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Experimental and Theoretical Study of an Actively Lubricated LEG Tilting Pad Bearing

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Abstract
This work presents theoretical and experimental results regarding the feasibility of introducing an active lubrication concept in tilting pad journal bearings (TPJBs) featuring the leading edge groove (LEG) lubrication system. The modification of the oil flow into each pad supply groove by means of servovalves renders the bearing active, due to the resulting alteration of the oil film pressure field. Experimental results obtained in a suitable test rig enables verification of the concept feasibility at the component level, considering a single tilting pad, load on pad configuration, supporting a rigid rotor. Secondly, theoretical results portray the application of this technology in a 5-pad, load between pad LEG TPJB, aimed at performing active control of vibrations in a flexible rotor setup. The proposed technology exhibits benefits in terms of the ability to excite system dynamics to perform in-situ identification experiments, as well as to reduce the rotor vibration amplitude.

1 INTRODUCTION
Among the different fluid film bearing designs, the oil lubricated tilting pad journal bearing is widely used in the industry. Key features from this machine element are its ability to endure high rotational speed and high load applications, its damping characteristics due to oil film squeeze effect, and its enhanced stability properties [1] when compared to other oil film bearing designs.

Since its introduction during the first half of the 20th century, its basic design has been continuously improved, based on the knowledge gained from theoretical and experimental investigations, as well as the experience gained from their industrial usage. The latest publications related to this bearing design deal with a number of issues: the effect of reduced oil supply rates over its steady state and dynamic behavior [2, 3], the influence of the pivot design over the bearing properties [4], the usage of superconductor materials to incorporate magnetic forces to the bearing load carrying capacity [5], the effect of composite liners installed on the pad surface [6]. The common denominator for these research activities revolves around the need to expand the operational envelope of the tilting pad bearing, enabling it to operate at even higher rotational speed, higher load, lower oil supply rate with a high level of reliability and low power consumption.

Some key parameters that have been the target of the bearing optimization process are the oil film temperature, oil flow consumption and oil shear viscous losses. In order to obtain an energy efficient bearing, it is desirable that all three of these parameters are diminished. Several configurations have been tested, being the so called "directed lubrication” or “leading edge groove” technology one of the alternatives widely implemented in the industry. As its name indicates, it resorts to a deep groove machined close to the pad leading edge to feed the bearing clearance with the lubricant. Its performance has been studied [7, 8, 9, 10] in terms of oil consumption rates and bearing viscous losses, exhibiting significant benefits when compared to other tilting pad bearings lubrication methods.

Together with its characteristics related to energy efficiency, the improvement of the tilting pad bearing dynamic properties has been widely investigated. In general, for rotating machinery that is usually designed to operate in the supercritical rotational speed range, it is desirable that the damping provided by the tilting pad bearing is high
enough to ensure safe crossing of critical speeds, and to counteract the onset of instability due to destabilizing forces acting over the supported rotor [11, 12]. Several strategies can be followed to achieve this objective, being the introduction of the “mechatronic” approach one that has received attention in the late years. The mechatronic approach consists of the synergistic coupling of traditional mechanical elements with sensors, processing units and actuation elements derived from electronic or electromechanical engineering [13]. In the case of the studied bearing, the presence of the tilting pads entails some advantages for the application of this approach. Among the different designs that have been tested over the years to render the tilting pad bearing “mechatronic” or “active” one can mention: linear and rotational actuators acting over the bearing pads [14, 15, 16], pads with embedded magnetic actuators [17], and pads with controllable oil injection. The later design, denominated “active lubrication” system [18, 19], entails the usage of servovalve controlled injection of pressurized oil into the bearing clearance through nozzles located radially across the bearing pads. The consequent modification of the oil film flow and pressure field gives the possibility to alter the resulting force over ther rotor, as a function of the servovalve electrical control signals. An extensive series of studies has dealt with the improvement of the mathematical modelling of such active bearing design [20, 21, 22], as well as with its experimental application to control the dynamics of a rotor system [23, 24, 26, 25] and to perform parameter identification procedures in situ [27, 28]. Within this framework, a recent research effort has focused on attempting to combine the mechatronic characteristics arising from the “active lubrication” technology, with the energy efficient features of the leading edge groove tilting pad bearing. The basic concept entails the introduction of servovalves into the oil supply system for the pad leading edge groove, yielding active regulation of the inlet flow and oil film pressure field. Therefore, the leading edge groove tilting pad bearing can be transformed into a mechatronic machine element in a non-invasive fashion, since only the oil supply system needs to be modified. Two steps have been accomplished within this research effort. In [29], the mathematical model for the LEG tilting pad bearing with active lubrication was presented, featuring an elastohydrodynamical lubrication model, coupled with the dynamics of the servovalve controlled oil supply system. Theoretical results portrayed the viability of the proposed active bearing design. Followingly, in [30] the mathematical model was validated at the component level, in a test rig featuring a single pad, and LEG inlet flowregulated by a single servovalve. The objective of the current article is to provide insights into the actively lubricated LEG tilting pad bearing technology (ALEG TPJB). Firstly, experimental and theoretical results are presented, depicting the capabilities of the mechatronic pad design as a calibrated actuator. For this purpose, the subject of investigation corresponds to a simplified setup, consisting of a single active LEG tilting pad supporting a rigid rotor in load on pad configuration. Secondly, theoretical results portray the application of the active bearing design in a configuration resembling in a closer manner an industrial application. The studied setup consists of a flexible rotor supported by two TPJBs, featuring 5 pad, load between pads configuration, where one of them features the ALEG system. Active control schemes synthetized via pole placement and LQR techniques are simulated, depicting the feasibility of reducing the rotor vibrations via the proposed mechatronic bearing design.

2 STUDIED SETUPS

2.1 SETUP 1 (DTU-MEK, Denmark)

The first system to be experimentally and theoretically analyzed is the one depicted in Figure 1. Key parameters of the setup are listed in Table 1. It consists of a rigid rotor supported by a single tilting pad in load on pad configuration. The tilting pad features the leading edge groove oil supply configuration. One end of the oil supply line is connected to the groove chamber, and the other one to the exit port of a high response servovalve, governed by electrical signals synthetized in a control unit. Consequently, the groove inlet flow is actively regulated. The rigid rotor is attached via roller bearings to a lever arm, which features a pivot point and a free side. The free side is used to install displacement sensors and accelerometers to measure indirectly the rotor vertical position. Furthermore, it is used to apply vertical static loading via helical springs, as well as vertical dynamic loading by means of a stinger connected to an electromagnetic shaker. The lever arm is mounted so that the horizontal movement of the rotor is negligible. All analyses are restricted to the frequency range where the lever arm flexible dynamics is not dominant, hence the system is modelled considering only one degree of freedom (arm tilting angle), where the rotor only exhibits vertical motion. A detailed depiction of the LEG tilting pad can be observed in Figure 1. The pad includes the leading edge groove system to provide the oil supply, via the feedline which is connected to the servovalve exit port to render the system controllable. Since the full amount of the oil supply flow reaching the LEG cavity passes through the servovalve, care must be taken to provide a sufficiently high bias voltage to the control signal. This ensures that the pad does not suffer oil starvation during testing. Refering to Figure 1, the pad is instrumented with a piezoelectric pressure sensor to measure directly the pressure
Figure 1: Setup 1: Test rig for the component level study of the ALEG tilting pad. It is composed by: (1) servovalve, (2) LEG tilting pad, (3) lever arm pivot point, (4) lever arm free end, (5) rigid rotor, (6) leading edge groove, (7) LEG oil supply line, (8) piezoelectric pressure transducer, (9) thermocouple within the groove chamber. Furthermore, a thermocouple installed in the pad trailing edge enables to characterize the oil temperature.

2.2 SETUP 2 (PUCV, Chile)

The second system under analysis is depicted in Figure 2. It corresponds to a flexible rotor supported by two identical tilting pad journal bearings featuring the LEG supply configuration. The bearing on the rotor free end includes the ALEG setup. Nevertheless, any of the two bearings are suitable to implement the active system, since the modifications required for it take place outside the bearing itself. The parameters that define the bearing configuration are listed in Table 2. The test rig is currently being assembled and tested, hence for this article only simulation results are presented.

Setup 2 clearly differs from Setup 1, aiming at providing a closer approximation to a real implementation of the studied mechatronic bearing design. It also differs in the manner in which the hydraulic supply system is configured. In Setup 1, the full amount of oil supply reaching the LEG cavity passes through the servovalve mounted in the system, forcing to impose a sufficiently large bias voltage to avoid oil starvation issues. In Setup 2, a “passive” and “active” hydraulic system coexist, enabling to operate the servovalve with zero bias voltage, while preventing oil starvation.

Refering to Figure 3, the passive lubrication system (in black) provides oil supply at all times for each LEG pad. The active lubrication system (in red and blue) enables to modify the oil supply flow reaching each LEG pad via electrical control signals. The output ports of servovalve 1 (SV1, marked in blue) are connected to the supply lines for “upper” pad and the two “lower” ones, whereas servovalve 2 (SV2, marked in red) is associated with pads “left” and “right”. Check valves are installed between the passive and the active lubrication system, in order to prevent counterflow towards the “passive” supply pumps.

The presence of the “passive” supply system enables to operate the servovalves in the vicinities of their closed position (zero bias voltage). Furthermore, the two output ports from the servovalve are connected to “opposite” pads, enabling actuation over the rotor in all relevant directions. For instance, for SV2 a positive voltage signal directs the active flow to the “left” pad, whereas a negative value redirects it to the “right” pad. A similar operation mode is achieved for SV1 and the “upper” and two “lower” pads.

3 MATHEMATICAL MODEL

The mathematical model of the actively lubricated LEG tilting pad bearing technology (ALEG TPJB) has been thoroughly presented in [29, 30]. Consequently, the following presentation is constrained to provide the key points
Figure 2: Setup 2: Test rig for the system level study of the ALEG technology. On the right, a detailed view of the pad arrangement in each bearing is provided. The test rig consists of: (1) ALEG TPJB, (2) Excitation Bearing, (3) Unbalance Disk, (4) Passive LEG TPJB, (5) Driving Transmission and Electric Motor.

Figure 3: Setup 2: Schematics of the hydraulic system providing the oil supply for the ALEG TPJB.

The global linearized mathematical model for the two setups under study is formulated with the following structure:

\[
\dot{\mathbf{X}} = [\mathbf{A}] \{\mathbf{X}\} + [\mathbf{B}] \{\mathbf{U}\} \\
\{\mathbf{Y}\} = [\mathbf{C}] \{\mathbf{X}\} \\
\{\mathbf{X}\} = \{\delta x_r, \delta b, \delta q_x, v\}^T \\
\{\mathbf{U}\} = \{\delta u_V, f_r\}^T
\]

(1)

In Equation (1), the system state vector \(\mathbf{X}\) is defined by the degrees of freedom related to the rotor \(\delta x_r\), the tilting pads \(\delta b\), and the servovalves spool driven flow \(\delta q_x\). The state variables are denoted with the \(\delta\) operator to indicate that they have been linearized around the steady state equilibrium of the system. The system inputs \(\mathbf{U}\) consist of the servovalves control signals \(\delta u_V\) and external excitation forces \(f_r\) applied to perturb the rotor equilibrium position in the experimental setup. The system measurements \(\mathbf{Y}\) are provided by the displacement sensors registering the rotor movement at specific positions.

In open loop operation, a predefined servovalve input signal \(\delta u_V\) affects the servovalve flow \(\delta q_x\), with the consequent modification of the oil supply flow reaching each pad LEG. As a result, the rotor and pads states \(\delta x_r, \delta b\)
Figure 4: Setup 2: Schematics of the flexible rotor arrangement and node numbering for its finite element discretization.

are altered, due to the modification of the oil film and LEG pressure field. In closed loop operation, the servovalve input signals $\delta u_V$ are synthetized by means of suitable control gains. Such controller requires the output of an observer, fed with the measured rotor states $Y$.

Equation (1) corresponds to a linearized mathematical model, suitable to apply linear control theory. Such model stems from the linearization of the following relationships:

1. **Rotor Model**: The modelling of the rotor states $\delta x_r$ differs in the two analyzed setups. In Setup 1, Newton Second Law is applied to model the vertical movement of the rigid rotor supported by the tilting pad. The rotor inertia is corrected to account for the tilting frame that supports it. Consequently, a single degree of freedom representation is obtained. In Setup 2, the finite element method considering Euler-Bernoulli beam elements is applied, to obtain a discretized set of equations. The model accounts for the rotor inertial, gyroscopic and lateral flexibility effects. Figure 4 depicts the rotor finite element discretization and further details are provided in Table 3. The ALEG bearing is mounted at node 2 close to the rotor free end, whereas the passive LEG TPJB is located at node 10, close to the driven end. The sensors measuring the rotor horizontal movement are located at nodes 4, 6 and 8. The excitation disk mounted at node 5 is meant to be used in conjunction with an electromagnetic shaker, in order to provide controlled perturbation forces over the rotor. Furthermore, an unbalance disk is mounted at node 7, with the purpose of incrementing the bearing static load and allowing to apply unbalance loads to the rotor.

2. **Tilting Pads Model**: The tilting pads are discretized using the finite element method in two dimensions (radial and circumferential direction), by means of plane stress assumption and second order quadrilateral elements. Linear elasticity equations and boundary conditions related to the tilting motion around the pivot point are imposed. The obtained finite element model is defined by an equivalent mass and stiffness matrix $M_b$ and $K_b$. Such model is then used to obtain mode shapes vectors related to tilting motion, pivot flexibility and first bending mode, which are arranged in a pseudomodal matrix $V$. Consequently, the pad dynamics can be modeled in a reduced manner using the pseudo modal reduction scheme [31, 21] considering only three degrees of freedom per pad, arranged in a modal coordinates vector $b$ as follows:

$$V^T [M_b] [V] \{b\} + [V]^T [K_b] [V] \{b\} = \{f_b(p)\}$$

(2)

3. **Oil Film Pressure Field**: In order to calculate this field in the circumferential $\hat{x}$ and axial direction $\hat{z}$ of the bearing clearance, the Reynolds Equation for incompressible lubrication is employed:

$$\frac{\partial}{\partial \hat{x}} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial \hat{x}} \right) + \frac{\partial}{\partial \hat{z}} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial \hat{z}} \right) = \frac{\Omega R}{2} \frac{\partial h}{\partial \hat{x}} + \frac{\partial h}{\partial t}$$

(3)

Boundary conditions are imposed to solve numerically Equation (3) via the finite element method. Zero pressure is imposed at the pad surface limits, except for the one between the bearing clearance and the LEG cavity. In this boundary, the imposed value is equal to the pressure at the LEG cavity plus a correction value due to the Rayleigh step effect. A simplified model developed by [32] is implemented to determine it. The viscosity $\mu$ used for Equation (3) is corrected due to thermal effects, based on the oil temperature field.
determined by means of a thermal model. Such model is stated following [33], considering the coupled finite element solution of an energy equation for the oil film and heat conduction through the pad via Fourier law. The formulation determines the temperature field in circumferential and radial direction.

4. Leading Edge Groove Flow and Pressure: Within the leading edge groove cavity, the mass conservation principle dictates the following relationship:

\[
q_V + q\text{lead} - q\text{trail} - q\text{side} = 0
\]

\[
q\text{lead} = \int_{L\text{lead}}^L \left[ \frac{\Omega R h}{2} - \frac{h^3}{12 \mu} \frac{\partial p}{\partial \hat{x}} \right] \text{d}\hat{z}
\]

\[
q\text{trail} = \int_{L\text{trail}}^L \left[ \frac{\Omega R h}{2} - \frac{h^3}{12 \mu} \frac{\partial p}{\partial \hat{x}} \right] \text{d}\hat{z}
\]

\[
q\text{side} = \int_{L\text{lead}}^{L\text{trail}} h^3 \frac{\partial p}{12 \mu} \text{d}\hat{x}
\]

Equation (4) dictates that the oil supply flow \( q_V \) reaching the LEG through the inlet must be equal to the oil flow entering or leaving the cavity through its leading, trailing and side boundaries. Such flow is modelled by means of the Poiseuille and Couette flows, based on the Reynolds model for incompressible lubrication. Consequently, Equation (3) and (4) must be solved in a coupled manner.

5. Hydraulic System Model: the oil supply flow \( q_V \) provided by the hydraulic system to the LEG cavity can be modelled as:

\[
q_V = q_V^* + q_{SV} - K_{pq}p_{port}
\]

\[
p_{port} - p_{LEG} = \frac{8\mu L_{line}}{\pi R_{line}^2} q_V
\]

Equation (5) states that the LEG supply flow \( q_V \) is modelled following a first order linearization, composed of a steady state flow \( q_V^* \), a servovalve spool driven flow \( q_{SV} \) and pressure dependent term given by the \( K_{pq} \) coefficient and the pressure at the servovalve port \( p_{port} \). The pressure difference between the servovalve port \( p_{port} \) and the LEG cavity \( p_{LEG} \) is quantified following the Hagen-Poiseuille model, whereas the relationship between servovalve spool driven flow \( q_{SV} \) and control signal \( u_V \) is given by a second order differential equation that captures the frequency dependant relationship between these parameters.

4 SETUP 1: COMPONENT LEVEL RESULTS

4.1 ALEG Pad as Calibrated Actuator: Theory and Experiment

The first of results deal with the application of the ALEG pad as a calibrated actuator. The experimental and theoretical results aim at evaluating the capability of this mechatronic machine element to generate controllable forces in a wide frequency range. In this case, such forces are applied vertically over the rigid rotor in Setup 1. The experimental arrangement involves fixing the free end of the tilting arm in Setup 1, see Figure 1, with a load cell and an adjustment screw. The arrangement is used to set the rotor vertical position on top of the LEG tilting pad, applying a static load over it. Followingly, a chirp voltage signal is fed to the servovalve, and the resulting force over the rotor is measured indirectly by means of the tilting arm load cell. The postprocessing of these two experimental signals delivers a frequency response function in force/voltage units, that informs how much force is applied over the rotor by the active LEG pad at different frequencies, as well as the phase relationship between input voltage and rotor force.

These experimental frequency response functions can be compared with the ones delivered by the linearized model presented in Equation 1. Since this model includes the dynamics of the hydraulic supply system and the tilting pads, it should be able to portray the frequency dependant relationship between servovalve input voltage and resulting force over the rotor, both in amplitude as well as in phase. The comparison between theory and experiment is provided in Figure 5 and 6. It can be seen that the theoretical model is able to provide a good prediction of the active force generated at the active LEG pad. A critical issue for obtaining good resemblance between theory and experiment is to provide the model with a good estimation of the servovalve cut-off frequency \( \omega_V \) and the
Figure 5: Setup 1: Frequency response function between servovalve input signal and active vertical force over the rotor. A comparison between theoretical model and experimental results is depicted, for rotational speed 1000 RPM and three different static loads over the ALEG pad.

Voltage flow linearized coefficient $R_V$. This is a hidden cost of resorting to a simplified linear model strategy. For higher frequencies, the coherence between theory and experiment diminishes. This is an artifact of the simplified mathematical model that governs the relationship between servovalve spool driven flow and input signal, see Equation 5. The implemented second order model provides better coherence with the servovalve behavior for lower frequencies than higher ones. Furthermore, in the experimental setup the flexibility of the arm becomes relevant as an additional dynamic effect at higher frequencies. Such effect is not accounted for in the theoretical model.

These results prove that the active LEG pad is able to provide measurable forces over the rigid rotor in a wide frequency range. It can be seen that the active force over the rotor tends to diminish for higher static loads, due to predominance of the oil film pressure field in comparison with the pressure developed within the LEG cavity. On the other hand, when varying the rotor rotational speed the frequency dependency between input signal and active force tends to be modified. This is an effect of the alteration of the pad dynamics when incrementing the rotational speed. When analyzing the active bearing as linear system, the pads transfer function is a key component of the global transfer function between servovalve input signal, and rotor resulting force.

4.2 Obtaining Frequency Response Functions by means of the ALEG Pad as Excitation Source

A direct application of the active LEG pad capabilities as a calibrated actuator is to use it as excitation source to perform modal testing in Setup 1. A chirp signal is fed into the servovalve, and the resulting movement of the tilting arm free end is measured by means of displacement sensors. Consequently, a frequency response function can be obtained in displacement/force units, if the active LEG pad calibration function is obtained as portrayed in the previous section. These results can be compared with the benchmark obtained when exciting the tilting arm free end with an electromagnetic shaker equipped with a piezoelectric load cell.

The comparison between the benchmark results and the ones obtained with the active LEG pad as the excitation source are portrayed in Figure 7. It can be observed that the active pad can effectively replace the electromagnetic shaker to perform modal testing in the studied setup, achieving good coherence and good resemblance with the benchmark results. This set of results implies that a bearing equipped with active LEG pads could be employed to perform modal testing in an industrial rotor without the need of installing additional equipment. However, it also requires a good theoretical prediction of the active bearing calibration function, i.e., the transfer function between servovalve input signal and active force over the rotor. This fact justifies the effort in obtaining a good mathematical model of the studied mechatronic bearing.

5 SETUP 2: SYSTEM LEVEL RESULTS

5.1 ALEG Bearing as Calibrated Actuator: theoretical results

The mathematical model for the ALEG bearing can provide a prediction of the active force that can be exerted over the rotor. This model was validated considering the results for Setup 1 presented in the previous section.
Figure 6: Setup 1: Frequency response function between servovalve input signal and active vertical force over the rotor. A comparison between theoretical model and experimental results is depicted, for rotational speed 3000 RPM and three different static loads over the ALEG pad.

Figure 7: Setup 1: Experimental frequency response function of the rotor vertical displacement versus applied force. Benchmark results are obtained using an electromagnetic shaker as excitation source, whereas the ALEG results resort to the controllable forces generated by the active pad. Two operational conditions are tested, characterized by two rotational speeds and applied static loads.

Hence, it could be inferred that the model should also be able to simulate the behavior of the ALEG bearing in Setup 2, regarding the frequency response functions between servovalve input signal and resulting force, in a similar manner to the analysis presented before for Setup 1. Figure 8 and 9 portray the frequency response functions for the simulated bearing between servovalve control signals and active force over the rotor. Two different rotational speeds are included in the simulations. In both cases, it can be seen that the hydraulic arrangement illustrated before in Figure 3 enables to obtain almost purely vertical active forces as actuation for servovalve 1, whereas horizontal forces are obtained by means of servovalve 2. In both cases, some actuation is obtained over the “other” direction, but its magnitude is negligible.

In general, an increment of the rotational speed entails a reduction of the active force magnitude. This is due to the increment of the oil film pressure in comparison to the LEG cavity pressure, as well as the reduction of the LEG cavity pressure. Since the journal retrieves oil from the groove at a faster rate for higher rotational speeds, a lower
Figure 8: Setup 2: Frequency response function between the two servovalve input signals and active force over the rotor in horizontal and vertical direction. Theoretical model results are depicted for rotational speed of 3000 RPM.

Figure 9: Setup 2: Frequency response function between the two servovalve input signals and active force over the rotor in horizontal and vertical direction. Theoretical model results are depicted for rotational speed of 9000 RPM.

pressure is developed in it for the same active input flow. The simulation indicates that the actuation capabilities of the ALEG bearing extend to a wide frequency range, making it suitable to implement active control strategies to limit the rotor vibrations.

5.2 Active Rotor Vibration Control via ALEG Bearing

In this case, the active LEG bearing actuation capability is aimed at reducing the amplitude of the rotor response around resonant areas. For doing so, suitable control laws must be established to synthetize the servovalves control signals. In particular, two methods are implemented to determine the control law: pole placement and LQR. The study is performed by theoretical means, taking advantage of the linearized state space model already presented in Equation 1.
The simulation is setup by applying simultaneously a perturbation force in horizontal and vertical direction over node 5 of the rotor finite element model. The force corresponds to a chirp of amplitude 50 N and frequency ranging from 5 Hz to 200 Hz. The response is measured in node 4, where displacement sensors are mounted in the experimental setup. The time domain simulation results include displacement in the vertical direction at that node, plus the servovalve signals depicting the control effort necessary to alter the rotor dynamics. The frequency domain results deal with the frequency response function between applied load in vertical and horizontal direction, and the resulting displacements in the same directions (i.e. only the direct terms of the frequency response function are plotted). The control objective is to decrease the rotor response amplitude around the resonant areas, while keeping the servovalve input signals within the ±1V range. This is to ensure that the actuation system operates within the linear range of these devices.

Figures 10 and 11 depict the results obtained at two different angular speeds for the passive and active lubrication regimes. In order to close the loop, a pole placement strategy was firstly implemented. The control objective to define the pole placement was to increase by 10\% the damping ratio for the poles with damping ratio below 0.5, and to increase their natural frequency in the same order of magnitude, in order to maintain the amplitude of the system response at lower frequencies.

Figure 10 shows that the system frequency response at the resonance around 40 Hz is significantly reduced when the control strategy is implemented. Additionally, the time domain results show that the maximum system response does not surpass the two tenths of milimeter in the vertical direction nor horizontal (unshown). The control effort is bounded between ±1V for both servovalves. Similar results are obtained at 9000 RPM in Figure 11 regarding reduction of system response around resonance, although with an increased control effort which is slightly above ±1V.

In order to diminish the control effort required to improve the rotor dynamic behavior, a LQR control law is implemented instead of the previous one. For obtaining the LQR control, the weighting matrices are selected in such a way that the control effort does not surpass ±1 V and the response from the system poles in the studied range are equally penalized. Figure 12 and 13 show that LQR results are comparable to figures 10 and 11 in terms of reducing the rotor response in the resonant areas. However, some benefits can be observed in terms of reduced control effort, ensuring that the servovalves operate within their linear range.

6 CONCLUSIONS AND FUTURE PERSPECTIVES

This work provided insights into the actively lubricated LEG tilting pad bearing technology (ALEG TPJB), by means of experimental and theoretical results. This technology derived from the extensive previous research effort on the active lubrication concept for TPJBs, and aims at introducing it in a non invasive manner in passive industrial bearings featuring the leading edge groove design. The main focus of this article was set on presenting the capabilities of this novel mechatronic pad design as a
calibrated actuator, at the component and system level. Firstly, a theoretical and experimental study was carried out for an experimental facility representing a component level implementation of the studied bearing design. The obtained results portrayed the feasibility of employing the active LEG pad to apply controllable forces over a rotor in a wide frequency band. A good resemblance between experimental and theoretical results was obtained, particularly for the lower frequency range, regarding the active LEG pad calibration function, i.e. the transfer function between servovalve input signal and resulting force over the rotor. Experimental results also validated the capabilities of the active LEG pad as an actuator for performing in-situ and non invasive modal testing in rotors. Secondly, simulations of a flexible rotor supported by active LEG bearings were performed, in order to portray an industrial application of the studied mechatronic bearing design. No experimental results were included since the test rig is still under assembly process. It was theoretically demonstrated that the 5 pad ALEG bearing is able to exert controllable forces over a rotor in a wide frequency range. Such capability was then applied to simulate the feasibility of reducing the vibration amplitudes of a flexible rotor, when state feedback control strategies are implemented.

Figure 11: Setup 2: Active rotor vibration control using the ALEG bearing as actuator, for 9000 RPM. Control law synthetized via pole placement.

Figure 12: Setup 2: Active rotor vibration control using the ALEG bearing as actuator, for 3000 RPM. Control law synthetized via LQR.
The next step within this research effort is to obtain experimental data to validate the ALEG TPJB concept at the system level, concerning the active vibration control of a flexible rotor supported by this novel mechatronic machine element. The experimental facilities to carry out this objective are currently being assembled at PUCV Chile, and should deliver results to complement the theoretical results presented here for Setup 2.

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**REFERENCES**


### Table 1: Parameters for Setup 1

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<th>Parameter</th>
<th>Value</th>
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<td>Pad inner radius</td>
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<tr>
<td>Journal radius</td>
<td>49.692 mm</td>
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<td>Bearing axial direction length</td>
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<tr>
<td>LEG axial length</td>
<td>50 mm</td>
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<tr>
<td>LEG angular position (measured from pad edge)</td>
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<td>LEG circumferential direction width</td>
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</tr>
<tr>
<td>Oil Supply Pump Maximum Pressure</td>
<td>250 bar</td>
</tr>
<tr>
<td>Oil Supply Pump Maximum Flow</td>
<td>2.5 liters per minute</td>
</tr>
<tr>
<td>Servovalve cut-off frequency $\omega_V^1$</td>
<td>150 Hz</td>
</tr>
<tr>
<td>Servovalve leakage flow $q_{pq}^V$</td>
<td>Variable</td>
</tr>
<tr>
<td>Servovalve flow pressure coeff. $K_{pq}^V$</td>
<td>1e-12 m³/(s Pa)</td>
</tr>
<tr>
<td>Servovalve flow voltage coeff. $R_{V}^1$</td>
<td>Variable</td>
</tr>
<tr>
<td>Servovalve damping ratio $\xi_V^1$</td>
<td>0.95</td>
</tr>
</tbody>
</table>

### Table 2: Parameters for Setup 2

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad inner diameter</td>
<td>99.840 mm</td>
</tr>
<tr>
<td>Journal diameter</td>
<td>99.347 mm</td>
</tr>
<tr>
<td>Preload</td>
<td>0.35</td>
</tr>
<tr>
<td>Bearing axial direction length</td>
<td>50 mm</td>
</tr>
<tr>
<td>LEG axial length</td>
<td>45 mm</td>
</tr>
<tr>
<td>LEG angular position (measured from pad edge)</td>
<td>8 °</td>
</tr>
<tr>
<td>LEG circumferential direction width</td>
<td>8 mm</td>
</tr>
<tr>
<td>LEG depth</td>
<td>10 mm</td>
</tr>
<tr>
<td>Number of pads</td>
<td>5</td>
</tr>
<tr>
<td>Pad arc</td>
<td>65 °</td>
</tr>
<tr>
<td>Offset</td>
<td>0.5</td>
</tr>
<tr>
<td>Load Angle Between pad</td>
<td>Between pad</td>
</tr>
<tr>
<td>Applied Load</td>
<td>500 N</td>
</tr>
<tr>
<td>Pad thickness</td>
<td>16 mm</td>
</tr>
<tr>
<td>Oil type</td>
<td>ISO VG32</td>
</tr>
<tr>
<td>Pad material</td>
<td>Brass</td>
</tr>
<tr>
<td>Pivot insert material</td>
<td>Steel</td>
</tr>
<tr>
<td>Pivot design</td>
<td>Rocker</td>
</tr>
<tr>
<td>Servovalve cut-off frequency $\omega_{V1}$</td>
<td>150 Hz</td>
</tr>
<tr>
<td>Servovalve cut-off frequency $\omega_{V2}$</td>
<td>150 Hz</td>
</tr>
<tr>
<td>Servovalve 1 flow pressure coeff. $K_{pq1}$</td>
<td>1e-12 m³/(s Pa)</td>
</tr>
<tr>
<td>Servovalve 2 flow pressure coeff. $K_{pq2}$</td>
<td>1e-12 m³/(s Pa)</td>
</tr>
<tr>
<td>Servovalve flow voltage coeff. $R_{V1}$</td>
<td>Variable</td>
</tr>
<tr>
<td>Servovalve flow voltage coeff. $R_{V2}$</td>
<td>Variable</td>
</tr>
<tr>
<td>Servovalve damping ratio $\xi_{V1}$</td>
<td>0.95</td>
</tr>
<tr>
<td>Servovalve damping ratio $\xi_{V2}$</td>
<td>0.95</td>
</tr>
</tbody>
</table>
Table 3: Finite element discretization for the flexible rotor in Setup 2

<table>
<thead>
<tr>
<th>Element number #</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter [mm]</td>
<td>99.350</td>
<td>99.350</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>99.350</td>
<td>99.350</td>
<td>80</td>
<td>40</td>
</tr>
<tr>
<td>Length [mm]</td>
<td>50</td>
<td>50</td>
<td>20</td>
<td>180</td>
<td>100</td>
<td>200</td>
<td>180</td>
<td>20</td>
<td>50</td>
<td>50</td>
<td>80</td>
<td>50</td>
</tr>
</tbody>
</table>