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ACTIVE VIBRATION CONTROL IN ROTATING SHAFTS

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Abstract

In this paper, we examine the spatial behaviour of rotating shafts in the neighbourhood of the resonance frequency. In particular, we study the effect of active magnetic damping on the maximum amplitude of shaft vibrations. We formulate a mathematical model describing the motion of the center of a typical cross section of the shaft with and without damping, which can be used to optimize the damping strategy. We compare the measured result and the result of finite element method and the numeric simulation.

Keywords: shaft vibration, vibration control, eddy current probe.

1. Introduction

The main goal of thermal plants and district heating centers is to provide reliable electric power and heat. The shaft of turbine-generators turned over the resonance, vibration is at maximum when speed passes the resonance range. The load on bearings and the lifetime of the equipment will be reduced and the likelihood of defects increases. A well-established technology to achieve that goal is to use active magnetic bearings on turbine-generators what leads to reduced loss from friction. Unfortunately, there are some notable shortcomings of magnetic bearings, like limited load tolerance, high energy use, need of continuous regulation due to instability [3] just to name a few.

In this paper, we propose a novel approach: we keep the traditional bearings and in addition we apply a device we call magnetic vibration damper (see *Fig.1*) which in activated state reduces unwanted shaft vibrations. As a byproduct we are able to minimize reaction forces on the bearings. Roughly speaking, the piecewise constant periodic control force (see *Fig.2*. $F(t)$) generated by the magnetic vibration damper tends to decrease the shaft deflection due to the 180° phase angle unbalanced shaft mass and the control force. In this paper, we use the filling factor (i.e. the portion where $F(t) = 0$ compared to the length of the period) of the square wave and the amplitude of the control function as parameters, and determine near maximum values of the corresponding vibration reduction.

2. Description of the Model

Fig. 1 presents a schematic drawing of the generator model. The model is driven by a 220 V motor with carbon brushes. The operation range of this equipment is 0–6600 RPM. An annular coil serves to regulate the speed of rotation. The motor is connected to the shaft by a flexible clutch. The generator is housed in a PDN 306 unit, which is supported by two rigidly fixed bearings. The shaft section of the generator is connected to the shaft section of the turbine by a flexible clutch.

The shaft is included in a self adapting ball bearing with an adapter sleeve on the clutch side and a hydrostatic slip bearing in a PDN 206 bearing house. The distance between the two bearings is 550 mm. The diameter of the flexible shaft is ⌀12 [mm].

The sets of blades to be found on a real turbo generator are substituted by a rotating disc in the model because in the current study we are only interested in issues related to shaft vibrations.

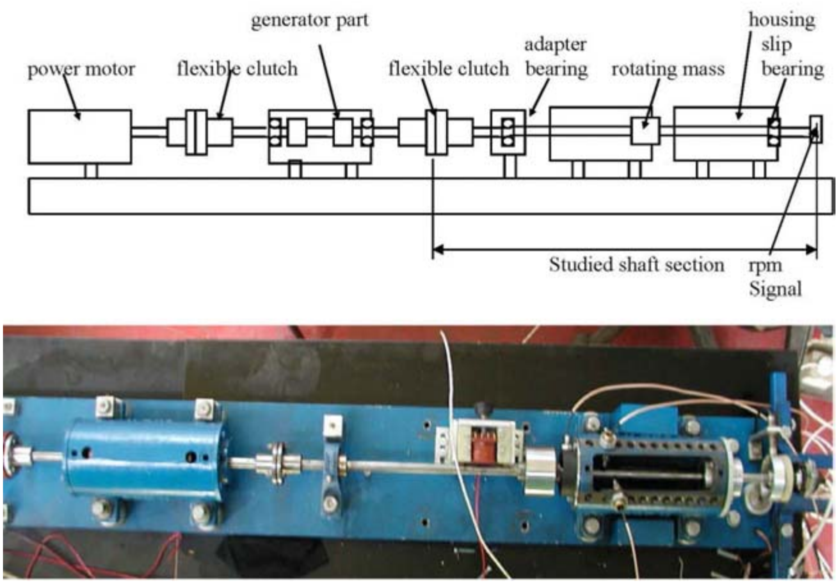

Fig. 1. Structure of Model

In order to allow easy evaluation of measurement readings, 18 pairs of CMSS668 type of eddy current sensors are installed at 17 mm distance from each other along the bearing house. The drill holes for the sensor pairs form 90° angles. An RPM signalling instrument is placed at the end of the shaft to provide information on the angular position of the shaft as compared to the dead point position.

An active magnet to control resonance is installed horizontally on the studied shaft section at 200 mm from the adapter bearing.

3. Evaluation of Measured Results

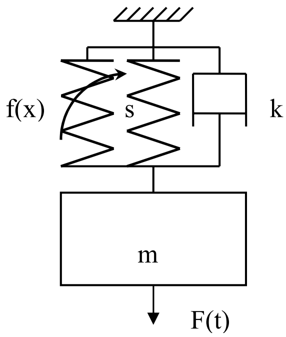

Fig. 2. Mechanical model of the shaft

The 1 DOF (see *Fig. 2*) equation of the controlled system can be written in the following form [1, 2]:

$$m \cdot \ddot{x} + k \cdot \dot{x} + s \cdot x = f(x) + F(t), \tag{1}$$

where m is the mass [kg], x is the displacement [m], $s+$ is the spring rate [N/m] and k is the damping factor [N/(m/s)], and $F(t)$ and $f(x)$ are the exciting and control functions, respectively.

We have created a vibration model to optimize the parameters (control amplitude, filling time of the square wave) of vibration damping. We have used the method of 3 dB band width on the vibration curve to define the relative damping factor ($D = 0.032$, $k = 17.1$ N/(m/s)). The measured spring stiffness value is $s = 59569$ N/m. Mass equals $m = 1.22$ kg. In the diagram of *Fig. 3* we analogize the binary $f(x)$ belonging to different voltages in the algebraic equation, see *Table 1*.

Table 1.

Input voltage	function of the magnetic force
30 [V]	$f(x) = 0.002116 \cdot x^{-1.184}$
40 [V]	$f(x) = 0.009973 \cdot x^{-1.241}$
50 [V]	$f(x) = 0.001702 \cdot x^{-1.130}$
60 [V]	$f(x) = 0.0000553 \cdot x^{-1.491}$

The excitation functions $F(t)$ and $F2(t) = f(t) + F(t)$ and the control force

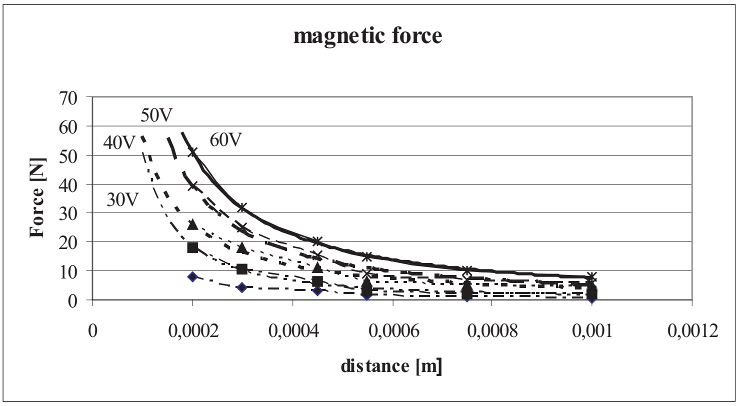

Fig. 3. Variation of the magnetic force

$f(t)$ are presented below:

$$F(t) = A \cdot \cos(\omega \cdot t - \phi) \tag{2}$$

$$f(t) = \left[\frac{(\omega \cdot \tau) \cdot H}{\pi} + \frac{-2 \cdot H}{\pi} \cdot \left[\sum_{j=1}^{300} \frac{\sin[j \cdot (\omega \cdot \tau)] \cdot \cos[j \cdot (\omega \cdot t - \phi)]}{j} \right] \right] + \left[\frac{(\omega \cdot \tau) \cdot H}{\pi} + \frac{2 \cdot H}{\pi} \cdot \left[\sum_{j=1}^{300} \frac{\sin[j \cdot (\omega \cdot \tau)] \cdot \cos[j \cdot (\omega \cdot t - \phi)]}{j} \right] \right].$$

3.1. Results of Simulation Studies Performed with a Mechanical Swing System

Fig. 5 presents the degree of attenuation of vibrations as a function of the change of the filling factor of the square wave and the change in the ratio of the amplitude of excitation and attenuation. *Fig. 6* shows how the mid-point of the shaft moves (Lisajoues curves) [4] with and without using unilateral magnetic vibration damping. The arrow shows the direction of magnetic intervention.

The comparison of the measured result, the arithmetical result of the finite element and the numerical simulation, without using magnetic intervention can be seen in the *Fig. 7*. The difference between the two methods: I use the AMESIM finite element software for optimization of the space factor of the square wave as long as the square wave shape of AMESIM is ideal, till wave shape of numeric simulation is distorting from nonlinear magnet characteristic. The operation speed of the shaft is 25 Hz and 30 Hz. The results show that the base accord harmonic is 0.4 mm in each case. In *Fig. 8*, one can observe absorption of vibration, which is caused by a

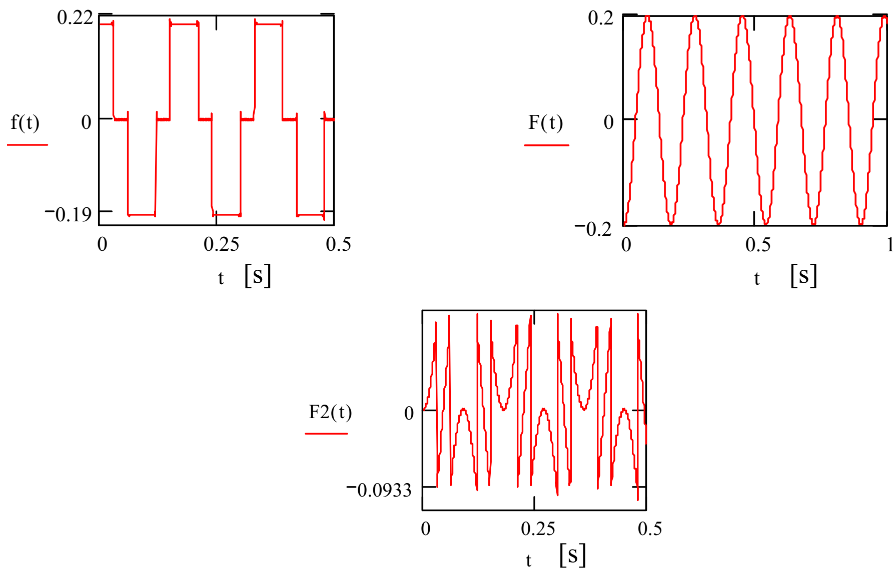

Fig. 4. Excitation functions

one-side magnetic intervention. Both the measuring and the numeric semi-implicit results show the decrease of the base harmonic value and upper harmonic appears a bit.

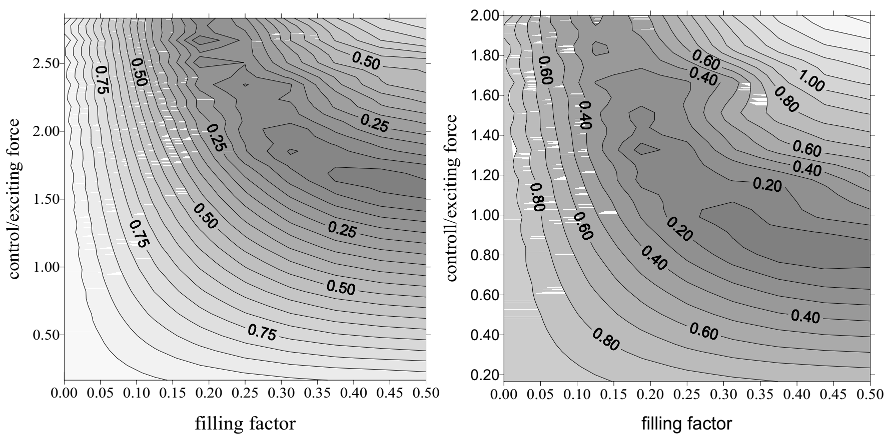

Fig. 5. Optimizing swing reduction parameters with one (right side) and two (left side) magnets

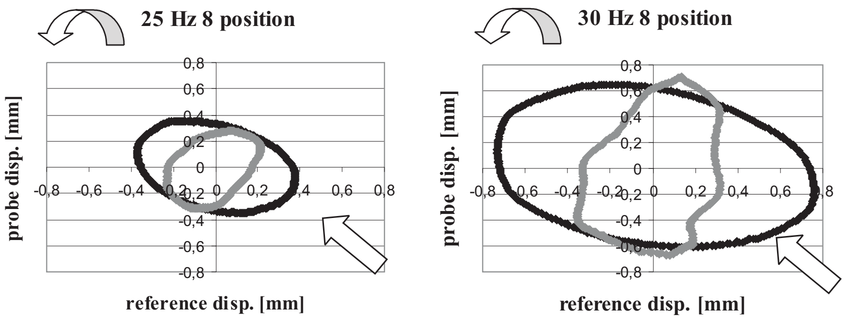

Fig. 6. The spatial position of the shaft at rotations of 25 and 30 Hz. With (gray line) and without damping (black line)

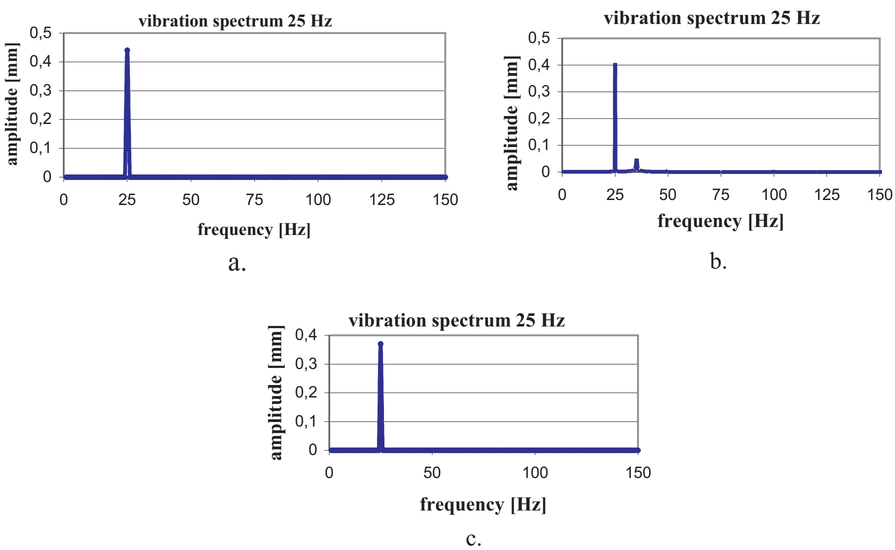

Fig. 7. Comparison of the experimental result without using magnetic intervention. a. measured result, b. finite element method, c. numeric simulation.

4. Summary

Proper use of a single magnetic damping device may reduce amplitude of unwanted vibrations substantially. If we use one magnet the vibration is minimal when the

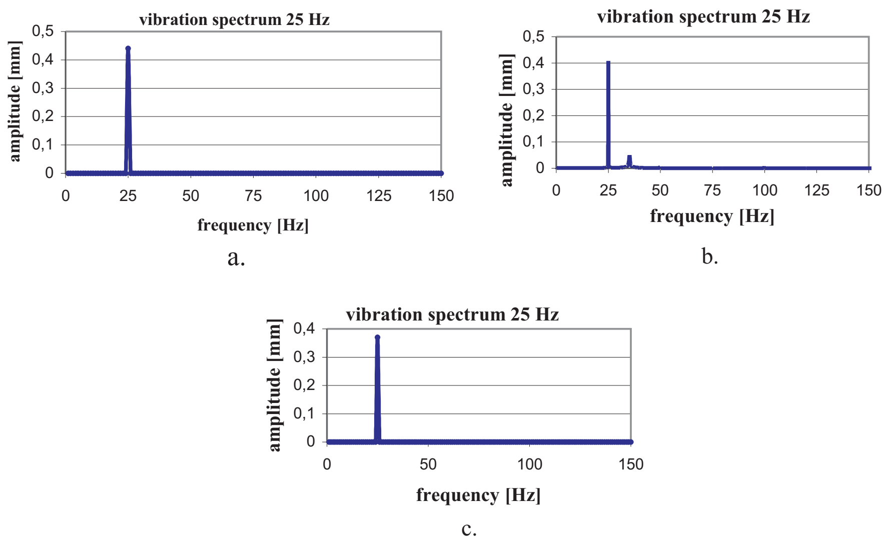

Fig. 8. Comparison of the experimental result with one-side magnetic intervention. a. measured result, b. finite element method, c. numeric simulation.

filling factor of the square wave is 0.5 and the ratio of the amplitudes of the control/exciting forces is 1.5. Measured values indicate a rate of reduction at 50% in the vicinity of resonance. The amplitude of vibration may be reduced even more when more than one magnets are applied. If we use two magnets, the vibration is minimal when the filling factor of the square wave is 0.5 and the ratios of the amplitudes of the control/exciting forces are 0.8. In turn, the load on the bearings gets reduced, what ensures smooth operation and longer life span for the equipment.

Acknowledgement

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Nomenclature

m: mass [kg]
k: damping factor [Ns/m]
D: damping ratio [–]

s: spring stiffness [N/m]
ϕ: phase angle [rad]
A: exciting amplitude [N]
H: control amplitude [N]
ω: frequency [Hz]
τ: filling time of the square wave [s]
t: time [s]

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