# Design and evaluation of two types of active tilting pad journal bearings

**I.F.Santos**
State University of Campinas, Brazil

**SYNOPSIS** Recent experimental and theoretical investigations have shown that an increase in damping and stabilization is possible by applying control techniques using hydraulic servo systems. For control purposes two types of active tilting pad journal bearings have been designed: a TPJB combined with a hydraulic chamber system fed by a servo valve and a proportional valve, and a TPJB with active oil film, also fed by a servo valve and a proportional valve. In both cases the proportional valve is used to change the damping and stiffness of the bearing and the servo valve for controlling vibrations of the rotating parts of the machines. Details about the design, operation, modelling and performance of this active bearings are given. Theoretical and experimental results are also presented and discussed.

## NOTATION

| | |
|---|---|
| $d_{ij}$ | damping coefficient - bearing |
| $d_o$ | diameter of the orifices |
| $h_o$ | initial thickness of oil film |
| $h(x,t)$ | oil film thickness |
| $k_{ij}$ | stiffness coefficient - bearing |
| $l$ | length of the orifices |
| $m_S$ | mass of pad and connections |
| $n$ | order of the state matrix |
| $o$ | number of orifices |
| $p(x,z,t)$ | distribution of pressure |
| $r$ | number of active elements |
| $\bar{r}$ | rotor radius |
| $\bar{r}_S$ | pad radius |
| $s$ | number of sensors |
| $t$ | time |
| $x, y, z$ | coordinate system fixed on the pad |
| $\mathbf{u}(t)$ | control vector |
| $\mathbf{x}(t)$ | state vector - complete system |
| $\mathbf{y}(t)$ | output vector |
| $A$ | membrane area |
| $\mathbf{A}$ | state matrix - complete system |
| $B$ | width of the journal bearing |
| $\mathbf{B}$ | control matrix |
| $\mathbf{C}$ | measure matrix |
| $F_P(t)$ | piezoactuator force |
| $I_A$ | lever-rotor moment of inertia |
| $I_S$ | pad and sensor moment of inertia |
| $K_M$ | membrane stiffness |
| $K_P$ | piezoactuator stiffness |
| $K_{PQ}$ | leakage coefficient |
| $K_V$ | $= \omega_V^2 \cdot Q_N/U_N \cdot \sqrt{\Delta P_S/\Delta P_N}$ |
| $L$ | length of the pads |
| $L_{AP}$ | length AP - lever |
| $L_{AR}$ | length AR - lever |
| $L_P$ | piezoactuator length |
| $P$ | supply pressure |
| $\bar{P}_o$ | inlet pressure |
| $P_1(t)$ | pressure - chamber 1 |
| $P_2(t)$ | pressure - chamber 2 |
| $Q_1(t)$ | oil flow to chamber 1 |
| $Q_2(t)$ | oil flow to chamber 2 |
| $Q_N$ | nominal flow |
| $Q_V(t)$ | oil flow - unload servo valve |
| $R$ | pressure in the resevoir |
| $U$ | circumferential rotor speed |
| $U_N$ | nominal voltage - servo valve |
| $U_P(t)$ | piezoactuator voltage |
| $U_V(t)$ | control voltage - servo valve |
| $V_{Lo}$ | oil volume (chamber + piping) |
| $Y_R(t)$ | rotor vertical displacement |
| $Y_{S1}(t)$ | pad 1 linear displacement |
| $Y_{S2}(t)$ | pad 2 linear displacement |
| $\beta_{K'}$ | compressibility module |
| $\mu$ | oil dynamical viscosity |
| $\xi_V$ | damping in the servo valve |
| $\rho$ | oil density |
| $\varphi_{S1}(t)$ | tilt motion - pad 1 |
| $\varphi_{S2}(t)$ | tilt motion - pad 2 |
| $\omega_V$ | eigenfrequency of the servo valve |
| $\Delta P$ | pressure drop in the valve |
| $\Delta P_N$ | nominal pressure difference |
| $\Delta P_S$ | supply pressure difference |
| $\Delta s$ | pad thickness |
| | time derivative |

## 1 INTRODUCTION

Tilting Pad Journal Bearings can be found in a variety of industrial applications due to their stabilizing effects and good damping behavior. This particular type of hydrodynamical bearing in its passive form has been investigated by many authors since 1953, on the basis of advanced models and experiments (Boyd (2), Lund (11), Malcher (12), Klumpp (9), Springer (20) (21), Jones (8), Ettles (5), Allaire (1), Parsell (13), Rouch (15), Glienicke (7), Flack (6), Someya (19), Brockwell (3), Lie (10), Earles (4), Rodkiewicz (14), All of the mentioned investigations concentrate on the properties of such bearings when they operate passively (or without control).

A further increase in damping and stabilization can be obtained when these bearings operate actively (or with control). The first ideas about an active tilting pad journal bearing were presented by Ulbrich (22), where the pads are actively controlled through piezoactuators. The first theoretical and experimental investigations about these active bearings (using hydraulic actuators connected to the bearing pads) can be found in Santos (16), (17), (18)).

This paper gives theoretical and experimental contributions to the problem of tilting pad journal bearings connected to the control system design and to the problem of the active modification of the properties of hydrodynamical bearings. Different options for increasing the bearing damping and the stable range of operation of a rotor-bearing system are investigated: (a) by means of adjusting the bearing gap, (b) with help of control techniques and (c) by influencing directly the velocity and pressure distribuition of the oil film in the radial and axial direction. Cases (a) and (b) are theoretically and experimentally investigated with help of a TPJB connected to a hydraulic chamber system, and case (c) only theoretically, using a TPJB with active oil film.

## 2 ACTIVE TPJB WITH HYDRAULIC CHAMBER SYSTEM

### 2.1 Design

The journal bearing under investigation has 4 pads, 2 in the vertical direction and 2 in the horizontal direction (see Fig.1). The active tilting pad journal bearing shown possesses only 2 active pads arranged in the vertical direction.

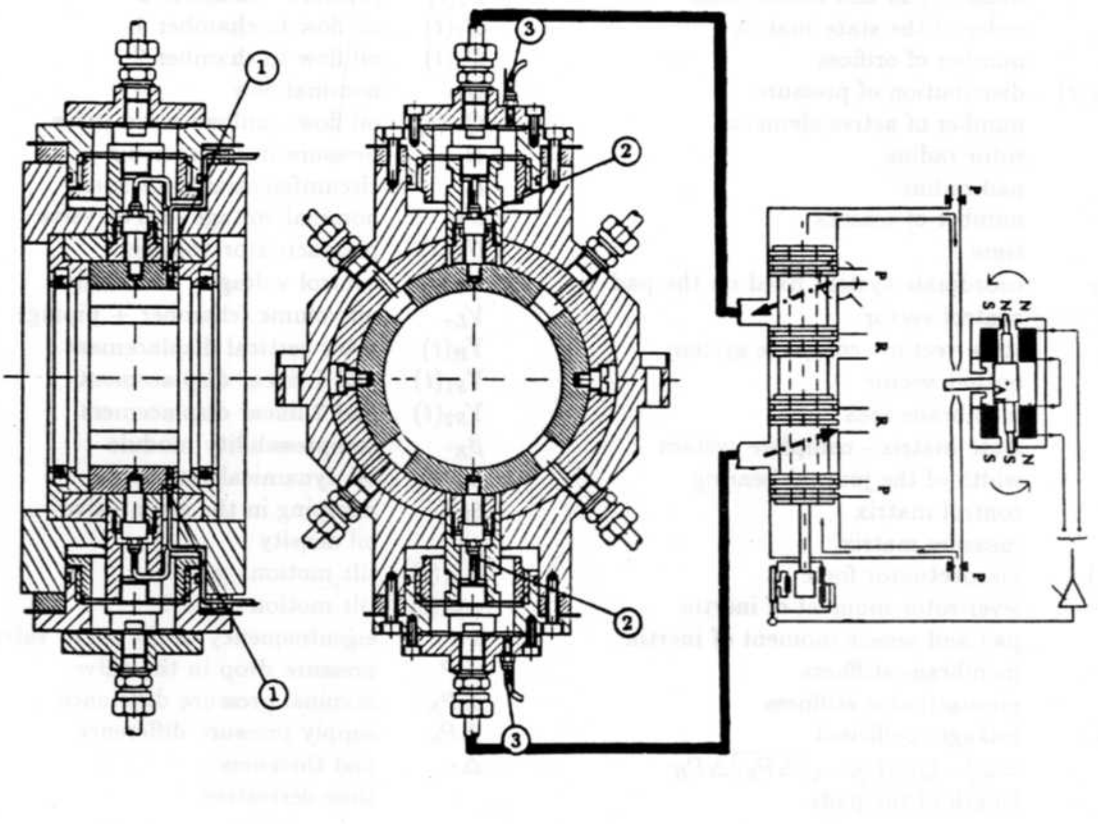
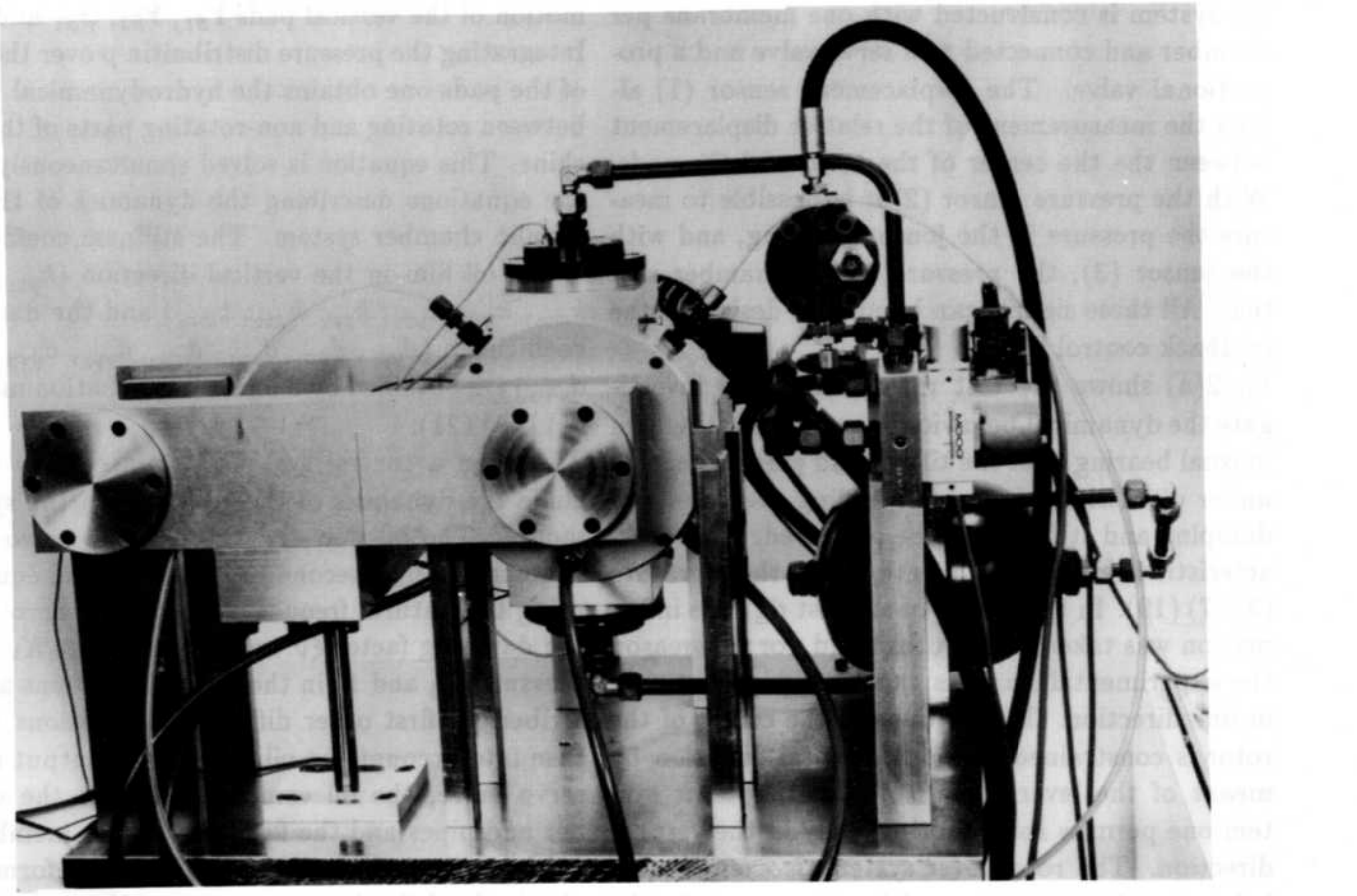



Figure 1: *TPJB connected to hydraulic chambers and servo valve*

(a)



(b)

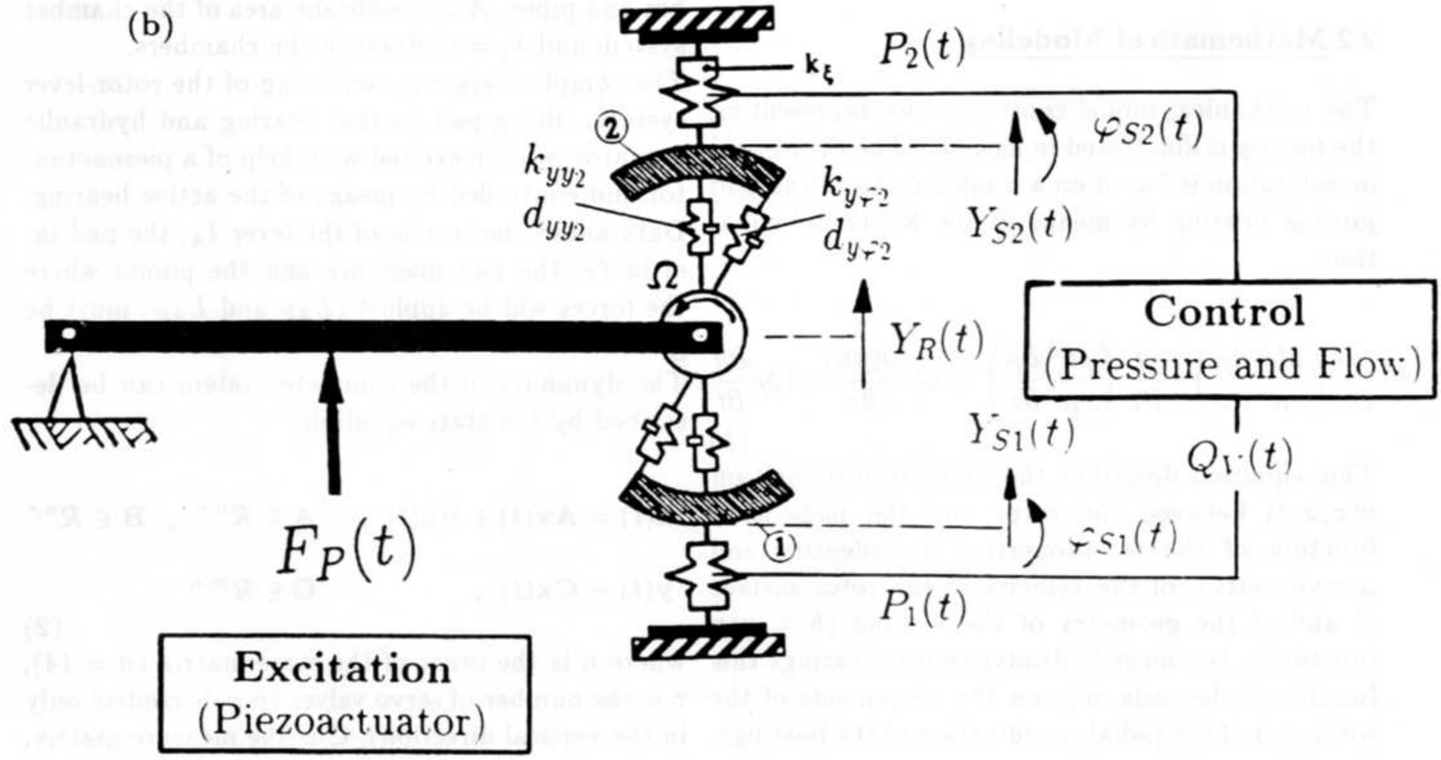



Figure 2: *Test rig of the active tilting pad journal bearing.* (a) *Photograph of the test device.* (b) *Mechanical model of the test rig*

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 sensor (1) allows the measurement of the relative displacement between the 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 the feedback control law.

Fig.2(a) shows the test rig developed to investigate the dynamical behavior of an active tilting pad journal bearing. For the tilting pad journal bearing under discussion, the cross coupling coefficients of damping and stiffness can be neglegted. This characteristic is extensively mentioned in the literature (3) (7) (19). In the design of the test rig, this information was taken into account and, for this reason the experimental analysis can be carried out only in one direction. The motions of the center of the rotor is constrained in the horizontal direction by means of the lever system. With this lever system one permits rotor motions only in the vertical direction. The rotor-lever system, is excited with help of a piezoactuator and its movements in the vertical direction are controlled through the active TPJB.

## 2.2 Mathematical Modeling

The mechanical model contructed for representing the test rig is illustrated in fig.2(b). The theoretical investigation is based on a model of the tilting pad journal bearing by means of the Reynolds' equation:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{\mu}\frac{\partial p}{\partial z}\right)=6U\frac{\partial(\rho h)}{\partial x}+12\rho\frac{\partial h}{\partial t} \tag{1}$$

This equation describes the pressure distribuition $p(x,z,t)$ between the rotor and the pads as a function of the oil properties ($\rho$ =density and $\mu$ =viscosity), of the velocity of the rotor surface $U$ and of the geometry of the bearing ($h$ = gap function). For most hydrodynamical bearings this function $h$ depends only on the movements of the rotor and of the radial coordinate $x$ of the bearings. For tilting pad journal bearings supported by flexible elements (membrane of the hydraulic actuator). this function depends also on the movements of the pads.

For our analysis $h$ depends on the vertical displacement of the rotor $Y_R$ and on the linear and tilt motion of the vertical pads $Y_{S1}$, $Y_{S2}$, $\varphi_{S1}$ and $\varphi_{S2}$. Integrating the pressure distribuitin $p$ over the area of the pads one obtains the hydrodynamical forces between rotating and non-rotating parts of the machine. This equation is solved simultaneously with the equations describing the dynamics of the hydraulic chamber system. The stiffness coefficients of the oil film in the vertical direction ($k_{yy_1}$, $k_{y\varphi_1}$ $k_{\varphi y_1}$, $k_{\varphi\varphi_1}$ $k_{yy_2}$, $k_{y\varphi_2}$ $k_{\varphi y_2}$, $k_{\varphi\varphi_2}$) and the damping coefficients ($d_{yy_1}$, $d_{y\varphi_1}$ $d_{\varphi y_1}$, $d_{\varphi\varphi_1}$ $d_{yy_2}$, $d_{y\varphi_2}$ $d_{\varphi y_2}$, $d_{\varphi\varphi_2}$) are obtained by a linear perturbation method (1) (18) (21).

To design a control system, it is necessary to include the dynamics of the actuators in the system model. The oil flow $Q_V$ through the servo valve is described by a second order differential equation using the natural frequency $\omega_V$ of the servo valve, the damping factor $\xi_V$ and the constant $K_V$. The pressures $P_1$ and $P_2$ in the chamber systems are described by first order differential equations which take into account the oil flow at the output of the servo valve, the oil compressibility in the chamber and pipes and the flexibility of the membrane. In this way, it is necessary to obtain information about the following parameters: $K_{PQ}$ = leakage coefficient of the servo valve, $\beta_K$ = compressibility module of the oil, $V_{Lo}$ = oil volume in the chamber and pipes, $A$ = membrane area of the chamber system and $k_\xi$ = stiffness of the chambers.

The complete system, consisting of the rotor-lever system, tilting pad journal bearing and hydraulic actuator, will be excited with help of a piezoactuator and controlled by means of the active bearing. Data about the inertia of the lever $I_A$, the pad inertia $I_S$, the pad mass $m_S$ and the points where the forces will be applied ($L_{AP}$ and $L_{AR}$) must be given.

The dynamics of the complete system can be described by the state equation:

$$\begin{aligned}\dot{\mathbf{x}}(t)&=\mathbf{A}\mathbf{x}(t)+\mathbf{B}\mathbf{u}(t)\ , &\mathbf{A}\in\mathcal{R}^{n,n}\ ,\ \mathbf{B}\in\mathcal{R}^{n,r}\\ \mathbf{y}(t)&=\mathbf{C}\mathbf{x}(t)\ , &\mathbf{C}\in\mathcal{R}^{m,n}\end{aligned} \tag{2}$$

where $n$ is the order of the state matrix ($n$ = 14), $r$ is the number of servo valves ($r$ = 1, control only in the vertical direction), $\mathbf{C}$ is the measure matrix, $\mathbf{y}(t)$ is the measure vector and $m$ is the number of measured signals.

The vector $\mathbf{x}(t)$ is the state vector of the complete system:

$$\mathbf{x}(t) = \left\{ Y_R\,,\, Y_{S1}\,,\, Y_{S2}\,,\, \varphi_{S1}\,,\, \varphi_{S2}\,,\, \dot{Y}_R\,,\, \dot{Y}_{S1}\,,\, \dot{Y}_{S2}\,,\, \dot{\varphi}_{S1}\,,\, \dot{\varphi}_{S2}\,,\, Q_V\,,\, \dot{Q}_V\,,\, P_1\,,\, P_2 \right\}^T. \tag{3}$$

The matrix $\mathbf{A}$ is the state matrix of the complete system:

$$\mathbf{A} = \left[ \begin{array}{cccccccccccccc}
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\
-\frac{L_{AR}^2}{I_A}k_0^* & \frac{L_{AR}^2}{I_A}k_{yy_1} & \frac{L_{AR}^2}{I_A}k_{yy_2} & \frac{L_{AR}^2}{I_A}k_{y\varphi_1} & \frac{L_{AR}^2}{I_A}k_{y\varphi_2} & -\frac{L_{AR}^2}{I_A}d_1^* & \frac{L_{AR}^2}{I_A}d_{yy_1} & \frac{L_{AR}^2}{I_A}d_{yy_2} & \frac{L_{AR}^2}{I_A}d_{y\varphi_1} & \frac{L_{AR}^2}{I_A}d_{y\varphi_2} & 0 & 0 & 0 & 0 \\
\frac{1}{m_S}k_{yy_1} & -\frac{1}{m_S}k_1^* & 0 & -\frac{1}{m_S}k_{y\varphi_1} & 0 & \frac{1}{m_S}d_{yy_1} & -\frac{1}{m_S}d_{yy_1} & 0 & -\frac{1}{m_S}d_{y\varphi_1} & 0 & 0 & 0 & -\frac{1}{m_S}A & 0 \\
\frac{1}{m_S}k_{yy_2} & 0 & -\frac{1}{m_S}k_2^* & 0 & -\frac{1}{m_S}k_{y\varphi_2} & \frac{1}{m_S}d_{yy_2} & 0 & -\frac{1}{m_S}d_{yy_2} & 0 & -\frac{1}{m_S}d_{y\varphi_2} & 0 & 0 & 0 & -\frac{1}{m_S}A \\
\frac{1}{I_S}k_{\varphi y_1} & -\frac{1}{I_S}k_{\varphi y_1} & 0 & -\frac{1}{I_S}k_{\varphi\varphi_1} & 0 & \frac{1}{I_S}d_{\varphi y_1} & -\frac{1}{I_S}d_{\varphi y_1} & 0 & -\frac{1}{I_S}d_{\varphi\varphi_1} & 0 & 0 & 0 & 0 & 0 \\
\frac{1}{I_S}k_{\varphi y_2} & 0 & -\frac{1}{I_S}k_{\varphi y_2} & 0 & -\frac{1}{I_S}k_{\varphi\varphi_2} & \frac{1}{I_S}d_{\varphi y_2} & 0 & -\frac{1}{I_S}d_{\varphi y_2} & 0 & -\frac{1}{I_S}d_{\varphi\varphi_1} & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\omega_V^2 & -2\xi_V\omega_V & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & \frac{\beta_K}{V_{Lo}}A & 0 & 0 & 0 & \frac{\beta_K}{V_{Lo}} & 0 & -\frac{\beta_K}{V_{Lo}}K_{PQ} & \frac{\beta_K}{V_{Lo}}K_{PQ} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{\beta_K}{V_{Lo}}A & 0 & 0 & -\frac{\beta_K}{V_{Lo}} & 0 & \frac{\beta_K}{V_{Lo}}K_{PQ} & -\frac{\beta_K}{V_{Lo}}K_{PQ}
\end{array} \right] \tag{4}$$

where

- $d_1^* = (d_{yy_1} + d_{yy_2})$,
- $k_0^* = (\frac{L_{AP}^2}{L_{AR}^2}K_P + k_{yy_1} + k_{yy_2})$,
- $k_1^* = (k_{yy_1} + k_\xi)$ and
- $k_2^* = (k_{yy_2} + k_\xi)$.

The matrix $\mathbf{B}$ is the control matrix:

$$\left[ \begin{array}{cccccccccccccc} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & K_V & 0 & 0 \end{array} \right]^T \tag{5}$$

Next the design of the control system for the rotor-lever system supported by the bearing is discussed.

The first step for the control design is to define the objectives of the control system. The goal here is to optimize the damping to reduce vibrations. The **output feedback** for the servo valve is constructed here with the following signals: difference of pressures in the chamber system $(P_1(t) - P_2(t))$, and vertical velocity and displacement of the center of the rotor ($\dot{Y}_R(t)$ and $Y_R(t)$). In this case, $m = 3$ and $n = 14$, the measure matrix $\mathbf{C} \in \mathcal{R}^{m,n}$ is given as:

$$\left[ \begin{array}{cccccccccccccc}
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & -1
\end{array} \right] \tag{6}$$

The optimal gains $\mathbf{K}$ for the feedback

$$\mathbf{u}(t) = -\mathbf{K}\mathbf{y}(t) \quad (7)$$

is obtained when the real part of the eigenvalues of the closed-loop model

$$\dot{\mathbf{x}}(t) = (\mathbf{A} - \mathbf{BKC})\mathbf{x}(t) = \bar{\mathbf{A}}\mathbf{x}(t) \quad (8)$$

achieves a maximum negative value.

## 2.3 Performance

In these experimental studies the rotor ($\bar{r}$ = 0.04937 $m$) operates at rotational speeds of 20 and 40 $Hz$, and with an initial oil film tickness of 110 $\mu m$. The bearing possesses 4 pads with $L$ = 0.05264 $m$, $B$ = 0.05600 $m$ and $\bar{r_S}$ = 0.05027 $m$. The oil used for lubricating the bearing is ISO VG46 ($\mu = 0,070\ Ns/m^2$ and $\rho = 900\ kg/m^3$).
Fig.3 shows the transfer function of the rotor-lever system supported by the tilting pad journal bearing in 5 cases :

(a) The TPJB operates without control.

(b) The chamber pressure is fixed at about 4 $bar$ - without control.

(c) In this case the feedback is constructed with the signal of the pressure sensors in the chamber systems. One changes the gain of the pressure feedback and finds an optimal value for the gain.

(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 the rotor is calculated.

(e) The feedback is also designed with the signal of the displacement of the center of the rotor. The optimal gain of the pressure and of the velocity feedback 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.

The damping and stiffness of the bearing depend on the thickness of the oil film. Through the proportional valve, one can change the static pressure in the chamber system and the oil film thickness between rotor and pads to modify the damping and stiffness coefficients of the bearing. Such modifications of the bearing coefficients are realized without feedback control, using only the non-linearities of the bearing coefficients, namely their dependency on the bearing gap.Fig. 4 illustrates the increase of the damping and stiffness of the journal bearing for 3 pressure values in the chamber system: 0, 4 and 14 $bar$.

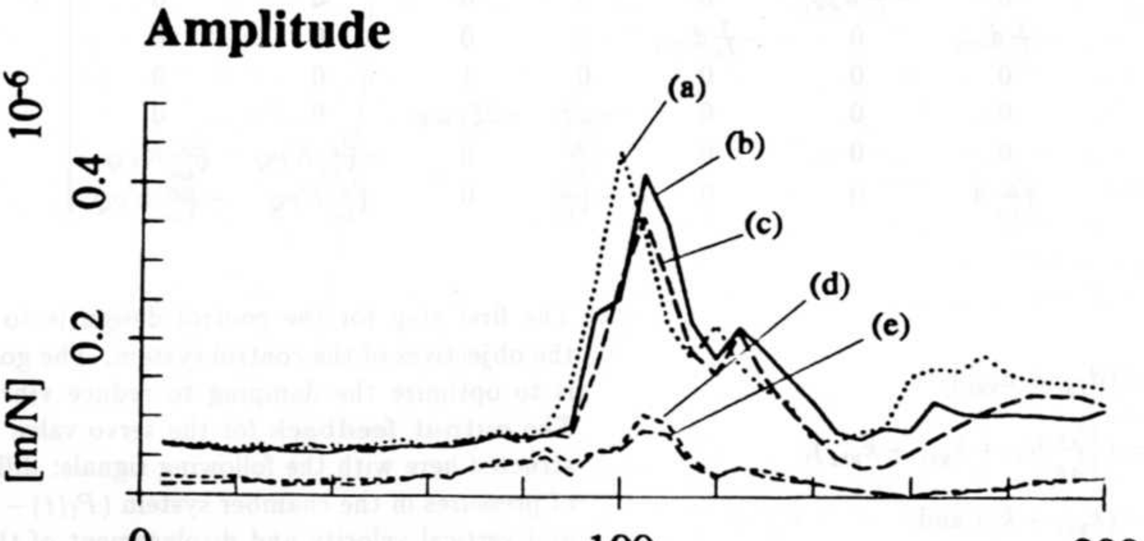



Figure 3: *Experimental transfer function $Y_R/F_P$ of the rotor-lever system excited with help of the piezoactuator in different conditions: (a) passive. (b) chamber pressure of 4 bar, but without control. (c) feedback: chamber pressures. (d) feedback: chamber pressures and rotor linear velocity. (e) feedback: chamber pressures, rotor linear velocity and displacement - Rotor speed: 40 Hz.*

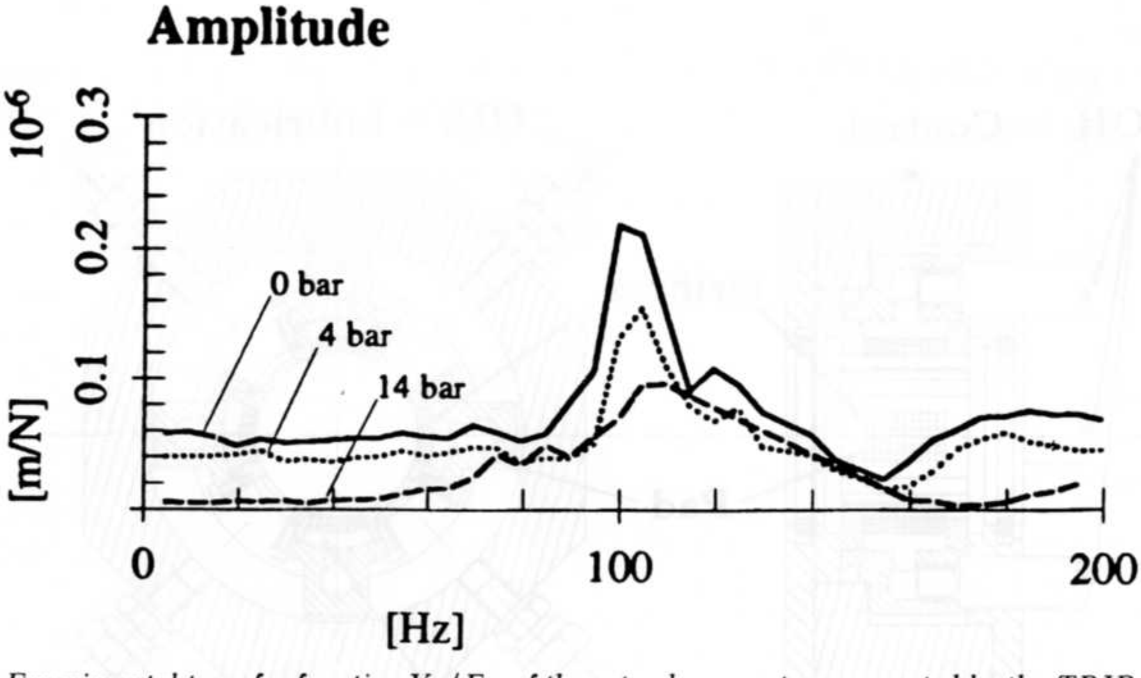



Figure 4: *Experimental transfer function $Y_R/F_P$ of the rotor-lever system supported by the TPJB connected with the hydraulic chambers, in the cases of different pressures in the chambers (without feedback control) - Rotor speed:* 20 $Hz$.

## 3 TPJB WITH ACTIVE OIL FILM

Another way to increase the damping of the bearing is to influence directly the kinematics of the fluid flow in the bearing gap. This can be done in the case of a TPJB with active oil film by adjustment of the velocity and pressure distribution of the fluid in the radial and axial directions. This adjustment is made by changing the inlet control pressure $\bar{P}_o$ in the gap through these orifices, with help of a proportional valve and a servo valve. These orifices are distributed over the surface of the pads (see Fig.5). This new type of active hydrodynamical bearing is under theoretical investigations and details about its design and operation follows.

### 3.1 Design

The TPJB with active oil film has 4 pads, 2 in the vertical direction and 2 in the horizontal direction (see Fig.5). These pads are mounted in the housing without hindering the tilting motion. The pads are constructed with many small orifices. These are connected to two servo valves (for controlling the rotor vibrations in the vertical and horizontal directions) and one proportional valve (for changing the bearing coefficients). Information about the absolute displacement of the rotor and the pressure in the pads $\bar{P}_o$ can be used in designing the feedback control law.

### 3.2 Modeling

This theoretical investigation is based on a model of the tilting pad journal bearing using the Navier-Stokes and the continuity equation. Considering the same assumptions used for obtaining the Reynolds' equation and taking into account the velocity distribuition of the fluid in the radial direction (through the orifices), one obtains the following partial differential equation:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{\mu}\frac{\partial p}{\partial z}\right)+\frac{12}{\mu l}\sum_{i=1}^{o}\mathcal{F}_i(x,z)\cdot p =$$

$$= 6U\frac{\partial(\rho h)}{\partial x}+12\rho\frac{\partial h}{\partial t}+\frac{12}{\mu l}\sum_{i=1}^{o}\mathcal{F}_i(x,z)\cdot\bar{P}_o \quad (9)$$

where

$$\mathcal{F}_i(x,z)=\begin{cases} =\frac{d_o^2}{4}-(x-x_i)^2-(z-z_i)^2 & \text{, if } (x-x_i)^2-(z-z_i)^2\leq\frac{d_o^2}{4} \\ =0 & \text{, if } (x-x_i)^2-(z-z_i)^2\geq\frac{d_o^2}{4} \end{cases} \quad (10)$$

and $(i=1,...,o)$ is the number of orifices over the surface of a pad, $(x_i,z_i)$ are the coordinates of the center of the $i$-th orifice, $d_o$ is the diameter of the orifices and $l$ is the length of the orifices.

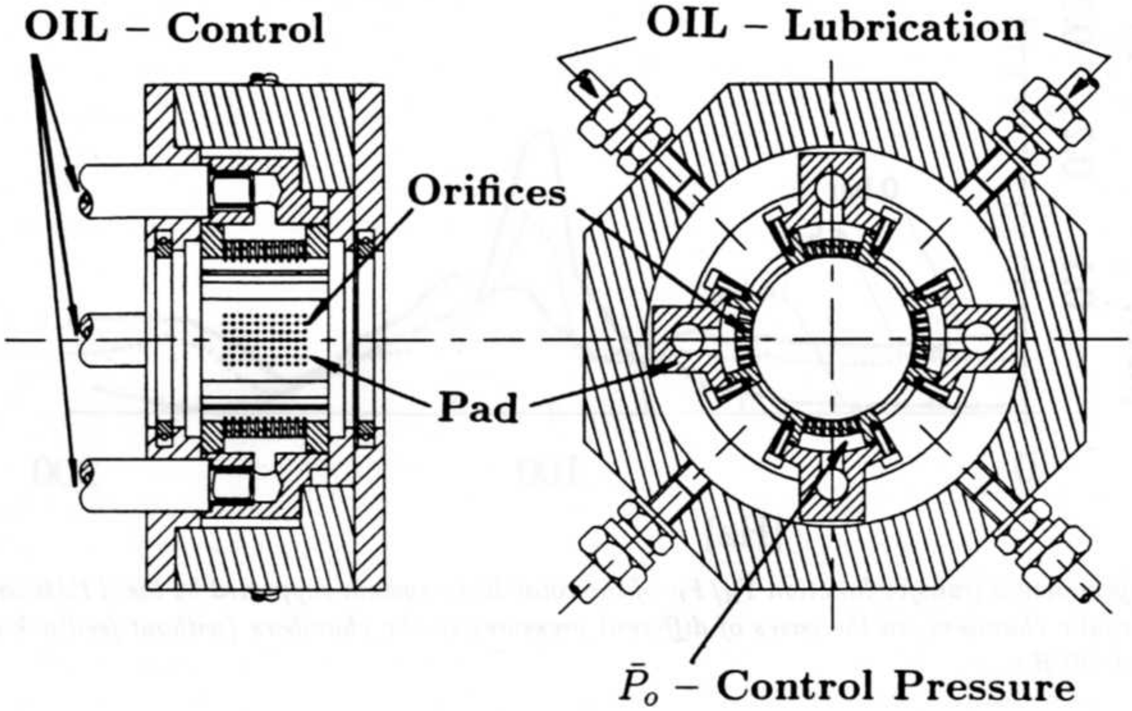



Figure 5: *TPJB with active oil film*

The eq.(9) describes the pressure distribuition $p(x, z, t)$ between the rotor and the pads as a function of the oil properties ($\rho$ and $\mu$), of the velocity of the rotor surface $U$, of the geometry of the bearing $h$ and finally, as a function of the inlet pressure $\bar{P}_o$. When one compares eq.(1), used for modeling a normal hydrodynamical bearing, and eq.(9), used for modeling the hydrodynamical bearing with active oil film, one can see that the pressure and velocity distribution of the lubricant can be changed by means of the inlet pressure $\bar{P}_o$. Consequently, it is possible to modify the hydrodynamical forces and the coefficients of the tilting pad journal bearing by using these special kinds of pads connected to a proportional valve and/or a servo valve.

## 4 CONCLUSIONS

The adjustment of the gap between the rotor and pads with help of the proportional valve permits an increase in the damping and stiffness properties of the tilting pad journal bearings, because these properties depend strongly on the bearing clearance. Examining the kinematics and continuity of the fluid flow in the bearing gap, it is possible to verify that increasing the damping only with an increase in the static pressure of the chamber system and the respective reduction of this gap is limited mainly to high speed rotors. A further increase in damping is realized with help of a servo valve and the design of an optimal control system. An experimental comparision between the passive and the active behavior of a tilting pad journal bearing is performed. By using this type of bearing on a rotating system one can get a significant reduction of vibration amplitude of rotors crossing critical speeds. The theoretical and experimental results show an excellent increase in damping and stabilization by using active tilting pad journal bearings. A further increase in damping and stabilization seems possible by directly influencing the fluid flow in the bearing gap. This theoretical investigations is being carried out.

## 5 ACKNOWLEDGEMENTS


The author wish to thank the Brazilian foundation FAPESP (Fundação de Amparo à Pesquisa do Estado de São Paulo), the German foundation DFG (Deutsche Forschungsgesellschaft) and the LBM (Lehrstuhl B für Mechanik der Technische Universität München) for their assistance and support.

## REFERENCES


(1) ALLAIRE, P.E., PARSELL, J.A. and BARRETT, L.E., A Pad Perturbation Method for the Dynamic Coefficients of Tilting Pad Journal Bearings, *Wear*, 1981, 72, 29-44.

(2) BOYD, J., and RAIMONDI, A.A., An Analysis of the Pivoted-Pad Journal Bearing, *Mech. Engrg.*, 1953, 75, N.5, 380-386.

(3) BROCKWELL, K. and DMOCHOWSKI, W., Experimental Determination of the Journal Bearing Oil Film by the Method of Selective Vibration Orbits, *12th Biennial ASME Conference on Vibration and Noise - Montreal/Canada*, 1989, 245-259.

(4) EARLES, L.L., PALAZZOLO, A.B. and ARMENTROUT, R.W., A Finite Element Approach to Pad Flexibility Effects in Tilt Pad Journal Bearings: Part I - Single Pad Analysis, *ASME Journal of Tribology*, 1990, 112, 169-177.

(5) ETTLES, C.M.McC., The Analysis and Performance of Pivoted Pad Journal Bearings considering Thermal and Elastic Effects, *J.Lubr. Tech., ASME Trans.*, 1980, 102, N.2, 182-194.

(6) FLACK, R.D. and ZUCK, C.J., Experiments on the Stability of Two Flexible Rotor in Tilting Pad Journal Bearing, *Tribology Trans.*, 1988, 31, N.2, 251-257.

(7) GLIENICKE,J. Stabilitätsprobleme bei Lagerung schnellaufender Wellen - Berechnung, Konstruktion und Verhalten von Mehrflächen- und Kippsegmentlagern, *Technische Akademie Wuppertal*, 1987, Sem-Nr 111107.

(8) JONES, G.J. and MARTIN, F.A.,Geometry Effects in Tilting Pad Journal Bearings, *ASLE Trans.*, 1979, 22, N.3, 227-246.

(9) KLUMPP,R., Ein Beitrag zur Theorie von Kippsegmentlagern, *Dissertation - Universität Karlsruhe*, 1975.

(10) Lie,Y., You-Bai,Z.J. and Damou,Q., Experiments on the Destabilizing Factors in Tilting Pad Journal Bearings, *Tribology International*, 1989, 22, N.5, 329-334.

(11) LUND,J.W., Spring and Damping for the Tilting Pad Journal Bearings, *ASLE Trans.*, 1964, 7, N.4, 342-352.

(12) MALCHER, L., Die Federungs- und Dämpfungseigenschaften von Gleitlagern für Turbomaschinen - Experimentelle Untersuchungen von MGF- und Kippsegmentlagern, *Dissertation - Universität Karlsruhe*, 1975.

(13) PARSELL J.K, ALLAIRE P.E. and BARRETT L, Frequency Effects in Tilting Pad Journal Bearing Dynamic Coefficients, *ASLE Transactions*, 1983, 26, 2 222-227.

(14) RODKIEWICZ, Cz.M., KIM, K.W. and KENNEDY, J.S., On the Significance of the Inlet Pressure Build-Up in the Design of Tilting-Pad Bearings, *Trans.ASME Journal of Tribology*, 1990, 112, 17-22.

(15) ROUCH, K.E. Dynamic of Pivoted Pad Journal Bearings, Including Pad Translation and Rotation Effects, *ASLE Trans.*, 1983, 26 N.1, 102-109.

(16) SANTOS,I.F. and ULBRICH,H., Actively Tilting Pad Journal Bearings; *ISROMAC - The International Symposium on Transport Phenomena and Dynamics of Rotating Machinery* - Honolulu, Hawaii,USA; 5-8 Abril 1992, 423-431.

(17) SANTOS,I.F. und ULBRICH,H., Zur Anwendung von Regelungskonzepten für aktive Kippsegmentlager; *ZAMM - Zeitschrift für Angewandte Mathematik und Mechanik*, 1993, 4; No.73, 241-244.

(18) SANTOS,I.F.; Aktive Kippsegmentlagerung – Theorie und Experiment, *VDI – Fortschritt-Berichte*, 1993, 189, Reihe 11: Schwingungstechnik, VDI Verlag.

(19) SOMEYA, T., Journal Bearing Data Book, *Springer Verlag*, 1989.

(20) SPRINGER,H., Zur Berechnung hydrodynamischer Lager mit Hilfe von Tschebyscheff-Polynomen, *Forsch. Ing.-Wes.* 1978, 44, Nr.4, 126-134.

(21) SPRINGER,H., Dynamische Eigenschaften von Gleitlagern mit beweglichen Segmenten, *VDI - Berichte*, 1980, 381, 177-184.

(22) ULBRICH,H. and ALTHAUS,J., Actuator Design for Rotor Control, *12th Biennial ASME Conference on Vibration and Noise - Montreal/Canada*, 1989, 17-22.