Sound radiation of a baffled plate; theoretical and numerical approach

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Chapter 1

Introduction

Nowadays for manufactures it is of a high importance to produce low noise products. Policies exist to minimize the sound power radiated by machinery, airports and highways. Besides these policies set by the government manufactures need to comply with the demands of the costumor, who often prefer low noise designs. On the other hand there are also products that have to produce sound for functional reasons, for example a speaker system. For these reasons it is important to understand the behavior of mechanical structures that produce sound, whether it is functional sound or noise.

To answer the need of manufactures to be able to quantify the sound production of their product on beforehand without the use of a prototype, computer software is developed. With the ever increasing computational power of modern computers more powerful software is introduced to perform simulations to calculate the radiated sound power of complex structures. Moreover, manufactures are able to integrate the aspect of sound production into an early development stage of their product design.

In this report theory about the radiated sound of a baffled rectangular plate is compared with simulation results obtained from computer software. In this way conclusions can be drawn about the accuracy of the model that is used in the simulations. First the theory about the structural behavior of a simply supported plate is presented. Secondly some theory about radiation efficiency and radiation modes are presented. In Chapter 3 these theoretical results are compared with the results from the numerical model. Chapter 4 will give more insight in the process of modeling, exporting and importing results from one software package into another one. And finally in Chapter 5 some discussions and recommendations will be done.

Chapter 2

Theory

2.1 Baffled rectangular plate

To estimate the radiated sound power by a vibrating structure computer software can be used. In order to be sure that a correct model is used, the results have to be compared with theory. Theory exist about simple structures like a rectangular plate. Therefore in this report a baffled rectangular plate (without damping) is investigated, see Figure 2.1(a). The plate is simply supported at the edges; there are no translational degrees of freedom at the edges, see Figure 2.1(b). Throughout the report the properties, mentioned in Table 2.1, of an aluminium plate are used.

The natural eigenfrequencies and eigenmodes of this simply supported rectangular plate can be calculated. From theory [Fah87] the eigenfrequencies equal

$$\omega_r = \sqrt{\tfrac{D}{m}}\left[\left(\tfrac{r_1\pi}{a}\right)^2 + \left(\tfrac{r_2\pi}{b}\right)^2\right], \tag{2.1}$$

where r_1 and r_2 are the modal indices of the r-th mode and m and D represent the mass per unit area ($m = \rho h$) and the bending stiffness per unit length ($D = Eh^3/12(1-\nu^2)$) respectively. Physically the numbers r_1 and r_2 represent the number of half sine-waves in x and y direction of the plate in the corresponding mode shape. In Table 2.2 the first 10 eigenfrequencies corresponding with the (r_1, r_2)-th mode are denoted. The corresponding structural mode shapes can be calculated with

Property	Symbol	Value	Property	Symbol	Value
Length [m]	a	0.414	Density [kg/m^3]	ρ	2700
Width [m]	b	0.314	Youngs Modulus [GPa]	E	71
Thickness [mm]	h	2.0	Poisson ratio [-]	ν	0.33

Table 2.1: Properties of the aluminium plate.

(a) A rectangular plate in an infinite baffle. (b) Structural mode shapes of a simply supported plate.

Figure 2.1: The rectangular baffled plate.

Mode	Eigenfrequency [Hz]	Mode	Eigenfrequency [Hz]
(1,1)	79	(3,2)	459
(2,1)	165	(1,3)	478
(1,2)	229	(4,1)	510
(3,1)	309	(2,3)	565
(2,2)	315	(4,2)	660

Table 2.2: First 10 analytical eigenfrequencies of the rectangular simply supported plate.

$$\phi_r(x, y) = 2\sin\left(\frac{r_1 \pi x}{a}\right)\sin\left(\frac{r_2 \pi y}{b}\right). \tag{2.2}$$

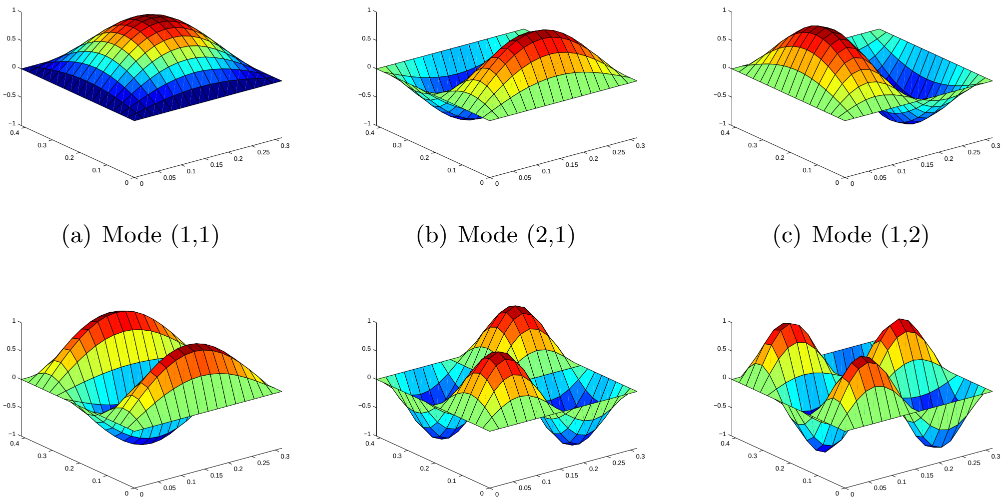
The first six eigenmodes are plotted in Figure 2.2.

2.2 Boundary Element Method

With the results of a vibrational analysis it is possible to compute acoustic pressures in the surrounded fluid by a structure using the Helmholtz differential equation. Only the surface of a vibrating structure in contact with the fluid, also called wetted surface, is able to transfer energy to the fluid. Therefore the Helmholtz differential equation can be reduced to an integral equation that covers only the boundary surface S. The complete derivation can be found in [Vis04]:

$$\alpha(\vec{x})p(\vec{x}) = \oint_S \left(\frac{\partial G(r)}{\partial n_y} p(\vec{y}) + i\omega\rho_0 G(r) v_{n_y}(\vec{y})\right) dS + p^{in}(\vec{x}). \tag{2.3}$$

In (2.3) the acoustic pressure and normal velocity are related to the radiated pressure field in the fluid domain. The term $\alpha(\vec{x})$ is a geometry related coefficient, $\vec{y}$ is a point on the boundary surface S and $\vec{x}$ is a field point in the fluid domain. The unit normal to the surface at source point $\vec{y}$, denoted as $\vec{n}_y$, is pointed into the fluid domain. The distance r is the length of vector $\vec{r}$ that is directed from the source point $\vec{y}$ to the field point $\vec{x} : r = ||\vec{x} - \vec{y}||$. The term p^{in} represents the incident acoustic wave in the case



(a) Mode (1,1) (b) Mode (2,1) (c) Mode (1,2)

(d) Mode (3,1) (e) Mode (2,2) (f) Mode (3,2)

Figure 2.2: First six structural eigenmodes of the simply supported rectangular plate.

of a scattering analysis and $G(r)$ is the Green's function, which represents the effect observed at point $\vec{x}$ created by a unit source located at point $\vec{y}$.

In order to solve the Helmholtz integral equation (2.3) for a field point $\vec{x}$ the normal velocity and pressure at the surface S should be known. If only the normal velocities are known, first the pressures at the surface have to be calculated by replacing $\vec{x} = \vec{y}$ in (2.3). Secondly the pressures at the field points can be calculated.

For the flat aluminium plate the partial derivative of the Green's function to the unit normal will be zero. Further, the radiation into the free field is investigated, so incident acoustic waves will not be present. Now the Helmholtz integral equation can be reduced to the first Helmholtz integral equation:

$$\alpha(\vec{x})p(\vec{x}) = \oint_S \Big(i\omega\rho_0 G(r) v_{n_y}(\vec{y}) \Big) dS. \tag{2.4}$$

2.2.1 Numerical implementation

The analysis of vibrations and the resulting radiated sound can be done with the use of sophisticated computer software, e.g. for calculating the dynamics of a practical structure a Finite Element (FE) Model with appropriate boundary conditions can be used. This demands a discretization of the structure into a number of finite elements. If the structural dynamics are of interest all elements, interior and boundary, are to be accounted for, as they are a measure of the total mass and stiffness of the system.

For the total radiated sound power however, only the elements on the boundary have a contribution. Only these elements are in contact with the fluid to which energy is transferred. For the purpose of determining the radiated sound by a structure a Boundary Element (BE) model will suffice. The advantage of a BE model is that less equations are to be solved, as in general there are less nodes in a BE model compared with a FE model. In this report ABAQUS is used for the FE analysis and LMS Virtual.Lab together with SYSNOISE for the BE analysis. In Chapter 4 the implementation of a model of the baffled plate is discussed in more detail.

From the structural vibration data at the nodes the acoustic pressures at the surface can be calculated with the first Helmholtz integral equation (2.4). This requires a discretization of this continuous equation, see [Vis04]. This results into the following matrix calculations:

$$\mathbf{A}\mathbf{p} = \mathbf{B}\mathbf{v}, \tag{2.5}$$

where the matrices $\mathbf{A}$ and $\mathbf{B}$ are the influence matrices. These matrices are dependent on the geometry of the structure and comply with the Helmholtz integral equation (2.4). Since the partial derivative of the Green's function to the unit normal is zero for the flat plate, the matrix $\mathbf{A}$ will be the identity matrix $\mathbf{A} = \mathbf{I}$. The influence matrices are calculated inside a BE package. The vectors $\mathbf{p}$ and $\mathbf{v}$ contain the sound pressures and normal velocities at each node respectively. So with the velocities from the structural vibration data the sound pressure at each node can be calculated.

2.3 Radiation Efficiency

A useful measure of the effectiveness of sound radiation by a vibrating surface is the total radiated sound power normalized with respect to the specific acoustic impedance of the fluid medium, the structure area and the velocity of the surface vibration, which is defined as the radiation efficiency. A commonly used measure of the surface vibration is the space-average value of the time-averaged squared vibration velocity defined by

$$\overline{v_n^2} = \frac{1}{S}\int_S \left(\frac{1}{T}\int_0^T v_{n_y}^2(\vec{y})dt\right)dS, \tag{2.6}$$

where T is a suitable period of time over which to estimate the mean square velocity $v_{n_y}^2$ at a point $\vec{y}$ and S extends over the total vibrating surface.

The radiation efficiency is defined by reference to the acoustical power radiated by a uniformly vibrating baffled piston at a frequency for which the piston circumference greatly exceeds the acoustic wavelength k: $ka \gg 1$. For the radiated power of a baffled piston the following relation holds:

$$\overline{P} = \tfrac{1}{2}\rho_0 c S \overline{v_n^2}. \tag{2.7}$$

The definition of the radiation efficiency is thus:

$$\sigma = \overline{P}/\rho_0 c S \overline{v_n^2}. \tag{2.8}$$

The radiation efficiency is below unity for frequencies lower than the critical frequency $\omega_c = c^2(m/D)^{1/2}$. In this equation c is the speed of sound in air (340 m/s). At the critical frequency the structural (bending) wavelength k_b equals the acoustic wavelength k. For the baffled plate the critical frequency in Hz equals $f_c = \frac{\omega_c}{2\pi} = 5866$ Hz. At the critical frequency the radiation efficiency will exceed unity and at higher frequencies it will be close to unity [Fah87].

2.4 Radiated Power

In the case of a structure modeled within a FE software package the structural vibration of this structure with R elements can be calculated with FE method. In this way it is possible to obtain a column vector of complex velocities at each node caused by a harmonic point force. Theory [Fah87] however gives the relation between the normal velocities and sound pressures at the center of an element. These quantities at the center of an element can be calculated from the quantities at the surrounding nodes using bilinear interpolation functions. When relatively small elements (small compared to the acoustic wavelength, $\sqrt{A_e} \ll \lambda$) are used the difference between the quantities at the nodes or center of an element can be neglected. The velocities are grouped in a column vector, like:

$$\mathbf{v}_e = \begin{bmatrix} v_{e1} & v_{e2} & \dots & v_{eR} \end{bmatrix}^T. \tag{2.9}$$

With the calculated velocities the sound pressure and radiated sound power can be calculated within a BEM package. The obtained sound pressure at each element is also grouped in a column vector:

$$\mathbf{p}_e = \begin{bmatrix} p_{e1} & p_{e2} & \dots & p_{eR} \end{bmatrix}^T. \tag{2.10}$$

From a BE model the relation between the elemental velocities and sound pressures can be found. As a result of (2.5) the sound pressures can be denoted as

$$\mathbf{p}_e = \mathbf{A}^{-1}\mathbf{B}\mathbf{v}_e \tag{2.11}$$

Analytical equations [Fah87] for the radiated sound power give the relationship between the velocities and sound pressures as in

$$P(\omega) = \sum_{r=1}^{R} \tfrac{1}{2} A_e Re(v_{er}^{*} p_{er}) = \tfrac{S}{2R} Re(\mathbf{v}_e^H \mathbf{p}_e), \tag{2.12}$$

where A_e and S are respectively the areas of each element and of the whole structure. Substituting (2.11) into (2.12) result in

$$P(\omega) = \tfrac{S}{2R} Re(\mathbf{v}_e^H \mathbf{A}^{-1} \mathbf{B} \mathbf{v}_e) = \mathbf{v}_e^H \mathbf{R} \mathbf{v}_e, \tag{2.13}$$

with $\mathbf{R}$ the radiation resistance matrix for the elementary radiators. From (2.13) the radiation resistance matrix can be calculated with

$$\mathbf{R} = \tfrac{S}{2R} Re(\mathbf{A}^{-1}\mathbf{B}). \tag{2.14}$$

For the baffled finite plate it holds that $\mathbf{A}^{-1} = \mathbf{A} = \mathbf{I}$. Combining this result with analytical expressions for the radiation resistance matrix [Fah87], this results in:

$$\mathbf{R} = \tfrac{S}{2R} Re(\mathbf{B}) = \tfrac{\omega^2 \rho_0 A_e^2}{4\pi c} \begin{pmatrix} 1 & \frac{sin(kR_{12})}{kR_{12}} & \cdots & \frac{sin(kR_{1R})}{kR_{1R}} \\ \frac{sin(kR_{21})}{kR_{21}} & 1 & \cdots & \vdots \\ \vdots & \vdots & \ddots & \vdots \\ \frac{sin(kR_{R1})}{kR_{R1}} & \cdots & \cdots & 1 \end{pmatrix}. \tag{2.15}$$

In (2.15) R_{ij} is the distance between the centers of the i-th and j-th element. Note that because of reciprocity the radiation resistance matrix $\mathbf{R}$ is symmetric.

2.5 Radiation Modes

The eigenvalues of the radiation resistance matrix $\mathbf{R}$ are proportional to the radiation efficiencies and the eigencolumns are known as the radiation modes. These radiation modes are frequency dependent and indicate the contribution of each element to the total radiated sound power. For the aluminium plate with the properties listed in Table 2.1 the shapes of the first six radiation modes and their corresponding eigenvalue λ_r, which are proportional to the radiation efficiencies, have been calculated, see Figures 2.3, 2.4 and 2.5. The eigenvalues λ_r are linear increasing with the frequency at low frequencies. For frequencies under 1 kHz the first radiation mode is by far the most efficient one. Above 1 kHz the different modes are equally efficient. The radiation modes are frequency dependent and are plotted at two frequencies, $f_1 = 100$ Hz and $f_2 = 800$ Hz. Figures 2.4 and 2.5 clearly shows that the first, and most efficient mode,

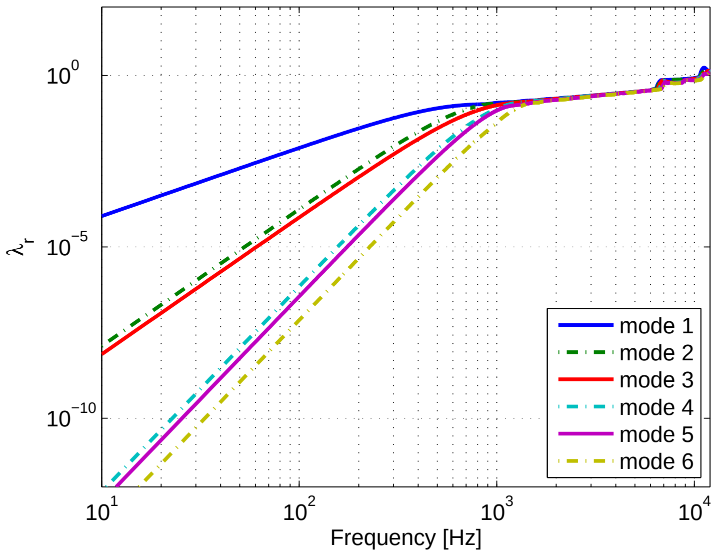
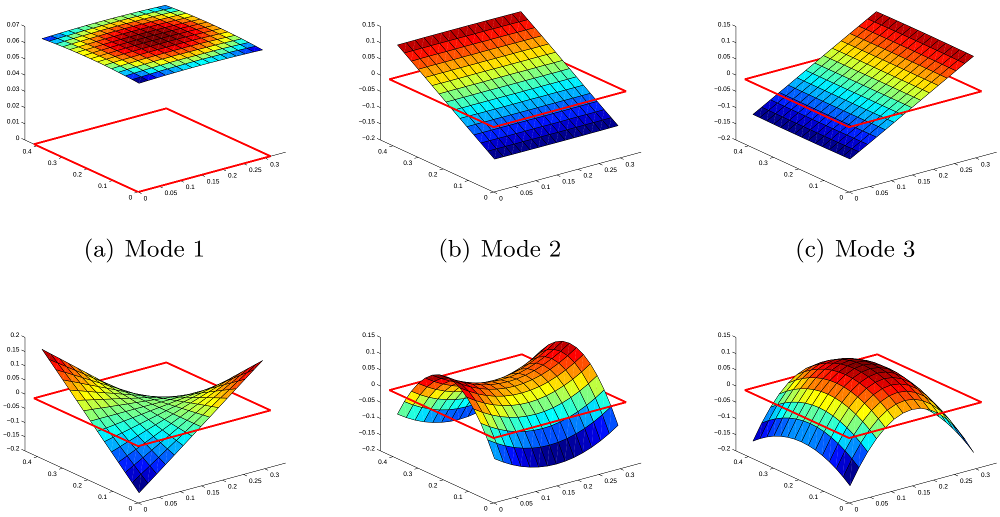


Figure 2.3: First six eigenvalues of the radiation resistance matrix for the baffled plate.

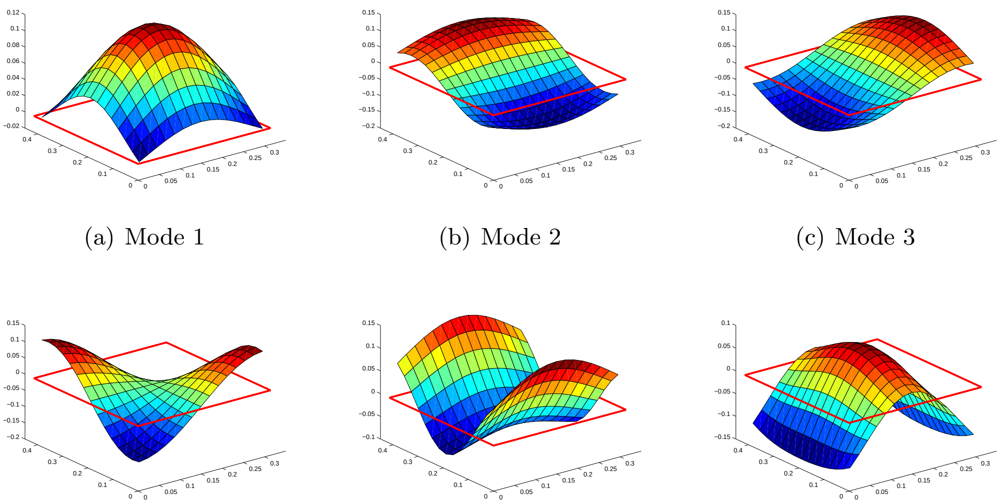
corresponds to a piston-like mode. At low frequencies the velocity distribution is nearly uniform over the surface of the panel; as frequency increases the piston-like shape is gradually distorted towards a domeshape. The following two modes are rocking-type modes oriented along the two axes of the panel.



(a) Mode 1 (b) Mode 2 (c) Mode 3

(d) Mode 4 (e) Mode 5 (f) Mode 6

Figure 2.4: Radiation modes of the baffled plate at 100 Hz.



(a) Mode 1 (b) Mode 2 (c) Mode 3

(d) Mode 4 (e) Mode 5 (f) Mode 6 @

Figure 2.5: Radiation modes of the baffled plate at 800 Hz.

Chapter 3

Numerical results

In literature [Fah87] the eigenfrequencies and radiation efficiencies for a finite baffled plate can be found and are mentioned in Chapter 2. By discretizing the plate into finite elements numerical methods in software packages can be used to calculate the eigenfrequencies and radiation efficiencies and they are compared with the theoretical values.

3.1 Eigenfrequencies and mode shapes

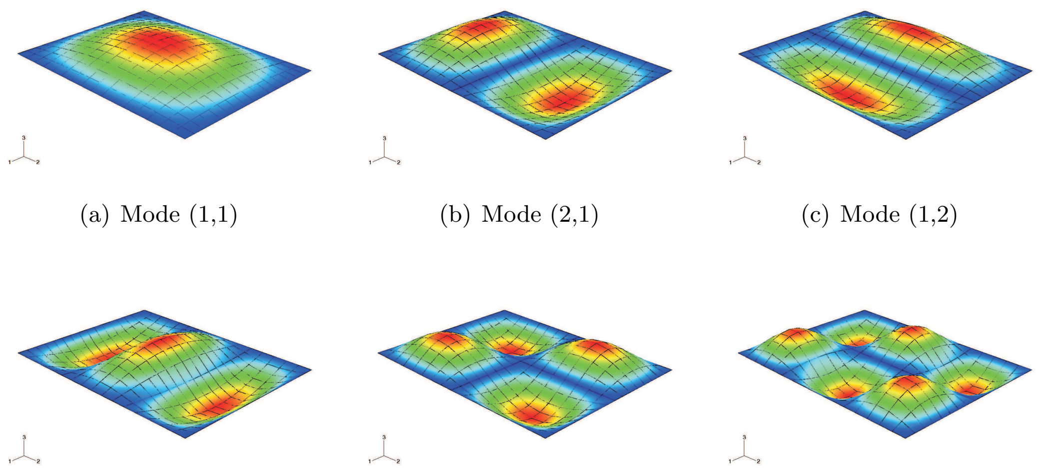
In Chapter 2 the theory about the eigenfrequencies of a simply supported finite plate is presented. In ABAQUS this simply supported plate is modeled with 16 by 16 shell elements. In Appendix A the input file that is used in ABAQUS can be found. In Figure 3.1 the mode shapes calculated with ABAQUS can be found and in Table 3.1 the theoretical and numerical (ABAQUS) eigenfrequencies are denoted. As can be seen the relative error between the eigenfrequencies is bigger for higher frequencies. The numerical eigenfrequencies are higher than the theoretical eigenfrequencies, because the discretized model has a certain number of degrees of freedom (dof's). When more elements would be used this number of dof's will increase, which results in a less stiff plate and consequently lower eigenfrequencies. Also the number of elements per structural wavelength should be examined; at least six elements per wavelength should be used.

3.2 Radiation efficiencies and modes

As described in Chapter 2.5 the radiation efficiency is proportional to the eigenvalues of the radiation resistance matrix $\mathbf{R} = \frac{S}{2R} Re(\mathbf{B})$. To compare the eigenvalues of the radiation resistance matrix obtained from an BE analysis the (complex) influence matrix $\mathbf{B}$ is exported from LMS Virtual.Lab. This is done for 10 different frequencies at

Mode	Eigenfreq. [Hz] theory	Eigenfreq. [Hz] numerical	Rel. error [%]
(1,1)	79	79	0.0
(2,1)	165	167	1.2
(1,2)	229	232	1.3
(3,1)	309	320	3.6
(2,2)	315	320	1.6
(3,2)	459	472	2.8
(1,3)	478	498	4.2
(4,1)	510	546	7.1
(2,3)	565	585	3.5
(4,2)	660	697	5.6

Table 3.1: First 10 analytical eigenfrequencies of the rectangular simply supported plate.



(a) Mode (1,1) (b) Mode (2,1) (c) Mode (1,2)

(d) Mode (3,1) (e) Mode (2,2) (f) Mode (3,2)

Figure 3.1: First six structural eigenmodes of the simply supported rectangular plate calculated with ABAQUS.

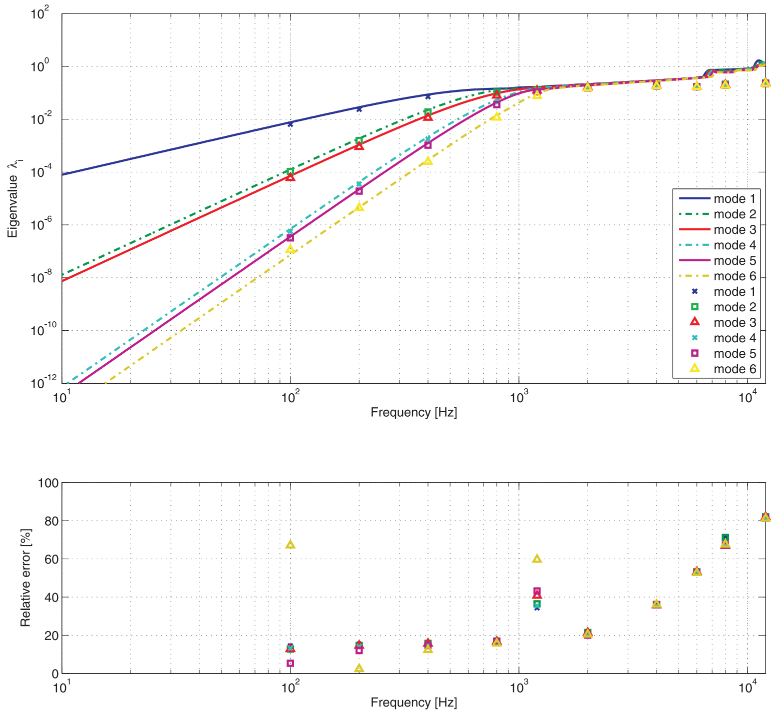


Figure 3.2: First six eigenvalues of the radiation resistance matrix for the baffled plate. The lines represent the theoretical values and the markers the numerical ones. The graph at the bottom represents the relative error between the theoretical en numerical eigenvalues.

$100, 200, 400, 800, 1200, 2000, 4000, 6000, 8000$ and 12000 Hz. In Appendix B it can be found how this is done. In Figure 3.2 the eigenvalues and the relative error with the theoretical eigenvalues are plotted. As can be seen the error between the theoretical and numerical values varies a lot. Up to 800 Hz the error is between 5 and 17 percent (the error for the 6th mode at 100 Hz neglected) and increases for higher frequencies, with an unexpected high error at 1200 Hz. This result is due to the fact that the theory is based on the positions of the element centers, while the numerical results are based on the positions of the nodes. At higher frequencies this will cause a bigger mismatch. The total radiation efficiency can also be exported from LMS Virtual.Lab directly and is shown in Figure 3.3.

The eigenvectors of the radiation resistance matrix at 100 Hz and 800 Hz, also known as the radiation modes, are plotted in Figure 3.4 and Figure 3.5. These radiation modes can be compared with the radiation modes presented in Chapter 2.5. The shapes of the modes agree with the theoretical ones, except the sixth mode at 100 Hz. As can be seen

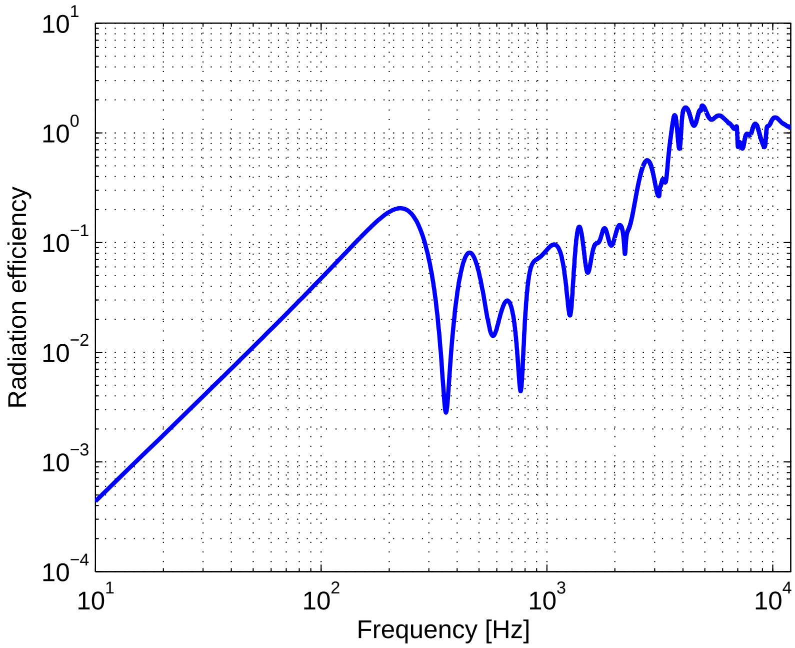


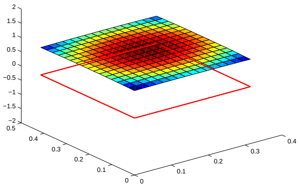
Figure 3.3: Radiation efficiency exported from LMS Virtual.Lab.

in Figure 3.2 the eigenvalue for this mode does not agree with the theoretical value as well. Within this report there is not an explanation for this result.

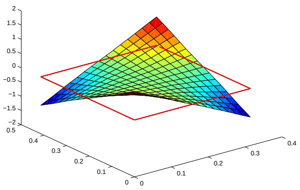
3.3 Radiated Sound Power

The total sound power radiated by the baffled plate for a given velocity distribution can be calculated using (2.13). A harmonic point force with an amplitude of 1 N at the center and perpendicular to the plate is used to calculate the velocity distribution at the nodes of the plate within ABAQUS. A frequency range of 1 Hz to 12001 Hz with a linear step-size of 2 Hz is used. This velocity distribution is interpolated bi-linearly to the element centers, because the theoretical radiation resistance matrix $\mathbf{R}$ is defined using the element centers and is given as in (2.15). From simulations with the BE software package LMS Virtual.Lab the radiated sound power can be exported. These simulations use Gaussian quadrature rules for the numerical integration of the Helmholtz integral equation. In LMS Virtual.Lab the quadrature, i.e. the number of Gauss points used in one direction for the numerical evaluation of integrals, are by default set to the values of 2, 2, 1 for the different regions (near, medium and far respectively). In this report only the near field region is examined. In Figure 3.6 the results for the total radiated sound power using both the theory and the numerical results are shown.

As can be seen in Figure 3.6 the radiated power is characterised by a sequence of

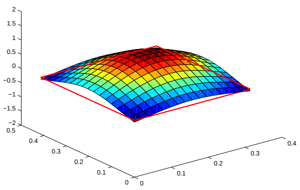


(a) Mode 1 (b) Mode 2 (c) Mode 3

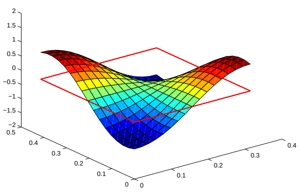


(d) Mode 4 (e) Mode 5 (f) Mode 6

Figure 3.4: Radiation modes of the baffled plate at 100 Hz.



(a) Mode 1 (b) Mode 2 (c) Mode 3



(d) Mode 4 (e) Mode 5 (f) Mode 6 @

Figure 3.5: Radiation modes of the baffled plate at 800 Hz.

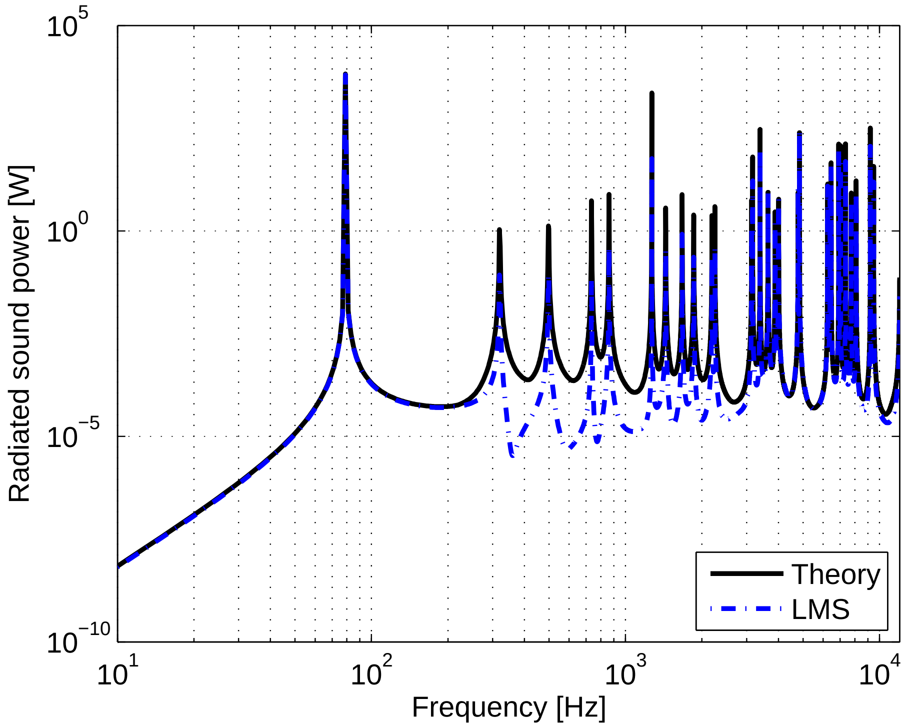


Figure 3.6: Radiated sound power by the baffled plate, both the theoretical as the numerical determined powers, using 1st and 2nd order Gauss quadrature rules, are plotted.

peaks corresponding to the eigenfrequencies of the plate. Note that the peaks do not correspond with all structural eigenfrequencies. The first four peaks correspond with the (1,1), (3,1), (1,3) and (3,3) mode respectively. These odd-odd modes are symmetric modes and radiate sound efficiently. The even (asymmetric) modes do not radiate sound efficiently. For example, the (2,1) mode shape is an asymmetric mode, that causes a high pressure at one half of the plate and a low pressure at the other half at the same time. In this way the variation in pressure at the surface caused by the two parts of the plate cancel each contribution to the total radiated power.

Between 200 and 3000 Hz a large mismatch between the theoretical and numerical sound power can be seen. This is due to the different methods of interpolation that are used. In theory the element centers are used. Therefore the velocities at the nodes are bi-linearly interpolated to the element center. This means that the four surrounded nodes each have the same contribution to the resulting value at the center of the element. In Gaussian quadrature, both the nodes and the weights are optimally chosen to maximize the degree of the resulting quadrature rule [Hea02]. In LMS Virtual.Lab two Gauss point are used, resulting in quadrature rules of degree three. Gaussian quadrature rules therefore have a higher accuracy than the bi-linearly interpolation used for the theoretical results.

Chapter 4

Implementation

In this Chapter the implementation of a model of the baffled plate in the various software programs is described in more detail. Moreover, the various steps of modeling, exporting and importing the results is described in a chronological way.

4.1 ABAQUS

First to obtain the vibrational data that causes the radiated sound, a simply supported rectangular aluminium plate is modeled within ABAQUS. An input file with extension `.inp` is written, see Appendix A. In this input file first a grid of 17 by 17 nodes is defined. Next step is defining the 16 by 16 quadrilateral shell elements, with a virtual 3th dimension with a size of 2 mm (the thickness of the plate). The edges of the plate are simply supported, so a boundary condition is defined to set all translational (so not rotational) degrees of freedom at the corresponding nodes to zero. Finally two steps are applied, the first to calculate the first 10 eigenfrequencies and the second to calculate the velocities at the nodes generated by a harmonic point force at the center of the plate for different frequencies.

This input file can be run in the ABAQUS command window by typing

```
abaqus job = filename .
```

Now ABAQUS will generate an database file with extension `.odb` containing the results. Also an `.dat` file will be created, which may be handy for debugging the input file. The `.odb` file can be opened in the ABAQUS Viewer to see the results. Moreover, the calculated velocity distribution can be exported to a text file (`.txt`), which can be imported in Matlab.

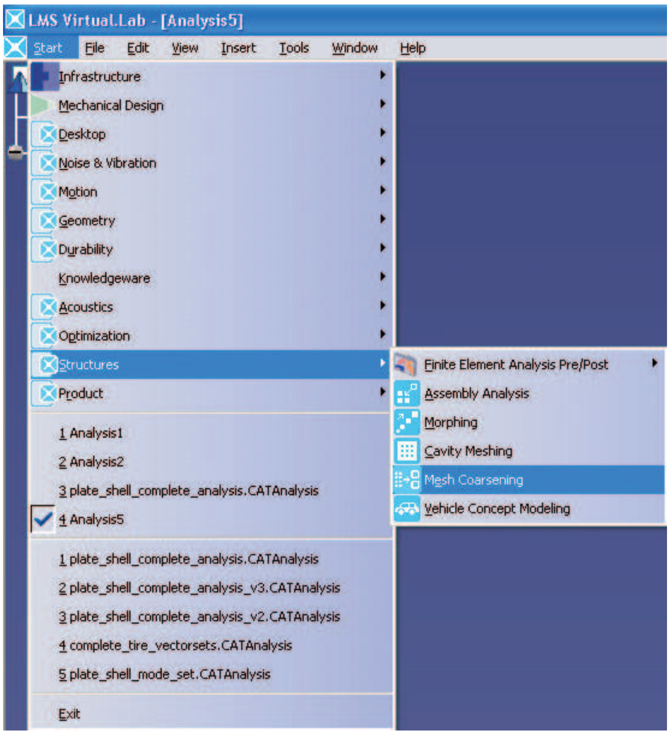


Figure 4.1: Select the Mesh Coarsening workbench.

4.2 LMS Virtual.Lab

A very common application of LMS Virtual.Lab is to compute the sound field radiated by a vibrating structure. To characterize the structural vibrations in LMS Virtual.Lab a structural finite element analysis, like the `.odb` file from ABAQUS, can be imported. In this way the mesh and analysis steps from ABAQUS can be used in LMS.

4.2.1 Mesh Coarsening

First the imported mesh is converted into a mesh that is suitable for Boundary Element Method (BEM). In other words, the structural mesh of any structure is converted into a mesh of 2D elements that represents only the part of the structure that is in contact with the fluid to which sound is radiated. In LMS Virtual.Lab this is done with the Mesh Coarsening workbench in the Structures toolbox, see Figure 4.1. When importing the `.odb` file a menu appears where only the mesh has to be selected, see Figure 4.2. Of course, it is important to use consistent units (meter, kg, s).

Next a Skin Mesh is applied on the structural mesh, that covers only the envelop of the structural mesh. Note that in the case of the thin rectangular plate (modeled with 2D shell elements) the Skin Mesh is the same as the structural mesh. However, later on it is

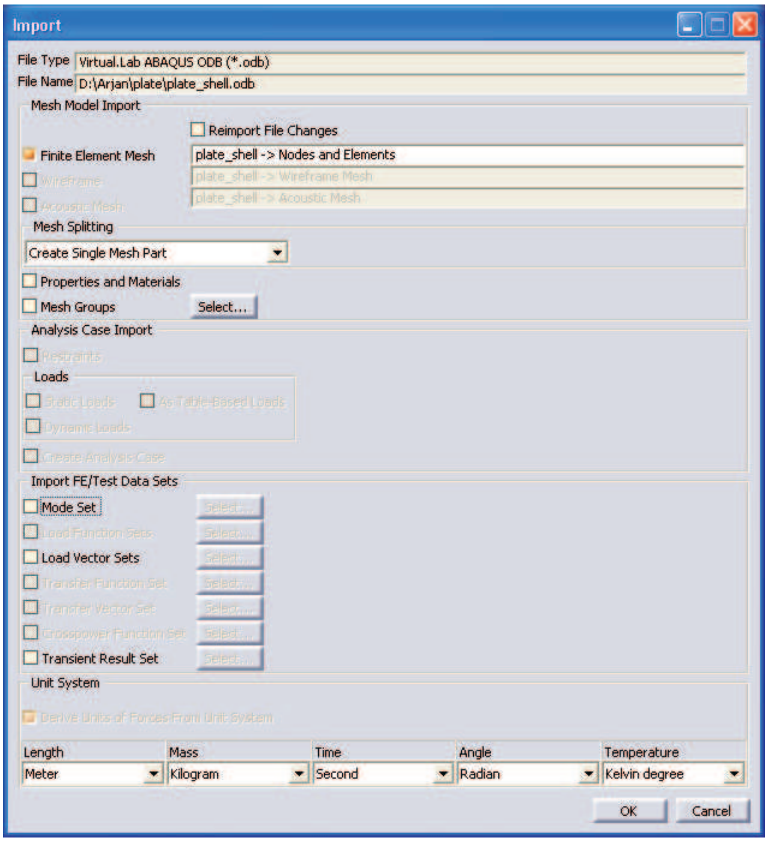


Figure 4.2: Import the structural mesh.

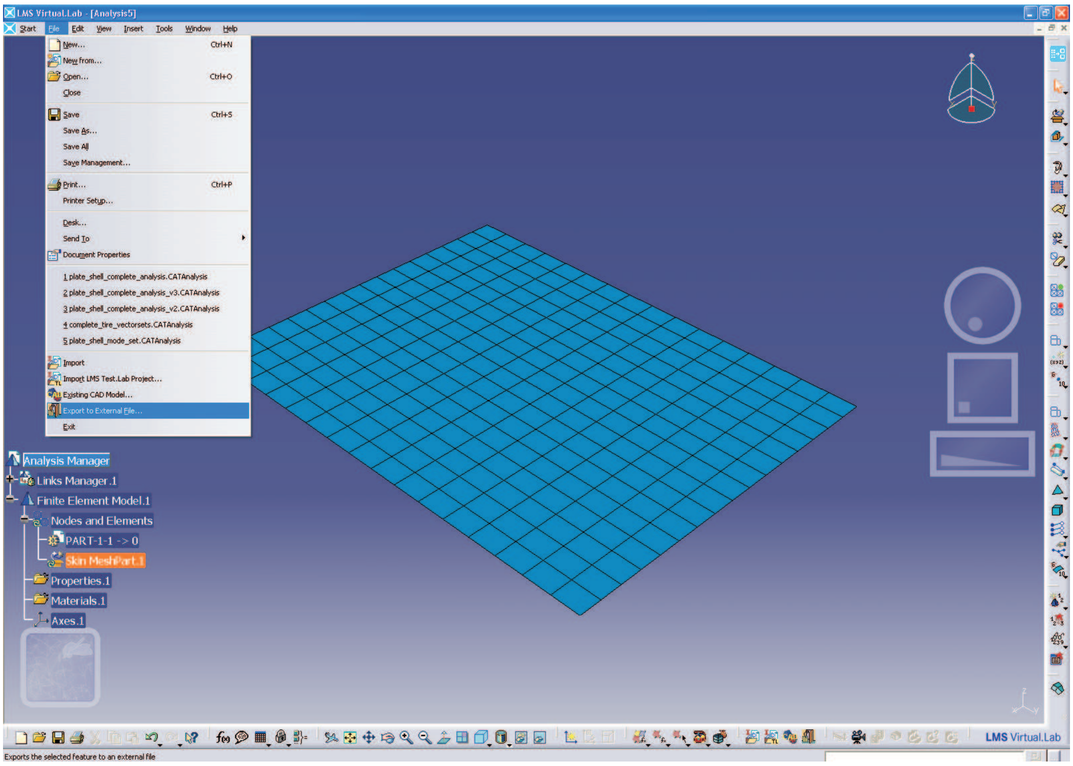


Figure 4.3: Export the Skin Mesh.

needed to have both a Skin (acoustic) Mesh to apply the acoustic boundary conditions to and a structural mesh with the vibrational data. For more complex structures it can be desired to fill holes, convert element types or delete unwanted elements to create an optimal acoustic mesh. This is also possible in the Structures toolbox.

Finally the Skin Mesh part has to be exported to an external file, e.g. a Nastran bulk file (`.bdf`), see Figure 4.3.

4.2.2 Load Vector Set

The structural vibrational data from ABAQUS is imported as a Load Vector Set into LMS Virtual.Lab. A new System Analysis with the Noise and Vibrations toolbox is started and the `.odb` file is imported. This time the mesh and Step 2 from the Load Vector Sets are toggled on. This will create a Load Vector Set in the specification tree. The definition type of this Load Vector Set is set to velocities, as it contains the velocities calculated with ABAQUS, by double clicking on the Load Vector Set, see Figure 4.4. By generating an image of the Set, the velocities can be visualized in LMS Virtual.Lab. Finally this analysis is saved as a CATAnalysis document.

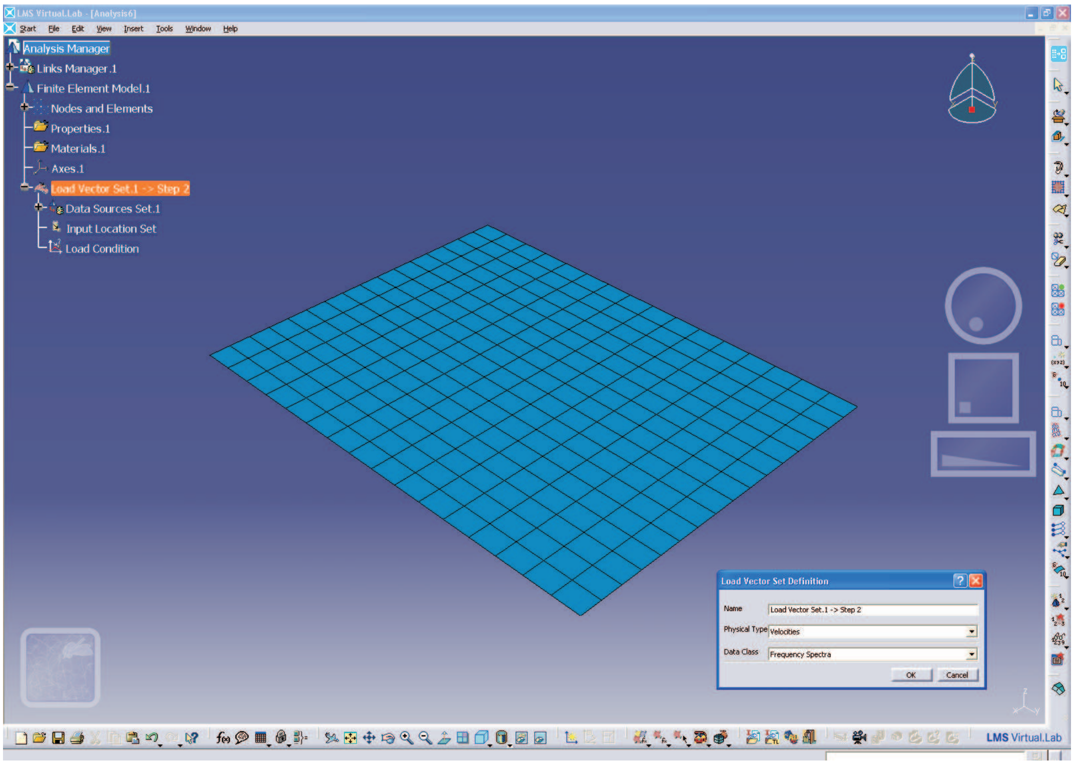


Figure 4.4: Set the definition type of the Load Vector Set to velocities.

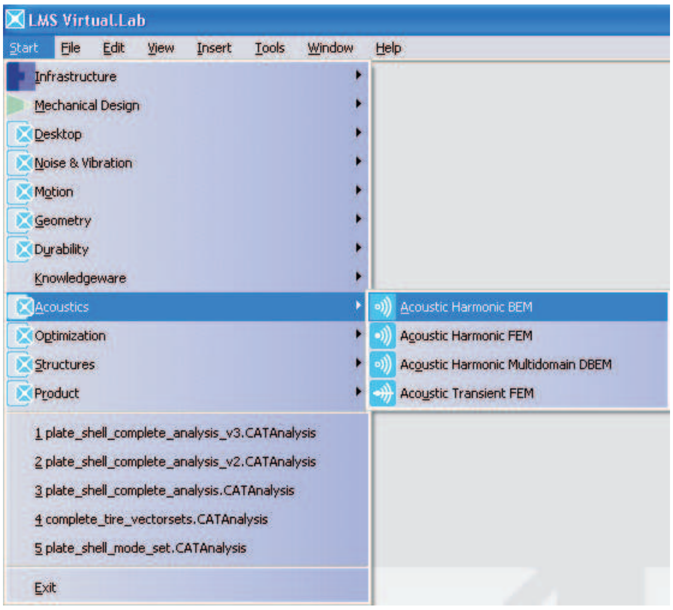


Figure 4.5: Select the Acoustic Harmonic BEM workbench.

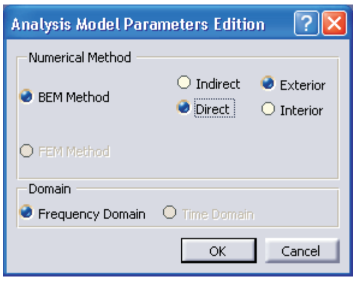


Figure 4.6: Select the BEM direct model type definition.

4.2.3 Acoustic Response Analysis Case

With the Skin Mesh part and the Load Vector Set created, an Acoustic Response Analysis case can be assigned. First, however, in LMS Virtual.Lab the Acoustic Harmonic BEM workbench is started. Then, before importing the acoustic mesh, the model type definition is set to BEM direct, as only the exterior problem has to be solved.

Now the Skin Mesh (Nastran bulk file `.bdf`) can be imported (only import the mesh). Next step is to set the Skin Mesh as acoustical mesh part by right clicking on the mesh in the specification tree. After this a symmetry plane can be inserted. The Z-plane is used as a symmetry plane to represent the infinite baffle, see Figure 4.7. Physically this represents a rigid surface where the normal velocities equal zero. So on both sides of the symmetry plane sound will be radiated as a monopole.

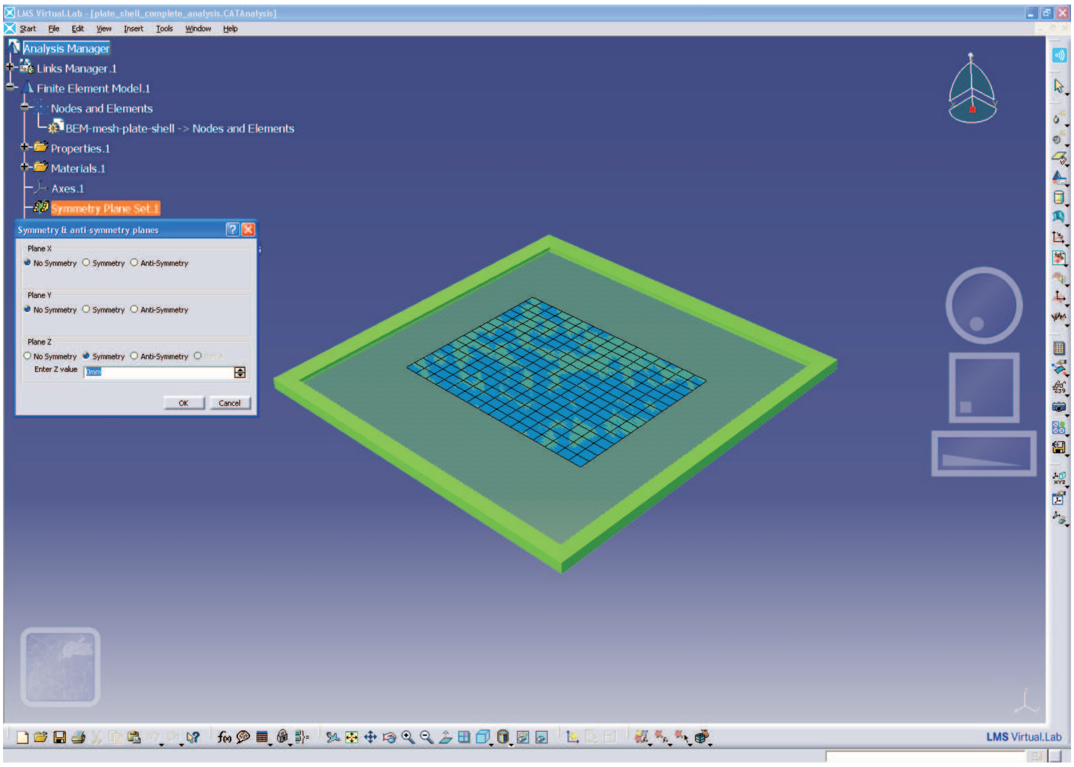


Figure 4.7: The symmetry plane that represents the infinite baffle.

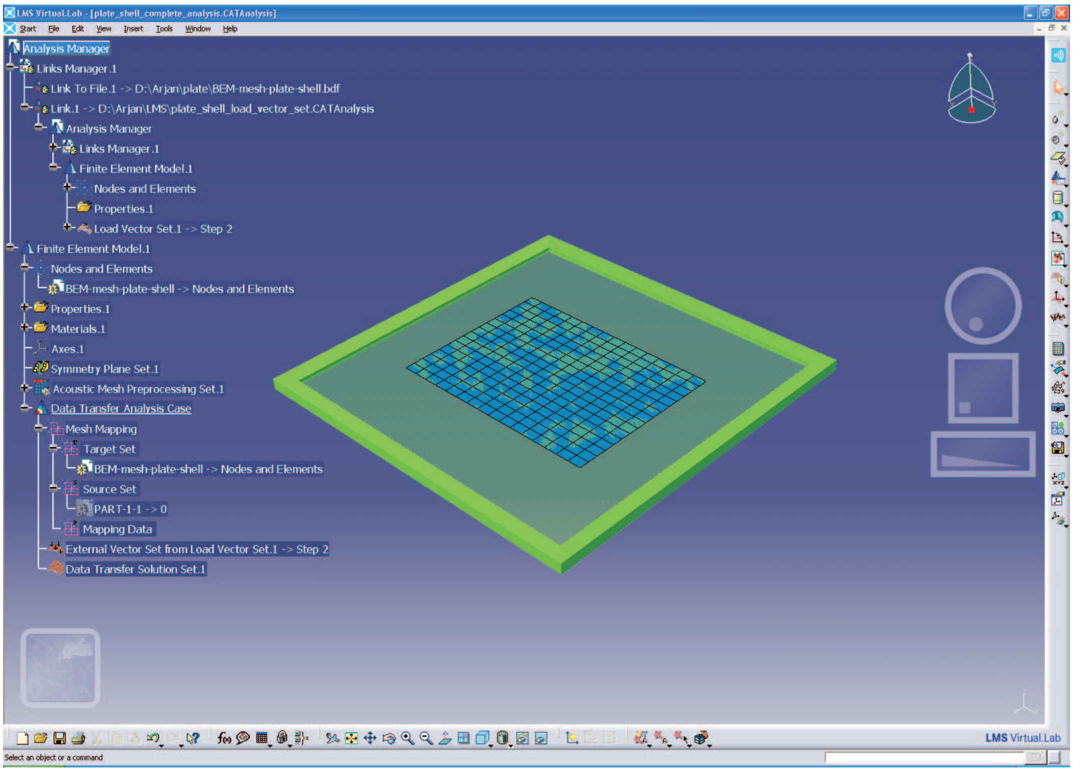


Figure 4.8: The Data Transfer Analysis case.

The BEM direct methodology requires a strictly consistent mesh definition. The direction of the element normal vectors have to be consistent and on free edges the free edge boundary condition should be applied. This is achieved by the Mesh Preprocessing Set. LMS Virtual.Lab will automatically prepare the acoustic mesh such that it complies with these requirements. This is achieved by inserting a Mesh Preprocessing Set, selecting the acoustic mesh and clicking update.

Next step is to apply a new fluid material and property. In this case the fluid medium is air and it can be inserted from the insert menu.

The following step is to insert a Data Transfer Analysis case to transfer the vibrational data from the Load Vector Set to the acoustic mesh. First the structural CATAnalysis file that contains the Load Vector Set is imported into the recent analysis. Next the Data Transfer Analysis Case is created. In the Data Transfer Analysis Case dialog, Create a New Mesh Mapping Set is selected and the Load Vector Set is selected to transfer. This will create a new Data Transfer Analysis Case in the specification tree. Now the acoustic mesh is set as the Target Set and the structural mesh (in the links manager) as the Source Set. In the Mapping Data feature the MaxDistance algorithm is used to transfer the structural vibrations data to the acoustic mesh. In Figure 4.8 it can be seen what the specification tree looks like at this moment.

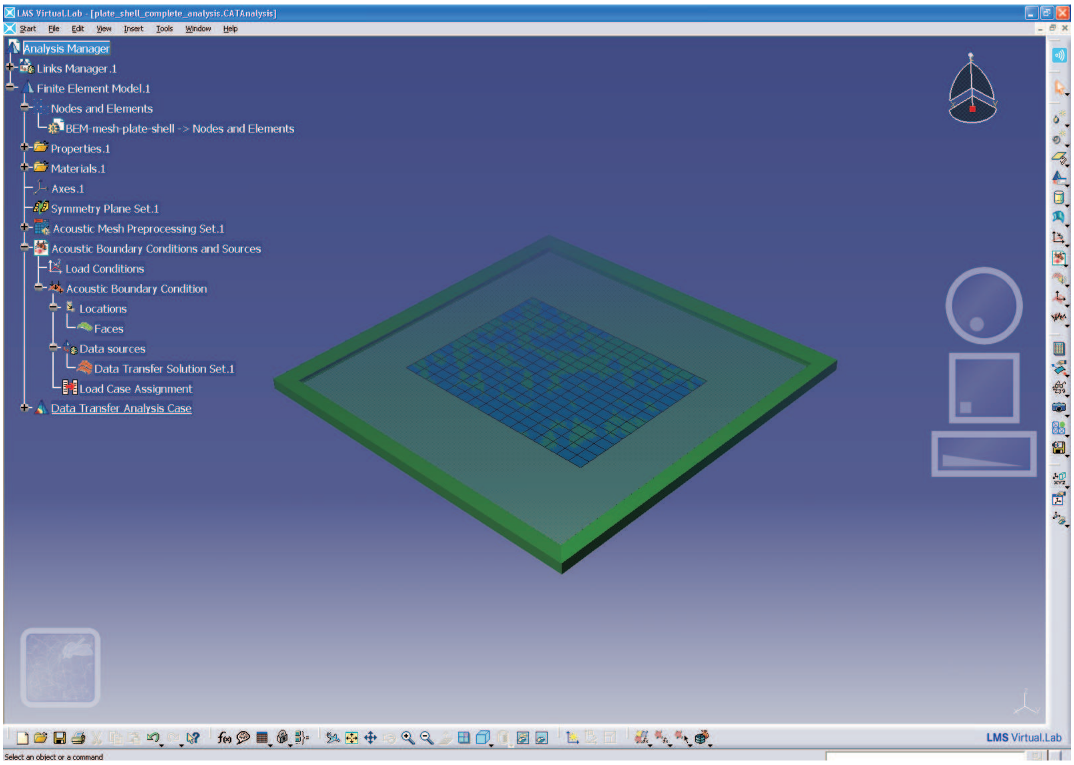


Figure 4.9: The specification tree after inserting the Boundary Condition Set.

To define vibration boundary conditions a Boundary Condition Set feature is inserted in the specification tree. The Load Conditions are set to Automatic and the load conditions to DataSource + Test/Subcase. Then a Boundary Condition, with velocities as physical type and vector as mathematical type, are added. The acoustic mesh part is assigned under the Faces feature and the Data Transfer Set is assigned under the Data Source feature. This completes the Boundary Condition Set and the specification tree becomes like in Figure 4.9.

Finally the Acoustic Response Analysis Case is created. In the Acoustic Response Analysis Case dialog the just created Boundary Condition Set can be assigned. The solution parameters of the Acoustic Response Analysis Case can be adjusted, e.g. for the frequency spectra the values from the boundary conditions can be used. Also it can be adjusted whether SYSNOISE performs the computations in synchronous mode (LMS waits for SYSNOISE to complete the job, while it is possible to use other software simultaneously) or in manual mode (LMS creates files to be executed by SYSNOISE separately, outside LMS). The latter one will use all computer power available and consequently will be faster. Moreover, some adjustments can be made to these files for extra options, like exporting the BEM matrices **A** and **B**. This is discussed in more detail in Appendix B.

Chapter 5

Conclusion and Recommendations

In this report numerical results for the sound radiation of a baffled rectangular plate is compared with theory. For the structural vibration analysis ABAQUS is used to determine the response of the plate to a harmonic point force. Moreover, ABAQUS is used to calculate the first ten eigenfrequencies of the plate. These eigenfrequencies match the theoretical eigenfrequencies up to 92.9 %. To get a better match between the numerical and theoretical eigenfrequencies, especially for the higher frequencies, more elements should be used.

For the acoustic response analysis LMS Virtual.Lab and SYSNOISE are used. The theoretical and numerical eigenvalues of the radiation resistance matrix are compared. They match up to 83 % for frequencies below 1200 Hz. Above this frequency the error will grow. The numerical results for the total radiated sound power agrees well with the theory for low frequencies, below 240 Hz. Above this frequency the results match less. This is due to the different interpolation functions that are used. The theoretical results are obtained by a bilinear interpolation of the velocities at the nodes to the element centers, while the numerical integration uses Gauss quadrature rules using two Gauss points, which will give more accurate results. Besides the use of more accurate quadrature rules, more elements will give better results in the acoustic analysis for higher frequencies, so that more elements per wavelength are used.

Bibliography

[Fah87] F. Fahy and P. Gardonio. Sound and structural vibration, radiation, transmission and response. 2:145–175, 1987.

[Hea02] M.T. Heath. Scientific computing, an introductory survey. 2:351–353, 2002.

[Vis04] R. Vissers. A boundary element approach to acoustic radiation and source identification. 1:22–52, 2004.

Appendix A

ABAQUS input file

The input file for ABAQUS that is used for the numerical results in the structural analysis of the plate is listed below.

```
*HEADING
PLATE
*NODE, NSET=ENDS
1, 0., 0., 0.
17, 0.314, 0., 0.
273, 0., 0.414, 0.
289, 0.314, 0.414,0.
*NGEN
1, 17, 1
1, 273, 17
17, 289, 17
273, 289, 1
18, 34, 1
35, 51, 1
52, 68, 1
69, 85, 1
86, 102, 1
103, 119, 1
120, 136, 1
137, 153, 1
154, 170, 1
171, 187, 1
188, 204, 1
205, 221, 1
222, 238, 1
239, 255, 1
256, 272, 1
273, 289, 1
*NSET,NSET=RAND,GENERATE,UNSORTED
 1, 17, 1
 18, 256, 17
 273, 289, 1
 34, 272, 17
*NSET,NSET=BASE,GENERATE,UNSORTED
 1, 289, 1
*ELEMENT, TYPE=S4
```

```
1, 1, 2, 19, 18
*ELGEN
1, 16, 1, 1, 16, 17, 16
*ELSET, ELSET=PLATE, GENERATE
1, 256, 1
*SHELL SECTION, ELSET=PLATE, MATERIAL=ALU
0.002
*MATERIAL, NAME=ALU
*ELASTIC
71.E9, 0.33
*DENSITY
2700
*BOUNDARY
 RAND, 1,3
********************************************
********************************************
*STEP,INC=1,NLGEOM=YES
 1:  FREQUENCY_ANALYSIS
*FREQUENCY, EIGENSOLVER=LANCZOS
 10
*BOUNDARY,OP=NEW
 RAND, 1,3
*EL PRINT,FREQ=0
*OUTPUT,FIELD,FREQ=1
*NODE OUTPUT
 U
*OUTPUT,HISTORY,FREQ=1
*END STEP
*********************************************
*********************************************
*STEP,NLGEOM=YES
 2:  FREQUENCY RESPONSE: STEADY STATE DYNAMICS,
     DIRECT
*STEADY STATE DYNAMICS,DIRECT,
 INTERVAL=RANGE,FREQUENCY SCALE=LINEAR
 1,12001,6000
*CLOAD
 145,3,1.
*OUTPUT,FIELD,FREQ=1
*NODE OUTPUT
 V
*OUTPUT,HISTORY,FREQ=1
*END STEP
```

Appendix B

Export of BEM matrices

The acoustic response calculations are executed with LMS Virtual.Lab and SYSNOISE. The calculated A and B matrices in (2.5) can be exported to a file in a so called free format. This is only possible to calculate for a direct BEM uncoupled analysis. To calculate the matrices the Acoustic Response Analysis Case has to be executed manually resulting in a new `.cmd` and `.bat` file in the specified folder. Now the `.cmd` file has to be adapted with the following command:

```
COMPUTE ABMATRIX
 FILE "filename" FORMAT Free
 FREQUENCY "beginfrequency" TO "endfrequency" LINSTEP "frequencystep"
 NEAR 2
 FAR 5
 QUADRATURE 2 2 1
 RETURN
```

Next step is to execute the `.bat` file by double clicking this file. It will use SYSNOISE with no graphical user interface installed in the LMS Virtual.Lab destination folder to create a sysnoise database file with extension `.sdb`, that has to be attached to the Acoustic Response Analysis Case in LMS Virtual.Lab.

The used `.cmd` file for the numerical results from this report is listed below:

```
ENVIRONMENT SECTION SETUP USRDIR 'D:\Arjan\Sysnoise\' RETURN
ENVIRONMENT SECTION SETUP TMPDIR 'D:\Arjan\Sysnoise\' RETURN
Open Model 1 File s030293-638-Acoustic.sdb Original Return
Extract Summary Return
Environment Section SETUP BELL 'on' Return
Environment Section SETUP GEO_TOLERANCE '0.001' Return
Parameter Model 1
  Physical

  Save Potentials Step 1
  Save Results none
  Store Results none
  Return
```

```
 Near 2
 Far 5
 Quadrature 2 2 1
 Return
Solve
 Frequency 0 1 3 5 7 9
 ...
 Frequency 11989 11991 11993 11995 11997 11999
 Frequency 12001
Return
Save Return
COMPUTE ABMATRIX
 FILE bemmatrices FORMAT Free
 FREQUENCY 2000 TO 8000 LINSTEP 2000
 NEAR 2
 FAR 5
 QUADRATURE 2 2 1
 RETURN
Close Return
New Model 1 File s030293-638-SignalFile.sdb Return
Save Return
Exit
```

This script will create the files `bemmatrices.fre` and `s030293-638-SignalFile.sdb` containing the BEM matrices and the results of the Acoustic Response Analysis Case respectively.

The `bemmatrices.fre` file has the following layout:

```
Header Section

TITLE   Title
NNBEM   Number of BEM nodes (number of elements in contact with the air)
LNUSR   List of BEM nodes (external numbers)

Matrix Section

Freq    Frequency
AMATR   [A] matrix written column by column
BMATR   [B] matrix written column by column
```

The header section is written only once. The matrix section can be written several times, once for each frequency. On each line three complex coefficients are written. So, in the case of the plate with 289 nodes, each matrix is represented by 83521 (289 times 289) complex numbers. Consequently, the `bemmatrices.fre` file will contain 167042 (2 times 83521) complex coefficients for each the **A** and **B** matrix.

To import the results in Matlab, first for each frequency a new text file with only the contents of the matrices is distracted from the `bemmatrices.fre` file. Secondly, a new complex matrix can be distracted from this text file.