Challenges in Technical Acoustics: What Can Be Computed Today

Otto von Estorff

Hamburg University of Technology, Institute of Modelling and Computation
Hamburg, Germany

Abstract:

Numerical procedures, such as the finite element and boundary element methods, are known to be very suitable tools for the investigation of acoustical problems. It turned out, however, that early formulations of these methodologies are rather computer time consuming and therefore not used very often in practical applications. The contribution aims to give an overview of the current possibilities of numerical methods and shows some facts of today's real live applications. Also the fears that computer simulations will replace measurements completely will be addressed by means of representative examples. Finally, it will be discussed how in the future modern computations will be performed and which topics will be of particular interest in current research. All methodologies discussed in this contribution are introduced briefly and illustrated by examples which show how the approaches can be used and how accurate they are.

Keywords:

Acoustics, Wave Propagation, Fluid-Structure-Interaction, Sound Transfer, Finite Element Method, Boundary Element Method, Statistical Energy Analysis

1 Introduction

The increasing awareness of comfort amongst passengers of vehicles and aircrafts, the growing demand for acoustically optimized systems, as well as the overall need for noise reduction lead to the necessity of a continuous progress with respect to the development of simulation tools. These must enable the developers of technical systems to predict the vibroacoustic behavior of a component long before its prototype exists.

In order to be able to optimize the acoustics, it is essential to develop numerical tools which enable the vibroacoustic computation across the entire audible frequency range. Moreover, these tools must offer a high flexibility as well as the possibility to model the considered system in all its details.

Even by making use of the enormous computational power available today and by applying highly efficient algorithms, the development of such numerical models is rather difficult, in particular due to the wide frequency range and, in certain cases, the large size of the structures under investigation. Further difficulties may arise from complex material configurations which need to be modeled when investigating the dynamics of advanced components, e.g. light weight sandwich structures.

For the numerical analysis in the lower and medium frequency range, the usage of element based methodologies, such as the *Finite Element Method (FEM)* and the *Boundary Element Method (BEM)*, is quite feasible. However, in order to perform the numerical analysis in the high frequency range, where uncertainties regarding the structural properties and therefore the principle limitations with respect to the modeling accuracy result in a significant effect on the expected dynamical response of the structure, an approach based on the *Statistical Energy Analysis (SEA)* should be employed.

The present paper gives an overview of the methodologies currently available. Emphasis is placed on the *FEM* and the *BEM* in the frequency domain, but also *SEA* approaches are addressed briefly. Typical applications have been selected to demonstrate the capabilities but also the limitations of the methodologies, showing that measurements will stay an important part in the design process of acoustical systems. Additionally, it is tried to identify some future development tendencies.

2 Available Methodologies

When investigating acoustical problems, mainly discretization methods such as the *finite element method* [1], [5], [7], [15], [17] or the *boundary element method* [3], [9], [17], [19] are applied. Both methodologies are well suited for the investigation of those problems, where the physical behavior of an acoustic medium, e.g. air or water, can be described by the *Helmholtz* equation. Due to the differences in their formulation, the two approaches have in turn some advantages and drawbacks, which shall be summarized in the following.
At the end of this paragraph also some coupled *FEM/BEM* computations as well as the *SEA* [13] will be considered.

2.1 Finite Element Method

The finite element method is a domain type methodology which means that the complete three dimensional acoustic domain needs to be discretized using separate elements. For interior problems, e.g. noise reduction inside passenger compartments, this is a well introduced procedure and it is rather straightforward to take into account also absorbing boundary conditions or vibrating surfaces.

In the case of exterior problems, e.g. when the sound radiation of an engine component shall be investigated, waves radiating to infinity are difficult to treat. Since, however, the *FEM* has many advantages compared to other discretization techniques, some researchers started to develop so-called *"Infinite finite elements"* which work quite well [2], [7], [17]. Moreover, transient problems and inhomogeneous acoustical media can be handled in an uncomplicated manner.

The concept of the infinite elements is as follows: First, the vibrating system is surrounded by a closed surface, usually by a sphere or by an ellipsoid. Then the near field between the structure and the enclosing surface (envelope) is modeled with conventional acoustic *finite elements* which are coupled to the infinite elements at the envelope. These elements are formulated in such a way that in one direction the approximation and shape functions are able to account for the sound radiation to infinity. Details may be found in [2], [7], [17].

2.2 Boundary Element Method

The *boundary element method* belongs to the group of boundary type formulations since only the surface (boundary) of an acoustical fluid needs to be discretized. The basic idea is to express the physical phenomena inside the acoustic medium, for instance the propagation of the sound waves, by physical quantities (sound pressure, sound velocity, potentials) defined at its boundaries. Such a formulation comprises two significant advantages: Compared to a *finite element* discretization the number of degrees of freedom is reduced considerably and the radiation of sound waves to *"infinity"* is implicitly included in the formulation.

A major drawback of the *BEM* must be seen in the fact that the methodology leads to a fully populated system of equations which needs to be solved [9], [10], [17], [19]. Moreover, the *BEM* is not as popular as the *FEM* and most of the numerical algorithms developed in the past for symmetric and positive definite matrices cannot be used.

2.3 Coupling of *FEM* and *BEM*

To allow numerical investigations of the vibroacoustic behavior of thin walled structures or structures which are in contact with a heavy acoustic fluid, a computer model must be able to take coupling effects between the structure and the fluid into account. In the majority of cases, the structure is modeled using conventional *finite elements*, while the adjacent fluid is represented by a *boundary element* mesh. The coupling of both subsystems results in a coupled system of equations, whose solution leads to the structural behavior under the influence of the fluid as well as to the acoustic pressure inside the acoustic medium [9], [17].

The coupling procedure is of particular importance if sound transmission through a structure needs to be investigated. Here it is possible to directly compute the transmission through a closed structural wall, i.e. an iteration is not needed [9], [17].

2.4 Statistical energy analysis

The *statistical energy analysis* is a powerful method for predicting and analyzing the vibration behavior of coupled fluid-structure systems [13]. Its main use is for complex systems that can be considered as an assemblage of interconnected subsystems which are subjected to high frequency vibration sources. In fact, when such a structure is excited in several different ways by different sources, the *SEA* energy balance equations result in a set of linear equations that can be used to calculate loss factors, coupling loss factors or net energy flows and incoming powers. The major advantage of the *SEA* is its applicability in the high frequency range where the modal density is high and discretization methods like *FEM* and *BEM* cannot be used anymore. However, for mid and low frequencies, where the shapes of the occurring modes and not only their energy contents are of importance, the *SEA* is known to be less reliable.

3 General Procedure

Usually, the starting point of a computational investigation is a CAD model of a sound generating structural surface. By means of special interfaces, the geometrical data are either directly imported in a *finite element* code (e.g. *ABAQUS, ANSYS, NASTRAN, PERMAS*) or in a pre processor (e.g. *IDEAS, PATRAN*). In both cases, a *finite element* mesh is generated afterwards, where the overall computational and manual effort to complete this mesh strongly depends on the kind of elements (h-method, p-method) used.

Once the computational model is finished, mostly a *FE* analysis is performed in order to determine the velocities of the vibrating structural surface. Next, these surface results are imported together with the structural mesh in an acoustical computer software (e.g., *SYSNOISE*, *Virtual.Lab*). Here all quantities important for acousticians, such as sound pressure levels, velocities, intensities or radiated power, are calculated. Sometimes also some optimization algorithms are employed to directly seek for the best acoustical solution.

In real life applications, sound radiating systems are often very complex. Therefore, before starting with setting up a computational model, it is strongly recommended to investigate the prototype of an already existing system. Thus, by comparing measurements and computations, a numerical model can be developed which leads to rather reliable results with respect to the overall vibroacoustic behavior of the investigated system. During the course of this very first step, often a detailed investigation of single components and structural parts is needed. As soon as the prototype model and all its components are validated, variants, problems and new developments can be investigated and optimized by computations only. The complete procedure will be explained in more detail by means of an example where the sound radiation of a complete washing machine will be computed and compared to measurements.

4 Representative Examples

Next, the procedures summarized above will be discussed on the basis of representative examples. It should be mentioned that the objective of this contribution is to provide a general overview of today's possibilities with respect to computational methods in acoustics. Therefore, the examples are not described in all their technical details. These may be taken from the original references given throughout the text.

4.1 Sound pressure distributions in closed rooms

The sound pressure distribution inside closed rooms, e.g. passenger cabins or vehicle components, is mostly investigated using *finite elements*. In Figure 1 (left) the result of a typical *FE* analysis is shown: the acoustic mode of a passenger compartment. Acoustic modes clearly indicate, at which locations inside the compartment a significant increase of the acoustic pressure has to be expected as soon as the excitation frequency gets close to the respective eigenfrequencies. The calculation of modes is a standard option in the *FEM* and can be seen as an advantage of the *FEM* compared to the *BEM*.

It should be mentioned that also absorbing boundaries can be taken into account. In this case different surface impedances, measured before, e.g., in an impedance tube, can be prescribed at all surfaces of the interior air volume. Thus, an optimized vibroacoustic layout can be determined.

In the case of the passenger car compartment, the highest frequency that may be investigated using *finite elements* is at about 180 Hz. This upper bound is defined by the modal density of the coupled fluid-structure system. If higher frequencies shall be investigated, either *geometrical methodologies* or the *statistical energy analysis* have to be used.

Of course, also the *boundary element method* can be used to model the interior sound pressure distribution. However, since the acoustic fluid is represented only by a discretization of its surfaces, special points, so-called field points, have to be defined. These can be used later on to show the pressure distribution inside the fluid [9], [17]. In Figure 1 (right), four of these field point meshes, also referred to as „visualization surfaces", are depicted. They were chosen in order to display the acoustic pressure in a cabin section of an aircraft. It should be mentioned that the calculation of the sound pressure at these points is just another evaluation of the governing integral equations and not time consuming at all.

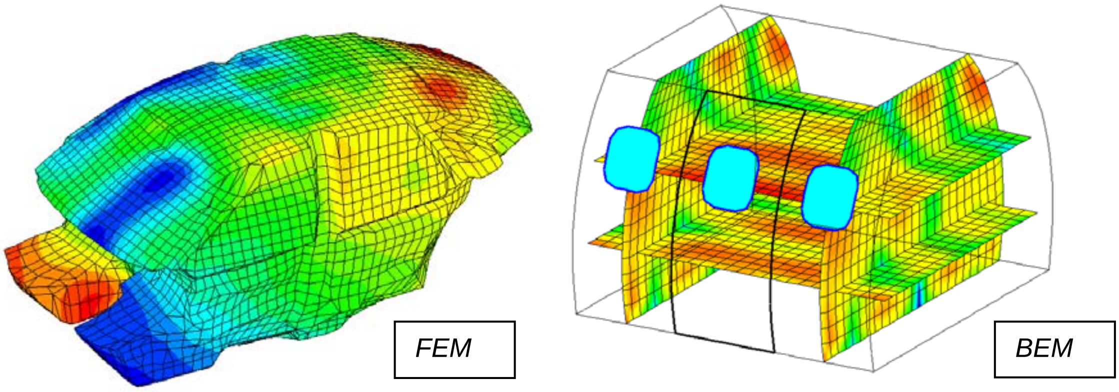

Figure 1: Sound pressure in a car and in a cabin section of an aircraft

4.2 Sound radiation

In the majority of cases, sound radiation problems are investigated by means of the *boundary element method*, since the energy loss at infinity is implicitly included in the formulation. A typical application is the aircraft fuselage shown in Figure 2. In a first step of the computation, the surfaces of the turbine and the fuselage have been modeled by means of *boundary elements*. This discretization was used afterwards to compute the sound pressure distributions for several frequencies. The objective of such calculations is the prediction and, based on that, the reduction of the noise radiating from the turbines. In particular, also the sound transmission into the cabin needs to be reduced following the demand for more passenger comfort.

The sound pressure distribution in the vicinity of a tire is given in Figure 3. It has been investigated by using *boundary elements* to model the surfaces of the tire and the road. It should be mentioned that the interaction of the tire and the road is of significant importance. Moreover, it is possible to take into account rigid as well as absorbing boundary conditions. The excitation of the tire has been extracted from measurements of the texture of the road [2].

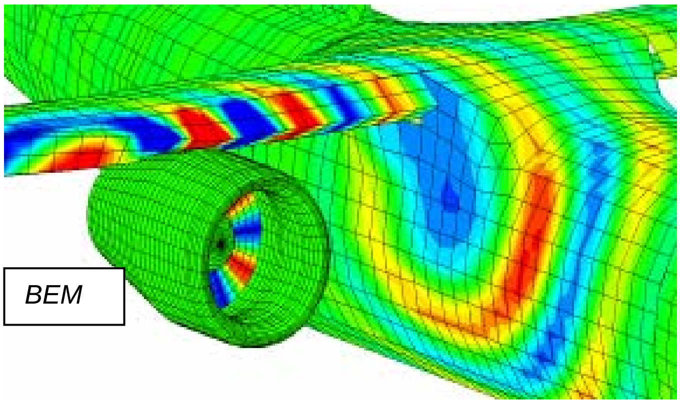

Figure 2: Sound pressure distribution at the surface of an aircraft section

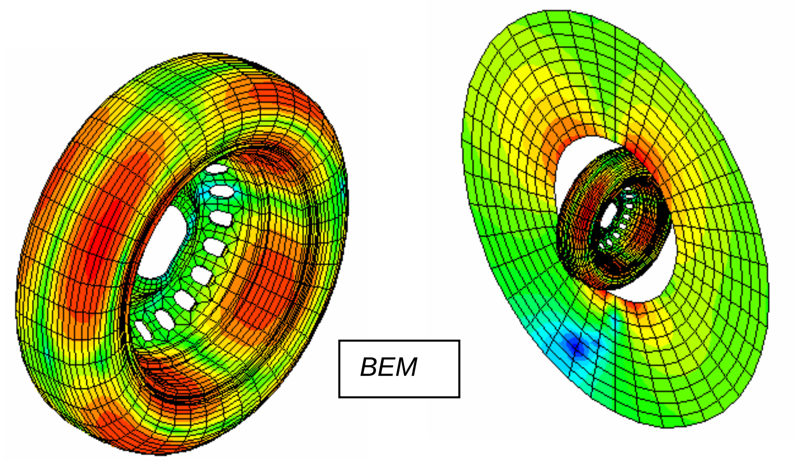

Figure 3: Sound pressure distribution at the surface and in the vicinity of a tire

Instead of using *boundary elements* for sound radiation investigations, also *infinite elements* can be used. The vibrating system, i.e. the tire, is first surrounded by a closed surface, usually by a sphere or by an ellipsoid. Then, the near field between the tire and the enclosing surface is modeled by conventional acoustic *finite elements* which are coupled to the *infinite elements* along the enclosing surface. A considerable advantage of infinite elements must be seen in the fact that they can be integrated easily in existing finite element programs. Moreover, the resulting systems of equations can be solved by means of iterative solvers [7]. In Figure 4 the infinite element model of a typical tire is shown.

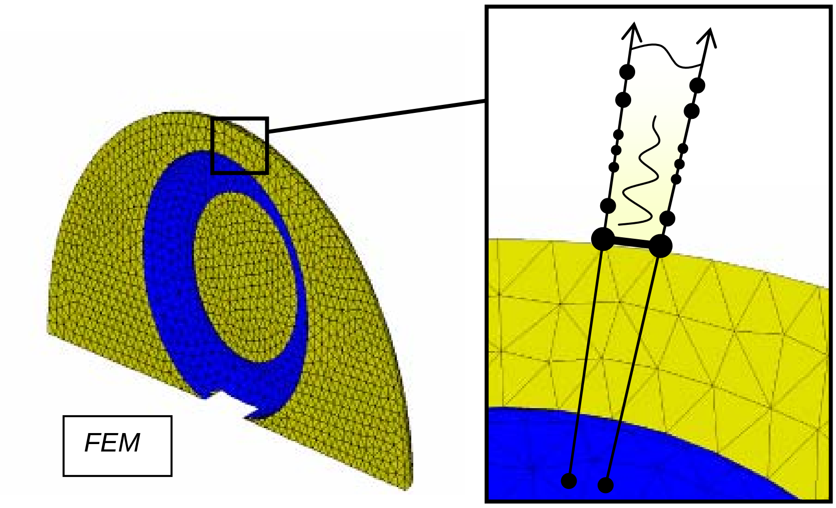

Figure 4: Discretization of the air around a tire using infinite elements

4.3 Sound transmission

Looking at the discussed features of the numerical methods, one interesting field of application is the sound transmission through walls, panels or windows. In particular, one is interested in the so-called transmission loss which characterizes quite well the vibroacoustic behavior and quality of a wall or panel. In this section, the models of two special wall constructions, namely a honeycomb sandwich plate and an aircraft side wall panel, will be discussed in more detail.

Interior aircraft lining elements often consist of honeycomb sandwich plates. The main advantage of such plates is their low weight in conjunction with a rather high mechanical stiffness [3], [11], [12], [14], [18]. A major drawback, however, is their low sound insulation.

The numerical model to determine the transmission loss of the sandwich plates can be divided into two parts: A finite element model describing the structural behavior of the honeycomb sandwich plate and a coupled acoustic model, based on the baffled formulation of the boundary element method as suggested by McCulloch [6]. More details can be found in [11].

Two different types of finite elements are used for the two materials of the sandwich plate. The facing material is modeled with linear shell elements. It is assumed that the woven structure of the facing material can be represented by an isotropic material model. The honeycomb core, however, is characterized by showing different properties in different directions. Compared to the skin material, it has a low tensile stiffness in the plane direction, but it is able to transmit high shear forces in the transversal direction. Therefore, the honeycomb core is modeled by linear brick elements, employing an orthotropic material model. Figure 5 shows the *finite element* mesh consisting of 8100 elements.

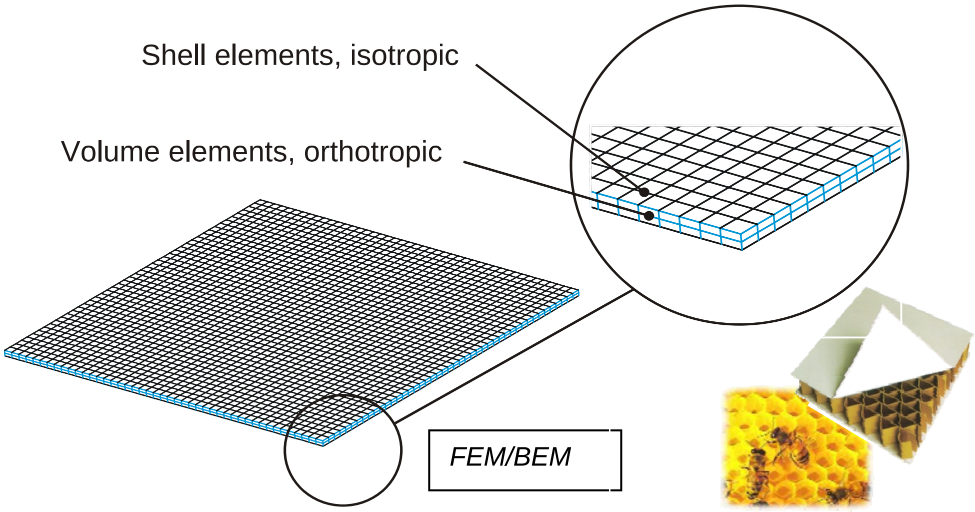

Figure 5: Structural finite element model of a honeycomb plate

To measure its transmission loss, the honeycomb plate has been installed in a test facility consisting of two rooms, separated by a heavy wall with a rectangular opening (window). The window is the only connection between the two rooms. While the source room is a reverberant chamber, where a diffuse sound field is generated by a random sound source, the receiver room is an anechoic chamber (see Figure 6). The honeycomb plate was fitted into the window and its active sound intensity has been determined. Moreover, in the reverberant room, the sound pressure has been measured. Once these values are known, it is straight forward to determine the transmission loss TL defined, e.g., in [11].

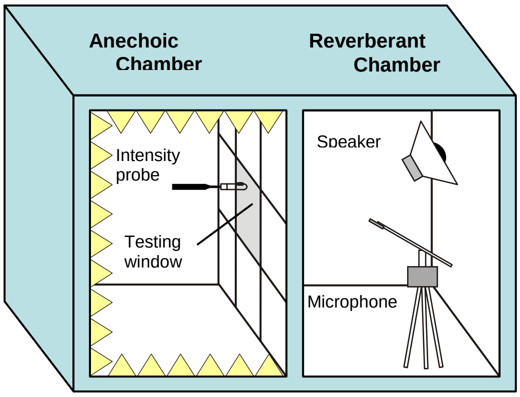

Figure 6: Anechoic and reverberant chamber to measure the transmission loss

For the calculation of the transmission loss, a coupled *FEM/BEM* model is used. The acoustic model employs a special adaptation of the indirect variational baffled *BEM* which is described in detail in [9], [17]. Figure 7 shows the transmission loss factor obtained by measurements and calculations. It is apparent that, over the whole frequency range considered, the computed results match very well with the measured ones.

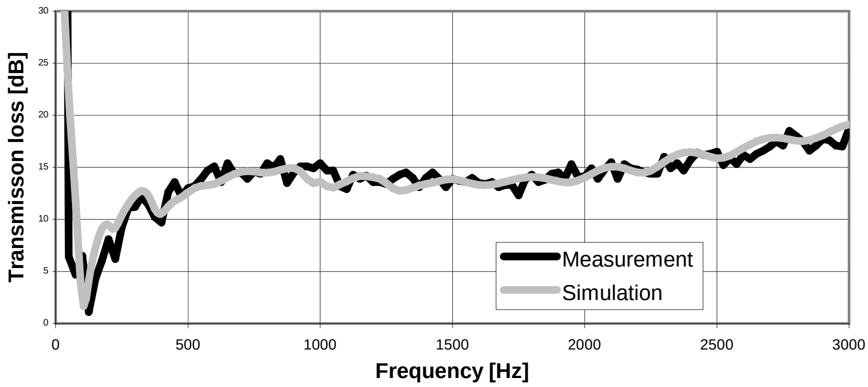

Figure 7: Measured and computed transmission loss of a honeycomb plate

Recently, the procedure described above also has been applied to more complicated structures, namely to a side wall panel of an aircraft as given in Figure 8 (left). While the *FEM* and the *BEM* have been used in the lower frequency range, *SEA* approaches implemented in the commercial code *AutoSEA* [8] and in the public domain software *OpenSourceSEA* [4], [20] have been employed at higher frequencies. Figure 8 (right) shows a typical *SEA* model where a detailed element discretization is not needed. The measured and computed transmission loss is given in Figure 9. Also in this case, the agreement is excellent. More details are given in [3].

Figure 8: Side wall panel of an aircraft and its SEA model

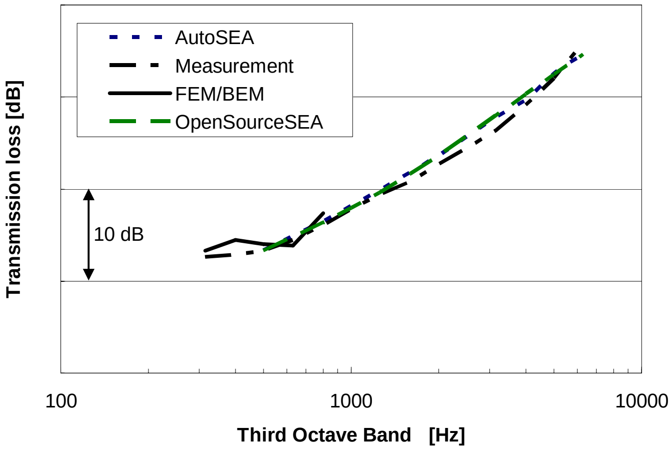

Figure 9: Measurement and computation of the transmission loss of an aircraft side wall panel

4.4 Investigation of complex systems

Due to the continuous improvements of the numerical methods and the fact, that the computer power increases continuously, the investigation of rather complex and realistic systems has become feasible. As a representative example, a complete washing machine will be discussed next. The example will show that even in the case of complicated technical systems a reliable vibroacoustic model can be obtained. One major prerequisite, however, is that there has to be a chance to investigate all the details needed for the overall model later on. The complete investigation is documented in [1].

When starting with the development of a complete vibroacoustic model for the washing machine, a major challenge was the identification of those subsystems which might have an influence on the acoustics. It became clear, that in order to obtain a generally applicable model, different computation procedures had to be combined with measured values. The final numerical model is based on a combination of the conventional *finite element method* to determine structural vibrations, *multi-body-dynamic (MBD) algorithms* to analyze the machine dynamics, and the *boundary element method* to compute the radiated acoustic waves. The dynamic characteristics of single components of the structural model have been specifically investigated by accompanying measurements and it took several steps until the complete model had been assembled.

The major steps and findings can be summarized as follows:

First, measurements of the acoustic pressure have been carried out at several spin-dry speeds of the investigated washing machine. The raw data of six microphones (time series) were analyzed and the characteristic features of the acoustic emission determined. In the course of these studies, it has been realized, for instance, that despite of a harmonical excitation due to an unbalanced drum, the acoustic pressure spectra show a strong poly-harmonical acoustic radiation, caused by poly-harmonical vibrations of the washing aggregate. Such a response to a harmonical excitation is usually caused by strong nonlinearities within the system, for which in this case the suspension system (springs and dampers) is responsible.

Second, the inputs of the single components of the washing aggregate (engine, leach tank, pump, and side panels) have been determined by means of a spectral analysis of the obtained signals. Of course, the determined contributions of the single components depend on the frequency.

Third, the dynamic behavior of the absorbers had to be investigated. Each of the dampers was fixed with one end at a rigid wall by a DMS-force sensor and the damper piston was excited by an electromagnetic shaker, see Figure 10 (left). The exciter has been controlled by a signal generator through a power amplifier, and the amplitude as well as the frequency of the excitation power could be controlled. The acceleration of the oscillating clamp of the exciter has been measured with an accelerometer. The measurements have been carried out at small (approx. 1 mm) and large (2-8 mm) piston amplitudes for those excitation frequencies which correspond to the spin-dry speed. As expected, the behavior was very different [1], a fact that had to be taken into account within the numerical model.

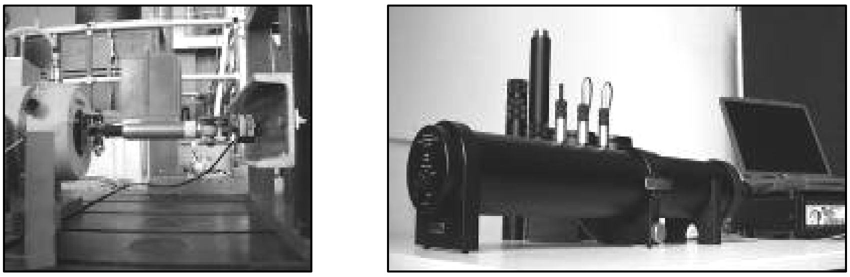

Figure 10: Measurement of damping characteristics and impedances

In the fourth step, a parameter determination for the acoustic model had to be performed. By means of an impedance tube, see Figure 10 (right), characteristic data of various fleece mats, which are applied inside of the investigated washing machine, have been determined. These cotton mats are important for the acoustic insulation of the washing machine and had to be considered in the acoustic model correspondingly. It has been shown that the absorption coefficient of a coated wall can be changed significantly by means of cotton fleece mats of different thicknesses.

Finally, based on the preparing measurements, a non-linear dynamic model of the washing machine had to be developed, in particular to take into account the transient loading characteristics of the machine. This is based on an *elastic multi-body-dynamics (EMBD)* model, where a flexible *FE* structure is integrated in a *multi-body-dynamics* simulation. In particular, the coupled vibrations of the rigid aggregate and the flexible case of the machine are linked by an orthogonal Craig-Bampton modal basis [1].

Additionally, the Craig-Bampton orthogonal modes should be computed for all elastic structures of the system. The integration of the elastic body structure in the *MBD* model is depicted in Figure 11.

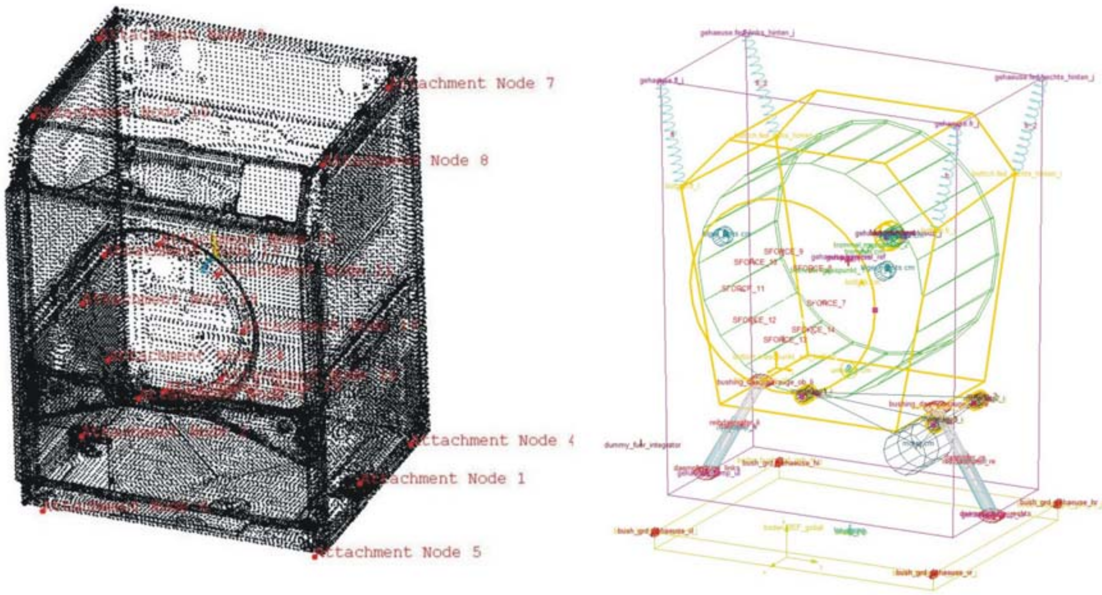

Figure 11: Integration of the elastic body structure in the MBD simulation

In order to transfer the simulation results of the dynamic model into the acoustic models, a numerical interface has been developed. This directly allows, for instance, the creation of the modal matrix in a form that can be transferred into the acoustic mesh as well as the determination of the eigenvectors of the dynamic model, projected onto the acoustic mesh. Furthermore, the complex amplitudes of the surface velocities are determined at the nodes of the acoustic mesh at those frequencies, at which the acoustic propagation shall be computed. Note that these data are used as input values, which incorporate the motion of the sound radiating surfaces into the acoustic model of the washing machine.

The computational investigation of the acoustic radiation is carried out by using the *indirect boundary element method*, which allows - contrary to the *direct BEM* - also to compute structures with free edges as well as branched surfaces.

Based on the structural CAD mesh of the washing machine, the boundary element mesh has to be developed as a pure surface mesh (see Figure 12). This includes a reduction of the degrees of freedom by eliminating small structural details, which certainly have an influence on the dynamic behavior of the washing machine, but do not effect its acoustic radiation. Further details are given in [1].

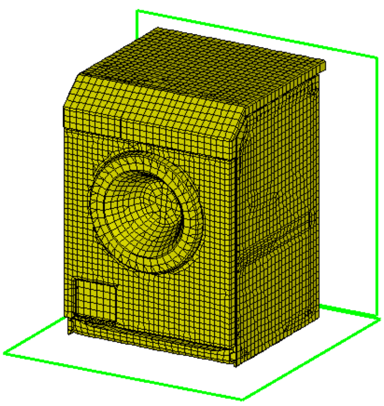

Figure 12: MBD/FEM/BEM model of a washing machine

During the measurements, the washing machine has been placed on a rigid floor in front of a reverberant wall. Floor and wall have been modeled by means of so-called "symmetry plane conditions", especially provided for such [17] and marked with squares. The remaining walls of the test chamber were fully absorbing, which means that there was no need to model them.

On the inner walls of the computational machine model, admittance boundary conditions have been defined according to the distribution of the mats of cotton fleece.

In order to evaluate the computational results in front of the washing machine, six field points have been defined. Their positions correspond to the positions of the microphones during the sound pressure measurement. To get information about the overall noise which is radiated by the washing machine, the locally determined acoustic pressures can be converted into the acoustic power by means of a standardized approximation formula. This conversion formula is given by the standard EN 60704-1, which was also used to determine the positions of the six field points (microphone positions) needed [1].

Figure 13 shows the third octave bands of the measured and computed radiated power of the complete washing machine. In the whole frequency range considered, the agreement is excellent. The largest difference of just under 3 dB appears in the 400 Hz third octave band. In the majority of frequencies it is considerably smaller.

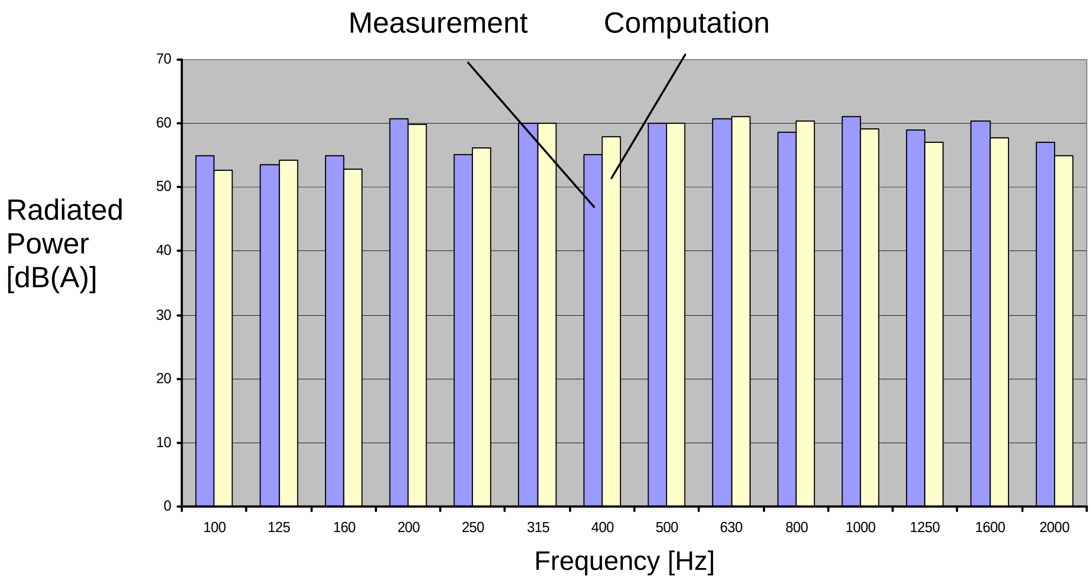

Figure 13: Comparison of measured and computed sound power

5 Future Developments

Looking at today’s developments, a clear trend towards making numerical computations faster [10] and easier to handle [17] can be observed. Having these overall objectives in mind, the following aspects might be of interest in the future:

- Completion of the existing numerical methods, e.g., with respect to the time domain
- Seeking for alternative solution strategies, such as iterative procedures, multigrid approaches, multipole formulations, and multi frequency algorithms
- Combination of existing methodologies (hybrid formulations) and usage of efficient substructure techniques
- Further development of mesh free formulations
- Simplification of computational models without loosing the required overall accuracy
- Usage of improved optimization techniques

- Development of system-oriented computation strategies with an automatic discretization and choice of the best suited methodology based on the CAD model.

In addition to the aforementioned activities, researchers are very concerned with the development of algorithms to bridge the so-called “mid frequency gap”. This is the frequency range between the low frequencies, where *FEM* and *BEM* can be applied, and the higher frequencies, where the usage of the *SEA* is recommended. More details can be found, for instance, in [16], [21], [22], [23].

6 Conclusions

The objective of the present contribution was to give an overview of standard numerical methods in acoustics and to discuss their accuracy and applicability by means of representative examples. The following conclusions could be drawn:

- *FEM* and *BEM* are well suited to handle rather complex acoustic as well as vibroacoustic problems.
- Often, the *FEM* is employed for interior problems. In combination with infinite elements, however, it can be also used to investigate sound radiation phenomena. The solution of the resulting system of equations can be obtained efficiently by iterative solvers.
- The *BEM* is mainly used to consider sound radiation problems, but it is also applicable in interior acoustics. The method is known to be highly flexible and also well suited to compute the transmission loss of walls or panels.
- Comparison between measurements and computations show that the numerical methods are sufficiently accurate. It is extremely important, however, to ensure a reliable input to the acoustic models. This means that the structural behavior has to be validated as well.
- Measurements are of particular importance not only if it comes to the validation of new algorithms, but also if very sophisticated mechanical systems shall be considered. In particular, if a new model is assembled, critical components and later on the complete system should be investigated by measurements in order to make sure that all necessary details have been taken into account in a proper way and to gain experiences with the new configuration. Based on these activities, modifications and further developments of the modeled system can be simulated without the necessity to measure in parallel.

It should be pointed out that there is no reason to be afraid that in the future all measurements will disappear. Only the combination of measurements and computations will lead to an efficient and successful technical design process.

The computational methods discussed above are already used extensively in many industrial applications. Nevertheless, the development of more efficient algorithms will be a continuous effort of all researchers. Moreover, further improvements of the overall solution process with respect to handling will be needed. It should be aimed to get closer to the dream of the user just to read a CAD model into his acoustic software, add some excitations, and obtain the solution of the corresponding acoustical problem within seconds.

It should be mentioned that most of the computations have been conducted using the acoustic software SYSNOISE [17].

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