EXPERIMENTAL IDENTIFICATION OF DYNAMIC PARAMETERS FOR ACTIVE MAGNETIC BEARINGS

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In the paper, an experimental identification procedure of dynamic parameters for active magnetic bearings is presented. In the proposed method, analysis of variable components of displacements of the rotor for an excitation with the rotating vector force of unbalance, is conducted. Proportionality of the bearing magnetic response force to the excitation force in the whole range of rotational frequency, is assumed. It allows for identification of equivalent dynamic coefficients of the bearing. The experimental investigations were carried out on test rigs with the shaft supported in active magnetic bearings.

Key words: magnetic bearing, identification, dynamic properties

1. Introduction

In rotor dynamics, characteristics of bearings exert decisive influence on modes of vibrations and critical frequencies of the rotating shaft. In a system of active support of the rotor in the magnetic field, there is a strict connection between dynamic parameters of the model and the shape of its frequency characteristics obtained for the fixed rotor. This feature is used in situations when it is necessary to test different control algorithms in order to optimise dynamic characteristics of the designed machine without the risk connected with its start-up and shut-down. Thus, it is very important from the viewpoint of the measurement procedure to elaborate a method which allows for interpretation of the obtained characteristics and whose aim is to find dynamic parameters of the system (stiffness, damping) determining stable machine operation in

the assumed range of rotational frequency. This is a complex problem that demands proper measurement and calculation techniques to be developed, and also access to defined levels of software of the bearing control system is needed.

In the present paper, a concept of the method and the investigation results obtained by the authors through the experimental procedure aimed at the determination of dynamic coefficients of the bearing are presented. Its important feature consists in the possibility of realization under nominal operating conditions of the bearing, and, therefore, reliable identification of actual values of these coefficients is possible.

2.   Concept of the method

In the measurement method, a dependence of the vector of the resultant magnetic force acting on the machine shaft as a function of the journal position in the magnetic radial bearing and the currents that flow through the windings of actuators (electromagnets) is employed. The magnetic bearing response vector is a sum of forces generated by bearing electromagnets and alters in each control cycle (Kozanecki and Kozanecka 2003).

The magnetic response components $F_{X\,mag}$, $F_{Y\,mag}$ for each control axis are related to the measured mean values of the current controlling the electromagnets I_{XT}, I_{XB}, I_{YT}, I_{YB} in a given control period, values of the magnetic gaps s_{XT}, s_{XB}, s_{YT}, s_{YB} and to the values of the electromagnet constants K_{XT}, K_{XB}, K_{YT}, K_{YB} (top – index T, bottom – index B). Values of the magnetic gaps are found on the basis of measurements of instantaneous positions of the journal with respect to the centre of the bush of the known clearance. The magnetic response components $F_{X\,mag}$, $F_{Y\,mag}$ for the axis X, Y are determined by the following relationships

$$F_{X\,mag} = K_{XT}\frac{I_{XT}^2}{s_{XT}^2} - K_{XB}\frac{I_{XB}^2}{s_{XB}^2} \qquad\qquad F_{Y\,mag} = K_{YT}\frac{I_{YT}^2}{s_{YT}^2} - K_{YB}\frac{I_{YB}^2}{s_{YB}^2} \quad (2.1)$$

Equation (2.1) holds on the assumption that the linear dependence of the magnetic flux on the induction is maintained. It means that the bearing operates according to that part of the characteristics which is distant enough from the state of magnetic circuit saturation, when the induction does not exceed 50% of the saturation induction for the core material. The value of the constant K depends on electromagnet design parameters and can be calculated

theoretically (Schweitzer *et al.*, 1993). However, in the actual design of the journal bearing, the constants K_{XT}, K_{XB}, K_{YT}, K_{YB} can slightly differ for each pair of electromagnets of the bush. In order to increase the accuracy of the proposed measurement method, the constant values are verified experimentally for each electromagnet, and their real values are taken into account in the calculations (Kozanecki and Kozanecka, 2003). If the journal motion parameters are known and the magnetic response force is determined by an indirect method (Kozanecka *et al.*, 2003a), it is possible to find the bearing dynamic parameters that relate the magnetic response force to the journal motion parameters.

The idea of the method is based on the relationship between stiffness and damping coefficients and the linear response force for the bearing in which interactions between the axes do not occur. In the journal active magnetic bearing system, the interactions between the control axes X and Y can be neglected for small displacements of the journal (Kozanecka, 2001; Kozanecki and Kozanecka, 2003). In the case of the bearing under investigation, the control system performs two independently imposed control algorithms for both the axes X, Y, which also allows one to neglect the interactions between them. For one control axis, the relation between the stiffness coefficients K_{XX} and the damping coefficients C_{XX} and the linear magnetic response force of the bearing $F_{X\,lin}$ has the following form

$$F_{X\,lin} = K_{XX}x + C_{XX}V_X \tag{2.2}$$

Changes in the nonlinear magnetic response force $F_{X\,mag}$ measured for the given control axis and being the response of the bearing system to the assumed synchronous excitation F_Z are approximated with the linearised harmonic function $F_{X\,lin}$. A difference between the nonlinear magnetic response force of the bearing along the given axis $F_{X\,mag}$ that is known from the model calculations and its linearised form $F_{X\,lin}$ determined on the basis of formula

$$F_{X\,lin} - F_{X\,mag} = \Delta F_i \tag{2.3}$$

is sought with the least square method in such a way that $\sum \Delta F_i^2 = \min$.

Thus, the equivalent values of the bearing dynamic parameters K_{XX}, C_{XX} for the axis X are obtained. An analogous situation refers to the coefficients K_{YY}, C_{YY} for the axis Y. The calculations are conducted for stable bearing operation, where the journal position oscillates around the assumed point of equilibrium (Fig. 1).

Figure 2 presents a comparison between the measured magnetic response force F_{Xmag} and a theoretical function, which is a sum of the forces of stiffness
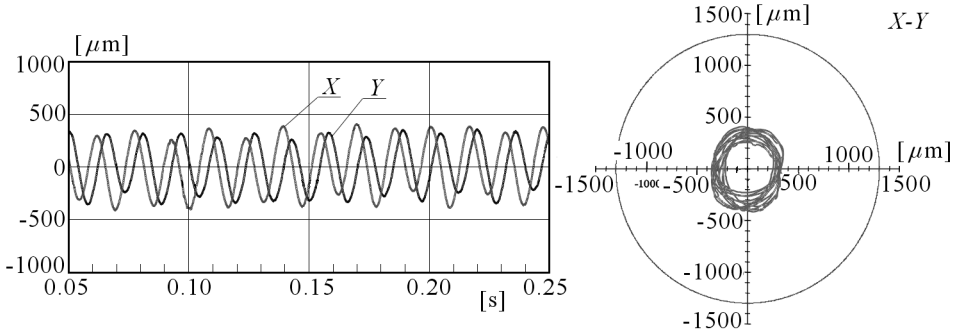
Fig. 1. Displacement versus time and the orbit

and damping $F_{X\,lin} = K_{XX}x + C_{XX}V_X$. The curve $F_{X\,lin}$ has been plotted on the basis of the measured journal displacement X (Fig. 2) and the journal velocity V_X obtained through digital differentiation of the displacement and a selection of suitable values of the dynamic stiffness coefficients K_{XX} and the damping coefficients C_{XX} in such a way as to make the sum of squares of differences minimal for the selected part of the time history.
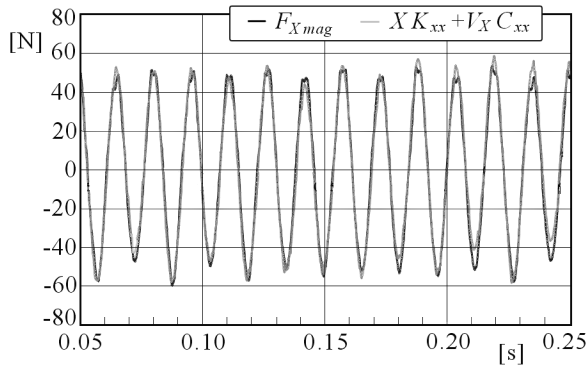

Fig. 2. Measured magnetic response component along the control axis $X - F_{Xmag}$ and its modelled time history $F_{X\,lin}$ with the identified dynamic coefficients K_{XX}, C_{XX}

In the method of identification of the bearing dynamic coefficients, it is required that the theoretically calculated magnetic response force is the closest approximation of its function obtained in the measurements and that the share of synchronous components in the curves of displacement, current and magnetic response force is dominant (Kozanecka, 2005).

3.   Test rig

The test rig of the long flexible shaft line has a module structure and allows one to investigate systems of a different number of supports and of various shaft lengths within the assumed range of frequency of rotations. It is driven by an electric motor connected to the shaft through an elastic membrane coupling with smooth rotation control and fed by a frequency converter.

In Fig. 3, the magnetic bearing is one of two supports of the shaft part whose length is approx.1000 mm, the thin-wall pipe diameter is $\phi = 54$ mm and the wall thickness is equal to 2 mm. There is a disk at the free end of the shaft, on which the masses of test unbalancing can be mounted.
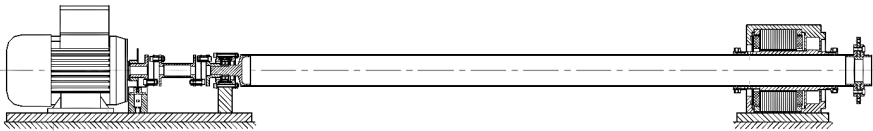

Fig. 3. Test rig

In the second configuration, the test rig consists of the horizontal flexible power-transmission shaft supported on two rolling bearings mounted at both ends. An active magnetic bearing operates as an auxiliary bearing that modifies the dynamic properties of the shaft line. Between the magnetic bearing and the shaft right end, there is a rigid disk which allows one to mount the balancing weights for the real structure. The mass of the rotating system is equal to 4.85 kg, the shaft line length equals 1923 mm. The test rig allows one to investigate the effects of the magnetic bearing on dynamic properties (vibration level, displacement and the coefficient of vibration amplification of subsequent critical frequencies) and to control vibrations of the long flexible rotor (Kozanecka, 2005).

4.   Results of the investigations

The model test rig of the flexible power-transmission shaft was used to carry out an experiment whose aim was to verify the identification procedure of dynamic coefficients of the active magnetic bearing. A kinematical exciter was fixed on the disk, on which the masses of test unbalancing can be mounted. After introducing a selected program for magnetic bearing control, harmonic vibrations of the shaft of frequencies $(10, 20, 30, \ldots, 80)$ Hz and the assigned

amplitude were excited. For each frequency under analysis, the time histories
of displacements and currents in the magnetic bearing, which were subject to
respective calculation procedures, were recorded, and then the bearing dyna-
mic parameters were estimated. To conduct the measurement and calculation
procedures, a measurement system with DBK 15 input systems made by IO-
tech, operating with a PC and employing the Daq/112B type PCMCIA me-
asurement card of the resolution equal to 12 bites and the maximum sampling
frequency of 100 kHz, was applied.

The voltage time histories corresponding to displacements (positions) of
the journal along both the control axes X, Y were recorded on-line on re-
spective inputs of the measurement-control module. These were two voltage
signals 0-24 V from Bently-Nevada type 3300 eddy-current transducers of rela-
tive vibrations. The voltage time histories corresponding to currents flowing in
electromagnets were measured and recorded. These were four voltage signals
0-5 V from current-voltage LEM type transducers (Kozanecka, 2005).

The DaqView v.7.9.8 software was used for recording purposes. There were
4000 measurements made, at the sampling frequency of 8 kHz/channel. The
results were stored in binary files of a data acquisition system, and then conver-
ted into text files. The programs for analysis of dynamics and identification
procedures of the bearing dynamic parameters, according to the methodology
proposed, were developed with the MS Excel spreadsheet.

Exemplary time histories of the quantities measured are shown in Fig. 4 and
Fig. 5, and of those calculated in Fig. 6 and Fig. 7 for the magnetic bearing of
the selected configuration of the control program for the kinematic excitation
of the frequency 40 Hz and the assigned amplitude, whose value was such as
to obtain the dominant share of synchronous components in the time histories
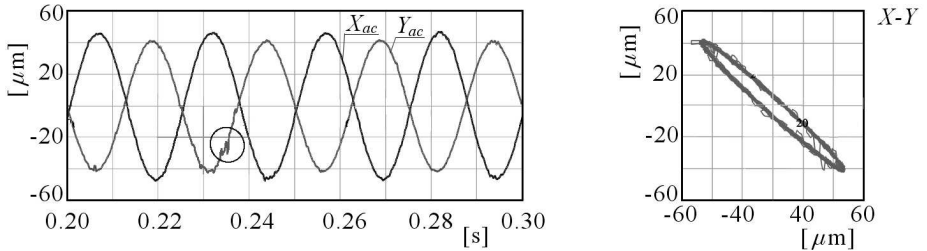under analysis and to obtain the linear range of magnetic response forces.

Fig. 4. Displacements for both the control axes X, Y and the shaft motion
trajectory

The occasional disturbances which occur in the recorded time histories of
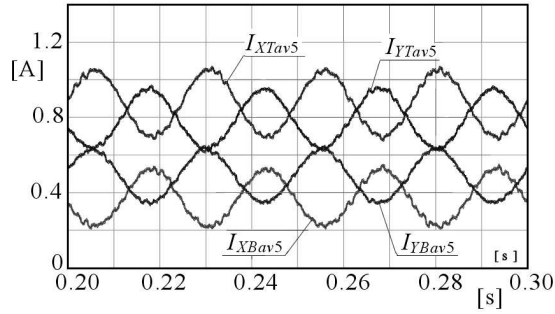displacements (Fig. 4) are amplified by digital differentiation, and the effect

Fig. 5. Time histories of the currents in electromagnet windings I_{XT}, I_{XB}, I_{YT}, I_{YB} – averaged
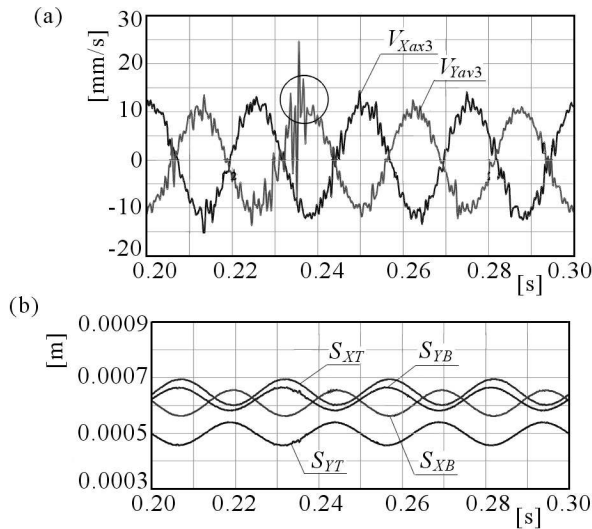
Fig. 6. (a) Velocity components for both the control axes V_X, V_Y – averaged; (b) variable gaps for individual electromagnets s_{XT}, s_{XB}, s_{YT}, s_{YB}

of these disturbances is very distinct in the time history of the velocity component V_Y (Fig. 6a). This does not affect the calculation accuracy, where the characteristics are approximated with the harmonic time history.

The measurement and calculation cycles were conducted for various excitation frequencies in the range (10-80) Hz, which allowed one to build functions representing values of the bearing dynamic coefficients, namely: the stiffness coefficients K_{XX}, K_{YY} and the damping coefficients C_{XX}, C_{YY}, as functions of the frequency at the given excitation frequency and the given configuration of the control program (Fig. 8).
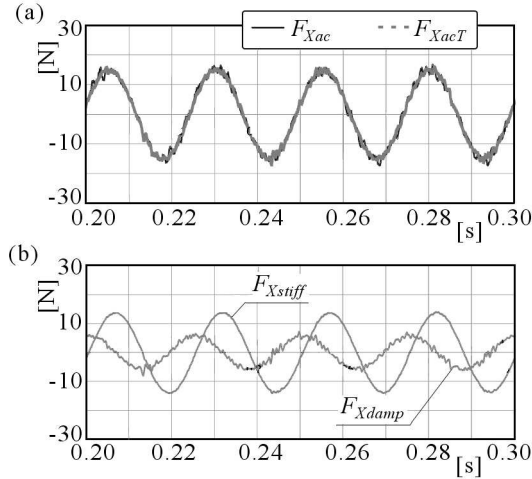
Fig. 7. (a) Measured magnetic response component F_{Xac} and the force
$F_{XacT} = F_{Xlin}$ modelled with the identified dynamic coefficients K_{XX}, C_{XX};
(b) components of the magnetic response related to the stiffness F_{Xstiff} and to the
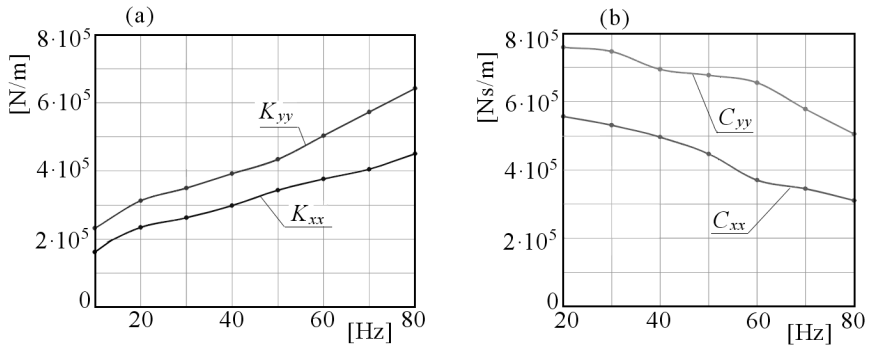damping F_{Xdamp}

Fig. 8. (a) Stiffness K_{XX}, K_{YY} versus frequency; (b) damping C_{XX}, C_{YY} versus
frequency

It means that the start-up and shut-down characteristics of the model shaft
line under such magnetic bearing operating conditions should be overcritical
characteristics. The experimentally determined dynamic coefficients of the bearing show some anisotropy of the properties for individual axes $K_{XX} \cong K_{YY}$,
$C_{XX} \cong C_{YY}$. The analysis of this state showed that for isotropic properties of
the energy transmission systems (symmetrical saturation-control current characteristics), this scattering resulted from the scattering of constant values of
electromagnets for the given axis. This conclusion was confirmed by calcula-

tions conducted with the simulation model of the magnetic bearing, however its further verification is still needed (Kozanecka, 2005).

5.   Conclusions

The experiment was conducted for various configurations of the program controlling the auxiliary magnetic support used in the model shaft line system. The determined dynamic coefficients were employed in subsequent stages of the investigations in the modelling and numerical calculations of the start-up and shut-down characteristics of the flexible power-transmission shaft. The generated numerical characteristics were verified experimentally through comparison to the start-up and shut-down curves recorded for the real model shaft line with supports of identified dynamic properties.

The proposed methodology of measurement of the response and dynamic coefficients of the magnetic bearing is a very important tool in designing dynamics and vibration control of machine rotors in which active magnetic bearings are applied. It allows one to find analogies to classical bearing systems and to employ professional calculation codes for evaluation of the effects of modification in the dynamic properties of shaft lines introduced through changes in the configuration of the program controlling its active magnetic supports.

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Eksperymentalna identyfikacja parametrów dynamicznych aktywnego łożyska magnetycznego

Streszczenie

W artykule przedstawiono procedurę eksperymentalnej identyfikacji parametrów dynamicznych aktywnego łożyska magnetycznego. Zaproponowana metoda wykorzystuje analizę zmian przemieszczenia wirnika dla wymuszenia wirującym wektorem niewyważenia. Założono proporcjonalność siły reakcji magnetycznej na wymuszenie w zadanym zakresie częstości. Pozwoliło to na identyfikację dynamicznych współczynników łożyska. Badania eksperymentalne przeprowadzono na stanowisku badawczym układu wirującego podpartego w aktywnym łożysku magnetycznym.