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Experimental evaluation of a modal parameter based system identification procedure

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Correct modelling of the foundation of a rotor bearing foundation system (RBFS) is an invaluable asset for the balancing and efficient running of turbomachinery. Numerical experiments have shown that a modal parameter based identification approach could be feasible for this purpose but there is a lack of experimental verification of the suitability of such a modal approach for even the simplest systems. In this paper the approach is tested on a simple experimental rig comprising a clamped horizontal bar with lumped masses. It is shown that apart from damping, the proposed approach can identify reasonably accurately the relevant modal parameters of the rig; and that the resulting equivalent system can predict reasonably well the frequency response of the rig. Hence, the proposed approach shows promise but further testing is required, since application to identifying the foundation of an RBFS involves the additional problem of accurately obtaining the force excitation from motion measurements.

1. Introduction

Correct modelling of a rotor bearing foundation system (RBFS) is an invaluable asset for the balancing and efficient running of turbomachinery, particularly for existing installations [1–6]. A promising approach for such modelling uses motion measurements of the rotor and foundation at the bearing supports and at select points on the foundation to identify the relevant modal parameters for an equivalent foundation, defined as a foundation which, when substituted for the actual foundation, reproduces the vibration behaviour of the RBFS over the operating speed range of interest. If successful, such an identification technique would be applicable to the supporting structures of existing turbomachinery installations using readily available monitoring instrumentation.

Traditional modal approach usually solves the non-linear modal analysis equation for the modal parameters of all vibration modes simultaneously [6–9]. In our earlier work, we developed an approach to decouple the modal analysis equation so that each vibration mode could be solved independently; thereby, the number of unknowns in each individual identification equation was significantly reduced and the solving procedure was simplified [10]. The developed approach successfully identified, via numerical experiments, the modal parameters of an equivalent foundation for a relatively simple RBFS consisting of an unbalanced rotor supported by two hydrodynamic bearing pedestals fixed to an undamped flexibly supported flexible foundation block, using as input data the numerically generated motion of the foundation and forces transmitted to the foundation at the bearing supports. However, these numerical experiments did not consider adequately

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the problems associated with experimental measurement data generation and processing. Hence, experimental evaluation of the proposed modal parameter identification technique is in order.

Since an actual RBFS is a relatively complex system and the application of the proposed identification technique has many potential sources of measurement error, this paper, as a first step, evaluates the ability to successfully identify, in a laboratory environment, the modal parameters defining the equivalent system for a flexible flat bar with lumped masses, designed to have five degrees of freedom (DOF) in the horizontal plane over the frequency range of interest. Such a setup will allow one to assess the ability of the technique to cater for practical instrumentation errors, instrumentation limitations as well as to assess data recording and data processing capabilities.

Such a preliminary experimental evaluation of the identification technique is necessary because its application to an RBFS where in the rotor remains in situ has significant additional sources of measurement error. Thus, the force excitation is due to rotor unbalance, it is no longer possible to measure the excitation force directly. Instead, one needs to rely on an accurate dynamic model of the rotor. Also, the unbalance state is generally unknown, so one requires motion measurements at the same speeds for both the original unbalance state and a subsequent unbalance state when a known unbalance is added; and these motion measurements need to be subtracted [11]. Finally, one needs to measure the absolute motion of the rotor at the bearing connexion points. Hence, experimental evaluation of the identification technique for application to as complex a system as an RBFS is outside the scope of this paper and left for future work.

2. Identification equations

As shown in the Appendix A, the equations of motion of a stable n DOF linear system subjected to harmonic excitation forces can be written as:

$$(-\Omega^2 \boldsymbol{I} + i\Omega\,\boldsymbol{\zeta} + \boldsymbol{\lambda})\boldsymbol{A}^T\tilde{X} = \boldsymbol{m}^{-1}\boldsymbol{\Phi}^T\tilde{F}. \tag{1}$$

Eq. (1) comprises the n identification equations ($k = 1, \ldots, n$):

$$\left(-\Omega^2 + i\Omega\zeta_k + \lambda_k\right)\sum_{j=1}^{n} a_{jk}X_j - \sum_{j=1}^{n}\Phi_{jk}F_j/m_k = 0 \tag{2}$$

Knowledge of the measured values of F_j and X_j at a sufficient number of excitation frequencies Ω suffices to identify the elements of $\boldsymbol{\zeta}$, $\boldsymbol{m}$, $\boldsymbol{\lambda}$ and $\boldsymbol{\Phi}$ [10]. These parameters define the desired equivalent system.

3. Experimental procedure

3.1. Experimental rig setup

Figs. 1 and 2 show the experimental rig which consists of a 3 mm by 20 mm rectangular steel bar with five masses clamped along its length. The ends of the bar are clamped in vices using pairs of 'L' keys and each vice is fixed to the laboratory floor via two bolts with a nominal clamping torque of 40 N m. The end fixities ensure that the bar is in tension when clamped in the vices and that there is no axial motion of the bar ends during the experiments. Thus, a steel strip is welded to the left end of the bar and bears against the end face of the left vice; and wedges bear against the end face of the right vice and against a steel pin located through the right end of the bar.

Fig. 3 and Table 1 show the weights of the masses and their locations. These were selected arbitrarily to form a system with 5 natural frequencies with horizontal mode shapes within the planned excitation frequency range. Since the steel bar is flat and thin, the natural frequencies of the vibration modes in the horizontal direction are expected to be lower than those in the vertical direction. Accordingly, all force excitation and displacement measurements were to be in the horizontal plane.

Fig. 1. The general view of experimental rig.

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