Development of Tools to Calculate the Vibroacoustic Performance of Electrical Machines in Lift Installations

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Abstract. Improper operation of the traction machine of a lift installation causes energy waste, vibrations and noise. The design of the machine must be optimum if energy efficiency and comfort specifications have to be satisfied. The vibrations and noise frequency spectra of electrical machines present manifest peaks at certain frequencies, multiples of the fundamental electrical frequency, that depend on the machine topology and its rotation velocity. Changes in its topology or in its mechanical properties (geometry, size, materials…) must be done in order to reduce the magnitude of peaks at certain excitation frequencies or to locate the excitation frequencies far from the natural frequencies of the structure or the lift installation. Machine designers need tools to calculate their vibroacoustic response once a certain design has been proposed, so they can modify it before a prototype is built in case the response is not acceptable. Numerical and analytical models to calculate the vibroacoustic response of electrical machines have been developed and experimentally validated. In this paper, the authors summarise the state of the art in modelling the vibroacoustic performance of electrical machines, provide some guidelines regarding the values to be assigned to the machine componets, and show some of the results obtained in their research work.

Keywords: lift, comfort, vibration, noise, electrical machine, machine design.

1 INTRODUCTION

The traction machine of a lift installation is a source of vibrations and noise that cause discomfort to the lift passengers and of the neighbours living at flats close to the lift well. Therefore, the machine design must be optimised (e.g. for power, size, cost, vibrations and noise) in order to conform to the riding comfort standards. Furthermore, the machine should not be designed without considering the lift installation, the whole assembly, because vibrations generated at the machine are born through the structure to the cabin. Consequently, the machine design is conditioned too by the lift installation where it will be placed.

Tools to predict the vibroacoustic performance of an electrical traction machine in a certain lift installation are necessary to achieve an optimum design and to avoid, as much as possible, the prototyping stage. The first step is to predict its performance on a test bench but the final goal must be to predict it in the installation.

This document reviews the state of the art corresponding to that first step and describes the procedure to be carried out to compute the vibroacoustic performance of an electrical machine.

2 VIBROACOUSTIC PERFORMANCE OF AN ELECTRICAL MACHINE

Vibrations and noise of an electrical machine can be originated by the electromagnetic forces at the air-gap, by mechanical defects associated to the rotating parts (bearings, shaft), or by the air flux, when the machine has a fan for cooling purposes (see Fig. 1). Below 1000 Hz and in low to medium speed rated machines, electromagnetic forces are the main sources of vibrations and noise [1]. The frequency spectra of electromagnetic vibrations and noise are very tonal and particularly annoying for lift passengers and neighbours close to the installation. This paper reviews the procedure followed to calculate the vibroacoustic response of electrical machines due to the radial electromagnetic forces generated at the air-gap.

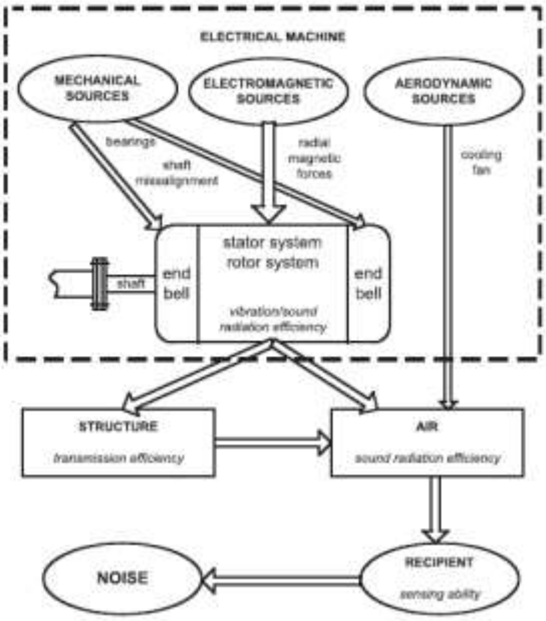

Figure 1 Noise generation and transmission in electrical machines [2]

The procedure consists of three parts (see Fig. 2). First: calculating the radial electromagnetic forces. Second: applying them to the machine structural model to obtain the vibratory response of its outer surface. Finally, computing the acoustic power it radiates [2].

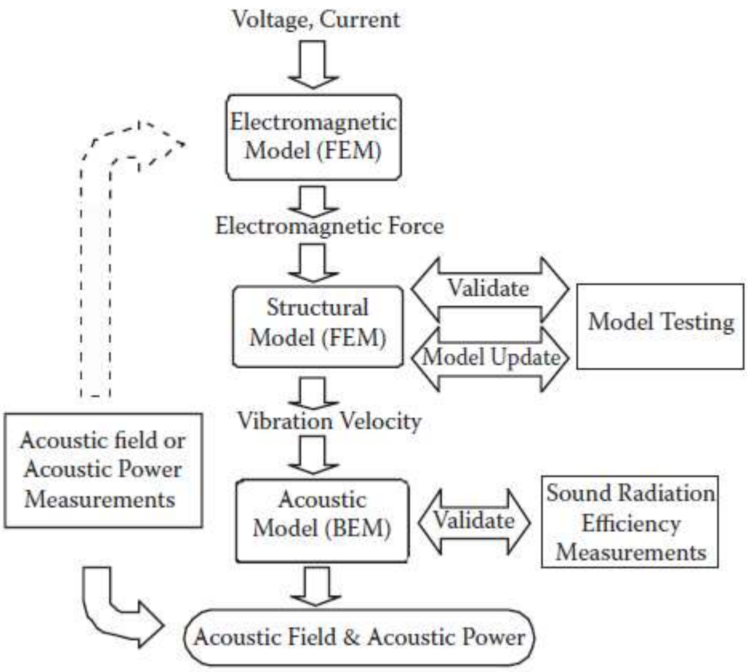

Figure 2 Procedure for predicting acoustic power from an electric machine [2].

2.1 Computing the Maxwell Forces

According to Le Besnerais Maxwell forces [3], normal to the front surface of the stator teeth, are the main contributors to the machine vibrations [4], particularly in Permanent Magnets

Synchronous Motors (PMSM) [5], very common in lift installations. If the rotor is not skewed and the end effects are neglected, the pressure distribution is independent of the motor axial direction, so 2D calculations are adequate. In order to obtain the magnetic pressure on the teeth and on the poles of the machine the magnetic field originated by the permanent magnets and the armature is calculated and the magnetic pressure is derived applying the Maxwell stress tensor. The electromagnetic field at the air-gap can be calculated either by Finite Element Models (FEM) or analytical methods, and a reasonable agreement between them is achieved [6]. For instance a FEM model developed in FLUX is shown in Fig. 3.

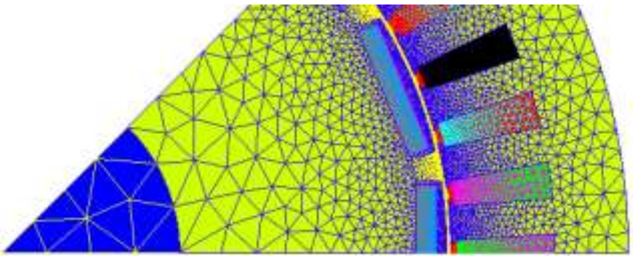

Figure 3 Electromagnetic FEM model developed in FLUX

The whole machine does not have to be modelled as symmetry can be applied according to the number of teeth and poles. The electromagnetic flux is obtained as in Fig. 4.

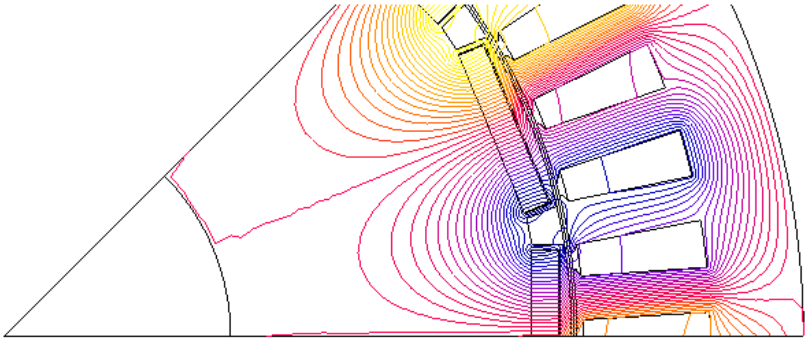

Figure 4 Magnetic field calculated by FLUX

Next, the pressure applied on the surfaces of the teeth and of the poles at the air-gap, due to the electromagnetic field is calculated. By using the two-dimensional Fourier transformation the pressure is converted into separate rotating force waves, defined by frequency, spatial harmonic number on the circumference, amplitude, phase angle, and rotation direction [7]. The magnetic pressure is mainly dominated by the magnetic field generated by the magnets, so that the spatial and time domain variation of the pressure is related to the number of harmonics of the number of poles in space and to the number of poles and the rotational frequency in time. In Fig. 5 the pressure in one tooth is plotted, where the horizontal axis are the angle θ and time.

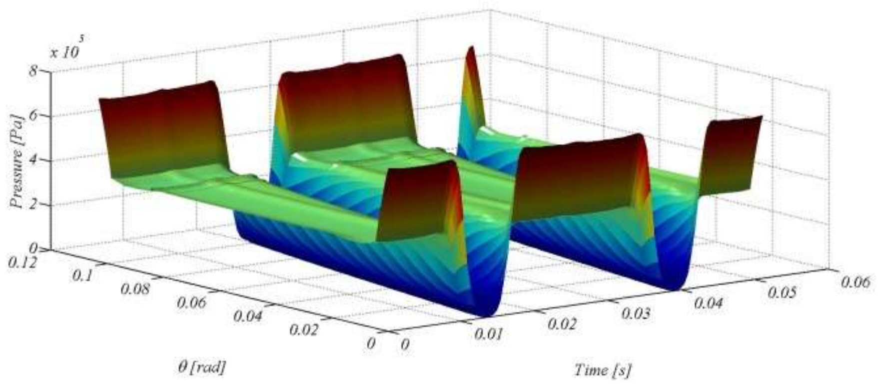

Figure 5 Magnetic pressure on a tooth tip

2.2 Structural Model

Electrical machines are composed of several parts. Some parts are free to rotate, as the shaft and rotor assembly, due to the bearings placed in the end-shields which are joined to other static parts such as the stator and windings. Stator and windings are the main components as they are fed by currents in order to generate the desired torque. Thus, the main forces arise on the stator of the machine which is usually one of the biggest components and consequently contributes to vibrations and noise the most. The stator and rotor consist of a stack of laminates, electrically isolated, and consequently they show an orthotropic behaviour. Similarly, the windings are made of many copper wires inserted in the slots of the stator which has a particular shape depending on the machine design.

In order to develop a computationally efficient finite element model, the stator and rotor are modelled as solid structures. The same assumption is needed when developing simplified analytical models. Because of this orthotropic nature, there is uncertainty in the values of some mechanical properties. Axial stiffness of the stator increases if the clamping pressure applied when joining them is increased [8]; high axial stiffness implies high values of the natural frequencies associated to the axial modes; however, natural frequencies of the radial modes hardly vary [9].

In addition, there is no elastic connection between adjacent stator sheets and some slip is allowed between them. The slip is the main contributor to damping, not only in the case of the axial modes but also in that of the whole structure [10]. Damping is another key parameter, uncertain as well, that is expected to affect the amplitude of vibrations.

When the windings are added to the stator, the values of the natural frequencies of the assembly decrease, but they increase again when the assembly is impregnated with the isolation varnish [8].

2.2.1 Finite element models

The process to develop a FE model of an electrical machine was explained in [11, 12] and the results obtained showed that some issues have to be overcome according to the nature of the different components, specially the stator and windings which are the most considerable and the hardest to model.

Proper values must be assigned to the mechanical parameters of the equivalent solid systems representing the stator and windings (see Fig. 6). The laminates of the stator are made of steel of known density; however, as the stator is a stack of steel laminates, the density of the whole body is lower than that of the steel, due to air gaps between the laminates. In regard to the windings, the components are the wires, made of copper, and the electrical insulation material, whose densities and the volume they fill are known as well. Thus, the equivalent density can be estimated according to the slot filling factor.

Figure 6 Stator-windings assembly

Concerning the elasticity parameters of the stator, that is to say, elasticity modulus, shear modulus and Poisson's ratio, different values must be assigned in the axial direction and radial-tangential plane. Some authors [8, 9] provide approximate values of those uncertain parameters of the machine; otherwise, the model of the structure can be updated based on the results obtained from experimental modal analysis [9, 11, 12], which provides the natural frequencies and mode shapes of the structure.

In the radial-tangential plane, the elasticity and shear modulus and the radial Poisson's ratio are assumed to be those of the steel; nevertheless, the axial elasticity is around 3% and the axial shear modulus below 6% of those of the steel; the Poisson's ratio is usually negligible in the axial direction.

A question to be answered is how the vibratory performance of the stator – windings assembly changes as the values of the previously mentioned parameters change. Regarding the stator alone, the changes in the natural frequencies corresponding to the modes shown in Fig. 7 (pure radial or both radial-axial deformation shapes) with respect to changes in the parameter values have been studied.

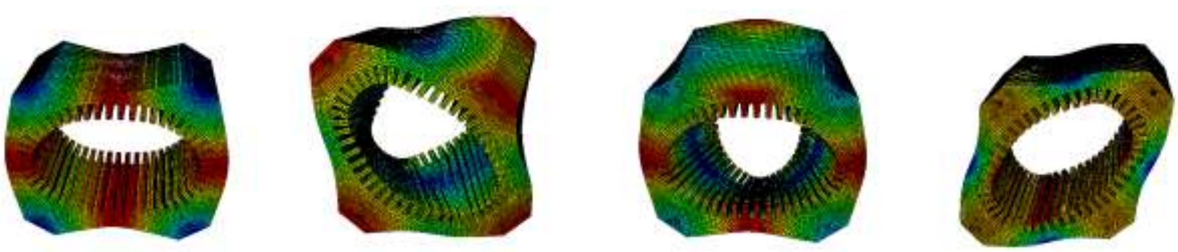

Figure 7 From left to right, modes 1 to 4

The modulus of elasticity in the radial-tangent plane affects most modes 1 and 4 (pure radial modes) whereas the axial elasticity modulus does hardly affect them; the shear modulus in the radial-tangent plane affects most modes 2 and 4 (those modes undergo shear deformation as well) whereas the shear moduli in the axial direction only affect the radial-axial modes (modes 2 and 3).

Concerning the elasticity properties of the windings, it has been concluded that their changes does not affect much the values of the natural frequencies compared to variations of the properties of the stator. Consequently, elasticity modulus in the radial-tangential plane and shear moduli in the axial direction of the stator are mainly the key parameters when developing the FEM model of the stator-windings assembly.

Another point to be considered in the FEM model is the mesh; the size of the elements have to be carefully selected in order to achieve reliable results up to a certain frequency. Depending on the zone of the assembly, different sizes should be applied. The number of elements in the tooth tip has to be selected according to the spatial distribution of the pressure. Once the bandwidth of the spectrum to be calculated is defined, the number of nodes per tooth must be set according to the number of spatial periods of the highest pressure component (decomposed by FFT as mentioned in section 2.1) in order to avoid spatial aliasing. On the contrary, the size of the elements in the stator core, far from the teeth could be larger (see Fig. 8).

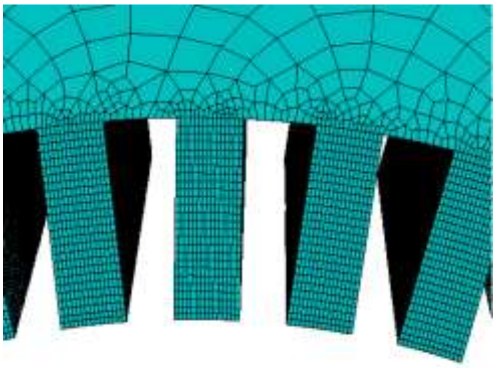

Figure 8 Mesh of the stator

Despite the bigger influence of the stator and windings in the vibratory behaviour, three-dimensional FEM models enable the user to take into consideration machine components that would be simplified in analytical models, as the end-shields, the frame, the support or the rotor [11, 12, 15, 16]. The biggest modelling issue when these elements are introduced are the bearings, which are generally modelled by springs [11, 12, 17] to which the stiffness value of the radial and axial equivalent stiffness of the bearing has to be applied. Particularly, in some applications rotor vibrations should be taken into account [17] (see Fig. 9).

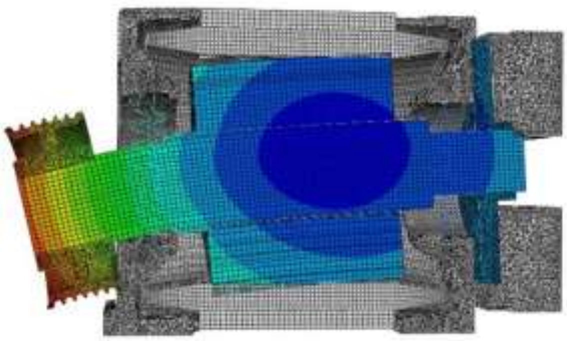

Figure 9 A FE model of the whole machine

2.2.2 Analytical models

The FE models, described in the previous section, are closer to the real structure, but they require much bigger computation effort than analytical models.

In consequence, analytical models could be convenient for machine design optimisation purposes, although less accurate. A number of them have been proposed in the literature to calculate the natural frequencies of electrical machines. The stator-windings assembly is commonly modelled as a lumped parameter model, with two circular cylinders attached to each other, one of them corresponding to the stator and the other one to the teeth-windings part [2]. For the case of short machines (the machine length to diameter ratio less than or equal to one) [2], the axial dimension is negligible and a two-dimensional ring model can be used. If the machine is not short enough, instead of as a double ring, it is modelled as a double cylinder, either of infinite [2] or finite length [13, 14].

Regarding the mechanical properties, all that was said in Section 2.2.1 is applied in the analytical models. The values assigned to the material of the outer ring are those of the steel, but those assigned to the inner ring are estimated. The orthotropic nature of some of the properties could be accounted for as well.

2.2.3. Vibratory response of the machine

Once the structural model has been developed (section 2.2), previously computed forces are applied on the surfaces of the teeth and poles (section 2.1), and the vibratory response of the outer surface of the machine is obtained. The modal superposition theorem [18] is used to compute the global displacement of a certain point of the outer surface. This theorem uses the mode shapes as a vector basis to calculate the response of the system to a harmonic forcing load vector. The method allows computing the global displacement of a given point by summing the displacements caused by individual modes [19]. The number of modes to be considered depends on the frequency range to be analysed and the spatial distribution of the forces.

2.1.Acoustical Model

If the vibratory response of the outer surface of the machine is known, the acoustic power radiated by it can be obtained. The key parameter to determine the acoustic power is the sound radiation efficiency, defined as the ratio between the acoustic power and the radiation power of the surface [20].

If the geometry of the machine is idealised, analytical expressions to obtain the sound radiation efficiency are available. If the length of the machine is similar to its circular section diameter, an acoustic spherical model can be assumed (the radiated sound waves approximate to the spherical waves radiated by a vibrating sphere) [21]. If the length to diameter ratio is much bigger than one, an infinitely long cylindrical model can be used [22]. If it is not so big, the finite length circular-section cylindrical shell model [23] is usually a better approach.

To deal with complex topologies and to take end-plates and other details into account, numerical methods have to be used to calculate the sound radiation efficiency and radiated acoustic power [24].

3. ACCURACY OF SIMULATION RESULTS AND CURRENT RESEARCH

From a quantitative point of view, there are usually differences between the computed and measured acoustic power spectrum, at least at certain third-octave bands, due to the assumptions considered at the modelling phase. There is uncertainty regarding the electromagnetic forces applied; validation tests reported in the literature are commonly based on vibration measurements but not on measurements of the forces themselves. In the structural model, there is uncertainty too regarding certain mechanical parameters (such as elasticity modulus and damping) and in the boundary conditions assumed. Nevertheless, the developed tools provide interesting results regarding relative analysis, that is to say, to compare different designs, to make sensitivity analysis, to understand which modes are excited by which forces, and to choose the best slot pole combination.

Let us show some results for the sake of illustrating the previous paragraph. Fig. 10 shows an experimental set up to test vibrations of the machine in operation, where the machine is tied to the floor and accelerometers are placed on its outer faces. The vibrations are measured by the accelerometers for certain operating conditions. Fig. 11 shows the comparison between the power spectral densities (PSD) of the measured (blue line) and calculated (by a FE model, green line) accelerations at the top surface of the machine. We have orders in the horizontal axis instead of frequency. One order corresponds to the rotation frequency of the machine multiplied by the number of pole pairs. Only vibrations of electromagnetic origin are calculated but all vibrations are measured, those of mechanical origin as well.

Figure 10 Experimental set up

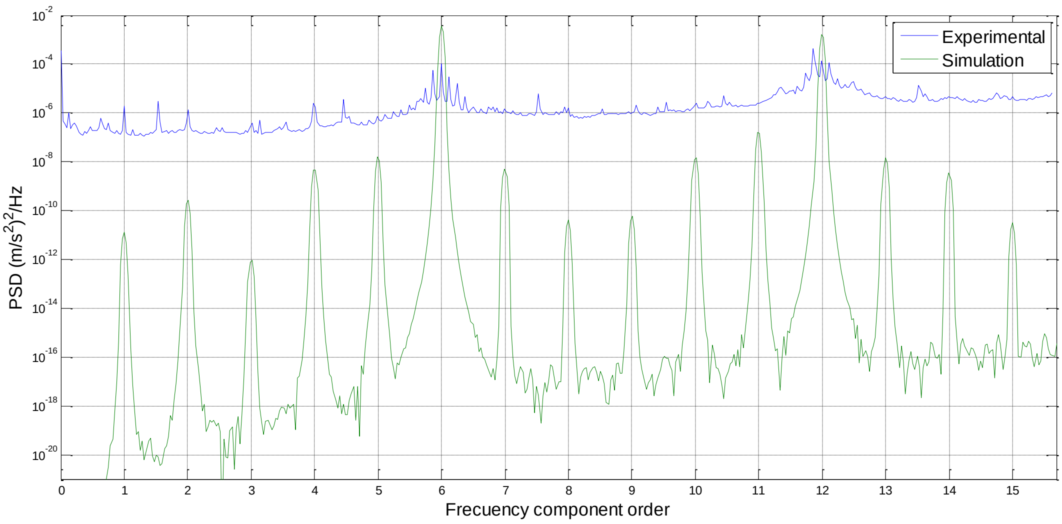

Figure 11 Comparison between measured (blue) and calculated (green) accelerations

The model identifies the vibration frequency components with the highest amplitudes (orders 6 and 12), although there are considerable differences in the amplitudes of most of the peaks. Close to the two highest orders smaller peaks can be observed. They are due to certain eccentricity of the rotor. Eccentricity always causes the increase of the amplitude of the peaks, particularly at the main orders. As the measurement includes all vibration (not only that of electromagnetic origin), it was expected its spectrum to be over that one of the calculated vibration at all intervals between orders.

With respect to the results provided by the analytical models compared to those obtained in experimental measurements, PSD of the acceleration vs. frequency harmonics (see Fig. 12), where the vibration responses provided by two models of only the stator-windings assembly of the machine are compared. The continuous blue line corresponds to the experimental signal. The points correspond to the results provided by a two-dimensional model with isotropic properties and a three-dimensional one with orthotropic properties.

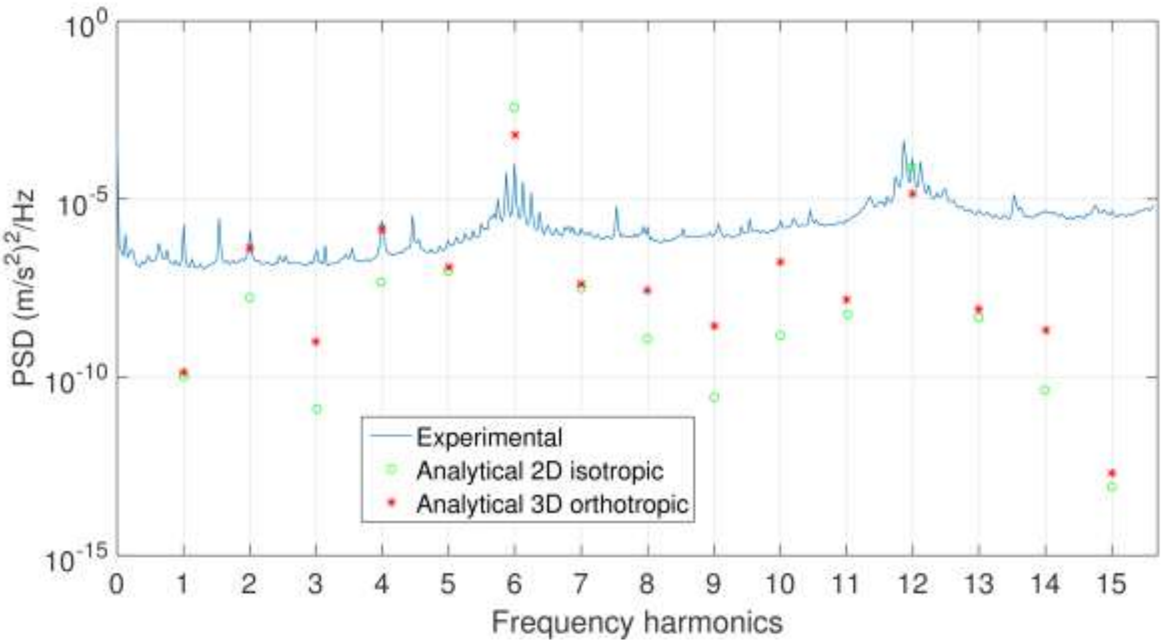

Figure 12 Comparison between analytical and FEM models

There are considerable differences in the amplitudes of the peaks, but it can be observed that the results of the models considering axial modes are closer to the FEM.

To conclude, it seems difficult to obtain accurate models from a quantitative point of view, but interesting conclusions can be obtained from a qualitative one (main frequencies, shape of the response, comparison between different design).

The final question is how the machine will operate once it has been installed, because a discarded machine, based on tests carried out on a test bench, could behave properly in a certain installation. Thus, any tool developed to compute the machine behaviour should consider the whole assembly it belongs to.

Consequently, nowadays, there are two main areas of research: improvement of the machine models regarding all uncertainty aspects previously mentioned and behaviour of the whole installation due to electromagnetic excitations generated at the air-gap of the machine.

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