

IDENTIFICATION OF BEARING FORCE COEFFICIENTS IN FLEXIBLE ROTORS –EXTENSIONS TO A FIELD METHOD

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ABSTRACT

Rotor-bearing system characteristics, such as mode shapes and their associated natural frequencies and damping ratios are essential to diagnose and correct vibration problems during system operation. Of the above characteristics, reliable identification of fluid film bearing force parameters, i.e. stiffness and damping coefficients, is one of the most difficult to achieve, in particular during field operation. Results of an enhanced method to estimate support force coefficients in flexible rotor-bearing systems based on imbalance response measurements obtained near the bearing locations are presented herein. The procedure can be conducted on site with minimal instrumentation. A test flexible rotor mounted on two-lobe hydrodynamic bearings is used to validate the identification procedure. Imbalance response measurements for various imbalance magnitudes are obtained near the bearing locations and also at rotor mid-span. At shaft speeds around the bending critical speed, the displacements at rotor mid span are an order of magnitude larger than the shaft displacements at the bearings. The identification procedure renders reliable bearing force coefficients for shaft speeds between 1 krpm and 4 krpm. The sensitivity of the method and derived parameters to noise in the measurements is also quantified.

INTRODUCTION

Dynamic forces in oil lubricated bearings are represented in terms of (linearized) stiffness and damping coefficients, $\{K_{ij}, C_{ij}\}_{ij=X,Y}$, i.e.

$$-\begin{Bmatrix} F_X \\ F_Y \end{Bmatrix} = \begin{bmatrix} K_{XX} & K_{XY} \\ K_{YX} & K_{YY} \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} + \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} \quad (1)$$

These coefficients, depending on the bearing configuration and applied static load, are strictly valid for small amplitude motions about an equilibrium condition.

Identification of bearing force coefficients requires of measurement (or estimation) of the excitation forces and measurement of the ensuing rotor response [1]. Several excitation methods such as unidirectional harmonic loads [2], random excitation loads and transient (impact) loads [3] are typical. Tiwari et al. [4] present the most comprehensive review on identification methods, and recommending as the simplest, a procedure that relies on the synchronous response to known calibrated imbalances. De Santiago and San Andrés [5, 6] introduce robust identification methods from imbalance response measurements in rigid rotor and flexible rotor-bearing systems, respectively. The original approach assumes shaft displacements are recorded at the bearing center locations, a condition difficult to realize in practice. In flexible rotors, the rotating structure model is needed for adequate identification of system parameters. Presently, the enhanced method uses analytical transformations to estimate the shaft displacements at the bearing center planes from

measurements at nearby locations, thus making the procedure even more appealing for ready field implementation. Reference [6] details the rotor-bearing system equations of motion and the method to identify bearing stiffness and damping coefficients from imbalance response measurements.

Clearly, the results from the identification method depend on the magnitude and location of the imbalance masses used to excite the system. If the calibrated masses are too small, the shaft amplitudes of motion are also small, even while crossing a critical speed; and the detection of phase angle is inaccurate since the noise to signal ratio (NSR) could be too large. On the other hand, too large imbalances produce large amplitudes of motion, in particular around critical speeds. This may lead to a regime of operation that exacerbates the fluid film bearings' nonlinearities, thus violating the fundamental assumption for estimation of force coefficients, as defined by Eqn. (1).

TEST MEASUREMENTS AND IDENTIFIED COEFFICIENTS

Figure 1 shows the laboratory test rig for conducting the experiments, see [6] for details. A variable speed motor drives, through a flexible coupling, a long flexible rotor (11.2 kg) supported on two, two-lobed fluid film bearings ($\Phi=25.4$ mm). Three pairs of eddy-current probes measure the rotor displacements outboard of the bearing housings and close to the rotor mid span. A commercial DAQ system collects and processes the data, delivering the rotor synchronous response (amplitude and phase relative to a keyphasor).

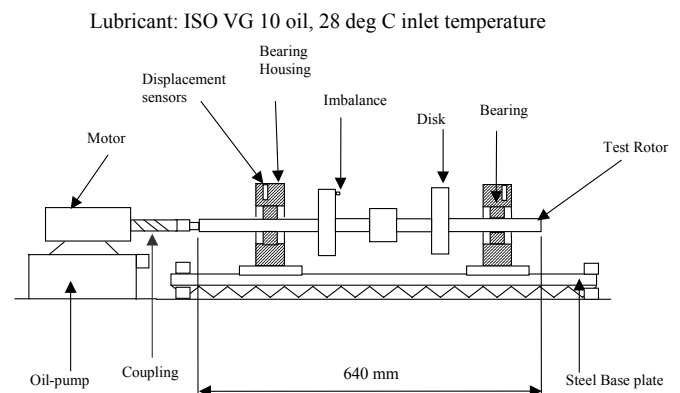


Fig. 1 Test rig for identification of bearing force coefficients

The test bearings have equal dimensions and carry almost identical static loads. Lubricant inlet temperature and pressure remain invariant during the tests. The parameter identification is carried out from measurements obtained from increasing imbalance levels and for shaft speeds to 6 krpm. The masses, attached to the left and right disks,

equal 3.9, 7.2 and 10.5 gram. The tests aim to evidence the dependence of the identified parameters upon the recorded amplitudes of shaft motion. Proper identification of the full set of eight parameters, $\{K_{ij}, C_{ij}\}_{ij=X,Y}$ for each bearing, demands of two independent rotor imbalance responses [5].

Figure 2 depicts the amplitudes of shaft motion recorded at shaft mid span for the three imbalances. The amplitudes are normalized with respect to the smallest imbalance mass. Note that the shaft motion amplitude appears to be proportional to the imbalance magnitude, except for the largest mass when approaching the system critical speed. Thus, the identification procedure will render different bearing force coefficients in this region. Although the test results evidence deviation from the desired (theoretical) linear behavior, the displacements at the bearing locations are an order of magnitude smaller, thus enabling adequate identification of force coefficients.

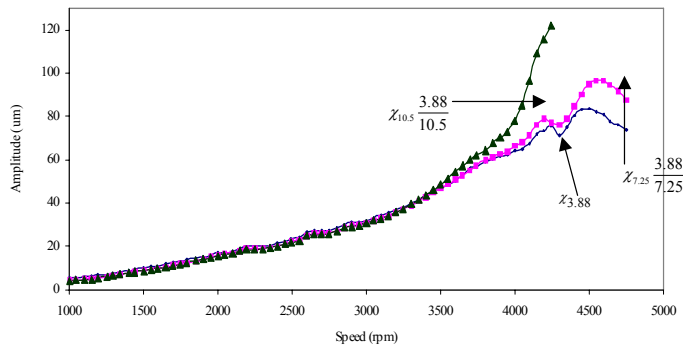


Fig. 2 Normalized amplitude of motion recorded at rotor mid span vs. shaft speed for three levels of imbalance

Figure 3 shows the identified bearing direct stiffness and damping coefficients as obtained from synchronous shaft responses due to the largest imbalance. The force coefficients derived from measurements with smaller imbalances differ little. Figure 3 includes the force coefficients previously identified with the same imbalance amount but with measurements assumed to take place at the bearing locations [7]. The current procedure corrects slightly the stiffness in the vertical direction K_{yy} , (orthogonal to the static load direction). Most noticeably, however, is the difference in damping, C_{yy} , at shaft speeds close to the system critical speed. The identified cross-coupled coefficients are not shown for brevity. Figure 4 shows the direct stiffnesses identified from the test data with added Gaussian-type noise of 10% NSR. These force coefficients, when compared to the ones depicted in Figure 3, evidence the robustness of the identification method, even when significant noise is present in the measurements.

CLOSURE

The paper presents further results on the experimental identification of bearing force coefficients in a flexible rotor supported on a pair of oil lubricated bearings. The method, relying on the measurement of the rotor synchronous motion, is easy to apply as an in-situ diagnostic tool. Tests with increasing imbalance masses establish a limit for the validity of the linearized bearing force coefficient model. Presently, the measurements show the relative insensitivity of the estimated bearing parameters to the amount of imbalance. The method avoids the need to measure shaft motions at the bearing center locations; thus enabling its ready applicability as a field identification method with further potential to provide bearing condition in actual operation.

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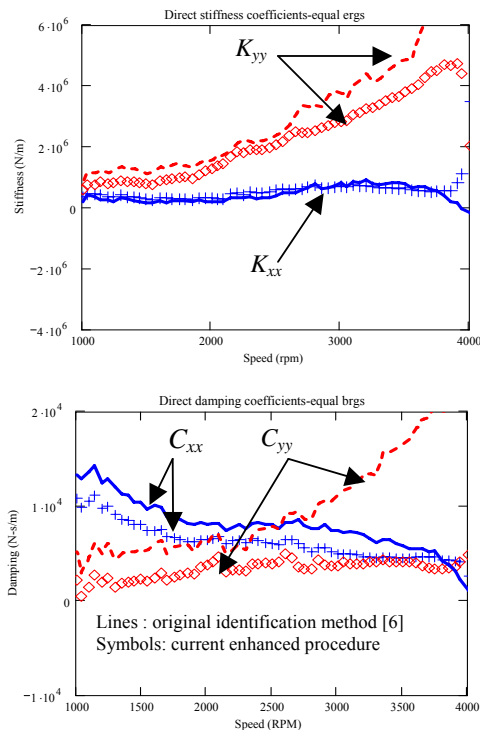


Fig. 3 Identified bearing direct force coefficients from synchronous shaft response due to a 10.5 gram imbalance

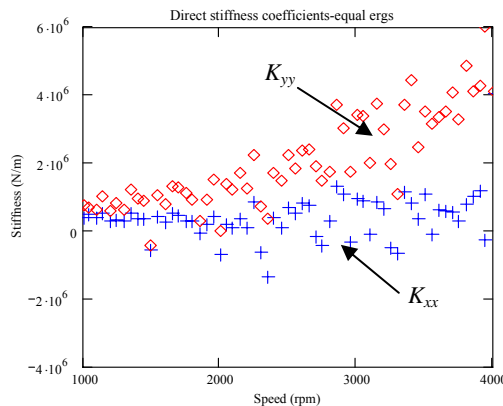


Fig. 4 Effect of Gaussian noise (NSR = 10%) on identified direct stiffnesses from synchronous shaft response due to 10.5 gram imbalance