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IMPROVING S.E.A. PREDICTION OF STRUCTURE-BORNE NOISE BY DESCRIBING JUNCTIONS DETAILS WITH F.E.

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Abstract

The use of Statistical Energy Analysis to analyse the structure-borne dynamics of complex engineering systems is sometimes limited because of the difficulty to predict the coupling loss factors of the complex junctions encountered between structural subsystems. This paper demonstrates how a finite elements model can be used to capture the complexity of the junctions, and can be seamlessly incorporated into a standard SEA model using a Hybrid FE-SEA approach. The modelling process is illustrated on a simplified mock-up of an optical space platform, and the improved SEA prediction is compared to test results.

INTRODUCTION

Statistical Energy Analysis (SEA) is a well-proven technique for predicting the vibro-acoustic response of complex engineering systems subjected to high-frequency broadband excitation [1]. SEA is used routinely in many different industries to create models of air-borne and structure-borne transmission. While SEA has been successfully applied to many different structure-borne noise problems, problems are sometimes encountered that contain complex junctions that can be difficult to model using the library of "coupling loss factors" found in a traditional SEA code [2]. This paper discusses how existing SEA models can be improved to accurately model such transmission paths using a fully coupled Hybrid FE-SEA approach.

The Hybrid FE-SEA method rigorously couples SEA and FE descriptions [3,4] and therefore provides a way to introduce as much details as desired into a SEA model. The particular application in this paper uses FE to provide a description of local junction details in an SEA model, and it is shown how the transmission at the junction between SEA subsystems can be estimated however complicated the junction is.

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STRUCTURE, TEST DATA, SEA MODEL

The structure used in the study is a simplified mock-up of an optical space platform and consists of a cylinder with three connected plates. It was initially built and tested to experimentally validate a variance theory for SEA, and more details on the structure and experimental setup can be found in [5].

Cylinder-plate Structure

The structure shown in figure 1 comprises three plates connected to a circular bracket attached at one end of a cylinder. The plates are identical in terms of material, thickness, damping treatment and area, whereas the exact geometries differ. All components are of steel, with the properties given in Table 1. The loss factors were measured by the power injection method [1] and they were found to be approximately independent of frequency. Some details of the bracket connections and cylinder ends are also shown in figure 1. It can be seen that the brackets are welded to the cylinder, while two bolts are used to connect them to each plate.

An ensemble of random systems was generated by attaching masses at random locations on each substructure. Four masses were attached to each plate and ten masses to the cylinder, and the total amount of mass used is shown in Table 1.

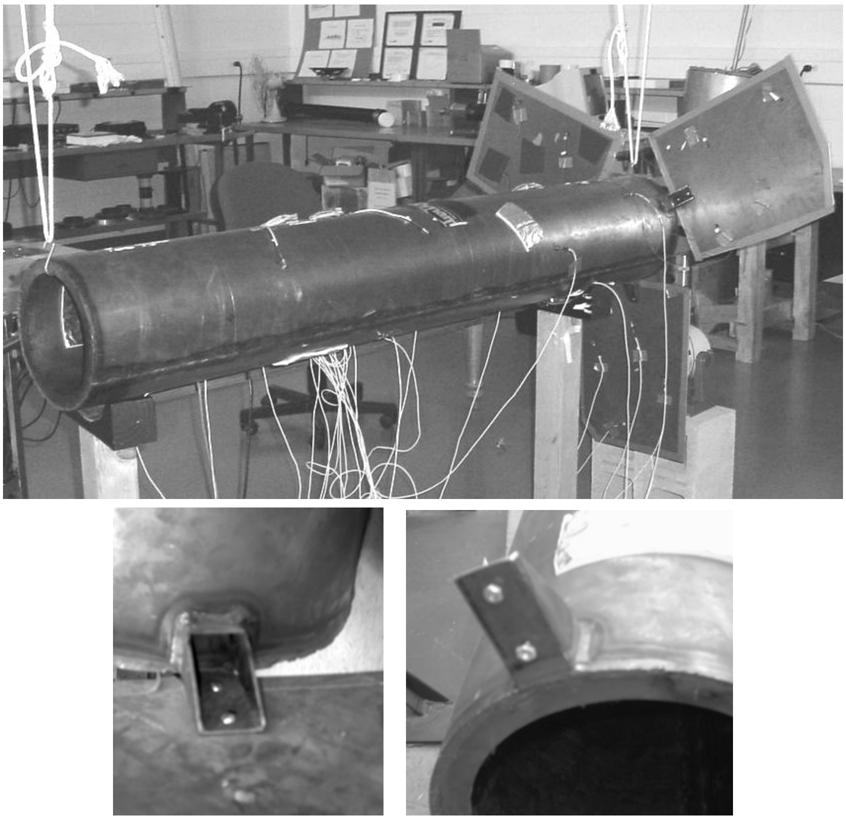

Figure 1 – Cylinder-plate structure: the top picture shows the experimental setup with the lower plate driven by a shaker (point force); the bottom pictures show details of the bracket junction and cylinder end.

Table 1: Properties of the cylinder and attached plates.

	Material (steel)	Dimensions	Damping factor η	Mass	Total added masses
Plates	E=2.1 10^{11} N m^{-2}, ν=0.3, ρ=2800 kg m^{-1}	a~0.4m, b~0.5m, h=1mm	1.5 %	1.5 kg	0.34 kg (22%)
Cylinder		r=0.14m, l=1.83m, h=1mm	0.6 %	13.6 kg	3.6 kg (26%)

Using asymptotic formula, the estimated number of modes below 500 Hz is 160 (based on the number of modes, it might be expected that a narrowband FE prediction of the response is unlikely to be accurate across the entire frequency range). The mode count increases to 885 below 2000 Hz, and 3380 below 5000 Hz.

Experimental Data

The assembled structure was suspended at two points as shown in figure 1. The experimental set-up consisted of one shaker, one impedance head at the excitation point, ten accelerometers scattered on the cylinder, and four accelerometers scattered on each plate. A white noise signal was applied to the shaker, which was attached to one of the plates. Valid data was obtained up to 5000 Hz (limitations arise from the signal to noise ratio). The measurements were repeated for an ensemble of 25 different locations of the point masses. The experimental ensemble-averaged energy frequency responses of the cylinder and three plates are shown in figure 2.

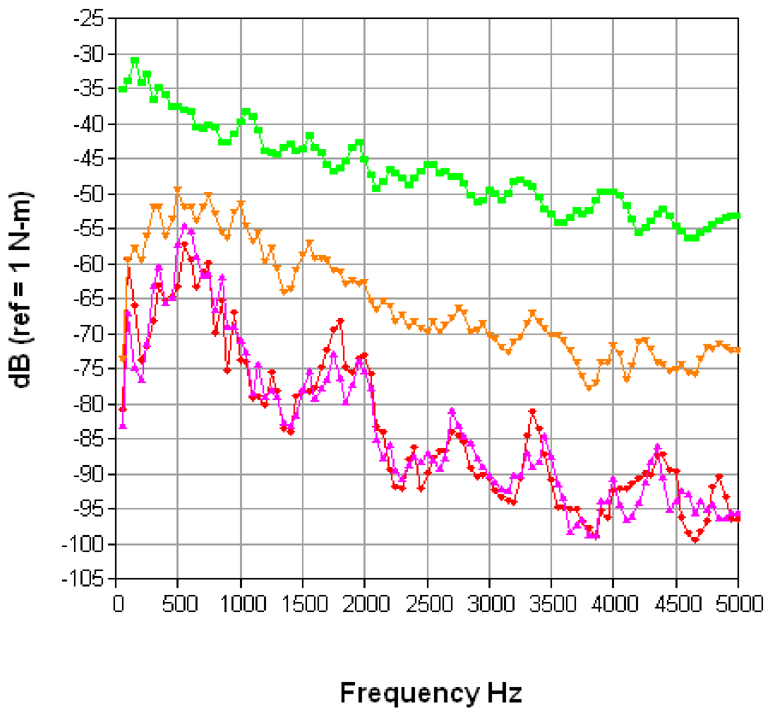

Figure 2 – Experimentally measured ensemble-averaged energy of the cylinder (orange) and three plates, due to a point force applied on plate 1 (green curve).

It can be seen that the ensemble-averaged response of the driven plate is a fairly smooth function of frequency (as expected from an ensemble-averaged response at high frequency). On the other hand, the ensemble-averaged responses of the two transmitted plates show some distinct peaks at 550, 1750, 2700, 3400 and 4400 Hz. Some of the peaks can also be seen on the response of the cylinder, although less obvious.

SEA Models

An initial SEA model of the system was built with the AutoSEA2 software [6] by describing the connections between the plates and the cylinder as point junctions, since for much of the frequency range the bending wavelength in the cylinder and plates is much greater than the bracket dimensions (0.04 m). The mean external input power used in the SEA subsystem was based on the relevant theoretical result for an infinite plate, with a mass correction factor accounting for the presence of the impedance head: $\mathrm{E}[P_{in}] = P_{\mathrm{in}}^{\infty} / |1 + i2\omega M_a P_{\mathrm{in}}^{\infty}|^2$, where M_a is the mass of the device between the force sensor and the subsystem (estimated at 4.5 g), and P_{in}^{∞} is the infinite plate power input per unit force. The mass effect of the impedance head was found to be significant (more than 10 dB at 5000 Hz).

A second SEA model was built in which an artificial direct point junction was added between the three panels (although they are connected physically only through the cylinder). This apparently add-hoc additional junction tries to account for the stiffening ring at the end of the cylinder (see figure 1). It is indeed expected that the ring produces a direct transmission between the plates, short-circuiting the resonant modes of the cylinder. Both SEA models are shown in figure 3 below.

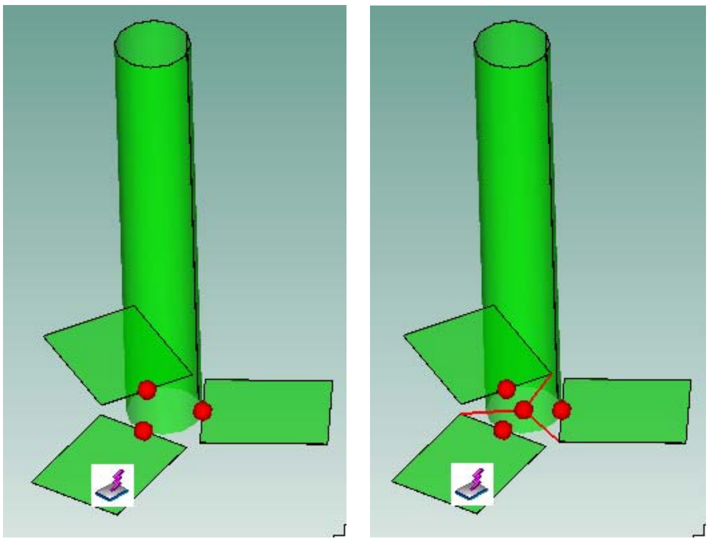

Figure 3 – Two SEA models of the cylinder-plate structure.

The predictions obtained with both models are shown in figure 4. Both models yield the same energy for the driven plate and cylinder, but predictions differ for the transmitted plates. The mean energies are reasonably well predicted by the second SEA model, while the first one underestimates the transmitted panels mean energies. Due to symmetry, plates 2 and 3 should have the same mean response, and this is found to be the case in both the experimental and theoretical results.

Although the second SEA model yields satisfactory predictions, it can be seen that it is not able to capture the peaks in the transmission between the plates. Moreover, the additional point junction provides a convenient way of accounting for the direct transmission between plates, but it is questionable how rigorous its use is. These two limitations of the SEA modeling motivated the development of a model where the junction details are described with FE.

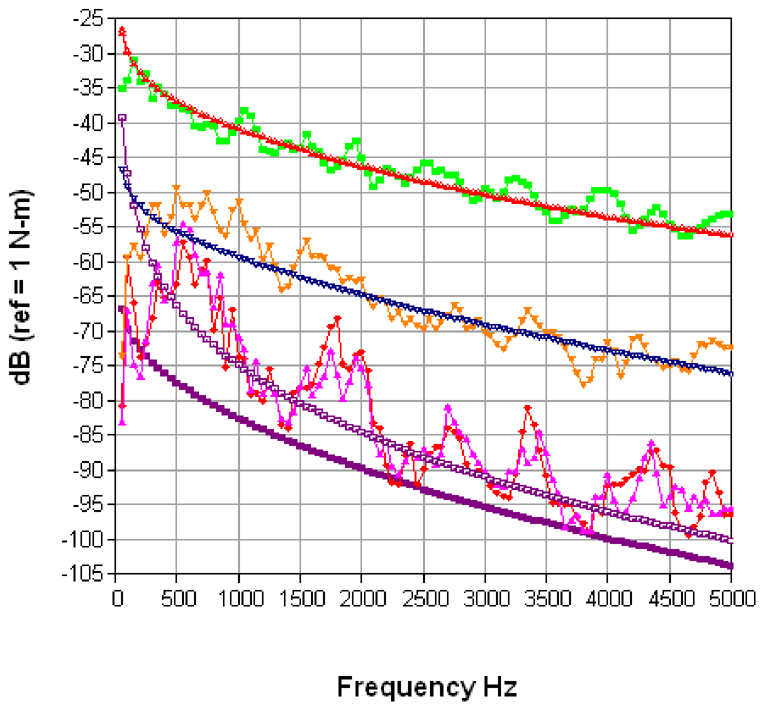

Figure 4 – Measured and predicted ensemble-averaged energy frequency response of the driven plate 1 (top curves), the cylinder (middle curves), and plates 2-3 (bottom curves). The two SEA predictions only differ for plates 2-3.

HYBRID FE-SEA MODEL

A Hybrid FE-SEA model of the structure was built with the VA One commercial software [7] in order to better describe the structure-borne transmission and more rigorously capture i) the impact of the stiffening ring at the end of the cylinder, ii) the peaks of the energy response of the transmitted panels.

An FE model of the bottom part of the cylinder, the stiffening ring, the brackets and the bolts was thus created as shown in figure 5. The model is composed of CTRIA3, CQUAD4 and CBAR elements (using the inbuilt Nastran solver within VA One), and there are 1354 nodes. The details of the bracket-panel connection are shown in figure 5: the bracket (orange) and panels (green) are described by different sets of finite elements; Each bolt (blue) is modeled as a single CBAR finite element, whose properties were computed for a plain circular section with the diameter of the physical bolt; the additional stiffness due to the non-perfect contact of the bracket and panels is modeled by a set of soft and massless CBAR elements. The four Hybrid junctions between the finite elements and the three SEA panels and the SEA cylinder are shown on the figure as bold blue lines.

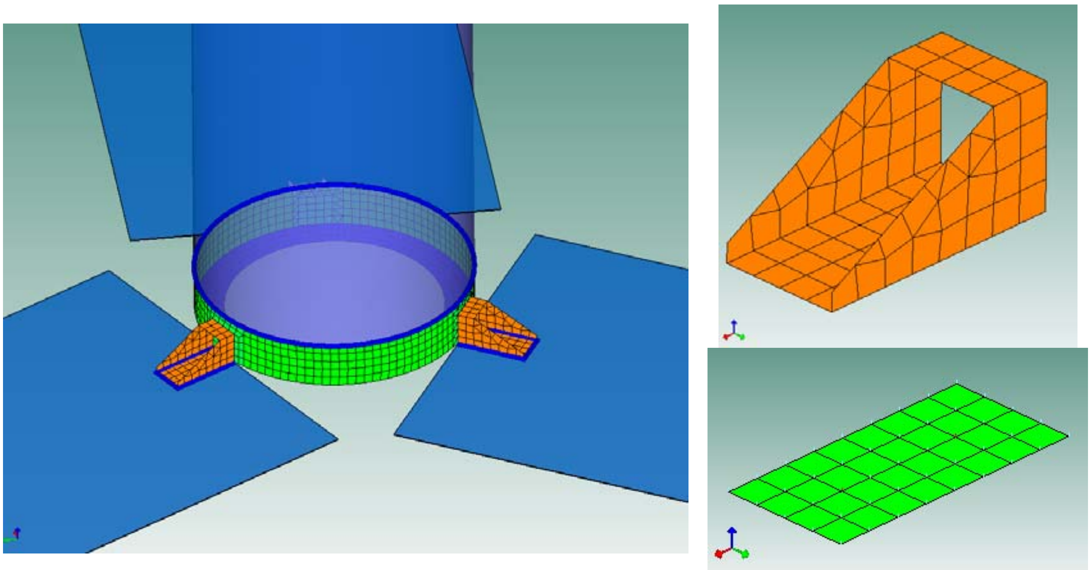

Figure 5 – Details of the Hybrid FE-SEA model (the blue subsystems are modeled with SEA while the meshed subsystems are modeled with FE; the thick blue lines represent Hybrid junctions between the FE and SEA subsystems). The FE model of the bracket-panel connections is also shown.

140 modes of the FE junction were extracted below 5300 Hz (as compared to more than 3380 for the complete structure). Examples of the modes are shown in figure 6. In the Hybrid FE-SEA model, the coupling loss factors between SEA subsystems are computed by fully accounting for the dynamics of the FE components (including damping), together with the reactive and resistive effects of the SEA subsystems impedances along the Hybrid junctions. The predicted coupling loss factors between the flexural wavefields of the driven plate and of the cylinder and plate 3 are shown in figure 7. The energy frequency responses (ensemble-averaged) predicted by the Hybrid model are shown in figure 8.

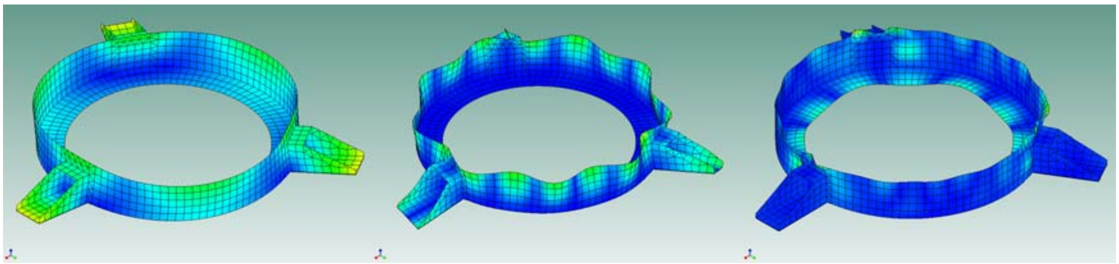

Figure 6 – Three modes of the connection described by FE.

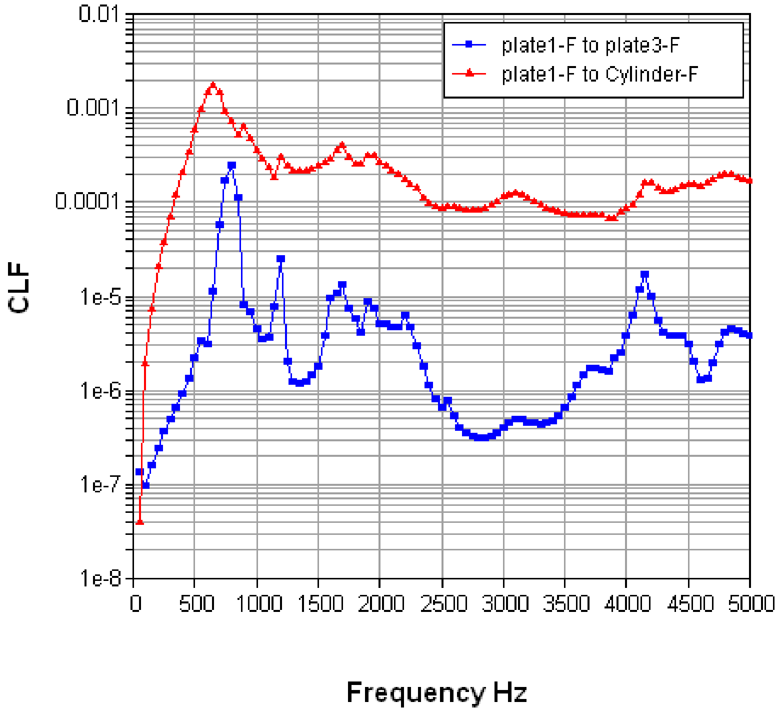

Figure 7 – Predicted coupling loss factors between the flexural wavefields of the driven plate 1 and the cylinder and plate 3.

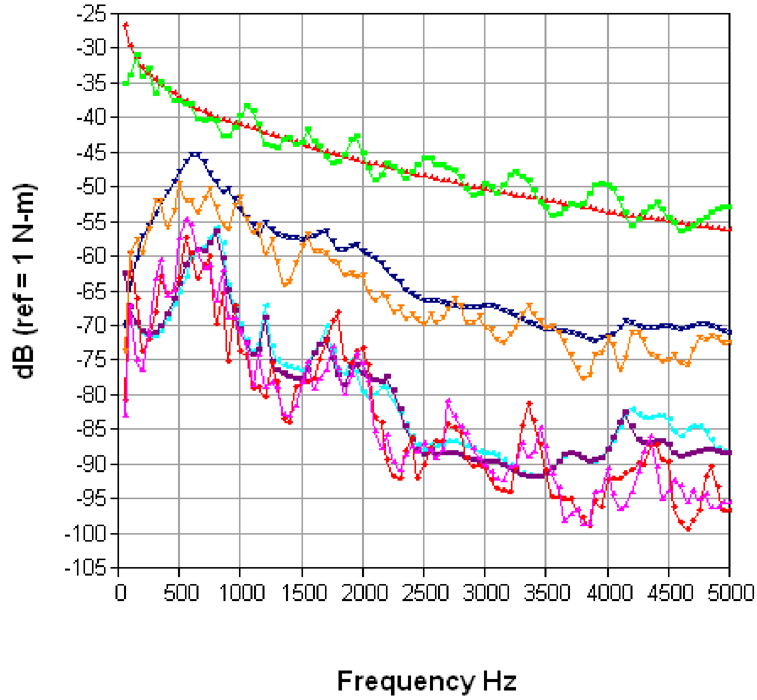

Figure 8 – Measured and predicted ensemble-averaged energy frequency response of the driven plate 1 (top curves), the cylinder (middle curves), and plates 2-3 (bottom curves).

It can be seen that the use of finite elements to describe the subsystems connection improves the prediction of the transmission and allows to capture some of the peaks in the transmission. It is straightforward to correlate the peaks with the “modes” of the junction. As an example, the transmission peak at 1200 Hz appears to be due to the second mode shown in figure 6. The reason this mode gives rise to high transmission is that the displacement along both the panels junctions and cylinder junction are significant.

Some of the peaks are not captured by the model (at 2800, 3400 Hz for example) and it is believed that those are due to longitudinal modes and global bending modes of the cylinder. Indeed, those global modes have a long wavelength (the third rigid-section bending modes occur at 2400 Hz, while the third longitudinal mode occurs at 3270 Hz), and are thus expected to be fairly robust to the randomization of the structure. Their response should consequently be fairly distinct since the ensemble-averaging process does not affect them.

CONCLUSIONS

It is demonstrated in this paper how a Hybrid FE-SEA method can be used to introduce detailed junction models into an existing SEA model. The technique was demonstrated on a simplified mock-up of an optical space platform where the structural junctions are complex and difficult to describe using standard analytical SEA coupling loss factors. The method provides a rigorous and generic prediction of the coupling loss factors between SEA subsystems, and also provides insights into the physics of the structure-borne transmission.

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