DEVELOPMENT OF FEM/BEM AND SEA MODELS FROM EXPERIMENTAL RESULTS FOR STRUCTURAL ELEMENTS WITH ATTACHED EQUIPMENT

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ABSTRACT

This work focuses on the analysis of a structural element of MetOP-A satellite. Given the special interest in the influence of equipment installed on structural elements, the paper studies one of the lateral faces on which the Advanced SCATterometer (ASCAT) is installed. The work is oriented towards the modal characterization of the specimen, describing the experimental set-up and the application of results to the development of a Finite Element Method (FEM) model to study the vibro-acoustic response. For the high frequency range, characterized by a high modal density, a Statistical Energy Analysis (SEA) model is considered, and the FEM model is used when modal density is low. The methodology for developing the SEA model and a compound FEM and Boundary Element Method (BEM) model to provide continuity in the medium frequency range is presented, as well as the necessary updating, characterization and coupling between models required to achieve numerical models that match experimental results.

1. INTRODUCTION

The vibro-acoustic response of spacecraft structures and payloads is a key element when studying the behaviour of such systems during the launch phase, mainly due to the noise loads generated at the boundary layer and the rocket engines.

In this regard, one issue which proves to be of special interest is the study of the influence of equipment installed on structural elements on the latter. One example of this is the Advanced SCATterometer antenna (ASCAT), which is installed on the MetOP-A satellite (Fig. 1) by means of a deployment mechanism directly attached to one of the lateral sides of the satellite.

Analysing the aforementioned problems can be approached studying the different elements of the system under acoustic loads, i.e. characterizing the structure and the fluid domain in terms of their modal response and their response to random acoustic loads.

Since acoustic loads extend over a broad range of frequencies, depending on the modal density of the elements different methodologies should be used when modelling the system, if it is desired to study the problem in the whole frequency range.

Figure 1. MetOP-A satellite (Credits: ESA-Silicon World)

At low frequencies, the noise and vibration response can be predicted accurately by using the Finite Element Method (FEM) for the structure, while the fluid domain may be modelled with either FEM or the Boundary Element Method (BEM). The choice between these two methods for modelling the fluid domain – finite or semi-infinite – depends on its geometry and size.

However, at high frequencies these methods would involve high computational efforts due to the excessive number of degrees of freedom that would be required to capture the short wavelength deformation, and for this reason the Statistical Energy Analysis (SEA), which has a simpler mathematical formulation based on system energies rather than displacements and velocities, is preferred [1, 2].

Since there is no absolute frequency range associated with each of these methods, as much depends on the structure itself, a unique method cannot be used to model all the elements of the system. This leads to the need of using hybrid methods at medium frequencies, where both FEM and SEA methods are used in the same model to define the different parts of the system [3]. The criterion for choosing the adequate method for each element has typically been its modal density, using FEM when it is low and SEA when it is high.

Proc. '12th European Conference on Space Structures, Materials & Environmental Testing'
Noordwijk, The Netherlands, 20–23 March 2012 (ESA SP-691, July 2012)

This paper is organized as follows. In Section 2 the specimen under study is described. In Section 3 information about the modal test performed to characterise the specimen is given. Section 4 presents the three numerical models proposed in this paper – FEM-BEM model, Hybrid model and SEA model – along with a preliminary SEA model with which analyse the modal density of the different elements and determine the frequency application range of each analysis method. In Section 5 response continuity over the frequency range of interest resulting from the proposed methods is presented, as well as correlation between experimental and numerical results for the low frequency range. Information about the acoustic test performed to obtain the aforementioned experimental results is also provided in this section. Finally, Section 6 presents a set of conclusions derived from this work.

2. SPECIMEN DESCRIPTION

The specimen under study consists of a structural panel, which is part of one of the lateral faces of the MetOP-A satellite, and the latch support of the ASCAT antenna arm deployment system, which is attached to the panel as Fig. 2 shows.

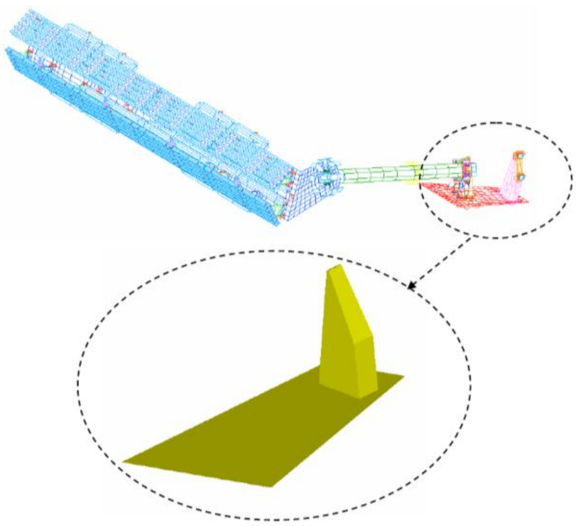

Figure 2. Specimen under study as part of the ASCAT antenna

The structural element – the 'support panel' from now on – is an aluminium honeycomb core and CFRP skin sandwich panel of approximately 467 x 733 mm. The core thickness is 18 mm and the skin thickness is 2 mm, leading to a total thickness of 22 mm. This panel includes 42 inserts (12 blind and 30 fully-potted).

The latch support – the 'equipment' from now on – is a 4-mm-thick titanium piece, which is attached to the support panel by means of 4 blind inserts. The distance between the bottom of the equipment and the support panel is approximately two times the equipment thickness.

Fig. 3 shows a picture of the specimen under study.

Figure 3. Specimen under study

3. SPECIMEN CHARACTERISATION

In order to characterize the specimen, a modal test was carried out at the Instituto Nacional de Técnica Aeroespacial (INTA).

In this test specimen was hanging from two points under free-free conditions. Structural excitation for modal characterization was performed through an electromagnetic shaker fed with a wideband white noise signal. Eigenfrequencies, eigenvectors and modal damping output data obtained from this test allow a fine-tuning of the FEM model proposed in this paper.

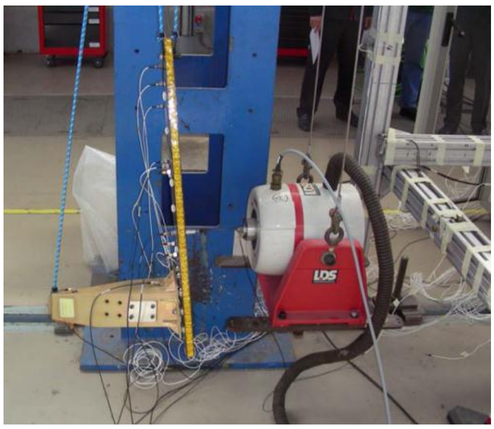

Figure 4. Modal test for specimen characterization

4. NUMERICAL MODELS

4.1 Preliminary SEA model for modal density analysis

In order to determine the frequency application range of each analysis method a preliminary SEA model has been performed, so as to calculate the modal density of each element of the system under study. A deep study of the influence of elements modal density on the design of numerical models can be found in [4].

In this SEA model the support panel has been modelled as one single sandwich plate with an isotropic material for the skin, due to restrictions of the commercial code. Physical properties of this isotropic skin have been chosen to match those of an orthotropic skin. In addition to this, the core density has been increased with respect to the nominal value, so as to account for the mass of the inserts included in the panel.

The equipment has been modelled as a set of uniform plates. Its modal density has been assumed to be approximately equal to that of the largest face of the equipment – the front face – and the contribution of the rest of the faces to the number of modes in band barely perceptible.

To model the air located inside the equipment, an acoustic cavity has been placed within it. However, results showed that the influence of this cavity is negligible.

Finally, in this model the equipment has not been attached to the support panel, since only modal density results are desired.

The resulting preliminary model is shown in Fig. 5.

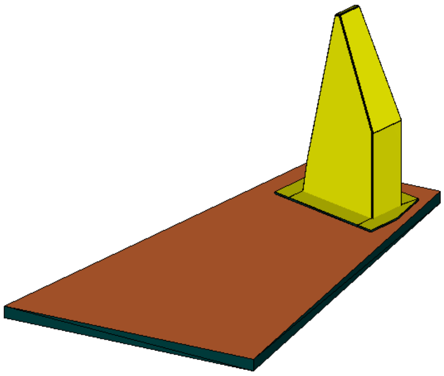

Figure 5. Preliminary SEA model for modal density analysis

Fig. 6 shows the modal count of the support panel, the acoustic cavity and the equipment. From this analysis, bands centre frequencies above which it is possible to use a SEA formulation may be obtained. Using a minimum of 5 modes per one-third-octave band as criterion, these frequencies are 3000 Hz for both the support panel and the acoustic cavity and 8000 Hz for the equipment.

Regarding these frequencies, three different numerical models are considered. The first model uses FEM formulation to model the support panel, the equipment and the acoustic cavity and BEM formulation to model the fluid domain. The second model is a hybrid model which uses FEM formulation for the equipment and SEA formulation for the support panel, the acoustic cavity and the fluid domain. Finally, the third model uses SEA formulation for the whole system.

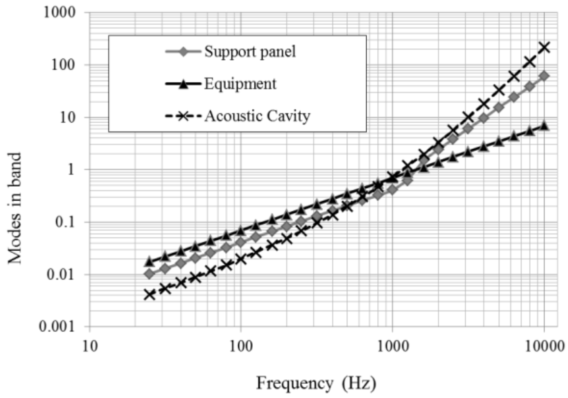

Figure 6. Modes per one-third-octave band of the support panel, the equipment and the acoustic cavity

In order to allow punctual junctions to be created between the equipment and the support panel when the latter is modelled with SEA, the panel has been divided into six different parts, reproducing thus the geometry of the equipment base (Fig. 7). So as to preserve the wave behaviour, SEA subsystems dimensions must be corrected to reproduce the behaviour of the whole support panel. This division has also been done in the FEM and hybrid models to allow comparison of results between the different models.

The acoustic load considered in all models is a diffuse acoustic field of an intensity of 1 Pa^2/Hz.

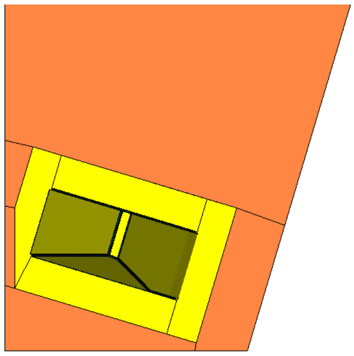

Figure 7. Support panel division

4.2 FEM-BEM model

For the low frequency range the plates of the support panel and the equipment are modelled with FEM, along with the acoustic cavity. The element size is set to ensure enough acoustic resolution, with six finite elements per wavelength up to the maximum frequency of interest. This leads to a finite element length of approximately 23 mm for the support panel and 10 mm for the equipment – the support panel is modelled with FEM up to 3000 Hz whereas the equipment is modelled with FEM up to 8000 Hz. The length of the acoustic cavity finite elements is also set to 10 mm, so as to ensure a high quality junction between it and the equipment faces.

Mechanical properties for this model have been obtained from the specimen specifications. A non-structural and uniformly distributed mass has been included in the support panel so as to take into account the inserts, which are not included in the model. Support panel and equipment are joined by means of three rigid junctions. The damping loss factor assigned to the structural elements is 0.1 % in the whole frequency spectrum, whereas the acoustic cavity has a damping factor of 1 %.

In this model, the fluid domain is formulated through a BEM fluid linked to the external faces of the structure. The diffuse acoustic field is applied through the superposition of 50 plane waves uniformly distributed in five latitude values and ten longitude values (Fig. 8).

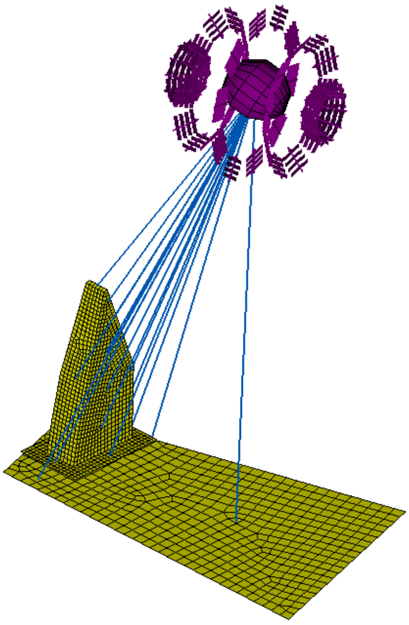

Figure 8. FEM-BEM model

4.3 Hybrid model

For frequencies between 3000 and 8000 Hz a hybrid model is performed. In this model the equipment is not modified with respect to the FEM model, while the support panel is modelled as a set of six SEA plates whose mechanical properties are the same as those determined for the preliminary SEA model. The acoustic cavity is also modelled with a SEA formulation.

Support panel and equipment are joined by means of three punctual hybrid junctions as shown in Fig. 9. Again, the damping loss factor is 0.1 % for the structural elements and 1 % for the acoustic cavity.

The fluid domain is modelled through a VA One Semi-Infinite Fluid [6,7] linked to the external faces of the structure. The acoustic load is applied through several VA One diffuse acoustic fields [6,7], one for each of the external faces of the structure.

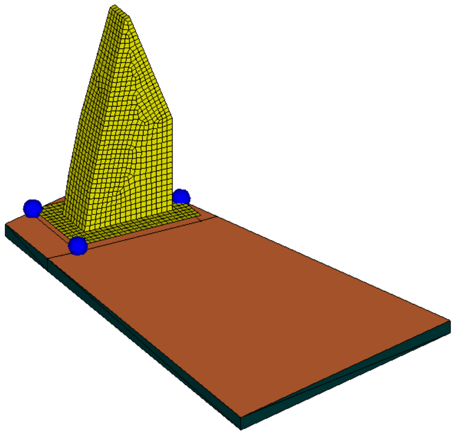

Figure 9. Hybrid model

4.4 SEA model

For frequencies above 8000 Hz a full SEA model is performed (Fig. 10), where the whole structure is modelled as a set of SEA plates. The formulation used for modelling the acoustic cavity, the fluid domain and the acoustic load is the same as the one used in the hybrid model.

Three punctual SEA junctions are used between the support panel and the equipment. The damping loss factor is 0.1 % for the structural elements and 1 % for the acoustic cavity.

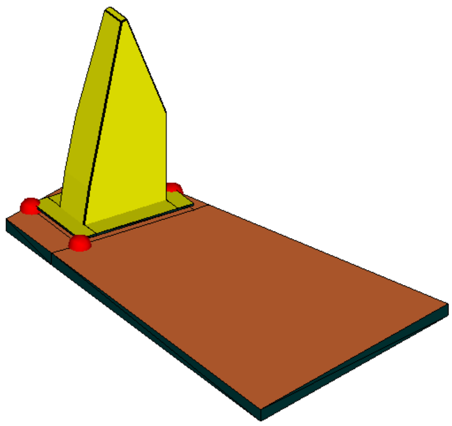

Figure 10. SEA model

5. RESULTS

5.1 Response continuity over the frequency range

In order to evaluate the adequacy of the different models proposed in this paper the continuity of their response to the acoustic load is analysed.

Figs. 11 and 12 show the acceleration response of the equipment – its front face – and the support panel – the plate located under the equipment – respectively, through the studied frequency range and the corresponding proposed models.

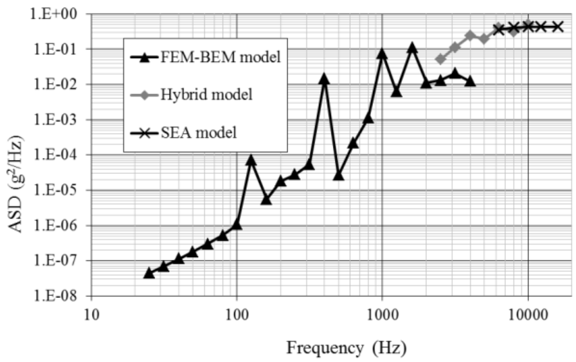

Figure 11. Acceleration response continuity of the equipment through the FEM-BEM, Hybrid and SEA models

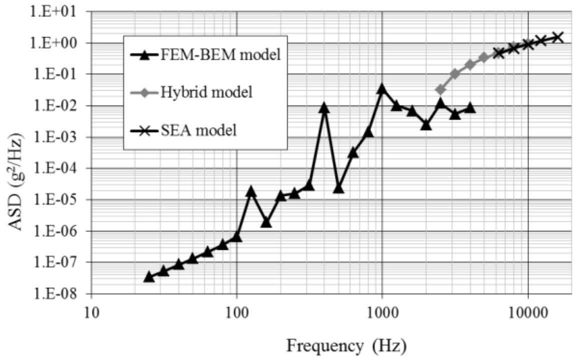

Figure 12. Acceleration response continuity of the support panel through the FEM-BEM, Hybrid and SEA models

5.2 Correlation with low frequency experimental results

An acoustic test was performed within the "Random Vibration Environment Derivation by Vibro-Acoustic Simulation" (RANDERIV) project [5]. The test was carried out in the reverberation room of the Instituto de Acústica – CSIC, which has a volume of approximately 200 m^3. Specimen was hung-up under free-free conditions in the central area of the room to guarantee a random acoustic load all around the structure under test. The acoustic load was set to 1Pa^2/Hz.

5 microphones were used to measure the room sound pressure levels, 2 microphones were placed close to the structure to measure the specimen near field, and 9 accelerometers were used to measure the vibration acceleration on the panel. In addition a triaxial accelerometer was put up on the front face of the equipment. Fig. 13 shows the position of these accelerometers on the specimen.

Sound pressure and specimen acceleration captured by the transducers were transmitted to an analyser were pressure levels and power spectral densities were calculated in narrow band.

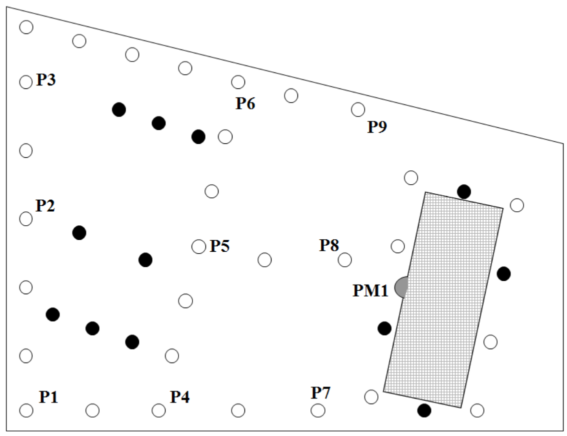

Figure 13. Accelerometers installed on the specimen during acoustic test

Figs. 15 to 18 show the correlation between these experimental results and those obtained from the FEM model. For this purpose, 10 velocity sensors were placed on the surface of the model reproducing the same position of the accelerometers in the acoustic test (Fig. 14).

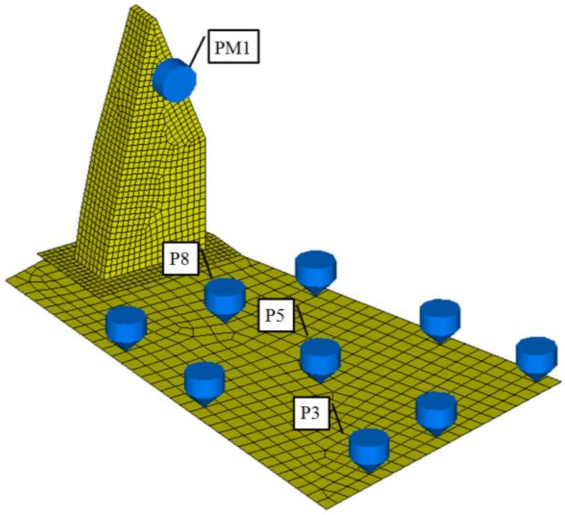

Figure 14. Velocity sensors installed on the FEM-BEM model for correlation with experimental results

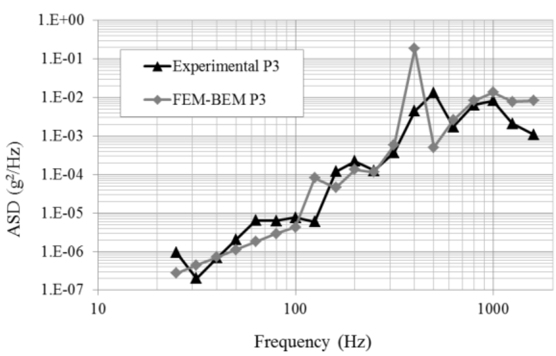

Figure 15. Correlation between P3 simulated and measured acceleration response

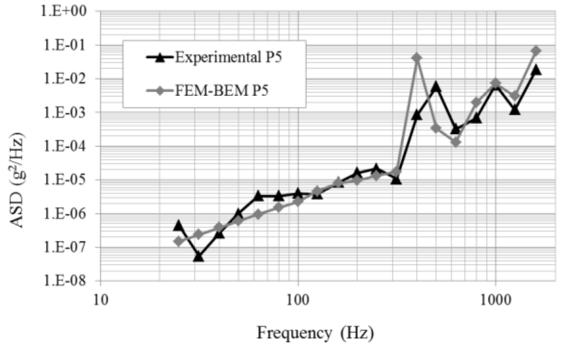

Figure 16. Correlation between P5 simulated and measured acceleration response

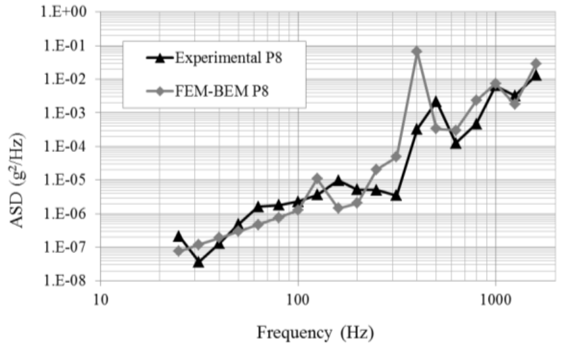

Figure 17. Correlation between P8 simulated and measured acceleration response

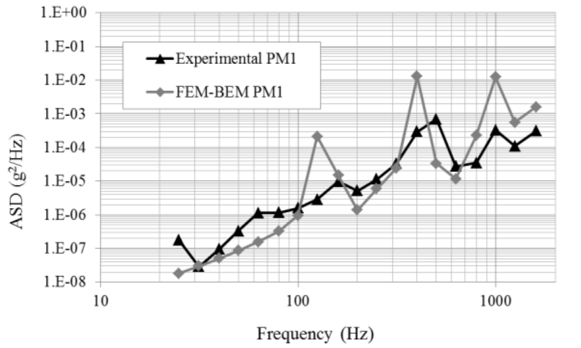

Figure 18. Correlation between PM1 simulated and measured acceleration response

6. CONCLUSIONS

Results obtained in this work show that the FEM-BEM model for the low frequency range approximates fairly well the actual specimen response. This validates the modelling of the support panel inserts as a non-structural and uniformly distributed mass and the joining between the equipment and the panel as three rigid junctions.

In regard to the response continuity achieved with the proposed numerical models, results from the simulations show that transition between low and medium frequency ranges is not adequately reproduced, since there is an order-of-magnitude difference in the acoustic response between the FEM-BEM and the hybrid models at its crossover frequency (3000 Hz). This might be a consequence of considering the specimen model as a set of only two subsystems – the support panel and the equipment – using the same modelling method for each one, instead of distinguishing between each of the plates comprising them. This concern is especially important for the equipment, since its shape is far from that of a rectangular plate, the basic geometry in the SEA formulation. For this reason, a multi-hybrid modelling approach as the one proposed in [4] would probably provide better continuity results.

AKNOWLEDGEMENTS

This work has been supported by the European Space Agency within the framework of the RANDERIV project. The work has been partially funded by the Spanish Secretaría de Estado de Investigación, Desarrollo e Innovación under the research programs DPI2005-08276; tools were also developed under the research program PB98-0725. Authors wish to thank Dr. Julián Santiago-Prowald, from ESA, and Dr. Ricardo Atienza and Mr. Díaz Robisco, from INTA, for their helpful comments during the development of the present work.

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