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CONSIDERATIONS ON THE MODAL ANALYSIS AND VIBRATION SIMULATION OF INDUCTION MOTORS

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Abstract: *The article is observing the vibration numerical simulation of induction electric motors by using finite elements method, with particular focus on the modal analysis. Several other dynamic simulations like the frequency responses to harmonic, random excitations and mechanical shocks are shortly presented. The importance of the modal analysis is highlighted from theoretical and experimental point of view, on one hand, and by performing and commenting several simulations on electric motors components (housing, rotor and stator) and the assembled structure, on the other hand.*

Keywords: *induction electric motor, frequency response, modal analysis, experimental modal analysis.*

1. INTRODUCTION

Simulation of electrical motors is a high-priority task due to the continually extended use of electric motors. The vibration problems of these motors are coming from mechanical, magnetic and fluid-structure interaction sources. When observing the motor's vibration, the presence of the beats (oscillatory amplitude of vibration) is indicating a combined problem coming from mechanical and magnetic sources, while the absence of the beats indicates a mechanical problem. By removing the electrical power, the magnetic problems are eliminated immediately, being able to observe the mechanical problems alone [2].

The mechanical vibration problems are caused by the motor's shaft misalignment, eccentric mass, a bent shaft, bearings failures or mechanical looseness between the motor's components [2]. When measuring the vibrations on an electric motor mounted and functioning, registered mechanical vibrations are often coming from external components of the system like driven equipment (harmonic excitations), misalignment with the driven equipment, gears, couplings and so on.

Induction motors is one of the most exploited family of the industrial electrical motors. They are widely used in industries like: automotive (electric propulsion), food and beverage, packaging, printing, plastics and rubber, port logistics (cranes), test rigs, textile, glass, industrial refrigeration (compressor mounted), marine (for pumps or propulsion), metals, mines, quarries and cement, water, oil and gas, and last but not least, nuclear industry (power plants). For this last industry in particular, predicting the vibration behavior of the electric motors by conducting numerical simulation is even more critical, since the consequences in case of failure could be catastrophic. In the majority of cases, motors drive pumps responsible for cooling or evacuating water in the case of a natural disaster. The most important excitation the motors could be subjected to is the seismic load. Therefore, in order to avoid the occurrence of the resonance phenomenon during an earthquake, electric motors have to be designed in such way that they have no critical frequencies in the seismic spectrum frequency range.

Modal analysis and harmonic excitation response analysis, both in the bandwidth of interest, are required in the motors' vibrations study. Also of interest are the system's response to random excitations and the special case of the mechanical shock response.

In the case of electric motors, there are several types of vibrations that are to be analyzed through numerical simulation. Thus, we mention:

• The vibrations of the entire structure; in this case the most dangerous are the tilting modes that have a high amplitude at low frequencies.

• Vibration of the rotor when it rotates at a speed close to its critical speed (synchronism speed).

• Stator vibrations; in this case we are referring to the vibration modes of electromagnetic nature. These are very common and can't be avoided, due to the power supply. They cause noise during operation.

• The vibrations of some flexible components; these are not dangerous, causing only noise.

2. DYNAMIC SIMULATIONS OF INDUCTION MOTORS

Dynamic simulations study the behavior of systems subjected to loads which vary with time.

The vibration behavior of induction motors can be characterized by conducting several types of dynamic simulations. Modal analysis is the basic type of dynamic analysis, used to determine the motor's modal parameters: the natural frequencies and their associated mode shapes. The harmonic response analysis is used to determine the motor's response to harmonic loads (e.g. the excitation generated by a compressor with a known pulsation). A response spectrum analysis can be used to determine how a motor responds to random or time-depending loading conditions, while all its resonances are excited in the same time (e.g. the shocks generated by ocean waves, earthquakes or wind loads). A random vibration analysis is a particular type of response spectrum analysis and can be used to determine how the motor's structure responds to random excitations (e.g. the vibrations caused by the road in the case of an electric car). Finally, the transient analysis can be used to accurately calculate a motor's response to time varying loads (e.g. the motor's behavior during start-up when coupled to a large fan). Among all these types of dynamic simulations, the transient simulation is the most expensive one in terms of computational resources.

Like anticipated, the modal analysis can serve as a standalone analysis whose results (modal parameters) are exploited likewise with the objective of avoiding resonance occurrence. Through design evolutions, the motor's natural frequencies could be moved away from the excitation frequencies and their harmonics.

On the other hand, modal results of the motor's structure are often used as input data for other frequency response analyses (spectral analyses) with the goal of studying the dynamic behavior of its structure subjected to external loads, such as harmonic vibrations, random vibrations (power spectral density), mechanical shocks or transient vibrations.

The general principle, from a numerical point of view, for conducting frequency response analyses, is:

- in a first stage, a modal analysis on the motor's structure is conducted, with the goal of calculating the modal parameters, on a given frequency interval;
- in a second stage, the modal analysis is coupled with a frequency response analysis, in such way that the results of the modal analysis are imported as input data for this second analysis;
- the coupled simulations are run, in the coupling order, and results are analyzed.

The quality of the results obtained for the frequency response analysis will be directly influenced by the truthfulness of the simulation model used in the modal analysis. Therefore, defining a reliable numerical model to calculate the modal parameters is essential in characterizing the dynamic behavior of the motor's structure subjected to external loads.

In the case of dynamic simulations, inertia, elasticity and damping (if present) of the system play an important role.

Damping is a set of energy-dissipation mechanisms that causes free vibration movement to decrease in amplitude over time. Introducing the correct value for the damping coefficient plays an important role in measuring the motor's structure correct frequency response (nodal displacements, velocities and accelerations). Stresses and relative deformations will depend directly proportional with the damping of the structure, increasing with the diminishing of the damping coefficient and vice-versa. When analyzing the impact of the damping coefficient on the harmonic response analysis results for a 45 kW electric motor mounting flange, for a compressor application, it can be observed that the maximum principal stresses vary from 92 MPa for a 2% damping coefficient (Fig. 1, up) to 37 MPa for a 5% damping coefficient (Fig. 1, down), for the same excitation.

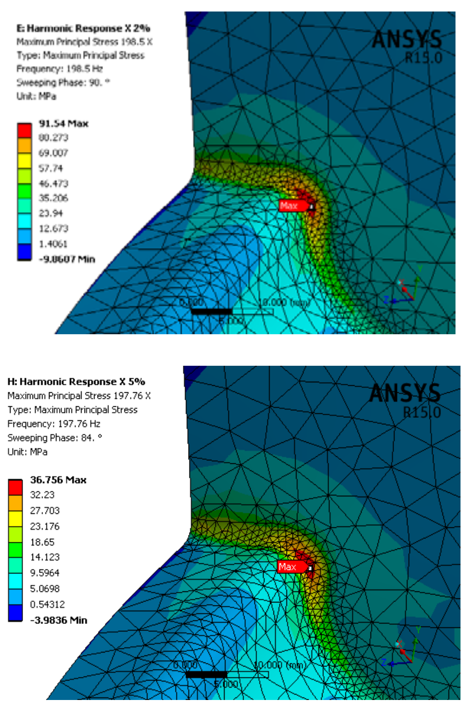

Fig. 1. Variation in stress levels for different damping coefficients

The value of the global damping coefficient is specific to each electric motor, in function of its size and construction (materials used, connections between components, tightening torques applied to the assembly elements, functional plays, tightening between assembled parts, etc.). For most part of the mechanical structures, the damping coefficient is considered to be around 5%. Working with a more precise value would assume determining it by experimental means using the impact hammer method, for example.

3. MODAL ANALYSIS OF THE INDUCTION MOTORS' STRUCTURE

3.1. NUMERICAL MODAL ANALYSIS

The dynamical equations of some mechanical systems can be written in matrix form (1), as follows [14]:

$$M\ddot{Q} + C\dot{Q} + KQ = F \tag{1}$$

where Q is the vector of the system generalized coordinates, M, C, K are the mass (inertia), viscous damping and stiffness matrices and F is the vector of generalized forces. When doing the normal mode analysis of the system, the damping is neglected and the external forces are not acting, resulting (2):

$$M\ddot{Q} + KQ = 0 \tag{2}$$

Assuming a harmonic and synchronous motion in the structure, when all coordinates perform in time the motion in phase or out of phase, the following solution is proposed:

$$Q(t) = u \cdot \cos(\omega\, t - \varphi) \tag{3}$$

where ω is considered the natural frequency of the whole system and u is a constant n–vector of amplitudes. By replacing the proposed solution in the system of differential equations, results:

$$Ku - \omega^2 Mu = 0 \ or \ (K - \omega^2 M) \cdot u = 0 \qquad (4)$$

The set of the homogeneous algebraic equations (4) has the unknown vector u. Considering $\lambda = \omega^2$ as a parameter, one get:

$$Ku = \lambda Mu \qquad (5)$$

known as the eigenvalue problem when trying to determine λ values for which the system (4) has nontrivial solutions. By solving for λ and nontrivial solution ($u \neq 0$), the following characteristic equation (6) is obtained:

$$\det(K - \lambda M) = 0 \qquad (6)$$

where λ_r ($r=1,2\ldots,n$) values are the eigenvalues (or characteristic values) of the system. The natural frequencies of the system are derived:

$$f_r = \sqrt{\lambda}_r / 2\pi \quad r = 1,2,\ldots n \qquad (7)$$

For each determined eigenvalue λ_r, a vector u_r ($r=1,2\ldots,n$) named eigenvector, defining the associated mode shape of vibration, is satisfying the following equation:

$$Ku_r = \lambda_r Mu_r \quad r = 1,2,\ldots,n \qquad (8)$$

Numerically, there are a series of hypotheses that are implemented, valid for all modal analyses:

- there are no nonlinearities admitted in the numerical model, therefore materials are linear (the stiffness matrix [K] and the mass matrix [M] are constant, the material behavior is linear elastic, no plasticity being allowed), all contacts have to be linear (bonded) and the nonlinear contacts (frictional or frictionless) will be automatically transformed into linear ones;
- small deformations theory is used;
- the damping matrix [C] is not present, therefore damping is not taken into consideration;
- there is no external excitation on the motor's structure (free vibrations);
- the motor's structure may or may not be constrained (geometrical boundary conditions present or not).

The required and sufficient characteristics to define a material for a modal analysis are the density and any two of the following parameters: the modulus of elasticity (Young's modulus, E), the compressibility modulus (bulk, K), the shear modulus (G) and the Poisson's ratio (□) in the form of constant values.

In a modal analysis, only linear elastic, isotropic or orthotropic materials are accepted.

The boundary conditions for continuous systems are [7] either natural, which must be satisfied as a result of the balance of forces and moments, or geometric, which must be satisfied due to the imposed geometric constraints.

The modal analysis is the analysis of free vibrations, considering that the external forces are null. Therefore, the boundary conditions are natural, the derivatives of the displacements of the nodes found on the solid's outer surface being null.

Furthermore, referring to the geometric boundary conditions, if the system is not constrained or insufficiently constrained, rigid body modes may occur. Also, the choice of the geometric boundary conditions will affect the shape of the vibration modes and the associated frequency values. In Fig. 2, comparative results are presented for two different boundary conditions setups, in the case of a standard 4 poles, 250 kW, 355 mm frame size induction motor: 0 degrees of freedom on the feet fixing holes (left side images – setup no. 1) versus a more realistic modelling including a chassis on which the motor is fixed through four bolts, having integrated axial pretension forces of 130000 N per bolt, corresponding to the bolts tightening torque value (right side images – setup no. 2). Plus, the contact between the motor's feet and the chassis is assumed to be a frictional one, with a frictional coefficient chosen for the cast iron – steel contact pair, while the chassis is considered fixed and infinitely rigid. Knowing that the stiffness matrix of the system for the setup no. 2 is different, a static structural analysis has to be run prior to the modal analysis in order to calculate the new deformation state generated by the bolts pretension forces.

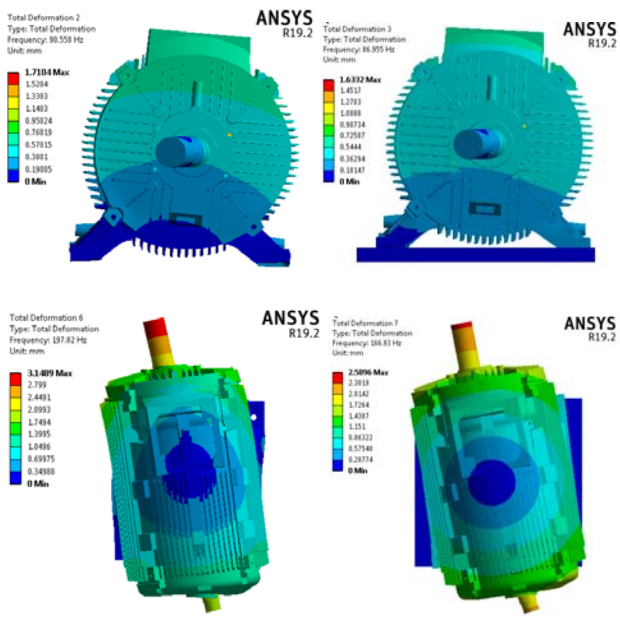

Fig. 2. Variation in eigenfrequencies values for different boundary conditions: lateral tilting mode at 91 Hz (setup no. 1) vs. 87 Hz (setup no. 2), torsional mode at 198 Hz (setup no. 1) vs. 167 Hz (setup no. 2)

Due to the fact that there is no excitation in the model, the modal shapes are relative. Therefore, the displacements of the model do not have real values, but only provide information about the eigenvector of the structure for each mode and the associated frequencies. In Fig. 3, the first three natural mode shapes for the housing of a standard 80 mm frame size electric motor are exemplified, the corresponding critical frequencies being calculated at 196 Hz, 437 Hz and 633 Hz.

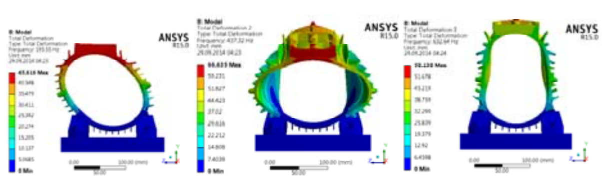

Fig. 3. Exemplifying the first three natural mode shapes for the housing of a standard 80 mm frame size motor

There are several numerical methods available to solve the dynamical equations described above. The eigensolver used to describe the vibration behavior of induction motors in the present research is the Block Lanczos, under the ANSYS Workbench platform. It is suitable for calculating many modes for large numerical models, even if the model consists of poorly shaped solid and shell elements. The Block Lanczos eigenvalue solver uses the Lanczos algorithm, where the Lanczos recursion is performed with a block of vectors. The Block Lanczos method uses a sparse matrix solver. The method is especially powerful when searching for eigenfrequencies in a given frequency range of a system [24].

3.2. EXPERIMENTAL MODAL ANALYSIS

It is possible to describe the dynamic behavior of any mechanical structure by means of modal parameters, namely resonant frequencies, damping at resonance and the associated vibration mode shapes (characterized by amplitudes and phases). Experimental modal analysis is a set of methods to determine the modal dynamic characteristics of a structure. Through experimental modal analysis one can measure the dynamic response of the motor's structure under the action of forces that are supposed to act upon it in its working environment [1].

Experimental modal analysis (EMA) is the process of extracting modal parameters of a mechanical structure starting from vibration data measured on it [12], [13], [14], [15], [18], [22], [23].

There are several different ways to extract the modal parameters from the measurements made on a structure. The main difference between them refers to the domain in which the answers are dealt. From this point of view, there are methods based on measurements in the time domain and in the frequency domain.

When the frequency response functions (FRFs) are used, the extraction process is called frequency response domain analysis. FRF is a description of the input-output relationship as a function of frequency between two degrees of freedom of the structure. When the impulse response functions (IRFs) are used, the approach is called time domain modal analysis. The mathematical model in case of frequency domain modal analysis is an analytical expression of a FRF best fitted with the measured FRFs. In reality a structure has an infinite number of DOF and poles [17].

In both cases of time domain and frequency domain measurements, there are direct methods (assessing mass, stiffness and damping matrices without further calculating the modal parameters [17]) and indirect (which estimate the modal parameters from the physically measured response and which allow the construction of the modal model [17]). Over a certain frequency range, there are usually more resonances. On one hand, some methods allow the extraction of the modal parameters taking into account all resonances simultaneously. These are called methods with more degrees of freedom (MDOF, such as, for example, impact hammer testing, in which all modes are simultaneously excited). On the other hand, there are some indirect methods in the frequency domain with which the modal parameters can be estimated by carrying out a progressive analysis until the entire frequency domain is covered. These are called single degree of freedom methods (SDOF, such as, for example, testing on a shaker) [17].

Lightly damped modes have narrow peaks in frequency domain described by a few points, hence the modal parameters are not so easy to be determined. On the contrary, in time domain, the signal of the impulse response has long duration decay and is easier to be identified. Over the last years, techniques for parameter identification only from output data (input being the operational data) have been developed. One can mention Auto-Regressive Moving Averaging models (ARMA), Natural Excitation Technique (NexT), stochastic subspace methods and others [14].

The accuracy of the dynamic model and the accuracy of the measurements are directly linked, therefore it is important to choose the most appropriate test method, depending on the type and size of the structure, whether the structure behaves linear or not, the destination of the experimental results and resources available.

Experimental modal analysis on induction motors could have the following usages [1]:

- detecting noise and vibration problems;
- design optimization;
- monitoring;
- validation of analytical or finite element simulation models.

Detecting noise and vibration problems: the source of noise and vibrations is identified by conducting vibration measurements while the motor is in operation. In most cases the problem in caused by a resonance of the system.

Design optimization: by proposing and testing modifications on the corresponding dynamic model, the resonance could be removed or moved away from the offending frequency by implementing solutions on the structure without dismantling the motor from its working environment (e.g. by mounting stiffening elements or, on the contrary, vibration absorbers).

Monitoring: modal parameters' evolution in time could be used to monitor the dynamic behavior of the motor's structure. Their evolution suggests a change in stiffness, a structural weakness. In most cases, bearings' failure is the cause of the altered dynamic behavior of the motor.

Validation of analytical or finite element simulation models: after the construction of a prototype and the carrying out of a modal analysis on it, the experimental results can be compared with the numerical results. In this way, the analytical or finite element model can be improved, thus improving the dynamic model without further physical testing.

A special type of Experimental Modal Analysis (EMA) is the Operational Modal Analysis (OMA). Operational modal analysis is a technique by which modal parameters are estimated from the structure's response without knowing the excitation force.

In the case of induction motors, structures with moving parts, the classic experimental modal analysis might have a series of disadvantages [20]:

- the structure usually has a large mass, which makes difficult the excitation of all modes simultaneously;
- the modal parameters are determined on a static structure. Therefore, the influence of temperature during operation, for example, could not be quantified.

Operational modal analysis enables dynamic behavior to be determined without decommissioning of the electrical machine. Therefore, OMA could have the following benefits [20]:

- the test procedure is simpler and the operation of the tested electric motor does not have to be interrupted;
- reduced costs since the motor runs without interruption;
- the modal parameters are determined in real machine operating conditions; for example, dependence on the operating temperature can be investigated.

3. 3. VIBRATION BEHAVIOR OF THE INDUCTION MOTOR

Vibrations of the entire structure of the motor are those induced through the chassis that they are mounted to the final customer on. Chassis vibrations excite the natural vibration modes of the motor and the whole system suffers the resonance phenomenon. Generally, these vibrations occur at low frequencies, so they could be avoided by a robust design and a smaller mass of the electric motor, both of which have the effect of increasing the frequencies of the tilting modes of the motor (the first modes that appear, having low frequencies and large amplitudes). Some examples of situations where induction motors could be subjected to external loads causing resonance phenomenon are: an electric motor mounted on a pump operating at a given frequency, a motor operating on a vessel excited at a certain frequency by the waves, a motor in operation on an excavator whose structure is excited by the vibrations of an internal combustion engine, vibrations transmitted through a chassis from neighboring equipment, seismic loads absorbed during an earthquake, and so on.

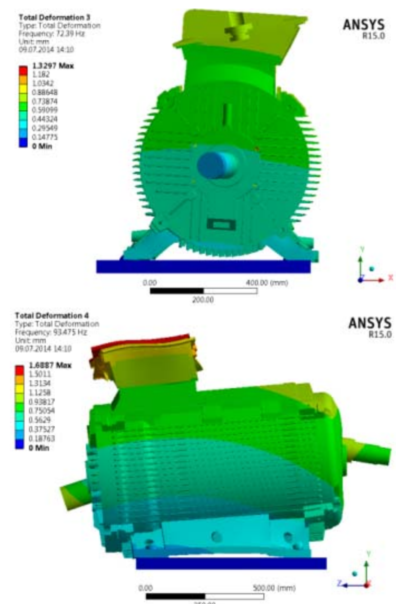

Fig. 4. First two tilting modes (important mass excited)

In Fig. 4, the first two tilting modes of a standard, 4 poles, 250 kW, 355 mm frame size induction motor are offered as an example: the tilting mode in the transverse direction and a second one, in the longitudinal direction.

The rotor's vibrations are some of the most difficult to control, as they are influenced by a set of constructive elements:

- The shaft: the larger its diameter, the more the frequencies of the bending modes of the rotor are increased, but the dimensioning of the electrical heart restricts its design; the larger the distance between the bearings, the more the frequencies of the bending modes of the rotor are decreased and vice-versa. Fig. 5 illustrates the first three typical bending modes of a rotor.

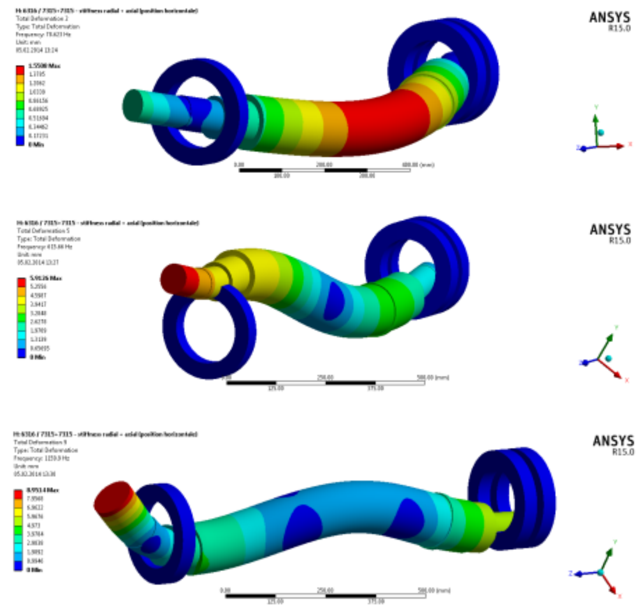

Fig. 5. First three typical bending modes of a rotor supported by flexible bearings

- The rotor stack: adds mass to the shaft, so it lowers the critical frequencies of the rotor. In addition, the only construction solution that adds a certain stiffness to the shaft is the one with injected aluminum bars (squirrel cage type) that gives the lamination stack a certain bending strength, so robustness. Reducing mass through electrical design would have a beneficial effect. Considering the variety of materials (steel and aluminum for the rotor core) and the very large number of components (hundreds of stacked laminations), it is difficult to approximate the physical and mechanical properties of the rotor stack, such as the density of a single equivalent material or the stiffness brought to the shaft, elements that influence the dynamic behavior of the overall assembly. Therefore, there is a need to define simplified computational models, integrated into the global numerical model. In [8] and [9], authors propose solutions to model an equivalent material and the contact between the laminations, on one hand, and a model describing the stiffness of the stack, on the other hand.
- Bearings: they are elastic, flexible elements that lower the stiffness of the rotor. This is why the tendency is to use bearings with increased stiffness given by construction and axial pretension. However, the tendency in the industry nowadays is to increase in speed, which requires the use of smaller bearings, thus more flexible, with a conservative effect on the rotor's natural frequencies. Therefore, a more expensive version would be the roller bearings having a radial stiffness greater than in the case of the radial ball bearings. The problems that arise in this case refer to the need of a very precise machining of the motor's flanges, the deviation of the concentricity between the two being in the order of the hundredths of a millimeter. A first approach in approximating the stiffness of the radial ball bearings is the one by which the radial stiffnesses of the bearings are calculated, the stiffness they oppose with when the inner and outer rings are displaced one towards the other radially. This stiffness depends only on the following parameters: the balls diameter (D_w), the number of balls (Z) and the radial forces the bearings are subjected to (F_r). Thus, the expression of the displacement (x) of the two rings with respect to the radial force applied and the geometrical parameters is [4]:

$$x = \frac{0.00043 \cdot 5^{\frac{2}{3}}}{Z^{\frac{2}{3}}} \cdot \frac{F_r^{\frac{2}{3}}}{D_w^{\frac{1}{3}}} \tag{9}$$

There is a non-linear dependence between the applied radial effort and the displacement of the rings. Therefore, the stiffness of the bearing is not constant, but depends on the static load to which it is subjected. In order to capture as accurately as possible the influence of the bearings' flexibility on the bending modes of the shafts, it is necessary to calculate the torsional stiffnesses of the bearings, as well as the axial ones.

- Loss of the rotor's balancing: is very common for variable speed motors. The rotors being balanced at a certain speed imposed by the balancing machine, as the speed increases, imbalances occur which may be compensated or improved by pre-balancing the rotor in multiple planes.

• Low flanges stiffness: this flaw must be detected from the design stage and improved by stiffening the flanges. They directly influence the rotor bending modes in a conservative way.

The stator's vibrations have electromagnetic sources and can be deepened by studying the interface between the stator and the housing. It has been observed that an adequate tightening reduces the acoustic level of the electromagnetically-excited vibration modes occurring in the stator. A comparative study is presented in this paper, in order to demonstrate the influence of the interface pressure on the natural frequencies of the stator's vibration modes. Fig. 6 – up illustrates the geometry of the stator itself, while Fig. 6 – down illustrates the stator-housing assembly (liquid cooled motor application). The tightening between the two parts (0.18 mm / diameter, for a Ø695 mm stator) has been previously solved through a static structural analysis, coupled with the modal one.

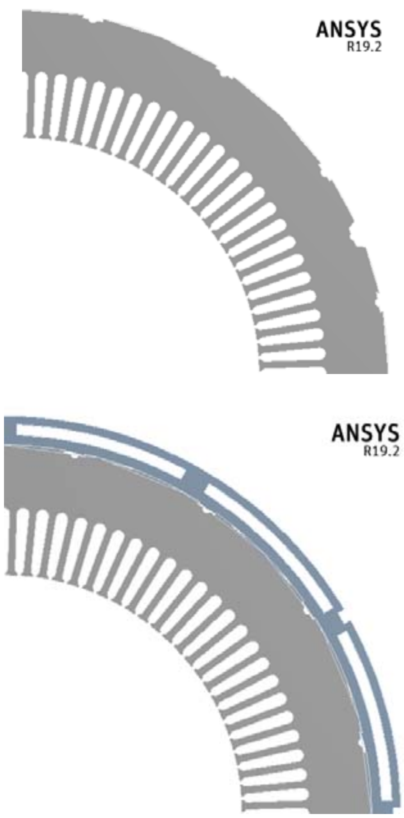

Fig. 6. The 2D geometries of the stator

Figure 7 presents the mode shapes and the associated natural frequencies for the stator itself, calculated through a 2D analysis, while Fig. 8 presents the results for the stator-housing pretensioned assembly.

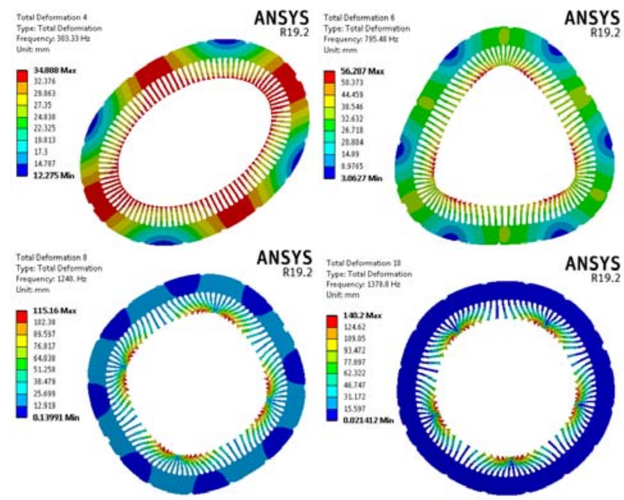

Fig. 7. First four modes for the stator itself at 303 Hz, 795 Hz, 1240 Hz and 1379 Hz

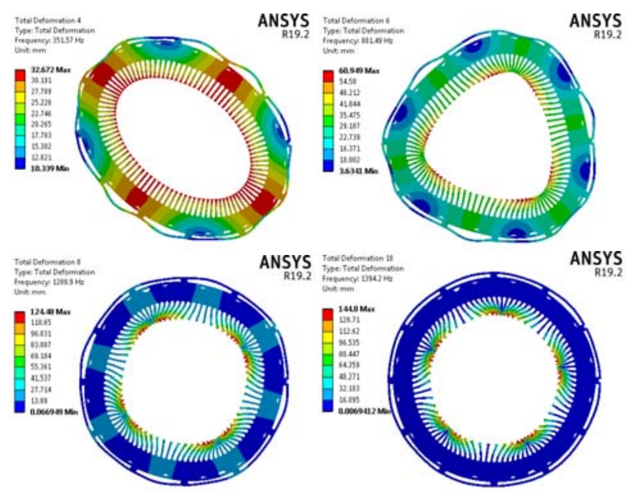

Fig. 8. First four modes for the stator-housing assembly at 352 Hz, 882 Hz, 1290 Hz and 1395 Hz

Vibrations of the flexible components refer to the vibrations of fan covers, terminal boxes, forced-ventilation sub-assemblies, and so on, that may occur. Generally these are not dangerous, causing only high levels of vibration and noise during operation, therefore must be corrected in order to improve the NVH (Noise and Vibration

Harshness). Fig. 9 illustrates some of the vibration mode shapes for the plastic fan cover on a standard 80 mm frame size, flange mounted induction motor, the corresponding critical frequencies being calculated at 303 Hz, 542 Hz and 555 Hz.

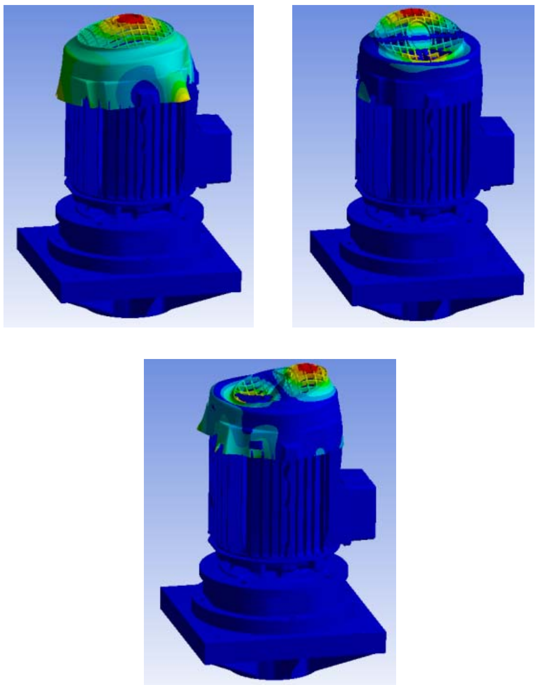

Fig. 9. Vibration modes for a plastic fan cover

4. CONCLUSIONS

The modal analysis of the motor's structure can serve as a standalone analysis whose results could be exploited with the objective of avoiding resonance occurrence through design evolutions. Furthermore, modal analysis results could be used as input data for other frequency response analyses for studying the dynamic behavior of its structure subjected to external loads.

As seen in Chapter 3.3, motor's vibrations could be studied on individual components, sub-assemblies or even on the entire structure.

Studying vibrations phenomena on individual components shouldn't pose important problems, neither from experimental, nor from numerical point of view, since mass and stiffness data could be extracted from EMA.

However, the uncertainties multiply once one starts to study the vibrational behavior of some sub-assemblies (rotor, stator) or even the whole motor assembly. These uncertainties may come from unknown material mechanical properties, interfaces between components (frictional coefficients, contact pressures, stiffness induced by press fits, stiffness induced by the bolted assemblies, stiffness induced by thermal expansions of the parts, and so on) and realistically imposed boundary conditions.

Overcoming these uncertainties would imply developing and validating advanced numerical models describing the fundamentally non-linear behavior of some sub-assemblies, such as the bearings and the rotor and stator lamination stacks. Once these models validated, they should be integrated in the motor's global dynamic model. For the validation of the latter, authors of this paper propose a sequential validation technique, which consists in starting from one single part, such as the housing (for which a numerical model could be easily built using data from EMA) and progressively build up the complete model by adding other parts or sub-assemblies. It would be mandatory to fine tune each intermediary model with data from EMA conducted on the corresponding prototypes.

Authors will approach this consequent work in order to develop and validate the induction motor's global dynamic model with a certain level of universality, regardless of the frame size.

5. ACKNOWLEDGMENTS

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Considerații cu privire la analiza modală şi simularea vibrațiilor motoarelor cu inducție

***Rezumat**: In articol sunt urmărite simulările vibrațiilor motoarelor electrice cu inducție, prin metoda elementelor finite, cu o preocupare pentru analiza modală. Sunt trecute în revistă şi o serie de alte simulări dinamice, comentându-se importanța acestora: analiza răspunsului în frecvență a structurii la solicitări armonice, răspunsul structurii la solicitări aleatoare şi la şoc mecanic. Importanța analizei modale este pusă în evidență din punct de vedere teoretic şi experimental, pe de o parte, şi prin realizarea şi comentarea unor simulări pentru componentele motorului electric (carcasă, rotor şi stator) şi pentru structura asamblată, pe de altă parte.*

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