Correlation of the Vibroacoustic Response of Structural Panels with Isight for Use in Statistical Energy Analysis in Aerospace Applications

Cory J. Rupp[1] and Lina Maricic[2]

ATA Engineering, Inc.

[1] 1687 Cole Blvd. Suite 125, Golden, CO 80401

[2] 11995 El Camino Real, Suite 200, San Diego, CA 92130

Abstract: With the increasingly common use of advanced composite materials in the design of aerospace structures, vibroacoustic loads have become an important factor for structural design and integrity. Vibroacoustic analysis at high frequency is often performed by statistical energy analysis (SEA) using software products such as ESI's VA One. A difficulty in performing SEA on complex aerospace structures is ensuring that the SEA subsystem (e.g., a structural panel) properly represents the actual structure. The model is often checked by comparing the response of a subsystem to that of a refined finite element (FE) mesh of the subsystem. Unfortunately, modifying the SEA properties to improve the correlation between the SEA and FE responses is often cumbersome and time consuming. To alleviate the difficulties in this process, we have developed an interface between SIMULIA's Isight simulation management software and VA One that automates the correlation process. The details of this interface are shown and several correlation examples of varying complexity are provided. Using this improved process will speed up workflow and improve the accuracy of SEA simulations.

Keywords: Aerospace analysis, Optimization, Vibroacoustics.

1. Introduction

Aerospace structures including aircraft, satellites, and space vehicles are increasingly making use of highly engineered materials and components to reduce weight and improve performance. In doing so, these structures are also becoming more sensitive to acoustic environments, to the point where the vibroacoustic response of the structure is a major concern for the integrity and reliability of the structure and the equipment attached to it. Vibroacoustic analysis generally spans a large frequency range of interest, from 10s of Hz up through 10,000s of Hz. At low frequencies, a finite element model can be used to represent the structure and its vibroacoustic response. At higher frequencies, however, the finite element (FE) discretization necessary to accurately resolve the eigenmodes, as well as the sheer number of modes necessary to reach high frequency, becomes unwieldy, and the vibroacoustic problem quickly becomes intractable. At these high frequencies, vibroacoustic analysis is often performed using the statistical energy analysis (SEA) method, where the structure of interest is divided into a small set of subsystems that represent whole panels, beams, or acoustic cavities within the structure. The result is a coarse representation of the

structure that balances the vibroacoustic energy, power inputs, dissipation, and power transmission within and between the subsystems in a statistical sense.

The foremost difficulty in creating an SEA representation of any structure is ensuring that the SEA subsystems properly represent the actual structure. In general, SEA subsystems are created to span large, more-or-less physically continuous sections of a structure so that the assumptions of SEA are valid with boundaries usually placed at physically relevant locations such as connections to beams or other panels. If a subsystem is too small, its response and the transfer of energy via the panel will be inaccurate, thereby resulting in downstream inaccuracies in addition to the local inaccuracies in response. This, combined with the fact that there can only be one set of averaged properties per subsystem, often poses a difficulty in defining the set of properties used to represent an SEA subsystem because most structures, in particular highly engineered aerospace structures, are not uniform over large regions. Features such as stiffeners or holes further complicate the creation of subsystems. Some techniques to address this issue include spatial averaging of the properties and using the most represented property.

The accuracy of an SEA subsystem can be checked by comparing its response to that of a refined FE mesh of the same subsystem. Updates can then be made to the SEA subsystem properties in order to improve the correlation between the two models. Unfortunately, this updating and correlation process is often cumbersome and time consuming due to the number of possible variable combinations and the sometimes non-intuitive nature of how the response changes due to a change in variable. In previous projects where this procedure was performed, correlating a single panel could take several hours to a whole day to complete, depending on the complexity of the panel. To alleviate the difficulties in this process, we have developed a methodology and interface between SIMULIA's Isight simulation management software and the SEA software package VA One that allows the automation of this correlation process.

2. Model setup: Isight + Matlab + VA One

The setup for automating the SEA-to-FE vibroacoustics correlation process consists of three parts: (1) using Isight to drive an optimization algorithm and make changes to design variables, (2) using VA One to solve the vibroacoustic problem for a set of design variables and return the vibroacoustic response, and (3) using Matlab as an interface between Isight and VA One and to calculate the objective function. Figure 1 shows the Isight sim-flow diagram with an optimization module encompassing a Matlab module. Within the Matlab module, the design variables are first mapped to the Matlab workspace, then a script is executed that calls functions from the VA One Matlab API, and finally the responses and objective function are mapped back to Isight. The commands in this script update model parameters via the design variables, solve the problem, compute the value of the objective function, and store the resulting solution in Matlab variables. The dataflow diagram in Figure 2 shows the data mappings between Isight and Matlab. An example Matlab script showing a typical set of API calls to VA One is provided in Appendix A.

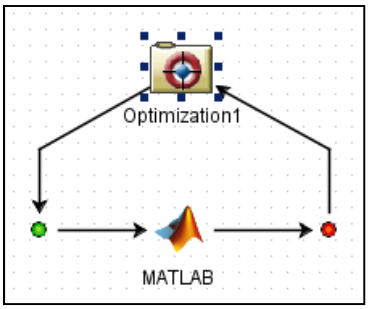


Figure 1. Sim-flow diagram of the model setup in Isight connects an optimization loop to Matlab.

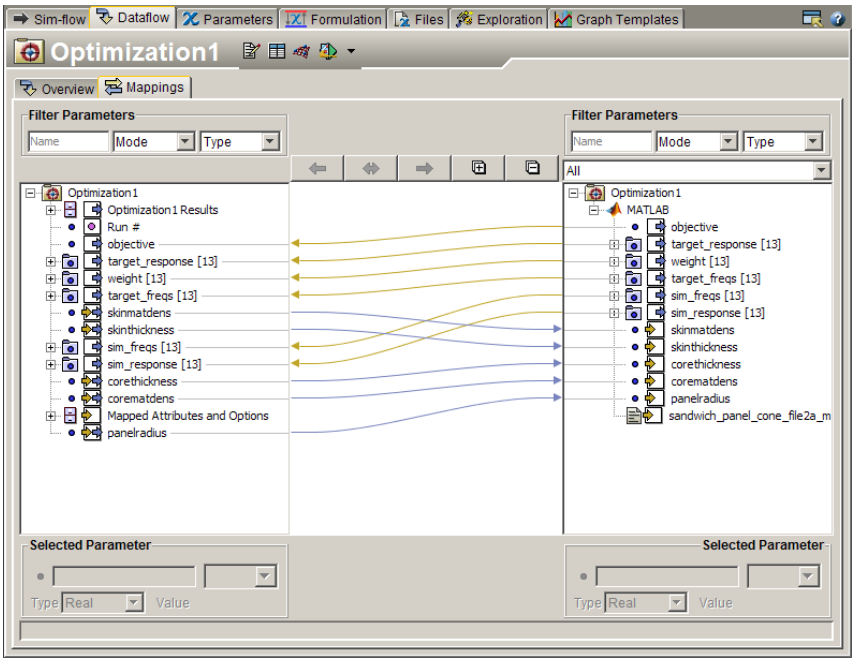


Figure 2. Dataflow diagram showing the flow of design variable and optimization parameter data between Isight and Matlab.

Several objective function formulations were tested when developing the methodology for the correlation process. The formulation that was found to perform the best resembles a weighted least-squares approach; it is written as

$$f = \sum_{i=1}^{nfreq} w_i * \left(\log_{10} R_{i,sim} - \log_{10} R_{i,target}\right)^2,$$

where $R_{i,sim}$ is this simulation response at the *i*th frequency band (in PSD units), $R_{i,target}$ is the same for the target response, w_i is a weight function, and nfreq is the number of frequency bands in the analysis. This function takes into account the logarithmic scaling of the PSD response and can also

be tailored to individual problems via the weight function. In the examples provided in this paper, the weight function is written as

$$w_i = i,$$

which places more emphasis on correlation at higher frequencies. This weight function is used for two reasons: (1) the SEA method is more accurate at higher frequencies, so accurate correlation at low frequency may have little meaning; and (2) the correlated SEA subsystem will be used to predict responses at frequencies higher than those used for this problem, so it is more important to have better correlation at high frequencies.

Isight enables the user to select from a variety of optimization algorithms to use in the optimization module. Because of different model formulations for different types of SEA subsystems in VA One, there is no single optimization algorithm that works best for the correlation problem. The examples that follow will illustrate two approaches to this problem.

3. Examples

Two example problems that illustrate the correlation process are presented here. The first example is a simple conical section panel of sandwich panel construction. Sandwich panels are extensively used in aerospace applications for their high stiffness-to-mass ratios; unfortunately, this also makes them more susceptible to vibroacoustic excitation. The second example is a complex cylindrical section of a titanium ribbed panel. This example panel is typical of highly engineered aerospace structures and illustrates the particular difficulty in determining an appropriate vibroacoustic representation of such panels.

3.1 Conical sandwich panel section

The conical sandwich panel example is used to illustrate a difficulty in accurately modeling certain types of vibroacoustic panels in VA One. In general, modeling sandwich panels in VA One is a straightforward process as long as the panel geometry is simple. A conical section represents one of the cases where the geometry cannot be well represented and inaccuracies in panel response due to vibroacoustic loads can arise. Specifically, conical panels are not explicitly supported in VA One and are most closely represented by curved panels with a single radius of curvature. Conical sections do not have a single radius of curvature, so defining this value is the primary point of ambiguity in this problem.

The example panel consists of a sandwich panel with a honeycomb aluminum core 0.75 inches thick, and two carbon-fiber composite facesheets totaling 0.04 inches thick. The panel has a maximum radius of 50 inches, and it is 30 inches in height at an angle of 30 degrees. The panel spans a 60-degree conical section. A finite element (FE) model of the panel was created using composite shell elements. Figure 3 shows the panel geometry.

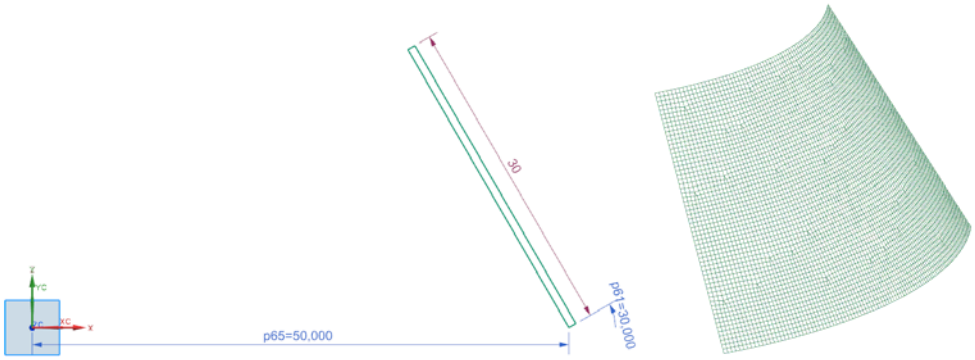
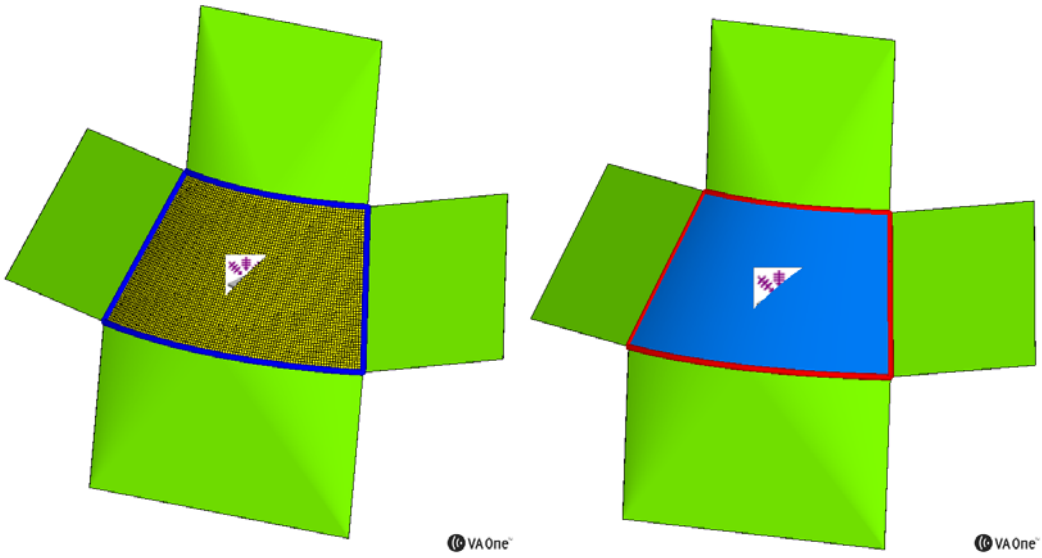


Figure 3. Sandwich panel geometry and FE model.

In order to calculate the vibroacoustic response of the FE panel, the free-free modal frequencies and displacements are first computed, and both the model and modes are imported into VA One. In general, and depending on the problem, when choosing the maximum frequency for the eigenmode solution a balance is struck between having a sufficient frequency range for vibroacoustic analysis and having a reasonable number of modes such that the analysis can be solved in a practical amount of time. In this example, modes were calculated using standard practices up to 4,600 Hz, which covered up to the 4,000 Hz 1/3-octave band and resulted in 109 modes. In VA One, an FE subsystem is created using the model, and several "dummy" SEA panel subsystems are connected to it as shown in Figure 4(a). This step is necessary because the vibroacoustic response of FE subsystems in VA One is dependent on the boundary conditions, while this is not the case for SEA subsystems. Additionally, the dummy panels should have realistic properties comparable to the connecting structure (alternatively, one could use a static reduction of the rest of the structure). This step essentially ensures an appropriate comparison between the SEA and FE models. The last step in setting up the vibroacoustic model is to apply a diffuse acoustic field (DAF) excitation to the sandwich panel. The model is then solved in 1/42-octave bands (narrow band) and the spatially-averaged vibroacoustic acceleration response is recovered; it is subsequently converted to 1/3-octave bands. An SEA model of the same panel is created by copying this model, including the dummy panels, and converting the FE subsystem into an SEA subsystem and updating the properties to a reasonable "initial guess" value. This model is shown in Figure 4(b). The SEA version of the model is solved in 1/3-octave bands.



(a) (b)

Figure 4. Vibroacoustic models of the sandwich panel in (a) FE subsystem and (b) SEA subsystem representations.

The goal at this point is to match the response of the SEA subsystem representation of the sandwich panel to the high-fidelity FE subsystem representation. The initial comparison of the responses is shown in Figure 5. While the general trend of the curves is reasonably close, it is clear that the SEA subsystem is underpredicting the vibroacoustic response above 800 Hz. We will now use Isight to improve this correlation.

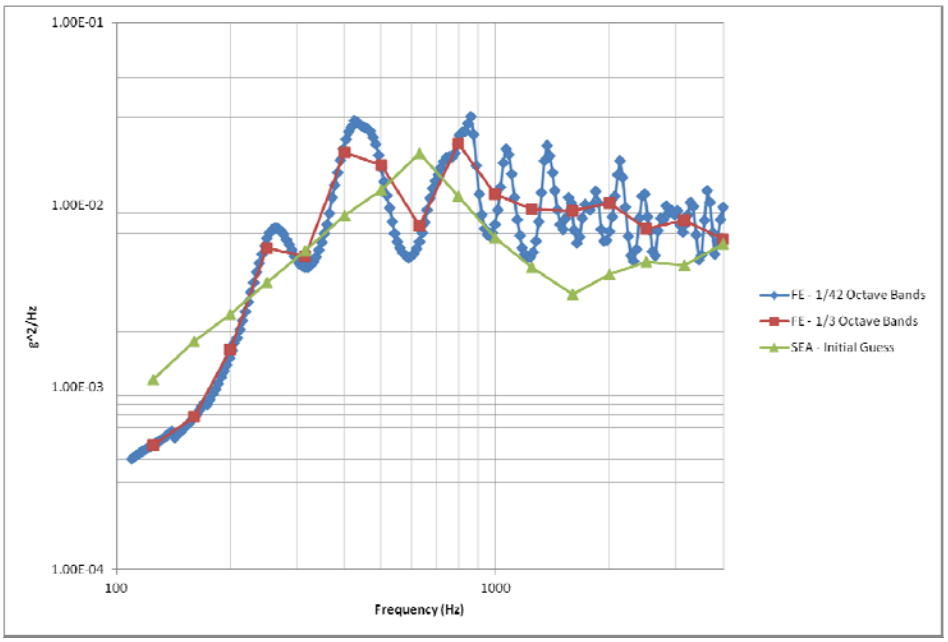


Figure 5. Initial comparison between FE subsystem and SEA subsystem sandwich panel vibroacoustic responses.

The procedure described in Section 2 was used to improve the correlation between the FE and SEA subsystem sandwich panel responses. The SEA panel properties that were varied (i.e., the design variables) include the material density and thickness of both the core and the facesheets as well as the panel radius. Two optimization algorithms were tested: the modified method of feasible directions (MMFD) and the downhill simplex (DS). MMFD is a gradient-based algorithm, while DS is a simplex-based exploratory algorithm. Both algorithms work well for the continuous, well-behaved optimization landscape that is expected for this problem. The default algorithm parameters are used for both of these problems. The results of these optimization problems are shown in Figure 6. Both algorithms showed improvement in the correlation to the FE subsystem, with MMFD using more function evaluations than DS (89 and 64, respectively), but this algorithm also had a slightly better objective function and could have been stopped much earlier (at ~40 iterations) with little change in objective if a better convergence criterion had been used. Table 1 shows the structural parameter values (the design variables) in the SEA panel for the initial guess and after both optimization runs. In general, the optimal property values are similar for both algorithms relative to the initial guess, indicating that they found the same local optimum.

Overall, both algorithms performed equally well for this task. The low-frequency response (below ~600 Hz) is largely unchanged because this is a region where the number of modes-in-band for both the FE and SEA subsystems are getting too low for the SEA method to be valid. In these regions, VA One approximates the response with analytical relationships that are unable to conform to the variation in the FE response. This behavior is expected.

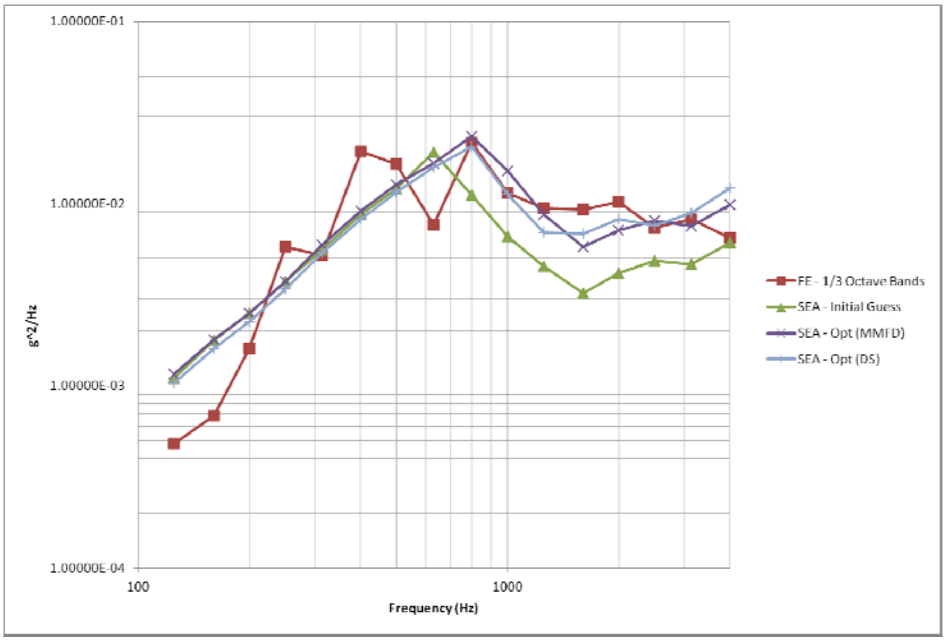


Figure 6. Comparison of sandwich panel vibroacoustic responses before and after optimization using Isight show improved correlation.

Table 1. SEA sandwich panel properties before and after optimization.

Sandwich panel property	Initial guess	Opt – MMFD	Opt – DS
Core thickness (in)	0.75	0.54	0.47
Core density (lbm/in^3)	0.0018	0.0016	0.0017
Facesheet thickness (in)	0.04	0.042	0.050
Facesheet density (lbm/in^3)	0.0654	0.0623	0.0706
Radius of curvature (in)	46.75	39.37	40.39

3.2 Cylindrical ribbed panel section

The next example consists of a complex cylindrical ribbed panel that illustrates how difficult it is to correlate an SEA panel subsystem to an FE representation. The ribbed panel has several features found in typical panels in aerospace structures including varying skin and rib thicknesses as well as a hole. These features make defining SEA panel properties a very difficult and error-prone task.

The sample ribbed panel consists of a varying-thickness facesheet with varying-thickness ribs spaced 4.5 inches apart in each direction. The panel has a 50-inch radius of curvature, it is 36 inches high, and it spans 60 degrees. A hole of about 115 in^2 is located off-center in the panel. Both the facesheet and ribs are made of titanium. The FE model of the panel, created using shell elements, is shown in Figure 7.

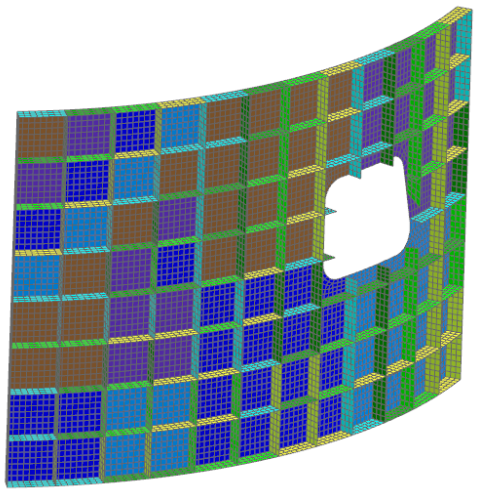
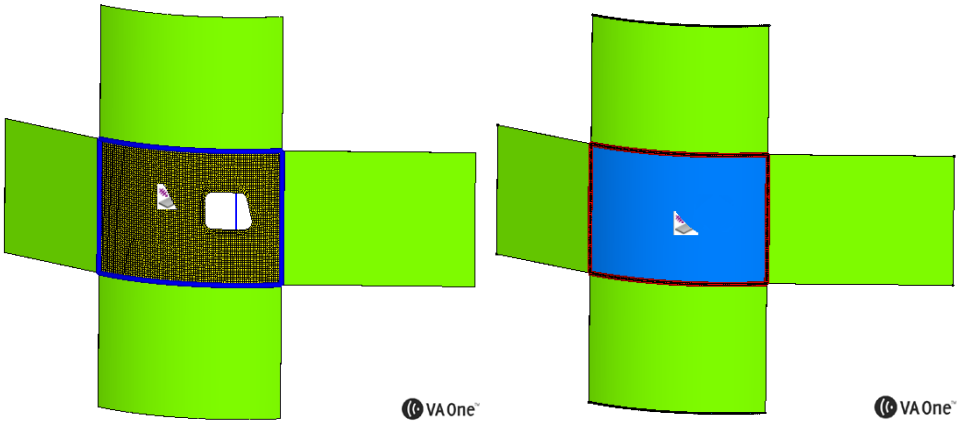


Figure 7. FE model of the cylindrical ribbed panel section – example shown with thickness properties varying by element color.

The setup procedure for this example is similar to the previous problem. Free-free modes of the panel were calculated and imported along with the model into VA One. In all, 368 modes were calculated up to 3,700 Hz. This covered up to the 3,150 Hz 1/3-octave band. The FE subsystem was again connected to “dummy” SEA panel subsystems, and a corresponding SEA version of the panel was created with “initial guess” properties. The initial guess properties were calculated via an area average of all the properties in the panel (e.g., average rib thickness of all the ribs). In the SEA subsystem, the hole is also removed. DAF loading was then applied to the panels as shown in Figure 8. Again, the FE model is solved in 1/42-octave bands, the results are reduced to 1/3 octave, and the SEA model is solved in 1/3-octave bands. The initial comparison of the responses is shown in Figure 9. It is clear that the response of the SEA subsystem does not match the response of the FE subsystem, especially at higher frequencies. Unfortunately, it is unclear how to improve the correlation based on the available SEA subsystem properties.



(a) (b)

Figure 8. Vibroacoustic models of the ribbed panel in (a) FE subsystem and (b) SEA subsystem representations.

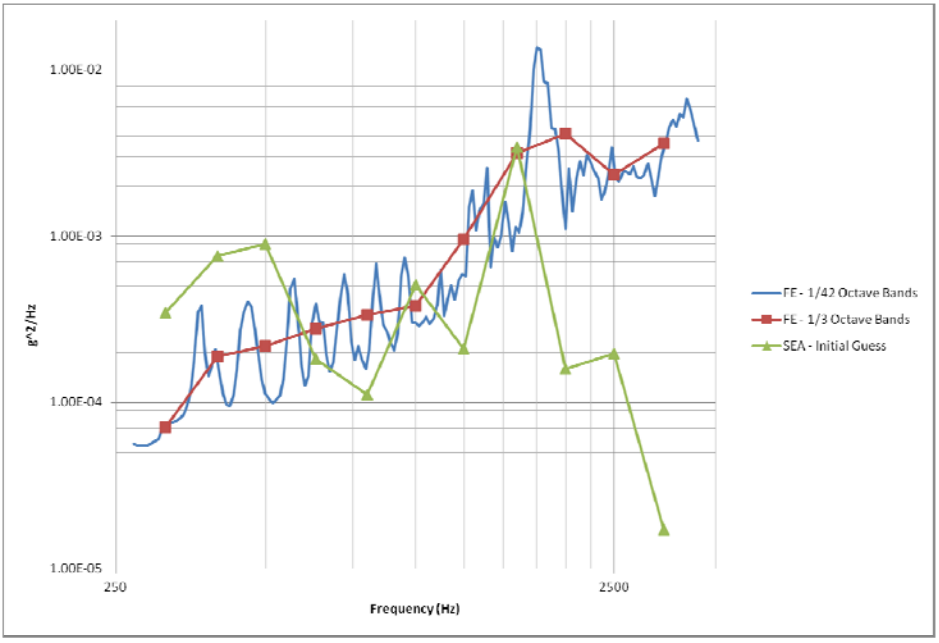


Figure 9. Initial comparison between FE subsystem and SEA subsystem ribbed panel vibroacoustic responses.

A similar procedure to that used in the previous example was applied to improve the correlation between the two models. In this case, the SEA properties that were varied were the material density of the titanium, the offset of the ribs from the facesheet, the spacing between the ribs, and

the facesheet thickness. The primary difficulty in this problem is derived from the method with which VA One calculates the vibroacoustic response for ribbed panels. For this type of panel, VA One uses an internal modal solution to compute the vibroacoustic response so that contributions from the ribs are included. As a result, the response quantities of the SEA panel are not a smooth function of the physical parameters. Because of this, it is necessary to use non-gradient-based optimization algorithms within Isight to solve the correlation problem. Again, two optimization algorithms were tested: a multi-island genetic algorithm (MIGA) and the Pointer method. Both algorithms work well with the discontinuous optimization landscape present in this problem. In the MIGA, the number of islands was reduced to five, while all other parameters were left at their default values. The Pointer algorithm, which uses a combination of several different types of algorithms, has an allowable job time parameter that was set to 1 hour, with all other parameters untouched.

An important aspect of the formulation for this problem that differs from the previous example is the penalization of zero modes-in-band (MIB) responses from the SEA model. A ribbed SEA panel may have zero MIB if the simple closed-form solution indicates that there are no eigenvalues within a given 1/3-octave band. The result of this is zero response in that band, which is clearly incorrect (as illustrated by the response of the FE subsystem). In the formulation of the optimization problem, we want to penalize the occurrence of zero-MIB responses so the optimizer will stay away from them. To do this, a two-pronged approach is taken. First, if the response of the SEA panel is zero in a given band (indicating a zero MIB solution), the response of that band is redefined at a value approximately an order of magnitude above the highest response of the FE subsystem. Second, a constraint is placed on the response of each band that is slightly lower than this redefined response, thereby indicating that such a design point is infeasible. These two factors together (a higher objective value and an infeasible design point) help the optimizer avoid zero-MIB solutions.

The solutions provided by these two algorithms for this correlation problem are shown in Figure 10. Both algorithms showed significant improvement in the correlation to the FE subsystem. The high-frequency correlation of the solutions is quite good, while the mid-frequency correlation is maintained. The discrepancy at low frequency occurs in a region with few MIB and is not as important as good correlation at high frequencies (hence the higher weighting in the objective function for high frequency results). The MIGA algorithm used 501 function evaluations, while the Pointer algorithm used 692 function evaluations. Both algorithms could likely have used fewer iterations, but as with many exploratory-type algorithms such as these, it is difficult to know how many iterations are necessary beforehand. SEA panel properties before and after optimization are provided in Table 2. The design variable values (i.e., panel properties) found by the two algorithms are quite different, which is an indication of multiple local optima. This behavior is expected and is one of the primary reasons why the presented optimization algorithms were chosen. Overall, both algorithms provided improved correlations for this problem and took much less effort to find a good solution than it would have taken for a manual guess-and-check approach.

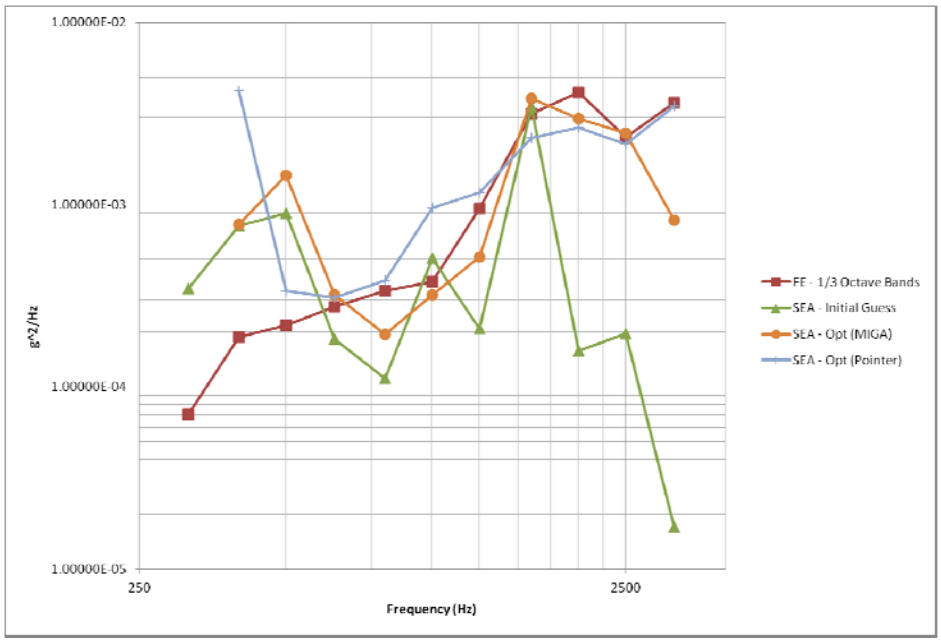


Figure 10. Comparison of ribbed panel vibroacoustic responses before and after optimization using Isight showing improved correlation.

Table 2. SEA ribbed panel properties before and after optimization.

Ribbed panel property	Initial guess	Opt – Pointer	Opt – MIGA
Panel skin density (lbm/in³)	0.16	0.17	0.14
Panel skin thickness (in)	0.17	0.14	0.16
Rib 1 spacing (in)	4.5	7.05	3.32
Rib 1 offset (in)	0.75	1.83	2.76
Rib 2 spacing (in)	4.5	11.98	38.77
Rib 2 offset (in)	0.75	0.17	0.09

4. Other potential use cases for Isight in vibroacoustic analysis

This work presents the start of an exciting new frontier in engineering design with vibroacoustic analysis. Potential future applications of the presented methodology include its incorporation into multidisciplinary design optimization where vibroacoustic analysis is an integral part of the solution. A tool does not currently exist that can perform such an analysis, but in combining Isight with VA One this may be possible. An example would be to have design parameters in Isight update an FE model used for static, random vibration, and other analyses. The methodology

presented here would then correlate the SEA model to a new FE model definition in a secondary loop within Isight. Such a setup would enable optimal design within a vibroacoustic setting.

Additional future work related to the methodology presented here may include sensitivity analysis of SEA responses due to a variety of input factors. Such input factors may include the input parameters for environment definitions, thereby revealing the range of responses that may result from uncertainty in the environment. Another option would be to find the sensitivity of cabin noise levels or panel responses to structural inputs such as panel stiffness or mass factors. Such sensitivity information would be useful in guiding design efforts to create lighter, safer, and more reliable structures.

5. Conclusions

We have shown through the examples that Isight can be effectively used to improve the correlation between SEA and FE vibroacoustic models. Using our methodology, the effort necessary to complete any panel correlation project can be cut from up to a day of manual guess-and-check work down to about half an hour of setup time plus an hour of unattended solve time. This represents a significant reduction in the amount of engineering effort and underlying vibroacoustics knowledge that are necessary to carry out such a correlation, thereby reducing costs and resulting in a more accurate SEA model.

6. Appendix A

The following is a Matlab script that uses the VA One API to update vibroacoustic model parameters, solve the model, store the response, and compute an objective function. This script is executed within the Matlab module in Isight. This script was used for the first example problem.

```
% The following commands will open a VA-1 database, modify a panel's material
property and recover the response of the panel.

% On first run store data that does not change
if ~exist('started','var')
    % Open the VA-One database
    filename = ‘sandwich_panel_cone_SEA.va1';
    db=va1_open_db(filename);

    % Get the pointer to the panel of interest
    plate_name = 'SEA sandwich panel';
    plate_ptr = pi_fNeoDatabaseFindByName(db,pi_fPlateGetClassID,plate_name);
    spec_ptr = pi_fResultsGetSpectralFunction(2,plate_ptr,2);

    % Get the list of solution frequencies and associated data
    sim_freqs = va1_get_freqs(db);
    nfreq = length(sim_freqs);
    Noct = 3; % Assume 1/3 octave bands
    bws = sim_freqs .* (2^(1/2/Noct) - 2^(-1/2/Noct));

    % Get pointers to property data
    propref = pi_fPlateGetSection(ref);
    cmref = pi_fSandwichXSectionGetCoreMaterial(propref);
    smref = pi_fSandwichXSectionGetSkinMaterial(propref);
```

```
    % Import the target data
    target = importdata('sandwich_panel_cone_target_data.txt',' ');
    target_freqs = target(:,1);
    target_response = target(:,2);
    target_responseL10 = log10(target_response);

    % Set weights for objective function
    weight = (1:length(freq))';

    % Indicate end of first run
    started = 1;
end

% Set parameters from design variables

% Curvature
pi_fPlateSetCylinderOverrideCurvature(plate_ptr,1/panelradius);

% Skin
pi_fSandwichXSectionSetSkinThickness(propref, skinthickness);
pi_fIsotropicSolidSetDensity(smref, skinmatdens);

% Core
pi_fSandwichXSectionSetCoreThickness(propref, corethickness);
pi_fRealOrthotropicSolidSetDensity(cmref, corematdens);

% Solve & save
pi_fDatabaseSolve(db);
va1_save_db(db);

% Get responses
sim_response=zeros(nfreq,1);
for j=1:nfreq
    sim_response(j)=double(pi_fFloatSpectralFunctionGetValSI(spec_ptr,j-1));
end

% Convert to g^2/Hz
response = (response .* 2*pi.* sim_freqs / 9.81).^2 ./ bws; % Flexural
sim_responseL10 = log10(sim_response);

% Correction for null response due to zero MIB in simulation
ind = sim_response==0;
sim_response(ind)=1;

% Compute objective function
sqdiff = (log10(sim_response)-log10(target_response)).^2;
objective = sum(sqdiff.*weight);
```