Optimum Design of Traction 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 and show some of the results obtained in their research work.

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 (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 also conditioned by the lift installation in which 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.

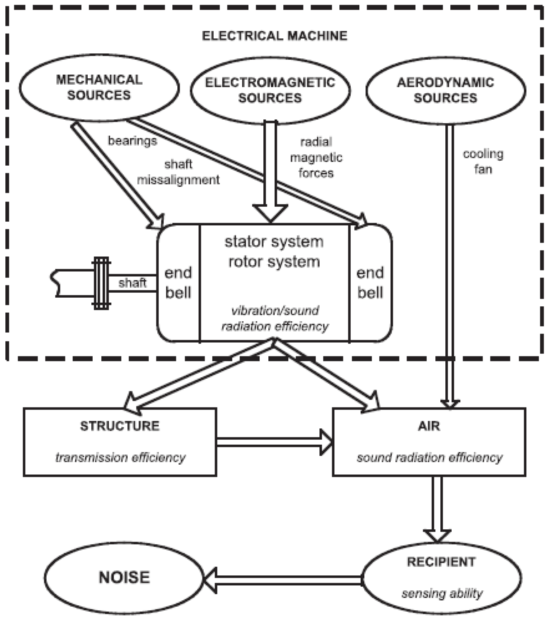

Figure 1: Noise generation and transmission in electrical machines [2].

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].

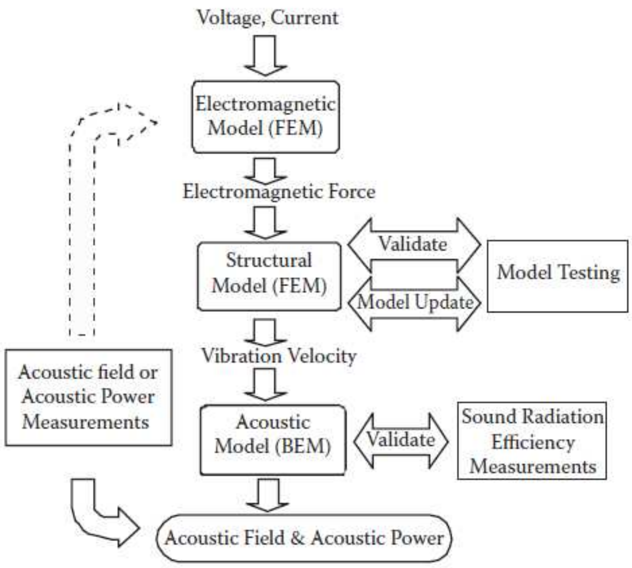

Figure 2: Procedure for predicting acoustic power from an electric machine [2].

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. The electromagnetic field at the air-gap is calculated either by Finite Element Models (FEM) or analytical ones, and a reasonable agreement between them is achieved [6]. Next to be calculated is the pressure applied on the surfaces at the air-gap, due to the electromagnetic field. The pressure, by the two-dimensional Fourier transformation, is converted into separate rotating force waves, defined by frequency, spatial harmonic number on the circumference, amplitude, phase angle, and rotation direction [7].

2.2 Structural Model

The electrical machine is composed of static (stator-windings) and rotating parts (shaft-rotor). The transversal section of the stator is constant in the axial direction and has a particular shape with a number of teeth and slots, where the windings are inserted.

Regarding the mechanical behaviour of electrical machines, it is worth mentioning some particularities.

The stator and rotor consist of a stack of laminates, electrically isolated, and consequently they show an orthotropic behaviour. 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.

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 common modal analysis [9, 11], which provides the natural frequencies and mode shapes of the structure (see Fig. 3).

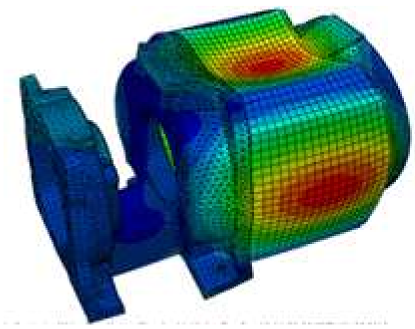

Figure 3: Modal shapes of the structure [11]

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].

With respect to modelling the mechanical structure, either finite element (FE) or analytical models have been proposed. FE models 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 analytical models 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 [12, 13].

Mechanical properties are commonly assumed to be isotropic. The values assigned to the material of the outer ring are those of the steel, but those assigned to the inner ring are estimated. A more accurate model should consider that the components are orthotropic and assign different values to the radial and axial elasticity moduli. For a laminated structure, the elasticity modulus in the axial direction is much smaller than that in the circumferential direction [15].

Two-dimensional models neglect several aspects as three-dimensional mode shapes, axial vibrations and rotor vibrations, and, consequently give inaccurate results in the case of machines with significant axial and rotor vibrations [14]. For these cases FE models can be more appropriate, as they allow considering other components as end-shields [15], the rotor [16], the frame or the support, and provide the possibility to assign orthotropic properties to the materials (see Fig. 4).

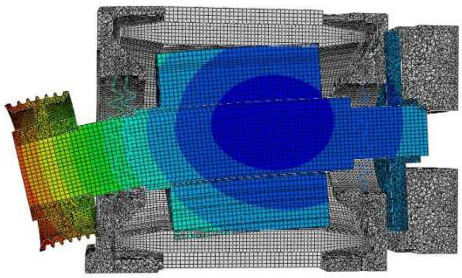

Figure 4: a FE model of the whole machine.

Once the structural model has been developed, previously computed forces are applied on the surfaces of the teeth and poles, and the vibratory response of the outer surface of the machine is obtained. The modal superposition theorem [17] 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 [18].

2.3 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 [19].

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) [20]. If the length to diameter ratio is much bigger than one, an infinitely long cylindrical model can be used [21]. If it is not so big, the finite length circular-section cylindrical shell model [22] 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 [23].

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 (elasticity modulus, 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, to choose the best slot pole combination…

Let us show some results for the sake of illustrating the previous paragraph. Fig. 5 shows an experimental set up to test vibrations of the machine in operation. Fig. 6 shows the comparison between the power spectral densities (PSD) of the measured (blue line) and calculated (green line) accelerations (by a FE model) 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, including those of mechanical origin.

Figure 5: Experimental set up.

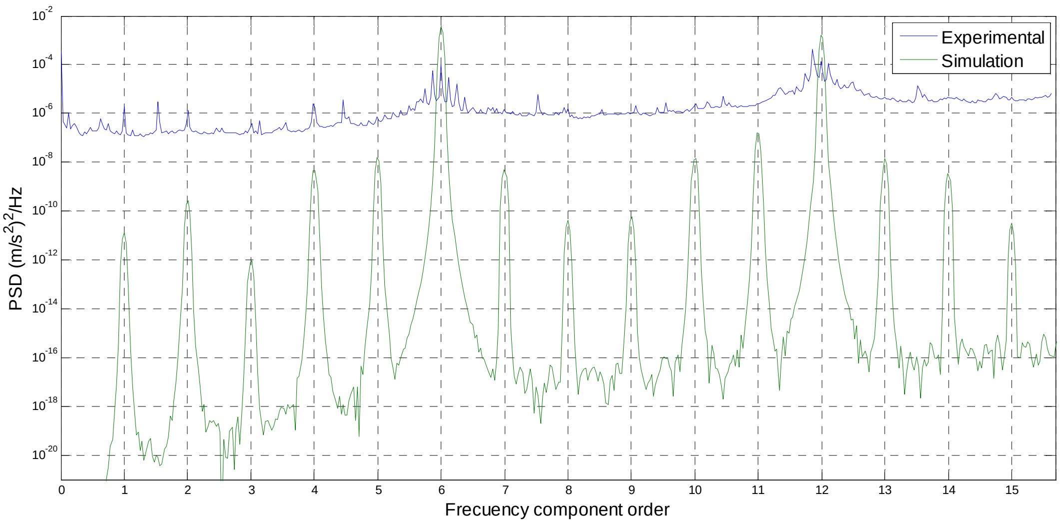

Fig. 6: Comparison between measured (blue) and calculated (green) accelerations.

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 axis.

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), its spectrum was expected 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 provided by the FEM (displacement vs. orders), see Fig. 7, where the vibration responses provided by several models of only the stator-windings assembly of the machine are compared. The continuous red line corresponds to the FEM. Three analytical models have been used: a double ring of circular shape (ring means that axial modes are not considered), a single cylinder (including stator and windings whose properties have been assigned average values) and a double cylinder. All properties are assumed to be isotropic.

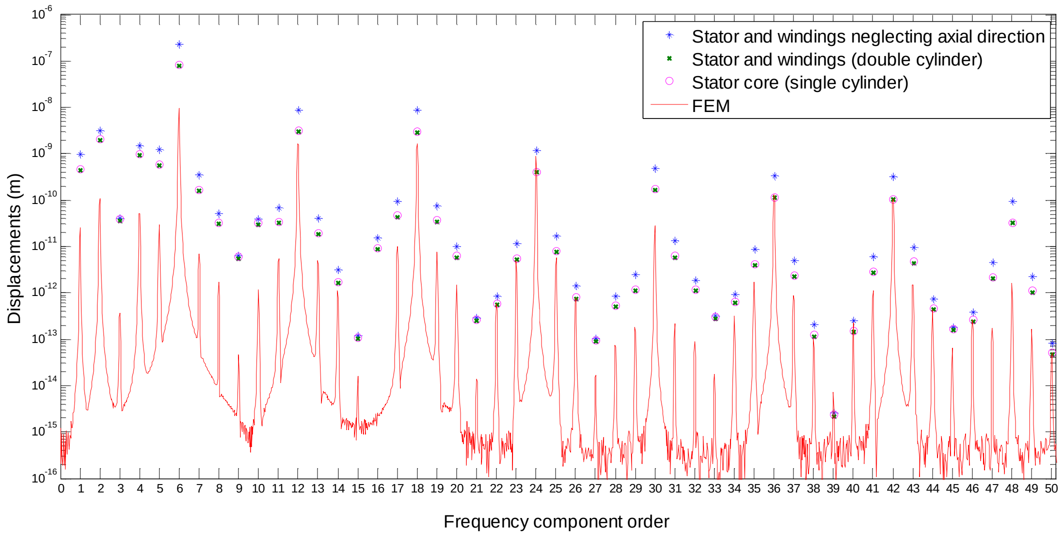

Figure 7: Comparison between analytical and FEM models.

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 designs…).

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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