Experimental identification of an Aerostatic Thrust Bearing

Federico Colombo, Luigi Lentini, Terenziano Raparelli, Vladimir Viktorov

Abstract Conventional approaches to design aerostatic bearings are based on their static characteristics. However, dynamic predictions are fundamental in order to obtain very precise positioning since air bearings are frequently subjected to dynamic load variations. This paper presents a simple linear time invariant model for aerostatic thrust bearings and the related experimental identification procedure. The experimental procedure identifies the stiffness and damping coefficients of the air gap model. The model validation was obtained comparing the experimental and theoretical bode diagrams of the studied system. The comparison demonstrates that the model leads to good results when neighbours of equilibrium conditions are considered.

1 Introduction

Aerostatic thrust bearings are characterized by low friction, high precision and they represent a competitive clean alternative to conventional rolling and oil bearings. For this reason, aerostatic bearings are widely used in applications where high level of accuracy is required, e.g., measuring and tool machines. Conventional approaches to design aerostatic thrust bearings are based on their static characteristics. However, the prediction of their dynamic features is fundamental to obtain high precision bearings since they are frequently subjected to dynamic load variations. Literature presents several kinds of analytical and numerical approaches [1] to predict the dynamic behaviour of air bearings. In particular, the identification of the air gap stiffness and damping coefficients represents the main goal of this kind of study. Aguirre et al. [2] and Al-Bender [3] proposed a multi-physical FE model of an active aero-

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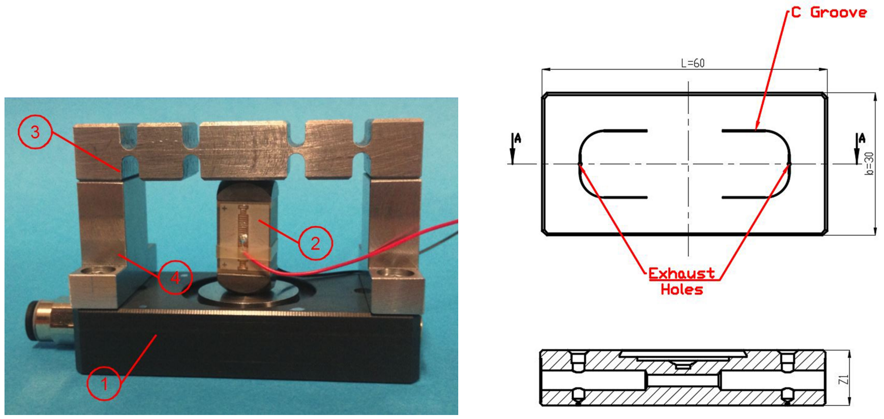
static thrust bearings with a conicity compensation. In their work, they investigate the air bearing dynamic behaviour considering the interactions among mechanical, electrical and pneumatic phenomena. Matsumoto et al. [4], [5] developed a physical model of an ultra precision straight motion system with many degrees of freedom.
Air gap models are often experimentally verified and investigated through different techniques. Impact hammer [6] and modal exciters [7] are currently adopted for the dynamic identification of bearing stiffness and damping coefficients. These two techniques mainly differ in the amplitude of the excited frequency bandwidths and the measured signal-to-noise ratio of the system's responses. When impact hammer tests are carried out, large bandwidth of frequencies are excited. However, the dynamic identification of the stiffness and damping coefficients is hard task due to the low signal-to-noise ratio of the measured spectrum. Conversely, shakers can be used to excite systems in a controlled frequency bandwidth. The amplitude of the excited bandwidth depends on the basis of the selected excitation function, e.g., sweep, pseudo-random periodic signal, chirp. In this case, the identification limitations are due to the interactions between the modal frequencies of the test rig and investigated system.
This paper introduces an experimental procedure to identify dynamic coefficients of an aerostatic thrust bearing by using a step force excitation. After illustrating the identification procedure, the dynamic coefficients are used to built a simple linear time invariant model of the air system. In the first parts of the paper, the main features of the aerostatic bearing and the test rig are discussed. Subsequently, the system physical model and its governing equation are described. Finally, the experimental identification and validation tests are shown.

2 The Active Aerostatic Thrust Bearing

The proposed experimental identification procedure has been performed on the active aerostatic bearing shown in Figure 1. This bearing consists of a conventional air pad 1 integrated with a multilayer piezo actuator 2 (see Figure 1(a)). This integration is possible thanks a compliant mechanism 3 and two supporting element 4 which hold the air pad and actuator together. Moreover, the compliant mechanism has the function to guide and preload the actuator when is running out. As it is possible to see, (Figure 1(b)) the air pad has a rectangular base of $60x30$ mm^2 and two exhaust holes each one inside a "C shaped" groove.

3 Experimental Set-up

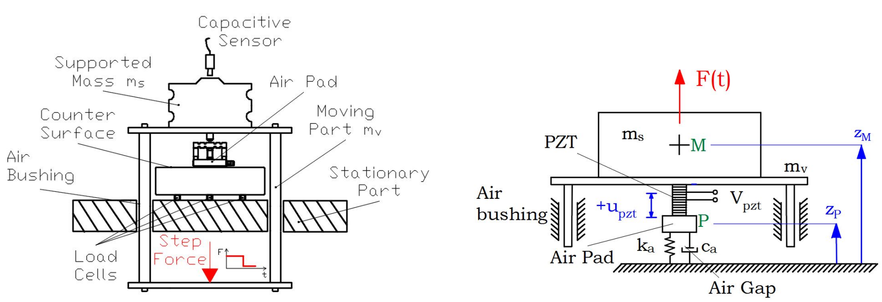
A scheme of the adopted experimental set-up is shown in Figure 2(a). The test bench consists of a moving and stationary parts. The moving part has a mass m_v and is composed of two horizontal and two vertical crossbars which have rectangular and



(a) A photo of the proposed ATB. (b) The air pad lower surface.

Fig. 1 The proposed active aerostatic thrust bearing.

circular cross sections respectively. The presence of two air bushings allows vertical motions of the moving part with respect to the stationary one. The bearing was placed between the upper crossbar of the movable structure and the bearing counter surface. External static loads are imposed on the system through the weight of mass m_s, which is fixed on the top of the moving structure. The load acting on the system is currently measured by means of three load cells (KISTLER Type 9313AA1) which are circumferentially and symmetrically bolted between the air pad counter surface and the stationary part.



(a) A scheme of the test bench adopted for the air pad dynamic identification. (b) The physical model of the identified system.

Fig. 2 System mechanical and physical configurations.

System displacements are measured at the center of the supported mass by means of a capacitive sensor (MICROSENSE 8810 with a 2823 probe). The dynamic identification test was performed by applying a broad band excitation to the system and by evaluating its response in the frequency domain. In particular, step force excitations were chosen in order to exploit the test bench features and because this simple procedure makes it possible to impose an almost identical input to the air pad. Step force was obtained by suspending a small weight to a rope at the lower rectangular cross bar of the moving part. A negative step force occurs almost instantaneously when the connection rope is cut.

4 System Modelling

The motion of the system is described considering only its vertical translation as degree of freedom, due to the presence of two air bushings. In fact, these frictionless constraints do not allow additional displacements and rotations. The independent physical coordinate to describe the motion of the system can be arbitrary chosen among the absolute displacements z_p or z_m which are relative to the air pad (point P) and mass (point M). Relative displacements u_{pzt} correspond to the piezo actuator stroke changes and allow air gap variations to be compensated for. Changes of the actuator stroke are achieved by modifying the actuator control voltage V_{pzt} on the basis of the piezo electric material features. In this instance, z_m has been selected to describe the system motion. The relationship among displacements z_m, z_p and u_{pzt} is the following:

$$z_m = z_p + u_{pzt} \tag{1}$$

Concerning the model's parameters, the mass $m = m_s + m_v$ is the total and only one considered. It is realistic to treat the air pad and the piezo actuator ($k_{pzt} = 267 \cdot 10^6 N/m$)as rigid bodies. The air gap was modelled as a spring-damper system which has k_a and c_a as stiffness and damping coefficients respectively. $F(t)$ represents the external force acting on the system. By considering the non-linear nature of the air gap features, the presented physical model is feasible only in neighbourhoods of an equilibrium conditions. The system governing equation 2 can be easily achieved through a suitable combination of the equilibrium equations and by applying the Laplace transform.

$$Z_m(s)[ms^2 + c_a s + k_a] = F(s) + U_{pzt}(s)[c_a s + k_a] \tag{2}$$

Figure 3 shows the block diagram representation of the investigated system which is considered as MISO (Multiple Input Single Output). Indeed, the mass displacement $Z_m(s)$ is expressed as the sum of two contributions relative to external force disturbances $F(s)$ and the actuator stroke $U_{pzt}(s)$. The actuator strokes is proportional to the control voltage $V_{pzt}(s)$ applied at the actuator terminals. $K_{V/\mu m}$ is the static gain between the actuator control voltage $V_{pzt}(s)$ and its resulting stroke $U_{pzt}(s)$.

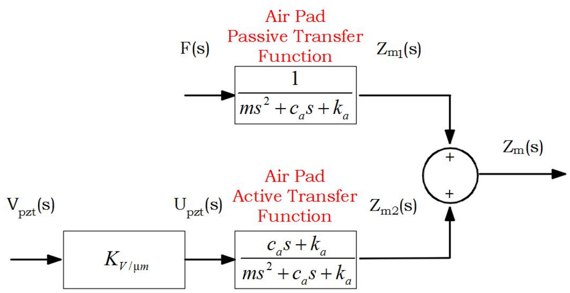


Fig. 3 The block diagram of the investigated system.

5 Identification

The aim of the presented experimental identification procedure is to assess the stiffness and damping coefficients (k_a and c_a) of the above described physical model in a neighbour of an equilibrium condition (P_s= 4 absolute bar and an air gap height of 10 μm). Figure 2(a) shows the scheme of the adopted test bench. The experimental identification was performed by imposing a step force to the moving part of the test bench (as described in Section 3) and measuring the time responses $z_{m,Air}(t)$ and $z_{m,NoAir}(t)$ which correspond respectively to the responses in presence and absence of the air pad supply. The power spectra of these two responses were compared in order to identify the first natural frequency of the air physical model. These spectra are shown in Figure 4 and they are obtained by using the MATLAB® pwelch function.

The first natural frequency ω_n of the system was identified at 683.6 rad/s whereas, other spectra are related to the test bench modes. Given m= 20.40 kg, it is possible to compute the stiffness coefficient of the air gap k_a as:

$$k_a = \omega_n^2 \cdot m = 9.531 \cdot 10^6 N/m \tag{3}$$

The air gap damping coefficient was computed estimating the damping ratio ζ by using the half-power bandwidth method (also called 3 dB method) [8]:

$$\zeta \simeq \frac{\Omega_2 - \Omega_1}{2\omega_n} = 0.04182 \tag{4}$$

$$c_a = \zeta \cdot 2\sqrt{k_a m} = 1.166 \cdot 10^3 Ns/m \tag{5}$$

where, Ω_2=711.3 rad/s and Ω_1=654.1 rad/s are the frequencies corresponding to a magnitude of -3dB with respect to the amplitude at the natural frequency ω_n. This method is a consolidate frequency domain method for damping estimation in presence of several different peaks in FRFs.

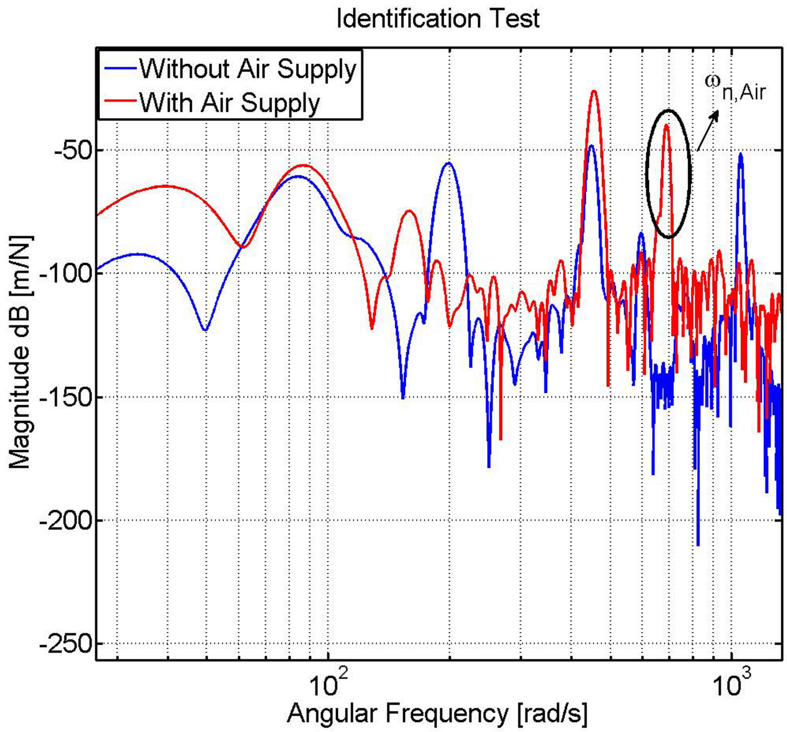


Fig. 4 The power spectra of the responses in presence and absence of the air pad supply and their difference.

6 Experimental Validation

The system model was validated by comparing bode diagrams of the theoretical and experimental FRFs of the active transfer function (see Figure 3):

$$G_{Active}(s) = \frac{Z_{m2}(s)}{U_{pzt}(s)} = \frac{K_{V/\mu m}(c_a s + k_a)}{ms^2 + c_a s + k_a} \tag{6}$$

The theoretical FRF was obtained through a MATLAB® LTI (Linear Time Invariant) model by considering the estimated k_a and c_a coefficients. On the other hand, the experimental bode diagram was obtained by imposing sine control voltage waves $v_{pzt}(t) = Asin(\omega t)$ as input to the system and evaluating the output $z_{m2}(t)$. The imposed sine waves have small amplitudes (A=10 V) in order to study the neighbourhood of an equilibrium condition and an excitation angular frequency ω . After this, the FFTs (Fast Fourier Transform) of $v_{pzt}(t)$ and $z_{m2}(t)$ signals were performed at each ω. Magnitudes were obtained by computing the ratio Abs[FFT($z_{m2}(t)$)] / Abs[FFT($v_{pzt}(t)$)] at the considered excitation frequencies. Similarly, the associated phases were assessed by computing the difference between the Angle[FFT($z_{m2}(t)$)] and Angle[FFT($v_{pzt}(t)$)]. Figure 5 shows both the theoretical and experimental Bode diagrams. As it appears from the results, the adopted system model is in good agreement with the experimental data.

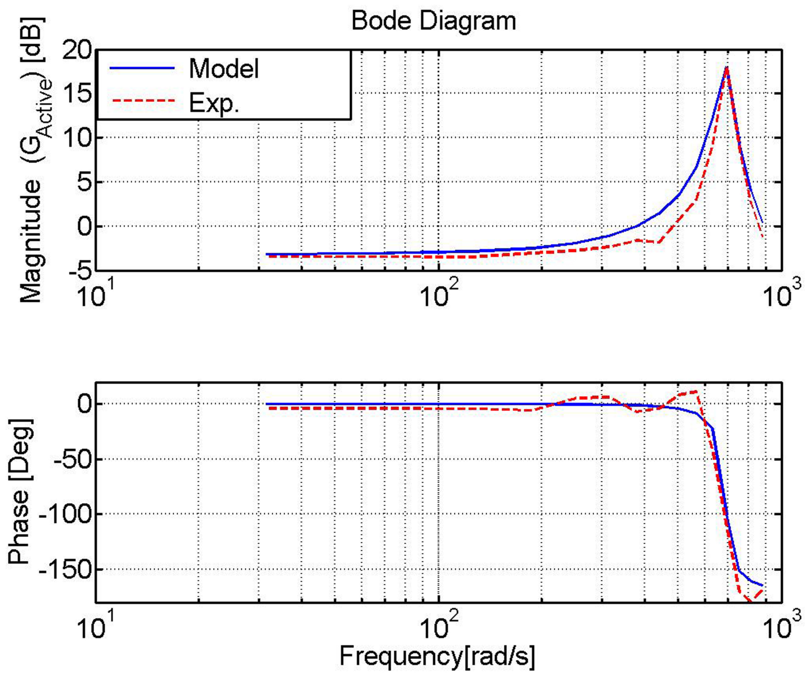


Fig. 5 The theoretical and experimental Bode diagrams.

7 Conclusion

This paper presents a physical model of an aerostatic thrust bearing. This simple LTI model results feasible in neighbourhoods of equilibrium positions. An experimental identification procedure has been developed in order to identify the system's stiffness and damping coefficients. This procedure consists in the application of a step force to the system both in presence and absence of the air supply and by computing the power spectrum of the responses. Firstly, the system's natural frequency was evaluated by comparing the computed spectra. Secondly, the air gap stiffness and damping coefficients were identified using the half power bandwidth method. Concerning the system modelling, the active transfer function $G_{Active}(s)$ was theoretically and experimentally by the respective Bode diagrams. The comparison shows that the physical model results are in good agreement with the experimental ones.

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