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Discussion

Authors' reply

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The authors appreciate the comments. It is evident that Dr. Larsson's remarks are due to misunderstandings about basic aspects of the model-based technique used to identify faults in rotating machines. These basics are not contained in the paper since they were described in detail in our cited Refs. [1–5].

In the introduction of the paper, to which Dr. Larsson refers, dynamic effects of machine faults are simulated by means of sets of equivalent forces and moments applied to the nodes of the finite element model of the rotor train. This approach can also be used to take into account the dynamic effects induced by geometrical deformations of the shaft like a bow.

The effects of most typical faults of rotating machines can be simulated by means of suitable sets of equivalent excitations, forces or moments, that are applied to nodes of the finite element model of the shaft-train. Beam finite elements are used. Each node of the beam elements has 4 degrees of freedom: two orthogonal radial displacements (lateral vibrations) and rotations (associated with a shaft bending in two orthogonal planes). The equivalent excitations are referred to an absolute reference system and are denoted $\mathbf{F}_n(\Omega)$ in Eq. (1) of the paper. This vector can be also written as

$$\mathbf{F}_n(\Omega) = \mathbf{F}(n\Omega) = \mathbf{F}_0(\Omega)\,\mathrm{e}^{\mathrm{i}n\Omega t}.$$

Depending on the type of fault the equivalent excitations can be expressed by a rotating vector, with a certain harmonic content, or by a fixed vector.

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In general, in the absence of nonlinear effects, the frequency spectrum of the equivalent excitations contains significant harmonic components limited to the rotating frequency ($1\times$) and to integer multiples of this frequency (i.e. $2\times$, $3\times$).

The magnitude $|\mathbf{F}_0(\Omega)|$, as well as the phase, of each term of vector $\mathbf{F}_n(\Omega)$ can be constant or it can vary with the rotating speed. In the case studies that are shown in our paper, the magnitudes of the terms of vector $\mathbf{F}_n(\Omega)$ were constant.

The dynamic effects of a shaft bow can be modelled by means of pairs of opposite bending moments that are applied to suitable nodes of the finite beam model of the shaft-train.

When the longitudinal (axial) plane that contains the shaft bow is fixed with respect to the shaft, the moments are represented by vectors that rotate with the rotating speed (see Fig. 1); thus a shaft bow causes a synchronous excitation ($1\times$).

The magnitude of the equivalent bending moments can change during the machine transients (runups and coastdowns) depending on the evolution of the physical phenomenon that causes the shaft bow (i.e. rub, axial asymmetry of the thermal distribution, etc.).

Anyhow, the magnitude of these moments cannot be null even at very low rotating speeds and the system vibrations can be negligible, significant or considerable, depending on the mechanical properties of the system and the magnitude of the equivalent excitations. On the contrary, when the rotating speed is low, most inertia effects and the magnitude of unbalance force would be negligible and unimportant, because they are proportional to Ω^2. This is in accordance to Dr. Larsson's statements.

In the case study described in the paper, the equivalent excitations contain only bending moments whose magnitude was constant, that is it was not proportional to the rotating speed.

Contrary to Dr. Larsson's Ref. [1], the case study analysed in the authors' paper does not show spiral vibrations occurred at a constant rotating speed (e.g. the machine operating speed). The shaft bow was caused by an axial asymmetry of the thermal distribution of the generator rotor induced, during the on-load operating condition, by a not homogeneous flow of the rotor current through the rotor windings as well as by different friction forces between the copper bars and the slots. Only transient vibrations collected during coast-downs have been analysed and a sub-set of vibration data collected in the range from 120 to 1800 rpm has been considered. These speed

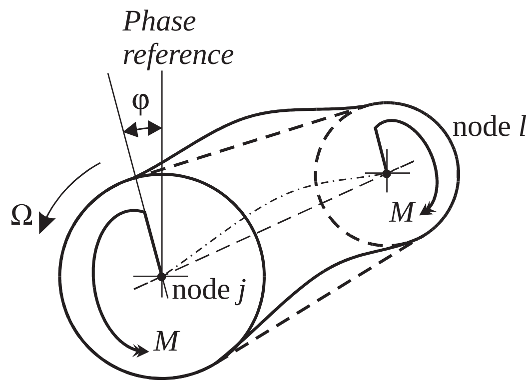

Fig. 1. View of a single or grouped beam elements of the shaft portion affected by the bow. The dynamical effect of the bow is modelled by means of equivalent bending moments.

transients were rather short (see Fig. 4 of our paper); therefore, the magnitude of the equivalent excitations (opposite bending moments) that simulate the dynamic effects of the shaft bow has been assumed to be constant. In the speed range below 500 rpm these moments were not null; in fact they caused vibrations with small but not null amplitude (see Figs. 11–12 of our paper).

References

[1] B. Larsson, Journal asymmetric heating, Part II: alteration of rotor dynamic properties, *Journal of Tribology* 121 (1) (1999) 164–168.