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Noise radiated by electric motors: simulation process and overview of the optimization approaches

Henri Saucy[1], Jean-Baptiste Dupont[1], Pascal Bouvet[1]

[1]*Vibratec SA, 28 Chemin du Petit Bois, 69131 ECULLY, henri.saucy@vibratec.fr*

Summary

Electrification of automotive powertrains has brought new challenges to the industry. High expectations in interior comfort and control of pass-by noise require careful design and integration of traction and hybridization motors. New tools have to be developed to set and meet acoustic targets compatible with safety, power, and weight constraints. This paper introduces an acoustic optimization scheme based on state of the art simulation workflows, and its application to an industrial case.

Keywords: case-study, motor design, optimisation, powertrain, simulation

1 Introduction

1.1 Noise from electric vehicles

The powertrain of electric (EV) and hybrid vehicles (HEV) is noticeably quieter than the traditional internal combustion engine (ICE), but is not silent. Electrical machines have a lower overall noise level and usually display a tonal quality that makes it more annoying to the human ear. Next to that, the overall acoustic level of the main propulsion system being lower, noise from ancillary systems becomes more noticeable: accessories, pumps, heating and ventilation (HVAC), etc.

The push towards electric propulsion requires a paradigm shift in acoustic design of vehicles: artificial exterior noise synthesis at low speed for safety combined with a quiet environment for the passengers. Acoustic signals have to be provided to the driver to provide necessary information such as motor speed, road surface, possible malfunction, etc. The experience accumulated with ICE must be translated for EVs. On this respect, autonomous vehicles offer an even greater challenge of comfort and sensory information.

1.2 Acoustic design of vehicles

This difficult task has now landed on the desk of the industry’s noise, vibration, harshness (NVH) engineers. A vehicle can be seen as a dynamic system where energy is converted from one form to the other. Typically, electrical energy is turned into mechanical energy, resulting in vibration propagating through the vehicle structure, and, eventually, into noise emission at the interface with air, and its propagation in the environment. The task of NVH engineers is to understand those propagation phenomena, characterize sources and transfer properties, and ultimately design vehicles and subsystems according to noise and comfort guidelines. The typical workflow consists in setting-up specifications for the whole vehicle, breaking them down in specific targets for each subsystem, choosing and evaluating solutions, and integrating those components in the vehicle.

1.3 Electromagnetic noise

Noise and vibrations generated by an electric motor can be divided into three main contributions related to three distinct sources [1]: Mechanical, aerodynamic, and electromagnetic.

Mechanical noise and vibration is related to assembly conditions: friction, loss of contact etc. at bearings, gears, and other interfaces. Those phenomena are highly dependent on rotational speed. So is aerodynamic fan noise, whose acoustic power is related to the power 5 of the rotational speed [2].

The electromagnetic excitations result not only in an overall noise level, but in tonal contributions: the engine "whistles". Even if the sound power radiated is lower than that of an ICE, the noise can be extremely annoying: psychoacoustics, noise perception have to be taken into account.

Between the source and the passenger or bystander, there are several transfer paths for noise and vibration generated by an electric motor (Fig. 1):

- The airborne noise directly radiated by the motor frame,
- The structure-borne noise due to vibration transmitted to the vehicle structure by the stator through its mountings,
- The noise due to the excitation of gearboxes and couplings downstream from the motor, and caused by torque ripple.

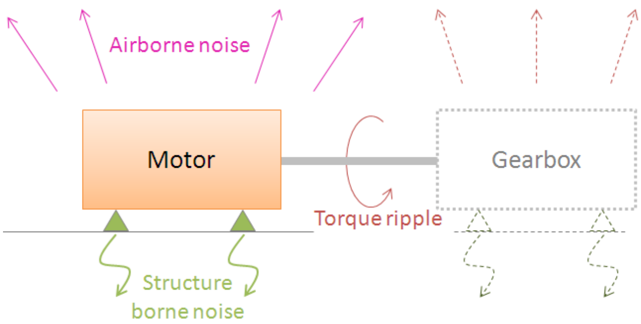

Figure1: Main vibro-acoustic transfer paths

Automotive NVH engineering requires several type of specific tools: measurement methods to characterize materials and structures, simulation tools to estimate the contribution of a component and set-up targets, as well as to anticipate integration in the full vehicle. At the current time, acoustic powertrain optimization methods are much desired to meet acoustic targets, but is still a complex task due to a large amount of constraints to be simultaneously handled: structural strength and stiffness for safety, weight, mechanical power for energy efficiency , thermal properties etc. [13]

Meeting NVH acoustic and comfort targets when the electrical machine has been primarily designed to meet those safety and efficiency constraints is hard and expansive. More often than not, it results in added mass to damp unwanted noise. Suppliers may lose markets after accomplishing all the development effort because the last item on the list, the acoustic target, is not met.

This paper focuses on noise from electromagnetic excitation and transmitted through airborne and structure borne transfer paths. It introduces tools to design acoustically sound machines and integrated, multi-physics optimization approaches [3].

2 Acoustic simulation of electric machines

2.1 Principle

Simulating noise emission due to electromagnetic excitation in an electric powertrain requires coupling several phenomena, usually handled by very different physical models as seen in Fig. 2: energy supply, electromagnetic forces, vibration response of the structure, acoustic propagation. Standard tools exist to measure and simulate each step. The last decade has seen the development of specific methodology to combine those models into a seamless simulation workflow [3].

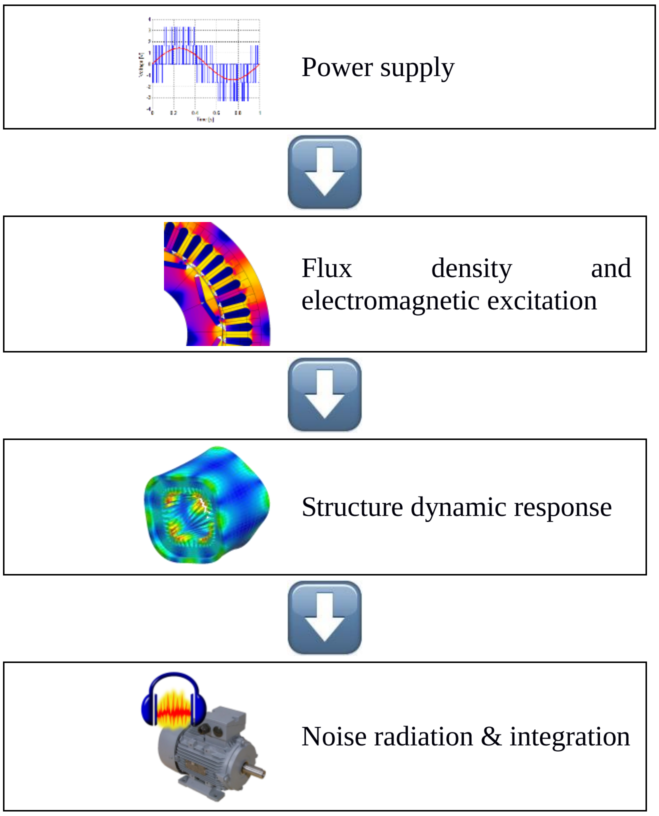

Figure 2: Principle of noise generation from electromagnetic sources

This methodology requires the estimation of the forces applied to the stator, to calculate the dynamic response of the housing, and then deduce the radiated noise. This multi-physics problem requires to combine elements of electromagnetic calculations, dynamic calculations and acoustic calculations performed using numerical finite element or boundary element methods. This type of calculation has been presented and implemented by several authors. Neves et al. [4,5] applied it to a switched reluctance motor (SRM) by using simplifying assumptions. They showed the relevance of the approach. Furlan et al. [6] used this methodology to calculate the noise radiated by a DC electric motor. Schlensok et al. [7] applied it to an induction machine with a squirrel-cage rotor. Rainer et al. [8] computed the dynamic response of a skewed induction machine and studied the accuracy in the frequency domain. Pellerey et al. [9] also applied this methodology to a wound-rotor synchronous motor. Arabi et al. [10] proposed some enhancement in the simulation process.

The implementation of the simulation method has already been detailed [11]. This article focuses on the optimization opportunities given by this simulation method to reduce the noise radiated by electrical machines.

The basic principle of the calculation is to perform a weakly coupled electromagnetic-dynamic calculation [9]. The electric motor is modeled using a finite element electromagnetic software program in order to calculate the electromagnetic excitations applied to the stator (see Fig. 3).

This excitation data is projected onto the structural mesh of the e-machine with the aid of a dedicated mapping tool and the dynamic response of the structure can be calculated using a finite element method. In an acoustic scope, the output value of interest is the vibration velocity of the stator outer shell. The last step is the calculation of the vibrating structure acoustic radiation. For estimation of the airborne noise emission, an acoustic finite element method (FEM) is used. Alternatively, a boundary element method (BEM) or an analytical model can also be used. Vibration velocity is taken into account as a boundary condition. The output data is the sound power radiated by the machine. Since this is a typical calculation, this step is not detailed in this paper. Fig 3 shows a scheme of the 3-step multi-physics calculation procedure. A corresponding finite element of the supporting structure including an adequate description of the dynamic behavior of the interface is required to calculate the structure borne noise transfer.

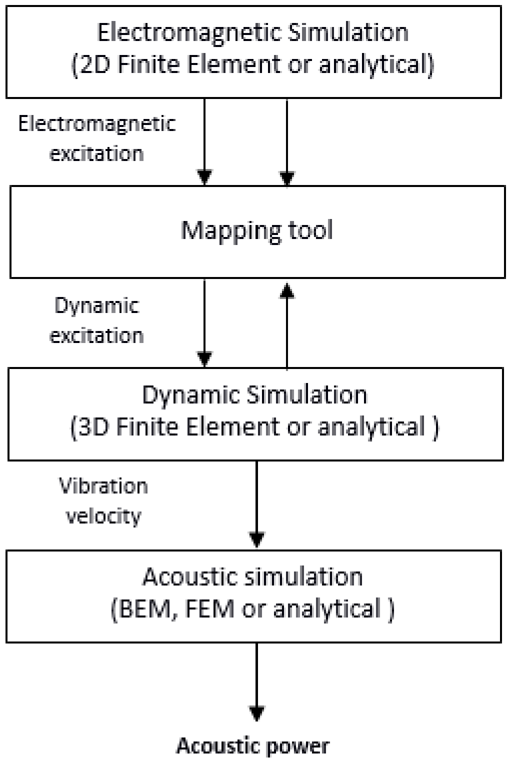

Figure 3: Basic principle of the calculation procedure

2.2 Electromagnetic excitation

Maxwell pressure in the airgap is taken as the main phenomenon responsible for the stator vibration and the stator radiation [1, 12]. When a magnetic flux crosses the interface between two materials of different ferromagnetic properties, a surface force density appears at this interface dependent of the airgap flux density. This Maxell pressure acts as a dynamic excitation on the structure of the machine, mainly in the radial direction, while its tangential component create the machine torque. Flux density and thus Maxwell pressure can be calculated using finite element electromagnetic solvers.

Maxwell pressure is calculated numerically for virtual sensors located inside the air gap. Since the rotor is moving, the electromagnetic solver is based on a time resolution so that the electromagnetic calculation results in a time evolution of the magnetic excitation at each virtual sensor. Thus, for each motor speed, the electromagnetic simulation provides two time-space excitation matrices (one related to the radial component, the second related to the tangential component). The influence of every relevant parameter is contained in this simulation: number of poles, of stator slots, and rotor slots, current shape, eccentricity, and saturation of the magnetic core. These parameters affect the excitation content in the time domain as well as its spatial distribution.

2.3 Excitation projection onto the structural model

When linking results of the electromagnetic simulation with a finite element model of the dynamic response of the structure, several difficulties appear:

- The mesh sizes of electromagnetic and structural models are incompatible,
- Electromagnetic simulation is a 2D simulation while the dynamic simulation is 3D,
- Electromagnetic simulation provides excitation as pressure while dynamic simulation requires a force,

- Electromagnetic simulation is performed in the time domain while the frequency domain is preferred for dynamic and noise calculations.

The first step is to transform the time excitations into a sum of frequency excitations with the aid of Fourier series. Then, the Maxwell pressure (expressed in N/m²) has to be projected onto the structural mesh and transformed into forces (in N). For each interface node (Interface nodes are the nodes located close to the air gap, where an excitation is applied) of the structural mesh, the corresponding surface force density is multiplied by the area of the surface located around the node (this value is calculated and depends on the topology of the mesh structure). Thus, two excitation spectra (radial and tangential) are calculated independently for each interface node of the structural mesh. Excitation thus calculated is then applied to the structural model in order to obtain dynamic response in the frequency domain. This mapping algorithm can be applied to any type of structural model.

2.4 Dynamic and acoustic response

The dynamic calculation is based on the modal frequency response. In a first step, the modal basis of the structure is calculated. By knowing the applied loads, it is used in order to estimate the structural response of the stator. The structural velocity can be used as a boundary condition to calculate the radiated power.

In most application cases, the structure's geometry can be approximated with a cylinder, the stator deformation modes (examples shown on Fig. 4) can be described using two integers (m,n) giving:

- m: the number of maxima over the circumference. It is called circumferential spatial order.
- n: the number of nodal circles (sections with a null displacement) in the length.

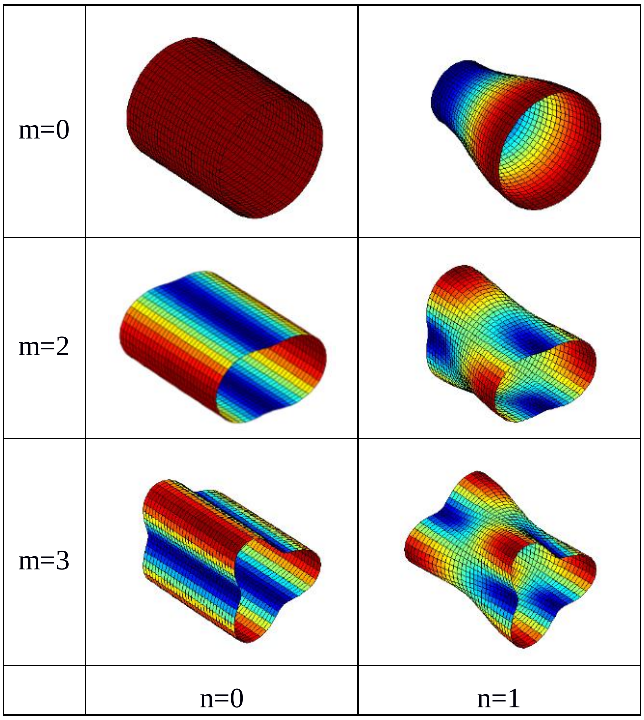

Figure 4: First modal shapes of a cylindrical structure

Highest levels of dynamic response are observed when system resonances occur, i. e. when there is both a space and frequency coincidence between an excitation contribution and a structure modal behaviour. In order to analyse and understand the dynamic behaviour of the machine structure under realistic electromagnetic excitation, both the modal basis and the electromagnetic excitation must be considered.

Radiation efficiency reflects the ability of a structure to turn vibration into noise. On this regard, the most acoustically significant modes are the low spatial order deformation modes. If those modes are excited, a large fraction of the mechanical energy will be transformed into acoustic energy. Thus, for a given excitation frequency, mostly the low spatial order modes contribute to the radiated sound power, typically for an uniformly excited cylindrical shape: Stator breathing (0,0), stator bending (2,0), and stator bending (3,0).

In order to predict resonance, one has to to compare the excitation and the cylinder modes. In practice, the electromagnetic excitation is decomposed into elementary rotating forces characterized by their frequency f and their spatial order m, leading to a spatial order-frequency excitation matrix that conveys the content of the electromagnetic excitation and providing a powerful tool to estimate the critical speeds and frequencies.

In this analysis the electromagnetic excitation is actually made by a small number of contributions: the stator is excited for a few spatial orders and a few frequencies dependant on the main characteristics of the machine (number of poles, number of slots, eccentricities…). These elementary rotating forces are the cause of the dynamic response of the stator and its acoustic radiation.

This computational scheme has been validated and applied to several industrial cases in the last years [12].

2.5 Industrial application

The acoustic level of a newly designed alternator was found by its designer, VALEO, to exceed the OEM targets at certain speeds within the range of interest. The previously exposed methodology is applied to predict acoustic performance of refined solutions aiming at reducing electromagnetic excitation [3].

Because of the non-prismatic shape of the machine rotor (Fig 5), the electromagnetic excitation has to be simulated with a 3D software, and projected on a mechanical model build by VIBRATEC.

Figure 5: Alternator rotor and its claw shape poles

The same electromagnetic model is used by VALEO to predict the performance of the machine (resistive torque, generated current, etc.).

Prototypes of the first version of the alternator have already been manufactured at that stage. This allows to perform an experimental modal analysis to tune the structural finite element model used to simulate the dynamic response, noise emission, and operating deflection shapes.

Fig. 6 shows the Campbell diagram (equivalent radiated power in function of machine speed and acoustic frequency) in the range of interest identified by the manufacturer, between 2100 rpm and 2400 rpm. The critical speeds are easily identified (yellow circles) along the machine harmonics, as well as the associated deflection shape (ODS) resulting from dynamic FE simulation. This established multi-physics modelling workflow allows engineers to test design variations virtually in a very quick and efficient way, without the need for physical prototypes, saving high costs and reducing delays.

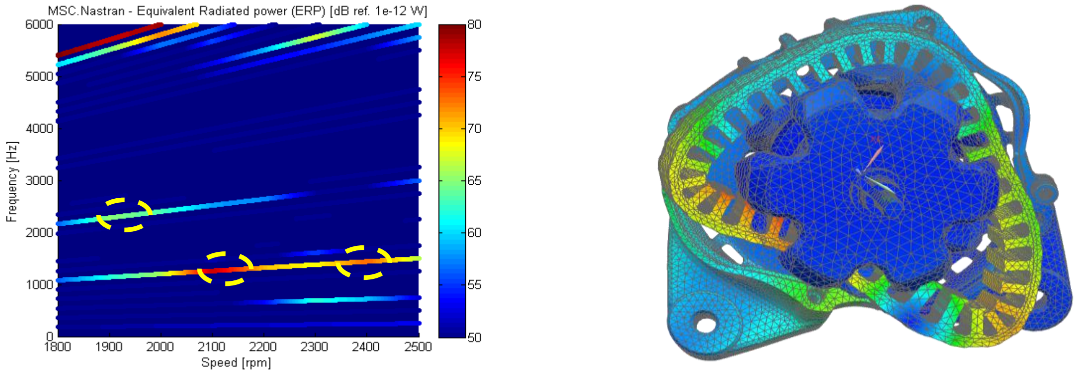

Figure 6: Left - Simulated run-up and critical speeds in yellow circles. Right - ODS of harmonic H72 at 1900 rpm

At that stage, margins for modification of the design are already very narrow. Most of the exterior design of the machine is frozen and weight constrains strongly limit stiffening of the housing. A step by step, manual optimization of the excitation was undertaken, focused on small alteration of the shape of the rotor claws. After each design iteration, electrical performance and noise radiation are evaluated in parallel, using identical simulation tools. A convergence criterion based on sound emission at the main harmonics of the machine and the acoustic targets imposed by the OEM customer is used.

Several optimization steps have been performed and evaluated in simulation towards the acoustic and performance criteria. A prototype of the most promising configuration was built and the acoustic emission measured shown in green on Fig. 7, with overall sound power level plotted against rotational speed. The initial configuration is displayed as a reference in red. Two noise emergences between 2000 rpm and 2500 rpm are clearly identified. In accordance with simulation results, a reduction of about 10 dB is observed in the speed range of interest. Following up on this promising first results, Valeo kept modifying the claw shape along the same trend, producing a second prototype (in green). This second optimization provides a better acoustic behaviour in the lower speed range, for an even better electrical performance.

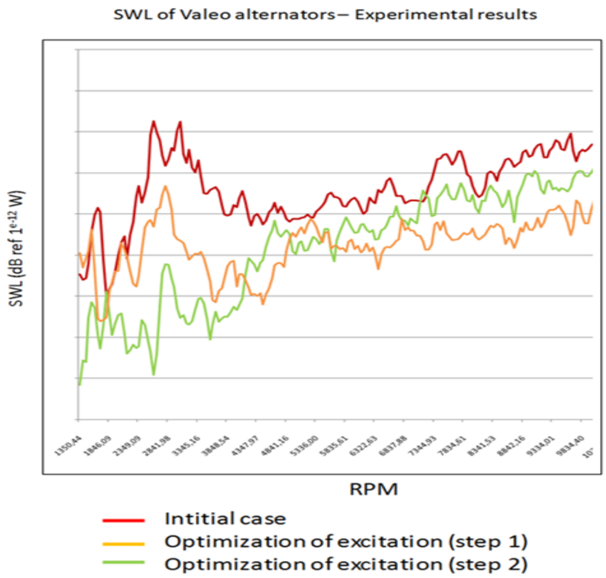

Figure 7: improvement of the sound emitted by the alternator, experimental results.

This example, among many other cases, shows the high potential of coupled electromagnetic and dynamic simulation methods to help design electrical machines with improved acoustic characteristics. It also points towards the need to integrate this workflow in a complete, multi-physics evaluation and optimization process in order to produce better, more robust designs from the beginning. This approach is described in the next section.

3 Acoustic optimization of electric machines

3.1 Method

In principle, it is possible to take action at each step of the noise generating process described in section 2 to improve the acoustic behaviour of electric machines, namely optimizing the power supply strategy, the electromagnetic flux in the air gap between stator and rotor, the vibratory behaviour of the machine structure, or its radiation efficiency for airborne transfer and its interface with the vehicle structure for structure-borne transfer respectively.

Optimizing the power supply, for example the pulse width modulation (PWM) strategy, by acting on the switching frequency and shape of the input voltage has proven significant results in industrial cases [14]. Modifying the shape of the rotor and stator active parts, such as described earlier for an alternator, with the aim of minimizing vibratory energy transfer to the machine from the Maxell pressure in the air gap requires a geometric optimization. This process is described below, and is valid for any type of electric machine. Acting on the machine structure itself to avoid resonances provides good results when significant changes to the machine housing are allowed. An automatic procedure requires topologic optimisation schemes, and is computationally costly. Finally, it is also possible to work on the transfer of the machine vibration to its environment. Optimization of radiation efficiency offers limited gains since most electric motors have already compact cylindrical shapes. On the other hand, optimization of the coupling elements to minimize structure-borne transmission is a demanding, yet routinely performed engineering task.

Optimization of the active electromagnetic parts of the machine requires as input three sets of parameters: the acoustic target to achieve as objective function, the free geometric parameters of the machine, and the other, non-acoustic targets and specifications as constraint function. In industrial cases, acoustic targets are typically of two kinds, depending on the original diagnosis and specification: a large reduction of sound level in a narrow frequency band at a specific point of operation, or an overall noise performance spread over the entire spectrum and range of operation. Those criteria can also be combined.

At first approximation, geometric optimization can be confined to a careful selection of the slot-pole arrangement. However, in practice, this approach must be refined and include several geometric parameters in the objective function. For example, Lin et Al. use slot opening and skewing angle [15], while Wang et Al. take pole arc ratio and slot opening width into account [16]. Depending on the application, constraint function may take into account mean torque, max torque, efficiency, etc.

The workflow presented in Fig. 3 is applied to compute electromagnetic excitation, transfer it to the structure, and compute the dynamic and acoustic response of the structure. The optimization algorithm calculate the objective function (acoustic power level, depending on the targets set) and finds an optimum in the parameter space defined by the free geometric parameters. At each iteration, the constraint function resulting from non-acoustic targets and limitations is evaluated.

3.2 Case-study

This optimisation algorithm has been used on a 4-poles, 48-slots wound-rotor synchronous machine (WRSM, overview on Fig. 8). This automotive traction machine is designed to operate between 0 and 12000 rpm.

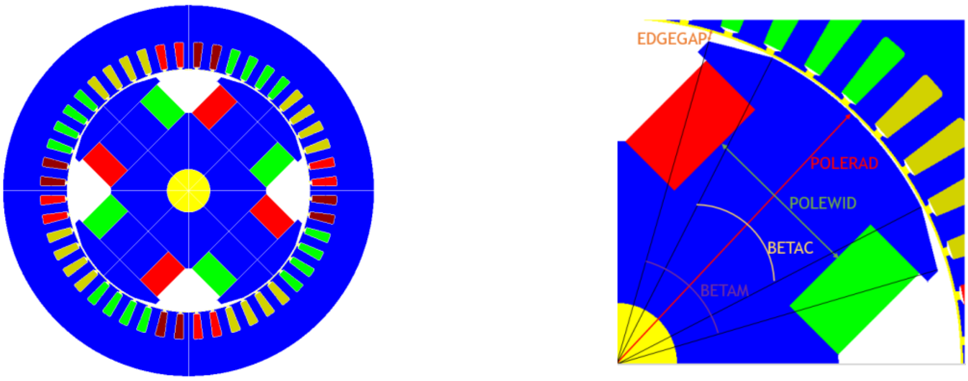

Figure 8: Left - overview of the active parts of the WRSM to optimize. Right – geometric parameters considered

A first simulation of its vibro-acoustic behaviour in starting configuration shows significant contributions from several engine orders and resonances due to the excitation of the 4-lobes mode by H20, H44, H52 and H76 at 5750 Hz, and resonance due to 0-lobe mode by H48 at 6140 Hz. Two critical speeds are identified at 6500 and 7700 rpm. Equivalent radiated power (ERP) is outside the acceptable limits at both speeds, with strong tonal noise emission around 6 kHz. A multi-speed overall acoustic criterion (ERP) is chosen as objective function. Customer accepts a 2% reduction of the mean torque as optimization constraint.

Five geometric parameters are defined in agreement with the customer manufacturing capabilities, describing a correction of the pole shape: radius of curvature, chamfer size and angle (Fig. 8).

Optimization of the machine helped achieve a strong reduction of the acoustic emission. Overall noise level show an improvement of 8.5 dB at 6500 rpm and 3 dB at 7700 while maintaining an average torque of 98% of its original value (Fig. 9). Harmonic ERP at the critical speeds, but also at 4500 and 12000 rpm, is strongly improved (Fig. 10).

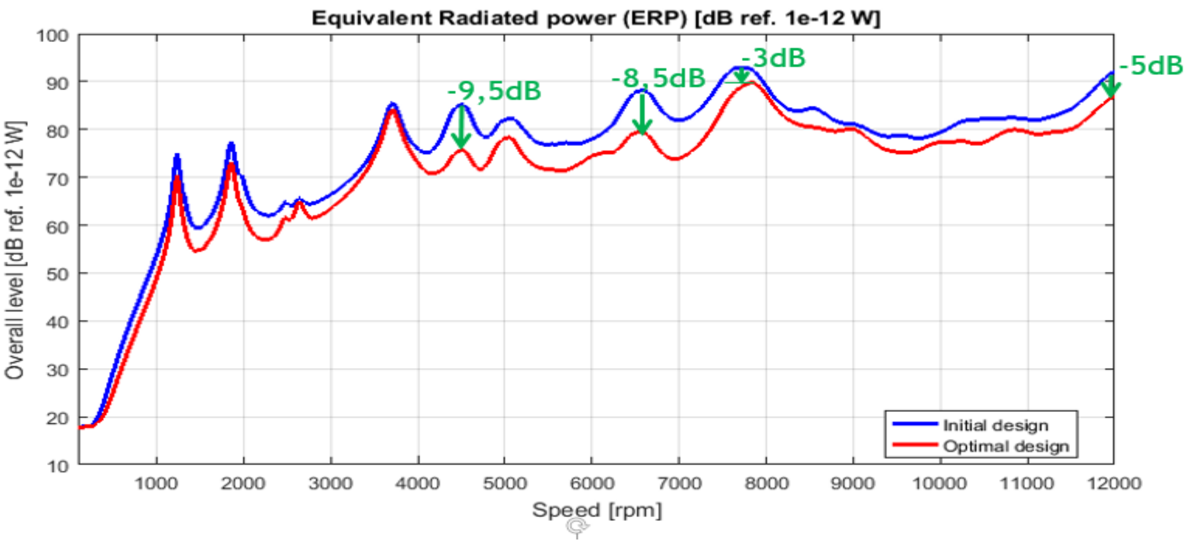

Figure 9: Overall noise level over entire speed range, before and after optimisation.

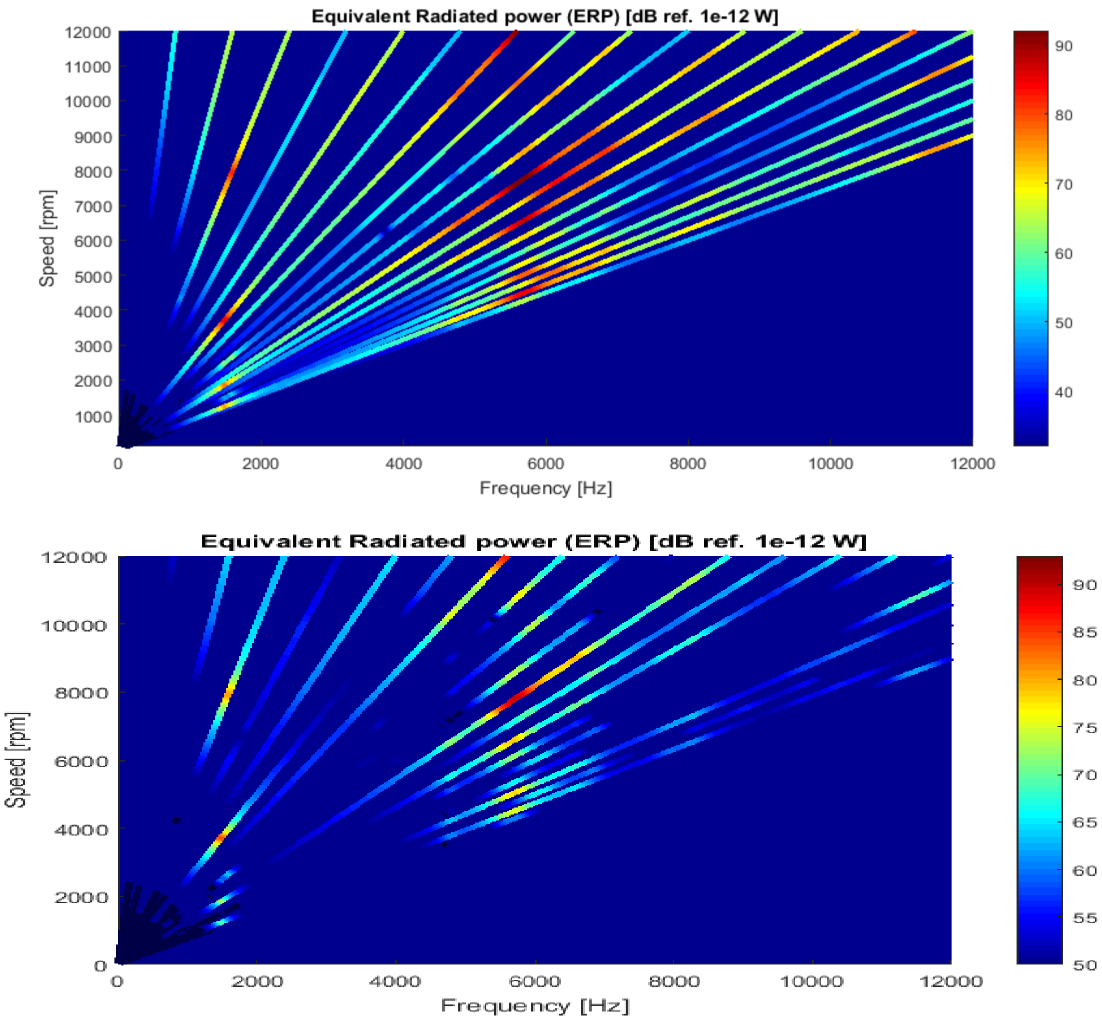

Figure 10: Campbell diagram of the machine before (top) and after (bottom) geometric optimization

3.3 Conclusion and outlook

In order to meet the needs of the automotive industry, a methodology to optimize the acoustic behaviour of all types of electrical machines, based on multi-physics simulation techniques, has been presented. This method focuses on the geometric definition of the active parts of the machine, and allows for optimization of the noise level over the overall speed range, or at targeted critical speeds, under consideration of other design constraints such as efficiency, performance, weight, etc.

This method originates from a long experience with acoustic simulation of electric motors and has been applied to several industrial cases with significant results. It can be advantageously combined with an optimization of the supply strategy (PWM frequency) and housing geometry to provide a complete toolbox for pre-design, design, and vehicle integration of electric machines.

VIBRATEC is currently improving the method, especially the robustness of the optimum solutions, in the framework of the French FUI research project E-SILENCE.

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Authors

Henri Saucy received the M. E. degree in mechanical engineering from the Swiss Federal Institute of Technology, Zurich, in 1999. After working as a design engineer for different industrial sectors, he joined VIBRATEC in 2015. There, he’s working as a consultant for the automotive and railway industries, focusing on the acoustic behaviour of electrical machines, the improvement of railway rolling noise, and railway maintenance.

Jean-Baptiste Dupont received the M.E. and Ph.D. degrees from Ecole Centrale de Lyon, Lyon, France, in 2003 and 2007 respectively. He joined Vibratec, Ecully, France, in 2007. He is the head of the simulation team and his main missions are related to computational acoustics, vibro-acoustics and dynamics. Since 2009, he is in charge of the development of simulation methods of the noise radiated by electrical machines.

Pascal Bouvet received the M. E. degree from University of Technology, Compiegne, France in 1989, and the Ph.D. degree from CSTB, Grenoble, France, in 1996. He joined Vibratec, Ecully, France, shortly after as an automotive acoustics expert. As head of Vibratec Transportation department, he has managed several collaborative research projects related to rolling noise, ground vibration, automotive interior acoustic comfort and powertrain acoustic. Pascal Bouvet is member of the Acoustic Commission of the SIA (French Automotive Engineer Society).