# ACOUSTIC PROPERTIES OF A HONEYCOMB-SILICONE RUBBER ACOUSTIC METAMATERIAL

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In order to overcome the influence of mass law on traditional acoustic materials and obtain a lightweight thin-layer structure which can effectively isolate the low frequency noises, a honeycomb-silicone rubber acoustic metamaterial was proposed. Experimental results show that the sound transmission loss of acoustic metamaterial in this paper is greatly higher than that of monolayer silicone rubber metamaterial. Based on the band structure, mode shape shapes, as well as the sound transmission simulation, the sound insulation mechanism of the designed honeycomb-silicone rubber structure was analyzed from a new perspective, which had been validated experimentally. Side length of honeycomb structure and thickness of the unit structure would affect sound transmission loss in damping control zone. Relevant conclusions and design method provide a new concept for engineering noise control.

Keywords: low frequency sound transmission loss; acoustic metamaterial; bandgap

## 1. Introduction

Bodies of the vehicles (cars, planes and trains) are mainly made up of thin wall panel. These panels and sound-insulation inner decorations will have obvious resonances below 500Hz, which leads to poor sound insulation performance of bodies. Considering the body size and weight, thickness of sound insulation parts usually is required less than 5mm. This target poses challenges for sound insulation design. membrane and slab acoustic metamaterials have the advantages of low surface density and small thickness, and these acoustic metamaterials are expected to be the optimal choices in low frequency vibration noise problems. Periodic holes or columns have been arranged, hence formed the whole thin slab acoustic materials. Thickness of the slab, column size and shape of the hole are adjustable [1-2]. Membrane acoustic materials are different from thin slab type， these are usually splitted out lots of single unit cells by using frame. Due to added mass block hugging the membrane, the stiffness of membrane is not enough to overcome its own gravity. And because of this, tensioning membrane could be used in the experiment. By changing the shape, weight, number and position of the mass block, unit resonant frequency and acoustic properties can be adjusted. In 2008, ZY Yang et al proposed membrane acoustic metamaterial which has outstanding sound insulation ability in low frequency [3]. Membrane acoustic metamaterial first proposed to get preferable sound insulation only within a very narrow band near the antiresonant frequency. Through the way of multi-layer membrane superposition, membrane acoustic metamaterial can get high acoustic attenuation in range of 50Hz ~ 1500Hz [4]. Naify et al studied the effects of frame to sound insulation, revealed that unqualified frame would produce overall structural effects which can increase the corresponding effective sound insulation band width [5]. In

2014, Chen et al studied the energy dissipation mechanism of membrane and slab acoustic metamaterials through theoretical derivation. P Shen et al proposed a broadband sound absorption performance of membrane dark acoustic metamaterial [6], enriched the application of low frequency acoustic metamaterial in noise control solutions.

Although the above researches show that membrane acoustic metamaterials have a good application in low frequency vibration and noise attenuation, material weakness is still prominent. The stiffness of membrane is low and its acoustic performance is not stable, so sound insulation of membrane acoustic metamaterial is strongly dependent on the tension. During the experiment, membrane tension is difficult to be quantitative controlled. Even if silicon rubber is not affected by load, the material itself is also very easy to aging which will greatly shorten the material life.

In order to solve these problems, there are three kinds of methods: Put forward a new method of accurate control of membrane tension; Put forward a structural design of weakening or eliminating tension dependence; Find the alternative materials of silicone rubber membranes. Active control acoustic mematerial offers a solution of accurate controlling membrane tension through the electric or magnetic field [7]. However, active control system needs to add complex components which would have to bring many new problems. Honeycomb material has many advantages structurally: low density, large porosity, high strength, high flatness, good impact resistance. Owing to their superiorities, honeycomb structures are widely used in aerospace industry [8].

Inspired by this, this paper proposed a new honeycomb-silicon rubber acoustic metamaterial to weaken or eliminate membrane tension dependency. It is structured by perfusing honeycomb unit cell using silicon rubber. Compared with the single membrane, the structural mechanics behavior of acoustic metamaterial shifts from membrane to thin slab type. Mechanical equation of the thin slab belongs to the fourth order partial differential, but that of membrane is second order.

Differences are caused by the structure stiffness representation form. If flexural stiffness on membrane thickness direction cannot be ignored, membrane mechanical equation would be restored to slab mechanical equation. Based on this, this paper introduced honeycomb structure into membrane to weaken membrane itself tension.

## 2. Simulation model and experiment sample

Structure schematic diagrams of honeycomb-silicone rubber acoustic material in this paper is as shown in Figure 1. Structure size is: $L_a$=$L_b$=5mm, $H_a$=$H_b$=2mm, and $t$=0.5mm. Figure 1 shows the first Brillouin zone of triangular lattice structure. By using COMSOL Multiphysics to solve structure eigenfrequency, and scanning frequencies along the first Brillouin zone path ($\Gamma \rightarrow X \rightarrow M \rightarrow \Gamma$). If three different direction eigenfrequencies are arranged together, the band structure is obtained. The structure average mass is small, and its surface density is only 1.77$kg/m^2$.

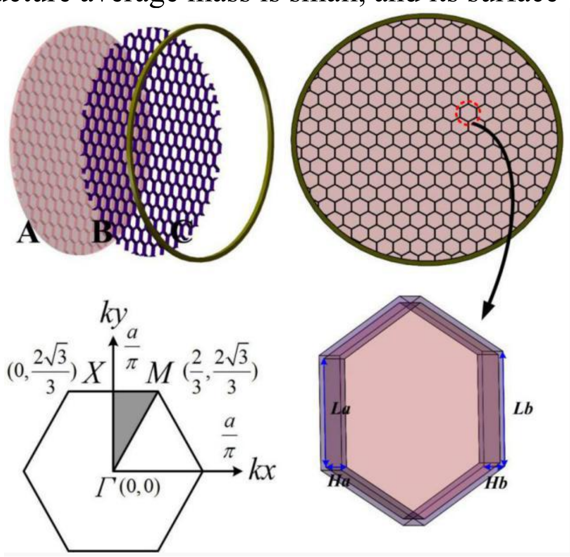


Figure 1: Structure schematic diagrams, size marking and first Brillouin zone

STL is calculated by COMSOL Multiphysics Acoustic solid coupling module. In the process of simulation, the computational model has two calculation domains, air domain and solid domain respectively. The frequency-domain sound Helmholtz equation for air domain is

$$\nabla\cdot(-\frac{1}{\rho_c}\nabla p)-\frac{\omega^2 p}{\rho_c c^2}=0 \tag{1}$$

Here the acoustic pressure is a harmonic $p=p_0 e^{i\omega t}$ quantity, $p=p_0 e^{i\omega t}$ , $p$ is the pressure(N/m$^2$), $p_c$ is the density(kg/m$^3$), $\omega$ is angular frequency(rad/s), and $c$ is the speed of sound(m/s). On the outer surface of the air domain, we specify an incident plane wave to represent an incoming sound wave. On coupling boundary condition between the air domain and solid, we set the boundary load $F=-n_s p$ (force/unit area) on the solid structure, where $n_s$ is the outward-pointing unit normal vector seen from inside the solid domain.

While on the fluid side the normal acceleration experienced by the air, it is set equal to the normal acceleration of the solid. Mathematical expression is as follows:

$$-n_a\cdot(-\frac{1}{\rho_0}\nabla p+q)=\alpha_n \tag{2}$$

Where $n_a$ is the outward-pointing unit normal vector seen from inside the acoustic domain, and normal acceleration $a_n$ is equal to $(n_a\cdot u)\omega^2$ , q is solid domain reaction to fluid domain. $u$ is the calculated harmonic displacement vector of the solid structure. In this model, we set a fixed constraint on the solid boundaries. In this case $u=0$ . It reduces the above condition $(\alpha_n=0)$ to the sound hard boundary condition:

$$n_a\cdot(-\frac{1}{\rho_0}\nabla p+q)=0 \tag{3}$$

For the solid domain, membrane can be regarded as thin slab abstractly, so the flexural wave of a thin solid elastic membrane satisfies the bi-harmonic equation as follows:

$$\nabla^4 z-(\rho h/D)\omega^2 z=\overline{f} \tag{4}$$

Where $z$ is the normal displacement of the membrane, $\overline{f}=p_i+p_r-p_t$ is force on the surface of a membrane, with $p_i$, $p_r$, $p_t$ representing incident sound pressure, reflect sound pressure and transmission sound pressure. $D=Eh^3/12(1-\upsilon^2)$ is the flexural rigidity and $h$ is the thickness of the membrane, with $\rho$ , $E$, and $\upsilon$ being the Mass density, Young's modulus, and Poisson's ratio, respectively. The equation for STL can be written as

$$Sound\,transmission\,loss(STL)=10\log(\frac{W\_in}{W\_out}) \tag{5}$$

$W\_in$ and $W\_out$ represent incidence and exit sound pressure energy respectively. By varying the excitation frequencies of the incident waves, the STL can be obtained. Elastic constants of materials are shown in table 1.

Table 1: Material properties

| Material Category | Young's modulus(Pa) | Poisson's ratio | Mass density(kg/m3) |
|---|---|---|---|
| Aluminum alloy | $7.76\times10^{10}$ | 0.352 | 2730 |
| Silicon rubber | $1.175\times10^{5}$ | 0.469 | 1300 |
| Nylon | $2\times10^{9}$ | 0.4 | 1150 |

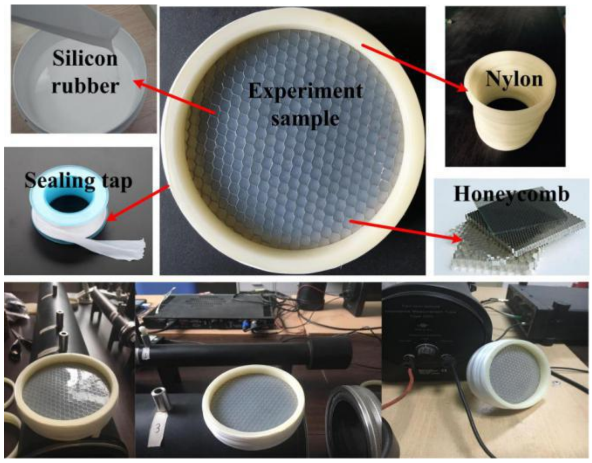



Figure 2: Experiment samples and acoustic impedance experiment

In order to obtain sound insulation performance reliable data of honeycomb membrane acoustic metamaterial and test the simulation results, corresponding experiment samples are made and shown in Figure 2. Normal incidence sound transmission measurements for the honeycomb-silicon rubber acoustic metamaterial structures were conducted using an impedance tube (*Brüel and Kjær* 4206) . Figure 3 shows a schematic of the impedance tube used for measuring STL of the 100mm diameter circular samples over a frequency range of 100 Hz ~ 1600Hz. In order to ensure the stability of the samples in tube, two hard nylon border (Figure 1) was made to fix structure unit cell and could provide the stiffness at the circumference. Multiple sealing tap is designed, and prevent sound leakage, as shown in Figure 2. Two microphones were positioned upstream of the sample to measure the incident sound pressure level, while two additional microphones were situated downstream of the sample to measure the transmitted sound pressure level. An anechoic termination (in the form of a foam plug) on the receiver side of the sample prevented transmitted sound from being reflected back to the sample. The STL of the structure was calculated by using a transfer matrix method.

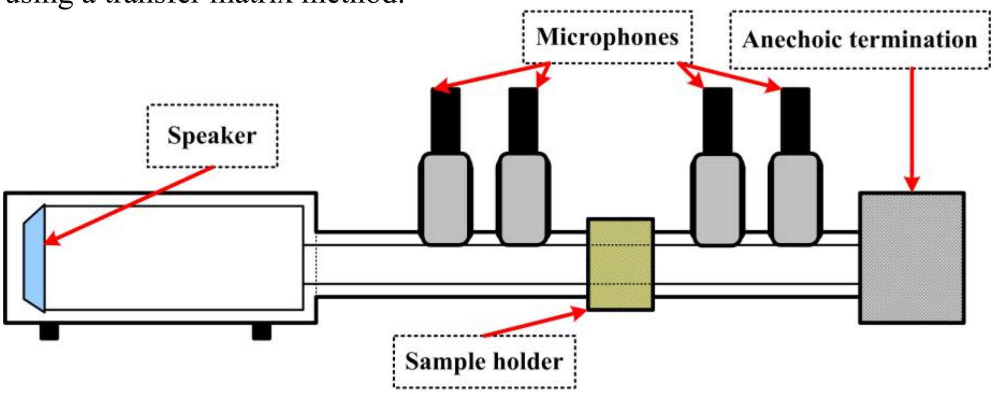



Figure 3: Schematic of small-diameter impedance tube showing locations of speaker, microphones, and sample.

Sound insulation ability of acoustic metamaterial can be reflected by breaking mass density law, and thus measured STL should compare with the predicted STL by mass density law to highlight itself outstanding acoustic performance. Mass density law is as follows:

$$STL = 20\lg\left(\omega\rho_s h / 2\rho_0 c\right) \approx 20\lg m_s + 20\lg f - 42 \tag{6}$$

Where $\omega=2\pi f$, $\rho_s$ and $h$ are the unit cell density and thickness respectively, $\rho_0$=1.225kg/m$^3$, $c$=340m/s, $ms=\rho sh$ is surface density.

# 3. Results and discussions.

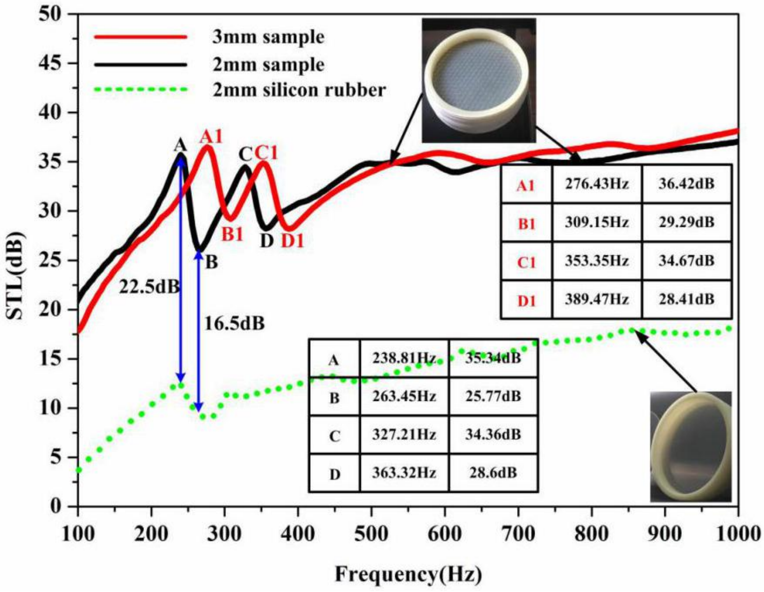



Figure 4: STLs comparing of different samples

## 3.1 Manuscript Title

In order to study the influence of membrane thickness on the STLs, Figure 4 gives experimental STLs of two different type honeycomb-silicone rubber acoustic metamaterials. Red and black full lines represent the STLs of corresponding 3mm and 2mm samples respectively. In order to highlight the sound insulation advantage of proposed structure in this paper, monolayer silicon rubber membrane sample was made and tasted itself sound insulation ability, the results are shown as green full line in Figure 4. It can be seen from the Figure 4 clearly, red and black full lines are higher (15dB) than green full line in range of 100Hz ~ 1000Hz on average, with significant difference between the three lines. According to acoustics theory, STL curves in Figure 4 can be very clearly divided into stiffness, damping, and mass control zone along with the increase of frequency. In mass control zone (above 400Hz), 3mm (red full line) and 2mm samples are about the same sound insulation ability, however these are higher around 20dB than green full line. In damping control zone (238Hz ~ 400Hz), the STLs of two type samples appeared two crests and troughs obviously, which are marked out in Figure 4. Crest and trough values corresponding frequencies and STLs are marked in Figure 4. It is different that only one pair of obvious crest and trough appears in green full line in damping control zone, this illustrates the addition of honeycomb aluminum would change damping control characteristic of the membrane structure itself. According to thin slab sound insulation theory, damping control zone corresponds to first resonance frequency of structure itself, and bandwidth of first resonance zone is associated with structural surface density, shape, installation, and damping coefficients. In damping control zone, first crest and trough of 3mm sample's STL is higher than that of 2mm sample. On account of monolayer membrane thickness (2mm) and through comparison, the crest of black full line is higher around 22.5dB than green full line, and its trough is also higher around 16.5dB than green full line. Then comparing the second crest and through of 2mm and 3mm samples, the results are similar. Because damping effect of 3mm structure itself is larger than that of 2mm sample, STL of 2mm sample has more violent shocks in damping control zone, but that of 3mm sample is smoother. That is why the crest values (A1 36.42dB, C1 34.67dB) of red full line is same to that (A 35.34dB, C34.36dB) of black full line, but their troughs (B 25.77dB; B1 29.29dB) values are different. The STLs in Figure 4 also illustrates the addition of damping would lead to structural damping control zone move to high frequency range. For stiffness control zone (below 238Hz), STL of 2mm and 3mm samples are higher that of monolayer silicone rubber membrane, the reason is that increase of thickness would also affect itself stiffness.

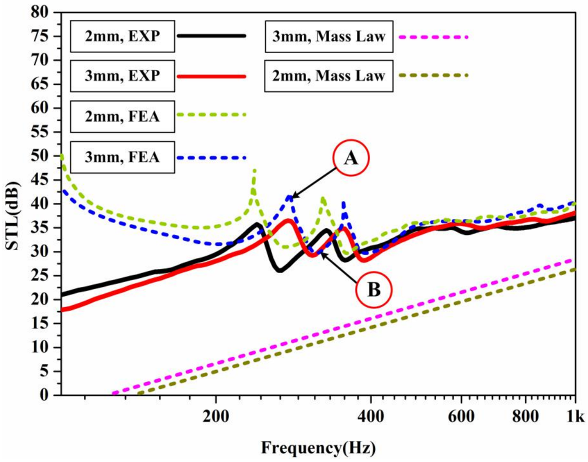
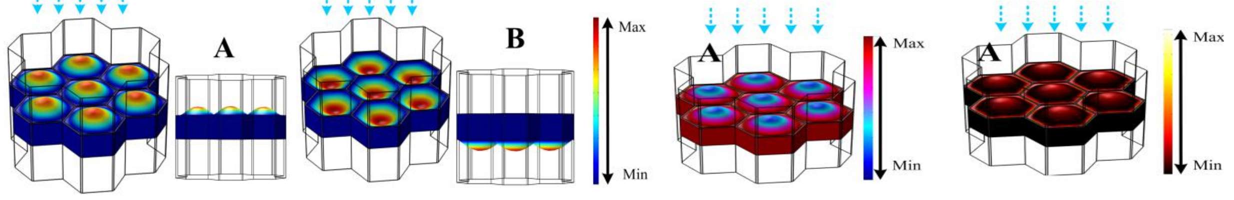



Figure 5: STLs calculated by FEA and experiments

Figure 5 gives the comparison of experimental and FEA results. Unit structure size and material properties in FEA simulation are consistent with experiment samples. This paper adopts seven units according to the hexagonal array, for reducing calculation cycle. Black and red full lines in Figure 5 represent 2mm and 3mm experiment results respectively, cyan and bule dotted lines represent FEA results, fuchsia and yellow dotted line represent the mass law results. In damping control zone, STLs of FEA also have two crest and troughs, these are the same to the experiment results, but FEA results are higher. The main reason for these is that, the silicon rubber is considered as linear elastic material in FEA, but actual material should be viscoelastic material, in addition, random errors also exist in experimental process. FEA results show a close agreement with experiment results in mass control zone, but this agreement is not very well in stiffness control zone. As previously mentioned, sound insulation test system in this paper adopts transfer function method with four microphones. In this test method, periphery of samples are free to move on axial direction in order to reduce sound insulation difference between small samples and actual large structure. This test method is different from the previous researches [9], which adopts improved sound insulation test method [10]. Fixed support is on the sample around in these improved methods, that is different from this paper, but FEA calculation in this paper adopts fixed boundary. These are the reasons why the FEA results are higher than experiment results in stiffness control zone. In the field of acoustic metamaterial, especially for membrane metamaterial, most of the existing researches adopt fix boundary in experiment test, so the STL can achieve very high value in low frequency, even more than 60dB. In this paper, test experiment samples have hard nylon to tighten themselves, but instabilities exist, resulting in the low STL below 200Hz.




(a) (b) (c) (d)

Figure 6: (a) and (b) are displacement mode shapes; (c) is acceleration mode shapes; (d) is strain energy density mode shapes.

Figure 6 are the displacement, acceleration, strain energy density mode shapes of A and B points marked in Figure 5, blue arrows represent the incident sound wave. It can be seen clearly from Figure 6(a) that maximal displacement occurs in silicone rubber membrane on sound insulation high spot (A point), and honeycomb aluminum is in static state relatively. At the moment, silicon rubber membrane displacement direction is opposite to incident sound wave direction, this situation is equivalent to there is a reactive force to incident sound wave. In addition the silicon rubber material itself has a certain sound insulation effect, these explain physically why the acoustic metamaterial structure proposed in this paper can has high sound insulation in these frequencies. On sound insulation low spot(B point), silicon rubber membrane displacement direction in Figure 6(b) is same to incident sound wave direction, and honeycomb aluminum is also static, so the reactive force to incident sound wave would not exist, corresponding STL decreases. Figure 6(c) and (d) are the acceleration, strain energy density mode shapes of sound insulation high spot respectively. Under the influence of the incident sound wave, high density honeycomb aluminum is equivalent to "mass block", which has large acceleration and its direction is same to incident sound wave direction; high elasticity silicon rubber is equivalent to "spring", which has small acceleration and large strain energy in central area. These motion states are analogous to "dynamic vibration absorber", which illustrates sound insulation nature of honeycomb silicon rubber metamaterial.

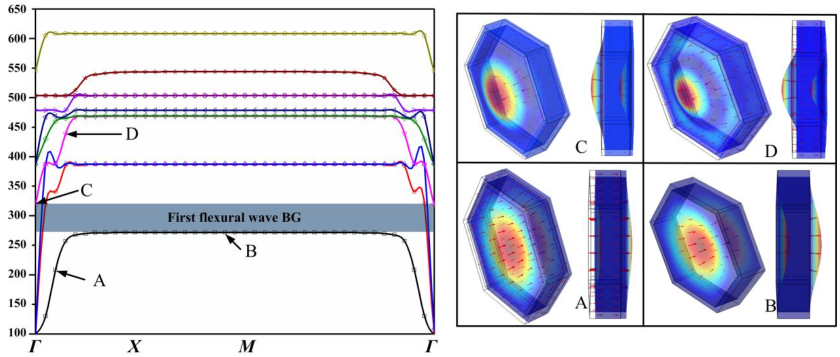



Figure 7: Band structure and corresponding displacement mode shapes

Next this paper discusses the band structure of honeycomb silicon rubber acoustic metamaterial unit. The FEA calculation results from Figure 7 manifest that the unit structure can unfold flexural bandgap from 275.4Hz to 324Hz. From the displacement mode shapes of A and B points in Figure 7, the maximal displacement deformation are concentrated at the center of the silicone rubber membrane, its direction is perpendicular to the XY plane, at this moment, the overall membrane presents the anti-symmetric flexural vibration. Observing the displacement mode shapes of C and D points on forth band, the membrane also presents the anti-symmetric flexural vibration, but its direction is opposite to that of A and B points. So the Z mode vibration coupling effect is the reason of flexural bandgap. From displacement mode shapes in Figure 7, it could be also concluded that, the frame can be seen as rigid foundation, which makes internal vibration mode be localized by unit structure fully. When the frame is connected to vibration absorption slab, the frame and slab are not vibrated, the vibration of membrane section dissipative energy continuously to achieve purpose of the vibration damping noise reduction. Analogously, when the sound acts on the unit structure in the form of air incentive, this action would arouse the internal vibrations of silicon rubber, and dissipative energies. Because the unit structure vibration abate the Sound or vibration excitation energies, the sound would not pass through the structure in certain frequency, so the noise and vibration isolation can be achieved. In a word, elastic wave propagation properties and shear action in silicon rubber are the main reason of flexural bandgap, because now the incident sound wave only have the effect of stirring structure vibration, almost all the vibration are concentrated on the internal. For the acoustic metamaterial in this paper, addition of honeycomb aluminum reduces tension dependence of overall structure, so the membrane vibration is close to thin slab. According to the thin slab vibration theory to analyze bandgap mechanism, there are a series of symmetric and anti-symmetric Lamb wave and shear horizontal wave modes. When the sound wave on arbitrary directions acts on the unit structure, this is equivalent to an incentive. Under the action of external force, elastic medium between adjacent parts would generate additional forces (internal force) and deformation (strain), and make the overall structure go into the fluctuating state. Because of the shearing action, symmetric, anti-symmetric Lamb wave and shear horizontal wave modes would be motivated, resulting in the flexural wave in slab. Reference to the relevant theory of locally resonant phononic crystal, locally resonant bandgap is owing to the coupling effects between the locally resonant modes and traveling wave modes in matrix.

Figure 7 and Figure 5 indicate that the STL curves and band structures have a strict corresponding relation in range of flexural bandgap. Frequency range of flexural bandgap is between the first troughs (B point in Figure 5) and second crest(C point in Figure 5), and the STL has a rapid growth within this range. In the range of flexural bandgap, STL would increase from trough critical point in damping control zone to a new crest, and reflect a relatively strong antiresonance effect. Deviating from the upper edge of flexural bandgap, STL variation trend would turn and its amplitude would be reduced. This kind of structure has an important potential application value in low frequency vibration and noise reduction, and has an advantage of flexible layout.

## 4. Conclusions

In this study, through honeycomb silicon rubber acoustic metamaterial with excellent sound insulation ability, the following novel results are experimentally and theoretically obtained: 1. For the structure of this paper, sound insulation experimental tests show that this kind of low surface density sample can show an excellent low frequency sound insulation performance. STL crest and troughs of 2mm and 3mm samples are higher around 20dB than monolayer membrane structure on average. 2. For the acoustic metamaterial in this paper, addition of honeycomb aluminum reduces tension dependence of overall structure, so the membrane vibration is closed to thin slab. STL curves and band structures have a strict corresponding relation in range of flexural bandgap. Frequency range of flexural bandgap is between the first troughs and second crest, and the STL has a rapid growth within this range.

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