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Mechatronics Applied to Machine Elements with Focus on Active Control of Bearing, Shaft and Blade Dynamics

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Mechatronics Applied to Machine Elements with Focus on
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Lars Pallesen
Rektor

Martin P. Bendsøe
Dekan

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Dean

Preface

This work is based on the collection of papers from the period of 1994-2009. The theoretical and experimental investigations are based on my research work developed at three different universities: at the Technical University of Munich (Germany), at the State University of Campinas (Brazil) and at the Technical University of Denmark (Denmark).

The former Head of the Department of Mechanical Engineering at the Technical University of Denmark, Professor Preben Terndrup Pedersen and Head of Section Professor Viggo Tvergaard are gratefully thanked for their support, allowing my sabbatical at the Catholic University of Rio de Janeiro (Brazil) from September 2006 until January 2007, where this manuscript could start to be written.

I wish to thank Professor Friedrich Pfeiffer and Professor Heinz Ulbrich, from the Technical University of Munich, for their inspiring suggestions during the initial stage of my research work. I specially thank Professor Hans Ingo Weber, initially at State University of Campinas and later at Catholic University of Rio de Janeiro, for the pleasure of the technical cooperation and for many stimulating conversations, not only in the field of Rotordynamics, along the last twenty years. Thanks are also due to Professor Peder Klit, my closest colleague at the Technical University of Denmark, for the nice suggestions and discussions about some design solutions.

The term “mechatronics” has been used for about 40 years. It is derived from the observation of the synergy achieved through the integration of mechanical, electronic and information technologies. Practicing mechatronics obviously requires multidisciplinary expertise from mechanical, electronic, and information technologies to decision making theories. All my co-authors, master and Ph.D. candidates from the two research teams built by myself at State University of Campinas and Technical University of Denmark, are gratefully thanked for the fruitful synergy, among them, Fábio H. Russso, Alexandre Scalabrin, Rodrigo Nicoletti, Cristina M. Saracho, Flávio Y. Watanabe, Jonas T. Smith, Jacob Eiland, René H. Christensen, Niels Heinrichson, Klaus Kjølhede, Edgar A. Estupiñan, Martin Asger Haugaard and Stefano Morosi.

Dr.-Ing. Axel Fuerst (Alstom Power Ltd., Switzerland), Dr. Claus Myllerup (ODS-Llodys Register A/S, Denmark), the group of engineers led by M. Sc. Rogério Ribeiro Tacques (Oil Company Petrobras S.A., Brazil), Dr. Dexter Johnson (Structural Systems Dynamics Branch at NASA Glenn Research Center, USA) and Professor Minel J. Braun (University of Akron, Ohio, USA) are also kindly thanked for the motivating discussions about the feasibility of industrial applications of the controllable fluid film bearings, in different stages of this work.

Experimental work demands patience and determination, and it is normally extremely time consuming, involving design, calibration, modifications, re-calibration, until final tests can be performed. Nevertheless, “theoretical and experimental works” can be compared to the “two wings of a bird”. The careful development and strengthening of both wings, the “experimental”

and the "theoretical", allows us flying higher and more stable inside of our research field. It helps us understanding more deeply what is important and what is not while developing theories and mathematical models for representing the dynamics of machines and structures. Hereby I would like to thank the technicians from the three different workshops whom I have worked with, for their invaluable help and support during the different phases of the experimental work: Julius Kriese, Georg Mayr, and Walter Wöß from the Technical University of Munich; Eli P. Souza, Mauricio O. Santanna, Rosangelo W. A. Ferreira and Almiro F. Silveira Jr. from the State University of Campinas; Torben B. Christensen, Per Bo Nielsen, Viggo Nielsen and Mogens P. Frank from the Technical University of Denmark.

Carrying out experimental work demands also economical backing. I would like to thank all the research agencies which have sponsored the experimental work along the years: Deutsche Forschungsgesellschaft – DFG (Germany); The State of Sao Paulo Research Agency – FAPESP, National Council for Scientific and Technological Development – CNPq, Foundation for High-Education and Human Resource Development of Brazilian Ministry of Education – CAPES (Brazil); Myhrwolds Foundation, Toubro Foundation and Otto Mønsted Foundation (Denmark).

I thank my two best friends in Denmark, Simon Bjerrum Lidegaard and Jakob Thyssen Valerius, for their support and for always being on my side.

Finally, I dedicate this work in memorial of my father, an example of simplicity and humbleness, and to my mother, an example of perseverance and dedication.

Rio de Janeiro, January 20, 2009.

Ilmar F Santos

Ilmar Ferreira Santos

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Chapter 1

Introduction

Rotating machines like turbo-generators, compressors, turbines, and pumps, are often vital elements in the production process, for example in the oil and gas industry. Therefore, these machines must have not only high performance, but also high availability. In many situations, this availability depends on the adaptation capability of the machine to fast changing demands. Moreover, the need of rotating machines of high efficiency, working at severe pressure and flow condition, demands continuous monitoring and control of vibration levels. Due to aerodynamic excitations during full load condition (high maximum continuous speed), it is difficult to ensure a reasonable stability margin for such machines.

Furthermore, while designing such rotating machines, one of the most useful norms, namely the API 617 [46], becomes hard to satisfy, even when parameter optimization techniques are applied Ahn et al. [37]. In order words, such machines have to operate at higher angular velocities and, simultaneously, be as light as possible. Such a combination of design demands leads to severe vibration problems. The flexibility of their components can not be neglected. Thus, the development of new machine components for dissipating vibration energy is of fundamental importance in order to keep low levels of machine vibration.

This claim can be easily verified by checking the number of new passive and active elements under development stages in the last 20 years. Among the passive elements the squeeze-film dampers cited by Thomsen and Andersen [255], Thichy [256], San Andrés and Lubell [224], Pietra and Adiletta [203] [35], Defaye et al. [84] and the seal dampers investigated by Vance and Li [266] are the most important ones. Among the active and semi-active elements (mechatronic components) one can cite: the hybrid squeeze-film dampers in El-Shafei and Hathout [95], the controllable squeeze film damper using electro-rheological fluids in Nikolajsen and Hoque [188], Morishita and Mitsui [180], Jung and Choi [151], the controllable squeeze film damper using magneto-rheological fluids in Zhu et al. [279] [280], the active hydraulic chamber systems connected to different types of bearings in Ulbrich and Althaus [259], Althaus [42], Santos and Ulbrich [226], Santos [225], the variable impedance hydrodynamic journal bearings using accumulators in Goodwin et al. [125], the piezo-controlled bearings in Palazzolo et al. [196], Alizadeh et al. [40], the combined piezoelectric-hydraulic actuators in Tang et al. [253], the active lubricated bearings in Santos [1] [3] [6], the active-controlled fluid bearings in Bently et al. [53] [54], Peltier [200], the magnetized journal bearings lubricated with ferro-fluids in Osman et al. [194], the disk-type electro-rheological dampers in Vance and Ying [267], Vance et al. [268], the disk-type magneto-rheological fluid dampers in Zhu et al. [281], the foil-magnetic hybrid bearings in Heshmat et al. [140] and bearings composed of flexible sleeves controlled by means of hydraulic chambers in Krodkiewski and Sun [161] [248].

1.1 State of the Art

The relative maturity of many traditional technologies within different technical areas, implies little or no potential for significant improvements of machine performance. Consequently, to deal with the technical challenges related to some new requirements of safety, quality, reduced weight, low vibration and noise levels, non-conventional techniques have to be introduced where the conventional ones have already reached their limits, Olsson [192]. This is especially true in the field of conventional hydrodynamic bearings. Thus, the combination of tribology, control techniques and informatics (mechatronics) enables the development of "smart" bearings able to deal with multi-objective functions, not only the function of supporting a rotating shaft. The synergy among different sub-areas of mechanics, control techniques and informatics enables significant improvements and balancing of various contradictory properties, closely associated to one or several interactive subsystems. Moreover, mechatronics has proven to be able of overcoming limitations associated with the traditional "passive" solutions.

One of the most attractive features of mechatronics is its ability to render products "smart". Such a smart machine makes use of the built-in active control to incorporate additional or higher performance functions. Thus, the machine may acquire higher precision and the ability for self-diagnosis. The machine can calibrate itself. It can give a prognosis about its future ability to function in a satisfactory way, or about its remaining lifetime, and possibly, it could suggest a correction measure, a "therapy", or even induce it itself. It is the mechatronic structure of the machine, the built-in control, its sensors, processors, actuators, and above all, its software, which enable these novel features. This is a way to design machines and products with higher performance, less maintenance costs, longer lifetime, and an enhanced customer attraction Schweitzer [233]. Mechatronics has made major impacts in a variety of industries such as automotive Hiller et al. [142], consumer electronics, biomedical and robotics/automation. Research topics relevant to mechatronics are diverse and include actuators/sensors, microelectromechanical systems (MEMS), mechatronic devices/machines, control of mechatronic systems, human-machine interface/haptics, embedding computing and software engineering and design/integration methodologies for mechatronic systems.

From the viewpoint of rotating machines one of the most well-known mechatronic component is the active magnetic bearing Schweitzer et al. [232]. The concept of magnetically suspending a rotor began to receive serious attention in the mid-nineteenth century. U.S. patents dating back to the early 1800's incorporated the use of "magnetic bearings" to support rotating shafts, reduce mechanical friction, and improve the efficiency of electrical meters. In 1842, Ernshaw analytically demonstrated the inherent instability in totally passive (permanent) magnetic bearing systems and presented the requirement of at least one active degree of freedom to achieve rotor stability Moses et al. [181]. Nevertheless, since the beginning of the nineties a spinning top is offered in toy shops. This top is able to rotate for minutes over a permanent magnetic base plate - contactless. The interaction of the six degrees of freedom of this fast top with the magnetic field is discussed in details in Gash and Lang [117] with the classical methods of rotor dynamics. Surprisingly linearized differential equations are sufficient to find the stable zone of rotational speeds.

The advancements in magnetic materials, servo circuitry, and power electronic components have provided many enhancements to the magnetic bearings technology. Today private corporations and institutes around the world are applying magnetic bearing devices as diversified as centrifugal compressors Forster et al. [111] Dell et al. [85], turbogenerators Canders et al. [73], helium circulators, x-ray tubes, high vacuum pumps Moses et al. [181], machine tools spindles Brunet

[65], rocket engine turbopumps Girault [122], space guidance Bichler and Eckhardt [55] and energy storage flywheels Nakajima [184]. While active magnetic bearings have a relatively long history of development and tests and their industrial application to rotating machines is nowadays a well-established technology, the development of new types of mechatronic components, as those previously listed, has a shorter history and is still "creeping" in laboratories of universities and companies.

1.2 Motivation of the Work

The development of active lubricated bearings (ALB) and bearings controlled by means of active chamber systems, also typical mechatronic devices, has only a little more than one decade of history. There are obviously advantages and drawbacks when compared to active magnetic bearings, depending on the application purposes. Research activities have shown the potential of such new mechatronic devices and this work is intent to give an overview about the new achievements in this promising field, with special emphasis on active lubricated bearings and bearings actively controlled by means of hydraulic chamber systems. Active magnetic bearings and magnetic actuators are also presented. The focus of the research is though on the feasibility of measuring and calibrating magnetic forces with high accuracy to used active magnetic bearings and magnetic actuators as calibrated shakers toward model parameter identification and fault diagnose in machines.

Active elements, such as active lubricated bearings, bearings controlled by means of active chamber systems, active magnetic bearings and magnetic actuators are used to control the static and dynamic properties of machine elements as bearings, rigid and flexible shafts and flexible rotating blades. The main theoretical and experimental achievements are summarized and the advantages and limitations of such a new devices are elucidated, with focus on the vibration control of flexible machine elements, i.e., shafts and blades. The needs of further development, the feasibility of industrial application and trends are also discussed.

With such active elements it is intent to

- control of the lateral vibration of flexible rotating shafts and flexible rotating blades;
- modify bearing dynamic characteristics, as stiffness and damping properties;
- increase the rotational speed ranges by increasing damping and eliminating instability problems, e.g., by compensating cross-coupling destabilizing effects.
- reduce simultaneously start-up torque and energy dissipation in the bearings;
- compensate thermal effects;
- develop "smart" components able to avoid unexpected stops of plants;
- identify model parameters "on site" with help of active bearings used as a calibrated shaker.

1.3 Structure of Chapters and Original Contribution

This manuscript is divided into five chapters and an appendix. In the first chapter a short introduction, a brief state-of-the-art, the motivation of the present work and the structure of the following chapters are presented.

In chapter 2 the main contribution is of experimental nature. The chapter is devoted to the description of six different mechatronic systems, designed and built with the goal of verifying the feasibility of adjusting bearing static and dynamic properties and controlling the dynamics of rigid and flexible rotating shaft and blades. Moreover, the test rigs are used to validate the mathematical models developed to predict the dynamic behavior of the different mechatronic systems. The six different mechatronic devices are: (I) oil film bearings controlled via active chamber systems, (II) oil film bearings controlled via active lubrication mechanism – injector connected laterally to tilting-pads, (III) oil film bearings controlled via active lubrication mechanism – injector connected radially to tilting-pads, (IV) oil film bearings controlled via active lubrication mechanism – injector connected radially to active pockets, (V) active magnetic bearings with slot and hall sensors, and finally (VI) active rotor-blade system controlled by magnetic actuation. Advantages and drawbacks of the different mechatronic devices are critically discussed.

In chapter 3 the main original contribution is of theoretical nature. The first part of this chapter has its focus on the modeling of oil film bearings, rotor-blade coupled dynamics and rotordynamics. The derivation of the modified Reynolds equation for active lubrication and the rotor-blade models with different levels of complexity are generally explained. All details about the derivation of such mathematical models are explained in the appendix, where the thesis papers can be found. The second part of this chapter is devoted to control system design for actively attenuating vibrations of machine elements, as rigid and flexible shaft and flexible rotating blades. The design of centralized and decentralized controllers for flexible rotating shafts mounted on active lubricated bearings is presented. Advantages and drawbacks of both type of controllers are critically discussed. The design of controller for rotor-blade systems capable of dealing with time-variant parameters are also explained based on time-variant modal analysis.

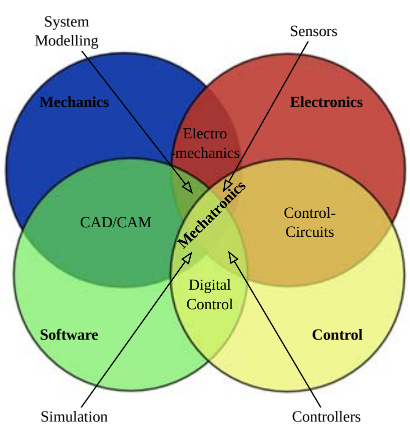
In chapter 4 an overview of the main theoretical and experimental achievements are presented and critically discussed. The feasibility of industrial application of such new mechatronic devices is elucidated and challenges to be overcome are highlighted. In chapter 5 the main conclusions are addressed and future aspects of the research activities are outlined.

Finally, in appendix, a collection of 33 articles are included, 25 journal papers and 8 international conference papers. All of them are reviewed by international referees and published in different technical books and journals in the U.S., the U.K., Germany and Brazil. The last article "Trends in Controllable Oil Film Bearings" is based on an invited keynote presented by myself at the *IUTAM Symposium on Emerging Trends in Rotor Dynamics*. Such an article is intended to foresee "the future of the controllable oil film bearings and the smart machines" and draws up a plan for new research activities in the field. All topics, briefly presented in the chapters 2, 3 and 4 of this manuscript, are detailed in the 33 articles. It is worth highlighting that the manuscript is written to give the readers an overview of the original achievements of the work inside a wide international context.

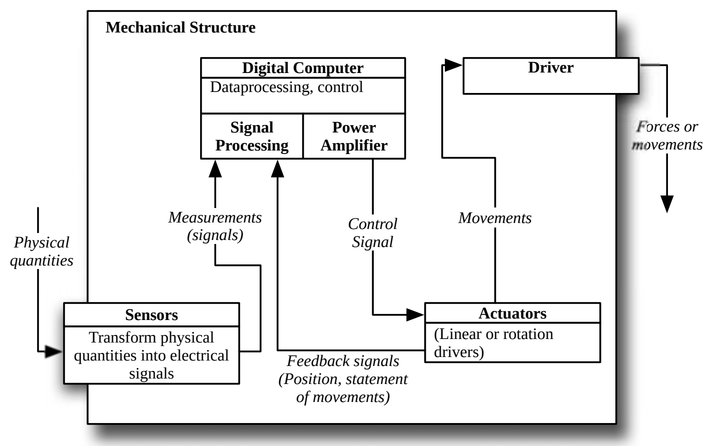
Chapter 2

Design of Mechatronic Systems – Experimental Work

"Mechatronics is the synergetic integration of mechanical engineering with electronic and intelligent computer control in the design and manufacturing of industrial products and processes" (IEEE/ASME Transactions on Mechatronics 1996). Mechatronics is an interdisciplinary area of engineering based on disciplines from mechanical engineering, electrotechnics and informatics. Figure 2.0.1(a) illustrates its schematically. A typical mechatronic system acquires electrical signals from sensors and transducers, treats and processes them and sends them out to actuators and drivers, so that such signals are transformed into forces and movements as shown in figure 2.0.1(b).



(a)



(b)

Figure 2.0.1: (a) Mechatronics – synergy of different areas and disciplines; (b) basic structure of a mechatronic system.

Significant contributions in the literature Bremer [64] Schweitzer [231] Isermann [146] Pelz [201] Nordmann [190] Roddeck [212] give inspiring overview about the development of mechatronics along these 40 years, since the term "mechatronics" was defined in Japan in 1969 by Mr. Tetsuro Mori, a senior engineer of the Japanese company Yaskawa. New electro-mechanical systems for mechanical engineering are nowadays developed in an integrated form as self-sufficient mechatronic systems. A recent overview about the development of new actuators is given in Janocha [148], where material properties can be properly changed by means of electronics. Actuators based on electro-rheological and magneto-rheological fluids, piezo-electrical material properties, electro-magnetic principles as well as materials with shape memory are cited in Janocha [149]. The trends in actuator design for application to control rotating machines are very effectively described in Ulbrich [263] Ulbrich and Bormann [264]. The advantages and drawbacks of different types of actuators, i.e., magnetic, hydraulic and piezoelectric, are carefully described in Ulbrich [260].

As mentioned before and illustrated in figure 2.0.1(a), mechatronics is an intersection of different areas and disciplines. The basic structure of a mechatronic system is also schematically presented in figure 2.0.1(b). Such a structure will be recognized and carefully described in all devices described in the next sections, namely:

- (I) oil film bearings controlled via active chamber systems;
- (II) oil film bearings controlled via active lubrication mechanism – injector connected laterally to tilting-pads.
- (III) oil film bearings controlled via active lubrication mechanism – injector connected radially to tilting-pads.
- (IV) oil film bearings controlled via active lubrication mechanism – injector connected radially to active pockets.
- (V) active magnetic bearings with slot and hall sensors;
- (VI) active rotor-blade system controlled by magnetic actuation.

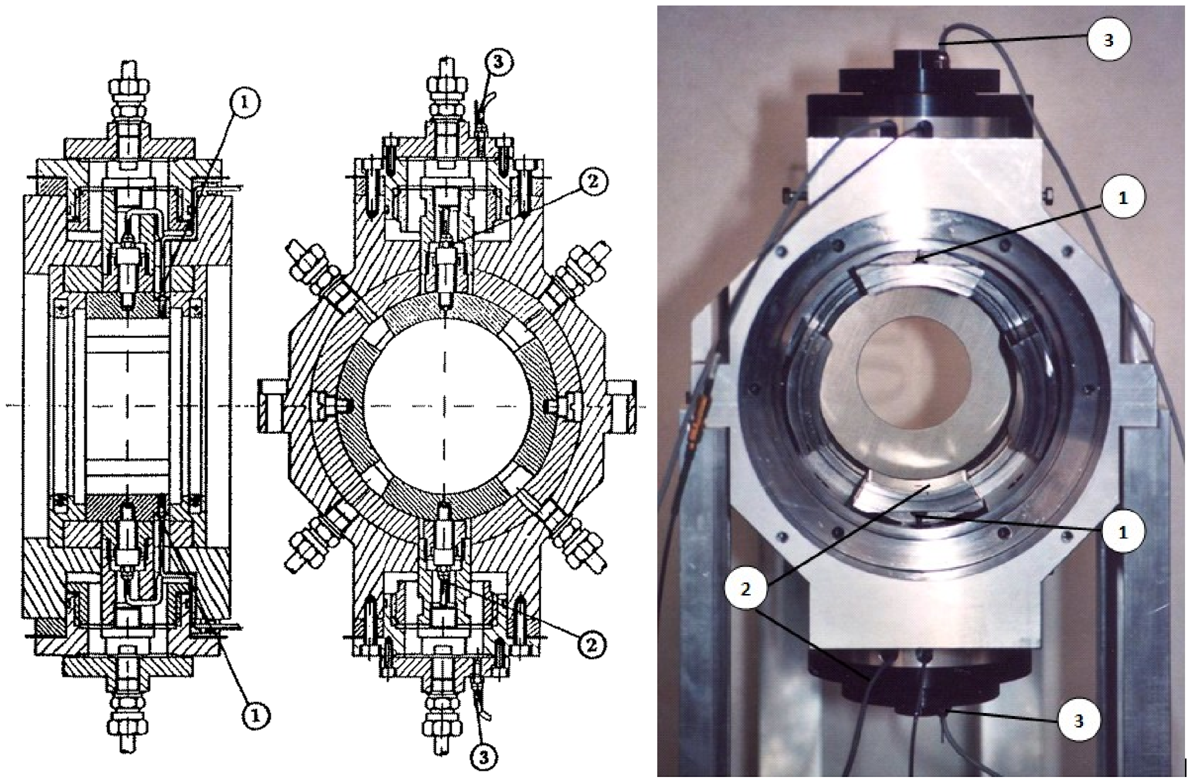
A description of these mechatronic devices are presented in the following sections.

2.1 Oil Film Bearings Controlled via Active Chamber Systems

Tilting-pad journal bearings are normally used in high-speed machineries, particularly in applications where plain cylindrical bearings might present problems with self-induced vibrations. The static and dynamic behaviors of such bearings have been investigated by many authors since 1953 Boyd and Rumondi [57] Lund [167]. Tilting-pad journal bearings have less damping than predicted by the theoretical models Lund [167] Malcher [172] Klumpp [155] Glienicke [123] Lund [169] Earles et al. [93] [94] Chan and White [75] White [270] Parkins and Horner [198] Pagano et al. [197] Monmousseau and Fillon [179] Guijosa and Feng [127]. A further increase in damping and stabilization can be obtained when these bearings operate actively.

The first ideas about an active tilting-pad bearing were presented in Ulbrich and Althaus [259], where the pads are actively controlled through piezoactuators. The first theoretical and experimental investigation about these active bearings, using though hydraulic actuators connected to the bearing pads, can be found in Santos and Ulbrich [226] Santos [225]. Further theoretical investigations of active tilting-pad bearings have been carried out later by other authors Al-Issa and Forte [39] Deckler et al. [81] [82] [83] Cai and Khonsari [70]. In the last five contributions, different control strategies are developed when piezoelectric actuators are used to control tilt angles or translations of the pads.

Figure 2.1.1 illustrates the design of the oil film bearing controlled by hydraulic chamber systems. The oil film bearing under investigation is a tilting-pad bearing with four tilting-pads, two in the vertical direction and two in the horizontal direction. The active tilting pad bearing possesses only two active pads arranged in the vertical direction. These pads are attached to the chamber system, without hindering the tilting motion. The chamber system is constructed with one membrane per chamber and connected to a servo valve and a proportional valve. The displacement sensors **1** allows the measurement of the relative displacement between the center of the rotor and the pads. With the pressure sensor **2** it is possible to measure the pressure in the journal bearing and with the sensor **3** the pressure in the chamber system. All these signals can be used in designing feedback control laws.



(a) (b)

Figure 2.1.1: Oil film bearings controlled via active chamber systems – (a) scheme illustrating the chamber system and sensing system built by displacement sensors **1** for measuring relative movement between rotor and pads, pressure sensors **2** for measuring oil film pressure and pressure sensors **3** for measuring the hydraulic pressures in the chambers; (b) picture of the bearing demounted illustrating the sensing system.

The first main objective of this mechatronic component is to adjust the dynamic coefficients of a tilting-pad bearing and obtain a significant reduction of the vibration amplitudes of rotor-bearing systems. The damping and stiffness coefficients of an oil film bearing depend on the thickness of the oil film. Through the proportional valve the static pressures in the chamber system are modified. This produces a deformation of the elastic membranes of the chamber. The bearing pads attached to the membranes are moved and the oil film thicknesses between rotor and pads are consequently modified. It is important to highlight that the adjustment of the bearing coefficients are achieved without feedback control, using only the non-linearities of the the bearing coefficients, namely their dependency of the bearing gap. The main results are presented in Santos [2] and [4].

Further increase in damping and stabilization is possible by using control techniques. In such an operational mode the device operates fundamentally as an active system. By measuring a) the lateral displacement of the rotor, b) the pressure in the chambers and c) the pressure in the oil film, different types of feedback control laws can be defined and different types of controllers can be designed. The control signal is then generated as a linear combination of the signals a), b) and c) and sent to the servo valve as input signal. It means that the hydraulic pressure in the chambers and the membrane deformations are dynamically modified, resulting in active forces able to control rotor lateral vibrations. The main results are presented in Santos [1] and [3].

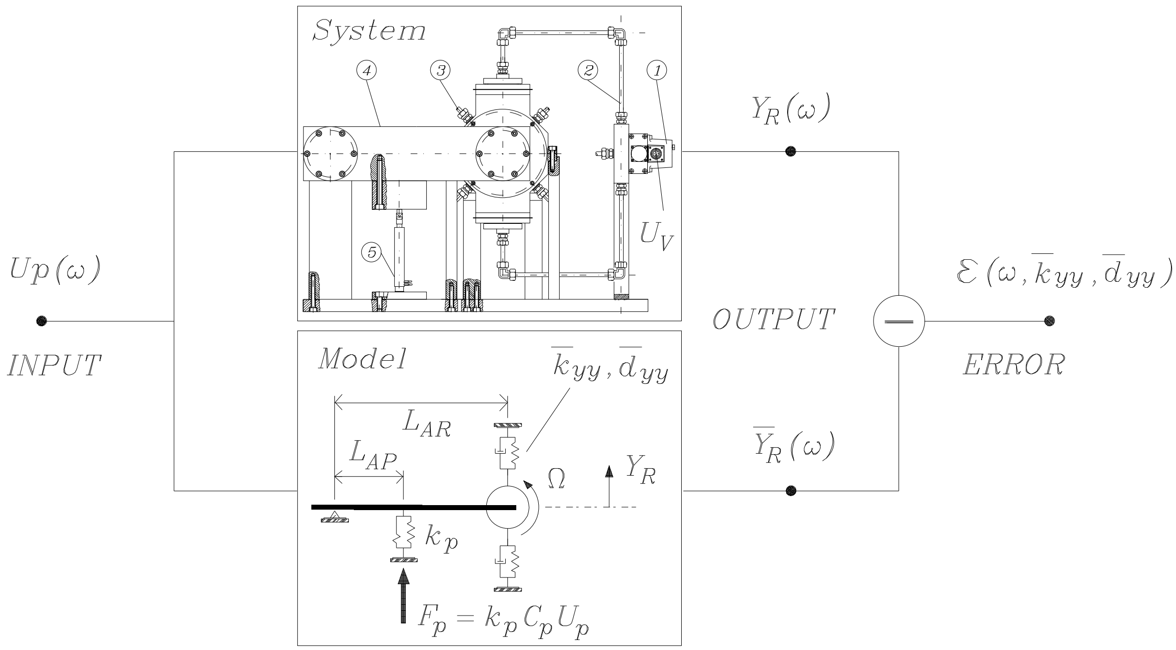


Figure 2.1.2: Scheme of the operational principle of the test rig for experimental investigation of the oil film bearing controlled by active chamber systems – System composed of: **1** servovalve, **2** pipelines connected to the chamber system (high pressure) , **3** oil supply to the tilting-pad bearing (conventional hydrodynamic lubrication), **4** rotor-lever- system, **5** piezo-actuator; Model built by a rigid rotor-lever system connected to adjustable springs and dampers (oil film coefficients) and excited by the piezoactuator force F_p.

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Figure 2.1.2 illustrates a scheme of the operational principle of the test rig used to evaluate the bearing performance. For the tilting-pad journal bearing under discussion the cross-coupling coefficients of damping and stiffness can be neglected. This characteristic is extensively mentioned in the literature Brockwell and Dmochowski [62] Glienicke [123] Someya [240]. In the design of the test rig this information was taken into account and, for this reason, the experimental analysis could be carried out only in one direction, namely the vertical. The motion of the center of the rotor Y_R is constrained in the horizontal direction by means of a lever system. The rotor-lever system is excited in the vertical direction with help of a piezoelectric actuator. The electrical excitation signal U_p is generated by means of a signal generator and used as input signal for the piezoelectric actuator. The piezoelectric actuator force is represented by $F_p = k_p \cdot C_p \cdot U_p$, where k_p is the piezoelectric actuator stiffness and C_p is its piezoelectric constant. The displacement obtained by the piezoelectric actuator is amplified at the rotor location with help of the lever system L_{AP} and L_{AR}. The rotor-lever movements are then controlled by an active bearing, where the control signal U_v is the servo valve input.

In figure 2.1.3 a picture of such a test rig is shown. The main typical elements of the mechatronic system are highlighted. A parallel between figure 2.1.3 and figure 2.0.1 presented in the introduction of this section can be easily made.

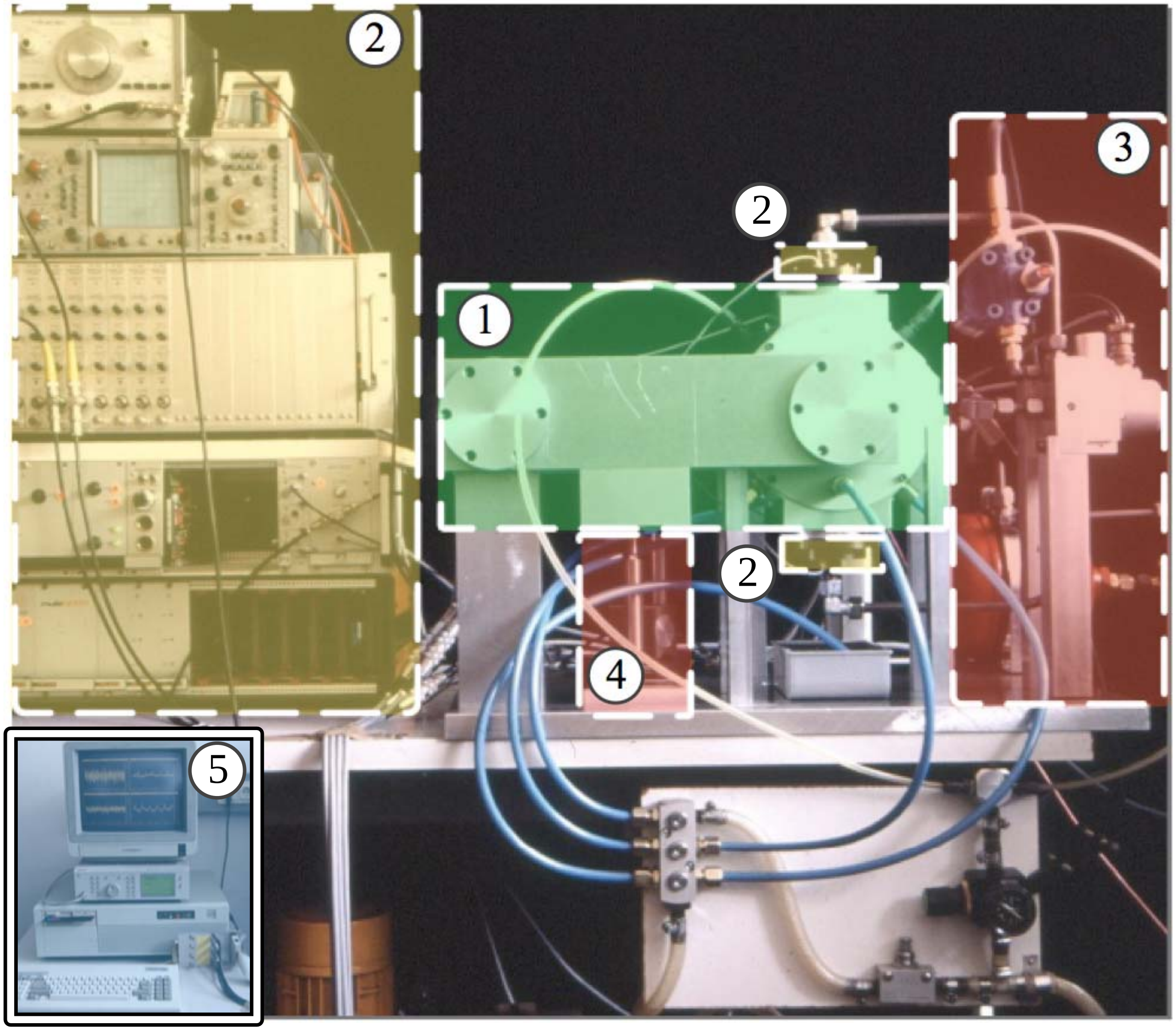

Figure 2.1.3: Mechatronic system – Test rig for experimental investigation of the oil film bearing controlled by active chamber systems; **1** – Mechanics: rotor-level system pivoted on the left hand side. **2** – Electronics: power supply and amplifiers of the displacement and pressure sensors mounted in the bearing interacting with the journal bearing controlled via active chamber systems. **3** – Mechanics: servovalve, accumulator and pipelines. **4** – Electronics: piezoactuator used to excite the rotor-level system. **5** – Control and Software: signal processing and control unit (PC-computer).

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The main advantages of the design are the capability of controlling rotor lateral vibrations by actively moving the pairs of pads in orthogonal directions. The amplitude of such movements will be dictated by the threshold strain on the membranes. The membranes can be mounted in parallel allowing large pad movements. Another advantage is the possibility of eliminating self-excited vibrations and flutter by changing the bearing gap, as investigated theoretically and experimentally in Santos and Nicoletti [5]. Nevertheless, the deformation and threshold strain on the membrane of the chamber system limit the modification of the oil film thickness. It is known that, in case of rotating systems operating at very high speeds, the stiffness of the oil film is high and the damping level is very low. To change the oil film thickness, it is necessary to have high pressure values in the chambers. Due to stress and strain limitations in the actuator membranes, such high pressures become unpractical, Santos [2]. An alternative approach is to act directly on the flow behavior of the lubricant fluid through tilting-pad journal bearings with active oil film, Santos [3]. Such a operational principle is presented in the next section.

2.2 Oil Film Bearings Controlled via Active Lubrication Mechanism

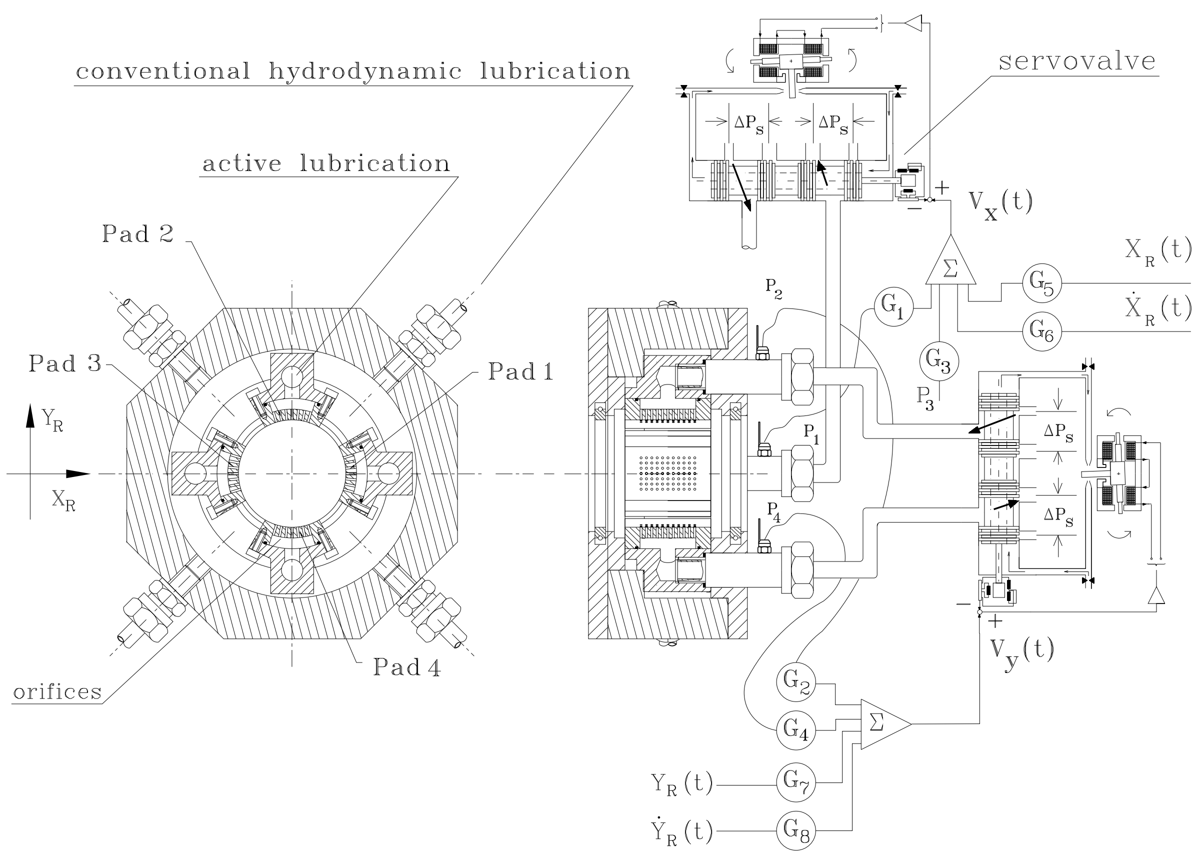
When the hydrostatic and the hydrodynamic lubrication are simultaneously combined in a journal bearing, with the aim of reducing wear between rotating and stationary parts, one refers to the hybrid lubrication, which offers the advantages of both lubrication mechanisms. When part of the hydrostatic pressure is also dynamically modified by means of hydraulic control systems, one refers to the active lubrication or active oil film[1]. By the combination of fluid power, electronics and control theory, the active lubrication makes simultaneously feasible the reduction of wear and the attenuation of vibration. The direct influence on the oil flow behavior in the bearing gap is achieved by machining orifices along the bearing surface or pad surface and connecting them to two servo valves via pipelines. The servo valves enable changes of the fluid injection pressure directly in the bearing gap. By changing the flow characteristics by means of the electronic injection, one attempts to control the rotor movements, modify the stiffness and damping of the oil film or induce controllable shaft movements.

2.2.1 Injectors Connected Laterally

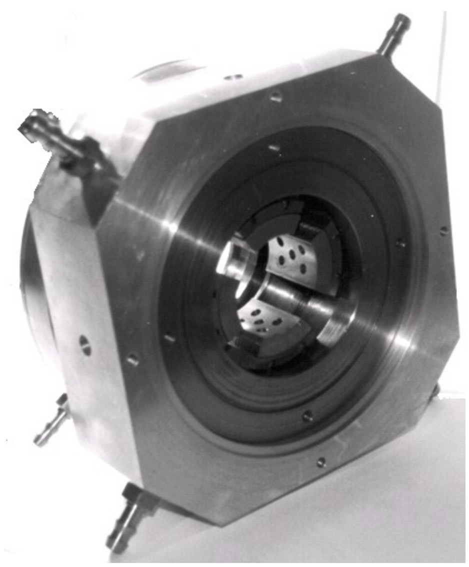
Figure 2.2.1(a) shows the operational scheme of two distinct mechanisms of lubrication applied to a special type of journal bearing, namely the tilting-pad journal bearings. It is possible to see the conventional passive hydrodynamic lubrication between pads and the active lubrication through the orifices machined over the four tilting-pad surfaces. Figure 2.2.1(b) illustrates the designed tilting-pad bearing with four pads and the five orifices machines over each pad surface. Figure 2.2.1(c) shows the oil injectors connected laterally to the bearing pads and bearing housing. Two servo valves controlled by the electrical signals $V_X(t)$ and $V_Y(t)$ are connected to the pair of pads in the vertical and horizontal directions by pipelines. A differential principle is used. Through the orifices along the pad surfaces the oil is injected with pressures $P_1(t)$, $P_2(t)$, $P_3(t)$ and $P_4(t)$, which can be measured by pressure sensors mounted at the pipelines. These injection pressures are controlled by the electrical signals V_X and V_Y. Such control signals are generated as a combination of measurement signals, coming from sensors mounted in the rotor-bearing system.

The goal of the control system is among others to influence the lateral rotor motions, increasing the oil film damping when the gain coefficients ($G_i \quad i = 1, ..., 8$) are properly chosen. For instance, the input signal $V_Y(t)$, which acts on the vertical rotor displacements, is defined as a combination of the pressure signals $P_2(t)$ and $P_4(t)$ and the rotor vertical displacement and velocity signals, $Y_R(t)$ and $\dot{Y}_R(t)$, measured by displacement and/or velocity sensors. The constants G_2, G_4, G_7 and G_8 are the gain coefficients of the control law. These coefficients can be chosen, adjusted, calculated and optimized with the aim of lubricating the sliding surfaces and reducing rotor vibrations at the same time. The input signal $V_X(t)$, which influences the horizontal rotor displacements, can be similarly defined, using the pressure signals $P_1(t)$ and $P_3(t)$ and the rotor

[1]Brazilian Patent: "Mancal Segmentado com Filme de Óleo Ativo", SEDAI:18.566, INPI 9403084-7



(a)



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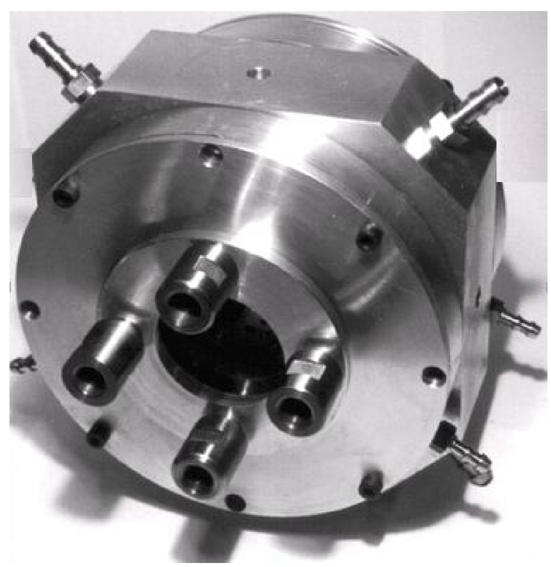


(c)

Figure 2.2.1: (a) Scheme of a tilting-pad journal bearing controlled connected to servo valves by means of pipelines; (b) picture of the open bearing with 4 pads and 15 orifices machined on the pad surface; (c) picture of the injectors connected laterally to the pads and bearing housing.

horizontal displacement and velocity signals $X_R(t)$ and $\dot{X}_R(t)$ and choosing appropriate values for the gain coefficients G_1, G_3, G_5 and G_6.

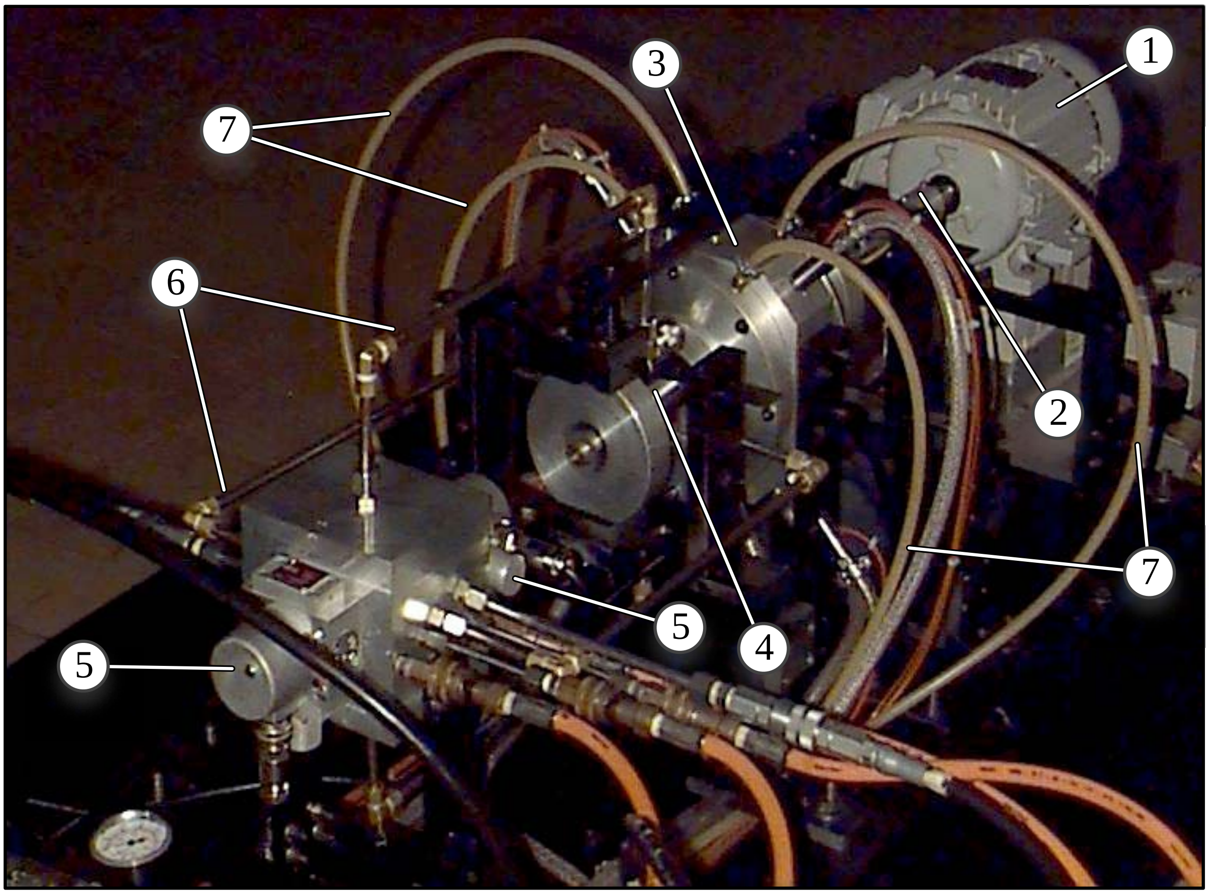


Figure 2.2.2: Test rig used to experimentally evaluate the dynamic performance of active lubricated tilting-pad journal bearing: **1** – motor, **2** – flexible coupling, **3** – tilting-pad bearing with active oil film, **4** – rigid shaft, **5** – servo valves, **6** pipelines (high pressure up to 220 bar – control), **7** pipelines (low pressure up to 3 bar – conventional hydrodynamic lubrication.

The test rig designed to experimentally evaluate the performance of such an active bearing can be seen in figure 2.2.2. The rigid shaft **4** is driven by an electric motor **1**. The rigid rotor is connected to the motor using a flexible coupling **2**. The rotor is supported by flexible mountings at its extremities. The rotor is supported in the middle by the active lubricated tilting-pad journal bearing **3** described in details and illustrated in figure 2.2.1. The oil supplied through pipelines **7** will generate the conventional hydrodynamic lubrication, whereas the pressurized oil from the servo valves **5** through pipelines **6** will generate the active lubrication.

The most important advantages in applying such a type of active bearing to rotating systems are: the reduction of lateral vibrations of machine rotating parts; the increase in the stable operational speed ranges; the possibility of modifying bearing coefficients and improvement of the damping characteristics. Moreover, the reduction of the required startup torque of the motor due to the elimination of dry frictions between rotor and pads by means of the radial oil injection before the rotor starts, and the compensation of thermal effects are also possible.

Conversely, the main disadvantage is the reduction of bearing load capacity, specifically in the case of control system failure. Long pipelines (elements **6** in figure 2.2.2) lead also to limitation of the operational frequency ranges. Some of these limitations are eliminated in the "second generation" of active lubricated tilting-pad journal bearings described in the following section.

2.2.2 Injectors Connected Radially

The main advantage of the bearing illustrated in figure 2.2.3(a) is the compactness toward industrial application. The long pipelines connecting the servo valves and oil film gaps are significantly reduced by machining channels inside the bearing housing, as illustrated in details in figure 2.2.3(b).

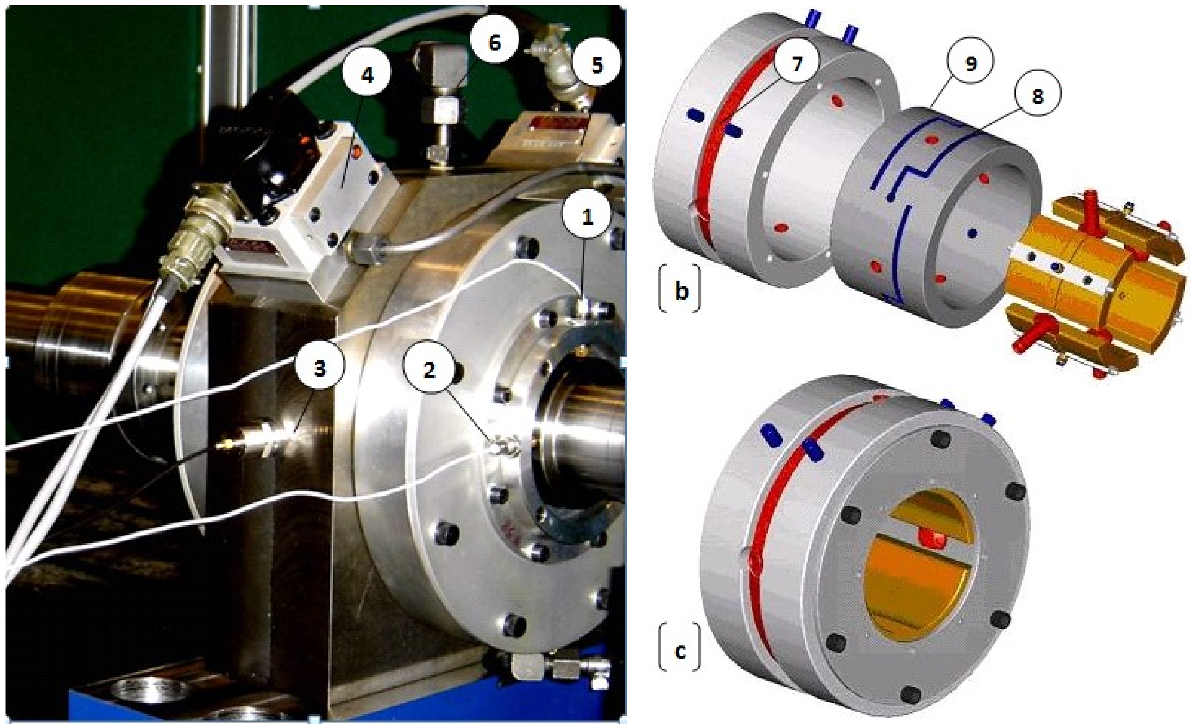
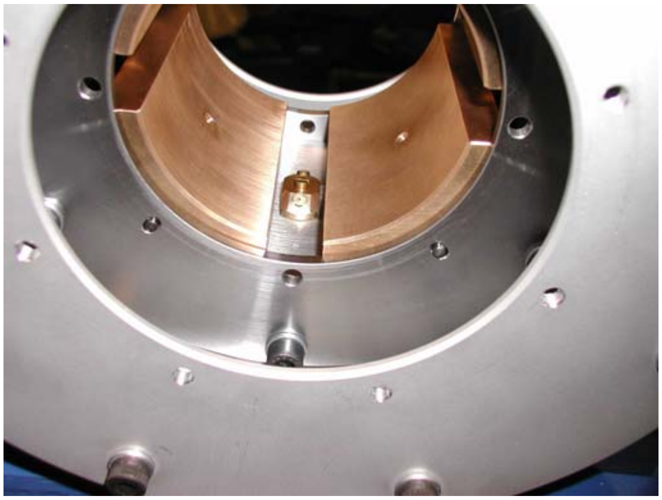


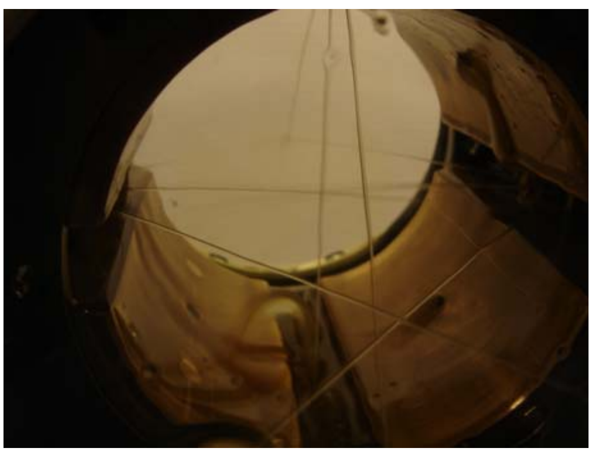
Figure 2.2.3: Active lubricated bearing – (a) active lubricated bearing as a mechatronic device: **1** – displacement sensor (eddy-current) to measure rotor vibrations in the vertical direction; **2** – displacement sensor (eddy-current) to measure rotor vibrations in the horizontal direction; **3** – accelerometer to measure housing vibrations; **4** and **5** – servo valves; **6** – oil supply pipeline (conventional hydrodynamic lubrication). (b) exploded view of the bearing illustrating the channels machined inside of the housing with the aim of achieving a compact bearing: **7** channel build circumferentially to feed oil between the pads (conventional hydrodynamic lubrication); **8** and **9** – channels inside of the bearing housing connected pairwise with the the servo valves (active lubrication). (c) 3D-view of the bearing as a mechanical component.

The “second generation” of active lubricated bearing as a mechatronic device is shown in figure 2.2.3(a). The displacement sensors **1** and **2** capture the lateral vibrations of the rotor in the vertical and horizontal directions respectively. The accelerometer **3** mounted on the the bearing housing detects bearing housing vibrations in the horizontal direction. The servo valves **4** and **5** are used to control the pressurized oil injection. The pressurized oil flows through the channels machined inside the bearing housing, namely channels **8** and **9**, through the orifice machined in the middle of the pads until the bearing gap. Such a pad orifice are normally used in industrial bearings for generating the hydrostatic lubrication during startup conditions. The channels **8** and **9** connected to the servo valves **4** and **5** by a differential principle make possible the generation of time-dependent control forces. The oil injection will occur with an angle of 45^o from the horizontal or vertical directions because the pads are mounted in a load-between-pads configuration. The oil injection will be controlled using the information coming from the sensor signals **1**, **2**, **3** and others mounted along the flexible rotating shaft. The bearing can operate passively as well as actively. The (conventional) hydrodynamic lubrication will be always turned on and will be created by feeding the oil with low pressure (up to 2 bars) via the pipeline connector **6** and the channel **7**. The oil will achieve the bearing gap through holes machined between the four pads. The orifices are machined (90^o from each other) connecting the channel **7** and the space between the four tilting-pads.



(a)

(b)



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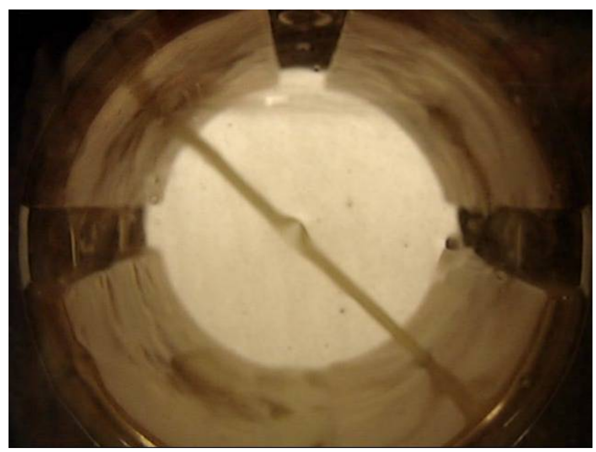

(d)

Figure 2.2.4: (a) Interior of the active lubricated tilting-pad bearing, where it is possible to see the orifice machined in the middle of the pad surface (active lubrication) and the injector between pads for the conventional lubrication (hydrodynamic lubrication); (b) compactness of the active lubricated tilting-pad bearing composed of 4 pads and two servo valves mounted on the bearing housing; (c) conventional lubrication between pads using nozzles and low pressure and (d) active lubrication (high pressure) through a single orifice machined in the middle of pad surfaces.

In figure 2.2.4(a) is possible to see the orifice machined in the middle of the pad surface (active lubrication). In the same figure is also possible to see injectors between pads for the conventional lubrication (hydrodynamic lubrication). Figure 2.2.4(b) illustrates the bearing composed of 4 tilting-pads (configuration load-between-pads) with the two servo valves mounted on the top of the bearing housing. Compactness is one of the main design advantages. Figure 2.2.4(c) shows the conventional lubrication between pads. Nozzles are used to redistribute the oil along the pad inlet edge. A picture of the hydrodynamic lubrication is shown in a “frozen” instant of time. Finally, figure 2.2.4(d) shows the picture of the active lubrication (high pressure) through a single orifice machined in the middle of pad surfaces “frozen” in a instant of time. A low frequency sinusoidal signal is sent to only one of the two servo valves. Although no signal is sent to the other servo valve, oil leakage can be seen in the orifices in the middle of the pads (weak flow), typical when using under-lapped servo valves.

The test rig designed to evaluate the performance of rotor-bearing performance is shown in figure 2.2.5. An electric motor **7** with speed controller and a belt transmission is used, aiming at achieving angular velocities until 10,000 rpm. The rotating shaft **2** is supported at one of its ends by a ball bearing **1**. Such a shaft can behave as a rigid body or as a flexible body, depending on the frequency range and the number of rigid discs attached to its free end, close to the auxiliary bearing **4**. The active tilting-pad journal bearing **3** built by 4 pads in load-between-pad configuration is the mechatronic device to be tested. The oil to build the hydrodynamic lubrication is supplied through pipelines **10**. The electronic radial oil injection (active lubrication) is built using two servo valves **8** as actuators and displacement sensors **9** as input signals to the controller. Acceloremeters are also mounted on the auxiliary bearing **4**. Acceleration signals can also be used to define different control strategies. An excitation station built by an auxiliary bearing **4** connect the electromagnetic shaker **5** and the force transducer **6** allows also dynamic testing. A hydraulic system built by a pump, accumulators, oil filters and servo valves **8** and a digital control unit built by a PC computer with DSPACE cards are used. A dedicated software for digital signal processing Simulink-based is used to design and investigate different types of control strategies.

2.2.3 Mutirecess Journal Bearing with 2 Pairs of Active Pockets

Tilting-pad journal bearings are frequently used in high speed machines with the aim of avoiding oil-whirl instabilities. They do not have significant cross-coupling effects. It means the control energy can be saved and used to increase the damping properties of the bearing instead of suppressing or eliminating the cross-coupling effect. Tilting-pad bearings are though more expensive than conventional journal bearings, because they are built by extra accurately machined components, namely the pads. Multirecess journal bearings are hybrid bearings which allow a significant adjustment of the bearing stiffness and due to the design simplicity they are cheaper than tilting-pad journal bearings.

The active hybrid journal bearing under investigation in this work has four recesses, aligned in pairs in the horizontal (Y) and vertical (X) directions and numbered as shown in figure 2.2.6(a). The conventional passive bearing operation is warranted by fluid injection into the recesses through capillary orifices, at a constant pressure supply . Additionally, for active dynamic control of the fluid pressure and compensation of cross-coupling effects, the flow is injected into the opposed bearing recesses pairs, each one of them are connected to servo control systems, as explained before. The servo control system is constituted by electro hydraulic servo valves, journal displacement and/or velocity transducers and PD feedback controllers (gains g_{1X}, g_{2X}, g_{3X}, g_{4X}, g_{1Y}, g_{2Y}, g_{3Y} and g_{4Y}). The servo valves are controlled by electrical voltage signals, U_X and U_Y, generated by the combination of journal dynamic displacement and velocity measurement signals in X and Y directions, mounted on the bearing housing as illustrated in figure 2.2.6(b). The main geometric characteristics of a four recesses hybrid journal bearing are represented in figure 2.2.6(c). The operational principle of the active lubrication is the the same as presented

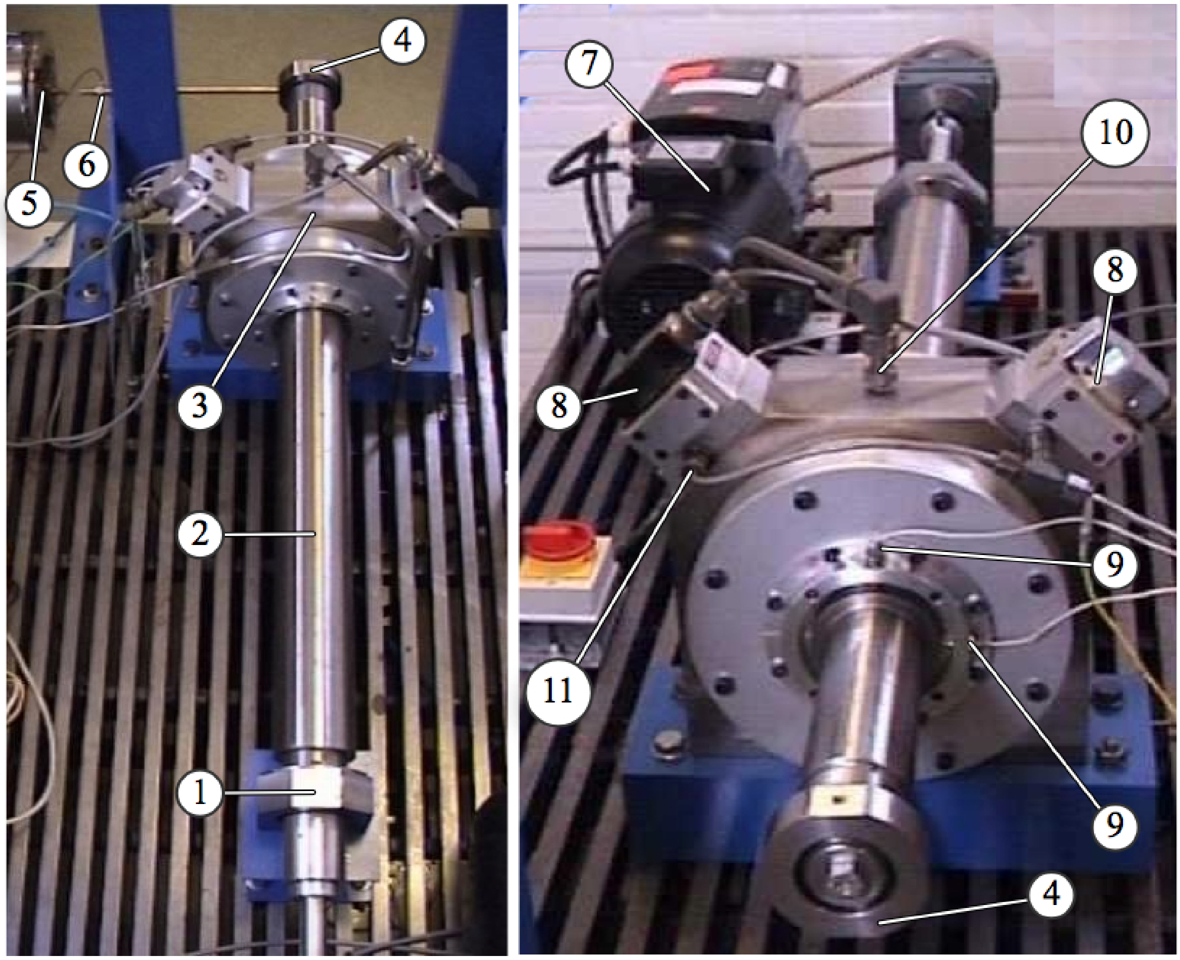

Figure 2.2.5: Test rig to experimentally evaluate the performance of active lubricated bearings to control rigid and flexible rotating shafts: **1** – pedestal and ball bearing ; **2** – flexible shaft; **3** – active lubricated tilting-pad bearing **4** – auxiliary bearing used as excitation and measurement station; **5** – electromagnetic shaker; **6** – force transducer; **7** – electric motor; **8** – servo valves; **9** – displacement sensors without contact (eddy-current); **10** – pipeline for low pressure injection (hydrodynamic lubrication) and **11** – pipeline for high pressure injection (active lubrication).

in the last section, but the bearing geometry changes. Multirecess bearings have relative good damping characteristics in the passive form. Nevertheless, at high angular velocities the cross-coupling effect increases and affects significantly the rotor-bearing stability. The feasibility of applying multirecess journal bearings to industrial compressors is theoretically investigated in Santos and Watanabe [23] and experimentally illustrated in Bently et al. [53] [54]. The application to rigid rotating shafts seems to work very well due to strong changes of the direct stiffness and damping coefficients. Nevertheless, such a kind of bearing may not be so well suitable to control lateral movements of flexible shafts. Such a claim will be deepened in chapter 4 based on the theoretical results obtained in [23].

Modifications of the direct stiffness coefficients are achieved by means of proportional as well as derivative controllers. The largest modifications of the direct stiffness are achieved by means of the proportional controllers. However, significant modifications of the direct damping coefficients are only achieved by using derivative controllers, Santos and Watanabe [12]. It means that the linear velocity of the journal center has to be measured or estimated.

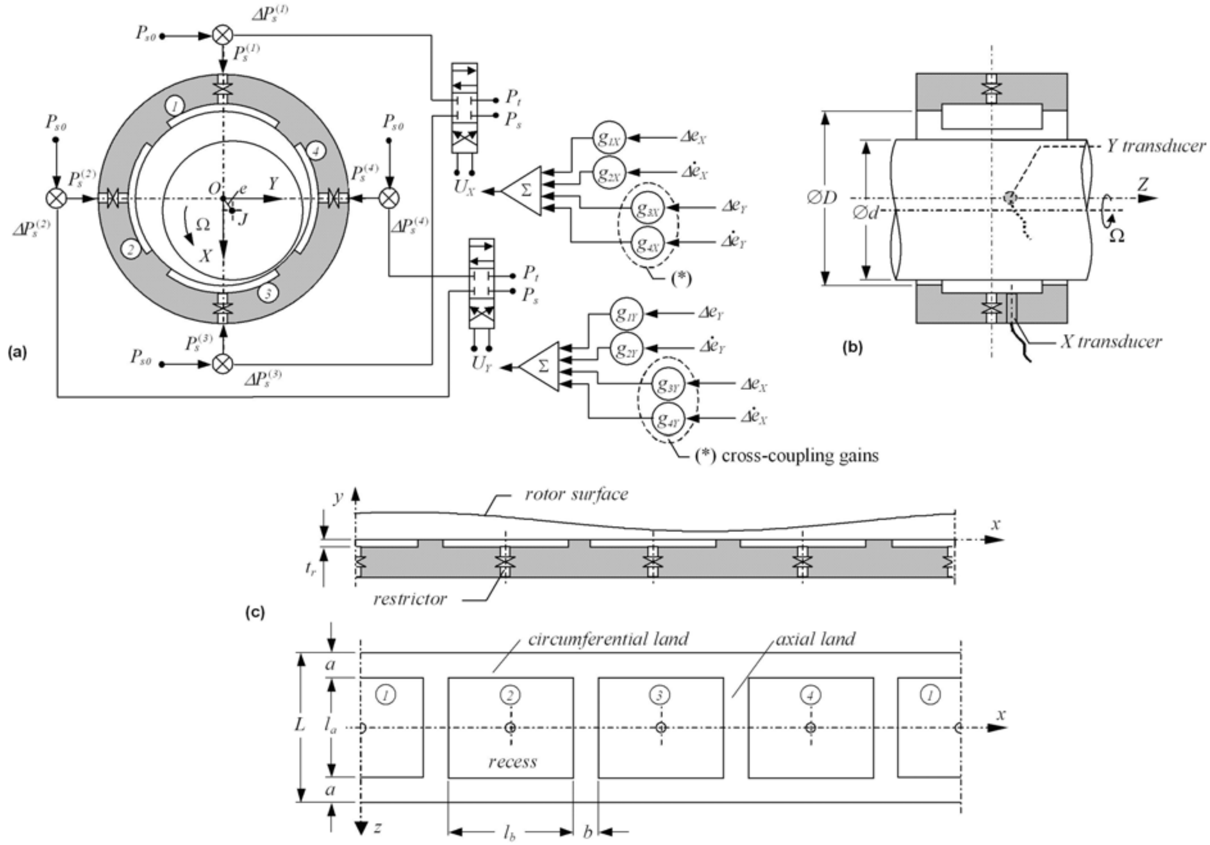

Figure 2.2.6: Schematic view of a multirecess journal bearing with 2 pair of "active pockets": (a) bearing section with control scheme; (b) rotor and displacement sensor mounted onto the bearing housing and (c) planar view of the multirecess journal bearing composed of 4 pockets (recesses) and landing surface.

If an X-Y uncoupled PD controller is used, the cross-coupling stiffness and damping coefficients cannot be influenced by the control gains, whereas if an X-Y coupled PD controller is used, a compensation of the cross-coupling effects is possible. The significance of the modification of the bearings properties achieved by the active lubrication is evaluated by the behavior of the whirl frequency rate and critical mass parameter, Lund and Thomsen [168]. By properly designing a X-Y coupled PD-controller for the active lubrication system, it is possible to reduce the whirl frequency ratio and increase the critical mass parameter. The crossed retrofitting of the rotor linear displacement signals with a simultaneous inversion of their signals allows almost the cancelation of the cross coupling stiffness in a tiny range of proportional gain values, Santos and Watanabe [15]. From the point of view of control implementation in variable-speed rotating machines, an adaptive self-tuning controller, with a variable gain is necessary to compensate the cross coupling effect.

Aiming at experimentally investigating multirecess journal bearings under active lubrication, a test rig was designed and is being built at the Department of Mechanical Engineering of the Technical University of Denmark, as it is illustrated in figure 2.2.7. Figure 2.2.7 shows the design of a multirecess journal bearing with two pair of active pockets. Two servo valves **1** and **2**, four force transducers **6**, **7**, **8** and **9** and three pressure sensors **3**, **4** and **5** are shown (the fourth pressure sensor is not illustrated). One of the active pockets **10** is also illustrated. The four force transducers will be used to measure the reaction forces between rotating shaft and bearing pockets, i.e. the active fluid film forces. The four pressure transducers will allow to establish a

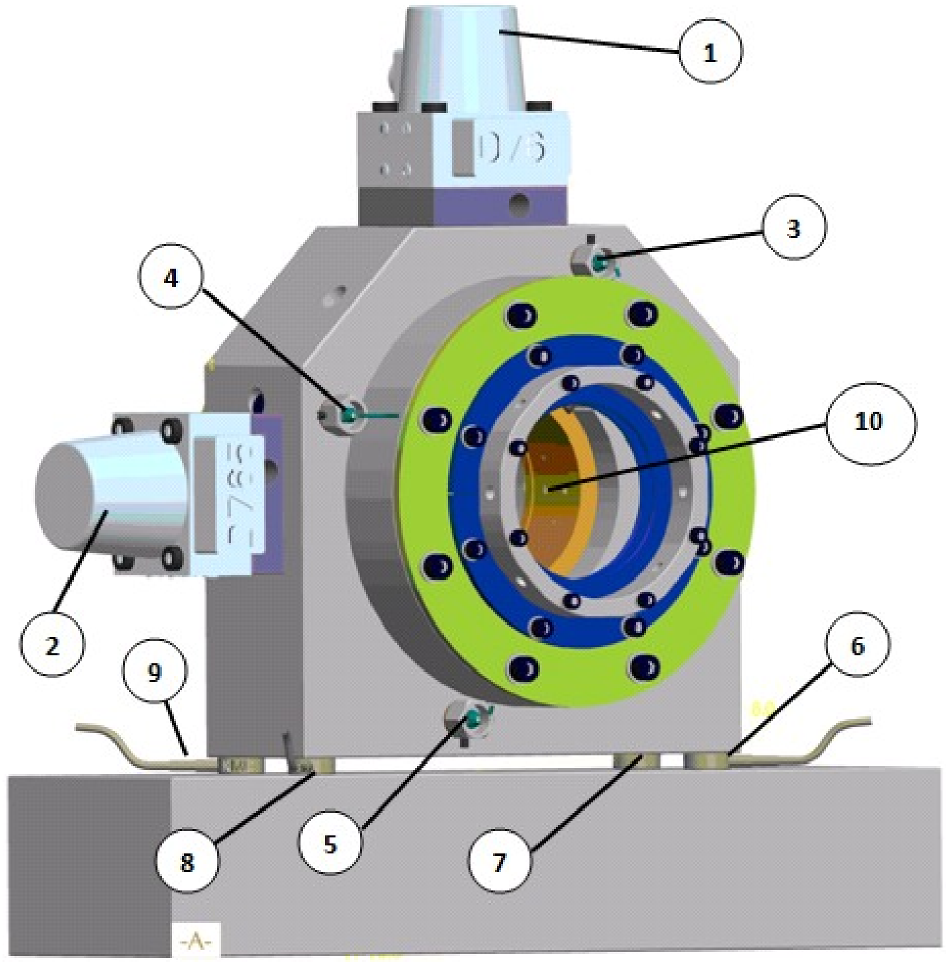

Figure 2.2.7: View of the designed multirecess journal bearing with 2 pair of "active pockets" to be built at the Department of Mechanical Engineering of the Technical University of Denmark in the near future: **1** – servo valve, **2** – servo valve, **3** – sensor to measure the pressure in the vertical upper pocket, **4** – sensor to measure the pressure in the horizontal left pocket, **5** – sensor to measure the pressure in the vertical lower pocket, **6**, **7**, **8** and **9** – force transducer to measure the reaction force between housing and foundation, which is influenced by the active oil film forces, **10** – "active pocket".

correlation between pressure in the pockets and reaction forces on the bearing housing.

It is imperative to understand that the high precision calibration of the active oil film forces is a very challenging problem and, if overcame, will open new possibilities of dynamic testing rotating machines in situ via their fluid film bearings, Santos [30]. For example when a comparison with active magnetic bearings is made, the accurate prediction of the magnetic forces make feasible the utilization of such bearings as calibrated shakers. This will also be a trend in controllable fluid film bearings, Santos [33]. Insights into the calibration of active magnetic forces of active magnetic bearings are elaborated upon in the following script.

2.3 Active Magnetic Bearings with Slot and Hall Sensors

Active magnetic bearings (AMB) have normally been used as simple bearings, and not as an integrated identification and diagnosis tool. To fully exploit the potential of active magnetic bearings it is necessary to think about them as both actuators and as bearings, capable of not only supporting the rotating shaft but also of exciting the shaft with defined signals. This unique feature facilitates measurements of input-output (displacements and forces) relations and not only measurements of the system response. The measurement of the output signals (displacements) is already integrated in active magnetic bearings as a part of the control loop to stabilize the system when they are used as simple bearings, and the precision of the output measurement has reached a sufficient level when utilized eddy current probes, Aenis and Nordmann [36].

The measurement of the input signals (forces) is still subjected to research and different methods have been proposed Marshall et al. [174] Aenis and Nordmann [36] Knight et al. [156]. Generally the different force measurements in an active magnetic bearing can be divided into two main groups: (a) measurement of coil currents and rotor displacements (i-s-method, reluctance network model) and (b) direct measurement of the magnetic flux density B with Hall sensors. The two groups have different advantages and drawbacks but when it comes to accuracies the Hall sensor method has shown promising results [36]. The drawback in using Hall sensors is that the air gap has to be enlarged to integrate the Hall probes resulting in a decrease of load capacity of the bearing. Knopf and Nordmann [157] [158] and Gahler [115] give very good theoretical and experimental contributions to the problem of rotor dynamic testing and control with active magnetic bearings. Gähler [115] uses Hall sensors and applies dynamic forces at frequencies from 20 to 200 Hz with constant amplitude and obtains a force error of 11% of load capacity. Dynamic forces with various amplitudes are sequentially applied to the rotor at 120 Hz, and the force error is reduced to 2% of the load capacity. Knopf and Nordmann [157] report 1% of load capacity for static loads but 5% with increasing eccentricity and speeds.

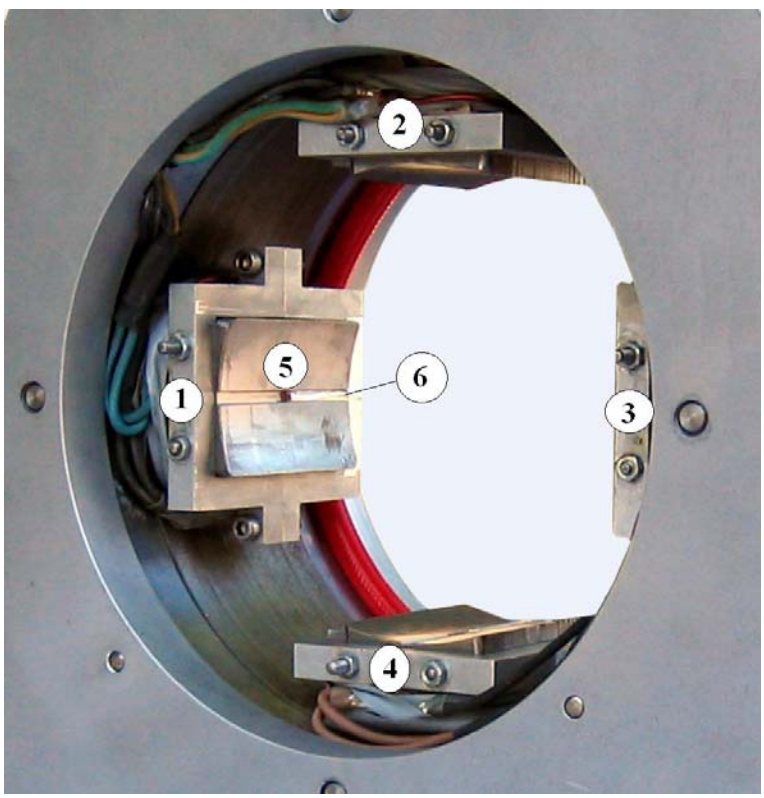

Figure 2.3.1: View of the active magnetic bearing composed of 4 poles (elements **1**, **2**, **3** and **4**) and one hall sensor per pole **5** mounted at the center of the pole surface and inside of the longitudinal slot **6**.

A different Hall sensor mounting principle is proposed in Kjølhede and Santos [24], which does not require enlargement of the air gap. Earlier mounting principles are based on attaching the Hall sensor directly to the pole surface resulting in an unintended enlargement of the air gap. The mounting principle described in [24] involves attaching the Hall sensor in an especially post-manufactured slot at the pole surface, as it is illustrated in figure 2.3.1. The active magnetic bearing designed and built is a four pole radial bearing with a rotor diameter of 140 mm and a shaft diameter of 90 mm. The stator and rotor parts are built as laminating sheets. Each lamination sheet has a thickness of 0.5 mm and both the stator and the rotor is composed of 140 lamination sheets. The surfaces of the lamination sheets have been oxidized to prevent any electrical contact between the sheets. The nominal radial air gap between the rotor outer diameter and the stator legs is 0.25 mm and the AMB was designed for a load capacity of approximately 1900 N, Andersen [44]. The bearing has a flux splitting coil configuration and it is operated in differential mode, Schweitzer et al. [232], where one magnet is driven with the sum of bias current i_0 and control current i_x, and the other one with the difference. The operating point of the AMB is determined by the bias current and the AMB is originally designed for a bias current of 8 A. In practice the AMB can be operated with a bias current up to 10 A and to investigate the influence of the bias current the experiments are carried out for bias currents of 6, 8 and 10 A. The maximum bearing force is reached if for one coil the resulting current is equal to zero.

The main advantage of installing the hall sensors in the slot is the protection of the sensors against rubbing and shock. Due to the machined slots on the poles, a reduction of the magnetic forces are expected. Nevertheless, the reduction observed is very small, less than 1% of the bearing capacity as reported in Kjølhede and Santos [24]. Once the slots are machined on the the poles, a further experimental calibration of the hall sensors is needed. In chapter 4 rotordynamic testing is performed using the bearing as a calibrated shaker. The results are promising and are validated using another experimental procedure.

Magnetic actuation is also used to excite and control coupled rotor-blade vibrations. Following details about two rotor-blade test rigs are presented. The main interest is the physical understanding coupled rotor-blade dynamics and the feasibility of applying active vibration control techniques to improve its dynamic behavior.

2.4 Dynamics and Control of Rotor-Blade Systems

In this section two experimental setups are presented. The first one is designed to validate mathematical models for rotor-blade systems with different levels of complexity. Parametric vibrations, stiffening effect and veering phenomena are clearly understood and physically demonstrated. After understanding the dynamic coupling between rotor and blades, a new test rig is designed with the aim of investigating active control of rotor-blade vibrations and experimental modal analysis in time-varying mechanical systems.

2.4.1 Dynamics of Rotor-Bearing-Blade Systems

The test setup is presented in figure 2.4.1(a) and schematized in figure 2.4.1(b). A rotating disc (disc + motor) is fixed rigidly to a foundation. The disc-motor-foundation builds a single concentrated mass, and it is connected to the inertial frame by means of 4 flexible beams with adjustable length, i.e., adjustable springs. This arrangement allows only linear motion of the rotor center and eliminates the gyroscopic effects due to angular motion of the rotor-foundation.

Thus, the mass-spring system has only one degree of freedom, i.e., linear displacement in the horizontal direction.

(a)

(b)

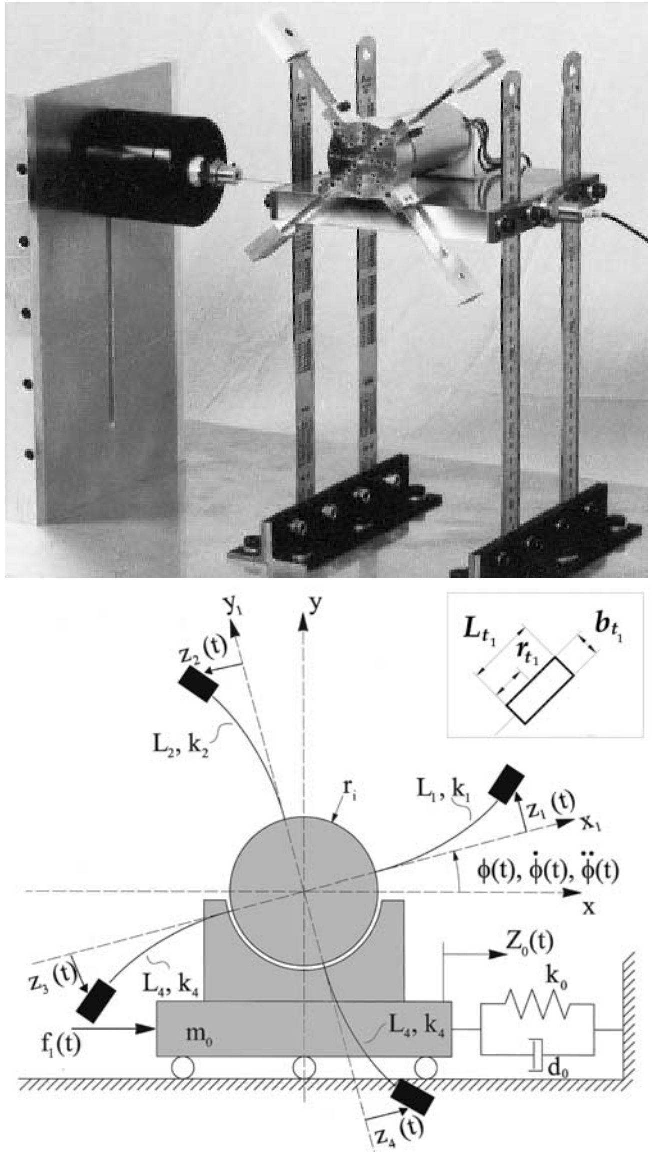

Figure 2.4.1: (a) Test rig built by: rotor-foundation supported by flexible beams; 4 flexible rotating beams with tip masses attached to their ends; electromagnetic shaker attached to foundation by means of a wire; and acceleration sensors fixed to foundation - (b) mechanical model illustrating: reference frames I and B_1; degrees of freedom $z_o(t)$, $z_1(t)$, $z_2(t)$, $z_3(t)$, $z_4(t)$; and main reference frames: inertial I (x, y) and moving B_1 (x_1, y_1).

Four masses (particles) are connected to the rotating disc by means of flexible beams (blades) with adjustable length. The movements of the tip masses are described by linear displacements in the rotating reference frame. Only the first beam bending mode is of importance in the defined range of frequencies. An electromagnetic shaker is used to excite the mass-spring system in the horizontal direction. An acceleration sensor is used to measure the linear movements of the mass-spring system and indirectly detects changes of the blade dynamic behavior. The test rig damping, specifically the damping factor related to the blade mode shapes, is kept extremely small. This facilitates the measurements while capturing the blades vibrations by means of the acceleration sensor attached to the non-rotating part of the test rig.

Two points should be shortly highlighted: 1) the tip mass considerably influences the dynamic response of the beam, once it strengthens the coupling between the dynamic terms and the elastic ones, Yigit et al. [275] Fallahi and Lai [103]; 2) the restriction of the motion in the vertical plane allows that theoretical as well as experimental analyses can be carried out with a 2D model in the frequency range of 0 to 50 Hz.

2.4.2 Control of Rotor-Bearing-Blade Systems

Another experimental test setup is designed and built with sensors and actuators attached to the rotor and to the blades. The rotor is allowed to perform lateral movements in both orthogonal directions. The eddy-current probes, strain gages and magnetic actuators are capable of sensing and acting on all degrees of freedom of the test setup, making feasible the experimental investigation of active rotor-blade vibration control using different control strategies.

The rig is composed of a rigid rotating disk mounted in a flexible suspended hub. The flexible suspended hub can perform lateral movements in both orthogonal directions, as already mentioned. For simplicity, the suspension is designed to allow only disc lateral movements. It means disc angular rotations and gyroscopic effects are avoided. Four flexible blades are radially attached onto the disc. Different schemes of blade configurations, with and without mistuning, can be easily generated by substituting or rearranging the blades or by changing the weight of the tip masses.

To drive the rotor-blade system an AC-motor is used. The motor speed is controlled by a frequency converter. The motor torque is transmitted to the flexible suspended bladed rotor with help of an auxiliary subsystem built by a pulley and two flexible couplings. The flexible coupling arrangement allows “free” rotor/hub lateral movements.

Various sensors are built into the test rig to monitor the rotor and blade vibration levels. The blade deflections, the rotor lateral movement and the rotor angular position and speed are monitored, see figure 2.4.2. The rotor/hub lateral movement is monitored by two eddy current displacement probes, elements **7** and **8** in figure 2.4.2. The rotor angular position and speed are also measured by means of an eddy current displacement probe attached to the hub, element **9**. Such a sensor detects a mark at the shaft surface and its signal is used as trigger. Based on the time-history of the trigger signals the rotor angular speed and angular position can be calculated.

Each blade is instrumented with a pair of strain gages, allowing the measurements of blade deflections. The strain gages are connected to analog amplifiers (Wheatstone bridges) and filters. Electronic circuit boards have been developed for this particular test rig and built into the rotating disk. The measured strain gage signals are amplified and transmitted from the rotating disk to an external control unit, located outside the rotor in the inertial non-rotating frame, through an arrangement built by twelve slip rings.

Six pairs of non-contact electro-magnetic actuators are used to control the blade and rotor vibrations. Four sets of actuators are built into the rotating disk, acting directly onto each one of the blades **3**, **4**, **5** and **6**. The remaining two sets of actuators, **1** and **2**, are mounted in the inertial frame and act directly upon the hub. It means, these two pairs of actuators work as an active radial magnetic bearing, controlling the rotor lateral movements. The strain gage

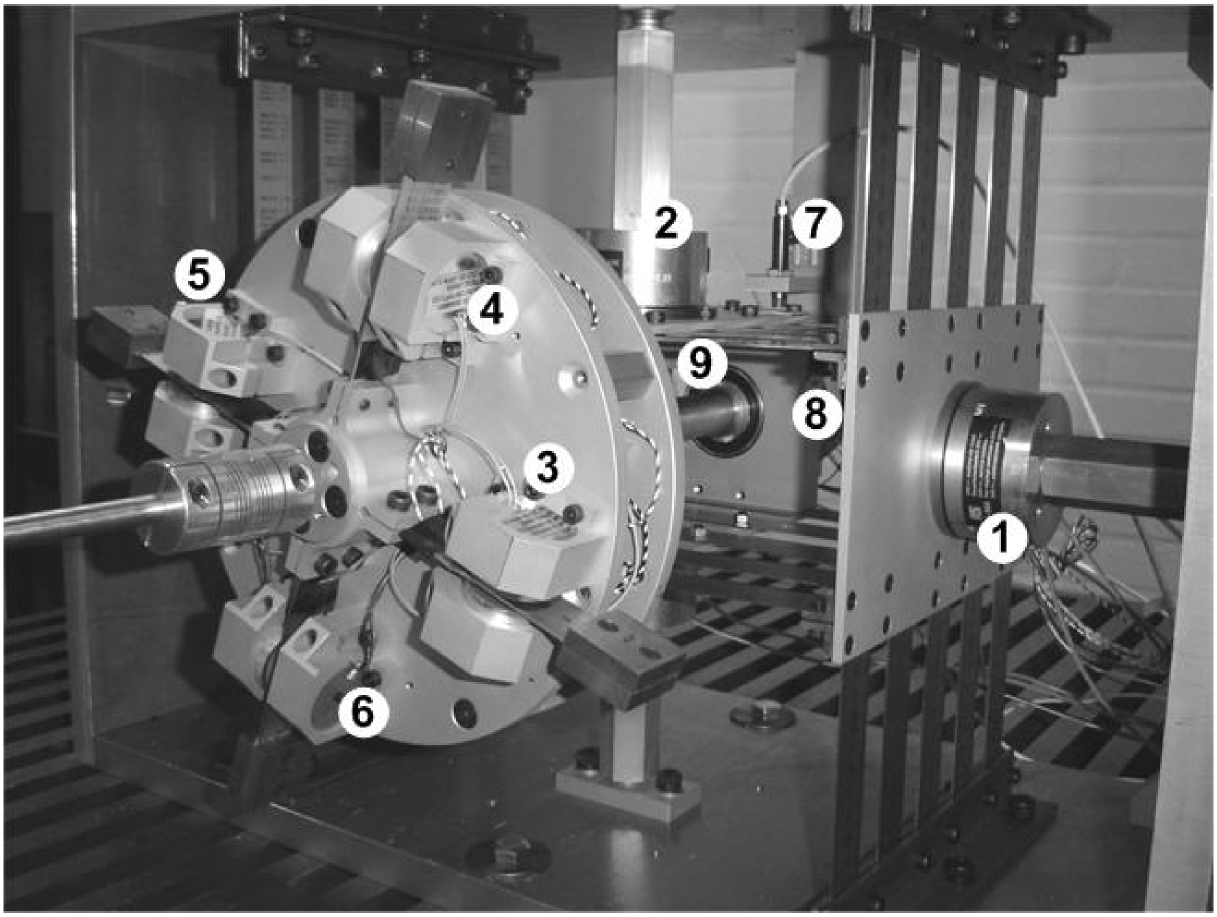

Figure 2.4.2: Picture of the rotor-blade test rig composed of one rotor-hub subsystem capable to vibrate horizontally and vertically and connected to four flexible rotating blades (or four beam-tip-mass subsystems): **1** electromagnetic actuator used to excite the hub in the horizontal direction; **2** electromagnetic actuator used to excite the hub in the vertical direction, **3** – electromagnetic actuator used to excite the blade-tip mass subsystem 1; **4** – electromagnetic actuator used to excite the blade-tip mass subsystem 2; **5** – electromagnetic actuator used to excite the blade-tip mass subsystem 3; **6** – electromagnetic actuator used to excite the blade-tip mass subsystem 4; **7** – eddy-current sensor used to measure the vertical movements of the hub; **8** – rotor-hub subsystem focusing the ball bearing housing; **9** – rigid shaft.

measurement signals and the control signals to the magnetic actuators, all of them related to the four flexible blades (**3**, **4**, **5** and **6**), are transmitted from the inertial reference frame to the rotating frame through the slip ring assembly.

Chapter 3

Mathematical Modeling – Theoretical Work

3.1 Introduction

The main original contribution of this chapter is of theoretical nature. Initially a short summary of the historical development of bearing-influenced rotor dynamics is presented in order to place the theoretical contribution of this work in a more general framework. Stodola [246] in 1925 idealized the concept of "oil spring" as a nonlinear spring depending on the rotor eccentricity. One of the original contributions is related to the combination of hydraulic servo systems and control theory to actively modify the "oil spring" coefficients and "oil damping" coefficients.

The oil film coefficients can be calculated by using the Reynolds equation and the perturbation technique Lund and Thomsen [168], after the equilibrium position of rotor-bearing system is found. The modification of the oil film coefficients and the control of rotor vibrations can be achieved by two different ways: a) by varying the bearing gap, i.e., the gap function h of the Reynolds equation or b) by changing the hydrodynamic pressure distribution between rotor and housing, i.e., the pressure function p of the Reynolds equations. In the first, the gap function h is modified by forces applied on the back of the tiling-pad. In the second case, the pressure distribution between rotor and sliding surface is modified by injecting pressurized lubricant through one or multiple orifices machined on the bearing surface. Such a principle is also applied to a multirecess bearing with 4 pockets. After presenting the modified equation for active lubrication taking into consideration isothermal conditions, a thermohydrodynamic model is developed considering a hybrid lubrication (hydrostatic + hydrodynamic) Santos and Nicoletti [8] [9]. Nevertheless, oil film coefficients are until now calculated only using isothermal theory. Contributions to hybrid and active elastohydrodynamics, i.e., elastic deformations of the pads together with hybrid and active lubrication can be found in Haugaard and Santos [135] [136] [137].

The equations for describing the dynamics of hydraulic servo systems and rigid rotor are coupled to the Reynolds equation. After describing the dynamics of the global system built by rigid rotor, oil film and hydraulic actuators, control techniques can be applied to achieve more stable rotating systems.

After focusing on bearing and rotor-bearing modeling, emphasis is given to the rotor-bearing-blade systems. Mathematical models with different levels of complexity are derived and the range of application for such models are elucidated. At the end of this chapter the contribution to control system design is mentioned, focusing on centralized and decentralized rotor-bearing systems and rotor-bearing-blade systems.

3.2 Bearing-Influenced Rotor Dynamics – A Short Historical Summary

A very good and detailed historical overview about oil film bearings and rotor dynamics can be found in Dowson [92]. A short historical summary is presented here in order to situate the theoretical contribution of the present work in a global framework.

Rankine[1] in 1869 first drew attention to vibrations which could be encountered in rotating machinery at certain critical speeds. Dunkerley[2] in 1884 carried out a wide range of experiments suggested by Osborne Reynolds and developed an approximate method for finding the natural frequencies or critical speeds of more complicated rotor systems. Jeffcott[3] in 1919 did much to establish the present-day concepts of shaft whirling. The 1920s witnessed a considerable development of the steam turbine and it was during this period that the special influence of fluid-film bearings upon rotor dynamics was clearly recognized. Stodola [246] in 1925 developed an iterative technique in which the mode shape of the vibration was assumed and successively improved, and which later formed the basis for the matrix iteration procedures. In his textbook[4] dated of 1927 Stodola associated the rotor vibration with the bearing stiffness, and introduced his “oil spring” concept. He suggested that a fluid-film bearing acted like non-linear springs, with a definite stiffness associated with each eccentricity.

Newkirk and Taylor[5] in 1925 reported a slight increase in the amplitude of a synchronous vibration when running at the shaft critical speed, but a remarkable and totally unsuspected increase in the severity of the vibration when the rotational speed of the shaft was about twice the first critical speed. The vibration was not synchronous, but almost equal to the first critical speed, or half the shaft running speed. Newkirk[6] in 1931 confirmed experimentally the existence of half-speed whirl with very stiff shafts running at speeds well below twice the first critical speed, and in due course it was recognized that half-speed whirl could occur over a wide speed range. The most comprehensive theoretical study prior to the Second World War was presented by Swift[7] in 1937. His general analysis of non-steady conditions in journal bearings provided the foundation for later analytical studies, even though it applied only to bearings of infinite width and excluded the possibility of cavitation. In the 1930s Smith[8] also analyzed non-symmetrical rotors, considering both bearing and rotor flexibility and found four critical speeds where there had been one for the symmetrical case.

Since the 1940s, following the advent of the digital computer, the literature on rotor dynamics has been largely concerned with the improved modeling of the rotor system and alternative procedures for predicting rotor response and stability. Support flexibility and damping,

[1]Rankine, W. M. M., 1869. ”On the Centrifugal Force of Rotating Shafts”, *The Engineer*, **27**, pp.249.

[2]Dunkerley, S., 1894. ”On the Whirling and Vibration of Shafts”, *Trans. R. Soc.* , **A185**, pp.279-360.

[3]Jeffcott, H. H., 1919. ”The Lateral Vibration on Loaded Shafts in the Neighbourhood of a Whirling Speed - The Effect of Want of Balance”, *Phil. Mag.* , **37**, pp.304-314.

[4]Stodola, A., 1927. ”Steam and Gas Turbines”, trans. L. C. Leowenstein, *McGraw Hill*, New York.

[5]Newkirk, B. L. and Taylor, H. D., 1925. ”Shaft Whipping Due to Oil Action in Journal Bearings”, *Gen. Elect. Rev.*, **28**, pp.559-568.

[6]Newkirk, B. L., 1931. ”Whirling Balanced Shafts”, *Proceeding of the Third International Congress of Applied Mechanics*, Stockholm, **3**, pp.105-110.

[7]Swift, H. W., 1937. ”Fluctuating Loads in Sleeve Bearings”, *J. Instn. Civ. Engrs.*, **5**, pp.161-195.

[8]Smith, D. M., 1933. ”The Motion of a Rotor Carried by a Flexible Shaft in Flexible Bearings”, *Proc. R. Soc.* , **A142**, pp.92-118.

distributed-mass systems, gyroscopic and inertia effects, external forcing, environmental and internal damping were considered. Various mathematical procedures built upon consistent mass representations (lumped-mass, transfer matrices and finite-element techniques) have been evaluated. Linear and non-linear dynamic response has been approached through developments of the early numerical methods, modal analysis, direct integration, finite-element and perturbation techniques. Balancing of rotating machinery has become the most important aspect of the successful operation of present-day equipments. Nowadays very good rotor dynamics textbooks extensively deal with all mentioned topics Dimentberg [87] Tondl [257] Rao [207] Dimarogonas and Paipetis [86] Krämer [159] Rieger [209] Vance [265] Lalanne and Ferraris [162] Frêne et al. [112] Krämer [160] Childs [79] Ulbrich [262] Ehrich [97] Gasch et al. [118] Yamamoto and Ishida [273] Genta [119] Muszynska [183].

3.3 Passive and Active Oil Film Bearings

Following two types of oil film bearings are in evidence: the tilting-pad journal bearings (TPJB) and the multirecess journal bearings (MRJB). A short summary of the historical development and improvements of mathematical models is given, followed by the theoretical contribution of this work.

3.3.1 Reynolds Equation and Oil Film Bearings Controlled by Active Chambers

The static and dynamic behaviors of tilting-pad bearings have been investigated by many authors since the end of the 1940s. Hagg[9] included oil-film damping in his studies of the behavior of a rigid shaft supported by various forms of bearings. In an associated experimental investigation he confirmed the stabilizing characteristics of tilting-pad bearings. Boyd [57] in 1953 investigated the static properties of such bearings. Lund [167] was one of the first authors who solved the Reynolds equation for theoretically calculating the dynamic coefficients of tilting-pad journal bearings. Klumpp [155] and Malcher [172] investigated theoretically and experimentally the behavior of such coefficients on the basis of advanced experiments. They mentioned some discrepancies between theoretical and experimental values of the damping coefficients. One of the reasons for explaining these discrepancies is the flexibility of the pivot to which the pads are connected. Jones and Martin [150] presented a detailed theoretical study to show the influence of bearing geometry on the steady-state and dynamic behavior of these bearings. Including the flexibility of the pivot in their models, Springer [241] [242] [243] Rouch [215] and Parsell et al. [199] investigated theoretically the dynamic properties of these bearings for predicting rotor instabilities. Additional information concerning instabilities in tilting-pad bearings can be found in experimental studies by Flack and Zuck [108], Lie et al. [164] and Olsson [193].

Combining pivot flexibility with the thermal effects in one theoretical model, Ettles [98] concluded that the thermal effects can reduce the damping properties of the bearing. Experience suggests that tilting-pad journal bearings have less damping than predicted by the theory Lund [169] White [270]. For more realistic results, the flexibility of the pads and the pivot has to be taken into account in the modeling [169]. The influence of other parameters on the steady-state behavior and dynamic coefficients of tilting-pad journal bearings is also mentioned in the literature. For example, the inlet pressure by Rodkiewicz et al. [213], the perturbation frequency by

[9] Hagg, A. C., 1946. "The Influence of Oil-Film Journal Bearings on the Stability of Rotating Machines", *Journal of Applied Mechanics*, A13, **68**, pp. 211-220.

Springer [243], Parsell et al. [199], Santos [2], Dmochowski [88], Childs [38], the thermal effects and distortions of the bearing by Brockwell and Dmochowsky [62]. All of the above mentioned papers concentrate on the properties of such bearings when they operate passively or without control.

In Santos [1] [2] [3] the Reynolds equation (3.3.1) is solved taking into account that the oil film function h_j for the j-th pad is described as function of the linear movements of rotor and pad pivot point.

$$\frac{\partial}{\partial \bar{y}}\left(\frac{h_j^3}{\mu}\frac{\partial p_j}{\partial \bar{y}}\right)+\frac{\partial}{\partial \bar{z}}\left(\frac{h_j^3}{\mu}\frac{\partial p_j}{\partial \bar{z}}\right)=6\,U\,\frac{\partial h_j}{\partial \bar{y}}+12\,\frac{\partial h_j}{\partial t}\qquad (j=1,2,3,4) \tag{3.3.1}$$

The static equilibrium position of the pads depends on the resultant hydrodynamic force over the pad and the applied control force at the pivot position, as illustrated in figure 3.3.1.

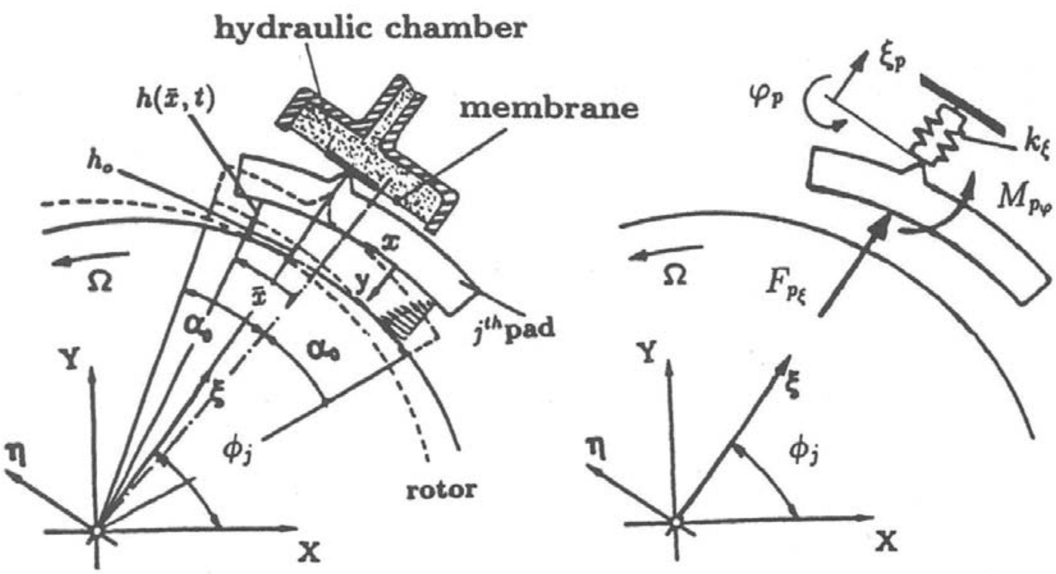

Figure 3.3.1: Hydraulic chamber system composed of a flexible membrane connected to the pivot point of a tilting-pad – changing the bearing properties by changing the oil film thickness h.

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To obtain the hydrodynamic forces the Reynolds equation is numerically solved. Two degrees-of-freedom are used to describe the lateral movements of the shaft center and two degrees-of-freedom to address the movements of each pad, which are able to simultaneously translate and tilt. Applying Newton-Euler method, a system of $(2 + 2N)$ nonlinear equations is achieved, where N is the number of bearing pads. Such a system of nonlinear equations is solved using Newton-Raphson procedure. After achieving the equilibrium position of shaft and pads, a small perturbation is introduced and the stiffness and damping coefficients of the bearing calculated. Such coefficients are not only a function of the Sommerfeld number and pre-load factor but also of the hydraulic control pressure in the hydraulic chamber. Such a control pressure can be changed statically or dynamically. By static changes of the chamber pressure, new static equilibrium positions for the rotor and pads are achieved and, consequently, new values of damping and stiffness coefficients are calculated. Such studies are carried out in Santos [2]. By dynamic changes, the mass and inertia of the pad have to be taken into consideration. If a linearized model is used, the system of dynamic equations can be written in the state-space form and different control strategies developed. Such studies are conducted by Santos [1]. The control pressure in the chamber can be mathematically written as a linear combination of the lateral movements of rotor, the pressure in the journal and in the chamber. Such signals can be multiplied by the appropriate control gains and the global coefficients of the bearing become a function of the control gains. By changing the control gains, it is possible to modify the stiffness and damping coefficients of the bearing. Such studies are conducted by Santos [4] in the frequency domain, where a parameter identification procedure based on least-square method is developed.

3.3.2 Modified Reynolds Equation for Active Lubrication

The derivation of the basic equations for the Active Lubrication is carefully presented in Santos and Russo [7], Santos and Scalabrin [11] and Santos et al. [10]. Zero-lapped servo valves [7] [11] and under-lapped servo valves [10] are considered and leakage flow through the servo valves is taken into account while writing the modified Reynolds equation. Aided by the reference frames illustrated in Figure 3.3.2 the velocity profiles of the lubricant in $\bar{x}$, $\bar{y}$ and $\bar{z}$ directions can be mathematically expressed under the consideration of laminar flow. Using the continuity equation and integrating the equations in the normal direction $\bar{x}$ to get the modified Reynolds equation for active lubrication as a function of the bearing gap h, one achieves a set of four hydrodynamic pressure distributions $p_i(\bar{y}, \bar{z})$ $(i = 1, 2, 3, 4)$ over the four pad surfaces.

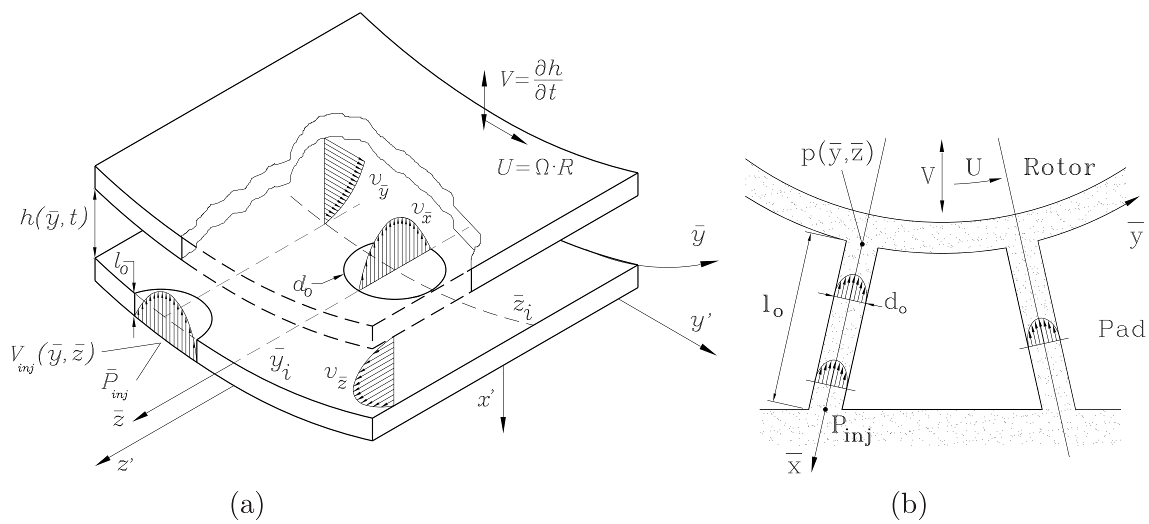

Figure 3.3.2: (a) Assumed velocity profiles in the bearing gap; (b) assumed velocity profile along the orifice channel.

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Such a set of partial differential equations can be calculated by solving a pair of coupled partial differential equations for each orthogonal direction. For the horizontal direction $(i = 1, 3)$, the pair of equations is given by equations (3.3.2) and (3.3.3):

$$\frac{\partial}{\partial \bar{y}}\left(\frac{h_1^3}{\mu}\frac{\partial p_1}{\partial \bar{y}}\right)+\frac{\partial}{\partial \bar{z}}\left(\frac{h_1^3}{\mu}\frac{\partial p_1}{\partial \bar{z}}\right)-\frac{3}{\mu l_o}\sum_{m=1}^{no}\mathcal{F}_m\left[p_1\left(1-C_1\right)-C_2\,p_3\right]=6\,U\,\frac{\partial h_1}{\partial \bar{y}}+12\,\frac{\partial h_1}{\partial t}$$

$$-\frac{3}{\mu l_o}\sum_{m=1}^{no}\mathcal{F}_m\left[P_{inj_{st1}}+C_3\,\omega_v^2\,K_V\,Y_h\sqrt{\frac{G_{1y}^2+\left(\omega\,G_{2y}\right)^2}{\left(-\omega^2+\omega_v^2\right)^2+\left(2\,\xi_v\,\omega_v\,\omega\right)^2}}\,e^{j(\omega t+\phi_y)}\right] \tag{3.3.2}$$

$$\frac{\partial}{\partial \bar{y}}\left(\frac{h_3^3}{\mu}\frac{\partial p_3}{\partial \bar{y}}\right)+\frac{\partial}{\partial \bar{z}}\left(\frac{h_3^3}{\mu}\frac{\partial p_3}{\partial \bar{z}}\right)-\frac{3}{\mu l_o}\sum_{m=1}^{no}\mathcal{F}_m\left[p_3\left(1-C_1\right)-C_2\,p_1\right]=6\,U\,\frac{\partial h_3}{\partial \bar{y}}+12\,\frac{\partial h_3}{\partial t}$$

$$-\frac{3}{\mu l_o}\sum_{m=1}^{no}\mathcal{F}_m\left[P_{inj_{st3}}+C_3\,\omega_v^2\,K_V\,Y_h\sqrt{\frac{G_{1y}^2+\left(\omega\,G_{2y}\right)^2}{\left(-\omega^2+\omega_v^2\right)^2+\left(2\,\xi_v\,\omega_v\,\omega\right)^2}}\,e^{j(\omega t+\phi_y)}\right] \tag{3.3.3}$$

where p is the hydrodynamic pressure distribution, h is the gap function, μ is the oil dynamic viscosity, $\mathcal{F}_m$ is function responsible for describing the orifices positioning over the pad surface, l_o is the length of orifice, U is the linear velocity at the rotor surface, $P_{inj_{st}}$ is the hydrostatic (leakage) part of the radial oil injection, ω_v and ξ_v are the natural frequency and damping factor of the servo valve, K_V is the servo valve constant, Y_h is the amplitude of the rotor oscillation in Y direction, G_{1y} and G_{2y} are the proportional and derivative control gains, ω is the excitation frequency and ϕ_y, given by

$$\phi_y=\tan^{-1}\left[\frac{-G_{1y}2\,\xi_v\,\omega_v\,\omega+\omega\,G_{2y}(\omega_v^2-\omega^2)}{G_{1y}(\omega_v^2-\omega^2)+\omega\,G_{2y}2\,\xi_v\,\omega_v}\right] \tag{3.3.4}$$

is the phase that describes the delay between the input signal of the servo valve and the output flow towards the bearing orifices.

For the vertical direction (Z direction), the pair of coupled equations are similar to equations (3.3.2) and (3.3.3), but referring to pads 2 and 4 instead of pads 1 and 3. These coupled equations are called the modified Reynolds equations for the active lubrication. Such equations relate the pressure distribution over each pair of pads arranged in Y and Z directions, to the gains of the adopted PD controller by considering the servo valve dynamics. G_{1y} and G_{1z} are the proportional gains, whereas G_{2y} and G_{2z} are the derivative gains. The servo valve dynamics is described using the constants K_V, ω_v, ξ_v, and K_{PQ}. The coeficients C_1, C_2 and C_3 are given by:

$$C_1=\frac{\displaystyle\sum_1^{no}\frac{\pi\,d_0^2}{128\,\mu\,l_0}-K_{PQ}}{\displaystyle\sum_1^{no}\frac{\pi\,d_0^2}{128\,\mu\,l_0}-2\,K_{PQ}};\quad C_2=\frac{-K_{PQ}}{\displaystyle\sum_1^{no}\frac{\pi\,d_0^2}{128\,\mu\,l_0}-2\,K_{PQ}};\quad C_3=\frac{1}{\displaystyle\sum_1^{no}\frac{\pi\,d_0^2}{128\,\mu\,l_0}-2\,K_{PQ}} \tag{3.3.5}$$

The coeficients C_1 and C_2 are the coupling terms between the hydrodynamic pressures over the pair of pads in a given direction. The coeficient C_3 multiplies the term related to the active radial oil injection.

Hence, by a given set of controller gains (G_{1y}, G_{1z}, G_{2y} and G_{2z}), it is possible to calculate the resultant oil film pressure distribution over each pair of pads, considering the effects of the additional radial oil injection.

In Santos and Russo [7] the fundamental set of equations to describe a controllable radial oil injection into the bearing gap using servo valves and a simple feedback controller was presented. However, the procedure assumes an almost constant pressure in the orifice region in order to allow coupling between servo valve flow and journal pressure. The formulation was not clearly explained in all details, which may lead to confusion. It is important to point out that in [7] the journal pressure in the orifice region was Taylor expanded and only the term of order zero was taken into account. This is justified based on the claim that the orifice area is much smaller than the pad surface area and the journal pressure variation over this small area is almost negligible when compared to the journal pressure variation over the entire pad surface area. The control parameters were later introduced directly into the modified Reynolds equation for active lubrication Santos et al. [10] and Santos and Scalabrin [11] under such an approximation. This provided insights into the influence of the control terms, since they appeared explicitly in the modified Reynolds equation.

The dynamic coefficients of the bearing with active lubrication can be estimated by applying a Taylor series and assuming harmonic variation of the oil pressure distribution Lund and Thomsen [168] Hamrock [134] Ghosh et al. [121]. Thus, the oil pressure distribution over the i-th pad can be approximated by:

$$p_i = p_i\left(\bar{x}, \bar{z}, t\right) = \mathcal{P}_{st_i} + \mathcal{P}_{y_i}\, \Delta Y_h\, e^{j\omega t} + \mathcal{P}_{z_i}\, \Delta Z_h\, e^{j\omega t} + \mathcal{P}_{\alpha_i}\, \Delta A_i\, e^{j\omega t} \tag{3.3.6}$$

where:

$$\begin{aligned} \mathcal{P}_{st_i} &= p_i|_{eq} & \mathcal{P}_{y_i} &= \left.\frac{\partial p_i}{\partial Y_h}\right|_{eq} + j\,\omega\, \left.\frac{\partial p_i}{\partial \dot{Y}_h}\right|_{eq} \\ \mathcal{P}_{z_i} &= \left.\frac{\partial p_i}{\partial Z_h}\right|_{eq} + j\,\omega\, \left.\frac{\partial p_i}{\partial \dot{Z}_h}\right|_{eq} & \mathcal{P}_{\alpha_i} &= \left.\frac{\partial p_i}{\partial \alpha_i}\right|_{eq} + j\,\omega\, \left.\frac{\partial p_i}{\partial \dot{\alpha}_i}\right|_{eq} \end{aligned} \tag{3.3.7}$$

The terms $\mathcal{P}_{st_i}$, $\mathcal{P}_{y_i}$, $\mathcal{P}_{z_i}$, and $\mathcal{P}_{\alpha_i}$ are obtained by applying small perturbations to the equilibrium position of the bearing system, and then solving the Reynolds equation for each pair of pads in the horizontal and vertical directions. Since there are four terms to be calculated for each pad, and the active bearing has four pads, one arrives to a set of sixteen partial equations, as described in Santos et al. [10]. By solving numerically this system of equations, one can build for each pad the following matrix:

$$\mathbf{A}_i = \int_{\bar{x}} \int_{\bar{z}} \begin{bmatrix} \mathcal{P}_{y_i} \cos\left(\frac{\bar{z}}{R_s}\right) & \mathcal{P}_{y_i} \sin\left(\frac{\bar{z}}{R_s}\right) & \mathcal{P}_{y_i} \left(R_s + \Delta s\right) \sin\left(\frac{\bar{z}}{R_s}\right) \\ \mathcal{P}_{z_i} \cos\left(\frac{\bar{z}}{R_s}\right) & \mathcal{P}_{z_i} \sin\left(\frac{\bar{z}}{R_s}\right) & \mathcal{P}_{z_i} \left(R_s + \Delta s\right) \sin\left(\frac{\bar{z}}{R_s}\right) \\ \mathcal{P}_{\alpha_i} \cos\left(\frac{\bar{z}}{R_s}\right) & \mathcal{P}_{\alpha_i} \sin\left(\frac{\bar{z}}{R_s}\right) & \mathcal{P}_{\alpha_i} \left(R_s + \Delta s\right) \sin\left(\frac{\bar{z}}{R_s}\right) \end{bmatrix} d\bar{z}\, d\bar{x} \tag{3.3.8}$$

Such a matrix is an implicit function of the Sommerfeld number $\left(S = \frac{2\mu L \Omega R^3}{|W|(R_s - R)^2}\right)$, the excitation frequency (ω), the servo valve dynamics (K_V, ω_v, ξ_v, K_{PQ}), and the control gains of the PD controller (G_{1y}, G_{1z}, G_{2y}, G_{2z}). The excitation frequency ω can be considered synchronous with the unbalance frequency, i.e., $\omega = \Omega$, or asynchronous, i.e., $\omega \neq \Omega$. The stiffness and damping matrices of the $i-th$ rotor-pad subsystem are obtained by the real and imaginary parts of $\mathbf{A}_i$, as follows:

$$\mathbf{K}_i = \Re\left(\mathbf{A}_i\right) \qquad \mathbf{D}_i = \frac{1}{\omega}\, \Im\left(\mathbf{A}_i\right) \tag{3.3.9}$$

By using a transformation matrix $\mathbf{T}_i$ which depends on the position of the pad in the bearing and on the angular displacement of the pad Allaire et al. [41], the stiffness and damping coefficients can be calculated in the inertial referential system, for the global rotor-bearing system:

$$\mathbf{K} = \sum_{i=1}^{4} \mathbf{T}_i^T \, \mathbf{K}_i \, \mathbf{T}_i = \begin{bmatrix} k_{yy} & k_{yz} & k_{y\alpha_1} & k_{y\alpha_2} & k_{y\alpha_3} & k_{y\alpha_4} \\ k_{zy} & k_{zz} & k_{z\alpha_1} & k_{z\alpha_2} & k_{z\alpha_3} & k_{z\alpha_4} \\ k_{\alpha_1 y} & k_{\alpha_1 z} & k_{\alpha_1\alpha_1} & 0 & 0 & 0 \\ k_{\alpha_2 y} & k_{\alpha_2 z} & 0 & k_{\alpha_2\alpha_2} & 0 & 0 \\ k_{\alpha_3 y} & k_{\alpha_3 z} & 0 & 0 & k_{\alpha_2\alpha_3} & 0 \\ k_{\alpha_4 y} & k_{\alpha_4 z} & 0 & 0 & 0 & k_{\alpha_4\alpha_4} \end{bmatrix} \tag{3.3.10}$$

$$\mathbf{D} = \sum_{i=1}^{4} \mathbf{T}_i^T \, \mathbf{D}_i \, \mathbf{T}_i = \begin{bmatrix} d_{yy} & d_{yz} & d_{y\alpha_1} & d_{y\alpha_2} & d_{y\alpha_3} & d_{y\alpha_4} \\ d_{zy} & d_{zz} & d_{z\alpha_1} & d_{z\alpha_2} & d_{z\alpha_3} & d_{z\alpha_4} \\ d_{\alpha_1 y} & d_{\alpha_1 z} & d_{\alpha_1\alpha_1} & 0 & 0 & 0 \\ d_{\alpha_2 y} & d_{\alpha_2 z} & 0 & d_{\alpha_2\alpha_2} & 0 & 0 \\ d_{\alpha_3 y} & d_{\alpha_3 z} & 0 & 0 & d_{\alpha_3\alpha_3} & 0 \\ d_{\alpha_4 y} & d_{\alpha_4 z} & 0 & 0 & 0 & d_{\alpha_4\alpha_4} \end{bmatrix} \tag{3.3.11}$$

where:

$$\mathbf{T}_i = \begin{bmatrix} \cos(\varphi_i + \alpha_i) & \operatorname{sen}(\varphi_i + \alpha_i) & 0 \\ -\operatorname{sen}(\varphi_i + \alpha_i) & \cos(\varphi_i + \alpha_i) & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{3.3.12}$$

In the case of bearings without tilting-pads, the stiffness and damping matrices reduce to the size 2×2, by disregarding the variable α_i.

Once the oil injection pressure is mathematically written as a linear combination of the lateral movements of rotor multiplied by the control gains, the oil film coefficients become also a function of the control gains. By changing the control gains and using a mechatronic approach, it is possible to modify and adjust the stiffness and damping coefficients of oil film Santos [2] Santos et al. [10] Santos and Scalabrin [11], the same "oil spring" coefficients idealized by Stodola in 1925, that until 1994 were only a function of bearing geometry and rotor-bearing operational conditions Santos [1].

3.3.3 Mutirecess Journal Bearing with Two Pairs of Active Pockets

The importance of hybrid journal bearings as potential application as load support elements in high-speed turbomachinery and machine tool spindles has steadily grown over the past years, Sawicki et al. [228], San Andres and Childs [220], Fayolle and Childs [104]. Hybrid bearings enable high-load capacity with large direct stiffness, accuracy of positioning, low friction and long life. Such properties make them attractive for applications to reactor coolant pumps and precision grinder spindles. The static and dynamic characteristics of multirecess hybrid journal bearings have been extensively investigated by many authors with help of different mathematical models and sophisticated experimental test rigs.

Stodola's work [246] was one of the first report related to the modeling of journal bearing and to the investigation of critical speed of a shaft supported by hydrodynamic bearings. Hagg and Sankey [131] and Sternlich [245] investigated the linearization of bearings around different equilibrium positions and its effects on the dynamics of rigid rotors. A more complete handling of hydrostatic and hydrodynamic (hybrid) journal bearing dynamic properties was presented by

Adams and Shapiro [34]. According to Rohde and Ezzat [214], the effects of lubricant compressibility in recesses and supply lines lead to a sharp increase of the stiffness and an abrupt reduction of damping above a “break frequency”. Theoretical stiffness and damping results for hybrid bearings were presented by Rowe [216] and Rowe and Chong [217], with emphasis on the non-symmetric portion of the stiffness matrix, responsible for self-excited vibration in rotor-bearing systems. Rowe and Chong [218] and Chen et al. [78] presented a numerical solution using finite difference method to predict the dynamic coefficients of multirecess hybrid journal bearings and a method to determine the stability threshold speed of a rotor-bearing system, including the rotor flexibility.

Ghosh and Viswanath [120], Ghosh et al. [121] and Guha et al. [126] included the effect of fluid compressibility and fluid inertia into the modeling of hybrid journal bearings with deep recesses, taking into account also the dependency of the bearing dynamic coefficients on the perturbation frequency. In high speed hybrid journal bearings for cryogenic applications, the fluid inertia and turbulent flow effects must be included into the modeling, aiming at obtaining a more consistent handling of bearing dynamic characteristics, as presented by Artiles et al. [48] using an incompressible lubricant and by San Andres [221] [222] using a compressible lubricant. Yang et al. [274] extended the turbulent flow by including the thermal effects in the recesses. Variable lubricant viscosity with pressure and temperature and also flow pattern in the recesses are extensively treated by Braun et al. [58] [59] [60]. The elastohydrostatic performance of hybrid bearings is presented by Sinhasan et al. [238] and the effect of bearing shell deformation is taking into account in the modeling. Effects of misalignment on static and dynamics characteristics of hybrid bearings are theoretically as well experimentally investigated by Bou-Saïd and Nicolas [56]. The effects of unconventional recess geometries on the minimum film thickness, flow rate, rotordynamic coefficients and threshold speed are investigated by Sharma et al. [234].

As it can clearly be seen, sophisticated computational models have been used and still being elaborated and improved, aiming at correctly describing the dynamic coefficients of multirecess hybrid journal bearings. A major drawback of hybrid bearings is related to the instabilities resultant from internally developed cross-coupled forces of the bearings, Fayolle and Childs [104]. Lund [170] introduced the so-called whirl frequency ratio, which is generally about 0.5 for smooth-land hybrid bearings. It means that a rotor supported by such a bearing will be limited to speeds lower than twice the first critical speed. To reduce these cross-coupled forces and raise the onset speed of instability, San Andres and Childs [223] investigated the angled injection- hydrostatic bearings. The analysis revealed that the fluid momentum exchange at the orifice of discharge produces a pressure rise in the hydrostatic recess, which retards the shear flow induced by the journal rotation, and thus, reduces cross-coupling forces. The predictions as well as the measurements demonstrate that the advantages of using angled injection in hybrid bearings are lost as the journal speed increases and brings dominance of hydrodynamic over hydrostatic effects. Also aiming at reducing the cross-coupled forces, the use of deliberately roughened stators, already successfully tested for liquid “damper” seals, is theoretically as well as experimentally investigated by Fayolle and Childs [104]. As a result, the new design is more stable, allowing a significant increase of the onset speed of instability.

In such a framework the main theoretical contributions of Santos and Watanabe [12] [15] [23] are related to the search of control strategies associated to active lubrication, aiming at reducing the cross-coupling forces in multirecess hybrid journal bearings. The results related to the application of active lubrication to hydrostatic bearings presented by Bently et al. [53] [54] are predominantly experimental and gives an excellent contribution to the field of active lubrication or actively controlled hydrostatic bearings. Such results show clearly the feasibility of changing the transfer functions and root locus plots of the rotor-bearing systems using a proportional controller. Nevertheless, a lack in the theory of actively controlled hydrostatic bearings can be detected in the literature. The main contribution of the papers [12] [15] [23] is to fulfill this lack, where decentralized [12] [15] and centralized [23] controllers are developed. In Santos and Watanabe [12] [15] the mathematical model presented by Ghosh et al. [121] is extended by including the dynamics of the servo control system, as presented by Santos and Russo [7]. It means that turbulent flow, inertia effects, bearing shell deformation, misalignment effects and temperature variations are not taking into account in [12] [15] [23]. By controlling part of the hydrostatic pressure and flow into opposed bearing recesses with help of servo control systems, significant modification of fluid film flow and forces can be achieved. A multirecesses hybrid journal bearing with active lubrication is termed active hybrid journal bearing. The feasibility of influencing the bearing dynamic coefficients (reduction of cross-coupling stiffness and increase of direct damping coefficients) by means of a controlled fluid injection is the main focus of the papers [12] [15]. Bearing coefficients can be numerically calculated as a function of Sommerfeld number, perturbation frequency and control gains. The journal bearing stability is analyzed by using the concepts of critical mass and critical whirl frequency ratio Lund [170]. Details about the mathematical modeling can be found in Santos and Watanabe [12] [15] [23].

3.3.4 Modified Reynolds Equation and Energy Equation in Bearings with Multiple Orifices

The prediction of temperature distribution in journal bearings has been the goal of many research works. To determine the oil temperature distribution, the energy equation and the Reynolds equation have to be simultaneously solved, considering the lubricant properties. The theoretical study of the temperature distribution in the gap of journal bearings has begun in the early 60‘s Hunter and Zienkiewicz [144]. Since Dowson [90] published the generalized Reynolds equation, several papers have been presented applying the thermo-hydrodynamic (THD) theory to journal bearings Dowson and March [91] and more specifically to tilting-pad journal bearings (TPJB) Jones and Martin [150] Pinkus [204] Brockwell and Dmochowski [63] Fillon et al. [105] Fillon and Khonsari [106] Ha and Kim [128] Ha et al. [129]. Ettles [98] presented theoretical and experimental results of the dynamic coefficients for a four pad journal bearing, taking into account the pad flexibility. Heshmat and Pinkus [141] published a thorough study of the mixing inlet temperature, considering cavitation zones as well. Tanigushi et al. [252] applied the laminar and turbulent theory to TPJBs operating at high rotor velocities. Gadangi and Palazzolo [114] and Monmousseau et al. [178], by using elastohydrodynamic theory, analyzed the transient dynamic behavior of TPJB. Fillon and Khonsari [106] presented very complete and helpful design charts of maximum operational temperatures of journal bearings with five shoes. Based on the literature review, one observes that previous investigators took into account conventional lubrication mechanisms. The main theoretical contribution of Santos and Nicoletti [8] [9] is the application of THD to hybrid tilting-pad bearings lubricated through multiple orifices. Such studies are directed to solve the energy equation and the modified Reynolds equation presented in Santos and Russo [7], considering the further effects of oil viscosity variation along the pad surface. Adiabatic boundary conditions are adopted, and a two-dimensional model is used to represent the fluid flow behavior in the bearing gap. After the THD investigation, one can conclude that, for controlling and cooling the bearing simultaneously, it is better to machine the orifices close to the edges. The cooling effect of such a kind of lubrication is restricted to the areas in front of the orifices, following the fluid flow in $\bar{y}$ direction, according to figure 3.3.2.

Reynolds Equation, Energy Equation and Structural Deformation in Bearings with Multiple Orifices and Pockets

Many authors have dealt with modeling of hydrodynamic thrust bearings. Huebner [143], Kim et al. [154] and Ettles and Anderson [99] presented models including 3-dimensional treatment of the oil temperature. These models do not include the effects of a pocket. Hemmi and Inoue [139] using an isothermal model investigated the influence of the pocket size in a centrally pivoted bearing. Ettles and Donoghue [100], Braun and Dzodzo [61] and Helene et al. [138] used Navier-Stokes equation solvers to investigate the flow pattern in bearing pockets or hydrostatic oil injection and its influence on the pressure distribution.

The main contribution of Santos and Fuerst [13], Heinrichson et al. [25] [26] and Heinrichson and Santos [29] is related to the modeling of hydrodynamic thrust bearings taking into account thermal effects and structural and thermal deformations in high loaded bearings. The theoretical results are validated against experimental results obtained in two different environments: an instrumented thrust bearing of a vertical hydrogenerator [13] and a very well equipped test rig [26] designed with the aim of investigating hybrid thrust pads. The theoretical and experimental works were done in co-operation with Alstom Power Ltd. in Switzerland.

More specifically in Santos and Fuerst [13] an adiabatic thermohydrodynamic model is used to describe static and thermal properties of the bearings under hybrid lubrication using a circular pocket. The minimum oil film, minimum startup hydrostatic pressure in the circular pockets and temperature distribution are calculated. The theoretical results are compared to experimental results obtained with help of an instrumented thrust bearing of a vertical hydrogenerator. From the comparison it was possible to conclude that the thermal and structural deformations could not be neglected in the case of highly loaded vertical water turbines. The mathematical modeling is expanded and improved in [25]. A three-dimensional thermo-elasto-hydrodynamic (TEHD) laminar flow model based on the Reynolds equation was developed. The model is described in detail in [25]. In [26] the model results are compared to experiments showing good agreement between theoretical and experimental pressure curves for a test-pad with an oil injection pocket at the pivot point. It is demonstrated that shallow pockets with depths of the same order of magnitude as the oil film thickness can have a positive effect on the bearing performance as they increase the minimum oil film thickness while slightly reducing the friction loss. In such recesses, there is no recirculation flow and the recesses have characteristics similar to those of parallel-step bearings.

The main contribution given in Heinrichson and Santos [29] is related to the study of the influence of equipping tilting-pad bearings with deep recesses in the high-pressure region. It is shown that a significant reduction of the friction loss is possible compared to conventional tilting-pad bearings and it is shown that recesses placed in the high-pressure region can be used for high-pressure jacking. A three-dimensional thermo-elasto-hydrodynamic model is applied to the analysis of tilting-pad bearings with spherical pivots and equipped with deep recesses in the high-pressure regions. A potential for a 10 - 20 % reduction in the friction loss compared to conventional plain bearing pads is documented. Design suggestions minimizing the power loss are given for various length-to-width ratios. The tilting angle in the sliding direction is more sensitive to correct positioning of the pivot point than conventional bearing pads. Improving the performance by equipping a tilting-pad bearing with a deep recess therefore requires accurate analysis and design of the bearing. Similarly, a high sensitivity perpendicular to the sliding direction suggests that this method of reducing friction is more feasible when using line pivots or spring beds than when using spherical pivots for controlling the tilting angle.

In deeper recesses (one or more orders of magnitude deeper than the oil film thickness), there is recirculation and the recess has no positive effect on the pressure build-up. The pressure distribution is therefore scalable with the Sommerfeld number and the design suggestions stated in this paper are generally applicable - with the reservation that the influences of thermal effects vary with the operating conditions. In the volume of a deep recess, the reductions of the flow equations leading to the Reynolds equation are not strictly valid. In particular, the assumption of inertialess flow seems unreasonable. The recirculation flow accelerates the oil entering the recess causing an inertial pressure drop at the recess leading edge. At the trailing edge of the recess, the oil is stagnated leading to an inertial pressure rise. In tilting-pad bearings with recesses, this may cause a smaller tilting angle than the one predicted by using the Reynolds equation. In conventional bearings the effects of inertia in the groove between the pads lead to an inlet pressure build-up at the leading edge. In large and slow speed tilting-pad bearings the inlet pressure build-up is usually negligible indicating that, for such bearings, the effect of inertia on the pressure in a recess is also likely to be small. In sector shaped bearing pads the centrifugal term may cause a shift of the pressure maximum towards the outer radius resulting in a radial tilt of the pad. Shinkle and Hornung [235] showed the recess flow to be turbulent at $Re = \rho U h_p/\mu > 1.0 \cdot 10^3$. For some of the recess geometries and velocities used in [29], turbulence is to be expected. The function of a deep recess is to create a low friction zone. When inertial end effects in the recess are negligible, the pressure inside it is close to constant regardless of whether the flow is laminar or turbulent. Although the use of the Reynolds equation inside the recess is not strictly valid, the inaccuracies arising from using this equation in a laminar formulation are therefore small. Determining the friction loss in the recess poses a more delicate problem. The friction loss in the bearing is calculated from the shear stress at the collar surface. Due to the thick oil film in the recess, the friction is small in this area when the flow is laminar and inaccuracies have a negligible influence on the calculation of the total friction loss. In turbulent flow recesses, the friction is larger and may significantly influence the total friction loss. Based on the work by [235] the inaccuracy on the recess and total friction loss is analyzed later in [29].

In Haugaard and Santos [135] [136] [137] the elasticity of the pads is taken into account using elasto-hydrodynamic models. Such models are being developed with the aim of achieving the dynamic coefficients of radial tilting-pad journal bearings under active lubrication, i.e., the dynamics of the servo valves is incorporated into the mathematical model. Promising results are being obtained and documented in [135] [136] [137]. The possibility of applying active lubrication to deform the bearing pad and change the bearing preload factor is also a new trend in controllable fluid film bearings as cited in Santos [33].

3.4 Passive and Active Rotor-Bearing Systems

As mentioned in section 3.2 very good rotor dynamics textbooks extensively deal with all topics related to passive rotor-bearing system dynamics, namely [87] [257] [207] [86] [159] [209] [265] [162] [112] [160] [79] [262] [97] [118] [273] [119] [183].

Related to active rotor-bearing systems, the most common controllers found in the industry are the PD and PID. Due to their simplicity of implementation, the PD and PID controllers are used in most engineering areas where control systems are applied. State feedback can also be applied to rotating systems and several investigations can be found in the literature. A summary of state feedback applied to rotating systems is presented by Stanway and Burrows [244] and Ulbrich [258]. Experimental results showing the efficiency of such control method in a horizontal rotor can be seen in Fürst and Ulbrich [113], regarding impulse response, unbalance response, start-up transient condition and instability condition. However, state feedback implies the consideration of all system parameters in the control system design. This can be a drawback to many applications since the number of parameters may be big and not all of them are available for observation and measurement, or the number of sensors is limited. In order to overcome this disadvantage, Firoozian and Stanway [107] design observers for the unavailable parameters of rotating systems. This strategy requires that rotor system modeling be strongly reliable and precise, otherwise the designed observer may provide misleading data to the control feedback.

An alternative way is the adoption of output feedback. Burrows et al. [67] present a detailed description of modeling, design, and identification of rotating systems with active control and output feedback. The same approach in discrete domain is presented by Lin et al. [165]. Regarding that only a limited number of parameters is considered in the control design, the decision of sensor position is of great importance. An analysis of sensor location can be found in the work of Palazzolo et al. [195].

Linear control techniques applied to non-linear systems can be efficient only for the operational condition that the controller was designed for. In rotating machines during transient conditions, where the coupling stiffness between orthogonal directions may increase significantly, an adaptive controller may be more efficient Lang et al. [163]. Examples of that application can be found in Sun et al. [247], El-Shafei [96], Lum et al. [166], Reinig and Desrochers [208], Janecki and Gosiewski [147].

Other types of controllers have also been investigated in the literature, being on-off control El-Shafei and Hathout [95], parametric control Rahn and Mote [206], and synchronous proportional control Rho and Kim [210] [211] just a few examples of them. El-Shafei [96] presents a comparison among different kinds of controllers and their effect in rotating machines.

Considering the controller gains calculation, different approaches can be found in the literature, as it can be seen in Bremen [64], Ulbrich [258], Lunze [171], Slotine and Li [239], Ogata [191], Zhou [278] among others. Sahinkaya and Burrows [219] present an algorithm of pole location for rotor-bearing systems that does not depend on the calculation of system eigenvalues. The algorithm determines the necessary changes in the state matrix for the system to present desired eigenvalues. In the following section the main contribution of the present work towards control design is elucidated.

3.4.1 Centralized and Decentralized Controllers for Active Lubricated Bearings

The design of the feedback control laws for active lubricated bearings is not a trivial task. The active (oil film) control forces are strongly coupled to the hydrodynamic forces via fluid continuity equation and Modified Reynolds Equation Santos and Russo [7]. There are different ways of designing the feedback control law:

- The first one is to predefine the structure of the control loop, for example, defining a PD-control (retrofit of displacement and velocity signals coming from the rotor lateral movements). The gain of such feedback signals can be explicitly included into the Modified Reynolds Equations [7]. By using the Modified Reynolds Equation and Perturbation Techniques the damping and stiffness of the bearing can be calculated as a function of the excitation frequency ω, the Sommerfeld number So and the feedback control gains G_P and G_D. Such coefficients can be introduced into the linearized mathematical model of rigid or flexible rotors. Eigenvalues and eigenvectors of the rotor-bearing system can be calculated. Keeping the attention on the real part of the eigenvalues, it is possible to calculate the gains in order to obtain a more stable rotor-bearing system. Such an approach is detailed in Santos et al. [10] and Santos and Scalabrin [11]. The drawback of such an approach is that only decentralized controllers can be designed. The main advantage is that the frequency-dependency of the active oil film forces is taken into account. Moreover, it makes feasible to elaborate tables for representing stiffness and damping coefficients as a function of Sommerfeld number, rotor eccentricity and control gains G_P and G_D. Such coefficients can be easily coupled to rotordynamic simulation programs to predict the dynamic behavior of rotor-bearing systems under conventional and active lubrication conditions.

- Another useful approach is to decouple the active oil film forces and conventional hydrodynamic forces by using pesudo-static tests and approximate the active oil film force by a linear function of the input signal U_v. This means, the problem is mathematically summarized by finding a linear coefficient λ that leads to $F_A = \lambda \cdot U_v$ where F_A is the active oil film force and λ is a coefficient obtained from pseudo-static tests and different values of U_v. Obtaining the coefficient λ is not an easy task, as discussed in Nicoletti and Santos [20]. Nevertheless, it allows the direct application of modern control theory to rotating shafts supported by active lubricated bearings. In Nicoletti and Santos [28] though such a coefficient is made depending on Sommerfeld number and excitation frequency, i.e., $\lambda = \lambda(So, \omega)$, and feedback control gains are calculated using modern control theory. The object of that study was an industrial gas compressor (flexible rotor-bearing system) of about 400 Kg.

- Another approach is to explore the nonlinear relationship between damping and stiffness coefficients on the rotor eccentricity (or equilibrium position inside the bearing) and to develop nonlinear controllers, as shown in Nicoletti and Santos [14].

3.4.2 First Approach – Equivalent Dynamic Coefficients obtained from the Equations of the Active Lubrication

A way of designing the feedback control gains is by adopting the global equivalent dynamic coefficients of the active lubrication, Santos et al. [10] and Santos and Scalabrin [11]. These coefficients are calculated by simultaneously considering the contributions of conventional (hydrodynamic) and active lubrication. In such a case the Reynolds equation is modified and an additional term is included, in order to consider the controlled radial oil injection into the bearing gap, Santos and Russo [7]. By also considering the dynamics of the servo valves and the adopted control strategy, it is possible to obtain the equivalent dynamic coefficients of the oil film as function of the control gains. It is important to highlight that the dynamics of the hydraulic system (servo valve parameters, pressure-flow relationship, pipeline and orifices dimensions) plays an important role in the oil film global coefficients. The estimation method of these global coefficients is thoroughly deduced in Santos [6], Santo et al. [10] and Santos and Scalabrin [11].

By adopting the global dynamic coefficients, the hydrodynamic forces considering both active and conventional lubrication are given by:

$$\begin{aligned} F_{H_y} &= -\bar{K}_{yy}\, y_H - \bar{D}_{yy}\, \dot{y}_H \\ F_{H_z} &= -\bar{K}_{zz}\, z_H - \bar{D}_{zz}\, \dot{z}_H \end{aligned} \tag{3.4.1}$$

Hence, the equations of motion of the rotor-bearing system (Eq. (3.4.4)) changes to:

$$\mathbf{M}\,\ddot{\mathbf{x}} + (\mathbf{G} + \bar{\mathbf{D}})\dot{\mathbf{x}} + \bar{\mathbf{K}}\,\mathbf{x} = \mathbf{f}_E \tag{3.4.2}$$

where the term $\mathbf{f}_A$ is suppressed and the contribution of the hydrodynamic forces, considering both active and conventional lubrication, is inserted into matrices $\bar{\mathbf{D}}$ and $\bar{\mathbf{K}}$ through the global dynamic coefficients. It is important to emphasize that the matrices $\bar{\mathbf{D}}(So, \omega, G_P, G_D)$ and $\bar{\mathbf{K}}(So, \omega, G_P, G_D)$ are dependent on the Sommerfeld number So, excitation frequency ω, the control loop gains G_P (proportional gain) and G_D (derivative gain). When such gains are set to null, Eq. (3.4.4) in the passive case and Eq. (3.4.2) become the same. A typical behavior of the global coefficients as a function of the proportional gain G_P will be illustrated in figures 4.1.3 and 4.1.2 in chapter 4.

3.4.3 Second Approach – Linearization of the Active Forces by means of Quasi-Static Experimental Tests.

The oil film forces can be split into two basic components: the hydrodynamic force originated from the conventional lubrication, and the active force resulting from the oil injection system. The hydrodynamic forces in a tilting-pad journal bearing can be represented by equivalent dynamic coefficients, neglecting cross-coupling effects, as follows:

$$\begin{aligned} F_{H_y} &= f_{static_y} - K_{yy}\, y_H - D_{yy}\, \dot{y}_H + F_{A_y} \\ F_{H_z} &= f_{static_z} - K_{zz}\, z_H - D_{zz}\, \dot{z}_H + F_{A_z} \end{aligned} \tag{3.4.3}$$

In the case of tilting-pad journal bearings, the cross-coupling coefficients (K_{yz}, K_{zy}, D_{yz}, D_{zy}) can be neglected when compared to the direct coefficients (K_{yy}, K_{zz}, D_{yy}, D_{zz}). Thus, only the direct coefficients are considered in Eq. (3.4.3).
The linearized equations of motion for a rigid or a flexible rotor-bearing system can be written as:

$$\left.\begin{array}{ll} \bullet\ \textit{active case} & \mathbf{M}\,\ddot{\mathbf{x}} + (\mathbf{G} + \mathbf{D})\,\dot{\mathbf{x}} + \mathbf{K}\,\mathbf{x} = \mathbf{f}_E + \mathbf{f}_A \\ & \\ \bullet\ \textit{passive case} & \mathbf{M}\,\ddot{\mathbf{x}} + (\mathbf{G} + \mathbf{D})\,\dot{\mathbf{x}} + \mathbf{K}\,\mathbf{x} = \mathbf{f}_E \end{array}\right\} \tag{3.4.4}$$

The term $\mathbf{f}_E$ is related to the excitation force applied to the shaft by means of the electromagnetic shaker. The gravity force $\mathbf{f}_O$, not presented in Eq. (3.4.4) is cancelled by the static part of the hydrodynamic forces, presented in Eq. (3.4.3). The force from the active lubrication $\mathbf{f}_A$ is linearized and considered as an external force in the mathematical model. Details about how to experimentally obtain such a force is presented in Nicoletti and Santos [20] [28]. The contribution of the conventional hydrodynamic lubrication is inserted into the model by using the matrices $\mathbf{K}$ and $\mathbf{D}$ of Eq. (3.4.4), which contain the oil film equivalent dynamic coefficients (Eq. (3.4.3)). In the literature, there are several approaches to estimate the equivalent dynamic coefficients K_{yy}, K_{zz}, D_{yy} and D_{zz} in Eq. (3.4.3) of conventionally lubricated bearings, for example, Lund [167], Lund and Thomsen [168], Springer [243], Someya [240] and Allaire et al. [41]. It is important to emphasize that the matrices $\mathbf{D}(So, \omega)$ and $\mathbf{K}(So, \omega)$ are dependent on the Sommerfeld number So and the excitation frequency ω. Such an approach, separating passive and active lubrication mechanisms, allows the application of modern control theory, Bremer [64]. However, in such an approach servo valve dynamics is neglected and discrepancies between theoretical and experimental results can be found in high frequency ranges [20]. The dynamics of a rigid rotor supported by an active lubricated tilting-pad bearing is theoretically as well as experimentally investigated in [20]. Later, in [28] such an approach is expanded by including the servo valve dynamics while representing the active oil film forces. The dependency of the active oil film forces on the frequency is taken into account and modern control theory is used to design different types of controllers for a flexible rotor supported by active lubricated tilting-pad bearings.

3.4.4 Third Approach – Exploring the Nonlinear Bearing-Rotor Relationships

The relationship between bearing (damping and stiffness) coefficients and bearing gap is highly nonlinear. The bearing gap is depending on the static rotor position inside of the bearing. The third approach explores the nonlinear relationship between damping and stiffness coefficients on the rotor eccentricity (rotor equilibrium position inside of the bearing) while designing nonlinear controllers, Nicoletti and Santos [14]. Special attention should be paid to integral controllers which can lead to less stable rotor-bearing systems when the integral part of the controller brings the rotor back to the bearing geometric center. This leads to a reduction of the static eccentricity and, consequently, a significant reduction of the (passive) bearing damping coefficient. To achieve more stable rotor-bearing systems the integral part of the controller should not reduce the rotor eccentricity. Ideally, the integral part should bring the rotor as close as possible to the bearing surface, increasing the static eccentricity. Nevertheless, in the case of high eccentricities, thermal effects may lead to a reduction of the (passive) damping coefficients, as mentioned by Ettles [98].

3.4.5 Comparison Among Different Control Design Approaches

It was shown in Nicoletti and Santos [20] that the estimation of oil-film (passive) coefficients is appropriate and the bearing coefficients are well tuned in the case of conventional hydrodynamic lubrication. In the active case (active lubrication), the second approach, based on the linearized active forces and the linear coefficient λ, presents resonance peaks with amplitudes 40% higher than those obtained experimentally. Although such an approach is very practical and extremely easy to be implemented, the value of the coefficient λ was poorly estimated by the quasi-static test at constant angular velocity. The parameter λ is frequency-dependent. As a result, the adopted value of λ in the simulations led to inaccurate results. The conventional and active lubrication forces may be accurately decoupled using the coefficient λ, only if such a coefficient also becomes dependent on the frequency and on the Sommerfeld number, i.e., $\lambda = \lambda(So, \omega)$. It means that λ shall be estimated by a series of tests in the operational frequency range and not only by a quasi-static test at low frequency.

The first approach, based on the global equivalent dynamic coefficients of the oil film, and obtained from the modified Reynolds equation for the active lubrication, leads to good agreement with the experimental results. Such an approach takes into account both lubrication mechanisms, i.e. conventional and active, the dynamics of the hydraulic system and the control gains. For this reason this approach leads to a more accurate mathematical model. The global dynamic coefficients, obtained from the modified Reynolds equation for the active lubrication, represent in an accurate way the rotor-bearing dynamics under active lubrication.

As mentioned while describing the third approach, special attention should be paid to integral controllers. For optimal conditions, the integral part should bring the rotor as close as possible to the bearing surface, increasing the rotor eccentricity and exploring the nonlinear relationship between damping and eccentricity. Sequentially, the PD-part of the controller shall stabilize the rotor-bearing system even more. If the integral part of the controllers brings the rotor back to the center, the PD-part of the controller will have to compensate for the loss of (passive) damping. It means that the control energy will be used to achieve a rotor-bearing system less stable or a little more stable than the original passive rotor-bearing system. As already mentioned, in the case of high eccentricities, thermal effects may lead to a reduction of the (passive) damping coefficients, as mentioned by Ettles [98]. Such a theoretical investigation should be carried out to obtain further insights into the problem.

3.5 Passive and Active Rotor-Bearing-Blade Systems

This section is divided in two parts. One is dedicated to the passive (without feedback control) behavior of rotor-bearing-blade systems and the other to the active control of rotor-bearing-blade systems.

3.5.1 On the Coupled Rotor-Bearing-Blade Dynamics

The main change detected in the dynamic behavior of a rotating beam is the increase of the natural frequencies with the rotational speed, an effect known as centrifugal stiffening. This effect is only correctly described by a mathematical model if non-linear terms related to the beam deformation are taken into account Anderson [45]. At least a second order linearization scheme of the deformation vector must be employed to ensure that the effect of centrifugal stiffening is appropriately taken into account, once those non-linear terms are responsible for transferring part of rotation energy to the bending motion Simo and Vu-Quoc [236]. Due to the terms of centrifugal stiffening, the application of such a linearization of second order introduces a matrix of geometric stiffness. When the blades are subjected to constant rotational speeds, this matrix may be calculated either from the solution of the initial stress problem or from the normal force acting on the blade Baruh [51]. However, in the most general case, this matrix is not constant, as it depends on the longitudinal displacements of the beam Bakr and Shabana [50].

The influence of the derivative of the geometric stiffness matrix and other higher order non-linear terms, which are usually neglected, can sometimes lead to different solutions Mayo and Domínguez [175]. In contrast, after including those high order non-linear terms into the formulation, the solution becomes extremely sensitive to the shape function used for modeling the axial displacement. Thus, it is necessary to include a large number of axial modes to correctly model the interaction between the axial and transverse motion, otherwise the numerical solution becomes unstable. Mayo et al. [176] discuss the numerical efficiency of the non-linear model for modeling rotating beams. Another approach to this problem is to assume that the axial displacements are originated from transverse deformation without axial deformation. This approach leads to a more efficient and stable numerical model. Moreover, it is not necessary to include any axial mode, depending on the range of frequencies. This kind of formulation, in which the displacement and the axial deformation are uncoupled, is valid only for systems whose axial stiffness is much higher than the transverse one, for example in windmills. Therefore, the correct choice of the beam model is essential to ensure that coupling between the elastic deformation and rotational motions is properly taken into account [236]. The equations of motion for a single cantilever beam, parametrically excited, is presented by Cartmell [74] [110] using Lagrange's formulation. The intention of that work [74] is to illustrate the use of classical engineering theories in the accurate modeling of a very simple structure, and to highlight the conceptualization of such a three-dimensional problem.

In Santos et al. [16] the equations of motion of four rotating cantilever beams, attached to tip masses and to a mass-spring system, are obtained using the Newton-Euler-Jourdain Method. This two-dimensional model is employed to analyze how the lateral vibrations of the mass-spring system (non-rotating parts of a structure) influence the bending vibrations of the blades (rotating parts) and vice-versa. The blade (beam) deformations are obtained using three different approaches: (a) only small displacements are assumed, leading to a linear deformation vector and thus to a linear model of the whole assembly; (b) large displacements are included, but to obtain the deformation vector only the non-linear terms of second order are considered, resulting in linear equations with a new matrix due to the geometric stiffness; (c) a fully non-linear model for the deformation of the beam. Experimental and numerical results obtained with these three different models are presented, clarifying their limitations and the range of application. Comparing the results coming from the three different mathematical models to the experimental ones, the second order linearized and fully non-linear models are those which correctly predict the stiffening effect and coupled parametric vibrations of the rotor-blade assembly. The linear model can correctly estimate parametric vibrations only in a short range of low rotational speeds, where the stiffening effect is of minor importance. Super-harmonic vibrations associated to the movement of the flexible rotating parts (with large amplitude) can only be detected with help of the fully non-linear model.

From the viewpoint of vibration monitoring in rotating flexible structures one can conclude that the positioning of sensors at stationary or rotating reference frames will lead to peaks in different frequencies. With sensors attached to the non-rotating part of the structure (inertial reference frame) one will detect parametric vibrations related to the coupled mode shapes of the flexible rotating parts and flexible non-rotating parts. Such information is very important when feedback control systems have to be designed.

In rotor-bearing systems coupled to flexible blades, the lighter the flexible blades are, the closer are the BB-modes and the BR-modes. It is important to point out that BB-modes are the modes where the structure does not vibrate and only blade vibrations are observed. BR-modes are the modes where the movements of the structure and the rotating blades are coupled. For more details, see Santos et al. [16]. It means that a long acquisition time is necessary in order to achieve a reasonable precision in the frequency domain and to correctly identify both frequencies and modes.

3.5.2 On the Control Design for Rotor-Bearing-Blade Systems

Active modal control of vibrations in flexible structures has been extensively studied for several decades and applied to various types of systems Balas [49] Firoozian and Stanway [107] Kaneko and Kano [152] Chantalakhana and Stanway [76] Khulief [153] suppressing vibrations among others in flexible rotors, plates and rotating beams. Nevertheless, most of the reported work only deals with linear time-invariant systems. Active control of vibrations in flexible rotating structures, such as flexible bladed rotors or rotating flexible beams, has been also extensively investigated by many authors under the assumption of time-invariant as well as time-variant systems. Vibrations in rotating flexible blades using active piezoelectric actuation have been theoretically as well as experimentally investigated among others by Chen and Chopra [77] Baz and Ro [52]. Active modal control of vibrations in a simple rotating cantilever beam is studied by Khulief [153]. In all three reported works the rotor lateral movement is neglected. When such a movement is also considered and the rotor operates at constant angular velocity the system becomes periodic time-variant. In this case, a lack in the literature dealing with the problem of control design with focus on blade vibration can be detected.

Flexible rotating blades interacting with rotor lateral movement present some special dynamic peculiarities, which have to be clearly understood before active systems for controlling vibrations can be properly designed and applied. As already mentioned, due to the centrifugal effects the blade natural frequencies may significantly change depending on the rotor angular velocity, the so-called stiffening effect. Moreover, the variation of the blade position, when operating at constant angular velocities, leads to a periodic variation of the inertia distribution in time, which can induce the appearance of parametric vibrations. Furthermore, the coupled rotor-blade mode shapes can occur extremely close to the flexural blade modes, hindering a precise identification of the different frequencies and introducing beating phenomena. The smaller the mass relationship between blades and rotor is, the closer the coupled rotor-blade modes and the flexural blade modes are, Santos et al. [16].

Several control methodologies, directly pointed towards periodic systems, have been reported during the last two decades. An optimal periodic controller for such a system is designed by solving a periodic Ricatti equation Arcara et. al [47]. Lyapunov-Floquet transformation of the periodic system into a time-invariant form is used by other authors. By using this approach, linear time-invariant control technique can be applied to design controllers and their optimal gains. A technique for designing a pole placement state feedback controller by gain selection based on a modal transformation of the periodic system is described in Calico and Wiesel [71]. In order to avoid real time estimation of state variables in the periodic system, a design technique for an optimal output feedback controller with fixed gains based on a modal transformation of the periodic system is proposed in Calise et. al [72]. A method for designing periodic state feedback controllers using a Lyapunov-Floquet transformation method based on Chebyshev polynomials is reported in Sinha and Joseph [237]. The efficiency of this methodology has been investigated in different types of systems, i.e., to control parametric vibrations in a single rotating flexible beam Marghitu et. al [173] and to reduce vibrations in bladed disks by shaft based actuation Szász and Flowers [251].

In Christensen and Santos [18] a periodic time-variant modal controller is designed, aiming at attenuating vibrations in a coupled rotor-blade system built by four flexible rotating blades. While designing modal controllers, it is not always desired to address and suppress all vibration modes, but only those which can cause significant damages to the machines. Such a selection minimizes the order, complexity and energy consumption of the controller. By combining elements from the previously reported works, a state feedback controller is designed similar to the method presented in [237], based on Chebyshev polynomials. However, in order to address the modal control concept, the system is transformed into a time-invariant form by the modal transformation technique for periodic systems, as presented in [72] and described in detail in Xu and Gasch [272]. Among the advantages of using the modal transformation for the controller design one can mention that it allows a straightforward method for analyzing the modal controllability and observability of the time-varying system. Moreover, it gives better physical interpretation and visualization of the time-varying mode shapes which are very useful e.g. for decisions regarding actuator and sensor placement.

Implementation of active vibration control into any mechanical structure implies that actuators and sensors have to be distributed in the structure. Numerous difficulties relate to the practical implementation of actuators and sensors into rotating frames, i.e., rotor-blades. Therefore, it is of significant interest to study where to locate sensors and actuators in order to be capable of monitoring and controlling all vibration levels of a bladed disc. It is worth analyzing whether direct blade actuation and sensing are absolutely necessary or blade as well as rotor vibration can be controlled by means of only shaft actuation. The controllability and observability analysis of bladed discs, where flexible blades interact with disc lateral movement, presents several difficulties to be overcome. The dynamics of this kind of system is characterized by peculiarities such as centrifugally stiffened blades, a time-variant dynamic model, parametric vibration modes and the frequency veering phenomenon, as already mentioned. These dynamical characteristics imply that the levels of controllability and observability become time-variant and dependent on the angular velocity. In the literature, active vibration control has been applied to various types of rotating flexible bladed machines, among those helicopter rotors. Typically, the vibrations of such kind of rotating flexible systems are controlled using actuators and sensors attached directly to each blade. Moreover, no vibration coupling among rotor lateral movement and flexible blade motion is typically considered. The controllability and observability of a bladed disc via shaft-based actuation and sensing have only been studied by very few researchers, among them Szász and Flowers [249] [251] Szász et al. [250].

In Christensen and Santos [21] the controllability and observability are investigated by analyzing the system eigenvectors (mode shapes) but considering only the constant parts of mode shapes. The parametric time-variant rotating parts of mode shapes are not considered. It is shown that disc and blade vibrations can be monitored and controlled by using only shaft-based actuation and sensing if the blades are properly mistuned. In Christensen and Santos [19] though the controllability and observability of a mistuned bladed disc are studied in details aided by time-variant modal analysis. The time-variant parametric mode shape components are considered.

In Christensen and Santos [22] one of the aims of the work is to show how and why bladed discs become controllable and observable using only shaft-based actuation and sensing if the blades are properly mistuned via the presence of vibration coupling and parametric modes. The modal controllability and observability of the bladed disc depend significantly on disc angular velocity. At certain angular velocities basic and parametric vibration mode components will “coincide” and interact resulting in varying degrees of controllability and observability. The degree of controllability and observability will also vary with angular velocity due to centrifugal stiffening of the blades. The results reveal the theoretical potential of controlling blade vibrations by using shaft-based actuation, i.e., active electro-magnetic or hydraulic bearings. However, it also shows the complexity of such a methodology. The levels of controllability and observability are relatively small and dependent on their angular velocity, making practical use of such vibration control methodology questionable on a short view.

In Christensen and Santos [21] [22] experimental and practical aspects related to the implementation of active vibration controllers for coupled rotor-blade systems have been studied. In [21] horizontal rotating machines are investigated. The flexible blades are periodically excited by the gravity force. In [22] though vertical rotors are focused. The blades are not significantly excited by the gravity force and the controllers designed are not using energy to overcome such periodical excitations. The experimental tests are conducted using the test setup shown in figure 2.4.2. The tests are split into three types; i) experimental tests of a non-controlled system; ii) tests of an actively controlled tuned system and iii) test of an actively controlled mistuned system.

Chapter 4

Theoretical and Experimental Achievements

This chapter is intended to give an overview about the theoretical and experimental achievements toward the application of the mechatronic devices presented in the last two chapters. This chapter is divided in four sections. The first is devoted to the application of mechatronics to bearings. The second is dedicated to the active vibration control of rigid and flexible rotating shafts. The third section deals with the active control of rotor-blade vibrations, and finally the fourth section gives an overview about the feasibility of industrial applications.

4.1 Bearings

4.1.1 Modification of Oil Film Stiffness and Damping Coefficients

As previously mentioned, one of the different ways of modifying the oil film coefficients is by using constant hydraulic pressure in the chamber systems to adjust the bearing gap. In figure 4.1.1(a) it is possible to compare some theoretical and experimental values of stiffness and damping coefficients of the controllable tilting-pad bearing illustrated in figure 2.1.1. The bearing coefficients are changed only in the vertical direction. Two different rotor angular frequencies are chosen: 1200 rpm (20 Hz) and 2400 rpm (40 Hz). Two different hydraulic pressures are applied to the chambers: 0 and 4 bar. The experimental values of the bearing coefficients were obtained by means of an identification procedure based on the frequency domain and using least-square method Santos [4]. The identification procedure uses the transfer function as data, i.e., amplitude and phase of the rotor-lever-bearing system illustrated in figure 2.1.2. The rotor-lever-bearing-system is excited by the piezoelectric actuator from 0 until 200 Hz. Rotor-lever-bearing transfer functions are illustrated in figure 4.1.1(b).

The experimental damping and stiffness coefficients can be well predicted by means of the theoretical model presented in Santos [2]. The behavior of such coefficients in the range of the perturbation frequencies of 0 until 200 Hz is shown in figure 4.1.1(a), when the rotor operates at a constant speed of 1200 rpm (20 Hz) and 2400 rpm (40 Hz). The experimental damping coefficients are smaller than the coefficients predicted by the theoretical model. This claim is also mentioned by Ettles [98], Lund [169], Earles et al. [93] [94] and White [270]. The discrepancies between theoretical and experimental values around the frequency of 105 Hz, i.e., the resonance of the rotor-lever-bearing system, can be explained through the non-linearities of the bearing coefficients regarding the dependency of such coefficients on the displacements of the rotor and of the pads. Around this frequency, the vibration amplitude of the rotor increases, and averaged values of damping and stiffness are identified. Moreover, small discrepancies between theoretical and experimental transfer functions influence the identification procedure significantly. Such discrepancies are more clear around the first natural frequency of the test rig. The discrepancies between theoretical and experimental damping coefficients are bigger than the discrepancies between theoretical and experimental stiffness coefficients.

Figure 4.1.1(b) shows the vertical transfer function of the rotor-lever-bearing system, when the rotor operates at a constant angular velocity of 2400 rpm (40 Hz), and the perturbation frequency introduced by means of the piezoelectric actuator is varied from 0 until 200 Hz. Three values of the chamber pressure are selected, aiming at modifying the bearing coefficients and reducing system vibrations: 0, 4 and 14 bar. Analyzing figure 4.1.1(b) it is possible to recognize the increase of bearing stiffness through the reduction of the vibration amplitudes of the rotor in the low frequency range. The increase of the damping coefficient can be observed by means of the reduction of the vibration amplitudes of the rotor around the resonance of the system.

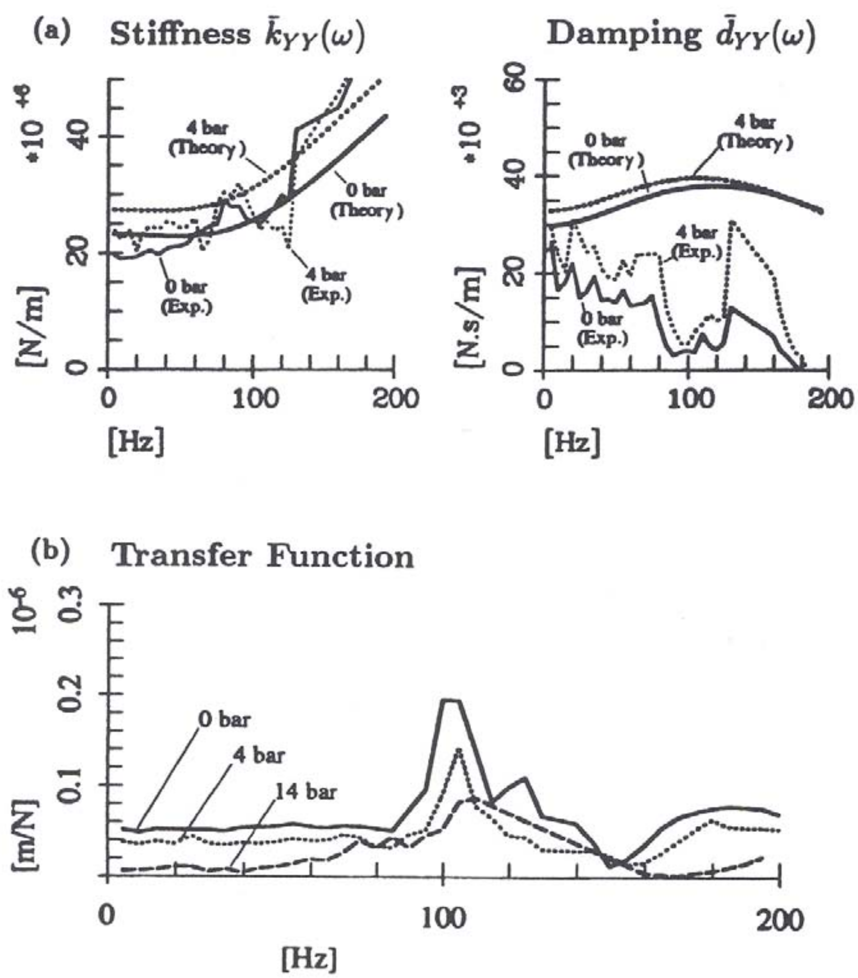


Figure 4.1.1: Modification of the reduced coefficient of the tilting-pad journal bearings at the rotor speed of 40 Hz – (a) behavior of the vertical coefficients of the bearing as a function of the perturbation frequency for two different values of hydraulic chamber pressures: 0 and 4 bar; (b) transfer function of the rotor-lever-bearing system in the vertical direction as a function of the perturbation frequency at a rotor speed of 40 Hz and chamber pressures of 0, 4 and 14 bar.

Another application of constant chamber pressures is demonstrated theoretically as well as experimentally in Santos and Nicoletti [5]: the elimination of pad flutter by adjusting the bearing gap. Pad flutter is typically seen when the bearing gap between rotor and some of the pads becomes too large due to, for example extreme high static loading, uncorrect design of bearing pre-load factor and positioning pivot angles. Pad flutter is measured using the eddy-current displacement sensors installed in the pads, as illustrated in figure 2.1.1. The mathematical model is developed in [5] and shows the dependency of the self-excited vibrations on the bearing coefficients. Some experimental results, illustrating the elimination of flutter, can be seen in figures 7 and 8 of the paper [5].

As explained in chapter 2, by using constant hydraulic pressure, the membrane is deformed, pushing the pads against the rotor and changing the bearing oil film. This results in an increase of the global damping coefficient and in an increase in the stability reserve of the rotor-bearing system, as illustrated before. However, the deformation and threshold strain of the membrane limits the modification of the oil film thickness. It is known that, for rotating systems operating at very high speeds, the stiffness of the oil film becomes very high and the damping very low. To change the oil film thickness, it is necessary to have extreme high pressure values in the chambers. Such high pressures are unpractical due to strain and stress in the actuator membrane. Two alternative approaches are: (a) to act directly on the flow behavior of the lubricant in the bearing gap or (b) to use feedback control strategies and operate actively.

By using constant hydraulic pressures, the limitations in changing the bearing coefficients are associated to the operational conditions of the rotor-bearing system, expressed in terms of the Sommerfeld number. At very high angular velocities, it is difficult to vary or to adjust the bearing gap towards obtaining significant changes in the damping coefficients. By using feedback control strategies, alternative (b), the limitations are associated to the actuator dynamics (frequency range) and the energy used by the controller. Results will be presented and discussed in section 4.2. The results obtained using the alternative approach (a) are elucidated, as follows.

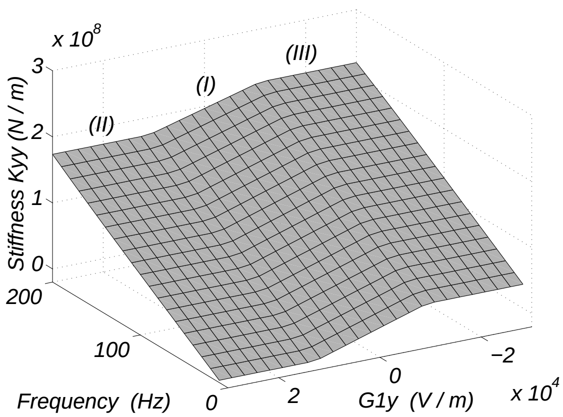
The influence of the pressure and flow in the bearing gap can be achieved by the application of hybrid and/or active lubrication techniques. By setting the control gains G_{1y}, G_{1z}, G_{2y} and G_{2z} in equations (3.3.2) and (3.3.2) to zero, and varying the values of the static injection pressure ($P_{inj_{st}}$), the bearing operates under hybrid lubrication mode. One sensitively modifies the bearing stiffness coefficients, whereas damping coefficients remain almost constant. By setting the control gains different from zero, the active lubrication mode is obtained. The operational linear range of servo valves is limited to 5% of its nominal maximum control signal Schäfer [230] Althaus [42]. Besides, dynamic coefficients are only theoretically valid for infinitesimal displacements. However, according to Lund and Thomsen [168], dynamic coefficients may be used in practical applications for amplitudes up to 50% of the bearing clearance. With this in mind, the following restrictions of control signal and vibration amplitude were applied to the dynamic coefficients calculations:

$$
\begin{aligned}
\frac{|u_y|}{|Y_h|} &= \sqrt{G_{1y}^2 + \omega^2\, G_{2y}^2} \leq \frac{0.25}{0.3\, h_0} \\
\frac{|u_z|}{|Z_h|} &= \sqrt{G_{1z}^2 + \omega^2\, G_{2z}^2} \leq \frac{0.25}{0.3\, h_0}
\end{aligned}
\tag{4.1.1}
$$

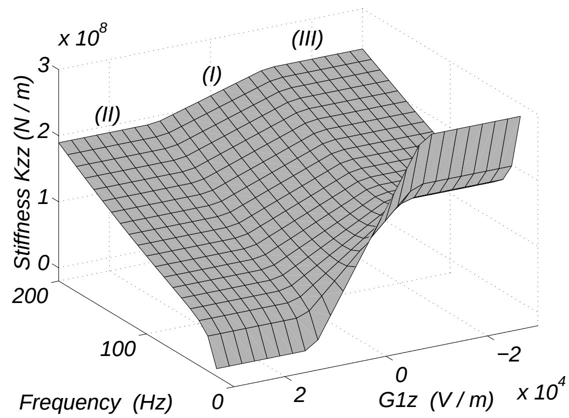
where an amplitude limitation of 30% of the assembled clearance is chosen.

Active Lubrication using P-Controllers – by adopting static pressures of 0.1 MPa in all pads, and varying the proportional control gains (G_{1y} and G_{1z}), while keeping the derivative gains

(G_{2y} and G_{2z}) as zero, one achieves a modification of the stiffness coefficients, while the damping is not much altered. Figures 4.1.2(a) and 4.1.2(b) illustrate the behavior of the horizontal and vertical main stiffness coefficients as a function of the proportional gains and rotating frequencies. In these figures, for a given control gain, the coefficients K_{yy} and K_{zz} have a standard behavior as function of the rotating frequency, i.e., they increase with the frequency Someya [240]. At a given frequency and varying the control gain, one can detect three different regions in these same figures: region (I) with control gains between $-1.5 \cdot 10^4\ V/m$ and $+1.5 \cdot 10^4\ V/m$; region (II) with control gains smaller than $-1.5 \cdot 10^4\ V/m$; and region (III) with control gains larger than $+1.5 \cdot 10^4\ V/m$. In region (I), the coefficients K_{yy} and K_{zz} vary linearly as function of the control gains. When the control voltage reaches the limits established by equation (4.1.1), the coefficients stop varying and remain constant in the so far achieved values, thus forming regions (II) and (III). The damping coefficients are not much altered by the proportional gains, thus being not presented here. The cross-coupling stiffness coefficients are also not presented since they are negligible, as in the conventional lubrication case of tilting-pad bearings.



(a)

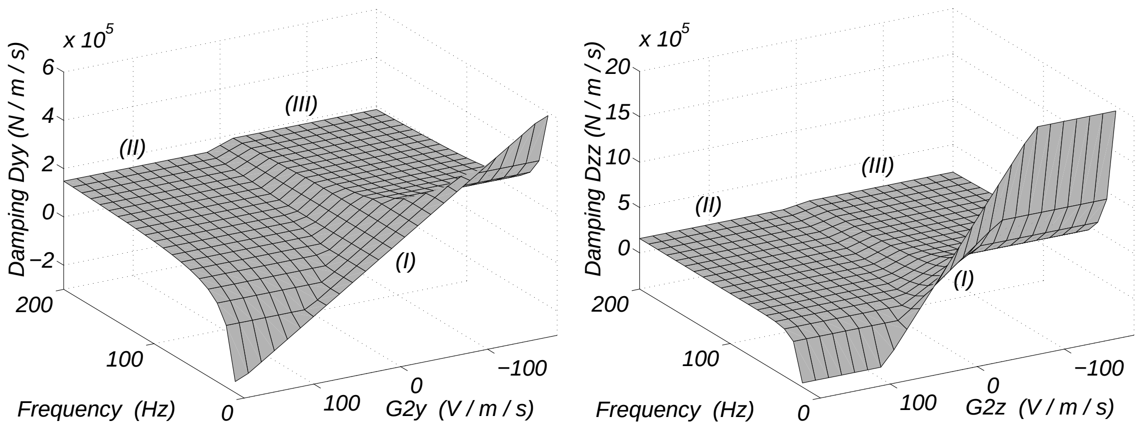


(b)

Figure 4.1.2: Active lubricated tilting-pad journal bearings – (a) synchronous bearing stiffness coefficient in horizontal direction as a function of the rotating frequency and proportional gain G_{1y}; (b) synchronous bearing stiffness in vertical direction as a function of the rotating frequency and proportional gain G_{1z}.

Active Lubrication using D-Controllers – By setting once again the static pressures to 0.1 MPa, but varying the derivative control gains (G_{2y} and G_{2z}), while keeping proportional gains as zero, one achieves a sensitive modification of the damping coefficients. By this time, the stiffness coeficients are not much altered. Figures 4.1.3(a) and 4.1.3(b) illustrate the behavior of the horizontal and vertical main damping coefficients as a function of the derivative gains and rotating frequency.

In these figures, when the control gain is zero, the coefficients D_{yy} and D_{zz} have a standard behavior as function of the rotating frequency, i.e., they decrease with the frequency [240]. However, for a given frequency and varying the control gain, one can detect three different regions in figures 4.1.3(a) and 4.1.3(b) similar to those found in figures 4.1.2(a) and 4.1.2(b). In region (I), the coefficients D_{yy} and D_{zz} vary linearly as function of the control gains. Such a region narrows with the increase of the rotating frequency. When the control voltage reaches the limits established by equation (4.1.1), the coefficients stop varying and remain constant in the so far achieved values, thus forming regions (II) and (III). The narrowing of the region (I) is



(a) (b)

Figure 4.1.3: Active lubricated tilting-pad journal bearings – (a) synchronous bearing damping in horizontal direction as function of the rotating frequency and derivative gain G_{2y}; (b) synchronous bearing damping in vertical direction as function of the rotating frequency and derivative gain G_{2z}.

caused by the term ω in equation (4.1.1), leading the maximum allowed derivative gain to be a function of the inverse of the frequency $\left(G_{2y}^{max} = \frac{|u_y|}{|Y_h|\cdot\omega} = \frac{0.25}{0.3\,h_0\cdot\omega}\text{ and } G_{2z}^{max} = \frac{|u_z|}{|Z_h|\cdot\omega} = \frac{0.25}{0.3\,h_0\cdot\omega}\right)$.

The stiffness coefficients are not much altered by the derivative gains, thus being not presented here. The cross-coupling damping coefficients are also not presented since they are negligible, as in the conventional lubrication case of tilting-pad bearings.

The experimental validation of such curves is not a trivial task. Obtaining the bearing stiffness and damping coefficients by means of identification procedures and inverse methods demands extremely precise calibration of all sensors. Moreover, identification procedures are normally based on dynamic models which always present some levels of inaccuracy. Sensitivity and error propagation analyzes are fundamental to assure correct interpretation of experimental results. Up to now it was not possible to identify such active bearing coefficients with appropriate accuracy to clearly detect the dependency of the active bearing coefficients on the excitation frequency. In all test rigs, the non-modeled dynamics heavily influences the identification of bearing parameters in the frequency domain. It is possible to obtain with a good accuracy the values of damping ratio ξ of rotor-bearing systems using the information around the resonance ranges, as it will be illustrated in the section 4.1.5, but the variation of parameters according to the excitation frequency is still a challenge. Efforts are still made to experimentally obtain figures 4.1.2 and 4.1.3.

One of the main advantages of incorporating the PD-control gains inside the modified Reynolds equation and obtaining the bearing coefficients using perturbation technique is the feasibility of elaborating "controllable journal bearing tables". Journal bearing tables for conventional hydrodynamic bearings are presented by Lund and Thomsen [168] and Someya [240]. Such tables do not take into consideration the supply pressure and the control gains. In the near future, the control parameters may be incorporated into "controllable journal bearing tables", and the integration of mechanics, electronics and informatics (mechatronics) may become even more clear in the field of Tribology.

4.1.2 Compensation of Cross-Coupling Effects

Tilting-pad bearings do not have a significant cross-coupling effect, but multirecess journal bearings do, specially at high angular velocities. The cross-coupling effect comes mainly from the hydrodynamic effect in the landing areas, see figure 2.2.6(c). In Santos and Watanabe [12] detailed theoretical investigation is carried out in order to achieve appropriate mathematical models to describe the behavior of multirecess journal bearings under active lubrication conditions. Santos and Watanabe [15] show that the gain factor k_g, illustrated in the schematic view of a multirecess journal bearing with 2 pair of active pockets, figure 2.2.6(a), can significantly influence the cross coupling stiffness of the bearing. Furthermore, when varying the gain of the proportional controller using cross-coupling compensation, significant modifications of the direct and cross-coupling stiffness coefficients are achieved, but the damping coefficients are insignificantly affected by the proportional controller. The corresponding critical mass and critical whirl frequency ratio can be very much improved. When the proportional gain is varied, whereas $k_g = 0$, i.e., no compensation of cross-coupling stiffness, no significant changes of critical whirl frequency ratio are achieved. The critical whirl frequency ratio varies slightly around 0.47. The critical mass values can be slightly increased or reduced, depending on the positive or negative value of the proportional gain g_1. When $k_g \neq 0$ (controller with compensation of cross-coupling stiffness), significant changes of critical whirl frequency ratio and critical mass can be achieved. It is shown in Santos and Watanabe [15] that for $k_g = 3$ and a proportional gain $g_1 = 10^4\ V/m$, the critical whirl frequency ratio is reduced from 0.471 (passive case) to 0.170 while the critical mass is also strongly increased from 2.15 to 10.9. Such achievements towards to a more stable rotor-bearing system are obtained due to three simultaneous actions: (a) reduction of the direct stiffness coefficients; (b) slight increase of the direct damping; (c) a significant modification of the cross-coupling coefficients. The crossed retrofitting of the rotor linear displacement signals with a simultaneous inversion of their signals almost allows a cancelation of the cross coupling stiffness in a tiny range of proportional gain values. From the point of view of control implementation in a variable-speed rotating machine, an adaptive self-tuning controller, with a variable gain k_g is necessary to compensate the cross coupling effect.

4.1.3 Compensation of Thermal Effects

In Santos and Nicoletti [8] thermo-hydrodynamic analysis is carried out for hybrid lubricated tilting-pad journal bearings with multiples orifices. At lightly loading conditions, low rotor angular velocities and low injection pressures one obtains the best cooling effects on the pad surface by the radial oil injection. Increasing the injection pressure for low rotor angular velocities, one achieves an increase of the oil temperature inside of the orifices due to an increase of the viscous dissipation (high injection velocities). By increasing the rotor angular velocity and consequently the hydrodynamic pressures, outlet flow into the orifices can be detected if the injection pressure is low. Moreover, no cooling effect over the pad surface is observed in such a case. Although the injection pressure could overcome the hydrodynamic pressure, resulting in inlet flow of the cooled oil into the bearing gap, the cooled area over the pad would be restricted to the tangential direction. Such an effect is a consequence of the predominant lubricant flow in the tangential direction and is clearly illustrated in figures 4.1.4(a) and 4.1.4(b), with 5 and 15 orifices distributed over the pad surface.

It is shown in Santos and Nicoletti [9] that the positions of the orifices play a very important role on the cooling effect over the pads, but such effect is locally detectable forward the orifices, in the pad longitudinal direction. For the temperature distribution, it is better to machine the orifices close to the edges, where the hydrodynamic pressures are lower and an inlet flow of the

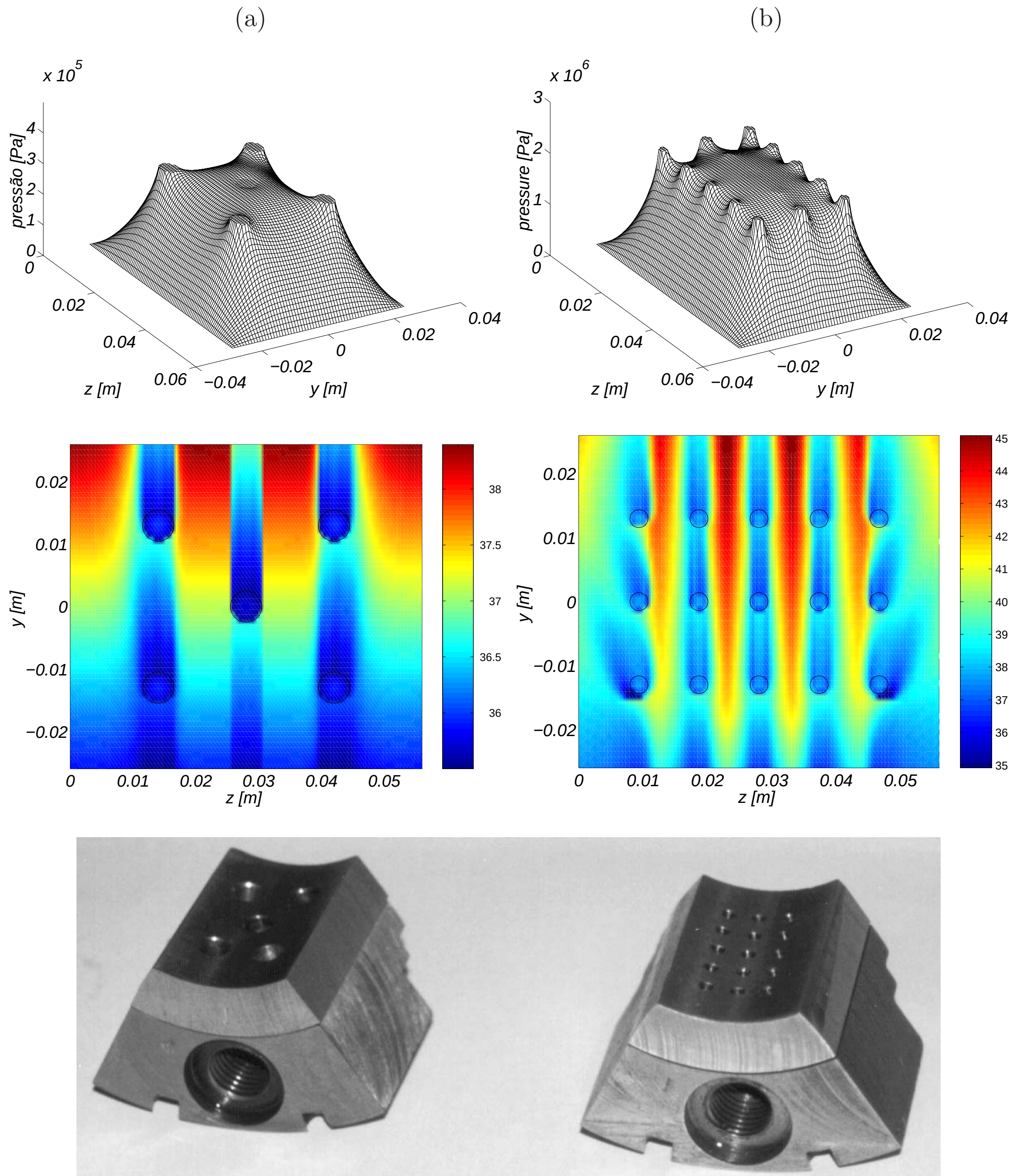


Figure 4.1.4: Theoretical results - compensation of thermal effects: (a) pad surface with 5 five orifices; (b) pad surface with 15 orifices.

cooled oil into the bearing gap is more easily ensured. Regarding the dimension of the orifices, a strong reduction of their diameter may cause an increasing of the viscous dissipation inside of the orifices and thus, a further reduction of the bearing cooling effect is expected.

4.1.4 Reduction of Friction Between Rotating and Stationary Parts

In the theoretical contribution presented by Heinrichson and Santos [29] a 3-dimensional thermo-elasto-hydrodynamic laminar flow model based on the Reynolds equation has been used to study the effect of equipping tilting-pad thrust bearings with deep recesses with the purpose of reducing the friction loss.

It can be concluded that, in a bearing pad of finite width the potential for reducing friction is smaller. A study of rectangular bearing pads shows a 10 % reduction with a length-to-width ratio of 2.0 while a larger energy saving is possible for low length-to-width ratios. At a ratio of 0.2, a saving of 24 % is possible. Optimal values of recess lengths and widths, and the corresponding optimal positions of the pivot points are stated for length-to-width ratios ranging from 0.2 to 2.0. An analysis shows that the sensitivity of the bearing pads to the circumferential positioning of the pivot is greater than in conventional bearings. Therefore, in order to achieve a superior performance compared to a conventional bearing, it is imperative that the pivot is placed close to the optimal position [29].

The design suggestions given for rectangular pads have subsequently been applied to sector shaped thrust bearings. For the common 6 to 8-pad bearings an energy saving of 10 - 15 % is obtained. The bearing pads are very sensitive to the radial position of the pivot point, as mentioned in [29]. Spring bed bearings or Michell type bearings with line pivots are therefore preferable to bearings with spherical pivots.

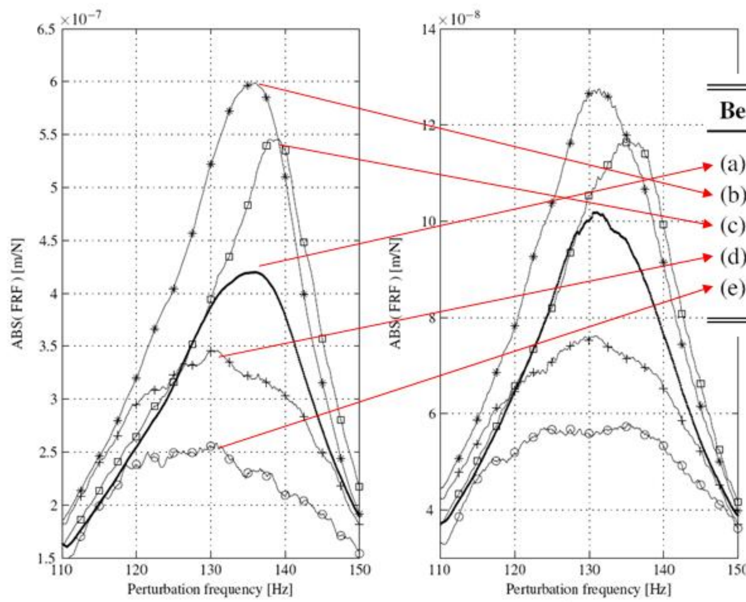
One of the main contributions given by Heinrichson and Santos [29] is the design charts intended to state the necessary geometric changes when redesigning a tilting-pad thrust bearing from a conventional design with plain pads to a design with deep recesses in the high-pressure regions. Recesses with the suggested sizes and positions can be used as oil injection pockets for hydrostatic jacking at start-up. Because they are positioned to the trailing edge side of the pivot point, the pads may however tilt backwards at start-up. This could lead to pad flutter and mixed lubrication when accelerating the rotor. Using recesses with larger circumferential dimensions can solve this problem by guaranteeing positive circumferential tilt at lifting off.

4.1.5 Smart Bearings – Rotordynamic Testing and Parameter Identification

When hydraulic or magnetic active forces are measured with high accuracy and, simultaneously, displacement, acceleration and force transducers are properly adapted to the system, it is possible to develop “smart bearings” able to properly perform rotordynamic tests and aid the identification of system parameters. It is shown for active magnetic bearings and active lubricated bearings in the following.

Using the magnetic bearing with hall sensor illustrated in figure 2.3.1, carefully calibrated in Kjølhede and Santos [24], and an electromagnetic shaker with a force transducer attached between shaker and rotor, the identification of system damping ratio ξ is performed. The experimental results are illustrated in figure 4.1.5. The rotor operates with a constant angular velocity of 1000 rpm and the tilting-pad bearing operates under the following conditions: (a) conventional hydrodynamic lubrication; (b) hybrid lubrication with $P_{inj} = 70\ bar$; (c) active lubrication with a P-controller and supply pressure of 70 bar; (d) active lubrication with a PD-controller and supply pressure of 70 bar; (e) active lubrication with a PD-controller and supply pressure of 100 bar; The reduction of vibration amplitudes around the resonance of the rotor-bearing system can be clearly seen. The increase in the damping factor obtained via active lubrication is experimentally demonstrated and presented in the table incorporated into figure 4.1.5.

As mentioned before, the identification of damping ratio ξ is done with good accuracy. The two experimental procedures are compared. Nevertheless, the identification of damping and stiffness coefficients as a function of the excitation frequency and control gains is still very poor and



Damping Factor ξ (experimental identification)

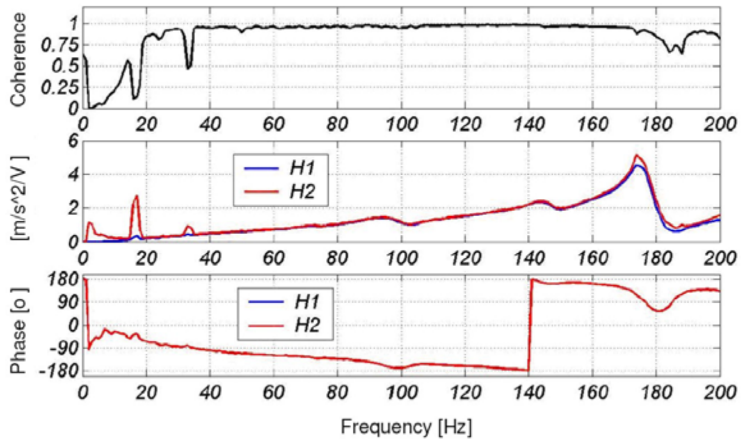
Bearing config.	Shaker	AMB
(a) TPJB	0.071	0.074
(b) ALTPJB, 70 bar	0.060	0.072
(c) ALTPJB, 70 bar, $P_{contr.}$	0.056	0.063
(d) ALTPJB, 70 bar, $PD_{contr.}$	0.120	0.122
(e) ALTPJB, 100 bar, $PD_{contr.}$	0.125	0.130

Figure 4.1.5: Experimental results - identification of damping ratio ξ of the rotor-bearing system at constant angular velocity of 1000 rpm and different lubrication conditions using a "smart active magnetic bearing" and an electromagnetic shaker: (a) conventional hydrodynamics; (b) hybrid lubrication with $P_{inj} = 70\ bar$; (c) active lubrication with a P-controller and supply pressure of 70 bar; (d) active lubrication with a PD-controller and supply pressure of 70 bar; (e) active lubrication with a PD-controller and supply pressure of 100 bar.

the results are not presented here. Research activities toward the identification of frequency-dependent coefficients are still being carried out.

The characterization of active oil film forces is a complicated task and depends on many parameters: Sommerfeld number, i.e., journal angular velocity, bearing load, oil viscosity, bearing gap, bearing dimensions, pre-load factor, orifice diameter, excitation frequency, feedback control gain, as well as on the dynamic parameters of the servo valves, i.e., their natural frequencies, damping factors and pressure-flow coefficients.

In Santos [30] the rotor-bearing system illustrated in figure 4.1.6(b) is excited by the active oil film forces using a slow-sine sweep function as servo valve input signal. The experimental frequency response function (FRF) is illustrated in figure 4.1.6(a). The estimators H1, H2 and the coherence function are defined in Ewins [101]. Both estimators H1 and H2 are presented and given in acceleration units divided by volts. Good coherence is achieved between 40 and 170 Hz, as it can be seen in figure 4.1.6(a). The feasibility of using the active lubricated bearing as a "calibrated shaker" is strongly dependent on the "active force maps" presented in figure 4.1.7. Such maps translate the servo valve input signal (given in volts) into values of force in Newtons. In that way the functions H1 and H2 can be given in acceleration units divided by Newtons. By using the least-square method the error between the theoretical and experimental frequency response function (FRF) is minimized and the behavior of the active oil film force is determined at different operational conditions. In figure 4.1.7 the theoretical and experimental behaviors of the active oil film forces as a function of the excitation frequency and the servo valve gain are illustrated. The rotor-bearing system operates with a constant angular velocity of 1000 rpm

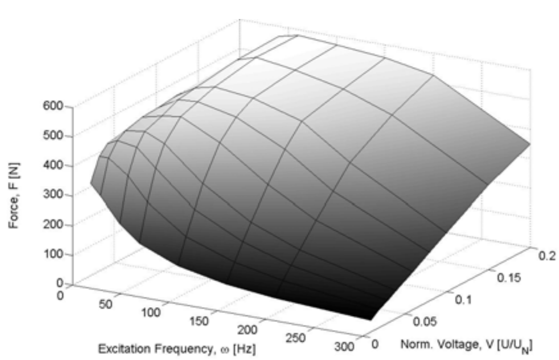


(a)

(b)

Figure 4.1.6: Experimental results - active lubricated bearing used as shaker: (a) experimental frequency response function obtained via active lubrication; (b) flexible rotating shaft supported by an active lubricated TPJB.

and is excited in the range of $0 - 200\ Hz$ via slow-sine sweep signals using active oil film forces supplied by a pressure of 10 MPa. The gain is varied generating servo valve input signals in a range of 0 to 20% of the nominal voltage U_N. The experimental vibration response of the rotor is measured using acceleration transducers attached to an auxiliary bearing fixed at one of the shaft extremities. The theoretical FRF is obtained by solving the modified Reynolds equation for the active lubrication coupled to a Finite Element Model (FEM) for the flexible shaft. By minimizing the error between the theoretical and experimental FRFs the behavior of the active oil film forces is characterized and the diagram of active oil film force versus frequency versus gain is built. The theoretical and experimental diagrams show that the active oil film forces decrease "exponentially" as a function of the excitation frequency and increase linearly as a function of the input signal gain. Saturation ranges are also identified. Differences between the FEM theoretical model and the test rig dynamics lead to severe discrepancies between the theoretical and experimental diagrams. Based on this claim, theoretical results are presented only up to 150 Hz. In order to achieve more accurate diagrams in frequency ranges over 150 Hz the theoretical FEM-model shall be improved and adjusted based on experimental dynamic tests.



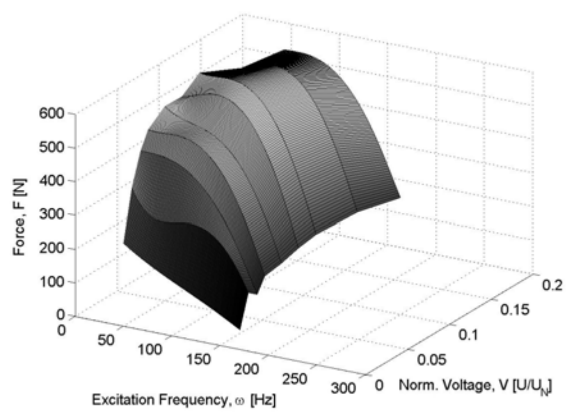
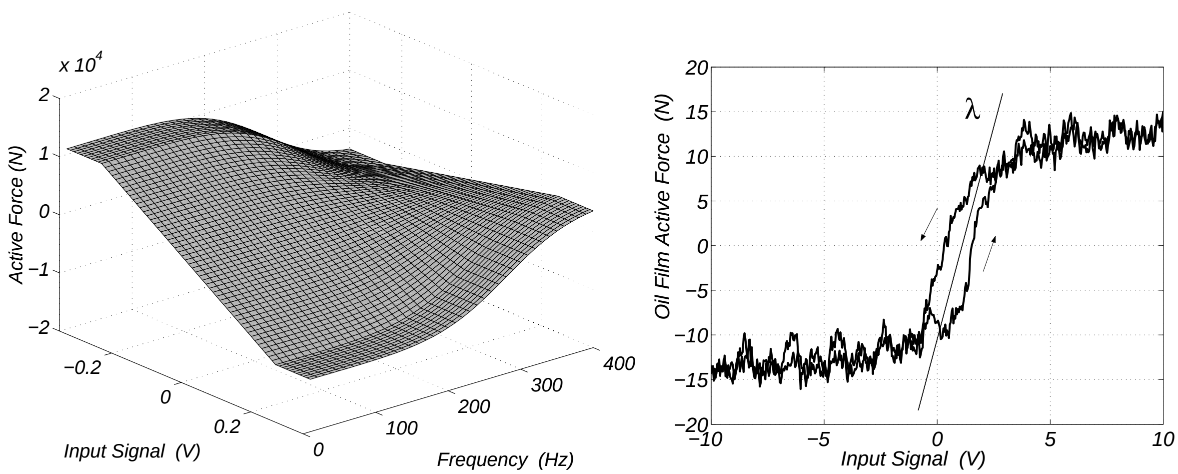


Figure 4.1.7: (a) Theoretical and (b) experimental characterization of the active lubrication forces as a function of the excitation frequency and signal amplitude used as servo valve input.

Figure 4.1.8(a) illustrates the theoretical behavior of the active lubrication forces as a function of the excitation frequency and amplitude of the input signal to the servo valve given in Nicoletti and Santos [28] and figure 4.1.8(b) shows the experimental results coming from the characterization of the active lubrication forces using a quasi-static test and the determination of the coefficient λ given in Nicoletti and Santos [20]. As mentioned in the last chapter, the conventional and active lubrication forces may be accurately decoupled using the coefficient λ, only if such a coefficient also becomes dependent on the frequency and on the Sommerfeld number, i.e., $\lambda = \lambda(So, \omega)$. It means that λ shall be estimated by a series of tests in the operational frequency range, not only by a quasi-static test at low frequency, as it is illustrated in figure 4.1.8(b). Moreover, the influence of hysteresis effect as a function of the excitation frequency has to be carefully investigated, aiming at correctly dealing with the nonlinear characteristics of the hydraulic control system.



(a) (b)

Figure 4.1.8: Qualitative comparison of the active oil film forces and the experimental determination of the coefficient λ used to linearized the oil film active forces – (a) theoretical behavior of the active lubrication forces as a function of the excitation frequency and amplitude of the input signal to the servo valve; (b) experimental characterization of the active lubrication forces using a quasi-static test and determination of the coefficient λ.

4.2 Shafts

4.2.1 Active Chamber Systems – Active Vibration Control of Rigid Shafts

Figure 4.2.1 shows the transfer function of the rotor-lever-bearing system supported by the tilting-pad journal bearing in five cases: (a) the tilting-pad journal bearing operates without control at constant angular velocity of 2400 rpm (40 Hz); (b) The chamber pressure is fixed at about 4 bar (0.4 MPa) without feedback control; (c) in this active case, the feedback is constructed with the signal of the pressure sensors mounted in the chamber system. The optimal gain of the pressure feedback is found with help of the theoretical model; (d) The optimal gain of the pressure feedback is fixed and an optimal value of the feedback of the derivative of the displacement signal of the center of rotor is calculated; (e) The feedback is also designed with the signal of the displacement of the center of the rotor. The optimal gains of the pressure and the velocity feedback signals are fixed and the gain of the feedback of the linear displacement of the center of the rotor is changed. One can see that the rotating system performance with the active bearing in case (e) is considerably improved.

As previously mentioned, the limitation of the approach is not related to the rotor-bearing operational conditions (Sommerfeld number) but related to the operational frequency range of the hydraulic actuators built by servo valves and chambers.

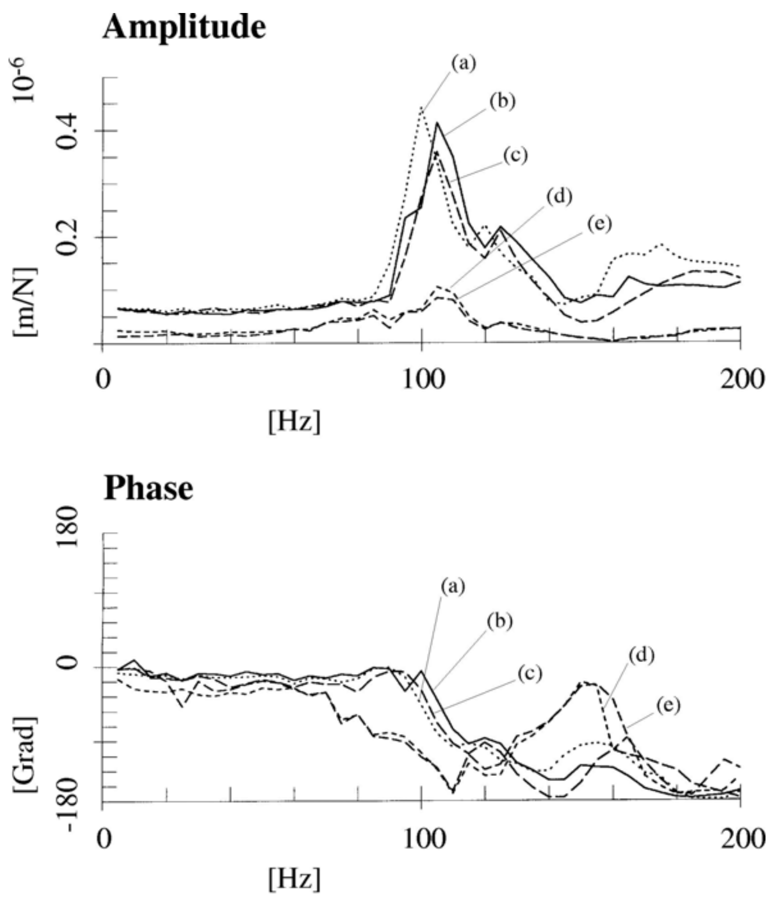


Figure 4.2.1: Experimental transfer function of the rotor-lever-bearing system operating at constant angular velocity of $40Hz$ ($2400\ rpm$) and excited with help of the piezoelectric actuator in different conditions – (a) passive; (b) static pressure of 4 *bar* ($0.4\ MPa$) in the chamber with no feedback control; (c) feeding back the chamber pressures; (d) feeding back chamber and pressures and rotor linear velocity; (e) feeding back chamber pressures, rotor linear velocity and displacement.

4.2.2 Active Lubrication – Active Vibration Control of Rigid Shafts

The first experimental results showing vibration reduction in time domain via active lubrication are presented in Santos and Scalabrin [11]. The test setup used is shown in figure 2.2.2. This test rig was built by a relatively light rotor, which operates at a low rotational speed of 650 *rpm*. Such a combination of parameters inevitably leads to nearly over-damped vibration responses. Despite these conditions, a further increase of damping is achieved when the active lubrication is turned on. The experimental results presented in figure 4.2.2 illustrate the rotor unbalance response in time domain and show a nearly 50% reduction of vibration amplitude, after transient fades away. In this figure, the control system is activated after 0.5 *s*.

The test rig is modified, the rotor mass is increased and the first theoretical and experimental results in the frequency domain are presented in Nicoletti and Santos [20]. Figure 4.2.3 illustrates the theoretical and experimental comparison. In figure 4.2.3(a) and (b) the rotor-bearing system operates at the constant angular frequencies of 15 Hz (900 rpm) and 30 Hz (1800 rpm). In the active mode, the active lubricated bearing operates with a very simple P-controller. Figures 4.2.3(c) and (d) show the predicted theoretical frequency response functions obtained by two different control design approaches. The approach based on the global coefficients (modified Reynolds equation for active lubrication and perturbation technique) leads to better results than the approach decoupling the hydrodynamic and active oil film forces by using the coefficient λ (obtained from a quasi-static test).

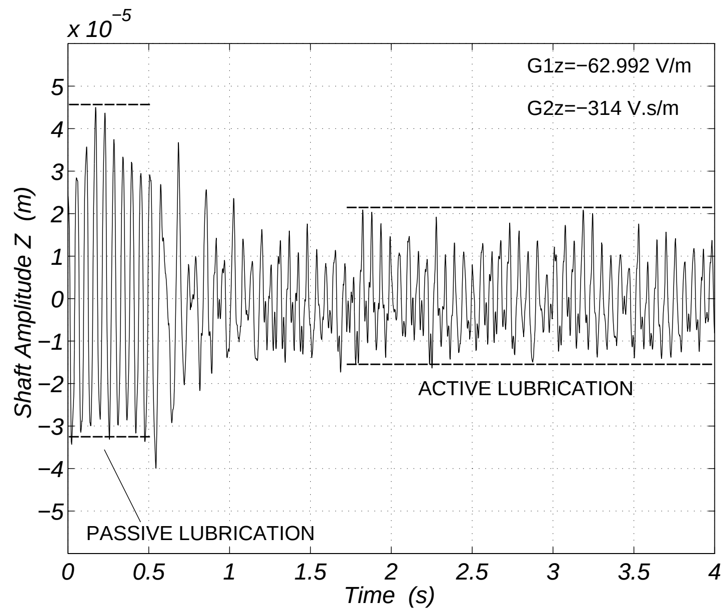


Figure 4.2.2: Experimental results in time domain – reduction of lateral vibration (unbalance response) of the rigid rotor supported by an active lubricated tilting-pad bearing.

Using the test rig illustrated in figure 2.2.5 the feasibility of controlling lateral vibrations were demonstrated in the frequency domain using the "second generation" of active lubricated bearings shown in figure 2.2.3. In the test rig the rotor is supported by two bearings: a ball bearing and a tilting-pad journal bearing. Due to the flexibility of the the ball bearing pedestal the shaft behaves as a rigid body and the rotor-bearing system eigenfrequency is 69 Hz. An electromagnetic shaker and a force transducer is connected to the auxiliary bearing as shown in figure 2.2.5. A sinusoidal excitation signal with frequency varying from 0 until 100 Hz is used as input to the shaker. The test is performed when the rotor operates passively (conventional hydrodynamic lubrication) as well as actively using a centralized PD-controller. The lateral acceleration of the rotor extremity (point 1), where the auxiliary bearing is mounted, is used as the feedback signal to the centralized controller. The acceleration signal is integrated twice in order to obtain displacement and velocity signals and build the structure of the PD-controller. A strong reduction of the resonance peak is seen at the shaft extremity (point 1).

The effect of the angular velocity is investigated in the figures 4.2.5 and 4.2.6 when the rotor operates at the velocities of 600 rpm and 1800 rpm. The same controller scheme is used and the measurements are carried out in the ball bearing pedestal (point 9). The reduction of damping as a function of the angular velocity can be clearly detected by analyzing the frequency range from 60 to 80 Hz in both figures, where the resonance amplitude in both graphics occurs. The resonance amplitude for the passive cases increases from 0.37 to 0.41 $m/s^2/N$ (11%) when the

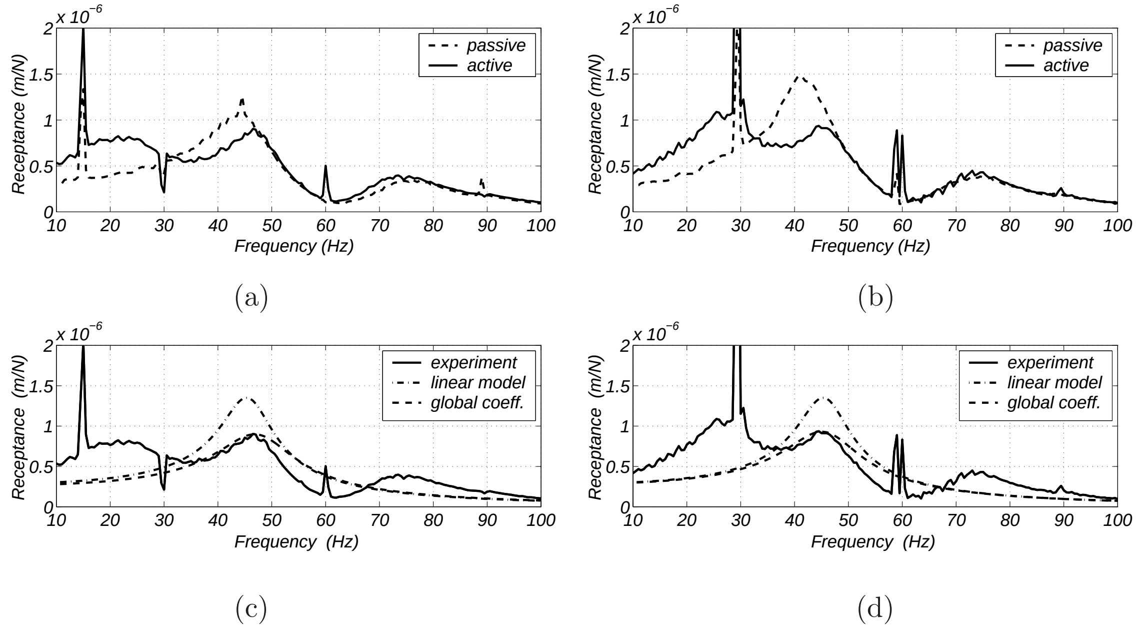


Figure 4.2.3: Experimental results in frequency domain – (a) comparison of passive and active behavior of the rotor-bearing system rotating at 15 Hz (900 rpm); (b) comparison of passive and active behavior of the rotor-bearing system rotating at 30 Hz (1800 rpm); (c) experimental and theoretical frequency response functions of the active system operating at 15 Hz (900 rpm); (d) experimental and theoretical frequency response functions of the active system operating at 30 Hz (1800 rpm).

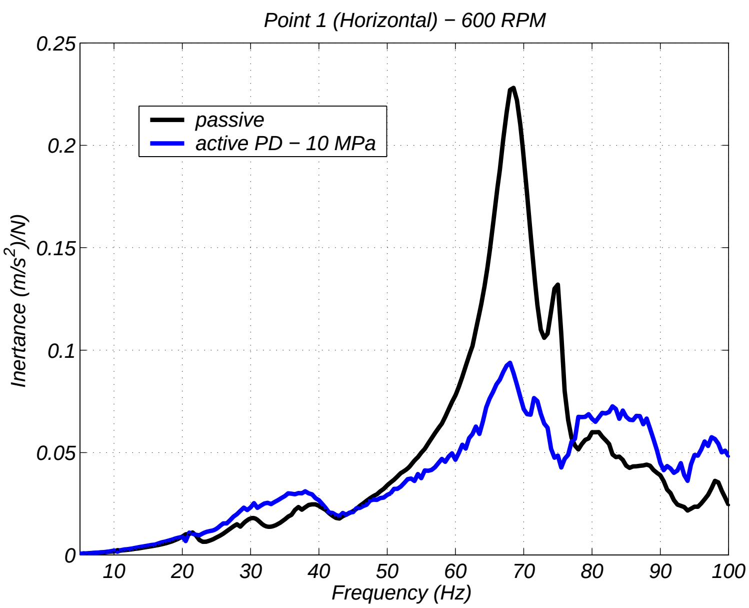


Figure 4.2.4: Experimental results – frequency response function of node 1 when the rotor-bearing system operates at rotational speed of 600 rpm in two different cases: (a) passively or conventionally lubricated and (b) actively lubricated using a PD-feedback control.

angular velocity changes from 600 to 1800 rpm. The variation of the stiffness can be hardly detected by analyzing the frequency where the maximum amplitudes occur. Nevertheless, the resonance peak when the rotor operates at 1800 rpm occurs in a frequency a little bit higher than when it operates at 600 rpm. By using a PD-controller with two different supply pressures, namely 60 bar and 100 bar, the reduction of the resonance peaks are clearly demonstrated. In the case of 600 rpm, the reduction is from 0.37 to 0.15 $m/s^2/N$ with a PD-controller and a supply pressure of 100 bar (10 MPa). It means a reduction of 59% in the resonance amplitude. In the case of 1800 rpm the reduction of the resonance peak from 0.41 to 0.25 $m/s^2/N$ is observed, leading to a reduction of 39% in the resonance amplitude.

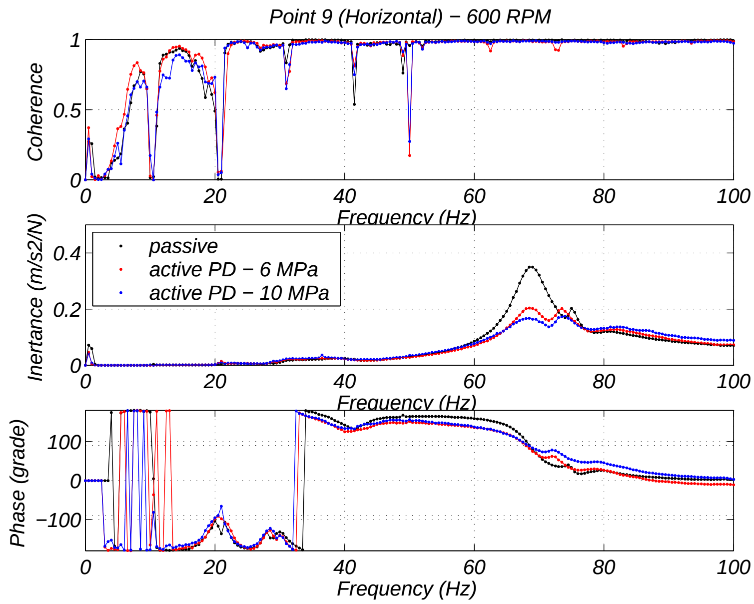


Figure 4.2.5: Experimental results – frequency response function of node 9 (ball bearing pedestal) when the rotor-bearing system operates at rotational speed of 600 rpm in three different cases: (a) passively or conventionally lubricated; (b) actively lubricated using a PD-feedback control and a supply pressure of 60 bar (6 Mpa); (c) actively lubricated using a PD-feedback control and a supply pressure of 100 bar (10 MPa).

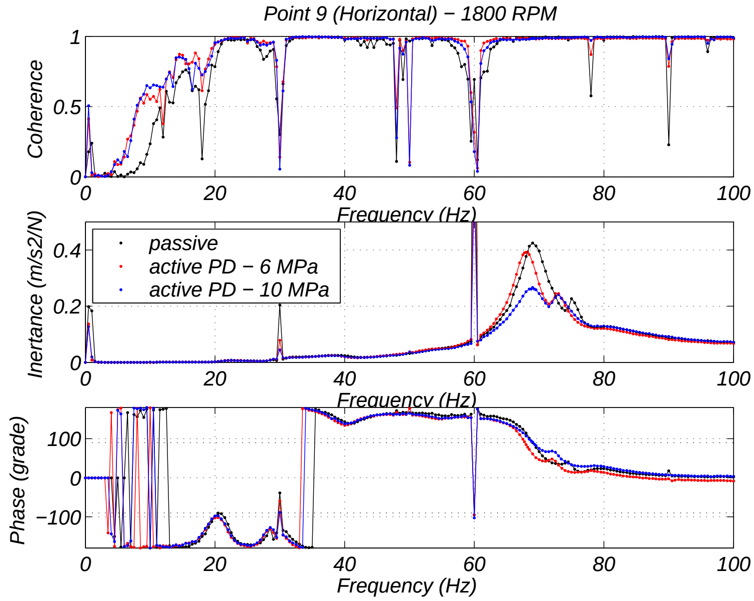
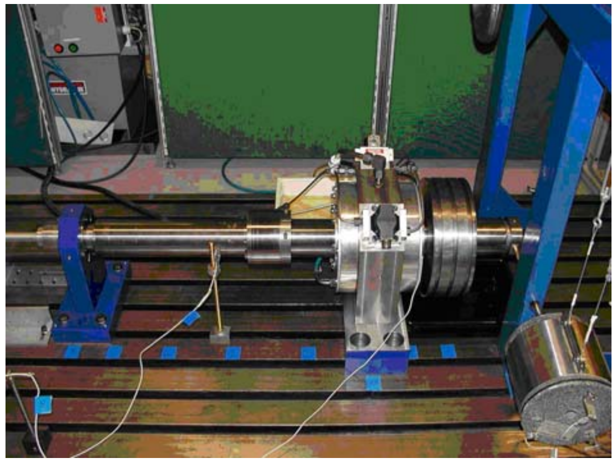


Figure 4.2.6: Experimental results – frequency response function of node 9 (ball bearing pedestal)when the rotor-bearing system operates at rotational speed of 1800 rpm in three different cases: (a) passively or conventionally lubricated; (b) actively lubricated using a PD-feedback control and a supply pressure of 60 bar (6 MPa); (c) actively lubricated using a PD-feedback control and a supply pressure of 100 bar (10 MPa).

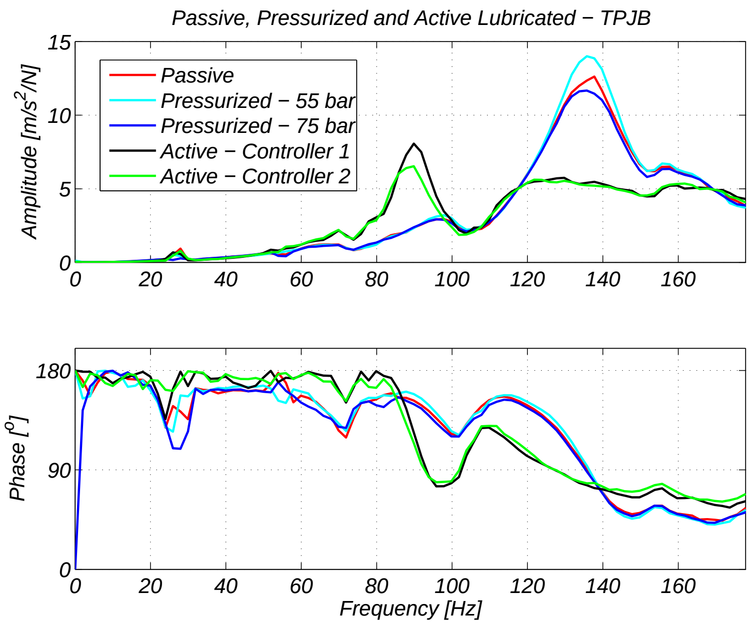
4.2.3 Active Lubrication – Active Vibration Control of Flexible Shafts

The pedestal where the ball bearing housing is mounted was significantly stiffened, and additionally, a heavy inertia (disc) was mounted to the shaft. Moreover, the rotor of a magnetic bearing and the magnetic bearing with hall sensor was mounted between the ball bearing and the tilting-pad bearing, to be used among others as a calibrated shaker without contact. After all these changes, the rotating shaft behaves as a flexible body and different control strategies for flexible rotors can be experimentally investigated. The results were already presented in section 4.1.5, see figures 4.1.5.

The design and implementation of active control of flexible rotating shafts via active bearings has to be carefully carried out. Normally such a design is based on adjusted and updated mathematical models. It avoids a series of implementation problems, among others instability and spill-over, a problem well-known in the literature. Moreover, significant reduction of damping factor of specific mode shapes can occur, while other mode shapes become very well damped. Figure 4.2.7 illustrates the experimental frequency response function for the flexible rotating shaft controlled by the active lubricated tilting-pad journal bearing. Figure 4.2.7(a) highlights the test rig with the electromagnetic shaker, the excitation and the measurement point 1, where the auxiliary bearing is placed. Figure 4.2.7(b) shows the experimental frequency response function of node 1 (auxiliary bearing) when the rotor-bearing system operates at rotational speed of 3000 rpm (50 Hz) in five different cases: (i) passively or conventionally hydrodynamic lubricated; (ii) hybrid lubricated (pressurized) with a hydrostatic pressure of 55 bar (5.5 MPa); (iii) hybrid lubricated with a hydrostatic pressure of 75 bar (7.5 MPa); (iv) actively lubricated using a PD-feedback control and a supply pressure of 75 bar (7.5 MPa); (v) actively lubricated using a PD-feedback control and a supply pressure of 100 bar (10 MPa). The system resonance amplitude around 130 Hz is very significantly attenuated (almost by a factor 2) when the system operates under active lubrication regime. Nevertheless, the other resonance around 90 Hz is amplified.



(a)



(b)

Figure 4.2.7: Experimental results for a flexible rotating shaft controlled by means of an active lubricated bearing – (a) test rig illustrating electromagnetic shaker, excitation and measurement point 1 (auxiliary bearing); (b) experimental frequency response function of node 1 (auxiliary bearing) when the rotor-bearing system operates at rotational speed of 3000 rpm (50 Hz) in five different cases: (i) passively or conventionally lubricated; (ii) hybrid lubricated with a hydrostatic pressure of 55 bar (5.5 MPa); (iii) hybrid lubricated with a hydrostatic pressure of 75 bar (7.5 MPa); (iv) actively lubricated using a PD-feedback control and a supply pressure of 75 bar; (v) actively lubricated using a PD-feedback control and a supply pressure of 100 bar (10 MPa).

New control strategies shall be investigated for attenuating the vibration of flexible rotating shaft based on Nicoletti and Santos [28].

4.3 Blades

4.3.1 Passive Rotor-Bearing-Blade Dynamics – Stiffening, Veering and Parametric Vibrations

In Santos et al. [16] the comparison of the results coming from the three different mathematical models to the experimental ones, shows that the second order linearized and fully non-linear models are those which correctly can predict the stiffening effect and coupled parametric vibrations of the rotor-blade assembly. The linear model can correctly estimate parametric vibrations only in a short range of low rotational speed, where the stiffening effect is of minor importance. Super-harmonic vibrations associated to the movement of the flexible rotating parts (with large amplitude) can only be detected with help of the fully non-linear model.

From the viewpoint of vibration monitoring in rotating flexible structures one can conclude that the positioning of sensors at stationary or rotating reference frames will lead to peaks at different frequencies. With sensors attached to the non-rotating part of the structure (inertial reference frame) one will detect parametric vibrations related to the coupled mode shapes of the flexible rotating parts and flexible non-rotating parts. In rotor-bearing systems coupled to flexible blades, the lighter the flexible blades are, the closer are the modes (BB) and (BR). It means that a long acquisition time is necessary in order to achieve a reasonable precision in the frequency domain and correctly identify both frequencies and modes.

In Christensen and Santos [21] experimental tests of the non-controlled rotor-blade system were carried out and frequency response functions for the system rotating at speeds in the span of 0-600 rpm were measured. The peculiar characteristics of this special kind of system, like the presence of parametric vibration modes and frequency veering, were clearly presented, theoretically as well as experimentally, considering not only one lateral movement in the horizontal direction, but both in the horizontal and vertical directions.

4.3.2 Active Vibration Control of Blades via Actuators Mounted on the Blades

For a tuned rotor-blade system, two different control strategies were implemented and tested in Christensen and Santos [21]. A scheme of six independent decentralized PD-controllers allows the reduction of rotor as well as blade vibrations very efficiently. Until this stage of the experimental research, the implementation of the more complex periodic modal state feedback control scheme proposed in Christensen and Santos [18] has shown its limitations and difficulties. Too many state variables and too many actuators are coupled in the controller design resulting in a controller very susceptible to model imperfections. Therefore, the performance of the centralized modal controller became poor and it could not compete with more simple decentralized PD-controllers [21]. Different control strategies towards more robust controllers [80] shall be investigated in the near future.

4.3.3 Active Vibration Control of Blades via Bearings

A mistuned horizontal rotor-blade system was controlled using only radial shaft actuation in Christensen and Santos [21]. The performance of this system shows that, in practice, blade vibrations can be controlled using only the shaft-based actuation. The results presented reveals the potential of controlling rotor-blade vibrations via shaft-based actuation, if the blades are properly mistuned. Despite the implementation difficulties and the high sensitivity to the modeling imperfections, it is also possible to conclude that the periodic modal control methodology applied to controller design works in practice and may be applicable to other periodic time-variant systems. Additionally, when designing active controllers for this special kind of system the modal transformation based on modal analysis in time-variant systems is very useful: (a) it gives a mathematically explanation of all the frequencies and vibration modes of the system, i.e. basis and parametric modes; (b) it allows the possibility of order reduction so only troublesome modes are controlled; (c) it provides an easy method for identification of optimal actuator and sensor placement via controllability and observability.

A numerical study of a bladed disc modal controllability and observability is presented in Christensen and Santos [19] and shows that for the tuned case with identical blades, it is necessary to apply blade as well as shaft-based sensors and actuators to monitor and suppress all vibration levels. However, if the disc is properly mistuned the results show that all vibration levels can be monitored and controlled by using only shaft-based sensors and actuators. This fact is explained via the presence of parametric vibration modes, which occur due to the vibration coupling. Moreover, the numerical analysis presented in Christensen and Santos [18] shows that the modal controllability and observability of the bladed disc depend significantly on disc angular velocity. At certain angular velocities basis and parametric vibration mode components will almost "coincide" and strongly interact resulting in varying degrees of controllability and observability. The degree of controllability and observability will also vary with angular velocity due to centrifugal stiffening of the blades. The levels of controllability and observability are relatively small in some angular velocity ranges. In Christensen and Santos [22] further investigations of the dynamic characteristics of a mistuned vertical bladed rotor show how, why and when a bladed rotor becomes controllable and observable if properly mistuned. The levels of controllability of different bladed disc with different numbers of blades are critically discussed. As part of such investigation the modal controllability and observability of a tuned as well as a mistuned coupled vertical rotor-blade system are analyzed based on the time-variant controllers designed based on time-variant modal analysis as theoretically proposed in [18].

4.4 Feasibility of Industrial Application

Active magnetic bearings Forster et al. [111] Canders et al. [73] Nakajima [184] Bichler and Eckhardt [55] Girault [122] Dell et al. [85] Brunet [65] Moses et al. [181], hybrid tilting-pad thrust bearings Santos and Fuerst [13] Heinrichson et al. [25] [26], ball bearings controlled by active chamber systems Althaus et al. [43] Hagemeister [130] are already under operation in some types of rotating machines.

Active Tilting-Pad Journal Bearings – In terms of industrial application, it is important to highlight that tilting-pad bearings are the most stable among the different types of journal bearings. Normally, when vibration instabilities arise in a rotating machine supported by hydrodynamic bearings, the journal bearings are exchanged to tilting-pad bearings. If further actions shall be taken towards to improve rotor-bearing stability, control techniques can be applied to a journal bearing that already presents the best stability properties.

The feasibility of reducing lateral vibrations of an industrial gas compressor by using an active lubricated tilting-pad journal bearings is investigated in Santos et al. [17]. The rotating machine whose dynamics is analyzed is a gas compressor composed of five impellers, which weights 391 kg and operates in the range of 6942 rpm (115.7 Hz) to 10,170 rpm (169.7 Hz). By applying the finite shaft elements proposed by Nelson and McVaugh [186] [187], the compressor is modeled with 56 elements (57 nodes) as it is shown in figure 4.4.1. Impellers and other machine elements attached to the shaft are considered as rigid discs, whose dynamics are incorporated into the model by adding inertia to the respective nodes. Hence, in the model, the impellers are at nodes 20, 24, 28, 32, and 36; bushes are at nodes 22, 26, 30, and 34; a thrust disc sleeve is locate at node 3; a balance piston is located at node 38; seal bushes are located at nodes 12 and 46; and the coupling is at node 55. The bearings are located at nodes 8 and 50. The active lubricated tilting-pad journal bearing used in the nodes 8 and 50 are illustrated in figure 2.2.1, namely with 5 orifices per pad.

The feasibility of reducing lateral vibrations of an industrial gas compressor by using an active lubricated tilting-pad bearing is illustrated in figure 4.4.2. The unbalance response at the shaft center (node 28) is shown when the compressor operates passively and actively lubricated. Figure 4.4.2 also illustrates the first bending mode shape of the gas compressor. One of the most important conclusions is related to the feasibility of controlling the first bending vibration mode of a flexible rotor via its bearings. Despite the limitation of rotor displacements in the bearing gap (30% of the assembled clearance), it is possible to reduce the rotor vibration by active forces (active lubrication) acting on the bearings. Such a claim can be verified by analyzing the unbalance response of node 28.

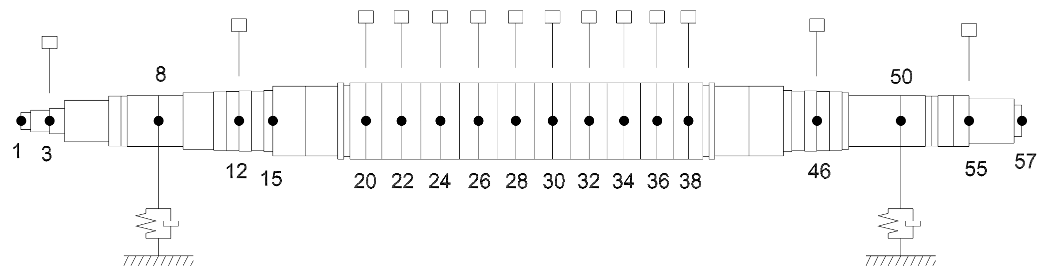


Figure 4.4.1: Finite element model of the gas compressor.

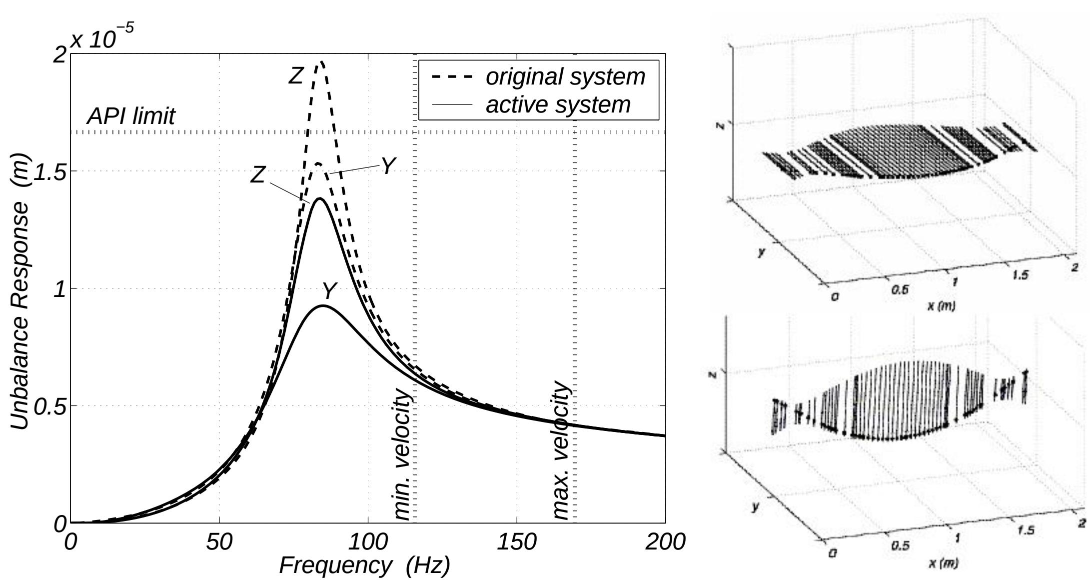


Figure 4.4.2: Theoretical results – gas compressor unbalance response according to API 617.

By using the modified Reynolds equation for the active lubrication the gains of the PD-controller are explicitly written inside of the equation. Combining such an equation with perturbation techniques, damping and stiffness can be predicted as a function of the controller gains. Figure

4.4.3 illustrated the damping factor associated to the first bending mode of the gas compressor as a function of the bearing stiffness and damping. It is possible to see the improvement of the damping factor related to the original design, when different supply pressures are used in combination with the PD-controller.

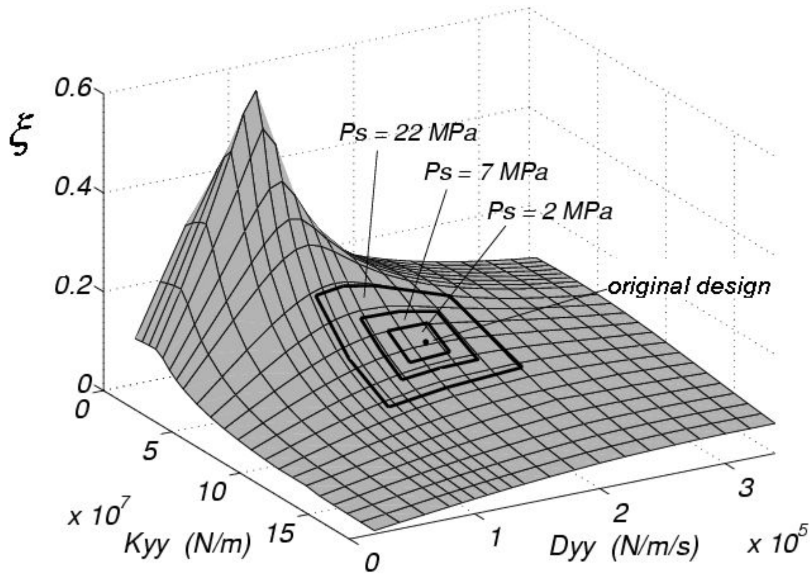


Figure 4.4.3: Theoretical results – feasibility of modifying damping factor associated to the first mode shape of a gas compressor by means of an active lubricated tilting-pad bearing - damping factor as a function of the bearing stiffness and damping coefficients and supply pressure Ps.

Multirecess Journal Bearings with Two Pair of Active Pockets – In terms of industrial application, the importance of hybrid journal bearings as potential application as load support elements in high-speed turbomachinery and machine tool spindles has steadily grown over the past years. Hybrid bearings enable high-load capacity with large direct stiffness, accuracy of positioning, low friction and long life. Such properties make them attractive for applications to reactor coolant pumps and precision grinder spindles. Further improvement of rotor-bearing stability using active lubricated multirecess bearings is discussed in Santos and Watanabe [23], using the same industrial gas compressor shown in figure 4.4.1. According to the API 617 norm, the vibration amplitude limit for the case of the gas compressor under investigation is $L_v = 16.65\mu m$. The unbalance response of node 28 (largest amplitude) in the horizontal direction (X direction) is illustrated in figure 4.4.4 in case of using only hybrid lubrication (without feedback control). The bearing operates with different values of supply pressure . One can notice that, for low values of 1.5 or 2.0 MPa, the unbalance responses of the compressor are in accordance with API limit, even at critical speed, but for higher values of 3.0 or 4.0 MPa the unbalance response of the compressor at critical speeds are higher them the API limit. In the special case of multirecess journal bearings, an increase of the supply pressure will necessarily lead to a simultaneous increase of the stiffness coefficients of the bearing. By increasing the stiffness of the bearings at the nodes 8 and 50, only small relative movements between compressor shaft and bearing housing will be possible in such nodes. This will deteriorate the capability of dissipation of vibration energy by squeezing the oil film in the bearings and will reduce the damping ratio. Such a claim can be clearly seen in Figure 4.4.4, once the damping factor will define the height of the resonance peaks. From the viewpoint of active lubrication applied to multirecess journal bearings with the aim of controlling lateral dynamics of flexible rotors a paradigm is reached:

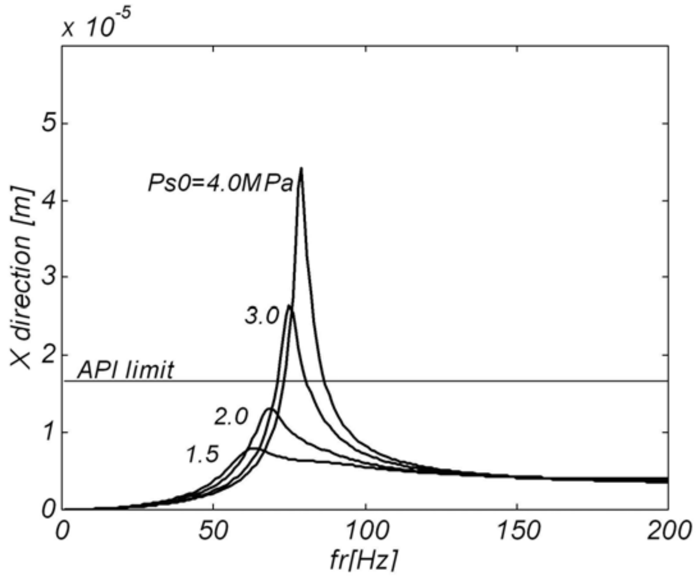


Figure 4.4.4: Theoretical results - feasibility of reducing lateral vibrations of an industrial gas compressor by using a multirecess journal bearing under hybrid lubrication (without feedback control).

a) for improving the performance of the active lubrication in such special kind of bearings high values of supply pressure are needed, once the saturation of control signals retrofitted to the servo valves limits the range in which the bearing dynamics coefficients can be modified. Nevertheless, by increasing the supply pressure the main stiffness coefficients of the bearing will also increase; b) for reducing lateral vibrations of the flexible rotating machine it is very important to allow relative large amplitudes of vibration in the bearings, in order to increase the dissipation of vibration energy by squeezing the oil film. It means that it is important to keep low values of main stiffness coefficients, which coerces into working with low values of supply pressure. With low supply pressure the efficiency of the active lubrication will be significantly reduced or unnecessary.

Active Lubricated Plain Bearings of Reciprocating Machines – The feasibility of applying active lubrication techniques to reciprocating machines with different sizes are investigated in Estupiñan and Santos [27] [31] [32]. Depending on the system size, servo valves can be substituted by piezo-injectors in order to obtain more compact systems. Increasing the oil film thickness and reducing the rotor orbits inside of the bearings can lead to a significant reduction of friction and vibrations.

Chapter 5

Concluding Remarks

The work gives an overview about the theoretical and experimental achievements of mechatronics applied to machine elements with focus on active change of bearing, shaft and blade static and dynamic properties. A chronological overview is given, and the main theoretical and experimental contributions of this work based on 33 articles are highlighted.

In [1] the design and evaluation of two different types of active tilting-pad bearings are presented. The control design of tilting-pads supported by hydraulic chamber systems is the main focus and the idea of controlling the rotor movements via active lubrication is shortly mentioned. In [2] the possibility of adjusting journal bearing properties, exploring the non-linear relationship between damping and stiffness coefficients on the bearing gap, is theoretically as well as experimentally investigated. In [3] strategies for increasing the damping properties of rotating systems supported by tilting-pad bearings is discussed in details from the viewpoint of active lubrication. In [4] an identification procedure based on least-square methods and frequency response functions is developed aiming at identifying stiffness and damping coefficients of active tilting-pad journal bearings controlled by means of hydraulic chamber systems. In [5] self-excited (flutter) vibrations of tilting-pads are theoretically as well as experimentally investigated. The self-excitation mechanism is elucidated and the dependency on the bearing dynamic coefficients shown. The elimination of the self-excited vibrations (pad flutter) is experimentally proven by exploring the dependency of damping properties on the bearing gap and the possibility of changing the bearing gap via controllable hydraulic chambers.

In [6] the modified Reynolds equation (for the hybrid case) is solved with variable steps in order to properly deal with the orifices machined on the pad surfaces. Moreover, the design of tilting-pad bearings with active oil film based on the modified Reynolds equation is illustrated. In [7] the modified Reynolds equation is coupled to the equation of the servo valves, respecting the pressure-flow relationships and continuity equation. Results are presented for the hybrid case, taking into account different bearing pre-load factors and radial injection pressures. The importance of the hydrostatic lubrication (intended to be controllable) in comparison to the hydrodynamic lubrication are numerically investigated. The case in which the control system fails is numerically investigated, showing a reduction of bearing load capacity. In [8] the energy equation under adiabatic condition is solved together with the Modified Reynolds equation and insights into temperature distribution and cooling effect obtained via hybrid/active lubrication are obtained. In [9] the influence of orifice distribution on the thermal and static properties of hybrid lubricated tilting-pad bearings is investigated in detail. In [10] the most significant theoretical contribution to "Active Lubrication Theory" is given, but including the dynamics of the servo system and the feedback control law inside of the modified Reynolds equation and sequentially applying the perturbation technique to predict the behavior of the bearing stiffness and damping coefficients as a function of the Sommerfeld number, pre-load factor, excitation frequency and control gains. The proportional part of the PD-controller contributes to modify the stiffness of the oil film and the derivative part of the PD-controller to modify the damping of the oil film. One of the main advantages of incorporating the PD-control gains inside of Modified Reynolds equation and obtaining the bearing coefficients using perturbation technique is the feasibility of elaborating "controllable journal bearing tables", in the same way as the "journal bearing tables" presented by Lund and Thomsen (1978) and Someya (1989), but with the difference that supply pressure and control gains become new design parameters. In the near future, "controllable journal bearing tables" may be used to design rotor-bearing systems operating under active lubrication regime, and the integration of mechanics, electronics and informatics (mechatronics) may become even more clear in the field of Tribology.

In [11] the focus is on the control loop design of decentralized controllers using the modified Reynolds equation for active lubrication. An experimental example illustrates the reduction of lateral vibration of rigid rotor supported by active lubricated tilting-pad bearing in the time domain. In [12] the feasibility of influencing the dynamic coefficients of a multirecess journal bearing by means of active hybrid lubrication is theoretically investigated. Such bearings are composed of 4 pockets and hydrostatic part of the lubrication mechanism plays an important role. The mathematical models of hybrid bearings are expanded by coupling the dynamics of the servo systems with the continuity equation. In [14] linear and non-linear control techniques are applied to active lubricated journal bearings with the aim of controlling the lateral dynamics of a rigid rotor and eliminating oil-whirl phenomena. The integral part of the controller shall be carefully designed, avoiding centering the rotor inside of the bearing. Such a control action will significantly reduce the damping coefficient of the oil film. The integral part of a PID-controller shall be responsible for increasing the rotor eccentricity inside of the bearing, leading to a more damped and stable system. In [15] pressure saturation zones due to nonlinear behavior of servo valves are included into the modeling and analysis of active lubricated multirecess journal bearings. Sequentially, the compensation of cross-coupling stiffness and increase of direct damping in multi-recess journal bearings are investigated. The control strategies for efficiently eliminating cross-coupling effect are elucidated.

In [20] theoretical and experimental frequency response analysis of rigid rotor supported by an active lubricated tilting-pad bearing is shown. Moreover, different control strategies are experimentally tested, showing the procedure presented in [10] [11] being more efficient. In [17] the feasibility of applying active lubrication to a tilting-pad bearing aiming at reducing lateral vibration in an industrial gas compressor is carefully investigated. Active lubricated bearings can become a powerful mechatronic tool for a new generation of machines. Furthermore, in [23] the feasibility of applying active lubrication to multirecess journal bearings of an industrial gas compressors is also studied. The bearing operating in its hybrid mode is very powerful and able to control rigid rotor lateral vibrations.

From the viewpoint of active lubrication applied to multirecess journal bearings with the aim of controlling lateral dynamics of flexible rotors a paradigm is reached: a) for improving the performance of the active lubrication in such special kind of bearings high values of supply pressure are needed, once the saturation of control signals retrofitted to the servo valves limits the range in which the bearing dynamics coefficients can be modified. Nevertheless, by increasing the supply pressure the main stiffness coefficients of the bearing will also increase; b) for reducing lateral vibrations of the flexible rotating machine it is very important to allow relative large amplitudes of vibration at the bearing location, in order to increase the dissipation of vibration energy by squeezing the oil film. It means that it is important to keep low values of main stiffness coefficients, what coerces into working with low values of supply pressure. With low supply pressure the efficiency of the active lubrication will be significantly reduced. The design of controllers for flexible rotors supported by active lubricated bearings is further investigated in [28], by decoupling the hydrodynamic and active hydraulic forces and applying modern control theory. Moreover, a better description of the active forces and its dependency on the servo valves parameters and excitation frequency are taken into account.

The feasibility of using active lubrication in reciprocating machines is also theoretically investigated in [27] [31] [32]. In [27] a mathematical model of hermetic reciprocating compressors are developed, combining multibody dynamics, finite element method and fluid film lubrication. In [31] different constraint equations are used to accommodate multibody dynamics and fluid film bearings and correctly describe the dynamic behavior of such a type of compressors. In [32] the equations describing the dynamics of hermetic compressor components (obtained based on a multibody system approach) is coupled to the controllable thin fluid film and different control strategies are simulated and discussed. In [30] an overview about the the application of active lubrication techniques to different types of journal bearings is given.

In [13] theoretical and experimental aspects of heavily loaded tilting-pad journal bearings are considered. By using the developed thermo-hydrodynamic models to describe the properties of the bearings of a hydro-generator, the necessity of including structural and thermal deformations is evaluated. Startup hydrostatic pressures, minimum oil film and maximum temperature in heavily loaded bearings are theoretically and experimentally presented. The results are validated against experimental data obtained from a hydro-generator from Alstom Power Ltd. In [25] the influence of injection pockets on the performance of heavily loaded tilting-pad thrust bearings is investigated using a very complex elasto-thermo-hydrodynamic model, built with basis on the model developed in [13]. The results are validated based on the same data obtained from a hydro-generator from Alstom Power Ltd. In [26] the influence of injection pockets on the performance of tilting-pad thrust bearings is also investigated and the comparison between elasto-thermo-hydrodynamic model and experimental results are done using a test rig designed by Alstom Power Ltd. In [29] the "calibrated" elasto-thermo-hydrodynamic model is used to design enclosed recesses and reduce friction in heavily loaded tilting-pad.

In [16] a contribution to the experimental validation of linear and non-linear dynamic models for representing rotor-blades parametric coupled vibrations is given. The rotor-blade dynamics is described by using three models with different levels of complexity followed by experimental validation of such models. A deeper physical understanding of the dynamic coupling and the behavior of the parametric vibrations are achieved. Such an understanding is of fundamental importance while developing active control strategies. In [18] the design of time-variant modal controllers is focused. Time-variant modal analysis and modern control theory are integrated in an elegant way allowing the development of new control strategies. The modal controllability and observability of bladed discs are strongly dependent on the angular velocity, a detailed analysis of such a dependency is presented in [19]. To control rotor and blade vibration using only shaft actuation is a very challenging problem. In [21] such a problem is investigated theoretically as well as experimentally using different control strategies. The electromagnetic actuators are used to control a horizontal rotor-blade system (blades periodically excited by the gravity). In [22] the same problem is theoretically as well as experimentally investigated and new strategies are developed to control vertical rotor-blade systems. In the case of vertical bladed-rotors the controller does not deal with periodic excitations coming from the gravity force. Nevertheless, in both investigations the efficiency of the shaft-based actuation will be dependent on the level of mistuning among the blades.

In [24] an experimental contribution to high precision characterization of magnetic forces in active magnetic bearings is given. The goal is to develop calibrated shaker without contact to accurately identify damping and stiffness of active lubricated bearings in the frequency domain. Using the calibrated magnetic bearing excellent results related to the identification of the damping factors of active lubricated rotor-bearing systems are achieved. Although such magnetic forces are obtained with high level of accuracy in static tests, uncertainties are attached to the results when a dynamic calibration procedure is carried out. Until now the accurate experimental identification of the dynamic coefficients of active lubricated bearings and their dependency on the frequency and control gains is still a challenge to be overcome. Modern emerging trends in controllable fluid film bearings are elucidated in [33].

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Dansk Resumé (Summary in Danish)

Denne afhandling giver et overblik over teoretiske og eksperimentelle fremskridt indenfor mekatronik anvendt på maskinelementer, med fokus på regulering af de dynamiske og statiske egenskaber af lejer, aksel og blade. Der gives en kronologisk gennemgang, og afhandlingens vigtigste teoretiske og eksperimentelle bidrag, baseret på 33 artikler, fremhæves.

I [1] præsenteres design og analyse af to forskellige aktive radiale vippeskolejer. Design af reguleringssystemer til vippesko monteret på hydrauliske kamre er hovedfokus, og regulering af selve rotorbevægelsen nævnes kort. I [2] undersøges muligheden for at justere lejeegenskaberne, og det ikke lineære forhold imellem lejekoefficienterne og lejegapet undersøges teoretisk såvel som eksperimentelt. I [3] diskuteres strategier for at øge dæmpningen af roterende systemer, understøttet af vippeskolejer, med udgangspunkt i aktiv smøring. I [4] udvikles en indentifikationsprocedure baseret på minimering af den kvadratiske fejl (least-square), samt responsfunktioner, med henblik på at identificere lejekoefficienter for vippeskolejer understøttet af hydrauliske kamre. I [5] undersøges selveksitation (flutter) af vippesko teoretisk og eksperimentelt. Selveksitationsprocessen belyses, og afhængigheden af lejekoefficienterne vises. Forebyggelse af selveksitation udføres eksperimentelt ved hjælp af udnyttelse af sammenhængen imellem lejedæmpning og filmtykkelse, og ved at styre filmtykkelsen ved hjælp af de hydrauliske kamre.

I [6] løses den modificerede Reynolds ligning (i det hybride tilfælde) med et variabelt net, for at håndtere området omkring indsprøjtningsdyserne hensigtsmæssigt. Ydermere illustreres designet af vippeskolejer med aktiv oliefilm. I [7] kobles Reynolds' ligning til en ligning der beskriver servoventilers dynamik. Koblingen foretages med udgangspunkt i kontinuitetsligningen, samt en konstitutiv tryk-flux sammenhæng. Resultater præsenteres for det hybride tilfælde, med hensyntagen til forskellige forspændingsfaktorer og radiale indsprøjtningstryk. Vigtigheden af regulerbar hydrostatisk smøring i forhold til hydrodynamisk smøring undersøges numerisk. Tilfældet hvor reguleringssystemet bryder sammen undersøges ligeledes numerisk, her ses en reduktion af lejets lasteevne. I [8] løses energiligningen under adiabatiske randbetingelser sammen med den modificerede Reynolds ligning, hvilket skaber et indblik i temperaturfordelingen samt den via indsprøjtning opnåede køling. I [9] undersøges indflydelsen af dysefordelingen på de termiske og statiske egenskaber af hybridlejer grundigt. I [10] gives det mest vægtige teoretiske bidrag til teoretisk aktiv smøring, her reguleres Reynolds ligning af servoventiler. Lejekoefficienter findes ved at inkludere servoventilernes dynamik direkte i Reynolds' ligning, hvorefter en perturbationsmetode anvendes. Lejekoefficienterne afhænger af Sommerfeld tallet, forspændingen, eksitationsfrekvensen og reguleringsforstærkningerne. Den proportionelle del af PD regulatoren påvirker lejestivheden, og den afledte del påvirker lejedæmpningen. En af de største gevinster ved denne fremgangsmåde er at tabeller over regulerede vippeskolejer kan opstilles, efter samme facon som de tabeller over passive lejer der præsenteres af Lund og Thomsen (1978) og Someya (1989), med den tilføjelse at tilførselstryk og reguleringsforstærkninger nu er designparametre. I den nærmeste fremtid kan tabeller over regulerbare lejer benyttes til at designe aktivt smurte rotor-leje systemer, og integreringen af mekanik, elektronik og informatik (mekatronik) kan vinde

endnu mere indpas i tribologien.

I [11] ligger fokus på designet af kontrolløkker til decentraliserede regulatorer ved hjælp af den til aktiv smøring modificerede Reynolds ligning. Et eksperimentelt eksempel i tidsdomænet illustrerer reduktionen af laterale vibrationer af en stiv rotor understøttet af aktivt smurte vippeskolejer. I [12] undersøges muligheden for at påvirke de dynamiske koefficienter af et fire aksialt not hydrodynamisk leje ved hjælp af regulering teoretisk. Sådanne lejer består af fire lommer, og den hydrostatiske del af smøringen spiller en vigtig rolle. De matematiske modeller for hybridlejerne udvides til at omfatte servosystemet ved hjælp af kontinuitetsligninen. I [14] benyttes lineære og ikke lineære reguleringsmetoder til aktivt smurte lejer med det mål at regulere den laterale dynamik af en stiv rotor, og for at undgå "oil-whirl" ustabilitet. Integralleddet i regulatoren bør vælges med omhu, for at undgå centrering af rotoren i lejet. Integralleddet bør øge rotoreccentriciteten, hvilket vil føre systemet over i en mere stabil og dæmpet tilstand. I [15] inkluderes tryksatureringszoner, foranlediget af den ikke lineære opførsel af servoventilerne, i modelleringen og analysen af aktivt smurte fire aksialt not lejer. Efterfølgende undersøges kompensering af krydskoblingsled samt forøgelse af direkte dæmpning i fire aksialt not hydrostatiske lejer. Reguleringsstrategier til effektiv eliminering af krydskoblingseffekten fremhæves.

I [20] gives teoretiske og eksperimentelle frekvens respons funktioner for aktivt smurte vippeskolejer. Ydermere undersøges forskellige reguleringsstrategier eksperimentelt, hvilket viser at procedurerne i [10] og [11] er mest effektive. I [17] undersøges muligheden for at anvende aktiv smøring af vippeskolejer med henblik på reduktion af laterale vibrationer i en industriel gaskompressor. Aktivt smurte lejer kan blive et kraftfuldt mekatronisk værktøj til en ny generation af maskiner. Ydermere studeres i [23] muligheden for anvendelse af aktiv smøring i fire aksialt not hydrostatiske lejer til industrielle gaskompressorer. Lejet er i sin hybridtilstand meget kraftfuldt og i stand til at regulere laterale bevægelser af stive rotorer.

Indenfor aktiv smøring af fire aksialt not hydrostatiske lejer, med henblik på regulering af lateral dynamik af fleksible rotorer nåes et paradigme: a) ved forbedring af præstationen af den aktive smøring i sådanne lejer kræves store tilførselstryk så snart satureringssignalet for servoventilerne nåes. Ved forøgelse af tilførselstrykket, vil lejestivheden øges; b) ved reduktion af laterale vibrationer af flexibelt roterende maskineri er det vigtigt at tillade en relativt stor amplitude ved lejet, for at dissipere vibrationsenergi ved sammentrykning af oliefilmen. Det vil sige at det er vigtigt at fastholde lave værdier af stivhedskoefficienter, hvilket taler for at arbejde med lave værdier af tilførselstrykket. Ved lavt tilførselstryk reduceres effektiviteten af aktiv smøring betragteligt. Design af regulatorer til fleksible rotorer understøttet af aktivt smurte lejer undersøges yderligere i [28] ved dekobling af hydrodynamiske og aktive kræfter, og ved anvendelse af moderne reguleringsteori. Ydermere benyttes en bedre beskrivelse af de aktive kræfter og deres afhænghed af servoventilernes parametre samt eksitationsfrekvensen.

Muligheden for anvendelse af aktiv smøring i reciprokerende maskineri undersøges teoretisk i [27] [31] [32]. I [27] udvikles en matematisk model af hermetiske reciprokerende komporessorer, som kombinerer multilegemedynamik, finit element metoden, samt fluidfilm smøring. I [31] benyttes forskellige bindingsligninger til at beskrive multilegemedynamik i samspil med fluidfilm lejer og beskrive sådanne kompressorer korrekt. I [32] kobles hermetiske kompressorers bevægelsesligninger (fundet ved hjælp af en multilegeme tilgang) med en regulerbar oliefilm og forskellige reguleringsstrategier simuleres og diskuteres. I [30] gives et overblik af anvendelsen af aktiv smøring i forskellige lejetyper.

I [13] undersøges teoretiske og eksperimentelle aspekter af kraftigt belastede aksiale vippeskolejer. Ved hjælp af en termohydrodynamisk model til beskrivelse af en hydrogenerators lejer, undersøges nødvendigheden af at tage højde for statiske og termiske deformationer. Tilstanden under start af svært belastede lejer, det være sig hydrostatisk tryk, minimum filmtykkelse og maksimum temperatur, præsenteres. Resultaterne valideres imod eksperimentel data fra en hydrogenerator fra Alstom Power Ldt. I [26] undersøges indflydelsen af indsprøjntningslommer på præstationen af et kraftigt belastet aksielt vippeskoleje ved hjælp af en omfattende termoelastohydrodynamisk model udviklet på grundlag af modellen i [13]. Resultaterne valideres med data fra en hydrogenerator fra Alstom Power Ltd. I [26] undersøges indflydelsen af indsprøjtningslommer på aksielle vippeskolejer ligeledes. Her sammenlignes resultaterne fra den termoelastohydrodynamiske model med eksperimentelle resultater fra en testopstilling fra Alstom Power Ltd. I [29] benyttes den "kalibrerede" termoelastohydrodynamiske model til design af aflukkede forsænkninger og reduktion af friktion i kraftigt belastede vippesko.

I [16] gives et bidrag til den eksperimentelle validering af lineære og ikke lineære dynamiske modeller af rotor-blad koblede parametriske vibrationer. Rotor-blad dynamikken beskrives af tre modeller af forskellig kompleksitetsniveau, og efterfølges af en eksperimentel validering. En dybere fysisk forståelse af den dynamiske kobling og opførslen af parametriske vibrationer opnås. En sådan forståelse er af fundamental betydning under udvikling af reguleringsstrategier. I [18] ligger fokus på design af tidsvarierende modalregulatorer. Tidsvarierende modalanalyse og moderne reguleringsteori undersøges på elegant facon, og tillader udvikling af nye reguleringsstrategier. Den modale kontrollerbarhed og regulerbarhed er stærkt afhængig af vinkelhastigheden og en analyse af denne afhængighed præsenteres i [19]. At regulere rotor og bladvibrationer udelukkende ved hjælp af aktuering af akslen er et meget udfordrende problem. I [21] undersøges et sådant problem teoretisk såvel som eksperimentelt ved hjælp af forskellige reguleringsstrategier. Elektromagnetiske aktuatorer benyttes til regulering af et horisontalt rotor-blad system (bladene eksiteres periodisk af tyngdekraften). I [22] undersøges det samme problem teoretisk og eksperimentelt og nye strategier udvikles til regulering af rotor-blad systemer. For vertikale rotor-blad systemer skal regulatoren ikke håndtere periodisk eksitation fra tyngdekraften. Ikke desto mindre ses det i begge undersøgelser at effektiviteten af akselbaseret eksitation vil afhænge af niveauet af "mistuning" imellem bladene.

I [24] gives et eksperimentelt bidrag til præcisionskarakteristik af magnetiske kræfter i aktive magnetiske lejer. Målet er at udvikle en kalibreret ryster uden kontakt, med henblik på præcis identifikation af stivhed og dæmpning i aktive lejer i frekvensdomænet. Ved hjælp at den kalibrerede magnetiske ryster opnås gode resultater for identifikationen af dæmpningsfaktorer af aktivt smurte rotor-leje systemer. Selvom sådanne magnetiske kræfter kan måles med høj præcision i statiske forsøg, er der usikkerheder knyttet til resultaterne når en dynamisk kalibreringsprocedure udføres. Hidtil er det stadig en udfordring at identificere de dynamiske koefficienter af aktivt smurte lejer eksperimentelt og klarlægge deres afhængighed af frekvensen samt reguleringsforstærkningerne. Moderne fremvindende tendenser i kontrollerbare glidlejer fremhæves i [33].

Thesis Papers [1-33]