Sound Transmission Loss predictions of aircraft panels: an update on recent technology evolutions

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ABSTRACT

In view of optimizing aircraft structural panels, that is, in view of finding the right balance between weight and noise reduction, simulation technologies are of significant value. The prime technologies in this context are based on Finite Element Methods for the low to mid frequency range and Statistical Energy Analysis for the mid to high frequency range. The continuous improvements of these technologies allow today to simulate noise transmission through complex panels not only with good accuracy but also efficiently, across the entire frequency range of interest. This paper will address the latest improvements for simulating sound transmission loss. The aircraft panels taken into consideration in this paper are real-life industrial panels, including outer panel, inner trim, structural connectors, structural damping, acoustics & thermal insulation packages etc. The modeling process, possibilities and assumptions will be highlighted and discussed, including comparison between different modeling approaches. To the extent possible, simulation results are compared with measured data.

Keywords: Sound, Insulation, Transmission

1. INTRODUCTION

Efficient interior acoustic design of vehicles and aircrafts in particular requires efficient and precise methods for vibro-acoustic simulation. This means the implementation of several material models comprising linear fluids and isotropic- and anisotropic solids, as far as porous and viscoelastic material with strong damping and frequency dependent complex properties. Deterministic approaches like finite element methods (FEM) must deal with different mesh sizes due to different order of magnitudes for the wave lengths, whereas statistical energy methods (SEA) must handle complex systems in a realistic manner. Thus, several specific features must be implemented efficiently in a software environment to cope with the challenges of realistic systems.

The most important barrier separating the exterior excitation from the cabin inside is the aircraft sidewall consisting of the primary fuselage structure made out of aluminum or carbon fibre reinforced plastics (CFRP) and the secondary structure, i.e. cabin lining for decoration and additional acoustic and thermal isolation. This setup constitutes a so called double wall system. A useful criterion for evaluating the performance of such a system is the sound transmission loss (STL). The determination of the STL requires the implementation of elements that can handle radiation into the acoustic half space.

This paper deals with the FEM and SEA simulation of typical aircraft sidewalls including all above...
mentioned challenges. For method evaluation and benchmarking this paper starts with a simplified double wall model that includes all features of an aircraft sidewall but allows for fast simulation. Here, the selection of diffuse sound field realization, coupling formulation and the design of FEM-based acoustic radiation methods are investigated in detail. For SEA the model serves as test case for various implementations of the so called non-resonant paths and different approaches to model the double wall effect in an SEA model.

The paper concludes with a real size 2m x 2m model of an aircraft sidewall proving that the described methods can deal with a realistic structure including all the details like skin, stiffeners, windows, lining, fiber isolation, shock-mounts, sandwich panels, etc. in reasonable time.

2. SOUND TRANSMISSION LOSS SIMULATION WITH FEM

2.1 Introduction

The main advantage of using FEM for simulating Sound Transmission Loss (STL) of panels (aircraft and others) is related to the fact that one can deal with complex, sophisticated layouts of panels, taking into account curvatures, structural connections, local damping, corrugations and windows etc. all of which are important in the context of simulating noise inside an aircraft. Being able to define such geometrical details allows predicting Sound Transmission Loss more accurately compared to analytical methods with their inherent assumptions. On the downside for FEM, one can note that these methods take potentially significantly more CPU time solving larger systems and also the meshing and modelling can be time consuming especially for complex panels. In recent times, there have been substantial improvements on these two matters, both related to hardware and software improvements. Whereas a simulation of the STL of a typical aircraft panel would take ‘days’ just a decade ago, today, full FEM based calculations can be done in hours down to even minutes for small panels. This is especially true when all aspects of the numerical simulation process are carefully optimized. The usage of advanced direct solvers such as MUMPS, fully optimized for multi-CPU's, usage of latest technologies for exterior radiation with FEM and more, allow reducing computation times significantly for FEM-based analyses.

The paragraphs below address different aspects of the FEM-based STL simulation based on a simplified but representative test case and can be used as guidelines. The model which is considered for these analyses is a double wall panel configuration with glass wool and 2 air gaps between the panels as schematically illustrated below. The panel measures 1100 x 800 mm and is simply supported. All materials have 2% damping.

![Figure 1: double wall panel lay-out](image)

The glass wool is modelled by an equivalent fluid material with Limp frame formulation with the following properties: Flow Resistivity: 20 000 rayls, Porosity: 0.9964; tortuosity: 1; viscous length: 0.137 mm; thermal length: 0.274 mm.

The Trim panel is a sandwich panel composed of a 5mm core (32kg/m^3 and E-modulus 5.2e7 Pa) and 2 sheets of 0.2 mm (2000 kg/m^3 and E-modulus 2.9e10 Pa) covering the core.

For the FEM analyses, the aluminum skin panel is modelled by linear shell elements. Both the skin and the trim panel are considered simply supported. The interior trim panel is modelled by a combination of shells for the skin of the sandwich and a solid part of 5mm for the core. Air gaps and glass wool are modelled by 3D fluid elements.

The FEM model is shown below and contains 23kNodes and 22kElements modelled as QUAD4 and HEXA8 elements.

2.2 Plane wave approach

For simulating a diffuse field Sound Transmission loss, a standard approach is based on exciting
the panel by a set of uncorrelated plane waves.

![Figure 2: Uncorrelated plane waves representing a diffuse field.](image)

As the excitations are uncorrelated, they can be analyzed independently (in terms of uncorrelated, different load conditions) and results can be summed energetically in terms of power. The STL is then readily given by the formula:

\[
STL = 10 \log_{10} \left( \frac{P_{\text{incident}}}{P_{\text{transmitted}}} \right)
\]  

(1)

For FEM approaches, the incident power is directly deduced from the given amplitude of the incident waves. The transmitted power is obtained from the results of the FEM analyses. Typically this is done by analyzing the total sound power radiated directly by the panel in the receiving room.

The incident ‘diffuse’ field room is not explicitly modelled. The set of plane waves applied directly to the panel represent a good approximation of a diffuse field. At the receiving side, the anechoic room is not explicitly modelled either. A PML boundary condition is applied at the receiving side as explained in the next paragraph.

One aspect to take into account when dealing with STL calculations is the number of plane waves to take into account. A single wave will not be able to represent diffuse field behavior. On the other hand, too many plane waves result in more computational resources, slowing down the calculations. Through experience, for aircraft panels, we consider 12 plane waves – randomly distributed - to be a good guideline, compromise between accuracy and CPU time. The picture below shows the STL for different analyses considering different plane waves.

![Figure 3: Effect of the number of uncorrelated, distributed plane waves on STL results.](image)

As can be noticed from this graph, the ‘normal wave STL’ does differ from the diffuse field STL. Considering only 3 plane waves is not sufficient for a good diffuse field performance. From typically 12 plane waves onwards, the results converge. For further calculations, we can consider the nominal condition of 12 plane waves, sufficient to represent a diffuse field for the frequency range of interest. Calculation times in the case with 12 plane waves (nominal case) and for 21 frequencies, is approximately 3 minutes on a 20CORE system.

### 2.3 Incident and Receiving acoustic field

When applying a diffuse field represented by a set of plane waves onto the panel (Aluminum skin), it is needed to consider an acoustic mesh at the incident side of the panel, essentially representing the domain in which the diffuse field will act. The ‘extend’ of this acoustic mesh and the ‘boundary condition’ of this acoustic mesh should be carefully considered regarding the overall simulation performance. An incident field acoustic mesh that is very large will result in long CPU times. A technique called ‘AML’ (developed by SIEMENS PLM Software and based on the PML technology) allows considering very small acoustic meshes both for the incident side as well as for the receiving side. For the incident side – which is mostly convex for real life aircraft panels (single curved at the
outside) – one typically only need to consider ‘ONE’ layer of acoustic elements.

Two elegant approaches are often used in FEM acoustics to obtain the anechoic boundary condition: using Infinite Wave Envelope Elements (IFEM) or using a Perfectly Matched Layer (PML) technology at the outer FEM boundary. These techniques allow for this boundary to be much closer to the scattering object, resulting in smaller models and therefore smaller computational efforts. For standard PML, one has to create explicitly the absorbing layer. The thickness of the PML layer should be sufficient large in order to absorb the acoustics waves to a sufficient extend. This minimal thickness of the PML layer does depend on the frequency of analysis as it depends on the wavelength. SIEMENS PLM Software developed a technology called AML, which is based on PML and essentially does not allow to explicitly model the PML layer. The solver itself will ‘virtually’ construct the PML layer, with a dimension optimized for each particular frequency of analysis, resulting in significantly lower computation times [1], [2].

Specifically for the double wall panel, one should consider 2 separate FEM AML domains: the incident side and the receiving side, colored ‘pink’ in the below picture. By applying such an AML boundary condition, one can accurately obtain the radiation efficiency and joint acceptance of the 2 sides of the double wall panel. Even with a very small FEM mesh at incident and receiving side, one can predict well the input, transmitted and radiated sound from the panel. To illustrate the fact that one should only consider a very limited FEM AML mesh, we created 2 models: one model incorporates a small FEM AML mesh, the other model incorporates a more extensive FEM AML mesh as shown below:

In the left model, the exterior FEM domains, bounded by an AML boundary condition (colored Pink) are limited in size. The model on the right shows more extensive FEM domains with AML boundary conditions (colored pink). The green plane shown in the model on the right side represents an infinite reflecting plane, referred to as baffle. From computational robustness perspective, the larger model could be considered as more stable and accurate due to the larger exterior FEM domains. Despite, when considering the STL curves, it can be noted that both models show similar STL curves. The conclusion is that for such a double wall panel, one should not necessarily have large exterior FEM domains. Even with just one layer of elements and with the AML boundary condition applied to the free exterior face, one can already obtain good convergence of the STL. The ultimate benefit is that the global Double Wall Panel model can be reduced significantly in size. The exterior fluid FEM domains add only a limited number of degrees of freedom (DOF), thereby improving the efficiency of the calculation. In other words, calculating accurately the radiation of such panels from numerical models goes at a marginal, incremental cost.
Figure 6: STL of the double wall comparing two Exterior Acoustic FEM models
NOTE: When considering curvature effects the exterior FEM mesh at the incident side (outside of aircraft) can also be modelled by just one or two element layers because of the convexity. However at the receiving side, the shape is concave and therefore more elements are required to fill the space of the concave interior surface.

2.4 Direct versus modal-based FEM approaches

Both ‘direct’ and ‘modal’ approaches are relevant for STL simulations. However, depending upon the scope and characteristics of the study, one approach is to be preferred, whether for accuracy reasons or for efficiency reasons. For example, if one considers low frequency transmission loss where the number of modes will be limited and if one considers global structural damping then most likely modal superposition techniques will be preferred because they are the most efficient ones. However, for mid to higher frequencies with an increased number of modes direct (sometimes referred to as ‘physical’) approaches are most applicable. Also the direct approach allows taking into account local, frequency dependent damping such as visco-elastic damping treatment (see picture below) and porous materials such as often used in aircrafts for the acoustic and thermal insulation packs (see picture below).

Figure 7: Acoustic (LHS) and damping (RHS) treatment of panel
For the double wall panel considered before, in the frequency range 20-2000Hz, the difference between the direct and the modal approach is negligible because the damping is considered constant and global. Note that for the modal superposition approach, one can actually take modes up to 2000Hz, and still have acceptable STL results up to 2000Hz.

Figure 8: Model versus direct approach in view of STL prediction
For the direct calculation, the problem is solved by the SYSNOISE Vibro-Acoustic solver, which integrates both an advanced structural element library and an acoustic library including AML. For the modal-based solution, the solution is split between 2 solvers: NX NASTRAN for the structural modes using SOL103 (for the aluminum skin panel and the trim panel) and the SYSNOISE Acoustic FEM solver for the acoustic domains (interior as well as exterior) and for the coupling with the structure.
2.5 Fully coupled versus weakly coupled approach

With the ultimate objective to reduce calculation time of the numerical analyses, it could be interesting to consider weakly coupled formulation in contrast to fully coupled simulations. More specifically for the double wall considered here, the coupling between the 2 mm aluminum panel to the air outside and the coupling between the trim sandwich panel and the receiving room could potentially be considered as weak and as such resulting in a series of smaller analyses and hence overall faster calculation times. Obviously, the (inside) air gaps and the panels (aluminum and the trim) needs to be fully coupled as that is the only energy transmission path from source to receiver.

![Figure 9: Panel layout including exterior acoustic cavities](image)

Using weakly coupled formulations to couple the panels with the outside air is more interesting from the computational perspective, yet accuracy could be lower. The main question is therefore on the quantification of the effect of weak coupling versus strong coupling in view of accuracy.

An investigation has been carried out, analyzing the sound power through the double wall panel for the following 2 cases:

- Fully coupled case: all acoustic domains (incident, air gaps, glass wool and receiving) are fully coupled with the structural parts. The full set of equations in the physical domain is solved using a direct solver.
- Weakly coupled case: in this case at first, the blocked pressure is calculated onto the aluminum panel resulting from the plane waves (12 plane waves are considered). Then, these blocked pressures are applied to the aluminum skin panel of the double wall configuration. For this calculation, the outside air (incident and receiving) are not considered. From this 2nd calculation, the vibrations of the trim panel are obtained. Finally, in a 3rd step, these vibrations of the trim panel at the receiving side are used to calculate the sound power from the trim panel using FEM AML technology in the receiving room. Schematically this gives for the weakly coupled case the following process steps:

**Step 1**: calculation of the blocked pressure on the aluminum plate. Right: an example, resulting image of the blocked pressure calculation.

**Step 2**: calculation of the vibrations of the double wall due to the blocked pressure. Right: an example, resulting image of the vibration shape of the trim panel.
Step 3: calculation of the radiated sound power by the trim panel from the given vibrations of step 2. On the left side: FEM AML acoustic mesh (just one single layer of elements). On the right side: acoustic radiation result.

At the excitation side (aluminum skin panel), the vibrations in a few reference points is shown below, comparing the fully coupled case and the results from step 2 of the weakly coupled case.

![Vibration results of incident panel for weakly versus fully coupled case](image1)

This result shows that a relatively small error is made of about 3dB max in the lower frequency range. The effect of the weakly versus fully coupled simulation is less apparent in the higher frequency range, above 500Hz.

At the receiving side, a similar graph is made, showing a typical vibration result for the weakly coupled case versus strongly coupled case:

![Vibration results for receiving side between weakly coupled and fully coupled case](image2)

This vibration result shows that the effect at the receiving side of the fully coupled versus weakly coupled case is relatively small yet somewhat apparent in the entire frequency range. Again, the lower frequencies show the highest effect.

Comparing the radiated Sound Power (or transmitted power) from the plane waves, for both the fully and the weakly coupled case, this gives:

![Transmitted sound power comparing weakly versus strongly coupled case](image3)
Also for the transmitted output power, the difference between weakly and fully coupled simulations range from 1dB to 7dB. Conclusion: the dynamic effect of the air at the incident and at the receiving side should not be neglected. Therefore, a fully coupled analysis is desirable for similar cases.

3. SOUND TRANSMISSION LOSS SIMULATION WITH SEA

3.1 Introduction

Sound transmission loss modelling based on SEA has the particular advantage of efficiency (fast simulations) and the possibility to address a wider frequency band, typically from the first eigenmodes onwards to high frequencies. SEA-based STL has proven to be accurate, provided the SEA modelling is properly done for the particular structure. While as with FEM, an engineer mostly focuses on the geometry and meshing, the situation is different for SEA. For SEA, it is more about modelling the physics. A proper understanding of the key physical phenomena is essential in order to obtain acceptable, valuable results. In the following paragraphs, 2 key modelling aspects will be highlighted, namely the non-resonant path and the modelling using Transfer Matrix Method, referred to as TMM.

3.2 Mass Law

Considering sound transmission through a flat panel, below critical frequency, the STL is dominated by a Mass Law curve, corresponding to the non-resonant energy path between the incident and receiving room. Above the critical frequency, the STL is dominated by the resonant path as shown in the picture below.

![Figure 13: Sound Power radiated by the TRIM panel showing contributions of mass law and resonant path](image)

The mass dominated path (blue curve) is often referred to as the indirect path, because the energy exchange between the 2 rooms happens indirectly though non-resonant motion of the partition. Such phenomenon does not only occur for Sound Transmission through panels but also happens between pure structural connections. While as the indirect path in the case of a simple partition is easy to formulate (straight ‘mass’ line in blue in the graph above), this is not necessarily the case for complex partitions (such as double wall panels) or structurally connected subsystems. In an effort to generalize the indirect connections and hence extending the scope of the indirect transmission from standard mass law to generalized connections, the concept of non-resonant energy of subsystems is introduced. Conceptually it means that when a partition is connected to 2 rooms, the energy of the plate radiating into either cavity is the combination of the resonant energy, the non-resonant energy from the plate modes excited above their eigenfrequency (mass controlled) and the non-resonant energy from the plate modes excited below their eigenfrequency (stiffness controlled). Essentially the plate has 3 distinct energies in the frequency band of interest: the resonant energy, mass controlled non-resonant energy and stiffness controlled non-resonant energy. In standard SEA, only the mass law is typically considered and mostly and only for panel transmission. This is simplified to a straight line (mass law line). A more general formulation is where the 3 types of energies are considered through calculating also the effective radiation efficiencies of these off-resonant excited modes as opposed to approximating the mass law by a single straight line. Not only does this approach provide higher accuracy but also a major benefit is that the indirect connections should not be modelled explicitly as they are implicitly taken into account through the non-resonant energies of the subsystems.
Figure 14: Left side: standard SEA model with explicit modeling of the indirect path (colored in red). Right side: The non-resonant paths are automatically taken into account. No need to model the indirect path.

For a single panel, the STL calculated by Standard Mass law versus the new approach based on non-resonant energies is shown below. Both models match fairly well.

Figure 15: STL curves comparing mass law with new approach based on non-resonant energies.

In the case of a double wall panel, the new approach based on the ‘non-resonant energies of the subsystems’ is preferred as it avoids explicit modelling the ‘several’ indirect paths. In particular, for curved panels, it is a preferred technique. Due to curvature, their non-resonant mass behavior not only becomes dependent on mass density but also on both modal density and radiation efficiency of all non-resonant modes in frequency bands below critical frequency. The SEA+ non-resonant energy approach simulates this behavior without any extra CPU load due to vectorization of the radiation efficiency algorithm as radiation efficiency of all modes below the analyzed frequency band need to be computed.

3.3 Transfer Matrix Method

The TMM has been applied for panel transmission for quite some time and has proven its usefulness. Despite, there are assumptions with the TMM. The most stringent assumptions are dealing with the fact that TMM assumes infinitely extended domains and also in its general formulation, curvature remains a tough topic. The fact that the domain is considered ‘infinite’ implies that e.g. some modal effects in the cavities are not properly taken into account.

Consider the case of the double wall panel with properties given earlier in this paper. There are a variety of options to choose from in order to create an SEA model, of which we consider 3 possible models:

- **Model 1** (shown below): the entire partition (ALU panel, glass wool, air gap, trim) represents ‘one’ Transfer Matrix (one TMM). In other words, this model does not incorporate structural subsystems as such.

- **Model 2** (shown below): consider the Aluminum skin as a subsystem. The air gap, glass wool, trim are modelled as TMM:
Model 3 (shown below): consider the trim panel as a subsystem. The air gap, glass wool and ALU panel are then modelled as TMM.

SEA is not an exact science and all these models carry assumptions with them, more particular related to the TMM modelling.

The results in terms of STL and obtained by the SEA+ software from InterAC are shown hereafter for these different models.

Figure 16: Comparing STL for different SEA model

Which of these models is to be preferred is cannot be easily answered. On the one hand, one could say that modelling the entire double wall as a TMM, carries most assumptions, approximations with it (because ‘infinite’ by default). On the other hand, modelling the trim panel (a lightweight sandwich panel) as a separate subsystem (model 3) is advisable because the trim panel has the lowest model density and also has a coincidence frequency of roughly 2000Hz, much lower than the Aluminum skin panel. As such, from SEA perspective, the trim panel is the most ‘critical one’ and modelling the interior trim sandwich panel as a separate subsystem is therefore favored.

Finally, a comparison is made between the STL calculated by FEM (using LMS Virtual.Lab Acoustics from SIEMENS) and SEA (using SEA+ from InterAC). From the previous SEA Models, the Model 3 (trim as separate subsystem) is taken and compared to the FEM-based STLs (see before).

Figure 17: Comparing STL between SEA-based FEM-based analyses

These results show that the FEM-based STL prediction corresponds quite well with the SEA-based STL for Model 3 (Trim panel as subsystem). In the very low frequency range (below 50Hz), the difference between the FEM-based STL and the SEA-based STL is larger due to the low number of modes in this frequency and therefore the difference is quite understandable.
4. AIRCRAFT PANEL TL

A real-life aircraft panel section is considered for the following set of simulations. The panel incorporates all key ingredients to make it realistic: outer skin, windows, inner trim, structural connections, glass wool (porous materials) etc. For such a study, an FEM-based approach is preferred. The frequency range extends to 2kHz (high number of modes) and some materials incorporate frequency dependent properties (such as the glass wool), a direct solver should be taken therefore. Finally, as indicated above, for such calculations, it is better to use a fully coupled approach (2 way coupling) where all acoustic domains are fully coupled with all relevant structural parts.

LMS Virtual.Lab Acoustics is used for this study as it embeds all of the previous and can address these requirements in ‘one single vibro-acoustic FE model and solution sequence’.

The original structural FEM model (NASTRAN format) was extended to include the acoustic relevant domains such as the double wall cavity, the air between the window partitions and also the air representing the aircraft interior and exterior as well as the underfloor cavity. The exterior air domains also carry an AML boundary condition to allow for proper ‘infinite’ radiation and scattering, avoiding spurious reflections back to the panel.

Figure 1:  Left: Original FEM model    Right: with added acoustic (FEM and AML) domains.

Model characteristics: 50k structural nodes and 212k acoustic nodes. A diffuse field modelled through a set of 12 plane waves is applied to the outer skin (represented by the hemisphere below).

Figure 19: Application of 12 plane waves simulating diffuse field onto the aircraft panel

The glass wool within the double wall cavity is provided as frequency dependent, complex speed of sound and densities, calculated through an equivalent fluid model with Limp formulation for the frame. Characteristics of the glass wool are: Flow resistivity (20 000 rayls), Tortuosity (1), Porosity (0.964), Bulk density air and fiber (6.9 kg/m³), Viscous length (0.137 mm), thermal length (0.274 mm).

The model is solved by the Virtual.Lab Vibro-Acoustic solver (embedding SYSNOISE Acoustic FE technology and SAMCEF structural FE technology) using the Direct MUMPS technology for all 1/12 octave frequencies between 100 and 2000Hz. This solver is optimized for working with multi-CPU systems using a combination of frequency and matrix parallelization resulting in reduced calculation times. For this case, a 20 CORE workstation equipped with 128GB RAM is used taking 40sec / frequency line resulting in a total of 35 minutes to get the complete STL curve of the panel. As the solution is linear, and perfect to run in parallel, one can reduce computation time even further in case of availability of high performant hardware.

The resulting Transmission Loss is directly provided by Virtual.Lab Acoustics and shown below, comparing with the measured data. For reasons of confidentiality, absolute scaling cannot be given. See also [3] for further data and results discussion on this particular panel. NOTE that: Actually it is supposed that the lower part of the panel in the tests were not perfectly isolated which leads to bad high frequency results. However, this must be further investigated.
5. CONCLUSION

The STL of typical aircraft sidewalls under diffuse sound field excitation can be precisely and efficiently modelled using Virtual.Lab Vibro-Acoustic solver up to 2 kHz. The software masters the several challenges of vibro-acoustic simulation comprising excitation, various material formulations and the radiation into infinite space. Based on such a model hot spots can be identified and design optimization of the primary and secondary structure is possible.

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REFERENCES