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Copenhagen, Denmark – 10th to 13th June 2025

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Title: Investigation of acoustics inside wind turbine blades and how it effects the outside

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Investigation of acoustics inside wind turbine blades and how it effects the outside

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Summary

Wind turbine blades emit tonal and broadband noise in the far field due to air flow around the blade but also due to structural vibration coming from generator, gear box or transformer. So far, the acoustic behaviour of the air inside the blade is neglected assuming that it has no amplifying effect on the outside. In this study measurements of transversal and longitudinal standing waves, its wavelength, frequencies and reverberation times for a 57m blade are presented. It is shown that strong longitudinal standing waves are excited between the webs of the blade. The waves even excite the soft webs and with them the whole blade vibrates and can emit noise. It is also shown with structural and acoustic excitation that damping measures inside the blade (absorber mats and resonators) have an effect on the outside. Numerical investigations with boundary element - and finite element methods of a generic blade submerged in an acoustic field help to understand and confirm the effect of these countermeasures.

1. Introduction

On the European market the noise of wind turbines is a key selling point, i.e. it has strong influence on the decision for or against a wind turbine. The sound power level is determined by the broadband noise coming from the aerodynamics of blades (see e.g. [1]) but also from the tonal noise that is generated in the generator and gear box and that is emitted through tower, nacelle and/or blades (see [2]). The emission of tonal noise by the blades is demonstrated by the Doppler frequency shift measured on the ground (see [3]).

The blades play an important role in the acoustics of wind turbines. First the air flow around the blade generates broadband noise and second the blade is a large resonance body that turns structural vibrations into pressure fluctuations on a very large area. The surface structure of the blade is usually a thin shell that has low eigenfrequencies and that can easily be excited by structural vibrations but also by external or internal sound pressure. It is the goal of this investigation to show that the effect of the blade internal sound effects the surrounding acoustics.

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The aerodynamic noise imposes strong restriction on blade design and in particular on the blade tip (see [4]). It also limits the rotor speed and therefore the energy production of the turbine. Counter measures against noise are welcome that do not affect the air flow around the blade.

2. Blade-internal acoustics

The inside of a blade is a closed room that is framed by thin shells and that includes one, two or three webs going straight from blade root to the blade tip (see Figure 1). Also, the inner side of the foil consists of smooth surfaces that can be considered as acoustically hard walls. The thickness of the walls varies a lot but in general they can be considered as shells with thin thickness (so that the sun is shing through it as can be seen in Figure 1). By design, the blade reduces its diameter from 2-4m at the blade root to zero at the blade tip. It therefore has a conical shape in the longitudinal direction and round/oval shape in a cross section.

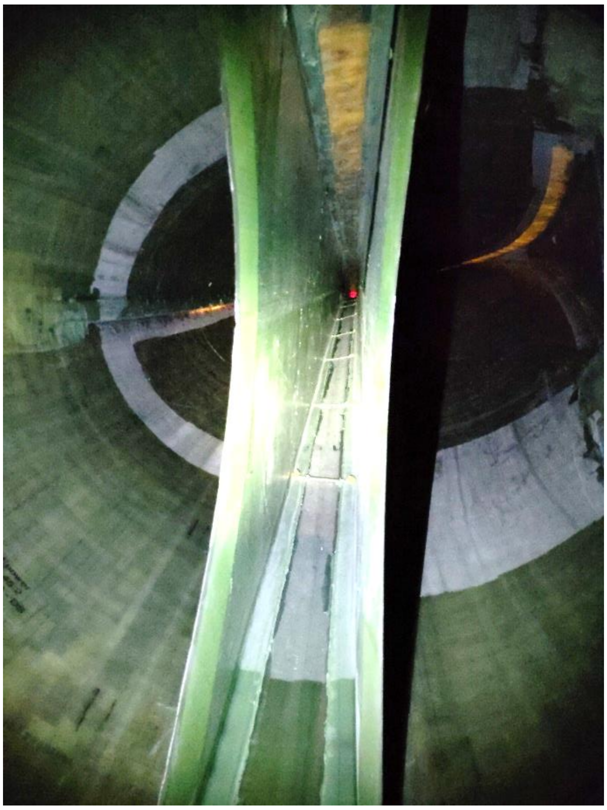

Figure 1: Inside of 45m blade of an ESM-owned turbine with two webs.

The acoustics inside the blade show the following characteristics:

1. The round and smooth inner walls of the blade allow good reflection of sound waves with small diffusion.
2. The room modes in the transvers direction of the blade depend on the diameter of the respective cross section. In a blade with 3m root diameter for example the room mode was measured to be 69Hz (see Figure 2) which fits quite well to the calculated room mode frequency in the round cross section.

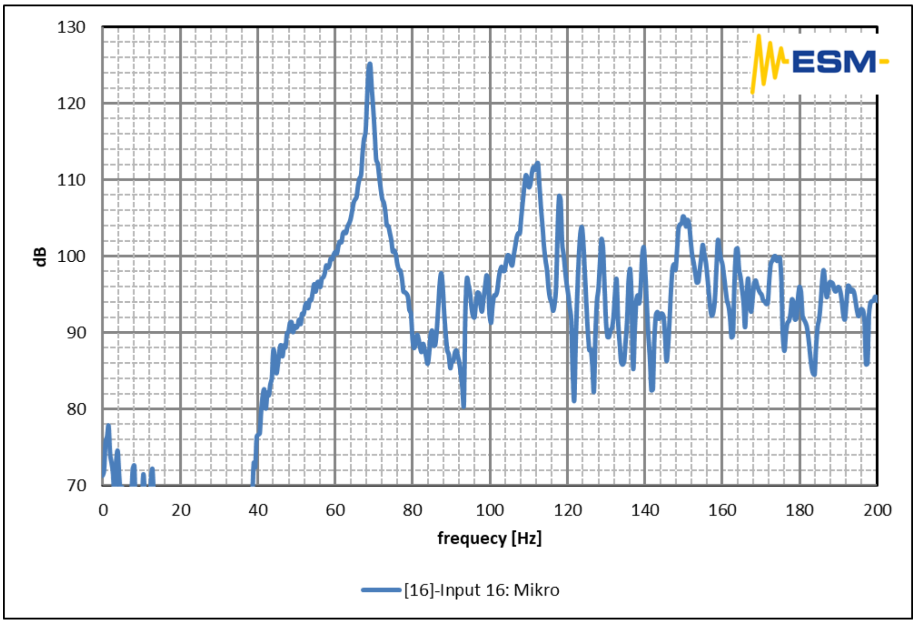

Figure 2: sound power vers. excitation frequency measured with a microphone at 3m diameter root inside a blade excited by a 30Hz-200Hz frequency sweep with a vibration isolated loudspeaker standing inside the blade at the root.

Also, the trend of the transvers room modes along the blade fits well to the calculated room modes (see Figure 3). In a single degree of freedom sound wave the relation between sound frequency f, wave speed c and the wave length λ is:

$$f = \frac{c}{\lambda} \tag{1}$$

A room mode or standing wave in a closed room is present if a half wavelength of the sound fits between the walls of the room.

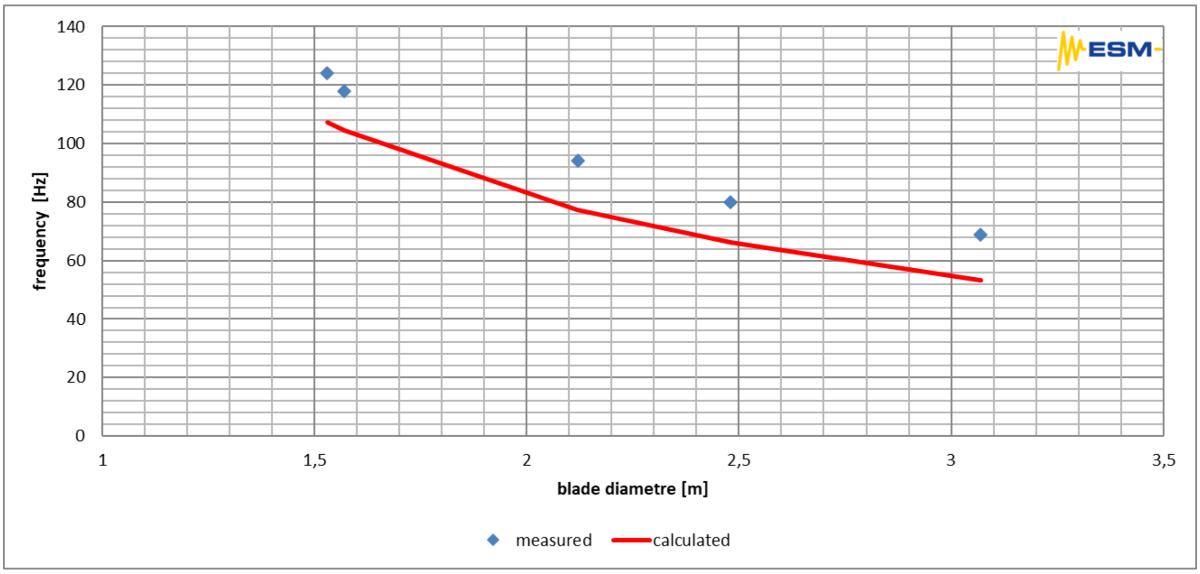

Figure 3: room mode frequency vers. blade diameter at the measured position (blue dots) and the calculated frequencies for the different blade diameters using equation (1).

3. The room mode along the blade axis between two webs (see Figure 1) start at much lower frequencies because in a blade of e.g. 60m a sound wave with frequency 2.75Hz fits between the root and the blade tip. As the harmonics to this sound wave also fit into the same blade length also the harmonics show up in the measurements (see Figure 4). The room modes along the blade axis move more air mass then in the transvers direction and therefore much more kinetic energy is active

in room modes along the blade axis. In addition, the space between the webs is a long equidistant channel with no disturbance in the width direction and conical shape in the height direction (see Figure 1). This geometry reduces the dispersion of sound waves and increases the capacity to store vibrational energy.

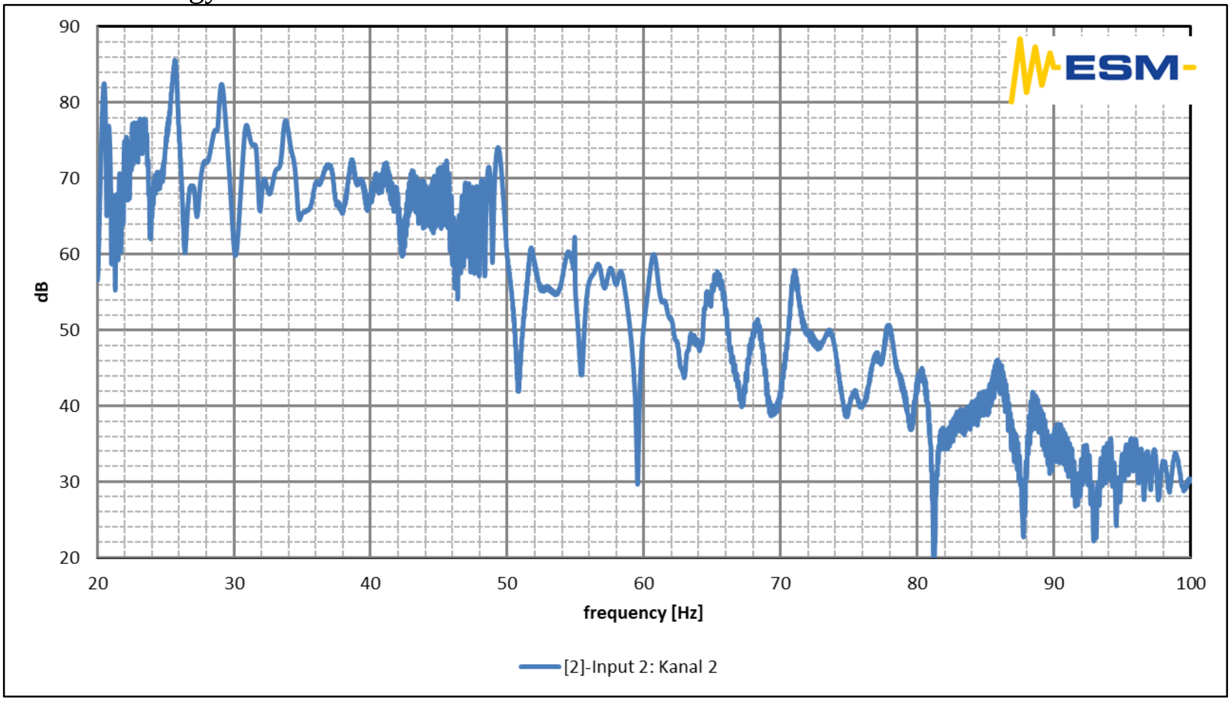

Figure 4: sound power vers. excitation frequency measured with a microphone between two webs excited by a 20Hz-100Hz frequency sweep with a vibration isolated loudspeaker standing at the blade root.

4. The damping ratio of a 30Hz room mode along the blade axis is measured to be 0,005 which corresponds to a reverberation time of T60 = 6s (see Figure 5). This small damping allows large kinetic energies in the longitudinal room modes because only a small amount of energy is dissipated. The reason for the low damping ratio is the absence of sound dispersion and the acoustically hard walls.

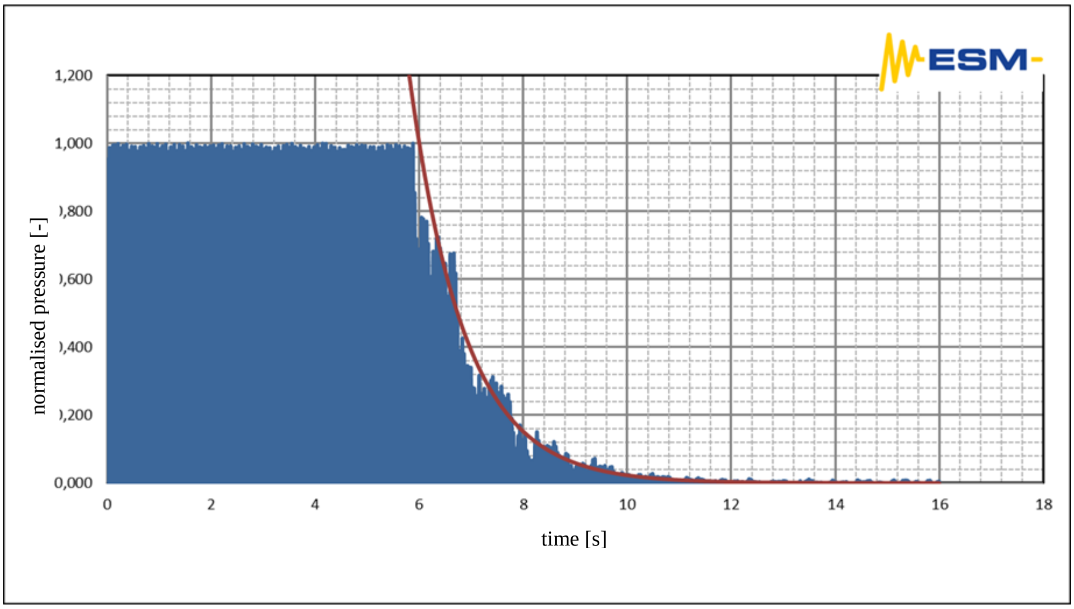

Figure 5: normalised pressure decay curve (blue line) measured with a microphone between two webs excited by a vibration isolated loudspeaker at 30Hz standing at the blade root; exponential decay curve (red line)

Room modes have the property that the velocity of the air molecules is zero at the walls but the pressure amplitude is maximum there. Consequently, the walls of the blade experience the room modes as strong pressure oscillations. From the above analysis it is therefore expected that especially the longitudinal room modes lead to strong pressure fluctuations on the blade walls, because due to low damping and high vibrating masses the kinetic energy in these type of room modes is high.

5. At higher frequencies (above the Schröder frequency [5]) room modes cannot be identified anymore but the above results for high kinetic energy in the longitudinal direction of the blade remains true.

The above analysis of the acoustics in blades neglects the structural eigenmodes of the blades. As the used loudspeaker is isolated from the structure it does not directly excite the structural eigenmodes but the thin shells of the blade can easily be excited by the in-blade acoustics. In this case a structural eigenmode can give a sound amplification back into the room. Consequently, the amplification in Figure 4 can also be structural eigenmodes instead of room modes. Here and in the following study a differentiation between room modes and structural modes can only be done by exciting the structure with a loudspeaker or a mobile shaker. Removing the blade internal sound using noise-absorbent mats or resonators also helps to differentiate between structural eigenmodes and room modes in the blade.

3. Effect of acoustics inside the blade on the outside for frequencies higher than 500Hz

To show the effect of the inner acoustic of a blade on the outside, noise-absorbent mats are positioned at the inside of a 57m blade close to the blade root (see Figure 6).

Figure 6: 12m² noise-absorbent mats of 5cm thickness at the blade root of 57m long blade.

The blade is excited by a loudspeaker inside the blade and by structural shaker at the root. The sound is measured with a microphone outside the blade. The comparison of outside sound power with and without noise-absorbent mats results in the following observations:

1. Exciting the inside of the blade with a loudspeaker and measuring the sound power outside the blade for both configurations, i.e. with and without noise-absorbent mats, shows that in the frequency range where the noise-absorbent mats are active (above 500Hz), the sound power reduces up to 6dB (see Figure 7). This observation demonstrates that the blade shell is not an acoustically

isolating structure, but it can transfer the inner acoustic to the outside of the blade. As the noise-absorbent mats are only active for frequencies above 500Hz this observation holds only true in the frequency range the noise-absorbent mats are active.

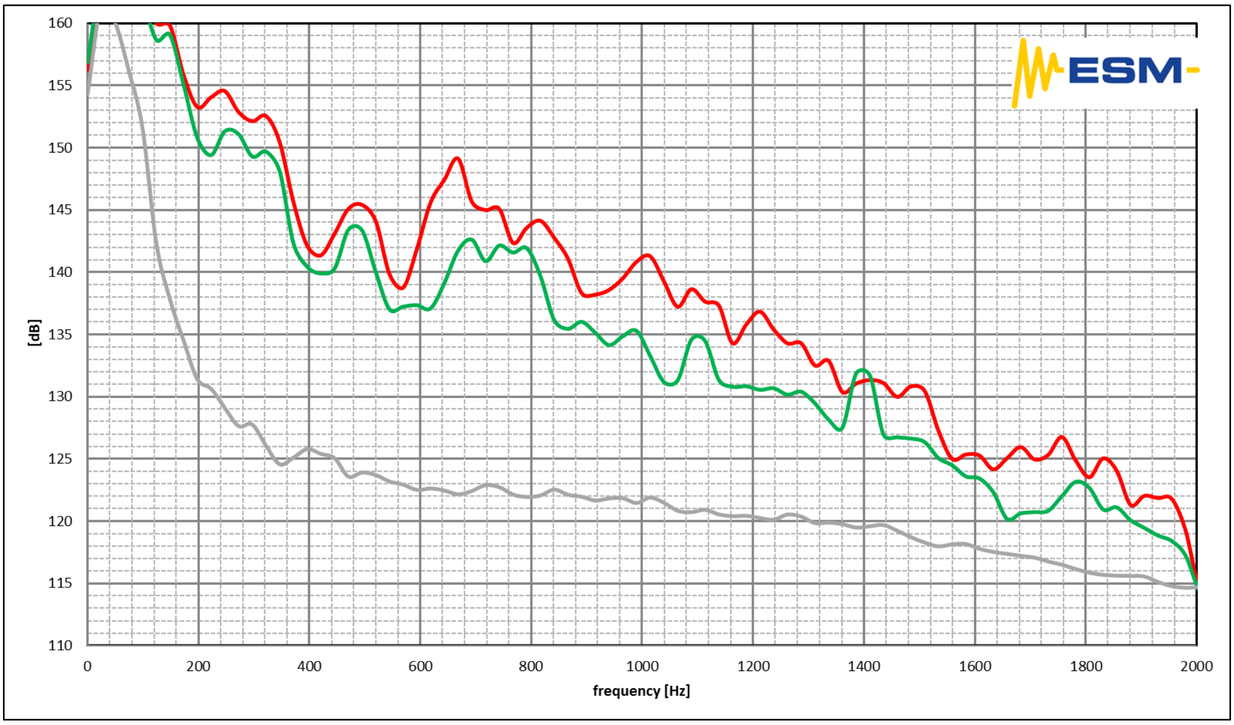

Figure 7: sound power vers. frequency measured with a microphone outside the blade excited by 70Hz-2000Hz frequency sweep using a vibration isolated loudspeaker inside the blade without noise-absorbent mats (red line) with noise-absorbent mats (green line) and without excitation (grey line).

2. Exciting the blade structure with a structural shaker at the blade root gives a less pronounced result compared to the excitation of loudspeaker but still a reduction of up to 2dB can be observed (see Figure 8). The reason for the reduced effect of the noise-absorbent mats on the outside acoustic – compared to the loudspeaker excitation – is that with shaker excitation two transfer paths exist to the outside microphone. The direct transfer path through the blade structure into the outside air is the most obvious one that is usually expected to be the only one. With this measurement it is shown that there is also a second transfer path where the vibrations go from the shaker into the structure, into the inside air of the blade, back into the structure and only then into outside air. The later path is not obvious, but it exists because of the large area of inner blade walls, because of the low damped air acoustic inside the blade that collects a lot of kinetic energy (see section 2) and because of the low stiffness walls of the blade that easily transfer the inner sound to the outside.

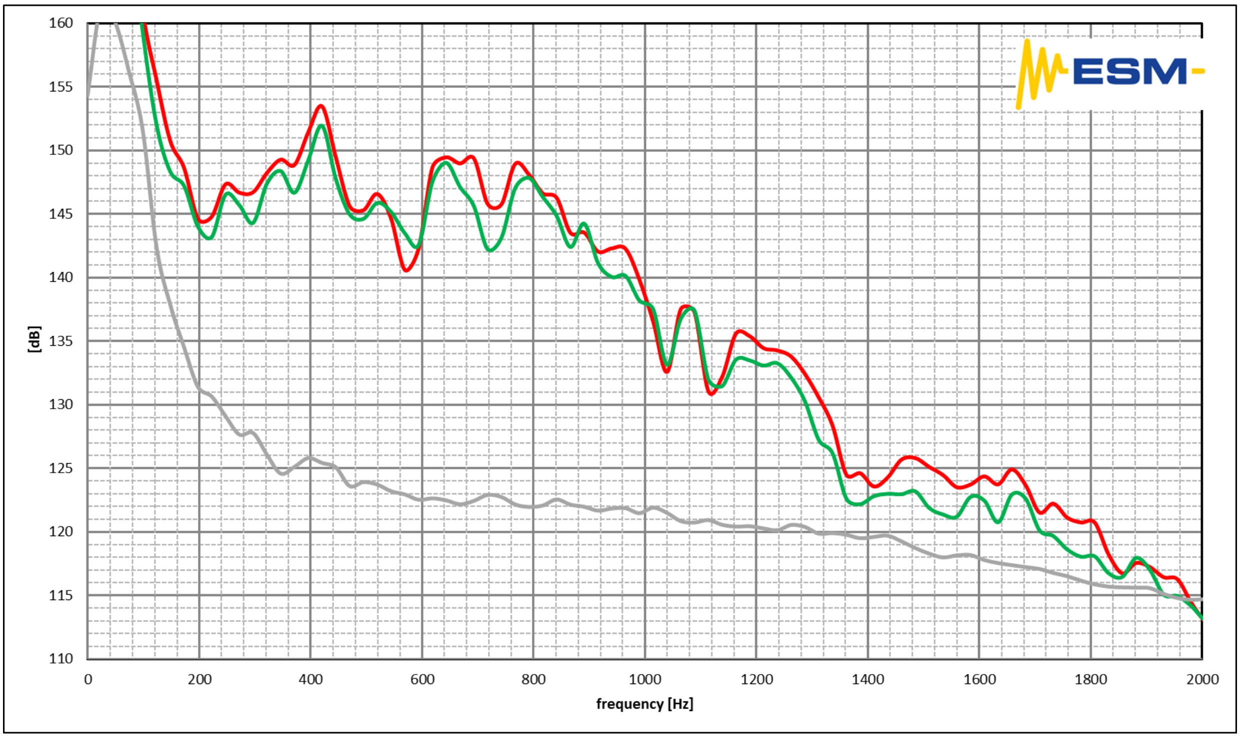

Figure 8: sound power vers. frequency measured with a microphone outside the blade excited by 70Hz-2000Hz frequency sweep using a structural shaker at the blade root without noise-absorbent mats (red line), with noise-absorbent mats (green line) and without excitation (grey line).

The above analysis also shows that noise-absorbent mats can be used to reduce emission of broadband noise coming from blades because it reduces the second transfer path. Noise-absorbent mats can be positioned close to the blade root because due to the conical shape of the blade the sound energy is concentrated at the blade root. They also have the advantage that they don't effect the flow around the blade as for example serrations or vortex generators do.

In operating wind turbines the broadband noise is generated in the air flow at the tip and at the trailing edge (see [1]). It is commonly assumed that the noise is directly emitted to the far field. The above evidences allow the hypothesis that there is another transfer path from the noise generation, via the blade structure, into the blade internal air, back into the blade structure and from there into the surrounding air towards the far field. This hypothesis needs to be proven by future tests on operating turbines. It is expected that this more complicated path is only possible because of the large blade emitting areas towards the inside and the outside air, because of the low damped room modes and acoustics inside the blade and because of the thin shells of the blade.

4. Effect of low frequency room modes inside the blade on the outside

To show the effect of low frequency room modes inside the blade on the outside, frequency tuned and damped resonators are distributed in a lying blade so that room modes are damped. The effect of the resonator is checked by loudspeaker excitation in the blade. In Figure 9 it is observed that in the frequency range between 85Hz and 105Hz the sound power is reduced by up to 10dB using the resonators. This observation demonstrates that the room modes can also have an effect on structural vibrations coming from the generator and gear box and being amplified by the standing waves in the blade.

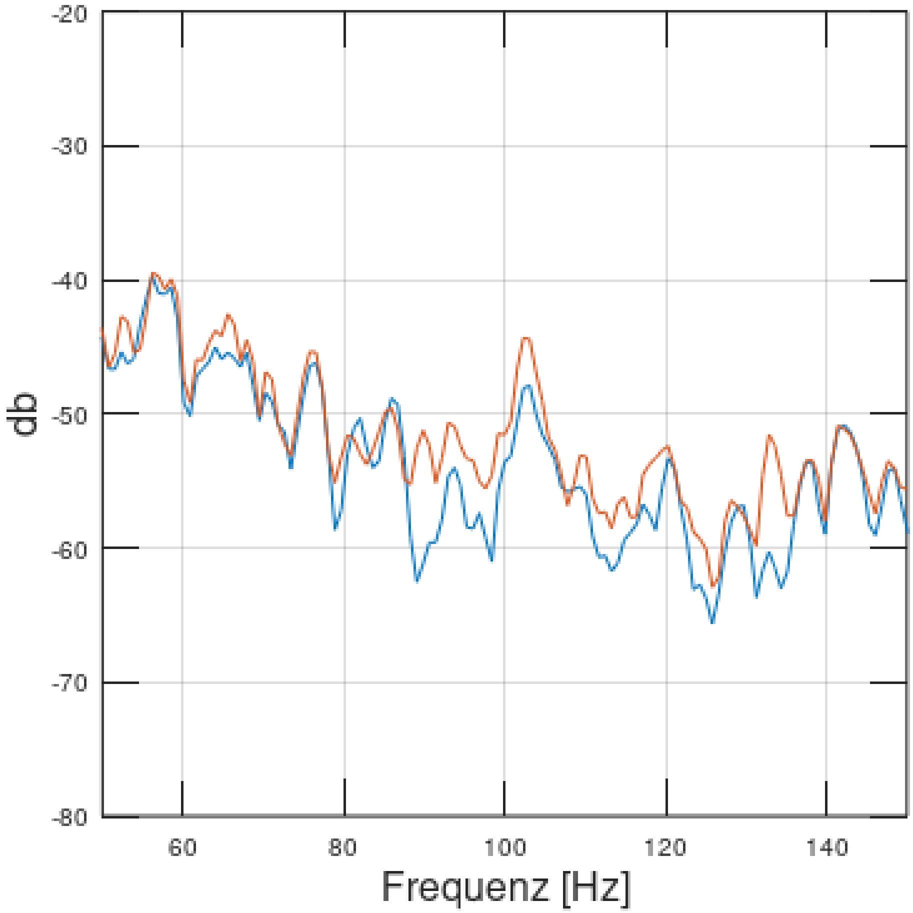

Figure 9: sound power vers. frequency measured with a microphone outside of the blade without resonators (orange) and with resonators (blue).

5. Numerical investigation of internal blade acoustics

The Simulation plays a crucial role in understanding the dynamic behavior of wind turbine blades, enabling optimized designs from the outset and correlation with test data. This approach not only ensures efficient performance but also helps achieve compliance with external noise regulations.

A vibroacoustic simulation model of a generic single wind turbine blade has been developed in ESI VA One 2024 (see [6]) to highlight the importance of incorporating noise control treatments, consequently, additional damping to reduce noise emissions, particularly under structural loads. The wind turbine blade is 27 meters long and its weight is 3000kg (*Figure 10*).

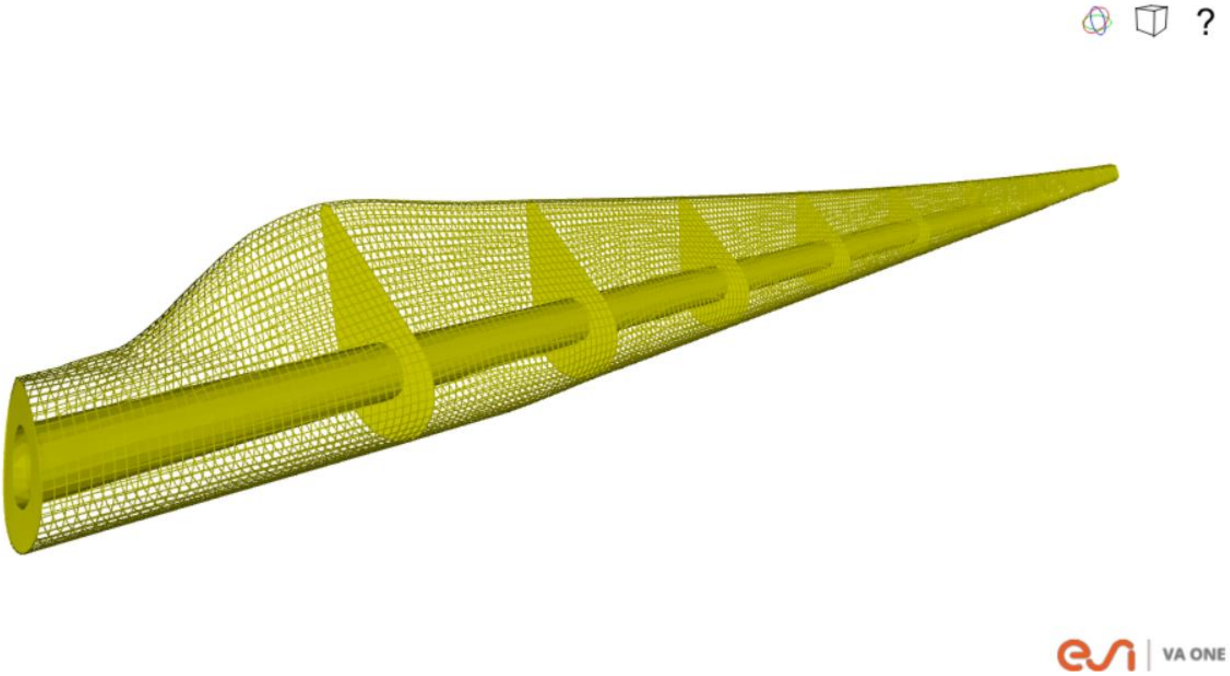

Figure 10 – Wind Turbine Blade Finite Element Model

The vibroacoustic model consists of a structural finite element (FE) representation comprising 14184 elements, acoustic FE representation for eight interior cavities with a total of 222947 elements (*Figure 11*), unit structural loads applied in three directions at the hub connection, and a semi-infinite fluid (SIF) (*Figure 12*) domain to enable exterior noise radiation analysis.

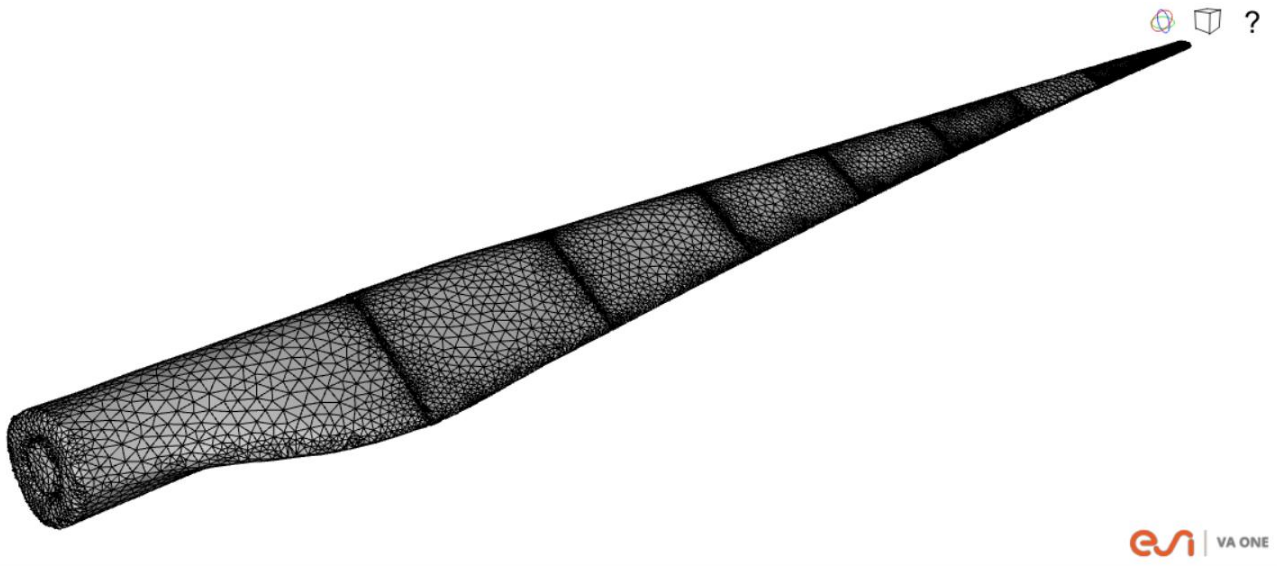

Figure 11 - Acoustic Finite Element Model

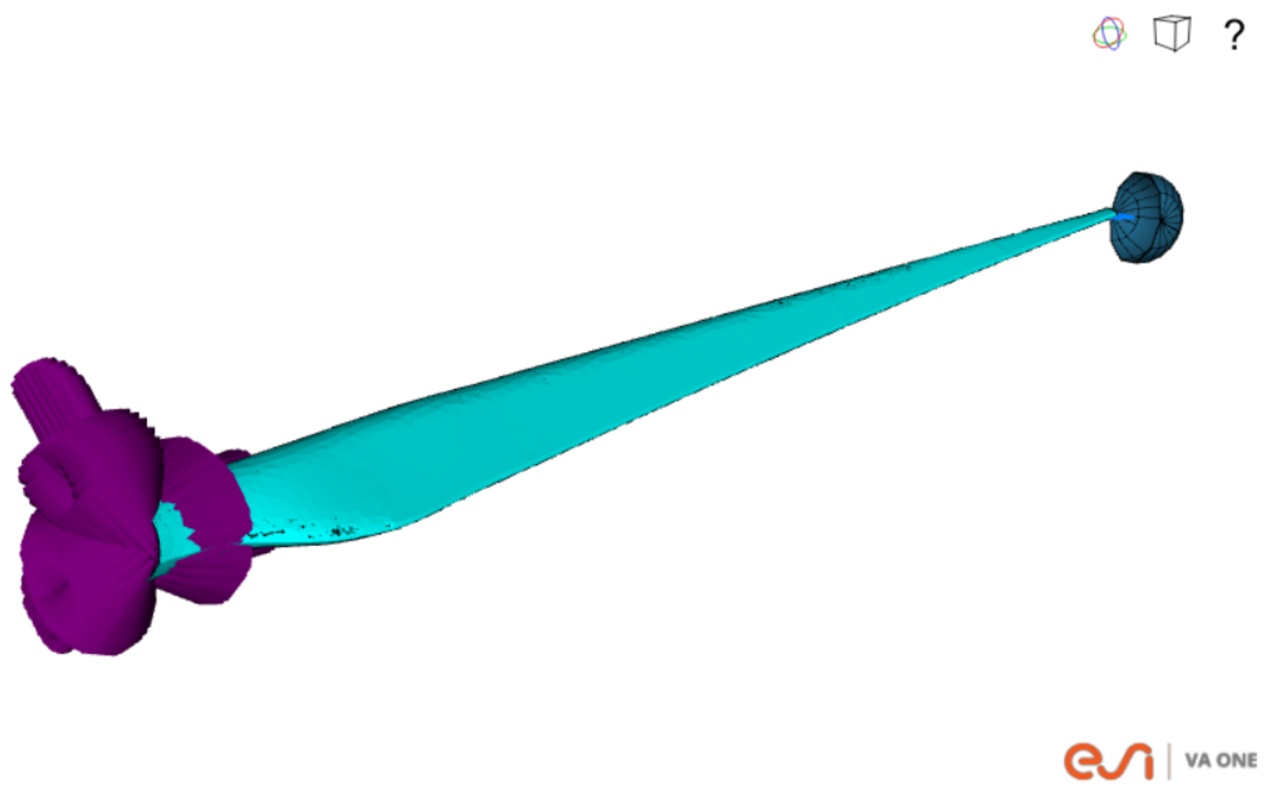

Figure 12 - Vibroacoustic Model with Loads and SIF

Structural and acoustic normal modes (*Figure 13* and *Figure 14*) have been computed using ESI VPS solver, ensuring the model's validity within a frequency range of 300 Hz. The analysis has been conducted in narrowband with a resolution of 1 Hz. If an extended frequency range is required, Statistical Energy Analysis (SEA) method available in ESI VA One is recommended.

Figure 13 - First Structural Bending Mode

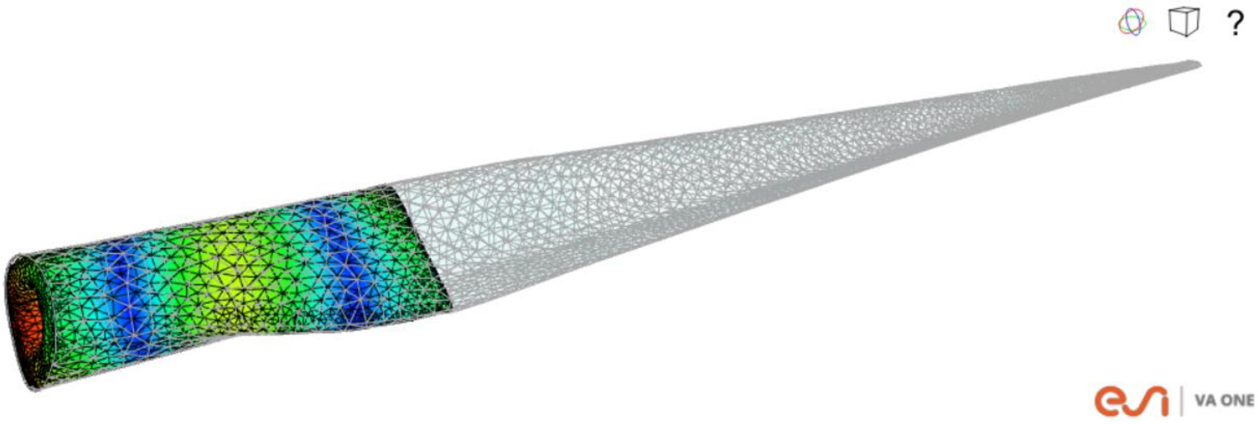

Figure 14 - Acoustic Mode of the First Cavity

Additionally, a second model has been developed where the interior of the blade has been treated with 5 cm noise control treatment of a low-density foam (8 kg/m3) to introduce absorption and damping (*Figure 15*). *Figure 16* presents a comparison of the sound power emitted by the blade with empty cavities versus the blade with poroelastic (foam) material applied to it. The results indicate that the addition of foam leads to a reduction in noise emission across the entire frequency range, attributed to both increased absorption and additional damping. However, it is important to note that incorporating foam increases the blade's weight by 39 kg, about 1.3% of the total weight of the blade, which could impact other performance factors. Therefore, selecting the appropriate poroelastic material is critical to balancing all key performance criteria under operational conditions.

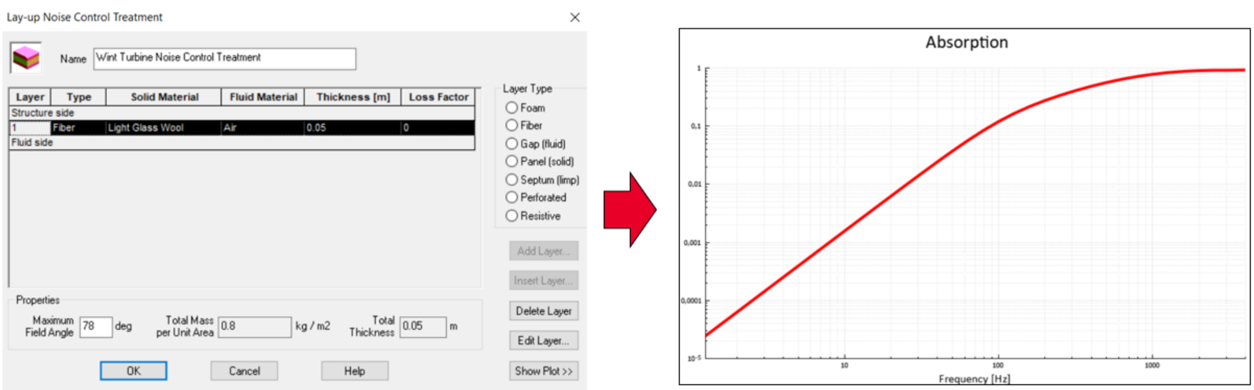

Figure 15 - Poroelastic Model (PEM) properties

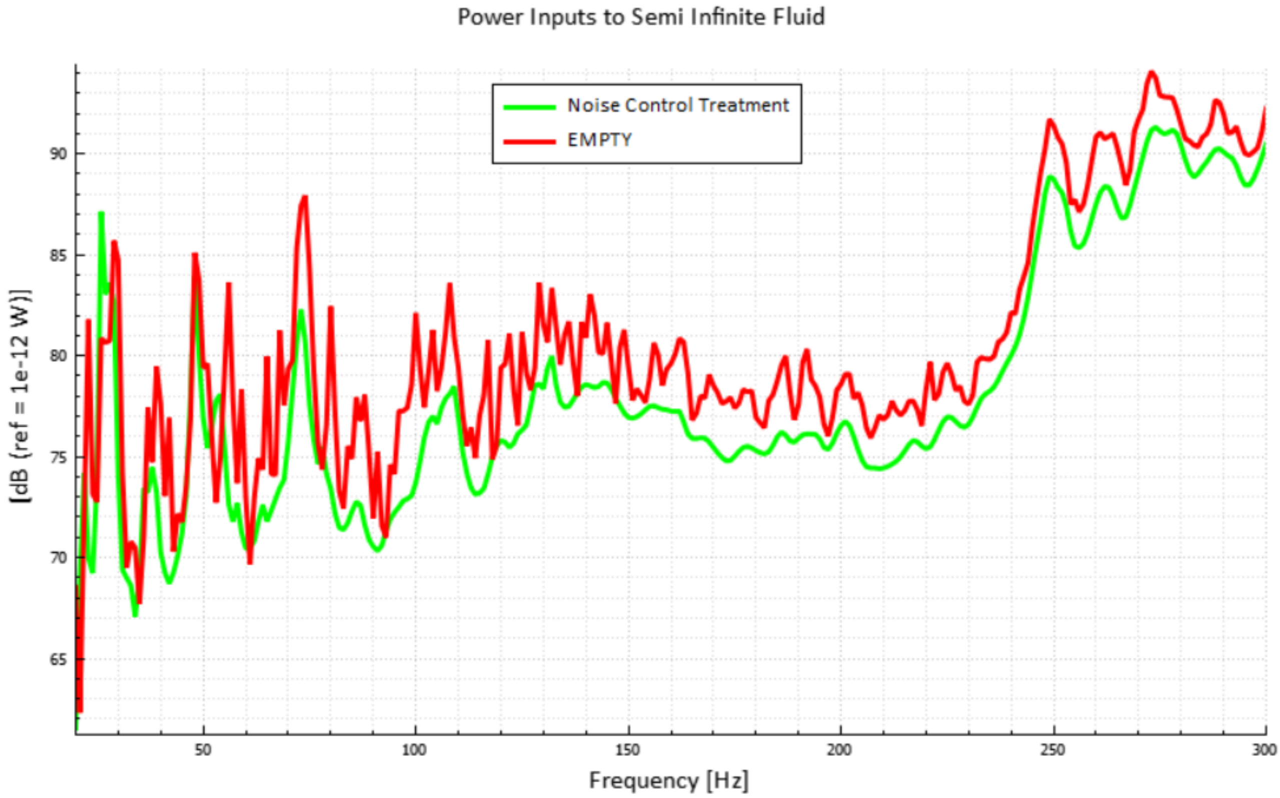

Figure 16 - Acoustic Power Emitted

For the peak at 74 Hz, as an example, the SIF models were converted into boundary element models (BEM) to gain deeper insights into blade radiation (see *Figure 17*). Furthermore, four virtual microphone planes were integrated into the model to analyze radiation patterns, confirming that the blade with noise control treatment applied exhibits superior vibroacoustic performance, as evidenced by reduced noise emissions (see *Figure 18*).

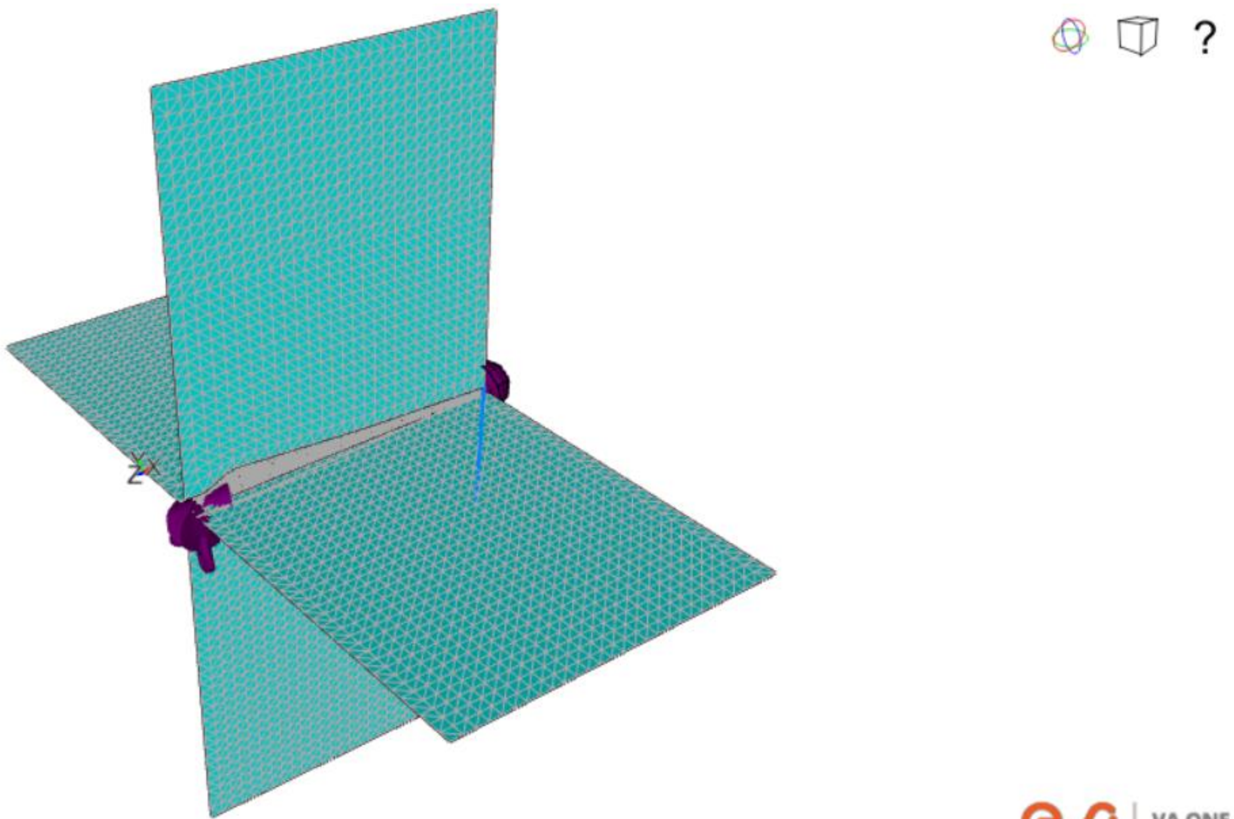

Figure 17 - Boundary Element model (BEM)

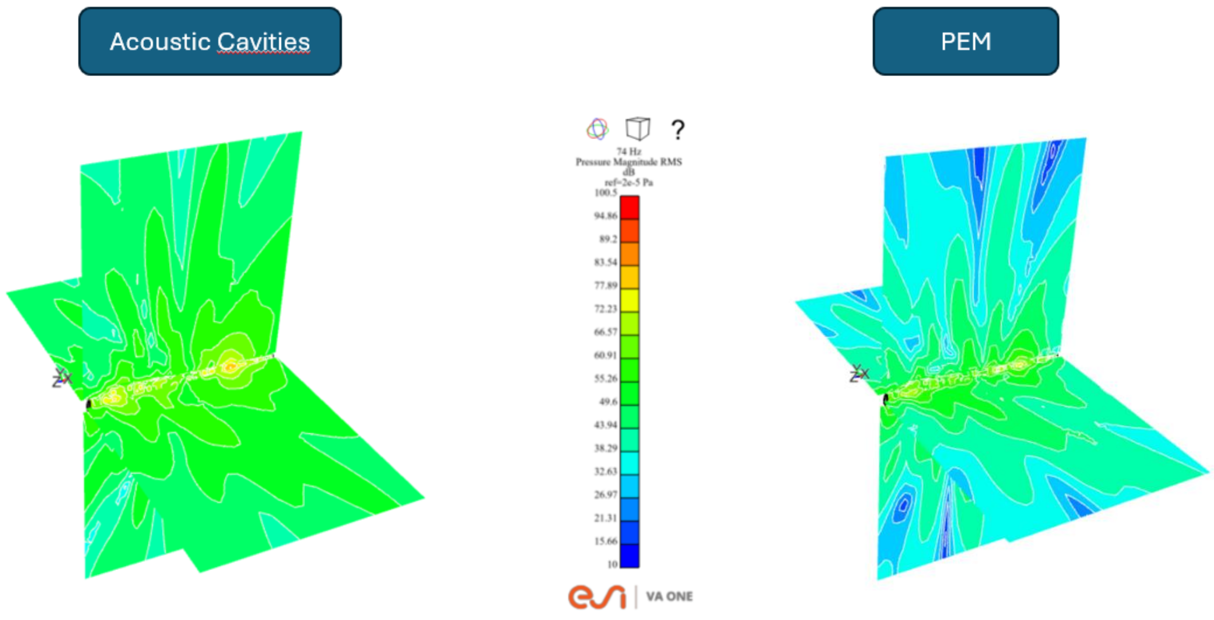

Figure 18 - Comparison between the model with noise control treatment (PEM) and the one with air in the cavities at 74 Hz

6. Conclusions

Experimental and numerical investigations give strong evidence that the acoustic of the room inside the blade has an effect on the far field acoustic of a wind turbine. It is also shown that noise-absorbent mats at the blade root and resonators distributed in the blade have a positive effect on the broadband and tonal noise behaviour of a blade. Sound measurements on an operating turbine and numerical investigation of the corresponding blade are planned to complete this analysis.

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