 6.7(c), these chambers are sequentially numbered from 1 to 9. Specifically, chambers 1, 3, 7, and 9 are isolated MPPAs. The perforation diameter and number of each MPP in this Chapter are variable. These dimensions are denoted as $d_{(i)}$ and $n_{(i)}$, respectively. Conversely, chambers 2, 4, 6, and 8 function as Helmholtz resonators, coupled by chamber 5, which also has a perforated top plate. Each resonator is characterized by a neck that can extend into its cavity with unique length and internal diameter. These dimensions are denoted as $L_{n(i)}$ and $d_{n(i)}$, respectively. The solid regions on the top surface of the HMHA design have the potential to reduce aerodynamic drag, which is particularly relevant for applications in flow ducts or aircraft engine liners.

The total specific acoustic impedance of the HMHA, as derived from the equivalent circuit model, depicted in Figure 6.7(c), can be calculated using the following equation:

$$z_{\text{HMHA}} = \frac{1}{1/Z_5 + 1/Z_1 + 1/Z_3 + 1/Z_7 + 1/Z_9} \times A_{\text{surface}} \quad (6.22)$$

where $A_{\text{surface}} = (3 \times L_1 + 2\tau') \times (3 \times W_1 + 2\tau')$ and

$$\begin{cases} Z_5 = Z_{\text{mpp}(5)} + 1/(1/Z_{\text{ac}(5)} + 1/Z_{\text{hr}(2)} + 1/Z_{\text{hr}(4)} + 1/Z_{\text{hr}(6)} + 1/Z_{\text{hr}(8)}) \\ Z_1 = Z_{\text{mpp}(1)} + Z_{\text{ac}(1)} \\ Z_3 = Z_{\text{mpp}(3)} + Z_{\text{ac}(3)} \\ Z_7 = Z_{\text{mpp}(7)} + Z_{\text{ac}(7)} \\ Z_9 = Z_{\text{mpp}(9)} + Z_{\text{ac}(9)} \end{cases} \quad (6.23)$$

where the term $Z_{\text{mpp}(5)}$ denotes the acoustic impedance of the MPP in chamber 5. The acoustic impedances of the Helmholtz resonator in chambers 2, 4, 6, and 8 are represented by $Z_{\text{hr}(2)} = Z_{\text{n}(2)} + Z_{\text{ac}(2)}$, $Z_{\text{hr}(4)} = Z_{\text{n}(4)} + Z_{\text{ac}(4)}$, $Z_{\text{hr}(6)} = Z_{\text{n}(6)} + Z_{\text{ac}(6)}$, and $Z_{\text{hr}(8)} = Z_{\text{n}(8)} + Z_{\text{ac}(8)}$, respectively. Similarly, $Z_{\text{mpp}(1)}$, $Z_{\text{mpp}(3)}$, $Z_{\text{mpp}(7)}$, and $Z_{\text{mpp}(9)}$ correspond to the acoustic impedances of the MPP in chambers 1, 3, 7, and 9, respectively. Furthermore, $Z_{\text{ac}(1)}$, $Z_{\text{ac}(3)}$, $Z_{\text{ac}(7)}$, and $Z_{\text{ac}(9)}$ indicate the acoustic impedances of the air cavity in chambers 1, 3, 7, and 9, respectively. Finally, the normal incidence SAC of HMHA, α , can be determined by

$$\alpha = 1 - \left| \frac{z_{\text{HMHA}} - \rho_0 c_0}{z_{\text{HMHA}} + \rho_0 c_0} \right|^2 \quad (6.24)$$

In a similar fashion to the previous section, an optimisation is performed on the HMHA to maximise sound absorption within a predefined frequency range. The optimisation algorithm, cost function, and geometric parameter configuration for this process are similar to those described in Section 6.3.2. The only difference is that the parameter τ is set to be 1mm throughout to prevent irregularities on the top surface of HMHA.

Figure 6.8(a) presents the optimised theoretical SAC curves of the HMHA within the frequency range of 200 to 500 Hz and for varying total thicknesses between 11 and 51 mm. Figure 6.8(b) illustrates the colormap corresponding to each absorption curve depicted in Figure 6.8(a). The analysis indicates that, similar to MHA, the thickness of the absorber still remains a critical determinant of HMHA's sound absorption efficacy. Moreover, Table 6.7 delineates the specific c_α values for Figure 6.8 across three distinct frequency ranges: 200 to 500 Hz, 250 to 500 Hz, and 300 to 500 Hz, as a function of D_{max} . It is evident that the c_α value varies with the chosen frequency range and the value of D_{max} used for evaluation. Notably, c_α values exceeding 0.8 can be achieved across a low and broadband frequency range through strategic selection of frequency range and D_{max} . For instance, a HMHA with a thickness of merely 51 mm can theoretically achieve an average SAC of over

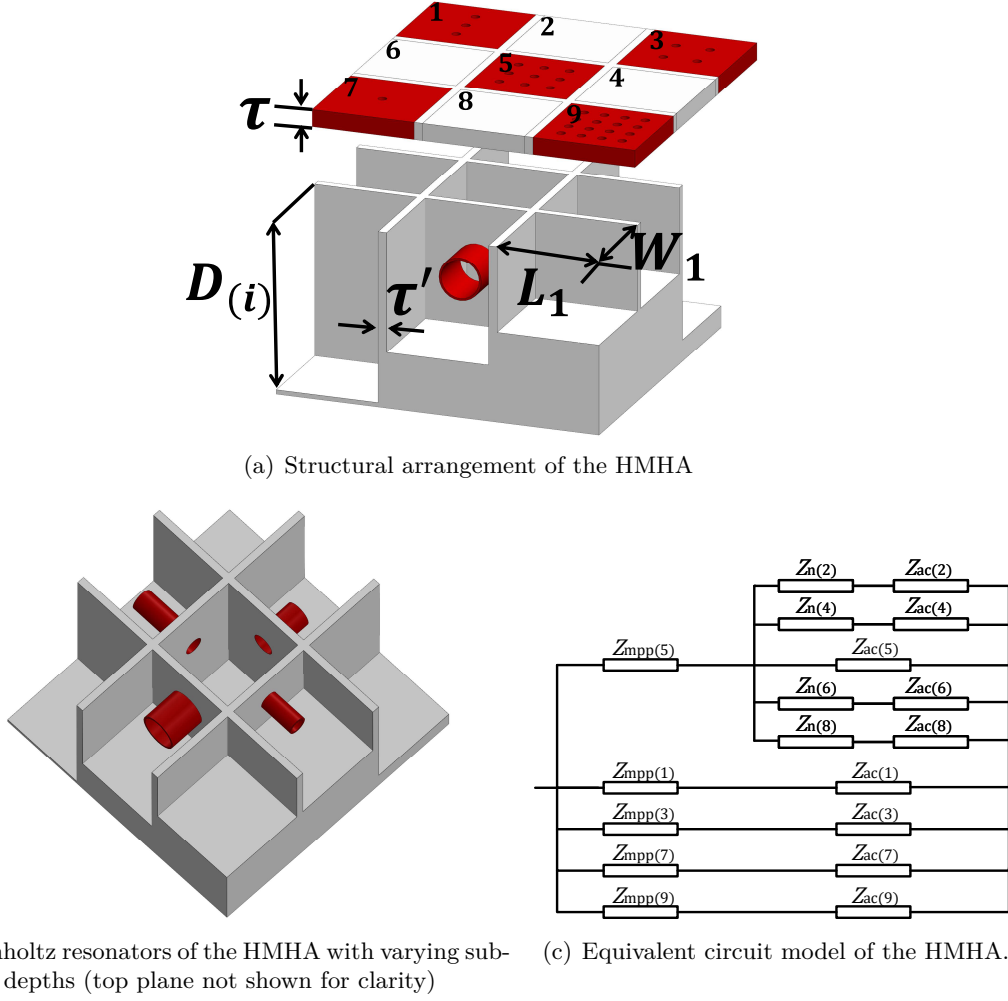


Figure 6.7: HMHA design.

0.75 at 200 to 500 Hz, approximately 0.82 at 250 to 500 Hz, and even surpass 0.84 at 300 to 500 Hz. A comparison of Figures 6.4 and 6.8 reveals that, despite having the same thickness, the low-frequency sound absorption bandwidth of the HMHA is significantly broader than that of the MHA. Figure 6.9 illustrates a comparison between the theoretical and numerical results, the latter obtained via COMSOL simulations, within identical frequency ranges for the HMHA variants with different absorber thicknesses. The numerical data corroborates the theoretical findings presented in Figure 6.8, thereby bolstering confidence in both the theoretical model and the optimisation algorithm employed.

6.4.2 Experimental validation of the optimised HMHA

An experimental validation of the HMHA design, featuring a maximum cavity depth of 50 mm (overall thickness of 51 mm), was conducted. Figure 6.10 displays a photograph of a

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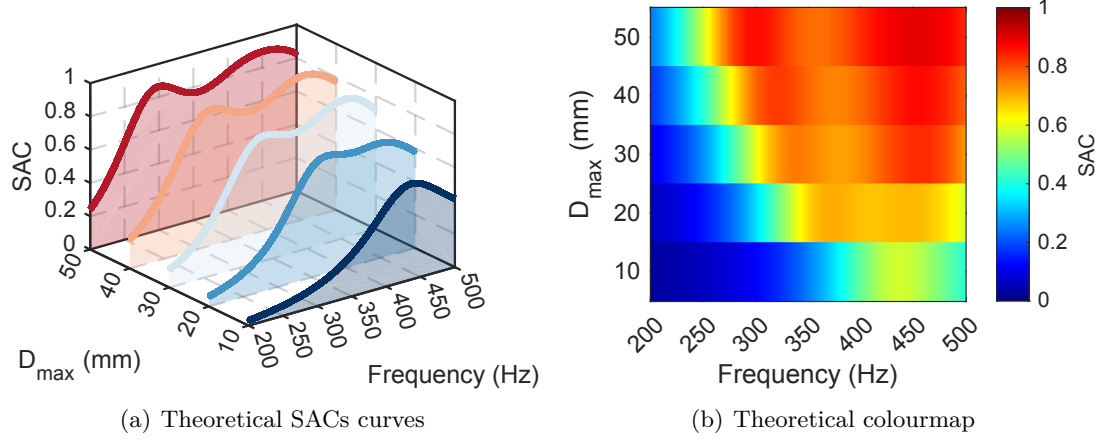


Figure 6.8: Theoretical SAC curves and colormap for optimised HMHA in [200 500] Hz with D_{max} : [10 50] mm. Corresponding geometric parameters are in Tables E.14 - E.18.

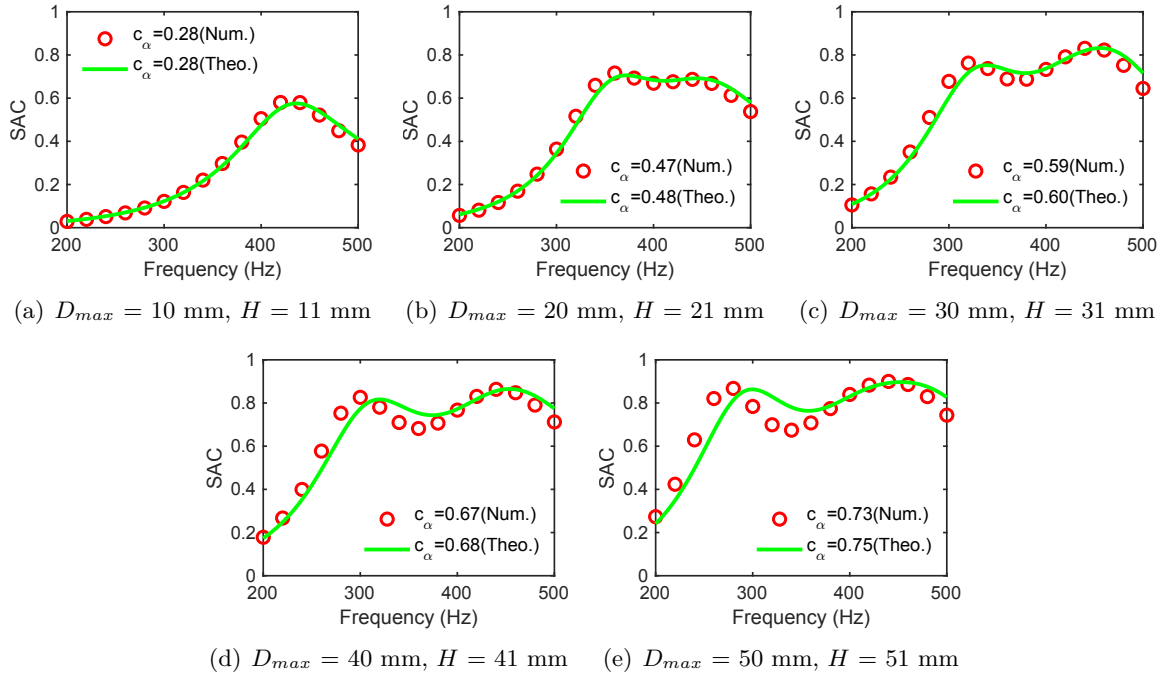


Figure 6.9: Theoretical and numerical SACs of HMHAs with different overall absorber thicknesses.

fabricated HMHA sample. As illustrated in Figure 6.10, the entire HMHA sample was mSLA 3D-printed as a single component, with the faceplate integrated directly into the print. This approach significantly reduces air leakage by eliminating the gap between the faceplate and the backing cavity, which was an issue in previous studies [55, 57, 58]. In addition, due to the intricate structure of the HMHA's core with extended neck HRs, which is significantly more complex than that of the MHA, the FDM 3D printing technology used for the MHA

Table 6.7: Average theoretical SACs of the HMHAs in different frequency ranges (data correspond to the SACs curves in Figure 6.8).

D_{max}	c_α in [200 500] Hz	c_α in [250 500] Hz	c_α in [300 500] Hz
10 mm	0.28	0.33	0.39
20 mm	0.48	0.55	0.63
30 mm	0.60	0.69	0.76
40 mm	0.68	0.76	0.80
50 mm	0.75	0.82	0.84

is unsuitable for the HMHA. The HMHA sample is placed at the end of an impedance tube (40 mm $\times$ 40 mm), and measurements are performed using the two-microphone method in accordance with ISO 10534-2 [153]. Details of the experimental setup are provided in B.1. The geometric parameters of the fabricated HMHA are presented in Table 6.8.

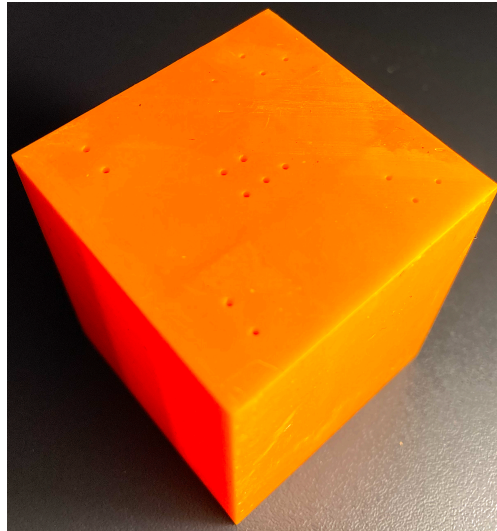


Figure 6.10: Photograph of a fabricated HMHA sample.

The experimental SAC curve, as shown in Figure 6.11(a), demonstrates that the HMHA achieves sound absorption of over 84% at the frequency of 305.2 Hz, despite its minimal thickness of only 51 mm—approximately $1/22.5$ of the sound's wavelength at this frequency. Moreover, it absorbs over 70% of sound within the frequency range of 273 Hz to 500 Hz, and over 50% from 246 Hz to 500 Hz. Notably, the HMHA achieves a high average SAC of 0.8 across the frequency range of 254 Hz to 500 Hz. The theoretical and experimental SAC curves agree very well.

The good agreement between the experimental and theoretical data of the HMHA is further demonstrated by examining its acoustic reactance and resistance, as shown in Figure

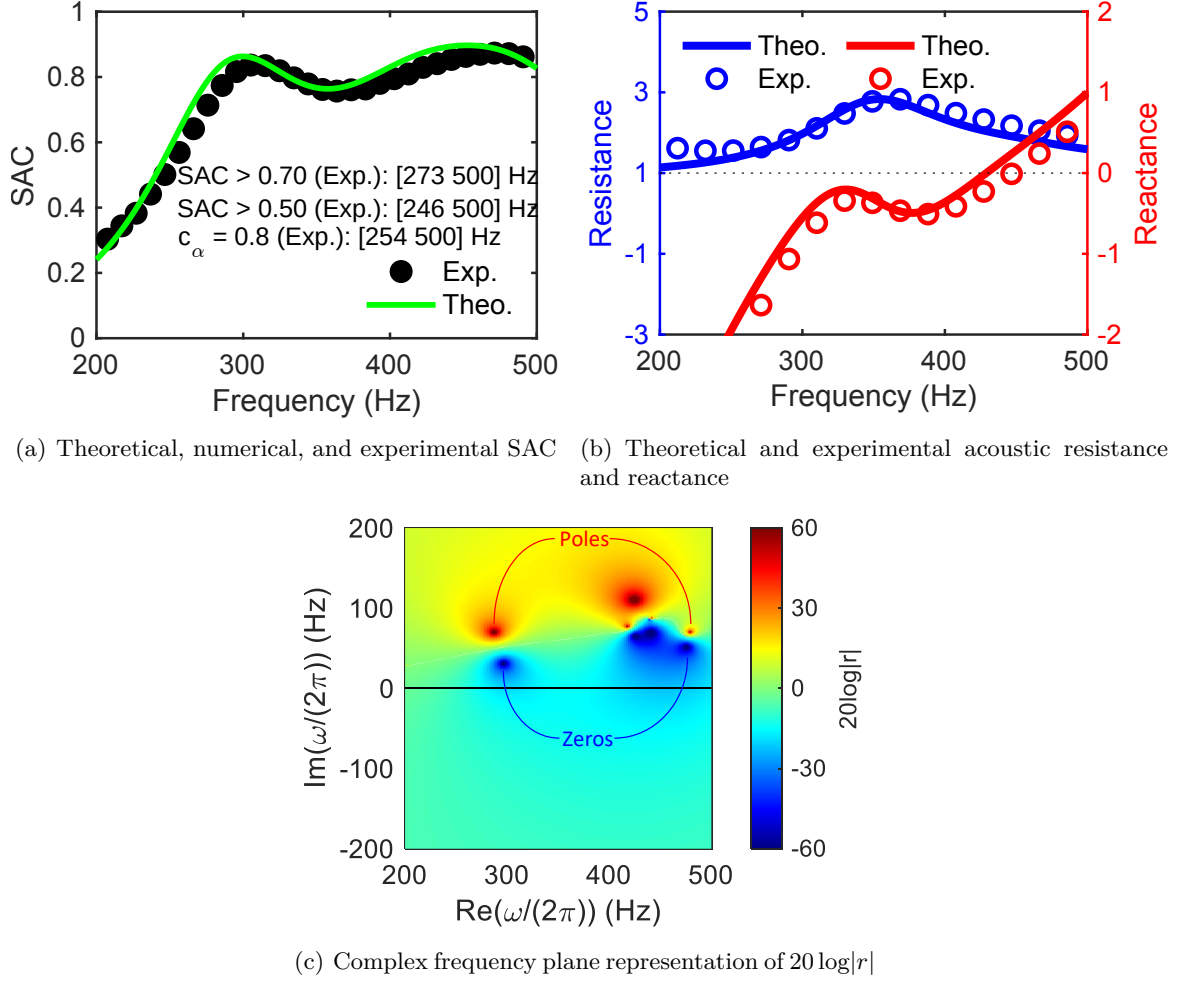


Figure 6.11: Acoustic performance the fabricated HMHA sample (51 mm thickness).

6.11(b). In Figure 6.11(b), the theoretical and experimental acoustic resistance and reactance agree well, corresponding to the well-matched theoretical and experimental SAC curves in Figure 6.11(a), thus confirming the accuracy of the proposed model. To be expected, the SAC value is highest when *both* the value of resistance and reactance are near their ideal values of 1 and 0 respectively.

To investigate the absorption characteristics and elucidate the relationship between energy leakage and energy loss, the framework of the complex frequency plane [163–166] is adopted. This approach introduces an imaginary component, $\text{Im}(\omega/(2\pi))$, to the frequency, allowing the theoretical model to be examined as a function of a complex frequency. This formulation enables the representation of the reflection coefficient, $20 \log(|r|)$, in the complex frequency domain, revealing its intrinsic physical properties. In this context, the points where $20 \log(|r|)$ reaches its minimum and maximum values are referred to as “zeros” and “poles”, respectively.

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It is important to note that the signs of the imaginary parts of these poles and zeros are determined by the time convention used—in this Chapter, $\exp(i\omega t)$. For a lossless system, the zeros and poles of the reflection coefficient would be symmetrically distributed about the real frequency axis, indicating that energy leakage is not counteracted by energy loss. Such a symmetry would indicate poor sound absorption performance. If energy loss were to increase or decrease, the zeros would shift upward, or downward, respectively. When the zeros approach the real frequency axis, energy loss nearly equals energy leakage, satisfying the critical coupling condition and achieving near-perfect absorption at that frequency. In our case, as shown in Figure 6.11(c), the zeros of the HMHA system are located above the real frequency axis, signifying that the system experiences greater energy loss than energy leakage (an overloss condition).

Table 6.8: Geometric parameters of the fabricated HMHA.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
0.79	2	0.92	2
$d_{(5)}$ (mm)	$n_{(5)}$	$d_{(7)}$ (mm)	$n_{(7)}$
0.85	5	0.67	3
$d_{(9)}$ (mm)	$n_{(9)}$	—	—
0.68	3	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
3.11	2.65	2.99	2.80
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	$d_{n(8)}$ (mm)	$L_{n(8)}$ (mm)
4.12	6.43	4.52	10.14
Cavity depths and panel thicknesses			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
50	50	50	50
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	$D_{(7)}$ (mm)	$D_{(8)}$ (mm)
50	50	50	50
$D_{(9)}$ (mm)	τ (mm)	τ' (mm)	—
50	1.00	1.00	—

6.4.3 Comparative analysis

In acoustic metamaterials, three performance metrics of interest include the target frequency range, the low-frequency sound absorption coefficients, and the overall structural thickness. Figure 6.12 displays a comparison of the sound absorption bandwidth (SAC > 0.7) for the MHA, HMHA, and other typical low-frequency and broadband HR-type sound absorbers found in the literature. In this figure, the red bars illustrate the data for the MHA and HMHA, sourced from Figure 6.6 and 6.11. The blue bar represents a compact acoustic absorber utilizing a Helmholtz resonator with an embedded spiral neck, as described by Guo et al. [167]. The yellow bar illustrates a sound-absorbing metasurface comprised of coupled Helmholtz resonators, explored in Li et al. [168]. The magenta bars denote two acoustic absorbers based on Helmholtz resonators with coiled-up channels, according to Wang et al. [169]. The green bar depicts an acoustic absorber that combines Helmholtz resonators with embedded necks and micro-perforations in parallel, as investigated in Ryoo et al. [101]. The cyan bars represent acoustic absorbers that incorporate a modified Helmholtz resonator, termed HREN, detailed in Vergara et al. [170]. It is crucial to acknowledge that all compared data are experimental and were extracted directly from figures in the cited literature using WebPlotDigitizer [124]. The relative acoustic performance of both the MHA and HMHA compare favourably when considering the bandwidth and low frequency range.

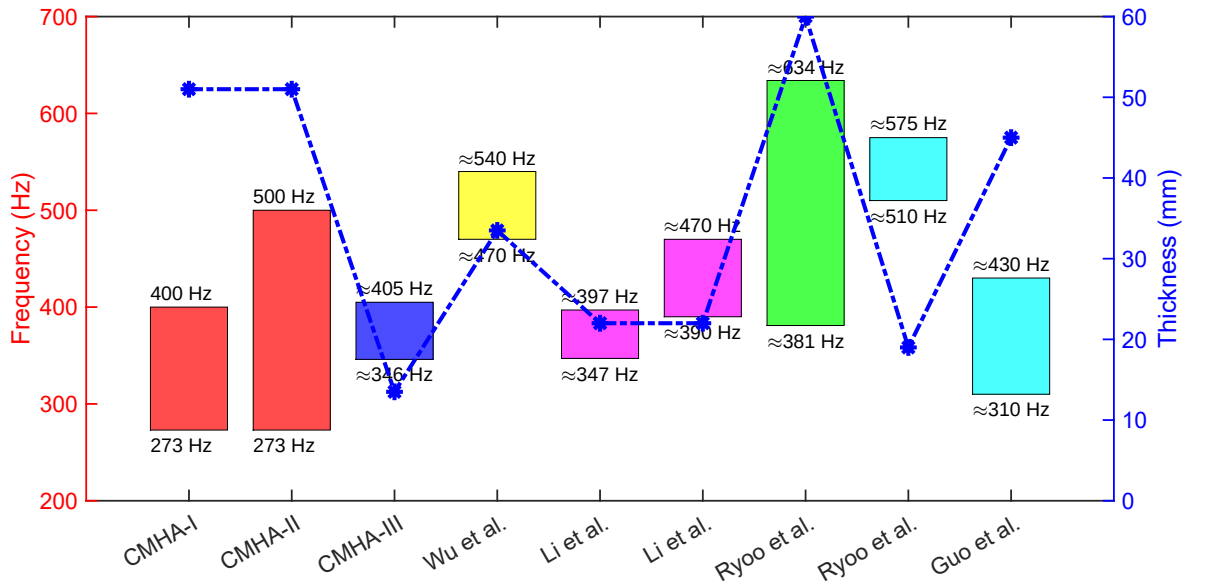


Figure 6.12: Comparison of sound absorption bandwidth (SAC > 0.7) among MHA, HMHA and other low-frequency and broadband HR-type sound absorbers.

6.5 Summary

This Chapter introduces an acoustic metamaterial that combines the sound absorbing characteristics of micro-perforated panels with extended neck Helmholtz resonators, referred to as MHA. To achieve a broader sound absorption bandwidth at low frequencies for the same foot print, a novel variant called the HMHA, comprising nine chambers, is developed. The HMHA can be theoretically conceptualized as an integration of four coupled MHAs and four micro-perforated panels absorbers into a more compact configuration, thereby enhancing acoustic resonance within the low-frequency spectrum and facilitating a coupling effect among its components. Both the MHA and HMHA are theoretically modeled using the equivalent circuit model method based on electro-acoustic analogies. The findings include:

1. **Efficient use of Space**

The proposed MHA and HMHA designs are novel due to their implementation of deep subwavelength scaling, which facilitates operation at low frequencies and enhances broadband noise absorption capabilities. Especially for the HMHA, which can be theoretically conceptualized as an integration of four MHAs and four micro-perforated panel absorbers into a more compact configuration, thereby maximizing space utilization and enhancing acoustic resonance within the low-frequency spectrum, while also facilitating coupling effects among its components. These designs are not only straightforward and simple to fabricate but are also expected to be stable over extended periods, making them practical for long-term applications in addition to being rigid and thus can be used as load bearing elements in designs such as liners.

2. **Tunable perfect low-frequency and broadband sound absorption.**

Theoretically, an MHA can achieve various degrees of sound absorption, which are contingent on its total thickness and the specific target frequency. The potential to tune such absorbers for perfect absorption makes them highly attractive for numerous practical applications. By varying the thickness of the MHA, one can finely adjust its absorption properties. Specifically, increasing the MHA's total thickness from 11.2 mm to 51.0 mm allows the peak absorption frequency to shift from 426 Hz to 334 Hz. Experimental results demonstrate that at a central frequency of 329.6 Hz and with a thickness of 51 mm —approximately $1/20.4$ of the wavelength of the sound at this frequency —the MHA achieves near-perfect sound absorption, exceeding 99.9%.

Additionally, it exhibits 70% sound absorption within the 273 Hz to 400 Hz frequency range, and 50% from 249 Hz to 447 Hz.

3. Broader low-frequency sound absorption of HMHA

Expanding on the MHA design, the HMHA significantly enhances the operational bandwidth, achieving effective sound absorption across a broader low-frequency range. Similar to the MHA, the HMHA also utilizes a deeply subwavelength thickness, making it ideal for compact applications. Experimental results confirm that the HMHA achieves substantial sound absorption, exceeding 84% at the central frequency of 305.2 Hz, with a minimal thickness of only 51 mm—approximately $1/22.5$ of the wavelength at this frequency. Additionally, it absorbs 70% of sound within the 273 Hz to 500 Hz frequency range, and 50% between 246 Hz to 500 Hz. Notably, the HMHA maintains an impressive average SAC of 0.8 across the 254 Hz to 500 Hz range.

4. Sensitivity Studies

Sensitivity studies of geometric parameters are conducted as an effective method to accurately determine the key parameters that significantly influence the absorption properties of the MHA. It is noted that when the frequency range is sufficiently large the absorption peaks de-couple which facilitated the sensitivity analysis on a parameter-by-parameter basis. By examining a smaller frequency range, the responses couple allowing the absorption range to be broader which is highly advantageous.

5. Various Industrial Applications

Both the MHA and the HMHA absorbers demonstrate significant potential for various industrial applications, including noise control, duct noise reduction e.g. HVAC (Heating, Ventilation, and Air Conditioning) or aeroengine liners, acoustic filtering, and room noise attenuation. Their effectiveness in these capacities could drive substantial advancements in the field due to their robust and relatively clean design.

Our results indicate that both the MHA and the HMHA offer promising solutions for enhancing acoustic absorption across various environments. MHA features tunable, perfect low-frequency and broadband sound absorption within compact and efficient designs. Similarly, HMHA provides broader low-frequency operational sound absorption bandwidth alongside compact, efficient frameworks.

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The small foot print and low frequency range means that the technology can be treated as a locally reacting absorber above the appropriate frequency threshold.

Chapter 7

Deep Sub-Wavelength Broadband Microperforated Acoustic Metasurfaces

7.1 Introduction

In Chapter 6, an acoustic metasurface combining a micro-perforated panel with an extended-neck Helmholtz resonator was developed to achieve broadband sound absorption below 500 Hz. This was an encouraging result for addressing low-frequency noise control.

However, developing low-frequency broadband absorbers at a deep subwavelength scale—where the physical thickness of the absorber is significantly smaller than the wavelength of the targeted sound—remains one of the foremost challenges in acoustic engineering. This Chapter further explores the potential of the MHA-based acoustic metasurface to achieve efficient sound absorption at ultra-low frequencies, specifically below 200Hz at a deep subwavelength scale, while also maintaining broadband absorption performance in the 200–1000Hz range. In addition, the sound transmission loss performance of the proposed metasurface is experimentally evaluated under different grazing flow conditions. The results demonstrate that broadband transmission loss below 1000 Hz can be achieved despite its compact thickness of only 32 mm.

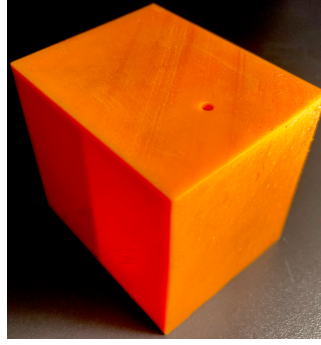
The proposed design not only operates effectively at a deep subwavelength scale but also exhibits robust broadband transmission loss performance, making it a promising candidate for next-generation acoustic liners in practical engineering applications.

7.2 Deep Sub-Wavelength Design

7.2.1 Experimental validation

In this Chapter, a single MHA experimental sample, designated MHA-I, was optimised for the single target frequency of 150 Hz and subsequently fabricated. The MHA-I sample was 3D-printed using an mSLA printer, with a photograph to be seen in Figure 7.1(a). The maximum depth of the sub-chamber air cavity is 45 mm, and the total thickness, including the facesheet thickness of τ , is 48 mm. The optimisation process and optimised geometric parameters of the MHA-I sample are provided in the Appendix E.6. The sound absorption coefficient and acoustic impedance of the MHA-I experimental sample were measured using the two-microphone tube in accordance with the ISO 10534-2 standard [153]. Details of the experimental setup are provided in the Appendix B.1. To further validate the theoretical model, a three-dimensional finite element model was developed using the thermo-viscous acoustic module in COMSOL Multiphysics 6.0, as illustrated in Figure 7.2(a). As for this case, the MHA-I model exhibits symmetry in the xy-plane, to optimise computational efficiency,

only one half of the MHA-I geometry was simulated, with symmetry boundary conditions applied to the corresponding plane. The thermo-viscous acoustic module was implemented to model all air domains, including the Helmholtz extended neck, aperture, air cavities (AC1 and AC2), and the upstream region of the tube. A normally incident plane wave pressure field was generated upstream of the test section. All structural walls of the MHA are represented by sound hard boundaries. Figure 7.2(b) illustrates the details of the meshed domains. Details of the mesh generation process can be found in the Section 7.2.3.



(a) Photograph of MHA-I with a thickness of 48 mm

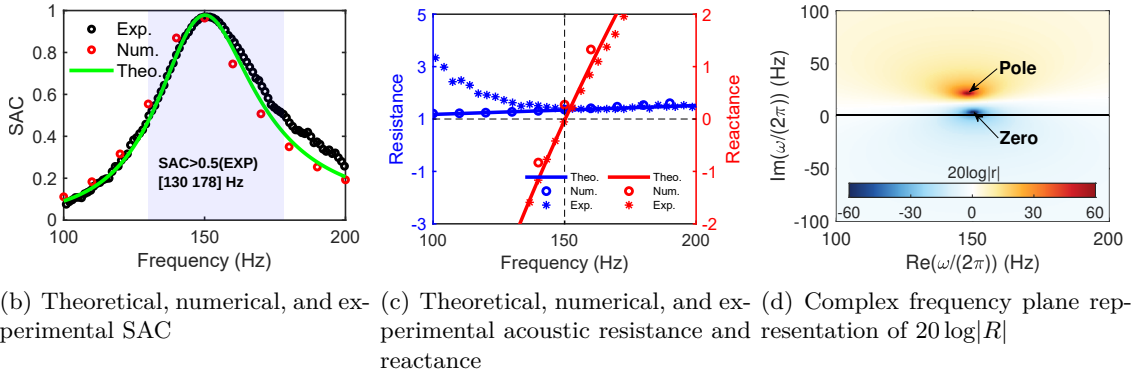


Figure 7.1: Photograph and acoustic performance of MHA-I with a thickness of 48 mm.

7.2.2 Deep sub-wavelength acoustic absorption

Figure 7.1(b) illustrates the theoretical, numerical, and experimental sound absorption coefficients of MHA-I. The theoretical and numerical results show excellent agreement with the experimental data. This consistency validates the accuracy of both the theoretical model and the finite element model. The fabricated MHA-I sample demonstrates near-perfect sound absorption ($SAC > 0.97$) at its resonance frequency, $f_r=150$ Hz, with a thickness of only 48 mm (less than two inches), which corresponds to approximately $1/48$ of the operating wavelength. To characterize the subwavelength nature of the resonator while explicitly considering

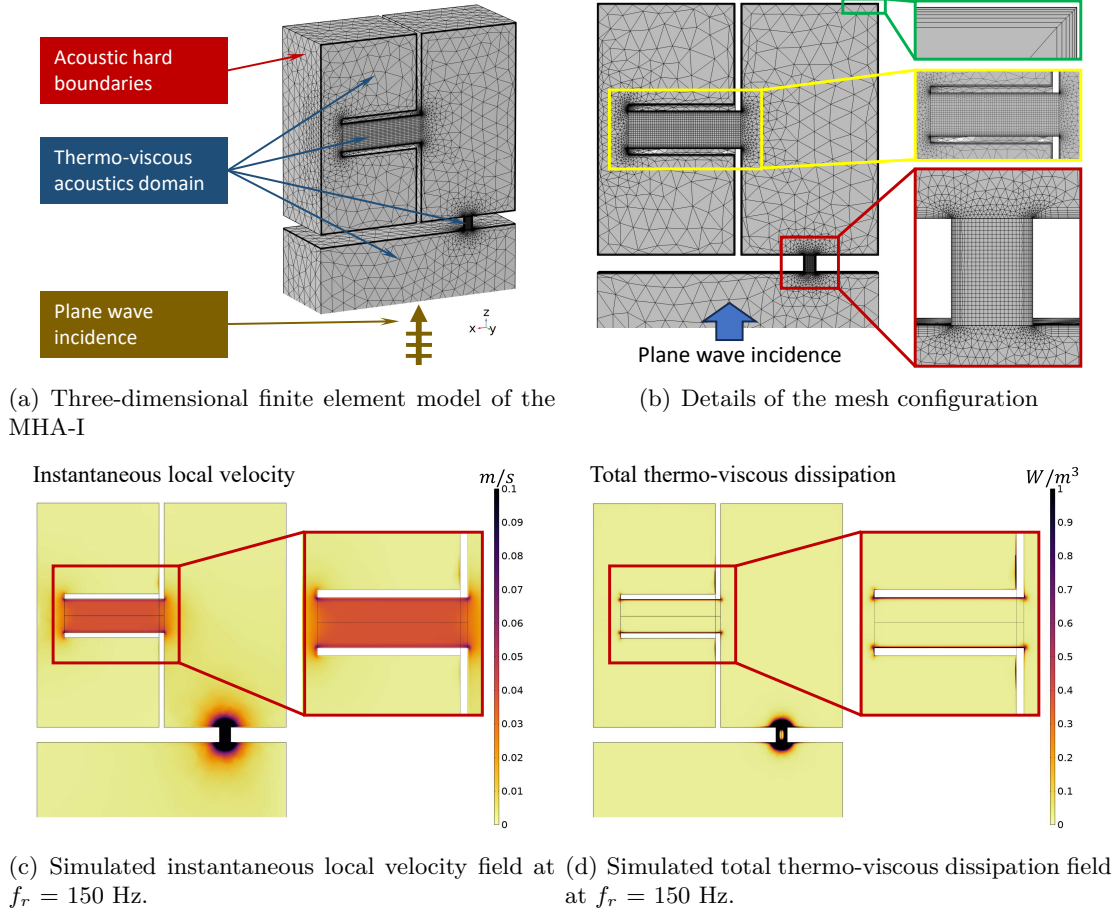


Figure 7.2: Three-dimensional finite element model of MHA-I and the corresponding simulation results.

its total volume, a volume-normalized wavelength ratio is defined as

$$\Gamma = \frac{V^{1/3}}{\lambda} \quad (7.1)$$

where V represents the total volume of the resonator. This parameter reflects the overall volumetric constraint, rather than focusing solely on the resonator's thickness. A smaller Γ indicates a more compact subwavelength design, demonstrating the capability to achieve low-frequency resonance within a minimized resonator volume. The total volume of MHA-I is $V = 40 \text{ mm} \times 50 \text{ mm} \times 48 \text{ mm}$, and thus the volume-normalized wavelength ratio is $\Gamma = 1/51$.¹

¹In the published article (<https://doi.org/10.1063/5.0258251>), the total volume of MHA-I was incorrectly reported as $V = 40 \text{ mm} \times 40 \text{ mm} \times 48 \text{ mm}$, leading to an erroneous value of $\Gamma = 1/54$. This has been corrected in the present thesis.

Additionally, the sample exhibits a sound absorption coefficient exceeding 0.5 over the frequency range of 130–178 Hz, yielding a frequency interval $\Delta f = 178 - 130 = 48$ Hz. This corresponds to an absorption bandwidth of 32% ($\Delta f / f_r$), indicating excellent low-frequency sound absorption performance at sub-wavelength scales. The acoustic absorption capabilities of the MHA-I can also be explained by its acoustic reactance and resistance, as illustrated in Figure 7.1(c). An ideal impedance match is achieved when the normalized acoustic reactance is zero, and the acoustic resistance equals one. As shown in Figure 7.1(b), at $f_r = 150$ Hz, the experimental acoustic reactance approaches zero, and the experimental acoustic resistance nears one, consistent with the experimental absorption curve at this frequency, where the experimental SAC is close to unity.

To effectively analyze absorption characteristics and illustrate the relationship between energy leakage and energy loss, along with their influencing factors, the concept of the complex frequency plane [163–165] is employed. It is crucial to note that the signs of the imaginary parts of poles and zeros depend on the time convention, specifically $\exp(i\omega t)$ in this Chapter. In the current work, as shown in Figure 7.1(d), a zero of the MHA-I is located near the real frequency axis at $f_r = 150$ Hz, where the system’s energy loss nearly balances its energy leakage. This balance satisfies the criteria for critical coupling, resulting in near-perfect absorption at this frequency.

7.2.3 Numerical validation

To further investigate the perfect sound absorption mechanism of the MHA-I at the resonance frequency $f_r = 150$ Hz, Figure 7.2(c) and 7.2(d) present the instantaneous velocity field and the total thermo-viscous dissipation rate distribution in the symmetry plane of the MHA-I model. At the resonance frequency, as shown in Figure 7.2(c), the maximum instantaneous velocity is observed within the perforation, followed by the walls of the Helmholtz neck. Significant velocity is also detected near the entrance and exit edges of the perforation and the Helmholtz neck, which can be attributed to geometric discontinuities, commonly referred to as radiation effects [167]. Similarly, Figure 7.2(d) shows that most acoustic energy dissipation occurs near the walls of the perforation and the Helmholtz neck. Energy dissipation is also significant at the entrance and exit edges of the perforation and the Helmholtz neck, corresponding closely to the regions of strong velocity shear shown in Figure 7.2(c). Compared to a conventional Helmholtz resonator, the inclusion of the perforation in the proposed absorber markedly increases the system’s viscous losses. Another point of interest to be seen in Figure

7.2(d), is that losses can be also observed on the cavity wall. As the cavity wall losses are usually not modeled in such studies, this justifies the extra effort in doing so in this current study.

A user-controlled mesh was employed in this Chapter. The overall meshing strategy reflects a physics-driven, region-specific refinement approach, designed to balance the need for accuracy in capturing thermoviscous effects with the need for practical computational efficiency.

In the bulk region of the computational domain, a free tetrahedral mesh was used, with a maximum element size of $1/507$ of the wavelength, which is considerably finer than the commonly recommended $1/12$ of the wavelength in COMSOL's acoustics module. This conservative setting was chosen to minimize numerical dispersion and phase errors, particularly at low frequencies.

In the mesh generation process, the viscous boundary layer thickness, δ_{visc} , defined as: $\delta_{\text{visc}} = \sqrt{\frac{2\mu}{\omega\rho_0}}$ is an important controlling parameter. As the frequency range in this initial study is $[100, 200]$ Hz, at the upper limit of 200 Hz, when the wavelength is shortest, $\delta_{(\text{visc}, 200 \text{ Hz})}$ is approximately 0.155 mm.

In regions near the boundary layer, where viscous and thermal dissipation effects are significant, a sufficiently refined mesh is required to accurately resolve the thermoviscous effects. In the current study, for the walls of the perforations, the Helmholtz neck where the velocity gradient is steep and for the internal cavity surfaces, the meshing sequence used *boundary layer operations* in COMSOL to ensure sufficient elements. The *boundary layer operations* settings are typically optimised as follows:

1. Number of boundary layers: 6 layers to resolve the acoustic boundary layer near the neck.
2. Stretching factor: A value of 1.2, means that the thickness increases by 20% from one layer to the next. This field is used to specify the increase in thickness between two consecutive boundary layers as a scaling factor.
3. First layer thickness: The first layer near the neck should be small enough to capture the acoustic effects, typically set to $\delta_{(\text{visc}, 200 \text{ Hz})}/5 = 0.031$ mm, which was considered to be sufficiently fine.

The *Swept* mesh technique in COMSOL was employed for the perforations and the Helmholtz neck. This approach reduces the total number of mesh elements, particularly in regions such as the perforations of the micro-perforated panel and the neck of the Helmholtz resonator, where the geometry is relatively simple and elongated. By sweeping the mesh from one end of the neck to the other, this method optimises the element count, enhances simulation speed, and reduces memory usage, all without sacrificing accuracy. By combining free tetrahedral meshing, boundary layer meshing, and swept meshing techniques, the final mesh configuration was optimised to capture both the global acoustic field and the localized thermoviscous boundary layer effects with minimal numerical error and practical computational cost.

7.2.4 Effect of extended neck position on MHA's acoustic performance

To further investigate the effect of the extended neck position on the acoustic performance of the structure, additional simulations were conducted to examine the influence of various neck positions. In these simulations, three extended neck positions were tested, as shown in Figure 7.3:

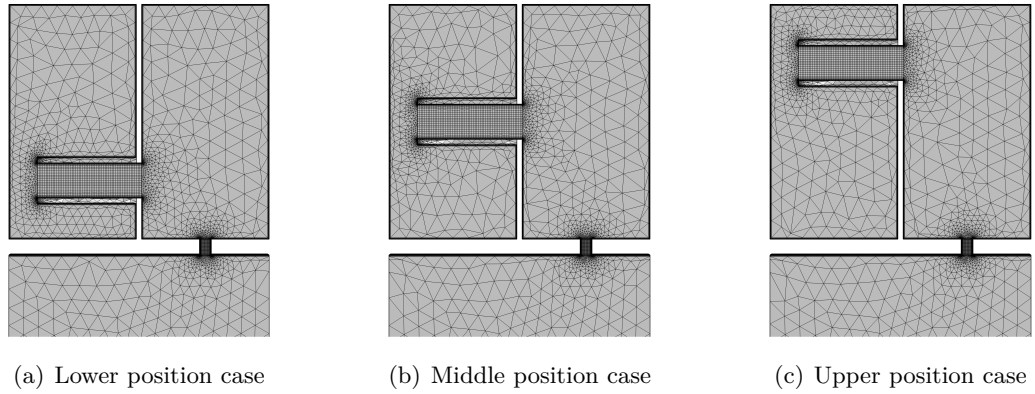


Figure 7.3: Three different MHA's extended neck position cases.

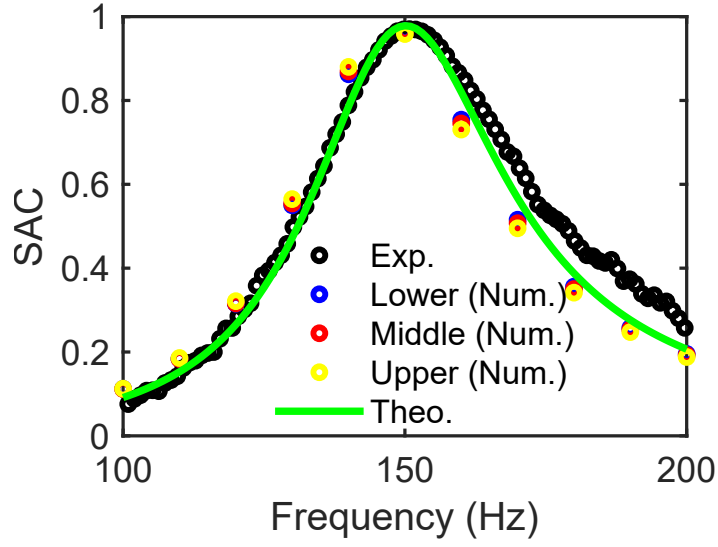
1. Lower position case: The distance from the center axis of the neck to the bottom of the AC2 is $3/4$ of D_{ac2} .
2. Middle position case: The distance from the center axis of the neck to the bottom of the AC2 is $1/2$ of D_{ac2} .
3. Upper position case: The distance from the center axis of the neck to the bottom of the AC2 is $1/4$ of D_{ac2} .

Figure 7.4(a) presents the numerical SAC results obtained for the three cases. The differences among these cases are minimal, and all numerical results align well with both the theoretical and experimental SAC results. Additionally, it can be observed that the numerical SAC of the Lower Position case is slightly closer to the theoretical and experimental results, although the advantage is marginal.

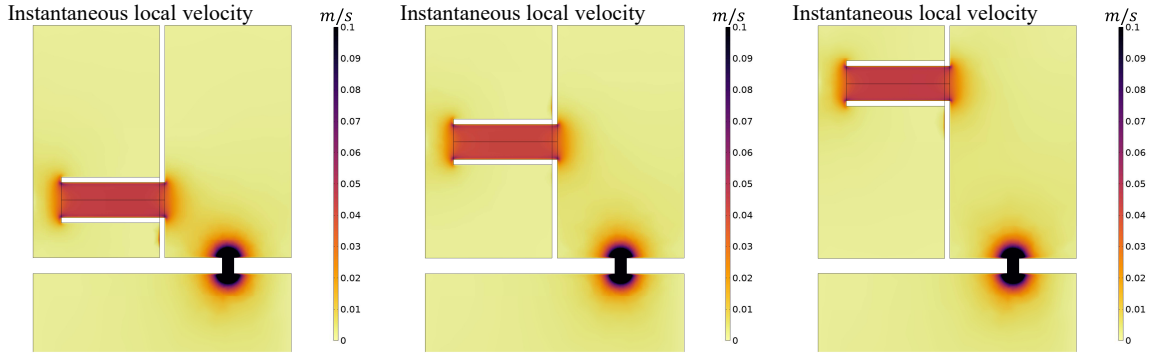
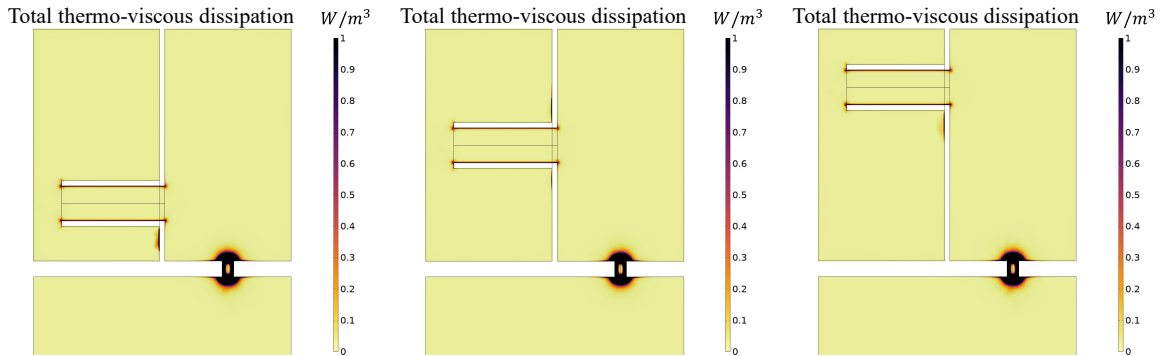
Figures 7.4(b)–7.4(g) show the instantaneous velocity field and the total thermo-viscous dissipation rate distribution in the symmetry plane of the MHA-I model for three cases.

In all three cases, as shown in Figures 7.4(b)–7.4(d), the maximum instantaneous velocity at the resonance frequency occurs within the perforation, followed by the walls of the Helmholtz neck. Significant velocity is also observed near the entrance and exit edges of both the perforation and the Helmholtz neck. Additionally, instantaneous velocity can also be observed on the cavity walls in all cases.

In all three cases, as shown in Figures 7.4(e)–7.4(g), acoustic energy dissipation at the resonance frequency primarily occurs near the walls of perforation and the Helmholtz neck. Additionally, substantial dissipation is observed at the entrance and exit edges of both the perforation and the Helmholtz neck, coinciding with regions of strong velocity shear. Energy dissipation can also be observed on the cavity walls. This is most significant on the AC2 cavity wall immediately below the neck which is a point of interest.



(a) Numerical SAC for three position cases

(b) Simulated instantaneous local velocity field at $f_r = 150$ Hz in the Lower position case(c) Simulated instantaneous local velocity field at $f_r = 150$ Hz in the Middle position case(d) Simulated instantaneous local velocity field at $f_r = 150$ Hz in the Upper position case(e) Simulated total thermo-viscous dissipation field at $f_r = 150$ Hz in the Lower position case(f) Simulated total thermo-viscous dissipation field at $f_r = 150$ Hz in the Middle position case(g) Simulated total thermo-viscous dissipation field at $f_r = 150$ Hz in the Upper position case**Figure 7.4:** Numerical SAC, along with the simulated instantaneous local velocity field and total thermo-viscous dissipation field at $f_r = 150$ Hz for three position cases.

7.3 Compact MHA

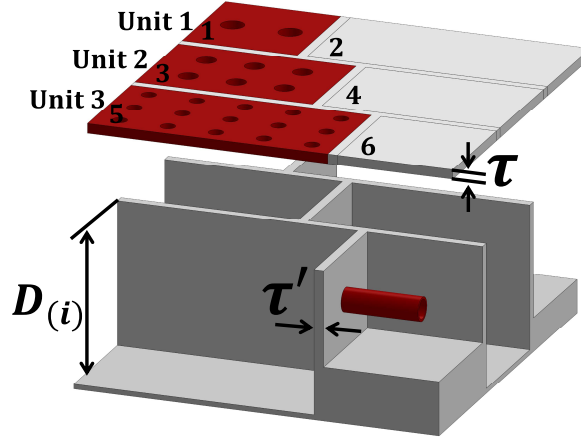
7.3.1 Design

In order to broaden the sound absorption bandwidth of the MHA while keeping the total area constant and with the additional objective of further reducing the total thickness, a variation involving three individual MHA units, as illustrated in Figure 7.5, was developed, termed the Compact MHA (CMHA). As shown in Figure 7.5(a), the chambers are sequentially numbered from 1 to 6. Each chamber in this design can have a unique depth, denoted as $D_{(i)}$, where i is the chamber number. The number of perforations and their diameter for each MPP are allowed to vary. These dimensions are denoted as $n_{(i)}$ and $d_{(i)}$ respectively. The internal diameter and length of each Helmholtz resonator's neck in this Chapter are also variable. These dimensions are denoted as $d_{n(i)}$ and $L_{n(i)}$, respectively. The lengths of each chamber are denoted by L_1 to L_6 . The width of each chamber is identical, denoted by W_1 . The thickness of the *top* plate in the CMHA is symbolized as τ , whereas the thickness of the *internal* adjacent panels and the wall thickness of the Helmholtz resonator's neck are represented by τ' .

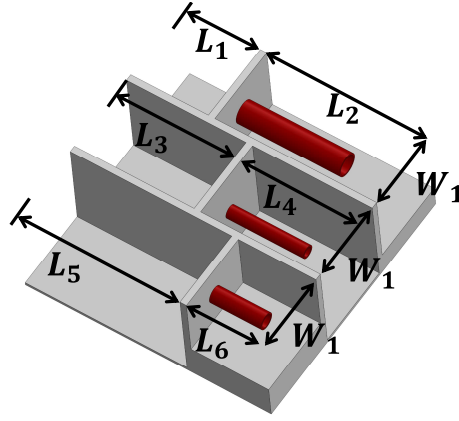
The total thickness, H , of the CMHA is obtained by summing τ and the maximum sub-chamber depth, $D_{\max}$. The total specific impedance of the CMHA can be derived from the equivalent circuit model, depicted in Figure 7.5(c).

7.3.2 Experimental validation

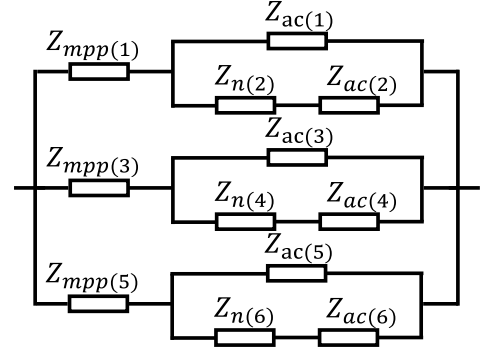
In this Chapter, three CMHAs are designed and fabricated, designated as CMHA-I, CMHA-II, and CMHA-III. CMHA-I is optimised to achieve broadband sound absorption within the range of [200–500] Hz, CMHA-II within [500–1000] Hz, and CMHA-III within [200–1000] Hz. The broadband optimisation process and the optimised geometric parameters of the CMHA samples are detailed in the Appendix E.6. The CMHA samples are manufactured once again using the mSLA 3D printing. Figure 7.6(a) shows a photograph of the fabricated CMHA-II. The theoretical and experimental sound absorption curves for CMHA-I, CMHA-II, and CMHA-III are shown in Figures 7.6(b)–7.6(d), demonstrating generally excellent agreement. However, some discrepancies exist between the theoretical and experimental results, particularly for CMHA-I below 250 Hz and CMHA-III below 300 Hz. The primary factors contributing to these discrepancies include (1) potential sealing issues between the sample and the impedance tube, and (2) uncertainties in the manufacturing process. The



(a) Structural arrangement of the CMHA



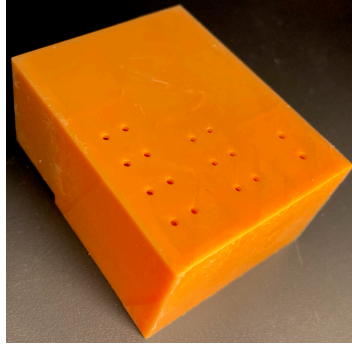
(b) Helmholtz resonators of the CMHA with varying sub-chamber depths (top plane not shown for clarity)



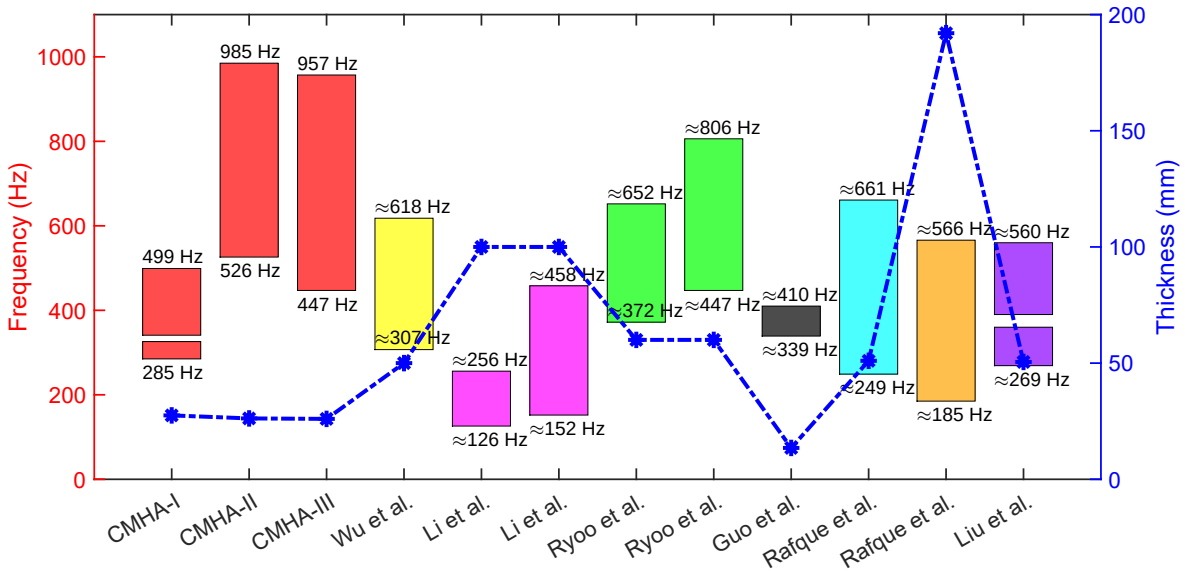
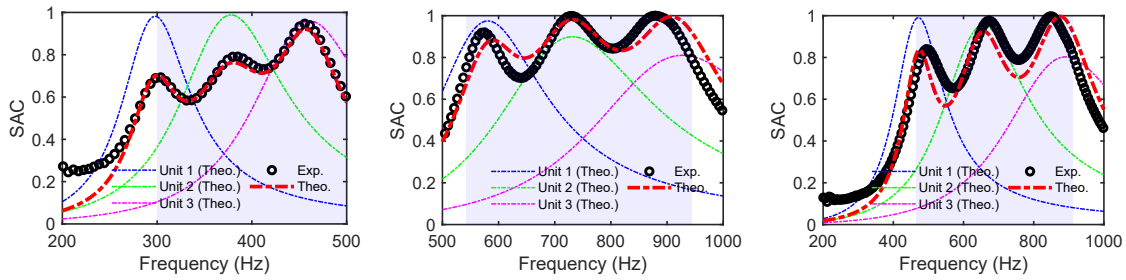
(c) Equivalent circuit model of the CMHA

Figure 7.5: CMHA design.

theoretical absorption curves of individual units in the CMHA are also presented in order to provide some insight into the benefit of the coupling effect. When these units are integrated, the relative absorption bandwidth is significantly broadened in all three CMHA cases due to coupling mechanisms between the units. Moreover, integration results in a reduction in the magnitude of the low-frequency absorption peaks, while the high-frequency absorption peaks tends to increase compared to the magnitude of the individual unit responses. Effective tunable sound absorption can be achieved by varying the geometric parameters of the CMHA. Experimental results demonstrate that CMHA-I, with an overall thickness of 27.5 mm, achieves an average SAC exceeding 0.74 in the frequency range of 300–500 Hz. Similarly, CMHA-II, with a thickness of 26.2 mm, achieves an average SAC exceeding 0.88 in the range of 543–945 Hz. CMHA-III, with an overall thickness of 26.0 mm, achieves an average SAC exceeding 0.84 in the range of 467–912 Hz.



(a) Photograph of CMHA-II



(e) Comparison with the literature of experimental results

Figure 7.6: Photograph of CMHA-II, theoretical and experimental SACs of CMHA-I, CMHA-II, and CMHA-III, and comparison of experimental results with literature data.

7.3.3 Comparative analysis

Figure 7.6(e) compares the sound absorption bandwidth (SAC > 0.6) of CMHA samples with other typical low-frequency, broadband MPP and Helmholtz resonator acoustic metamaterials

reported in the literature. The evaluation of acoustic performance is based on the target frequency range, low-frequency sound absorption coefficients, and overall structural thickness. In Figure 6.12, the red bars represent the data for the CMHA samples, extracted from Figures 7.6(b)–7.6(d). The yellow bar corresponds to a hybrid sound absorber composed of a micro-perforated panel and coiled-up Fabry–Pérot channels, as investigated by Wu et al. [111]. The magenta bars indicate two acoustic absorbers consisting of a perforated panel with parallel-arranged extended tubes, either with or without an MPP, as described by Li et al. [171]. The green bar denotes an acoustic absorber integrating Helmholtz resonators with embedded necks and micro-perforations in parallel, as studied by Ryoo et al. [101]. The dark bar represents a compact acoustic absorber employing a Helmholtz resonator with an embedded spiral neck, as reported by Guo et al. [167]. The cyan bar corresponds to a single-layer absorber comprising four parallel-arranged inhomogeneous micro-perforated panels, examined by Rafique et al. [172]. The orange bar signifies a double-layer absorber consisting of multiple parallel-arranged inhomogeneous micro-perforated panels, as explored by Rafique et al. [173]. The violet bar corresponds to a sound absorber based on MPP and impedance matching coiled-up cavity, examined by Rafique et al. [172]. The violet bar represents a sound absorber formed by the parallel connection of different units based on an MPP and an impedance-matching coiled-up cavity, as examined by Liu et al. [174]. In Figure 6.12, the blue asterisks represent the corresponding total thickness of the absorber. Notably, all compared data are experimental and were extracted directly from the cited literature using WebPlotDigitizer [124]. The CMHA samples exhibit favorable acoustic performance in terms of bandwidth, low-frequency absorption, and overall thickness compared to the referenced designs.

7.4 Acoustic Behavior Under Grazing Flow Conditions

The sound transmission loss performance of the CMHA under grazing flow conditions was subsequently evaluated. The optimisation was carried out using the Cremer impedance-based cost function proposed in Section 5.4, which aims to optimise the absorber’s impedance over a broadband frequency range of interest, considering various grazing flow Mach numbers. As previously demonstrated, the CMHA exhibits excellent low-frequency noise absorption performance. Therefore, the target frequency range was set from 200 Hz to 1200 Hz, and the maximum liner thickness was limited to only 32 mm. Details of the optimised CMHA liners

are provided in Table 7.1. Subsequently, the liners CMHA-00 and CMHA-25 were fabricated using a Masked Stereolithography Apparatus 3D printing technique. The grazing flow test duct in the acoustics laboratory at Le Mans University was then used to measure the STL under grazing flow conditions. Photograph of the tested liner CMHA-25 can be find in Figure 7.7. The grazing flow facility test matrix can be find in Table 7.2. The employed 3D printing method and the detailed description of the grazing flow test rig can be found in Section 5.6.

Table 7.1: Details of optimised CMHA liners, $H=32$ mm.

Liner	Target frequency range	Target Mach number	Geometric parameters
CMHA-00	[200 1200] Hz	$M=0$	Table E.23
CMHA-25	[200 1200] Hz	$M=0.25$	Table E.24

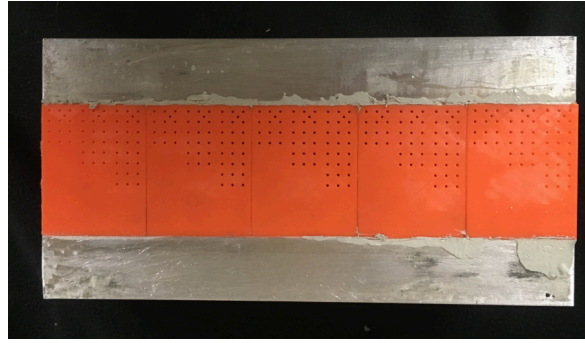


Figure 7.7: Photo of liner CMHA-25.

Figure 7.8 presents the experimentally measured TL results of the four liners tested, see Table 7.2, in the grazing flow rig but in quiescent flow conditions $M = 0$. In all cases, the TL decreases with increasing incident acoustic level. For both the CMHA-00 and CMHA-25 liners, the TL responses are quite similar, with values ranging between 10 dB and 23 dB. Although CMHA-25 was originally designed for a flow condition of $M=0.25$, it is applied here in a no-flow scenario. Both designs achieve broadband TL. A closer examination reveals that CMHA-00 provides a broader TL bandwidth compared to CMHA-25. For instance, at an incident acoustic level of 100 dB, CMHA-00 exhibits TL greater than 5 dB over the frequency range of approximately [540 Hz, 1055 Hz], while CMHA-25 exceeds 5 dB TL in the narrower range of approximately [540 Hz, 990 Hz].

Figure 7.9 shows the measured transmission loss at Mach numbers 0, 0.15, and 0.25. Measurements were conducted at SPLs of 130 and 140 dB to ensure adequate signal-to-noise ratio, with the source placed both upstream and downstream of the liner. For cases

Table 7.2: Grazing flow facility test matrix (Check marks indicate tests conducted at corresponding Mach numbers and sound pressure levels).

Mach number	without flow				$M=0.15$		$M=0.25$	
Sound pressure level	100 dB	120 dB	130 dB	140 dB	130 dB	140 dB	130 dB	140 dB
CMHA-00 liner	✓	✓	✓	✓	✓	✓	✓	✓
CMHA-25 liner	✓	✓	✓	✓	✓	✓	✓	✓

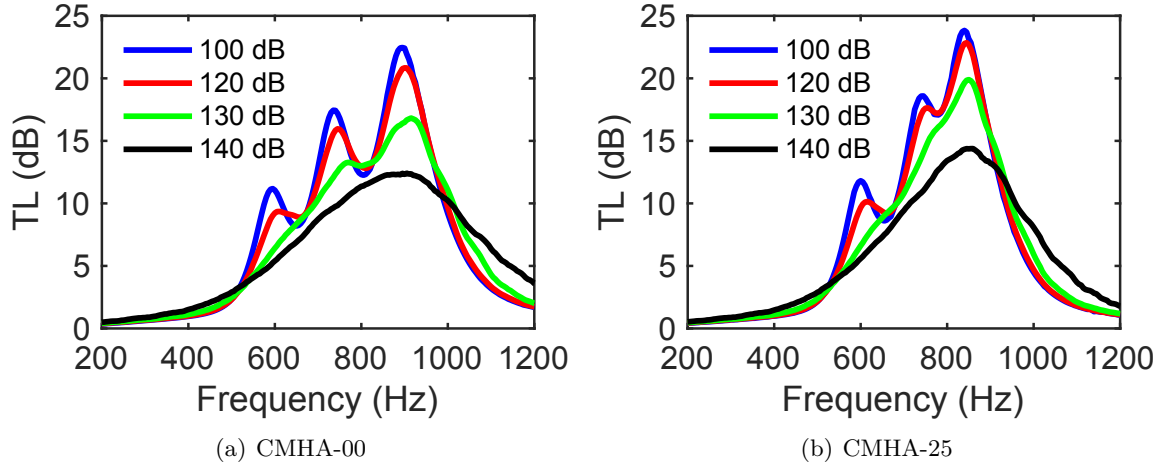


Figure 7.8: Experimental results. $M = 0$. TL as a function of SPL. Liners CMHA-00 and CMHA-25.

with flow ($M = 0.15$ and 0.25), the transmission loss remains similar between the two SPL levels, regardless of source position, aligning with previous findings that flow dominates over nonlinear acoustic effects [155].

When the acoustic source is located upstream, increasing the Mach number reduces the transmission loss and shifts the frequency response towards higher frequencies. In contrast, when the source is placed downstream of the liner, the results differ significantly. As shown in Figure 7.9, for liners CMHA-00 and CMHA-25, an increase in flow speed does not lead to a notable reduction in transmission loss; instead, a broadening of the effective frequency bandwidth is observed. Regarding TL levels, under the highest SPL and Mach number conditions, a value of approximately 5 dB is obtained when the source is upstream, whereas a substantially higher TL of around 15 dB is achieved when the source is downstream, indicating effective attenuation under challenging flow conditions.

Figure 7.10 presents a comparison between the predicted and experimental results. Further details about the numerical prediction model can be found in Section 5.5, while the specific modeling of the CMHA within this framework is described in Section 6.2.2. For the MPPs, the boundary condition of the internal perforated thin panel, as defined by Eq. (6.3) in

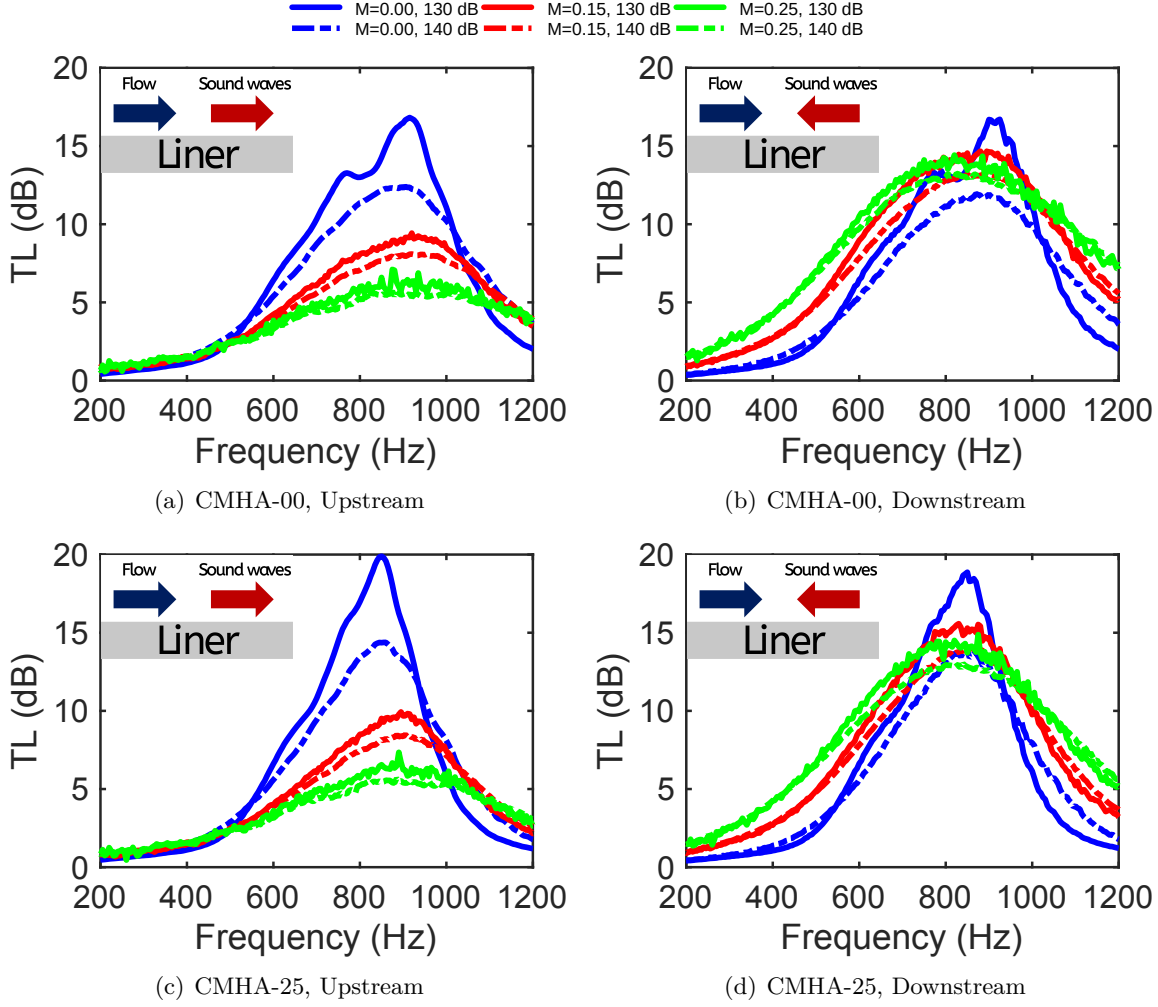


Figure 7.9: Experimental results. $M = 0$, $M = 0.15$, $M = 0.25$. TL as a function of Mach number and of SPL. Liners CMHA-00, and CMHA-25. Upstream: acoustic waves in the direction of flow, Downstream: acoustic waves against flow.

COMSOL, is applied. For the resonator neck, the Johnson–Champoux–Allard (JCA) model is employed to characterize sound propagation. The predicted results show good agreement with the experimental data for cases without flow.

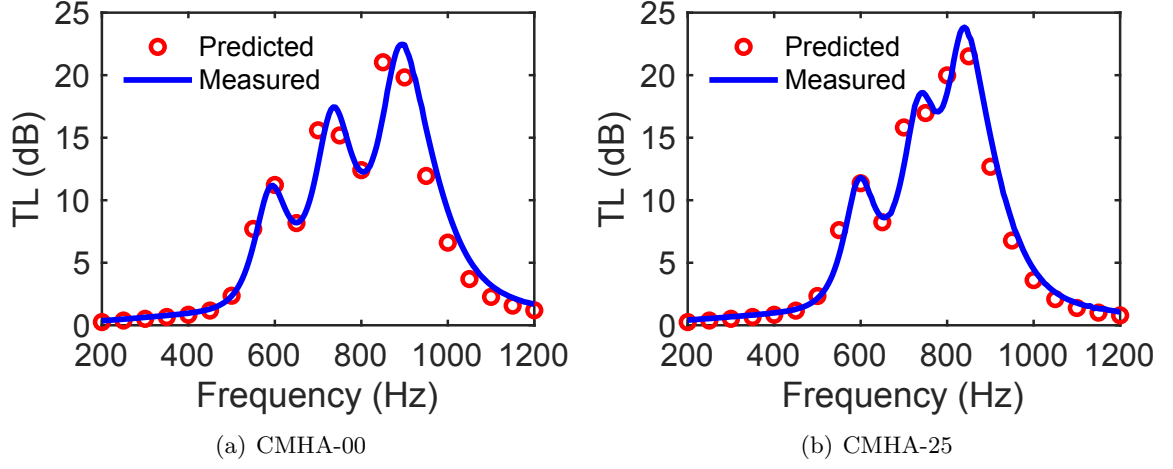


Figure 7.10: Validation results. $M = 0$. Comparison between predicted and experimental results for TL. Liners CMHA-00 and CMHA-25.

7.5 Summary

This Chapter investigates the design and evaluation of a low-frequency, broadband, noise-absorbing acoustic metasurface. A basic unit of the metasurface consists of a micro-perforated panel and an extended-neck Helmholtz resonator, referred to as MHA. Experimental results show that near-perfect sound absorption of over 97% can be achieved at 150 Hz with a thickness of 48 mm (sub-wavelength ratio = $1/48$). To further broaden the sound absorption bandwidth of the MHA while keeping the total area constant and reducing the total thickness, a variation involving three coupled MHA units, referred to as CMHA, is proposed. The experimental results demonstrate that the CMHA achieves an average SAC exceeding 0.74 within the 300–500 Hz frequency range, with an overall thickness of only 27.5 mm. Furthermore, the CMHA design achieves an average SAC exceeding 0.88 within the 543–945 Hz range with a reduced thickness of 26.2 mm, and an average SAC exceeding 0.83 within the 467–912 Hz range with a further reduced thickness of 26 mm. The TL performance of the CMHA liners under grazing flow was experimentally evaluated at the LAUM facility in Le Mans. Tests considered the effects of grazing flow (up to Mach 0.25), acoustic excitation level (up to 140 dB), and source position relative to the liner. Despite a total thickness of only 32 mm, both CMHA-00 and CMHA-25 achieved broadband TL between 10–23 dB below 1000 Hz under no-flow conditions, with CMHA-00 offering a slightly broader bandwidth. In grazing flow, TL below 1000 Hz remained effective. When the source was upstream, increasing Mach number slightly reduced TL and shifted the response to higher frequencies.

In contrast, with the source downstream, TL was maintained or enhanced, with bandwidth broadening and levels reaching 15 dB even under high SPL and flow—demonstrating robust low-frequency attenuation in a compact design.

The findings make both MHA and CMHA particularly suitable for practical applications where space limitations and low-frequency noise control are crucial. The demonstrated performance—broadband transmission loss below 1000 Hz despite a compact 32 mm thickness—highlights their potential for use in tightly constrained flow duct environments, such as aircraft engine nacelles or ventilation ducts, where conventional bulky liners are impractical.

Chapter 8

Conclusions and Future Work

8.1 Conclusions

This thesis has presented a comprehensive study on the development of compact broadband microperforated acoustic metasurfaces, with a particular focus on multi-chamber configurations and hybrid acoustic metamaterials incorporating extended-neck Helmholtz resonators, targeting applications in aeroengine noise control. The work spans theoretical modelling, numerical optimisation, experimental validation, and performance evaluation under realistic flow conditions. The main conclusions drawn from this research are summarized as follows:

Key conclusions of this thesis include:

1. **A novel multi-chamber micro-perforated panel absorber concept was proposed and validated.** This configuration enables multiple resonances within a compact structure, effectively broadening the absorption bandwidth while maintaining a compact design. A two-point impedance method based on acoustic network theory was developed to accurately model and optimise the impedance characteristics of MC-MPPAs. Experimental results showed that the MC-MPPA test pieces achieved impressive sound absorption coefficients exceeding 0.8 across the entire frequency ranges of 397–1000 Hz and 698–1895 Hz, corresponding to subwavelength ratios of 1/18.2 and 1/10.5, respectively.
2. **A micro-perforated acoustic metasurface incorporating extended-neck Helmholtz resonators was developed for deep-subwavelength low-frequency sound absorption.** To overcome the limited low-frequency performance of MC-MPPAs, particularly below 500 Hz, an acoustic metasurface composed of micro-perforated panels intergrated with extended-neck Helmholtz resonators was designed to achieve deep-subwavelength broadband absorption in this range. Three different configurations were developed and experimentally validated. Experimental results showed that:
 - One deep-subwavelength configuration achieved over 99.9% absorption at 329.6 Hz with a total thickness of 51 mm. Additionally, another design reached >97% absorption at 150 Hz with only 48 mm thickness, corresponding to a sub-wavelength ratio of 1/48.
 - A hybrid configuration demonstrated an average absorption coefficient of 0.8 within the 254–500 Hz range with 51 mm thickness.

- A compact configuration achieved an average absorption coefficient exceeding 0.74 in the 300–500 Hz band with a thickness of 27.5 mm, and over 0.88 in the 543–945 Hz range with a thickness of just 26.2 mm.

3. **Effects of perforated partitions on the optimised MC-MPPA.** The incorporation of internal perforated partitions provides clear acoustic advantages, particularly at higher frequencies and lower porosity levels—for example, in the frequency ranges of [20–2000] Hz and [20–3000] Hz at porosities of 0.5% and 1.0%, configurations with perforated partitions exhibit more pronounced absorption peaks and broader bandwidths compared to their non-partitioned counterparts. These results indicate that equivalent or superior acoustic performance can be achieved with fewer top-plate perforations when internal partitions are employed. This design strategy may also reduce aerodynamic drag, offering significant potential for noise control applications under grazing flow conditions, such as in aero-engine acoustic liners.
4. **The acoustic performance of proposed microperforated acoustic metasurface was extended to grazing flow environments through a novel impedance-based optimisation strategy.** A Mach-number-dependent impedance model was integrated into a graph-theory-based two-point impedance method, enabling accurate prediction of the total acoustic impedance under grazing flow conditions. A novel Cremer-impedance-inspired cost function was introduced to guide the optimisation of the absorber’s impedance across a target frequency range. The resulting design parameters—including sub-chamber depths, perforation diameters, and porosities—were used to fabricate 3D-printed liner samples. These 200 mm samples were tested under grazing flow up to Mach 0.25 and high sound pressure levels up to 140 dB. Experimental results demonstrated that the structure achieved up to 25 dB of transmission loss at an incident sound pressure level of 140 dB and a flow Mach number of 0.25 in the frequency range from 600 Hz to 2000 Hz. Additionally, the transmission loss performance of the CMHA liner was evaluated under grazing flow conditions. Despite its compact size (32 mm thickness), the CMHA achieved broadband transmission loss (10–23 dB) below 1000 Hz in the absence of flow. Moreover, its transmission loss performance remained effective under grazing flow. In the presence of downstream sources and under the highest SPL and Mach number conditions, the TL reached approximately 15 dB below 1000 Hz, further extending the effective noise reduction range. These results

demonstrate that the CMHA liner exhibits excellent low-frequency sound attenuation capabilities regardless of the presence of grazing flow, effectively compensating for the limited low-frequency performance of the MC-MPPA liner.

5. **All proposed designs were experimentally validated under realistic conditions.** Extensive impedance tube and grazing flow tests were carried out. The close agreement between predicted and measured acoustic performance confirms the accuracy of the theoretical modelling and optimisation frameworks developed in this thesis. The practical feasibility of 3D-printed fabrication techniques was also demonstrated.

Overall, this thesis advances the understanding and engineering of microperforated-based acoustic metamaterials for low-frequency broadband noise control. The proposed multi-chamber and hybrid designs offer compact, high-performance solutions suitable for aeroacoustic environments, ventilation systems, and industrial noise control. The developed modelling tools and optimisation strategies contribute valuable methodologies for the future design of acoustic metasurfaces in complex flow and acoustic environments.

8.2 Future Work

While this research has laid a solid foundation for the development of compact and broadband microperforated acoustic metasurfaces, and demonstrated their great potential in aircraft engine noise reduction, several directions can be pursued to further advance this field:

1. **Nonlinear acoustic effects:** In complex flow and acoustic environments, nonlinear acoustic effects are often non-negligible. Incorporating nonlinear terms into the current analytical models is therefore essential to improve the predictive accuracy of the proposed theoretical framework under realistic operating conditions.
2. **Active-passive hybrid systems:** Incorporating active control techniques into the microperforated acoustic metasurfaces framework could enable adaptive noise control, especially in dynamic environments where noise characteristics vary over time. Future work could explore hybrid liners that combine passive broadband absorption with active tuning mechanisms.
3. **Extension to curved surfaces:** Real-world applications, such as engine nacelles or fuselage liners, often involve curved surfaces. Future studies should investigate the

performance of MC-MPPA and MHA designs on non-planar geometries and assess their structural integration and manufacturability.

4. **Application-specific design and scalability:** The scalability and modularity of the proposed designs open the door for customized solutions. Future work may focus on integrating these absorbers into larger systems, such as duct networks or full-scale engine nacelles, and optimising their performance under practical constraints.
5. **Advanced manufacturing and materials:** The use of advanced 3D printing technologies has shown great potential in prototyping complex geometries. Future research may explore the use of novel materials (e.g., metal, flexible or temperature-resistant polymers) and fabrication methods to enhance performance, sustainability, and cost-effectiveness.

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Appendix A

Numbering Convention for MPPs in the MC-MPPA Configuration

A.1 Numbering Convention for MPPs in the MC-MPPA Configuration

Figure A.1 provides a visual representation of the numbering convention used for the top plate and the internal walls of the MC-MPPA. There are nine uniquely perforated surfaces for the top MPPs, and twelve for the internal MPPs. For the MC-MPPA-II configuration in Chapter 5, no internal MPPs are present.

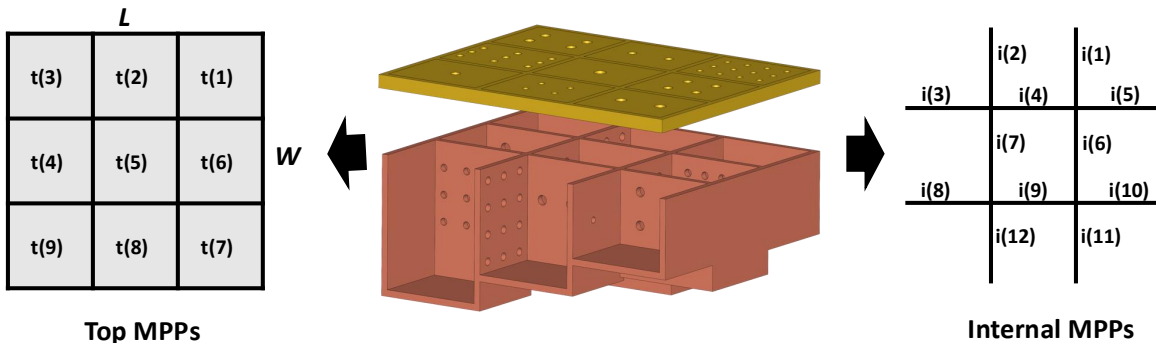


Figure A.1: Numbering convention for MPPs in the MC-MPPA configuration

Appendix B

Impedance Tube Measurements

B.1 Impedance Tube Measurements

Three rectangular impedance tubes are used to experimentally measure the normal incidence sound absorption coefficients of the proposed microperforated acoustic metasurfaces. The internal cross-sectional dimensions (e.g., 80 mm $\times$ 50 mm) and the corresponding cut-off frequencies are listed in Table B.1. The 80 mm $\times$ 50 mm rectangular impedance tube is shown in Fig. B.1.

Table B.1: Internal cross-sectional dimensions of the impedance tubes and their corresponding cut-off frequencies.

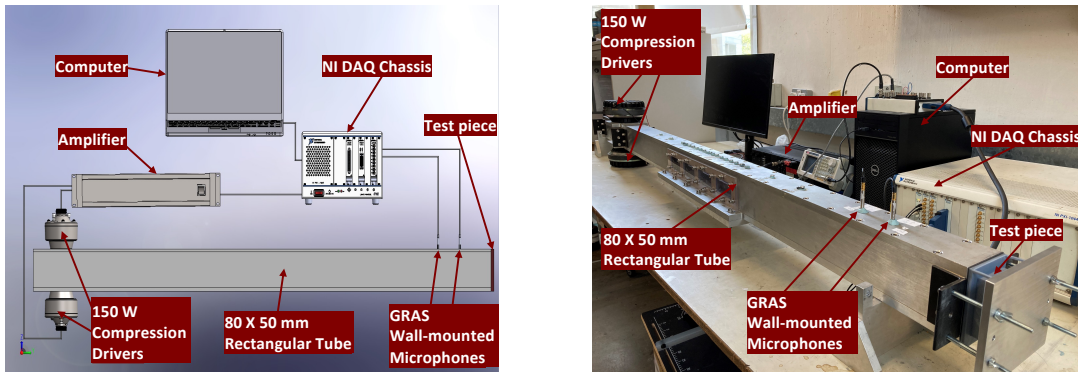
Internal cross-sectional dimensions	Corresponding cut-off frequencies
80 mm $\times$ 50 mm	2144 Hz
50 mm $\times$ 40 mm	3430 Hz
40 mm $\times$ 40 mm	4288 Hz

The experimental set-up is in accordance with the ISO 10534-2:1998 Standard [153]. The plane wave cut-off frequency of this experimental rig is approximately 2100 Hz.

Two 150 W BMS 4591 2" compression drivers are placed at one end of the tube to generate planar sound waves and test samples are mounted to the other end.

The microphones used in the measurements were G.R.A.S 40PH microphones. The 40PH has a flat frequency response from 20 Hz to 20 kHz, in excess of the frequency range of the acoustic drivers and the analysis conducted.

The incident sound pressure level was set to 90 dB or 100 dB to avoid nonlinear acoustic behavior of the perforated structure.



(a) Illustration of the 80 mm $\times$ 50 mm rectangular (b) Photo of the 80 mm $\times$ 50 mm rectangular impedance tube used to measure MC-MPPA's sound impedance tube absorption coefficients

Figure B.1: Impedance tube measurements.

Appendix C

Perforated Partition Effects on the Optimised MC-MPPA

C.1 Perforated Partition Effects on the Optimised MC-MPPA

To further investigate the influence of internal perforated partitions on the acoustic performance of the optimised MC-MPPA, a theoretical analysis was carried out by comparing two representative configurations introduced in Chapter 5: MC-MPPA-II and MC-MPPA-III. Both configurations feature variable sub-cavity depths and identical outer geometry; the key distinction lies in the presence of internal perforated partitions—MC-MPPA-II is designed without partitions, while MC-MPPA-III incorporates internal perforated partitions.

For consistency across the comparative study, the micro-perforation diameter was fixed at 1 mm, the thickness of the top MPP at 2 mm, the thickness of the internal MPP at 1 mm, and the maximum depth of the sub-cavities at 35 mm. Optimisation was performed over porosity ranges of [0, 5%], [0, 1%], [0, 0.5%], and [0, 0.1%] within three frequency bands: [0 1000] Hz, [0 2000] Hz, and [0 3000] Hz.

The results, illustrated in Fig. C.1, reveal that the inclusion of perforated partitions offers distinct advantages in higher frequency ranges and at lower porosity levels. Notably, within the frequency bands of [20–2000] Hz and [20–3000] Hz, and at porosities of 0.5% and 1.0%, the perforated partition configuration exhibits more pronounced absorption peaks and broader absorption bandwidths compared to the configuration without internal perforations.

These findings suggest that the use of internal perforated partitions can achieve equivalent or even improved acoustic performance with fewer perforations in the top plate. This reduction in the number of holes could lead to lower aerodynamic drag, making the design particularly attractive for applications under grazing flow conditions, such as in aero-engine acoustic liners.

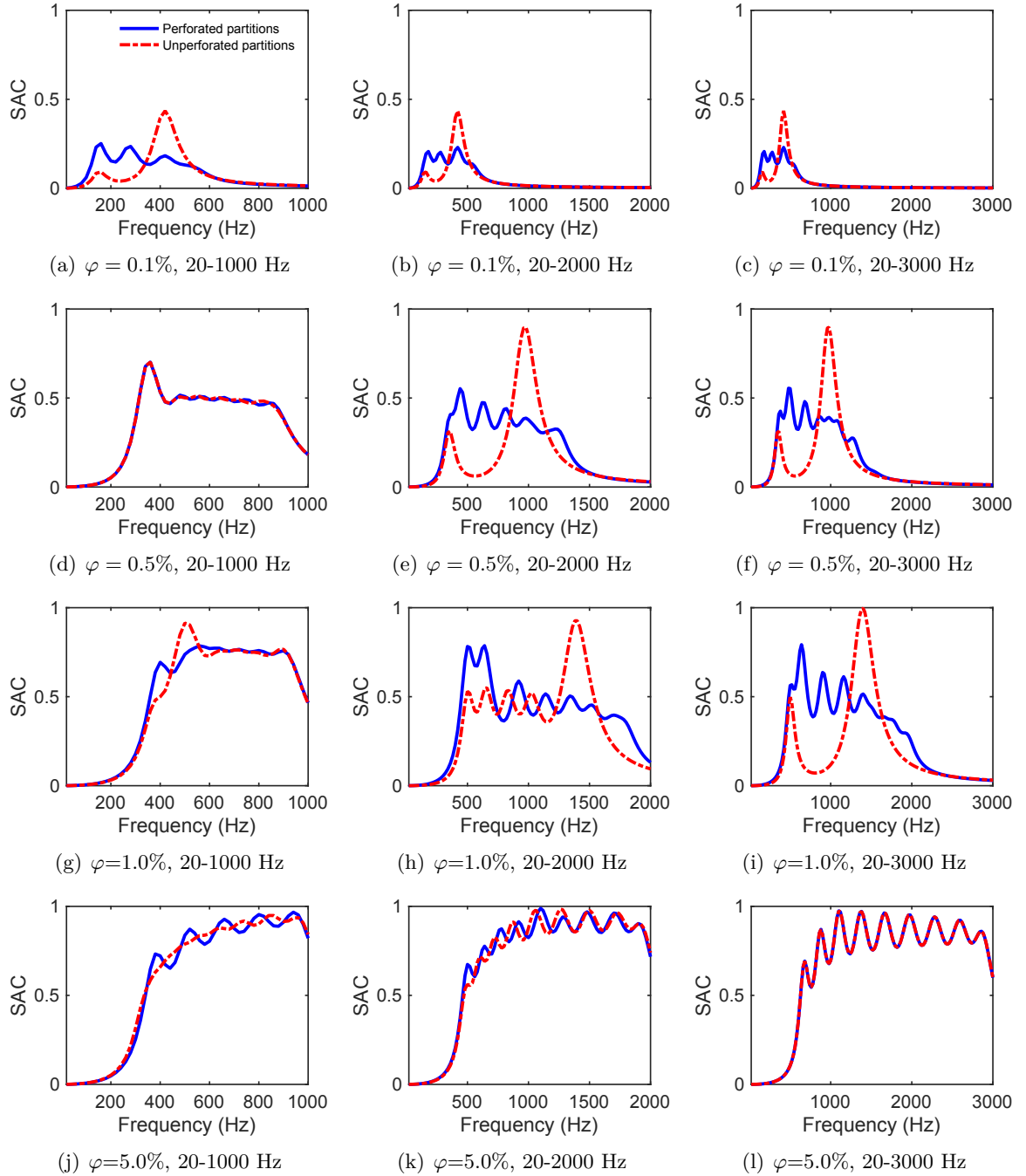


Figure C.1: Sound absorption coefficients of the optimised (3×3) MC-MPPA with and without perforated partitions at different porosities across various frequency ranges.

Appendix D

Sensitivity Analysis of Geometric Parameters of MC-MPPA in Chapter 4

D.1 Sensitivity Analysis of Geometric Parameters of MC-MPPA in Chapter 4

In Chapter 4, in the analysis of each geometric variable, other variables were controlled to ensure consistency. Specifically, in examining the impact of chamber depth on sound absorption performance, all chamber depths were considered as a single variable with a variation range of 10 to 50 mm. During this process, all other parameters were held constant (i.e., $d_{t(1-9)} = d_{i(1-12)} = 1$ mm, $\phi_{t(1-9)} = \phi_{i(1-12)} = 3\%$, $\tau_t = 2$ mm, and $\tau_i = 1$ mm).

In investigating the sensitivity of the sound absorption performance to the perforation diameters of the top MPPs, all perforation diameters of the top MPPs were treated as the same variable with a variation range of [0.5 1.0] mm, while all other parameters were held constant ($D_{(1-9)} = 30$ mm, $d_{i(1-12)} = 1$ mm, $\phi_{t(1-9)} = \phi_{i(1-12)} = 3\%$, $\tau_t = 2$ mm, and $\tau_i = 1$ mm).

In investigating the sensitivity of the sound absorption performance to the perforation porosity of the top MPPs, all perforation porosities of the top MPPs were treated as the same variable with a variation range of [0.5 5.0]%, while all other parameters were held constant ($D_{(1-9)} = 30$ mm, $d_{t(1-9)} = d_{i(1-12)} = 1$ mm, $\phi_{i(1-12)} = 3\%$, $\tau_t = 2$ mm, and $\tau_i = 1$ mm).

In investigating the sensitivity of the sound absorption performance to the thicknesses of the top MPPs, all thicknesses of the top MPPs were treated as the same variable with a variation range of [0.5 2.5] mm, while all other parameters were held constant ($D_{(1-9)} = 30$ mm, $d_{t(1-9)} = d_{i(1-12)} = 1$ mm, $\phi_{t(1-9)} = \phi_{i(1-12)} = 3\%$, and $\tau_i = 1$ mm).

In investigating the sensitivity of the sound absorption performance to the perforation diameters of the internal MPPs, all perforation diameters of the internal MPPs were treated as the same variable with a variation range of [0.5 1.0] mm, while all other parameters were held constant ($D_{(1)} = 50$ mm, $D_{(2)} = 17$ mm, $D_{(3)} = 5$ mm, $D_{(4)} = 6$ mm, $D_{(5)} = 32$ mm, $D_{(6)} = 5$ mm, $D_{(7)} = 7.8$ mm, $D_{(8)} = 47$ mm, $D_{(9)} = 11$ mm, $d_{t(1)} = 0.5$ mm, $d_{t(2)} = 0.8$ mm, $d_{t(3)} = 0.65$ mm, $d_{t(4)} = 0.7$ mm, $d_{t(5)} = 1$ mm, $d_{t(6)} = 0.65$ mm, $d_{t(7)} = 0.7$ mm, $d_{t(8)} = 0.8$ mm, $d_{t(9)} = 0.5$ mm, $\phi_{t(1-9)} = \phi_{i(1-12)} = 3\%$, $\tau_t = 2$ mm, and $\tau_i = 1$ mm).

In investigating the sensitivity of the sound absorption performance to the perforation porosity of the internal MPPs, all perforation porosities of the internal MPPs were treated as the same variable with a variation range of [0.5 5.0]%, while all other parameters were held constant ($D_{(1)} = 50$ mm, $D_{(2)} = 17$ mm, $D_{(3)} = 5$ mm, $D_{(4)} = 6$ mm, $D_{(5)} = 32$ mm, $D_{(6)} = 5$ mm, $D_{(7)} = 7.8$ mm, $D_{(8)} = 47$ mm, $D_{(9)} = 11$ mm, $d_{t(1)} = 0.5$ mm, $d_{t(2)} = 0.8$ mm,

$d_{t(3)} = 0.65$ mm, $d_{t(4)} = 0.7$ mm, $d_{t(5)} = 1$ mm, $d_{t(6)} = 0.65$ mm, $d_{t(7)} = 0.7$ mm, $d_{t(8)} = 0.8$ mm, $d_{t(9)} = 0.5$ mm, $d_{i(1-12)} = 1$ mm, $\phi_{t(1-9)} = 3\%$, $\tau_t = 2$ mm, and $\tau_i = 1$ mm).

In investigating the sensitivity of the sound absorption performance to the thicknesses of the internal MPPs, all thicknesses of the internal MPPs were treated as the same variable with a variation range of $[0.5 \text{ } 2.5]$ mm, while all other parameters were held constant ($D_{(1)} = 50$ mm, $D_{(2)} = 17$ mm, $D_{(3)} = 5$ mm, $D_{(4)} = 6$ mm, $D_{(5)} = 32$ mm, $D_{(6)} = 5$ mm, $D_{(7)} = 7.8$ mm, $D_{(8)} = 47$ mm, $D_{(9)} = 11$ mm, $d_{t(1)} = 0.5$ mm, $d_{t(2)} = 0.8$ mm, $d_{t(3)} = 0.65$ mm, $d_{t(4)} = 0.7$ mm, $d_{t(5)} = 1$ mm, $d_{t(6)} = 0.65$ mm, $d_{t(7)} = 0.7$ mm, $d_{t(8)} = 0.8$ mm, $d_{t(9)} = 0.5$ mm, $d_{i(1-12)} = 1$ mm, $\phi_{t(1-9)} = \phi_{i(1-12)} = 3\%$, $\tau_t = 2$ mm, and $\tau_i = 1$ mm).

Appendix E

Geometric Parameters of Tested Samples

E.1 Geometric Parameters of Tested MC-MPPA Samples in Chapter 3

Table E.1: Geometric parameters of the (3×3) MC-MPPA of test sample 50 in Chapter 3.

Perforation diameter (mm)									
$d_{t(1)}$	1.00	$d_{t(2)}$	1.00	$d_{t(3)}$	1.00	$d_{t(4)}$	1.00	$d_{t(5)}$	1.00
$d_{t(6)}$	1.00	$d_{t(7)}$	1.00	$d_{t(8)}$	0	$d_{t(9)}$	1.00		
$d_{i(1)}$	0.50	$d_{i(2)}$	0.50	$d_{i(3)}$	1.00	$d_{i(4)}$	0.50	$d_{i(5)}$	0
$d_{i(6)}$	0	$d_{i(7)}$	1.00	$d_{i(8)}$	0	$d_{i(9)}$	1.00	$d_{i(10)}$	0.50
$d_{i(11)}$	0.50	$d_{i(12)}$	0.50						
Perforation porosity (%)									
$\phi_{t(1)}$	0.75	$\phi_{t(2)}$	4.54	$\phi_{t(3)}$	4.56	$\phi_{t(4)}$	4.13	$\phi_{t(5)}$	3.97
$\phi_{t(6)}$	0.45	$\phi_{t(7)}$	0.46	$\phi_{t(8)}$	0	$\phi_{t(9)}$	0.43		
$\phi_{i(1)}$	0.37	$\phi_{i(2)}$	1.56	$\phi_{i(3)}$	5.00	$\phi_{i(4)}$	5.00	$\phi_{i(5)}$	0
$\phi_{i(6)}$	0	$\phi_{i(7)}$	5.00	$\phi_{i(8)}$	0	$\phi_{i(9)}$	0.55	$\phi_{i(10)}$	1.93
$\phi_{i(11)}$	0.07	$\phi_{i(12)}$	0.28						
Perforation number									
$n_{t(1)}$	2	$n_{t(2)}$	12	$n_{t(3)}$	12	$n_{t(4)}$	11	$n_{t(5)}$	10
$n_{t(6)}$	1	$n_{t(7)}$	1	$n_{t(8)}$	0	$n_{t(9)}$	1		
$n_{i(1)}$	12	$n_{i(2)}$	50	$n_{i(3)}$	51	$n_{i(4)}$	204	$n_{i(5)}$	0
$n_{i(6)}$	0	$n_{i(7)}$	40	$n_{i(8)}$	0	$n_{i(9)}$	6	$n_{i(10)}$	79
$n_{i(11)}$	2	$n_{i(12)}$	9						
Other geometric parameters									
L	50 mm	W	40 mm	D	50 mm	τ_i	1 mm	τ_t	2 mm

Table E.2: Geometric parameters of the (3×3) MC-MPPA of test sample 20 in Chapter 3.

Perforation diameter (mm)									
$d_{t(1)}$	0.98	$d_{t(2)}$	0.63	$d_{t(3)}$	0.64	$d_{t(4)}$	0.90	$d_{t(5)}$	0
$d_{t(6)}$	0	$d_{t(7)}$	0	$d_{t(8)}$	0.50	$d_{t(9)}$	0.50		
$d_{i(1)}$	0.50	$d_{i(2)}$	0.50	$d_{i(3)}$	0	$d_{i(4)}$	0.50	$d_{i(5)}$	0.50
$d_{i(6)}$	0	$d_{i(7)}$	1.00	$d_{i(8)}$	0	$d_{i(9)}$	0	$d_{i(10)}$	0.50
$d_{i(11)}$	0	$d_{i(12)}$	0.51						
Perforation porosity (%)									
$\phi_{t(1)}$	5.00	$\phi_{t(2)}$	5.00	$\phi_{t(3)}$	5.00	$\phi_{t(4)}$	5.00	$\phi_{t(5)}$	0
$\phi_{t(6)}$	0	$\phi_{t(7)}$	0	$\phi_{t(8)}$	4.22	$\phi_{t(9)}$	4.22		
$\phi_{i(1)}$	5.00	$\phi_{i(2)}$	5.00	$\phi_{i(3)}$	0	$\phi_{i(4)}$	0.58	$\phi_{i(5)}$	1.47
$\phi_{i(6)}$	0	$\phi_{i(7)}$	1.28	$\phi_{i(8)}$	0	$\phi_{i(9)}$	0	$\phi_{i(10)}$	5.00
$\phi_{i(11)}$	0	$\phi_{i(12)}$	5.00						
Perforation number									
$n_{t(1)}$	13	$n_{t(2)}$	32	$n_{t(3)}$	31	$n_{t(4)}$	16	$n_{t(5)}$	0
$n_{t(6)}$	0	$n_{t(7)}$	0	$n_{t(8)}$	43	$n_{t(9)}$	43		
$n_{i(1)}$	64	$n_{i(2)}$	64	$n_{i(3)}$	0	$n_{i(4)}$	10	$n_{i(5)}$	24
$n_{i(6)}$	0	$n_{i(7)}$	4	$n_{i(8)}$	0	$n_{i(9)}$	0	$n_{i(10)}$	81
$n_{i(11)}$	0	$n_{i(12)}$	61						
Other geometric parameters									
L	50 mm	W	40 mm	D	20 mm	τ_i	1 mm	τ_t	2 mm

E.2 Geometric Parameters of Tested MC-MPPA Samples in Chapter 4

Table E.3: Geometric parameters of Test Piece 1000.

Perforation diameter (mm)									
$d_{t(1)}$	1.00	$d_{t(2)}$	1.00	$d_{t(3)}$	1.00	$d_{t(4)}$	1.00	$d_{t(5)}$	1.00
$d_{t(6)}$	1.00	$d_{t(7)}$	1.00	$d_{t(8)}$	1.00	$d_{t(9)}$	1.00		
$d_{i(1)}$	0	$d_{i(2)}$	1.00	$d_{i(3)}$	0.50	$d_{i(4)}$	0	$d_{i(5)}$	1.00
$d_{i(6)}$	0.50	$d_{i(7)}$	0.50	$d_{i(8)}$	0.50	$d_{i(9)}$	0	$d_{i(10)}$	0.50
$d_{i(11)}$	1.00	$d_{i(12)}$	0.50						
Perforation porosity (%)									
$\phi_{t(1)}$	3.14	$\phi_{t(2)}$	5.11	$\phi_{t(3)}$	0.39	$\phi_{t(4)}$	0.79	$\phi_{t(5)}$	1.96
$\phi_{t(6)}$	2.36	$\phi_{t(7)}$	3.53	$\phi_{t(8)}$	0	$\phi_{t(9)}$	0.39		
$\phi_{i(1)}$	0	$\phi_{i(2)}$	0.14	$\phi_{i(3)}$	5.00	$\phi_{i(4)}$	0	$\phi_{i(5)}$	5.02
$\phi_{i(6)}$	0.91	$\phi_{i(7)}$	0.80	$\phi_{i(8)}$	0.19	$\phi_{i(9)}$	0	$\phi_{i(10)}$	5.00
$\phi_{i(11)}$	0.14	$\phi_{i(12)}$	0.04						
Perforation number									
$n_{t(1)}$	8	$n_{t(2)}$	13	$n_{t(3)}$	1	$n_{t(4)}$	2	$n_{t(5)}$	5
$n_{t(6)}$	6	$n_{t(7)}$	9	$n_{t(8)}$	0	$n_{t(9)}$	1		
$n_{i(1)}$	0	$n_{i(2)}$	1	$n_{i(3)}$	183	$n_{i(4)}$	0	$n_{i(5)}$	46
$n_{i(6)}$	26	$n_{i(7)}$	23	$n_{i(8)}$	7	$n_{i(9)}$	0	$n_{i(10)}$	183
$n_{i(11)}$	1	$n_{i(12)}$	1						
Sub-chamber depths (mm)									
$D_{(1)}$	45	$D_{(2)}$	45	$D_{(3)}$	45	$D_{(4)}$	45	$D_{(5)}$	45
$D_{(6)}$	45	$D_{(7)}$	45	$D_{(8)}$	45	$D_{(9)}$	45		
Additional parameters related to geometry (mm)									
L	50	W	40	τ_i	1	τ_t	2		

Table E.4: Geometric parameters of Test Piece 2000.

Perforation diameter (mm)									
$d_{t(1)}$	1.00	$d_{t(2)}$	1.00	$d_{t(3)}$	0.50	$d_{t(4)}$	0.58	$d_{t(5)}$	0.50
$d_{t(6)}$	0.50	$d_{t(7)}$	0.63	$d_{t(8)}$	1.00	$d_{t(9)}$	1.00		
$d_{i(1)}$	0.50	$d_{i(2)}$	0	$d_{i(3)}$	0.50	$d_{i(4)}$	0	$d_{i(5)}$	0.50
$d_{i(6)}$	0	$d_{i(7)}$	0	$d_{i(8)}$	0.50	$d_{i(9)}$	0	$d_{i(10)}$	1.00
$d_{i(11)}$	0	$d_{i(12)}$	0.50						
Perforation porosity (%)									
$\phi_{t(1)}$	1.57	$\phi_{t(2)}$	0.79	$\phi_{t(3)}$	5.00	$\phi_{t(4)}$	4.97	$\phi_{t(5)}$	5.00
$\phi_{t(6)}$	1.96	$\phi_{t(7)}$	4.93	$\phi_{t(8)}$	5.11	$\phi_{t(9)}$	5.11		
$\phi_{i(1)}$	0.03	$\phi_{i(2)}$	0	$\phi_{i(3)}$	0.23	$\phi_{i(4)}$	0	$\phi_{i(5)}$	0.03
$\phi_{i(6)}$	0	$\phi_{i(7)}$	0	$\phi_{i(8)}$	0.51	$\phi_{i(9)}$	0	$\phi_{i(10)}$	0.33
$\phi_{i(11)}$	0	$\phi_{i(12)}$	5.05						
Perforation number									
$n_{t(1)}$	4	$n_{t(2)}$	2	$n_{t(3)}$	51	$n_{t(4)}$	38	$n_{t(5)}$	51
$n_{t(6)}$	20	$n_{t(7)}$	32	$n_{t(8)}$	13	$n_{t(9)}$	13		
$n_{i(1)}$	1	$n_{i(2)}$	0	$n_{i(3)}$	4	$n_{i(4)}$	0	$n_{i(5)}$	1
$n_{i(6)}$	0	$n_{i(7)}$	0	$n_{i(8)}$	6	$n_{i(9)}$	0	$n_{i(10)}$	3
$n_{i(11)}$	0	$n_{i(12)}$	45						
Sub-chamber depths									
$D_{(1)}$	45	$D_{(2)}$	45	$D_{(3)}$	28.1	$D_{(4)}$	21.8	$D_{(5)}$	45
$D_{(6)}$	45	$D_{(7)}$	45	$D_{(8)}$	14	$D_{(9)}$	14.3		
Additional parameters related to geometry (mm)									
L	50	W	40	τ_i	1	τ_t	2		

E.3 Optimized Geometric Parameters of MC-MPPA in Chapter 4

Tables E.5 - E.8 present the geometric parameters of the MC-MPPA after the optimization across four distinct frequency ranges: [20 1000] Hz, [20 2000] Hz, [20 3000] Hz and [20 4000] Hz.

Table E.5: Optimized geometric parameters of MC-MPPA in the frequency range 20–1000 Hz. All cavity depths, panel thicknesses, and micro-perforation diameters are given in millimeters (mm).

$D_{\max}$	10	15	20	25	30	35	40	45
τ_i	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$d_{i(1)}$	1.000	0.520	0.740	0.600	1.000	0.500	0.500	0.500
$\phi_{i(1)}$	0.078	0.000	0.000	0.000	0.000	0.000	0.018	0.001
$d_{i(2)}$	1.000	0.500	0.500	0.540	1.000	0.500	0.950	1.000
$\phi_{i(2)}$	4.009	5.000	0.000	0.000	2.043	0.273	0.000	0.129
$d_{i(3)}$	1.000	0.500	1.000	1.000	1.000	0.500	0.500	0.500
$\phi_{i(3)}$	0.195	4.978	0.117	2.635	0.129	5.000	0.000	5.000
$d_{i(4)}$	0.890	0.500	1.000	0.500	0.500	0.500	0.500	0.520
$\phi_{i(4)}$	0.000	4.741	1.011	0.204	0.000	0.225	0.386	0.000
$d_{i(5)}$	1.000	0.500	0.500	0.500	0.500	0.500	1.000	1.000
$\phi_{i(5)}$	0.017	0.720	0.035	0.040	0.065	5.000	5.000	5.000
$d_{i(6)}$	0.500	0.500	1.000	1.000	0.500	1.000	0.500	0.500
$\phi_{i(6)}$	0.000	1.027	0.210	0.232	0.167	0.000	0.170	0.916
$d_{i(7)}$	0.500	0.500	1.000	1.000	0.500	0.500	0.500	0.500
$\phi_{i(7)}$	0.243	5.000	0.305	0.401	0.029	5.000	2.003	0.799
$d_{i(8)}$	1.000	0.500	1.000	1.000	1.000	0.500	0.500	0.500
$\phi_{i(8)}$	0.086	4.831	0.085	3.295	5.000	5.000	4.796	0.178
$d_{i(9)}$	1.000	0.500	0.500	1.000	1.000	0.500	0.500	0.500
$\phi_{i(9)}$	0.115	4.961	5.000	2.573	4.691	2.524	5.000	0.003
$d_{i(10)}$	1.000	0.500	0.570	0.730	0.500	0.500	1.000	0.500
$\phi_{i(10)}$	0.043	0.131	0.000	0.000	5.000	0.955	0.037	5.000

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Table E.5 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$d_{i(11)}$	1.000	0.840	0.500	1.000	0.840	0.500	0.890	1.000
$\phi_{i(11)}$	0.327	1.751	0.191	5.000	0.000	0.470	5.000	0.119
$d_{i(12)}$	1.000	0.500	1.000	0.990	0.500	0.500	1.000	0.500
$\phi_{i(12)}$	0.010	5.000	0.037	0.000	0.182	5.000	5.000	0.045
$d_{t(1)}$	1.000	0.740	1.000	1.000	1.000	1.000	1.000	1.000
$\phi_{t(1)}$	0.648	0.000	0.402	0.414	0.373	0.171	0.586	3.327
$d_{t(2)}$	1.000	1.000	1.000	1.000	1.000	0.500	0.520	1.000
$\phi_{t(2)}$	0.869	1.028	0.973	0.752	2.899	0.219	0.000	5.000
$d_{t(3)}$	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$\phi_{t(3)}$	0.870	1.051	1.833	2.173	2.861	2.612	4.565	0.336
$d_{t(4)}$	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$\phi_{t(4)}$	0.822	1.040	1.558	2.137	1.766	2.569	1.931	0.730
$d_{t(5)}$	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$\phi_{t(5)}$	0.822	1.230	1.041	1.165	0.834	2.588	2.847	1.874
$d_{t(6)}$	1.000	1.000	1.000	1.000	1.000	1.000	0.590	1.000
$\phi_{t(6)}$	0.294	0.382	0.317	0.270	0.652	0.187	0.000	2.344
$d_{t(7)}$	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$\phi_{t(7)}$	0.439	1.069	0.734	1.106	0.631	0.874	2.890	3.472
$d_{t(8)}$	1.000	1.000	1.000	1.000	1.000	1.000	0.930	0.530
$\phi_{t(8)}$	0.597	1.026	0.962	1.109	1.183	2.713	0.000	0.000
$d_{t(9)}$	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$\phi_{t(9)}$	0.368	1.051	1.795	2.167	1.685	2.555	3.105	0.463
τ_t	2.000	2.000	2.000	2.000	2.000	2.000	2.000	2.000
$D_{(1)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(2)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(3)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(4)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(5)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(6)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000

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Table E.5 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$D_{(7)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(8)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(9)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000

Table E.6: Optimized geometric parameters of MC-MPPA in the frequency range 20–2000 Hz. All cavity depths, panel thicknesses, and micro-perforation diameters are given in millimeters (mm).

$D_{\max}$	10	15	20	25	30	35	40	45
τ_i	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$d_{i(1)}$	0.500	0.500	1.000	0.500	0.500	0.990	1.000	0.500
$\phi_{i(1)}$	0.176	0.254	1.614	0.000	0.000	0.000	0.077	0.021
$d_{i(2)}$	0.500	0.500	0.860	0.730	0.500	1.000	1.000	0.610
$\phi_{i(2)}$	5.000	1.325	0.000	0.000	0.000	0.000	2.442	0.000
$d_{i(3)}$	0.500	0.500	0.500	0.500	0.500	0.500	0.500	0.500
$\phi_{i(3)}$	5.000	5.000	0.810	5.000	1.220	5.000	1.166	0.212
$d_{i(4)}$	0.500	0.500	0.750	0.810	0.500	0.990	0.500	0.560
$\phi_{i(4)}$	5.000	0.171	0.000	0.000	0.628	0.000	1.041	0.000
$d_{i(5)}$	0.500	0.500	0.500	0.560	0.500	0.500	0.500	0.500
$\phi_{i(5)}$	0.910	0.375	3.490	0.858	0.455	0.129	0.080	0.023
$d_{i(6)}$	0.500	0.500	0.500	0.500	0.500	1.000	0.500	0.810
$\phi_{i(6)}$	5.000	0.706	1.396	5.000	0.683	2.458	0.740	0.000
$d_{i(7)}$	0.500	0.500	0.500	0.650	0.910	0.500	1.000	0.820
$\phi_{i(7)}$	5.000	1.581	5.000	0.000	0.000	0.437	0.000	0.000
$d_{i(8)}$	0.500	0.500	0.500	0.500	0.500	1.000	0.500	0.500
$\phi_{i(8)}$	5.000	5.000	5.000	5.000	5.000	0.000	5.000	0.485
$d_{i(9)}$	0.500	0.610	0.500	0.500	0.990	0.500	0.500	0.970
$\phi_{i(9)}$	5.000	0.000	5.000	0.775	0.000	0.028	0.637	0.000
$d_{i(10)}$	0.500	0.500	0.500	0.620	0.850	0.500	0.500	1.000

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Table E.6 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$\phi_{i(10)}$	1.184	5.000	0.289	0.200	0.000	0.181	0.135	0.323
$d_{i(11)}$	0.500	0.500	0.500	0.500	0.500	0.500	0.660	0.970
$\phi_{i(11)}$	0.982	5.000	0.360	0.098	5.000	0.000	0.000	0.000
$d_{i(12)}$	0.500	0.500	0.500	0.500	0.500	0.500	1.000	0.500
$\phi_{i(12)}$	5.000	5.000	5.000	0.701	0.838	0.072	0.000	5.000
$d_{t(1)}$	0.600	1.000	0.710	0.960	1.000	1.000	1.000	1.000
$\phi_{t(1)}$	0.000	0.199	0.000	0.000	0.528	0.644	0.672	1.603
$d_{t(2)}$	1.000	0.500	1.000	1.000	0.560	1.000	0.500	1.000
$\phi_{t(2)}$	2.634	0.000	5.000	5.000	5.000	5.000	5.000	0.976
$d_{t(3)}$	1.000	1.000	0.500	0.500	0.630	0.500	0.830	0.500
$\phi_{t(3)}$	2.758	3.006	2.503	5.000	0.000	5.000	5.000	5.000
$d_{t(4)}$	1.000	1.000	0.770	0.500	0.500	0.500	1.000	0.580
$\phi_{t(4)}$	2.693	3.344	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(5)}$	1.000	0.500	0.810	0.850	0.680	0.860	0.500	0.500
$\phi_{t(5)}$	2.559	0.463	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(6)}$	1.000	1.000	0.850	0.970	1.000	0.500	1.000	0.500
$\phi_{t(6)}$	1.754	4.060	0.000	5.000	0.770	0.000	1.088	1.922
$d_{t(7)}$	0.500	1.000	0.670	1.000	0.920	0.660	0.680	0.630
$\phi_{t(7)}$	0.206	4.290	0.000	0.319	5.000	1.542	1.789	5.000
$d_{t(8)}$	1.000	1.000	0.800	0.500	0.920	0.590	0.500	1.000
$\phi_{t(8)}$	2.468	4.286	5.000	0.000	5.000	5.000	5.000	5.000
$d_{t(9)}$	1.000	1.000	0.770	0.540	0.500	0.760	1.000	1.000
$\phi_{t(9)}$	2.680	3.866	5.000	5.000	5.000	5.000	5.000	5.000
τ_t	2.000	2.000	2.000	2.000	2.000	2.000	2.000	2.000
$D_{(1)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(2)}$	10.000	15.000	20.000	12.900	30.000	12.950	30.200	45.000
$D_{(3)}$	10.000	15.000	20.000	25.000	30.000	35.000	20.590	28.100
$D_{(4)}$	10.000	15.000	20.000	25.000	30.000	35.000	14.970	21.820
$D_{(5)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000

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Table E.6 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$D_{(6)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(7)}$	10.000	15.000	20.000	25.000	14.650	35.000	40.000	45.000
$D_{(8)}$	10.000	15.000	20.000	25.000	15.320	22.750	40.000	14.030
$D_{(9)}$	10.000	15.000	20.000	25.000	30.000	16.960	14.560	14.320

Table E.7: Optimized geometric parameters of MC-MPPA in the frequency range 20–3000 Hz. All cavity depths, panel thicknesses, and micro-perforation diameters are given in millimeters (mm).

$D_{\max}$	10	15	20	25	30	35	40	45
τ_i	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$d_{i(1)}$	0.500	0.500	1.000	0.770	0.640	0.760	0.500	0.820
$\phi_{i(1)}$	0.501	0.296	0.000	0.000	0.000	0.000	0.000	0.000
$d_{i(2)}$	0.500	1.000	0.510	0.950	0.660	0.500	0.500	0.820
$\phi_{i(2)}$	0.958	1.567	0.000	0.000	0.000	0.142	0.000	0.000
$d_{i(3)}$	0.500	0.920	0.660	0.500	0.800	0.500	0.500	0.500
$\phi_{i(3)}$	0.568	0.000	0.574	0.094	0.000	0.055	0.377	0.238
$d_{i(4)}$	1.000	0.720	1.000	0.960	0.500	0.500	0.500	0.950
$\phi_{i(4)}$	3.847	0.000	5.000	0.000	0.236	0.045	1.072	0.000
$d_{i(5)}$	0.500	0.700	1.000	0.500	0.760	0.780	0.990	1.000
$\phi_{i(5)}$	2.289	0.606	0.000	0.000	0.000	1.977	0.000	0.079
$d_{i(6)}$	0.500	0.580	1.000	0.830	0.830	0.750	0.500	0.720
$\phi_{i(6)}$	5.000	0.000	2.070	0.000	0.000	0.000	5.000	0.000
$d_{i(7)}$	0.500	0.850	0.500	0.560	0.770	0.950	0.500	0.500
$\phi_{i(7)}$	0.653	1.451	0.000	0.000	0.000	0.000	2.051	0.000
$d_{i(8)}$	0.500	0.880	0.550	0.500	0.950	1.000	1.000	1.000
$\phi_{i(8)}$	0.000	0.000	0.000	0.090	0.514	0.668	0.000	0.000
$d_{i(9)}$	0.790	0.820	1.000	0.500	0.500	0.500	0.500	0.930
$\phi_{i(9)}$	5.000	0.000	0.000	0.505	0.008	0.054	0.858	0.000

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Table E.7 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$d_{i(10)}$	0.660	0.770	1.000	0.540	0.790	0.950	0.500	0.500
$\phi_{i(10)}$	5.000	0.000	2.637	0.000	0.000	0.000	5.000	0.832
$d_{i(11)}$	0.500	0.500	0.500	0.500	0.780	0.710	0.550	0.870
$\phi_{i(11)}$	5.000	0.023	1.813	0.432	0.000	0.000	0.000	0.000
$d_{i(12)}$	0.830	0.980	0.990	0.610	0.940	0.700	0.500	1.000
$\phi_{i(12)}$	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
$d_{t(1)}$	0.500	0.500	0.520	0.910	0.500	0.500	0.780	0.500
$\phi_{t(1)}$	0.000	1.098	5.000	1.428	5.000	5.000	1.862	3.363
$d_{t(2)}$	0.500	0.600	1.000	0.500	0.550	0.810	0.500	1.000
$\phi_{t(2)}$	0.608	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(3)}$	0.840	0.500	0.610	0.810	1.000	0.590	0.500	0.510
$\phi_{t(3)}$	0.000	2.587	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(4)}$	0.720	0.500	1.000	0.670	0.590	0.530	0.500	0.710
$\phi_{t(4)}$	5.000	0.000	0.526	5.000	5.000	5.000	5.000	5.000
$d_{t(5)}$	0.750	0.620	0.980	0.500	0.820	0.680	0.810	0.580
$\phi_{t(5)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(6)}$	0.750	0.560	0.500	1.000	0.680	0.500	0.890	0.500
$\phi_{t(6)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(7)}$	0.800	1.000	0.500	0.500	0.560	1.000	0.910	0.820
$\phi_{t(7)}$	5.000	5.000	5.000	5.000	3.344	5.000	5.000	5.000
$d_{t(8)}$	0.810	0.740	0.500	0.500	0.620	0.600	0.500	0.620
$\phi_{t(8)}$	5.000	5.000	0.896	0.999	5.000	5.000	5.000	5.000
$d_{t(9)}$	0.650	0.500	0.710	0.570	0.500	0.500	0.550	0.780
$\phi_{t(9)}$	3.780	5.000	5.000	5.000	0.778	0.721	3.292	1.878
τ_t	2.000	2.000	2.000	2.000	2.000	2.000	2.000	2.000
$D_{(1)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(2)}$	10.000	15.000	6.880	25.000	17.910	7.830	16.850	6.180
$D_{(3)}$	10.000	15.000	20.000	7.810	6.130	18.790	40.000	27.050
$D_{(4)}$	10.000	15.000	20.000	10.260	30.000	35.000	12.480	10.210

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Table E.7 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$D_{(5)}$	10.000	15.000	7.490	25.000	7.860	9.920	9.370	17.790
$D_{(6)}$	10.000	15.000	16.270	6.180	9.570	35.000	7.640	45.000
$D_{(7)}$	9.890	6.200	20.000	25.000	30.000	6.160	7.200	8.910
$D_{(8)}$	10.000	7.860	20.000	25.000	12.640	13.190	26.070	12.820
$D_{(9)}$	10.000	15.000	8.830	13.740	30.000	35.000	40.000	45.000

Table E.8: Optimized geometric parameters of MC-MPPA in the frequency range 20–4000 Hz. All cavity depths, panel thicknesses, and micro-perforation diameters are given in millimeters (mm).

$D_{\max}$	10	15	20	25	30	35	40	45
τ_i	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
$d_{i(1)}$	1.000	0.710	0.570	0.900	0.890	0.770	0.700	0.500
$\phi_{i(1)}$	1.671	0.000	0.000	0.000	0.000	0.000	0.000	0.000
$d_{i(2)}$	0.500	0.500	1.000	0.990	1.000	0.750	1.000	0.570
$\phi_{i(2)}$	0.575	1.896	3.465	4.187	4.050	0.000	4.103	0.000
$d_{i(3)}$	0.750	0.500	0.560	0.900	0.630	0.630	0.500	0.990
$\phi_{i(3)}$	0.000	0.000	0.000	0.000	0.000	0.000	0.191	0.000
$d_{i(4)}$	0.500	0.500	0.740	0.960	0.600	0.680	0.770	0.500
$\phi_{i(4)}$	5.000	5.000	0.000	0.005	0.000	0.000	0.000	0.834
$d_{i(5)}$	0.750	1.000	0.510	0.540	0.890	0.790	0.830	0.980
$\phi_{i(5)}$	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
$d_{i(6)}$	0.890	0.980	0.780	0.880	0.500	1.000	0.500	0.510
$\phi_{i(6)}$	0.000	0.000	0.000	1.231	5.000	0.131	0.013	0.000
$d_{i(7)}$	1.000	0.500	0.670	0.540	0.500	1.000	0.710	0.920
$\phi_{i(7)}$	3.099	0.514	0.000	0.013	0.668	4.306	0.000	1.566
$d_{i(8)}$	0.500	0.500	0.770	0.970	0.560	0.500	0.500	0.500
$\phi_{i(8)}$	0.779	0.000	0.000	0.000	0.000	0.127	0.040	2.088
$d_{i(9)}$	1.000	1.000	0.520	0.580	0.500	0.650	0.870	0.910

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Table E.8 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$\phi_{i(9)}$	0.891	3.061	0.560	0.662	0.972	0.000	0.000	0.000
$d_{i(10)}$	0.980	0.630	0.630	0.530	0.500	1.000	0.770	0.500
$\phi_{i(10)}$	0.000	0.000	2.181	0.013	1.158	1.488	0.000	0.068
$d_{i(11)}$	1.000	0.500	0.900	0.500	0.910	0.500	0.730	0.500
$\phi_{i(11)}$	0.573	0.287	0.000	0.069	0.000	0.697	1.791	0.035
$d_{i(12)}$	0.500	0.540	0.690	1.000	0.500	0.790	0.600	0.820
$\phi_{i(12)}$	2.314	1.977	0.000	0.000	0.025	0.000	0.000	0.000
$d_{t(1)}$	0.660	0.790	0.700	0.660	0.610	0.610	0.570	0.620
$\phi_{t(1)}$	0.847	1.769	2.293	2.375	2.781	2.767	2.784	2.927
$d_{t(2)}$	0.770	0.760	0.790	0.870	0.890	0.660	0.920	0.760
$\phi_{t(2)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(3)}$	0.500	0.500	0.500	0.500	0.500	0.570	0.500	0.600
$\phi_{t(3)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(4)}$	0.500	0.600	0.640	0.630	0.560	0.500	0.500	0.870
$\phi_{t(4)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(5)}$	0.770	0.780	0.700	0.740	0.730	0.940	0.610	0.500
$\phi_{t(5)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(6)}$	0.500	0.500	0.750	0.520	0.720	0.750	0.650	0.560
$\phi_{t(6)}$	4.126	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(7)}$	0.690	0.690	0.500	0.650	0.500	0.530	0.760	0.660
$\phi_{t(7)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$d_{t(8)}$	0.500	0.500	0.520	0.510	0.500	0.500	0.500	0.540
$\phi_{t(8)}$	5.000	5.000	5.000	5.000	5.000	5.000	4.999	5.000
$d_{t(9)}$	0.500	0.500	0.560	0.500	0.500	0.500	0.500	0.500
$\phi_{t(9)}$	0.424	1.397	3.742	3.981	5.000	5.000	4.999	5.000
τ_t	2.000	2.000	2.000	2.000	2.000	2.000	2.000	2.000
$D_{(1)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000
$D_{(2)}$	5.000	5.000	5.000	5.000	5.000	5.000	5.000	5.000
$D_{(3)}$	10.000	12.170	9.410	24.370	15.010	9.210	18.510	7.570

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Table E.8 – continued from previous page

$D_{\max}$	10	15	20	25	30	35	40	45
$D_{(4)}$	10.000	6.520	5.840	6.550	6.070	26.210	25.230	5.000
$D_{(5)}$	5.000	5.000	5.000	5.000	5.020	5.000	6.790	27.130
$D_{(6)}$	10.000	15.000	5.000	9.640	5.000	5.000	5.170	10.520
$D_{(7)}$	5.000	5.000	20.000	5.080	9.090	8.010	5.000	5.820
$D_{(8)}$	10.000	15.000	12.580	13.820	21.920	14.420	10.710	15.100
$D_{(9)}$	10.000	15.000	20.000	25.000	30.000	35.000	40.000	45.000

E.4 Geometric Parameters of Tested MC-MPPA Liners in Chapter 5

Table E.9: Geometric parameters of the sample II-00 liner.

Perforation diameter (mm)									
$d_{t(1)}$	1.00	$d_{t(2)}$	1.00	$d_{t(3)}$	1.00	$d_{t(4)}$	1.00	$d_{t(5)}$	1.00
$d_{t(6)}$	1.00	$d_{t(7)}$	1.00	$d_{t(8)}$	1.00	$d_{t(9)}$	1.00		
Perforation porosity (%)									
$\phi_{t(1)}$	3.88	$\phi_{t(2)}$	3.88	$\phi_{t(3)}$	3.88	$\phi_{t(4)}$	3.88	$\phi_{t(5)}$	3.88
$\phi_{t(6)}$	3.88	$\phi_{t(7)}$	3.88	$\phi_{t(8)}$	3.88	$\phi_{t(9)}$	3.88		
Perforation number									
$n_{t(1)}$	10	$n_{t(2)}$	10	$n_{t(3)}$	10	$n_{t(4)}$	10	$n_{t(5)}$	10
$n_{t(6)}$	10	$n_{t(7)}$	10	$n_{t(8)}$	10	$n_{t(9)}$	10		
Air cavity depth (mm)									
$D_{(1)}$	30.00	$D_{(2)}$	30.00	$D_{(3)}$	24.50	$D_{(4)}$	30.00	$D_{(5)}$	15.20
$D_{(6)}$	17.30	$D_{(7)}$	13.90	$D_{(8)}$	19.30	$D_{(9)}$	21.70		
Other geometric parameters									
L	50 mm	W	40 mm	τ_i	1 mm	τ_t	2 mm		

Table E.10: Geometric parameters of the sample II-25 liner.

Perforation diameter (mm)									
$d_{t(1)}$	1.00	$d_{t(2)}$	1.00	$d_{t(3)}$	1.00	$d_{t(4)}$	1.00	$d_{t(5)}$	1.00
$d_{t(6)}$	1.00	$d_{t(7)}$	1.00	$d_{t(8)}$	1.00	$d_{t(9)}$	1.00		
Perforation porosity (%)									
$\phi_{t(1)}$	3.88	$\phi_{t(2)}$	3.88	$\phi_{t(3)}$	3.88	$\phi_{t(4)}$	3.88	$\phi_{t(5)}$	3.88
$\phi_{t(6)}$	3.88	$\phi_{t(7)}$	3.88	$\phi_{t(8)}$	3.88	$\phi_{t(9)}$	3.88		
Perforation number									
$n_{t(1)}$	10	$n_{t(2)}$	10	$n_{t(3)}$	10	$n_{t(4)}$	10	$n_{t(5)}$	10
$n_{t(6)}$	10	$n_{t(7)}$	10	$n_{t(8)}$	10	$n_{t(9)}$	10		
Air cavity depth (mm)									
$D_{(1)}$	30.00	$D_{(2)}$	30.00	$D_{(3)}$	30.00	$D_{(4)}$	30.00	$D_{(5)}$	30.00
$D_{(6)}$	30.00	$D_{(7)}$	30.00	$D_{(8)}$	30.00	$D_{(9)}$	30.00		
Other geometric parameters									
L	50 mm	W	40 mm	τ_i	1 mm	τ_t	2 mm		

Table E.11: Geometric parameters of the sample III-00 liner.

Perforation diameter (mm)									
$d_{t(1)}$	1.00	$d_{t(2)}$	1.00	$d_{t(3)}$	1.00	$d_{t(4)}$	1.00	$d_{t(5)}$	1.00
$d_{t(6)}$	1.00	$d_{t(7)}$	1.00	$d_{t(8)}$	1.00	$d_{t(9)}$	1.00		
$d_{i(1)}$	0.80	$d_{i(2)}$	0.83	$d_{i(3)}$	0.00	$d_{i(4)}$	0.55	$d_{i(5)}$	0.00
$d_{i(6)}$	0.55	$d_{i(7)}$	0.75	$d_{i(8)}$	0.76	$d_{i(9)}$	0.52	$d_{i(10)}$	0.00
$d_{i(11)}$	0.74	$d_{i(12)}$	0.00						
Perforation porosity (%)									
$\phi_{t(1)}$	3.88	$\phi_{t(2)}$	3.88	$\phi_{t(3)}$	3.88	$\phi_{t(4)}$	3.88	$\phi_{t(5)}$	3.88
$\phi_{t(6)}$	3.88	$\phi_{t(7)}$	3.88	$\phi_{t(8)}$	3.88	$\phi_{t(9)}$	3.88		
$\phi_{i(1)}$	0.61	$\phi_{i(2)}$	0.27	$\phi_{i(3)}$	0.00	$\phi_{i(4)}$	0.48	$\phi_{i(5)}$	0.00
$\phi_{i(6)}$	0.13	$\phi_{i(7)}$	1.69	$\phi_{i(8)}$	0.30	$\phi_{i(9)}$	0.07	$\phi_{i(10)}$	0.00
$\phi_{i(11)}$	0.23	$\phi_{i(12)}$	0.00						
Perforation number									
$n_{t(1)}$	10	$n_{t(2)}$	10	$n_{t(3)}$	10	$n_{t(4)}$	10	$n_{t(5)}$	10
$n_{t(6)}$	10	$n_{t(7)}$	10	$n_{t(8)}$	10	$n_{t(9)}$	10		
$n_{i(1)}$	3	$n_{i(2)}$	1	$n_{i(3)}$	0	$n_{i(4)}$	6	$n_{i(5)}$	0
$n_{i(6)}$	1	$n_{i(7)}$	9	$n_{i(8)}$	2	$n_{i(9)}$	1	$n_{i(10)}$	0
$n_{i(11)}$	2	$n_{i(12)}$	0						
Air cavity depth (mm)									
$D_{(1)}$	30.00	$D_{(2)}$	19.60	$D_{(3)}$	16.00	$D_{(4)}$	18.60	$D_{(5)}$	18.60
$D_{(6)}$	14.90	$D_{(7)}$	30.00	$D_{(8)}$	30.00	$D_{(9)}$	21.30		
Other geometric parameters									
L	50 mm	W	40 mm	τ_i	1 mm	τ_t	2 mm		

Table E.12: Geometric parameters of the sample III-25 liner.

Perforation diameter (mm)									
$d_{t(1)}$	1.00	$d_{t(2)}$	1.00	$d_{t(3)}$	1.00	$d_{t(4)}$	1.00	$d_{t(5)}$	1.00
$d_{t(6)}$	1.00	$d_{t(7)}$	1.00	$d_{t(8)}$	1.00	$d_{t(9)}$	1.00		
$d_{i(1)}$	0.58	$d_{i(2)}$	0.76	$d_{i(3)}$	0.94	$d_{i(4)}$	0.81	$d_{i(5)}$	0.52
$d_{i(6)}$	0.96	$d_{i(7)}$	0.97	$d_{i(8)}$	0.86	$d_{i(9)}$	0.78	$d_{i(10)}$	0.96
$d_{i(11)}$	0.95	$d_{i(12)}$	0.81						
Perforation porosity (%)									
$\phi_{t(1)}$	3.88	$\phi_{t(2)}$	3.88	$\phi_{t(3)}$	3.88	$\phi_{t(4)}$	3.88	$\phi_{t(5)}$	3.88
$\phi_{t(6)}$	3.88	$\phi_{t(7)}$	3.88	$\phi_{t(8)}$	3.88	$\phi_{t(9)}$	3.88		
$\phi_{i(1)}$	0.14	$\phi_{i(2)}$	0.72	$\phi_{i(3)}$	1.16	$\phi_{i(4)}$	0.86	$\phi_{i(5)}$	0.22
$\phi_{i(6)}$	1.52	$\phi_{i(7)}$	1.36	$\phi_{i(8)}$	0.36	$\phi_{i(9)}$	0.30	$\phi_{i(10)}$	1.06
$\phi_{i(11)}$	0.37	$\phi_{i(12)}$	0.27						
Perforation number									
$n_{t(1)}$	10	$n_{t(2)}$	10	$n_{t(3)}$	10	$n_{t(4)}$	10	$n_{t(5)}$	10
$n_{t(6)}$	10	$n_{t(7)}$	10	$n_{t(8)}$	10	$n_{t(9)}$	10		
$n_{i(1)}$	2	$n_{i(2)}$	6	$n_{i(3)}$	8	$n_{i(4)}$	8	$n_{i(5)}$	5
$n_{i(6)}$	8	$n_{i(7)}$	7	$n_{i(8)}$	3	$n_{i(9)}$	3	$n_{i(10)}$	7
$n_{i(11)}$	2	$n_{i(12)}$	2						
Air cavity depth (mm)									
$D_{(1)}$	30.00	$D_{(2)}$	30.00	$D_{(3)}$	30.00	$D_{(4)}$	30.00	$D_{(5)}$	30.00
$D_{(6)}$	30.00	$D_{(7)}$	30.00	$D_{(8)}$	30.00	$D_{(9)}$	30.00		
Other geometric parameters									
L	50 mm	W	40 mm	τ_i	1 mm	τ_t	2 mm		

E.5 Geometric Parameters of Tested MHA and HMHA Samples in Chapter 6

Table E.13: Optimised geometric parameters of MHA corresponding to Figure 6.4, $L_{ac1} = W_{ac2} = 19.5$ mm and $W = 40$ mm.

D_{ac1}	n	τ	d	d_n	L_n	D_{ac2}	τ'
10 mm	3	1.2 mm	1 mm	5.9 mm	2 mm	10 mm	1 mm
20 mm	10	1.6 mm	0.8 mm	10 mm	2 mm	20 mm	1 mm
30 mm	10	1 mm	0.8 mm	10 mm	2 mm	30 mm	1 mm
40 mm	10	1 mm	0.86 mm	10 mm	2 mm	40 mm	1 mm
50 mm	10	1 mm	0.89 mm	10 mm	2 mm	43 mm	1 mm

Table E.14: Optimized geometric parameters of the HMHA corresponding to Figure 6.8, with $L_1 = W_1 = 12.67$ mm and $D_{\max} = 10$ mm.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
0.52	1	0.52	1
$d_{(5)}$ (mm)	$n_{(5)}$	$d_{(7)}$ (mm)	$n_{(7)}$
0.87	2	0.52	1
$d_{(9)}$ (mm)	$n_{(9)}$	—	—
0.52	1	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
2.00	2.82	2.00	2.76
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	$d_{n(8)}$ (mm)	$L_{n(8)}$ (mm)
2.00	2.93	2.00	2.92
Cavity depths and panel thicknesses			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
10	10	10	10
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	$D_{(7)}$ (mm)	$D_{(8)}$ (mm)
10	10	10	10
$D_{(9)}$ (mm)	τ (mm)	τ' (mm)	—
10	1.00	1.00	—

Table E.15: Optimized geometric parameters of the HMHA corresponding to Figure 6.8, with $L_1 = W_1 = 12.67$ mm and $D_{\max} = 20$ mm.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
0.76	1	0.77	1
$d_{(5)}$ (mm)	$n_{(5)}$	$d_{(7)}$ (mm)	$n_{(7)}$
0.86	3	0.76	1
$d_{(9)}$ (mm)	$n_{(9)}$	—	—
0.76	1	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
2.00	2.38	2.00	2.72
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	$d_{n(8)}$ (mm)	$L_{n(8)}$ (mm)
2.20	3.51	2.00	2.30
Cavity depths and panel thicknesses			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
20	20	20	20
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	$D_{(7)}$ (mm)	$D_{(8)}$ (mm)
20	20	20	20
$D_{(9)}$ (mm)	τ (mm)	τ' (mm)	—
20	1.00	1.00	—

Table E.16: Optimized geometric parameters of the HMHA corresponding to Figure 6.8, with $L_1 = W_1 = 12.67$ mm and $D_{\max} = 30$ mm.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
0.92	1	0.87	1
$d_{(5)}$ (mm)	$n_{(5)}$	$d_{(7)}$ (mm)	$n_{(7)}$
0.77	5	0.98	1
$d_{(9)}$ (mm)	$n_{(9)}$	—	—
0.99	1	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
2.04	3.48	2.11	3.46
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	$d_{n(8)}$ (mm)	$L_{n(8)}$ (mm)
2.60	6.10	2.18	3.78
Cavity depths and panel thicknesses			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
30	30	30	30
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	$D_{(7)}$ (mm)	$D_{(8)}$ (mm)
30	30	30	30
$D_{(9)}$ (mm)	τ (mm)	τ' (mm)	—
30	1.00	1.00	—

Table E.17: Optimized geometric parameters of the HMHA corresponding to Figure 6.8, with $L_1 = W_1 = 12.67$ mm and $D_{\max} = 40$ mm.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
0.76	2	0.76	2
$d_{(5)}$ (mm)	$n_{(5)}$	$d_{(7)}$ (mm)	$n_{(7)}$
0.81	5	0.77	2
$d_{(9)}$ (mm)	$n_{(9)}$	–	–
0.61	3	–	–
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
2.78	3.05	2.76	2.18
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	$d_{n(8)}$ (mm)	$L_{n(8)}$ (mm)
2.90	2.99	2.34	2.01
Cavity depths and panel thicknesses			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
40	40	40	40
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	$D_{(7)}$ (mm)	$D_{(8)}$ (mm)
40	40	40	40
$D_{(9)}$ (mm)	τ (mm)	τ' (mm)	–
40	1.00	1.00	–

Table E.18: Optimized geometric parameters of the HMHA corresponding to Figure 6.8, with $L_1 = W_1 = 12.67$ mm and $D_{\max} = 50$ mm.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
0.79	2	0.92	2
$d_{(5)}$ (mm)	$n_{(5)}$	$d_{(7)}$ (mm)	$n_{(7)}$
0.85	5	0.67	3
$d_{(9)}$ (mm)	$n_{(9)}$	–	–
0.68	3	–	–
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
3.11	2.65	2.99	2.80
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	$d_{n(8)}$ (mm)	$L_{n(8)}$ (mm)
4.12	6.43	4.52	10.14
Cavity depths and panel thicknesses			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
50	50	50	50
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	$D_{(7)}$ (mm)	$D_{(8)}$ (mm)
50	50	50	50
$D_{(9)}$ (mm)	τ (mm)	τ' (mm)	–
50	1.00	1.00	–

E.6 Geometric Parameters of Tested MHA and CMHA Samples in Chapter 7

Table E.19: Optimized geometric parameters of the MHA-I sample. $L_{ac1} = L_{ac2} = 24.5$ mm and $W = 40$ mm.

D_{ac1}	n	τ	d	d_n	L_n	D_{ac2}	τ'
45 mm	10	3 mm	2.29 mm	6.76 mm	20 mm	45 mm	1 mm

Table E.20: Optimized geometric parameters of the CMHA-I sample.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
1.65	1	1.19	3
$d_{(5)}$ (mm)	$n_{(5)}$	—	—
1.74	2	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
5.28	5.00	4.33	5.00
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	—	—
6.39	5.00	—	—
Cavity depths and panel thicknesses			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
25	25	25	25
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	τ (mm)	τ' (mm)
25	25	2.46	1.00

Table E.21: Optimized geometric parameters of the CMHA-II sample.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
1.01	6	1.05	8
$d_{(5)}$ (mm)	$n_{(5)}$	—	—
1.27	7	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
8.47	5.90	9.90	5.00
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	—	—
9.49	5.47	—	—
Cavity depths and panel thickness			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
25	25	25	25
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	τ (mm)	τ' (mm)
23.9	14.7	1.23	1.00

Table E.22: Optimized geometric parameters of the CMHA-III sample.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
1.33	2	1.02	6
$d_{(5)}$ (mm)	$n_{(5)}$	—	—
1.16	8	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
9.91	5.90	9.94	5.00
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	—	—
9.99	5.00	—	—
Cavity depths and panel thickness			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
25	25	25	25
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	τ (mm)	τ' (mm)
25	22.3	1.00	1.00

E.7 Geometric Parameters of Tested CMHA Liners in Chapter 7

Table E.23: Geometric parameters of the sample CMHA-00 liner.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
9.76	13	9.81	19
$d_{(5)}$ (mm)	$n_{(5)}$	—	—
9.87	25	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
7.99	5.00	8.00	5.00
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	—	—
8.00	5.00	—	—
Cavity depths and panel thickness			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
30	30	30	30
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	τ (mm)	τ' (mm)
30	30	2.00	1.00

Table E.24: Geometric parameters of the sample CMHA-25 liner.

Perforation diameters and numbers			
$d_{(1)}$ (mm)	$n_{(1)}$	$d_{(3)}$ (mm)	$n_{(3)}$
9.78	13	9.85	19
$d_{(5)}$ (mm)	$n_{(5)}$	—	—
9.88	25	—	—
Neck diameters and lengths			
$d_{n(2)}$ (mm)	$L_{n(2)}$ (mm)	$d_{n(4)}$ (mm)	$L_{n(4)}$ (mm)
8.00	5.00	7.99	5.00
$d_{n(6)}$ (mm)	$L_{n(6)}$ (mm)	—	—
5.90	5.02	—	—
Cavity depths and panel thickness			
$D_{(1)}$ (mm)	$D_{(2)}$ (mm)	$D_{(3)}$ (mm)	$D_{(4)}$ (mm)
30	30	30	30
$D_{(5)}$ (mm)	$D_{(6)}$ (mm)	τ (mm)	τ' (mm)
30	30	2.00	1.00