LDV-based vibration measurement of a stiffened plate covered by a rubber coating with multi-layered periodic porous in air

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ABSTRACT

To control underwater acoustic radiation of ships and protect against detection, coatings are attached to the vibrating wet surface of the hulls. Coatings with different pores or scatterers enclosed will attenuate vibration and suppress sound radiation with different mechanisms. However, there are very few experimental studies concerning vibration measurement of a coated structure in water, mainly due to the water environment. A compromised way to reveal the attenuation mechanism is to measure the vibration in an anechoic chamber. A stiffened plate covered by a rubber coating with multi-layered periodic porous is measured by a LDV scanning system in this study, which is aimed to minimize added mass effect of contact measurement. Comparative measurements on bare stiffened plate and stiffened plate covered with solid rubber are also carried out. Vibration transmission from the stiffened plate to the coatings as well as vibration modes on the radiating surface for the three tested objects is obtained. The vibration and sound attenuation mechanism of the proposed rubber coating is unveiled.

Keywords: Rubber coatings, LDV, periodic porous   I-INCE Classification of Subjects Number(s): 54.1

1. INTRODUCTION

To control underwater sound radiation of ships or submarines, acoustic coatings are developed, which is covered on the wet surface of hulls to reduce vibration or attenuate under-water acoustic radiation. Acoustic coatings can be classified into anechoic layer, decoupling layer, damping layer and the like. The main functions of anechoic layer are absorbing the incident acoustic waves for protecting against detection by active sonar. Decoupling layer isolates the fluid from the vibrating of the structure to reduce vibration and to attenuate sound radiation. Damping layer usually employed to reduce vibration. This investigation is concentrated on a layer which is designed to serve the sound and vibration insulation purpose, aiming at lowering the underwater sound radiation.

For the purpose of underwater acoustic attenuation, the material should be compatible with water in terms of the acoustic impedance. Use of multilayer absorbers has become increasingly important in noise abatement. These layers are made of rubber material for its convenience of manufacturing and prominent damping. The solid rubber is not an ideal one because of its impedance match with that of water and the artificial rubber is proved to be the best. Therefore, the rubber material is designed with internal structures. Among them, two kinds of internal structures are studied extensively. One is with voids filled with air; the other one is cavities filled with scatterers (1). The vibration and sound insulation mechanisms are different for them. Some of the studies are summarized as follows.

Maidanik and Tucker (2) studied the relations between coupled vibration and sound radiation for coated panels immersed in fluid media. Foin et al. (3) presented a coupled model to study the vibration and acoustic response of a rectangular plate covered by a decoupling coating and immersed in a heavy acoustic fluid. Berry et al. (4) analyzed the vibroacoustic of a finite rectangular plate covered with a layer of decoupling material and immersed in a heavy fluid based on three-dimensional theory of elasticity. Tao (5) built the vibroacoustic coupled model to predict the noise reduction performance of an infinite bare plate covered with a void decoupling layer. From the literatures above, it can be seen

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that there are still few experimental studies concerning detailed vibration measurement of a coated structure in water, usually only the pressure is given. It is of high necessity to conduct experiments on this aspect to validate the numerical or analytical simulation and reveal the attenuation mechanism. A compromised way to reveal the attenuation mechanism is to measure the vibration in an anechoic chamber. In this study, a stiffened plate covered by a rubber coating with multi-layered periodic porous is measured by a LDV scanning system, with the aim to discover the attenuation mechanism.

2. MANUFACTURING AND MEASUREMENT SYSTEM

2.1 Test Samples

The test sample is a shown in Figure.1. It is a stiffened plate covered with coating, simulating a ship hull covered with coatings. Two kinds of coatings are considered, they are solid rubber and a multi-layer periodic porous coating. To illustrate the effect of coatings, a stiffened plate without coatings is tested as a comparative example.

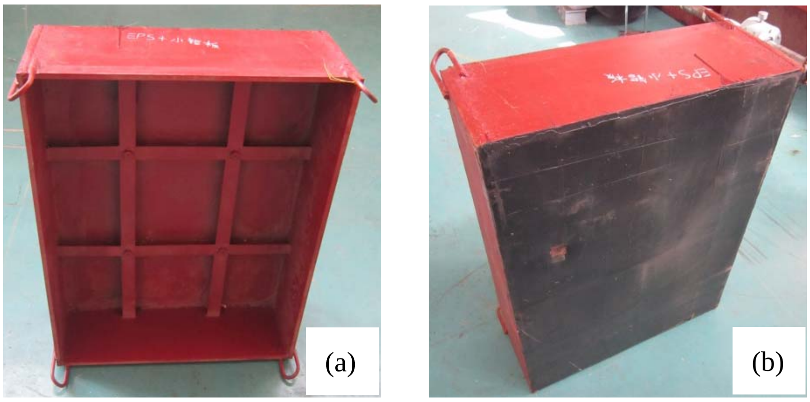


Figure 1 – Test samples: (a) stiffened plate; (b) stiffened plate with coating

The stiffened plate is a rectangular plate with web stiffeners. The dimensions of the stiffened plate are 800mm×600mm×3mm (length×width×thickness).There are two stiffeners in both directions with equal spacing. The stiffeners in the two orthogonal directions are T type stiffener with dimensions 35mm×35mm×3mm×3mm (the height of the web×the height of the flange×the thickness of the web×the thickness of the flange). Four plates with thickness 12mm and height 250mm are welded along the four sides to simulate the fixed boundaries for the stiffened plate and make the test object easy to be fixed. The stiffened plate is made of steel. The density, the elastic modulus and the Poisson ratio of steel are 7850 kg/m^3, 210 GPa and 0.3, respectively.

The periodic porous coating is a three-layer rubber slice with circular and diamond voids enclosed, which is shown in Figure.2. The overall dimensions for the coating are 200mm×100mm×50mm (length×width×height). Each layer is 13mm in thickness. The voids are of circular and diamond shapes. The circular void is 4mm in diameter and 13.5mm apart. The diamond void is 4mm×12mm (length× width). These enclosed voids are combined to form chiral structures. The density, the elastic modulus and the Poisson ratio are 1200 kg/m^3, 5 MPa and 0.48, respectively.

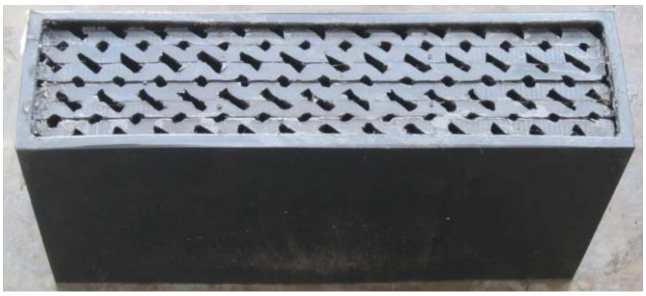


Figure 2 –The photograph of the coating

2.2 Measurement Configuration

To lower the influence of the ambient vibration and eliminate the contamination from surrounding objects to the sound results, measurement is carried out in a semi-anechoic chamber. The background noise of the chamber is 15.6(A)dB. The dimensions of the chamber are 8.4m×6.2m×4.8m (length×width×height). The frequency range for realizing a free field is 0.1k-10kHz. The test samples are located in the center of the chamber. The test samples are supported on a porous material to simulate the free boundary conditions. The test setup is shown in Figure.3.

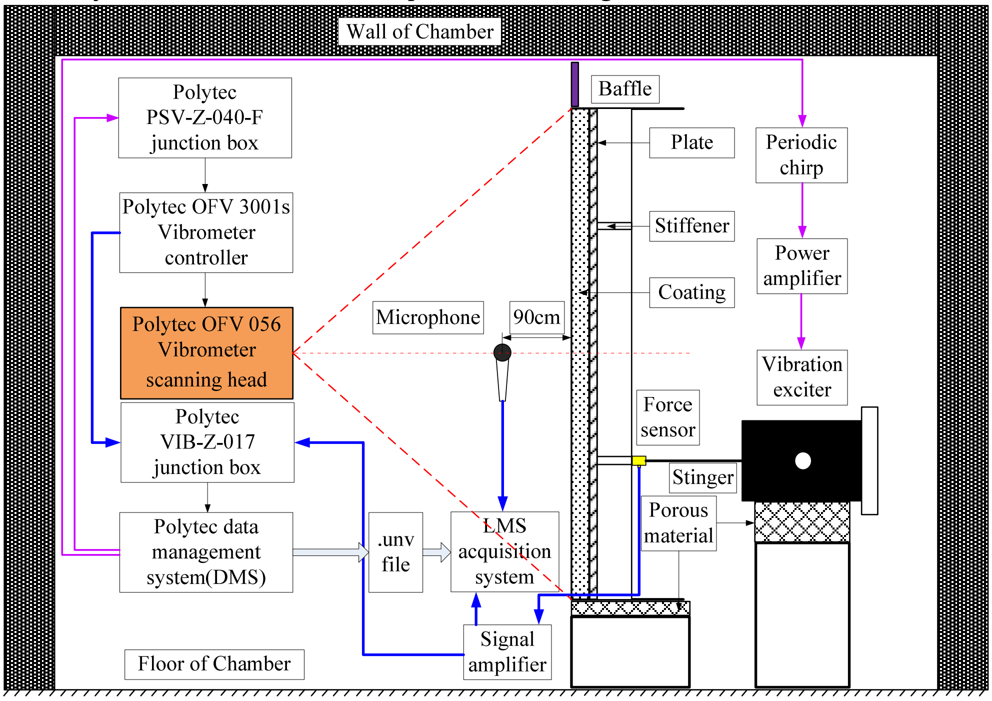


Figure 3 –The test setup

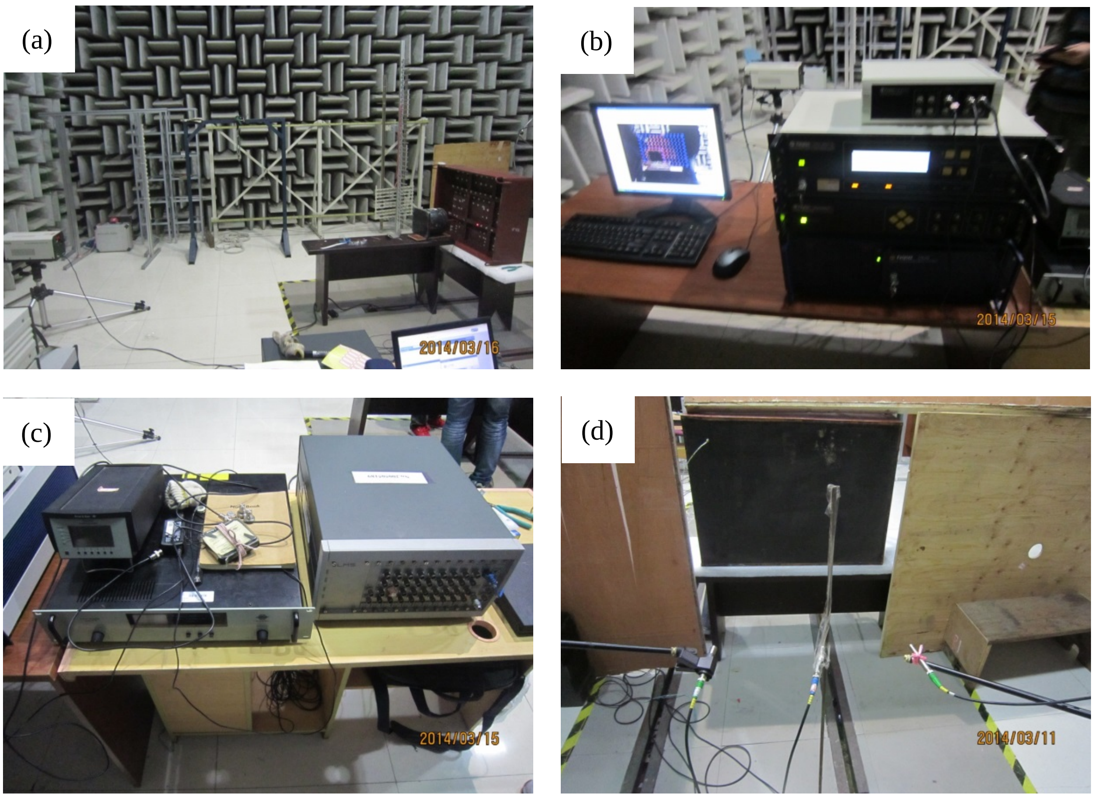


Figure 4 – (a) The configuration of the LDV and the test sample in the semi-anechoic chamber; (b) The

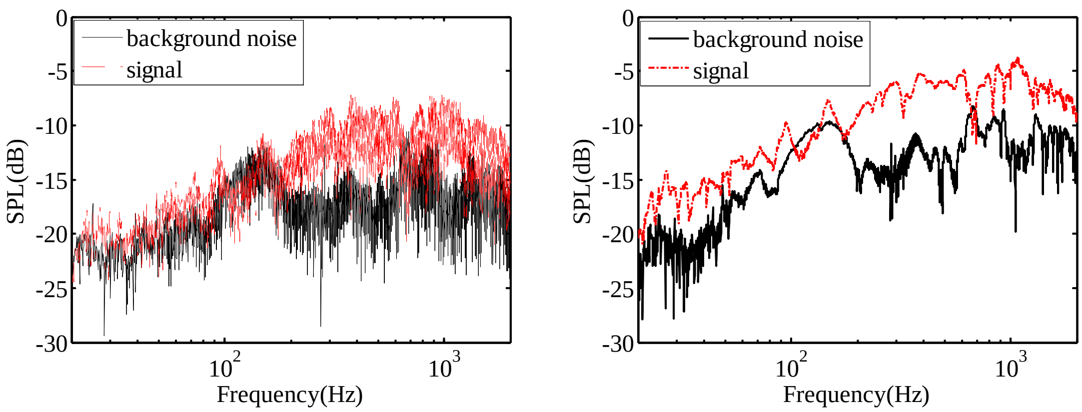
LDV system; (c) The LMS data acquisition system and (d) The microphone

The instrument equipment is shown in Figure.4. The stiffened plate and the coated plate were tested using a scanning laser Doppler vibrometer (Polytec PSV-Z-040-f). An electrodynamic shaker (B&K 4808), supported on a flexible cushion, was used to provide white noise excitation or periodic chirp excitation. A signal generator embedded in the Polytec data management system(DMS) provided input to the shaker through a power amplifier (B&K 2719) rated at 180VA. Input force was detected using a force transducer (B&K 1852401). The test samples were connected to the shaker using a stinger at the cross point of the two stiffeners. The velocity range for the scanning LVD corresponding to a vibrometer output voltage of 1V was 1mm/s. A FFT acquisition was performed within a selected bandwidth between 0 and 2kHz. The LVD was used to perform a modal analysis of the stiffened plate and the coated stiffened plate. For the modal analysis, 165 points on the surface of the test sample were used as measurement locations (11×15), each point scanning and FFT averaged 20 times. FRFs for each data point and average FRF were obtained in a universal file (.unv) for post-processing in a LMS data acquisition equipment (LMS SCADAS 310) by using the LMS modal analysis. A microphone is employed to measure the radiated sound, which is placed along the normal direction of the plate and 90cm away from the center of the surface of the plate (it is worth mentioning that this point is not a far-field point in its true sense). The center of the plate is 80cm higher than the floor of the chamber. To minimize the influence of the sound radiated by the shaker, some wooden plates are placed in front of the microphone.

3. RESULTS AND DISCUSSION

3.1 Background Noise

Firstly, background noise is measured for both white noise and periodic chirp excitation. The Sound Pressure Level (SPL, dBA reference pressure=2×10^{-5}Pa) results for the stiffened plate coated with solid rubber are presented in Figure.5. It can be seen that (1) the differences for the SPL results between the signal and background noise are more than 10dB over 200Hz, which satisfies the measurement requirements; (2) the SPL results under periodic chirp excitation are more convincing than those obtained by white noise excitation. Thus, periodic chirp excitation is adopted.



(a) white noise excitation (b) periodic chirp excitation

Figure 5 – Measured SPL results for stiffened plate coated with solid rubber

3.2 Sound Pressure Transfer Function

The SPL for the stiffened plate, stiffened plate coated with solid rubber and multi-layered periodic porous under unit force excitation (i.e., normalized to the measured force, which is equivalent to the sound pressure transfer function) are shown in Figure. 6. It can be seen that sound pressure transfer function of the stiffened plate is greatly suppressed after layered with both coatings. The suppression of sound pressure is reflected in the whole frequency range, the effect is especially remarkable in the higher frequency range (above 200Hz) for the evidence that the resonances have been attenuated mainly due to the added damping of the coatings. The periodic porous coating is superior to the solid coating almost in the whole frequency range. Thus, inclusion of the pores in coatings is beneficial for suppressing sound radiation.

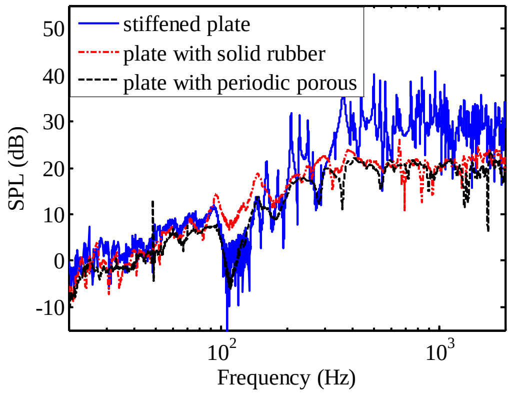


Figure 6 – Measured sound pressure transfer function results for the three tested objects.

3.3 Velocity Results on the Surface

To study the sound radiation suppression mechanism of coatings, the space averaged mean square velocity (MSV) results on the radiating surface for the stiffened plate, stiffened plate coated with solid rubber and multi-layered periodic porous are shown in Figure.7. It can be seen that MSV results of the stiffened plate is greatly suppressed above 200Hz. In the lower frequency range, the MSV on the solid rubber is amplified than that on the bare stiffened plate. The MSV on the solid rubber is almost larger than that on the periodic porous coating. Thus, the MSV result on the radiating surface is an indicator of the sound radiation ability. However, it cannot directly or totally reflect the true sound radiation in the far field. The reasons for this are as follows. Firstly, the sound radiation in the far-field is related to the radiation directivity of the field point; secondly, the sound radiation in the far-field is not only determined by the vibration amplitude, but also related to the vibration modes on the radiating surface, i.e., the velocity distribution on the surface. As a result, although the MSV is larger, the sound radiation in one field point is smaller. The attenuation mechanism is discussed from the two aspects, the vibration amplitude and the vibration distribution.

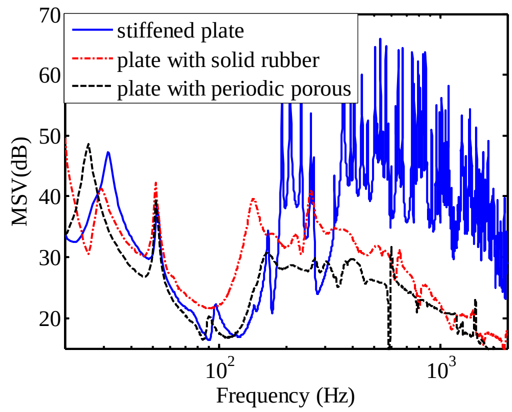


Figure 7 – MSV results on the radiating surface for the three tested objects

To investigate the vibration amplitude after layering the coating, the MSV results on the radiating surface and stiffened plate surface for the stiffened plate coated with solid rubber and multi-layered periodic porous are shown in Figure.8 and Figure.9, respectively. For the stiffened plate coated with solid rubber, it is seen that the vibration on the radiating surface is greatly attenuated. For the stiffened plate coated with periodic porous coating, it is seen that the vibration on the radiating surface is only suppressed above 300Hz. Another interesting thing is that MSV on the stiffened plate surface for the solid rubber is much greater than that for the periodic porous coating. This means that although the masses of the two coatings are almost the same, the suppression ability of vibration on the stiffened plate is not the same. From this aspect, periodic porous coating is stronger. As a result, although the transmission loss from the stiffened plate to the radiating surface is smaller for the stiffened plate coated with periodic porous coating, the response on the radiating surface is still smaller. Therefore, the vibration amplitude on the radiating surface is not only determined by the transmission loss from the stiffened plate to the radiating surface, but also related to the vibration attenuation ability of the coating on the stiffened plate.

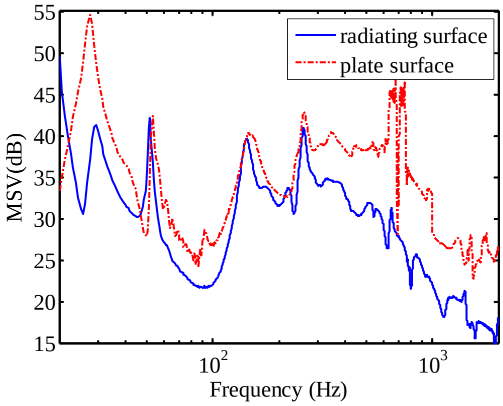


Figure 8 – MSV results on the radiating surface and the plate surface (coated with solid rubber)

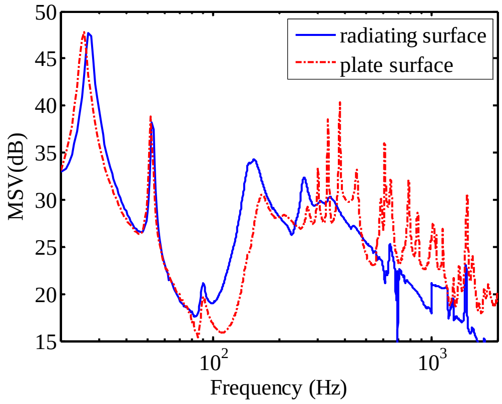


Figure 9 –MSV results on the radiating surface and the plate surface (coated with multi-layered periodic porous coating)

3.4 Velocity Distribution on the Surface

To study the influence of coating on velocity distribution, which influences the sound pressure in the far-field, the velocity distribution of each mode is identified. For the velocity distribution on the plate surface, the area that baffled by the exciter is measured by the acceleration transducer (B&W ICP transducer). The acceleration results are post processed to get the velocity results.

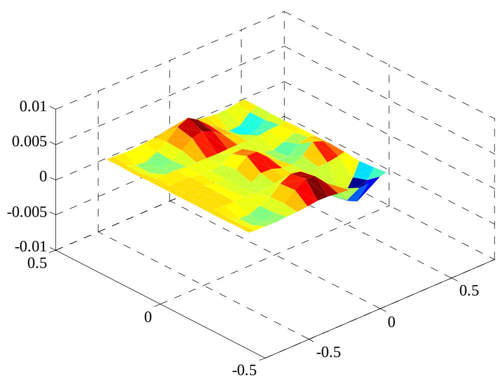
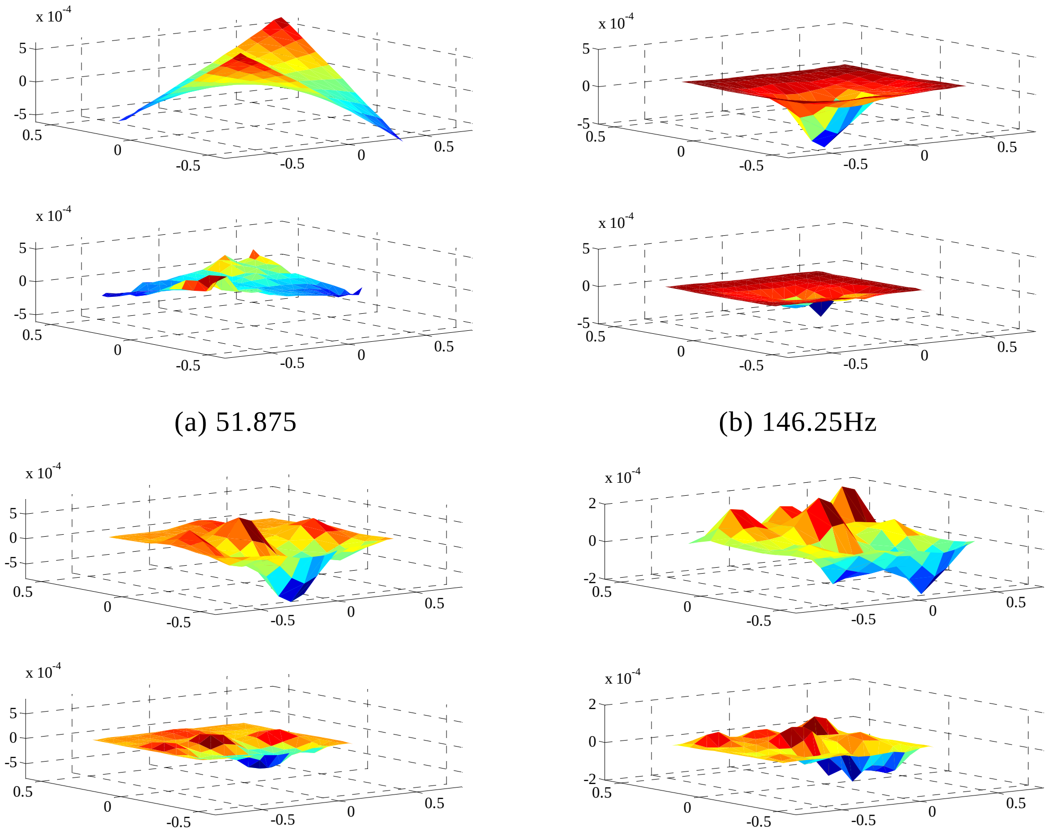


Figure 10 – Velocity distribution on the radiating surface of the stiffened plate(440.625Hz)

Firstly, the velocity distribution on the bare stiffened plate is given to give some illustration. One mode is given in Figure.10. It can be seen that the mode 440.526Hz is a typical mode of the stiffened plate, corresponding to the bumps of the nine small plates that are partitioned by the four web stiffeners. The phases between the adjacent areas are in-phase and out-of-phase along the two orthotropic directions, respectively. The phase distribution or the volume velocity distribution is a key factor that influences sound radiation in the far-field, which is mutually reinforced or cancelled, depending on the in-phase or out-of-phase distribution.

The velocity distribution on the radiating surface and the stiffened plate surface for the stiffened plate coated with solid rubber are shown in Figure.11. Only some modes that can be distinguished in an easy way are presented. It can be seen that vibration modal shape of each mode on the radiating surface is almost the same as that on the stiffened plate surface, which is verified by the fact that the relative ratio of the velocity of the two corresponding nodes on the two surfaces for each mode is the same. The absolute vibration amplitude does not have any physical meaning at all. That is to say, the solid rubber coating does not change velocity distribution on the two surfaces. This result indicates that the vibration mode of the solid rubber coating is following that of the stiffened plate.

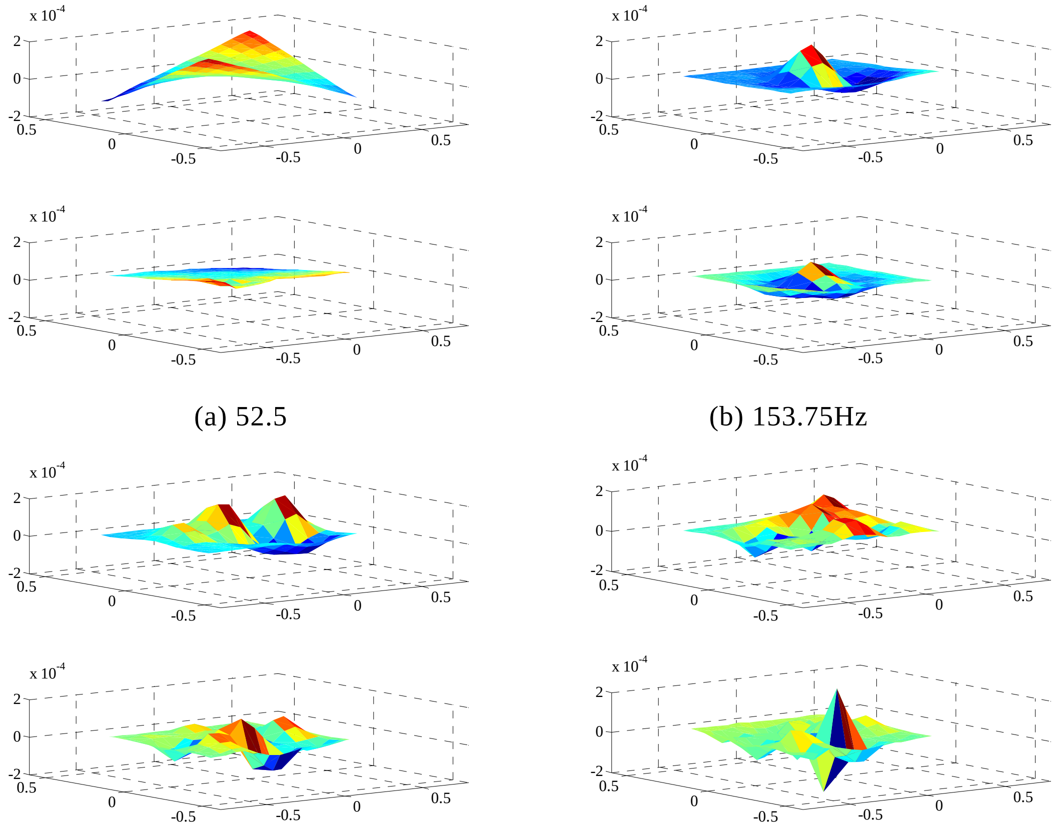


(a) 51.875 (b) 146.25Hz

(c) 259.375Hz (d) 313.75Hz

Figure 11 – Velocity distribution on the radiating surface (upper subfigure) and the stiffened plate surface (lower subfigure) (coated with solid rubber)

The velocity distribution on the radiating surface and the plate surface for the stiffened plate coated with solid rubber and multi-layered periodic porous are shown in Figure 12. It can be seen that unlike the solid rubber coating, the vibration modal shape of some mode on the radiating surface is not the same as that on the stiffened plate surface. This observation is more obvious for the mode with a higher frequency. Take 308.75Hz for example, the bumps and the sinks distribution on the radiating surface and the stiffened plate surface are different. Therefore, the multi-layered periodic porous coating will change vibration distribution between the two surfaces, which is fulfilled by the compatible deformation between the pores. Through careful design of pores, one can alter the phase distribution or the volume velocity distribution on the two surfaces, which can control the sound radiation in the far-field.



(a) 52.5 (b) 153.75Hz

(c) 260Hz (d) 308.75Hz

Figure 12 – Velocity distribution on the radiating surface (upper subfigure) and the stiffened plate surface (lower subfigure) (coated with multi-layered periodic porous coating)

4. CONCLUSIONS

The vibration of a stiffened plate, and a stiffened plate covered by a solid rubber coating as well as a multi-layered periodic porous coating is measured by a LDV scanning system in a semi-anechoic chamber. Vibration transmission from the stiffened plate to the coatings as well as vibration modes on the radiating surface and the plate surface is obtained. It is shown that sound radiation suppression of the coated stiffened plate is dependent on both the vibration amplitude and the vibration distribution. The periodic porous coating is superior in attenuating the vibration on the stiffened plate, while it is inferior in increasing the sound transmission loss from the stiffened plate to the radiating surface. The periodic porous coating can alter the velocity distribution between the stiffened plate surface and the radiating surface, which is beneficial to control the sound radiation in the far-field. However, solid rubber coating mainly follows the vibration mode of the stiffened plate.

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