„FE/SEA Coupled“, 10 years after first implementation

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Abstract

The evolution of vibro-acoustic simulation methods has allowed engineers to tackle applications that were ever increasing in complexity and sheer size. From the Nastran punch cards to today's cloud computing, the increase in possibilities is mind-blowing and the perspectives for the future of computing have never been so bright. In parallel, the Statistical Energy Analysis (SEA) method has enabled the prediction of broadband noise and vibration in the high frequency domain for an increasingly wider range of applications. Ten years ago, the coupling between FEM and SEA gave rise to a new method that further increased the range of application where vibro-acoustic response could be computed. This work summarizes the main features of this new "FE/SEA Coupled" prediction method, describes the theory governing the coupling between FEM and SEA and presents several examples of applications in the automotive, aerospace, train, marine and cabin noise industries.

Introduction

The world of vibro-acoustic simulation has evolved slowly but surely over the last fifty years. In the 1060's, Clough coins the phrase "Finite Element" while Lyon and Maidanik wrote the basic equation describing the energy exchange between two oscillators [1] which is considered the foundation of SEA still today. At the end of the decade, Zienkievicz publish the book "The Finite Element Method" which is still a standard reference textbook today.

The 1970's MSC Software launches its Finite Element Software Nastran which was based on the code developed by NASA: Cosmic Nastran, today a "public domain" software. For SEA, it was a decade of investigation on subject such as Power Injection Method (PIM) and other experimental techniques such as comparing experimental results with analytical calculations of impedances and modal densities.

SEA commercial software's such as VAPEPs, SEAM and AutoSEA were introduced on the market in the 1980's and the 1990's saw many improvements in the formulation of SEA parameters such as the Leppington/Broadbant/Heron radiation efficiency formulation, wavenumber-space approaches and more generic CLF calculations based on line wave impedances. In the 1990's and first half of the following decade, a further generalization and extensions of SEA theory based on wave approach took place, the extensions of the variance theory and the increase in generic subsystems (sandwich, composite…) based on wave methods allowed SEA to be adopted in several industries as the standard method for high frequency vibro-acoustic analysis.

The Finite Element Method (FEM) evolved at a more rapid pace, with new sophisticated material and physical property cards being developed, this simulation method became widely accepted by a majority of industries for a wide variety of applications and simulation domains. After all these years, it was clear that the FEM would be constrained to lower frequency due to its computer resource requirement and SEA would be restricted to higher frequency due to its inherent statistical character and lack of phase information. Since the frequency domains of the two methods were not overlapping, the gap between the upper frequency of FEM and lower frequency of SEA required a solution especially for structureborne excitation. This mid-frequency gap was covered by an innovative solution called "FE-SEA Coupled" methods (see Figure 1).

This method combines the deterministic representation of various components of a system in FEM and the representation of other more flexible components in SEA.

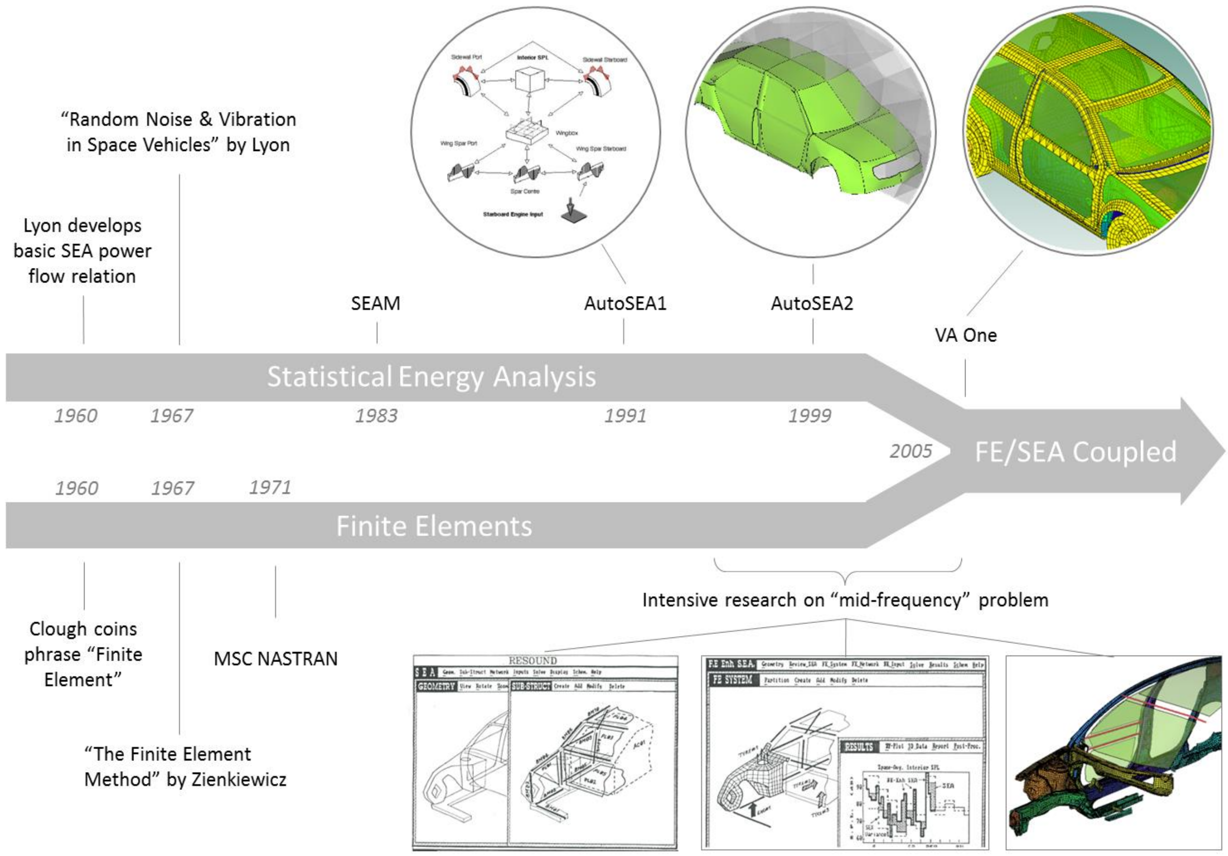

Figure 1: Evolution of classical vibro-acoustics methods culminated in the creation of the "FE/SEA Coupled" method.

Deterministic vs Statistical description of a subsystem

In the low frequency domain, the Finite Element Method (FEM) is well suited for structures and acoustic fluids where a low number of modes are present. It provides a good representation of the physics in a frequency range where boundary conditions (BC) has a non-negligible influence on the results. Boundary Element Method (BEM) is well suited for low frequency representation of fluids and is often combined to a FEM representation of the structure to compute radiated noise or Transmission loss. These methods are deterministic and usually computationally expensive but highly accurate [2][1].

In the high frequency domain, Statistical Energy Analysis (SEA) has been widely used for vibro-acoustic predictions on system and component studies. It is well suited to describe structures or acoustic fluids where a large number of modes are present. This method is extensively used in space, aircraft, automotive, rail and marine industry where complex structures and materials are often used [3],[4],[5],[6].

Both deterministic and statistical methods have different ways of representing the modes of a real system. In the FE method, the modes are represented by Eigen frequency and Eigen vectors and in the SEA method, modes are represented in terms of modal densities (Figure 2).

The modeling method proposed in this paper uses a combination of deterministic and statistical representation of a real system to understand its vibro-acoustic performance. The FEM and SEA content of a model are coupled together through the use of "FE/SEA Coupled" formulation (hybrid coupling).

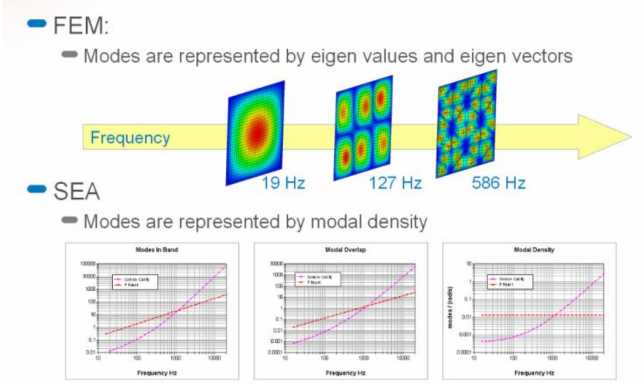

Figure 2: Representation of modes in FEM and SEA methods

Introduction to "FE/SEA Coupled" (From [7])

Hybrid FE/SEA method

A hybrid FE/SEA method ideally combines the low frequency performance of the FE method with the high frequency performance of SEA to produce a robust method that can be applied across the whole frequency range. However, the coupling of FE and SEA into a single model is difficult because the methods differ in two ways: (i) FE is based on dynamic equilibrium while SEA is based on the conservation of energy flow, and (ii) FE is a deterministic method while SEA is inherently statistical. Shorter and Langley [8] have developed a new method of realizing this coupling, which is based on wave concepts rather than the modal type of approach employed in reference [9]. At the heart of the method is a reciprocity result [10] regarding the forces exerted at the boundaries of an SEA subsystem. The method is briefly explained in the following paragraphs, it can be noted that references [8] and [10] contain a more formal and rigorous derivation of the "FE/SEA Coupled" method than that reported here.

In the mid-frequency range some components of a complex structure (for example thin panels) display short wavelength vibrations and are sensitive to the effects of random uncertainties, while others (for example beams) show little variation in their dynamic properties and are essentially deterministic. In the hybrid method proposed here, the deterministic components are modelled by using the finite element method, while the random components are modelled as SEA subsystems.

A key feature of the method is the concept of a "direct field" or "power absorbing" dynamic stiffness matrix associated with each SEA subsystem. Consider for example a thin plate that is excited at the boundaries. The excitation generates waves that propagate through the plate and are reflected repeatedly at the boundaries; the total dynamic stiffness matrix of the plate, phrased in terms of the edge degrees of freedom, has contributions from all of these reflections. Suppose now that the response is viewed in two parts: 1) the contribution from the initial generated waves, prior to any boundary reflections. This can be called the "direct field", 2) the contribution from waves produced on the first and all subsequent reflections. This can be called the "reverberant field". The direct field dynamic stiffness matrix can be defined as that resulting from the presence of the direct field waves – this matrix corresponds to "power absorbing" behavior, in the sense that the direct field waves all propagate energy away from the boundaries. Such a matrix can be found analytically for each of the subsystems by a variety of methods.

The Hybrid FE/SEA equations

The starting point for the hybrid method is to identify those parts of the system response that will be described by SEA subsystems. The remaining part of the system (which can be considered to be the "deterministic" part) is then modelled by using the FE method. For example, it might be decided that the bending motions of the panels of a structure have a short wavelength of deformation and will be described using SEA subsystems. The bending degrees of freedom of these panels will then be omitted from the FE model of the system, at all points other than the panel boundaries. The relevant "direct field" dynamic stiffness matrix is then added to the FE model at the panel boundaries, and this augmented FE model is then used in the subsequent analysis. If the degrees of freedom of the deterministic part are labelled q, then the governing equations of motion (for harmonic vibration of frequency ω) will have the form

$$\mathbf{D}_{tot}\mathbf{q} = \mathbf{f} + \sum_k \mathbf{f}_{rev}^{(k)} \;,\quad \mathbf{D}_{tot} = \mathbf{D}_d + \sum_k \mathbf{D}_{dir}^{(k)} . \tag{1,2}$$

The summation is over the number of SEA subsystems in the model, and $\mathbf{D}_{dir}^{k}$ represents the direct field dynamic stiffness matrix associated with subsystem k. Furthermore, $\mathbf{D}_d$ is the dynamic stiffness matrix given by the finite element model of the deterministic part of the system, $\mathbf{f}$ is the set of external forces applied to this part of the system, and $\mathbf{f}_{rev}^{k}$ represents the force arising from the reverberant field in subsystem k, which is not accounted for in $\mathbf{D}_{dir}^{k}$. The matrix $\mathbf{D}_{tot}$ is the dynamic stiffness matrix of the FE model (excluding the SEA subsystem degrees of freedom), when augmented by the direct field dynamic stiffness matrix of each SEA subsystem. It should be noted that equations (1) and (2) are exact – all that has been done is to split the forces arising from the SEA subsystems into a direct field part, which is accounted for by $\mathbf{D}_{dir}^{k}$, and a reverberant part which is carried to the right hand side of equation (1). The following result (Shorter and Langley[10]) is central to the development of the hybrid method:

$$\mathbf{S}_{ff}^{(k),rev} \equiv \mathrm{E}[\mathbf{f}_{rev}^{(k)}\mathbf{f}_{rev}^{(k)*T}] = \left(\frac{4E_k}{\omega\pi n_k}\right)\mathrm{Im}\{\mathbf{D}_{dir}^{(k)}\} . \tag{3}$$

Here E_k and n_k are respectively the (ensemble average) vibrational energy and the modal density of the k^{th} subsystem. Equation (3) implies that the cross-spectral matrix of the force exerted by the reverberant field is proportional to the resistive part of the direct field dynamic stiffness matrix, which is a form of diffuse field reciprocity statement. From equation (1), the response $\mathbf{q}$ can be expanded in the form

$$\mathbf{q} = \mathbf{q}_d + \sum_k \mathbf{q}^{(k)}, \quad \mathbf{q}_d = \mathbf{D}_{tot}^{-1}\,\mathbf{f}, \quad \mathbf{q}^{(k)} = \mathbf{D}_{tot}^{-1}\,\mathbf{f}_{rev}^{(k)} . \tag{4}$$

Now the time averaged power input to the direct field of subsystem j can be written as

$$P_{in,j} = \left(\frac{\omega}{2}\right)\mathrm{Im}\{\mathbf{q}^{*T}\mathbf{D}_{dir}^{(j)}\mathbf{q}\} = \left(\frac{\omega}{2}\right)\sum_{rs}\mathrm{Im}\,\{D_{dir,rs}^{(j)}\}S_{qq,rs} \tag{5}$$

where it has been noted that the dynamic stiffness matrix is symmetric. If the various contributions $\mathbf{q}_k$ that appear in equation (4) are taken to be uncorrelated and of zero mean, then the equation (3) – (5) yield

$$P_{in,j} = P_{in,j}^{ext} + \sum_k \omega\eta_{jk}n_j(E_k/n_k) \tag{6}$$

where

$$P_{in,j}^{ext} = (\omega/2)\sum_{rs}\mathrm{Im}\,\{D_{dir,rs}^{(j)}\}\left(\mathbf{D}_{tot}^{-1}\,\mathbf{S}_{ff}\mathbf{D}_{tot}^{-1*T}\right)_{rs} \tag{7}$$

$$\omega\eta_{jk}n_j = (2/\pi)\sum_{rs}\mathrm{Im}\{D_{dir,rs}^{(j)}\}\left(\mathbf{D}_{tot}^{-1}\,\mathrm{Im}\{\mathbf{D}_{dir}^{(k)}\}\mathbf{D}_{tot}^{-1*T}\right)_{rs} \tag{8}$$

Given that the dynamic stiffness matrices are symmetric, it is readily shown from equation (8) that reciprocity holds, in the sense that $\eta_{jk}n_j = \eta_{kj}n_k$. As will be shown in what follows, the terms η_{jk} are equivalent to the coupling loss factors that appear in SEA. The power output from the reverberant field in subsystem j can be written as the sum of: i) the power dissipated through damping, ii) the power transferred to the other subsystems, iii) the power dissipated in the deterministic system due to the response $\mathbf{q}^{(j)}$. Thus

$$P_{out,j} = \omega\eta_j E_j + \sum_k \omega\eta_{kj} n_k (E_j/n_j) + \omega\eta_{d,j} E_j \tag{9}$$

where

$$\omega\eta_{d,j} = (\omega/2\,E_j)\,\mathrm{Im}\{\mathbf{q}^{(j)*T}\mathbf{D}_d\mathbf{q}^{(j)}\} = \left(\frac{2}{\pi n_j}\right)\sum_{rs}\mathrm{Im}\{D_{d,rs}\}\left(\mathbf{D}_{tot}^{-1}\,\mathrm{Im}\{\mathbf{D}_{dir}^{(j)}\}\mathbf{D}_{tot}^{-1*T}\right)_{rs} \tag{10}$$

Equation (6) and (10) then lead to the following energy balance equation for subsystem j

$$\omega\left(\eta_j + \eta_{d,j}\right)E_j + \sum_k \omega\eta_{jk}\, n_j\left(\frac{E_j}{n_j} - \frac{E_k}{n_k}\right) = P_{in,j}^{ext} \tag{11}$$

Furthermore, the cross-spectral matrix of the response $\mathbf{q}$ can be derived from equations (3) and (4), which yields

$$\mathbf{S}_{qq} = \mathbf{D}_{tot}^{-1}\left[\mathbf{S}_{ff} + \sum_k\left(\frac{4E_k}{\omega\pi n_k}\right)\mathrm{Im}\{\mathbf{D}_{dir}^{(k)}\}\right]\mathbf{D}_{tot}^{-1*T} \tag{12}$$

Equations (11) and (12) form the two main equations of the "Hybrid FE/SEA" method. It is clear that these equations couple FE and SEA methodologies: equation (11) has precisely the form of SEA, but the coupling loss factors $\boldsymbol{\eta}_{jk}$ and loss factors $\boldsymbol{\eta}_{d,j}$ are calculated by using the FE model (augmented by the direct field dynamic stiffness matrices) via equations (8) and (10); furthermore, equation (12) has the form of a standard deterministic FE analysis, but additional forces arise from the reverberant energies in the subsystems. If no SEA subsystems are included then the method becomes purely FE; on the other hand, if only the junctions between the SEA subsystems are modelled by FE, then the method becomes purely SEA, with a novel method of computing the coupling loss factors.

The steps in the hybrid method

The hybrid method proceeds as follows:

1) A finite element model of the deterministic part of the system is constructed. All degrees of freedom associated with SEA subsystems are omitted from this model, other than those that lie on the subsystem boundaries.

2) A "direct field" dynamic stiffness matrix is constructed for each subsystem in terms of the relevant boundary degrees of freedom. These matrices are then coupled to the FE model to yield the total dynamic stiffness matrix $\mathbf{D}_{tot}$.

3) The various terms that appear in the SEA equation, equation (11), are calculated from equations (7), (8), and (10).

4) The SEA equations are solved to yield the subsystem energies E_j.

5) Given the subsystem energies, equation (12) is used to yield the response of the deterministic part of the system.

The terms E_j and S_{qq} then constitute the solution yielded by the method, and related quantities, such as energy flows or energy densities, can then be computed if required. A key feature of the method is that the FE mesh considered in step (1) does not need to capture the short wave response of the subsystems, and thus relatively few degrees of freedom are required compared to a conventional finite element model. As reported in a companion paper [11] this leads to very significant reductions in the computer time needed to solve the problem.

Structure to structure coupling

The combination of FEM and SEA representation for a structure is used in many industries; the space and automotive being the major industries sharing this common modelling approach [12],[13],[14],[15],[16]. Although this modelling approach yields accurate results, many users in the automotive industry are nowadays investigating the possibility of representing the full structure in FEM, the interior fluid as SEA and the acoustic trim with Transfer Matrix Method (TMM) [17,],[18]. The "FE/SEA Coupled" model can be created from the low and high frequency models using a remeshed FEM structural model, the same SEA cavities used in the pure SEA model and using a simplified trim description derived from the SEA model. The benefits of this new approach are numerous: i) The full vehicle model can be created in a few days instead of several weeks, ii) a non-expert in vibro-acoustics can create the FE/SEA model because it only involves coupling existing models together, iii) The cost to get an interior SPL prediction is converted from engineer modeling time to computer time, CPU and memory usage, freeing the engineer to do more valuable work than partitioning the structure into FEM and SEA.

Application to real world cases

Launcher, Spacecraft and Satellite

An example of a launcher simulation using FE/SEA Coupled is presented in [13]. Here care was taken to model the stiff part of the launcher with FEM and the more flexible parts using SEA. All important SEA subsystems are correlated with their FEM representation to make sure their modal densities and radiation efficiencies are correctly represented. Also, EFM is used in a critical connection to ensure proper flow of energy. According to the authors: "Considerable improvement in the quality of correlation with test data from pre-test to post-test has been made. The refined vibro-acoustic model is not only able to show good match with test data, but also predicts the dynamic load transfer paths from the external acoustic excitation to the interior of the CM (Crew Module) more accurately"

The NASA's Advanced Communications Technology Satellite's (ACTS) large parabolic reflector antennas were exposed to a reverberant acoustic loading to simulate the launch acoustic environment in the Space Shuttle Discovery payload bay. Several models were built: i) deterministic where structure is in FEM and fluid in BEM, ii) a statistical model where structure and fluid are represented as SEA and iii) a combined model where structure is FEM and fluid in SEA using a SIF (Semi-infinite fluid) and compared with test data. The authors concluded that: "At the majority of specific spatial locations, the Hybrid (FE/SEA Coupled) model predictions matched test accelerometer data very well…for the antenna structure studied, the Hybrid FE-SEA predictions matched the test data as well as the FE-BEM predictions, with the benefit of considerable computation time savings…SEA is still necessary to predict the responses at high frequencies due to limitations from FEM in capturing high-frequency modes…A combination of Hybrid and SEA methods could be used to cover the entire frequency range of interest for this and other problems." [19].

For several years, the simulation of acoustic tests on spacecraft or instruments are performed by Knockaert [16] and results are used to assess i) random levels at interfaces to define subsystem specifications, ii) load levels to ensure admissible loads are not exceeded and iii) stress levels to avoid damage during the acoustic test and launch. These simulations are more commonly used and especially for new developments where no predecessors structures are available. They represent a significant cost in terms of computer time and memory. Several methods have been compared for accuracy and computer resources requirement. Knockaert concludes: "The methods presented here have a different effect on the computation time. The orders of magnitude of the gains are the following ones: i) …simplified diffuse acoustic fields (Statistical -> FE/SEA Coupled) is much faster than coupled BEM/FEM computations…by a factor around 10, ii) the replacement of FEM subsystems by SEA subsystems has a more limited effect. Roughly speaking, the gain is proportional to the decrease in the overall number of nodes coming from this substitution…Even if vibro-acoustic simulations on large systems remain quite time consuming, recent simulation tools offer some possibilities to limit the computation time. It is now possible to perform simulations on a complete spacecraft or instrument in a reasonable amount of time”.

Aircraft fuselage, trim panel, luggage bin, counter rotating propeller

An extensive study of the best modelling guidelines using FE/SEA Coupled has been published in [20]. Various components and areas were studied such as i) luggage bin, ii) floor l, iii) Fuselage wall, iv) connection between floor and fuselage wall and finally v) fuselage wall and interior trim panel connections. Detailed analysis of the rationale behind the choice of modelling approach is described and approached are validated with measurements data. Authors conclude: “This paper has discussed the application of the Hybrid FE-SEA method and the periodic SEA subsystems to various structural-borne transmission problems. The examples demonstrate that the new methods can provide improvement to existing SEA models. For the problems considered, the methods are typically several orders of magnitude faster than a purely deterministic analysis, although they are more demanding than pure SEA. The methods are well suited to the analysis and design of structural-acoustic systems of practical interest”.

Peiffer [21] presented work done on interior noise prediction for the case of an aircraft equipped with counter rotating propellers. Since the propeller interaction tones change the directivity pattern of the excitation, it is inherently a deterministic source in mid and high frequency and a FEM fuselage wall at the rear of the aircraft and close to the propeller was used. The forward part of the aircraft was modelled as SEA. Peiffer concludes that: “The hybrid coupling of SEA cavities and FEM shells is precise enough and that in the ensemble average the hybrid SEA/FEM description provides correct results…The deterministic subsystems can be used for design studies of tonal noise control methods like tuned vibration absorbers (TVAs)”.

Automotive transmission loss, full vehicle structureborne and wind noise

Transmission loss (TL) is key to determining component acoustic performance in automotive. An investigation on the use of “FE/SEA Coupled” to component Transmission Loss is presented in [22]. The authors used SEA subsystems to represent the acoustic fluids and FE subsystems to represent the structure. The FE and SEA subsystems were coupled using Hybrid Area Junctions and a fully coupled analysis was performed. According to the authors: “…the potential benefits of the Hybrid FE/SEA approach over existing methods are that: (i) the Hybrid FE/SEA model was found to be several orders of magnitude faster than an equivalent BEM model (making it possible to compute the narrowband TL over a broad frequency range), (ii) the model does not require detailed geometric modeling of the contact surface between the structure and fluid (resulting in a simpler modeling approach than traditional low frequency methods) and (iii) the detailed results from a component Hybrid FE-SEA model can be used to update a …(SEA)… system level model (the use of an integrated environment simplifies model management). Local TL predictions were compared with test and good agreement was observed. The model was used to diagnose problematic frequencies in the TL and by performing a modal contribution analysis it was possible to identify how these problematic frequencies could be addressed”.

Because the structures are modelled in FEM, it is possible to compute TL for more exotic material using FE/SEA Coupled such as composite and honeycomb panels. [2] describes the “FE/SEA Coupled” models for two honeycomb panels transmission loss computations and compare results of simulation with test. Correlation is excellent with a deviation of less than 3 dB in a broad frequency range (Figure 3). The “FE/SEA Coupled” method is well adapted to compute Transmission Loss of complex material. The model building effort is low since it uses the existing FEM model for structural component (model usually available). It covers a wide frequency range, the accuracy is high and the method is at least 10 times faster than FEM-BEM.

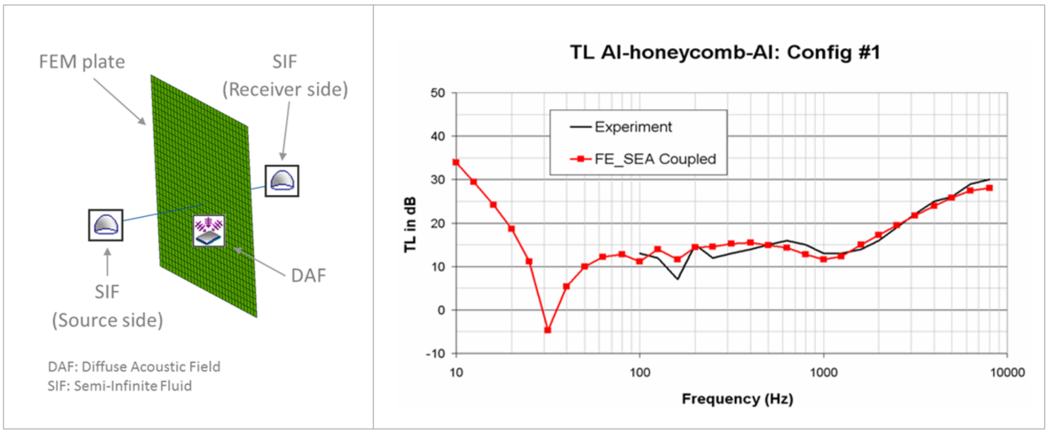

Figure 3: Comparison of Transmission Loss (TL) of a sandwich panel with Aluminum skins and honeycomb core. High level of correlation between measured (black) and predicted with "FE/SEA Coupled" (red). Note the large frequency domain the prediction can cover (10 to10 000 Hz).

Bocquillet recently published in [23] a study on “FE/SEA Coupled” where he investigates the ability of the method to predict insertion loss of a double-wall insulation system on a simplified representative model of a door. Frequency range of interest is 200 to 4000Hz. Once the model is setup and structural and acoustic damping’s are evaluated, he compares simulation with measurement and concludes: “simulation can match the measurement quite good, resonances are correctly represented…better accuracy of the results and interpretation than SEA”.

In [24], one can find a TL study aimed at better understanding the door seals. The study compares deterministic 2D and 3D FEM/BEM models with FE/SEA Coupled for various types of seals. The authors conclude that: …Hybrid FE-SEA models … were found to be approximately 50 times faster than the BEM models and are well suited to the prediction of TL. In the FE-SEA Coupled models, the source and receiving rooms of the TL suite were described with SEA, while the slit, channel and seal were described with acoustic FE. … The analysis of the transmission loss of seals showed that there are differences in the transmission loss between 2D and 3D models. A 3D model is needed to capture some of the features of the transmission, and the use of 2D models may lead to over-estimation of the transmission loss. It was also shown that the channel (gap where the seal is installed) can significantly reduce the transmission loss of the seal (up to 10 dB). Finally, the effects of the seal geometry (deformed versus undeformed) on the transmission loss were shown to be significant”.

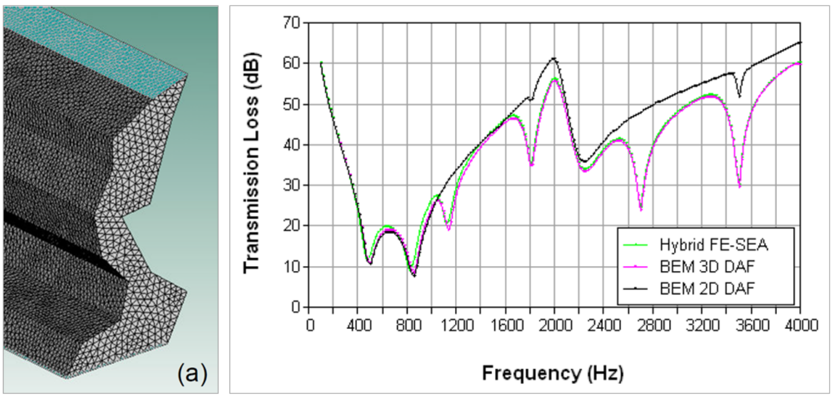

Figure 4: Transmission loss of a seal – Comparison between 2D-BEM, 3D-BEM and FE/SEA Coupled. Perfect agreement between FE/SEA Coupled and 3D BEM where 2D BEM fails

Several publications discussed the modelling of full vehicle vibro-acoustic response. In [14], a detailed description of the process of modelling a full vehicle in FE/SEA Coupled is given. Although users today do not tend to split the structure into FEM and SEA anymore but rather keep the whole structure in FEM, this paper is useful to understand the basics of the FE/SEA Coupled method: “By combining FE and SEA, this model is capable of predicting structureborne noise transmission over the range [200-1000] Hz. A process to analyse the dynamics of a complex structure and define the most appropriate model partitioning was established. Based on this partitioning, the regions of the structure that exhibit short wavelength behaviour were described with SEA. Only the stiff regions were left as FEM, making the resulting Hybrid (FE/SEA Coupled) model computationally efficient…It was shown how detailed local FE models of the BIW components could be used to efficiently calculate the SEA subsystem properties and SEA couplings loss factors across complex junctions…The Hybrid model is capable of predicting power inputs due to point force excitation at the engine mounts and shock tower locations within 3dB of tests for most frequency bands. Additionally, the Hybrid model is capable of predicting the velocity distribution on the floor and dash, not only for point force excitation on the front frame (shock towers and engine mounts), but also for spatially delta-correlated excitation ("rain on the roof") of any component of the vehicle…Additional numerical validations were performed on a half model (left side of a BIW)...The Hybrid model shows excellent correlation against the reference FE results for all the load cases studied and subsystems".

A more recent approach is described in [18], where the structure is fully described as FEM and the interior cavities as SEA and the trim in TMM. In addition, this innovative paper breaks down the prediction of interior noise by computing the radiated power from untrimmed body panels and adding on this path the component performance of each trim piece, such as the insertion loss and absorption coefficient. Authors mention: "it has been shown that synthesized 1/3 octave band results present a good correlation with vehicle measurements, not only for the incremental change in interior response associated with trim component effects, but also in the absolute value of interior response. The approach was successfully employed in an optimization study for trim component effects in the mid-frequency range. A total of 48 DOE runs corresponding to 12 trim iterations and 4 loadcases were done within a short period of time with reasonable accuracies".

Recent developments in wind noise contribution to interior Sound Pressure levels (SPL) have been published in [25]. It describes methods to characterize the turbulent source outside a side glass and combine this with a validated vibro-acoustic model to predict interior noise. Various modelling approaches are discussed. Recent results show high level of correlation for the case where the exterior turbulent flow is described using CFD compressible time domain histories, the side glass is modelled as FEM and the interior cavity is modelled in SEA (Figure 5).

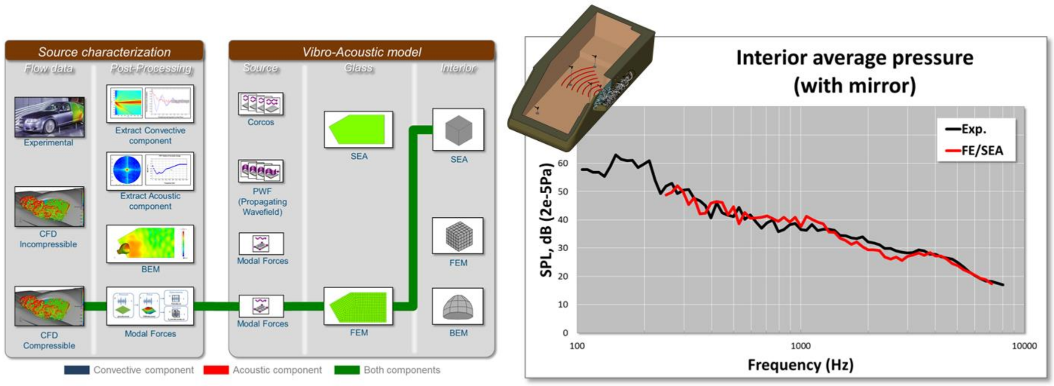

Figure 5: FE/SEA Coupled simulation combined with CFD representation of the exterior turbulences. Accuracy of the method is high and computation is several orders of magnitude faster than with BEM inside the vehicle.

Other industries: rail, elevators, HVAC, cabs…

In [15], a feasibility study investigating if FE/SEA coupled can be used for the prediction of a high speed train driver's cabin SPL was performed and documented. "The results obtained show normally a good accuracy for the vibration prediction at the frequency range of [100, 1000] Hz and the discrepancies between the models and the experimental curves are normally below 5 dB.

Furthermore, in those cases where the differences between the models and the tests are higher, the hybrid methodology seems to give a better agreement…"

In [26], the authors describe the various vibro-acoustic sources and transfer paths encountered in elevators. Since new towers are being designed and will soon reach 1000m in height, the design of elevators need to account for wind noise contribution generated by turbulences present on the outer surface of the cabin. Therefore, the coupling between CFD and vibro-acoustic is also investigated.

In 27, transmission loss (TL) predictions of large plena are computed using FE/SEA Coupled. The fluid inside the duct is modelled in FEM acoustic fluid and the excitation and the outer fluid are statistically represented. Authors note that: "The advantage of the approach is that the sound power is applied to the finite element model in a statistical sense…The simulation results compare well with published measured results. The results suggest that the methodology described in the paper can be applied to wide range of problems involving large mufflers and silencers. The suggested approach is especially advantageous above the plane wave cutoff frequency".

Finally, in [28] the use of "FE/SEA Coupled" is used to compute SEA canonical subsystem properties such as mass density, average stiffness and radiation efficiency to account for high complexities of stamped parts. The authors conclude: "In this paper, a relatively large SEA system model of a combine harvester cab was improved using these techniques. The (SEA) subsystems representing the major panels of the cab were modified, and the resulting model was compared against a series of laboratory tests representing idealized airborne loading of the cab exterior. The overall SPL predicted by the SEA model showed less than 0.5 dB deviation from the measured results, and the one-third octave band spectra agree favorably; the model captures spectral detail quite well. The modified SEA model can now be exercised with confidence using field insonifications gathered from several applications. This information will be employed to optimize noise control treatments for the cab".

Conclusion

This paper has described the birth and the evolution of the "FE/SEA Coupled" method. It has also presented the theory behind the coupling between FEM and SEA and has review several papers discussing the use of "FE/SEA Coupled" in real life application in the space, aeronautic, automotive and various other industries where vibro-acoustic simulation are of concern.

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