

2 Basics and Definitions

This chapter presents the necessary acoustic basics and definitions used in this work. It starts with a short overview on the basics of sound and signal processing in section 2.1, followed by the necessary definitions for the quantification of the transmission through sealing systems in section 2.2. At the end of this chapter, the used commercial simulation tools are briefly introduced in section 2.3.

2.1 Sound and Signal Processing

To judge the noise transmission of a system, one first has to define what “noise” is. Noise can be defined as undesired sound. Since this work focuses on exterior noise, which is perceived within vehicles and which is clearly undesired, throughout this work the quantities “noise” and “sound” are used equally.

The two main sources for aeroacoustic sound are unsteady motion in a fluid or the vibrations of a structure coupled to a fluid. Small amplitudes of vibrating structures may cause pressure disturbances in the coupled fluid. A propagating sound wave can be described as a pressure disturbance in a fluid superposed to its mean pressure:

$$p(x, t) = \bar{p}(x) + p'(x, t) \quad (2.1)$$

with $p(x, t)$ being the pressure in a fluid at position x and time t , $\bar{p}(x)$ the mean pressure and $p'(x, t)$ the fluctuating pressure. The effective amplitude of the fluctuating pressure $p'(x, t)$ can be described as its root-mean-square, $p_{rms}(x)$. The wavelength λ of a periodic pressure signal is defined as

$$\lambda = \frac{c}{f}, \quad (2.2)$$

with c , the speed of sound and f , the frequency of the signal. The speed of sound in air is dependent on multiple factors, such as the temperature of the air

and the ambient pressure. At a room temperature of 18 ° Celsius and an atmospheric pressure of 101325 Pa, the speed of sound has a velocity of about 342 m/s. Equation 2.2 is valid for tonal sound, meaning for sinusoidal signals. Usually, sound is a mixture of pressure variations that vary in phase, frequency and amplitude. However, this complex sound can be regarded as a superposition of sinusoidal components. This is called a frequency spectrum of the signal. Even if a random signal does not show periodic components, a Fourier analysis can be performed, which represents the signal by a superposition of waves of all frequencies. Such a Fourier analysis for measured signals is performed with the help of the discrete Fourier transformation (DFT). Although the time-signal only provides a finite number of data points, the DFT treats the data as if it was periodic. Due to this, discontinuities are introduced into the signal. These discontinuities can be suppressed by tapering the ends of the time signal toward zero with so-called windowing functions. A fast algorithm, which solves a DFT, is the fast Fourier transformation (FFT). All frequency spectra in this thesis are computed with the FFT. When only regarding a small portion of the time-signal, the calculated spectrum is very noisy. By computing the FFT over multiple equally-sized parts of the whole time-series, so-called windows, averages of these FFTs show a better signal-to-noise ratio. The previously mentioned windowing functions are again used for a correct combination of the signals.

Throughout this thesis, the following strategy concerning FFTs is used:

- Whenever transient surface pressures are used for a transmission simulation, the FFT is calculated from a time series of the length of only one window. The ends of this time series are tapered with a 5% taper window. Only one window is used to ensure that the phase information of the excitation is not corrupted by the averaging of multiple windows.
- For the visualizations of the excitation (e.g. dB-maps or the source microphone's *PSD*), a longer time series is divided in multiple windows which are combined with a Hanning window. With this method, the FFTs results are less noisy and the actual signal can be more clearly visualized.

The spectrum of the pressure time-series $p(t)$ consists of the p_{rms} values of the superposed sinusoidal waves. The amplitude of the DFT is dependent on the frequency resolution Δf . It does not make any sense to consider noise levels at exactly one frequency, since there is zero energy in any distinct frequency. Each value of the DFT summarizes the energy of its frequency band.

The sound pressure level (*SPL*) is a logarithmic measure of the effective sound pressure of a sound relative to a reference value. It is measured in dB and given by:

$$SPL = 10 \log_{10} \left(\frac{p_{rms}^2}{p_{ref}^2} \right) = 20 \log_{10} \left(\frac{p_{rms}}{p_{ref}} \right) \quad (2.3)$$

with p_{rms} being the root-mean-square sound pressure, measured in Pa, and $p_{ref}=2.0 \cdot 10^{-5}$ Pa, the reference sound pressure. This value of p_{ref} is usually considered as the threshold of human hearing.

A way to compare DFT results with different frequency resolutions is the power spectral density (*PSD*), the power per unit bandwidth in a signal. The *PSD* in a frequency band is found by simply dividing the power of that band by its frequency range Δf . In a level notation this corresponds to

$$PSD = 10 \log_{10} \left(\frac{\left(\frac{p_{rms}^2}{\Delta f} \right)}{p_{ref,psd}^2} \right), \quad (2.4)$$

with $p_{ref,psd}^2=4 \cdot 10^{-10}$ Pa²/Hz.

In this work, all *PSDs* are computed with a constant bandwidth of 100 Hz. This coarse frequency resolution is chosen mainly due to computational costs of the involved FEM simulations, where the direct frequency solver computes each frequency individually. Instead of using constant-bandwidth bands, referred to also as “narrow-bands”, it is also a common practice to use proportional bands. The benefit of this band type is an evenly-spaced representation of a spectrum in a logarithmic frequency plot. If the data involves experiments or simulations of production vehicles, the spectra are converted into proportional one-third-octave bands.

2.2 Sound Transmission

A method, to quantify the transmission through a sealing system, is the so-called “transmission loss” (*TL*). It is defined as

$$TL = 10 \log_{10} \left(\frac{W_{src}}{W_{rec}} \right) \quad (2.5)$$

with W_{src} , the incident power on the source side and W_{rec} , the transmitted power on the receiver side. To measure the power on either the source or the receiver side, intensity measurements have to be carried out at the openings of both cavities. This procedure is demanding. An easier way to quantify the transmission through a sealing system is the noise reduction (NR), defined as

$$NR = 10 \log_{10} \left(\frac{S_{src}^2}{S_{rec}^2} \right). \quad (2.6)$$

S_{src} denotes the spectral density of the sound pressure at the microphone at the source side of the obstacle, for which the NR is determined. In the current application this refers to the microphone in the outer door cavity. S_{rec} denoted the spectral density of the transmitted sound pressure at the receiver microphone behind the obstacle - here behind the seal in the interior of the vehicle. The disadvantage of the NR is the strong dependence of the position of the involved microphones. This is not the case for the TL , since the whole applied and transmitted energy is taken into account via the intensity measurements.

Since this work focuses strongly on simulations, another quantity is introduced, which is used only in simulations with finite element methods. The so-called vibro-reduction (VR) is defined as

$$VR = 10 \log_{10} \left(\frac{\langle d_{src}^2 \rangle}{\langle d_{rec}^2 \rangle} \right), \quad (2.7)$$

with $\langle d_{src} \rangle$ being the spatial average of the squared displacement magnitudes of all interface nodes wetted by the source fluid and $\langle d_{rec} \rangle$ the same quantity but for the interface nodes of the receiver fluid. The VR reflects a reduction of the displacements at the receiver side compared to the displacements at the source side and gives insights about the sound transmission through the structural parts of sealing systems.

A quantity, which is valuable for the scenario of the noise transmission through sealing systems is the acoustic impedance. The response of the seal to the excitation leads to vibrations at the boundaries to the adjacent air volumes. The impedance Z at the fluid-structure-interface is defined as

$$Z = \frac{p}{v}, \quad (2.8)$$

with v and p being the velocity and the pressure at each node of the fluid-structure-interface. Derived from this, a simplified “interface impedance” for

whole fluid-structure-interfaces can be defined as the ratio of the spatial averages of these quantities:

$$Z_{interface} = \frac{\langle p_{interface} \rangle}{\langle v_{interface} \rangle}. \quad (2.9)$$

This quantity describes, how much vibrations of the structure's surface are converted into pressure fluctuations which emit into the adjacent fluid.

2.3 Simulations Tools

Three commercial simulation tools are used in this thesis. PowerFLOW, a commercial CFD-code based on the Lattice-Boltzmann Method (LBM) is presented in 2.3.1. NASTRAN and ABAQUS are commercial codes based on FEM. Although the implementation and the available models of both codes are different, the basics of the FEM are the same. These basics are presented in section 2.3.2. The differences between both solvers are highlighted later in section 5.6.

2.3.1 Lattice-Boltzmann Method

The following section about the LBM solver PowerFLOW is based on the short descriptions of this code used in many publications by Exa Corporation, e.g. [20].

The software PowerFLOW 4.3d is used in the present study. The core physics of the software solve the Boltzmann equation for a distribution function on a hexahedral mesh. This mesh is automatically generated around bodies consisting of one or more connected solid parts. The distribution function represents the probability to find, in an elementary volume for a given time interval, a number of fluid particles with a certain velocity. The Boltzmann equation is solved by discretizing the particle velocity space into a prescribed number of values, in magnitude and direction. These discrete velocity vectors are defined such that, in a prescribed time step, one particle is advected from one point of the mesh to 19 neighboring points, including the point itself. It can be demonstrated that using 19 particle velocity states ensures sufficient lattice symmetry to recover the Navier-Stokes equations for an isentropic flow [4]. Once the distribution function is computed, the macroscopic flow quantities,

density and linear momentum, are simply determined through discrete integrations. All the other quantities are determined through thermodynamic relationships for an ideal gas.

Solving the Lattice-Boltzmann equation is equivalent to performing a direct numerical simulation (DNS) of the Navier-Stokes equations in the limits of the dynamic range (Mach number) that can be accurately covered by the number of discrete particle velocity vectors, and in the limits of the lattice resolution required to capture the smallest scales of turbulence. For high Reynolds number flows, turbulence modeling can be incorporated into the LBM scheme by changing the relaxation time in the collision operator that is computed according to a Bhatnagar-Gross-Krook (BGK) approximation ([4]). The turbulence kinetic energy and the turbulence dissipation are obtained by solving a variant of the RNG k - ϵ model for the unresolved scales ([57]). This approach is referred to as LBM very large eddy simulation (LBM-VLES) ([4, 5, 6]).

Since it is by far too expensive to resolve the wall boundary layer using a Cartesian mesh approach down to the viscous sub-layer in high Reynolds number applications, a wall function approach is used in PowerFLOW to model boundary layers on solid surfaces. The wall model implemented in the software is an extension of the standard formulation ([28]), but it includes the effects of favorable and adverse pressure gradients, and accounts for surface roughness through a length parameter.

The LBM scheme is solved on a grid composed of cubic volumetric elements (voxels). A variable resolution by a factor two is allowed between adjacent regions. Consistently, the time step is varied by a factor two between two adjacent resolution regions. Solid surfaces are automatically facetized within each voxel intersecting the wall geometry using planar surface elements (surfels). For the no-slip and slip wall boundary conditions at each of these elements, a boundary scheme ([6]) is implemented, based on a particle bounce-back process and a specular reflection process, respectively. Therefore, very complex arbitrary geometries can be meshed and simulated automatically by the LBM discretizer and solver.

The local character of the LBM scheme allows an efficient parallelization of the solver. Moreover, thanks to the intrinsic low dissipative properties of the scheme ([2]), LBM is particularly suited for aeroacoustic simulations. The code has proven its simulation performance in many application cases concerning the simulation of the excitation for the sound transmission into the interior ([34, 45]) or concerning the response of cavities under grazing flow, like the phenomenon of sunroof buffeting ([9, 10, 11]).

The post-processing tool PowerACOUSTICS 2.0b is used to perform statistical and spectral analysis of all the solutions generated by PowerFLOW.

2.3.2 Finite Element Method

The finite element method has become a well-developed engineering discipline as of the 1960s. It was originally developed for the modeling and analysis of complex structures. Nowadays, the finite element approach is applied to a wide variety of engineering applications, ranging from heat transfer, electromagnetics and fluid dynamics up to coupled multi-physics applications. The basis of FEM is a numerical technique for finding approximate solutions to boundary value problems for partial differential equations. The whole problem domain is subdivided into smaller and simpler parts, the finite elements. The main idea of FEM is that by connecting many simple element equations, which hold for the small finite elements, the more complex equation of the whole domain is approximated. This is done by minimizing an associated error function with the help of variational methods. The main commercially available codes, which are widely used in the industry, include:

- MSC/NASTRAN by MSC.Software,
- NX/NASTRAN by Siemens,
- ANSYS by ANSYS Incorporated and
- ABAQUS by Simulia.

The FEM simulations presented in this thesis use the codes MSC/Nastran MD 2010, which will be referred to as “NASTRAN” in the following, and ABAQUS. Since FEM is a well established simulation method, the basics of this method are widely available in the literature (see for example [30, 44, 47]) and will not be covered in detail in this thesis. In the following, only the basics of the sound transmission with FEM are presented, which is realized with a frequency response analysis.

The transient excitations, which are estimated with CFD, are transformed into the frequency domain and are applied as complex harmonic loads, equivalent to periodic repetitions of the transient signal. By assuming such harmonic excitation, the damped forced equation of motion is defined as

$$\mathbf{M}\ddot{\mathbf{d}}(t) + \mathbf{B}\dot{\mathbf{d}}(t) + \mathbf{K}\mathbf{d}(t) = \mathbf{P}(\omega)e^{i\omega t}, \quad (2.10)$$

with $\mathbf{M}$, the global mass matrix, $\mathbf{B}$, the global damping matrix and $\mathbf{K}$, the complex global stiffness matrix. With a harmonic solution of the form $\mathbf{d} =$

$u(\omega)e^{i\omega t}$, with $u(\omega)$ being the complex displacement vector, equation 2.10 is transferred into the frequency domain:

$$(-\omega^2\mathbf{M} + i\omega\mathbf{B} + \mathbf{K})u(\omega) = P(\omega) \quad (2.11)$$

This equation is solved by the so-called “direct frequency response solver” of the commercial FEM codes. This solver is preferable for simulation problems with a high modal density. For problems, where the modal density is low in the given frequency range, equation 2.11 may be reduced significantly by transforming the physical coordinates $u(\omega)$ to modal coordinates $\xi(\omega)$. This step needs the additional computation of the mode shapes of the involved fluids and structures, which may be expensive for problems with a high modal density. The mathematical transformations are described in detail in [35]. The so-called “modal frequency response solver” is used to solve the equations with a modal basis. Within the investigations of this thesis, both solvers are used, depending on the modal density of the setup.

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