coustics'08
Paris
June 29-July 4, 2008
www.acoustics08-paris.org

Sound Radiation of a Large Truck Oil Pan: Estimation and Experimental Investigation

Olaf Heintze[a], Volker Wittstock[b] and Carl Fredrik Hartung[c]

[a]German Aerospace Center (DLR), Institute of Composite Structures and Adaptive Systems, Lilienthalplatz 7, 38108 Braunschweig, Germany
[b]Physikalisch-Technische Bundesanstalt, Bundesallee 100, 38116 Braunschweig, Germany
[c]Volvo Technology Corporation, Götaverksgatan 10, 405 08 Göteborg, Sweden
olaf.heintze@dlr.de

The oil pan of large diesel engine trucks has been identified as a significant contributor to their external noise radiation. This undesired broadband noise is caused by the oil pan's structural vibration and can not be treated effectively by passive measures especially in the low frequency regime up to 500Hz. In order to address this challenge, an active structural acoustic control system consisting of structural sensors and actuators is suitable to alter the oil pans vibrations in a sound reducing manner. A first step is however to classify the broadband sound radiation such that it allows for a proper and efficient sound power estimation resulting from structural measurements. Therefore, an acoustical model was set up based on a geometrical surface scan of a serial production large truck oil pan mounted in a laboratory test stand. This model served for the numerical computation of a set of principle velocity patterns contributing independently to the active sound power, where its hybrid estimation has been performed employing additionally the measured structural response of the oil pan assembly due to a shaker excitation. Finally, the sound power radiation of the test stand has been measured in a reverberation room to validate this hybrid estimation.

1 Introduction

The constantly increasing road traffic within the European Union (EU) affects the undesired environmental noise in significant manner and causes financial losses as well as health problems [1]. About 80 million people are directly or indirectly impaired by an unacceptable level of road traffic noise, which continuously rose such that the noise level along major highways in Germany is increased by 2.5dB(A) since 1975 and in cities at night time even by 3.0dB(A) [1]. This trend has been mitigated by several successful measures of legislative nature on one hand and by technical improvements on the other. An example for legislative measures is the stepwise reduction of permissible noise emissions per different car types. The technical advances allowed manufacturers to produce even more silent vehicles ahead of these legislative restrictions. Therefore, nowadays cars are about 8dB(A) more silent than thirty years ago and trucks and busses about 11dB(A) [1].

The sources of road traffic noise can be mainly identified as of aeroacoustic origin or as structure borne sound, where the former results from, e.g., turbulent air flow phenomenon and the latter from structural vibrations of vehicle components. Due to future requirements for improved environmental compatibility of upcoming vehicle generations, even more lightweight designs and novel material systems and will be employed to lower fuel consumptions and eventually carbon dioxide emissions. Unfortunately, these lightweight structures are prone to vibrations and may result in even greater noise emissions without an appropriate treatment. Typically, standard measures fail especially in the low frequency regime and require alternative low noise treatment technologies [2]. The "Road Map for Future Noise Reduction Technologies for Road Traffic Noise" [3] outlines that such technologies are essential for a significantly lowered vehicle noise emission until the year 2020.

Major contributors to road traffic noise are heavy-duty trucks, which are present in great numbers and equipped with large diesel engines. It has been identified [3] that forty-three percent of the pass-by noise of a heavy-duty truck can be related to its engine and belonging components, whereas the component with the greatest contribution is the engines oil pan. Typically, the truck oil pan does not contribute to the engine stiffness and, thus, can be designed in a lightweight manner. Its noise radiation can not be efficiently treated by passive measures, where in the low frequency regime these measures are ineffective. Moreover, the lining of the entire engine and its components with passive damping material would result in a challenging heat management that provides a sufficient engine cooling. A possible solution for this challenge is the employment of an active structural acoustic control (ASAC) system that is lightweight and permanently alters the oil pans vibration in a noise reduction manner.

The noise emission of such a vibrating structure that is in direct contact to the ambient air can be immediately antagonized in his causal emergence by such an ASAC system. Structurally implemented sensors and actuators serve for the vibration detection and alternation, and it requires the permanent measurement of the oil pans vibration as well as an acoustic filtering of these signals in order to estimate an acoustic quantity representative for the noise radiation. This control principle has been proven successful for broadband and tonal noise in the frequency range up to 500Hz and even above [8,9,10,11,12].

A key component of an ASAC system is the acoustic filtering of the structures vibration that has to be implemented efficiently in control hardware allowing for a real time control. One possibility to account for this requirement is the employment of principle velocity patterns or radiation modes that independently contribute to the active sound power radiated. Their computation for a large diesel engine oil pan, the sound power estimation and experimental validation are quintessences of the work presented here.

In cooperation with Volvo Trucks, the oil pan of a Volvo MD13 engine has been chosen as the research basis. First, a boundary element mesh has been established from surface measurements of a hardware oil pan and the radiation modes as well as their contribution factors have been computed. This allows for an estimation of the active sound power that is radiated due to a structural excitation. Here, the broadband excitation up to a cut-off frequency of 500Hz was measured in a test stand and caused by a shaker device. Secondly, the actual sound power radiation of the truck oil pan in this previously designed test stand [4] was experimentally determined in a reverberation as well as in a hemi free-field room. Finally, the results of the hybrid estimation approach are compared to the measurements.

2 Simulations

2.1 Set up of boundary element mesh

In preparation for several measurement campaigns, a test stand has been previously designed [4] that allowed for mounting the oil pan assembly as well as the measurement of the frequency response functions (FRF) due to a shaker excitation. This measurement was performed with a laser scanning vibrometer (LSV) at discrete points on the oil pans surface, which coordinates were determined by a digitizer device.

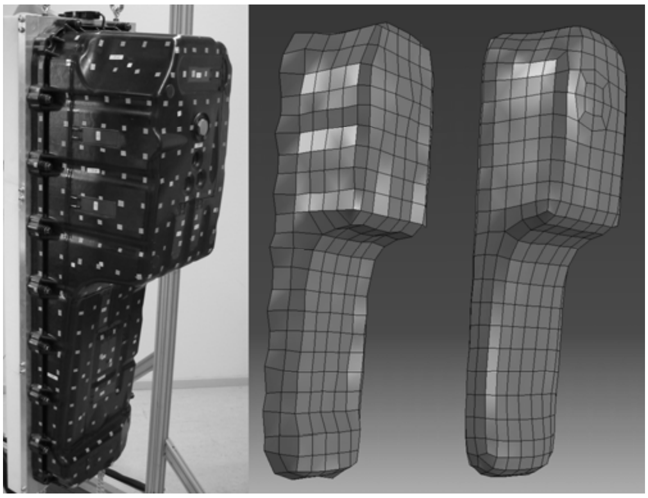

Fig. 1 Truck oil pan in test stand with LSV marks (left), derived intermediate (center) and final mesh (right).

The nodal coordinates of the measurement points formed the basis for an intermediate mesh of the oil pan that was finally transformed into a suitable boundary element mesh by the software VirtualLab.Acoustics. It is valid for frequencies up to 500Hz. Furthermore, the measured FRFs have been additionally interpolated to this final mesh.

2.2 Principle velocity pattern

The computation of the sound radiation is typically performed by a boundary element method for low frequency problems. Unfortunately, this procedure can not be implemented into control hardware and is only feasible on more powerful and stationary computer hardware. Hence, it is desired to extract the essential information for the structures sound radiation and derive a compact scheme that can be implemented in less powerful control hardware.

Several investigations have been published in the early 1990s by, e.g., Borgiotti, Photiadis, Cunefare, Elliott and Johnson [14,13,15,10]. One approach to the solution of this problem, which was presented by Elliott and Johnson, employs the specific acoustic impedance to express the active sound power of a radiating structure in terms of the normal surface velocity and a coupling matrix. This power coupling matrix relates the normal surface velocity and the sound pressure on a discretized surface to each other and can be challenging to compute with great accuracy by BEM. Nevertheless, it can be rearranged by an eigenvalue decomposition featuring an orthogonal set of eigenvectors with associated eigenvalues that contribute independently to the radiated sound power. Additionally, these so called radiation modes or principle velocity pattern are dependent on frequency but do not change in shape significantly within a certain frequency range. Thus, within this frequency range the radiation modes can be assumed as constant without a significant error in the sound power estimation. Such a set of radiation modes is finally suitable to implement into control algorithms and hardware.

Due to the numerical issues computing the radiation modes for a complex shaped object, the method proposed by Photiadis [13] is implemented here. Photiadis relates discrete measurements on the structures surface to the sound pressure in the far field und the radiated sound power. The equivalent set of radiation modes is found by a singular value decomposition since the number of microphone measurements does not necessarily equal that of the structures normal velocity. Expressing the sound power W_f in the far field as

$$\overline{W}_f = 1/2 \int_{\partial S_f} v_{nf}^H(\underline{x}_f) p_f(\underline{x}_f) dS \approx \frac{S_{Ef}}{2\rho_0 c_0} \underline{p}_f \underline{p}_f \quad (1)$$

where the particle velocity is denoted as v_{nf}, the sound pressure as p_f, x_f the position vector in the area spanned by the microphone mesh, dS the incremental area of the microphone mesh, S_{Ef} an elementary area in the fluid region, ρ_0 the fluid density and c_0 the speed of sound. Furthermore, the following relation between the structures normal surface velocity v_{ns} and the sound pressure in the far field holds:

$$\underline{p}_f = \underline{\underline{G}} \underline{v}_{ns} = \underline{\underline{U}}\,\underline{\underline{S}}\,\underline{\underline{V}}^H \underline{v}_{ns} \quad (2)$$

The Green's function G is computed by the BEM software Sysnoise and exported for postprocessing to the mathematic software Matlab. In Matlab the singular value decomposition is performed resulting in the unitary matrices U and V as well as the diagonal matrix S holding the real positive singular values. Therefore, Eq. (1) can be re-written in the following form:

$$\overline{W}_f = \frac{S_{Ef}}{2\rho_0 c_0} \underline{v}_{ns}^H \underline{\underline{V}}\,\underline{\underline{\Lambda}}\,\underline{\underline{V}}^H \underline{v}_{ns} \quad (3)$$

In Eq. (3) the diagonal matrix Λ contains the squared singular values that weight the independent contributions of the structures normal velocities filtered by the radiation modes matrix V. The first four radiation modes for the oil pan are depicted in Fig. 2 at the frequency of 50Hz.

Fig. 2 First four radiation modes of oil pan at frequency of 50Hz (order increases clockwise starting at the top left).

In Fig. 3 the respective contribution factors of the first four radiation modes to the active sound power are plotted, which are the greatest four weighted singular values represented by the expression $\left(S_{Ef}/2\rho_0 c_0\right)\underline{\underline{\Lambda}}$.

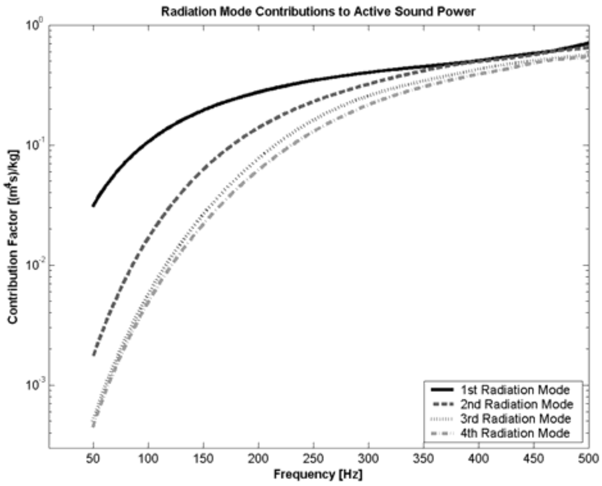

Fig. 3 Contributions of the radiation modes 1-4 to the active sound power.

2.3 Hybrid sound power estimation

In the laboratory test stand of the oil pan, a shaker device is employed to provide a structural excitation. This excitation has been measured by a load cell, and a band limited white noise force spectrum from 0-500Hz has been set for the entire test campaign including the sound radiation measurements. Previously derived frequency response functions were weighted by the force spectrum to achieve the structures normal surface velocities, and in combination with the set of radiation modes the sound power spectrum of the vibrating oil pan was computed. Employing Eq. (3) including the radiation modes and its contribution factors, the active sound power spectrum is depicted in Fig. 4. This active sound power spectrum features the frequency range from 50-500Hz with a sample frequency of 1Hz.

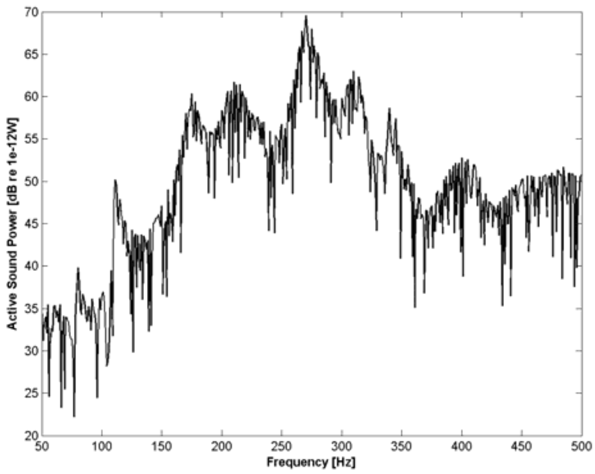

Fig. 4 Numerically computed active sound power employing the radiation mode approach.

3 Measurements

3.1 General remarks

The aim of the measurement campaign is the verification of numerically estimated sound radiation of the truck oil pan. The oil pan was mounted in a test rig and excited by shaker device in a broadband manner at frequencies between 0 and 500Hz. The excitation force was measured and no active control has been engaged. In preparation of measurements to determine the efficiency of an ASAC system, piezoceramic patch actuators have been already applied to the oil pan but were disabled during the measurements presented here.

3.2 Reverberation room

The sound power radiated by the oil pan was determined in a reverberant room at the Physikalisch-Technische Bundesanstalt (PTB). This non rectangular room has a volume of 204m³ and a surface of 210m². It is equipped with 8 stationary diffusers of about 1.5m² each. The measured reverberation time with the oil pan in the room is between 26s at 50Hz and about 7s at 1kHz (Fig. 5).

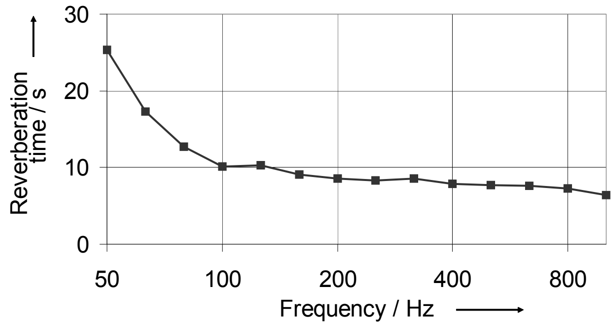

Fig. 5 Measured reverberation time in the non rectangular reverberation room at PTB.

The level of the active sound power was determined according to ISO 3741 [5] by

$$L_W = L_p - 10\lg\frac{T}{T_0}\,\mathrm{dB} + 10\lg\frac{V}{V_0}\,\mathrm{dB} + 10\lg\left(1+\frac{S\,\lambda}{8\,V}\right)\mathrm{dB} - 10\lg\left(\frac{B}{1000}\right)\mathrm{dB} - 14\,\mathrm{dB} \tag{4}$$

where the volume average of the sound pressure level is L_p, T the reverberation time, V the volume, S the surface area, λ the acoustic wavelength, and B the static pressure. The reference values are T_0 = 1s and V_0 = 1m³. The static pressure was 1002hPa and the temperature 17°C.

One source position and 6 microphone positions were used for the sound pressure level measurements.

3.3 Hemi free-field room

The measurement uncertainty of sound power determinations in reverberant rooms is relatively large for the low frequencies especially for tonal sources. To verify the measurement results and to obtain narrow band results, additional measurements were carried out in the hemi free-field room at PTB. This room has a volume of 190m³ and meets the requirements of Annex A of ISO 3745 [6] for tonal excitation between 60Hz and 20kHz for measurement distances up to 2m.

Measurements were carried out by the intensity method according to ISO 9614-1 [7]. The box shaped measurement surface had a distance of 10 cm to the reference box. It was covered by 48 measurement points, each representing a surface element of 0.3 *x* 0.3m². The measurement was carried out by a two-microphone probe with a 50mm spacer. It was oriented perpendicular to the measurement surface.

The measurement signals from the two-microphone probe were analyzed by two methods. A real time analyzer was used to determine one third-octave band values of the intensity and of the sound pressure directly. Simultaneously, the cross spectrum G_{12} of the two microphone signals as well as the auto spectra were determined by an FFT-analyzer. The frequency resolution was 1.25Hz in a frequency range between 0 and 1kHz. A Hanning window was used. The averaging time was 3 minutes for each point. The static pressure was 990hPa and the temperature 20°C.

The one third-octave band results were evaluated in accordance with ISO 9614-1 [7]. The active sound power is then

$$P = \frac{S}{N}\sum_{i=1}^{N} I_{n,i} \qquad (5)$$

where the $I_{\mathrm{n},i}$ are the measured intensities, S is the measurement surface and N the number of measurement points (N = 48). In addition, the reactive sound power was determined by

$$Q = \sqrt{\left(\frac{S}{N\,\rho\,c}\sum_{i=1}^{N} p_i^2\right)^2 - P^2}\,. \qquad (6)$$

Here, the sound pressures on the surface are p_i whereas ρ and c are the density and speed of sound in air, respectively.

The narrow band intensities are calculated from the imaginary part of the cross spectrum by

$$I = -\frac{1}{\rho\,\omega\,\Delta r}\mathrm{Im}\{G_{12}\} \qquad (7)$$

with the spacing between the microphones Δr and the angular frequency ω. The narrow band active and reactive sound powers are then determined by Eqs. (5) and (6).

3.4 Sound power results

The results of the active sound power levels determined in the reverberation room and in the hemi free-field room are in accordance (Fig. 6) for most of the one third-octave bands under consideration. The stated uncertainties are the 95% confidence intervals calculated from the standard deviations of reproducibility given in ISO 3741 (1) and ISO 9614-1 [7] by a coverage factor of 2. Even though there was no excitation above 500Hz the measured sound power levels were well above the background noise up to frequencies of 1kHz. The only significant discrepancy between the measured sound power levels is observed at 800Hz and 1kHz. Since the excitation in this frequency range is due to nonlinearities, a strong dependency on the exact mounting conditions of the shaker and the oil pan may be assumed. Since the whole setup had to be dismounted for the transport from the reverberation room to the hemi free-field room the mounting conditions may be different in both setups which explains the discrepancies.

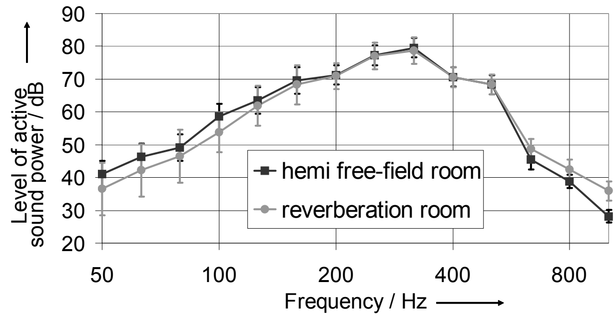

Fig. 6 Levels of the active sound power re 1 pW, one third-octave band results obtained in the hemi free-field room and in the reverberation room.

The narrow band results clearly show the cut-off frequency of the excitation at 500Hz (Fig. 7). It is furthermore observed that the reactive sound power is about 2 to 3dB larger than the active sound power. Due to the short measurement distance, the measurements are carried out in the near-field where the reactivity of the sound field is large. The same behaviour is observed for the one third-octave band results determined in the hemi free-field room (Fig. 8).

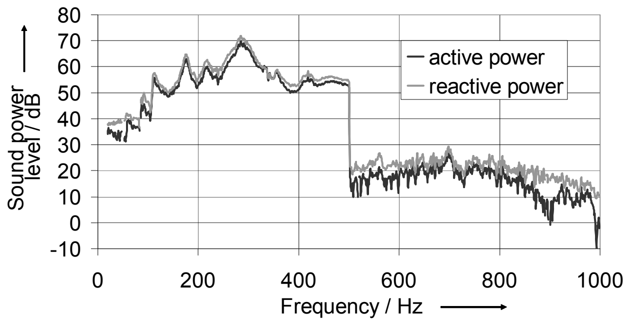

Fig. 7 Levels of the active and the reactive sound power re 1 pW, FFT results obtained in the hemi free-field room.

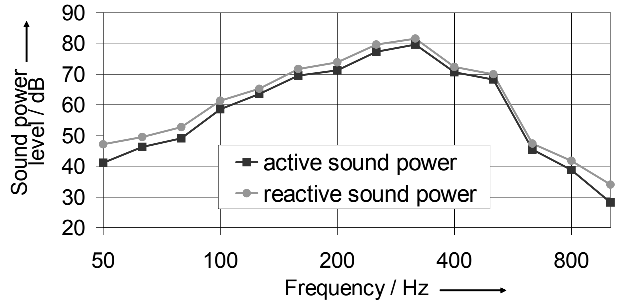

Fig. 8 Levels of the active and the reactive sound power re 1 pW, one third-octave band results obtained in the hemi free-field room.

4 Comparison between measurements and simulations

The measures active sound power and the numerically computed one for the band-limited shaker excitation up to 500Hz. In Fig. 9 these results are compared in third-octave bands and display a remarkable agreement well within the measurements uncertainty limits. Difference can furthermore be explained by comparing Fig. 4 and Fig. 7. The spectra reveal a shift in the major peaks, which results from a variation in the offline measured frequency response functions that altered in between the experiments.

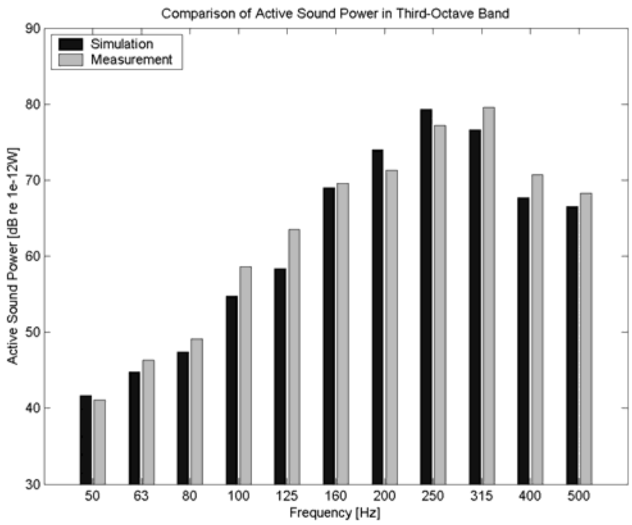

Fig. 9 Comparison of numerically computed and measured active sound power in third-octave bands.

5 Conclusion

The active sound power radiated by a truck oil pan has been computed, and the numerical estimation was validated by different measurements in a reverberation and a hemi free-field room. Furthermore, the numerical estimation approach presented employed the radiation mode technique suitable to implement the relevant information for the structures sound radiation into control hardware. Therefore, boundary element computations have been performed and post-processed to actually extract the radiation modes as well as its contribution factors to the radiated sound power. The comparison of simulation and measurement reveals a very good agreement and, thus, validates this compact expression for the sound power radiation applied to structures of complex shape. Future work will focus on the implementation of the numerically computed radiation modes into control hardware.

Acknowledgments

Part of the work was funded by the European research project InMAR. Measurements have been carried out by Christian Bethke and Heinrich Bietz (both Physikalisch-Technische Bundesanstalt, Braunschweig). The authors appreciate the productive cooperation with Volvo Trucks.

References

[1] J. Affenzeller and A. Rust, “Road Traffic Noise – a Topic for Today and the Future”, VDA – Technical Congress 2005, Ingolstadt, Germany, 16th and 17th March 2005 (2005)

[2] M. Redaelli *et al*, „Different Techniques for Active and Passive Noise Cancellation at Powertrain Oil Pan”, Adaptronic Congress 2007, 23 – 24 May, Göttingen, Germany (2007)

[3] N. N., “Research for a Quieter Europe in 2020”, Updated Strategy Paper of the CALM Network, February 2007, http://www.calm-network.com (2007)

[4] O. Heintze, “Active Structural Acoustic Control for a Truck Oil Pan: Actuator Placement and Efficiency Estimation”, Noise-Con 2007, October 21 – 24, Reno, NV, USA (2007)

[5] ISO 3741:1999 Acoustics - Determination of sound power levels of noise sources using sound pressure - Precision methods for reverberation rooms

[6] ISO 3745:2003 Acoustics - Determination of sound power levels of noise sources using sound pressure - Precision methods for anechoic and hemi-anechoic rooms

[7] ISO 9614-1:1993 Acoustics -- Determination of sound power levels of noise sources using sound intensity -- Part 1: Measurement at discrete points

[8] G. P. Gibbs, R. L. Clark, D. E. Cox and J. S. Vippermann, “Radiation Model Expansion for Acoustic Control”, *J. Acoust. Soc. Am.* 107(1), 332-339 (2000)

[9] M. H. H. Oude Nijhuis, “Analysis Tools for the Design of Active Structural Acoustic Control Systems”, PhD Thesis, University of Twente, Enschede, The Netherlands (2003)

[10] S. J. Elliott and M. E. Johnson, “Radiation Modes and the Active Control of Sound Power”, *J. Acoust. Soc. Am.* 94(4), 2194-2204 (1993)

[11] T. C. Sors and S. J. Elliott, “Volume Velocity Estimation with Accelerometer Arrays for Active Structural Acoustic Control”, *J. Sound Vibr.* 258 (5), 867-883 (2002)

[12] T. Weyer, E. Breitbach, O. Heintze, “Self-Tuning Active Electromechanical Absorbers for Tonal Noise Reduction of a Car Roof”, INTER-NOISE 2007, 28-31 August, Istanbul, Turkey (2007)

[13] D. M. Photiadis, “The Relationship of Singular Value Decomposition to Wave-Vector Filtering in Sound Radiation Problems”, *J. Acoust. Soc. Am.* 88(2), 1152-1159 (1990)

[14] G. V. Borgiotti, “The Power Radiated by a Vibrating Body in an Acoustic Fluid and its Determination from Boundary Measurements”, *J. Acoust. Soc. Am.* 88(4), 1884-1893 (1990)

[15] K. A. Cunefare, “The Minimum Multimodal Radiation Efficiency of Baffled Finite Beams”, *J. Acoust. Soc. Am.* 90(5), 2521-2529 (1991)