The Lindeman Hall of Oslo — Evidence of lowfrequency radiation from the stage floor

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The Lindeman Hall of Oslo — Evidence of low-frequency radiation from the stage floor.

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ABSTRACT

It is well known that plate radiation below the critical frequency is very poor, and therefore many stage floors dissipate low-frequency energy transmitted from double-bass and cello end pins rather than providing a tuning-fork/table-top effect. However, if the stage floor is well damped, so that the transverse amplitudes fade out quickly around the point of excitation, a significant net radiation can be experienced also for low frequencies, due to the piston/baffle effect. Measurements performed in the Lindeman Hall of the Norwegian Academy of Music, in Oslo, Norway, showed that vibrational amplitudes in the stage floor faded out at a nearly equal pace in all directions around the excitation points, leaving nearly circular, quasi isotropic patterns for most frequencies of interest. In the audience area no tendency of spectral roll off was seen in the low-frequency end down to 30 Hz, which may represent the lowest fundamental of modern double basses. Transfer functions from stage floor to audience (intensity vs. power, and sound pressure vs. transverse velocity) were calculated for a number of seats in the hall.

INTRODUCTION

This research group has previously investigated the stage floors of a number of modern concert halls with respect to low-frequency excitation through double-bass end pins [1], [2]. Surprisingly, given favourable floor impedances, the floor can at certain frequencies show amplitudes significantly higher than those excited at the instrument's bridge. Also: in certain cases as much as 40 percent of the excitation power was transferred to the stage floor. However, in these earlier reports no conclusions were drawn on how much of this energy would actually reach the audience. Since double basses due to their limited size are very poor radiators below their Helmholtz frequency at about 60 Hz, while still playing fundamental frequencies one octave below that, one can imagine that floor radiation in this region could be benefitting.

The group hence set forth to perform two kinds of measurements in the hall that seemed most promising with concern to low-frequency radiation: the Lindeman Hall of the Norwegian Academy of Music in Oslo. First to measure the vibrations in the floor, i.e., how amplitudes were distributed around the point of excitation, bending wavelengths, etc. Second to measure the transfer functions from the floor to a number of seats in the hall, with respect to intensity and sound pressure.

THE LINDEMAN HALL

The Lindeman Hall (LiH), opened in 1988, is the symphony hall of the Norwegian Academy of Music in Oslo. It seats up to 430 people in the audience section. Of the five halls we investigated in ref [2] LiH is considered to be the one with the most satisfying sound from the double basses: deep, but still transparent, and well balanced. The floor is 25 mm Merbau (*Intsia bijuga/Intsia palembanica*) parquet, resting on joist 30 cm apart. The joists are "floating" on ca 5 mm thick blocks of rubber. The cavities between joists are filled with rock wool. The joists are oriented from rear to front of the stage, with the parquet crossing.

Figure 1. The Lindeman Hall before the organ was mounted.

EXPERIMENTAL SETUP

In previous experiments we have mainly excited the floors and instruments with force (impact) hammers. To obtain better control of the sound/noise ratio, we now excited the floor with a heavy shaker. Sweeps from 20 to 500 Hz were performed in sequences of 30 seconds.

First experiment

In the first of the two present experiments we measured how amplitudes fade out around the point of excitation, i.e., which region of the floor was truly active. A reference accelerometer was positioned adjacent to the head of the shaker, while a second accelerometer was positioned 5, 10, 15, 20, 30, 40, 75, 125, 200, and 300 cm away from the excitation point, respectively, and in two directions: sideways in the direction of the stage centre, and forward in the direction towards the audience. In Figure 1, the white arrow indicates the point of excitation on the floor. The signals of the two accelerometers were recorded at a sampling rate of 22050 Hz.

Second experiment

In the second experiment, where we wanted to measure the sound intensity in the audience vs. input power to the floor, a three-channel setup was used: the signal from a force transducer attached between the shaker head and the floor, an accelerometer adjacent to the force transducer, and a calibrated Brüel & Kjær 4007 studio microphone. This time we had the floor excited in two series 15 cm apart: *between* and *on* the joists. The microphone was placed at distances 170 cm above, and 8, 13, and 19 meters away from the points of excitation, the first position representing the near field as experienced by a (double bass) player, the latter three representing normal head positions of listeners seated at three different rows. In each row two different seat positions were used, about 3 meters apart. Totally this makes 1 + 3 × 2 microphone positions and two shaker positions, with a grand total of 14 registrations.

AMPLITUDES IN THE FLOOR

By use of a shaker and two identical accelerometers, the vibrational activity of the point-driven floor could be measured. The amplitude ratios are shown in Figures 3 through 10 for eight 1/3 octave bands from 25 to 125 Hz. As can be seen, the –6 and –10 dB isodynamic lines are found rather close to the excitation point (the distance being ≤ 50 cm and ≤80 cm, respectively) in all cases. In a previous study (ref. [1]) we measured the point impedance and damping of the floor on and between joists. The table is reproduced in the section on "Impedance match" (see table 2). The loss factor is quite high. The calculation of isodynamic lines of Figures 3 – 10 are all based on excitation in a point between joists. In the figures half wavelengths are included for comparison, the calculation of which is based on phase differences between the signals of the two accelerometers. These values proved a little noisy, so in order to get a fair estimate, phase differences of frequencies in the entire (thirds of octave) band were included in a series of least-squares error calculations. In general, waves along the joists are slightly longer than those crossing. However, a classical estimate of the wave propagation in the (unbeamed parquet) plate ends up with wavelengths between the two series actually measured.

The thumb rule (derivable from ref. [3]) reads:

$$\lambda = \sqrt{\frac{1.8\, c_L h}{f}}, \qquad (1)$$

where

c_L is longitudinal propagation speed,

h is plate thickness,

f is frequency.

Parameters utilized were c_L = 4000 m/s and h = 0.025 m. With the same routine the critical frequency can thus be estimated:

$$f_{CRIT} = \frac{64222}{h\, c_L}, \qquad (2)$$

which in this case gives 642 Hz, well above the frequency range of interest.

The efficiency of radiation in the low-frequency range is a matter of great interest. The problem can be illustrated as in Figure 2, where the floor vibration for the frequency 31 Hz is compared to the wavelength in air. The floor's vibration amplitude normal to the surface is fading out as the distance from excitation point (r) increases, but on the other hand the vibrating area expands with the factor $2\pi r$.

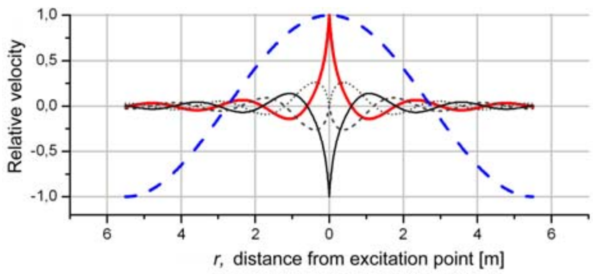

Figure 2. Point-excited bending waves (31 Hz) in the floor (solid red line and black trajectories) compared to the wavelength in air at the same frequency (blue).

It is often heard that below the critical frequency there will be little or no radiation from plates. Although this is often valid for plane waves, circular waves propagating from a single point of excitation *will* be radiated, but effectively only from the near or *direct* field. To give an estimation of sound power efficiency, both Cremer [3] and Skudrzyk [4] compare the area around the point of excitation (of an infinite plate) to a baffled piston of radius a, moving with the velocity of the driving point, and they make the following estimation:

$$a = \sqrt{\frac{2}{\pi^3}}\lambda_B = 0.254\lambda_B \approx \frac{\lambda_B}{4}, \qquad (3)$$

where λ_B is the bending wavelength of the plate.

In Figures 3 – 10, the Cremer/Skudrzyk piston diameters, $\lambda_B/2$ are indicated. The measured bending wavelengths of Lindeman Hall stage floor are:

Frequency [Hz]	λ_B (width) [cm]	λ_B (depth) [cm]
25	200	335
31	189	308
40	177	280
50	165	253
63	154	227
80	142	201
100	130	176
125	119	152

Table 1. Measured bending wavelengths of the stage floor. (Distance between joists is 30 cm.)

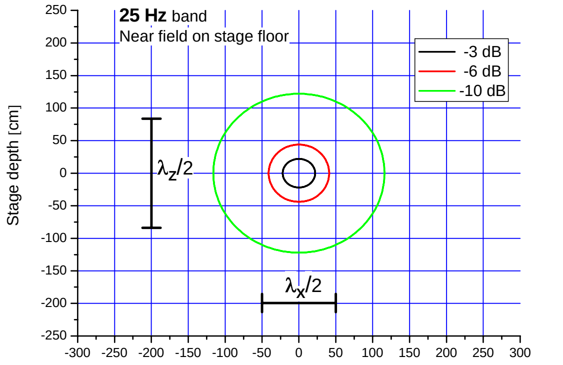

Figure 3. Amplitude distribution at the 25 Hz band.

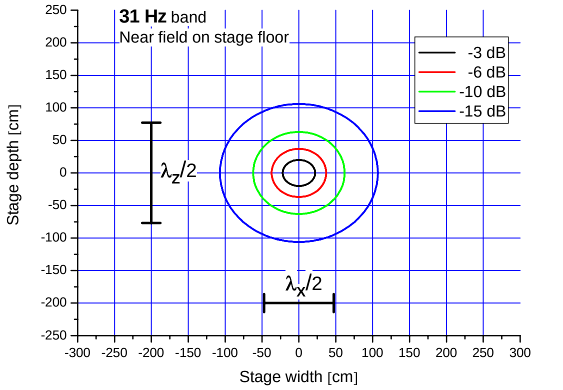

Figure 4. Amplitude distribution at the 31 Hz band.

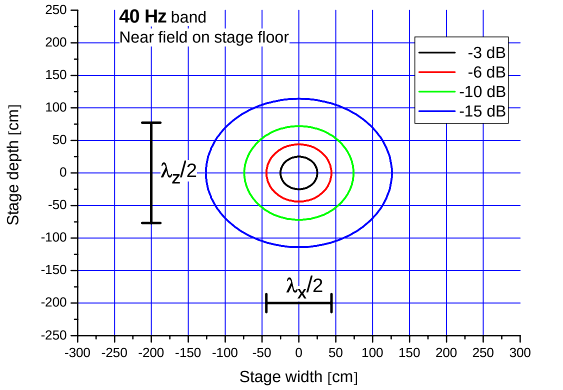

Figure 5. Amplitude distribution at the 40 Hz band.

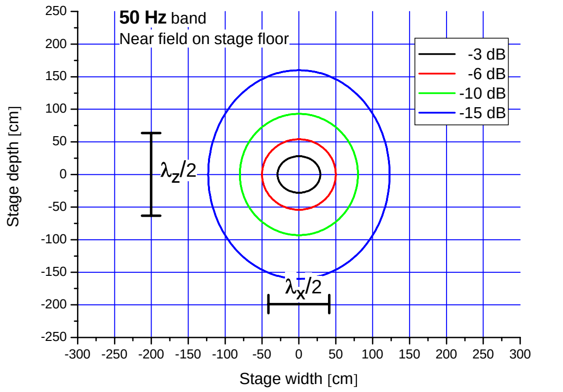

Figure 6. Amplitude distribution at the 50 Hz band.

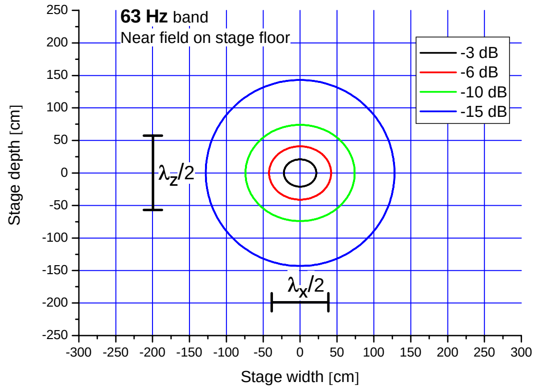

Figure 7. Amplitude distribution at the 63 Hz band.

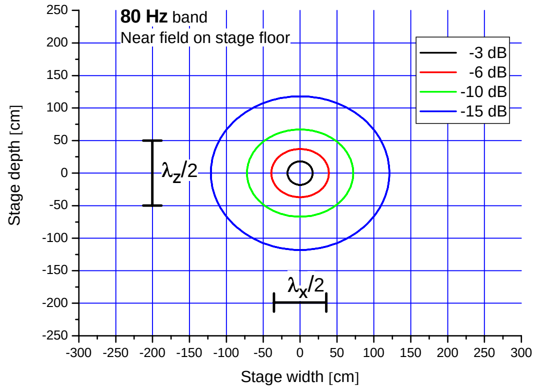

Figure 8. Amplitude distribution at the 80 Hz band.

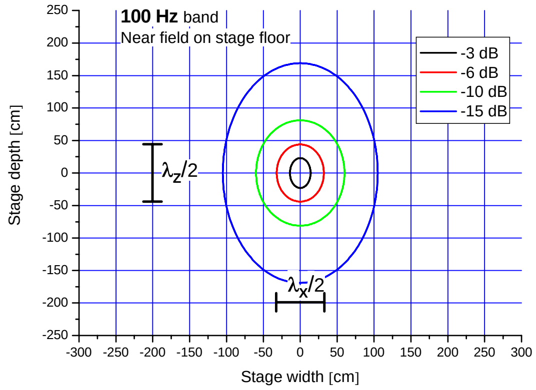

Figure 9. Amplitude distribution at the 100 Hz band.

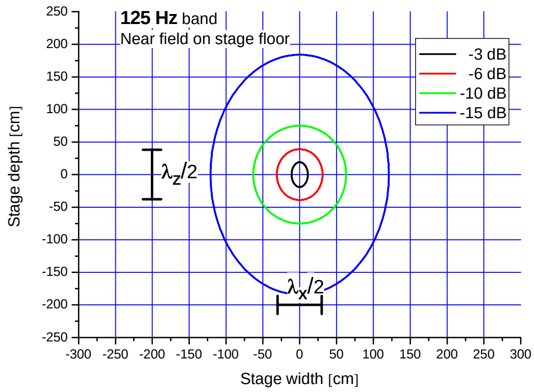

Figure 10. Amplitude distribution at the 125 Hz band.

One factor that plays a crucial role for radiation efficiency below the critical frequency is the damping. Figure 11 shows the *radiation factor*, i.e., the ratio between output sound power and input power, as function of normalized frequency and loss factor (η).

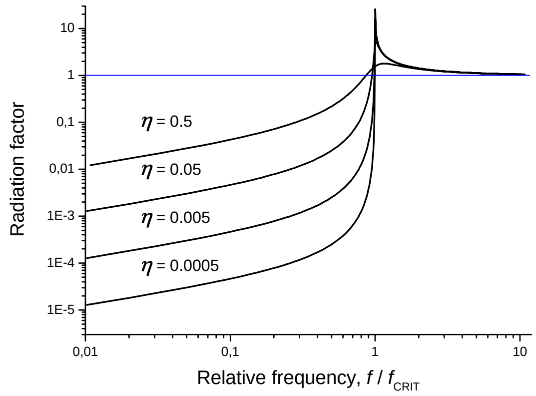

Figure 11. Radiation factor as function of loss factor and relative frequency

As is seen, the radiation factor is nearly proportional to the loss factor in an infinite plate. The equation here is [5]:

$$s = \mathrm{Re}\left(1 \Big/ \sqrt{1 - f_{CRIT} \big/ f} \right), \tag{4a}$$

where

$$f_{CRIT} = \frac{c^2}{2\pi}\sqrt{\frac{m''}{D_P}}, \tag{4b}$$

c is propagation speed in air,

m'' is mass per unit area,

$$D_P = \frac{E h^3}{12\left(1-\upsilon^2\right)}\left(1+j\eta\right), \tag{4c}$$

E is Young's modulus,

h is plate thickness,

υ is Poisson's number,

j is $\sqrt{-1}$

η is loss factor.

Notice that f_{CRIT} becomes complex when the loss factor η is introduced. Without the loss factor, the radiation factor, s, would have been zero whenever $f < f_{CRIT}$.

Concerning the Lindeman Hall, in the 20 – 125 Hz range the loss factor was on average 0.47 and 0.37 when measured *on* and *between* joists, respectively (see Table 2 in the section “Impedance match”).

For frequencies far below f_{CRIT} one may utilize the simple equation below for an estimate of the sound power radiated from the direct field, i.e., the field that is dominated by the point excitation so that the reverberation-field influence can be ignored [3]:

$$\bar{W}_{DIR} = \frac{\rho_0 \tilde{F}^2}{2\pi \left(m''\right)^2 c}, \qquad f \ll f_{CRIT}, \tag{5}$$

where

ρ_0 is density of air,

$\tilde{F}$ is input driving force.

In this equation, where the bending-stiffness term, D_P, has vanished, the power is inversely related to mass per unit area squared, and independent of frequency. Moreover, to find a simple expression for the input power, one can use [5]:

$$\bar{W}_{IN} = \mathrm{Re}\left(\frac{\tilde{F}^2}{8\sqrt{D_P m''}} \right). \tag{6}$$

(Notice that in this case, the denominator inside the parenthesis expresses the floor impedance.) Ignoring the joists and the reverberant field, and using c = 340 m/s, ρ_0 = 1.21 kg/m^3, m'' = 20 kg/m^2, and $f \ll f_{CRIT}$, an estimated sound-power radiation of the LiH stage floor can thus be calculated to

$$s = 10\log_{10}\left(\bar{W}_{DIR} / \bar{W}_{IN} \right) = -20.6 \text{ dB}.$$

Table 2 gives detailed information about the stage-floor impedances and losses, as well as averaged and linearized information on double-bass impedances for comparison.

SOUND INTENSITY IN THE HALL

A three-channel setup was utilized: (1) a force transducer mounted to the shaker head, (2) an adjacent accelerometer on the floor, and (3) a movable calibrated microphone. This permits the calculation of two transfer functions: sound intensity vs. input power, and sound pressure vs. floor velocity at the point of excitation (see Figures 12 –19; left column: between joists, right column: on joists). For each microphone position of we had some four seconds of of silence, from which the background noise of the ventilation system, etc. could be estimated and included in the plots. The S/N ratio (sound-pressure levels) seems not to have been influencing the outcome of these measurements in the range of main interest (i.e., above 30 Hz).

The 0 dB reference values in the plots are:

Intensity:	Sound pressure^2 (rel. 94 dB SPL) – 10 $\log_{10}$ (408).
Power:	1 W, calculated as: Re[Force(ω)×Velocity*(ω)], where * indicates conjugated.
Sound pressure:	94 dB SPL
Force:	1 N
Velocity (floor):	1 m/s.

In the far-field plots measurements two different seat positions in each row were averaged to minimize the effect of potential node-line dropouts caused by standing waves.

The plots show evidence of effective radiation down to low frequencies. All transfer functions show low-frequency roll off from just above 30 Hz. As can be seen, the sound-pressure/floor-velocity transfer function (red lines) is fairly flat from about 200 Hz down to, say, 30 Hz. The same is seen in sound-intensity/input-power transfer plots (green lines), but here a minor tilt towards the low-frequency end is noticeable (typically some –5 dB), as expected with the rising floor impedance. This tendency is visible at all distances.

When running the sweeps, the sound was audible from above some 35 Hz approximatey, but not continuously through all frequencies until, say, above 80 Hz. A significant rise in perceived level took place around 120 Hz–still some five times below the expected critical frequency. During the 20 to 500 Hz sweep the input power typically showed values around 1.0 mW. At favourable impedance conditions one double bass can deliver more than ten times this power in the 30-60 Hz region through its endpin.

Excitation between joists:

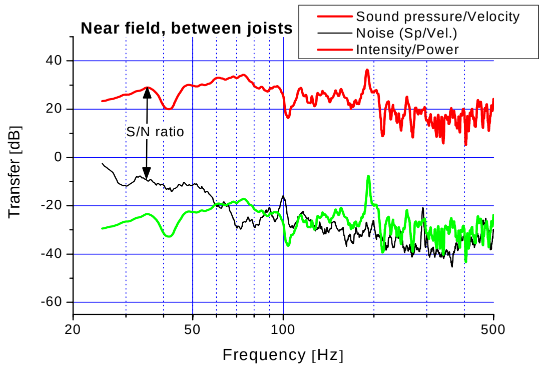

Figure 12. Near-field transfer functions.

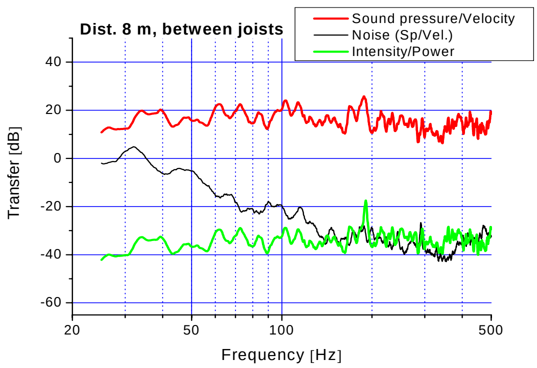

Figure 13. Transfer functions with microphone 8 m away from the excitation point.

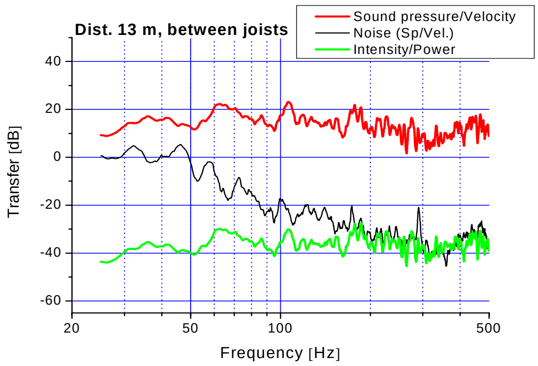

Figure 14. Transfer functions with microphone 13 m away from the excitation point.

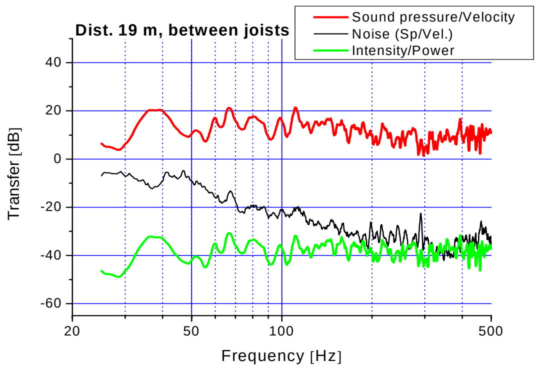

Figure 15. Transfer functions with microphone 19 m away from the excitation point.

Excitation on joists:

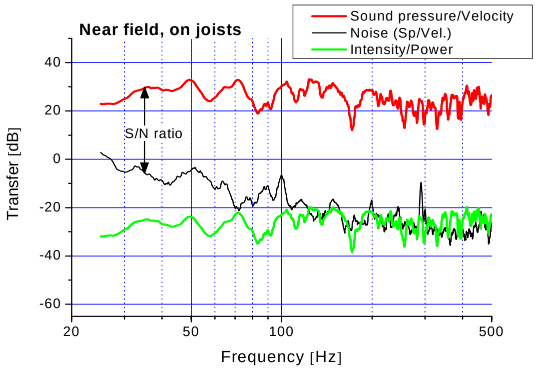

Figure 16. Near-field transfer functions.

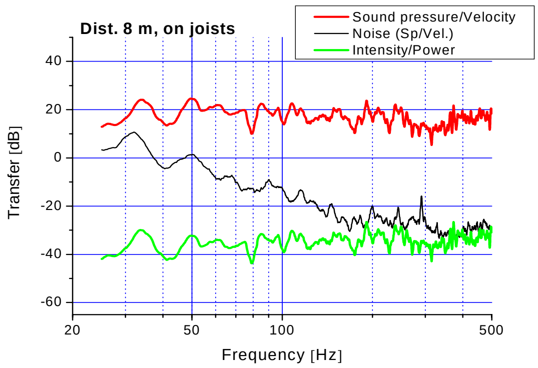

Figure 17. Transfer functions with microphone 8 m away from the excitation point.

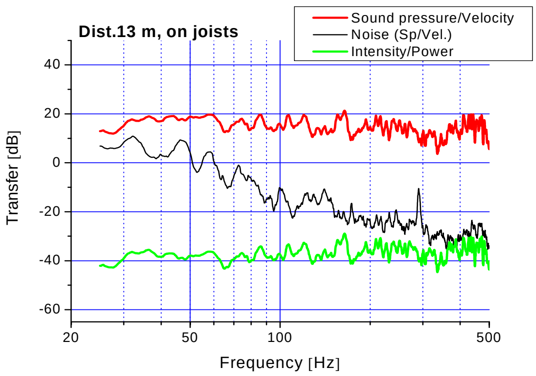

Figure 18. Transfer functions with microphone 13 m away from the excitation point.

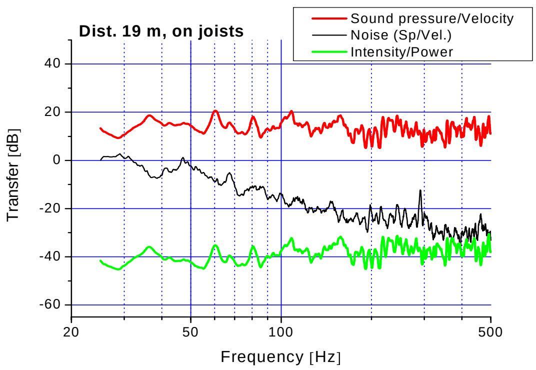

Figure 19. Transfer functions with microphone 19 m away from the excitation point.

IMPEDANCE MATCH

We have investigated five concert arenas with respect to the stage floor impedances, etc. (see [2]). With the exception of LiH, the impedances vary significantly with the choice of measuring position on stage: between joists, on a joist, on floor sections mounted on hydraulic risers, on rigid floor sections glued to concrete. Because of the unusual floor construction in LiH, where all joists rest on thin rubber blocks, the difference in impedances between the “pliant” and “rigid” areas proved considerably smaller here than in the other halls. (As far as we know, the rubber blocks were not put in there for acoustical reasons, but rather as a practical solution for straightening up small irregularities in the concrete foundation.)

If double basses (which due to their limited corpus size hardly radiate frequencies below the Helmholtz frequency at about 60 Hz) should benefit from the radiation potential of the floor, their endpin impedances must be fairly well matching those of the floor. In Table 2 representative (averaged) double-bass impedances, as measured into the end pin, are included for comparison with those of LiH. The Lindeman Hall provides excellent matches in the important octave 31 to 63 Hz. Three of the other stage floors we measured (The Oslo Concert Hall, and the rehearsal hall and the orchestra pit of the new Norwegian Opera) all showed higher impedances in this region: typically 8 – 15 dB above LiH in the rigid sections, and 3 – 8 dB in the pliant sections. A fifth hall, the Berwald Hall of Stockholm was originally constructed with the stage-floor parquet glued directly onto bedrock with asphalt. After massive protests from the musicians the fundament of the entire stage was caved out to give space for hydraulic lifts, supporting smaller stage-floor sections. This implied that the impedance at 40 Hz was reduced from about 4 000 000 to 800 kg/s, or –37 dB, and the frequency of equal impedance magnitude of bass and floor, landed at 27 Hz, which may be a little on the low side.

The fact that the impedance of most floors is “springy” in the frequency range below 100 Hz, while the bass, as seen into the endpin, is predominantly a mass in the same range, enables very efficient transfer of vibrations through the end pin. For certain frequencies the vibrational velocity of the floor is often seen to reach significantly higher values than at the bridge, through which the instrument body is excited. The transfer equation can be expressed:

$$\boldsymbol{H}(\omega) = \frac{\boldsymbol{v}(\omega)_{FLOOR}}{\boldsymbol{v}(\omega)_{BASS\_CORPUS}} = \frac{\boldsymbol{z}(\omega)_{END\_PIN}}{\boldsymbol{z}(\omega)_{END\_PIN} + \boldsymbol{z}(\omega)_{FLOOR}} \qquad (7)$$

The transfer spectrum will typically have a peak at the frequency where impedance magnitudes are equal, while approaching unity above, and roll off by some –18 dB/oct below.

CONCLUDING DISCUSSION

This investigation has shown that low-frequency radiation from a stage floor indeed is possible if conditions are favorable. These results are not in conflict with established theories within the field.

From a musical point of view it is likely that this radiation is quite desirable as long as the reverberation in this frequency range is not excessive. In the LiH, where the lower part of the sound spectrum sounds balanced and transparent, the reverberation times are 1.9, 2.2, 2.1, and 1.8 for the octave bands 63, 125, 250, and 500 Hz, respectively. (No estimation of the 31 Hz band was ever done. Even the 63 Hz octave band is often omitted in such analyses.)

In order for double basses—or other low-frequency instruments with floor contact—to take advantage of the radiation properties of a suitable floor, it is a prerequisite that the floor be light enough to facilitate a sufficient vibration transfer in the frequency region of interest. In the Lindeman Hall of Oslo, all parameters seem to be well matched for supporting this phenomenon.

REFERENCES:

1 K. Guettler, Askenfelt, A.., and Buen, A.. "Double basses on the stage floor" *Proc. ISMA'07* (2007), Barcelona.

2 K. Guettler, A. Buen and A. Askenfelt, "An in-depth analysis of the double bass-stage floor contact" Proc. 7th International Conference on Auditorium Acoustics (2008), Oslo, Norway, Institute of Acoustics.

3 L. Cremer and M. Heckl, "*Structure-borne sound*" (second edition) Springer-Verlag New York, Inc. (1987).

4 E. Skudrzyk, "*Simple and complex vibratory systems*" Pennsylvania State University Press (1968).

5 H. Bodén, U. Carsson, R. Glav, et al., "*Ljud och vibrationer*" Kungl. Tekniska Högskolan, Stockholm (2001).

Lindeman Hall (LiH), On/Between Joists									Double bass	
Freq. band [Hz]	Decay$_{-60dB}$ [ms]		Loss factor		Impedance magn. [kg/s]		phase [degrees]		Imp. [kg/s]	phase [deg.]
20	215	356	.518	.313	9950	3554	-105	-124	1000	88
25	197	255	.449	.347	7951	3293	-106	-126	2286	88
31	203	189	.346	.371	6016	3155	-111	-129	3571	88
40	144	140	.388	.397	5850	3090	-120	-129	4857	88
50	85	110	.519	.402	4680	2785	-112	-128	6143	88
63	69	55	.511	.639	3706	2208	-118	-135	7429	88
80	52	87	.537	.320	3141	2434	-122	-157	8714	75
100	34	105	.657	.211	2667	2985	-129	-151	10000	-125
125	36	54	.494	.327	2632	2995	-136	-156	10000	-125
160	34	68	.409	.204	2523	3487	-132	-152	Descending impedances with higher frequencies	
200	30	72	.366	.153	2165	3465	-126	-142		
250	35	38	.248	.229	1653	3274	-130	-146		
320	23	49	.304	.142	1430	2754	-144	-125		
400	27	48	.201	.115	1795	2472	178	-140		
500	25	48	.172	.091	5085	2521	-176	-141		

Table 2. Acoustical properties of the LiH stage floor. Notice the high loss factors. The rightmost two columns show typical double-bass endpin impedances, averaged and linearized.