# ACTRAN – POWERFUL VIBRO-ACOUSTIC FINITE ELEMENT SOFTWARE

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**Abstract**

Actran is finite element software which can be used for modelling sound propagation, transmission and absorption in an acoustic, vibro-acoustic, or aero-acoustic context. This paper will give two examples where the software has been used. The material library and available boundary conditions proved to be very useful, and in both cases Actran has proven to be a very powerful tool.

## 1 INTRODUCTION

The ability of Actran vibro-acoustic finite element (FE) software to accurately model sound radiation and internal acoustic problems was tested on two very different industrial silencers. The first case is a first stage exhaust silencer, which under certain operating conditions had noise problems resulting in a complaint from nearby the testing facility. Actran was used to model the silencer, explain the cause of the complaint, and indicate the noise reduction of potential modifications. The second case modelled was that of an air filter silencer leading to the intake of a diesel engine turbocharger. This was by far a more complex problem, but several steps were taken to simplify the model without a loss in accuracy as compared with measurements. The following sections describe more thoroughly the approach taken to solve these two problems.

## 2 EXHAUST SILENCER

A night-time noise complaint was issued from a residential area near Wärtsilä laboratory testing facilities during the 1000 hr endurance test of the W6L32E engine, which was operating at full load and 750 rpm. Sound pressure level (SPL) was measured in the neighbourhood and at the test facility, indicating that the noise source was the 1$^{st}$ stage two-chamber exhaust silencer. The noise spectrum showed a dominant peak at approximately 94 Hz, which corresponds to engine order 7.5. Further temperature and SPL measurements were taken with the engine operating at varying speeds and load levels. During these tests it was noted that the exhaust gas temperature decreases with decreasing load, as does the sound radiating at 94 Hz.

## FEM Models

Actran software was used to create the vibro-acoustic model of the silencer shown in Figure 1, including both internal and external air volumes, as well as infinite elements for sound radiation computations. The mesh generated was intended for simulations up to frequencies of 150 Hz, and had 23.4k nodes. Either unit velocity or realistic pressure excitation was applied at the silencer inlet and a non-reflecting infinite duct boundary condition placed at the silencer outlet. A zero admittance boundary was applied at the location of the laboratory roof, which characterizes the surface as a rigid wall. A number of field points, or imaginary microphones, were positioned in and around the silencer for comparison with measurement.

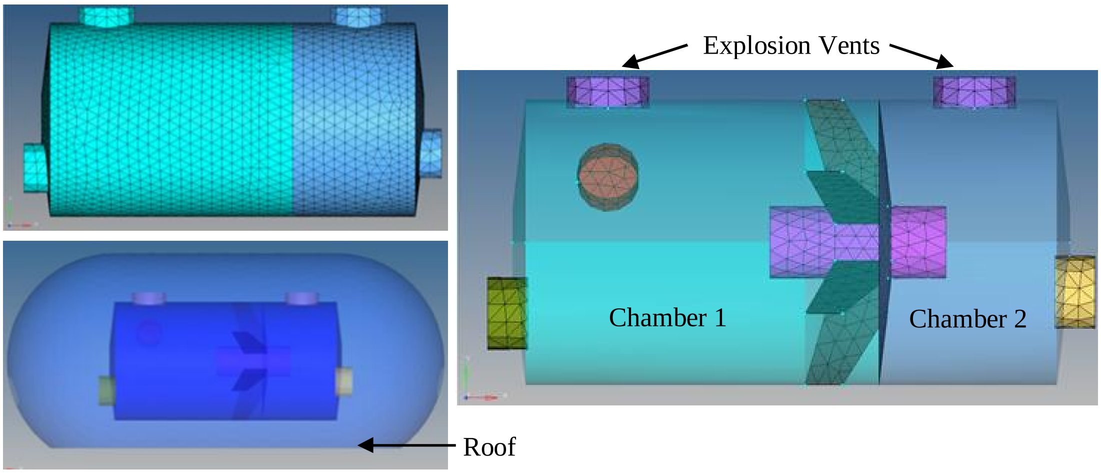



**Figure 1. Exhaust silencer FE model; interior air volumes (top left), external air volume and roof location (bottom left), and inlet, outlet, manhole, explosion vents, central duct, interior wall and supports (right) shown. In all images, inlet is on the left-hand side.**

Initial simulation results indicated a longitudinal standing-wave mode occurring in the first chamber at 75 Hz, and a cross-channel mode occurring in both chambers at 106 Hz. The theoretical modes of a duct with circular cross-section can be calculated with equations 1 and 2 for the n$^{th}$ longitudinal and cross-channel modes, respectively, where $f$ is the resonance frequency, $c_0$ is speed of sound, $L$ is duct (chamber) length, and $D$ is diameter. For the air temperature and silencer dimensions in the current problem, the first few resonance frequencies are computed in Table 1.

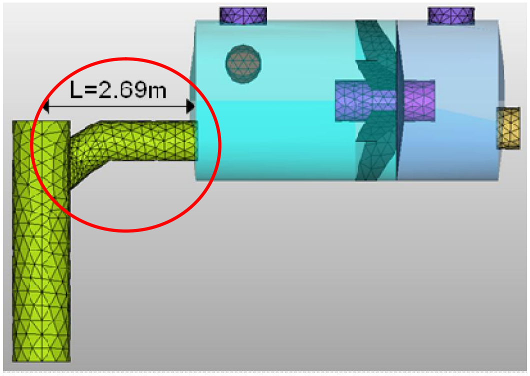
$$f_n = {}^{nc_0}\!/_{2L}\,. \tag{1}$$

$$f_{01} = {}^{1.841c_0}\!/_{\pi D}\,. \qquad f_{02} = {}^{3.054c_0}\!/_{\pi D} \tag{2}$$

The primary acoustic resonances predicted by the model matched theory extremely well, but failed to explain the source of the 94 Hz noise. The FE model was thus expanded to include some inlet-side ductwork. It was determined that a standing wave (longitudinal) resonance occurred in the horizontal ductwork leading to the silencer at 94 Hz, as shown in Figure 2.

**Table 1. Predicted resonance frequencies of the exhaust silencer for the engine operating at full load, which corresponds to exhaust gas temperature of 390 °C and corresponding $c_0$ = 524 m/s.**

| Silencer Diameter (D=2.8 m) | Longitudinal Modes | | Cross-Channel Modes | |
|---|---|---|---|---|
| | $f_1$ (Hz) | $f_2$ (Hz) | $f_{01}$ (Hz) | $f_{02}$ (Hz) |
| 1st Chamber (L=3.49 m) | 75 | 150 | 110 | 182 |
| 2nd Chamber (L=1.76 m) | 149 | 298 | 110 | 182 |



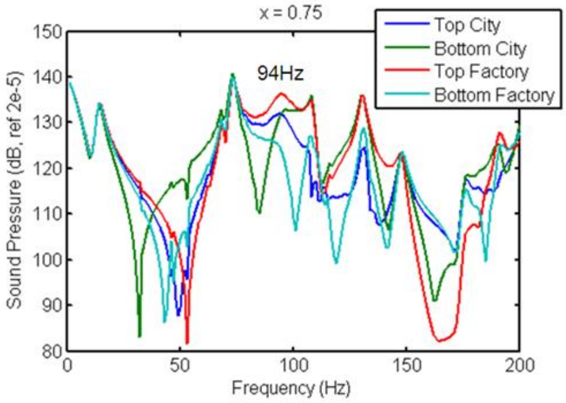




**Figure 2. Silencer model with extended inlet-side ductwork and simulation SPL results at field points outside the silencer.**

### Silencer Modification

The extension of the silencer inlet by 1.4 m to the centre-line of the first chamber was proposed as a solution to the noise problem for several reasons. First, doing so will effectively lengthen the horizontal ductwork leading to the silencer, thus changing the resonance frequency so that it no longer matches with the engine order at 94 Hz. Second, by having the extended inlet lie at approximately the nodal line of the first longitudinal and cross-channel modes of the first chamber, the effects of both will be substantially decreased. Finally, this particular modification is easy to perform.

The exhaust silencer was modified as prescribed, SPL measurements were repeated, and the silencer model was updated to reflect the changes. With realistic sound pressure level excitation applied, it was predicted that the sound level at 94 Hz would be reduced by 7 dB, and that significant reductions could be expected at frequencies near 75 and 106 Hz as well. The final measurement results showed that the modified silencer reduced noise in the problematic 100 Hz 1/3 octave band by 20 dB, and in total the noise was reduced by 10 dB. Having a model that was correctly able to explain the phenomenon causing the noise problem led to a very simple but extremely effective solution.

## 3 AIR FILTER SILENCER

A second problem was used to test the capabilities of Actran software. The transmission loss of an air filter silencer (AFS) leading to the turbocharger of a Wärtsilä diesel engine, having dimensions of diameter ≈ 0.9 m and length ≈ 0.4 m, was simulated up to 8 kHz. The purpose of the filter is to pass air to the turbocharger and engine at high speeds, while attenuating high frequency noise produced by the turbocharger. The silencer consists of a main cylindrically shaped body, in which there are a large number of curved structural and acoustic ribs that are positioned around the cylinder's centreline. A total of five structural aluminium ribs run through the main cylinder and connect a base plate to a top plate which includes a short central duct mounted to the turbocharger. The acoustic ribs consist of polyester fleece baffles that are wrapped in protective perforated sheets. The outer wall of the main cylinder is also made up of a perforated sheet, and it is through here that air enters the silencer and a majority of the sound exits.

The challenges in modelling this silencer were numerous, including complicated geometry, lack of symmetry, porous materials, perforates, and a high frequency range of interest. To meet these challenges, a number of approaches were considered and FE models produced. The following details some of the methods used and a summary of the results.

**Model Size**

The size of a given model depends on a number of factors, including maximum frequency range of interest, the acoustic fluid material, and temperature. A general rule of thumb is that a minimum of 6-8 linear elements are required per wavelength in order to capture the acoustic fluctuation. If second order elements are used, 4 elements per wavelength are sufficient. It is not currently practical to create a full vibro-acoustic model of the AFS that is valid above ~8 kHz.

Thus, a number of finite element models were created to test the influence of the vibrating structure and the use of varying amounts of rotational symmetry. In the reduced models, a periodic boundary condition was applied in Actran. This requires a periodic mesh (with identical node configuration) to exist on the two repeating surfaces of the model. This boundary condition dictates that in a fluid, the pressure degree of freedom (dof) at matching nodes be linked, while in a solid, the 3 solid displacement and 3 solid rotation dofs are linked.

A vibro-acoustic one-fifth model was generated, in which 72° of the silencer was considered, including one of the structural ribs and seven wool baffles. An acoustic one-rib model was also created, this time with only the wool baffle and a small segment of the fluid volume included.

**Table 2. Comparison of size and simulation time for several model configurations.**

| Model | Maximum Frequency (kHz)* | Number of Nodes (Millions) | Simulation Time (Per Frequency) |
|---|---|---|---|
| 1 Rib Acoustic | 8.0 | 0.025 | < 1 sec |
| 1 Rib Acoustic + Perforates | 8.0 | 0.032 | < 1 sec |
| Full Acoustic | 4.4 | 0.54 | 2.1 min |
| Full Vibro-Acoustic | 4.4 | 0.63 | 15.9 min |
| Fifth Acoustic | 8.0 | 1.4 | 18.5 min |
| Full Acoustic | 8.8 | 4.18 | 1.9 hr |
| Full Acoustic + Perforates | 8.8 | 6.4 | 3.7 hr |
| Full Acoustic | 8.0 | 7.9 | 6.4 hr |

**The maximum valid simulation frequency is determined based on a minimum of 8 elements per wavelength for linear elements and 4 elements per wavelength for second order elements.*

The time needed for geometry and mesh creation and simulation varied greatly depending on the models used. A comparison of the simulation time for models of different type and size is displayed in Table 2. The simulations times are based on use of an Actran license allowing parallelization on up to 8 CPU, and run on a high-powered Linux machine (32 CPU, 2.7GHz, 512GB RAM).

Note that the simulation times of the acoustic versions of the 1 rib, fifth and full models valid up to 8 kHz are < 1 sec, 18.5 min, and 6.4 hr per frequency, respectively. The large difference in solution time between acoustic and vibro-acoustic models can also be seen through a comparison of the full models valid up to 4.4 kHz (2.1 min/freq versus 15.9 min/freq).

**Porous Materials**

The polyester fleece used in the baffles was modelled as an equivalent fluid, with complex frequency-dependent speed of sound and density computed using Delany-Bazley empirical formulations [1,2]. This approach uses the assumption that the solid skeleton of the porous material is approximately rigid, and thus the elasticity of the skeleton is not taken into account. A more rigorous approach of using Biot theory to model the skeleton of the porous material as being elastic is possible with Actran software, but the properties necessary to define the material this way were unavailable.

**Perforates**

It is possible to use a full 3D model to analyse the effect of a thin perforated plate on sound propagation, however this requires a fine, complicated mesh which is time consuming for both model/mesh generation and simulation run-time. The alternative is to replace the perforate in the 3D model with a characteristic transfer admittance that is defined between the acoustic cavities that the perforate separates [1]. The coefficients of the transfer admittance matrix can be computed with Mechel's empirical formulas, and are dependent on properties of the fluid and the perforate geometry [3].

**Computing Transmission Loss**

A common measure of silencer performance is transmission loss. Transmission loss (TL) is the acoustical power-level difference between incident ($W_i$) and transmitted ($W_t$) waves of a silencer that is attached to infinitely long ducts or has anechoic termination at the outlet: $TL = 10\log(W_i/W_t)$.

Within Actran, a specific component (called a modal basis) exists to model semi-infinite ducts in which the sound field propagates along the duct axis. With the modal basis component, excitations are defined by setting the amplitudes of the incident duct modes, while the total incident and radiated powers on the modal duct surfaces is computed. In this case, a modal basis component was utilized across the outlet face leading to the turbocharger, unit intensity was applied, and the incident power was computed. Around the circumference of the baffles, infinite elements are placed and the radiated power is determined ($W_t$).

**Results**

The transmission loss computed based on the results from several of the models (without perforates) can be compared with measured insertion loss (IL) in Figure 3. All three sizes of FE model produced TL curves that match the measured IL quite well, and all three geometry types produce similar results. It should also be noted that in both full and

1/5th models, the simulated TL of acoustic and vibro-acoustic models are nearly identical. Thus it was concluded that structural vibration has very little effect on TL, and the smallest one rib model might be a good option for testing possible design changes to the silencer.

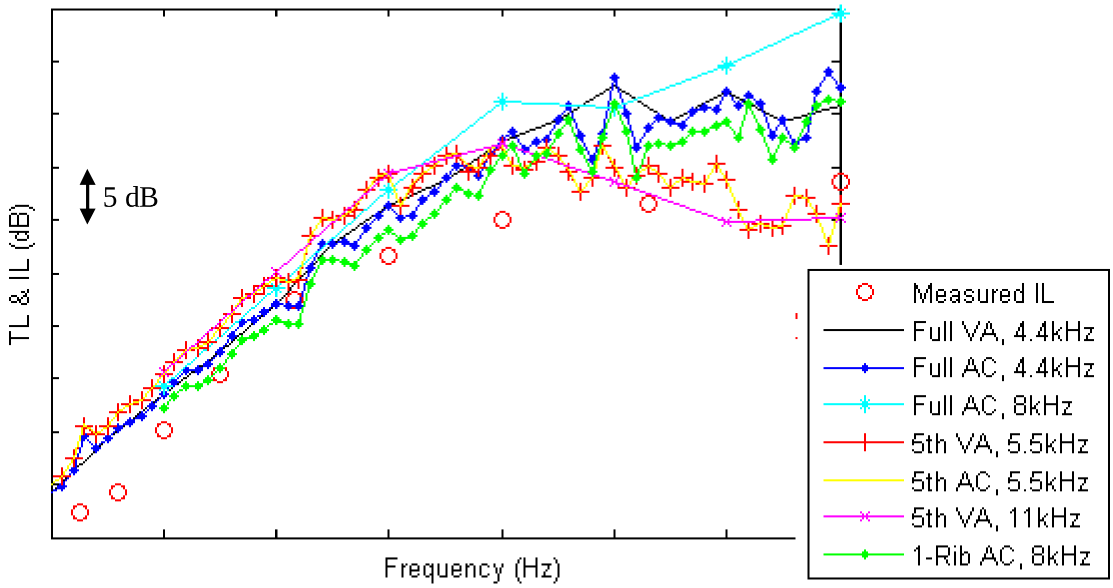



**Figure 3. Measured insertion loss compared to simulated transmission loss for a variety of AFS models (none of which include perforates). The legend describing the curves gives model type (full 3D, 1/5th, or 1 rib) and the maximum valid frequency based on mesh element size.**

The effect of perforated sheets around the wool ribs and silencer exterior was also considered, and the results from some examples can be seen in Figure 4. At low frequencies, they have little effect on TL, at mid-range frequencies they appear to increase TL by a few dB, and then at high frequencies the models with perforates estimate a lower TL value than those without. Again, there is good agreement between the full and reduced (1 rib) model results.

Perhaps the best way to visualize the difference between the results of the 1 rib and full models is by viewing the pressure distribution inside the silencers, as in Figure 5. The SPL (dB) at 2 and 5 kHz is shown for both models in 3D and on a 2D cut-plane. Note that the color representation of the pressures is identical in all of the views. From this it becomes very clear that the 1 rib reduced-size model can emulate the full 3D models, but with significant time reduction.

## 4 CONCLUSIONS

Actran finite element software was used to simulate the performance of two very different industrial silencers. In both cases, the simulations were found to match measured results while clearly explaining the phenomena. The boundary conditions and material representations available within Actran allowed for straight-forward model creation, and also allowed for a number of simplifications that greatly saved time needed for geometry/mesh generation and simulation. In the case of the exhaust silencer, the model helped pinpoint the source of a noise problem and also led to the discovery of a straightforward and effective solution. In the air filter silencer case, a number of techniques were used that reduced an unmanageable model size to one that is small enough to effectively be used within an optimization loop.

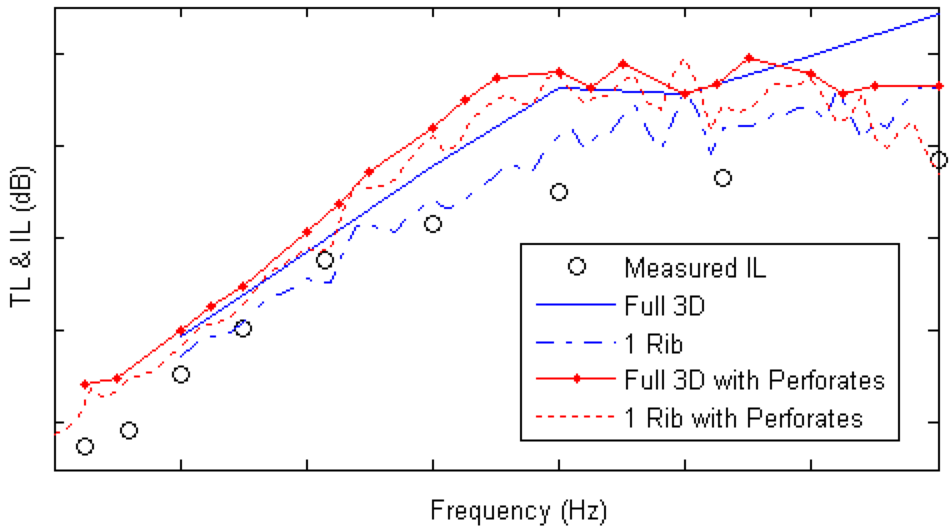



**Figure 4. Measured insertion loss compared to simulated transmission loss of full 3D and 1 rib models with and without perforates included.**

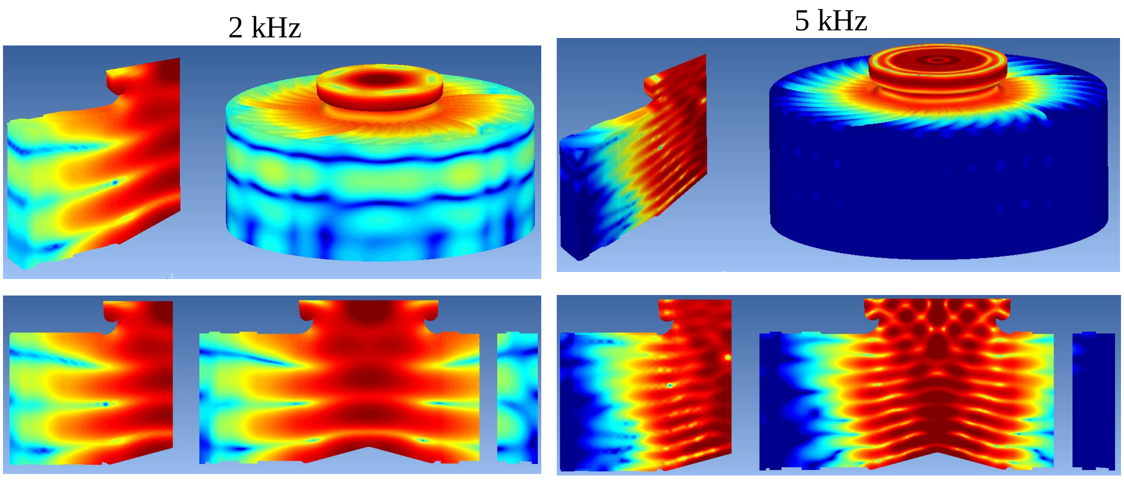



**Figure 5. Sound pressure (dB) colormaps at two frequencies created from simulations of the AFS with the one rib model (left) and full 3D model (right) in each image. The top images show results on the models in their entirety, while the bottom images are of a 2D cut-plane.**

# References

[1] Free Field Technologies, Actran 13.0 User's Guide, 2012.

[2] Delany M, Bazley E, Acoustical properties of fibrous absorbant materials, Applied Acoustics, 3(1970), 105-116.

[3] Mechel F, Ed., Formulas of Acoustics, 2nd ed., Springer, 2008.