Chapter 28
Vibrational Model Updating of Electric Motor Stator for Vibration and Noise Prediction

M. Aguirre, I. Urresti, F. Martinez, G. Fernandez, and S. Cogan

Abstract In order to improve the comfort of passengers in electrical vehicles, it is increasingly important to consider the vibroacoustic behavior of electrical machines during the design phase. In this work, a weakly coupled multiphysical model for electrical machine vibration and noise prediction is presented and applied to a 75 kW railway traction motor. The main objectives of the model are to obtain firstly the vibrational level and secondly the acoustic pressure level predictions. The multiphysical model includes an electromagnetic 2D model, a 3D structural vibrational model and an acoustic model, all of them based on the finite element method. The work is focused on the validation of the modal analysis and vibrational models, using a bottom-up approach. Experimental modal analyses at different assembly stages are performed in order to update uncertain input parameters of the structural model at those levels. An anisotropic damping model is developed and updated in order to obtain adequate FRF amplitudes and the mean squared error (MSE) metric is employed to quantify the correlation between the experimental and numerical results. Finally, vibrational spectra under nominal operational conditions of the motor are used to demonstrate the adequacy of the vibrational model.

Keywords Electrical machine • Multiphysical model • Vibrational model • Anisotropic damping • Mean squared error metric

28.1 Introduction

For some years now, noise has become a major factor in the ambient quality. Several laws and standards have been written to limit the total amount of noise to a safe and comfortable level [1, 2]. This is why engineers are constantly challenged to reduce the levels of vibration and noise, increase the life expectancy of components and improve the efficiency of machines. Since electrical machines are complex systems where a great amount of physical phenomena are produced simultaneously, a detailed multidisciplinary approach taking into account the coupling between different physical fields is needed in order to implement a fast, accurate, reliable and optimized design methodology. This paper presents a sequentially coupled multiphysical model where electromagnetic, structural vibrational and acoustic aspects are taken into account for a Permanent Magnet Synchronous Machine (PMSM). This type of calculation has been presented and implemented by several authors. For example, Rainer et al. [3] computed the dynamic response of a skewed induction machine and studied the accuracy in the frequency domain. Pellerey et al. [4] also applied this methodology to a wound rotor synchronous motor. Dupont et al. [5] applied to electric motors from automotive industry trying to validate the dynamic response of the stator, and Abrahamsson et al. [6] calibrated a finite element model of a car front subframe against test data.

In the field of structural dynamics, reliable finite element (FE) response predictions are becoming increasingly important to industry and there is a real interest to improve these in the light of measured frequency response functions (FRFs). Model updating using FRFs is a modern design technology which improves the predictive capabilities of computer-based models of structural dynamics problems. Grafe [7] emphasized the importance of using FRF based correlation and model updating formulations instead of modal-based formulations for reliable finite element (FE) response predictions. In this paper three different FRF correlation metrics are presented and an amplitude correlation based metric is employed in the analysis. This correlation measure may be used across the full measured frequency range and any complex response can be

M. Aguirre (✉) • I. Urresti • F. Martinez • G. Fernandez
IK4-Ikerlan Technology Research Centre, Mechanics Area, P J.M. Arizmendiarrieta, 2, 20500, Mondragón, Spain
e-mail: maguirre@ikerlan.es; iurresti@ikerlan.es; Felix.Martinez@ikerlan.es; gfernandez@ikerlan.es

S. Cogan
FEMSTO-ST, Département de Mécanique Appliquée, Université de Franche-Comté, 24 Chemin de l'Epitaphe, Besançon 25000, France
e-mail: scott.cogan@univ-fcomte.fr

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summarized with an error value. Moreover, the importance of giving suitable damping properties is highlighted in this paper. An anisotropic damping model is developed and updated in order to obtain adequate FRF amplitudes. This innovative form of applying different damping values at different directions allows to better correlate the experimental data.

Therefore, the main objectives of this paper are: (a) implement a weakly coupled multiphysical model where electromagnetic, structural vibrational and acoustic aspects are taken into account, (b) perform an experimental analysis in the so called IkerMAQ electrical machine both in free and operational conditions, (c) update mass and stiffness of the model to correlate better the peak frequency values, (d) develop the anisotropic damping model to give directional dependent damping values, (e) perform a sensitivity analysis to analyze the influence of the different parameters and (f) adjust parameter values to improve experimental correlation.

28.2 Multiphysical Model

To model the PMSM from a global point of view, a multiphysical design methodology is developed where elements of electromagnetic finite element calculations, dynamic finite element calculations and acoustic finite element calculations are combined. To implement this calculation methodology, the ANSYS WORKBENCH software [8] for finite element modeling is used. Figure 28.1 shows the three main groups composing the model. In the first group a 2D electromagnetic Maxwell model is defined where electromagnetic forces are calculated. These results are then inserted into the dynamic model where modal and harmonic simulations are performed. The last step consists in the acoustic simulation. This paper focuses on the second group where the vibrational behavior of the electrical machine is simulated and updated.

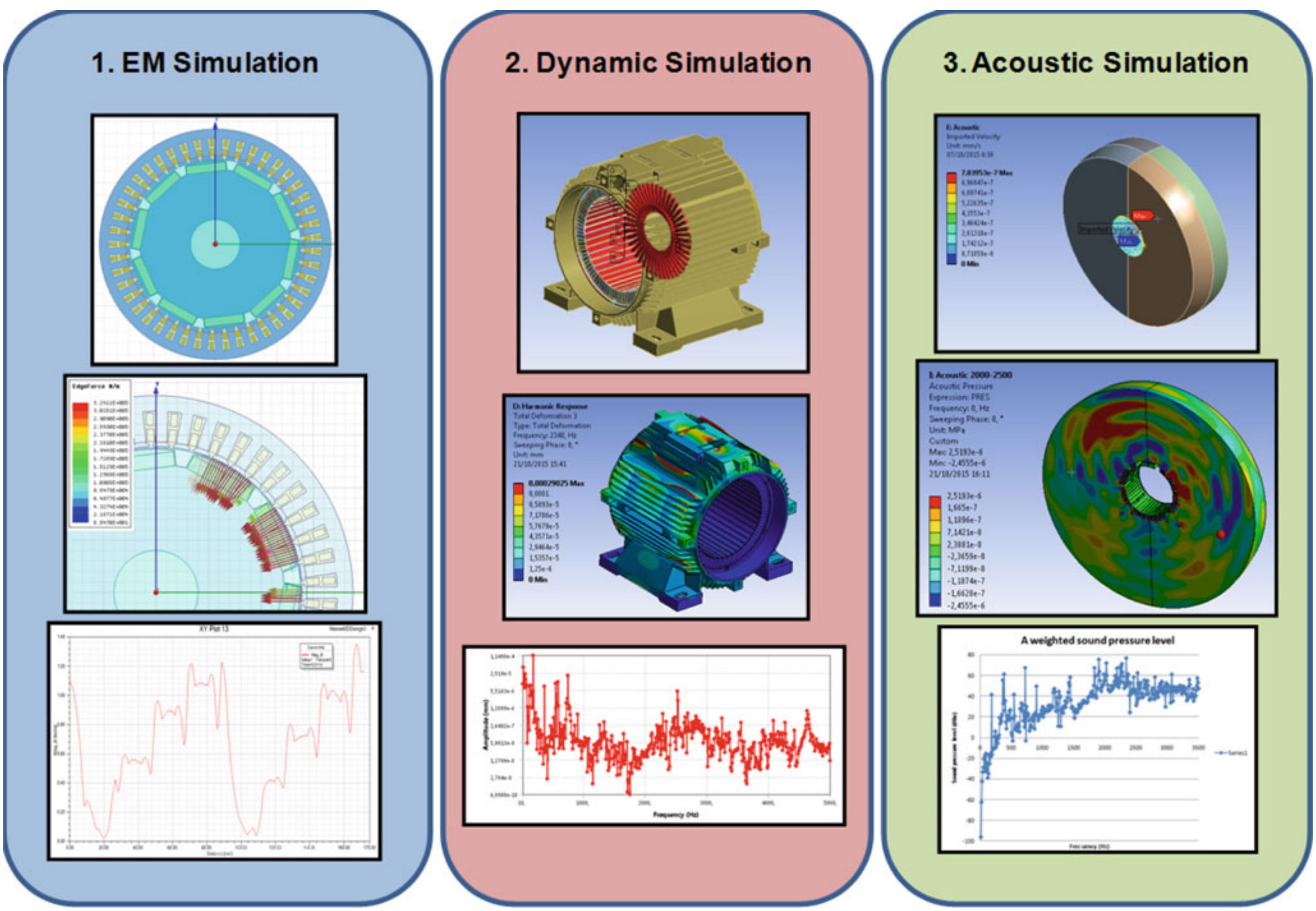

Fig. 28.1 Multiphysical model for PMSM complete modeling

28.2.1 Vibrational Model

Among all components of the electrical machine, the stator is the major responsible of the acoustic noise; this is why the vibrational model analysis is focused on this component. The model includes the housing and the stator core with coils. The material of each element is properly selected; therefore realistic orthotropic behavior of the structure is taken into account. This fact affects the dynamic behavior of the structure, so the damping properties are appropriately defined as presented in Sect. 28.5. Solid elements (solid 186) have been used to build up this model with a total number of 193,225 nodes and 68,437 elements. Harmonic analysis is performed taking into account the electromagnetic forces calculated in the Maxwell model.

28.3 Experimental Campaign

For the experimental campaign, a PMSM (Permanent Magnet Synchronous Machine) was fully designed in IK4-Ikerlan. After an extensive study, the final prototype, called IkerMAQ [9] is a machine with 45 slots and 5 pole pairs, this is, a Q45p5 machine. The nominal electrical output power is 75 kW with a nominal speed of 1080 rpm; this establishes a nominal torque of 700 N.m. The chosen magnets are N40H.

In the work described in this paper, a bottom-up approach is employed. First, single components are measured, correlated and adjusted. After that, the same procedure is applied to subassemblies with increasing number of components. Finally, the whole system is analyzed. For the study of the vibratory behavior of the structure, two different modal analyses are performed: Experimental Modal Analysis (EMA) and Operational Modal Analysis (OMA). EMA is carried out using a shock hammer to apply the external force, while OMA is done during IkerMAQ's normal operation. During the tests the Brüel & Kjær platform PULSE is used and piezoelectric triaxial accelerometers are placed at different locations.

The EMA's main objective is the identification of the modal shapes of the structure under test including the damping factors and resonance frequencies of each mode. Two stator EMAs were carried out during IkerMAQ manufacture process.

Once the test campaign is done, it is possible to identify the different experimental modal shapes with its resonant frequencies and the damping factors and correlate them with those simulated in FEM. All the studied modes are vibrating only in the radial direction and they have both, tangential and axial dependency. For this particular situation, the only modes of interest are the radial ones as they have a major contribution in noise emission.

As vibrational behavior and, hence noise, are going to be maximum at nominal values, a test campaign at this working conditions is performed with the PMSM in generator mode. Figure 28.2 shows the test configuration.

28.4 Baseline Model Definition

For a reliable model of the electrical machine, the first step consists on building a suitable FE baseline model. If the frequency values of the radial forces are close to any eigenfrequency of the stator, the stator could vibrate in resonance propitiating deformations, vibrations and acoustic noise. Therefore, the first step for the complete modelization of the electrical machine

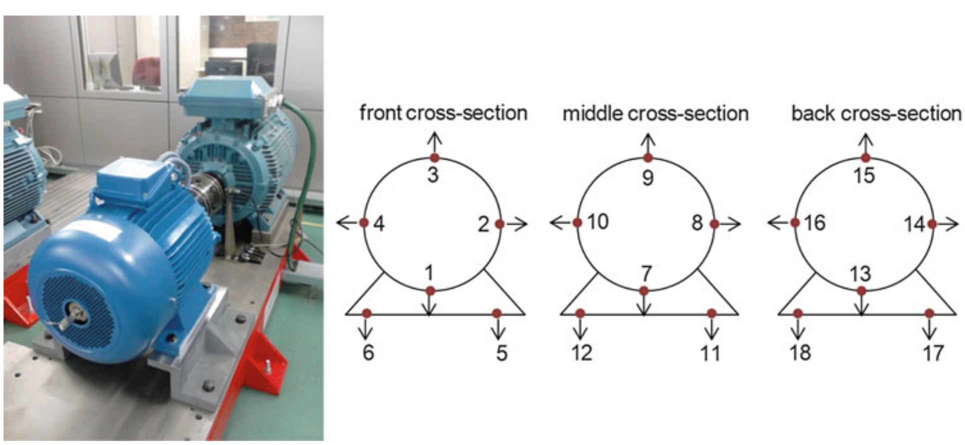

Fig. 28.2 Operational modal test in the IkerMAQ machine with accelerometers position and direction schedule

is the prediction of the natural frequencies and mode shapes by a modal analysis. For this reason, a model for vibroacoustic simulation in the frequency domain is developed. The updating of the mass, stiffness and damping values are performed taking into account EMA results.

The winding and core are the two elements which have more contribution on the weight of the stator. Also, considering that the stator has an axis-symmetrical geometry, it is considered that the mass has a slight and homogeneous influence on the modal frequencies and shapes. Knowing the total mass of the stator (available at IK4 Ikerlan) and taking into account the different elements it is composed with, the densities of the different parts of the model are defined.

The second step is the adjustment of stiffness in the model. For that purpose, first a sensitivity analysis is performed to adjust the materials elastic properties. Sensitivity analysis is the systematic investigation of the model responses to quantitative (input parameters) or qualitative (structure, etc) factors of disturbance. Once the most influential parameters are selected, an optimization process is performed via FEMTools [10] and an adjusted model is obtained.

The natural frequencies of the model depend on the mass and stiffness of the materials. However, the amplitude of vibration, and thus the noise emitted at each frequency, depends largely on the damping values introduced to the model. Therefore, there exists a need to design a model adjusted both in frequency and in amplitude to be able to predict the vibrational behavior and thus, the noise.

The updating procedure of the damping is not as straightforward as with mass and stiffness values. Figure 28.3 shows an example of the FRF curves obtained both in the experimental campaign (red line) and in simulation (blue line) at a specific point. The simulated FRF has no damping; therefore, there are sharp frequency peaks. However, analyzing the experimental curve, it is possible to see that there exist some peaks with non-negligible damped values. These peaks are related through the longitudinal direction of the stator; the resin that there is placed between the different sections makes to increase the damping in this direction. For that reason, damping is directional dependent and this fact is mandatory to take it into account.

Ansys has an option to insert a global damping to the model called constant damping ratio [11]. This is the simplest way of specifying damping in a structure. It represents the ratio of actual damping to critical damping. However, this parameter applies the damping globally in the model and thus, its suitability is limited. Therefore it seems evident the necessity of defining an anisotropic damping that can give different damping values in the different directions.

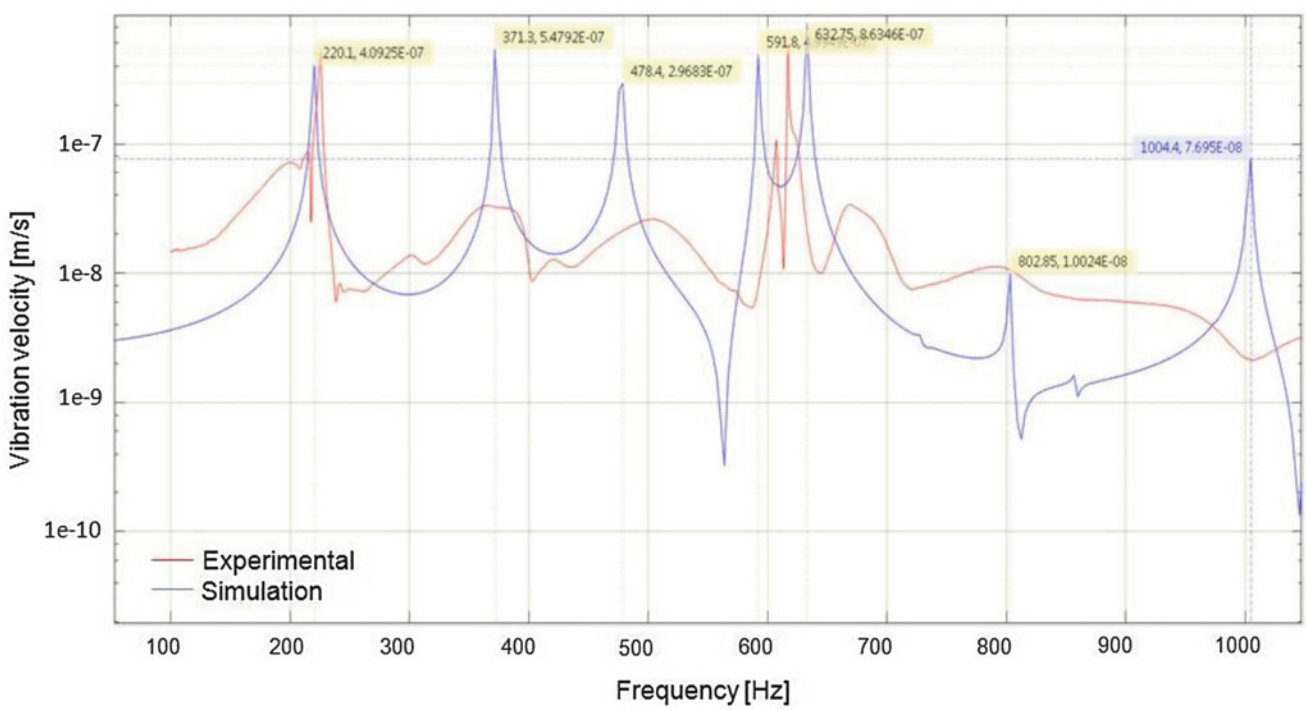

Fig. 28.3 Experimental and numerical (without damping) FRF curves from 0 to 1000 Hz

28.5 Anisotropic Damping

The most appropriate solution to have different damping values at different directions is to develop a FE model with an anisotropic damping. For that purpose, in parallel to the original model, a virtual material is created [12]. These two parallel materials are connected at every node.

$$[K]\{x\} + [C]\{\dot{x}\} + [M]\{\ddot{x}\} = \{0\} \tag{28.1}$$

$$[K]^{'}\{x\} + [C]^{'}\{\dot{x}\} + [M]^{'}\{\ddot{x}\} = \{0\} \tag{28.2}$$

where $[K]$ is the damping matrix, $[C]$ the stiffness matrix and $[M]$ the mass matrix of the original model, $[K]$', [C]', $[M]$' are the damping, stiffness and mass matrices respectively of the virtual material and $\{x\}$, $\{\dot{x}\}$, $\{\ddot{x}\}$ are the displacement, velocity and acceleration vectors respectively.

The mass and stiffness matrices of the first model are already updated as described in the previous section. Therefore, the mass and stiffness values of the virtual material need to be negligible, so: $[K]$' < < $[K]$ and $[M]$' < < $[M]$.

On the other hand, the first material has a negligible damping, while the damping of the virtual material is defined by the stiffness and mass damping matrices:

$$[C] = [0] \tag{28.3}$$

$$[C]^{'} = \beta^{'}[K]^{'} + \alpha^{'}[M]^{'} \tag{28.4}$$

As directional dependence is needed, stiffness damping is only used, $\alpha' = 0$, thus:

$$[K]\{x\} + \beta^{'}[K]^{'}\{\dot{x}\} + [M]\{\ddot{x}\} = \{0\} \tag{28.5}$$

β' is defined as a constant number while in $[K]$' there are 4 independent parameters that need to be adjusted: a1 defines the Young's Modulus in x direction ($Ex = Ey$), a2 is the Young's Modulus in z direction (Ez), a3 refers to the shear modulus in xy direction (Gxy) and a4 is the shear modulus in xz direction ($Gyz = Gxz$). Taking into account the mode shapes from 0 to 1000 Hz from EMA test campaign, an updating procedure of the parameters β' and $[K]$' is performed. This time, the Frequency Response Function obtained in the simulation is closer to the experimental data than before (see Fig. 28.4).

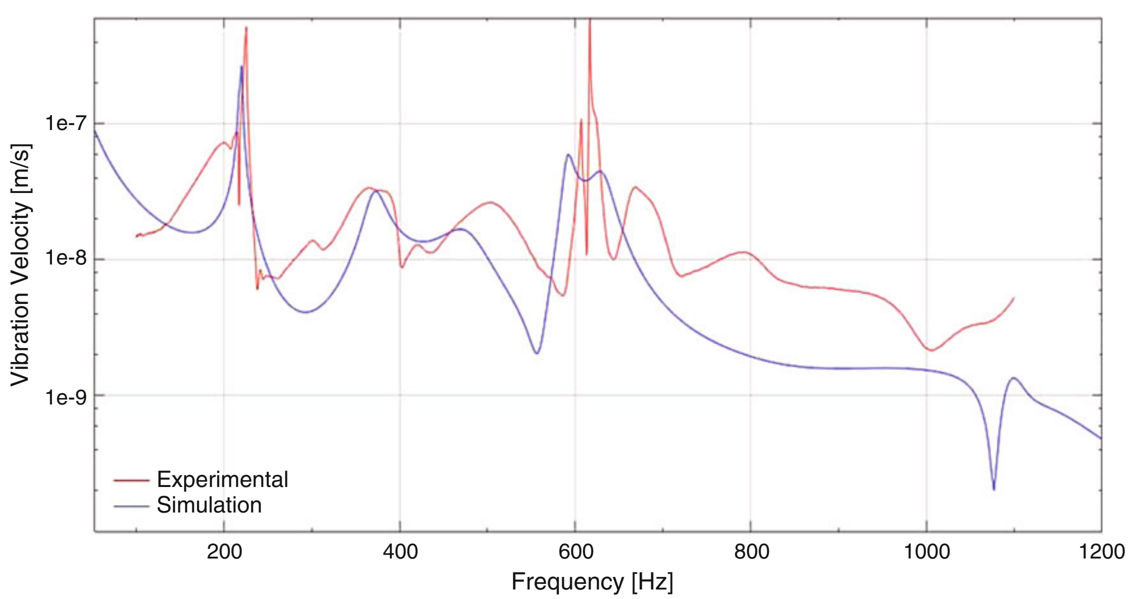

Fig. 28.4 Experimental and damping updated numerical FRF curves from 0 to 1000 Hz

It can be concluded that when the anisotropic damping is defined, the correlation between the simulated and experimental data is better. Next step consists on the correlation and posterior updating analysis between the OMA results and the simulated model.

28.6 Operational Correlation and Updating Analysis

To obtain a reliable FE model of the electrical machine compared to OMA results, different steps need to be followed. First, it is mandatory to define a valuable metric. This metric will give a quantitatively measure of the difference between the simulated and the experimental data. Three different metrics are described in this paper and the most suitable one is chosen for this specific case. After that a sensitivity analysis is carried out to identify the most influential parameters to finally improve the correlation between the simulated and the experimental data. As experimental data, the response in the radial direction of 18 accelerometers located at different places of the machine is chosen (see Fig. 28.2).

28.6.1 Frequency Response Calibration Metrics

This section describes three different metrics for frequency responses. They are described and the most suitable one is chosen for this specific case.

28.6.1.1 Frequency Response Assurance Criterion (FRAC)

Two frequency response functions, simulation (A) and experimental (X), representing the same input-output relationship, can be compared using a technique known as the Frequency Response Assurance Criterion (FRAC) which was proposed by Heylen and Lammens [13]. The basic assumption is that the measured frequency response function and the synthesized frequency response function should be linearly related (unity scaling coefficient) at all frequencies. The FRFs can be compared over the full or partial frequency range of the FRFs as long as the same discrete frequencies are used in the comparison. This procedure is particularly effective when the shape comparison of the FRFs is the major issue and the amplitude correlation is left aside.

Instead of handling the 18 FRF signals individually, which results interpretation can be of high complexity, the 75 percentile is calculated for all signals frequency by frequency and a unique FRF signal, $H_{75}{}^{X}(\omega)$, is then obtained and evaluated. The same procedure is applied to the simulation results obtaining one FRF signal, $H_{75}{}^{A}(\omega)$. It should be noted that the frequency spectrum ω goes from 0 to 5000 Hz with a resolution of 10 Hz.

$$FRAC = \frac{\sum_{\omega} \left| H_{75}{}^{X}(\omega) \cdot \overline{H_{75}{}^{A}(\omega)} \right|^2}{\left(\sum_{\omega} H_{75}{}^{X}(\omega) \cdot \overline{H_{75}{}^{X}(\omega)} \right) \cdot \left(\sum_{\omega} H_{75}{}^{A}(\omega) \cdot \overline{H_{75}{}^{A}(\omega)} \right)} \tag{28.6}$$

For identical FRFs, the FRAC value is unity and zero if the responses are uncorrelated. The authors, however, point out that a global shift in frequency between the experimental (X) and analytical (A) FRFs leads to a biased correlation value even if the FRFs are otherwise identical.

28.6.1.2 Square Deviation (SD)

This metric instead of using the 75 percentile of the signals, all signals are arranged in one vector one after another, such as:

$$H^{X}(\omega) = \begin{bmatrix} H_1{}^{X}(\omega) \\ H_2{}^{X}(\omega) \\ \vdots \\ H_{18}{}^{X}(\omega) \end{bmatrix} \tag{28.7}$$

The same procedure is applied to the signals from FE calculations. This metric [6] does not discriminate against deviations at frequencies where the structural response is small is the quadratic functional:

$$\delta = \frac{\sum_{\omega} \overline{\varepsilon(\omega)} \cdot \varepsilon(\omega)}{N} \tag{28.8}$$

where N is the number of elements of that vector. The deviation vector ε is computed for each frequency ω as:

$$\varepsilon(\omega) = \log_{10}\left(H^{A}(\omega)/H^{X}(\omega)\right) \tag{28.9}$$

28.6.1.3 Mean Squared Error (MSE)

The third metric used is the Mean Squared Error (MSE). This time again, percentile 75 of the signals are calculated; ${H_{75}}^{X}(\omega)$ for experimental and ${H_{75}}^{A}(\omega)$ for simulation.

$$MSE = \frac{\sum_{\omega} \left|{H_{75}}^{A}(\omega) - {H_{75}}^{X}(\omega)\right|^{2}}{\sum_{\omega} \left|{H_{75}}^{X}(\omega)\right|^{2}} \tag{28.10}$$

28.6.1.4 Correlation Metric Selection

The FRAC metric (Sect. 28.6.1.1) is mainly concerned to the shape comparison of the FRFs, while the SD metric (Sect. 28.6.1.2) gives more relevance to the relation between amplitudes of the FRFs. However, there is still a drawback in this metric. The correlation between the resonant peaks needs to be more important than the correlation between the anti-resonant peaks as the priority of this study is the correlation of the vibration that most influence in acoustic. This is why the Mean Squared Error (MSE) metric is the best option in this case. As the differences between the experimental and the simulated values are squared, an error in a peak will be more important than an error in the valleys.

28.6.2 Construction Parameters and Boundary Conditions

Before proceeding with the sensitivity analysis itself, it is mandatory to fix some parameters first. These are the eccentricity of the rotor, the boundary conditions and the introduction of the covers to the stator. The influence of these three design parameters has been studied first. Regarding eccentricity of the rotor, it is possible to introduce this effect in the electromagnetic calculation part of the analysis (see Fig. 28.1). Different eccentricity values have been applied and the value that gives the minor error MSE value has been chosen as reference for further research. Eccentricity values of 0.03 mm, 0.05 mm, 0.07 mm, 0.1 mm, 0.2 mm, 0.4 mm and 0.8 mm corresponding to 1.3%, 2.2%, 3.0%, 4.3%, 8.7%, 17.4% and 34.8% of the air gap have been calculated and 0.05 mm eccentricity has been chosen. So as to boundary conditions, in the experimental set-up the machine is bonded to the ground. However, this joint is not very stiff. To evaluate this joint, the same calculation is performed twice, one using bonded conditions and another one in free-free conditions. MSE values are calculated and in overall, the errors are lower in the free-free configuration. Therefore, the following analysis is performed in free-free conditions. Last point to take into account is that in the experimental set-up the IkerMAQ machine has covers at both sides (see Fig. 28.2). In order to see if this variable affect to the responses, the same calculation is done, one with covers and another one without them. Analyzing MSE values of the different points, it can be concluded that there is no much difference in using them or not. Thus, in order to simplify the FE model, the following calculations are performed without covers.

28.6.3 Sensitivity Analysis of Damping Parameters

Once the FE baseline model is completely defined, it is possible to perform a sensitivity analysis taking into account the different damping parameters as input parameters. In this paper, a One-Factor-at-a-Time (OFAT) approach has been employed

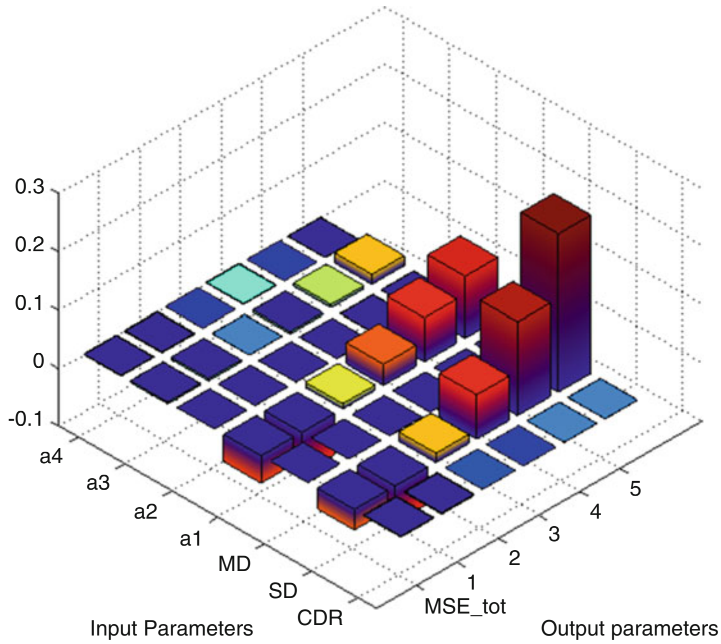

Fig. 28.5 Input parameter influence in the response

to analyze the effect of one input parameter at a time. In total, apart from the baseline model calculation, seven different calculations are performed as there are seven different damping input parameters: the Constant Damping Ratio (CDR), the Stiffness Damping (SD), the Mass Damping (MD) and four parameters that defined the Anisotropic Damping (a1, a2, a3, a4). The reference values for the baseline model are obtained from the conclusions in Sect. 28.4.

Figure 28.5 shows the sensitivity analysis results. The results are given in normalized values to avoid unit problems. A normalized sensitivity shows the percentage change of the response value for one percent change of the parameter value. Normalized sensitivities are dimensionless and thus ideal to be used when comparing different combinations of responses and parameters. The x axis represents the different input parameters (CDR, SD, MD, a1, a2, a3, a4). The y axis represents the Mean Squared Error (MSE) split into several frequency ranges; the first one is the total error obtained from full range (0–5000 Hz), second one is the MSE value from 0–1000 Hz, the third one from 1000–2000 Hz and so on. Finally, the z axis represents the values of the errors. Analyzing the figure it is possible to conclude that the most influential parameters are the stiffness damping (SD) and the anisotropic damping a1 and a3. Therefore, these are the parameters that are going to use in the model calibration.

The sensitivity result shows that parameters SD, a1 and a3 have higher influence in output parameters than the rest. Thus, these three input parameters will be used in the next sections to improve FE model's correlation with the experimental measurements.

28.6.4 Surrogate Models

To obtain the most adequate values for the three input parameters defined in the previous section, this is, to perform an optimization procedure, it is necessary to build a metamodel or surrogate model. A surrogate model can be defined as an approximation model of the original model (in this case, a FE model), which is built from sampled data obtained by randomly probing the design space (called sampling via Design of Experiment (DoE)). Once the surrogate model is built, the optimization procedure is easier to execute as the computational cost associated with the search based on the surrogate model is generally negligible. A suitable approach to choose sample points in the design space is the Design of Experiments (DoE). In this case the Latin Hypercube Sampling procedure from Matlab [14] has been employed and a total number of 30 simulations have been carried out. There are several approaches to build a surrogate model. Maybe the most popular ones are polynomial response surfaces (RSM), kriging, support vector machines, space mapping and artificial neural networks. In this paper two approaches are used and compared; RSM and kriging.

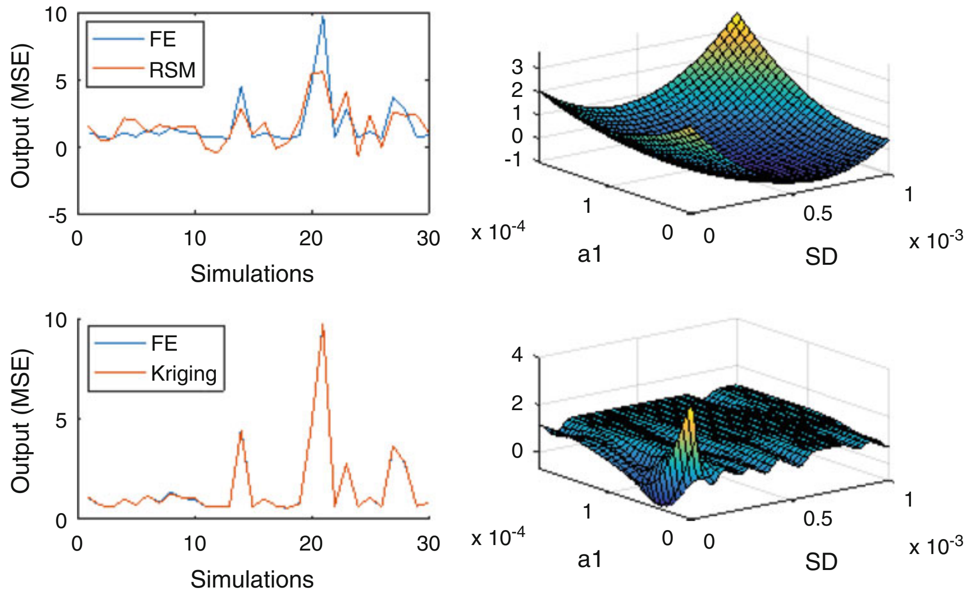

Fig. 28.6 Surrogate model precision with (**a**) RSM and (**c**) kriging, and the obtained response surface keeping the third parameter constant in its reference value for (**b**) RSM and (**d**) kriging

RSM [15] denotes a polynomial approximation model in which the sampled data is fitted by a least-square regression technique. In RSM-based optimization applications, the quadratic polynomial model usually provides the best compromise between the modeling accuracy and computational expense, when compared with the linear or higher order polynomial models. An advantage of RSM is that it can smooth out the various scales of numerical noise in the data while captures the global trend of the variation, which makes it very robust and thus well suited for optimization problems in engineering design. In this case, Fig. 28.6a shows that the generated surrogate model cannot approximate exactly the FE responses. Nevertheless, as a practical example, plotting two input parameters (SD and a1) and keeping the third one (a3) in its reference value, it is possible to see that smooth surface is obtained applying this methodology (see Fig. 28.6b).

Kriging [16] is an interpolating method which features the observed data at all sample points. This method provides a statistical prediction of an unknown function by minimizing its Mean Squared Error (MSE). It can be equivalent to any order of polynomials and is thus well suited for a highly-nonlinear functions with multiple extremums. This time, the surrogate model can resemble the FE responses perfectly (see Fig. 28.6b). However, looking at the response surface generated from two input parameters, keeping the third one constant, it is possible to see that this methodology sacrifices smooth responses in order to resemble the FE responses as much as possible (see Fig. 28.6d). This type of surrogate model generally gives local minimums and they are more complicated to use.

28.6.5 FRF Updating

Once surrogate models are built, they are employed in the optimization app from Matlab [14]. The objective of the optimization procedure is to find the minimum of the constrained nonlinear multivariable function, in order to minimize the MSE. Minimum and maximum bounds are defined to avoid having results outside the design space. Using the RSM surrogate model, regardless the starting point, a minimum is reached. These parameters values are then inserted into FE model in order to assess that the obtained results are in concordance with the FE results. For the case of the kriging surrogate model, as there are many local minimums, the starting point influences the obtained result. Therefore three different starting points are chosen and the optimization process gives three different results. These parameter values are then inserted into the FE model and simulation is run. Although the four simulation options give better results that the first baseline model, thus, the MSE value is reduced, there is one DOE simulation that has a lower error value. Therefore it seems that these 2 surrogate models are not accurate enough. In this paper, the DOE simulation that gives minimum error is selected. Figure 28.7 shows the obtained 75 percentile FRF signals for the baseline model and for the minimum MSE simulation case. These two curves

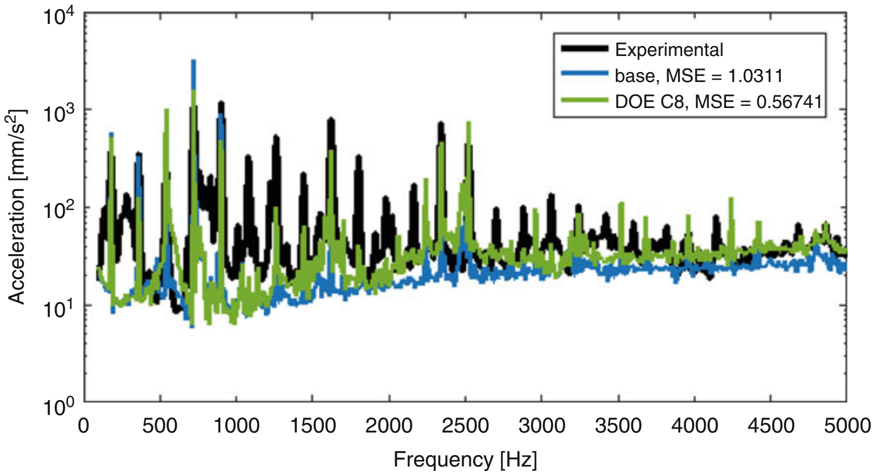

Fig. 28.7 FRF comparison between baseline model and minimum MSE simulation against experimental data

are compared against test data. It is possible to see that in the first half of the graph, this is, from 0–2500 Hz, the peak amplitudes are closer to the experimental ones for the DOE case simulation and this is which most influences on the obtained error value.

28.7 Conclusions

A weakly coupled multiphysical methodology for the calculation of the dynamic response of an electric motor stator has been presented in this paper. This method can also lead to the calculation of the acoustic power radiated by the motor. However, the first step, and the main objective of this paper, is to correlate the vibrational response of the machine. For that purpose, an electrical machine called IkerMAQ is built and a bottom-up approach is employed. First, single components are measured, correlated and adjusted. Finally, the whole system is analyzed testing it at nominal speed values.

As the electric machine is built with different materials with different properties, the damping values are directionally dependent. Therefore, the main contribution of this paper is to develop an anisotropic damping model that can apply different damping values at different directions. This model provides a better approximation to the available test data.

A complete sensitivity analysis is performed and the influence of the diverse simulation parameters is analyzed in this paper. Although the generated surrogate models are not capable of correctly representing the FE model, the objective of decreasing the MSE value is reached. Therefore, it can be concluded that the vibrational response of the electrical machine is correlated relatively well. Nevertheless, for future work the development of a suitable surrogate model is recommended. Once the vibrational model is calibrated, the next step consists in correlating the acoustic model.

To study the global vibrational behavior of the experimental and simulation cases, the Root Mean Square (RMS) values are calculated. In the case of the experimental 75 percentile signal, a RMS value of 130.0 mm/sˆ2 is obtained. On the other hand, the baseline 75 percentile signal gives a RMS of 164.1 mm/s^2 while the optimized one (minimum MSE signal) gives a RMS of 114.3 mm/s^2.

In summary, the overall suitability of the study carried out in this research, the errors between the experimental RMS and simulated RMS are calculated. While the baseline error is 26.2%, the optimized signal error is 12.1%, thus reducing the error considerably.

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