Design and development of 3D-printed polymeric structures for acoustic absorption

Yang, Wenjing

2020

Yang, W. (2020). Design and development of 3D-printed polymeric structures for acoustic absorption. Doctoral thesis, Nanyang Technological University, Singapore.
<https://hdl.handle.net/10356/147846>

<https://hdl.handle.net/10356/147846>

<https://doi.org/10.32657/10356/147846>

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**DESIGN AND DEVELOPMENT
OF 3D-PRINTED POLYMERIC STRUCTURES
FOR ACOUSTIC ABSORPTION**

YANG WENJING

**SCHOOL OF MECHANICAL AND AEROSPACE
ENGINEERING**

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**SCHOOL OF MECHANICAL AND AEROSPACE
ENGINEERING**

A thesis submitted to the Nanyang Technological University in
partial fulfilment of the requirement for the degree of
Doctor of Philosophy

2020

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research, is free of plagiarised materials, and has not been submitted for a higher degree to any other University or Institution.

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Yang Wenjing

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Supervisor Declaration Statement

I have reviewed the content and presentation style of this thesis and declare it is free of plagiarism and of sufficient grammatical clarity to be examined. To the best of my knowledge, the research and writing are those of the candidate except as acknowledged in the Author Attribution Statement. I confirm that the investigations were conducted in accord with the ethics policies and integrity standards of Nanyang Technological University and that the research data are presented honestly and without prejudice.

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Zhou Kun

Authorship Attribution Statement

This thesis contains material from two paper(s) published in the following peer-reviewed journal(s) / from papers accepted at conferences in which I am listed as an author.

Chapter 3 is published as **W. Yang**, X. Bai, W. Zhu, R. Kiran, J. An, C. K. Chua and K. Zhou. 3D printing of polymeric multi-layer micro-perforated panels for tunable wideband sound absorption, *Polymers*, 2020. **12**:360.

The contributions of the co-authors are as follows:

- I developed the experimental methods, obtained and analyzed the results of the simulation and the experiments, and drafted the manuscript.
- Prof. Zhou initiated the research idea, supervised the research, and revised the manuscript.
- Prof. Chua initiated the research idea, supervised the research, and revised the manuscript.
- Ms. Bai provided the technical support for the simulations.
- Dr. Zhu provided the insights to improve the manuscript.
- Dr. An provided the insights to improve the manuscript.
- Mr. Kiran revised the manuscript.

Chapter 4 is published as **W. Yang**, J. An, C. K. Chua and K. Zhou. Acoustic absorptions of multifunctional polymeric cellular structures based on triply periodic minimal surfaces fabricated by stereolithography, *Virtual and Physical Prototyping*, 2020. **15**(2):242-249.

The contributions of the co-authors are as follows:

- I developed the experimental methods, obtained and analyzed the experimental results, and drafted the manuscript.
- Prof. Zhou initiated the research idea, supervised the research, and revised the manuscript.
- Prof. Chua initiated the research idea, supervised the research, and revised the manuscript.
- Dr. An provided the insights and revised the manuscript.

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Acknowledgements

I would like to express my sincere gratitude and appreciation to my supervisor Associate Professor Zhou Kun and co-supervisor Professor Chua Chee Kai for their invaluable guidance and constant supports during my Ph.D. study. I give my heartfelt gratitude to Prof. Zhou for his rigorous and supportive supervision. I have immensely benefitted from his precious suggestions on scientific thinking, writing and presentation. He has always devoted his resources to supporting my research. I also appreciate Prof. Chua for providing crucial advice on my research. He has always encouraged me with patience and guided me to the correct path when I encountered difficulties.

I would like to express my appreciation to Associate Professor Tor Shu Beng and Assistant Professor Song Xu for serving as my thesis advisory committee. I owe a great deal of gratitude to Dr. An Jia and Dr. Yuan Shangqin for the valuable discussions about my research. My thanks also go to Dr. Tan Pengfei, Dr. Zhu Wei, Mr. Li Weidong, Ms. Bai Xueyu, Mr. Raj Kiran for their suggestions and support.

I gratefully acknowledge the Singapore Centre for 3D Printing at the Nanyang Technological University, Singapore for providing the research scholarship and continuous support during my Ph.D. study.

Last but not least, I wish to express my utmost gratitude to my family and friends. My family, especially my mother, have always loved and encouraged me. My friends Mr. Liu Likun and Mr. Zhou Xiaosong have provided me technical supports. I would like to give special thanks to my friend Ms. Chi Yuchen for her company and Mr. Johnny Kitagawa for bringing me hope and light during the tough time in my entire journey.

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Abstract

In acoustic engineering, noise control is of concern in various fields including industrial machines, vehicles, architectures and home appliances. Sound absorption is one of the major solutions to control noise and the efficiency of sound absorption can be evaluated by the acoustic absorption coefficient and the absorption frequency bandwidth. The development of novel acoustic absorbers is crucial in enhancing the efficiency of sound absorption and fulfilling various environmental requirements of acoustic absorption in different frequency ranges. The enhancement of functional properties of absorbers should also be considered by innovative designs. Additive manufacturing (AM) has been introduced to reduce the prototyping costs and lead time of sound absorbers, while the potential of AM for fabricating sound absorbers with complex structures has not been explored. This Ph.D. study designs and develops different types of novel acoustic absorbers with complex structures to enhance the efficiency of sound absorption in various frequency ranges by achieving wideband absorption. The development of the absorbers involves two different polymer AM processes owing to additive manufacturability of the complex designs as well as prototype cost. Three novel absorber prototypes with improved performances have been successfully developed.

Firstly, for sound absorption enhancement in the range from low frequency to middle frequency, innovative multi-layer micro-perforated panels (MPPs) with adjustable geometries for tunable wideband absorption are designed and fabricated by selective laser sintering. The finite element method is applied to provide numerical predictions of the performances of the designed structures. The effects of structural

parameters of multi-layer MPPs on absorption coefficients and frequency bandwidths are analyzed by theoretical predictions, numerical simulations and experiments. The results reveal that the absorption frequency bandwidths of MPPs are broadened by the multi-layer designs, while the absorption coefficients remain comparable or even higher. The frequency ranges can be tuned by varying the air gap distances and the inter-layer distances. An optimization method, in which the area under the sound absorption curve as the criterion to evaluate the sound absorption ability is maximized, is introduced to design acoustic absorbers with the most effective sound absorption.

Secondly, for sound absorption enhancement in the range of high frequencies, triply periodic minimal surface (TPMS) structures that have shown potential absorption abilities but have not been experimentally validated are designed for characterization of the acoustic performances of innovative acoustic absorbers with multifunctionality. Three typical types of TPMS structures: Primitive (P), Gyroid (G) and Diamond (D) are designed and fabricated by stereolithography apparatus using polymeric resins. D structures exhibit superior absorption abilities in terms of high absorption coefficients and wide bandwidths. A series of D structures are designed to study the effects of the unit cell size, volume fraction and height on acoustic absorption. The analysis of the results provides an optimized design of geometric parameters with a small unit cell size and a large volume fraction for optimal acoustic absorption.

Finally, for further enhancement of sound absorption of the TPMS absorbers, innovative micro-perforated TPMS absorbers are designed and investigated by combining MPPs and TPMS structures. Significant improvements on acoustic absorption are achieved by the micro-perforated TPMS structures, especially for P and G

structures with lower absorption coefficients compared to those of D structures. In addition, the effects of the perforation ratio of P and G structures are studied. The increase of the perforation ratio in the observed range exhibits positive effects on improving the acoustic absorption of the structures. The fabrication materials are proved to have no major effects on the acoustic absorption performances of the TPMS structures and therefore can be selected according to the application environments and requirements.

The study of innovative acoustic devices is significant in acoustic engineering and is facilitated by AM that efficiently manufactures the absorber prototypes with complex structures. AM is promising to be applied as an enabling technology that accelerates the development of acoustic engineering by fabricating innovative and customized designs.

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List of Symbols

Symbol	Physical meaning
t	Panel thickness
d	Perforation diameter
p	Perforation ratio
D	Air gap distance
θ	Perforation angle
h	Height
v	Velocity
u	Average velocity
ρ_0	Density of air
η	Velocity of air
ω	Angular frequency, periodicity
P	Sound pressure
Z	Acoustic impedance
c_0	Speed of sound in air
f	Frequency
α	Sound absorption coefficient
q	Perforation constant
μ	Coefficient of the kinematic viscosity of air
$\text{Re}(Z)$	Real part of the impedance
$\text{Im}(Z)$	Imaginary part of the impedance
L	Total length

l	Unit cell size
ρ	Volume fraction
c	Controller of the volume fraction
R	Reflection coefficient
k_0	Wave number
J	Bessel's function

Chapter 1 Introduction

This Ph.D. dissertation aims to design and develop innovative 3D-printed polymeric acoustic absorbers. In this chapter, the research background is first stated in Section 1.1. The motivations and objectives of the research are then elaborated in Sections 1.2 and 1.3, respectively. The outline of the dissertation is provided in Section 1.4.

1.1. Background

In acoustic engineering, noise control is of concern in various fields including industrial machines, vehicles, architectures and home appliances. Sound absorption is one of the major solutions to control noise by viscous dissipation, heat conduction or resonance (Yang and Sheng, 2017). The efficiency of sound absorption can be evaluated by the acoustic absorption coefficient and the absorption frequency bandwidth. The acoustic absorption coefficient is the ratio of the absorbed energy to the incident energy and is dependent on the acoustic frequency (Fan and Fu, 2016). The absorption frequency bandwidth is the difference between the highest and the lowest frequency in a continuous range of frequencies at which the sound energy is effectively absorbed so that the absorption coefficients are no less than a target value.

Conventional acoustic absorbers can be classified into porous absorbers and resonant absorbers according to the different mechanisms used to reduce acoustic energy. Porous absorbers that consist of porous materials such as foams, fibrous materials and lattices reduce acoustic energy by viscous dissipation and heat conduction when the acoustic wave passes through the materials (Yang and Sheng, 2017). Conventional resonant absorbers such as Helmholtz resonators, membrane resonators and degenerate

resonators convert acoustic energy into mechanical oscillatory energy through incident sound pressure on the absorbers (Oldfield, 2006). Micro-perforated panels (MPPs) are special resonant absorbers with wider absorption bandwidths. A micro-perforated panel (MPP) system comprises a thin panel with sub-millimeter perforations, a rigid wall and an air gap between the panel and the wall to form air columns. Acoustic energy is transferred to thermal energy by the viscous effects of resonant vibrations of the air columns. Energy dissipation is achieved by the structural configuration and is independent of the material of the MPP panel (Johnson *et al.*, 1987; Maa, 1975). Within the audible frequency range of 20–20000 Hz, porous absorbers are applied for absorption in high-frequency ranges, while resonant absorbers are used for absorption in low-frequency ranges. The absorption bandwidth of conventional resonant absorbers is inherently narrow because of the high resonant frequencies of the structures on the sub-wavelength scales (Yang and Sheng, 2017), and thus MPPs have been developed for wideband absorption in the range from low frequency to middle frequency.

Besides conventional acoustic absorbers, the increasing concern about noise control has accelerated the development of novel acoustic absorbers to fulfill the requirements of different environmental conditions, various target absorption frequencies, minimized manufacturing time and cost, enhanced absorption efficiency and improved mechanical structural properties. For example, acoustic metamaterial absorbers that enhance the energy density and sound dissipation have been developed to achieve near-perfect absorption performances within deep-subwavelength volumes (Duan *et al.*, 2015; Jiang *et al.*, 2014; Mei *et al.*, 2012) and to extend the applications in wide acoustic fields including acoustic cloaking (Faure *et al.*, 2016; Zigoneanu *et al.*, 2014) and one-way

transmittance (Fleury *et al.*, 2014; Liang *et al.*, 2010). For simple-structure absorbers such as MPPs, innovative designs have been investigated for absorption with high efficiency or wide bandwidth in comparison with those of conventional absorbers (Ruiz *et al.*, 2011). The geometric variations of MPPs include non-circular perforations (Ning *et al.*, 2016; Randeberg, 2002), modified shapes of the air gap (C.Wang *et al.*, 2010; C.Wang and L.Huang, 2011; Sakagami *et al.*, 2009a), partitioned air gap (Gai *et al.*, 2017; Toyoda and Takahashi, 2008; Yairi *et al.*, 2006) and multi-layer configurations of the structures (Ruiz *et al.*, 2011; Sakagami *et al.*, 2010; Sakagami *et al.*, 2009b). Furthermore, the micro-slit or micro-slot MPPs with perforations in easy-drilling shapes have been studied for easy fabrication and therefore are able to reduce the manufacturing cost (Maa, 2000; Randeberg, 2002). However, the simplifications of the geometric designs of absorbers for manufacturing cost reduction are only applicable for limited absorbers with simple structures, and the undiminished absorption efficiency of the replacement designs needs to be verified. Advanced manufacturing processes are necessary for the cost reduction in manufacturing and the development of absorbers with complex structures. Since the absorption abilities of resonant absorbers and MPPs are independent of the materials, replacing conventional materials with less costly materials can further contribute to cost reduction without compromising performance enhancement.

Conventional manufacturing processes require several steps including cutting, drilling and assembling to fabricate an absorber. Figure 1.1 shows the strip slotting method that fabricated a Helmholtz resonator liner in several steps (Dannemann *et al.*, 2018). The costly manufacturing methods such as laser cutting, micro-punching and electro-etching have been applied for the challenging fabrication of sub-millimeter

perforations on the pre-manufactured panels of MPPs (Arenas and Crocker, 2010). Metals, especially steels, have been extensively utilized as the materials of the manufacturing methods mentioned above. Recently, as an ideal solution to complex structures for multi-discipline applications including bioactive implants and tissue scaffolds through the layer-by-layer buildup of pre-designed computer models (An *et al.*, 2015; Lee *et al.*, 2016a; Shuai *et al.*, 2018; Yang *et al.*, 2019), additive manufacturing (AM), or three-dimensional (3D) printing, has been increasingly applied to fabricate resonant absorbers, MPPs and prototypes of novel absorbers in a single step. 3D printing and AM are utilized interchangeably in this dissertation. Among various AM systems, selective laser sintering (SLS), stereolithography apparatus (SLA) and Polyjet have mostly been introduced to produce acoustic devices with polymeric materials for reduction of costs, processing time and the total weight of structures (Godbold, 2008; Liu *et al.*, 2016, 2017; Setaki *et al.*, 2014; Setaki *et al.*, 2016).

Figure 1.1. Fabrication process of a Helmholtz resonator liner: (a) preparation of adhesive bonding of films, (b) assembling of support structures, (c) attachment of the face sheet of an acoustic liner sample and (d) completion of the sample (Dannemann *et al.*, 2018).

The introduction of AM has not only facilitated the manufacturing of existing structures but also enabled the fabrication of complex structures that could not be realized by other manufacturing methods. Recently, many theoretical studies have been conducted on cellular structures based on triply periodic minimal surfaces (TPMS). TPMS are mathematically defined continuous surfaces in various types depending on the equations describing them. The surfaces exhibit periodicity in three perpendicular dimensions with zero mean curvatures and locally minimized surface areas (Chen *et al.*, 2009; Lord and Mackay, 2003). The unique geometries of TPMS structures have enabled them as a novel type of porous materials for multifunctional applications. These structures have demonstrated superior mechanical properties including high stiffness and strength (Abueidda *et al.*, 2017; Yan *et al.*, 2014), energy absorption ability (Maskery *et al.*, 2017; Zhang *et al.*, 2018), thermal and electrical conductivity (Torquato and Donev, 2004), fluid permeability (Jung and Torquato, 2005) and vibration isolation ability (Elmadih *et al.*, 2019). Recently, TPMS structures have achieved acoustic bandgaps that presented the prohibition of sound wave propagation in certain ranges of acoustic frequencies, which indicated the potential of TPMS structures as acoustic absorbers with multifunctionality, while the existing literature is limited and lacks experimental validations (Abueidda *et al.*, 2018; Chen *et al.*, 2013).

1.2. Motivation

In acoustic engineering, the development of acoustic absorbers is crucial in enhancing the efficiency of sound absorption and fulfilling various environmental requirements of acoustic absorption in target sound frequency ranges. As conventional

acoustic absorbers, porous absorbers are applied for absorption high-frequency ranges while resonant absorbers are used for absorption in low-frequency ranges with inherently narrow bandwidths. However, novel absorbers are necessary to be developed for sound absorption at various frequency ranges with wide bandwidths and high absorption coefficients, and the frequency ranges of absorption are ideally tunable by variations of the geometries of absorbers reducing different noises. The reduction of manufacturing costs and the enhancement of functional properties of absorbers should also be considered by innovative designs.

An MPP is an ideal absorber to be studied for cost reduction and development of manufacturing methods as its sound absorption performance is independent of the material but highly dependent on geometric characteristics (Johnson *et al.*, 1987; Maa, 1975). Metals are commonly utilized as manufacturing materials of conventional MPPs so that the perforations in millimeter-scale can be fabricated conveniently, while polymeric materials are promising candidates owing to their characteristics such as low cost and light weight as compared to metals. Meanwhile, in order to expand and tune the frequency range of acoustic absorption, novel MPPs with adjustable geometries for tunable wideband absorption in various frequency ranges are necessary to be developed. A multi-layer MPP with an adjustable number of panel layers and inter-layer distances is a promising solution. A method to evaluate the efficiency of designed structures and provide optimized structural parameters should be developed at the same time.

TPMS structures with a small unit cell size are able to exhibit acoustic bandgaps that indicate their potential as acoustic absorbers for absorption in the range of high frequencies, while the existing literature is limited and lacks experimental validations

(Abueidda *et al.*, 2018; Chen *et al.*, 2013). As promising acoustic absorbers, TPMS structures have several advantages compared to traditional cellular structures and materials. Traditional cellular structures and materials have been widely used as sound absorbers with strong absorption abilities while their poor mechanical properties have restricted the applications (Duan *et al.*, 2011; Liu and Chen, 2014b). TPMS structures are multifunctional cellular structures with excellent mechanical, fluid, thermal and electrical properties (Abueidda *et al.*, 2017; Jung and Torquato, 2005; Torquato and Donev, 2004; Yan *et al.*, 2014). In addition, for porous absorbers, while the acoustic absorption abilities, the ranges of acoustic absorption frequencies and the frequency bandwidth are determined by the intrinsic properties of traditional porous materials, they are tunable by adjusting the geometric parameters of TPMS structures.

To facilitate the design, prototyping and manufacturing of novel acoustic absorbers, a low-cost, fast-speed and high-quality manufacturing method to fabricate complex structures with various candidate materials is in demand. AM is a solution to be applied as an advanced manufacturing method that fabricates objects through layer-by-layer buildup of pre-designed computer models (ASTM International, 2015; Chua and Leong, 2017). AM is an effective method to produce polymeric MPPs with perforations in submillimeter and is the only method to fabricate TPMS structures. Among all AM systems, SLS has the advantages of excluding the design, printing and removal of supporting structures when manufacturing multi-layer structures. However, the feasibility of applying SLS in MPP manufacturing needs to be proved (Berretta *et al.*, 2014). Meanwhile, due to the small unit size and geometric accuracy requirements of the designed structures, SLA with the advantages of high accuracy, good surface finish and

low cost is selected to fabricate TPMS structures as acoustic absorbers (Melchels *et al.*, 2010; Yu *et al.*, 2019).

To sum up, introducing design complexity to sound absorbers such as multi-layer MPPs and TPMS structures could potentially solve current problems of narrow bandwidth and limited tunability while keeping high absorption coefficients of the conventional absorbers. To fabricate the complex absorbers, AM should be introduced as an effective, efficient and cost-saving manufacturing method compared to conventional manufacturing methods. Meanwhile, the validation of the acoustic properties of physical TPMS structures would fill a gap in the current acoustic research and enhance the multi-functionality of TPMS structures.

1.3. Objectives

The development of acoustic absorbers contributes significantly to noise control in acoustic engineering. The present Ph.D. study aims to design, evaluate and develop different types of novel acoustic absorbers for tunable wideband sound absorption in various frequency ranges. The introduction of different AM systems as the fabrication method and polymers as manufacturing materials for cost reduction and manufacturing effectiveness facilitates the development of the absorbers. The main objectives of the present research are as follows:

- To develop multi-layer MPPs as acoustic absorbers for tunable wideband sound absorption in the range from low frequency to middle frequency;
- To develop multifunctional acoustic absorbers based on TPMS structures for effective sound absorption in the range of high frequencies;

- To enhance the absorption performances of TPMS absorbers by adding micro-perforations;
- To apply AM as a solution to facilitate the development of acoustic engineering by efficiently manufacturing the designed acoustic devices with complex structures.

1.4. Dissertation layout

This Ph.D. dissertation is organized into six chapters. Following this chapter, Chapter 2 reviews the fundamentals of AM, noise control by sound absorption, acoustic absorbers, cellular structures and TPMS structures. In Chapter 3, multi-layer polymeric MPPs are developed for tunable wideband sound absorption in the range from low frequency to high frequency. Meanwhile, an optimization method is provided for structural designs of MPPs with the most effective sound absorption performances in target frequency ranges. In Chapter 4, TPMS structures are designed and evaluated as acoustic absorbers for wideband sound absorption in the range of high frequencies. Chapter 5 proposes TPMS absorbers with micro-perforations for absorption enhancement. In Chapter 6, the main achievements of this dissertation are concluded, and future works are recommended.

Chapter 2 Literature Review

This chapter consists of three sections. In Section 2.1, the fundamental understanding of additive manufacturing (AM) and the comprehensive review of three advanced AM systems: selective laser sintering (SLS), stereolithography apparatus (SLA) and fused deposition modeling (FDM) are provided. In Section 2.2, the fundamentals of sound absorption and the classification of acoustic absorbers are reviewed, followed by the demonstration of existing AM-manufactured acoustic absorbers, which links to Section 2.1. The design principle, the selected AM system for fabrication and the performance of each absorber are discussed. Furthermore, micro-perforated panels (MPPs) are singled out as the special acoustic absorbers. The working principles, theoretical backgrounds, absorption performances and manufacturing methods of MPPs are demonstrated. In Section 2.3, porous structures are reviewed and the structures based on triply periodic minimal surfaces (TPMS) that can only be fabricated by AM are emphasized due to their multifunctionality and unique geometric characters. The existing applications and research on porous structures including TPMS structures for acoustic absorption are discussed and linked to Section 2.2.

2.1. Additive manufacturing

AM is defined by the American Society for Testing and Materials (ASTM) as a manufacturing process to fabricate objects from three-dimensional (3D) model data by joining materials layer upon layer, which is opposite to the subtractive manufacturing process (ASTM International, 2015). The ability to turn digital model data into physical

objects with complex structures and shapes has enabled AM to revolutionize the manufacturing industry (Chua and Leong, 2017). It has fabricated extremely complex structures for multi-discipline applications (An *et al.*, 2015; Ng *et al.*, 2019; Yuan *et al.*, 2019a; Zhuang *et al.*, 2018). Increasing attention has also been demonstrated by studies on applying AM in developing the design and manufacturing of acoustic devices. Recent studies on the acoustic applications of AM will be described in detail in Section 2.2.2 (Godbold *et al.*, 2007; Liu *et al.*, 2016, 2017; Setaki *et al.*, 2014).

2.1.1. Basics of additive manufacturing

The process of AM is generally described as a layer-by-layer manufacturing process, which can be explained as a sequence of basic steps: (i) the parts that are represented as closed surfaces and enclosed volumes are modeled on a computer aided design computer aided manufacturing (CAD-CAM) system; (ii) the built models are converted into STereoLithography (STL) files which use triangles and polygons to approximate the surfaces; (iii) the models are analyzed and sliced into cross sections, and then reformed into 3D objects through binding of laminations or solidifications of liquids and powders (Chua and Leong, 2017). With the advantage of fabricating complex objects in a relatively short time, AM has enabled more complicated designs with simplified fabrication processes that reduce manufacturing costs (Chua *et al.*, 1998). Consequently, product designers can enhance structure functions by improving structure complexity without manufacturing limitations or extra work and costs.

For the convenience of selecting appropriate AM systems to meet manufacturing requirements, existing AM systems can be classified by different methods. One of the

simplest methods broadly classifies AM systems into three types by the initial form of the utilized material: liquid-based, solid-based and powder-based systems. Another method according to the ASTM standard of the manufacturing principle categorizes the systems into seven types: material extrusion, sheet lamination, material jetting, directed energy deposition, vat photopolymerization, binder jetting and powder-bed fusion (Standard, 2012). According to the ASTM standard, SLA is a vat polymerization system, SLS is a powder bed fusion system and FDM is a material extrusion system.

While the initial states of the printing materials are classified into three types, a wide range of candidate materials including metals, ceramics, polymers, wax, resins and paper are available for AM products. For the manufacturing of acoustic apparatus, polymers are the most common materials considering the requirements of relatively light weight and high rigidity of the structures. In addition, composite materials have the potential to improve the functions and effects of acoustic structures. Therefore, AM systems that use polymers or polymer composites as building materials and provide relatively good accuracy in geometric dimensions have been used in pioneering research for quick concept verification and the preparation of future manufacturing of acoustic applications (Godbold *et al.*, 2007; Liu *et al.*, 2016, 2017; Setaki *et al.*, 2014). The most-used polymeric materials are photosensitive resins and polymer powders (Tan *et al.*).

2.1.2. Selective laser sintering

SLS is a powder-based AM system that fabricates objects by a CO₂ laser beam sintering thermoplastic polymer powders layer by layer as illustrated in Figure 2.1 (Childs and Tontowi, 2001; Hao *et al.*, 2006; Yang *et al.*, 2002). The cross sections of the

parts are selectively sintered, while the un-sintered powders are served as supports and are highly recyclable. Compared to melting and fusing, laser sintering owns the advantage of avoiding distortions due to the flow of molten material during the fusing or melting process (Chua and Leong, 2017). The critical factors that have influences on input laser energy as a crucial process parameter include laser power, laser scanning speed, hatching space and laser beam size (Yuan, 2018). Besides the printing process parameters, functionalities and properties of the fabricated products are affected by building materials. While polyamides including Polyamide 11 and Polyamide 12 (PA12) are the most commonly utilized materials among the large candidate building materials of SLS, intensive studies have been conducted for developing the functionalities of current materials by introducing improved polymers and polymeric composites (Gaikwad *et al.*, 2012; Lao *et al.*, 2010). Thermoplastic elastomeric materials have also been used to manufacture flexible structures by SLS (Bates *et al.*, 2016; Raasch *et al.*, 2015; Verbelen *et al.*, 2017; Yuan *et al.*, 2016b; Yuan *et al.*, 2017). The development of new materials for SLS is a complicated process that involves the manufacturing of powders, characterization of powders, sintering, characterization of sintered parts and reliability verification (Yuan, 2018).

Figure 2.1. Schematic illustration of an SLS system (CustomPart.Net, 2008a).

The strengths of SLS compared to other AM systems are mainly concluded as (i) a wide range of candidate materials including ceramics, metals, polycarbonates and nylon in powder form can be processed by the sintering process (Chua and Leong, 2017; Goodridge *et al.*, 2012); (ii) there is no requirement of supporting structures to build complex or hollow objects as they are supported by un-sintered powders (Shahzad *et al.*, 2012; Shahzad *et al.*, 2014; Wen *et al.*, 2014); (iii) un-sintered powders (Berretta *et al.*, 2014; Schmidt *et al.*, 2007) are highly recyclable. Besides recycling the un-sintered powders during the manufacturing process, the laser-sintered composites can also be reused for other post processes to reduce manufacturing costs (Kenzari *et al.*, 2012;

O'Neill *et al.*, 1999; Song and Koenig, 1997). The end-use applications of SLS are mainly concept models, functional prototypes and tools in architecture, automobile and surgical platforms, and the applications have been extended to the components in aircraft and aerospace apparatus (Bai *et al.*, 2015; Yuan *et al.*, 2016a; Yuan *et al.*, 2016b).

2.1.3. Stereolithography apparatus

SLA is a liquid-based AM system that builds the objects by spatially controlled photopolymerization of liquid resins as illustrated in Figure 2.2 (Chua and Leong, 2017; Meicheis *et al.*, 2010; Skoog *et al.*, 2014). Part of the photosensitive resin is solidified when the laser beam is irradiated on the resin surface in a precise pattern that has been designed in STL files. A building platform where the first layer of the photopolymerized part is adhered to provides the support for the object. After polymerization of a layer, the platform is moved to a designed height for polymerization of the subsequent layer and the process is repeated until the designed object is complete. The depth of each layer of the resin to be cured during photopolymerization is a crucial parameter that is dependent on the sintering temperature and the energy of the laser beam determined by the power of the light source and the laser scanning speed. Post-processing is necessary to remove the supports for overhanging or undercutting designs and post-curing by ultraviolet light is often required to cure the objects completely and to improve the mechanical properties of the objects.

Figure 2.2. Schematic illustration of an SLA system (CustomPart.Net, 2008b).

SLA was developed by 3D Systems in 1988 as the first commercial AM system and has been developed to be superior in accuracy, resolution and surface finish to other AM systems. A feature size at $5\text{ }\mu\text{m}$ can be reached by SLA while the general feature sizes of objects printed by other AM systems are larger than $50\text{ }\mu\text{m}$ (Gittard and Narayan, 2010). Even structures in micron size with a sub-micron resolution of $0.43\text{ }\mu\text{m}$ have been realized by a developed SLA system (Maruo and Ikuta, 2002). Structures are fabricated in an upright approach that the building platform is under the resin surface or an upside-down approach that the building platform is dipped into the resin and is lifted by the designed height to cure successive layers of the resin material (shown in Figure 2.3). The upside-down approach, Digital Light Processing (DLP), has been widely developed as it

provides smoother surfaces with a smaller amount of resins when fabricating the same structures as the bottom-up approach and minimizes oxygen inhibition (Hornbeck, 1996; Meicheis *et al.*, 2010).

Figure 2.3. Schematic comparison of an upright system with a scanning laser (left) and an upside-down system with digital light projection (right) (Meicheis *et al.*, 2010).

Most of the commercially available materials of SLA are conventional monomers with low viscosity, high photosensitivity, oxygen inhibition and low sensitivity to temperature changes (Corbel *et al.*, 2011). The materials have been extended to biodegradable copolymers (Matsuda *et al.*, 2000), polymer-ceramic composites (Provin and Monneret, 2002), polymer-metal composites (Corbel *et al.*, 2011), etc., to create objects with superior properties (mechanical, thermal, electrical, etc.) of composite materials for functional applications (Mansour *et al.*, 2007). The SLA systems printing multiple resins at a time have been developed while complex platforms, polymerization progresses and rinsing steps are required (Choi *et al.*, 2011; Mapili *et al.*, 2005).

2.1.4. Fused deposition modeling

FDM is a solid-based AM system that extrudes a fiber of polymeric material by a moving nozzle to build the physical model via a layer-by-layer method (Chua and Leong, 2017; Razavi and Thakor, 2017). As illustrated in Figure 2.4, the nozzle head moves along X and Y directions to extrude melted filaments on a building platform according to the CAD designs of structures. The building platform is lowered down after the completion of each layer for fabrication of successive layers (Xu *et al*, 2005). The filaments are melted to a semi-liquid phase by the heating element in the liquefier head and the layer being processed should be completed before the solidification of the materials or the properties of the built structures will be affected (Solomon *et al*, 2021). Besides the condition of the extruded material, various parameters including the raster angle, air gap between two adjoining bed depositions, printing speed and extrusion temperature have significant effects on fabrication and the built structures. Building orientation, which describes the direction of structure placement on the building platform, affects the compressive strength and tensile strength of the printed structures (Ashtankar *et al*, 2013). Building layer thickness, which is dependent on the diameter of the nozzle tip, has effects on bending and impact properties. A small layer thickness will lead to better bending properties and poorer impact properties (Toro *et al*, 2019).

Figure 2.4. Schematic illustration of an FDM system (CustomPart.Net, 2008b).

Durable components are commonly fabricated by FDM using high-strength thermoplastics including polycarbonate, acrylonitrile butadiene styrene (ABS), polylactide (PLA) and polyphenylsulfone (Najmon *et al*, 2019; Deshmukh *et al*, 2020). ABS is highly resistant to heat with ductility and PLA is biodegradable with good rigidity (Dey and Yodo, 2019). For applications in tissue engineering, polymers with a low glass transition temperature and high decomposition temperature, such as polycaprolactone (PCL), have been commonly used in scaffold fabrication (Fairbanks *et al*, 2009; Razavi and Thakor, 2017). Biocompatible products with excellent mechanical properties and

interconnectivity of pores with high porosity can also be achieved (Bose and Bandyopadhyay, 2016).

Due to the advantages of simplicity and cost-effectiveness of the process, FDM has become one of the most widely used AM techniques for functional parts and prototypes (Sheoran and Kumar, 2020; Zhang *et al*, 2020). In addition, no organic solvent is required for post-processing and removal of supporting structures. However, poor surface finish and low dimension accuracy are two major issues of the structures fabricated by FDM (Mohamed *et al*, 2015; Kuo and Mao, 2016).

2.1.5. Summary

The basics of AM including the manufacturing process, the classifications of AM systems and processing materials have been provided. Furthermore, three AM systems, SLS, SLA and FDM, have been reviewed by demonstrating their working principles, crucial parameters in printing, materials and advantages.

2.2. Noise control by sound absorption and acoustic absorbers

In acoustic engineering, noise reduction has received considerable attention in the past years. Major solutions to control noise include sound insulation, sound absorption, vibration damping and vibration isolation. Sound absorption reduces the noises by the viscous dissipation in the vicinity of the solid surfaces of sound absorbers, heat conduction through the solids of absorbers, or local resonances (Yang and Sheng, 2017). Acoustic absorbers are devices for sound absorption such as porous absorbers, resonant absorbers and MPPs.

2.2.1. Fundamentals of sound absorption and classification of acoustic absorbers

The effect of sound absorption can be attributed to three major mechanisms. The first mechanism is the frictional losses of air molecules that are oscillated by the sound wave and interact with absorbing structures or materials. The second mechanism is heat losses due to the isothermal transformation of the energy from the air to absorbing materials. The third mechanism is the internal losses of acoustic materials subjected to forced mechanical oscillation caused by acoustic pressure, which mainly occurs in closed-cell porous materials (Wang, 2010).

Acoustic absorbers that have been extensively applied can be classified according to different principles of acoustic energy reduction. For porous materials, the viscous dissipation and heat conduction at the fluid-solid interface of the structures with micro-geometric parameters provide an effective reduction of energy. For conventional resonant absorbers including Helmholtz resonators, membrane resonators and degenerate resonators, local resonances and geometric constraints serve as the framework and convert acoustic energy into mechanical oscillatory energy by the sub-wavelength scale structures (Yang and Sheng, 2017). Effective absorption of conventional resonant absorbers is achieved in the vicinity of resonant frequencies, resulting in a relatively narrow-band absorption. For MPPs as special resonant absorbers, acoustic energy reduction is achieved by viscous effects of resonant vibrations of the air columns provided by the micro-geometric parameters.

The sound absorption coefficient, which is the ratio of absorbed energy to incident energy, is used to evaluate the efficiency of sound absorption of a material or a structure. The parameters that usually characterize porous absorbers in sound absorption include

porosity, airflow resistance, air permeability and tortuosity. Porosity is the ratio of the volume of voids in the material to its total volume (Allard *et al.*, 1989). Porosity describes the ratio of the spaces in which the sound wave can enter the structure to dissipate acoustic energy (Seddeq, 2009). Airflow resistance measures the resistance of the material that converts part of acoustic energy into heat energy. The value of the airflow resistance of a porous material can be measured by an impedance tube method (Standard, 1990). The airflow resistance per unit thickness is proportional to the coefficient of shear viscosity of air and inversely proportional to the square of the pore size (Ingard, 1994). Air permeability is the rate of airflow passing perpendicularly through an area under a prescribed air pressure differential between two surfaces of a material (Perlmutter, 2015). A smaller air permeability will reduce sound waves entering the material and thus enhance sound absorption. Tortuosity is the structure shape factor that describes the degree of twist of the structures as well as the paths of airflow passing through the structures. It is defined as the ratio of the channel length to the straight-line distance between its two ends (Johnson *et al.*, 1987). A larger tortuosity will increase the length of the path and the viscous losses of sound waves as the contacting time between the waves and the structures increases, thereby achieving larger acoustic absorption (Newell, 2017).

The acoustic absorption of MPPs, the parameters and the calculation of absorption coefficients will be reviewed in detail in Section 2.2.2. The examples of resonant absorbers and porous materials will be reviewed in Sections 2.2.3 and 2.3.3, respectively.

2.2.2. Micro-perforated panels

MPPs have been extensively applied in room acoustics, environmental noise barriers (Figure 2.5) and duct silencers (Figure 2.6). For room acoustics, MPPs are installed on walls and ceilings of rooms such as music halls, audio rooms and industrial rooms, to trap noises and reduce sound reverberation (Yang and Cheng, 2016). For outdoor acoustics, transparent MPPs made of glass and polycarbonate have been studied to be applied as anti-noise barriers. The light transmission of the glass-fabricated MPPs was measured by initial light tests and the results suggested that two layers of MPPs would be acceptable although the transmission was slightly less than that of a sheet of glass (Kang and Brocklesby, 2005). The transmittance and reflectance of the polycarbonate-fabricated MPPs were tested and only small influences of perforations on the normal incidence transparency were observed, suggesting an adequate optical performance (Asdrubali and Pispola, 2007). These studies also indicate the potential of MPP barriers to be installed at roadways or as windows, and their transparency ensures visibility. In addition, compared to conventional silencers with porous materials, MPP duct silencers with air gaps have the advantage of avoiding bacterial contamination and small particle discharge caused by porous materials. Thus, MPP duct silencers can be widely applied in hospitals, food industries, pharmaceutical industries and micro-electronics industries (Fuchs, 2001; Liu *et al.*, 2014; Yu *et al.*, 2014).

Figure 2.5. Prototype of the transparent sound-absorbing MPP employed for the reverberation room tests. (Asdrubali and Pispola, 2007).

Figure 2.6. (a) Schematic diagram of an MPP silencer (Yu *et al.*, 2014). (b) Rectangular silencer with MPPs.

An MPP system consists of a thin panel with perforations, a back wall, and an air gap. Acoustic resistance is provided by a lattice of perforations with sub-millimeter diameters on the thin panel. The wall is parallel to the thin panel, and the air gap between

the wall and the panel generates an acoustic stiffness dependent on the depth of the air gap. The entire MPP system provides resonant absorption of sound. There are variations on the designs of MPPs such as multi-layer MPPs and MPPs with non-circular perforations, which are developed to enhance the absorption efficiency and broaden the frequency bandwidth of absorption.

Maa (1975, 1987, 1998) developed the theory for acoustic impedance and sound absorption of single-layer MPPs. The theory summarizes that the absorption coefficient and the effectively absorbed range of frequencies are independent of materials but dependent on the geometric dimensions of MPPs. The thickness of the panel t , the perforation diameter d , the perforation ratio p , and the air gap distance D are the geometric variables determining the absorption efficiency at a given frequency. Figure 2.7 shows the schematic configuration of a single-layer MPP and the parameters t , d , and D .

Figure 2.7. Schematic diagram of an MPP consisting of a thin panel, an air gap, and a rigid wall (Yang *et al.*, 2020).

Acoustic waves are small harmonic variations of pressure in a fluid, which are superimposed on a background pressure. The fundamental equation governing this sound wave propagation in a short tube is calculated as (Crandall, 1926; Maa; Strutt, 1945)

$$\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} - \frac{j\omega\rho_0}{\eta} v = \frac{1}{\eta} \frac{\Delta P}{l_t} \quad (2.1)$$

where v is the particle velocity, ρ_0 is the density of air, η is the viscosity of air, ω is the angular frequency, ΔP is the sound pressure difference between the two ends of the tube, and l_t is the length of the tube's unit element. The particle velocity v is obtained analytically and is equal to zero at the tube walls. The total acoustic impedance Z_{total} is defined as the ratio of the sound pressure difference to the average velocity u :

$$Z_{\text{total}} = \frac{\Delta P}{u} \quad (2.2)$$

where the constant c_0 is the speed of sound in air and f is the frequency.

An electro-acoustical equivalent circuit model shown in Figure 2.8 was developed to derive the equations (Qian *et al.*, 2013). The specific acoustic resistance R and the reactance M are induced by the perforations. The voltage of the circuit is an analogy of the acoustic wave with a sound pressure of $2P$ (Qian *et al.*, 2013).

Figure 2.8. Schematic diagram of the equivalent circuit of an MPP (Qian *et al.*, 2013).

The total normalized specific acoustic impedance Z_{total} is calculated as

$$Z_{\text{total}} = Z_M + Z_D, \quad (2.3)$$

where Z_M and Z_D are the acoustic impedance of the panel layer and the air gap, respectively. The acoustic impedance of the air gap is expressed as

$$Z_D = -j \cot \frac{2\pi f D}{c_0}, \quad (2.4)$$

where the constant c_0 is the speed of sound in air and f is the frequency.

The model of a panel with an array of tubes is used to derive Z_M . Rayleigh *et al.* (1945) firstly studied the sound wave propagation in narrow tubes, and the formulas were further simplified by Crandall (1926) for wave propagation in tubes that are extremely short in comparison with the sound wavelength. Assuming that the time function is sinusoidal and the velocity is zero at the walls of the tubes, the acoustic impedance Z_{tube} of the tube can be derived as

$$Z_{\text{tube}} = \frac{\Delta P}{u} = j\rho_0\omega t \left[1 - \frac{2}{q\sqrt{-j}} \frac{J_1(q\sqrt{-j})}{J_0(q\sqrt{-j})} \right]^{-1}, \quad (2.5)$$

where the constant μ is the coefficient of the kinematic viscosity of air. J_0 and J_1 are the Bessel's function of the first kind and zeroth order, and the first kind and first order, respectively. The perforation constant q is calculated as

$$q = \sqrt{\frac{\omega}{\mu}} \frac{d}{2} \quad (2.6)$$

and the angular frequency is calculated as

$$\omega = 2\pi f. \quad (2.7)$$

The approximation formula by Crandall (1926) suggests the situations of narrow tubes ($q < 1$) and wide tubes ($q > 10$) as

$$Z_{\text{tube}} = \frac{4}{3} j\rho_0\omega t + \frac{32\rho_0\mu}{d^2} \quad (q < 1), \quad (2.8)$$

$$Z_{\text{tube}} = j\rho_0\omega t + \frac{4\rho_0\mu}{d} \sqrt{\frac{\omega}{2\mu}} (1+j) \quad (q > 10). \quad (2.9)$$

Maa (1975) developed an approximation solution for perforation diameters in submillimetre that makes $1 < q < 10$ as

$$Z_{\text{tube}} = \frac{32\mu t}{pc_0 d^2} \left(\sqrt{1 + \frac{q^2}{32}} + \frac{\sqrt{2}q}{8} \frac{d}{t} \right) + \frac{j\omega t}{pc_0} \left(1 + \frac{1}{\sqrt{9 + \frac{q^2}{2}}} \right). \quad (2.10)$$

End corrections should be considered due to sound radiation from both ends of the tube and the friction loss of airflow. The end correction of the acoustic mass due to sound

radiation increases the effective length of the tube by $0.85d$, while the end correction due to friction loss adds $2\sqrt{2\mu\omega\rho_0^2}$ to the acoustic resistance if the size of both panels is infinite at the end of the tubes (Qian *et al.*, 2013). The acoustic impedance of the panel normalized by the characteristic impedance of air can be calculated as

$$Z_M = \frac{32\mu t}{pc_0 d^2} \left(\sqrt{1 + \frac{q^2}{32}} + \frac{\sqrt{2}q}{8} \frac{d}{t} \right) + \frac{j\omega t}{pc_0} \left(1 + \frac{1}{\sqrt{9 + \frac{q^2}{2}}} + 0.85 \frac{d}{t} \right). \quad (2.11)$$

The sound absorption coefficient α of an MPP is calculated as

$$\alpha = \frac{4 \operatorname{Re}(Z_{\text{total}})}{[1 + \operatorname{Re}(Z_{\text{total}})]^2 + [\operatorname{Im}(Z_{\text{total}})]^2}, \quad (2.12)$$

where $\operatorname{Re}(Z)$ and $\operatorname{Im}(Z)$ represent the real part and the imaginary part of the impedance, respectively.

Besides the conventional single-layer MPP, Maa (1975, 1987) proposed a double-layer MPP backed with a rigid wall in his early research. Theoretical studies on multi-layer structures were conducted later to broaden the bandwidths of acoustic absorption (Guo *et al.*, 2017; Lee and Kwon, 2004; Ruiz *et al.*, 2011; Sakagami *et al.*, 2010; Zhang and Gu, 1998). The performance of the multi-layer MPP was predicted by Ruiz *et al.* (2011) using an electro-acoustic equivalent circuit model. For the multi-layer structure, Equation (2.12) still works for calculating the sound absorption coefficient.

Maa's theory simplifies multi-layer MPPs as a series-parallel connection system of acoustic elements (MPP and cavity)(Maa, 1975). The acoustic impedance of each series-connection subsection is calculated upon which the total acoustic impedance can be

obtained by applying the parallel-connection rule. The acoustic impedance at the input of the n^{th} series-connection MPP-cavity layer is calculated as

$$Z(n) = Z_M(n) + Z_C(n) , \quad (2.13)$$

where $n = 1, 2, \dots, N-1$ with N being the total number of the layers. $Z_M(n)$ and $Z_C(n)$ are the acoustic impedance of the MPP and cavity of the n^{th} layer, respectively, wherein the subscripts M and C represent the initials of the MPP panel and cavity. Figure 2.9 shows the schematic illustration of an N -layer MPP and the impedances of each layer of the panel and each cavity. The acoustic impedance $Z_C(n)$ of the n^{th} cavity is related to the acoustic impedance $Z(n+1)$ of the $(n+1)^{\text{th}}$ layer, which is calculated as

$$\frac{1}{Z_C(n)} = \frac{1}{Z_{CS}(n)} + \frac{1}{Z(n+1)} , \quad (2.14)$$

where the acoustic impedance of the single cavity of the N^{th} layer $Z_{CS}(N)$ is the same as the impedance Z_D in Equation (2.4). The calculation of the acoustic impedance $Z_M(n)$ of the n^{th} layer of the MPP is identical to the impedance Z_M in Equation (2.11).

Figure 2.9. Schematic illustration of the configuration of an N -layer MPP.

Besides the number of layers of the MPP system, geometric variations on the panel, such as the shape of perforations, can also result in variations on absorption efficiency and frequency range. While conventional perforations of MPPs are circular shapes, irregular shapes of perforation have also been studied for the convenience of manufacturing and the improvements on absorption efficiency and frequency bandwidth. Ning *et al.* (2016) summarized and compared the impedance and absorption formulas of MPPs with slit (Maa, 2000), triangular and square cross-sectional perforations (Stinson and Champoux, 1992). Both the theoretical analysis and finite element (FE) simulations concluded that the slit perforation provided lower impedances and absorption coefficients when the width of the slit was the same as the diameter of the circular perforation and the perforation ratio p was kept the same as that of the circular perforation. Meanwhile, as shown in Figures 2.10 and 2.11, the triangular shape provided the largest acoustic impedance, absorption coefficients and frequency bandwidths, when the circumference of the perforations was kept the same as that of the other perforation shapes, while the effects of different perforation shapes with the same sectional area were not obvious.

Figure 2.10. Comparison of (a) the real part and (b) the imaginary part of the relative acoustic impedance of MPPs with different perforation shapes but the same cross-sectional area; comparison of (c) the real part and (d) the imaginary part of the relative acoustic impedance of MPPs with different perforation shapes but the same circumference (Ning *et al.*, 2016).

Figure 2.11. Comparison of the absorption coefficients of MPPs with different perforation shapes. The red marks represent the half-value of the largest absorption coefficients. The width between the two marks on the same curve indicates the bandwidth of the absorption frequency (Ning *et al.*, 2016).

The acoustic impedance of the triangular-perforated MPP that provides the largest absorption in the widest frequency bandwidth can be calculated as (Stinson and Champoux, 1992)

$$Z_{\text{triangle}} = \frac{j\omega\rho_0 q^2}{q^2 - 3q \coth(q) + 3}, \quad (2.15)$$

where the perforation constant q for the triangular perforation is calculated differently from the circular perforation:

$$q = \frac{\sqrt{3}}{4} \sqrt{\frac{j\omega\rho_0}{\mu}} l_{\text{triangular}}, \quad (2.16)$$

where the variable $l_{\text{triangular}}$ is the side length of an equilateral triangle for the simplicity of calculation. Similar to the approximation of the circular perforation shown in Equation (2.9), the normalized acoustic impedance can be concluded as

$$Z_{\text{triangle}} = \omega \rho_0 t \sqrt{\left(\frac{315 \times 21}{441 q^2 + 4 q^6}\right)^2 + \left(\frac{3 q^3 / \sqrt{2} - 3 q^2}{(5/2 - 3 q / \sqrt{2})^2 + (q^2 - 3 q / \sqrt{2})^2}\right)^2} + i \omega \rho_0 t \sqrt{\left(\frac{315 \times 2}{441 + 4 q^4}\right)^2 + \left(\frac{q^4 - 3 q^3 / \sqrt{2}}{(5/2 - 3 q / \sqrt{2})^2 + (k^2 - 3 q / \sqrt{2})^2}\right)^2} \quad (2.17)$$

The end correction before normalisation is added as (Ingard, 1953)

$$Z_{\text{reac}} = 0.96 \sqrt{\frac{\sqrt{3}}{4}} l_{\text{triangular}}^2 \quad (2.18)$$

The total impedance and the absorption coefficient can be calculated by substituting Equation (2.17) into Equations (2.3) and (2.12).

For conventional manufacturing processes, the panel and perforations are fabricated in two separate steps. The original materials are firstly cut or milled to panels by tools such as the laser cutting machine or computer numerical control machines. The perforations are then fabricated by drilling, etching, micro-punching, etc. The performance of MPPs is material-independent, while metals are mostly selected as the materials for the convenience of cutting, milling and drilling.

With the development of manufacturing methods, studies have introduced a one-step fabrication of the panel and perforations such as infiltration and casting. Infiltration is a method to solidify the mixture of polymeric resin and selected salt grains with the size close to the required perforation size. After the salt grains were dissolved by the water, unevenly distributed perforations were left on the polymeric panel (Kristiansen and

Vigran, 1994). The infiltration method requires time-consuming pre-procedure of salt grains selection and counting of the number of perforations, and the perforation shapes were not in smooth-circular shape. Casting has also been commonly used for MPP manufacturing. The fabrication of the casting mould with a micro-cylinder lattice to produce perforations determines the high manufacturing cost and long processing time. Micro-Electro-Mechanical Systems, X-ray LIGA technology (LIGA is a German acronym meaning lithography, electroplating and moulding in English), UV LIGA technology, laser processing technology and electrical discharge machining, which use radiation or electrical energy to partially melt or cure the original materials for subtractive manufacturing, have been applied in mould fabrication (Qian *et al.*, 2015). This method produces MPPs and perforations simultaneously with high accuracy, while the subtractive manufacturing method results in high cost and a waste of materials, and the processing time is prolonged for the manufacturing of the mould. Recently, AM has been introduced for MPP manufacturing for the first time to fabricate polymeric MPPs with various perforation diameters and perforation ratios (Liu *et al.*, 2016, 2017). AM could manufacture products with high accuracy via a relatively low-cost and fast-speed process, which revealed its great potential in the manufacturing of MPPs and other acoustic structures.

2.2.3. Acoustic absorbers fabricated by additive manufacturing

AM was first introduced to acoustic applications in 2007 by Godbold *et al.* (2007). A novelly designed acoustic absorber (shown in Figure 2.12) based on passive destructive interference (PDI) was fabricated by SLS (Elliott and Nelson, 1993).

Different from the main types of acoustic absorbers, PDI absorbers reduced the noise through acoustic interference created by out-of-phase sound waves with a 180° phase difference from two paths with different lengths (illustrated in Figure 2.13) (Setaki *et al.*, 2014). A reduction of the total sound amplitude was caused by destructive interference. For comparison, a Helmholtz resonator acoustic absorber (Selamet *et al.*, 1997) was fabricated at the same time by SLS. The Helmholtz resonator absorber consisted of a narrow neck attached to a large cavity shown in Figure 2.14. The volume of the cavity and the neck dimension determined the resonant frequency of the resonator. Sound absorption reached a peak when the frequency of an incident sound wave matched the resonant frequency. Both absorbers were designed at geometries that provided the largest absorption coefficient at 300 Hz.

Figure 2.12. SLS-printed PDI absorber and internal structure design (Godbold *et al.*, 2007).

Figure 2.13. Schematic illustration of the principle of PDI absorbers. ΔL is the transmission length difference (Setaki *et al.*, 2014).

Figure 2.14. SLS-printed Helmholtz absorber and internal structure design (Godbold *et al.*, 2007).

The results of the acoustic tests showed that both structures exhibited good absorption performances in the range of low frequencies (100–1000 Hz), and both provided the largest absorption at around 300 Hz. However, the overall absorption coefficient of the novel PDI was much lower than that of the Helmholtz resonator (shown in Figure 2.15). As the structure of a Helmholtz resonator requires a large bulk cavity, PDI absorber without structure restriction is an optimal solution if the absorption performance can be improved. The newly introduced SLS provides the convenience of manufacturing absorbers with varying geometries and configurations for future research on improving the sound absorption performance.

Figure 2.15. Absorption coefficients of the designed PDI absorber and Helmholtz resonator (Godbold *et al.*, 2007).

In 2014, Setaki *et al.* (2014) developed various types of small PDI absorbers with complex internal paths to provide sound absorption in a wide frequency range (800–5000 Hz), as shown in Figure 2.16. SLS was selected as the manufacturing method owing to its economical and fast-processing advantages. The heights of the absorbers varied from 3 to 10 cm and the diameters varied from 2.8 to 9.8 cm. The internal air paths were designed in different geometric configurations and dimensions to create phase differences of sound waves.

Figure 2.16. Customized small PDI absorbers manufactured by SLS (Setaki *et al.*, 2014).

In the measurement range of 800–5000 Hz, all tested absorbers provided one, two or three peaks on absorption coefficient curves at distinct frequencies as shown in Figure 2.17. However, the overall sound absorption coefficients were not high. While the highest absorption coefficients reached above 0.8 or 0.9, the bandwidths of frequency ranges for effective absorption were not broad. The limitation of PDI absorbers was that the absorption performances were hard to predict by existing theories before physical acoustic tests were conducted.

Figure 2.17. Sound absorption coefficients of the eight designs of PDI absorbers (Setaki *et al.*, 2014).

Liu *et al.* (2016), as a pioneer, introduced AM into the manufacturing of MPPs in 2016. Samples shown in Figure 2.18 were manufactured by SLA with VisiJet-SL Material consisted of 50% propylene carbonate and 50% mixed triarylsulfonium salts. The thickness, diameter and perforation diameter of the fabricated MPPs were 10 mm, 29 mm and 0.8 mm, respectively. In the meantime, they studied the effects of perforation angle and air gap distance on absorption coefficients by acoustic absorption tests.

Figure 2.18. Illustration of the designed MPPs at (a) top view and (b) side view. The printed samples with perforation angles (c) $\theta = 0^\circ$ (Sample 3DP-1), (d) $\theta = 15^\circ$ (Sample 3DP-2), (e) $\theta = 30^\circ$ (Sample 3DP-3) and (f) $\theta = 45^\circ$ (Sample 3DP-4) (Liu *et al.*, 2016).

The results (shown in Figure 2.19) concluded that the absorption coefficient decreased as the perforation angle increased, and the micro-perforated panel (MPP) with the perforation angle $\theta = 0^\circ$ provided the highest sound absorption coefficient. For an MPP with the perforation angle, thickness, perforation diameter and perforation ratio kept at constant values, the increase of air gap distance shifted the peak absorption coefficient

towards a lower frequency and increased the value of the peak coefficient (shown in Figure 2.20). This study by Liu *et al.* (2016) revealed the potential of AM-manufactured MPPs that had better absorption performances than PDI absorbers and resonators, while the bandwidths of the absorption frequency and absorption coefficients could be further improved during future research. To design MPPs with high sound absorption coefficients, the perforation angle should be designed as 0° .

Figure 2.19. Comparison of the absorption coefficients of MPPs with different perforation angles (Liu *et al.*, 2016).

Figure 2.20. Comparison of the absorption coefficients of MPPs: (a) 3DP-1, (b) 3DP-2, (c) 3DP-3 and (d) 3DP-4 (Liu *et al.*, 2016).

Liu *et al.* (2017) continued studies on MPPs and introduced Polyjet as another AM system for MPP manufacturing in 2017. The effects of the perforation ratio p and porous materials added between the panel and the rigid wall were studied. Figure 2.21 shows the fabricated samples with different perforation ratios while the thickness and the perforation diameter were kept as constants. Figure 2.22 shows the MPP sample backed by a porous absorbing material. The 5 mm non-woven porous material was mainly made from recycled cotton fibers.

Figure 2.21. Illustration of the designed MPP at (a) top view and (b) side view. The printed samples with perforation ratio (c) $p = 0.90\%$ (Sample MPPA1), (d) $p = 1.59\%$ (Sample MPPA2), (e) $p = 2.63\%$ (Sample MPPA3) and (f) $p = 5.90\%$ (Sample MPPA4) (Liu *et al.*, 2017).

Figure 2.22. MPP backed by a porous absorbing material (Liu *et al.*, 2017).

The results showed that the increase of perforation ratio shifted the peak of the absorption curve to the right, indicating a range of higher absorption frequencies, and porous materials provided a wider bandwidth of absorption frequency compared to that of a conventional air gap (Figure 2.23). The results provided an instruction on the design of the perforation ratio to tune the bandwidth of absorption frequency. With the advantages of freeform fabrication and fast process, AM could be considered as the most promising method to manufacture MPPs.

Figure 2.23. Comparison of the absorption coefficients of MPPs backed by (a) an air gap and (b) porous materials (Liu *et al.*, 2017).

2.2.4. Summary

This section firstly reviewed the fundamentals of sound absorption. The acoustic absorbers according to the principles of acoustic energy absorption have been classified into three major types: porous absorbers, MPPs and resonant absorbers. The principle and crucial parameters for porous absorbers have been discussed. In addition, MPPs have been reviewed in detail as special absorbers. The comprehensive review has presented the

applications, theoretical backgrounds, absorption performances and manufacturing methods of different MPPs. Finally, linked to Section 2.1, AM-manufactured acoustic absorbers including PDI absorbers, Helmholtz absorbers and MPPs have been demonstrated and compared regarding their absorption performances.

2.3. Porous and triply periodic minimal surface structures

Architected cellular structures are cellular solids with voids being arranged in periodicity, for example, naturally honeycombs and synthetic lattices. The structures are kinds of porous materials with the feature of high-strength-low-weight and have been extensively used in various applications, especially as acoustic absorbers. Cellular structures based on TPMS that can only be fabricated by AM have been highly focused due to their multifunctionality and have been proved to be of great potential in acoustic applications.

2.3.1. Porous structures

Porous materials are the composite of solid phases and pores that differentiate them from solid materials. The most important parameter of porous materials is the porosity that significantly affects the mechanical, physical and chemical properties of materials. Other pore-related parameters including pore size, pore shape and specific surface area are also crucial to the properties of porous materials (Liu and Chen, 2014a). As sound absorption materials, porous materials can be classified into three types based on their microscopic configurations: cellular, fibrous and granular (Arenas and Crocker, 2010), as illustrated in Figure 2.24. Fibrous materials such as glass and mineral fibers consist of

tunnel-like openings formed by interstices in material fibers (Kazragis *et al.*, 2002). Granular materials such as asphalt, porous concrete and granular clays consist of rigid macroscopic bodies that largely exceed the sizes of the internal voids to form consolidated granular materials, or loosely packed assemblages of individual particles to form unconsolidated granular materials (Asdrubali and Horoshenkov, 2002; Magrini and Ricciardi, 2000). Cellular structures including both naturally occurring structures (honeycomb, sponge, cancellous bone, etc.) and synthetic structures (polymeric foams, metallic foams, etc.) are a subset of porous materials with low density, large specific surfaces and various special properties in mechanical, physical, electrical, thermal and acoustical fields (Heaney, 2018). Cellular structures with the periodic open-cell arrangement of two-dimensional configuration such as honeycombs or 3D configuration such as strut-based structures are referred to as lattice structures (Al-ketan *et al.*, 2018). These structures are potential replacements of bulk structures owing to their competitive mechanical properties while the weight and materials of the structures are reduced by voids (Al-ketan *et al.*, 2018; Gibson and Ashby, 1999).

Figure 2.24. Comparison of the configurations of cellular, fibrous and granular materials (Arenas and Crocker, 2010).

Studies have been focused on parametric designs of lattice structures for optimal properties. The properties of high stiffness and strength have been achieved by stretch-dominated lattices such as Kagome and octet-truss lattice (shown in Figure 2.25 (a)) as the stiffness and strength scale linearly as a function of the relative density of the structures, which are higher than those of the bending-dominated structures that the stiffness and strength scale quadratically with the relative density (Bendsoe and Sigmund,

1999; Deshpande *et al.*, 2001a; Deshpande *et al.*, 2001b; Ullah *et al.*, 2014). In contrast, bending-dominated lattices such as body-centered cubic (BCC) lattices (shown in Figure 2.25 (b)) and stochastic foams have large toughness and have been studied for energy absorption due to the long stress plateau of the structures under compression (Deshpande *et al.*, 2001a; Sun and Li, 2018). Relative density (volume fraction) is the ratio of the volume of the solid phase to the total volume of a porous material when no other phases such as glass fibers or solid fire-retardant additives are present, and the total volume of the porous material is the sum of the porosity and the volume fraction (Mills, 2007). When the material and the volume fraction of the cellular structures are fixed, topology optimization can be an alternative method to achieve optimal mechanical properties that highly depend on the architecture of the unit cell. For example, in tissue engineering, the optimal designs are the topologies that are able to mimic the nature of bones for bone scaffolds and orthopedic implants (Wang *et al.*, 2016). The recent research of the cell topology of cellular structures has shifted to mathematically defined architectures, and TPMS structures have been highly focused as the geometric characters are appealing for multifunction applications.

Figure 2.25. Cellular topologies of (a) an octet-truss lattice (Al-ketan *et al.*, 2018) and (b) a BCC lattice (Alwattar and Mian, 2019).

2.3.2. Triply periodic minimal surface structures

TPMS are mathematically defined continuous surfaces that exhibit periodicity in three perpendicular dimensions with zero mean curvatures and locally-minimized surface areas (Chen *et al.*, 2009; Lord and Mackay, 2003). These surfaces divide the space into two interlocked sub-domains while voids are maintained (Rajagopalan and Robb, 2006; Wohlgenuth *et al.*, 2001). The types of surfaces are determined by various surface equations describing them. Common types of TPMS include Primitive and Diamond surfaces discovered by Schwarz (1885) and Gyroid, I-WP and O, C-TO surfaces discovered by Schoen (1970). The TPMS-based lattice structures are created by thickening the surface or solidifying the volumes enclosed by the surfaces to create sheet-based or skeletal-based cellular structures respectively (Kapfer *et al.*, 2011). Figure 2.26 shows the samples of I-WP structures created in both methods.

Figure 2.26. Sheet-based and skeletal-based I-WP TPMS structures (Al-ketan *et al.*, 2018).

The unique geometries of TPMS structures have enabled their potential for multifunctional applications. TPMS structures can be employed for structural engineering as they show superior mechanical properties including compressive modulus and compressive strengths when compared to conventional cellular structures (Abueidda *et al.*, 2017), and these mechanical properties can even be customized upon different design requirements (Lee *et al.*, 2016b). Yan *et al.* (2014) concluded that the yield strength and Young's modulus of Gyroid TPMS structures increased as the volume fraction of the structures increased. These structures also exhibit optimal energy absorption abilities. Maskery *et al.* (2017) revealed that the energy absorption by Gyroid TPMS structures was around three times the absorption by common BCC lattices with the same volume fraction and manufacturing material. Zhang *et al.* (2018) extended the studies of energy absorption of TPMS structures to Diamond and Primitive structures and concluded that TPMS structures significantly outperformed BCC lattices with Diamond structures exhibiting the optimum ability. An energy absorption diagram was provided to select the

most efficient density of respective structure types. Besides superior mechanical properties and energy absorption abilities, the structures showed excellent thermal and electrical conductivities (Torquato and Donev, 2004), fluid permeability (Jung and Torquato, 2005) and vibration isolation abilities (Elmadih *et al.*, 2019). Vibration isolation was achieved by Gyroid structures that caused the mechanical bandgaps of elastic waves under 15k Hz, and the bandwidth of a bandgap could be tuned by adjusting the unit cell size and the volume fraction of the structures.

The research on TPMS was restricted to theoretical studies until the introduction of AM. AM has enabled the fabrication of complex structures for multi-discipline applications through a layer-by-layer buildup of pre-designed computer models (An *et al.*, 2015; An *et al.*, 2016; Cheah *et al.*, 2002; Lee *et al.*, 2017a; Lee *et al.*, 2017b; Lee *et al.*, 2016a; Low *et al.*, 2001; Ng *et al.*, 2019; Sing *et al.*, 2015; Yuan *et al.*, 2019a; Yuan *et al.*, 2019b; Zhuang *et al.*, 2018). The advanced AM systems, selective laser melting (SLM) and SLS, have been mostly utilized for the manufacturing of TPMS structures made of metal and polymers, respectively, for investigations on mechanical and multifunctional properties of the structures (Abueidda *et al.*, 2017; Al-ketan *et al.*, 2018; Elmadih *et al.*, 2019; Maskery *et al.*, 2018; Zhang *et al.*, 2018). Figure 2.27 shows the SLS-printed TPMS cubics using PA12 powder for investigations on compressive deformation behavior of Primitive, Gyroid and Diamond structures (Maskery *et al.*, 2018). Based on the results, the authors recommended Primitive structures for applications requiring high stiffness and strength in one well-defined loading direction.

Figure 2.27. Video frames showing the deformation of Gyroid ((a) and (b)), Diamond ((c) and (d)) and Primitive ((e), (f) and (g)) lattice structures at a series of compressive strains. The unit cell size is 10 mm and the cubic size is $40 \times 40 \times 40$ mm consisting of $4 \times 4 \times 4$ unit cells (Maskery *et al.*, 2018).

Figure 2.28 shows the SLM-printed TPMS cubics using gas atomized Maraging steel fine powder for microstructure characterization and mechanical tests (Al-ketan *et al.*, 2018). The authors concluded that the sheet-Diamond TPMS structure exhibited a stiffness behavior that was almost independent of relative density, making the structures

unique among cellular materials. Overall, the sheet-based TPMS structures showed superior mechanical properties.

Figure 2.28. SLM-printed TPMS structures with the unit cell size of 7 mm. A cubic consists of $6 \times 6 \times 6$ unit cells and the cubic size is $42 \times 42 \times 42$ mm (Al-ketan *et al.*, 2018).

In addition to metal and polymers, ceramic was applied as the material of TPMS structures as bone tissue scaffolds printed by FDM (Restrepo *et al.*, 2017). Figure 2.29 shows the samples fabricated by ceramic paste for mechanical characterization. The results revealed the potential of Primitive structures as scaffolds for bone repair with appropriate characteristics. The unit cell sizes of the structures fabricated by SLM, SLS and FDM were 7–20 mm, while SLA fabricated structures in smaller sizes (less than 5

mm in unit size) with high dimension accuracy and good surface finish (Melchels *et al.*, 2010; Yu *et al.*, 2019).

Figure 2.29. FDM-printed ceramic (a) Primitive, (b) Gyroid, and (c) Diamond structures. The unit cell size is 10 mm and the cubic size is 50×50×50 mm consisting of 5×5×5 unit cells (Restrepo *et al.*, 2017).

2.3.3. Porous structures for acoustic application

Resonant absorbers including Helmholtz resonators, PDI resonators and MPPs have been reviewed in Sections 2.2.2 and 2.2.3. As another type of acoustic absorbers, porous absorbers including foams, fibrous materials and lattices have been extensively used for acoustic absorption in the range of high frequencies. The most common materials for noise reduction are inorganic fibers including carbon fiber, glass wool and basalt fiber. Metallic fibers including stainless steel fiber, iron fiber, nickel fiber, aluminum and aluminum alloy fibers have been developed as absorption materials with the advantages of high resistance of temperature, moisture and corrosion for acoustic applications in extreme environments (Tang and Yan, 2017). Natural fibrous materials such as bamboo fiber, tea-leaf fiber, kapok fiber and sheep wool fiber have been increasingly investigated as environmental-friendly acoustic absorbers (Chen *et al.*, 2005; Ersoy and Kucuk, 2009;

Ganesan and Karthik, 2015). Figure 2.30 shows that the kenaf fiber and sheep wool fiber achieved the absorption coefficients that were no less than 0.8 in the range of low frequencies with a wide bandwidth (Berardi and Iannace, 2015). Synthetic fibrous materials have been increasingly applied to replace natural fibrous materials to improve the absorption efficiency because of their high structural diversity such as different cross-section shapes and total surface areas (Kino and Ueno, 2008; Tascan and Vaughn, 2008; Tascan *et al.*, 2011). Recently, composites with natural or synthetic fibrous materials as reinforcement have been studied for the improvement of acoustic absorption ability of the base material while maintaining the advantages of the base material and reinforcement materials (Jiang *et al.*, 2012; Li *et al.*, 2015). For example, Jayamani *et al.* (Jayamani *et al.*, 2016) investigated the zein reinforced oil palm empty fruit bunch fiber composites as a sustainable, renewable and biodegradable acoustic absorption material. Besides, adding nanofibers into porous materials is an alternative method to enhance the absorption efficiency at low frequencies and middle frequencies owing to the effects of air friction inside the nanopores, the vibrations of nanolayers and the interaction between acoustic waves and the large specific surface of nanofibers (Bahrambeygi *et al.*, 2013; Na *et al.*, 2012). Chang *et al.* (2016) proposed a novel technique to create porous materials with 3D nanofibrous networks and observed nontrivial improvements of sound absorption in the ranges from low frequency to middle frequency because of the nanofibrous networks added in the conventional cotton as shown in Figure 2.31. However, the application of nanofibers on acoustic absorption has been limited due to high cost and low efficiency of the manufacturing process.

Figure 2.30. Absorption coefficients of (a) kenaf fiber with different bulk density and thickness and (b) sheep wool with 4 cm and 6 cm thickness (Berardi and Iannace, 2015).

Figure 2.31. 3D nanofibrous networks (scale bar: 2.0 cm) with geometric shapes of (a) sphere, (b) cube, (c) triangular pyramid and (d) triangular prism, and (e) the experimentally measured sound absorption coefficients of 3D nanofibrous networks and the commercial sound-absorbing cotton (Chang *et al.*, 2016).

While traditional cellular structures have been extensively applied as acoustic absorbers and noise control solutions, the studies on acoustic properties of TPMS structures are at the beginning stage. In recent studies, TPMS structures have exhibited the potential in acoustic applications with multifunctionality. Zhang *et al.* (2018) reported that the TPMS sheet-based structures, especially the Diamond structure, showed a superior energy absorption ability compared to BCC lattices, indicating the potential of these structures for acoustic absorption. Elmadih *et al.* (2019) extended the applications of TPMS structures to the prohibition of elastic waves through a periodic medium by destructive interference of reflected waves in certain frequency ranges. Phononic bandgaps were formed at the frequencies of prohibited elastic waves. As shown in Figure 2.32, four bandgaps were achieved in the sub-15000 Hz region by the skeletal-based Gyroid structure with a unit cell size of 15 mm and a volume fraction of 20%. The broadest bandgap was formed from 7905 to 8952 Hz between the 6th and 7th wavebands, while a bandgap of a similar width spanned from 11349 to 12327 Hz between the 9th and 10th wavebands. The starting frequency, the bandwidth and the number of the bandgaps were tunable by varying the unit cell size and volume fraction. The bandgaps were obtained by theoretical and numerical calculations and the designed TPMS structures were fabricated by AM to prove the feasibility of producing the designed geometric parameters of structures.

Figure 2.32. Phonon dispersion curves for a network Gyroid structure. Wavebands are numbered from the 1st to the 11th waveband. The top axis shows the frequency f normalized to the unit lattice cell size L and the speed of the wave v in the material, while the right axis shows the wavenumber normalized to the lattice unit cell size (Elmadih *et al.*, 2019).

Similar to phononic bandgaps, acoustic bandgaps could be achieved by prohibiting the propagation of sound waves at certain frequencies. Abueidda *et al.* (2018) numerically investigated the effects of relative density of three TPMS structures (Primitive, I-WP and Neovius) with the same material and unit cell size on acoustic bandgaps. As shown in Figure 2.33, any incident wave with a frequency in the gray region would be prohibited from transmitting through the respective structures. The Neovius structure achieved two bandgaps in the observed frequency range and prohibited the most sound waves when compared to the other two structures. The widths of the

bandgaps at high frequencies increased as the relative density of the structures increased while the first bandgap at lower frequencies of the Neovius structure exhibited an insignificant decrease. The bandgaps of these TPMS structures were wider than those of the simple cubic lattice, BCC lattice and face-centered cubic lattice that have been studied by Chen *et al.*(2013). To achieve complete bandgaps, the relative densities of TPMS structures were lower than those of conventional lattices while the mechanical integrity of TPMS structures was not comprised as compared with conventional lattices.

Figure 2.33. Complete bandgap width distribution of Primitive, I-WP and Neovius structures as a function of relative density (Abueidda *et al.*, 2018).

2.3.4. Summary

Firstly, the fundamentals of cellular structures have been provided. Furthermore, TPMS structures have been reviewed by their definition, properties, advantages, applications and multifunctionality in detail as the special cellular structures that can only be fabricated by AM (reviewed in Section 2.1). Finally, the existing applications and research on various cellular structures including TPMS structures for acoustic absorption (reviewed in Section 2.2) have been demonstrated.

Chapter 3 3D-Printed Polymeric Micro-Perforated Panels for Tunable Wideband Sound Absorption

Micro-perforated panels (MPPs) are next-generation solutions for noise reduction with the potential of achieving high absorption coefficients for wideband sound absorption in the range from low frequency to middle frequency. In this chapter, multi-layer MPPs with adjustable geometries for tunable wideband absorption in various frequency ranges are designed and developed to enhance the absorption efficiency. The parameters of the structures are designed for controlling noises at around 1000 Hz, which is the frequency range of major traffic noises (Can *et al.*, 2010). Selective laser sintering (SLS) that utilizes polymers as the feedstock is applied for fabrication to reduce the costs of materials and manufacturing time. Furthermore, an optimization method is developed and validated experimentally to evaluate the efficiency of designed structures and to provide optimized structural parameters for future designs.

3.1. Design and fabrication of micro-perforated panels

Multi-layer MPPs for tunable wideband acoustic absorption in the target ranges were designed according to the effects of the geometric parameters of single-layer MPPs that were analyzed by preliminary numerical results based on Equations (2.1)–(2.12). SLS with the significant advantage of excluding the design, printing and removal of supporting structures for manufacturing of multi-layer structures was selected as the suitable additive manufacturing (AM) system to fabricate samples for acoustic absorption tests. The finite element method (FEM) was applied to predict the acoustic absorption performance of the designed structures.

3.1.1. Effects of geometric parameters on absorption coefficients of single-layer micro-perforated panels by theoretical predictions

According to Equations (2.1)–(2.12), based on Maa’s theory, for a single-layer MPP, the absorption coefficient α and the effective frequency f are dependent on four parameters: the thickness of the panel t , the perforation diameter d , the perforation ratio p and the air gap distance D . The schematic illustration of the geometric parameters is shown in Figure 2.7.

A preliminary numerical study on the effects of the parameters t , p , d and D on the acoustic absorption performance of a single-layer MPP according to Maa’s theory was conducted to guide the design of the single-layer MPP and the panels of multi-layer MPPs. Table 3.1 summarizes the values of the parameters t , d , p and D , one of which was set to be the variable in sequence while others were kept constant. The thickness t of the panel as a variable was observed from 1 mm to 5 mm to match the manufacturing resolution while $t = 1$ mm was preferred to reduce the weight and material usage of the structure. The range of the perforation diameter d as the variable was chosen to be 0.5–1.3 mm to provide effective absorption at the target frequency range and allow a reasonable range of the perforation ratio ($\leq 6\%$) (Stinson and Shaw, 1985). The predicted range of the perforation ratio p as a variable was 0.8%–5% to allow a reasonable number of perforations at a diameter of 0.9 mm. The value of the air gap distance D as a variable was observed from 1 cm to 5 cm so that D was large enough for frequency bandwidth differentiation to achieve a wideband or multiband absorption while the smallest value was favorable to minimize the volume of the whole MPP structure for future applications.

Table 3.1. Values of the variables and the respective constant parameters for preliminary numerical predictions of the geometric effects on absorption performances of single-layer MPPs.

Variable	Constant parameters
$t = 1 \text{ mm}, 2 \text{ mm}, 3 \text{ mm}, 4 \text{ mm}$ and 5 mm	$d = 0.9 \text{ mm}, D = 2 \text{ cm},$ $p = 1\%$
$d = 0.5 \text{ mm}, 0.7 \text{ mm}, 0.9 \text{ mm}, 1.1 \text{ mm}$ and 1.3 mm	$t = 1 \text{ mm}, D = 2 \text{ cm},$ $p = 1\%$
$p = 0.8\%, 1.0\%, 2.0\%, 3.0\%, 4.0\%$ and $5.0\%,$	$t = 1 \text{ mm}, D = 2 \text{ cm},$ $d = 0.9 \text{ mm}$
$D = 1 \text{ cm}, 2 \text{ cm}, 3 \text{ cm}, 4 \text{ cm}$ and $5 \text{ cm},$	$t = 1 \text{ mm}, d = 0.9 \text{ mm},$ $p = 1\%$

Figure 3.1 shows the preliminary results of the effects of varying geometric parameters on acoustic absorption of single-layer MPPs. As shown in Figure 3.1(a), as t increases from 1 mm to 5 mm, the peak of the curve shifts to a lower frequency range and the frequency bandwidth of the curve decreases, while the peak value of the absorption coefficient is kept high. Therefore, the thickness t of the panel should be designed as 1 mm to reach a wide frequency bandwidth and to minimize the weight and material usage. Figure 3.1(b) shows a decrease in the value of the absorption coefficient and the frequency bandwidth with a left shift of the peak on the curve as d increases. Therefore, small perforation diameter values are favorable for designed MPPs. However, in order to meet the manufacturing resolution and to reduce material usage, 0.9 mm is the suitable value of d for the designed structures. As shown in Figure 3.1(c), the peak value of the

absorption coefficient decreases significantly as p increases, which can be explained by the positive gradient of the material's flow resistance (Way *et al.*, 2014). The peak value of the absorption coefficient of the $p = 0.8\%$ curve is slightly higher than that of the $p = 1.0\%$ curve while the $p = 1.0\%$ curve has a larger bandwidth. Hence, p is suggested to be designed at a value close to 1.0% to obtain an effective absorption with a high absorption coefficient and a wide frequency bandwidth, subject to the perforation diameter d and the number of perforations. Figure 3.1(d) shows that the peak of the absorption curve shifts to a lower frequency while the bandwidth and the peak value of the absorption coefficient show limited changes as D increases. The effect can be explained by considering the perforation panels as the mass in a spring system, and the fluid air in the gap plays the role of an acoustic spring. The increase of the air gap distance reduces the stiffness of the spring, which causes the peak coefficient as the resonance of the spring and the mass system shift to a lower frequency range (Liu *et al.*, 2017). To achieve wideband and multiband absorption, structures with different combinations of D will be designed, and variations on D will tune the absorption frequency ranges.

Figure 3.1. Preliminary numerical predictions of the effects of (a) thickness t , (b) perforation diameter d , (c) perforation rate p and (d) air gap distance D on acoustic absorption of single-layer MPPs.

3.1.2. Sample preparation and acoustic absorption test

Among various AM systems, SLS was selected due to the significant advantage of excluding supporting structures when manufacturing multi-layer structures. In addition, SLS is able to manufacture the samples with designed sizes in one batch. Manufacturing costs and time can be reduced by using SLS due to the merits. Although the manufacturing accuracy is limited, the expected absorption capability can be achieved and will be discussed in Section 3.2.

Based on the effects of geometric parameters of single-layer MPPs that have been provided by the preliminary numerical results in Section 3.1.1, multi-layer structures with single, double, and triple layers of identical panels were designed for characterizing the sound absorption capability of MPPs. Figure 3.2 shows the geometric configurations of the structures. The geometric dimensions were designed to match the manufacturing resolution of SLS (Chua and Leong, 2017) and to provide tunable acoustic impedances and sound absorption coefficients within the target frequency range of 200–1600 Hz. As the arrangement of the perforations between the adjacent layers of the multi-layer MPPs has no effect on the absorption performance according to Equations (2.11)–(2.14), for the convenience of designing and drawing the structures by Solidworks software, the perforations of each layer were aligned identically. Meanwhile, as the arrangement of the perforations on the panel has no effect on the absorption performance as long as the ratio of the diameter to the distance between the centers of perforations is smaller than 0.2 (Ruiz and Cobo, 2011; Tayong *et al*, 2011), the perforations were aligned on each layer for design convenience, as shown in Figure 3.2.

Figure 3.2. Schematic demonstration of single-layer, double-layer and triple-layer MPPs.

The panel thickness t , perforation diameter d , perforation ratio p , and panel diameter were kept at 1 mm, 0.9 mm, 1.0%, and 100 mm, respectively. The arrangement was made such that the inter-layer distances (heights) h , h_1 and h_2 in Figure 3.2 could vary from 20 mm to 40 mm to quantify the influences of different h on the effective range of f and the absorption coefficient α . The geometric dimensions are shown in Table 3.2. All the structures were manufactured in one batch by an SLS P395 machine (EOS) with a CO₂ laser (wavelength: 10.6 μm , laser power: up to 50W). One of the commercial materials, PA12 (PA2200 from EOS) with advantages of a high melting flow rate, a low melting temperature, a low glass transition temperature and a low degree of crystallization temperature (Way *et al.*, 2014), was used as the feedstock. The mechanical properties and viscosity of laser-sintered PA12 are shown in Table 3.3. The commercial printing parameters for PA12 from the EOS P395 system including the laser scanning speed of 4000 mm/s, the laser power of 40 W, the layer thickness of 0.12 mm, and the hatching space of 0.3 mm were used.

Table 3.2. Geometric parameters of the tested specimens.

Sample number	Number of layers	Inter-layer distance
MPP_1	1	NA
MPP_2	2	$h = 30$ mm
MPP_3	2	$h = 40$ mm
MPP_4	2	$h = 50$ mm
MPP_5	3	$h_1 = 30$ mm, $h_2 = 40$ mm
MPP_6	3	$h_1 = 40$ mm, $h_2 = 40$ mm
MPP_7	3	$h_1 = 50$ mm, $h_2 = 40$ mm

Table 3.3. Mechanical and viscosity properties of PA12 (Yuan, 2018).

Tensile modulus (MPa)	Tensile strength (MPa)	Toughness (J/mm ³)	Elongation at break (%)	Viscosity (Pa·s) at 200 °C
1291 (± 12.1)	44 (± 1.3)	10.18 (± 0.9)	24 (± 0.8)	2616 (± 99)

The typical samples of AM-fabricated MPPs and the experimental setup are shown in Figure 3.3. A two-microphone impedance method using impedance tube type 4206 (Brüel and Kjær Sound and Vibration) was used to measure the sound absorption of the specimens. The impedance tube with an inner diameter of 100 mm measured the frequency range of 50–1600 Hz. The sound absorption coefficients were calculated by the

transfer function method (Chung and Blaser, 1980). The air gap distance D of MPPs could be adjusted to tune the absorption frequency range. Figure 3.4 shows the schematic configurations of the double-layer and triple-layer MPP structures including an air gap in the tube. The total length L was defined as the length of the MPP system including the thickness of the panels, the inter-layer distances, and the air gap distance.

Figure 3.3. AM-fabricated samples of MPPs and setup of the acoustic absorption test.

Figure 3.4. Schematic demonstrations of the designed (a) double-layer MPP and (b) triple-layer MPP in the impedance tube.

3.1.3. Finite element method to model the acoustic absorption of multi-layer micro-perforated panels

Different from the Maa's equivalent circuit model for multi-layer MPPs that simplified the panel-to-panel and panel-to-air gap interactions, 3D FEM-based numerical simulations were carried out to solve Equation (2.1) using the pre-set module Thermoacoustics in COMSOL Multiphysics software for more precise numerical results than those of the Maa's model. The velocity and pressure field in the MPPs were calculated, and the surface impedance was extracted through Equation (2.2).

Given the periodicity of the perforations, only a quarter of the unit cell containing one perforation and the corresponding incidence part of the back cavity was sufficient to model the whole MPP structure. In the finite element (FE) model, only the air gap as the fluid region with symmetrical boundaries and the air region of the quarter of perforation were created as the solid parts, as shown in Figure 3.5. The models of MPP_2 and MPP_3 are shown as examples of double-layer and triple-layer MPPs, respectively. The panels were built as voids between the air gap regions in the models. Sound-hard boundaries were imposed on the interfaces between air and the MPP plane because the fluid-solid coupling effect was ignored. In order to fully capture the influence of viscous boundary layers, the smallest mesh size inside the micro-perforations adjacent to the wall of the perforation was kept smaller than the thickness of the viscous boundary layer at the highest frequency. Moreover, hexahedron elements were used. A mesh sensitivity study indicated that the element sizes could effectively balance the simulation accuracy and efficiency, and thus guarantee the calculation convergence. A front view of the built models that have been meshed is shown in Figure 3.6. The quarter of the perforation at the corner in the model is highlighted in Figures 3.5 and 3.6.

Figure 3.5. Geometric structure of (a) MPP_2 and (b) MPP_6 built for FEM.

Figure 3.6. Front view of a built model with meshes.

The dependent variables of the Thermoacoustic module in COMSOL Multiphysics software are the acoustic pressure, the velocity field, and the acoustic temperature variation. This is a detailed acoustic model solving linearized Navier-Stokes equations for a compressible flow. With a normal incidence plane wave with an amplitude of 1 Pa impinging on the front surface of the tube, the pressure field and the velocity field are determined accordingly. Figures 3.7 and 3.8 show the pressure distribution and the velocity distribution around the first, second and third layer of MPP_6, respectively. The structure was observed from the side with the quarter of perforation to show the obvious variations of pressure and velocity distributions caused by the perforation. Both the pressure field and the velocity field are dependent on frequencies. The figures show the instant distributions at 1000 Hz.

Figure 3.7. Pressure distribution around each layer of MPP_6 at 1000 Hz.

Figure 3.8. Velocity distribution around each layer of MPP_6 at 1000 Hz.

The sound absorption coefficient was calculated by the transfer function method (Standard, 2001). Considering a plane incident sound wave normally entering into the acoustic impedance tube, the specific acoustic impedance Z_i at the front surface of the specimen can be determined by

$$Z_i = \rho_0 c_0 \frac{1+R}{1-R}, \quad (3.1)$$

where R is the reflection coefficient which is obtained by

$$R = \frac{H_{12} - H_i}{H_r - H_{12}} e^{2ik_0 s}, \quad (3.2)$$

where $H_{12} = p_2/p_1$ is the transfer function, $H_i = e^{-ik_0 s}$ and $H_r = e^{ik_0 s}$, k_0 is the wave number, p_1 and p_2 are the simulated sound pressure at the inlet and outlet of each panel, respectively. The sound absorption coefficient α of the MPP can be calculated as

$$\alpha = 1 - |R|^2. \quad (3.3)$$

The absorption coefficient is independent of the positions of the measured points at the inlet and outlet of the panel but only related to the distance s between them. The impedance of each layer of the panel was assumed to be identical so that the influence of a panel on the adjacent panels was ignored. The effects of the pillars between the layers were ignored in the Maa's model and the FE model, as the experimental results (would be discussed in Section 3.2) exhibited a negligible improvement in the absorption efficiency caused by the pillars.

3.2. Effects of structural parameters of micro-perforated panels on acoustic absorption

In this section, the effects of the structural parameters including the number of layers, the air gap distance and the inter-layer distance are compared and analyzed by the results of theoretical predictions, numerical simulations and experiments. The results of the theoretical predictions and the numerical simulations were calculated by Equations (2.11)–(2.14) and FEM simulations, respectively. The acoustic absorption tests were conducted through the impedance tube. Each experiment was repeated for three rounds to record the average values as the presented results.

3.2.1. Effect of the number of layers

Figure 3.9 shows the comparison of the absorption coefficients of MPP_1, MPP_2, and MPP_6. The air gap between the rigid wall and the panels was kept at 20 mm. The frequency range was restricted to 200–1600 Hz so that the noise could be filtered out at low frequencies (below 200 Hz).

The numerical simulation results have a reasonable correlation with the values from the theoretical predictions, which indicate the acceptable prediction capability of the developed FE model. The tendency of the experimental results agrees well with the theoretical and numerical results, while the experimental results exhibit better absorption performances. Both the double-layer and triple-layer structures tune the peaks of the absorption curves to different frequencies so that the frequency bandwidth is broadened with the absorption coefficients kept at 0.8 or above. An increase in the number of layers is able to add the number of the peaks on the absorption coefficient curves and shift the

effective frequency range. The absorption curves of the double-layer MPP and the triple-layer MPP observe two and three peaks, respectively, for the given geometric parameters.

Figure 3.9. Comparison of the absorption coefficients of the single-layer, double-layer, and triple-layer MPPs obtained by theoretical predictions, numerical simulations and experiments.

The experimental results with unpredicted noises and side peaks on the absorption curves show wider frequency bandwidths and higher peak values of the MPPs than the

theoretical and numerical results. The discrepancies are clearer at the second peak on the curves of the double-layer and triple-layer MPPs, respectively. It is observed that the second peak on the curve of the experimental results of the triple-layer MPP has slightly shifted to the higher frequency range as compared to those calculated by the theoretical predictions and FEM simulations. The discrepancies could be attributed to the following reasons:

- (i) The perforations in the printed structures, as shown in Figure 3.10(a), were not perfectly circular;
- (ii) Since the diameters of the perforations were all smaller than 1 mm, they were not consistent and not precisely equal to the designed values of 0.9 mm;
- (iii) As shown in Figure 3.10(b), the rough rims of the panels resulted in air spaces between the panels and the duct wall, which affected the wave flow during the acoustic tests but were not taken into account by the theoretical model or the FE model;
- (iv) The peripheral warpage was caused by the thermal gradient during the SLS printing process, which might have resulted in irregular gaps between the circular panel and the tube wall (Manshoori Yeganeh *et al.*, 2019);
- (v) The pillars between the panels were ignored by the theoretical model and the FE model;
- (vi) The sound speed C_0 in the air and the coefficient of the kinematic viscosity of air μ were assumed to be constant in the theoretical model and FEM simulations but could be varied due to the temperature change in experiments.

These causes of the discrepancies among the theoretical, FEM, and experiment results are also valid in the subsequent studies of tuning the absorption frequency ranges and optimizing the structural designs.

Figure 3.10. The microscope images of (a) two perforations in imperfect circular shapes and (b) a part of the rough rim with grains of a printed panel.

Besides the discussed causes of discrepancies, the panel can have effects on absorption coefficients that have not been considered by the theoretical or FE model. A solid panel without perforations was fabricated by the same material with the same diameter and thickness as MPP_1. The absorption coefficients of the solid panel with the air gap distance at 0–30 mm were tested. As shown in Figure 3.11, the panel exhibits no obvious absorption with no air gap or a 10 mm air gap. When the air gap is 20 mm, a number of irregular peaks with extremely narrow bandwidths are exhibited. When the air gap is 30 mm, the panel exhibits sound absorption at low frequencies while the bandwidths are narrow compared to MPPs. The absorption can be attributed to the polymeric materials as the thin panels made of other materials exhibited limited absorption performances (Arenas and Ugarte, 2016; Song *et al*, 2016).

Figure 3.11. Absorption coefficients of a solid panel with varying air gap distances obtained experiments.

3.2.2. Effect of the air gap distance

To study the effects of the air gap distance, MPP_2 and MPP_6 were considered with air gap distances of 20 mm, 30 mm and 40 mm, and the results are shown in Figure 3.12. The experimental results show similar trends as predicted by the theoretical and numerical results with varying air gap distances. However, the experimental results dictate a wider frequency bandwidth and a higher overall absorption coefficient.

Figure 3.12. Comparison of the absorption coefficients of MPPs with varying air gap distances obtained by theoretical predictions, numerical simulations and experiments.

For both double-layer and triple-layer structures, the increase of the air gap distances results in the left shift of the peaks. Thus, this tunes the effective absorption to a lower frequency range, which is similar to the single-layer structure that has been studied by Maa (1975, 1987, 1998). The width of the frequency range and the peak values, however, do not change significantly. It is interesting to note that, at the lower frequencies, the

absorption coefficients do not change obviously as the air gap distances increase. The effect can be explained by considering the perforation panels and the air column as the mass and an acoustic spring, respectively. The increase of the air gap distance reduces the stiffness of the spring, which causes the resonance of the spring and the mass system to shift to a lower frequency range. For the triple-layer MPP, as the air gap distance increases, the absorption coefficient corresponding to the third peak on the curve increases sharply.

3.2.3. Effect of the inter-layer distance

To study the effects of the inter-layer distance on the absorption coefficients, MPP_2, MPP_3 and MPP_4 were compared under the category of double-layer MPPs while MPP_5, MPP_6 and MPP_7 were compared under the category of triple-layer MPPs, as shown in Figure 3.13. The air gap distances were kept at 20 mm for all the cases under consideration. For both double-layer and triple-layer MPPs, the experimental results have similar trends, as predicted by the Maa's theory and FE simulations when the inter-layer distance is varied, while the absorption performances in the experiments are found to be more effective with higher absorption coefficients in wider frequency ranges.

Figure 3.13. Comparison of the absorption coefficients of MPPs with varying inter-layer distances obtained by theoretical predictions, numerical simulations and experiments.

The results also show that the increase of the inter-layer distance shifts the peaks on the absorption curves to a slightly lower frequency range for both double-layer and triple-layer MPPs. The first peaks on the curves of double-layer and triple-layer MPPs are not

significantly affected by the change of inter-layer distances. However, as the inter-layer distances increase, the values of the second peak on the curve of the double-layer MPPs decrease, while the second peak on the curve of the triple-layer MPPs shifts to a lower frequency with no significant change in the values of the absorption coefficients. The value of the third peak on the curve of the triple-layer MPP drops significantly when the inter-layer distance h_1 increases from 30 mm to 50 mm.

The effects can be explained by the changes of the resonance frequencies of the systems. Each peak on the absorption curve occurs at the resonance frequency (Lee and Kwon, 2004). The first peak is due to the effect of the perforation panel closest to the air gap and rigid wall (Panel 2 of the double-layer MPP and Panel 3 of the triple-layer MPP). The variation of inter-layer distance does not change the closest panel and the air gap, and therefore the first peak is not significantly affected by the inter-layer distance. The additional panels (Panel 1 of the double-layer MPP, Panel 2 and Panel 1 of the triple-layer MPP) provide higher resonant frequencies to the system due to the series-parallel connection according to Equations (2.13) and (2.14) and thereby adding the peaks (the second peak by Panel 1 of the double-layer MPP, the second and third peak by Panel 2 and Panel 1 of the triple-layer MPP, respectively) on the absorption curves at higher frequencies. Similar to the effects of increasing the air gap distance, the increase of the inter-layer distances reduces the stiffness and thus shifts the resonance frequencies to a lower frequency range and consumes less energy with smaller absorption coefficients. As the distance between Panel 1 of the MPPs and the rigid wall is determined by h_1 , the increase of h_1 consequently has more obvious effects on the second peak of the double-layer MPP and the third peak of the triple-layer MPP, respectively. Meanwhile, the

second peak of the triple-layer MPP is not affected significantly as h_2 was kept unchanged.

For installations of multi-layer MPPs, due to the constraints of the installation spaces and requirements, the adjustment of inter-layer distances is more feasible and easier than the change of air gap distances to tune the absorption frequency range and improve the absorption performance within a limited space.

3.3. Parameter design of the micro-perforated panels for optimal acoustic absorption

In Figures 3.9, 3.12 and 3.13, it is clear that the area under the curves depends on both the absorption coefficients and the frequency bandwidth, i.e., larger coefficients and a wider bandwidth give rise to a larger area. Therefore, the optimization of the sound absorption at the target frequency range can be presented by the maximization of the area under the sound absorption curve, which is defined as frequency coverage in this chapter.

The main target frequency range in this chapter was set to be 800–1200 Hz since most of the frequencies of the traffic noises were within this range (Can *et al.*, 2010). Sound absorption at other frequencies could also be reached at the same time as multiple peaks with large absorption coefficients were achieved using multi-layer MPPs.

Space constraints should also be considered in designing MPP structures for their installations on buildings, in rooms or small devices including the ducts. The total length L of an MPP system is determined by construction requirements. At a certain L , the optimized geometric parameters of multi-layer MPPs to achieve the largest frequency coverage at the target frequency range can be predicted according to the Maa's theory,

whose results agree well with the FEM results and indicate the same tendency as experimental results while the experimental results are expected to exhibit better absorption performances. In this study, L varied from 10 mm to 200 mm with an interval of 10 mm. The panel thickness t , perforation diameter d , and perforation ratio p of the panels were kept as 1 mm, 0.9 mm, and 1.0%, respectively.

The frequency coverage of the single-layer and double-layer MPPs with L varying from 10 mm to 200 mm is shown in Figure 3.14. The air gap distance of the single-layer MPP is fixed as L ($t = 1$ mm is ignored) and the largest value of the frequency coverage is reached at $L = 20$ mm. For the double-layer MPP, the sound frequency coverage at each L changes as the inter-layer distance h changes from 0 to L , and the peak value is reached at a certain value of h . The largest peak value of the frequency coverage of the double-layer MPP is reached at $L = 10$ mm and $h = 1$ mm, and the second-largest frequency coverage is reached at $L = 20$ mm and $h = 16.2$ mm.

Similarly, the frequency coverage with a peak value for the triple-layer MPPs at each L is calculated. Among all the peak values of the frequency coverage at different L , the largest peak value is reached at $L = 190$ mm. The frequency coverage of the triple-layer MPPs with varying inter-layer distance h_1 and air gap distance at $L = 190$ mm is shown in Figure 3.15. The largest frequency coverage is achieved at $h_1 = 28.5$ mm and $D = 19$ mm.

Figure 3.14. Frequency coverage of the single-layer and double-layer MPPs at each L varying from 10 mm to 200 mm.

Figure 3.15. Frequency coverage of the triple-layer MPPs at $L = 190$ mm.

The peak values of the frequency coverage of the double-layer and triple-layer MPPs are calculated and compared with the single-layer MPP at each L in Figure 3.16. The largest frequency coverage achieved by a single-layer MPP is found to be 282.9 at $L = 20$ mm. For the double-layer MPP, the largest frequency coverage is observed with $L = 10$ mm and $h = 1$ mm (shown in Figure 3.14), which is not feasible to manufacture and install in the real-life application. Thus, the optimized double-layer MPP is designed at $L = 20$ mm and $h = 16.2$ mm when the maximum frequency coverage is at the second-highest value of 285.3 (shown in Figure 3.14). For the triple-layer MPP with $L = 190$ mm, $h_1 = 28.5$ mm and $D = 19$ mm, the frequency coverage can reach up to 311.9 (shown in Figure 3.15), which is the largest among the single-layer and multi-layer structures with L varying from 10 mm to 200 mm.

Figure 3.16. The maximum frequency coverage of the single-layer, double-layer and triple-layer MPPs at a different L .

The parameters for the maximum frequency coverage of the double-layer MPP were found to be $h = 16.2$ mm and $D = 3.8$ mm, while the parameters for the maximum frequency coverage of the triple-layer MPP were $h_1 = 28.5$ mm, $h_2 = 142.5$ mm, and $D = 19$ mm. With the designed parameters, the sound absorption of the structures for the frequency range of 800–1200 Hz was optimized by the maximization of frequency coverage, which was the area under the absorption curve. Figure 3.17 shows the absorption coefficients of the optimized structures by the Maa's theory, FE simulations, and experiments. The frequency coverage of the optimized double-layer MPP by the experiment is 308.9, which is 8.3% larger than the theoretical prediction. The frequency coverage of the optimized triple-layer MPP by the experiment is 319.3, which is 2.4% larger than the theoretical prediction. In this case, the absorption curve of the double-

layer MPP only has one peak, while the absorption curve of the triple-layer MPP has three peaks with large absorption coefficients that indicate effective acoustic absorption at multiple frequency ranges by the selected parameters.

Figure 3.17. Absorption coefficients of the optimized (a) double-layer MPP and (b) triple-layer MPP obtained by theoretical predictions, numerical simulations and experiments.

The frequency coverage can be maximized by designing geometric parameters of MPPs to achieve highly effective sound absorption in different target frequency ranges

according to application requirements. The optimization method by maximizing frequency coverage can also be applied in a wider range of wave damping devices for effective wave selection and absorption.

3.4. Summary

Innovative multi-layer MPPs have been designed and fabricated by SLS for tunable wideband acoustic absorption in the range from low frequency to middle frequency. The effects of structural parameters of multi-layer MPPs have been analyzed by theoretical predictions, numerical simulations and experiments. The developed FE model has been proved to be feasible for the numerical simulation of multi-layer acoustic structures. Furthermore, the frequency coverage, which is the area under the sound absorption curve, has been introduced to design acoustic structures with the most effective sound absorption in target frequency ranges. The evaluation method was validated experimentally by the designed multi-layer structures manufactured by SLS.

Chapter 4 3D-Printed Polymeric Triply Periodic Minimal Surface Structures with Multifunctionality for Wideband Sound Absorption

Porous materials including foams, fibrous materials and lattices have been extensively used for acoustic absorption in the range of high frequencies. However, as multifunctional porous structures that exhibit superior properties, triply periodic minimal surface (TPMS) structures have not been studied as sound absorbers while their potential in wideband acoustic absorption has been indicated by existing literature. In this chapter, three typical types of TPMS structures are designed and fabricated by stereolithography apparatus (SLA) for acoustic absorption tests in the range of high frequencies. The feasibility of TPMS structures as sound absorbers is studied for the first time. Furthermore, the effects of parameters including the unit cell size, the volume fraction and the sample height on absorption efficiency are observed and analyzed to design geometric parameters for optimal absorption.

4.1. Design and fabrication of triply periodic minimal surface structures

Three types of surfaces, Primitive (P), Gyroid (G) and Diamond (D) were employed to create cellular structures for acoustic absorption. The three structures can be generated by the following approximation equations (Wohlgemuth *et al.*, 2001):

$$\Phi_P = \cos(\omega x) + \cos(\omega y) + \cos(\omega z) - c, \quad (4.1)$$

$$\Phi_G = \cos(\omega x)\sin(\omega y) + \cos(\omega y)\sin(\omega z) + \cos(\omega z)\sin(\omega x) - c, \quad (4.2)$$

$$\Phi_D = \sin(\omega x)\sin(\omega y)\sin(\omega z) + \sin(\omega x)\cos(\omega y)\cos(\omega z) + \cos(\omega x)\cos(\omega y)\sin(\omega z) - c, \quad (4.3)$$

where x, y, z are the spatial coordinates and ω is the function periodicity given as

$$\omega = \frac{2\pi}{l}, \quad (4.4)$$

where l is the length of the unit cell. The arbitrary parameter c is the controller of the volume fraction (the volume portion of the rigid parts of TPMS structures) of the lattice cell. The regions are defined to be solid when $\Phi \leq 0$ and to be void when $\Phi > 0$. The isosurface is defined at $\Phi = 0$ to separate the solid and void regions and the triangular mesh can be exported as a STereoLithography file to be processed by additive manufacturing. After being generated by Matlab software, the structure surfaces were converted to solid bodies in Blender software. Materialise Magics software was then used to offset and repair the structures. Figure 4.1 shows the unit cell and the designed cylindrical samples of P, G and D structures for acoustic tests.

Figure 4.1. Unit cell and the designed samples (top view) of P, G and D structures.

Air permeability and tortuosity are two of the important parameters that affect the sound absorption abilities of porous materials. Air permeability is the rate of airflow passing perpendicularly through an area under an air pressure differential between the top and bottom surfaces of a material (Perlmutter, 2015). The decrease of the air permeability reduces the sound waves entering the material and thereby improves sound absorption. Although no existing literature has measured or computed the air permeabilities of TPMS structures yet, Jung *et al.* (2005) have calculated the permeabilities that measure the rates of liquid fluid passing through the material of six TPMS structures including P, G and D structures. A large fluid permeability indicates a large air permeability due to the similar theory. Tortuosity is the structure shape factor that describes the degree of the twist of structures as well as the paths of air flow passing through the structures. A larger tortuosity can increase the length of the path and thus lead to the viscous losses of the sound wave, thereby achieving larger acoustic absorption (Newell, 2017). The tortuosities of P, G and D surfaces have been obtained by Nawada (2018) using Lattice-Boltzmann simulations and are listed in Table 4.1 with those of the fluid permeability computed by Jung *et al.* (2005). To compare the absorption abilities of the three surface types, four combinations of geometric parameters were designed. Table 4.2 lists the geometric parameters for each group of P, G and D structures. The number of cells varies to achieve different h at different l .

Table 4.1. Tortuosity and fluid permeability of P, G and D structures at $\Phi = 0$ and $\rho = 50\%$ (Jung and Torquato, 2005; Nawada, 2018).

	P	G	D
Tortuosity	1.03	1.26	1.29
Permeability	3.4765×10^{-3}	2.2889×10^{-3}	1.4397×10^{-3}

Table 4.2. Designed geometric parameters of four groups of P, G and D structures.

Group	Geometric parameters
Gr ₁	$\rho = 15\%$, $l = 5.6$ mm, $h = 11.2$ mm
Gr ₂	$\rho = 35\%$, $l = 5.6$ mm, $h = 11.2$ mm
Gr ₃	$\rho = 25\%$, $l = 7.0$ mm, $h = 11.2$ mm
Gr ₄	$\rho = 25\%$, $l = 5.6$ mm, $h = 16.8$ mm

Besides the surface types of the structures, geometric parameters of porous structures affect sound absorption by their effects on acoustic resistance, air permeability and tortuosity. The decrease of unit cell size and the increase of volume fraction decrease the pore size and result in the increase of the airflow resistance (Ingard, 1994), the

reduction of the permeability (Mishra and Militky, 2018) and the increase of the tortuosity, and consequently enhance the sound absorption ability (Newell, 2017). Liu *et al.* (2015) proved the positive effects of a higher volume fraction, sample thickness (height), tortuosity and airflow resistance on various porous materials for sound absorption by acoustic absorption tests, while the effects were varied at different target frequencies.

To study the effects of the geometric parameters of TPMS structures on the acoustic absorption coefficients and the effective ranges of frequency, D structures with the smallest permeability computed by Jung *et al.* (2005) and the largest tortuosity calculated by Nawada (2018) were selected to study the effects of the volume fraction ρ , the unit cell size l and the height h . The designed structures named as S₁–S₉ and their geometric parameters are listed in Table 4.3 as well as the grouping of samples to investigate the effects of different variables.

Table 4.3. Geometric parameters and grouping of the designed samples for the experiments on the effects of the various variables.

Sample	ρ (%)	l (mm)	h (mm)
S ₁	15	5.6	11.2
S ₂	25	5.6	11.2
S ₃	35	5.6	11.2
S ₄	45	5.6	11.2
S ₅	25	4.7	11.2
S ₆	25	7.0	11.2
S ₇	25	9.3	11.2
S ₈	25	5.6	5.6
S ₉	25	5.6	16.8
Group	Variable	Samples in the group	
Gr ₅	ρ	S ₁ , S ₂ , S ₃ , S ₄	
Gr ₆	l	S ₂ , S ₅ , S ₆ , S ₇	
Gr ₇	h	S ₂ , S ₈ , S ₉	

The samples were designed to be cylinders with a diameter of 29 mm and a wall thickness of 1 mm to fit the dimension of the experiment apparatus. The values of the unit cell size l were determined by the largest cell number of the top surface of the cylindrical samples. As the cell number decreased from 6 to 3, l increased from 4.7 mm to 9.3 mm. Similarly, the values of the height h were designed by multiplying the cell number by the unit cell size. As the number of the cells increased from 1 to 3, h increased from 5.6 mm to 16.8 mm when the unit cell size l was fixed at 5.6 mm.

Due to the small sizes of the designed structures with the extremely high complexity of TPMS structures, current powder-based and solid-based AM systems were not able to fabricate the designed samples. SLA, a liquid-based vat photopolymerization technology with the advantages of high accuracy, good surface finish and low cost (Chua and Leong, 2017; Lee *et al.*, 2018; Szymczyk *et al.*, 2020), was used to fabricate the small-size samples by photopolymerization of the polymer resin. The structural parameters were designed for sound absorption in the range of 2000–6400 Hz. The SLA printer Form 2 (Formlabs) with a laser power of 250 mW and a building speed of 6 mL/h along the z-axis was used and the printing material was a commercial resin (Tough FLTOTL04, Formlabs). The structures were cleaned by isopropyl alcohol and cured by a 405 nm UV light. Figure 4.2 shows the printed samples based on P, G and D structures.

Figure 4.2. Printed samples of P, G and D structures.

The two-microphone impedance method with impedance tube type 4206 (Brüel and Kjær Sound and Vibration) was used to measure the sound absorption. In this chapter, the impedance tube with an inner diameter of 29 mm (shown in Figure 4.3) was used to measure the frequency range of 2000–6400 Hz. Each experiment was repeated for three rounds to record the average values as the presented results. The sound absorption coefficients were calculated by the transfer function method (Chung and Blaser, 1980; Standard, 2001) with Equations (3.1)–(3.3).

The mechanical performances of the printed structures were not tested here as the focus was on the acoustic properties. There is extensive literature on the mechanical properties and the structure-property requirements of photopolymer-based cellular structures printed by SLA (Abueidda *et al.*, 2017; Dikshit *et al.*, 2017, 2018; Mondschein *et al.*, 2017; Yan *et al.*, 2014).

Figure 4.3. Experiment setup of the acoustic absorption test.

4.2. Effects of surface type and structural parameters of triply periodic minimal surface structures on acoustic absorption

In this section, the effects of the surface type, volume fraction, unit cell size and height of TPMS structures on acoustic absorption are compared and analyzed.

4.2.1. Effect of the surface type

Figure 4.4 shows the comparison of the acoustic absorption obtained from the three types of TPMS structures of Groups Gr₁, Gr₂, Gr₃ and Gr₄. Obviously, the D structures exhibit higher absorption coefficients than the G and P structures with the same parameters l , h and ρ in the target frequency range. The D structures hold the optimal absorption performance among the three types of structures despite the changes of the designed parameters l , h and ρ , while the G and P structures show similar performances with the G structures being able to achieve slightly higher absorption coefficients. The peak values of the absorption coefficients and the respective frequencies at the peak values are affected by the changes of l , h and ρ . The results prove that a larger tortuosity and a lower permeability will provide larger absorption coefficients. It can be concluded that type D with the largest tortuosity and the lowest permeability is optimal among the three candidate surface types for acoustic absorption. In the subsequent sections, to study the effects of parameters l , h and ρ , all the structures were designed in type D for highly effective acoustic absorption.

Figure 4.4. Comparison of the absorption coefficients of the designed structures from groups (a) Gr₁, (b) Gr₂, (c) Gr₃, and (b) Gr₄.

4.2.3. Effect of the volume fraction

Figure 4.5 shows the sound absorption coefficients of samples S₁, S₂, S₃ and S₄ from Group Gr₅, which are all D structures with the increasing values of ρ . The height h was kept at 11.2 mm and the unit cell size l was kept at 5.6 mm. The results show that the overall absorption coefficients increase as ρ increases from 15% to 45% while the effective absorption frequency ranges are kept at around 4000–6400 Hz. The significant enhancement of the peak value of the absorption coefficients can be observed by comparing the curves of $\rho = 15\%$ and $\rho = 45\%$ with similar absorption bandwidths. A

similar tendency can be observed from Figures 4.4(a) and (b) that both G and P structures exhibit higher absorption coefficients with a larger ρ . The increase of the volume fraction ρ has positive effects on the absorption coefficients because the airflow resistance and tortuosity are increased and the permeability is reduced at the same time. However, for the D structures that have the most complicated structural configurations among the three types, although the increase of the volume fraction can benefit acoustic absorption, the structures with ρ being larger than 50% are difficult to realize because the void parts of the structures are blocked by the materials of the solid parts. In addition, the reduction of the weight and manufacturing materials of acoustic absorbers is preferable in industrial and architectural applications. The structures that can achieve the target absorption at acceptable absorption coefficients with the lightest weight (smallest ρ) will be the ideal solution in multifunctional applications.

Figure 4.5. Comparison of the absorption coefficients of the D structures with various ρ .

4.2.3. Effect of the unit cell size

Figure 4.6 shows the sound absorption coefficients of samples S_2 , S_5 , S_6 and S_7 from Group Gr₆. They are all D structures with increasing l . The height h was kept at 11.2 mm and the volume fraction ρ was kept at 25% for these structures. It is obvious that as l increases, the absorption coefficients decrease while the effective frequency range of 4000–6400 Hz does not change. It can be explained by the decrease of airflow resistance and the increase of permeability due to the increase of l . However, when the largest cell number at the top surface increased to 7 and l decreased to 4 mm, the fabrication of the D structures failed due to the restricted spot size of the laser beam of SLA. For applications of D structures as acoustic absorbers, the designed l is suggested to be larger than 4 mm, or an advanced printer with a smaller laser spot size should be developed and applied for the manufacturing.

Figure 4.6. Comparison of the absorption coefficients of the D structures with various l .

4.2.4. Effect of the height

Figure 4.7 shows the sound absorption coefficients of samples S_2 , S_8 and S_9 from group Gr_7 , which are D structures with the increasing values of h . The volume fraction ρ was kept at 25% and the unit cell size l was kept at 5.6 mm. It can be observed that when h is 5.6 mm, the acoustic absorption is poor because the overall absorption coefficients are low. When h is 11.2 mm, the effective absorption frequency range is around 4000–6400 Hz and the absorption coefficients can achieve high values larger than 0.8. When h increases to 16.8 mm, the peak of the absorption curve shifts to the left, causing the effective absorption frequency range to shift to 3000–5000 Hz and the peak value of the absorption coefficients to decrease to 0.74. Figure 4.4(d) reflects similar shifts of the effective frequency ranges of the G and P structures with large h while the overall absorption coefficients are lower when compared with Figure 4.4(c). Thus, the effective absorption frequency ranges can be tuned by adjusting the height of the absorbers when applying the TPMS structures in acoustic absorption.

Figure 4.7. Comparison of the absorption coefficients of the D structures with different h .

However, the wall of the cylindrical samples for the closest contact between the testing tube and the structures can change the propagation routes of the sound wave in the cellular structures, and thus the height of the wall, which equals to the height h of the structure, is preferable to be small to reduce the effects of the wall for the testing samples with the restricted size. The designed structure sizes depend on the requirements of the applications, and the suitable h for the acoustic absorption in the target frequency ranges is varied for different structure sizes. Furthermore, the height should be minimized to reduce the space and weight of the structures for industrial and architectural applications.

4.3. Parameter design of the triply periodic minimal surface structures for optimal acoustic absorption

In Section 4.2, the effects of the surface type and the geometric parameters of TPMS structures on acoustic absorption have been analyzed by experimental results. It can be concluded that D structures with a large volume fraction and a small unit cell size were able to achieve optimal sound absorption in the target frequency range. A D structure S_{op} with a volume fraction of 45%, a unit cell size of 4.7 mm and a height of 12.0 mm was designed according to the analysis of the effects of geometric parameters considering the manufacturing feasibility. Figure 4.8 shows the absorption coefficients of S_{op} in comparison with S_4 ($\rho = 45\%$, $l = 5.6$ mm, $h = 11.2$ mm) and S_5 ($\rho = 25\%$, $l = 4.7$ mm, $h = 11.2$ mm), which exhibits the best absorption in Gr_5 (ρ as the variable) and Gr_6 (l as the variable), respectively. It can be observed that S_{op} achieves the largest frequency coverage (defined as the area under the absorption curve in Chapter 3) among the three structures while the respective frequency of the peak absorption coefficient has shifted

from over 4000 Hz to a lower frequency of 3800 Hz, which may be contributed by the small increase of the sample height from 11.2 mm to 12.0 mm and the interference of the effects of the maximized ρ and minimized l . The shift of the peak of the absorption curve does not change the effective absorption frequency range. The frequency coverage has been increased by 9% and 11% when compared with S_4 and S_5 , respectively, indicating the improvement on acoustic absorption by the selected parameters of the TPMS structure. As the acoustic absorption performance of S_4 and S_5 is closed to S_{op} , for the reduction of manufacturing time and material, it is suggested to design structures with either a high volume fraction or a small unit cell size that can be fabricated easily by conventional 3D printing techniques to achieve highly effective absorption while small changes of the structure height can be applied to tune the effective frequency range.

Figure 4.8. Comparison of the absorption coefficients of samples S_4 , S_5 and S_{op} .

4.4. Summary

Three types of TPMS structures have been designed and fabricated by SLA for acoustic absorption in the range of 4000–6400 Hz. When designed and manufactured with the same unit cell size, volume fraction and height, respectively, D structures exhibited superior absorption abilities in terms of high absorption coefficients and wide bandwidths when compared to P and G structures. A series of D structures have been designed and fabricated for acoustic absorption tests to study the effects of the unit cell size, volume fraction and height. The analysis of the results suggested the design of geometric parameters of a D structure for optimal acoustic absorption. The absorption coefficients of the designed structure have been tested and compared with those of the original D structures and the optimized structure has achieved the largest frequency coverage within the target frequency range.

Chapter 5 3D-Printed Polymeric Triply Periodic Minimal Surface Structures with Micro-Perforations for Sound Absorption Enhancement

Among the triply periodic minimal surface (TPMS) structures reported in Chapter 4, Diamond (D) structures have exhibited superior sound absorption compared to Primitive (P) and Gyroid (G) structures and are able to achieve optimal acoustic absorption with designed geometric parameters. As P and G structures have unique properties as the advantages, they are potential multi-functional sound absorbers for applications different from D structures if their acoustic absorption can be improved by novel solutions. In this chapter, micro-perforated TPMS structures are designed and developed for large enhancement of sound absorption of P and G structures and improvement of D structures without adjusting the geometric parameters. The perforations are produced at the same time with the samples by stereolithography apparatus (SLA). Before that, the effect of materials of TPMS structures is studied to select a proper material for fabrication of the micro-perforated TPMS structures.

5.1. Effect of the material of triply periodic minimal surface structures on acoustic absorption

For conventional porous absorbers, most of the parameters including porosity, airflow resistance, permeability and tortuosity that characterize their sound absorption abilities are intrinsic properties that are dependent on materials. However, these parameters are dependent on surface type and geometries but independent of materials for TPMS structures. Therefore, compared to geometric features, the fabrication materials

are assumed to have limited effects on acoustic absorption of TPMS structures, which is similar to those of micro-perforated panels (MPPs) and resonant absorbers such as Helmholtz resonators and passive destructive interference resonators.

To validate the assumption, TPMS structures were designed and manufactured with different materials for acoustic absorption tests. The SLA printer Form2 (Formlabs) with a laser power of 250 mW and a building speed of 6 mL/h along the z-axis was used to print the P and G structures with the unit cell size of 5.6 mm, the height of 11.2 mm and the volume fraction of 25% using three types of resins: Clear (FLGPCL02, Formlabs), Rigid (FLRGWH01, Formlabs) and Tough (FLTOTL04, Formlabs). According to the product information of Formlabs, Clear resin is widely applied for concept models and prototypes as the standard resin especially for parts requiring translucency. Rigid resin is reinforced with glass for high stiffness and a polished finish and therefore is highly resistant to deformation, and is suitable to print thin walls and features. Rigid resin is more brittle and less impact-resistant than Tough resin that is suitable for prototyping of assemblies, machining and snap-fits undergoing high stress and strain. The printed structures (shown in Figure 5.1) were cleaned by isopropyl alcohol and were cured by a 405 nm ultraviolet radiation light. The two-microphone impedance method with impedance tube type 4206 (Brüel and Kjær Sound and Vibration) was used to measure sound absorption at the frequency range of 500–6400 Hz. The sound absorption coefficients were calculated by the transfer function method (Chung and Blaser, 1980; Standard, 2001) with Equations (3.1)–(3.3). The experimental results are shown in Figure 5.2.

Figure 5.1. P and G structures printed by Tough, Clear and Rigid resins, respectively.

Figure 5.2. Comparison of the absorption coefficients of (a) P structures and (b) G structures with the same geometric parameters printed by Clear, Rigid and Tough resin, respectively.

It can be observed that the absorption curves of each material are almost overlapped for both P and G structures, respectively. Frequency coverage, which has been defined as

the area under the sound absorption curve in Chapter 3, is calculated to measure the absorption ability of the structures. The variations of the frequency coverage (P Clear: 1670; P Rigid: 1564; P Tough: 1628; G Clear: 2235; G Rigid: 2260; G Tough: 2124) are from 1% to 6%, which may be caused by the difference of contact between the sample wall and the wall of the impedance tube and are ignorable. As sound absorption of TPMS structures has not been obviously affected by the selected three types of materials, Clear resin with the lowest cost among the three types of commercial resins was utilized for the fabrication of the samples for the following experiments in this chapter.

5.2. Effect of adding perforations to triply periodic minimal surface structures on acoustic absorption

In Chapter 4, D structures have achieved superior acoustic absorption while P and G structures exhibited poorer absorption abilities. Considering the advantages of high specific surface area and low relative density with higher specific strengths of G structures and maximum stiffness of P structures over other TPMS types (Guest and Prévost, 2006; Li *et al.*, 2018), novel solutions to improve their acoustic absorption for multifunctional applications are necessary to be investigated.

To reduce the structure weight and material consumption, adding perforations to porous materials to improve their sound absorption has been proved to be highly efficient by introducing double porosity materials, in which two connected networks of pores of largely different characteristic sizes can be identified (Olny and Boutin, 2003). As porous materials are hardly used as the printing materials for small-scale and highly-accurate 3D printing, TPMS structures are difficult to achieve the conventional double porosity with

micro-pores in the porous materials and meso-pores added to the materials, but the positive effect of double porosity indicates the potential of adding micro-perforations to TPMS structures in improving acoustic absorption. The potential is additionally indicated by micro-perforated sandwich structures whose sound absorption has been dramatically improved by the perforations (Meng *et al.*, 2017; Wang *et al.*, 2020). Micro-perforated sandwich structures have been investigated as a set of MPPs connected in parallel and series and arranged in corrugated or face-centered cubic sandwich structures. Similar to micro-perforated sandwich structures, micro-perforated TPMS structures combining MPPs and TPMS structures were designed and developed for sound absorption enhancement in this chapter.

Four samples that have been designed and tested in Chapter 4 were selected for adding perforations. The types and geometric parameters of the renamed samples are listed in Table 5.1. The selected samples have exhibited relatively good acoustic absorption among P, G and D structures, respectively. As shown in Figure 5.3, several micro-perforations with a diameter of 0.9 mm were added at random positions of each sample while keeping the distances between the perforation centers larger than 0.18 mm. The random arrangement and the distances between the centers were designed according to the limited effects of the positions of perforations on MPPs when the ratio of perforation diameter to the distance of perforation center is smaller than 0.2 (Ruiz and Cobo, 2011; Tayong *et al.*, 2011). The diameter was designed the same as the micro-perforations of MPPs in Chapter 3 for consistency and due to the same reasons discussed in Chapter 3. Figure 5.4 shows part of the printed samples with perforations. The

perforations are not perfectly circular and the pores have defects due to the curved surfaces of the TPMS structures and manufacturing limitations.

Table 5.1. Surface types and geometric parameters of the unit cell size, volume fraction and height of the selected TPMS structures.

Sample	Type	ρ (%)	l (mm)	h (mm)
S _P	P	35	5.6	11.2
S _G	G	35	5.6	11.2
S _{D1}	D	25	4.7	11.2
S _{D2}	D	45	5.6	11.2

Figure 5.3. Top view of the designed micro-perforated TPMS samples. For the G structure, the newly added perforations are differentiated from the pores of the structure by red dash lines.

Figure 5.4. Microscopy images of the printed samples P (upside) and G (downside) with the designed perforations circled by red dash lines.

Figure 5.5 shows the absorption coefficients of the perforated samples compared to the respective samples without perforations. For samples S_P , S_G and S_{D1} , the perforations obviously increased the absorption coefficients in the target frequency range. For sample S_{D2} , as the original structure has achieved high absorption coefficients with a wide frequency bandwidth, the effect of adding perforations is difficult to be observed directly but can be reflected by the increase of the frequency coverage from 2774 to 3008. As the original D structures can achieve effective sound absorption while the original P and G structures have poorer absorption performances and have the potential to be largely

improved by adding perforations, the P and G structures are used by the following studies on the effect of perforation ratio. In addition, the lower complexity of the P and G structures than the D structures reduces manufacturing time and costs of the perforated samples.

Figure 5.5. Comparison of the absorption coefficients of the normal and perforated (a) sample S_P , (b) sample S_G , (c) sample S_{D1} and (d) sample S_{D2} .

To study the effect of the perforation ratio p of TPMS structures on sound absorption, different numbers of perforations with a diameter of 0.9 mm were added to the selected P and G structures ($\rho = 35\%$, $l = 5.6$ mm, $h = 11.2$ mm). The number of perforations and

the respective perforation ratios of the renamed samples are listed in Table 5.2. The perforation ratio p was calculated by considering the samples as MPPs with a panel diameter of 28 mm, though p was not the true ratio of the perforated area to the surface area of a TPMS structure as the surfaces are curved and porous. Unlike the panels of MPPs that have no pores but only the designed perforations, TPMS structures contain pores that result in limited surface areas for the additional perforations, and thus 10–30 perforations with the designed diameter of 0.9 mm are reasonable to be added without destroying the surfaces of the structures.

Table 5.2. Perforation number and the respective perforation ratio p of the P and G samples.

Type P	P ₀	P ₁	P ₂	P ₃
Type G	G ₀	G ₁	G ₂	G ₃
Perforation number	0	10	20	30
Perforation ratio p	0	1.0%	2.1%	3.1%

Figure 5.6 shows the comparison of the absorption coefficients of the original and the perforated P structures. The absorption coefficients, especially at the frequency range of 5000–6400 Hz, are obviously increased by the perforations. The enhancement improves as the perforation ratio increases from 1.0% to 3.1%. The perforation ratios of

1.0%, 2.1% and 3.1% have increased the frequency coverage of the original P structure by 30.1%, 41.7% and 51.3%, respectively. While the perforation ratio of 3.1% has increased the frequency coverage with the largest percentage, it can be observed that the increase of the frequency coverage is convergent as the perforation ratio increases, indicating the reduction of the effect of adding the same number of perforations on absorption enhancement.

Figure 5.6. Comparison of the absorption coefficients of the original P structure and P structures with the perforation ratios of 1.0%, 2.1% and 3.1%.

Figure 5.7 shows the comparison of the absorption coefficients of the original and the perforated G structures. The absorption coefficients, especially at the frequency range of 3500–5000 Hz, are obviously increased by the perforations. The enhancement improves as the perforation ratio increases from 1.0% to 3.1%. The perforation ratios of 1.0%, 2.1% and 3.1% have increased the frequency coverage of the original G structure

by 14.4%, 32.0% and 41.2%, respectively. The peak value of the 3.1% curve reaches 0.82, making the G structures more potential as sound absorbers than the P structures with the absorption coefficients lower than 0.8.

Figure 5.7. Comparison of the absorption coefficients of the original G structure and G structures with 1.0%, 2.1% and 3.1% perforations.

5.3. Summary

Three commercially used resins by SLA have been adopted for the fabrication of TPMS structures. The results of acoustic tests have suggested that the fabrication material has no major effects on the acoustic absorption performances of the TPMS structures. Therefore, Clear resin has been used to manufacture the micro-perforated TPMS structures for acoustic absorption enhancement due to its low cost. The results of the acoustic tests of the micro-perforated TPMS structures have shown obvious improvements on acoustic absorption by adding perforations, especially for P and G

structures with lower absorption coefficients in the target frequency range when compared to D structures. Furthermore, the effects of the perforation ratio of P and G structures have been studied and the increase of the perforation ratio in the observed range has shown positive effects on improving acoustic absorption of the structures. The improved G structures with much higher absorption coefficients than the P structures have more potential to be applied as sound absorbers.

Chapter 6 Conclusions and Recommendations

In this chapter, the main conclusions based on the works conducted through this Ph.D. study are given, and corresponding future works are recommended.

6.1. Conclusions

This Ph.D. study designs and develops innovative polymeric acoustic absorbers manufactured by additive manufacturing (AM) for tunable wideband sound absorption in various frequency ranges. Most of the conventional acoustic absorbers perform effectively in a narrow frequency range that cannot be tuned and the structural and functional limitations of the absorbers have restricted their applications. Therefore, novel absorbers are necessary to be developed to broaden the absorption frequency bandwidth and enhance functional properties. The reduction of prototyping costs of the novel structures should also be considered by utilizing new manufacturing processes and materials, and therefore AM and polymeric materials are introduced. The detailed conclusions are summarised as follows.

In this thesis, micro-perforated panels (MPPs) for tunable wideband sound absorption in the range from low frequency to middle frequency have been developed for absorption enhancement of the conventional MPPs. Innovative multi-layer MPPs with adjustable geometries were designed and manufactured by selective laser sintering (SLS), which is an AM system with the significant advantage of excluding the supporting structures when manufacturing multi-layer structures. The finite element method was applied to provide numerical predictions of the performances of the designed structures.

The effects of structural parameters of multi-layer MPPs on absorption coefficients and frequency bandwidths were analyzed by theoretical predictions, numerical simulations and experiments. The developed finite element model was proven to be feasible for the numerical simulation of multi-layer acoustic structures. To design acoustic absorbers with the most effective sound absorption in target frequency ranges, the maximization of the frequency coverage, which is the area under the sound absorption curve, was introduced as the criterion to design the selected parameters of the acoustic structures. The evaluation method was validated by the large improvements on the absorption performances of the optimized multi-layer MPPs.

Afterwards, the multifunctional triply periodic minimal surface (TPMS) structures were realized by AM and characterized as innovative acoustic absorbers for efficient absorption in the range of high frequencies. Three typical types of TPMS structures: Primitive (P), Gyroid (G) and Diamond (D), were designed and fabricated by stereolithography apparatus (SLA), which is an AM system with the advantages of high accuracy, good surface finish and low cost. D structures exhibited superior absorption abilities with high absorption coefficients and wide bandwidths when compared to P and G structures. Therefore, a series of D structures were designed and fabricated for acoustic absorption tests to study the effects of the geometric parameters including the unit cell size, the volume fraction and the height. The analysis of the results provided an optimized design of geometric parameters of a D structure for optimal acoustic absorption. The results of the acoustic tests suggested that the optimized structure was able to achieve the largest frequency coverage within the target frequency range when compared with other D structures. The large frequency coverage with high absorption coefficients in the target

frequency range achieved by D structures revealed the potential of TPMS structures as effective acoustic absorbers and thus extends the multifunctionality of TPMS structures to include absorption.

Finally, improvements on sound absorption of the TPMS structures were achieved by innovative micro-perforated TPMS structures which combined MPPs and TPMS structures. Before fabricating the samples for acoustic tests, the effect of the materials of TPMS structures was studied to select a proper material for manufacturing. Three commercial resins of SLA including Clear resin, Rigid resin and Tough resin were used for the fabrication of TPMS structures. The results of the acoustic tests suggested that the fabrication material had no major effects on the acoustic absorption performances of the TPMS structures. Therefore, Clear resin was used to manufacture micro-perforated TPMS structures due to its low cost. Obvious enhancement on acoustic absorption by adding perforations was shown by the results of acoustic tests of the micro-perforated TPMS structures, especially for P and G structures with lower absorption coefficients in the target frequency range when compared to D structures. In addition, the increase of the perforation ratio in the observed range exhibited positive effects on improving acoustic absorption of the P and G structures. The improved G structures with much higher absorption coefficients than the improved P structures have more potential as sound absorbers.

Innovative acoustic absorbers of multi-layer MPPs, TPMS absorbers and micro-perforated TPMS absorbers have been designed in this Ph.D. study. The absorption coefficients of the designed absorbers could reach nearly 1.0, which was comparably high among acoustic absorbers. For the absorption coefficients that reached 0.7 or above, the

corresponding frequency bandwidths of multi-layer MPPs were around 500 Hz and were tunable in the range of 500–1600 Hz, which were wider than those of the conventional absorbers. Superior corresponding frequency bandwidths of around 2000 Hz in the frequency range of 4000–6400 Hz were achieved by TPMS structures. AM systems using polymeric materials were applied for facilitating the development of the acoustic devices by efficient and low-cost prototype fabrication.

6.2. Recommendations

Innovative acoustic absorbers for effective sound absorption in various ranges have been designed and successfully manufactured by AM, which has indicated the potential of AM as the solution to facilitate the development of acoustic engineering by efficiently manufacturing the designed structures with complex structures. Nonetheless, the studies on the absorption performances of the designed TPMS absorbers and micro-perforated TPMS absorbers are based on acoustic tests, in which the numerical simulations and predictions on the absorption performances can be developed to guide the future designs according to application requirements. Furthermore, the application of AM on manufacturing the acoustic absorbers can be extended to manufacturing a wide range of multifunctional acoustic devices. The following research works are thereby recommended.

Firstly, theoretical modelings of TPMS and micro-perforated TPMS structures will provide effective design guidance for optimization of parameters. The pre-set module in COMSOL Multiphysics software that has been used for the simulation of multi-layer MPPs is not able to conduct the simulations of TPMS structures with the high complexity

of geometries. For each type of the TPMS structures, an approximate geometry model that calculates the permeability, porosity, airflow resistance and thereby the absorption coefficients using the geometric parameters is suggested to be built (Zhai *et al.*, 2018). An alternative solution that has been widely used for porous absorbers and is potential for TPMS absorbers is to calculate the impedance and the absorption coefficients of the structures by the Delany and Bazley model (Delany and Bazley, 1970) and the Miki model (Miki, 1990), which requires the values of the airflow resistivity of the structures that can be measured experimentally. Similar to the sandwich structures with micro-perforations (Wang *et al.*, 2020), micro-perforated TPMS structures can be modelled as a series of MPPs arranged in various configurations to calculate the absorption coefficients. Due to the curved surfaces of the TPMS structures, the air gaps of MPPs in the models should be considered to be graded or irregular-shaped and the acoustic impedances of the MPPs are calculated differently from the conventional MPPs (C.Wang *et al.*, 2010; Yang *et al.*, 2013). The systematic optimization of the structural design of absorbers for optimal acoustic absorption can also be facilitated by providing algorithms of geometrical parameters and sound absorption coefficients. For systematic optimization, the effects of all possible parameters should be studied individually and the interaction effects between different parameters should also be investigated.

Secondly, comprehensive understandings of the effects of structure size and materials can be provided by further experimental validations. The designed absorbers including multi-layer MPPs, TPMS absorbers and micro-perforated TPMS absorbers can be manufactured at the size of walls or large-scale constructions for validations of their absorption abilities in building and construction applications, especially for TPMS

absorbers and micro-perforated TPMS absorbers as promising structures to construct soundproof vehicles and buildings that do not require additional absorbers. The acoustic tests by the impedance tube method have shown the large potential of the designed structures in sound absorption, while the reverberation room method that closely simulates the conditions encountered in rooms and buildings can be applied to measure the random incidence absorption coefficients of the designed absorbers (ISO, 2003). The reverberation room can also facilitate the systematic study on the effects of partial cells at the edges of TPMS structures as the sample sizes will not be restricted by the testing machine. Meanwhile, various materials can be applied for sample fabrication to further validate the current observation of limited effects of materials on acoustic absorption of the designed TPMS absorbers. Although the absorption of MPPs is considered to be independent of materials, the materials have effects on the solid panel without perforations which can also be further studied.

Finally, the development of complex acoustic devices for advanced applications can be focused owing to the introduction of AM. For the absorbers studied in this dissertation, graded thickness, volume fraction and arrangement of perforations are able to be fabricated easily for further investigations. The effects of different shapes of perforations can also be studied because the MPPs with imperfectly circular perforations showed consistently better absorption than the theoretically predicted absorption of the circular perforations. Besides the absorbers, a wider range of applications of AM can be foreseen in acoustic engineering. Acoustic metamaterials are promising devices for extraordinary applications such as acoustic cloaking (Zigoneanu *et al.*, 2014), acoustic rainbow trapping (Zhou *et al.*, 2016) and enhanced acoustic sensing (Chen *et al.*, 2014). AM

provides the convenience and freedom of manufacturing various acoustic metamaterials whose properties are highly dependent on structural designs of the structures with high complexity. The wide range of the sizes of AM-manufactured structures from micro to macro scale induces acoustic responses at respective wavelengths and frequencies. The exploration of acoustic metamaterials fabricated by AM will promote the development of acoustic absorption and other applications in acoustic engineering.

The design and development of acoustic absorbers and other acoustic devices are significant in acoustic engineering and are accelerated by the development of AM as a promising mainstream technology for the manufacturing of customized and value-added products.

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- [1] **W. Yang**, J. An, C. K. Chua, K. Zhou. Acoustic absorptions of multifunctional polymeric cellular structures based on triply periodic minimal surfaces fabricated by stereolithography, *Virtual and Physical Prototyping*, 2020. **15**(2):242-249.
- [2] **W. Yang**, X. Bai, W. Zhu, R. Kiran, J. An, C. K. Chua, K. Zhou. 3D printing of polymeric multi-layer micro-perforated panels for tunable wideband sound absorption, *Polymers*, 2020. **12**(2):360.

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- [1] **W. Yang**, J. An, C. K. Chua, K. Zhou. “Wideband acoustic absorptions of polymer cellular structures based on triply periodic minimal surfaces”, Additive International, 8th–11th July, 2019, Nottingham, UK.
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- [3] **W. Yang**, S. Yuan, C. K. Chua, K. Zhou. “Wideband sound absorption of 3D printed multi-layer micro-perforated panels”, in proceedings of 3rd International Conference on Progress in Additive Manufacturing, 14th–17th, May, 2018, Singapore.