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Technical Briefs

Active Vibration Control of a Multimode Rotor-Bearing System Using an Adaptive Algorithm

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A method for minimizing forced harmonic vibration of a rotor-bearing system by the application of external control forces is presented. The frequency of the vibration is assumed known. In cases of mass unbalance or bend in the shaft this will be shaft rotation frequency and can usually be monitored without difficulty. The control forces could be provided by electromagnetic actuators. The control strategy presented does not require any knowledge of the system parameters and, provided the uncontrolled system is stable, cannot destablize the system. Results from a simulation are shown.

1 Introduction

Major sources of vibration in rotors are mass unbalance and/or bend of the rotor. The resulting forced harmonic excitation will occur at the rotation frequency which will either be known in advance or can be relatively easily monitored. Burrows and Sahinkaya [1] present a method for obtaining an open-loop control strategy which will minimize the vibration of a rotor-bearing system by the application of external control forces. Their method requires knowledge of the system parameters and the out-of-balance response of the system. Reliable estimates of these quantities may be difficult to obtain for some practical systems. In this paper we demonstrate a strategy which will minimize an arbitrary function of any measured states of the system without any knowledge of the system parameters or the out-of-balance response. Measurement of displacements can be achieved by noncontacting displacement sensors. The external forces required for the control could be provided by electromagnetic actuators [2].

2 Theory

Displacement measurements (x_i, y_i) are made at m stations on the rotor. The signals from the transducers are sampled every Δ seconds giving data (x_{ij}, y_{ij}). The rotation frequency ω is monitored and is assumed to be constant or to vary slowly compared with the speed of convergence of the algorithm. A typical function to be minimized would be a sum over time T of the form

$$Q=\sum_{j=1}^{N}\sum_{i=1}^{m} w_i (x_{ij}^2+y_{ij}^2),$$

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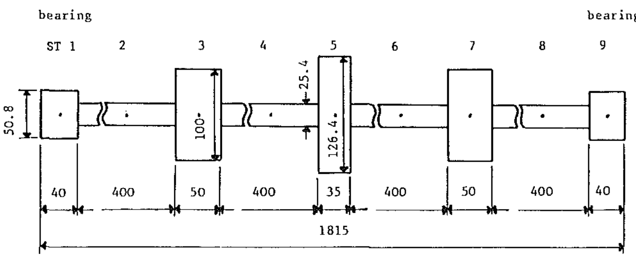

Fig. 1 Symmetrically supported flexible rotor carrying three rigid discs. All dimensions are in millimetres. The positions of stations 1 to 9 are shown above the rotor.

where w_i are weights and $T = N\Delta$.

Assume that there are p actuators each capable of providing a sinusoidal force u_k of frequency ω, variable amplitude a_k, and phase ϕ_k, directly to the shaft. That is

$$u_k=a_k \sin(\omega t+\phi_k) \quad : k=1, \ldots, p$$

Then Q will depend on a_k, ϕ_k and provided T is long enough for the transient response to become negligible the values of a_k and ϕ_k which minimize Q can be found by any numerical technique which does not require derivatives, e.g. [3]. If p does not exceed the number of modes and the control forces are not applied at nodes, these values are unique [4]. A controller based on this strategy cannot destabilize a stable linear system since it only changes the external forces applied. This advantage is also a feature of vibrational control [5] which modifies system parameters by applying high frequency vibrations.

3 Example

Consider the rotor in Fig. 1 used in [1] with mass unbalance distribution

[(4.0,0 deg) (3.3,73 deg) (9.6,196 deg) (2.9,118 deg) (11.8,7 deg) (2.3,266 deg) (9.3,302 deg) (1.6,116 deg) (2.7,160 deg)]

where the pairs represent mass (g) at periphery and angle for the nine stations.[3] No bend is modeled but it would not qualitatively affect the results. A rotation speed of 140 rad/s, slightly above the first resonance, is considered. We choose Q equal to the sum of squares of the x and y displacements at station 3 over a 0.5 s interval sampling at a rate of 600/s. This interval suffices for the transient response to become negligible. One actuator is situated at station 6. Using the simplex algorithm [3] the controller converged to the optimum within 30 s, Fig. 2. The magnitudes of the displacements at station 3 are reduced by factors exceeding 200. The magnitudes of the control forces were 7.1×10^{-3} N and 7.6×10^{-3} N in the x

[3]The differential equations describing the system are available from the authors.

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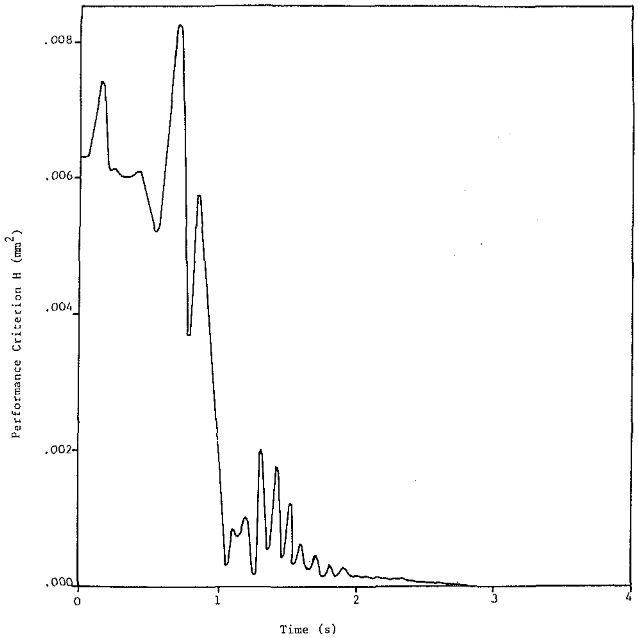

Fig. 2 Change in performance criterion with time

and y directions, respectively. Displacements at other stations were reduced by factors of at least 20 since modes higher than the first were hardly excited at the rotation speed considered. In other situations it would be necessary to include displacements at other stations in Q.

Despite the convincing result of the simulation the authors do not suggest that the control strategy of this paper would be an alternative to balancing a shaft. It could be a valuable supplement to other balancing techniques in applications where reduction of vibration is at a premium.

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