Real-time vibration control of rigid rotors using controlled supply pressure hydrostatic squeeze film dampers

Boubekeur Rokia
LIM research Laboratory,
Department of Computer science, Ibn Khaldun's University of Tiaret, Algeria
Boubekeur_rokia@yahoo.fr
Bouzidane Ahmed
Research Laboratory of Industrial Technologies, Department of Mechanical Engineering, Ibn Khaldun's University of Tiaret, Algeria
Ahmed.bouzidane@gmail.com

Abstract—**the aim of this research is to study the Real-time control vibration of rigid rotor using controlled supply pressure hydrostatic squeeze film dampers. A linear model of the hydrostatic squeeze film damper has been developed in order to study the effect of the supply pressure on the dynamic behaviour of a rigid rotor. A new control system is proposed to reduce the transient response of the rotor using controlled supply pressure in order to control stiffness hydrostatic bearings. The results show that it is possible to effectively monitor the supply pressure and the dynamic characteristics of the fluid inside the hydrostatic squeeze film dampers for a better control of rigid rotor vibration and bearing transmitted forces.**

Keywords— *Real time control vibration, hydrostatic journal bearing, linear dynamic behaviour, Squeeze film dampers, rotor dynamic*

XVII. Introduction

Many researchers have studied the effects of hydrostatic squeeze film dampers in order to use it as a device for actively controlling rotors. Burrowz et al. (1983) [1] have studied the possibility of controlling the pressure in an SFD as a mean for controlling rotating machinery. Mu et al (1991) [2] suggested an active SFD with a movable conical damper ring. San Andres (1992) [3] developed an approximate solution for the pressure field and dynamic force coefficients in turbulent flow, in a symmetric hydrostatic bearing with its journal centred within the bearing clearance. The model includes the effects of recess volume liquid compressibility and introduces the model for a HJB with end seals. The results of its investigation show that HJBs with end seals have increased damping, better dynamic stability characteristics than conventional HJBs. Braun et al (1994) [4] and (1995) [5] performed an extensive analysis of the variation in lubricant viscosity with pressure and temperature and also analyzed the flow pattern in the recesses. Hathout et al (1997) [6] summarized the modeling and control of hybrid squeeze film dampers for active vibration control of rotors exhibiting multiple modes. Sawicki et al (1997) [7] studied the effects of dynamic eccentricity ratio on the dynamic characteristic of a four–pocket, oil-fed, orifice-compensation hydrostatic bearing including the hybrid effects of journal rotation. Adams and Zahloul, (1987) [8] have invistegated the vibration of rotors by controlling the pressure in hydrostatic four-pad squeeze film dampers (SFDs). They showed that stiffness is quite controllable with supply pressure while damping is nearly insensitive to supply pressure changes using a linear method. Using a similar system, Bouzidane et al (2007) [9] studied the effects of film thickness, recess pressure and geometric configuration on the equivalent stiffness and damping of a four-pad hydrostatic journal bearing. Their results reveals that because of its higher stiffness, good damping and zero cross-coupling terms, the four-pad hydrostatic journal bearing has better dynamic characteristics and stability than the hybrid journal bearing. They found that an optimal equivalent stiffness of a four-pad hydrostatic journal bearing is obtained for a pressure ratio at the centered position ßo close to 0.67. Bouzidane et al (2008 -2013) [10-14] investigated the effect of electrorheological fluid, pressure ratio, supply pressure, viscosity, and rotational speed on the unbalance response and transmitted force of a rigid/flexible rotor supported by a four-pad hydrostatic squeeze film damper.

The objective of this research is to adapt supply pressure technology to hydrostatic squeeze film damper in order to control the vibration of high speed rigid rotors. A linear model of the smart fluid hydrostatic squeeze film damper using controlled supply pressure has been developed. The transient amplitude–speed responses are calculated for a rigid rotor supported by a smart hydrostatic squeeze film damper. A new smart hydrostatic squeeze film damper is proposed to reduce the transient response of the rotor vibration by applying a control strategic on supply pressure, which results in modifying its stiffness hydrostatic bearings. The results show that it is possible to effectively monitor the supply pressure for a better control of rigid rotor vibrations and bearing transmitted forces.

XVIII. Mathematical Modeling

A cross section of a new three-pad hydrostatic squeeze film damper in the eccentric case is shown in Fig. 1. This figure shows a vertical rigid rotor supported by a hydrostatic squeeze film damper composed of three-pads. All pad-geometries are identical and equally spaced around the journal. The indices 1, 2 3 and 4 refer to the characteristics of the ith hydrostatic bearing flat pad, respectively. Each pad is fed by a capillary restrictor through a recess, which is supplied with an external pressure P_s.

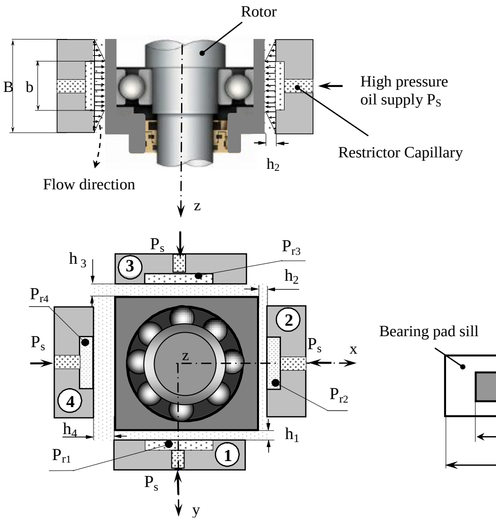


Fig.1: Hydrostatic journal bearing geometry and nomenclature

A Hydrostatic Squeeze Film Damper Characteristics

The calculation of the characteristics of the hydrostatic squeeze film damper can be obtained through the juxtaposition of three hydrostatic bearing flat pads (Fig.2). It is assumed that the fluid is incompressible and inertialess. The flow is laminar and the regime is steady state and isothermal.

B Reynolds Equation

For an incompressible, laminar, isoviscous, and inertialess fluid, the Reynolds equation may be written as [12]:

$$\frac{\partial}{\partial x_i}\left(\frac{\partial P_i(x_i, z_i, t)}{\partial x_i}\right) + \frac{\partial}{\partial z_i}\left(\frac{\partial P_i(x_i, z_i, t)}{\partial z_i}\right) = 12\frac{\mu}{h_i^3}\dot{h}_i \quad (1)$$

Note that the cavitations are not neglected when the thickness film increasing.

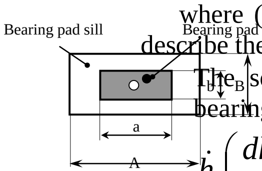
where:

- $0 \le x_i \le A$ and $0 \le z_i \le B$
- $P_i(x_i, z_i, t)$ is the hydrostatic pressure field of the i^{th} hydrostatic bearing pad;
- h_i is the film thickness of the i^{th} hydrostatic bearing pad ($h_i \neq f(x_i, z_i)$).
- (x_i, z_i, y_i) is the coordinate system used in the Reynolds equation, (i=1, 2 3, and 4).

- The film thickness h_i ($h_i = f(x, y) \neq f(x_i, z_i)$) is obtained as follows:

$$\begin{cases} h_1 = h_0 - y \\ h_2 = h_0 - x \\ h_3 = h_0 + y \\ h_4 = h_0 - x \end{cases} \quad (2)$$

where (x , y) is the coordinate system used to describe the rotor motion.



The squeeze velocity of the i^{th} hydrostatic bearing pad is determined as follows: $\dot{h}_i\left(\frac{dh_i}{dt}\right)$.

$$\begin{cases} \dot{h}_1 = -\dot{y};\ \dot{h}_3 = \dot{y} \\ \dot{h}_2 = -\dot{x};\ \dot{h}_4 = \dot{x} \end{cases} \quad (3)$$

C Recess Pressure

The recess pressure for each hydrostatic bearing pad is determined by resolving the following flow continuity equation:

$$Q_{ri} = Q_{oi} \quad (4)$$

where:

- $$Q_{ri} = \frac{\pi d_c^4}{128\,\mu\, l_c}\left(P_S - P_{ri}\right) \quad (5)$$

- $$Q_{oi} = Q_{vi} + Q_{oxi} + Q_{ozi} \quad (6)$$

- $$Q_{vi} = S_r \dot{h}_i \quad (7)$$

- $$Q_{oxi} = 2\int_0^B dz_i \int_0^{h_i} u_{xi}\,dy_i\,;$$
  $$Q_{ozi} = 2\int_0^A dx_i \int_0^{h_i} u_{zi}\,dy_i \quad (8)$$

- $$u_{xi} = \frac{1}{2\mu}\frac{\partial P_i}{\partial x_i}(y_i - h_i)y_i\,;$$
  $$u_{zi} = \frac{1}{2\mu}\frac{\partial P_i}{\partial z_i}(y_i - h_i)y_i \quad (9)$$

where d_c is the capillary diameter and l_c is its length; Q_{vi} represents the squeeze flow of the i[th] hydrostatic bearing pad; Q_{oxi} and Q_{ozi} are the oil flow of the i[th] hydrostatic bearing pad in the x_i, and z_i directions, respectively, Q_{ri} represents the flow through a capillary restrictor-type hydraulic resistance; u_{xi}, u_{yi} and u_{zi} are the flow velocities in the x_i, y_i and z_i directions, respectively.

III ROTOR DYNAMICS BEHAVIOR

In order to reduce the excessive high amplitudes of forced vibrations and the forces transmitted to the base, caused by rotor imbalance and passage through critical speeds, a study on the dynamic behavior of a rotor supported by hydrostatic squeeze film dampers based on linear methods was conducted.

The equations of the rotor motion can be expressed in Cartesian coordinates as follows (Fig.2):

$$\begin{cases} m\ddot{x} = F_x + me_x\,\omega^2\cos(\varphi) + me\ddot{\varphi}\sin(\varphi) \\ m\ddot{y} = F_y + me_y\,\omega^2\sin(\varphi) - me\ddot{\varphi}\cos(\varphi) \end{cases} \quad (10)$$

where m is the mass of the rotor; e is the eccentricity; ω is the excitation frequency and F_x and F_y are the hydrostatic forces in the x and y directions, respectively. $\ddot{\varphi}, \dot{\varphi}$ and φ represent the angular acceleration, angular velocity and angular displacement respectively, which are given by:

$$\begin{cases} \bullet\ \ddot{\varphi} = \text{constant} \\ \bullet\ \dot{\varphi} = \dot{\varphi}_0 + \ddot{\varphi}\,t \\ \bullet\ \varphi = \varphi_0 + \dot{\varphi}_0 t + \frac{1}{2}\ddot{\varphi}\,t^2 \end{cases} \quad (11)$$

A Forces Hydrostatics Bearings

- *Linear model*

The linear model is based on a small displacement and small speed hypothesis [9], and it is presented by linearizing the behaviour around an equilibrium state. The linear fluid film forces on the three-pad hydrostatic squeeze film damper in Cartesian coordinates (O_j, x, y) are obtained as follows:

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\underbrace{\begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}}_{[C_P]}\begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} - \underbrace{\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}}_{[K_P]}\begin{Bmatrix} x \\ y \end{Bmatrix} \quad (12)$$

where $[C_P]$ and $[K_P]$ represent the total hydrostatic bearing damping matrix and stiffness matrix, respectively, which are given as follows [10]:

$$[C_P] = \sum_{i=1}^{i=3} C_{Pi}\begin{bmatrix} \cos^2(\gamma_i) & -\cos(\gamma_i)\sin(\gamma_i) \\ -\cos(\gamma_i)\sin(\gamma_i) & \sin^2(\gamma_i) \end{bmatrix} \quad (13)$$

$$[K_P] = \sum_{i=1}^{i=3} K_{Pi}\begin{bmatrix} \cos^2(\gamma_i) & -\cos(\gamma_i)\sin(\gamma_i) \\ -\cos(\gamma_i)\sin(\gamma_i) & \sin^2(\gamma_i) \end{bmatrix} \quad (14)$$

with

$$K_{Pi} = -\left(\frac{\partial F_{Pi}}{\partial h_i}\right)_0 \quad ; \quad C_{Pi} = -\left(\frac{\partial F_{Pi}}{\partial \dot{h}_i}\right)_0 \quad (15)$$

where: K_{pi} and C_{pi} represent the stiffness and damping of the i[th] hydrostatic bearing pad, and F_{pi} is the hydrostatic force of the i[th] hydrostatic bearing pad. The partial derivatives are calculated numerically using the numerical differentiation method.

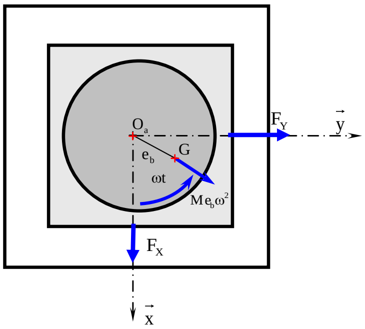


Fig2. Rigid rotor supported by four-pad HSFD

B. Solution procedure

The hydrostatic squeeze film damper effects on rotor dynamics are characterized by the hydrostatic forces generated by a pressure field. It must be noticed that these forces vary according to the position and velocity of the shaft center in the journal bearing. The calculation of the flow rate, vibratory responses and amplitude of transmitted forces due to a rotating unbalance vary depending on the rotational speed and are determined by resolving the equations of rotor motion (Eq. 10) using linear methods. The computed amplitudes are determined from direct numerical integration of the equations of motion using a step-by-step method. The hydrostatic forces are determined from equations (Eq. 12), which are based on the dynamic coefficients. The film thickness h_0 is determined by resolving the flow continuity equation from a given pressure ratio β_0 and using an iterative secant method, while the pressure is determined by resolving the flow continuity (Eq. [8]) by applying an iterative secant method.

The computation of the film thickness and recess pressure was performed using an iterative secant method after bounding the roots. The convergence tolerances of these computations were defined as follows:

- on the pressure: 0.01 $$\left(\left(\frac{P_{i,j}^{r}-P_{i,j}^{r-1}}{100\,P_{ri}}\right)_{\max}=0.01\right)$$
- on the film thickness: 10^{-6}
- on recess pressure: 10^{-6}

where $P_{i,j}^{r}$ represents the computed pressure at each mesh point (i,j) and r is the iteration number of the computation.

XIX. RESULTS AND DISCUSSION

As mentioned above, Figure 3 shows the transient response vibration amplitude versus time. It must be noticed from this figure that the envelope of displacements reveals the identification of critical speed (resonance). Not that the critical speed appears at 250 s. Fig. 4 presents the orbits around the critical speed. It can be seen from these results that the vibration amplitude come important around the critical speed.

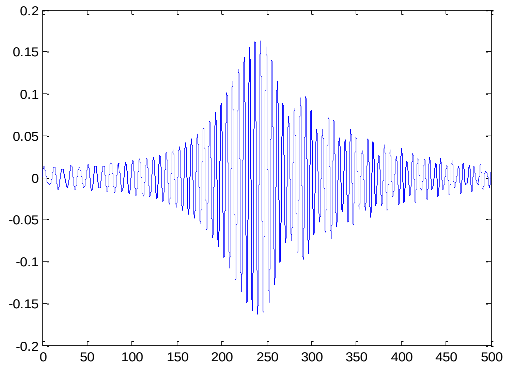


Fig.3 Transient response of the excitation versus time

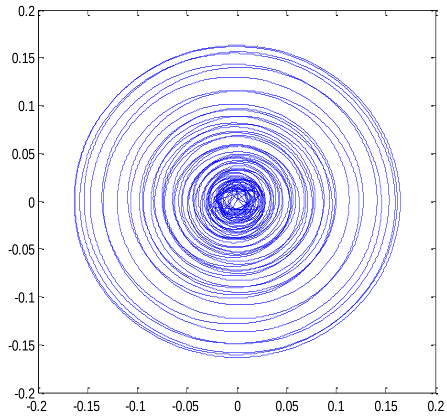


Fig.4 Orbits around the critical speed: X positions versus Y positions

Fig. 5 shows the effect of supply pressure on the responses due to rotational unbalance. This curve shows that when the pressure increases from 5 to 10 Bar, the amplitude and critical speed increase with an increase in supply pressure due to the increase in stiffness hydrostatic bearings. This leads an increase in rigidity zones since the critical increases with supply pressure.

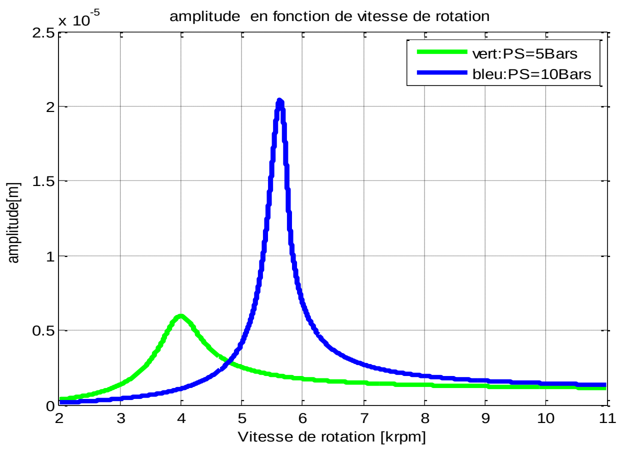


Fig.5 Amplitude versus rotational speed

A. Control method using controlled supply pressure

An of-on control system based on the control of supply pressure of a hydrostatic squeeze film damper can be developed to control rigid rotor vibration, reduce excessively high amplitudes of forced vibration and reduce the force transmitted to the bearing base. This control system functions by controlling supply pressure according to the operating speed of the rotor around the critical speed. An of-on control on supply pressure is proposed. Figure 6 demonstrates the use of the control system and the corresponding vibratory response. One can observe the effect of variations in the supply pressure on vibratory response versus speed and time in the HSFD. It can be seen from this results that it is possible to effectively monitor the supply pressure to reduce vibration amplitude when operating close to critical speed.

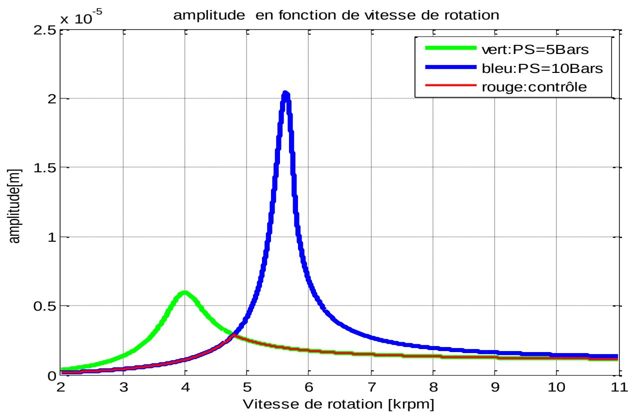


Fig.6 Control amplitude using controlled supply pressure

XX. CONCLUSIONS

Linear modeling of a hydrostatic squeeze film damper has been presented and applied in a control system to limit the vibration of high speed rigid rotor supported by a four-pad HSFD. The following conclusions can be obtained.

- Using controlled supply pressure of a hydrostatic squeeze film damper allows for achieving the objective to control the rotor vibration across the critical speeds. This effect is due to the fact that the stiffness hydrostatic bearing increases with an increase in supply pressure. Consequently the command law asked to control the supply pressure according to the operating speed of the rotor around the critical speed around the critical speed.
- The research shows that smart hydrostatic squeeze film dampers using controlled supply pressure has a promising potential future in vibration control of rigid/flexible rotor.

REFERENCES

[1] Burrows, C. R., Sahinkaya, M. N., and Turkaya, O.S., (1983),''An Adaptive Squeeze-Film Bearing,'' ASME Paper, **No. 83-Lub-23**, pp.1-7.

[2] Mu, C., Darling, J., and Burrows, C. R., (1991),''An appraisal of a Proposed Active Squeeze Film Damper,''. ASME J. Tribol., **113**, 4, pp.750-754.

[3] San Andres, L.A,(1992),'' Analysis of Hydrostatic Journal Bearings With End Seals,'' ASME J. Tribol., **114**, 7,pp. 802 – 811

[4] Braun, M. J., Zhou, Y. M., and Choy, F. K., (1994), ''Transient Flow Patterns and Pressures Characteristics in a Hydrostatic Pocket,'' ASME J. Tribol., **116**, pp.139 –146.

[5] Braun, M. J., Choy, F. K., and Zhu, N., (1995), ''Flow Patterns and Dynamic Characteristics of a Lightly Loaded Hydrostatic Pocket of Variable Aspect Ratio and Supply Jet Strength," Tribol. Trans., **38,** 1, pp.128–136.

[6] Hathout, J.P., El-Shafei, A. and Youssef, R.,(1997), ''Active Control of Multi-Mode- Rotor-Bearing Systems Using HSFDs,'' ASME J. Tribol., **119**, 7,pp. 49 – 56

[7] Sawicki, J.T., Capaldi, R.J. and Adams, M.L., (1997),''Experimental and Theoretical Rotordynamic Characteristics of a Hybrid Journal Bearing,'' ASME J. Tribol., **119**, pp.132-141.

[8] Adams, M.L., and Zahloul, H.,(September 1987),''Attenuation of Rotor Vibration Using Controlled-Pressure Hydrostatic Squeeze Film Dampers,'' Eleventh Biennial, ASME Vibrations Conference, Boston, MA, No.9, 99-108

[9] Bouzidane A. and Thomas M., (2007),''Equivalent Stiffness and Damping Investigation of a Hydrostatic Journal Bearing, ''Tribol. Trans., **50,** 2, pp.257-267.

[10] Bouzidane A., Thomas M. and Lakis A., (2008),''Non Linear Dynamic Behaviour of a Rigid Rotor Supported by Hydrostatic Squeeze Film Dampers,'' ASME J. Tribol. Trans., **130**, 4, pp. 041102-1-041102-9.

[11] Bouzidane, A., and Thomas, M., 2008. An Electro-Rheological Hydrostatic Journal Bearing for Controlling

Rotor Vibration, Computer & Structures, 86(3–5), 463–472.
[12] Bouzidane, A. and Thomas, M., 2012. Nonlinear Transient Response of a Flexible Shaft Controlled by Electro-Rheological Hydrostatic Squeeze Film Dampers, Proceedings of the 2nd Conference on Condition Monitoring of Machinery in Nonstationary Operations, T. Fakhfakh et al., eds., Hammamet, Tunisia, 33–40.
[13] Bouzidane A. and M. Thomas, 2013. Nonlinear Dynamic Behavior of a Flexible Shaft Supported by Smart Hydrostatic Squeeze Film Dampers, ASME Journal Tribology Transactions. 135 (5) 031701
[14] Bouzidane A. and Thomas M., 2013. Nonlinear Dynamic Analysis of a Rigid Rotor Supported by a Three-Pad Hydrostatic Squeeze Film Dampers, Tribology Transactions, 56: 717-727.