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Optimization of unbalance distribution in rotating machinery with localized non linearity

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ABSTRACT

Classical balancing techniques for rotating machinery consider that these systems are linear. However, if some nonlinearity appears in the structure, these techniques do not work properly and the results obtained regarding the correction weights and their corresponding angular positions are not satisfactory. This behavior is due to the fact that these techniques, such as the influence coefficients method, consider linear relations involving the unbalance excitations and the resulting vibration. On the other hand, the choice of the number and the repartition of the correction planes depends on the possible accessibility that varies for each machine. In this work, a new method dedicated to the identification of the rotating machinery and the unbalance distribution in linear and non linear conditions is realized through pseudorandom optimization methods and the system modeling is performed by using the well-known finite element method. The nonlinearity is introduced by using a frequency dependent bearing. Several computer simulations are performed for different rotor configurations (linear and nonlinear). The methodology is then validated through an experimental test rig. The results obtained demonstrate the effectiveness of the method developed.

1. Introduction

Unbalance is one of the most common problems found in rotating machinery. The vibration level increases due to the unbalance that can lead to bearing failure, incipient cracks and noise. The linearity existing between the vibration and the unbalance of rotating systems is the base for the influence coefficients method that is the most widely used technique for balancing. The popularity of this technique is related to the fact that it does not require a model of the machine since the system is represented by the influence coefficients, which are determined experimentally [1]. However, when the linearity between the unbalance and the vibration is not satisfied, this technique cannot be used successfully. Another aspect of the application of the influence coefficients approach is that trial weights are required at different rotation speeds imposing several stops of the machine along the balancing procedure, which make the methodology very time consuming. Even if balancing is one of the most studied subjects in the field of rotor dynamics, researches are still going on aiming at adapting the balancing strategies for new machines that are lighter, run faster and that could have frequency dependent characteristics.

Steffen and Lacerda [2] investigated the choice of the best balancing planes for the influence coefficients technique and concluded that the best planes are those that minimize the strain energy of the system. Foiles and Allaire [3] concentrated their efforts in developing a balancing technique that uses only the vibration amplitudes for balancing purposes so that the phase information is not required. Their technique needs, however, three trial weights for each balancing plane. Sperling et al. [4] and

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Kim et al. [5] proposed an automatic balancing technique based on the positioning of metallic spheres. Their approach requires an analytical model of the system.

Researchers also propose new approaches that reduce time consumption due to trial weights. Gnielka [6] proposed a modal balancing method without test runs. The modal characteristics of the model and the modal unbalance forces are identified for each considered mode by using an iterative procedure. Krodkiewski et al. [7] proposed an identification method of unbalance change using a non-linear mathematical model. Edwards et al. [8] presented a procedure to determine unbalance and support parameters simultaneously based on the least-squares method. El-Shafei et al. [9] proposed a modal based method without trial weights that is successful if the rotor model is correctly identified. Saldarriaga et al. [10] developed an optimization based identification methodology to determine the unbalance of linear rotating machines. The technique was validated by computer simulations and also using an experimental test rig. Tiwari and Chakravarthy [11] presented an identification algorithm for simultaneous estimation of residual unbalances and bearing dynamic parameters by using impulse response measurements for multi-degree-of-freedom flexible rotor–bearing systems. The algorithm identifies speed-dependent bearing dynamic parameters for each bearing and residual unbalances at predefined balancing planes. Mahfoud et al. [12] developed a technique for structural monitoring in which unbalance is identified by using a reduced model of the rotating system. The order of the model corresponds to the number of sensors used and the position of the unbalance forces have to be previously known. Finally, Castro et al. [13] present an identification approach for rotors mounted on hydrodynamic bearings. The parameters were identified using the measured orbits in the hydrodynamic bearings. The objective function was the difference between measured and simulated orbits, and its minimum corresponds to the identified unbalance. The optimization algorithms were based on genetic algorithm and simulated annealing.

The contribution presented in this paper is an alternative balancing methodology for rotating machinery, aiming first at overcoming the limitations faced by the influence coefficient technique due to the system nonlinearities; secondly at optimizing the balancing weights distribution. However, this alternative technique requires a reliable model of the rotating machine. The strategy developed passes through two steps. First, a model of the system studied is defined and identified, and then the unbalance distribution is identified. The identification for both steps is realized by using pseudo-random optimization techniques.

The principle of the identification approach consists in modeling the system studied by using “as many as needed” subsystems. In this work two submodels were necessary, one for the “linear part” and another for the non-linear part. Each submodel was identified separately. In this way the non-linear contribution is considered as restoring forces. The basic idea is to obtain a reliable model for the machine and then update this model along its operational life. The method proposed enables the balancing optimization of linear and nonlinear rotors and does not require trial weights for its implementation. In this contribution, each node is considered as a possible balancing plane.

In this paper, the method proposed and the system studied are presented first; then, the results obtained for the model identification are shown; for this step, only experimental investigations are reported in this work. Once the rotor is identified, numerical and experimental investigations are performed for the optimization of the unbalanced distribution. Finally, the results obtained are discussed in the conclusion.

2. System configuration

The system studied (Fig. 1) is a test machine composed of a horizontal flexible shaft of 0.04 m diameter containing two rigid discs. The rotor is driven by an electrical motor that can accelerate the shaft until a rotation of 10,000 rpm. The shaft is supported by bearings located at its ends, as follows: a roller bearing (B2) at one end and two ball bearings at the other end (B1). The roller bearing is located in a squirrel cage attached to the framework of the test bench by three identical flexible steel beams. The Electro-Magnetic Actuator (EMA) located on the external cage constitutes a smart active bearing and provides nonlinearity in the dynamics of the system. The active bearing is placed close the drive end. The displacements are measured by using four proximity sensors (Vibrometer TQ 103) arranged perpendicularly in two measurement planes located along the y-axis, namely, measurement planes #1 and # 2 (Fig. 1).

The geometries of the actuators are summarized in Fig. 2. Since an EMA can only produce attractive forces, four “identical” EMA supplied by constant currents are utilized. Each EMA is composed of a ferromagnetic circuit and an electrical circuit. The ferromagnetic circuit has two parts: an (E) shape, which receives the induction coil, and an (I) shape, which is fixed to the squirrel cage. Both parts are made of sets of insulated ferromagnetic sheets. The quality of the ferromagnetic circuit alloy is considered high enough and the nominal air gap between the stator and the beam is small enough to consider magnetic loss as negligible.

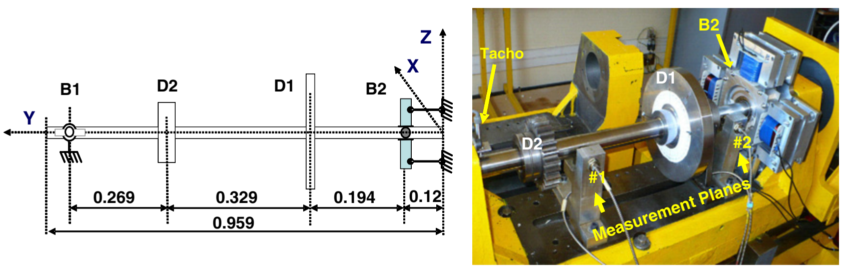

Fig. 1. Experimental test rig.