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Efficient Prediction and Analysis of the Noise Radiated by an Electric Powertrain

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Abstract

Reducing the emitted noise from vehicles is a primary issue for automotive OEMs due to the constant evolution of the noise regulations. In the context of electric powertrains, virtual prototyping has proven to be a cost-efficient alternative to the build-test process, especially in early design stage and/or if optimization is targeted. Due to the multiphysics nature of the model, the full simulation chain involves multiple components, each having its own specific modelling attributes. The difficulty then resides in the parts assembly, solving issues like mesh-to-mesh projections, time to frequency-domain transformation, 2d-axisymmetric to 3d mapping, data formatting and management, unit and local coordinate systems… This paper presents an environment that allows for the prediction and analysis of the noise radiated by electric automotive powertrains. The stator-rotor electro-magnetic behavior is represented by time-dependent forces applied on stator teeth. Transfer functions from structural modes to acoustic pressure describe the vibro-acoustic behavior and ensure a fast synthesis of the radiated noise. It is demonstrated how and where harmonic and space order decompositions are introduced in the computational process to deliver efficient and powerful analysis means to drive design decisions. As such, the workflow operation does not require deep expertise neither in electro-magnetic nor vibro-acoustic simulation. The proposed workflow, implemented in the Actran acoustic simulation environment, is finally used to obtain and analyze results on a typical industrial electric powertrain model.

1. Introduction

In the frame of electrification of vehicles, automotive industries have to switch from Internal Combustion Engines (ICE) to electric motors. Noise radiated by electric motors involves new challenges for NVH engineers. The main sources of noise are well known and clearly identified [1]. They are listed below:

- Electro-magnetic loads, especially applied on stator teeth.
- Mechanical loads: bearing loads and gear noise (whine noise)
- Aeroacoustic noise due to high speed rotating parts and potential presence of cooling fans

The contributions of each of these sources will depend on the rotation speed: whereas electromagnetic noise is dominant at low RPM, aeroacoustic noise may be dominant at high RPM.

Electromagnetic loads are tonal components and can be decomposed into harmonics of engine 1st order. Another source of electromagnetic noise is related to PWM harmonics [2]. This noise is especially dominant at low rotation speed [3] and can be annoying for psychoacoustic purpose (tonality, roughness) at engine start. PWM are not order based. Gear noise is also audible due to the presence of reducer. The reducer of e-motors is less complex than combustion engine gearboxes (for e-motor usually a 2 stages gear system is required without clutch). This noise can be addressed by the usage of multi-body dynamics simulation with accurate modelling of the gear contacts [4]. Aeroacoustic noise is mainly studied when cooling fans are mounted on the shaft, that is not the case still in automotive industry.

The mechanisms involving electro-magnetic forces (without PWM) exciting the stator, stator housing and radiating has been studied [1, 5, 6, 7], some tools can determine the noise radiated [8]. However, an integrated and systematic process to study the noise induced based on FE methods from A to Z is suitable for accurate results. Mechanical and acoustic engineers must couple their structure Finite Element (FE) model with electro-magnetic excitations. The acoustic radiation must be considered for accurate noise prediction.

The engineers also face a question of format exchange between the software and physics:

- Electro-magnetic FE data (excitation)
- Structure FE data (structure dynamics)
- Acoustic FE data (radiation efficiency)

Now how can an NVH engineer perform a e-motor NVH study efficiently? What kind of information is important to highlight for NVH design purposes? Dedicated analyses and graphs are required.

Based on this observation, an approach which is combining the three inputs together in a fairly flexible way is proposed. Electro-magnetic loads are provided by electric engineers, structural eigen modes are provided by NVH engineers and acoustic transfer functions are provided by acoustic engineers. The final acoustic response is the combination of all ingredients together within a dedicated interface. This interface is at the same time flexible in terms of inputs and dedicated in terms of analysis.

In the following sections, the process of combination of data is presented and applied on a realistic case. The paper is developed in two main parts:

- First part is explaining the theoretical background to couple electro-magnetic loads, structure modes and acoustic radiation;
- Second part is presenting an example of application of the strategy, based a realistic stator model.

Finally a conclusion will draw the advantages of the technique.

2. Theoretical Considerations

The synthesis of the acoustic noise that is generated by an electric motor involves multi-disciplinary models that have different attributes and that are generally not trivially assembled. Most of the time, it is a valid assumption to consider weak coupling between the electro-magnetic, vibratory and acoustic effects. In other words, the effect of the structural vibration on the electro-magnetic fields, as well as the effect of the acoustic pressure on the structural vibrations are reasonably neglected. As a consequence of this, a chained simulation workflow is traditionally put in place with the following main steps:

1. Solution of the electro-magnetic model with the objective of identifying of the Maxwell pressure field in the rotor/stator interface region that results from the rotation of the rotor at given rotation speeds.
2. Solution of the structural model with the objective of evaluating the vibration of the external stator skin that results from imposed electro-magnetic forces.
3. Solution of the acoustic model with the objective of evaluating the acoustic pressure field that results from an imposed vibration of the stator external skin.

Intermediate steps are sometimes needed in this workflow to accommodate modelling constraints. Usually, electro-magnetic simulations are performed on two-dimensional or sector models, taking advantage of symmetry properties, in the time domain. Vibro-acoustic simulations are on the contrary most of the time performed on three-dimensional models, in the frequency domain. Fourier transformations (with the underlying assumption of linearity) as well as full-field reconstruction techniques are therefore needed to transfer information from one discipline to another. Practical considerations on interfacing between EM, structural and acoustic solvers sometimes also require refining the above workflow by adding the necessary file conversion operations. Finally, the EM, structural and acoustic model have their own discrete support (most of the time finite element meshes), which a priori follow different meshing constrains, and therefore impose field mapping strategies between the different layers of the workflow.

As a consequence of this, the overall simulation workflow is most of the time complex and error prone. Packaging all operations into an automated workflow where user input is reduced brings a lot of value and indirectly increases the simulation reliability.

When it comes to performance aspects, advantage can be taken from the fact that the three main steps of the analysis (the EM simulation, the structural analysis and the acoustic radiation step) are somehow independent from each other. Indeed, if each of those steps is reworked into a step that generates well-chosen transfer functions, the overall computational cost of the simulation can be split into two parts:

1. Pre-computation steps:
   a. EM fields.
   b. structural modes, as transfer operator between the EM excitations and the structural vibration.
   c. acoustic transfer functions between the structural modes and acoustic data recovery points.
2. Synthesis of the structural vibration and acoustic response through transfer function manipulations.

The advantage of this reorganization is that the second part is light in terms of memory requirements and fast in terms of computational time. It also brings physical insights to the NVH analysis (modal and frequency content of the EM excitation as well as structural modes are directly exposed as design parameters). Finally, it eliminates the complexity of the EM, structural and acoustic models by moving them to a pre-computation step. Such models can therefore be prepared by field specialists, where the final NVH analysis is made accessible to non-experts or to specialists from other fields. By clearly splitting the three parts of the full NVH analysis, it also allows to anticipate EM noise analysis to early-design stages.

2.1. Electro-Magnetic Modeling

As the scope of this paper is the evaluation of the noise response of an electric motor, details are not given here on the

EM model itself. It suffices to mention here that the EM model should have as objective the generation of the Maxwell pressures that apply on the stator.

For the sake of analysis, it is convenient to decompose the spatially varying and time-dependent pressure field into space orders and harmonics. Assuming that the machinery has revolution symmetry properties and is rotating at an angular speed *RPM* revolutions per minute, the mechanical excitations (for radial flux machines) are decomposed into harmonics h and spatial orders o:

$$p(r,\theta,t) = Re\left(\sum_h \sum_o p_{ho} \exp\left(j\, 2\pi \frac{RPM}{60h} t\right) \cos\left(\frac{o\theta}{2\pi}\right)\right), \quad (1)$$

where r, θ are polar coordinates. Longitudinal orders have been omitted here for the sake of the presentation. A similar decomposition also applies for mechanical excitations in axial flux machines.

Truncating the above decomposition to a maximal harmonic value H and a maximal space order O, the excitation can be completely described by the matrix of values p_{ho} that only depends on the electric motor topology.

2.2. Structure Modeling

The structural model generally describes the stator part of the electric motor. For vibration analysis, a finite element mesh is generally created and allows to compute the eigenfrequencies and associated eigenmodes (f_i, φ_i) of the stator:

$$K.\varphi_i = \lambda_i M.\varphi_i, \quad (2)$$

where K and M are the stiffness and mass matrices of the FE model and, assuming mass normalization, $\lambda_i = (2\pi f_i)^2$ are the eigenvalues.

Once the modes are identified, the structural response for an arbitrary loading is obtained by solving for the structural modal participations $\alpha_i(\omega)$:

$$\varphi^T.\left(K - \omega^2 M\right).\varphi.\alpha(\omega) = \varphi^T.F(\omega), \quad (3)$$

where φ_i. $F(\omega)$ represents the modal force applied on mode i by the mechanical loading $F(\omega)$. In the above equation, no damping is considered for the sake of concision. Structural damping as well as modal dampings can be added without particular difficulty. Global structural damping will be reflected as a complex frequency-independent factor to the modal stiffness matrix. Modal dampings may be used to represent a frequency-dependent damping behavior and accommodate for either structural or viscous models. Finally, heterogeneous damping can be accommodated by the inclusion of a modal damping matrix, which can be assembled at modal extraction time. In this latter case, the diagonal nature of the modal impedance is lost and the solution for modal participations is less efficient.

The diagonalization properties of the modes (which remain valid in case of homogeneous or modal damping) allow to solve for the modal participations in a very efficient way. The operational deformation $U(\omega)$ is finally obtained by modal superposition:

$$U(\omega) = \varphi.\alpha(\omega). \quad (4)$$

It is noted that modal truncation is classically resorted to in order to reduce the dimensionality of the problem in modal coordinates. The number of modes considered in the analysis is therefore by far smaller than the number of degrees of freedom (dofs) in the analysis. The usual criterion for modal truncation (modes with associated eigenfrequency up to two/three times the maximal frequency of the analysis) holds.

From a computational perspective, the structural modal participation factors can be seen as transfer functions between modal forces and local response points. Also, if the electro-magnetic excitation is decomposed into harmonic and space orders, the modal forces can be pre-computed as well and the structural modal participation factors are transfer functions between the EM components p_{ho} and the local response points.

2.3. Acoustic Modeling

Similarly to what is done for structural modelling, the acoustic model is discretized through the finite element technique. The representation of the radiation in far field (so-called Sommerfeld boundary condition) is accommodated with techniques such as infinite elements (IE) or perfectly matched layers (PML). Given the damped nature of the model, a modal representation is not adequate, and the solution is generally searched for in physical coordinates. The acoustic pressure field $p(\omega)$ solves the following relation:

$$A(\omega).p(\omega) = V(\omega), \quad (5)$$

where $A(\omega)$ is the acoustic admittance matrix and $V(\omega)$ the acoustic excitation, directly obtained from the stator skin vibration.

In the case of electric-motor noise, the acoustic excitation results from the vibration of the stator on surfaces in contact with the acoustic fluid. Given the linearity of the problem, this excitation can be decomposed in the structural modal basis:

$$V(\omega) = V_\varphi.\beta(\omega), \quad (6)$$

where V_φ are the acoustic excitations resulting from the trace of the structural modes on the surfaces in contact with the acoustic fluid and $\beta(\omega)$ are the corresponding participation factors. This decomposition allows to rewrite the acoustic pressure field $p(\omega)$ as:

$$p(\omega) = p_\varphi(\omega).\beta(\omega), \quad (7)$$

with:

$$A(\omega).p_\varphi(\omega) = V_\varphi. \quad (8)$$

As the $p_\varphi(\omega)$ can be pre-computed and do not depend on the actual EM excitation field, they can be seen as transfer functions between the structural modes and the acoustic data recovery points.

2.4. Force Mapping

An important point concerns the mapping of the EM Maxwell pressures on the structural model. Though this point seems conceptually simple, it is not straightforward [9] as one needs to account for:

- the actual dimension of the Maxwell pressure field: based on the convention taken for storing the Maxwell pressures in the EM solver, or possible collocation and/or transport of the Maxwell pressures to forces in the EM solver, the exported mechanical excitation field may have dimensions that differ from distributed pressures.
- the region of application of this distributed pressure field: depending on the export options and data recovery domain selected in the EM solver, the distributed pressure field might be a distributed load (vector) rather than a pressure (scalar) field.
- the time to frequency domain transformation: discrete Fourier transform should account for Nyquist criterion and avoid any improper resampling that would break the tonal nature of the time-domain signals.
- the possible sector to full-circle data reconstruction.
- the possible 2D to 3D extrusion, with or without skewing effect.
- mesh mapping and possible aliasing effects: mesh resolution on the structural side should obviously respect constraints related to the representation of the structural waves, but also regarding the representation of the EM excitation.

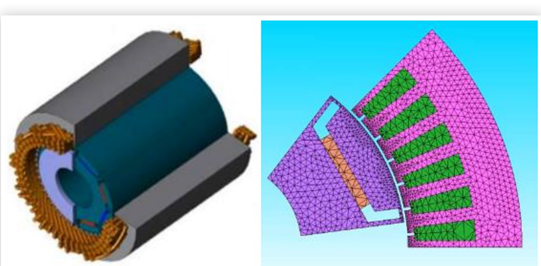
2.5. Response Synthesis

Given the choices made in the pre-computation steps, the synthesis of the radiated noise by the actual EM excitation field (or harmonics and/or space orders) simply appears as the combination of the structural modal forces, structural unit-modal force modal participations factors and acoustic modal transfer functions.

3. Application to e-Motor Stator

The process described in Part 1 is applied to a stator model (figure 1). The stator is part of a 48s8p Permanent Magnet (PM) machine containing stator, rotor, winding and magnets.

FIGURE 1 3D Permanent Magnet machine (left) and 2D electro-magnetic model associated (right).



3.1. Electro-Magnetic Loads

The magnetic transient analysis is performed with JMAG Finite Element software. Taking advantage of the machine symmetries, the model can be reduced to a 2D model and 1/8 of sector. The analysis is obtained for 10 angular speeds from 1000, to 10,000 RPM.

The output of the electro-magnetic calculation is the distributed load acting on stator mesh, expressed as forces in Newton. The loads are reconstructed during the export on a 360° 2D mesh (Figure 2).

Note that the electromagnetic is usually concentrated on the stator teeth so for NVH purposes, the mesh supporting the forces is reduced to the surface of the teeth in front of rotor (Figure 3).

It is usual to visualize the space and time dependency of electro-magnetic forces as a 2D diagram called spatiogram plot. Figure 4 displays this spatiogram for loads at 1000 RPM. X axis is the time series and Y axis represents the azimuthal direction in degrees. This diagram clearly shows the repetition of the patterns over time and space. The time series stops at one fourth of mechanical rotation. For the machine studied, the time domain minimal pattern as well as space minimal pattern in azimuthal direction is 8. Therefore, the following study will focus on engine order 8 and its harmonics.

To load 3D structure FE models, it is necessary to reconstruct a 3D field of excitation based on the 2D EM model. An extrusion of 0.2m of the 2D mesh of teeth surfaces is done (Figure 5). Additionally, the EM teeth mesh is aligned with structure teeth mesh to correctly project forces on the structure FE mesh.

FIGURE 2 Electro-magnetic forces applied on the stator teeth at t=0.0s. Overview and detail.

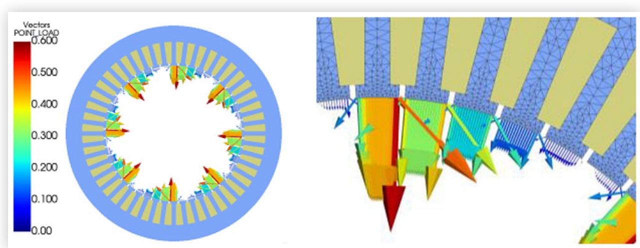
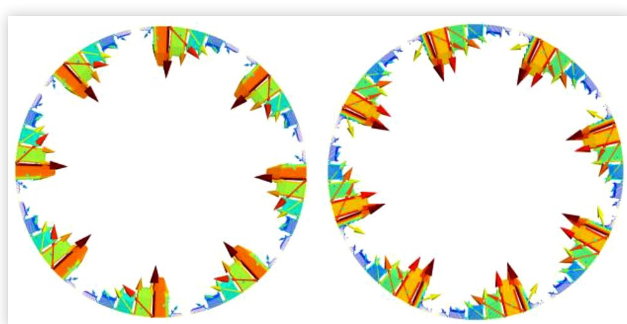


FIGURE 3 Reduction of the loads to the stator teeth at 2 time steps t=0.0s (left) and t=0.00312s (right).



The NVH analysis is usually performed in frequency domain, so the next step consists in transforming the data from time domain to frequency domain for each RPM. A Fourier Transform is applied at engine orders available for each RPM. The orders chosen here are 8 to 96 with a step of 8. The data are reorganized by engine orders leading to a spectrogram of excitations (Figure 6). Once working by engine order, it is possible to increase the resolution by interpolating results along the orders: from 10 initial input RPMs computed by EM calculation, the resolution is increased to 100 values per harmonics. The advantage is to better capture the resonances that will occur with structure (see next section). It can be observed that engine order 8 is dominant and the energy of each following harmonic is constantly decreasing.

FIGURE 4 Space-Time spatiogram of electro-magnetic forces at 1000 RPM.

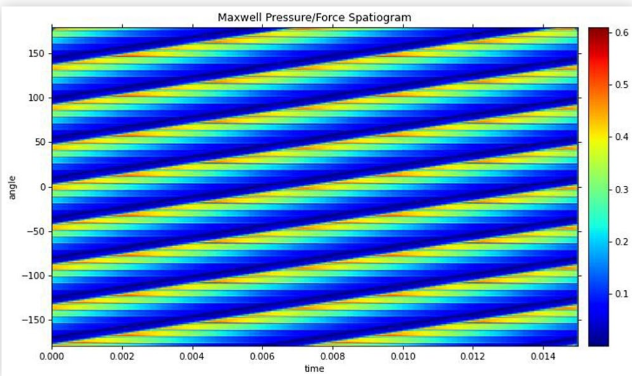


FIGURE 5 Extrusion of 2D EM loads to 3D field of loads and alignment with structure meh teeth.

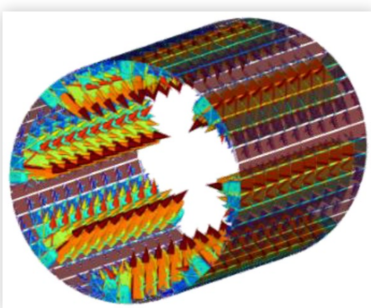
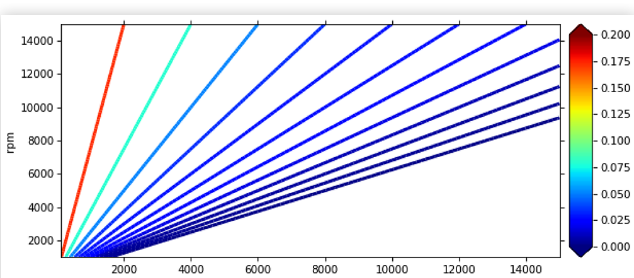


FIGURE 6 Spectrogram of electromagnetic excitation (RPM vs. frequency).



The spatial evolution of the electromagnetic field can also be studied. A space decomposition is performed by projecting the field of excitation on space orders shapes 0 to 96 by step 8, similarly to the frequency harmonics. A spatial representation of some of these orders is displayed in Table 1. These modes are rotating modes so they can rotate in positive (+) or negative (-) direction. Note that no decomposition is performed on radial and axial directions of the motor. As no skew is applied, the only axial order of interest is 0.

Once both frequency and space transformations are performed, it is possible to visualize the space/frequency diagram of excitation of electromagnetic loads (Figure 7). This type of representation is commonly used [5, 6, 7] to better understand the risk of resonance between the electromagnetic loads and the structure modes, in the early phase of e-motor NVH design. As expected for this type of machine topology, the spatial shape of each harmonic is mainly the same order value as the harmonics value. Nevertheless, some space harmonics are present leading to specific phenomena. For instance, it is visible that the harmonics 24, 48, 72 and 96 are containing an important pulsating component (space order 0). We can then expect that this component will excite the breathing mode of the stator.

TABLE 1 Example of Space orders selected from azimuthal orders 0 to 96.

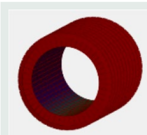
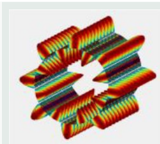
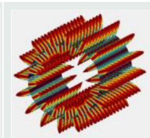
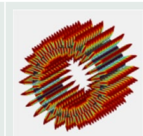
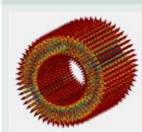
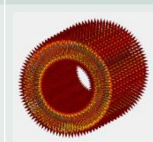
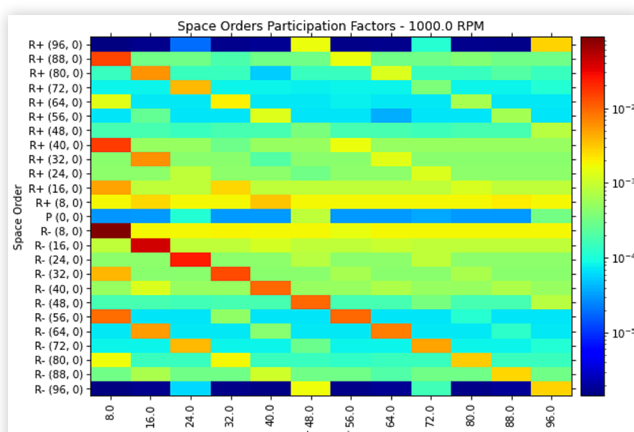
			
0	8	16	24
…		…	
…	48	…	96

FIGURE 7 Space/Harmonic diagram of electro-magnetic excitation.



3.2. Structural Model

The structural model consists in a stator piece made of steel. Windings are added and made of copper. The eigen modes of the structure are extracted from 0 to 15 kHz leading to 44 modes. The bending modes of cylindrical structures are usually defined using azimuthal *m* and axial *n* indices. These modes are referred in Table 2 with their eigen frequency. Additional modes can be identified like torsional modes and compression modes, but they are not presented here.

3.3. Structural Response

As seen in section 2.1 and 2.2 both electro-magnetic and structure bending modes can be identified by azimuthal and axial orders. This concept helps to understand that some structure modes can be highly excited by electro-magnetic field and resonate.

The structure response due to electro-magnetic excitation is computed as per equation (3) and with the mapping technique explained in section 1.4. The result is the participation factor of each structure mode due to each excitation. The structure results can be seen along the following dimensions:

- structure mode id (or eigen frequency)
- EM harmonic (or engine order)
- EM space order

We will focus here on the structure response by harmonic. Figure 8 is a representation of the participation factor for harmonic H48 and each of the 44 structure modes. This graph is available similarly for each harmonic from H8 to H96. A general analysis of these graphs leads to the following comments:

- The first 6 modes are the rigid body modes of the structure with eigen frequency of 0 Hz. They are mainly excited at low frequency.
- Harmonics H24, H48 and H96 seem to excite more the structure than other harmonics

TABLE 2 Structure bending modes.

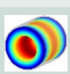
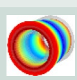
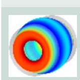
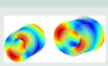
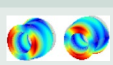
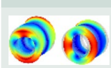
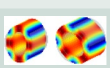
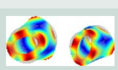
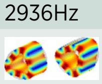
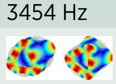
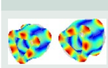
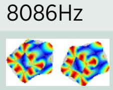
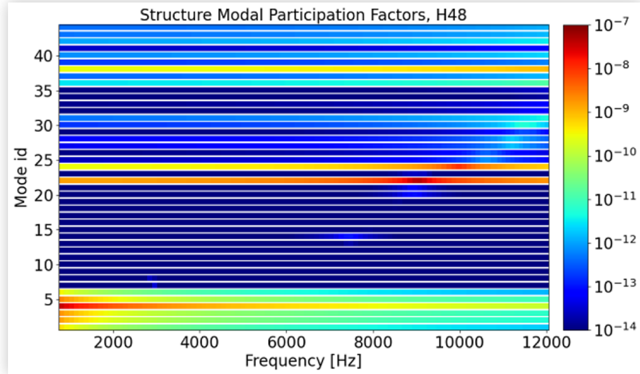
n→ m ↓	0	1	2
0	 9016 Hz	 9124 Hz	 9955 Hz
1	 7031 Hz	 8916 Hz	 13515 Hz
2	 2936Hz	 3454 Hz	 8521 Hz
3	 7438Hz	 8086Hz	 11205 Hz
4	 12650 Hz	 13242 Hz	

FIGURE 8 Participation factors of the structure to EM excitations (a. u.) represented by harmonic H48.



- 3 non-rigid modes are especially excited: modes 22, 24 and 38

The modes 22, 24 and 38 are represented on Table 3. They are identified as being linked to breathing, bending and compression modes of the stator. It is worth mentioning that the modes of azimuthal orders 1 or 2 are usually dangerous for noise emitted by electric motors. But they are not excited by the electromagnetic forces here because of high number of teeth (48) and pole pairs (8).

3.4. Acoustic Model

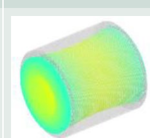
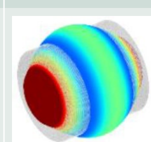
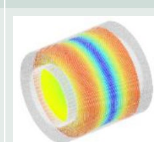
As stated in section 1.3 the radiation of each structure mode is performed using Finite Element method with Actran acoustic software [10]. This is providing the radiation efficiency of each mode over frequency. The pressure is obtained at 40 microphones located at 1m around the structure following the ISO3744 standard. In the current process this calculation can be seen as acoustic transfer functions between structure modes and microphones.

The radiation is computed using adaptive mesh and Infinite Elements method as non-reflecting boundary conditions.

The radiation efficiency σ is calculated according to the following formula:

$$\sigma = \frac{W_{acoustic}(\omega)}{W_{mechanical}(\omega)} \tag{9}$$

TABLE 3 Main structure modes excited by the electro-magnetic forces.

	Mode 22	Mode 24	Mode 38
Eigen frequency	9016 Hz	9955 Hz	14191 Hz
Mode shape			
Identification	Breathing mode (0,0)	Bending mode (0,2)	Compression mode

where $W_{acoustic}$ is the Sound Power Level radiated (in Watt) and $W_{mechanical}$ the mechanical power on the external surface of the stator (in Watt).

The result (Figure 9) shows the radiation efficiency for all 44 modes as well as the envelope. The general trend is an increase of the radiation efficiency with frequency. However, the increase of efficiency is not linear and depends on the mode. Some modes are especially efficient to radiate, and others are not as efficient. This is directly linked to the space pattern and deflection shape of the modes. Mode 15 which is a torsional mode has very low efficiency. But modes (0,0), (0,1) and (0,2) are very efficient to radiate, even at low frequency.

3.5. Acoustic Results

The final acoustic results are the combination of the following ingredients:

- Electromagnetic forces (engine order/frequency, space order)
- Structure participation factor per mode
- Radiation efficiency of each structure mode

The acoustic result is available for each microphone around the structure, expressed as Sound Pressure Level (SPL) in dBA. The Sound Power Level (SWL) in dBA is also obtained following the ISO3744 standard (Figure 10). As observed for structure response, the orders 24, 48 and 96 are dominant. However, the weightings due to the radiation efficiency and A-weighting lead to a dominance of harmonics 48 and 96. In a certain way, the noise emitted over frequency is inversed

FIGURE 9 Radiation efficiency of the modes as curve plot (top) and bar plot (bottom)

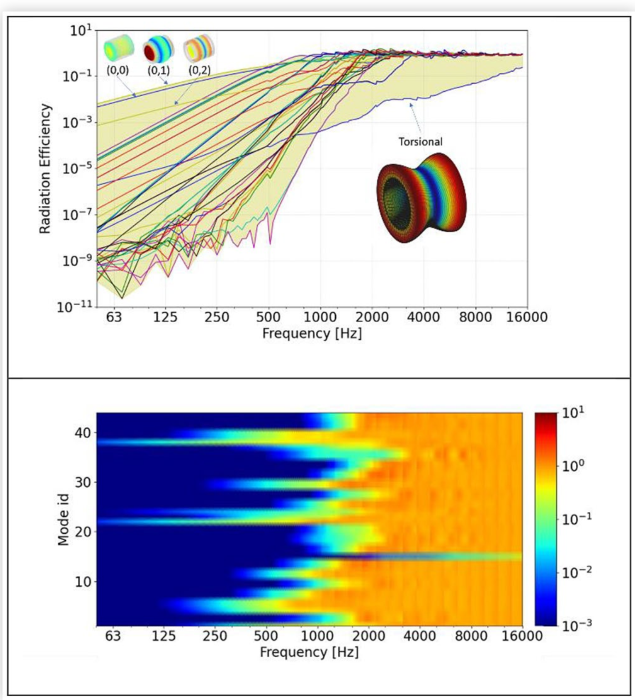
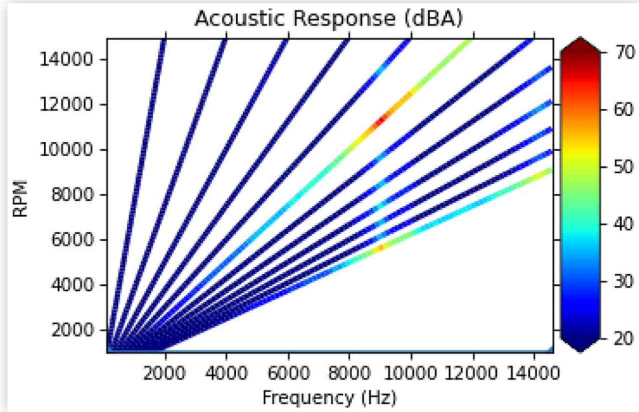


FIGURE 10 Sound Power Level (SWL) of the stator.



compared to EM excitation of Figure 6. The high level of excitation of order 8 is exciting the structure in a range of frequencies (0 to 2000Hz) where the structure radiation efficiency is poor. The two peaks observed around 9000Hz and 10000Hz are due to resonances of the modes (0,0) and (0,1). Contribution of compression mode is also visible around 14000Hz and is excited only by harmonic H96 because of its high frequency.

Finally, the Overall Sound Power Level (OSWL) over RPM is determined by summing the contributions of each harmonic for each RPM (Figure 11). The total noise emitted is mainly led by harmonics H48 and H96.

Thanks to the recombination approach, it is easy and fast to get results only for some space order modes. The study of the shape of the electromagnetic forces (Figure 7) had shown that harmonics 48 and 96 where mainly supported by shapes azimuthal orders 0, 48 and 96. These space orders are used separately to get their own contribution to the noise emitted (Table 4). It is clearly visible that the azimuthal order m=0 is the main contributor for both harmonics H48 and H96.

FIGURE 11 Overall Sound Power Level (OSWL) along rotation speed.

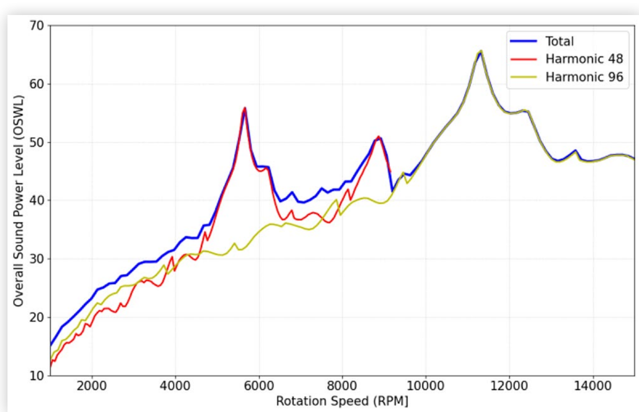
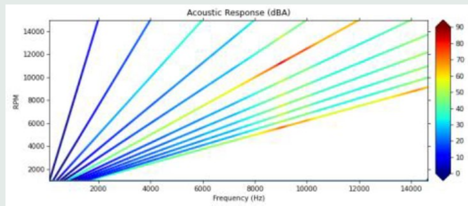
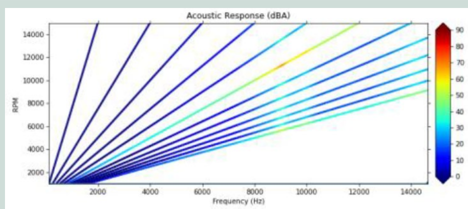
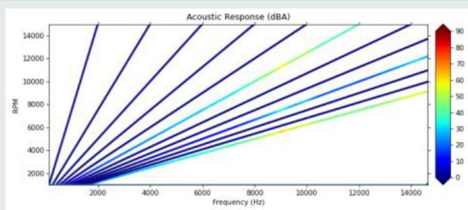


TABLE 4 Sound Power Level (SWL) of the stator for main space orders modes radiating.

mode (0,0)	
mode (48,0)	
mode (96,0)	

4. Discussion and Conclusion

A methodology has been presented to study the noise radiated by electric motors without any compromise on accuracy using Finite Element method:

- Electro-Magnetic FE data (excitation)
- Structure FE data (structure dynamics)
- Acoustic FE data (radiation efficiency)

The three ingredients are combined together in a single process including:

- Automatic pre-processing of electromagnetic forces:
  - 2D to 3D loads
  - Time domain to frequency domain transformation
  - Reorganization of input loads per engine order instead of RPM
  - Interpolation along orders to increase frequency resolution over RPM ramp-up
  - Space decomposition of electromagnetic loads that are used for further recombination with structure and acoustics
- Combination with structure modes to get structure vibration
- Combination with acoustic radiation efficiency to get acoustic results

This process is generic and allows inputs from various electromagnetic software and structure FE software available in the market. The acoustic radiation is based on Actran Finite Element solver which offers efficient capabilities to address acoustic radiation. The tool also comes with a dedicated graphical interface to ease the process and the analysis of the results.

The process itself is combining electromagnetic field on structure finite element model, structure modes and modal acoustic radiation. The first step consists in analyzing the electromagnetic loads both in frequency sand space domain; the second by analyzing the EM contribution applied on structure modes and the last step by weighting these results by acoustic radiation efficiency of the modes. This approach is very flexible and helps to determine the root cause of the noise radiated than can be either due to high EM loads or resonance of the structure, considering the radiation efficiency over frequency. Acoustic results can be obtained locally at any microphone in space or as Sound Power Level but can also be analyzed harmonic per harmonic or for any space order contribution.

The technique has been applied to an electric motor with a typical topology used in automotive industry (48s8p PM). It helps to understand the root cause of the noise thanks to a deep analysis of the results at each step and potential risks in noise emitted.

An important advantage is the pre-processing capability of input data from any EM software that is avoiding manual steps in the preparation of the loads for structure response, as well as the reorganization of the loads per engine order. Due to the flexibility of the technique, new results obtained from new set of electromagnetic would be easy to obtain. Indeed, there is no need for new structure modal extraction neither acoustic radiation, but only a recombination of the ingredients, which is fast to compute. The acoustic radiation can also be modified by including encapsulation effect from multi-layer acoustic treatment (typically foam + heavy layer) or to study propagation in modified surrounding environment such as engine bay cavity or microphones at pass-by locations instead of ISO3744 positions. Additional results are available and can be extracted to address transfer path analysis inside vehicle cabin: reaction force at mounting points of the electric motor (structure borne path) and noise at wall surface (firewall, airborne path) for further NVH analysis.

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Definitions/Abbreviations

ICE - Internal Combustion Engine
NVH - Noise Vibration and Harshness
EM - Electro-Magnetic
FE - Finite Element
SPL - Sound Pressure Level
SWL - Sound Power Level
OSPL - Overall Sound Pressure Level
OSWL - Overall Sound Power Level
RPM - Rotation Per Minute

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