SIMULATION AS A MEANS TO TEST ADVANCED CONTROL ALGORITHMS IN SUPPRESSING ROTOR VIBRATIONS IN ELECTRIC MACHINES

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Abstract

Active control of rotor vibrations in electrical machines is considered. The objective of the research described in the paper is to diminish unwanted forces generated by rotation and unbalanced mass of the rotor in an electrical machine. These forces, dependent of rotational speed, cause vibration that, when occurring in the natural frequency (or critical speed) of the rotor, cause severe problems. A new actuator (an extra winding) was designed to be installed in the stator of the machine, and it was controlled with frequency converter to create an opposite force to the vibration. The main task was to develop a controller for the system. The system was first modeled by first principles electromechanical equations, and based on FEM simulations more simplified state-space models were identified. The new models were analysed, and the model giving the best fit with regard to FEM simulations was chosen for controller design. Different controllers were then designed utilizing the identified model, and their performance was tested by using an accurate FEM model of the system. The results showed that the developed controller can be used for vibration control in electric machines operating near the critical speed. The next step in the research is to test the vibration suppressing controller in a real test machine.

Keywords: Electrical machines, Rotor dynamics, Active vibration control, Model-based control, LQG control, Convergent control.

Presenting Author's Biography

Kai Zenger was born in Helsinki, Finland, in 1958. He received the M.Sc., Lic.Sc. and D.Sc. degrees in electrical engineering, computer technology and automation and systems technology in 1986, 1992 and 2003, respectively. After an industrial career he has had different positions at Helsinki University of Technology, where he currently works as a Senior Researcher in the Department of Automation and Systems Technology. His main research areas are Control Engineering and System Theory generally, with applications in chemical process engineering, power electronics and mechanical engineering. He has specialized in the research of time-varying linear systems, periodic systems and adaptive and robust control methods. He has authored or co-authored more than 70 scientific publications.

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1 Introduction

Suppression of vibrations in different engineering systems and structures is an important topic of research both in industry and in academic research. The number of applications is growing vastly, since the precision and accuracy demands of different engineering solutions are getting more stringent [1]. As the allowed tolerances become smaller, the vibrations have a bigger impact on the total system. For example, air-gaps in electric drives need to be smaller to increase the overall efficiency of the machine, yet the vibrations hinder this goal as the air-gap varies along to the vibrations [2].

Because of vibration, rotating machines are usually designed to operate either at subcritical or supercritical range. Operation near the critical speed, where the vibration is heavily excited, is not possible. Passive damping has been widely used for many different applications, like in paper machines, but it can only be used to dampen vibrations at a certain frequency [3]. In high-performance applications active control methods are then needed, in order to be able to drive the system in high speed. In active control an external force is injected to the system according to a control law with the goal to minimize the vibrations. Usually, new mechanical constructions are needed to implement the active control methodology. For example, in small-power electrical machines a new force actuator is implemented by a supplementary winding installed in the stator slots [2].

The active control algorithm can be designed with traditional control methods to a certain extent, but the harmonics of the main vibration frequencies and the dynamic nature of the load disturbance may need more sophisticated control methods [4], [5]. It is then useful to study different modern control methods generated by using first-principle models of both the system and the new actuator. Classical LQ(G) control (Linear Quadratic Gaussian, see e.g. [6, 7]), robust and adaptive control methods ([8]), QFT-based control (Quantitative Feedback Theory, [9]), convergent control and repetitive control methods ([5, 4, 10]) have been tried for example.

Simulation is an effective tool in designing and verifying control designs for processes, which are not accessible for practical testing. This may be, because the real process does not allow practical testing, or the process does not even exist at the time that a controller for it has to be designed. In model-based control design suitable models are first constructed either based on physical modelling (first-principle equations) or input-output data generated e.g. from FE model generated to the process. Both approaches contain an identification step: the parameters of the first-principles model have to be estimated, or a structural model with parameter values has to be identified from input-output data. In any case a suitable low order model is then calculated for the purpose of controller design. The performance of the controller is then analysed and tested first by using the simple model, but later with the full-order model. The final test is finally carried out by using the controller in the real process.

In this paper active control of rotor vibrations in electrical motors is considered. A low-order linear time-invariant (LTI) state-space model of the force actuator is obtained through identification using finite element (FE) simulations of a test machine, which has been designed but has not been built yet. Controller design is carried out by using the identified model. The performance of the controllers is verified by simulations, first by using the identified LTI model of the actuator and machine, then by using the more accurate FEM models.

2 Physical model of the electrical machine and the actuator

The idea in the vibration control to be introduced is to generate a control force to the rotor through extra windings in the stator. This actuator generates a magnetic field that induces a force negating the disturbance force exited by the mass imbalance of the rotor. A similar approach has been used earlier by Chiba et al. [11], who constructed a two-pole winding to a four-pole induction motor.

A schematic layout of the system is presented in Fig. 1 and the related test machine data in Tab. 1.

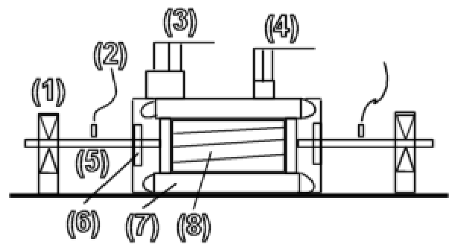

Fig. 1 The test machine (1=bearings, 2=movement sensor, 3=voltage control of stator winding, 4=voltage control of the new actuator winding, 5=rotor axis, 6=safety bearings, 7=stator, 8=rotor)

A 30 kW two-pole induction motor is studied, and a dynamic model starting from the electromechanical first principles equations has been built. An extra actuator for suppressing the rotor vibration (first mode at 49.5 Hz) has been constructed and modelled similarly. The modeling process is initialized by the first principles of electromechanical systems, see e.g. [2]. A separate electric model is created for the actuator generating control force with a magnetic field. An electromechanical model is then formed for the rotor that transfers the input forces into displacements.

In addition to the physical model of the system a FEM-

Tab. 1 Test machine data

Parameter	Value
supply frequency [Hz]	50
rated voltage (rms) [V]	400
connection	delta
rated current [A]	50
rated power [kW]	30
number of phases	3
number of parallel paths	1
number of poles	2
rated slip [%]	1.0
rotor mass (rotor core and shaft) [kg]	55.80
rotor shaft length [mm]	1560
bearing vertical stiffness [MN/m]	500
bearing horizontal stiffness [MN/m]	100
radial air-gap length [mm]	1.0
critical speed [Hz]	42.0
nominal speed [Hz]	49.5

model is created to help in the validation of the results, and to generate black box data [13]. The next phase is to manipulate the physical models in such way that they become standard transfer-function matrices. Alternatively, an augmented state-space representation that includes the actuator, disturbance and rotor models in a single composed model can be used. The state-space representation is constructed by selecting the state-variables from the physical model that yields an exact model of the system with some couplings between inputs. The other approach is to identify the system from the data achieved from FEM-simulations, which describe the system behavior very accurately. By identification the model can be simplified such that the couplings between inputs disappear, but the model still describes the system well enough. When the reduced model is ready, it is validated by simulations against the FEM-model. The FEM-model is very slow to simulate, so a Simulink-model with approximately the same behavior is needed.

The new actuator is driven by a separate frequency converter. The rotor unbalance force can be modelled by a two-dimensional (x and y directions) input entering in the rotor model input. The rotor model is a generalized form of the basic Jeffcott-rotor model. The actuator model can be expressed in the form (1) and the rotor model of the machine in the form (2) , which together can be changed into a general LTI state-space form (3), for details see [12].

$$\begin{aligned} \tfrac{di}{dt} &= A_{em}i + B_{em}v + S_{em}\dot{u}_{rc} + Q_{em}u_{rc} \\ f_c &= C_{em}i + P_{em}u_{rc} \end{aligned} \tag{1}$$

In Eq.1 the vector i contains currents in the rotor and stator, v the control voltages (two-dimensional), u_{rc} the rotor center position and f_c the control force generated to the rotor. In the mechanical model, Eq.2 the term η is the modal coordinate vector, f_{ex} is the disturbance force, and Φ_{rc}, Σ and Ω are matrices related to the generalized eigenfrequencies and damping coefficients of the rotor.

$$\begin{aligned} &\ddot{\eta} + 2\Sigma\Omega\dot{\eta} + \Omega^2\eta = \Phi_{rc}^T f_c + \Phi_{rc}^T f_{ex} \\ &u_{rc} = \Phi_{rc}\eta \end{aligned} \tag{2}$$

$$\begin{aligned} \tfrac{d}{dt}\begin{bmatrix} \xi \\ \eta \\ i \end{bmatrix} &= \begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix}\begin{bmatrix} \xi \\ \eta \\ i \end{bmatrix} \\ &+ \begin{bmatrix} 0 \\ 0 \\ B_{em} \end{bmatrix} v + \begin{bmatrix} \Phi_{rc}^T \\ 0 \\ 0 \end{bmatrix} f_{ex} \\ u_{rc} &= \begin{bmatrix} 0 & \Phi_{rc} & 0 \end{bmatrix}\begin{bmatrix} \xi \\ \eta \\ i \end{bmatrix} \end{aligned} \tag{3}$$

where

$$\begin{aligned} &a_{11} = -2\Omega\Sigma, a_{12} = \Phi_{rc}^T P_{em}\Phi_{rc} - \Omega^2, \\ &a_{13} = \Phi_{rc}^T C_{em}, a_{21} = I, a_{22} = 0, a_{23} = 0, \\ &a_{31} = S_{em}\Phi_{rc}, a_{32} = Q_{em}\Phi_{rc}, a_{33} = A_{em} \end{aligned}$$

A combined model including the dynamics of the actuator and rotor can now easily be formed. It is shown in Eq. 3.

3 Identification

The identification data was created by feeding zero mean pseudo random signals limited to 1V in amplitude into the control winding of the FEM-model. Pseudo random signal was chosen because of its properties. The signal can be limited to desired range; while the randomness guarantees that the whole frequency range is being used making the signal rich enough to identification purposes. The grid used by the FE simulation program [13] is shown in Fig. 2 and the input data (two directions) in Fig. 3. It turned out that the actuator is

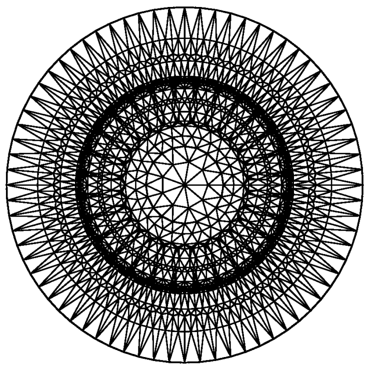

Fig. 2 A grid used for FE modelling of the new actuator

also highly dependent on the displacement o f the rotor. Therefore more inputs had to be added to the model. The signals for the feedback were generated by using the existing rotor model in such way that the forces generated by the actuator were fed into the rotor model generating the displacement, which is then fed back to the actuator. These inputs had to be generated by the model, because the noise signals as input were very ill-behaving, which is most likely due to the stiffness of

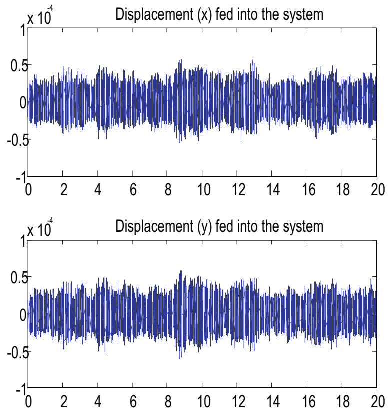

Fig. 3 Displacement input used in identification. Time in seconds.

the system (fast and slow dynamics occur in the system simultaneously). However the displacement given by the rotor model wasn't rich enough as it contained only certain frequencies, and it thus generated a linear relationship between the input and output of the model to be identified. The solution for this kind of problem is to modulate the rotor displacement by another pseudo random signal limited to 10% of the output of the rotor model. This leads to situation where the displacement has low frequency behavior suitable for the actuator model, yet the noise modulation guarantees that the inputs and outputs of the system are no longer linearly dependent and the system can be identified. The control force and its autospectrum have been presented in Figs. 4, 5 and 6. For validation a general model of the

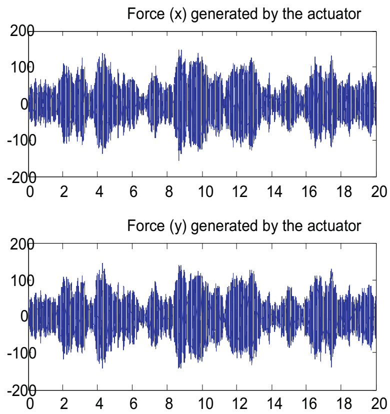

Fig. 4 Force output of the model

process has to be made. The complete process includes the actuator, rotor and the disturbance. For analyzing purposes a model that includes all of the sub-models will be made. For model comparison the disturbance model is left out and only the actuator dynamics with feedback from the rotor output is considered. In the latter model the inputs are voltage and disturbance in x-

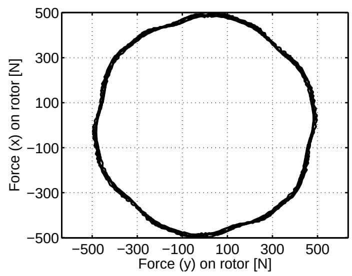

Fig. 5 Control force in x and y directions generated by the actuator

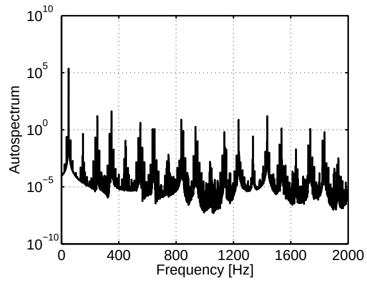

Fig. 6 Autospectrum of the control force

and y-directions, and the output is the displacement.

In order to find out whether the identified model is accurate enough, validation must be made. Because there is another model for the rotor-actuator system available (generated by subspace identification), cross-validation will be used to find out the true performance of the model. The models were compared to those obtained by FCSMEK, which is a program developed to do accurate FEM -based calculations on the behaviour of the electric machine.

In the cross validation all the input-output data available will be used. One of the data sets was used to identify this model and the other was used to identify the other model. Now it is easy to determine, which of the two models is more accurate to the given system. The validation will be made both in time-domain and frequency domain. In frequency domain the true frequency response of the system is available and the comparison will be made against it.

It turned out that the PEM identification (prediction error method) gave better results than subspace identification. In Figs. 7 and 8 frequency responses have been presented, in which the identified actuator models are compared to those calculated by using FCSMEK simulated data. The results are good, which can also be seen in the time series fit shown in Fig. 9.

4 Control

A layout of the rotor and the actuator is presented in Fig. 10. Three different modern control methods will be tested to dampen the vibrations. An LQ controller with integration added is first tried in order to dampen

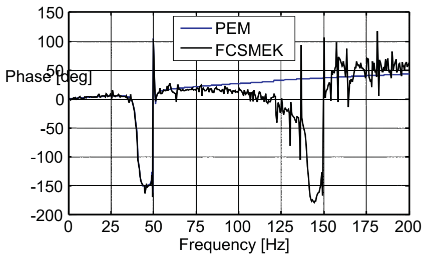

Fig. 7 Phase plot of the PEM model

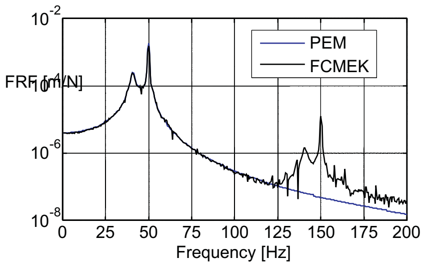

Fig. 8 Gain plot of the PEM model

the sinusoidal disturbance caused by the unbalanced rotor. Secondly, an LQ controller operating together with the so-called convergent control algorithm is tested [5]. Finally, a similar idea with a modified convergent control algorithm is designed. The LQ controller is based in classical state feedback, which thus needs a state observer [6]. In Fig. 11 a schematic layout has been presented, where the LQ controller (containing the state estimator and an integrator to hinder steady-state errors) is first used to dampen the resonant peak of the vibration and also to shift the peak in frequency domain. Another controller then attenuates the vibration to an acceptable level in all frequencies. In the design of the outer controller the combination of the process and the LQ controller is considered. The idea behind the modified convergent control (see Fig. 12) is that

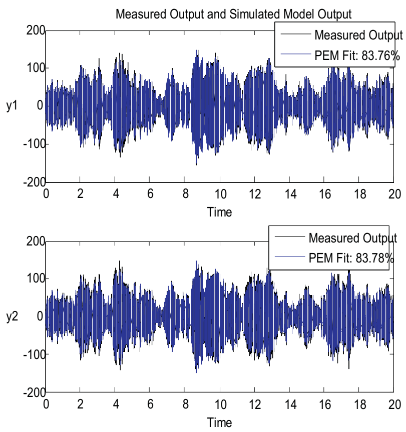

Fig. 9 PEM actuator model fit

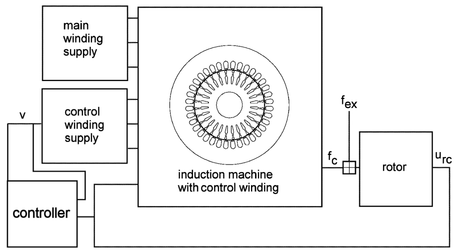

Fig. 10 Control configuration in the test machine

knowing the rotational speed of the rotor and the disturbance amplitude and phase, it is possible to calculate a control signal that gives a force with same frequency and amplitude as the disturbance but with 180 degree phase difference, and consequently completely compensate the disturbance. Assuming that the system is linear, frequency does not change, so only the amplitude and phase need to be controlled. Amplitude and phase can conveniently be presented as one complex variable. Performing all calculations in complex plane removes some numeric problems that the phase, being an angle, can cause. Also the identified system models can easily be converted to complex form simultaneously getting rid of all the derivatives (continuous-time models are used). All control designs achieved for the

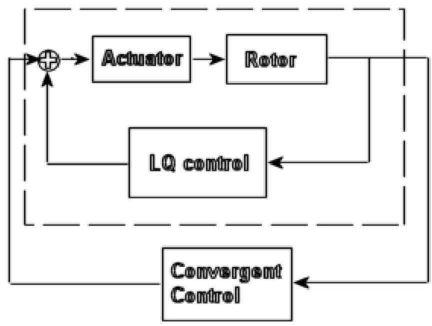

Fig. 11 Control schema of LQ control combined with convergent control

system are tested and validated by using the FEM process model. A control loop with controller tested in Simulink is included in the FEM-model and simulated. In order for the control models to be reliable they must have similar behavior in FEM-environment as they did in Simulink. In Fig. 13 simulation result (using FCSMEK as the process) has been presented, when the LQ controller augmented with integrative action has been used. It should be compared to the result in Fig. 14, in which the modified convergent control algorithm was used. The LQ controller seems to perform a little better, but the result needs further investigation. Also the fact that the convergent control alone did not work (not

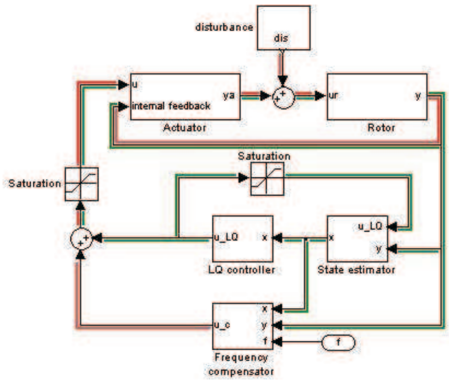

Fig. 12 Control structure. The frequency compensator is parallel to the LQ control. The compensator tries to compensate the disturbance force generated by the rotation of the rotor. LQ controller and the state estimator are unaware of the fact that the force is been compensated.

shown in figures) is a negative but premature result.

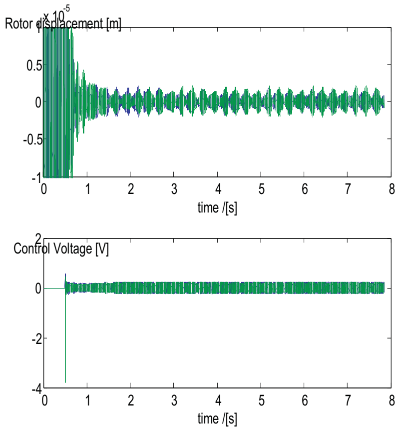

Fig. 13 Process controlled with an LQ controller

5 Conclusion

The possibility of driving an electrical machine in all frequencies, including the critical frequencies, were investigated in the paper by using analysis and simulation tools. The rotor and the new actuator were first modelled starting from first principle physical models, and the model parameters were estimated by using data generated by accurate FE models. A low-order state-space model was then constructed and validated in order to use it for controller design. An LQ controller and a modified convergent controller were then designed and tested by simulations, again using the accurate FE model as the process. The results show that the controllers can be used in vibration attenuation for all frequencies of the rotor. Further, it should be noted that the results have been obtained by using the test machine data only, because the real machine is not yet available. The potential of dynamical modelling and simulation as

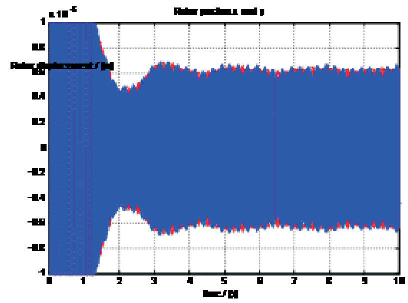

Fig. 14 Process controlled with a modified convergent control structure

a prototyping method for new controllers has therefore been demonstrated in this practical example case.

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