Exercise Report
(Sound and Vibration)

Exercise C: Measurement of Sound Radiation from Panels of a Box Structure

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1 Introduction

The aim of this laboratory exercise is to analyze the relation between flexural vibrations and radiated sound power from a vibro-acoustic complex structure. This is done by means of the radiation factor, which is the parameter that describes their relation.

For this purpose, the radiation factor of two of the six plates that conform a box structure are measured. In addition, the results are compared to the ones estimated with an analytical approximation that is derived from a modal-average approach.
2 Theory

The following theory is stated according to [Mørkholt, 2006] and [Jacobsen, 2006].

2.1 Temporal and spatial average considerations of vibrating panels

Temporally and spatially averaged mean-square values are often of interest when dealing with complicated structures. In a structure made of panels, each of them with a normal velocity $v(x, z, t)$, the stationary flexural vibration is often characterized by the temporally and spatially averaged mean-square velocity:

$$\langle v^2 \rangle = \frac{1}{S} \int_S v^2(x, z, t) \, dS \approx \frac{1}{m} \sum_{j=1}^{m} v_j^2,$$

(1)

where the approximation is the practical implementation for a substructure with constant thickness, and where $m$ is the number of discrete measuring points that are chosen randomly across the surface of concern.

2.2 Relation between vibration and radiated sound

The relation between the vibration of a surface and its radiated sound power is given by the radiation factor $\sigma$ (or radiation efficiency). This parameter is defined as:

$$\sigma = \frac{P_a}{\rho_0 c_0 S \langle v^2 \rangle},$$

(2)

where $P_a$ is the sound power radiated from the surface, $\rho_0 c_0$ is the specific acoustic impedance of the surrounding medium, $S$ is the area, and $\langle v^2 \rangle$ is the temporal and spatial mean-square normal velocity of the radiating surface.

This quantity is the ratio between the sound power radiated by a surface and the power that would be radiated by a uniformly vibrating piston with the same area and mean-square velocity as the surface. Physically, it expresses in which extent the vibration of the surface results in dissipation of sound power into the surrounding medium.

In the case of a complex structure composed of different plates, it can be assumed that each surface radiates independently of the others if the wavelength of the radiated sound is smaller than twice the length of the surface ($\lambda < 2L$). Hence, the total sound power that is radiated from the structure ($P_{a,total}$) is given by the summation of sound power radiated from each surface ($P_{a,i}$). This is,

$$P_{a,total} \approx \sum_{i=1}^{n} P_{a,i},$$

(3)

where $n$ is the number of surfaces that constitute the structure. As each surface presents its own radiation factor ($\sigma_i$) determined by equation (2), equation (3) can be written as

$$P_{a,total} \approx \rho_0 c_0 \sum_{i=1}^{n} S_i \langle v_i^2 \rangle \sigma_i.$$

(4)
The foregoing considerations have shown that it is possible to determine the radiation factor of a plate by measuring its radiated sound power (see equation 2). Besides, approximate analytical results for the overall radiation factor of plates have been derived by adopting a modal-average approach. The resulting formulas for the frequency-average radiation factor are:

\[
\sigma = \begin{cases} 
\frac{\lambda_c^2}{S} \left( 2g_1(f) + \frac{U}{\lambda_c} g_2(f) \right) & f < f_c \\
\sqrt{a/\lambda_c} + \sqrt{b/\lambda_c} & f \approx f_c \\
\frac{1}{\sqrt{1 - f_c/f}} & f > f_c,
\end{cases}
\] (5)

where \(\lambda_c\) is the acoustic wavelength at the critical frequency \(f_c\) of the plate, and \(U\) is the plate perimeter. The latter can be calculated by \(U = 2(a + b)\), where \(a\) and \(b\) are the dimensions of the plate. The critical frequency \(f_c\) is given by

\[
f_c = \frac{c^2}{\pi h} \sqrt{\frac{3\rho(1 - \nu^2)}{E}},
\] (6)

where \(c\) is the speed of sound in air, \(\nu\) is the Poisson ratio, and \(E\) is the Young’s modulus, [Rindel, 2008]. The critical wave length \(\lambda_c\) is approximately

\[
\lambda_c \approx \sqrt{1.8 \cdot c_{11} \cdot \frac{h}{f_c}}
\] (7)

where \(c_{11} = \sqrt{\frac{E}{\pi(1-\nu^2)}}\), which is the longitudinal wave speed in a flat, homogeneous plate, [Ohlrich, 2005].

The functions \(g_1(f)\) and \(g_2(f)\) are given by:

\[
g_1(f) = \begin{cases} 
\frac{4}{\pi^4} (1 - 2\gamma^2) \frac{1}{\gamma \sqrt{1 - \gamma^2}} & f < f_c/2 \\
0 & f_c/2 < f < f_c,
\end{cases}
\] (8a)

\[
g_2(f) = \frac{1}{4\pi^2} \left( 1 - \gamma^2 \right) \ln \left( \frac{1 + \gamma}{1 - \gamma} \right) + 2\gamma
\] (8b)

where \(\gamma = \sqrt{f/f_c}\).

### 2.3 Sound power determination using sound intensity

The sound power radiated by a structure can be determined by measuring either the sound pressure in free field or the sound intensity in the acoustic field around the structure. On the one hand, measuring the sound power with the first method requires an anechoic room to provide the free field conditions. On the other hand, with a sound intensity probe it is possible to measure the sound power in any room (in situ). In the present exercise, the sound intensity method is of interest.
The instantaneous sound intensity, which is a vector, is the product of the sound pressure and the particle velocity,
\[ \vec{I}(t) = p(t)\vec{u}(t). \] (9)
It expresses the magnitude and the direction of the instantaneous flow of energy per unit area.

The conservation of sound energy follows
\[ \nabla \cdot \vec{I}(t) = -\frac{\partial w(t)}{\partial t}, \] (10)
where \( w(t) \) is the total instantaneous energy density (energy per unit volume) which is the sum of instantaneous potential and kinetic energy densities. This equation shows that the rate of the flow of sound energy diverging away from a point in a sound field is equal to the rate of decrease of the sound energy density at this position.

Using the Gauss theorem
\[ \int_V \nabla \cdot \vec{I}(t) dV = \int_S \vec{I}(t) \cdot d\vec{S}, \] (11)
one can express equation (11) combined with (10)
\[ \int_S \vec{I}(t) \cdot d\vec{S} = -\frac{\partial}{\partial t} \left( \int_V w(t) dV \right) = -\frac{\partial E}{\partial t}, \] (12)
which shows that the rate of change of the sound energy \( E \) within a closed surface is identical with the surface integral of the normal component of the instantaneous sound intensity.

In practice, the time-averaged sound intensity in a stationary sound field is of interest. This is expressed as
\[ \vec{I} = p(t)\vec{u}(t). \] (13)
When equations (10) and (12) are examined, it can be concluded that the time-average of the instantaneous net flow of sound energy out of a given closed surface is:

- **Zero**, if there is no generation or dissipation of sound power within the surface. Hence,
\[ \int_S \vec{I} \cdot d\vec{S} = 0, \] (14)
because \( \nabla \cdot \vec{I} = 0 \). The validity of this expression is irrespective of the presence of sources outside the surface.
- **The sound power** \( P_a \), if the surface encloses a steady state source that radiates this sound power. That is,
\[ \int_S \vec{I}(t) \cdot d\vec{S} = P_a, \] (15)
irrespective of the presence of other steady sources outside the surface and irrespective of the shape of the surface. This is the expression that allows the determination of the sound power of a source using sound intensity.
3 Measurement procedure

3.1 Device list

**Brüel & Kjaer PULSE Analyzer System Type 3560-B-130.** Data acquisition system with 5 inputs and one generator output.

**Brüel & Kjaer NEXUS Conditioning Amplifier Type 2692.** Four-channel conditioning amplifier.

**Brüel & Kjaer Calibrator Type 4294.** Portable electrodynamic vibration calibrator that produces a tone at 159.2 Hz with a RMS velocity of 10 mm/s, i.e. a velocity level of 140 dB re 10^{-9} m/s.

**Brüel & Kjaer Accelerometer Type 4393.** Piezoelectric charge accelerometer that weighs 2.4 g. Its charge sensitivity (@ 159.2 Hz) is 3.1 ± 2% pC/g.

**Brüel & Kjaer Type Mini-Shaker 4810.** Shaker used for vibration testing of small objects. Its maximum force is 7 or 10 N depending on the frequency which can be up to 18 kHz.

**Brüel & Kjaer Type 3599.** Sound intensity probe kit. The probe set includes remote control unit ZH-0632 and a pair of 1/2-inch sound intensity microphones type 4197.

**Amplifier.** Exciter preamplifier.

**Computer.** To process the data obtained from the PULSE system. The used software is MATLAB.

Two different magnitudes are measured on panels number 5 and 6 of the system under study: Mean-square velocity and intensity. The setup used for the mean-square velocity measurement is shown in figure 1. As can be seen, the system is driven with a shaker which is controlled with the analyzer. The measured signal from the accelerometer is amplified with the Nexus amplifier and sent to the analyzer. The latter computes the autospectrum of the velocity and afterwards this information is processed with MATLAB.

The setup used for the intensity measurements is illustrated in figure 2. The system is excited in the same way as in the previous measurement. In this case,
the measured magnitude is the intensity, hence the transducer used to measure the velocity is changed for an intensity probe kit, which is directly connected to the PULSE system.

3.2 Calibration

Before starting the mean-square velocity measurements, the calibration of the accelerometer must be performed. This is done by connecting the accelerometer to an electrodynamic vibration calibrator, which gives a reference velocity of 10 mm/s. Besides, the sensitivity of the accelerometer is entered in the corresponding menus of the Nexus amplifier and the PULSE system. The latter must use a flat top window when calculating the autospectrum of the velocity signal.

Finally, it is confirmed that the correct level of 140 dB re $10^{-9}$ m/s is achieved.

No preliminary calibration is required for the intensity measurements.

3.3 Procedure

A sketch of the box-shaped structure under test is shown in figure 3. It consists of a frame formed by the surfaces $S_1$ to $S_4$, which are made of 3 mm thick steel.
and are welded together. The fifth panel (having the area $S_5$) is made of aluminium material and has a thickness of 3 mm. The sixth panel (denoted by $S_6$) is a 5 mm thick panel made of polycarbonate (Lexan). These two panels are screwed to the previously described frame along all four edges. The areas of the box are summarized in table 1. The exciter (shaker) is placed inside the box, namely on panel 1. White noise is used as an excitation signal.

### Table 1: Areas of the box.

<table>
<thead>
<tr>
<th>Panel #</th>
<th>Area in m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 and 2</td>
<td>$S_1 = S_2 = 0.109$</td>
</tr>
<tr>
<td>3 and 4</td>
<td>$S_3 = S_4 = 0.176$</td>
</tr>
<tr>
<td>5 and 6</td>
<td>$S_5 = S_6 = 0.246$</td>
</tr>
</tbody>
</table>

#### 3.3.1 Measurements

The velocity response of panels number 5 and 6 are measured by means of a small accelerometer (2.4 g) in order to minimize the additional mass load on each of the panels. The accelerometer signal is integrated, which yields the velocity instead of the acceleration. As the mentioned white noise is an aperiodic signal, the Hanning window is used when averaging data. A frequency range from 0 to 10 kHz is chosen, using a frequency resolution of 2 Hz. Each recording is an averaging process over several window lengths. A measurement time around 20 s per measurement point ensures the sufficient reduction of uncorrelated components, e.g. uncorrelated excitation by airborne background noise.

In addition to this temporal averaging, a spatial averaging per panel is also carried out. For this purpose, the measurement is interrupted after the temporal average for one accelerometer position is determined. Then it is placed at a different location on the panel and the measurement continues. In this way, an average over seven positions per panel is recorded.

According to equation (9), a sound intensity measurement requires to know the sound pressure and the particle velocity at the same point. The method used by the intensity probe is based on the $p-p$ principle. This method uses two closely spaced and phase-matched pressure microphones and relies on a finite difference approximation to the sound intensity. Then the particle velocity is:

$$
\hat{u}_r(t) = - \int_{-\infty}^{t} \frac{p_B(\tau) - p_A(\tau)}{\rho \Delta r} d\tau
$$

where $p_A$ and $p_B$ are the sound pressure signals provided by the two microphones, $\Delta r$ is the distance between the microphones (12 mm), $\rho$ is the mass density of air and $\tau$ is a dummy time variable. Note that the hat above the variable denotes that the obtained value is an estimate of the real one. The sound pressure at the center of the probe is:

$$
\hat{p}(t) = \frac{1}{2} (p_A(t) + p_B(t))
$$

From the above equations, the time-averaged sound intensity component in the axial
direction is
\[ \hat{I}_r = \bar{p}(t)\bar{u}_r(t) = \frac{1}{2\rho \Delta r} (p_A(t) + p_B(t)) \int_{-\infty}^{t} (p_A(\tau) - p_B(\tau)) \, d\tau \] (18)

The total intensity of each surface under test is determined by the 'scanning technique', i.e. the intensity probe is manually moved along the surface following two perpendicular motion patterns as shown in figure 4. The scan is performed during a period of 2 min, approximately. Besides, the following ideas should be kept in mind in order to ensure reliable measured data when proceeding to the scanning:

- The scanning speed as well as the distance between the intensity probe and the surface should be constant.
- The intensity probe must be kept perpendicular to the surface.
- The surface must be scanned until the edges due to the fact that close to them (about 2 cm) the panel radiates most. However, it is also important not to scan farther away from the edges, otherwise the intensity of another panel corrupts the measurement for the actual panel.

At the end, it is also required to calculate the radiation factor of each measured plate. According to equation (19), it is first necessary to determine the total radiated power and the temporal and spatial mean-square normal velocity of the plate. From the measurement, the mentioned equation can be rewritten as follows:

\[ \sigma = \frac{\hat{I}_r S}{\rho_0 c_0 S G_{vv}} = \frac{\hat{I}_r}{\rho_0 c_0 G_{vv}}, \] (19)

where \( \hat{I}_r \) is the temporal and spatial averaged intensity and \( G_{vv} \) is the autospectrum of the velocity (temporal and spatial averaged mean-square velocity in the frequency domain). Note that the total acoustical power radiated by the panel under test is assumed to be the product between the temporal and spatial averaged intensity and the area of the panel.
3.3.2 ‘Theoretical’ Calculations

After processing all the measured data, some ‘theoretical’ calculations are carried out. The required Matlab source code is explained step by step in the following.

First, the required physical parameters are defined in the following lines.

```matlab
1: c_air = 343;
2: rho_air = 1.204;
3:
4: rho = [2700 1120]; % kg/m^3 density (Al and Lexan)
5: a = 0.63; % m height
6: b = 0.39; % m width
7: h = [3e-3 5e-3]; % m thickness, (Al and Lexan)
8: S = a*b; % m^2 area
9: U = 2*(a+b); % m perimeter
10: E = [7.1e10 2.46e9]; % N/m^2 Young’s modulus (Al and Lexan)
11: Poisson = [0.33 0.3]; % (Al and Lexan)
12:
13: f = 1:10000;
14: tol = 4; % tolerance in % for f ~ f_crit

Except for the Poisson’s ratio of Lexan, all of them were given. Due to physical limitations the latter can only take values up to 0.5 (e.g. rubber) and only very few materials with negative values are known. Most materials have numbers around 0.3, which is assumed to be realistic for the Lexan plate. However, equation [6] shows that this number is squared, which makes the influence marginal. Note that the last line defines a certain tolerance, as the middle part of equation (5) is not specified for a defined frequency range around the critical frequency. Hence, this range can be adapted in order to achieve reasonable results.

The computation of the radiation efficiency is carried out in a for loop for both materials. First, the critical frequency \( f_c \) and the according wavelength are calculated according to equations (6) and (7) respectively:

```matlab
15: for mm = 1:length(rho)
16: f_crit(mm) = c_air^2/(pi*h(mm)) * sqrt((3*rho(mm)*(1-Poisson(mm)^2))/E(mm));
17: lambda(mm) = sqrt(1.8*sqrt(E(mm)/(rho(mm)*(1-Poisson(mm)^2))) * h(mm)/f_crit(mm));
```

Next, the first part of equation (5) (i.e. \( f < f_c \)) is implemented considering the tolerance range around \( f_c \). In order to define the function \( g_1(f) \), the used lower frequency range must be subdivided into two further parts, see equation (8a).

```matlab
18: % f < f_crit:
19: idx_low = find(f < (f_crit(mm)-(tol/100)*f_crit(mm)));
20: f_low = f(idx_low);
21:
22: % g1(f):
23: % f < f_crit/2:
24: idxl = find(f_low < f_crit(mm)/2);
25: f_lowl = f_low(idxl);
26: g1(idxl) = 4/pi^4 * (1-2*f_lowl/f_crit(mm)) .* (1./sqrt(f_lowl/f_crit(mm))).* sqrt((1 - f_lowl/f_crit(mm))));
27: % f > f_crit/2:
28: idx2 = find(f_low >= f_crit(mm)/2);
```

Trials with different realistic values showed that the influence is in fact negligible.
Exercise C: Measurement of Sound Radiation of a Box

```matlab
29: f_low2 = f_low(idx2);
30: g1(idx2) = 0;
31: % g2(f):
32: gamma = sqrt(f_low/f_crit(mm));
33: g2 = 1/(4*pi^2) .* (((1-gamma.^2) .* log((1+gamma)/(1-gamma))+2*gamma)
   ./ ((1-gamma.^2).^(3/2)));
34: sigma_low = lambda(mm)^2/S * (2*g1 + U/lambda(mm)*g2);
35: % f > f_crit:
36: idx_high = find(f > (f_crit(mm)+(tol/100)*f_crit(mm)));
37: f_high = f(idx_high);
38: sigma_high = 1./sqrt(1 - f_crit(mm)/f_high);
39: % f ~ f_crit (+- tolerance, see above):
40: f_approx_fcrit = f(idx_low(end)+1:idx_high(1)-1);
41: sigma_approx_fcrit(1:length(f_approx_fcrit)) = sqrt(a/lambda(mm)) +
   sqrt(b/lambda(mm));
42: % sigma_low, sigma_approx_fcrit, sigma_high:
43: sigma(mm,:) = [sigma_low, sigma_approx_fcrit, sigma_high];
44: end
```
4 Results and Discussion

4.1 Mean-square velocities of the panels

Figure 5 shows the mean-square velocity spectra of both panel 5 and 6. It becomes obvious that neither the aluminum nor the polycarbonate (Lexan) panel give a noteworthy response at frequencies below 100 Hz, i.e. the modal density is very low. At higher frequencies, more and more modes are excited, giving response levels of partly more than 100 dB. As stated in [Cremer and Heckl, 1995], the model that is to be simulated implies that only modes, whose resonance frequency is in the considered frequency range, contribute to the radiation. Therefore, frequencies below 100 Hz can be neglected.

By trend, the Lexan plate reaches higher velocity levels than the aluminum plate. However, in both cases the levels decay towards higher frequencies.

4.2 Sound intensity

Figures 6 and 7 on the following page show the measured sound intensities of panel 5 and 6 respectively. Note that the figures show the magnitudes of the measured intensities, even though negative values occurred at certain frequencies. These are marked by gray curves and crosses. Such values are measured because the flow of energy towards the plate is higher than the flow of energy radiated away from the plate at the given frequencies. These values should be neglected.

The big fluctuations of the intensity with respect to certain frequencies can be explained by the modal behavior of the plate. This means that the plate radiates strongly at certain frequencies, whereas it hardly emits energy at nearby frequencies.
Figure 6: Measured sound intensity of the aluminum plate. Gray curves and crosses mark negative values, which should be neglected.

Figure 7: Measured sound intensity of the Lexan plate. Gray curves and crosses mark negative values, which should be neglected.
4.3 Radiation factor

The measured and simulated radiation efficiencies for panel 5 (aluminum) and panel 6 (Lexan) are shown in figures 8 and 9 on the next page respectively. The black curves are the simulated values, whereas the measured ones are shown in dark gray. Again, values for which the intensity was negative are highlighted by light gray curves and crosses and should not be taken into account.

The agreement between simulations and measurements is good in the case of the aluminum plate (figure 8). Especially the frequency ranges below and above the critical frequency are approximated in a satisfying manner by the simulations. However, the measurements do not show a peak around \( f_c \approx 4 \text{ kHz} \). In general, the radiation efficiency increases up to the critical frequency, after which it decreases, forming a plateau.

In case of the Lexan plate, the degree of agreement is significantly lower. The valid values below 500 Hz are represented satisfyingly. But above this range, the simulations underestimate the measurements. A rather constant slope is measured, but not shown by the predicted data. Furthermore, a resonance behavior at \( f_c \approx 8.4 \text{ kHz} \) cannot be seen.

Comparable degrees of agreement can be seen, if the data is averaged in 1/3 octave bands. This is shown in figure 10 on page 16. As identical scales are used it becomes obvious that the aluminum plate radiates more efficiently than the Lexan plate on the entire frequency range.

In [Cremer and Heckl, 1995], a possible reason for the described disagreement is given: Equations (5) and (8) give the impression that the radiation efficiency only depends on frequency and mechanical data. However, especially the boundary conditions and the type of excitation are important for the prediction of this quantity below the critical frequency. Two assumptions of the model are not clearly defined in the given measurements:

- The plate is supposed to be excited punctually. In the given setup only an adjacent plate is excited.
- The plate is supposed to be mounted in a rigid wall and the edges to be supported rotatably. In this exercise, the plate is mounted to a box structure formed of vibrating panels. This is done by screwing it to a frame (i.e. the edges of four other panels), which does not allow a clear definition of the boundary conditions. They can neither be assumed to be free nor perfectly clamped nor supported.
Figure 8: Measured and simulated radiation efficiencies of panel 5 (aluminum). Light gray curves and crosses mark the frequencies at which the intensity was negative. The measured values at these frequencies should be neglected.

Figure 9: Measured and simulated radiation efficiencies of panel 6 (Lexan). Light gray curves and crosses mark the frequencies at which the intensity was negative. The measured values at these frequencies should be neglected.
Figure 10: Measured and simulated radiation efficiencies of panels 5 and 6, 1/3 octave band averaged.
5 Conclusions

In the present exercise, the usefulness of the radiation factor has been demonstrated. This parameter characterizes structures in terms of the efficiency to transform vibrational energy into acoustic energy.

This can be applied in building acoustics where the transmission of sound through plate-like structures is of interest in terms of sound insulation. While in this case a low efficiency is desired, high values are useful e.g. for panel loudspeakers. In contrast to conventional loudspeakers they consist of a plate whose modes are to be excited by a vibration source.

A modal approach of the radiation factor provides reasonable results on average. This can be used as a first estimate of the structure under analysis. However, this relies on a sufficient modal density. The frequency above which it is sufficiently high is determined by the dimensions of the structure.

The practical measurements, in particular the measurement of sound power, can be carried out rather easily using an intensity probe. In this way, neither anechoic nor perfectly diffuse conditions are required.

We herewith confirm the unassisted creation of this exercise report. All authors contributed in equal amount.

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