Statistical Energy Acoustic for high Frequency Analysis

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1. Introduction

For high frequency analysis, Statistical energy analysis (SEA) has proved to be a promising approach to the calculation of sound transmission in complex structures. In automotive industry and also in civil engineering, most of noise transmission is due to high-frequency structural vibrations, where the characteristic wavelength is small compared to the dimensions of the structure. For these applications classical methods of structural analysis, such as the finite element method (FEM), and Boundary Elements Method (BEM), cannot be used due to the large number of degrees of freedom required to model structural deformation. Statistical Energy Analysis (SEA) considers the vibrations of the structure in terms of elastic waves which propagate through the structure and are partially reflected and partially transmitted at structural connections. For the last few year there has been an increase in the application of SEA techniques to study noise transmission in motor vehicles.
In this paper simple examples are presented with SEA method, including a simplified vehicle model. The vehicle body is modeled as a collection of connected flat plates and cavities for predictive purposes. Each plate supports bending waves, longitudinal and transverse shear in-plane waves and assumed damping effects. The effects of curved plates and the transmission of waves through curves plates and beams are not included in the analysis. Numerical results in term of acoustic pressure inside the cavity, are in good agreement compared to analysis using other software published in the literature.

***KEYWORDS** : SEA, Statistical Acoustic, High frequency vibration

2. Theory of SEA

Statistical energy analysis is used to predict sound and vibration transmission of elastic waves which propagate through the structure that are partially reflected and partially transmitted into connected structure though structure discontinuity or line connection. In SEA method systems are considered to be divided into subsystems that are linearly coupled and exchange energy through resonant vibration modes. Subsystems can be plates, beams or acoustic rooms that have modes which are similar in nature, and where the primary variable of interest is energy. The theory of structural vibration transmission has been developed over many years, and detailed theory of SEA is described by Lyons in [1]. The earliest calculations were carried out by Langley et al. [2] and [3]. In SEA computation the system can be modeled by the intersection of a number of semi-inýnite plates which meet at a single line junction where there must be continuity of displacement, slope and equilibrium of moments and forces. Structural vibration transmission calculations are carried out as part of the calculation of sound and vibration transmission through structures. If two or more plates meet together to form a line junction then the properties of the junction can be given in terms of the transmission coefficient, defined as the ratio of the power transmitted across the junction to the power incident. The power may be transmitted by a transmission wave type that is different from the incident wave in which case there will be energy conversion from one wave type to another.
Such applications are be found in automotive industry as well as civil, naval and aerospace engineering for noise transmission in buildings, ships, aircraft and other structures as described in [4], and [5]. In this paper, the plates are assumed to be isotropic, thin and þat and to meet at a continuous line junction.

3. Numerical Tests.

For comparison to examples that been modeled using other software [5], in this paper we consider the following 3 different cases that have been modeled by FREE SEA code [5], and the results will be compared to the ones published by the author.

3.1 3-Plates connection

This example modelled in [5] consists of three plates which are connected at a common line, figure 1.

The 3 plates are all made linear elastic material with the following properties:

density = 7800 kg/m^3, Young Modulus = 210.10^9 Pas, Poisson Ration = 0.3

Each of the plates has a size of 1x1 m. Plate 1 and Plate 3 are 2 mm thick, Plate 2 is 3 mm.

Plate 1 is connected to plate 2 at an angle of 90°, plate 3 is connected at an angle of 210° .

All Plates have constant damping of 0.01.

Plate 1 is exited by a an input power of 1 Watt (in bending wave).

To goal is to calculate the mean velocity amplitude of the bending waves at all three plates, figure 2.

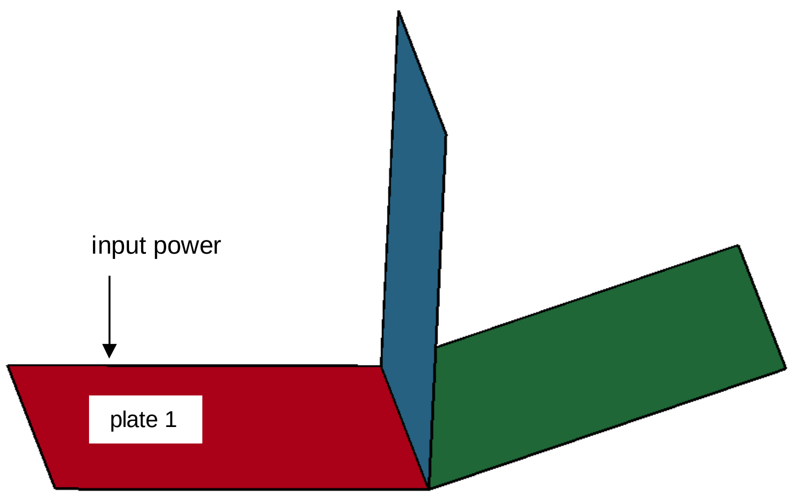
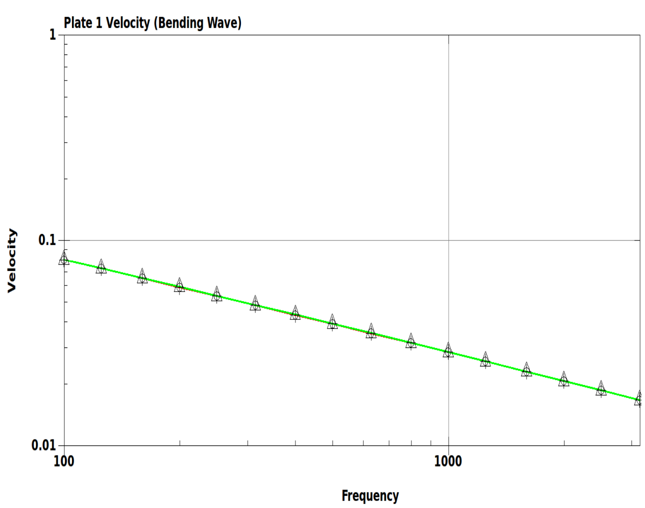


Figure 1. Description of the 3 plates model



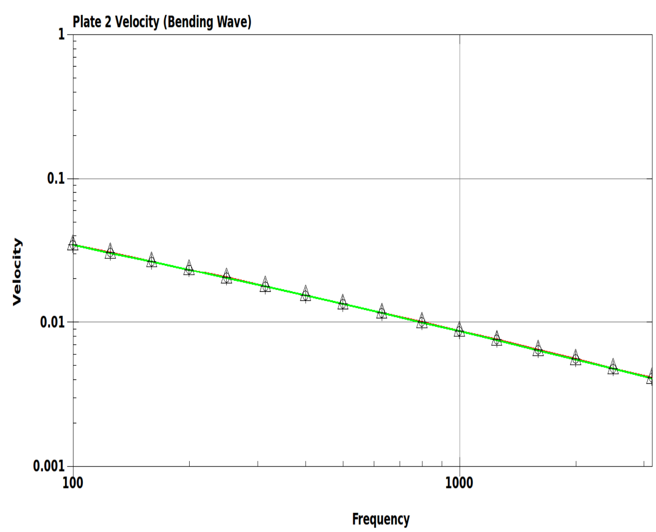


Figure 2. Comparison for Mean Velocity (Log-Log) on plate 1 and plate 2

3.2 Sound Transmission between 2 rooms

This example consists of 35 subsystems, 32 plates (walls) and 2 acoustic rooms figure 3 .

The 32 plates are all made concrete material modelled as linear elastic with thickness 0.19 m, area 13.95 m, and the following material properties:

density = 2300 kg/m^3 , Young Modulus = 2.8 10^{10} Pas, Poisson Ration = 0.3

The 2 rooms are modelled with ambient air material

An acoustic source of Power input 1 Watt into room 1 is assumed.

To goal is to calculate the acoustic sound pressure in both rooms, figure 4

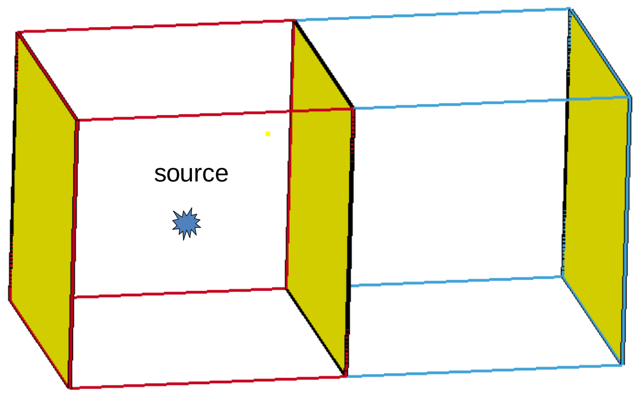


Figure 3. Problem Description for room 1 and 2

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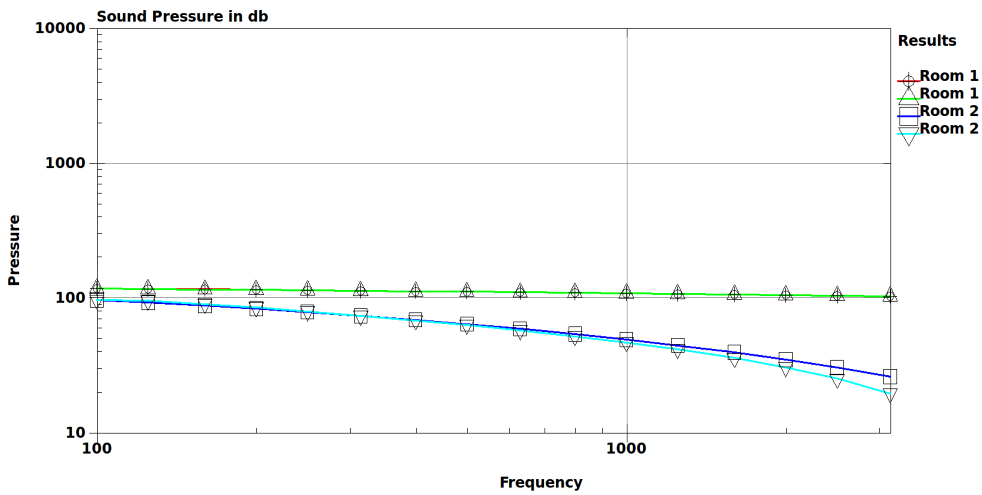

Figure 4. Comparison for acoustic pressure (Log-Log_) in room 1 and room 2

3.3 Sound Transmission inside a car

The car example consists of 26 subsystems, 18 Steel plates, 6 glass plates and 2 acoustic rooms, figure 5. The steel plates and glasses are modelled as linear elastic, and the following material properties:

density = 7800 kg/m^3 , Young Modulus = 2.1. 10^{11} Pas, Poisson Ration = 0.3

density = 2500 kg/m^3 , Young Modulus = 6.0 10^{10} Pas, Poisson Ration = 0.2

A real car would have some sort of a frame, pillars and stiffeners. However with only the information in build a simple SEA model

Input power applied in bending on the fender left and right front parts of the car as described in figure ?.

Sound Pressure in dB units in the passenger compartment. figire 6

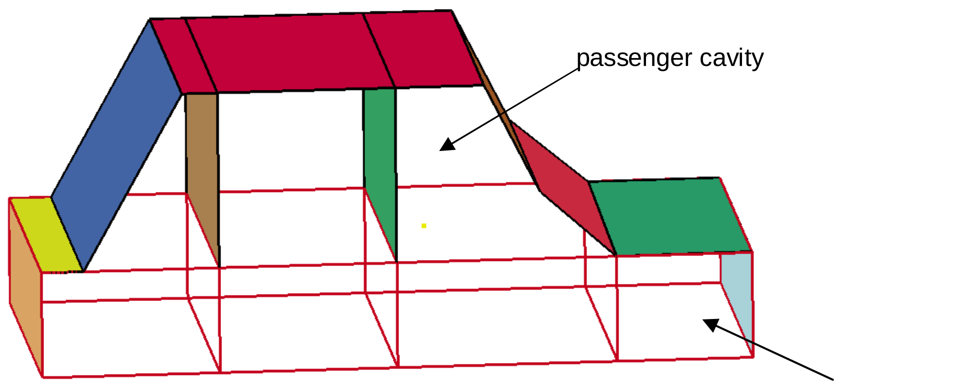

Figure 5. Problem Description of the subsystems

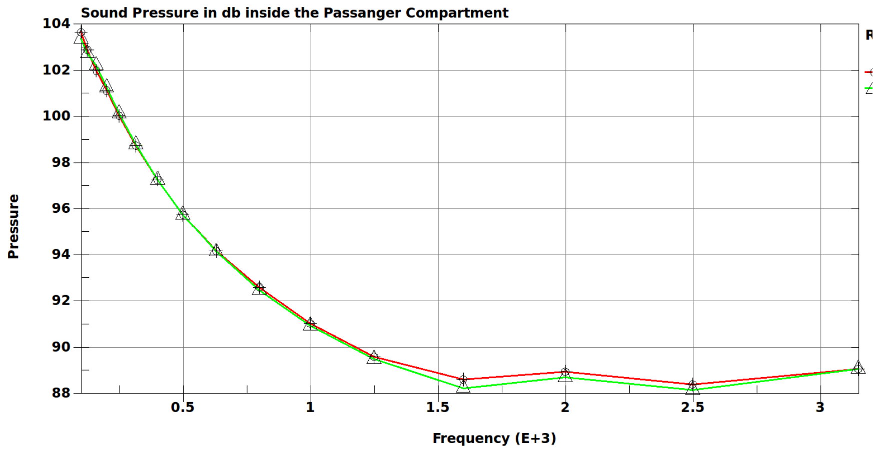

Figure 6. Comparison of acoustic pressure in passenger cavity

1 Literature

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