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A STUDY ON FLOOR IMPACT NOISE

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Abstract

In this paper, floor impact is studied by using 1-D wave model and predicted insertion loss is compared to the measurements done in the mock-up. A mock-up is built by using 6t steel plate, and two identical cabins are made where 25t or 50t panel is used to construct wall and ceiling inside the steel structure. Various floating floor structures are studied, in which mineral wool thickness, height, and stiffness changes are investigated. It is shown that the wave model and measurements are in good agreements in general, although there occur significant discrepancies in the low frequency range below 200 Hz.

INTRODUCTION

Floor impact noise is important not only in multi-dwelling buildings, but also in ships such as cruise ships, military ships, and ocean structures. The most common way to reduce the floor impact noise is to use floating floor, whose basic idea is vibration isolation, in which impact absorbing materials such as rubber or mineral wool are inserted between floor and upper surface board.

Most works on reduction of floor impact noise are concerned with measurements, while theoretical works on performance of floating floor are very rare. Johansson [1] studied low-frequency impact sound insulation of a light weight wooden joist floor, where he studied the effect of increasing the rigidity of joists and boards. Owaki *et al*. [2] considered floor impact sound in multiple-dwelling buildings, in which they measured the effect of various wooden floor coverings in actual buildings. Davern [3] carried out measurements of impact noise on two timber floors with vinyl floor coverings on resilient underlays. Ruthforth *et al*. [4] reported an investigation of impact sound insulation and viscoelastic properties of underlay manufactured from recycled carpet waste. They invented a small impact test rig whose sample size is 400 mm x 144 mm, instead of using full size laboratory test.

In this paper, we study the prediction method for performance of the floating floor, where mineral wool with thickness 25 mm to 100 mm is used for impact absorbing material. We compare predictions to the measurements, for which we built a mock-up for simulating cabins in cruise ships. The wall and deck are 6 mm steel plates and two identical cabins were constructed inside the lower space by using 25 mm panel, where wall and ceiling panels are separated from the steel plate by 300 mm and 1000 mm respectively.

THEORETICAL MODEL

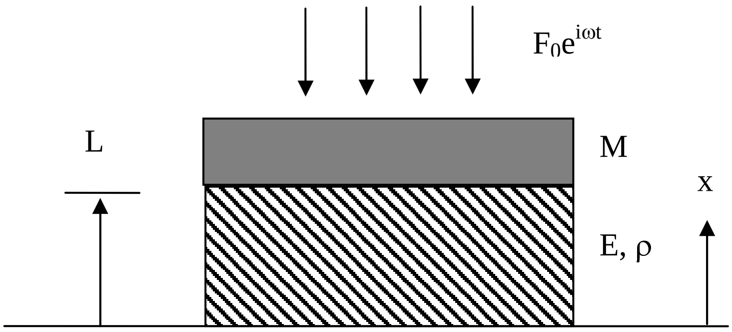

Figure 1 1-D wave model

We model the mineral wool and upper plate as a one-dimensional wave model as shown in Fig. 1. The mass of the upper plate per unit area is *M*, Young's modulus and density of the mineral wool are *E*, ρ respectively, and height is L. If *u(x,t)* is the displacement of the mineral wool, the equation of motion is [5]

$$\rho \frac{\partial^2 u}{\partial t^2} - E \frac{\partial^2 u}{\partial x^2} = 0 \ . \qquad (1)$$

We assume harmonic motion $u = Ue^{i\omega t}$. Eq. (1) becomes

$$\frac{d^2U}{dx^2} + (\omega / a)^2 U = 0 \, , \qquad (2)$$

in which $a^2 = E / \rho$.

When impact force $F_0 e^{i\omega t}$ is applied to the upper plate, the boundary conditions are

$$\text{at } x = 0: \ U = 0 \, , \qquad \text{at } x = L: \ E \frac{dU}{dx} = F_0 + \omega^2 MU$$

The solution of Eq. (1) satisfying boundary conditions is given by

$$U = \frac{aF_0}{E\omega} = \frac{\sin(\omega x / a)}{\cos(\omega L / a) - (\omega M / a)\sin(\omega L / a)} \qquad (3)$$

The ratio of forces $F_{x=0} / F_0$ is the transmissibility given by

$$\tau_{WAVE} = \frac{F_{x=0}}{F_0} = \frac{1}{\cos(\omega L / a) - (\omega M / a)\sin(\omega L / a)} \qquad (4)$$

We define the insertion loss IL_{WAVE} as

$$IL_{WAVE} = -20\log\tau_{WAVE} \qquad (5)$$

We consider following two cases: bare steel deck and deck + floating floor (mineral wool + upper plate) as shown in Fig. 2

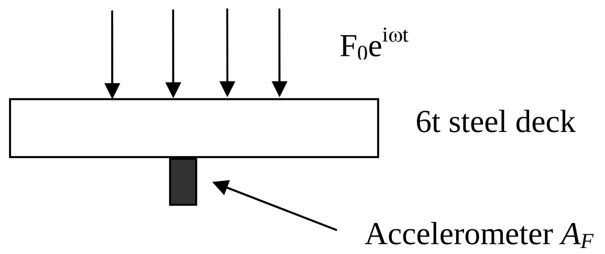

Figure 2a Bare steel deck (6 mm)

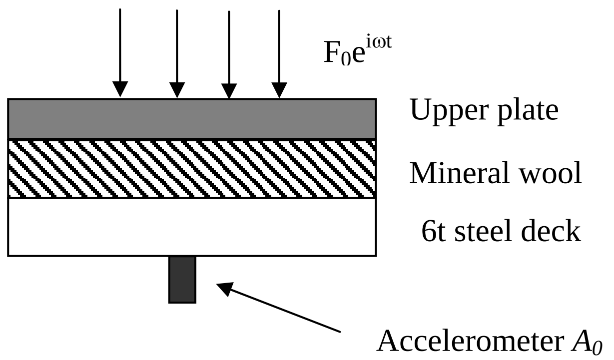

Figure 2b Bare steel deck (6 mm) + floating floor

We measured the acceleration of the deck, where accelerometer is located beneath the tapping machine. We can assume that the ratio of acceleration A_F / A_0 may be equal to the ratio of the forces $F_{x=0} / F_0$ provided that the tapping machine applies the same force on the deck (Fig. 2a) and on the upper plate (Fig. 2b).

$$\frac{F_{x=0}}{F_0} = \frac{A_F}{A_0}$$

in which A_F and A_0 represent acceleration with and without the floating floor.

If mineral wool is sufficiently soft and mass is smaller than that of the upper plate, we can even more simplify the model as 1-D mass spring system. In this case, the transmissibility and insertion loss are given by

$$\tau_{SPRING} = \left[\frac{1+\eta^2 r^2}{(1-r^2)^2 + \eta^2 r^2} \right]^{1/2} \qquad (6)$$

where η is loss factor of the mineral wool, r the ration of frequency and natural frequency f_n.

$$r = f / f_n, \qquad f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M_e}}$$

in which k is spring constant and M_e is the effective mass. Since mass of the mineral wool is comparable to that of upper plate, we include mass of mineral wool in computing effective mass M_e as

$$M_e = M + \rho L \qquad (7)$$

The spring constant k is $k = E / L$. We assumed the loss factor η=0.05.

COMPARISON OF PREDICTIONS AND MEASUREMENTS

In Fig. 3-5, we compared predictions from Eqs. (4) and (6) to the measurements for the case: deck + mineral wool (MW) + 3.2 mm steel plate. The density of the mineral wool is 140 kg/m^3 in Fig. 3 and 4, while 240 kg/ m^3 in Fig. 5. Fiber direction of the mineral wool is horizontal in Fig. 3 and 5, while vertical in Fig. 4. We measured Young’s modulus of the mineral wool with density of 140 kg/m^3 for horizontal and vertical fiber direction as

Horizontal : $E = 1.4 \times 10^5$ N/m^2, Vertical : $E = 1.7 \times 10^6$ N/m^2.

while Young’s modulus with density of 240 kg/m^3 for horizontal fiber direction is:

$E = 9.2 \times 10^5$ N/m^2.

In Figs. 3-5, the upper plate consists of many subplates where size of the single subplate is 420 mm x 800 mm. In Eqs. (3)-(7), it was assumed that whole area is under the impact force, which is not true in reality. Since only part of the subplate is excited by the tapping machine, we used *4M* instead of *M* in Eqs. (3)-(7). Figs. 3-5 shows that the wave model is in reasonable agreements with measurements except low frequency ranges below 200 Hz, while Eq. (6) for mass-spring model shows large discrepancies.

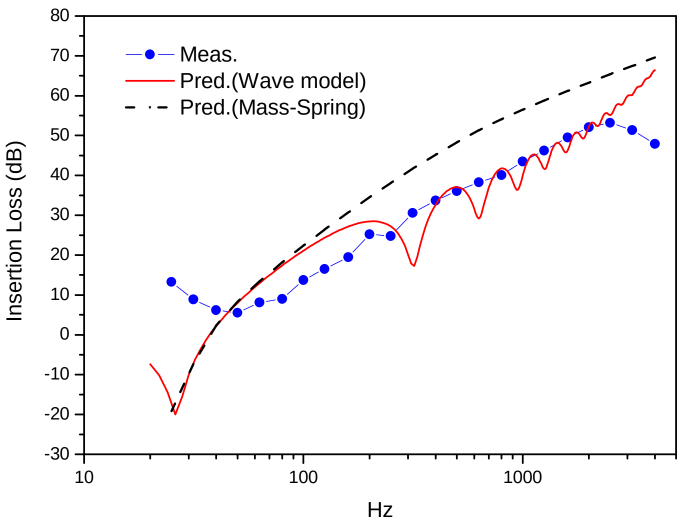

Figure 3 Insertion loss for deck + 50 mm MW (140H)+ 3.2 mm plate.

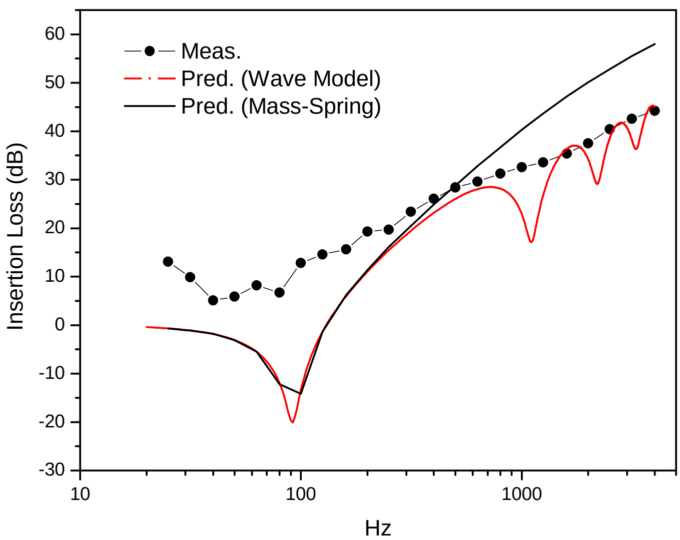

Figure 4 Insertion loss for deck + 50 mm MW (140V) + 3.2 mm plate.

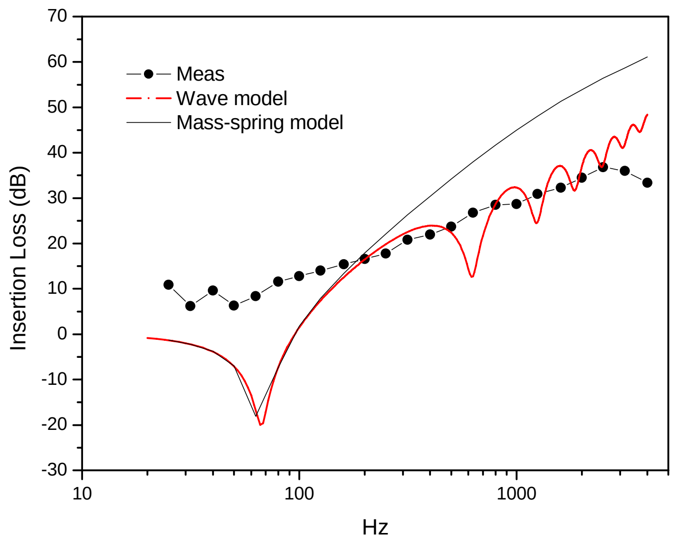

Figure 5 Insertion loss for deck + 50 mm MW (240H) + 3.2 mm plate.

In Figs. 6-8, we compared predictions to the measurements for the case: deck + mineral wool (MW) + 1.6 mm steel plate. The density of the mineral wool is 140 kg/m^3 and fiber direction is horizontal. The size of the single subplate is 450 mm x 450mm, and we assumed *2M* in Eqs. (3)-(7). Figs. 6-8 shows that mass-spring model is in good agreements with measurements except low frequency ranges below 200 Hz, while wave model severely underestimate the insertion loss.

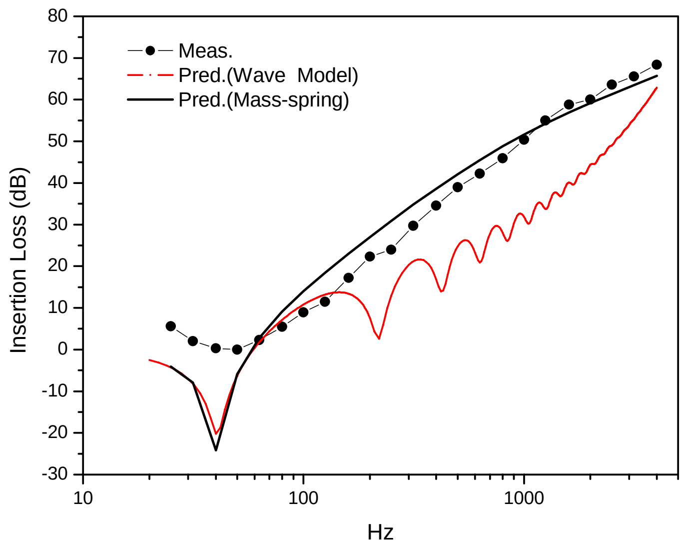

Figure 6 Insertion loss for deck + 75 mm MW (140H) + 1.6 mm plate.

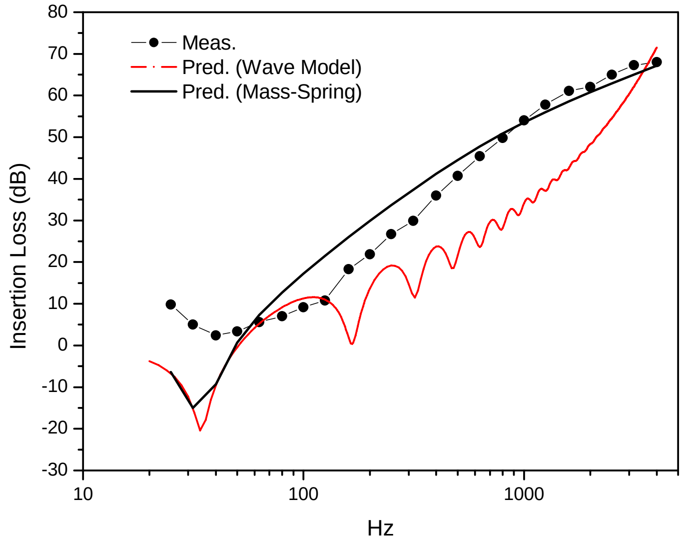

Figure 7 Insertion loss for deck +100 mm MW (140H) + 1.6 mm plate.

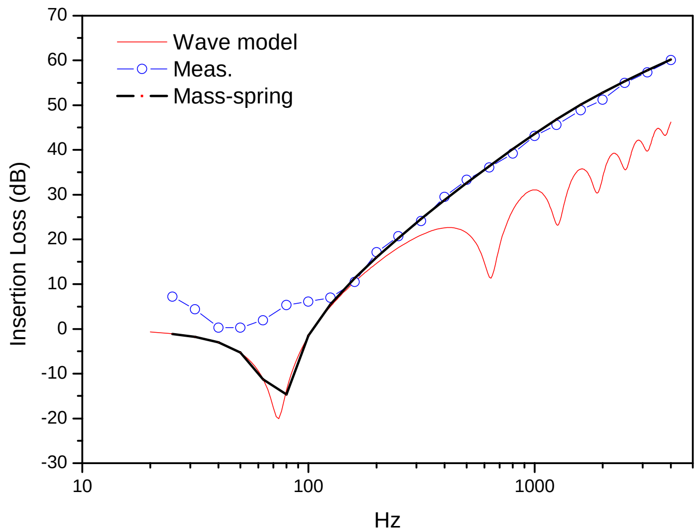

Figure 8 Insertion loss for deck +25 mm MW (140H) + 1.6 mm plate.

CONCLUSIONS

Comparisons of prediction by wave model and mass-spring model with measurements showed that when upper plate is large and heavy, wave model shows agreement with measurements, while for small and light upper plate, mass-spring model is more accurate. However, both predictions showed severe discrepancies from measurements in low frequency ranges below 100 Hz to 200 Hz. In predictions, we assumed that floor is infinitely rigid. However, 6 mm steel deck has elastic behavior, which cannot be neglected. It is expected that inclusion of elastic behavior of deck will improve the accuracy of the predictions.

ACKNOWLEDGEMENTS

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