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DYNAMIC LOADING OF A SQUIRREL CAGE MOTOR FOR VARIOUS VEHICLES

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Abstract: *Squirrel cage induction motors are employed in various machines and means of transport mostly as traction motors. The usage of this type of motors in vehicles can be connected with possible fatigue problems because of non-stationary loading and changing operational conditions. The complex dynamic model of an electro-motor rotor is described in this paper. It is based on solid and beam finite elements, on rigid bodies and on special couplings derived in a rotating coordinate system. The loading conditions and states are summarized and each dynamic excitation caused mainly by electro-magnetic field fluctuations and or track irregularities is described. The whole methodology was supported by experimental measurements and it was applied to a real motor.*

Keywords: ***Vibrations, electromotor, rotor dynamics, finite element method, steady state response.***

1. Introduction

An important example of rotating systems is a squirrel cage induction motor which is used in various machines and means of transport mainly as a traction motor. The operation of this type of motor in vehicles can be connected with possible fatigue problems because of non-stationary loading and changing operational conditions and therefore it is necessary to analyze dynamic response of the rotor.

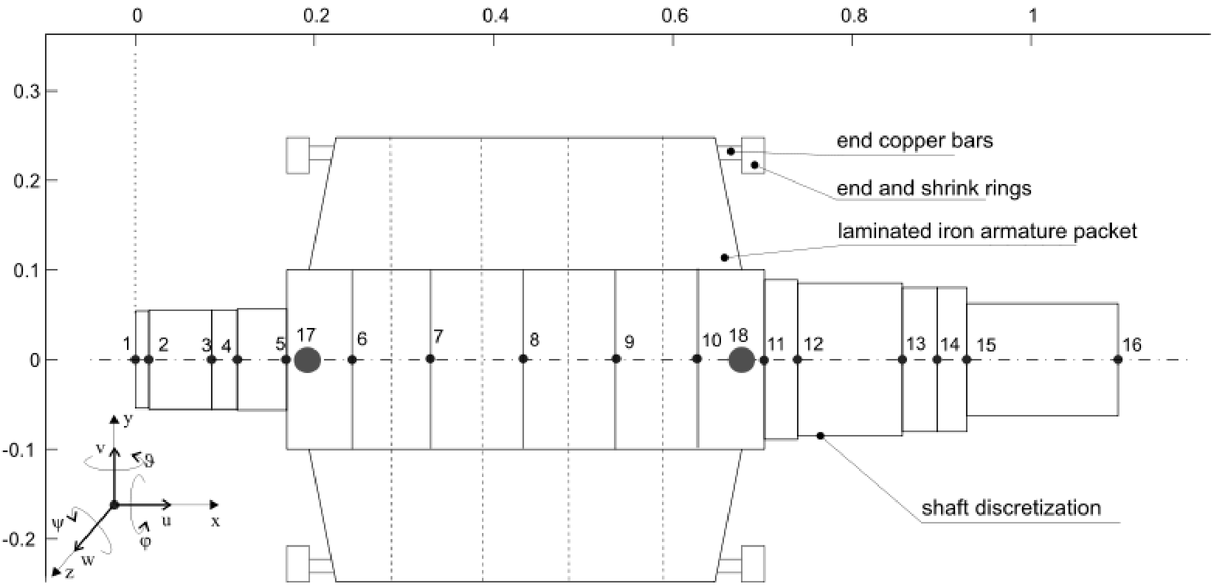

Fig. 1: Scheme of a rotor of a squirrel cage motor.

The scheme of the rotor of this studied type of electric motor is shown in Fig. 1. The rotor consists of a rotor shaft with laminated iron armature mounted in the middle shaft part. Laminations have the form of thin sheet metals with holes. The squirrel cage is composed of rotor bars (usually made of copper) passing through the laminations and joined by end rings on both ends (also made of copper). The end rings can be constricted by shrink rings made of certain more stiff material and mounted with initial

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tension. The short end pieces of rotor bars extruded from the armature are deformed during the rotor operation due to rings inertia and electromagnetic forces. Fatigue problems can be characterized by ruptures of the end ring and rotor bar connections.

This paper describes an advanced dynamic model of the rotor and summarizes the possible excitation modes. According to the literature review, e.g. Caruso et al. (2008) or Enge & Maisser (2005), the usage of such complex models of squirrel cage motor rotors is very unique.

2. Advanced dynamic model of a rotor of a squirrel cage motor

The originally developed complex dynamic model of the rotor of the squirrel cage motor is described in this section. The advanced model is characterized by the end and shrink rings considered as flexible bodies and copper bars end modelled as a set of beams revolved by defined radius. This model is intended for high-frequency and comprehensive dynamic analyses. The flexible shaft with the mounted laminated armature is modelled by the same way as in the basic dynamic model (see Hajžman et al., 2010a). The basic dynamic model was introduced in Hajžman et al. (2010b) and was characterized by considering ideally rigid end and shrink rings.

The end and shrink rings in the advanced dynamic model are considered to be 3D continuum and therefore they are modelled by solid finite elements derived in the rotating coordinate system, similarly as the shaft finite elements in Hajžman et al. (2009) and Šašek (2010). The eight-node rotating solid finite element with three translational displacements in each node is described in more detail e.g. in Šašek (2010).

The end and shrink rings (left denoted by subscript L, right denoted by subscript R) are represented in the advanced dynamic model by the mass matrices $\boldsymbol{M}_{L1}$, $\boldsymbol{M}_{R1}$, matrices of gyroscopic effects $\omega_0\boldsymbol{G}_{L1}$, $\omega_0\boldsymbol{G}_{R1}$ (ω_0 is the rotor angular velocity), stiffness matrices $\boldsymbol{K}_{L1}$, $\boldsymbol{K}_{R1}$ and spin softening matrices $\omega_0^2\boldsymbol{K}_{dL1}$, $\omega_0^2\boldsymbol{K}_{dR1}$. The order of these matrices is given by the number of degrees of freedom of the discretized rings and they are composed of particular element matrices. The model should be also extended by the vectors of centrifugal forces $\omega_0^2\boldsymbol{f}_{L1}$ and $\omega_0^2\boldsymbol{f}_{R1}$ (see Hajžman et al., 2009). The model of the end and shrink rings can be written in the form

$$\boldsymbol{M}_{L1}\ddot{\boldsymbol{q}}_{L1}(t)+(\boldsymbol{B}_{L1}+\omega_0\boldsymbol{G}_{L1})\dot{\boldsymbol{q}}_{L1}(t)+(\boldsymbol{K}_{L1}-\omega_0^2\boldsymbol{K}_{dL1})\boldsymbol{q}_{L1}(t)=\omega_0^2\boldsymbol{f}_{L1} , \tag{1}$$

$$\boldsymbol{M}_{R1}\ddot{\boldsymbol{q}}_{R1}(t)+(\boldsymbol{B}_{R1}+\omega_0\boldsymbol{G}_{R1})\dot{\boldsymbol{q}}_{R1}(t)+(\boldsymbol{K}_{R1}-\omega_0^2\boldsymbol{K}_{dR1})\boldsymbol{q}_{R1}(t)=\omega_0^2\boldsymbol{f}_{R1} , \tag{2}$$

where $\boldsymbol{B}_{L1}$ and $\boldsymbol{B}_{R1}$ denote material damping matrices.

End parts of flexible bars connecting end rings and laminated armature (see Fig. 1) are considered to be 1D continuum. Special beam finite elements which are rotating with defined radius were developed for this purpose (Šašek, 2010) and their model (L for left, R for right) is of the form

$$\boldsymbol{M}_{L2}\ddot{\boldsymbol{q}}_{L2}(t)+(\boldsymbol{B}_{L2}+\omega_0\boldsymbol{G}_{L2})\dot{\boldsymbol{q}}_{L2}(t)+(\boldsymbol{K}_{L2}-\omega_0^2\boldsymbol{K}_{dL2})\boldsymbol{q}_{L2}(t)=\omega_0^2\boldsymbol{f}_{L2} , \tag{3}$$

$$\boldsymbol{M}_{R2}\ddot{\boldsymbol{q}}_{R2}(t)+(\boldsymbol{B}_{R2}+\omega_0\boldsymbol{G}_{R2})\dot{\boldsymbol{q}}_{R2}(t)+(\boldsymbol{K}_{R2}-\omega_0^2\boldsymbol{K}_{dR2})\boldsymbol{q}_{R2}(t)=\omega_0^2\boldsymbol{f}_{R2} , \tag{4}$$

where the corresponding matrices is of the same meaning as in the previous case of rings but are derived for rotating beam finite elements.

The remaining part of the rotor is the shaft with the mounted laminated armature that is modelled using rotating shaft finite elements (see Hajžman et al., 2009) with defined rigid bodies of the discretized armature packet and defined elastic properties of the packet. The shaft and armature model can be written as

$$\begin{aligned}&(\boldsymbol{M}_S+\boldsymbol{M}_A)\ddot{\boldsymbol{q}}_S(t)+(\boldsymbol{B}_S+\boldsymbol{B}_A+\omega_0\boldsymbol{G}_S+\omega_0\boldsymbol{G}_A)\dot{\boldsymbol{q}}_S(t)+\\&+(\boldsymbol{K}_S+\boldsymbol{K}_A-\omega_0^2\boldsymbol{K}_S-\omega_0^2\boldsymbol{K}_A)\boldsymbol{q}_S(t)=\boldsymbol{0}\end{aligned} \tag{5}$$

and is created from the matrices representing the flexible shaft (subscript *S*) and the discretized armature with its elastic properties (subscript *A*). Bearing characteristics are included in damping matrix $\boldsymbol{B}_S$ and stiffness matrix $\boldsymbol{K}_S$.

The motion equations (1) to (5) belong to the uncoupled rotor subsystems. In order to obtain the whole rotor model, the transformation between generalized coordinates of the particular subsystems should be derived (e.g. the last nodes of the flexible bars in (3) and (4) should be connected with the appropriate nodes of the end rings in (1) and (2)). The whole transformation defined by transformation matrix $\boldsymbol{T}_R$ between original uncoupled general coordinates $\boldsymbol{q}(t)$ and new coupled general coordinates $\tilde{\boldsymbol{q}}(t)$ is $\boldsymbol{q}(t) = \boldsymbol{T}_R\,\tilde{\boldsymbol{q}}(t)$. Then the whole rotor model is (see Šašek, 2010) of the form

$$\tilde{\boldsymbol{M}}\ddot{\tilde{\boldsymbol{q}}}(t) + (\tilde{\boldsymbol{B}} + \omega_0\tilde{\boldsymbol{G}})\dot{\tilde{\boldsymbol{q}}}(t) + (\tilde{\boldsymbol{K}} - \omega_0^2\tilde{\boldsymbol{K}}_d)\tilde{\boldsymbol{q}}(t) = \omega_0^2\tilde{\boldsymbol{f}} + \tilde{\boldsymbol{f}}(t)\,, \tag{6}$$

where the general excitation vector $\tilde{\boldsymbol{f}}(t)$ was added. The parameter identification in the case of a real application is discussed in Hajžman et al. (2010b). The experimental modal analysis was employed in order to tune the proper eigenfrequencies of the numerical model. The illustration of the chosen tuned eigenmode characterized by dominant torsional deformation and higher eigenmode characterized by dominant deformation of the ring is in Fig. 2.

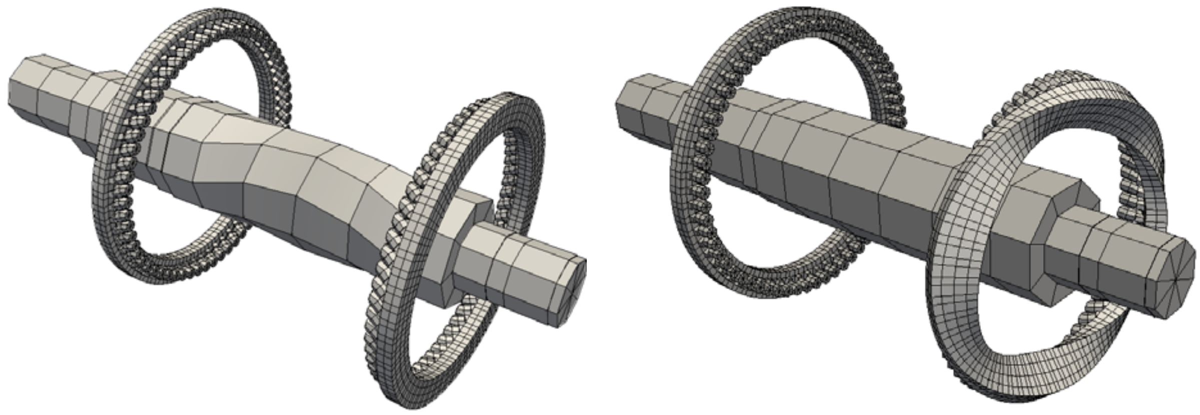

Fig. 2: Illustration of rotor eigenmodes (the laminated armature is not visualized).

3. Loading of a squirrel cage motor

The rotor of the squirrel cage motor can be loaded in various situations by various ways. The main excitation types are summarized in this chapter. It is considered that the motor is used as a traction motor in a rail vehicle. The definition of a motor loading is a principal step of dynamic analysis of the rotor of a squirrel cage induction motor.

The first possible excitation, which is rather static then dynamic one, is the centrifugal force loading represented by vector $\omega_0^2\tilde{\boldsymbol{f}}$. The analysis of a centrifugal response was presented in Šašek (2010). For the accurate and realistic solution of the rotor response to the static centrifugal excitation, it is necessary to solve detailed contact problem of the shrink and end short-circuit rings considering initial tension. This task is not the subject of this paper and should be performed by means of commercial software tools based on the finite element method.

One of the most common and most often loadings is the excitation by pulsating electro-magnetic torques which are excited by the electro-magnetic field of the motor. The detailed calculation of these torques for the real studied induction motor was shown in Bartoš & Janda (2008). The results of the electro-magnetic field analysis are the Fourier series of the pulsating electro-magnetic torques that can be understood as steady polyharmonic excitation. In this case, the rotor dynamic response can be also considered by polyharmonic one and due to the linearity of the model (6) each harmonic part can be solved separately. The solution of this problem together with preliminary analysis of stresses in the rotor bars is presented in Šašek (2010).

Another important type of excitations, which can arise during the standard motor operation, is the excitation in the course of driving through the curved track. Since the curves for the rail vehicles are not too aggressive as in case of road vehicles, the dynamic response is not so significant. However, the effect on the fatigue and service life can be evaluated. The gyroscopic torques for the particular curve was derived in dependency on the vehicle velocity, curve radius, moments of inertia of the end and shrink ring and drive gear ratio. These torques are acting on the end and shrink rings and are causing the dynamic loading because of the rotor rotation.

Similarly small vibration effects are caused by the kinematic excitation in the course of driving on the uneven rails in both horizontal and vertical directions. Byrtus et al. (2009) presents the possible method for such excitation determination.

The last considered excitation type is the electro-magnetic excitation due to the short circuit. Several types of short-circuit faults can be recognized. After electromagnetic field analysis (Bartoš, 2008), the excitation can be defined by time histories of short-circuit torques. The rotor model (6) can be solved by means of numerical integration in order to obtain the time histories of mechanical variables.

4. Conclusions

This paper presents the advanced dynamic model of the rotor of the squirrel cage electromotor. The model is characterized by the flexible end and shrink rings and therefore it is suitable for high-frequency analysis, where bending and torsional behaviour of the rings as well as of the shaft can be dominant.

The created computational model can be used for many types of dynamical analyses. The most important ones are modal analysis, steady-state response analysis and non-stationary time integration of motion equations. The excitation types in case of rail vehicles were summarized in the paper. It can be defined mainly on the basis of electromagnetic field calculation and track irregularities definition. Future work will be aimed at detail calculations of the response to these excitation types and successive output for stress and fatigue investigation.

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