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Agenor de T. Fleury
Domingos A. Rade
Paulo R. G. Kurka *Editors*

Proceedings of DINAME 2017

Selected Papers of the XVII International Symposium on Dynamic Problems of Mechanics

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ISSN 2195-4356 ISSN 2195-4364 (electronic)
Lecture Notes in Mechanical Engineering
ISSN 2524-6011 ISSN 2524-602X (electronic)
ABCM Series on Mechanical Sciences and Engineering
ISBN 978-3-319-91216-5 ISBN 978-3-319-91217-2 (eBook)
https://doi.org/10.1007/978-3-319-91217-2

Library of Congress Control Number: 2018941223

Printed on acid-free paper

This Springer imprint is published by the registered company Springer International Publishing AG part of Springer Nature
The registered company address is: Gewerbestrasse 11, 6330 Cham, Switzerland

Foreword

This book is the first volume of *ABCM Series on Mechanical Sciences and Engineering—Proceedings of International Symposium on Dynamic Problems of Mechanics—DINAME* and brings together the work of some Brazilian leaders on research concerning the broad area of Dynamics. This book presents a compendium of works presented at DINAME 2017, covering traditional subjects in the area such as dynamic systems, vibration, control, as well as new and non-less exciting subjects in contemporary Dynamics such as robotics and intelligent materials.

The *ABCM Series on Mechanical Sciences and Engineering* is a result of an agreement set by Springer and ABCM and its first volume was an initiative of the organizers of DINAME 2017 supported by the ABCM Direction Board—biennium 2015–2017.

The ABCM Direction Board expects that these proceedings become a disclosure vehicle for the best works presented in our events. We also hope that in a close future, the continuity of this series of books becomes reference material to graduate students, professors, and professionals developing research in the area of Dynamics.

The ABCM Direction Board and the authors and participants in DINAME 2017 gratefully acknowledge the support to this event received from Coordination for the Improvement of Higher Education Personnel—CAPES, National Council for Scientific and Technological Development—CNPq, and São Paulo Research Foundation—FAPESP, recognizing that without the success of this symposium, this book would not be possible.

The Board of Directors of ABCM is also grateful to the ABCM Committee of Dynamics and the professionals that have actively participated in the elaboration of this book as organizers authors, co-authors, and reviewers. In particular, we and the ABCM community thank our colleagues who took the responsibility of being the

chairs of the DINAME 2017 and editors of ABCM Series: Agenor de T. Fleury from University of São Paulo (USP), Domingos A. Rade from Aeronautics Institute of Technology (ITA), and Paulo R. G. Kurka from University of Campinas (UNICAMP).

São Carlos, SP, Brazil

Gherhardt Ribatski
(President of ABCM)

Preface

The International Symposium on Dynamic Problems of Mechanics—DINAME—is a biannual symposium promoted since 1986 by the Brazilian Society of Mechanical Sciences and Engineering (ABCM), and organized by its Committee of Dynamics.

Along the years, the Symposium became a vivid forum for scientists, academics, and practitioners to present and discuss developments related to dynamic problems of mechanics.

The meetings are traditionally held in quiet and pleasant sites, away from overcrowded areas, in a regime of immersion, which enables cross-fertilization of ideas, strong scientific exchanging and socialization among participants. In order to maximize the exchange of scientific ideas, achievements, and trends on topics related to the broad area of Dynamics, a single-session format is adopted for the oral presentations of the papers. The acceptance for presentation and inclusion of the papers in the proceedings is based on a two-phase peer-reviewing process of abstracts and full-length (10 pages) manuscripts.

The 2017 edition of DINAME was held from March 5 to March 10, 2017, at Beach Hotel Sunset and Cambury, in São Sebastião, which is located on the seashore of the State of São Paulo, in Brazil. The technical program comprised 100 regular papers, 2 invited papers, 7 keynote lectures, and 2 short courses, the latter intended for the education of young researchers in topics encompassed by the scope of the symposium.

In the context of an existing agreement between Springer and ABCM, an initiative was launched by ABCM Direction Board and the organizers of DINAME 2017 aiming at publishing the symposium proceedings as the first volume of the *ABCM Series on Mechanical Sciences and Engineering—Proceedings of DINAME 2017: Selected Papers of the XVII International Symposium on Dynamic Problems of Mechanics*. For this purpose, after DINAME 2017, the authors were invited to submit improved versions of their papers, which were subjected to another peer-reviewing process. This process led to the selection of 39 papers that compose the present volume, which are believed to be a representative sample of the best research works presented in the symposium.

The papers are organized according to the sessions established in the symposium, namely: Rotordynamics, Vibrations and Structural Dynamics, Robotics and Mechatronic Systems, Control of Mechanical Systems, Nonlinear Dynamics, Vehicle Dynamics and Multibody Systems, Wave Propagation, Acoustics and Vibroacoustics, and Uncertainty Quantification and Stochastic Mechanics.

The organization of DINAME 2017 had the financial support of the following Brazilian research agencies, to which the organizers are very grateful:

- CAPES Foundation (Brazilian Ministry of Education)
- National Council for Scientific and Technological Development—CNPq (Brazilian Ministry of Science, Technology and Innovation)
- São Paulo State Research Foundation—FAPESP

We sincerely hope that these proceedings will be useful to Brazilian and international readers interested in dynamic problems of mechanics.

São Paulo, Brazil — Agenor de T. Fleury
São José dos Campos, Brazil — Domingos A. Rade
Campinas, Brazil — Paulo R. G. Kurka

ABCM—Brazilian Society of Mechanical Sciences and Engineering

DINAME 2017 Chairmen

Prof. Agenor de Toledo Fleury
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Contents

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Part I
Rotordynamics

Estimation of Rotordynamic Seal Coefficients Using Active Magnetic Bearing Excitation and Force Measurement

Christian Wagner, Wataru Tsunoda, Tobias Berninger, Thomas Thümmel and Daniel Rixen

Abstract In high-speed rotational machinery such as pumps or compressors, contactless seals are commonly used to separate different fluids or gases and pressure levels. However, the presence of a leakage flow through the seal gap exerts forces on a rotor. These can culminate in stiffening, restoring, and damping effects as well as in unstable, self-excited vibrational behavior. The JEFFCOTT rotor model and rotordynamic seal coefficients are put under investigation to prevent instability in the rotating machinery and to determine the rotor-seal systems dynamic behavior. This paper focuses on an experimental methodology, examined on a flexible rotor-seal test rig using an active magnetic bearing for excitation. Coefficient identification problems due to unknown random force (noise) in the experiment are shown and a solution is described in detail and validated on the test rig. The presented methodology leads to a calculation of rotordynamic seal coefficients during safe operating conditions. They are ultimately used to describe the system's behavior and to predict the onset speed of instability.

Keywords Rotordynamic · Seal · Instability · Self-excited vibration
Active magnetic bearing · Coefficient estimation

1 Introduction

Seals in turbopumps are mostly used to minimize leakage flow from high pressure areas to low pressure parts. Because of the high rotational speeds of common cen-

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A. de T. Fleury et al. (eds.), *Proceedings of DINAME 2017*, Lecture Notes in Mechanical Engineering, https://doi.org/10.1007/978-3-319-91217-2_1

trifugal pumps or compressors, contactless seals, such as floating ring, labyrinth or small gaps are inserted between the rotating and the stationary parts. The ever-present clearance around these contactless seals permits fluid flow through the gap. For an eccentric rotor position, the fluid-velocity distribution inside the seal becomes unsymmetrical, which entails forces on the rotor. These can induce effects such as stiffening, damping, and added mass, but they can also end up in a rotor instability, a self-excited vibration which can destroy the machinery. Seal effects can engender the critical speeds that should be avoided in stationary operation. Thus, dry predictions of the rotor-system's behavior, such as those for critical speeds, damping or stability limits, are unusable under real operating conditions.

The seal forces within a rotor system are mostly modeled as rotordynamic coefficients:

$$-\mathbf{h}_s = \begin{bmatrix} m_{xx} & 0 \\ 0 & m_{yy} \end{bmatrix} \ddot{\mathbf{q}} + \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} \dot{\mathbf{q}} + \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{xx} \end{bmatrix} \mathbf{q} \tag{1}$$

with the motion of the rotor, $\mathbf{q}$, the seal reaction force, $\mathbf{h}_s$, and the rotordynamic seal coefficients m, c and k for added mass, the fluid's inertia, damping, and stiffness with cross-coupling parts. The coupling inertia terms in the mass matrix are neglected. The different types of seals in a pump can usually be simplified to a cylindrical annular seal for a rotordynamic analysis.

To ensure safe rotor-seal system operation, validated models and methods for characterizing the seals' behavior, the rotordynamic seal coefficients, in simulation and experiment are needed.

2 Literature Overview

The prediction of seal forces and effects mostly leads to a calculation of their rotordynamic coefficients. Several efforts have been made for theoretical and experimental prediction. Simple and fast models are based on bulk-flow theory, which is a simplification of the NAVIER-STOKES equations assuming a constant fluid velocity along the seals clearance.

One of the first efforts to determine restoring seal forces was made by [2]. They used the bulk-flow theory and incompressible fluid flow through a short annular seal. They express equilibrium through the axial momentum equation using turbulent wall friction models and the given pressure gradient over the seal as a boundary condition. For a centered rotor position, a perturbation analysis with small dynamic motion results in differential equations of the fluids motion. Moreover, a linearization of the fluid forces in reaction to the perturbation leads to the rotordynamic seal coefficients for the centered shaft position. The circumferential flow is supposed to be a fully developed, turbulent COUETTE flow. The assumption of constant fluid velocity in the axial and circumferential directions leads to a constant wall-friction factor, λ, for the whole seal [1].

Also based on the bulk-flow theory [3] introduced a closed-form analytical solution for the rotordynamic coefficients of a short, plain annular seal. He also used a perturbation analysis to solve the differential equations. The model's improvement is the consideration of fluid inertia terms and the inlet swirl, the circumferential fluid velocity at the seal's entrance [1]. The simulation agrees well overall with measurements [15].

Padavala and Palazzolo [10] developed a more detailed model, but with higher computational costs. Based on bulk-flow theory, the model discretized the annulus into finite parts to consider a variation of wall-friction factors in the circumferential and axial directions. In contrast to the finite difference methods used in [5] or [12], Padavala's model uses continuous functions, created by cubic splines to fit the distribution of the variables, pressure, velocity, and so forth. With this technique, it is possible to solve the bulk-flow equations for every finite part to get the pressure and velocity distribution within the seal. Although this model gives good to excellent agreement with measurements, its computational costs are high [15].

Further methods, based on finite volume CFD calculations like those in [17], or finite difference methods, or using the REYNOLDS equation known for journal bearings with turbulence correction factors and solved it with finite element methods, see [13], gives high quality results.

The consequence of seal forces acting on the rotordynamics of the whole system are well described in [4, 6]. The effect of self-excited vibrations and rotor instability like the "oil-whip" phenomenon are illustrated in [9] for a system with two degrees of freedom. Parametrization and variable description of the JEFFCOTT rotor model used are attributable to [11, 14].

Others focus on coefficient measurements using an AMB rotor system to measure transfer functions. Zutavern [18] for example gets good measurement results for frequency domain identification methods. The use of a rotor seal system in journal bearings with AMB excitation, in [7], with the variation of stiffness and damping of the AMB controller leads to the rotordynamic seal coefficients for steam turbine seals.

3 Modeling: Jeffcott Rotor Model

The simplified JEFFCOTT rotor model (see Fig. 1) with liquid annular seals is used for theoretical explanation, simulation and as far as possible, experiments on the test rig.

The JEFFCOTT rotor models a flexible, massless shaft with a mass disk symmetrically arranged between rigid bearings; see [6]. Here, the center of mass, S, has the distance ϵ from the disk's geometric center, M. Hence, $\mathbf{r}_M$ gives the position for M . $\mathbf{r}_S = \mathbf{r}_M + \epsilon$ is the position of S. Lumping the shaft stiffness, k_r, onto the rotor's center, M, and taking as degrees of freedom the two translations $\mathbf{q} = \mathbf{r}_M$ leads to dynamic equilibrium for the rotor with mass m_r and rotational speed Ω:

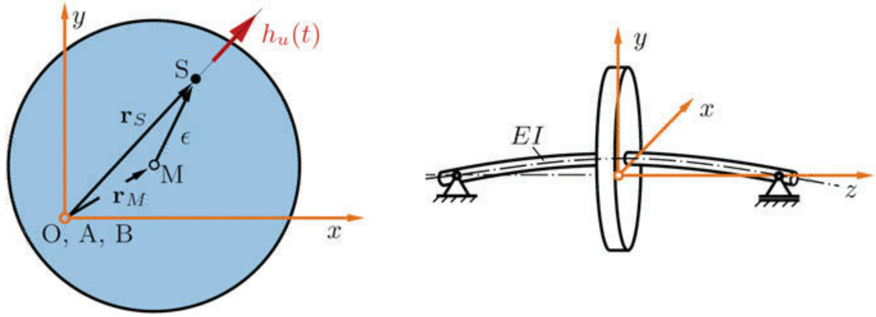

Fig. 1 JEFFCOTT rotor model according to [11, 14]

$$\begin{bmatrix} m_r & 0 \\ 0 & m_r \end{bmatrix} \ddot{\mathbf{q}} + \begin{bmatrix} k_r & 0 \\ 0 & k_r \end{bmatrix} \mathbf{q} = \mathbf{h} \tag{2}$$

$$\mathbf{h}_u = m_r \epsilon \Omega^2 [cos(\Omega t)\ sin(\Omega t)]^T \tag{3}$$

with the equivalent forces $\mathbf{h} = \mathbf{h}_u + \mathbf{h}_e + \mathbf{h}_s ...$ (unbalance, external forces, seal forces, and so forth). The rotor's natural frequency is $\omega_{crit} = \sqrt{\frac{k_r}{m_r}}$, its critical speed.

3.1 Contactless Seal: Minimal Model and Coupling to Rotor System

Defining the seal as system with spring, mass, damper and coupling to the rotor using force $\mathbf{h}_s$ leads to the dynamic equilibrium for the whole rotor seal system:

$$\begin{bmatrix} m_r + m_{xx} & 0 \\ 0 & m_r + m_{yy} \end{bmatrix} \ddot{\mathbf{q}} + \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} \dot{\mathbf{q}} + \begin{bmatrix} k_r + k_{xx} & k_{xy} \\ k_{yx} & k_r + k_{yy} \end{bmatrix} \mathbf{q} = 0 \tag{4}$$

Assuming that

$$\mathbf{q} = \hat{\mathbf{q}} e^{\lambda t} \tag{5}$$

yields the eigenvalue problem with eigenvalues $\lambda = \delta \pm j\omega$. For self-excited vibration, i.e. rotor instability, the positive real parts, δ, must be observed. The seal coefficients' speed dependency, mainly the increasing of cross-coupled parts of the stiffness, k_{xy} and k_{yx}, sets a speed limit for safe operation: the onset speed. A sub-synchronous, self-excited vibration at the rotors natural frequency arises when the onset speed is reached.

Calculating the seal coefficients is essential in this case to avoid rotor instability for safe operation.

3.2 *Bulk-Flow Modeling and Seal Simulation*

The bulk-flow theory is derived from the NAVIER-STOKES equations by neglecting all changes to the fluid flow parameters in radial direction and setting them to constant values or zero.

These assumptions lead to a pressure- and shear-driven fluid flow and a perturbation analysis about a steady state position leads to the fluid forces as a function of the rotor's movement. Simplifications made by Black and Jenssen [2], Childs [3] and Padavala and Palazzolo [10] are used to solve the fluid momentum equations to get the rotordynamic seal coefficients. The detailed description of the used equations and the solving process is well explained in the cited literature. The three models are implemented in MATLAB and called now as Black, Childs and Padavala model. The simulation results will be discussed in later chapters.

4 Experiments: Test Rig Setup

The test rig design is shown in Figs. 2 and 3. It is based on a flexible shaft 1 and a mass disc 2 rotating within a pressurized chamber with length l_c and clearance c_c. The shaft support is realized with stiff ball bearings 3 and the rig is driven by a servo motor 4. The fluid is injected into the chamber and flows through two symmetric

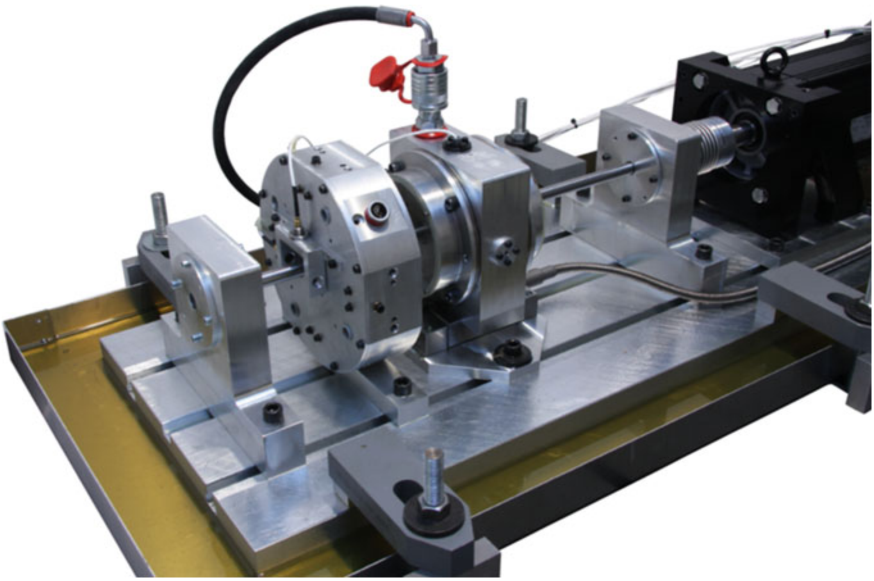

Fig. 2 Seals test rig, photograph

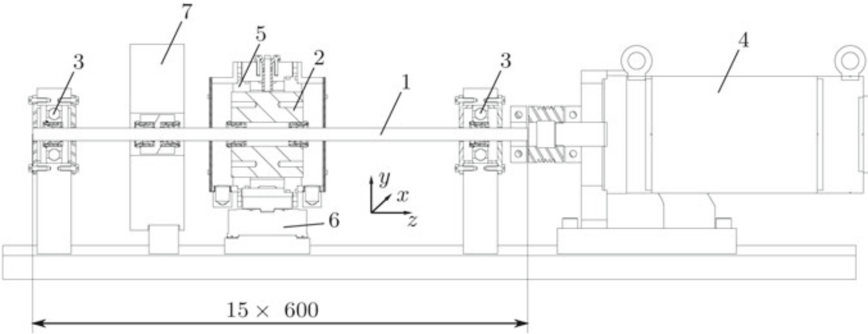

Fig. 3 Seals test rig, see [16]

Table 1 Test rig and seal parameters

Name	Value	Name	Value
Seal clearance	0.17 mm	Rotational speed Ω	0–100 rps
Chamber clearance c_c	2 mm	Rotor mass m_r	5 kg
Seal length l_c	20 mm	Shaft stiffness k_r	2.93×10^5 N/m
Chamber length	40 mm	Density at 40 °C	880 kg/m^3
Pressure difference at the seals Δp	2×10^5 Pa	Viscosity at 40 °C	0.04048 Pa s
Sealless natural frequency ω_0	38.6 Hz	Seal diameter	0.1 m

annular seals 5 to the environment. Two eddy current sensors measure the rotor's motion. A dynamometer 6 under the stator seal is used to get the seal's reaction forces. Further, the fluid inlet pressure, temperature, rotational speed, leakage flow and torque are measured. An active magnetic bearing 7 is used as an actuator for dynamic system excitation.

Table 1 list the test rig, the fluid and seal parameters used for simulation and measurements. The rotor's "dry" first natural frequency ω_0 is at 38.6 Hz. It is decreased by the seal influence to about half the rotational speed Ω, see Fig. 7.

4.1 Measurement Methods for Seal Coefficients

For a symmetrical rotor seal system, the coefficients in Eq. (1) can be written, according to [6]: $M_s = m_{xx} = m_{yy}$, $C_s = c_{xx} = c_{yy}$, $c_s = c_{xy} = -c_{yx}$, $K_s = k_{xx} = k_{yy}$ and $k_s = k_{xy} = -k_{yx}$. To determine the rotordynamic seal coefficients, the seal reaction forces in Eq. (1) are FOURIER transformed into the frequency, see [8]: