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Active tilting-pad journal bearings supporting flexible rotors: Part II–The model-based feedback-controlled lubrication

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ABSTRACT

This is part II of a twofold paper series dealing with the design and implementation of model-based controllers meant for assisting the hybrid and developing the feedback-controlled lubrication regimes in active tilting pad journal bearings (active TPJBs). In both papers theoretical and experimental analyses are presented with focus on the reduction of rotor lateral vibration. This part is devoted to synthesising model-based LQG optimal controllers (LQR regulator + Kalman Filter) for the feedback-controlled lubrication and is based upon the mathematical model of the rotor-bearing system derived in part I. Results show further suppression of resonant vibrations when using the feedback-controlled or active lubrication, overweighting the reduction already achieved with hybrid lubrication, thus improving the whole machine dynamic performance.

1. Introduction

The control of vibration in rotating machinery has been achieved both passively and actively. Passive elements, such as squeeze-film dampers [1] and seal dampers [2] introduce dissipative forces to the system counteracting destabilizing forces and consequently reducing vibrations. However, without simultaneous sensing and actuating capabilities their potential to adapt and perform adequately in a wide frequency range becomes limited. The active elements overcome this limitation by incorporating actuators and sensors, which in a closed-loop configuration provide adaptability to wider frequency ranges and higher efficiency toward vibration reduction. Most of the active elements for rotating machines are built upon bearings. The control of vibrations is attained by modifying the bearing properties accordingly to the excitation loads and operational conditions. A variety of bearing types (ball bearings, magnetic bearings, compressible and incompressible fluid-film bearings) combined with several actuator types (magnetic, piezoelectric, pneumatic, hydraulic) leads to a variety of mechatronic devices or simply active bearings [3–9]. Herein, an incompressible fluid-film bearing with hydraulic actuator, namely tilting-pad journal bearings with active lubrication [10–12], is theoretically as well as experimentally investigated. The dynamic characteristics of rotors supported by TPJBs can be controlled either by modifying the journal-pad clearance through pad pushers – piezoelectric [13] or hydraulic [11] – or by direct modification of the fluid-film pressure distribution [14] via the active lubrication. In [15] the authors simulated an active TPJB with piezoelectric pushers controlled by an LQR regulator, while in [10] the author investigated the lateral dynamics of a rigid rotor controlled by pads on hydraulic pushers and introduced the active lubrication principle applied to TPJBs.

Active TPJBs aiming at improving damping properties and the stability margin of rotating machines have been theoretically investigated in [16,17]. Bearing damping properties in active TPJBs are improved by featuring the feedback-controlled lubrication. Under this lubrication regime, the system response, namely the rotor lateral vibration, is utilized to generate suitable signals to command the servovalves that control the high pressurized oil flow. Synthesising PID controllers seems to be an adequate control design approach once the rotor-bearing system is manufactured. Some model-free approaches [18,19] or even “on-site” gain tuning can be used to obtain good controllers. The works [20–23] are focused on PID controllers for active TPJBs. Nonetheless, if the synthesis of the controller is to be considered as a part of the machine design process, i.e. before the whole rotating machine is manufactured, then a model-based approach becomes an attractive control design tool, allowing for an optimization of the whole electro-mechanical system dynamics. The accuracy in describing the dynamic behavior of passive TPJBs [24–29] and active TPJBs connected to hydraulic servosystems [14,30–33] makes feasible today's model-based control design approaches. Regarding the active control of flexible rotors supported by fluid-film bearings, a theoretical study on full and reduced modal state controllers was presented in [34]. In [6] the vibration suppression was theoretically and experi-

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mentally studied though using magnetic actuators. Different state-feedback controllers were studied and the system model reduction was based upon retaining dominant modes; no spillover problems were observed. [35] also emphasizes the modal reduction of large rotordynamics systems whilst an LQR controller was designed. Linked with active TPJBs, in [36] a compressor supported by this kind of bearing was theoretically studied. Output feedback control and pole placement methods were used. In all these contributions the need for reducing the size of the rotordynamics model by modal approaches is addressed, which might lead to spillover problems.

The main contribution of this work is to present the design and implementation of a model-based controller for a flexible rotor supported by an actively-lubricated TPJB. The objective of the controllers is to reduce the amplitude of the frequency response at resonance in closed-loop, while the TPJB is actively lubricated. From the theoretical standpoint, the rotor-bearing system modelling based on part I is hereby used. The model is modal-reduced and its states are further complex separated to design the LQG regulator. From an experimental standpoint, comparisons of the system response in closed-loop with LQG against the PID controller are presented. Additional tests on slightly modified system are carried out to check the actuator bandwidth. Finally, a discussion on additional pinpointed dynamics is presented.

2. Test rig facilities recap

The flexible rotor-bearing test rig was introduced in part I of this series of papers, more details can be found therein. It resembles a large overhung centrifugal compressor for which the rotor is supported by an actively-lubricated TPJB as the one shown in Fig. 1. The feedback-controlled regime, as defined in part I, is obtained by dynamically controlling the injection of pressurized oil directly into the bearing gap via servovalves. Three rotor configurations can be obtained by hanging different numbers of discs at the free-end, i.e. none, one or two discs. Consequently, three different levels of static bearing loading can be obtained, i.e. 400 N, 880 N or 1440 N, respectively. This also strengthens the gyroscopic effect due to the addition of the disc inertia and reduces the natural frequencies of the system. For instance, the first bending mode is reduced from 210 Hz to 150 Hz and 95 Hz by augmenting the disc number, respectively. The frequency bandwidth of the servovalves and consequently of the active forces strongly depends on the supply pressure P_s. For the current application high response servovalves are used, with a cutoff frequency of 350 Hz for a nominal pressure of 210 bar and 260 Hz at 100 bar, the maximum design pressure for the current system.

3. Linear model of the rotor-bearing system

The modelling of the flexible rotor-bearing system under different lubrication regimes was also presented in part I. The theoretical model is based on a finite element approach, in which each shaft element is represented by two nodes with 4 dofs each and where the active TPJB is included by using the full matrices of the dynamic linear coefficients obtained by an Elasto-Thermo-Hydrodynamic (ETHD) model of the bearing. In the case of the active lubrication regime, the equilibrium position ${\Pi_0}^{\alpha}$ can be regarded as the same as in the hybrid lubrication regime, which is defined by the force and thermal equilibria as well as by the supply pressure of the injection system. Therefore, the linear equation of motion at a defined rotational speed Ω and a given supply pressure P_s reads:

$$\mathbf{M}\ddot{\mathbf{x}} + (\mathbf{D}|_{\Pi_0^a} - \Omega\mathbf{G})\dot{\mathbf{x}} + (\mathbf{K} + \mathbf{K}|_{\Pi_0^a})\mathbf{x} = \mathbf{W}\mathbf{u} + \mathbf{f}_{ext} \tag{1}$$

where on the left hand side $\mathbf{M}$, $\mathbf{K}$ and $\mathbf{G}$ stand for the rotor mass, stiffness and gyroscopic matrices and $\mathbf{K}|_{\Pi_0^a}$ and $\mathbf{D}|_{\Pi_0^a}$ stand for the full dynamic coefficient matrices of the actively-lubricated bearing at the equilibrium condition ${\Pi_0}^{\alpha}$, which include the servovalves dynamics. The generalized coordinated vector is composed of the rotor and active tilting pad dofs: $\mathbf{x} = \{v_1\ w_1\ \gamma_{v1}\ \gamma_{w1} \ldots v_{ns}\ w_{ns}\ \gamma_{v_{ns}}\ \gamma_{w_{ns}},\ \theta_1 \ldots \theta_{np},\ \ldots \beta_1 \ldots \beta_{np},\ \eta_1 \ldots \eta_{np},\ q_{v1}\ q_{v2}\}^{\mathrm{T}}$. On the right hand side, $\mathbf{f}_{ext}$ stands for the external forces applied to the rotor and $\mathbf{u}$ for the 2×1 control signal vector containing the driven voltage of each servovalve, i.e. $\mathbf{u} = \{u_1\ u_2\}^{\mathrm{T}}$. $\mathbf{W}$ is a sparse input control matrix whose non zero elements are defined by the servovalve parameters dependent on P_s as:

$$\mathbf{W}(i_{q_{v1}}, 1) = \omega_{v1}^2 R_{v1} \quad \mathbf{W}(i_{q_{v2}}, 2) = \omega_{v2}^2 R_{v2} \tag{2}$$

Equation (1) can be rewritten as:

$$\mathbf{M}\ddot{\mathbf{x}} + \overline{\mathbf{D}}\dot{\mathbf{x}} + \overline{\mathbf{K}}\mathbf{x} = \mathbf{W}\mathbf{u} + \mathbf{f}_{ext} \tag{3}$$

4. Modal-reduced state-space model

In order to design a model-based controller, the system governing equation (3) must be rewritten in a state-space formulation. To reduce computational burden when implemented, a pseudo-modal reduction scheme is chosen which considers only the slowest eigenvalues. As a consequence, a separation of complex states is needed for implementing the controller in real time processors, which does not work with complex numbers.

By choosing the state displacement and velocity vector $\mathbf{X} = \{\mathbf{x}\ \dot{\mathbf{x}}\}^{\mathrm{T}}$, the state-space representation of the LTI system of Equation (3) is written as:

$$\dot{\mathbf{X}} = \mathbf{A}\mathbf{X} + \mathbf{B}\mathbf{u} + \mathbf{B}_{\mathbf{v}}\mathbf{v}_1 \tag{4a}$$

$$\mathbf{Y} = \mathbf{C}\mathbf{X} + \mathbf{v}_2 \tag{4b}$$

where for the state equation (4a) the state matrix $\mathbf{A}$, the input matrix $\mathbf{B}$ and the disturbance input matrix $\mathbf{B}_{\mathbf{v}}$ are defined as:

$$\mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\overline{\mathbf{K}} & -\mathbf{M}^{-1}\overline{\mathbf{D}} \end{bmatrix} \quad \mathbf{B} = \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{W} \end{bmatrix} \quad \mathbf{B}_{\mathbf{v}} = \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1} \end{bmatrix} \tag{5}$$

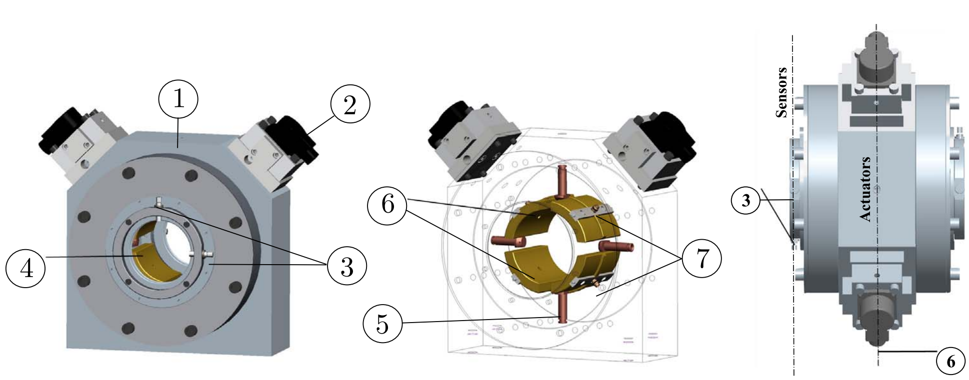

Fig. 1. Active TPJB. ① low pressure oil inlet. ② electrohydraulic servovalve. ③ proximity probes. ④ pad. ⑤ low pressure oil nozzle. ⑥ centred pad orifice. ⑦ high pressure oil nozzle.