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Operational Acoustic Modal Analysis: Sensitivity-Based Mode Shape Normalisation

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Summary
Nowadays Acoustic Modal Analysis often plays a significant role in the optimisation process of the acoustic parameters of all kinds of cavities, e.g. the interior of a car, a train, an airplane etc. A problem with the existing experimental Acoustic Modal Analysis techniques is that one has to use volume acceleration sources which interact with the acoustic system. The Operational Acoustic Modal Analysis (OAMA) technique doesn't suffer from this problem. An important disadvantage of the OAMA approach is that the estimated mode shapes are not correctly scaled. This scaling problem will be tackled in this paper. We will introduce a sensitivity based rescaling technique for the acoustic domain. Local volume changes of the cavity will make a correct estimate of the mode shapes possible. The technique will be validated by some simulations and experiments.

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1. Introduction

Today, Modal Analysis is a well known engineering tool. In mechanics, modal analysis is already well integrated in the product development process [1, 2]. Finite Element Analysis (FEA) guarantees robust virtual models and Experimental Modal Analysis (EMA) based on Frequency Response Functions (FRF) also is a well-known testing technique. Both can be combined in hybrid models [3]. Those models combine measurement data and simulation data. Also the updating of the analytical model based on experimental data has already been used in industrial applications [4, 5]. The fact that recent estimators can handle high damping values enlarges the domain of applications [6, 7, 8].

In acoustics, sound quality is a topic of increasing importance. As a result of this, also acoustic modal analysis (AMA) gains interest. Especially in the transport sector (cars, busses, trains, airplanes, ...) acoustic modal analysis and vibro-acoustic modal analysis are already well integrated in the product development process [9, 10, 11]. However there are some disadvantages which explain why experimental acoustic modal analysis is not yet widely used in a lot of application areas. A first difficulty is the fact that the equivalent of the mechanical force in structural modal analysis in acoustic modal analysis a volumetric acceleration is. So, to perform an experimental acoustic modal analysis one needs calibrated volumetric acceleration sound sources which are not widely available. Another problem concerning the sound sources is that they have to be omnidirectional. Almost all existing sound sources are directive [12, 13]. A second problem concerning the sources is that they always have a non-negligible size. So they change the volume of the cavity altering the acoustic system [14].

It is interesting to note that also in the domain of civil engineering it is more difficult to perform an EMA because an EMA needs input-output data [15, 16]. Thus one has to know the forces. Compared to small mechanical systems the forces have to be much bigger. Although some special impulse devices and large electrodynamic shakers have been designed to generate those big forces [15], EMA remains tough for big structures.

One can conclude that for both the domain of civil engineering and the domain of acoustics the problems with the sources make it difficult to perform an experimental modal analysis. Over the past years, a modal identification technique that uses only output data has been developed: operational modal analysis (OMA). It typically uses auto- and cross-power spectra in stead of frequency response functions (FRF's) [17]. The only disadvantage of OMA compared to EMA is that the mode shapes are not correctly scaled. However, for several applications this is not important. In these cases OMA is an interesting alternative for EMA [18]. Especially in the domain of civil engineering OMA has been welcomed as a powerful system identification tool [15].

As mentioned before, until recent it was impossible to calculate correctly scaled mode shapes from a structure using OMA. Recently, several rescaling methods have been introduced. A solution for the structural OMA is the sensitivity-based approach [19, 20]. This approach is based on the fact that when one adds known masses to the

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structure, the natural frequencies shift. By measuring the frequency shift it is possible to calculate the correct scaling factors. The advantage of this approach is that no additional forces have to be applied. This sensitivity-based rescaling method now makes it feasible to calculate all the modal parameters without using known forces. It is obvious that this solves the problem of EMA in the domain of civil engineering [21]. Recently, the technique has been validated on real-life civil structures (e.g. bridges) [22]. Also in the domain of mechanical engineering, OMA in combination with the rescaling method has been tested [23].

If it would be possible to find an equivalent approach for the domain of acoustics, one would not need the volume acceleration sources anymore. This would be very beneficial because then one could estimate all the modal parameters using only microphones and the operational sound sources. Also there would be no influence of the sound sources which one needs with an EMA. In this paper we will show that such an approach exists. The acoustic sensitivity-based rescaling method will be based on local volume changes in the cavity which is easy to apply.

2. Theoretical aspects

2.1. Analogy between acoustic and mechanical systems

Before we focus on the sensitivity based rescaling technique the analogy between acoustic and mechanical systems will be briefly recapitulated. Dynamic undamped mechanical systems can be modelled in the frequency-domain as:

$$\left(-\omega^2 \cdot \mathbf{M^s} + \mathbf{K^s}\right) \cdot \mathbf{x}(\omega) = \mathbf{f}(\omega), \tag{1a}$$

with

$\mathbf{M^s}$ the structural mass matrix,
$\mathbf{K^s}$ the structural stifness matrix,
$\mathbf{x}(\omega)$ the displacement vector,
$\mathbf{f}(\omega)$ the force vector. (1b)

For dynamic undamped acoustic systems the equations are:

$$\left(-\omega^2 \cdot \mathbf{M^f} + \mathbf{K^f}\right) \cdot \mathbf{p}(\omega) = \dot{\mathbf{q}}(\omega), \tag{2a}$$

with

$\mathbf{M^f}$ the acoustic equivalence of the mass matrix,
$\mathbf{K^f}$ the acoustic equivalence of the stifness matrix,
$\mathbf{p}(\omega)$ the pressure vector,
$\dot{\mathbf{q}}(\omega)$ the volume acceleration vector. (2b)

Note that the acoustic equivalence of the structural mass matrix in fact consists of acoustic compliances. This acoustic compliance is defined as the ratio between the volume displacement and the pressure [24, 25].

$$M^{\mathrm{f}}_{kl} = \begin{cases} 0 & \text{for } k \neq l, \\ \frac{V_k}{\rho c^2} & \text{for } k = l, \end{cases} \tag{3a}$$

Table I. Analogy between acoustic and mechanical systems.

Variable for Structural Modal Analysis	Variable for Acoustic Modal Analysis
$\mathbf{\Phi^s}$	$\mathbf{\Phi^f}$
$\mathbf{M^s}$	$\mathbf{M^f}$
$\mathbf{K^s}$	$\mathbf{K^f}$
$\mathbf{x}$	$\mathbf{p}$
$\mathbf{f}$	$\dot{\mathbf{q}}$
$\bar{m}^{\mathrm{s}}$	$\bar{m}^{\mathrm{f}}$

with

V_k the volume of element k of the cavity,
ρ the density of the medium,
c the speed of sound. (3b)

Also the FRF-matrices are comparable. One can prove that the FRF-matrix of an undamped structural system $\mathbf{H^s}(\omega)$ can be calculated as [26]:

$$H^{\mathrm{s}}_{kl}(\omega) = \sum_{n=1}^{N^{\mathrm{s}}_m} \frac{\Phi^{\mathrm{s}}_{kn} \cdot \Phi^{\mathrm{s}}_{ln}}{\bar{m}^{\mathrm{s}}_n \cdot ({\omega^{\mathrm{s}}_n}^2 - \omega^2)} \tag{4a}$$

with

$\bar{m}^{\mathrm{s}}_n$ the structural modal mass of mode n
$= \sum_{k=1}^{N^{\mathrm{s}}_m} \sum_{l=1}^{N^{\mathrm{s}}_m} \Phi^{\mathrm{s}}_{kn} \cdot M^{\mathrm{s}}_{kl} \cdot \Phi^{\mathrm{s}}_{ln}$,
Φ^{s}_{kn} the structural mode shape value of mode n at point k,
ω^{s}_n the eigenfrequency of mode n,
N^{s}_m the number of modes. (4b)

The acoustic equivalent equation is [27]

$$H^{\mathrm{f}}_{kl}(\omega) = \sum_{n=1}^{N^{\mathrm{f}}_m} \frac{\Phi^{\mathrm{f}}_{kn} \cdot \Phi^{\mathrm{f}}_{ln}}{\bar{m}^{\mathrm{f}}_n \cdot ({\omega^{\mathrm{f}}_n}^2 - \omega^2)}, \tag{5a}$$

with

$\bar{m}^{\mathrm{f}}_n$ the acoustic equivalence of the modal mass of mode n
$= \sum_{k=1}^{N^{\mathrm{f}}_m} \sum_{l=1}^{N^{\mathrm{f}}_m} \Phi^{\mathrm{f}}_{kn} \cdot M^{\mathrm{f}}_{kl} \cdot \Phi^{\mathrm{f}}_{ln}$,
Φ^{f}_{kn} the acoustic mode shape value of mode n at point k,
ω^{f}_n the eigenfrequency of mode n,
N^{f}_m the number of modes. (5b)

If one compares (4) to (5) one can conclude that the same formula's can be used for the sensitivity analysis if one uses the correct variables. Table I shows the equivalent variables.

Note that $\bar{m}^{\mathrm{f}}_n \cdot \rho \cdot c^2$ is nothing less than the volume matrix of the cavity weighted by the mode shape of mode n.

2.2. Sensitivity-based rescaling

For structural operational modal analysis a sensitivity-based rescaling method recently has been developed [19]. For an undamped system the sensitivity of the natural frequency ω_n of mode n to a local change in mass in degree of freedom i is given by [26]

$$\frac{\partial \omega_n^{\mathrm{s}}}{\partial m_i} = -\omega_n^{\mathrm{s}} \frac{{\Phi_{in}^{\mathrm{s}}}^2}{2 \cdot \bar{m}_n^{\mathrm{s}}}. \tag{6}$$

Solving this equation in a finite difference approximation results in

$$\Delta\omega_n^{\mathrm{s}} = -\omega_n^{\mathrm{s}} \frac{{\Phi_{in}^{\mathrm{s}}}^2}{2 \cdot \bar{m}_n^{\mathrm{s}}} \cdot \Delta M_i^{\mathrm{s}}, \tag{7a}$$

with

$$\begin{array}{ll} \Delta\omega_n^{\mathrm{s}} & \text{the shift in eigenfrequency } \omega_n^s \text{ of mode } n, \\ \Delta M_i^{\mathrm{s}} & \text{the change in mass in degree of freedom } i. \end{array} \tag{7b}$$

If the mass has been changed in N_c degrees of freedom, equation (7a) becomes:

$$\Delta\omega_n^{\mathrm{s}} = -\frac{\omega_n^{\mathrm{s}}}{2 \cdot \bar{m}_n^{\mathrm{s}}} \cdot \left(\sum_{i=1}^{N_c} {\Phi_{in}^{\mathrm{s}}}^2 \cdot \Delta M_i^{\mathrm{s}} \right). \tag{8}$$

Because the operational mode shapes $\boldsymbol{\Psi}^{\mathbf{s}}$ are not correctly scaled, one can write that each scaled mode shape value Φ_{in}^{s} equals the operational mode shape value Ψ_{in}^{s} multiplied by a scaling factor α_n^{s}:

$$\Phi_{in}^{\mathrm{s}} = \Psi_{in}^{\mathrm{s}} \cdot \alpha_n^{\mathrm{s}}. \tag{9}$$

Finally, the scaling factor can be calculated by combining equations (8) and (9)

$$\alpha_n^{\mathrm{s}} = \sqrt{\frac{-2 \cdot \bar{m}_n^{\mathrm{s}} \cdot \Delta\omega_n^{\mathrm{s}}}{\omega_n^{\mathrm{s}} \cdot \left(\sum_{i=1}^{N_c} {\Psi_{in}^{\mathrm{s}}}^2 \cdot \Delta M_i^{\mathrm{s}} \right)}}. \tag{10}$$

If one wants to use a mass-normalised scaling-scheme equation (10) becomes:

$$\alpha_n^{\mathrm{s}} = \sqrt{\frac{-2 \cdot \Delta\omega_n^{\mathrm{s}}}{\omega_n^{\mathrm{s}} \cdot \left(\sum_{i=1}^{N_c} {\Psi_{in}^{\mathrm{s}}}^2 \cdot \Delta M_i^{\mathrm{s}} \right)}}. \tag{11}$$

For Acoustic Operational Modal Analysis the same technique can be used. If one uses the acoustic equivalent variables that are listed in Table I equation (6) becomes

$$\frac{\partial \omega_n^{\mathrm{f}}}{\partial V_i} = -\omega_n^{\mathrm{f}} \frac{{\Phi_{in}^{\mathrm{f}}}^2}{2 \cdot \rho \cdot c^2 \cdot \bar{m}_n^{\mathrm{f}}}. \tag{12}$$

Using the same equivalent variables equation (10) becomes

$$\begin{aligned} \alpha_n^{\mathrm{f}} &= \sqrt{\frac{-2 \cdot \bar{m}_n^{\mathrm{f}} \cdot \Delta\omega_n^{\mathrm{f}}}{\omega_n^{\mathrm{f}} \cdot \left(\sum_{i=1}^{N_c} {\Psi_{in}^{\mathrm{f}}}^2 \cdot \Delta M_i^{\mathrm{f}} \right)}} \\ &= \sqrt{\frac{-2 \cdot \frac{\bar{V}_n^{\mathrm{f}}}{\rho c^2} \cdot \Delta\omega_n^{\mathrm{f}}}{\omega_n^{\mathrm{f}} \cdot \left(\sum_{i=1}^{N_c} {\Psi_{in}^{\mathrm{f}}}^2 \cdot \Delta \frac{V_i}{\rho c^2} \right)}} \\ &= \sqrt{\frac{-2 \cdot \bar{V}_n^{\mathrm{f}} \cdot \Delta\omega_n^{\mathrm{f}}}{\omega_n^{\mathrm{f}} \cdot \left(\sum_{i=1}^{N_c} {\Psi_{in}^{\mathrm{f}}}^2 \cdot \Delta V_i \right)}}, \end{aligned} \tag{13}$$

with

$$\bar{V}_n^{\mathrm{f}} = \sum_{k=1}^{N_m^{\mathrm{f}}} \sum_{l=1}^{N_m^{\mathrm{f}}} \Phi_{kn}^{\mathrm{f}} \cdot V_{kl} \cdot \Phi_{ln}^{\mathrm{f}}. \tag{14}$$

For acoustic mode shapes one can use the volume-normalised scaling scheme. Equation (13) becomes

$$\alpha_n^{\mathrm{f}} = \sqrt{\frac{-2 \cdot \Delta\omega_n^{\mathrm{f}}}{\omega_n^{\mathrm{f}} \cdot \left(\sum_{i=1}^{N_c} {\Psi_{in}^{\mathrm{f}}}^2 \cdot \Delta V_i \right)}}. \tag{15}$$

From equation (15) one can conclude that one now needs to add or subtract some local volume instead of some local mass. Doing this and measuring the frequency shift of the eigenfrequencies one is able to calculate the scaling factors of the acoustic mode shapes. Note that the second analysis is only needed to estimate the frequency shift and that no mode shape values are needed. So it is not necessary to measure in all the measurement locations!

From equation (15) one also can conclude that some local volume changes are more interesting than other ones. Knowing that the procedure is based on local volume changes, one wants to keep those changes as small as possible. On the other hand the frequency shift has to be as big as possible so that the influence of the frequency estimation errors are as small as possible. In fact the sensitivity which is defined in equation (12) has to be as big as possible. From this equation follows that the best locations for changing the volume are the maxima of the mode shapes (the antinodes). Note that one already knows the antinodes after the first step of the procedure. The sensitivity becomes zero for volume changes in the nodes. So applying local volume changes in the nodes has to be avoided.

Note finally that with this technique it is possible to calculate the FRF's without using a volumetric acceleration sound source.

3. Analytical validation

In this section the proposed procedure will be validated by means of an example which has an analytical solution. Suppose one wants to know the scaled mode shapes of a simple acoustic system: a box with dimensions $L_x \times L_y \times$

L_z. Suppose one adds some volume in the x-direction. This results in a box with dimensions: $(L_x+\epsilon)\times L_y \times L_z$. It is obvious that if the left hand side of equation (12) equals the right hand side, the proposed procedure will work. One can calculate the eigenfrequency of the unchanged box using the following analytical solution for the eigenfrequencies (with n_x, n_y and n_z non-negative integers):

$$\omega_n^f = \sqrt{\left(\frac{n_x\cdot\pi\cdot c}{L_x}\right)^2 + \left(\frac{n_y\cdot\pi\cdot c}{L_y}\right)^2 + \left(\frac{n_z\cdot\pi\cdot c}{L_z}\right)^2}. \quad (16)$$

The analytical solution for the mode shape values is:

$$\Phi^f(x,y,z) = \cos\frac{n_x\cdot\pi\cdot x}{L_x}\cdot\cos\frac{n_y\cdot\pi\cdot y}{L_y}\cdot\cos\frac{n_z\cdot\pi\cdot z}{L_z}. \quad (17)$$

So the left hand side of equation (12) becomes:

$$\frac{\partial\omega_n^f}{\partial V_i} = \frac{1}{L_y\cdot L_z}\cdot\frac{\partial\omega_n^f}{\partial L_x} = -\frac{1}{L_x\cdot L_y\cdot L_z}\cdot\frac{1}{\omega_n^f}\cdot\left(\frac{n_x\cdot\pi\cdot c}{L_x}\right)^2, \quad (18)$$

and the right hand side of equation (12) becomes

$$-\omega_n^f\frac{{\Phi_{in}^f}^2}{2\cdot\rho\cdot c^2\cdot\bar{m}_n^f} = -\omega_n^f\frac{\left(\cos\frac{n_x\pi x}{L_x}\cdot\cos\frac{n_y\pi y}{L_y}\cdot\cos\frac{n_z\pi z}{L_z}\right)^2}{2\cdot\rho\cdot c^2\cdot\bar{m}_n^f}. \quad (19)$$

One added some volume at the end of the box in the x-direction. At the location where the volume was added all mode shapes in the x-direction have an x-value $\cos\frac{n_x\pi L_x}{L_x} = 1$. The mode shapes in the y-direction and the z-direction will vary between -1 and 1. So one has to take the average of the sensitivity. So equation (19) becomes

$$-\omega_n^f\frac{{\Phi_{in}^f}^2}{2\cdot\rho\cdot c^2\cdot\bar{m}_n^f} = -\frac{\omega_n^f}{2\cdot\rho\cdot c^2\cdot\bar{m}_n^f}\cdot\frac{1}{L_y\cdot L_z}\cdot \int_{y=0}^{L_y}\int_{z=0}^{L_z}\left(\cos\frac{n_y\cdot\pi\cdot y}{L_y}\right)^2\cdot\left(\cos\frac{n_z\cdot\pi\cdot z}{L_z}\right)^2 \mathrm{d}y\,\mathrm{d}z. \quad (20)$$

In equation (20) one has to substitute $\bar{m}_n^f$:

$$\bar{m}_n^f = \frac{1}{\rho\cdot c^2}\left[\int_{x=0}^{L_x}\int_{y=0}^{L_y}\int_{z=0}^{L_z}\Phi_n^f(x,y,z)^2\right]. \quad (21)$$

The result of this calculation is

$$\bar{m}_n^f = \frac{L_x\cdot L_y\cdot L_z}{8\cdot\rho\cdot c^2}. \quad (22)$$

After substitution of $\bar{m}_n^f$ using equation (22) and substitution of ω_n^f using equation (16), equation (20) can be integrated,

$$-\omega_n^f\frac{{\Phi_{in}^f}^2}{2\cdot\rho\cdot c^2\cdot\bar{m}_n^f} = -\frac{1}{L_x\cdot L_y\cdot L_z}\cdot\frac{1}{\omega_n^f}\cdot\left[\left(\frac{n_x\cdot\pi\cdot c}{L_x}\right)^2 + \left(\frac{n_y\cdot\pi\cdot c}{L_y}\right)^2 + \left(\frac{n_z\cdot\pi\cdot c}{L_z}\right)^2\right]. \quad (23)$$

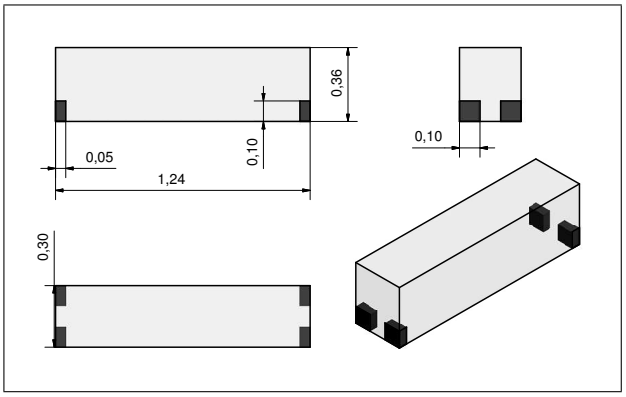


Figure 1. Box with local volume changes.

One can conclude that equation (18) only equals equation (23) when $n_y = 0$ and $n_z = 0$. The procedure will fail for modes that have a $n_y \neq 0$ or $n_z \neq 0$. The reason is that for those modes the volume change is not local anymore. The mode shapes in the y-direction and the z-direction will vary between -1 and 1 at the added volume. One can conclude that the procedure will work when all volume changes meet the condition that the mode shape is constant on the whole surface of each volume.

4. Simulations and experiments

In this section the proposed procedure will be validated. First it will be validated using a 3-D finite element simulation. In a second step the procedure is validated by an experiment. For both cases the acoustic system was a box with dimensions 1.24 m × 0.3 m × 0.36 m (see Figure 1 for the geometrical configuration).

4.1. 3-D computer simulation

In a first simulation the exact solutions have been calculated by means of a finite element analysis (FEA). The finite element model contained 39479 tetrahedral elements. Using this model the volume-normalised mode shapes ($\Phi_{kn}^f/\sqrt{\bar{V}_n^f}$) were calculated in two ways.

1. The first method used the raw FEA-data and also calculated $\bar{V}_n^f$ using the FEA-data and available integration functions. Note that the data of all nodes of the finite element model was used during the calculations.
2. The second method calculated the scaled mode shapes Φ_{kn}^f with the proposed procedure. So the simulation data of a first simulation (the box without the four volume changes) was compared to the data of the second simulation (the box with the four volume changes) (see Figure 1). During a real measurement it would be impossible to measure the pressure in all nodes. So a grid of 90 points (see Figure 2) was used to simulate a measurement and only those 90 points were used during the calculations. The frequency shift and the 4 known volume changes were used to calculate the scaling factors α_n^f using equation (15). Note that the $\bar{V}_n^f$ too were calculated using only the 90 grid points. Using the scaling

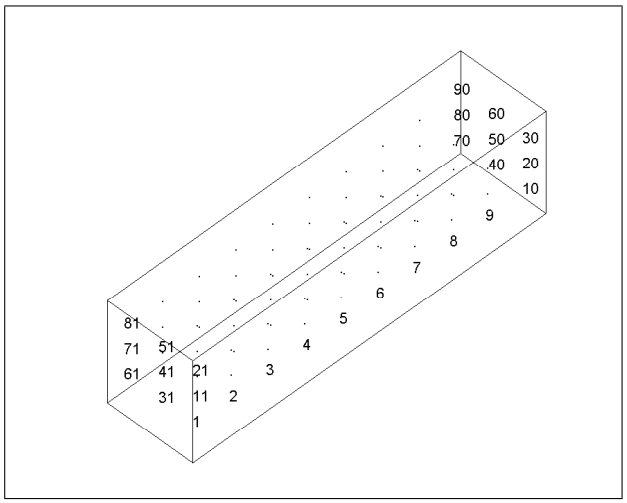


Figure 2. Grid of 90 points.

factors α_n^f and the mode shapes from the first simulation the volume-normalised mode shapes have been calculated.

In Figure 3 the results for the first longitudinal mode (138 Hz) are plotted. Note that the locations of the points are visualised in Figure 2. This plot shows that the sensitivity-based rescaling method results in correctly scaled mode shapes. Figure 4 shows the plot of the mode shape of the second longitudinal mode (277 Hz). This estimate too is quite good.

It is interesting to note that the errors become larger for lateral modes. The estimate of the first lateral mode (476 Hz) for example has already a significant error. This is shown in Figure 5. The reason of this increasing error is that for this mode the volume change cannot be assumed as a change in a point. For a longitudinal mode the pressure can be assumed constant at the inserted volumes (e.g. see Figure 3) while for the first lateral mode the pressure is not constant at all. If one mainly would be interested in the first lateral mode it would be smarter to place a volume with a big length, a small height and a big width. Also inserting smaller volumes could solve this problem. Note that the same behaviour has been observed in the past for the structural rescaling technique where the mass changes had to be point masses. It is important to note that the sensitivities are used in a first order approximation. If the modifications are too drastic, important estimation errors can occur. This was also noticed for the structural sensitivity based rescaling method [28].

4.2. Experimental validation

In a second step an experimental validation was done. We were mainly interested in the first longitudinal modes which resulted in the set-up of the acoustic system shown in Figure 6. The volume-normalised mode shapes resulting from the rescaling method were compared with those obtained by a FEA. To change the volume locally we used wooden blocks. Each wooden block had a volume of

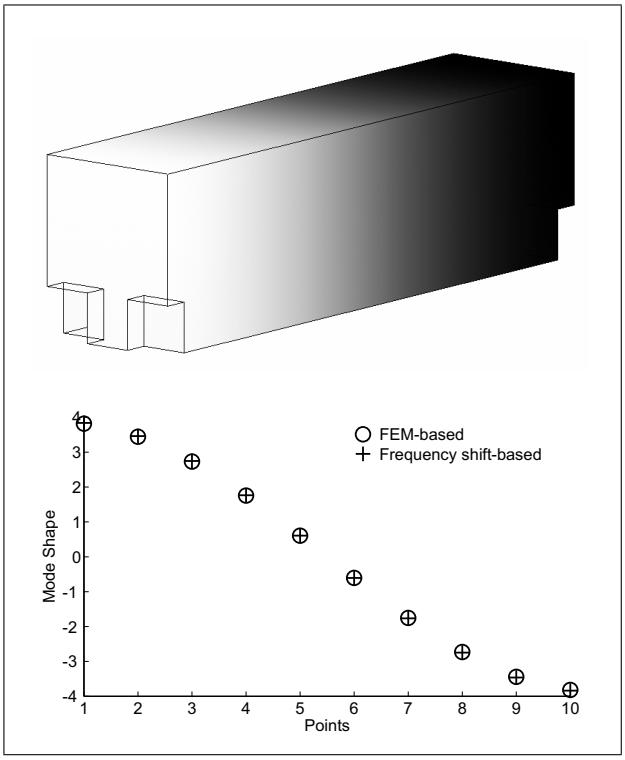


Figure 3. First longitudinal mode shape (138 Hz).

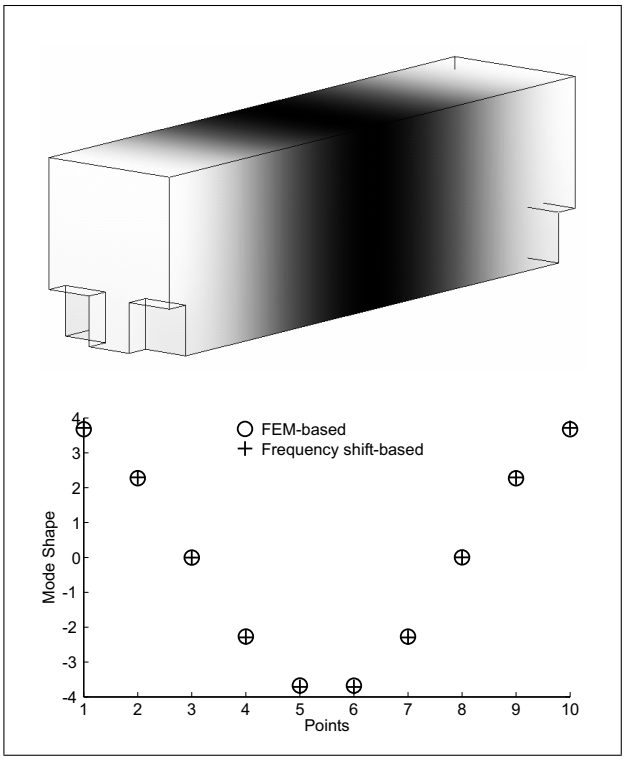


Figure 4. Second longitudinal mode shape (277 Hz).

7.5×10^{-4} m^3. Three different volume changes were considered: 1 block (7.5×10^{-4} m^3), 2 blocks (1.5×10^{-3} m^3) and 4 blocks (3×10^{-3} m^3). Those three situations are visualised in Figure 7.

Table II. Results of FEA. All frequencies in Hz.

Volume Change (m^3)	Reference Freq.	Changed Freq.	Freq. difference	Est. Scaled Mode Shape	Error (%)
7.5E-04	138.31	139.08	0.77	3.85	-0.2
1.5E-03	138.31	139.81	1.50	3.81	-1.4
3.0E-03	138.31	141.46	3.15	3.90	0.9
7.5E-04	276.62	278.07	1.45	3.73	-3.3
1.5E-03	276.62	279.54	2.92	3.75	-2.9
3.0E-03	276.62	282.69	6.07	3.82	-1.0
7.5E-04	414.94	416.75	1.81	3.41	-11.6
1.5E-03	414.94	418.57	3.63	3.42	-11.4
3.0E-03	414.94	422.06	7.12	3.38	-12.4

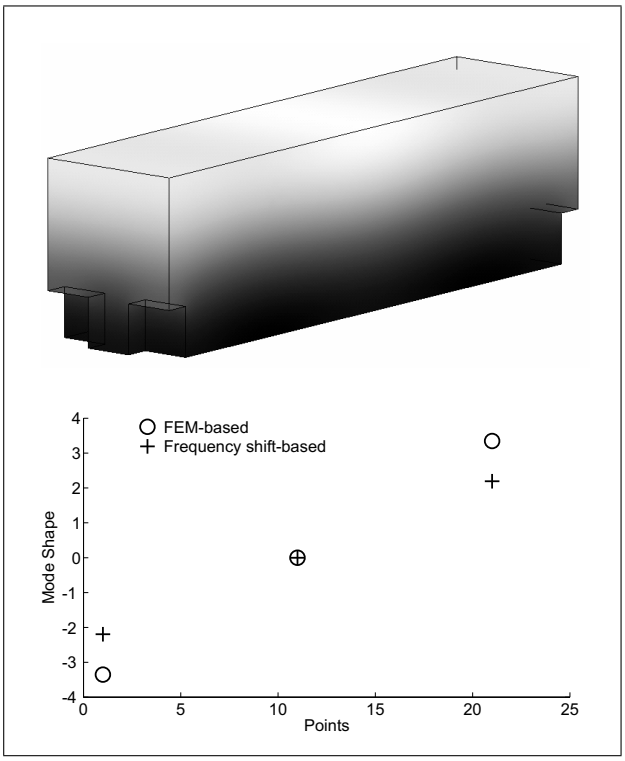


Figure 5. First lateral mode shape (476 Hz).



Figure 6. Experimental set-up.

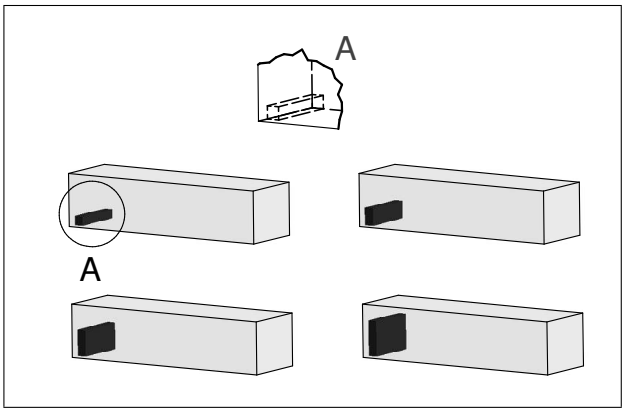


Figure 7. Volume changes.

The results of the FEA are summarised in Table II.

For the first two modes (138 Hz and 277 Hz) the errors are smaller than 4 %. For the third mode the errors are already above the 10 %. This is due to the fact that the volume difference is to big for this mode. So for the experimental validation only the first two modes were checked. The results of the experiment are summarised in Table III.

4.3. Applicability of the method

The experimental results show that the rescaling technique is applicable to a real acoustic system. Note that the volume changes have to be big enough because if the frequency shift is to small and if there is some noise on the measurements, the estimate of the frequency difference can be inaccurate. Knowing this and knowing that the volume changes also have to be small enough so one can use a linear approximation for the sensitivity analysis, one can conclude that there is an optimal size of the volume differences. As a rule of thumb one can say that if the frequency shifts one or two percent, the errors are smaller than 5 percent.

One can conclude that the rescaling technique gives satisfying results if the volume changes are wisely chosen. The same behaviour has been observed in the past for the structural rescaling technique. We only changed the vol-

Table III. Results of experiment. All frequencies in Hz.

Volume Change (m^3)	Reference Freq.	Changed Freq.	Freq. difference	Est. Scaled Mode Shape	Error (%)
7.5E-04	137.23	138.03	0.79	3.92	1.5
1.5E-03	137.22	138.72	1.50	3.82	1
3.0E-03	137.23	139.92	2.69	3.61	6
7.5E-04	276.15	277.80	1.65	3.99	3
1.5E-03	276.14	279.32	3.18	4.20	9
3.0E-03	276.13	279.41	3.28	2.81	27

ume in one location during the experiment. In real-life applications it is better to apply small volume changes in many points.

5. Conclusions

In this paper it has been shown that it is possible to estimate the acoustic FRF's using output-only data. The big advantage of this technique is that one does not need to use any volume acceleration sound sources. Based on a sensitivity analysis it is possible to calculate the scaled mode shapes. Some theoretical aspects have been discussed and the method has been validated by simple 3D-simulations and experiments. It has been shown that it is important to choose good volume changes and that it is important to start from good estimates. As a rule of thumb one can say that if the frequency shifts one or two percent, one gets the best results.

References

[1] H. Van der Auweraer, J. Leuridan: A new testing paradigm for today's product development process - Part 1. Sound and Vibration **39** (2005) 14–18.

[2] H. Van der Auweraer, J. Leuridan: A new testing paradigm for today's product development process - Part 2. Sound and Vibration **39** (2005) 18–22.

[3] K. Sumi, K. Yamamoto, L. Gielen, L. Meulewaeter: Powertrain noise and vibration simulation with hybrid models,. International Journal Of Vehicle Design **29** (2002) 149–159.

[4] S. H. Kim, J. M. Lee, M. H. Sung: Structural-acoustic modal coupling analysis and application to noise reduction in a vehicle passenger compartment. Journal of Sound and Vibration **225** (1999) 989–999.

[5] C. Schedlinski, F. Wagner, K. Bohnert, J. Frappier, A. Irrgang, R. Lehmann, A. Muller: Test-based computational model updating of a car body in white. Sound And Vibration **39** (2005) 19–23.

[6] B. Cauberghe: Applied frequency-domain system identification in the field of experimental and operational modal analysis. PhD thesis, Dept. of Mechanical Engineering, Vrije Universiteit Brussel, 2004.

[7] B. Cauberghe, P. Guillaume, P. Verboven, E. Parloo, S. Vanlanduit: A poly-reference implementation of the maximum likelihood complex frequency-domain estimator and some industrial applications. Proceedings of the 22th International Modal Analysis Conference, Dearborn, Michigan, 2004.

[8] P. Guillaume, B. Peeters, B. Cauberghe, P. Verboven: Identification of highly damped systems and its application to vibro-acoustic modeling. Proceedings of the 23th International Modal Analysis Conference, Dearborn, Michigan, 2004.

[9] M. Kronast, M. Hildebrandt: Vibro-acoustic modal analysis of automobile body cavity noise. Sound and Vibration **34** (2000) 20–23.

[10] M. Scionti, A. G. F. Vecchio, F. Petrone: Experimental road-noise analysis and acoustic cavity characterization of a series-hybrid electric passenger bus. Proceedings of the International Conference on Noise and Vibration Engineering (ISMA-2004), Leuven, Belgium, 2004.

[11] H. Van der Auweraer, K. Wyckaert, W. Hendricx: From sound quality to the engineering of solutions for NVH problems: Case studies. Acustica united with Acta Acustica **83** (1997) 796–804.

[12] X. Pelorson, J. Vian, J. D. Polack: On the variability of room acoustical parameters- reproductibility and statistical validity. Applied Acoustics **37** (1992) 175–198.

[13] J. D. Polack, L. S. Christensen, P. M. Juhl: An innovative design for omnidirectional sound sources. Acta Acustica united with Acustica **87** (2001) 505–512.

[14] K. Wyckaert, L. Meulewaeter: On the influence of finite acoustic source dimensions on acoustical frequency response functions inside an enclosed cavity. Proceedings of the International Conference on Noise and Vibration Engineering (ISMA-21) **Leuven, Belgium** (1996) 69–79.

[15] A. Cunha, E. Caetano: Experimental modal analysis of civil engineering structures. Sound and Vibration **40** (2006) 12–20.

[16] S. Zivanovic, A. Pavic, P. Reynolds: Modal testing and FE model tuning of a lively footbridge structure. Engineering Structures **28** (2006) 857–868.

[17] P. Guillaume, L. Hermans, H. Van der Auweraer: Maximum likelihood identification of modal parameters from operational data. Proceedings of the 17th International Modal Analysis Conference, 1999, 1887–1893.

[18] L. Hermans, H. Van der Auweraer: Modal testing and analysis of structures under operational conditions: Industrial applications. Mechanical Systems and Signal Processing **13** (1999) 193–216.

[19] E. Parloo, P. Guillaume, M. Van Overmeire, P. Verboven: Sensitivity-based operational mode shape normalization. Mechanical Systems and Signal Processing **16** (2002) 757–767.

[20] M. López Aenlle, R. Brincker, A. Fernández Canteli: Some methods to determine scaled mode shapes in natural input modal analysis. Proceedings of the 23th International Modal Analysis Conference, Orlando, Florida, 2005.

[21] R. Brincker, C. E. Ventura, P. Andersen: Why output-only modal testing is a desirable tool for a wide range of practical applications. Proceedings of the 21th International Modal Analysis Conference, Kissimmee, Florida, 2003.

[22] E. Parloo, B. Cauberghe, F. Benedettini, R. Alaggio, P. Guillaume: Sensitivity-based operational mode shape normalisation: Application to a bridge. Mechanical Systems and Signal Processing **19** (2005) 43–55.

[23] E. Parloo, P. Guillaume, J. Anthonis, W. Heylen, J. Swevers: Modelling of sprayer boom dynamics by means of maximum likelihood identification techniques: Part 2: Sensitivity-based mode shape normalisation. Biosystem Engineering **85** (2003) 291–298.

[24] F. Augusztinovicz, P. Sas: Acoustic modal analysis at low frequencies: similarities and differences in formulations. Proceedings of the International Conference on Noise and Vibration Engineering (ISMA-21), Leuven, Belgium, 1996, 1685–1699.

[25] P. Sas, F. Augusztinovicz: Acoustic modal analysis. NATO Advanced Study Institute Series E applied sciences, 1999. 487-506.

[26] W. Heylen, S. Lammens, P. Sas: Modal analysis theory and testing. Katholieke Universiteit Leuven, Departement Werktuigkunde, 1997.

[27] H. Kuttruff: Room acoustics. Applied Science Publishers LTD, Essex, 1979.

[28] E. Parloo, P. Verboven, P. Guillaume, M. Van Overmeire: Iterative calculation of non-linear changes by first order approximations. Proceedings of the 20th International Modal Analysis Conference, Los Angeles, California, 2002, 1084–1090.