Structure-Dynamic Analysis of an Induction Machine Depending on Stator–Housing Coupling

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***Abstract*—The estimation and the calculation of the acoustic sound of electric machinery are of particular interest nowadays. Various approaches have been presented, relying either on analytical or on numerical models. The analytical models presented here are based on the electromagnetic-field theory. Numerical models are applied to derive the exciting forces stemming from various sources and effects. The numerical results have to be verified. Hence, they are compared with the physically based analytical results. The radiated noise depends directly on the surface deformation of the machine. Therefore, the analysis is focused on the structure-dynamic vibration. The combined analysis presented here allows for the reduction of vibration and noise, optimizing the coupling of the machine's stator and housing. The studied induction machine's housing is mounted with six spiral-steel springs to the stator. With the presented method, the impact of different numbers of springs is analyzed exemplarily.**

***Index Terms*—Audible noise, deformation, finite-element method (FEM), induction machine, structure dynamics, vibrations.**

I. Introduction

THERE HAVE been several contributions to both the analytical [1]–[3] and numerical [4]–[6] approaches of estimating the radiated noise of electrical machinery. A comparison, as well as a combination, of both methods allows for more reliable predictions and faster improvements of the machine's structure. In this paper, an induction machine (IM) with squirrel-cage rotor is studied by means of analytical and numerical methods. At first, the applied models are introduced. In general, the structure of an IM is not purely cylindrical as the analytical models of [1]–[3] assume. For comparison reasons, different numerical finite-element (FE) models are introduced, and results are analyzed.

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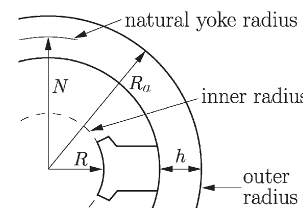

Fig. 1. Simplified analytical model of IM with teeth.

II. Analytical Model

The analytical model [1] is based on the analysis of the force-wave behavior resulting from the normal component of the air-gap flux density B_n depending on space x and time t

$$F_r(x,t) = \frac{B_\mathrm{n}^2(x,t)}{2\mu_0} \tag{1}$$

where μ_0 is the magnetic field constant. $B_\mathrm{n}^2(x,t)$ results from the fundamental and harmonic fields of the stator interacting with the induced fundamental and harmonic fields of the rotor. Three major effects are considered in the analytical model: the fundamental air-gap field, the saturation of the lamination, and the static and dynamic eccentricities. Each harmonic, i.e., each exciting force-wave frequency, results in oscillating space modes along the circumference of the stator at the air gap. The mode number r depends on the interacting field components of stator and rotor.

These force waves excite the structure of the machine, i.e., stator and housing. The analytical model simplifies the machine's structure to a cylindric ring, as shown in Fig. 1. In order to include the effect of slotting, the cylinder-ring model is modified, taking the teeth into account introducing the adjusting factor

$$\Delta = \frac{\text{yoke weight}}{\text{tooth weight} + \text{yoke weight}}. \tag{2}$$

The weight of yoke and teeth is the equivalent to the corresponding cross sections. The eigenfrequency of $r = 0$ reads

$$F_0 = \frac{C_\mathrm{s}}{2\pi \cdot N \cdot \sqrt{\Delta}} \tag{3}$$

with

$$C_\mathrm{s} = \sqrt{\frac{E}{\rho}}. \tag{4}$$

TABLE I
STATIC DEFORMATION FACTORS FOR DIFFERENT MODES r

r	0	1	2	3	4	5	6
$\eta_{r,stat}$	1.0	596.4	35.7	5.0	1.4	0.6	0.3

C_s is calculated by taking the mass density ρ and the Young's modulus E into account. With the analytical model, the deformation magnitude of the analyzed oscillation mode r is estimated on the outer radius of the stator R_a. For this, the static and dynamic deformation factors need to be calculated for an adequate cylinder ring. Since $r = 0$ results in pure tensile stress, the static deformation is calculated to

$$Y_{0,\mathrm{stat}} = \frac{R \cdot N}{E \cdot h} \cdot \sigma(f, r = 0) \tag{5}$$

with the natural yoke radius N, the height of the yoke h, and the inner radius of the stator R. The static deformation for mode number $r \geq 2$ is estimated with

$$Y_{r,\mathrm{stat}} = \frac{R \cdot N}{E \cdot h} \cdot \frac{\sigma}{i^2 (r^2 - 1)^2}, \qquad \text{for} \quad r \geq 2 \tag{6}$$

with

$$i = \left(\frac{1}{2\sqrt{3}}\right) \cdot \left(\frac{h}{N}\right). \tag{7}$$

The static factor as a ratio of $Y_{r,\mathrm{stat}}/Y_{0,\mathrm{stat}}$ reads

$$\eta_{r,\mathrm{stat}} = \frac{12}{(r^2 - 1)^2} \cdot \left(\frac{N}{h}\right)^2, \qquad \text{for} \quad r \geq 2. \tag{8}$$

Bending forces are generated by $r = 1$. In this special case, the corresponding static factor reads

$$\eta_{1,\mathrm{stat}} = \frac{4}{3} \frac{h \cdot l_\mathrm{Fe}}{N \cdot \left(\frac{d}{L}\right)^4 \cdot L} \tag{9}$$

where l_Fe is the effective stack length, and L is the distance between both bearings. For $r \geq 1$, the factors previously given are multiples of the deformation calculated for $r = 0$. Table I resumes the calculated static deformation factors for the studied IM.

The relative sensitivity of the structure γ is defined as the ratio of the force-wave harmonic f_r and the eigenfrequency F_0. With this and the bending and longitudinal oscillation frequencies f_r^B and f_r^L, respectively, the dynamic factor reads

$$\eta_{r,\mathrm{dyn}} = \frac{r^2 - \gamma^2}{\left[\gamma^2 - \left(\frac{f_r^\mathrm{B}}{F_0}\right)^2\right] \cdot \left[\gamma^2 - \left(\frac{f_r^\mathrm{L}}{F_0}\right)^2\right]}, \qquad \text{for} \quad r \geq 2. \tag{10}$$

In the special case $r = 1$, the lowest bending eigenfrequency is of interest

$$F''_\mathrm{b1} = \frac{1}{2\pi} \sqrt{\frac{c''_1}{m''}}. \tag{11}$$

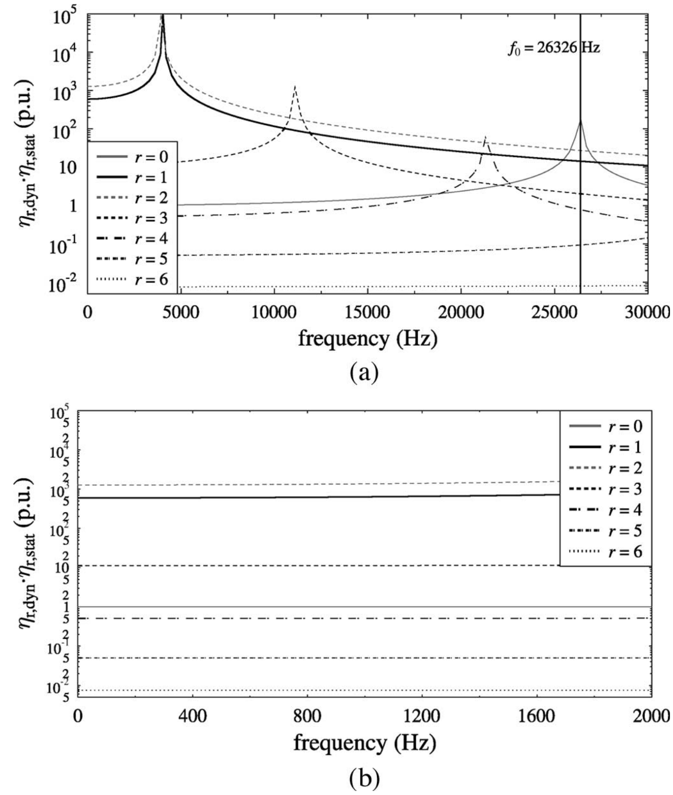

Fig. 2. (a) Resulting factor $\eta_\mathrm{stat} \cdot \eta_\mathrm{dyn}(r)$. (b) Resulting factor $\eta_\mathrm{stat} \cdot \eta_\mathrm{dyn}(r)$ in analyzed frequency range.

For a machine with the shaft diameter d, the spring constant c''_1 and the adequate mass m'' read

$$c''_1 = \frac{3\pi}{4} \cdot E \cdot \left(\frac{d}{L}\right)^4 \cdot L. \tag{12}$$

The adequate mass m'' is calculated by

$$m'' = \rho_\mathrm{Fe} \cdot \left\{ l \left[(2R)^2 - d^2\right] + \frac{1}{2} \cdot L \cdot d^2 \right\} \tag{13}$$

with the mass density ρ_Fe of the rotor. The dynamic deformation factor $r = 1$ reads

$$\eta_{1,\mathrm{dyn}} = \frac{1}{1 - \gamma^2 \cdot \left(\frac{F_0}{F''_\mathrm{b1}}\right)^2}. \tag{14}$$

Finally, the overall deformation amplitude is calculated by

$$Y_r = \eta_{r,\mathrm{stat}} \cdot \eta_{r,\mathrm{dynamic}} \cdot Y_{0,\mathrm{stat}}. \tag{15}$$

Fig. 2(a) shows the resulting behavior of the factor $\eta_\mathrm{stat} \cdot \eta_\mathrm{dyn}(r)$. Each mode number r shows a resonance. Due to the small size of the studied IM (800 W), these resonance frequencies are rather high. For $r \geq 4$, they are beyond the human ear's hearing ability. Next to this, the modes $r \geq 3$ produce rather small amplification factors throughout the spectrum. For the analysis of the studied machine, the spectrum is reduced to $f_\mathrm{max} = 1200$ Hz. Here, the entire range of frequencies shows