Experiences with characterising simple sources of structure-borne sound by the two plate method

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Introduction

The assessment of the ability of a vibratory source to inject sound power into a receiver is a major task in different fields of acoustics. Recently, a proposal has been made to characterise structure-borne sound sources by two properties, an activity and a mobility quantity [1]. They are determined by connecting the source under test to two different receiving plates with very different point mobilities. From the different sound powers injected into the different receiving plates, source quantities can be calculated.

At PTB, the method was used to determine the source characteristics of an electrodynamic shaker. This source has the advantage that it can be modelled analytically. Additionally, the power input into the receiving plates can be measured by an impedance head. Furthermore, impulse signals can be used with shakers as well as stationary signals which gave the opportunity of investigating whether the two plate method works in all cases. Afterwards, the source properties of tapping machines and modified tapping machines were determined.

The two plate method

The source under test is operated on two different receiver plates, whereby one of the plates must have a mobility Y_p which is significantly higher than the source mobility Y_s and the other one a significantly lower mobility. In both cases, the average velocity v_p on the plate is measured. Furthermore, the total loss factor η and the mass m of the plates have to be known, and the plates have to meet certain requirements regarding their modal characteristics. By measuring on the high mobility plate, the free velocity v_f of the source can be calculated by

$$v_f = \sqrt{\frac{P_{\text{p,high}}}{\text{Re}\left\{1/Y^*_{\text{p,high}}\right\}}} \quad (1),$$

where P denotes the sound power on the plate calculated from the average velocity and the plate parameters. The measurement on the low mobility plate provides the blocked force F_b:

$$F_b = \sqrt{\frac{P_{\text{p,low}}}{\text{Re}\left\{Y_{\text{p,low}}\right\}}} \quad (2).$$

Finally, the magnitude of the source mobility is

$$|Y_S| = \frac{v_f}{F_b} \quad (3).$$

The source mobility has, of course, to be estimated beforehand in order to select proper receiver plates. A prediction of the power P_{pred} emitted by the source into another receiving structure is then obtained from

$$P_{\text{pred}} = |v_f|^2 \frac{\text{Re}\left\{Y_{\text{pred}}\right\}}{|Y_S|^2 + |Y_{\text{pred}}|^2}\frac{1}{\Delta_1} = |v_f|^2 \frac{|Y_{\text{pred}}|}{|Y_S|^2 + |Y_{\text{pred}}|^2}\frac{1}{\Delta_2} \quad (4),$$

where Y_{pred} is the input mobility of the receiving structure and Δ_1 and Δ_2 are factors describing the quality of the prediction. Ideally, they should assume the value one. Eqs. (1) - (4) are valid for sources with one contact point with one degree of freedom. Nevertheless, they can be adapted to apply them to multiple contact sources with multiple degrees of freedom [1].

Receiving plates

Five different receiving plates were used for the investigations. Table 1 gives an overview of the plates used for the method verification, and Table 2 describes the plates used with the tapping machines. The steel plate is the same in both cases. It is completely embedded on an elastic foam layer. The concrete plate is the floor of the receiving room of a test facility for the determination of impact noise. It is connected to other structures at the edges. The high mobility plates are supported by small pieces of elastic foam. The loss factor of all the plates was determined by structure-borne reverberation times (Figure 1). It is high enough to ensure a sufficient modal overlap and low enough to enable far field velocity measurements outside the reverberation radius.

Table 1: Receiving plates used for verification

Usage	High *Y*	Low *Y*	Prediction
Material	Acrylic glass	Steel	Chipboard
S (m²)	0.5	1.45	1.45
d (mm)	2	10	8
m (kg)	1.3	114	8.2
E (GPa)	3.4	210	4.6
Y_{inf} (m/(Ns))	$5\cdot10^{-2}$	$2\cdot10^{-4}$	$4\cdot10^{-3}$

Table 2: Receiving plates used with tapping machine

Usage	High *Y*	Low *Y*	Prediction
Material	Plywood	Concrete	Steel
S (m²)	2.1	16.7	1.45
d (mm)	5	160	10
m (kg)	4.7	6400	114
E (GPa)	12	34	210
Y_{inf} (m/(Ns))	$7\cdot10^{-3}$	$2\cdot10^{-6}$	$2\cdot10^{-4}$

The receiving plates have to fulfil certain boundary conditions in order to be applicable for the characterisation and prediction procedures. One way to check their suitability is to compare the injected sound power measured by an impedance head to the sound power derived from the average velocity on the plate and the plate parameters.

Figure 2 shows the results for the receiver plates achieved with the experimental setup shown in Figure 3. The deviations are mostly between ±5 dB except for the low frequencies. For the further data processing, observed sound power deviations were used as a calibration factor.

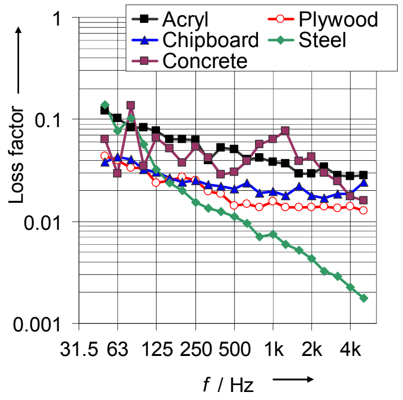

Figure 1: Loss factors of the receiving plates

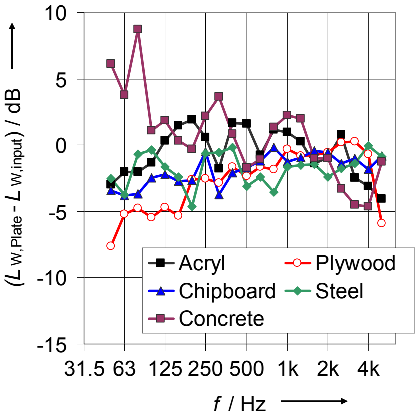

Figure 2: Difference between plate power and injected power

Verification of the method

Experimental setup

Figure 3 shows the experimental setup used for the verification. The shaker is coupled to the respective receiving plate by an impedance head. Thus, the injected mechanical power and the point mobility were directly measured. Furthermore, the voltage and current fed into the shaker were simultaneously measured to achieve a constant input power during the measurements. A complex spectral analysis in narrow bands was used for the aforementioned measurements. Real and imaginary parts of the quantities were separately summed up to one-third octave band values afterwards. All the further calculations were carried out in one-third octave bands. The average velocity on the receiving plates was measured by several accelerometers (5 or 6 positions) in one-third octave bands. The acceleration signal of the impedance head and the accelerometers was converted to velocity using complex integration in frequency domain. In a second step, the source was modified by inserting a rubber spring between the shaker and the plate. The shaker was usually operated using stationary noise signals, but in some cases impulse signals were also employed in order to verify the method for the case of non-stationary sources.

The three receiving plates described in Table 1 were used. The experimental setup was completely modelled using the lumped parameter model shown in Figure 4. A detailed description of the individual parameters will not be given here, but can be found in [4].

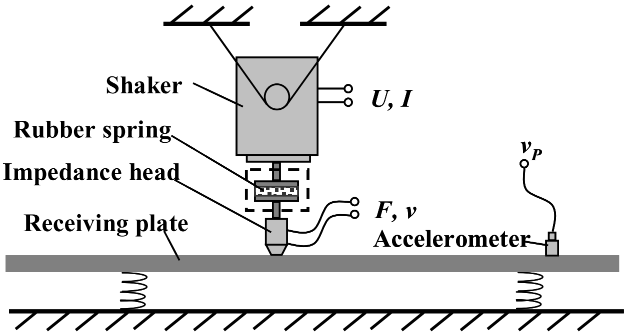

Figure 3: Experimental setup for verification

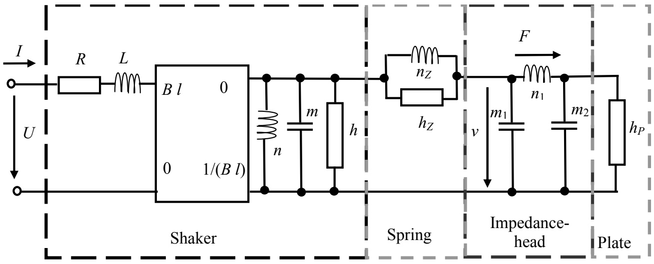

Figure 4: Lumped parameter model of the setup

Results

Figure 5 shows measured and calculated free velocities and blocked forces for the modified and unmodified shaker with stationary excitations. The deviations are a couple of dB at medium frequencies and somewhat higher in the high and low frequency ranges. In spite of the observed discrepancies, the overall tendency of the measured values is reflected by the lumped parameter model quite well.

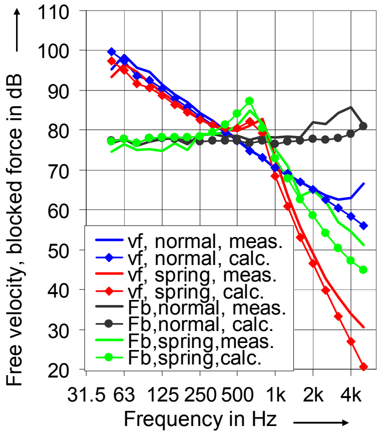

Figure 5: Measured and calculated free velocities (vf) in dB re $5 \cdot 10^{-8}$ m/s and blocked forces (Fb) in dB re $2 \cdot 10^{-5}$ N for the shaker with and without rubber spring

The difference between the measured and calculated source mobilities is between ±3 dB (Figure 6) for all three cases. So, the expected source mobilities are calculated well by the two plate method.

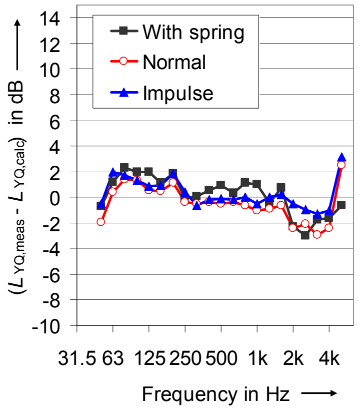

Figure 6: Difference between measured and calculated magnitudes of the source mobilities

With the necessary source characteristics at hand, the prediction (eq. (4)) was carried out for the chipboard. Figure 7 shows the difference between the predicted and the measured sound power level. Both approximations, Δ_1 and Δ_2, were used. Δ_1 takes into account the fact that only the magnitude of the source mobility is known, and not its complex value. With Δ_2 comes a further simplification, assuming that also only the magnitude of the receiver mobility is known. Figure 7 demonstrates that a prediction of the emitted sound power is possible on the basis of the source properties determined. But even under laboratory conditions, observed deviations between predicted and measured sound powers reach values of 2 dB and more.

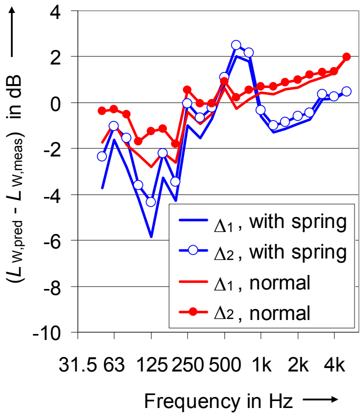

Figure 7: Difference between predicted and measured sound powers

Characterisation of ISO tapping machines

Tapping machines and modifications

In a first approach to characterise an unknown source, four tapping machines which meet the requirements given in [2] were selected as test objects. The chosen types were models 211 and 277 by Norsonic and 3204 and 3207 by Brüel & Kjaer. The tapping machines were operated as usual, but also in a modified way as described in [3]. In this case an elastic layer is placed under the tapping machine in order to give a better match of mobility in comparison to walking persons. Two different layers were used, one consisting of a 12.5 mm thick stripe of PEUR foam with the material properties s'=70 MN/m³ ± 10 % and η ≥ 0.2 and the other one made of cork and rubber granulate with a thickness of 16 mm (s'=46 MN/m³). Hard plastic washers were put under the feet of the modified tapping machines in order to achieve proper falling heights of the hammers.

Measurement setup

For source characterisation and the verification of the prediction method, the average velocity produced by the tapping machines was measured on the respective receiver plates. On the plywood and steel plate, one position for the tapping machine and five accelerometers were used. On the concrete plate, three source positions and six receiver positions were used due to the larger size of the plate. The signals from the accelerometers were recorded simultaneously using a multi-channel measuring system. The averaging time for each measurement was 32 s.

Results

The sound powers injected by the four different unmodified tapping machines are almost identical on all three receiving plates (Figure 8). Only at the very high frequencies, deviations of up to 3 dB occur. A modification with the granulate reduces the injected sound powers at high frequencies considerably, whereas the sound power remains more or less unchanged at the low frequencies. The reduction is less prominent for the plywood due to the fact that the input mobility is not changed very much by the presence of the granulate. It is furthermore observed that the four tapping machines show a different sound power emission if the granulate strip is put under the hammers. The reason is probably the sound power transmission by the feet. The results for the PEUR foam (not displayed) are similar to the results of the granulate.

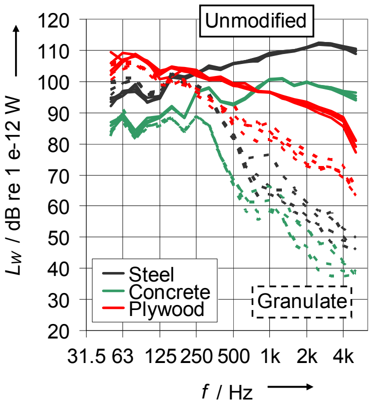

Figure 8: Sound powers of the unmodified tapping machines and modified tapping machines (granulate - dashed lines) on different receiving plates

Blocked forces (Figure 9) and free velocities are calculated from the sound powers and the input mobilities of the plates (eqs. (1) and (2)). These values are almost identical for the four unmodified tapping machines, whereas a 15 dB scatter is observed for the modified tapping machines.

The source mobility was afterwards calculated from the blocked forces and free velocities. The unmodified tapping machines show a mass-like mobility (Figure 10), which corresponds to a mass of about 700 g. Since the hammers have a mass of 500 g, this result is of an acceptable accuracy, though not perfect. The distance to the receiver mobility was too small at the very low frequencies. Here, the supporting foam elements decrease the mobility of the plywood. Therefore, the mobility in this frequency range deviates considerably from the mass-mobility. The deviation at the very high frequencies can be explained by different stiffnesses of the contact points when hammering on the concrete and on the plywood.

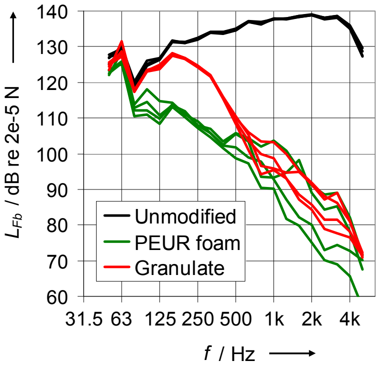

Figure 9: Blocked forces for different tapping machines

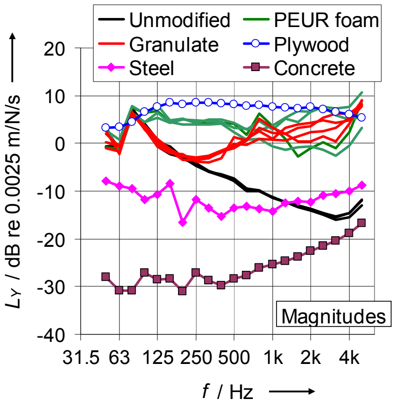

Figure 10: Source and receiver mobilities for unmodified and modified tapping machines

On the base of the source properties and the receiver mobility, the sound power emitted into the steel plate was predicted. It turns out that the predicted sound powers are on average about 3 dB larger than the measured sound powers (Figure 11). One reason is probably that the prediction formula (eq. (4)) works well for real receiver mobilities. The steel plate has a complex mobility with a large imaginary part due to the low damping of the steel plate (Figure 1). That is why the sound power levels are overpredicted for the steel plate whereas the prediction works better for the chipboard (Figure 7) due to the higher damping.

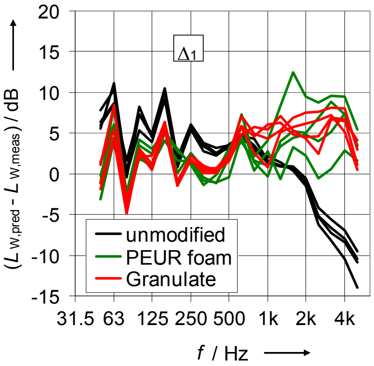

Figure 11: Difference between predicted and measured sound power levels on the steel plate

Conclusions

The two plate method is a very effective tool for the characterisation of structure-borne sound sources. It works for stationary and non-stationary sources and delivers the main quantities which are necessary for a prediction of the sound power output of structure-borne sound sources. The receiving plates have to be carefully designed. In particular, the trade-off between high mobilities on the one hand and the mechanical support of the sources on the other hand turned out to be difficult. The accuracy of the predictions still has to be improved, e.g. by excluding receivers where the prediction formula is not valid. This method will be used at PTB in the near future for the characterisation of walking persons.

Acknowledgements

The fruitful discussions with Werner Scholl are gratefully acknowledged.

References

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[2] ISO 140-6:1998 Acoustics -- Measurement of sound insulation in buildings and of building elements -- Part 6: Laboratory measurements of impact sound insulation of floors

[3] ISO 140-11:2005 Acoustics -- Measurement of sound insulation in buildings and of building elements -- Part 11: Laboratory measurements of the reduction of transmitted impact sound by floor coverings on lightweight reference floors

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