Experimental Evaluation of a Rotor Model Based Foundation Identification Procedure

M. Yu, N. Feng and E. Hahn

Abstract This paper evaluates experimentally a rotor model based foundation identification procedure (in terms of foundation modal parameters) for rotor bearing foundation systems (RBFS). Earlier experimental evaluations used a deficient rotor model and did not properly consider foundation damping. Demonstrated is the need for an accurate dynamic model of the rotor. It is shown that the proposed approach can identify an equivalent foundation which can predict reasonably well the unbalance response of an experimental RBFS rig over the speed range of interest. The proposed identification procedure shows promise for field applications, but further work is required to identify more accurately the modal damping.

Keywords Experimental evaluation · Foundation identification · Modal parameters · Rotating machinery

1 Introduction

Correct modelling of a RBFS is an invaluable asset for the balancing and efficient running of turbomachinery. A major problem is to properly identify the foundation of existing installations [1]. The approach investigated here requires an accurate rotor model and uses motion measurements at select points on the foundation to identify the modal parameters of an equivalent foundation. Such an approach is

M. Yu
Sun Yat-Sen University, Guangzhou, People's Republic of China

N. Feng
Shandong University, Jinan, People's Republic of China

E. Hahn (✉)
The University of New South Wales, Sydney, Australia
e-mail: e.hahn@unsw.edu.au

J. Beran et al. (eds.), *Advances in Mechanism Design II*,
Mechanisms and Machine Science 44, DOI 10.1007/978-3-319-44087-3_13

attractive, as it can utilize existing monitoring instrumentation. Numerical experiments on a simple RBFS indicate that such an approach is feasible [2], but experimental evaluation is needed to ensure that the approach can cope with the input measurement errors [3]. This paper describes this experimental evaluation.

To this end, an earlier rig [4], which had been used to evaluate the dynamic stiffness parameters of the bearing pedestals of a simple multi disk rotor, was modified to provide a flexible foundation for the rigid bearing pedestals. The modified rig, its commissioning, and the experimental procedure for obtaining the required measurements are described in a previous paper [5]. However, the results in Ref. [5] are only preliminary as they do not properly consider the effect of foundation damping. Also, they are based on a rotor model which does not account for rotor damping, resulting in incorrect force excitation input data. This paper addresses these potentially significant causes of identification error.

2 Notation

$\widetilde{F}$ Harmomic excitation force amplitude vector
$\boldsymbol{I}$ Identity matrix
$\boldsymbol{m}$ Modal mass matrix (diagonal); m_k = modal mass of the kth mode
$\widetilde{X}$ Displacement amplitude vector
$\boldsymbol{\Phi}$ Foundation modal matrix; $\Phi_k = k$th column of $\boldsymbol{\Phi}$
$\boldsymbol{\lambda}$ Eigenvalue matrix (diagonal); $\lambda_k = k$th eigenvalue = ω_k^2
Ω Excitation frequency
$\boldsymbol{\omega}$ Natural frequency matrix (diagonal); $\omega_k = k$th natural frequency
$\boldsymbol{\xi}$ Damping ratio matrix (diagonal); $\xi_k = k$th element of $\boldsymbol{\xi}$

3 The Identification Requirement

For a RBFS with an equivalent foundation having n degrees of freedom (DOF), a harmonic excitation frequency Ω, and periodic response with fundamental frequency Ω, it can be shown [2] that a knowledge of F_j and X_j at a sufficient number of frequencies Ω suffices to identify the elements of $\boldsymbol{\xi}, \boldsymbol{m}, \boldsymbol{\lambda}$ and $\boldsymbol{\Phi}$, which parameters define the desired equivalent foundation. The response is then given by:

$$\widetilde{X} = \boldsymbol{\Phi}\left[\boldsymbol{m}\left(-\Omega^2\boldsymbol{I} + 2i\Omega\boldsymbol{\omega}\boldsymbol{\xi} + \boldsymbol{\lambda}\right)\right]^{-1}\boldsymbol{\Phi}^{\mathrm{T}}\widetilde{F} \tag{1}$$

4 Experimental Procedure

Full details of the rig are given in Ref. [5] so only a brief summary is given here. The rig in Fig. 1 consists of a three disc rotor driven via a flexible coupling by a variable speed AC motor. The motor end of the rotor is supported by ball bearings; the other end by a plain journal bearing. Both bearings are mounted in aluminium pedestals which are bolted to an aluminium block which in turn is flexibly connected via steel bars to a heavy steel table. Steel weights bolted to the aluminium block allowed for some tuning of foundation natural frequencies. Motion measurements used appropriately positioned displacement transducers and accelerometers. Figure 2 shows the accelerometer locations and measurement directions. Displacement transducers were mounted at the bearings and facing a notch at the rotor end. Rotor speed, journal bearing oil temperatures and orbit size were monitored. Signals were processed using in house data processing software.

The natural frequencies of the foundation were determined by hammer tests. Six natural frequencies were found in the frequency range from 0 to 512 Hz with resolution of 0.125 Hz as shown in Table 1. None of the accelerometers was able to find all six. There was no longitudinal vibration frequency below 512 Hz. These natural frequencies formed the yardstick frequencies of the yet to be identified

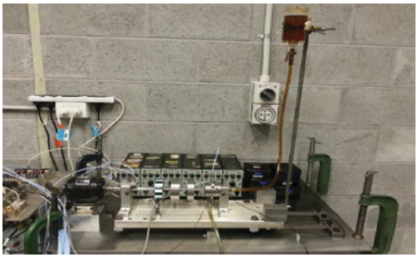

Fig. 1 Rotor bearing foundation rig [5]

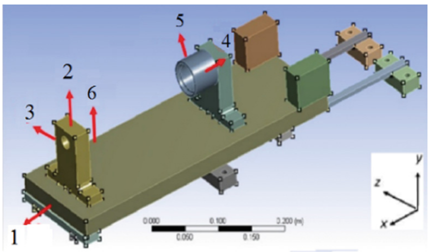

Fig. 2 Accelerometer locations and directions [5]

Table 1 Hammer test and FEM natural frequencies (Hz)

Mode	1st	2nd	3rd	4th	5th	6th
Hammer test	25.500	31.500	40.625	66.625	229.00	481.00
ANSYS FEM	26.102	29.103	49.602	68.137	242.30	462.69

foundation. Also, a finite element model (FEM) of the foundation was used to estimate the first six undamped natural frequencies and corresponding mode shapes of the foundation. Table 1 compares the hammer test and FEM natural frequencies. As expected, agreement is only approximate owing to the limited number of elements which could be accommodated in the FEM; but the results are close enough to provide a qualitative yardstick for the mode shapes. These mode shapes clearly showed that the aluminium block exhibited minimal rotation about the Z axis till the fifth mode and minimal flexure till the sixth mode [5].

The proposed identification procedure presumes a sufficiently accurate dynamic model of the rotor. A discretized rotor model which assumed zero rotor damping was initially accepted as adequate [5] since it had previously proved satisfactory [4]. However, in Ref. [4] all natural frequencies of the rotor were outside the operating speed range, whereas here the first natural frequency of the simply supported rotor (64.73 Hz) is within the operating speed range, necessitating the inclusion of damping in the rotor model to avoid inaccurate excitation force 'measurements' at speeds close to this frequency. With the rotor simply supported in its bearings, its damping ratio was estimated from hammer test signals in the frequency domain, resulting in a value of 0.020267 for the logarithmic decrement of internal shaft damping [6]. This value specified the damping in the updated rotor model.

5 Input Data

As the residual unbalance was unknown, measurements were taken at 'identical' speeds for two rotor rundowns over a speed range from 73 to 20 Hz with known unbalances of 15.189 g mm and 26.81 g mm added to rotor disks 1 and 3 respectively for the second rundown [5]. There were 67 'identical' speeds where speed differences in the two rundowns were less than 0.2 %. Figure 3 shows the magnitudes of the displacement differences obtained from the accelerometers while

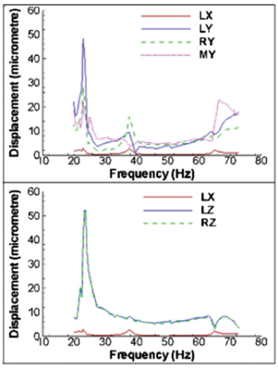

Fig. 3 Magnitude of displacement differences [5]

Fig. 4 Magnitude of excitation force differences

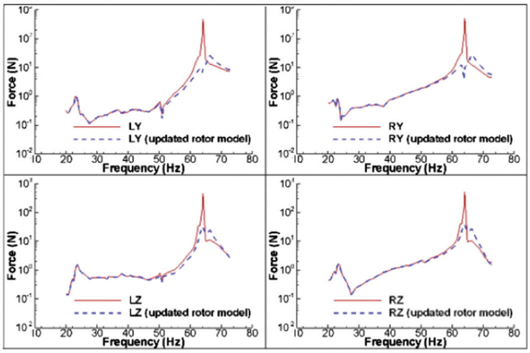

Fig. 4 shows the magnitudes of the transmitted force differences for both the undamped rotor model (used to obtain the preliminary results in Ref. [5]) and the now updated rotor model. Hereafter, for the sake of brevity, displacement differences and force differences will simply be referred to as displacements and forces.

In Fig. 4 one can see the significant effect (at speeds near the rotor natural frequency) of updating the rotor model on the transmitted forces, and hence on the force input 'measurements'. In Fig. 3, it can be seen the longitudinal displacement of the foundation (curve *LX*) is very small. The FEM analysis indicated that the longitudinal natural frequency was well above 512 Hz. Hence, accelerometer ***1*** signals could be ignored. Also, since ω_6 (430.75 Hz) was well above the upper bound of the operating speed range (73 Hz), ω_6 could also be ignored leaving the equivalent foundation to have at most 5 DOF. Also, there is minimal difference in the transverse horizontal displacements at the bearing connection points (curve *LZ*) and (curve *RZ*) so that one has, in effect, a 4 DOF foundation. As indicated in Fig. 3, peaks in the transmitted forces still occur around the rotor natural frequency and can be as high as 50 N. To allow for error in the calculated hysteretic damping in the rotor, acceptable data was restricted to speeds at which the transmitted forces were less than 10 N. Also, the selected speeds had speed spacings in excess of 2 rad/s to reduce error bias, leaving data for 42 speeds.

6 Results and Discussion

The identification procedure outlined in Ref. [2] was then applied to identify a 4 DOF equivalent foundation. The identified parameters and actual natural frequencies are given in Table 2. Figure 5 compares the predicted unbalance responses with the measured responses using Eq. (1). The agreement between actual and predicted responses is quite reasonable though there are large discrepancies between the responses around the fourth natural frequency (60–65 Hz), suggesting incorrectly identified damping coupled with error in the identified natural frequency. Even so, the agreement between the responses is far better than that

Table 2 Identified modal parameters (frequencies in Hz; masses in kg)

Mode	Actual	ω_k	Φ_1	Φ_2	Φ_3	Φ_4	ξ_k	m_k
1	25.5	29.55	−4.89	−3.10	−2.85	−2.28	−9.24E-02	117.19
2	31.5	36.04	−0.62	−0.74	−1.78	2.46	−4.73E-02	70.65
3	40.625	37.38	0.16	1.68	2.68	−4.20	1.03E-02	−382.32
4	66.625	64.71	5.50	3.99	5.85	−0.53	−1.80E-02	−2311.99

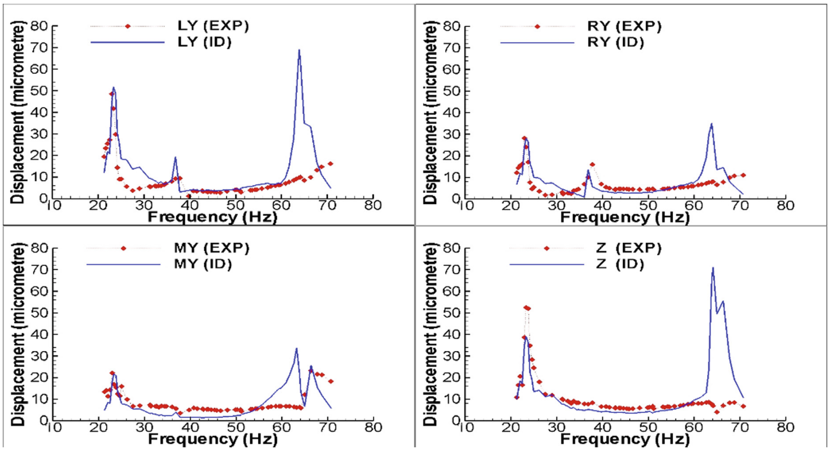

Fig. 5 Measured (EXP) and identified (ID) unbalance responses of the foundation

obtained in Ref. [5] where neither rotor damping nor foundation damping were properly accounted for.

The results in Table 2 and Fig. 5 suggest that the identification procedure needs further improvement and/or that there is too much error in the measurement data to enable better identification. The identified natural frequencies should agree better with the yardstick values since foundation damping is small. Negative damping ratios and modal masses indicate excessive round off error, exacerbated by errors in measurement data. To better understand these results, all modal parameters were reidentified using a different iterative procedure. Again, some of the identified damping ratios and modal masses were negative. The difficulty of identifying the damping ratio is highlighted in this alternative iterative procedure, where the other modal parameters are identified independently of the damping ratio. Its evaluation then involves subtracting two relatively large numbers both of which required extensive computation. Hence further work on perfecting the identification procedure is needed before it can be confidently applied in practice.

7 Conclusions

1. For a laboratory RBFS, a 4 DOF equivalent foundation has been identified which predicts reasonably well the measured unbalance response over the operating speed range.
2. An accurate dynamic model of the rotor, which included rotor damping, was essential to minimise inaccurate force excitation 'measurements' at speeds near the rotor natural frequency.
3. Further work is required to better evaluate the foundation modal damping parameter, which is very susceptible to computational round off errors.

References

1. Lees, A., Sinha, J., Friswell, M.: Model-based identification of rotating machines. Mech. Syst. Signal Process. **23**(6), 1884–1893 (2009)
2. Yu, M., Feng, N., Hahn, E.: An equation decoupling approach to identify the equivalent foundation in rotating machinery using modal parameters. J. Sound Vib. **365**, 182–198 (2016)
3. Yu, M., Feng, N., Hahn, E.: Experimental evaluation of a modal parameter based system identification procedure. Mech. Syst. Signal Process. **68–69**, 302–315 (2016)
4. Feng, N., Hahn, E.: Identification of pedestal parameters in rotor-bearing-pedestal systems using known unbalance, In: Proceedings of ASME DETC'01, 9 p (2001)
5. Yu, M., Feng, N., Hahn, E.: Experimental evaluation of a modal parameter based foundation identification procedure. In: Proceedings of the 9th IFToMM International Conference on Rotor Dynamics Mechanisms and Machine Science, 10 p (2014)
6. Lund, J.: Stability and damped critical speeds of a flexible rotor in fluid-film bearings. J. Eng. Ind. **96**(2), 509–517 (1974)