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INFLUENCE OF STRUCTURAL DAMPING ON RADIATION EFFICIENCY OF A MECHANICALLY OR ACOUSTICALLY EXCITED PANEL

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Abstract
Previous investigations have shown that damping is a good way to reduce structural vibration, but less effective for sound problems, since the radiation efficiency might be increased dramatically. This problem is further investigated in this paper. Results show that when a constrained damping layer is applied and when mechanical excitation is used, the radiation efficiency is increased by about 9 - 12 dB. This is mainly due to the near-field radiation around the excitation point, which is more important for a heavily damped structure than for a structure with little damping. The radiation efficiency of an acoustically excited panel, which is directly related to sound transmission, is also increased by about 6-9 dB when damping is added. Instead of radiation from near-field, forced transmission is now the main reason. Although resonant modes, which are the main contributors for vibration, are reduced greatly by added damping, the responses far away from the resonant frequency are little influenced, which are the main contributors for sound radiation and transmission.

INTRODUCTION

Damping treatment is a common way in industry to reduce structural vibration as well as sound radiation. However, this method does not always work, especially when sound radiation is concerned. Both successful and unsuccessful examples can be found in the literature. In order to reduce the transmission and radiation of structure-borne sound, designing engineers tend to use more and more damping materials. An example for that is a railway car, where more than one ton of damping materials can be used in a single carriage [1], without knowing exactly what the influence is.

When the vibration of a structure is concerned only, damping treatment is rather effective. However, if the noise radiated is also important, the effect of damping is often limited, because of the increase of the radiation efficiency [2-3]. The purpose of this paper is to further investigate the influence of structural damping on the radiation efficiency, when this is either mechanically or acoustically excited. Physics behind the phenomenon is discussed. From that it is hoped that some guidelines for practical applications of damping treatment can be drawn.

MEASUREMENTS OF RADIATION EFFICIENCY

Measurements were mainly performed for panels mounted in a frame in between a reverberation room and an anechoic room, with exactly the same arrangement as sound insulation measurement by using intensity method (ISO 15186-1:2000). In this arrangement the panel has effectively an infinite baffle. The panel can be excited either mechanically by using a shaker or acoustically by using the loudspeakers in the reverberation room. When shaker excitation is applied, it is attached to the fairly stiff frame to reduce the influence of the point of excitation. In addition, this arrangement can be viewed as representative for many real applications of panel mountings.

Two panels, with and without extra damping treatments, are tested for sound radiation efficiency in this arrangement: a 1-mm steel panel with the dimension of 1 m x 2 m (panel A) and a sandwich panel with the dimension of 0.39 m x 1.88 m (panel B). For both panels, a constrained layer damping (CLD) treatment is realized by adding a 0.60 mm rubber-like material with a 0.28 mm constraining steel sheet on top. A list of all panels tested and their structures are shown in Table 1 below.

Sound radiation efficiencies of freely-hanging panels have also been measured. Three configurations are compared for panel B: no damping treatment, 50% covered with CLD, and fully covered with CLD. For all three configurations, a shaker is used to excite the panel at same point, which is located at the un-damped half for the case of partially-damped panel. In addition, in order to compare sound powers radiated from near-field vibration and from resonant modes, a large sandwich floor panel (panel C) is also tested in freely-hanging condition. More detailed descriptions on this part of measurements can be found in reference [4].

Intensity method is used in most of cases to measure radiated sound power. Vibration velocity level is obtained by averaging results from at least ten accelerometer positions. The radiation efficiency is then obtained by using the formula

$$\sigma = \frac{P}{\rho c S < v^2 >} = \frac{\bar{I}}{\rho c < v^2 >} \tag{1}$$

Finally, two panels (panel D and E) with roughly same surface density but with totally different structures and loss factors are tested for sound reduction index, since this is closely related to the sound radiation efficiency when acoustic excitation is used. This test is made in accordance with standard ISO 15186-1:2000.

Table 1: List of panels and configurations tested

Panel	Size *m*	Construction	Condition	Density kg/m^2	Damping treatment	Loss factor
A	1 x 2	1 *mm* steel	Fixed	7.8	-	~0.005
B	0.39 x 1.88	0.8 *mm* plastic + 0.8 *mm* Al. + 0.8 *mm* plastic	Fixed or freely-hanging	4.8	-	0.02
Damped A	1 x 2	Same as A	Fixed	10.8	0.6 *mm* rubber-like material + 0.28 *mm* constraining steel	0.05
Damped B	0.39 x 1.88	Same as B	Same as B	7.8	Same as above	0.08
Partially -damped B	0.39 x 1.88	Same as B	Freely-hanging	-	Half of the panel covered by CLD	-
C	2 x 1.88	4 *mm* plywood +0.5 *mm* polymer + 9 *mm* plywood	Freely-hanging	8.3	-	0.2
D	0.96 x 0.96	1.5 mm steel	Fixed	11.4	-	~0.005
E	0.96 x 0.96	0.6 *mm* steel +0.5 *mm* polymer +0.6 *mm* steel	Fixed	10.9	-	0.25

The loss factors listed are averages of loss factors for all frequency bands of interest.

MECHANICALLY EXCITED PANEL

Figure 1 compares the sound radiation efficiency of panels with and without the damping treatment, when structural excitation is applied. The increase of the radiation efficiency is due to the decrease of the vibration velocity level while the radiated sound is roughly unchanged. The adding of the constrained damping layer has increased the average loss factor by a factor of four for the sandwich panel, and by a factor of ten for the steel panel. As a consequence, the radiation efficiency is increased by about 9 dB for the sandwich panel, and by about 12 dB for the steel panel. For all of the cases tested, the frequency range is far below the critical frequency (~ 10 kHz for the sandwich panel and ~ 12 kHz for the steel panel).

Increased bending stiffness when constrained layer is applied can be one reason for the increase of the radiation efficiency, since the critical frequency is reduced. It is estimated, from sandwich theory and from formula for radiation efficiency [5-6], that the increase of the radiation efficiency due to the increase of the bending stiffness is about 1.5 – 3 dB for the steel panel, and is less than 1.5 dB for the sandwich panel for the frequency ranges concerned. This increase is far below the values obtained from the measurements.

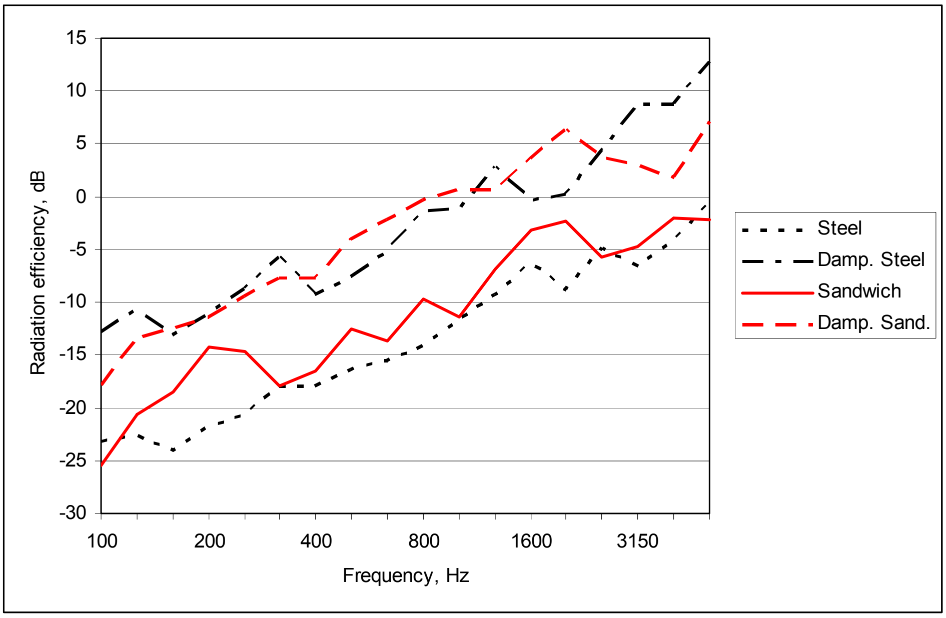

Figure 1 Radiation efficiency when mechanical excitation is applied (frame excitation) Panel mounted on a frame

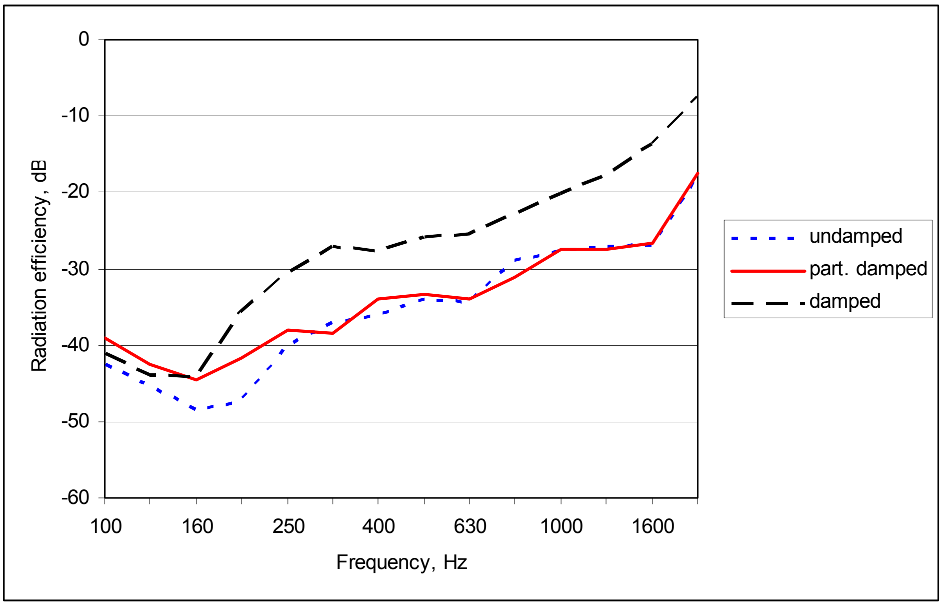

Figure 2 Comparison of radiation efficiency for a freely-hanging sandwich panel

The radiation efficiency of the same sandwich panel (panel B) has also been measured when it is freely-hanging, with the results shown in Figure 2. The difference between the radiation efficiency of the damped and un-damped panel is similar to that of the baffled structure. The partially-damped panel, however, reveals roughly same radiation efficiency as that of the un-damped panel, except for the frequencies below about 200 Hz. This is because that the excitation point is at the un-damped half, for that we may get some hint from the next example. The measurement uncertainty at 100 and 125 Hz bands is relatively big, which partly explains why the radiation efficiencies for the three configurations are similar.

In order to understand the phenomenon more clearly, the measured sound power radiated from a heavily damped panel, panel C, is compared to the sound power radiated from flexural near-field around excitation point. For this reason, the panel is freely-hanging and the excitation is applied at the middle of the panel. Results are compared in Figure 3, where the contribution of the radiation from the near-field, for an infinite panel, is calculated from [2]

$$P = \frac{\rho c k^2 \widetilde{F}^2}{2\pi\omega^2 m''^2} \tag{2}$$

where F is the applied point force and m'' is the surface density of the panel. The applied force is registered by using a force transducer at the excitation point.

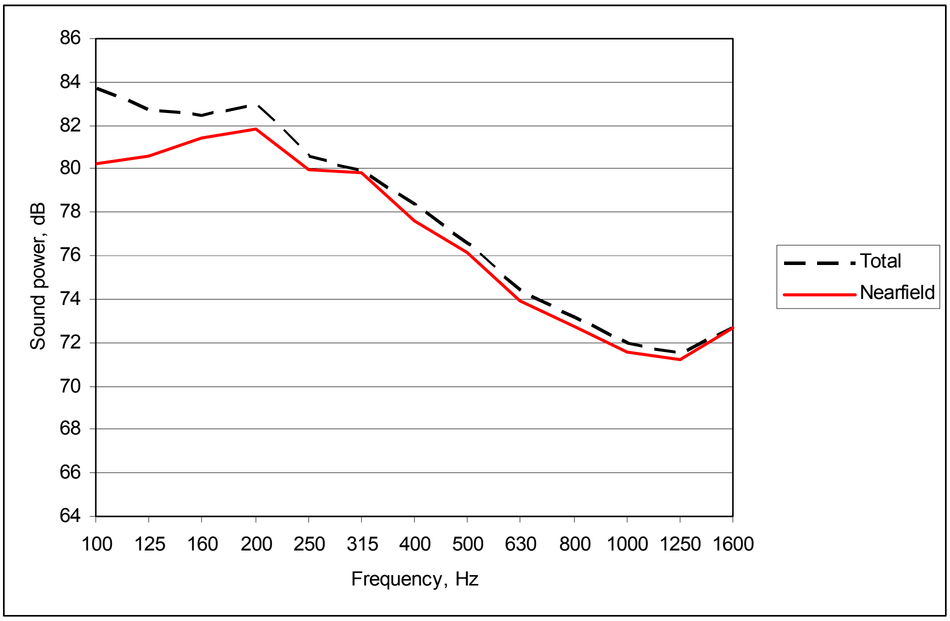

Figure 3 Measured sound power radiated from a sandwich panel excited at the centre compared with the calculated value from near field radiation of an infinite panel

For the panel concerned (loss factor of about 0.2), contribution from the near-field around the excitation point is very close to the total sound power when the frequency is higher than about 200 Hz. At low frequencies when the wavelength of bending wave is relatively long, the edge effect is relatively important, the near-field radiation and the edge radiation are comparable. At high frequencies when the wavelength of the bending wave is short, the edge radiation is much smaller compared to the near-field radiation, and the total sound power is determined by the sound power radiated from the area near the point source. This may explain the results in Figure 2, where partially damped panel has roughly the same radiation efficiency as the bare panel at relatively high frequencies, since the excitation is at the un-damped half of the panel.

When the excitation is applied at the frame, formula (2) may underestimate the near-field radiation, since the "near-field" in this case is much bigger and radiates more noise. It seems that for a heavily damped panel, excitation at a large area (with the same total force) might result in more noise radiation.

ACOUSTICALLY EXCITED PANELS

There are more discussions on radiation efficiency of a mechanically excited structure than on an acoustically excited structure. Very often people assume that the radiation efficiency of an acoustically excited structure is independent of the structural damping, since there is no near-field radiation around the excitation points and since the radiation efficiency of an acoustically excited infinite panel is in principle independent of the bending stiffness. However, Figure 4 shows that the radiation efficiency of a steel panel is increased by about 6 dB when a constrained layer damping is added, when the structure is excited acoustically. For the sandwich panel (panel B), the difference even reaches 9 dB.

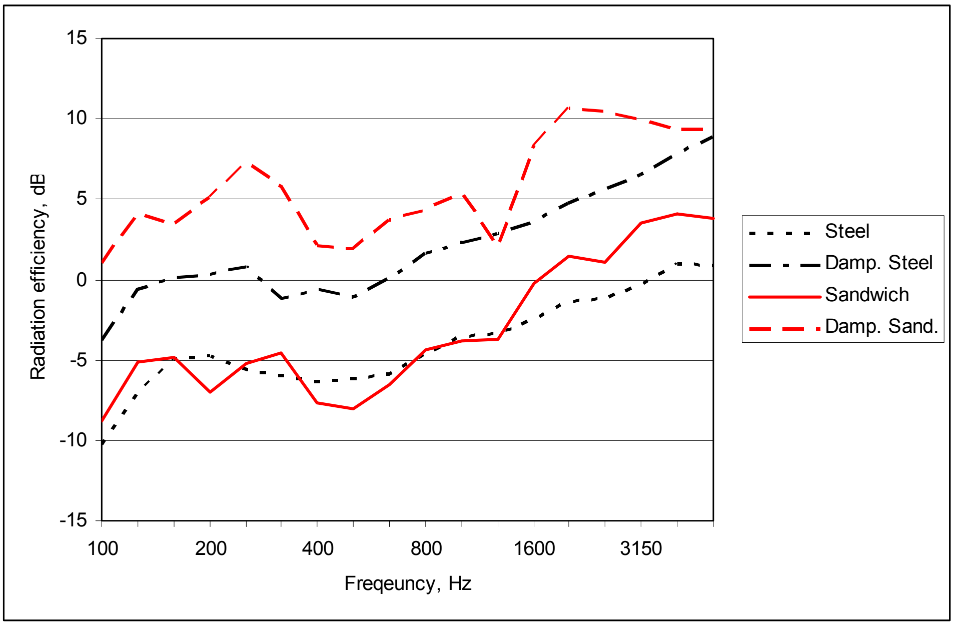

Figure 4 Radiation efficiency when acoustic excitation is applied
Panel mounted on a frame in between reverberation room and anechoic room

The measurements are made in exactly the same way as the measurement for sound transmission loss with intensity method, except that the averaged vibration is also measured here. It is noticed during measurement that the transmitted sound power for structures with or without damping treatment is almost identical, but the vibration velocity level is greatly reduced when the constrained damping layer is added.

The response of a finite plate to an acoustical excitation must have the same space distribution as that of the acoustic field and at the same time must fulfil the boundary conditions. As a result, it is a combination of forced vibration and resonant vibration. When damping is added, the peaks of resonant frequencies will be reduced greatly, so will the averaged vibration velocity level, while the responses far away from the resonant peaks will be little affected. However, for sound transmission or radiation, the main contributions are from the modes outside of the resonant region when it is lower than the critical frequency (forced transmission) [2-8]. Thus, we get much reduced vibration velocity level and roughly unchanged sound radiation for frequencies below the critical frequency. Radiation efficiency is then greatly increased. Figure 5 shows the sound transmission loss for panels D and E. They have roughly same surface density but with different structures and different loss factors. When it is lower than the critical frequencies of both panels, as in the case of the measurements, they show in principle identical sound transmission loss.

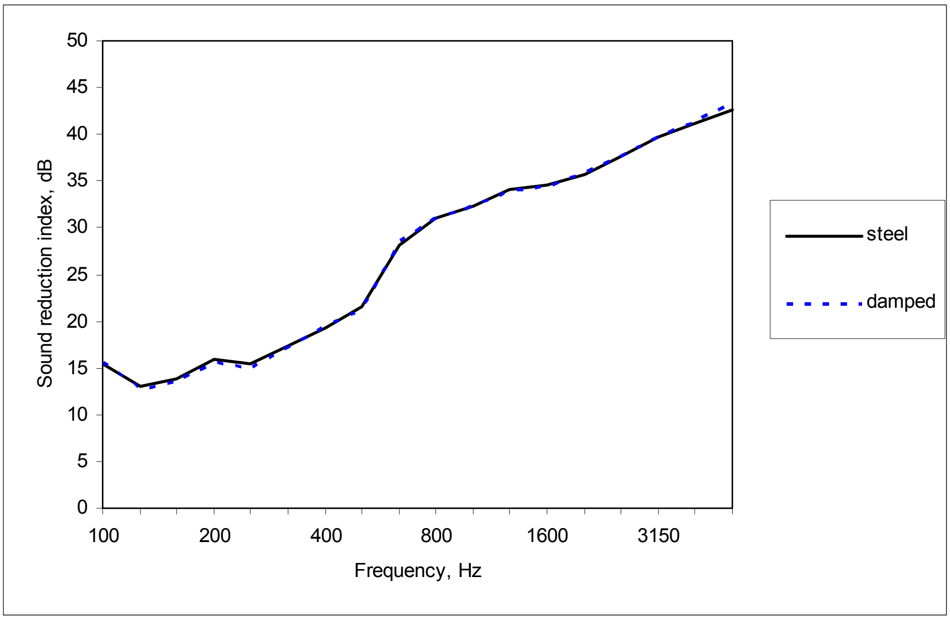

Figure 5 Comparison of sound reduction index of a damped panel and a steel panel with similar surface density

Successful example of damping treatment for sound transmission can also be found in literature [9], where the target product is an irregularly shaped oil pan with maximum outer dimension of 0.1 m x 0.27 m x 0.38 m. Because of the irregularity, the structure is strengthened, and the first resonant frequency is rather high. The contributions from modes outside the resonant frequency band are relatively small for the frequency ranges of interest. The increase of the sound radiation efficiency is not that big and it does not totally cancel the effect of reduced vibration. Accordingly, the radiated sound is also reduced significantly, though less than that of vibration velocity. Reference [2-3] has pointed out that the radiated sound power from a structure may be reduced significantly by added damping only when the size of the panel is less than about 5 bending wavelengths of the panel at the critical frequency. The situation in reference [9] is similar to that.

CONCLUSIONS

For mechanically excited panel, the radiation from the near-field around the excitation point is the main reason for the high radiation efficiency when the structural damping is high. The influence of the increased bending stiffness is less important. For a heavily damped panel, excitation at a large area may produce more noise.

For acoustically excited panel, the main reason for the increased radiation efficiency is the forced vibration, which is also the most important part of the sound transmission of a finite structure. For a uniformly damped structure, increased damping may reduce sound transmission greatly only when the size of the structure is much smaller than the bending wavelength at the critical frequency.

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