17. Investigation of loudspeaker cabinet vibration using reciprocity

H Alavi & K R Holland, ISVR, University of Southampton
E-mail: **Hessam.Alavi@soton.ac.uk**

This paper investigates the contribution of loudspeaker cabinet wall vibration to the sound pressure at a listening position, and draws conclusions about the importance of low-vibration cabinet construction. Although the vibration of the walls can be measured using standard techniques, assessing the contribution of that wall vibration to the radiated sound field requires details of the Green functions linking the vibrating wall to points in the field.

With all but the simplest geometries, these Green functions are generally not known. Here, the required Green functions are measured in-situ, by invoking the principle of vibroacoustic reciprocity. These Green functions are combined with measurements of the wall vibration when the loudspeaker is operated, to yield estimates of the contribution of parts of the cabinet to the overall sound pressure at the listening position. The Green functions are measured first in an anechoic chamber, and then in a typical domestic listening room.

INVESTIGATION OF LOUDSPEAKER CABINET VIBRATION USING RECIPROCITY

H Alavi   ISVR, University of Southampton, UK
K R Holland   ISVR, University of Southampton, UK

1   INTRODUCTION

There are many situations where a number of different unwanted noise and / or vibrational sources contribute to the total radiated sound by a loudspeaker. This paper explores the effect of the cabinet vibration on the sound radiated by a loudspeaker. For the purpose of this research a loudspeaker cabinet with a rectangular geometry and uniform thickness and density was studied. The number of drivers also was minimized to one drive-unit. The motivation behind this project was to find the contribution of the loudspeaker cabinet wall vibration to the pressure at the listening position also to compare those results with that of the driver contribution. An assessment for the importance, or otherwise, of rigid loudspeaker cabinet construction was also carried out.

The basic theory of vibroacoustic reciprocity has been briefly explained in the technical report *"A guide to the exploitation of vibroacoustic reciprocity in noise control technology" by Holland and Fahy 1997*[1] moreover the point-to-point acoustic reciprocity is presented in detail in *"The Theory of Sound" by J.W.S. Rayleigh 1896*[2]. For harmonically vibrating surfaces this has been explained in the aforementioned paper by *Holland and Fahy* which states that the pressure at the listening position is equal to the sum of the volume velocities of the individual elements on the surface of the loudspeaker cabinet multiplied by their corresponding Green functions. The volume velocity has also been equated to the product of the normal velocity of each surface element on the cabinet wall with its corresponding area for that individual element on the loudspeaker cabinet[1].

2   THEORY

Finding the total pressure at the listening position was the main interest in this study. However, the total pressure is equal to the integral of the total contributions of the cabinet and driver over the entire cabinet surface.

$$p(\mathbf{x}) = \int_S G(\mathbf{x}\,|\,\mathbf{S})u(S)dS \tag{2.1}$$

Where u is the normal velocity of the elements on the surface of the cabinet wall and G is the Green function from the surface (individual element) to the listening position $\mathbf{x}$. The integration over part of the loudspeaker cabinet gives its contribution to the total contribution. By dividing the entire surface of the cabinet into finite-sized elements the integration becomes a summation,

$$p(\mathbf{x}) = \sum_i G(\mathbf{x}\,|\,\mathbf{i})q_i \tag{2.2}$$

Where q_i is the volume velocity of each individual element on the surface of the cabinet,

$$q_i = u_i S_i \tag{2.3}$$

The volume velocity q_i can be directly measured using standard techniques such as accelerometer or laser vibrometer. Finding the Green functions G connecting each element on the surface of the loudspeaker cabinet wall to the listening position $\mathbf{x}$ requires the detailed geometry between the two

points, and these are not known in general. However, these Green functions were measured in situ by invoking the principle of vibroacoustic reciprocity.

2.1   Theory of reciprocity

The general principle of point-to-point acoustic reciprocity is presented in detail in *"The Theory of Sound" by J.W.S. Rayleigh 1896*[2]. The principle of vibroacoustic reciprocity in essence states that the acoustic pressure at a point in a fluid due to a harmonic point monopole acoustic source at another point in the otherwise still fluid is independent of an interchange of the positions of source and receiver, irrespective of the presence of any arbitrary, linearly reacting boundaries.[1]

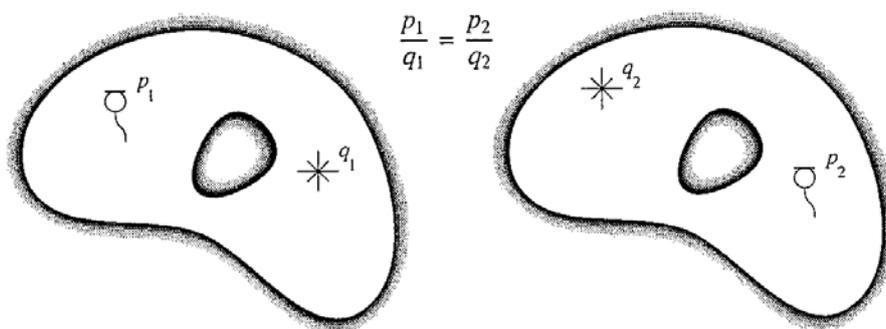$$G(\mathbf{x}\,|\,\mathbf{y}) = G(\mathbf{y}\,|\,\mathbf{x}) \qquad (2.4)$$

Figure 2.1 Shows acoustic reciprocity under arbitrary non-free-field condition[1]

By placing a sound source at the listening position the required set of Green functions can be measured using the principle of acoustic reciprocity.

$$G(\mathbf{x}\,|\,\mathbf{i}) = G(\mathbf{i}\,|\,\mathbf{x}) \qquad (2.5)$$

The pressure contribution of each element on the surface of the loudspeaker cabinet can then be calculated by multiplying the q_i by its corresponding Green function $G(\mathbf{x}\,|\,\mathbf{i})$ to give the contribution of the element i to the total pressure.

2.2   Spatial sampling

The main question in dividing each panel on the surface of the cabinet was the elemental size that would satisfy the maximum spacing requirements. Therefore the panels of the cabinet wall were divided into the finite-sized elements by applying the Nyquist sampling criterion to both vibration field and acoustic field to give the spatial sampling requirements.

$$d \leq \frac{\lambda_{\min}}{2} \qquad (2.6)$$

Where d is the spacing of the elements on the surface of the cabinet wall, and has to be smaller than half of the minimum wave length. The spatial sampling with regard to vibration field was determined using the equation:

$$\lambda_{\min} = \frac{2\pi}{\sqrt{\omega}}\left(\sqrt[4]{\frac{D}{m}}\right) \qquad (2.7)$$

Where m is the mass per unit area, and D is the bending stiffness of the plywood used in the cabinet construction.

$$D_{plywood} = \frac{Eh^3}{12(1-v^2)} \tag{2.8}$$

Where E is the Young's modulus, and v the Poisson's ratio, and h is the thickness of the plywood used in the cabinet construction. The smaller spacing d between the two considerations mentioned was considered for the elemental spacing, which in this case was the acoustic field.

3 METHOD OF MEASUREMENTS

3.1 Measured surface area

Each panel of the loudspeaker cabinet wall was divided in to the finite-sized square elements of equal cross-sectional area. The pressure contribution was assumed to be constant within each elemental area. As a result a single point per element was measured throughout this research. The drive-unit was divided to the similar size square elements as any other elements on the cabinet.

3.2 Grid divisions and spatial sampling

With the maximum frequency chosen to be 3.2*kHz* spatial sampling of the elements on the surface of the cabinet wall with regard to vibration field was found using equation (2.7) and was calculated to be:

$$d = \frac{\lambda_{min}}{2} = 10cm$$

The spatial sampling with regard to acoustic field consideration was found using equation (2.6) and worked out to be:
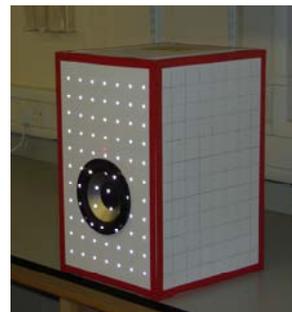
$$d \le \frac{c_0}{2f_{max}} = 5.31cm$$

Figure 3.1 Loudspeaker

The acoustic spatial sampling was the smaller spacing between the two considerations and was used as the spacing of the elements on the surface of the cabinet. Using the acoustic spatial sampling resulted in the Front, Sides and Back panels to be divided to 77 elements in 11 rows by 7 columns. The top panel was divided to 49 elements in 7 rows by 7 columns. The vibration of the bottom panel was assumed to be of not much significant since it was placed on the loudspeaker stand.

Table 1 Physical characteristic of the plywood used in the cabinet construction

Material used in the Cabinet Construction	Young's modulus E= [N m^{-2}]	Poisson's ratio v	mass per unit area m= [kg m^{-2}]	Thickness h= [m]
Plywood	12.4*GPa*	0.3	12.46	18*mm*

3.3 Acquisition setting for the laser vibrometer

Dual channel acquisition was carried out to record the signal from a laser vibrometer in measuring the volume velocities or the signal from miniature microphone in measuring the Green functions and the signal from the amplifier. The amplified signal was also used as the reference signal in order to find the Fast Fourier Transform (FFT) of the measurements. The recorded measurements were

automatically checked by the software of the laser vibrometer and re-measured incase it was necessary to ensure the optimum average signal was gathered.

The complex averaging for each element on the surface of the cabinet wall was set to 50 to ensure the optimum signal has been recorded. The frequency bandwidth was chosen to be from $20Hz - 3.2kHz$ with 6400 FFT line. The sampling frequency was found to be:

$$f_{sample} = 2.56 \times Bandwidth \qquad (3.1)$$

Where the bandwidth was considered from $0Hz - 3.2kHz$, therefore the sampling frequency became: $f_{sample} = 2.56 \times 3.2 = 8192Hz = 8.192kHz$. The sampling time was set according to:

$$t_{sample} = \frac{n_{FFT}}{BW} \qquad (3.2)$$

Where $n_{FFT} = 6400line$ is the number of FFT line, and $BW = 3.2kHz$ is the bandwidth.

$$t_{sample} = \frac{6400}{3200} = 2\sec$$

And finally the resolution was set by the ratio:

$$\Delta f = \frac{1}{t_{sample}} \qquad (3.3)$$

Where t_{sample} is the sampling time. The frequency resolution of the measurements was found using:

$$\Delta f = \frac{1}{2} = 0.5Hz$$

3.4   Acquisition of the Green functions

The Green functions were measured in two different conditions. First they were measured in the free-field condition which was simulated in the large anechoic chamber. Next the experiment was repeated in normal domestic listening room.

4   RESULTS AND DISCUSSIONS

4.1   Volume velocities

The loudspeaker was placed on a stand directly in front of the scanning laser head. The velocity within each element on the surface of the loudspeaker cabinet was considered to be uniform. As a result one point pre element was measured. The velocity of each individual point was measured using a scanning laser vibrometer. The velocity magnitude of the elements on the driver were fluctuating in the range of (0 to-40) dB relative to ms^{-1}/V, where as the velocity magnitude of the elements on the front panel were exhibiting (-30 to -100) dB relative to ms^{-1}/V fluctuations. The velocity magnitude of each point on the side panels was about 10 dB less than the points on the front panel.

Due to the pressures generated inside the cabinet by the drive-unit, the back panel exhibited similar vibrational magnitude as the front panel. The lower elements on the back panel showed less vibration since their vibration was greatly neutralized by the loudspeaker stand. Due to the same effect the bottom panel vibration was considered to be of no significant since it was placed on the stand. The difference in velocity magnitude of the elements on the front panel in comparison to the rest of the cabinet was expected since the source of structural vibration, the driver, was mounted on the front panel.

In finding the volume velocity of each element on the loudspeaker cabinet and the drive-unit, the measured velocity magnitudes were multiplied by their corresponding cross sectional area which was set by the acoustic spatial sampling. Since the cross sectional area was uniform for all the elements on the loudspeaker cabinet and the drive-unit, the volume velocities were following the same pattern as the point velocities, only all being multiplied by a constant (elemental area).
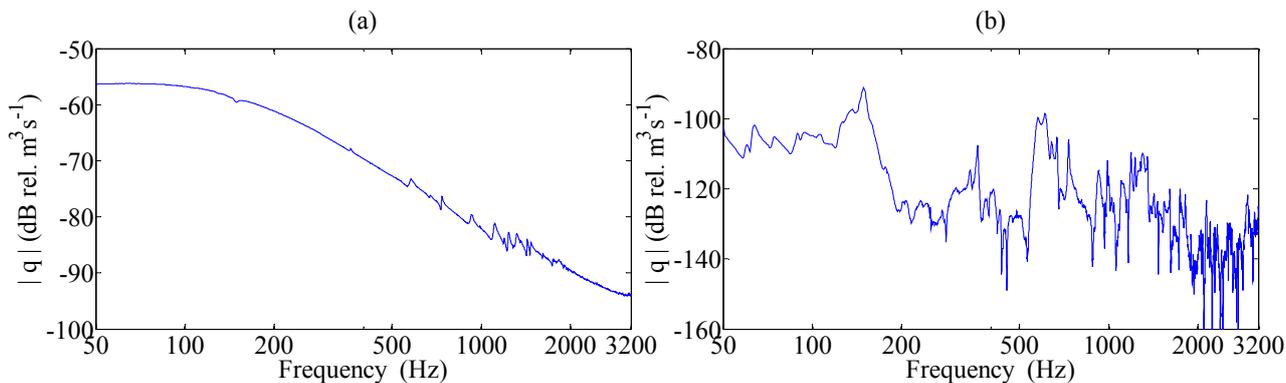

Figure 4.1 Volume velocity of point (2, 2) on the driver (a), and Volume velocity of point (2, 6) on the front panel (b).

The volume velocity of the individual elements on the drive-unit were in the range of (-50 to -100) dB relative to m^3s^{-1}, where as the volume velocity of the individual points on the front panel were fluctuating in the range of (-80 to -160) dB relative to m^3s^{-1}. The magnitude of measured volume velocities drop by about 20 dB above the frequency of 2000Hz. This indicates that above that frequency the volume velocity contribution of each element become insignificant. This result also confirms that the chosen frequency range of interest from 20 to 3200Hz contains all the significant bandwidth of energy.

4.2   Free-field Green functions (in anechoic chamber)

Next the corresponding Green functions for each individual point on the surface of the loudspeaker cabinet and drive-unit were measured in the free-field condition which was simulated in large anechoic chamber.
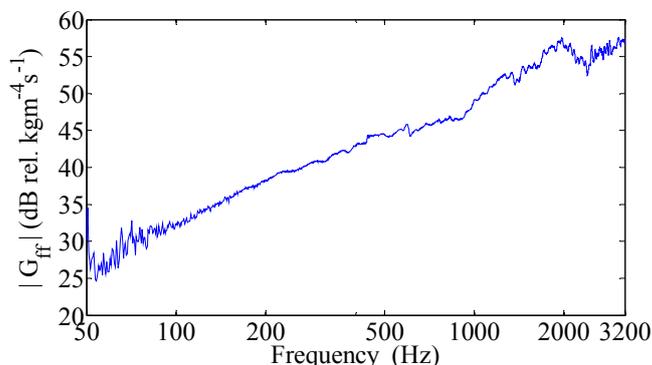
Figure 4.2 Free-field Green function of point (2, 6) on the front panel.

The delay in the low frequency range between signals from the microphone and the source to the control unit of the laser vibrometer caused low coherence in the frequency range of 50 to 100Hz which inevitably created fluctuations in the Green function magnitude of each point in that frequency

range. Above the 100*Hz* frequency the magnitude response of each points on the front panel were showing a gradual increase with increasing the frequency up to approximately 600*Hz*. After that a small dip of approximately 1.5dB arises. After that point the Green function magnitude exhibited a gradual increase up to about 1965*Hz* where it starts decrease of 5dB up to frequency of 2400*Hz* where the Green function start to increase again up to the maximum frequency range of interest.

Unlike the volume velocities the in situ measured Green functions demonstrating similar range of fluctuations for all the elements on the cabinet and driver. The Green function response of the elements on the side panels were similar to the front panel Green functions. However, the responses of the back panel showed much more pronounced fluctuation above 700 *Hz*. Since the size of the wave length becomes comparable to the actual size of the loudspeaker cabinet, the edge diffractions could be assume to be the cause of those strong fluctuations, above that frequency. In comparison to the rest of the panels, the Green functions of the elements on the top panel of the cabinet showed much more steady response after the 700*Hz* frequency.

4.3   Listening room Green functions

The experiment was repeated in the normal domestic listening room. The measurement procedure was exactly the same as the previous part in the anechoic chamber. The only difference was the location of the loudspeaker which was placed in the corner of the room on the stand as it would be in the most normal conditions. The loudspeaker was placed with 60° angle from the back wall, keeping similar distances from the loudspeaker center line to the back and side walls.
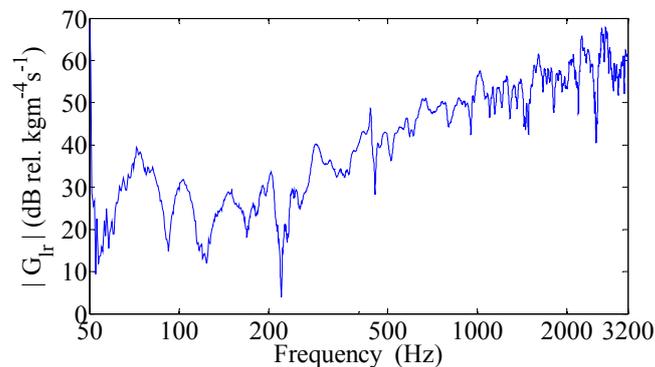

Figure 4.3 Listening room Green function of point (2, 6) on the front panel.

As can be seen the listening room Green functions exhibited much more fluctuation in comparison to the anechoic measurements due to the degrading effect of the room itself. The degrading effect of the room could also be noticed in the measured signal coherence.

4.4   Free-field pressure contributions

The pressure contribution created by each element on the surface of the loudspeaker cabinet and the drive-unit was then calculated. The complex pressure magnitude is the result of multiplying the volume velocity (q) of each individual point on the loudspeaker cabinet or driver by its corresponding Green function (G).

It can be seen from figure [4.4] that the on-axis pressure of the sample point on the driver exhibited lots of fluctuations up to 100*Hz* which was due to the Green function response of the corresponding element at that frequency range. From that frequency onwards the pressure magnitude started to rise gradually up to the 230*Hz* frequency. However, a small dip of about 1.5dB can be noticed in the

145 to 150*Hz*. From that point onwards the pressure started to decrease gradually with increasing the frequency. However, there were some small peaks in the response which they started to grow in terms of their magnitude as the frequency advanced. The first pronounced peak was observed at the 563*Hz* with the 2dB magnitude. The peaks followed at the frequencies of 728, 932, 1105, 1243, 1322*Hz* with the more or less 2dB magnitudes. From 1422 to 1769*Hz* the pressure response showed another increase of about 5dB. After that frequency, the pressure response decreased up to 2720*Hz* by about 10dB and eventually finished by a 3dB increase.
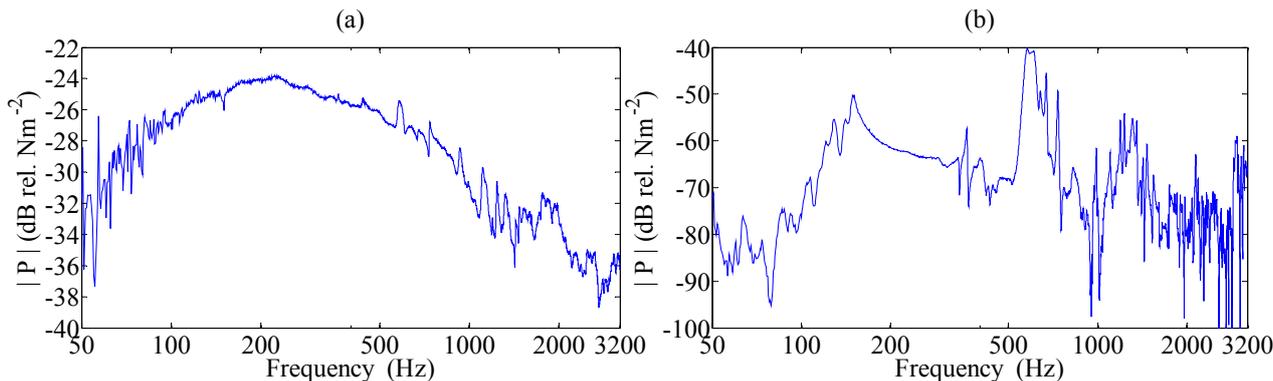
Figure 4.4 Free-field pressure response of point (2, 2) on the driver (a), and point (5, 4) on the front panel (b) measured in the anechoic chamber (note the different vertical scales in the presented figure).

The pressure response of the sample point on the front panel showed more erratic behavior than the elements on the drive-unit with the highest peak arising at the 577*Hz* with approximately 20dB peak. Overall the pressure contributions of the elements on the edges of each panel were higher compare to the rest of the elements on that panel, which was due the edge diffraction. The edge diffraction for the elements on the side panels were more pronounced than those points on the back panel edges. The elements on the front edge of the top panel were exhibiting higher magnitude compare to those at the far edge of the top panel. Overall, the interferences in all cases had degrading effect on the on-axis response of the loudspeaker.
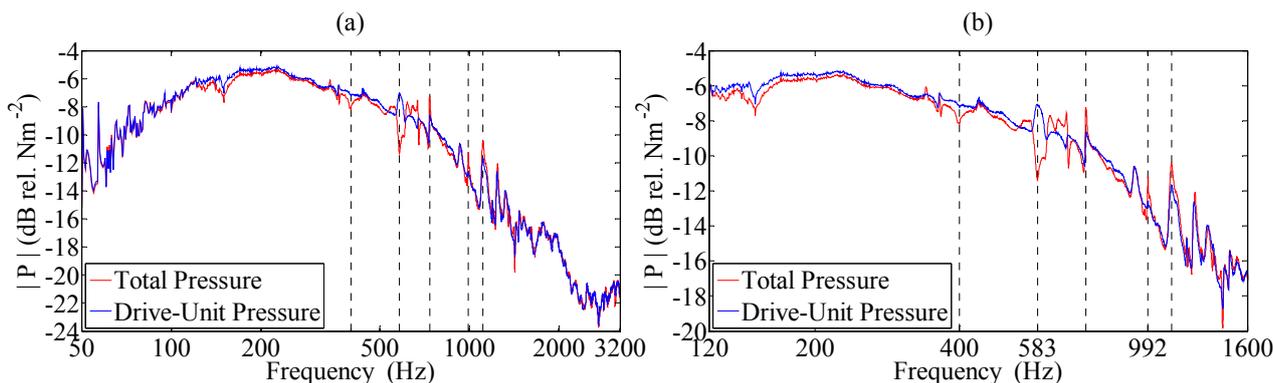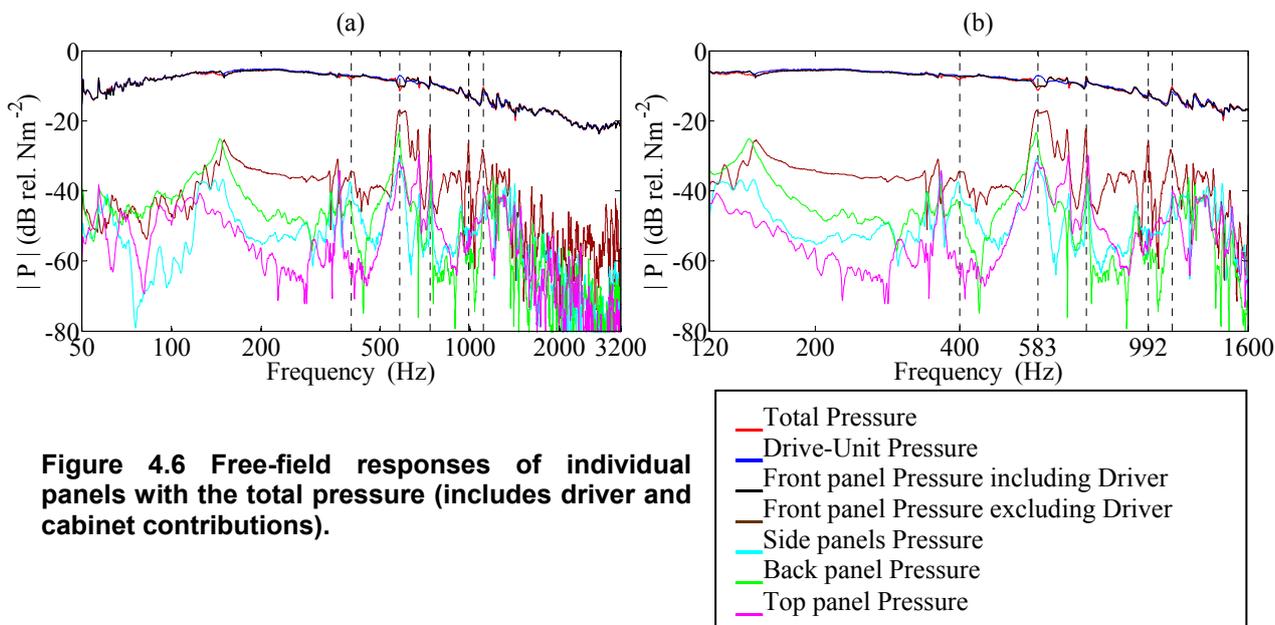

Figure 4.5 Total pressure response (includes driver and cabinet contribution) and the drive-unit response measured in anechoic chamber.

Figure [4.5] (a) compares the on-axis response of the total pressure which consist of driver and cabinet contributions with the driver contribution on its own. Figure (b) is the magnified view of 120 to 1600*Hz* frequency. The figure shows that at the frequencies of 400 and 583*Hz* the total loudspeaker pressure contribution is less than the driver contribution on its own by about 1 and 4dB respectively. In the frequency range of 610 to 680*Hz* the total contribution shows unwanted output

of about 1dB. In the frequencies of 735.5, 992 and 1111*Hz* the total response shows more or less 1dB peak. Overall the peaks in the total pressure of the loudspeaker appear at the most sensitive range of human's audible frequencies which are around 1000*Hz*. This result indicates that at these frequencies the effect of loudspeaker cabinet will degrade the channel quality and could be easily noticed.

The result also indicates that the drive-unit is the dominant source of the volume velocities. However, the cabinet response causes some destructive interference with the drive-unit output more or less through the whole frequency range of interest. These destructive interferences cause the total on-axis response of the loudspeaker to be lower than the output of the drive-unit on its own. However, in some frequencies these interferences become constructive and added some unwanted output to the total pressure response of the loudspeaker appearing as peaks.

Since the peaks are much more easily noticeable than the dips those troublesome frequencies have the priority to be canceled out. Also in the frequency range that the interferences were destructive, the total outcome of the loudspeaker was not offset uniformly. These behaviors clearly affect the desired flat response of the loudspeaker causing it to be less predictable.


Figure 4.6 Free-field responses of individual panels with the total pressure (includes driver and cabinet contributions).

The correlation between the troublesome frequencies in the on-axis total free-field response of the loudspeaker (which includes the driver and cabinet contribution) with the contribution of each individual panel can be observed from the figure [4.6]. Figure (b) is the magnified view of 120 to 1600*Hz* frequency. It can also be noticed that each individual panel exhibits peaks with high magnitude at each of the troublesome frequency. However, since pressure is a complex value at some frequencies the cabinet contributions add up to the driver response and create unwanted noise in the total pressure contribution. On the other hand the driver response cancels out at some other frequencies by the contribution of different panels causing lower total pressure. For the purpose of comparison the contribution of the driver on its own was also added to the figure to demonstrate, overall it is the dominant source of pressure contributions.
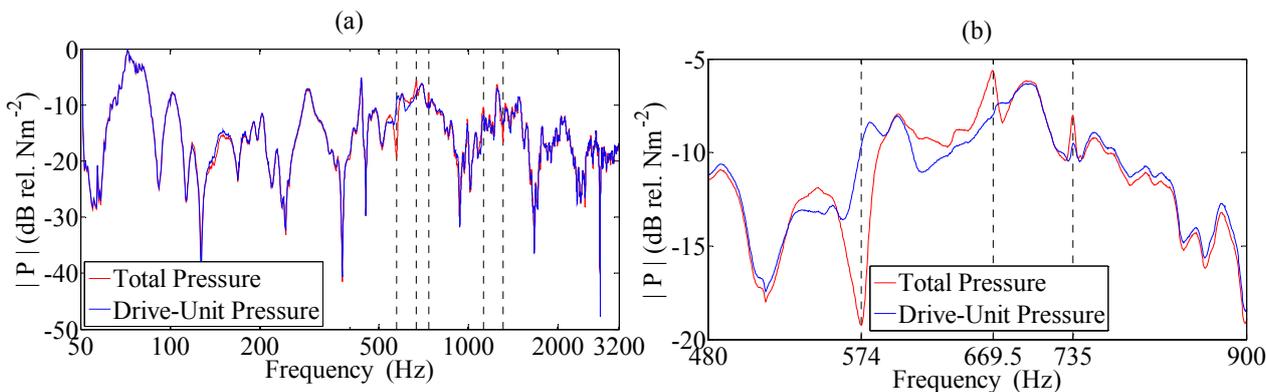
4.5   Listening room pressure contributions

The experiment was repeated at the normal domestic listening room and the pressure contribution of the loudspeaker was then calculated using the equation (2.2) in the theory section.
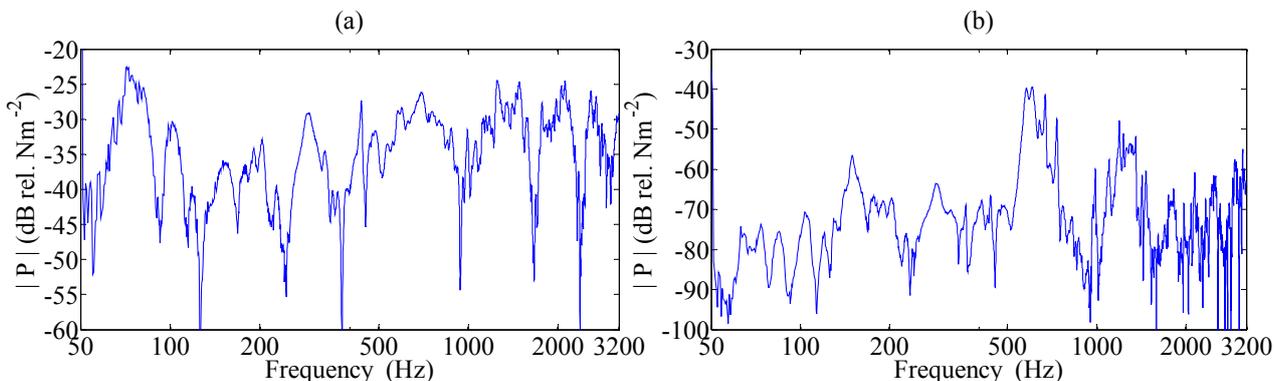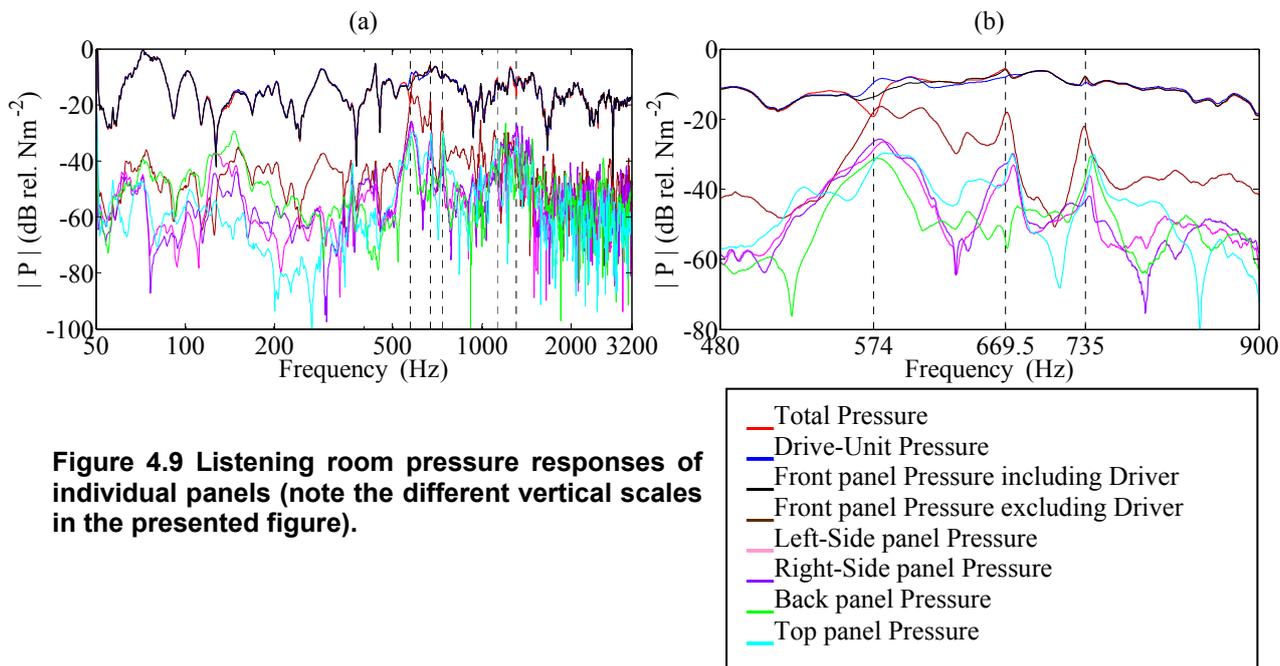

Figure 4.7 Listening room pressure response of point (2, 2) on the driver (a) and Listening room pressure response of point (5 4) on the front panel (b), (note the different vertical scales in the presented figure).

It can be seen from the presented figures that the listening room pressure responses are exhibiting much more erratic behavior. This is due to the effect of the listening room on the individual and invariably total responses. However, the listening room responses following the same pattern as the anechoic responses with drive-unit pressure contribution showing more steady response than any other elements on the loudspeaker cabinet and with higher magnitude.

Figure 4.8 Listening room total pressure response with the driver contribution.

Figure [4.8] shows the comparison between the total pressure (which includes the driver and cabinet response) with the contribution of the drive-unit on its own measured in the normal domestic listening room in most of the frequencies the total listening room response of the loudspeaker is less than the driver contribution on its own. The frequency range of 480 to 900*Hz* has been magnified alongside this figure.

Figure 4.9 Listening room pressure responses of individual panels (note the different vertical scales in the presented figure).

The correlation between the troublesome frequencies in the total listening room response of the loudspeaker (the driver and cabinet contributions) with the response of each individual panel can be observed from the figure [4.9]. The frequency range of 480 to 900*Hz* has been magnified alongside this figure. As in previous part it can also be noticed also that each individual panel exhibits peaks with high magnitude at each of the troublesome frequency. Again as in free-field responses there are some cancellations and some addition of the cabinet response with the driver response can be observed. The contribution of the driver on its own has also been added to this figure to demonstrate, overall it is drive-unit which is the dominant source of pressure contributions.

5   CONCLUSIONS

Through this project the pressure contribution of each individual element on the loudspeaker cabinet and the driver was measured in situ on the axis of the drive-unit in two different conditions. The experiment was first conducted in the free-field condition which was simulated in the large anechoic chamber. Next the experiment was repeated in the normal domestic listening room. The measurements were taken on the axis of the drive-unit since the loudspeaker becomes highly directional at higher frequencies and it is normally intended to be used pointing at the listener in order to create a sweet spot. In order to design a loudspeaker with high channel quality and capable of accurate reproduction of desired sound field, knowing the loudspeaker response becomes crucial. Overall flat response of the loudspeaker is one of the most desired characteristic for any loudspeaker. To achieve this it is important to be able to accurately predict the loudspeaker's response at different frequencies in order to make the necessary adjustments. Provided that the drive-unit of the loudspeaker has a flat response and is capable of correct reproduction of the desired sound field and the input signal to the driver is been adjusted accordingly then there is only the loudspeaker cabinet that has disturbing effect on the reproduced sound field.

It can be seen from the result that controlling the cabinet contribution on the magnitude of the total pressure response of the loudspeaker is necessary since the cabinet has degrading effect on the total outcome of the loudspeaker. The cabinet interferences have two different sorts. One of which is adding unwanted noise to the total pressure response of the loudspeaker. The other form of cabinet interference is cancelling out some of the driver contributions. The unwanted noises made by the cabinet create peaks on the response of the drive unit. Since peaks are generally much more

easily noticeable than dips therefore eliminating the unwanted noise of the loudspeaker cabinet is the first priority. The cabinet also has a destructive interference in some frequencies with the drive-unit contribution which causes some cancelation of the total response of the loudspeaker. Overall it can be concluded to achieve high channel quality from a loudspeaker for high-quality applications, capable of reproduction of the desired sound field a rigid loudspeaker cabinet is not only recommended but necessary.

6   REFERANCES

1. K. R. Holland and F. J. Fahy, *"A guide to the exploitation of vibroacoustic reciprocity in noise control technology,"* ISVR Technical Report, no. 264, pp. 26, 1997.
2. J. W. S. Rayleigh, *The theory of sound. Vol.2. (2nd revised edition)* New York, NY : Dover Publications, 1896.
3. K. R. Holland and F. J. Fahy, *"An investigation into spatial sampling criteria for use in vibroacoustic reciprocity,"* Noise Control Engineering Journal, vol. 45, no. 5, pp. 217-221, Sept.1997.
4. K. R. Holland and F. J. Fahy, *"Application of an Area-Integrating Vibration Velocity Transducer,"* Proceedings - International Conference on Noise Control Engineering, no. 5, pp. 2581, 1996.
5. K. R. Holland and F. J. Fahy, *"Radiation of sound through an aperture in a noise control enclosure via iteration around a finite element-boundary element loop,"* Noise Control Engineering Journal, vol. 44, no. 5, pp. 231-234, 1996.
6. R. S. Langley and V. Cotoni, *"Response variance prediction for uncertain vibro-acoustic systems using a hybrid deterministic-statistical method,"* Journal of the Acoustical Society of America, vol. 122, no. 6, pp. 3445-3463, 2007.
7. K. Bong-Ki and I. Jeong-Guon, *"In situ estimation of an acoustic source in an enclosure and prediction of interior noise by using the principle of vibroacoustic reciprocity,"* Journal of the Acoustical Society of America, vol. 93, no. 5, pp. 2726-2731, May1993.
8. Z. Engel and M. Gawlik. Vibroacoustic Reciprocity Principle in examination of musical instruments. 12th International Conference on Noise Control. Noise Control'01, 23-26 Sept. 2001. Archives of Acoustics 27[4], 263-276. 2002. Kielce, Poland, Polish Scientific Publishers PWN. Arch. Acoust. (Poland).
9. F. J. Fahy, *"Some applications of the reciprocity principle in experimental vibroacoustics,"* Acoustical Physics, vol. 49, no. 2, pp. 217-229, Mar.2003.
10. F. J. Fahy, *"Vibro-Acoustic reciprocity principle and applications to noise control,"* Acustica, vol. 81, no. 6, pp. 544-558, 1995.
11. F. J. Fahy, *Sound intensity, 2nd edition* E & FN Spon, 1996.
12. F. J. Fahy and F. Walker, *Advanced Applications in Acoustics, Noise and Vibration* Spon Press, 1996.
13. F. J. Fahy and P. Gardonio, *Sound and Structural Vibration Radiation, Transmission and Response 2nd edition* Oxford : Academic Press, 2007.
14. C. Gonzalez Diaz, P. Gardonio, and C. Paulitsch, *"Smart panel with decentralised inertial actuator active damping units,"* Journal of the Acoustical Society of America, 2006.
15. C. Gonzalez Diaz, C. Paulitsch, and P. Gardonio, *"Active damping control unit using a small scale proof mass electrodynamic actuator,"* Journal of the Acoustical Society of America, Aug.2008.
16. P. Newell and K. R. Holland, *Loudspeakers: For Music Recording and Reproduction* Amsterdam : Elsevier : Focal Press, 2006.
17. M. Colloms and P. Darlington, *High performance loudspeakers 6th edition* Chichester : Wiley, 2005.
18. Soedel and Werner, *Vibrations of shells and plates 3rd edition, rev. and expanded.* New York, [Great Britain] : Marcel Dekker, 2004.
19. A. W. Leissa, *Vibration of plates* [New York] : Acoustical Society of America through the American Institute of Physics, 1993.