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Active sound quality control of engine induced cavity noise

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ABSTRACT

Active control solutions appear to be a feasible approach to cope with the steadily increasing requirements for noise reduction in the transportation industry. Active controllers tend to be designed with a target on the sound pressure level reduction. However, the perceived control efficiency for the occupants can be more accurately assessed if psychoacoustic metrics can be taken into account. Therefore, this paper aims to evaluate, numerically and experimentally, the effect of a feedback controller on the sound quality of a vehicle mockup excited with engine noise. The proposed simulation scheme is described and experimentally validated. The engine excitation is provided by a sound quality equivalent engine simulator, running on a real-time platform that delivers harmonic excitation in function of the driving condition. The controller performance is evaluated in terms of specific loudness and roughness. It is shown that the use of a quite simple control strategy, such as a velocity feedback, can result in satisfactory loudness reduction with slightly spread roughness, improving the overall perception of the engine sound.

1. Introduction

The successful development of new products relies on the capability of assessing the performance of conceptual design alternatives in an early design phase. In recent years, major progress was made hereto, based on the extensive use of virtual prototyping, particularly in the automotive industry. The state-of-the-art in CAE modeling techniques which can be used for the analysis of time-harmonic acoustic problems is presented in [1]. An overview is given, with automotive interior noise applications, on recently investigated extensions and enhancements to enlarge the application range of different techniques. The efficiency of present CAE techniques allows the use of optimization, e.g., in improving the NVH characteristics of a full-scale engine [2] or a vehicle body [3]. The novelty on this framework is to account for the human perception when defining product performance criteria as in [4,5].

Additionally, active control has shown the potential to enhance system dynamic performance which allows lighter and improved products. Research done in the previous years on smart materials and control concepts has led to practical applications with promising results for the automotive industry [6]. However, to make the step to the design of *active sound quality control* (ASQC), the control schemes, along with appropriate simulation procedures, need to become an integral part of the product development process [7]. In other words, this requires: (i) the product performance metrics to be based on

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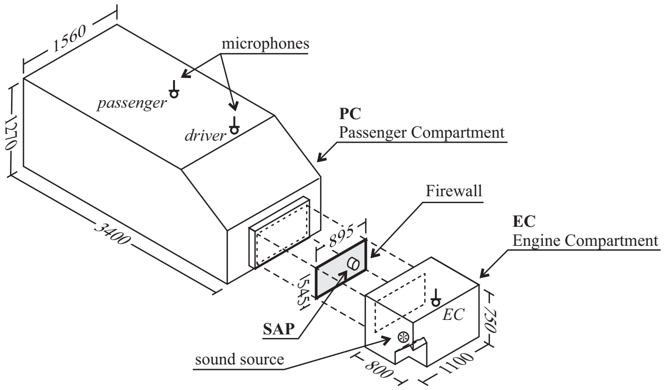

Fig. 1. Schematic view of the vehicle mock-up (dimensions in mm).

human perception attributes and (ii) the simulation models to support the specific aspects related to smart structures (active systems, actuators, sensors and control logic).

In order to demonstrate the proposed simulation procedure and evaluate the effect of active control on the perceived sound quality (SQ), a vibro-acoustic cabin mock-up is selected (Fig. 1). It consists of a simplified car cavity with rigid acoustic boundary condition. The passenger compartment (PC) and the engine compartment (EC) are connected through a flexible firewall which allows noise generated in the EC to be transmitted to the PC. A sound source placed in the EC works as a primary disturbance source. The primary source is driven by a real-time engine simulator, capable of delivering a harmonic excitation based on the engine orders' amplitude and phase [8]. The controller is based on a collocated structural sensor/actuator pair (SAP) connected to the firewall in a time-invariant velocity feedback loop.

The simulation procedure and experimental validation of the active structural acoustic control (ASAC) system are presented in Section 2. The SQ metrics and algorithms used in this paper are reviewed in Section 3. The results, concerning roughness, specific loudness and Zwicker loudness are treated in Section 4. Finally, some general conclusions are addressed in Section 5.

2. ASAC simulation scheme and experimental validation

The active control scheme adopted is an ASAC scheme involving a collocated velocity feedback controller with structural sensors and actuators. This choice is made given the advantages of such a scheme with respect to implementation, stability and reliability to system modification (or uncertainties) [9–11]. It is important however to mention that different control strategies can be adopted under the same simulation framework, which, in the end, reveals the functionality of such an approach to the assessment of conceptual design performance [7,12,13].

2.1. From vibro-acoustic finite element to state-space model

In order to bring active solutions to the level of industrial applications, the designer needs tools that allow the inclusion of sensors/actuators and control strategy in the product development phase. Eventually, one should be able to quantify the improvement in some performance design criteria (in this case SQ metrics) after adopting a certain control strategy.

One of the challenges resides in deriving reasonably sized models that integrate the structural and acoustic components along with the control algorithm. In order to fulfill this requirement, a fully coupled finite element (FE) model of the vibro-acoustic system is written as a modal state-space (SS) model. As a result of using the coupled vibro-acoustic modal base, any combination of structural and acoustic inputs/outputs can be used for the control design, e.g., an acoustic source in the EC, structural sensors and actuators on the firewall and microphones on the PC.

One of the possible coupled FE/FE formulations is the Eulerian, in which the structural degrees of freedom (DoFs) are displacement vectors, while the acoustic DoFs are expressed as scalar functions. The latter is usually the acoustic pressure [14–16]. If pressure is adopted, the system of equations yields non-symmetrical mass and stiffness matrices. The modal base resulting from such non-symmetric eigenproblem presents distinct left and right eigenvectors, denoted here, respectively, by $\mathbf{\Phi}_L$ and $\mathbf{\Phi}_R \in \mathbb{R}^{n\times 1}$ with n being the total number of structural and acoustic DoFs.

The adopted FE tool provides $\mathbf{\Phi}_R$ [17]. However, it has been indicated [18] that, particularly for the displacement/pressure Eulerian formulation, the left and right eigenvectors can be related as

$$\{\mathbf{\Phi}_L\}_r = \begin{Bmatrix} \{\mathbf{\Phi}_{Ls}\}_r \\ \{\mathbf{\Phi}_{La}\}_r \end{Bmatrix} = \begin{Bmatrix} \{\mathbf{\Phi}_{Rs}\}_r\omega_r^2 \\ \{\mathbf{\Phi}_{Ra}\}_r \end{Bmatrix}, \quad r = 1, 2, \ldots, N \tag{1}$$