Mechanical Systems and Signal Processing 70-71 (2016) 1150–1170

Contents lists available at ScienceDirect

Mechanical Systems and Signal Processing

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

Modelling and identification for control of gas bearings

Lukas R.S. Theisen [a,*], Hans H. Niemann [a], Ilmar F. Santos [b], Roberto Galeazzi [a], Mogens Blanke [a,c]

[a] Department of Electrical Engineering, Technical University of Denmark, DK 2800 Kgs. Lyngby, Denmark
[b] Department of Mechanical Engineering, Technical University of Denmark, DK 2800 Kgs. Lyngby, Denmark
[c] AMOS CoE, Institute for Technical Cybernetics, Norwegian University of Science and Technology, NO 7491 Trondheim, Norway

ARTICLE INFO

Article history:
Received 12 December 2014
Received in revised form
10 July 2015
Accepted 5 September 2015
Available online 1 October 2015

Keywords:
Active gas bearings
Experimental techniques
Grey-box modelling
Rotordynamics
System identification

ABSTRACT

Gas bearings are popular for their high speed capabilities, low friction and clean operation, but suffer from poor damping, which poses challenges for safe operation in presence of disturbances. Feedback control can achieve enhanced damping but requires low complexity models of the dominant dynamics over its entire operating range. Models from first principles are complex and sensitive to parameter uncertainty. This paper presents an experimental technique for "in situ" identification of a low complexity model of a rotor–bearing–actuator system and demonstrates identification over relevant ranges of rotational speed and gas injection pressure. This is obtained using parameter-varying linear models that are found to capture the dominant dynamics. The approach is shown to be easily applied and to suit subsequent control design. Based on the identified models, decentralised proportional control is designed and shown to obtain the required damping in theory and in a laboratory test rig.

1. Introduction

Passive and active gas bearings are receiving growing attention for their high speed operation capabilities. While passive gas bearings offer advantages of high speed operation, low friction, and clean and abundant air as lubricant, they suffer from low damping and vibration instabilities [1–3]. The damping and stability properties can be improved by two methods. One is through foil bearings [4–6] that exploit friction between bumps and foil. Such solutions are relatively cheap, but friction is a significant design challenge [7]. An alternative is to use a mechatronic approach in the form of active control of the gas bearing using piezoactuation [8] or active inherent restrictors [9]. The controllers for such systems could be tuned experimentally, with the uncertainty and lack of quality assurance this method implies, or they could be stringently designed based on dynamic models with documentable performance properties. The latter requires a suitable model, which in a simple manner describes the relation from actuator input to measured output, representing the dynamics of the journal in the frequency range where control is needed. Concerning modelling, air-injection actuators have only received sparse attention. In contrast, electromagnetical actuators and oil bearings have been well covered. Modelling and a linear parameter varying control design were presented for a rotor supported by an oil bearing and an electromagnetically actuated bearing in [10], which showed ability to reduce vibrations and to allow rub-free crossing of the first resonance frequency. Current state of the art models of controllable

* Corresponding author.
E-mail addresses: lrst@elektro.dtu.dk (L.R.S. Theisen), hhn@elektro.dtu.dk (H.H. Niemann), ifs@mek.dtu.dk (I.F. Santos), rg@elektro.dtu.dk (R. Galeazzi), mb@elektro.dtu.dk (M. Blanke).

http://dx.doi.org/10.1016/j.ymssp.2015.09.016
0888-3270/

gas bearings rely on solving the modified Reynolds equation [3], which emerges from including the external controllable lubricant injection into the Reynolds equation. No general closed analytical solutions exist for the equation considering bearings with finite dimension. Solutions are therefore found iteratively over time, and the input–output relationship between piezoactuator and rotor lateral displacement is not easily derived. Literature has therefore generally presented experimentally tuned controllers, e.g. in [8]. Some authors have proposed on-off control rules [11], where the opening of the valves changed the journal pressure, which in turn changed the critical speeds, allowing the rotor to cross them safely. Such approach, however, does not improve the damping characteristics of the gas bearing.

Models suitable for controller design can be developed using system identification. Such models can have low complexity and can yet provide a convenient basis for synthesising controllers [12]. Such models can leave out the details and high order associated with mechanical models based on first principles. Only few results exist for controllable gas bearings, whereas the literature is rich on active magnetic bearings (AMB). AMBs have inherently unstable dynamics [13–15] and therefore require stabilising controllers. For the non-rotating case, the horizontal and vertical AMB dynamics are uncoupled, therefore a model is developed for each of the two directions. In [16,14], a frequency based identification approach was used to develop black-box models of a rotor supported by AMBs. This allowed development of high order continuous time models for a non-rotating shaft supported by AMBs, which sufficed for controller design. In [17], a frequency based method was proposed for identification of the transfer function matrix model of a non-rotating shaft supported by AMBs. The method consisted of steps identifying the submodels separately and finally combining them together. In [13], a similar approach was proposed and deliberately poor controllers were used to allow identification of the poles on the real axis, which are in general not easily identified. In [18], a predictor-based subspace identification algorithm was proposed to identify the dynamics of a non-rotating AMB system, and the obtained model was used to design robust controllers. In [19] a simple black-box model was proposed to represent the vertical displacement of a simple non-rotating rigid shaft supported by AMBs, where the model parameters were estimated online. In [20], an iterative frequency based joint identification/controller design scheme for a non-rotating shaft supported by AMBs was applied using an LQ criterion.

Controllable gas bearings differ from AMBs in the sense that gas bearings can be designed to be open loop stable, hence open loop identification schemes can be used. The lateral dynamics is though coupled due to aero-static effects even in the non-rotating case. Recent work [21] showed that grey-box system identification could be a means to develop such models. The main contribution of this work relies on: (a) grey-box identification to develop low complexity models of the entire rotor–bearing–actuator system and (b) extension of the early results from [21] by investigating the system dynamics as function of both gas injection pressure and rotational speed, which are the two main variables that influence system dynamic behaviour when the static load and the bearing geometry are kept constant [22]. The earlier models from [21] need to be extended to include the dynamics of the piezoelectric actuators and to capture the delay between the displacement of the piezoelectric actuator and the pressure build-up in the journal. The experimental procedure is developed aiming at industrial applications to complex rotating systems supported by gas bearings, where first principles modelling is rarely simple and accurate enough for controller design.

The paper is structured as follows: a brief overview of the experimental test rig is given in Section 2. The piezoelectric actuators are then characterised. The static gain from piezoactuator position to disc position is experimentally characterised. Section 3 presents an experimentally based model of the rotor–bearing system obtained for a set of operational conditions through grey-box identification techniques. Regression techniques are used in Section 4 to fit polynomial surfaces to experimental data and build a linear parameter varying model of the entire controllable rotor–bearing system, which captures the essential behaviour across the operational range. Section 5 presents the design of a decentralised proportional controller to confirm the suitability of the identified models for controller design and the results are experimentally verified. Sections 6 and 7 evaluate critically the results, showing that the controller enhances the damping properties of the gas bearing as expected.

Notation: The paper uses upper case bold letters for matrices $\mathbf{A}$, lower case bold letters for vectors $\mathbf{a}$ and non-bold letters for scalars a or A. When relevant, clear distinctions are made to address time signals $a(t)$ and the Laplace transformed $a(s)$. Units for rotational speeds are listed in revolutions per minute ($1\ \text{rpm} = 1/60\ \text{Hz}$), and pressures are listed in bar ($1\ \text{bar} = 100{,}000\ \text{Pa}$).

2. Experimental setup of controllable gas bearing test rig

The experimental controllable gas bearing setup at hand is shown in Fig. 1. It consists of a turbine (1) driving a flexible shaft (2) supported by both a ball bearing (3) and the controllable gas bearing (4), in which pressurised air is injected through four piezoactuated injectors numbered as shown. The injection pressure P_{inj} is measured by a mechanical gauge before splitting up to the four piezoactuators. A disc (5) is mounted in one end to pre-load the journal. The horizontal and vertical disc movement $\mathbf{p} \triangleq [p_x, p_y]^{\mathrm{T}}$ is measured at the disc location using eddy current sensors (6) in the coordinate frame specified in the figure. The angular position of the rotor ϕ is measured by an optical quadrature encoder (7). The position of the i-th piezoactuator can be controlled through a voltage input $u_{p,i} \in [0;10]V$, where an increasing voltage expands the piezostacks by up to 45 μm, which closes the injector. Fig. 2 shows a CAD drawing of the test rig, where the gas bearing is cut in half to visualise the control mechanism. The nominal clearance of the gas bearing is 25 μm. Given the right conditions of sufficient injection pressure and sufficiently low rotational speed, the gas film generates restoring forces and thereby keeps