International Journal of Mechanical Sciences 111-112 (2016) 43–54

Contents lists available at ScienceDirect

International Journal of Mechanical Sciences

journal homepage: www.elsevier.com/locate/ijmecsci

Influence of cross-coupling stiffness in tilting pad journal bearings for vertical machines

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ARTICLE INFO

Article history:
Received 6 October 2015
Received in revised form
4 March 2016
Accepted 19 March 2016
Available online 26 March 2016

Keywords:
Vertical rotor
Journal bearing
Cross-coupling stiffness

ABSTRACT

This paper evaluates how cross-coupling coefficients affects the dynamics of a vertical rotor with tilting pad journal bearings. For vertical machines, the bearing properties are dependent on the bearing load and direction. As a result, it generally requires that bearing properties are calculated at each time step using the governing fluid dynamic equations. This method gives a good representation of the bearing but computational time increases and can cause stability problems. In this study the bearing properties are instead modelled as a function of eccentricity and its direction. Hence, the bearing properties can be evaluated at each time step without solving Navier–Stokes or Reynold's equation. The main advantage of using this method is to decrease the computational time. The cross-coupling stiffness and damping coefficients are usually neglected since they are small compared to the radial stiffness and damping coefficients. In this paper, the simulated unbalanced response is compared to experimental results and it is seen that the cross-coupling stiffness for vertical machines can influence the dynamics. It is shown that the cross-coupling can be of the same order of magnitude as the radial stiffness component depending on the shaft angular position in the bearing. Including cross-coupling increases higher frequency components and the experiments show similar behaviour. Hence the cross-coupling stiffness and damping coefficients should be included when simulating vertical machines subjected to high loads or when the detailed dynamical behaviour is important to investigate.

1. Introduction

Tilting pad journal bearings (TPJB) are widely used as supports for hydroelectric power plants because of their good stability. During the last decades, the hydropower industry has been shifting from base electricity production to more regulating power. This introduces some negative effects such as increased wear and fatigue. Building new machines requires better and more efficient models for optimizing the design. When performing rotordynamic analysis with TPJB's, the challenge is to describe the stiffness and damping coefficients dynamically. As early as 1964, Lund [1] published a paper with stiffness and damping coefficients for TPJB's in horizontal machinery. For vertical machines, it is hard to find a dynamical model for the TPJB's, in comparison to a horizontal machine where there exists a stationary working point. In horizontal machines the dead weight of the rotor is usually sufficient to describe the force in the bearings and a linear model is normally used to describe the dynamic behaviour. Many studies have been done for horizontal machines, Glienicke et al. [2] performed experiments on a horizontal rotor supported by TPJB's.

Studies on differences between load on pad (LOP) and load between pad (LBP) were performed by Childs [3]. In industry, most of the calculations on vertical machines are performed considering constant values for the bearing properties [4]. However, the stiffness and damping coefficients will depend on the shaft angular position in the bearing. Hence, since there is no static operating point the difference between LOP and LBP will introduce dynamic bearing coefficients. Similarities can be seen from cracked rotors which also introduce time dependent coefficients [5,6]. To describe the bearing coefficients dynamically in vertical machines, Navier–Stokes or Reynold's equations needs to be solved. It is possible to describe the bearing

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http://dx.doi.org/10.1016/j.ijmecsci.2016.03.017
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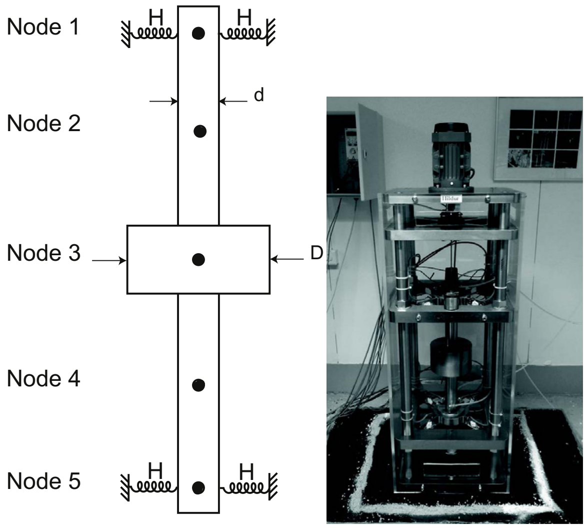

Fig. 1. Test rig setup.

dynamically by using the simplified Reynold's equation. Comparing with horizontal machines, there has not been as much studies performed on vertical machines with TPJB's. White et al. [7] investigated how radial clearance influenced the response using Reynold's equation to calculate the nonlinear bearing coefficients. Cha and Glavatskih [8] show the difference between horizontal and vertical orientation of TPJB's using Comsol Multiphysics where Reynold's equation is solved. Cardinall [9] modelled a hydropower rotor with TPJB's and used Reynold's equation to describe the coefficients. All of these studies require that Reynolds equation is solved at each time step to update the stiffness and damping coefficients. As a result, simulation time increases and performing parametric studies becomes impractical. As well, simulations of large hydropower systems with many degrees of freedom and large time periods requires a fast and good bearing description. A similar study has been performed by Nässelqvist et al. [10] where he proposed a method to model the bearings using a bearing description as a function of eccentricity and load angle. The method used in this paper is based on that model with some

Table 1
The test rig rotor and bearing properties.

Symbol	Description	Item	Value	Unit
Rotor				
d		Shaft diameter	49.84	mm
D		Disc diameter	168	mm
e_m		Mass unbalance radius	70	mm
H		Bearing bracket stiffness	500	MN/m
L		Shaft length	500	mm
Bearing				
R		Shaft radius	24.920	mm
R_b		Bearing radius	25.045	mm
R_p		Pad radius	25.075	mm
θ		Pad angle	72	deg
N		No. of pads	4	–
C_b	$R_b - R$	Radial bearing clearance	0.125	mm
C_p	$R_p - R$	Radial pad clearance	0.155	mm
m	$1 - C_b/C_p$	Preload factor	0.19	–
x	θ_b/θ	Pad pivot offset	0.6	–

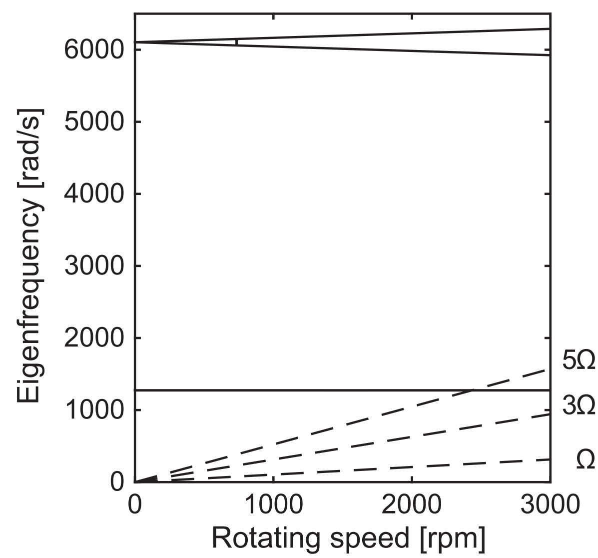

Fig. 2. Campbell diagram for the test rig considering the bearing coefficients fixed, here the dashed lines represent Ω, 3Ω and 5Ω.

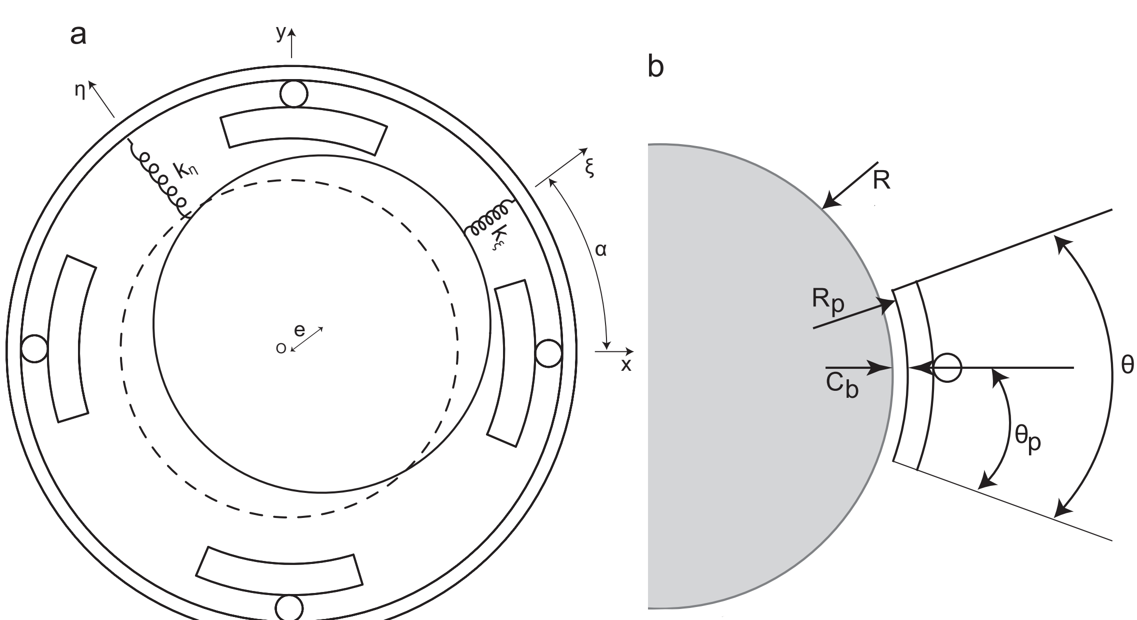

Fig. 3. Sketch of the test rig tilting pad bearing with a eccentricity e in the ξ-direction.