CHAPTER 6

HARMONIC RESPONSE OF Rotor–BEARING SYSTEMS

6.1 Introduction

Vibration analysis is a prerequisite for design and diagnosis of rotating machinery. Most vibrations in rotating machinery are induced by rotation–related sources: rotating unbalance is the major source of vibration synchronous to the rotational speed ($\Omega$); misalignment and cracks in shafts cause the vibration of frequency $i\Omega$ ($i$ is an integer); ball bearing defects cause vibration with frequency $n\Omega$ ($n$ is a real number), and so forth [1–6]. Thus, forced vibration analysis of rotating equipment subject to asynchronous harmonic excitation is essential for identifying the vibration sources or ensuring proper design of the equipment.

When a rotor bearing system is represented by a discrete model with constant matrices, its forced response calculation subject to any type of excitation is straightforward, either with direct method or modal analysis. However, since fluid film type bearings and gyroscopic effects play a significant role in the response characteristics as the rotational speed increases, the discrete model with matrices which are independent of rotational speed fails to represent the real system behavior accurately. The rotational speed dependency, mostly due to the gyroscopic effect and the bearing properties, has prohibited the direct use of standard modal analysis [7,8] for calculating the forced vibration response of the system, since it is found to be inefficient to calculate the modal responses every time the rotational speed is incremented and kept constant. On the other hand, one of the most commonly used techniques to calculate forced responses has been the direct computational method using the FEM[7,9] and the transfer matrix method[10,11]. Although the direct computational method avoids the difficulty in applying the modal analysis technique, in particular, to complex structures, the forced response calculation requires repetitive inversions of large complex matrices.

Critical speeds can be computed as the imaginary parts of the complex eigenvalues of the rotor–bearing system, under the constraint that the whirling frequency equals the frequency $n\Omega$ of an asynchronous harmonic excitation. However, the imaginary parts of the eigenvalues (modal frequencies) can change
with rotational speed due to the presence of the rotational speed dependent parameters such as the gyroscopic effects and journal bearing properties. This eigenvalue analysis [12] determines the critical speeds of the system without solving the non-homogeneous equations which include the asynchronous harmonic excitations, even though a critical speed is the speed at which response to asynchronous excitation is a local maximum. Instead, this is done by noting the speeds at which the eigenvalues are excited by the asynchronous harmonic frequency. Graphically the critical speeds are determined as the intersections of $n\Omega$ with $\text{Im}(\lambda)$ on the whirl speed chart. It may be more efficient computationally to preconstrain the frequency in the analysis to be $n\Omega$; this is known as critical speed analysis[12, 13]. Critical speeds can also be identified as the peaks of the whirl amplitude from a computation of asynchronous response. This approach has certain advantages peculiar to the rotor–bearing problem, and so has been widely used for critical speed and asynchronous response analysis.

In this chapter, we describe a method based on [14] which transforms the rotational speed dependent eigenvalue problem of the original system into the rotational speed independent eigenvalue problem of complex matrices by introducing a lambda matrix, so that the standard modal analysis technique can be directly applied to obtain the forced vibration response of the FEM rotational speed dependent rotor bearing system to an asynchronous harmonic excitation. The essential feature of the method is to obtain a new generalized eigenvalue problem(or latent value problem) not containing any rotational speed dependent parameters, but depending on the nature of excitation, assuming that the bearing dynamic properties are well approximated by polynomial functions of rotational speed. As for gyroscopic systems without other rotational speed dependent parameters, the method gives the exact solution since the gyroscopic effects are exactly represented by first order polynomial of rotational speed, and, furthermore, the critical speeds and the corresponding damping coefficients, which depend upon the frequency of excitation force, can be readily identified from the resulting eigenvalues. The method has significant advantages in computation and interpretation of the results compared with the standard modal analysis technique. In particular, critical speeds are readily available from the analysis. The limitation of the method, however, is the inability to deal with the stability problem for the system. In addition, this method permits the investigation of response sensitivity to a harmonic excitation or the sensor locations without calculating the response. In other words, the influence coefficient matrix is readily available from the result of the generalized modal analysis. It allows the rigorous development of the well–known theories such as balancing using influence coefficients[15,16] and modal balancing[17,18].