Abstract: - In this paper we propose a new lubricated journal bearing based on piezoelectric actuators. The actuators are used to generate traveling waves on the internal surface of the bearing. The radial displacements can influence the oil film thickness modifying the pressure distribution in the journal bearing gap. This link can be used to control the fluid film force in order to: 1) avoid instability of the rotor 2) to drive the rotor through of the friction force field generated on it.


1 Introduction
In rotating machinery the rotor is supported by bearings which can be classified into three categories:
1) Fluid bearings;
2) Roller bearings;
3) Magnetic bearings;
The basic structure of the foregoing components is indicated in the figures 1, 2 and 3, respectively.

The force field generated by the bearings transmits forces which can be cause of dynamic instability. These instability forms can cause malfunctions that can bring to crises of the whole system. In this paper we propose a new method to improve the stability characteristic of lubricated journal bearing. This method is based on the pressure control in the gap of the journal bearings. Furthermore the author are developing this method for other typologies of lubricated pairs.

2 Pressure Field Descriptions in Cylindrical Lubricated Journal Bearings
Pressure in lubricated journal bearings is obtained integrating the Reynolds equation.
\[
\frac{\partial}{\partial \vartheta} \left[ (1 + \varepsilon \cdot \cos \vartheta) \frac{\partial p}{\partial \vartheta} \right] + \nonumber \\
R^2 \frac{\partial}{\partial z} \left[ (1 + \varepsilon \cdot \cos \vartheta) \frac{\partial p}{\partial z} \right] = \nonumber \\
-6\mu \left( \frac{R}{C} \right)^2 \left[ (\omega - 2\varphi) \varepsilon \cdot \sin \vartheta - 2\dot{\varepsilon} \cdot \cos \vartheta \right] 
\] (1)

The variables used in (1) are indicated in Figg. 4 and 5 while the parameters \( \mu, \varepsilon, R, C, \) and \( p \) are dynamic viscosity, dimensionless eccentricity, bearing radius, radial clearance and pressure, respectively.

\[ \frac{\partial}{\partial \vartheta} \left[ (1 + \varepsilon \cdot \cos \vartheta) \frac{\partial p}{\partial \vartheta} \right] + \nonumber \\
R^2 \frac{\partial}{\partial z} \left[ (1 + \varepsilon \cdot \cos \vartheta) \frac{\partial p}{\partial z} \right] = \nonumber \\
-6\mu \left( \frac{R}{C} \right)^2 \left[ (\omega - 2\varphi) \varepsilon \cdot \sin \vartheta - 2\dot{\varepsilon} \cdot \cos \vartheta \right] \]

\[ \bar{p}(\vartheta, 0) = \bar{p}(\vartheta, L) = \bar{p}_{\text{am}} \]
\[ \bar{p}(0, z) = \bar{p}(\pi, 0) = \bar{p}_0 \]

Where \( \bar{p}_{\text{am}} \) is the atmospheric pressure and \( \bar{p}_0 \) is determined by the supply pressure to the bearing.

2.1 Short Bearing Model
For the short bearing model the first term of the equation (1) is omitted on the basis that it has a negligible effect on the flow compared to the other terms in the equation. Practically, this will occur when \( L/D \leq 0.25 \), where \( L \) is the axial length of the bearing and \( D = 2R \) is the journal diameter. The pressure distribution is given by:

\[ \bar{p}(\vartheta, z) = \bar{p}_{\text{am}} + \]
\[ \frac{3\mu(L^2 - 4z^2)(-2\dot{\varepsilon} \cdot \cos \vartheta + \varepsilon(\omega - 2\varphi)\sin \vartheta)}{4C^2(1 + \varepsilon \cos \vartheta)^3} \]

2.2 Approximate Finite Journal Bearing Model
In (4) is indicated the pressure distribution proposed by Warner [1] for finite journal bearing.

\[ \bar{p}_w(\vartheta, z) = \bar{p}_L(\vartheta) \gamma(\bar{z}, \lambda) \]

Where

\[ \bar{p}_L(\vartheta) = \bar{p}_w(\vartheta) - \bar{p}_c = 6\mu \left( \frac{R}{C} \right)^2 \left( \frac{\dot{\varepsilon}(1 - 2\varphi) \sin \vartheta \cdot \cos \vartheta}{2 + \varepsilon^2} \right) \times \]
\[ \left( \frac{1}{1 + \varepsilon \cos \vartheta} \right) + \left( \frac{1}{1 + \varepsilon \cos \vartheta} \right)^2 \]

and

\[ \gamma(\bar{z}, \lambda) = \left\{1 - \frac{\text{Cosh}(\bar{z}q\lambda)}{\text{Cosh}(q\lambda)}\right\} \]

are respectively the Sommerfeld solution for the cavitated infinite long bearing, and Warner’s flow correction factor, while \( q \) and \( \lambda \) indicated in the following relations:

\[ q^2 = \frac{\int_a^\alpha h^3 \left( \frac{dp_L}{d\theta} \right)^2 \ d\theta}{\int_a^\alpha h^3 \ p_L^2 \ d\theta}, \quad \lambda = \frac{L}{D} \]

Where the angle \( \alpha \) is calculated through of the following relations:

\[ \left[ (2 + \varepsilon^2) \varepsilon \cos \alpha - \varepsilon(1 - 2\varphi) \sin \alpha = 0 \right] \]
\[ \left[ (2 + \varepsilon^2) \varepsilon \sin \alpha + \varepsilon(1 - 2\varphi) \cos \alpha \geq 0 \right] \]

And \( \bar{P}_a \) is the pressure for \( \vartheta = 0 \).
3 Active Journal Bearing Based on Piezoelectric Actuators

When a vibration source is driven at one position on bearing at a frequency corresponding to the resonance of this bearing, only standing wave is excited, because the vibration propagates in two directions symmetrical to the vibration source and interferes with each other. When multiple vibration sources are installed on the bearing, displacement can be obtained by superposition all the waves. Using this principle, we can generate a propagating wave which is a rotation of the standing wave shape, even in a bearing. As shown in [1, 3] the pressure in traveling wave conditions for the short journal bearing model with the full boundary conditions is given by following relation:

\[
p_{c_{cc}}(\vartheta, \tau) = \frac{\psi a^{-1} \gamma \widehat{\delta}(h s_{1}) + 2\psi a s_{\tau}}{12h^3} + \frac{-\gamma \left( h \frac{\partial u_{1}}{\partial \vartheta} - u_{2} \frac{\partial h}{\partial \vartheta} \right) - 2v_{2}}{12h^3} \tag{9}
\]

With

\[
h(\vartheta) = 1 + \varepsilon \cos \vartheta + \beta_{R} \cos(n(\vartheta + \varphi) - \psi \tau)
\]
\[
s_{\tau} = \beta_{s} \sin(n(\vartheta + \varphi) - \psi \tau)
\]
\[
s_{R} = \beta_{s} \cos(n(\vartheta + \varphi) - \psi \tau)
\]
\[
v_{2} = \dot{\varepsilon} \cos(\vartheta) + \varepsilon(\dot{\varphi} - 1) \sin(\vartheta)
\]
\[
u_{2} = \gamma^{-1} + \dot{\varepsilon} \sin \vartheta - \varepsilon \dot{\varphi} \cos \vartheta \tag{10}
\]

The equation (9) can be simplified in the following equation:

\[
p_{c_{cc}}(\vartheta, \tau) = \frac{(1-2\dot{\varphi})e \sin \vartheta - 2\dot{\varepsilon} \cos \vartheta}{12(1+e \cos \vartheta + \beta_{s} \cos(n(\vartheta + \varphi) - \psi \tau)} - \frac{\beta_{R}(2\psi - n)\sin(n(\vartheta + \varphi) - \psi \tau)}{12(1+e \cos \vartheta + \beta_{s} \cos(n(\vartheta + \varphi) - \psi \tau))} \tag{11}
\]

The (11) for \( \beta_{R} = 0 \) becomes:

\[
p_{c_{cc}}(\vartheta, \tau) = \frac{(1-2\dot{\varphi})e \sin \vartheta - 2\dot{\varepsilon} \cos \vartheta}{12(1+e \cos \vartheta)} \tag{11}
\]

That is the well known expression of the pressure field in short journal bearing model obtained by Ocvirk.

The term \( \beta_{R} \) represent the dimensionless the radial displacement amplitude of the traveling wave induced by piezoelectric actuators.

4 Conclusion

In this paper we have proposed a new model of journal bearing in which the pressure field can be controlled by piezoelectric actuators placed opportunely on the external surface on the bearing. These actuators controlled in amplitude and phases excite traveling waves that influence the highness of the journal bearing gap. This influence is shown in the Figures. 6, 7 and 8. In Figures is indicated the pressure for traveling wave relative a vibration natural mode with three, five and seven lobes.

This research, proposed by the author in the year 2002, have been experimental verified in ambit of a research project supported by the MIUR (Italian Research Minister).
Fig. 8 - Pressure for traveling wave with seven lobes

References: