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Roger Ohayon and Christian Soize
Excerpt
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PRINCIPAL OBJECTIVES AND A STRATEGY FOR MODELING VIBROACOUSTIC SYSTEMS

In this book, we are interested in the analysis of *vibroacoustic systems*, which are also called *structural acoustic systems* or *fluid-structure interactions for compressible fluid* (gas or liquid). *Vibroacoustics* concerns noise and vibration of structural systems coupled with external and/or internal acoustic fluids. *Computational vibroacoustics* is understood as the numerical methods solving the equations of physics corresponding to vibroacoustics of complex structures. *Complex structures* are encountered in many industries for which vibroacoustic numerical simulations play an important role in design and certification, such as the aerospace industry (aircrafts, helicopters, launchers, satellites), automotive industry (automobiles, trucks), railway industry (high speed trains), and naval industry (ships, submarines), as well as in energy production industries (electric power plants).

Since we are interested in the analysis of general complex structural systems in the sense of computational methods defined here, we do not consider analytical or semianalytical methods devoted to structures with simple geometry, asymptotic methods mainly adapted to the high-frequency range (statistical energy analysis, diffusion of energy, etc.) and approaches that imply them. Concerning the latter, the coupling of the local dynamic equilibrium equation (finite element method) and power balances (implemented in the spirit of the statistical energy analysis) have been analyzed in Soize (1998); Shorter and Langley (2005); Cotoni et al. (2007). Nevertheless, this kind of approach is not purely "computational" in the sense described earlier and requires specific hypotheses concerning certain subsystems that are supposed to have a high-frequency dynamic behavior (high modal density).

By *advanced computational acoustics*, we mean computational methods that are adapted to the present and future generation of massively parallel computers and for which the formulations are adapted to analyze complex structural acoustic systems, and which will require a huge number of degrees of freedom (several billion) in order to correctly model structures with complex geometries and made up of different materials having complex microstructures such as composites, metamaterials and acoustic coating such as foams for soundproofing.

This book does not pretend to be a review of the existing approaches and methodologies devoted to the vibroacoustic field. Physical bases useful for vibroacoustics can be found in numerous books, such as in Morse and Ingard (1968); Cremer et al. (1988); Pierce (1989); Crighton et al. (1992); Landau and Lifchitz (1992); Junger and Feit (1993); Morand and Ohayon (1995); Ohayon and Soize (1998); Blackstock (2000); Bruneau (2006); Fahy and Gardonio (2007).

The book is relatively short in order to allow readers interested in computational vibroacoustics of complex structural systems to directly access computational methods that are adapted to the present and future evolutions of the general commercial softwares.

Finally, it should be noted that, obviously, a short book cannot include all the various methodologies in computational vibroacoustics of complex structural systems (but which have been mentioned and referenced throughout the text). More specifically, this short book proposes a unified strategy chosen by the authors and gives their view in this field in the context of the present and future generations of massively parallel computers. The methods proposed have effectively been validated and applied to complex structural systems and can partly be found in the references.

In the first section, the principal objectives of the book are presented. The next sections are devoted to a strategy for modeling complex vibroacoustic systems adapted to computational vibroacoustics. This strategy will be the guideline to be followed in subsequent chapters to present advanced computational vibroacoustics.

1.1 PRINCIPAL OBJECTIVES OF THE BOOK

The fundamental objective of this book is to present an advanced *computational method* for the prediction of the responses in the frequency domain of general *linear vibroacoustic systems*. The frequency domain is usually composed of three parts: the *low*-, the *medium*- and the *high*-frequency ranges which are defined in the next section. This book is devoted to computational vibroacoustics in the low- and medium-frequency ranges.

The system under consideration is constituted of a deformable *dissipative structure*, coupled with an *acoustic cavity*. This cavity is filled with a *dissipative acoustic fluid*, which is an inviscid acoustic fluid for which a damping model is introduced. In addition, *wall acoustic impedances* can be taken into account, which allow us to model the acoustic properties of the physical wall constituting a part of the geometrical interface between the internal acoustic fluid and the structure. The system is surrounded by an inviscid acoustic fluid occupying an infinite domain, called *external acoustic fluid*, which is coupled with the structure through the geometrical interface between the external acoustic fluid and the structure.

The vibroacoustic system is submitted to given *acoustic sources* (such as loudspeakers) in the cavity (internal acoustic sources) and in the external acoustic fluid domain (external acoustic sources), as well as given *mechanical forces* applied to the structure (such as surface and body forces).

The frequency responses of the vibroacoustic system are the *displacement field* in the structure, the *pressure field* in the acoustic cavity, and the *pressure fields* on the external fluid-structure interface and in the external acoustic fluid (near and far fields).

It is now well established that the predictions in the medium-frequency range must be improved by taking into account both the *system-parameter uncertainties* and the *model uncertainties* induced by *modeling errors*. Such aspects will be considered in Chapter 9, devoted to *uncertainty quantification* (UQ) in vibroacoustics (structural acoustics) and in fluid-structure interaction.

In this book, the presented formulations, which correspond to new extensions and complements with respect to the state-of-the-art, can be used for the development of a new generation of computational vibroacoustic softwares in particular, in the context of parallel computers. We present here an advanced computational formulation that is based on an efficient reduced-order model in the frequency range and for which all the required modeling aspects for the analysis of the low- and medium-frequency ranges have been taken into account. More precisely, we have introduced a frequency-dependent linear constitutive equation for modeling damping effects in complex structures, an appropriate dissipative model for the internal acoustic fluid including wall acoustic impedance, and, finally, a global model of uncertainty. It should be noted that model uncertainties must absolutely be taken into account in the computational models of complex vibroacoustic systems in order to improve the prediction of the responses in the low- and medium-frequency ranges.

The reduced-order computational model is constructed using the finite element method for the structure and for the internal acoustic fluid. The external acoustic fluid is treated using an appropriate boundary element method in the frequency domain.

Throughout the book, the finite element method (FEM) (see, for instance, Hughes, 2000; Zienkiewicz and Taylor, 2005) is used for the spatial discretization of the boundary value problems yielding the associated matrix equations. An alternative method to construct the matrix equations would consist in using the isogeometric analysis (see Hughes et al., 1996).

For some details of mathematical developments presented in this book, the reader is referred to Ohayon and Soize (1998), and for a brief general overview of the recent aspects, the reader is referred to Ohayon and Soize (2012).

In this book, we do not consider fluid flows (for fluid dynamics, see Batchelor 2000), such as the case of a structure in movements in an external fluid at rest (or a the case of a structure at rest in a flow) and the case of a structure with internal flows. For physical and modeling aspects related to flows in acoustics and fluid-structure interactions, we refer the reader to Howe (2008). For fluid-structure interaction with internal flow using nonlinear computational fluid dynamics, we refer the reader to Bazilevs et al. (2013).

Therefore, the considered modeling is carried out for an equilibrium state for which the acoustic fluid is at rest. Nevertheless, such a modeling can be used, for instance, for external flows around the structure or for internal flows inside pipes when the aeroelastic or the hydroelastic phenomena are decoupled from the vibroacoustic phenomena under consideration. Therefore, only the effects of the external flow in terms of the forces applied to the structure are considered; this is the case of the effects induced by a turbulent boundary layer on the structure.

1.2 DEFINITION OF THE DIFFERENT FREQUENCY RANGES: LF, MF, AND HF

The different types of vibration responses of a weakly dissipative complex structure lead us to define the frequency ranges of analysis. Let $u_j(\mathbf{x}, \omega)$ be the frequency response function (FRF) of a component j of the displacement $\mathbf{u}(\mathbf{x}, \omega)$, at a fixed point $\mathbf{x}$ of the structure and at a fixed circular frequency ω (in rad/s). Figure 1.1 represents the modulus $|u_j(\mathbf{x}, \omega)|$ of the FRF in log scale and Figure 1.2 represents the unwrapped phase $\varphi_j(\mathbf{x}, \omega)$ of the FRF such that $u_j(\mathbf{x}, \omega) = |u_j(\mathbf{x}, \omega)| \exp\{-i\varphi_j(\mathbf{x}, \omega)\}$. The unwrapped phase is defined as a continuous function of ω obtained in adding multiples of $\pm 2\pi$ for jumps of the phase angle. Three frequency ranges can then be characterized, as follows.

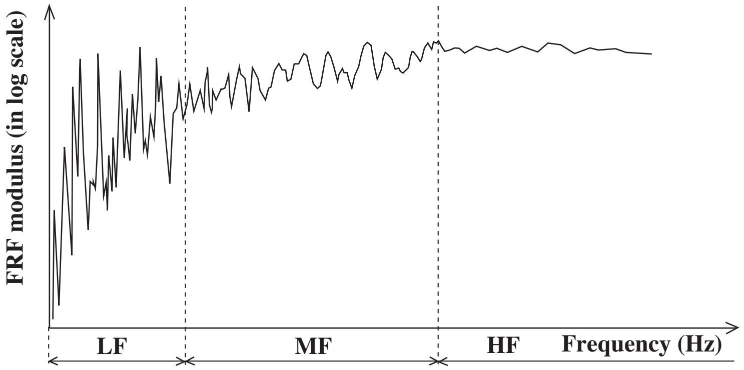

Figure 1.1 Modulus of the FRF as a function of the frequency. Definition of the LF, MF, and HF ranges.

(1) The *low-frequency range* (LF) is defined as the modal domain for which the associated conservative system exhibits isolated modes (low modal density). In this LF range the modulus of the FRF exhibits isolated resonances whose amplitudes are driven by the damping (see Figure 1.1) and the phase rotates of π at the crossing of each isolated resonance (see Figure 1.2).

(2) The *high-frequency range* (HF) is defined as the frequency band for which there is a high modal density that is approximatively constant

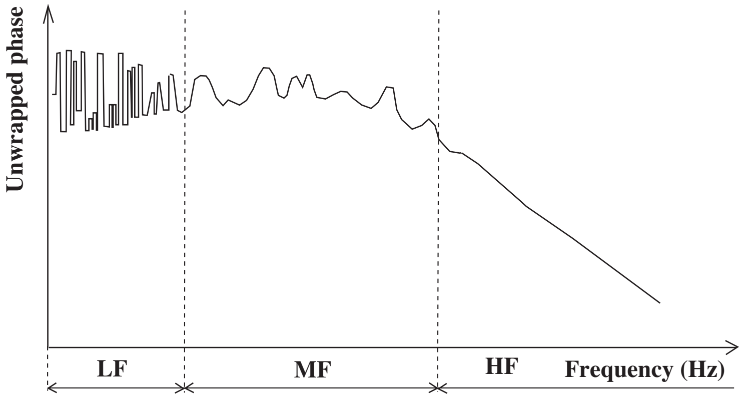

Figure 1.2 Unwrapped phase of the FRF as a function of the frequency. Definition of the LF, MF, and HF ranges.

as function of the frequency. Such an assumption is required to use asymptotic approaches and/or statistical descriptions (as carried out by the statistical energy analysis). In this HF range the modulus of the FRF varies slowly as the function of the frequency (see Figure 1.1) and the phase is approximatively linear (see Figure 1.2). It should be noted that this frequency range is not defined with respect to the absolute value of the frequency but is mainly related to the modal density of the system. This frequency range is outside the scope of this book.

(3) For a complex structure (complex geometry, heterogeneous materials, complex junctions, complex boundary conditions, several attached equipments, etc.), an intermediate frequency range called the *medium-frequency range* (MF) appears. This MF range does not exist for a simple structure (for example, a simply supported homogeneous straight beam). This MF range is defined as the intermediate frequency band for which the modal density exhibits large variations over the band. Contrary to the LF range, due to the effects of the damping, the frequency response functions do not exhibit isolated resonances (see Figure 1.1) and the phase slowly varies as a function of the frequency (see Figure 1.2).

This analysis presented for a structure can be extended for a complex vibroacoustic system. This book is devoted to LF and MF ranges analyzes for complex vibroacoustic systems.

1.3 STRATEGY FOR MODELING COMPLEX VIBROACOUSTIC SYSTEMS

Computational vibroacoustics is devoted to the computation of the responses of a vibroacoustic system (which can be a macrosystem or a microsystem). This vibroacoustic system is constituted of a deformable structure made up of metallic materials, heterogeneous composite materials, and, more generally, metamaterials. Let us also mention investigations of new types of materials, called metamaterials, for cloaking purposes connected to acoustic anechoicity (Milton et al., 2006; Fang et al., 2006; Chen and Chan, 2007; Pendry and Li, 2008; Cheng et al., 2008). It should be noted that the class of materials considered for the structure can be extended for the meta-smart-adaptive heterogeneous materials but are outside the scope of the book. The structure is coupled with an internal acoustic cavity and an external acoustic fluid. Acoustic coatings (soundproofing materials, sound-insulation layers, etc.), made up, for instance, with cellular materials such as porous materials, can be taken into account on the fluid-structure interfaces.

The responses of the vibroacoustic system are mainly driven by the structure and possibly by the internal acoustic cavity which are resonant systems, but not by the infinite external acoustic fluid which is not a resonant system.

(i) In the first step, let us precise the objectives of the vibroacoustic modeling, which consist of computing the responses of the vibroacoustic system submitted to prescribed excitations.
- The excitations are forces applied to the structure (forces generated by the environment and transmitted to the structure by solid paths such as shocks or vibration equipments or through fluid paths such as turbulent boundary layer effects), internal acoustic sources inside the acoustic cavity and external acoustic sources.
- The responses are the structural displacement field, the pressure field in the internal acoustic cavity for internal acoustic noise quantification, and the near and the far pressure fields in the external acoustic fluid. Those pressure fields can be decomposed as the sum of
  - an incident field due to external acoustic sources,
  - a scattering of the incident acoustic field by the rigid structure,

- a radiation acoustic field due to the deformation of the structure.

The so-called backscattered acoustic field is the sum of the scattering and radiation fields.

- The formulation presented in the book also includes the following cases:
  - the anechoicity analysis, which consists in studying the backscattered acoustic field of an incident acoustic field by the structure;
  - the acoustic stealth of the structure, which consists in analyzing the transmission into the external acoustic fluid, through the structure, of internal acoustic sources located inside the acoustic cavity or of mechanical forces applied to the structure.

(ii) In the second step, let us specify that the modeling strategy is based on a formulation in the frequency domain. In effect, there are many advantages to using a frequency domain formulation instead of a time domain formulation for linear vibroacoustic systems. The first one is the possibility to analyze the vibroacoustic system in terms of the frequency ranges, that is, to consider the low-, the medium-, and the high-frequency ranges. The second one is the capability to choose the vibroacoustic system modeling accordingly to frequency range. In particular, the damping models and the quantification of uncertainties must be taken into account in the medium-frequency range. Finally, the formulation in the frequency domain can easily be implemented in massively parallel computers because the calculations can be distributed frequency by frequency.

It should be noted that the low-frequency range is mainly driven by the resonances induced by the elastic modes of the structure, by the acoustic modes of the acoustic cavity, and by the elastoacoustic modes of the vibroacoustic system. Concerning the appropriate formulations for computing the elastoacoustic modes of the associated conservative vibroacoustic system, including substructuring techniques, and for the construction of the corresponding reduced-order model, we refer the reader to Morand and Ohayon (1995); Ohayon et al. (1997); Ohayon and Soize (1998); and Ohayon (2004a,b) (including the references).

(iii) In the third step, the strategy concerning the damping modeling is presented for the vibroacoustic system, as follows.

- *External acoustic fluid*. The viscosity of the fluid is negligible in the considered low- and medium-frequency ranges for usual acoustic fluids such as air or water. Therefore the acoustic fluid is inviscid and there is no damping inside the fluid. Nevertheless, since the external acoustic fluid occupies an infinite domain, the outward Sommerfeld radiation condition at infinity implies that the energy traveling towards infinity does not come back and consequently, corresponds to an energy loss. Due to the coupling between the structure and the external acoustic fluid, this phenomenon yields an apparent damping for the structure because the mechanical energy transmitted by the structure to the external acoustic fluid is lost for the structure.
- *Internal acoustic fluid*. The acoustic fluid in the cavity is dissipative and a damping model must be introduced. Generally, there are two main physical dissipations. The first one is an internal acoustic dissipation inside the cavity due to the viscosity and the thermal conduction of the fluid. The second one is the dissipation generated inside the wall viscothermal boundary layer of the cavity. In general the first one is negligible with respect to the second one. However, we propose to simply take into account the two sources of damping by introducing an equivalent damping term corresponding to an internal acoustic dissipation inside the volume of the cavity.
- *Structure*. For viscoelastic materials, a frequency dependent linear constitutive equation is introduced using the linear theory