Noise Reduction of Pipes under Sound Excitation via Various Attached Composite Structures

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Abstract. Pipe systems are broadly used in modern buildings and large transportation vehicles. The reduction of noise from pipes is one of the main concerns in their engineering applications. In this paper, the reduction of the radiated noise from pipes is studied experimentally using various attached structures. An experimental system is constructed in an anechoic room. Three types of attached structures are considered, including foam coatings, distributed absorbers and periodic absorbers. Via the comparison of basic pipe and pipes with different noise control designs, the reduction effects from the attached structures are compared and analysed. The results show that the radiated noise is effectively reduced by the foam coatings and distributed absorbers, especially, at most of the peaks of the noise. The distributed absorbers reduce the noise more on the whole. The low frequency peaks (below 200Hz) of the noise are reduced most by the periodic absorbers. Via the proper design of the structure types and corresponding parameters, the low frequency property, the broadband property and the reduction amplitude will be obviously enhanced.

Keywords: Noise reduction; Pipes; Foam coating; Distributed absorbers; Periodic design.

1. Introduction
Pipe structures are widely used in modern buildings and large transportation vehicles and so on. In these situations, pipe systems are one of the main noise sources and transmission paths for the sound environment. Parts of the noise are radiated into the surroundings through the vibration of the pipe walls and the others are through the outlet of the pipe systems. Correspondingly, to reduce the two parts of the noise, one method is the increasing of the sound insulation performance of the pipe walls, and the other is the reduction of sound transmission inside of the pipe lines and the reduction of noise emission from the outlets of the pipes. For the two aspects, much work has been done, and several techniques have been developed [1-3].

The pipe coating is one of the important methods for the first aspect [4-6]. Kanapathipillai et al [7] studied the effects of porous jackets on sound radiation from pipes. Guan et al [6] studied the vibration and noise reduction for the ventilation pipelines via the optimization design of the coating structures. The absorbers are also used broadly in the vibration reduction of machines, buildings, pipes with thick walls and so on, and are applied to reduce vibration and sound transmission of panels [8-10]. Recently, the distributed vibration absorbers receive much attenuation as their advantages of less added weight and wide effective frequency ranges for the reduction of panel vibration [11]. This provides a useful approach to reduce the vibration and noise of pipes. In addition, the artificial periodic structures/materials receive much attention, e.g., phononic crystals (PCs), acoustic metamaterials (AMs) and so on [12-14]. The periodic design has been also introduced into the field of vibration and noise.
control of pipe systems [2]. However, the studies on the noise reduction of thin wall pipes via periodic absorbers are seldom concerned in the published literatures.

In this paper, the sound properties for the pipes with various attached structures are studied by experiments. Four cases are considered and compared, including the basic pipe, the pipes with foam coatings, distributed vibration absorbers and periodic vibration absorbers. First, the experimental setups and the pipe structures are introduced. Next, the experimental results for sound properties of various pipe structures are analysed. The reduction of radiated noise from different design is compared.

2. Experimental Setups and Evaluation Parameters
An experiment system, including the pipe structure with end panels, foam base, microphone and corresponding supports, loudspeaker, power amplifier, date acquisition device, control and analysis computer and cables, is built inside an anechoic room. A foam block is installed at the surface of the top end panel. This aims at decreasing the effect from the noise transmitted through the end panel further. The sound excitation is generated by the loudspeaker which is installed at the center of the end panel at the bottom of the pipe. The control signal for the loudspeaker is pink noise signal from the computer and is amplified by the power amplifier. The same pink noise signal and the uniform parameter settings of amplifier are used for each test. The repeatability of the sound from the loudspeaker is confirmed. That is, the coincidence of the sound excitation for different pipes is good. Two 1/2 G.R.A.S microphones located outside of the pipes are used to collect the radiated noise. Both the two microphones are located at the position 1m far from the center of the wide pipe wall. The angles between the line from the microphone to the center point of wide pipe wall and the pipe wall are 0° and 45°, respectively, for microphones 1 and 2. The PAK-II date acquisition device and analysis software are used. Fig. 1 shows the photos of experimental setups.

Four pipe samples are constructed, that is, the basic pipe without attached structures, the pipes with foam coatings, with distributed vibration absorbers and with periodic vibration absorbers, as shown in Fig. 2. Among of them, the distributed vibration absorbers are composed of the foam components and aluminum surface panel; each of the periodic vibration absorbers are composed of the foam block and aluminum block, respectively, mainly as elastic component and mass component. The basic pipe is constructed by thin steel panels, and the end panels are 5mm thickness aluminum panels. The size of the pipes is 0.15m × 0.2m × 0.5m. The added masses from the attached foam, distributed absorbers and periodic absorbers are 10.5%, 29.7% and 12.5% of the mass of pipe walls of the basic pipe.

![Figure 1. Photos of the experimental setups.](image)

To evaluate the sound insulation performance of various pipes, the sound pressure level (SPL) of radiated noise is compared. That is,
\[ SPL = 20 \log_{10} \left( \left| \frac{p_{1,n}}{2\pi e^{-5}} \right| \right) \]

where \( p_{1,n} \) (n = 1, 2) is the radiated noise at microphone 1 or 2. In this study, the sound pressure level is modified by A-weighted method. In the following study, only the results from microphone 1 are compared. On one hand, the comparison results for most situations are similar for the two test points. On the other hand, the aluminum panels of distributed absorbers don’t cover the whole surfaces, and some sound is leaked. Correspondingly, the sound signals at microphone 2 are affected at some frequencies. This will affect the comparison of the noise from various pipes in individual cases.

Figure 2. Photos of various pipe structures. (a) Basic pipe, (b) pipe with foam coatings, (c) pipe with distributed absorbers, (d) pipe with periodic absorbers.

3. Results and Discussion

3.1. Reduction from Foam Coatings

Fig. 3 shows the radiated noise from the basic pipe and the pipe with foam coatings. The left figure shows the results for the frequency ranges from 0-600Hz, while the right one shows the results from 600-4000Hz. It can be seen that compared to the basic pipe, most of the peaks of the curves are effectively reduced. For instance, for the reduced peaks within low frequency ranges lower than 600Hz, the reduction amplitude is from 5dBA to 16dBA. However, there are also some specific peaks which are not reduced, and some new peaks are generated also. In addition, parts of the broadband noise is also reduced effectively, for instance, the noise in the ranges 550-650Hz, 760-1000Hz, 2700-2900Hz, 3450-3850Hz and so on.

To compare the reduction on the whole, several frequency ranges are defined and the total sound pressure is compared in these ranges, as shown in Fig. 4. Note that not only the results for the basic pipe and the pipe with foam coatings are presented; the results for the other two pipes are also shown, which will be analysed latter. It can be seen that the in the range 0-200Hz, the foam coating design nearly do not reduce the total radiated noise. This is mainly because some new peaks are generated and parts of the broadband noise are increased. In the range 0-600Hz, the noise is effectively reduced about by 4.5dBA. This value is obviously higher than that predicted by the mass law according to the increasing of total mass. In the range 600-2000Hz, the reduction value is quite small also. This is because the main peaks around 695Hz, 1053Hz, 1401Hz and 1774Hz are not reduced or some new main peaks are generated around these peaks. Besides, the broadband noise from 1430Hz to 1610Hz is increased. In the range 2000-4000Hz, the total sound pressure is reduced about by 2.5dBA. The decreasing of the reduction amplitude compared to the range 0-600Hz is because that the two groups of main peaks around 2400Hz and 2980Hz are not reduced effectively. The reduction in the range 0-4000Hz is similar with that in the
range 2000-4000Hz. This is because the noise from 2000-4000Hz plays a dominant role in the whole range in this study.

3.2. Reduction from Distributed Absorbers

Fig. 5 shows the radiated noise from the basic pipe and the pipe with distributed vibration absorbers. The left figure and right figure show the results for the frequency ranges from 0-600Hz and 600-4000Hz, respectively. It can be seen that compared to the basic pipe, most of the peaks of the curves are greatly reduced also. The number of reduced peaks and the reduction amplitude are bigger than that from single foam coating. For instance, the reduction for the main peak around 288Hz is reduced by 16dBA from the distributed absorbers, while this value is about 7.2dBA for the foam coatings. On the other hand, similar with the case with foam coatings, there are also some specific peaks which are not reduced and some new peaks are generated also. But the negative effects from the above two factors are smaller for case with distributed absorbers. In addition, the frequency ranges where the broadband noise is reduced effectively are also widened.

Similarly, the reduction for the total sound pressure within several frequency ranges is also compared. It can be seen from Fig. 4 that in the range 0-200Hz, the distributed absorber design reduces the total radiated noise about by 2.5dBA. In the range 0-600Hz, the noise is reduced about by 6.5dBA. The increasing for the reduction is mainly from the greater suppression on the main peaks around 248Hz. In the range 600-2000Hz, the reduction value is about 3.1dBA. The decreasing for the reduction amplitude compared to the former frequency range is because there are several main peaks which are not reduced effectively. Besides, small parts of the broadband noise are increased. In the ranges 2000-4000Hz as well as 0-4000Hz, the total sound pressure is reduced about by 6.6dBA. In addition, compared to the case with foam coatings, the reduction for the total sound pressure from distributed absorbers is increased effectively. And this improvement is better than that predicted by the mass law according to the mass increasing. It is also noted that due to the effects of the resonant modes and some other factors, the mass law does not work always for all the cases and at all the frequencies.

![Graph showing sound pressure vs frequency for basic pipe and foam coatings](image)

Figure 3. Comparison of the radiated noise from the basic pipe (No) and the pipe with foam coatings (Foam).
Figure 4. Comparison of the radiated noise from various pipes at different frequency ranges. Each group of the bars indicates the values for the basic pipe (left), pipe with foam coatings (middle-left), pipe with distributed absorbers (middle-right) and pipe with periodic absorbers (right).

3.3. Reduction from Periodic Absorbers
Fig. 6 shows the radiated noise from the basic pipe and the pipe with periodic vibration absorbers. The left figure and right figure show the results for the frequency ranges from 0-600Hz and 600-4000Hz, respectively. It can be seen that compared to the basic pipe, most of the main peaks of the curves at low frequencies are obviously reduced. However, there are also many peaks which are not reduced effectively or are even increased at middle to high frequencies. In addition, the frequency ranges where the broadband noise is lower are decreased obviously, while the frequency ranges where the broadband noise is higher are increased. Correspondingly, the reduction effect from periodic absorber design is weakened greatly on the whole. One of the main reasons is that the dynamic equivalent mass of the single absorbers as well as the whole periodic absorbers is decreased and is a negative value at middle to high frequencies. Correspondingly, some negative effects are generated to some extent.

Similarly, the reduction on the total sound pressure from periodic absorbers is also analysed. It can be seen from Fig. 4 that in the range 0-200Hz, the periodic absorber design reduces the total radiated noise about by 5.2dBA. This reduction is mainly from the great suppression on the peaks in this range, and is the highest one among the three types of attached structure design. The main reason is the existence of multi degree of freedom resonances of the absorbers. In the range 0-600Hz, the noise nearly is not reduced for the case with periodic absorbers. The main reasons are the increasing of broadband noise at specific ranges and the generation of new peaks. In the range 600-2000Hz, the radiated noise is increased about by 1.5dBA. In the ranges 2000-4000Hz and 0-4000Hz, the noise is also slightly increased. These phenomena indicate that the reduction effects from periodic absorbers are mainly presented at low frequencies, while it is not a good choice for reducing the total noise at the middle to high frequencies. Note that this is not an unalterable conclusion, and a better result can also be obtained at higher frequencies via proper design of periodic absorbers sometimes.

Figure 5. Comparison of the radiated noise from the basic pipe (No) and the pipe with distributed vibration absorbers (DVA).
4. Conclusions

In this paper, the reduction of radiated noise of the pipes is studied using various attached structures. An experimental system is built and the sound excitation is applied. Four cases are considered, including the basic pipe, the pipes with foam coatings, distributed vibration absorbers and periodic vibration absorbers. The reduction from different structural designs is compared and analysed. Compared to the basic pipe, most of the peaks of the radiated noise curves from low to high frequencies are reduced obviously by the using of foam coatings and distributed vibration absorbers. In addition, parts of the broadband noise are also reduced effectively by the two designs. Correspondingly, the total sound pressures in the whole frequency range and several divided wide ranges from low to high frequencies are also effectively reduced. Compared to the foam coatings, the number of reduced peaks and the reduction amplitude from distributed absorbers are bigger. The reduction from both the two designs is higher than that predicted by the mass law corresponding to the added masses. However, for the case with periodic absorbers, the obvious reduction is mainly obtained at low frequencies lower than 200Hz. And the reduction in this range is biggest among the three designs for sound radiation control. In the higher frequency ranges, the noise nearly is not reduced by the periodic absorbers, and is slightly increased to some extent. On the whole, a nice reduction result can be obtained by the proper application of the three designs, that is, foam coatings, distributed vibration absorbers and periodic vibration absorbers.

Acknowledgement

This research was financially supported by the National Science Foundation of China (Grant No. 51905532) and Research and Innovation Fund of Low Speed Aerodynamics Institute.

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