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Tilting pad gas bearing induced thermal bow- rotor instability (Morton effect)

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ABSTRACT

The Morton effect ME is a thermally induced, rotordynamic instability, which is frequently reported in overhung machines supported with oil-lubricated bearings. Synchronous journal whirling results in an uneven viscous shearing of the lubricant film and temperature variation around the journal circumference. This bends the shaft and under certain conditions causes increasing synchronous vibration. Previous research focused solely on oil-lubricated bearings. A transient, high-fidelity ME methodology is presented to expand the scope to tilting pad gas bearing (TPGB) systems. Three dimensional finite element models are established to predict the rotor and bearing temperature and dynamic responses in the time domain. Simulations indicate that the ME can occur in the TPGB systems and may be sensitive to imbalance and overhung mass.

1. Introduction

Gas bearings are widely used in micro-turbomachinery and some of their advantages over other bearing types include low cost, compactness and lightweight. Gas bearings do not require a complicated lubrication circulation system for oil supply or cooling [1,2] as compared with oil-lubricated bearings. In addition, gas bearings are less dependent on costly seals and more environmental friendly being especially suited for oil-free turbomachinery, such as aircraft air cycle machines, turbo-compressors, turbo-generators, etc.

Tilting pad bearings can significantly reduce bearing cross-coupled stiffness by allowing the pad tilting motions to adapt to external load. This aids in avoiding sub-synchronous vibration instability when the rotor is operated over the critical speed. However the pivot, with a typical rocker or spherical geometry may wear, and the resulting increased bearing clearance can dramatically change the bearing behavior, resulting in greater rotor vibration. Flexure pivot tilting pad bearings (FPTPBs) avoid pivot wear by integrating the bearing components into one single piece which is machined with electric discharge machining (EDM) [3–5]. The pad is connected to the bearing through a thin flexure web, and the pad tilting stiffness can be adjusted by changing the web thickness to ensure the suppression of cross coupled stiffness and sub-synchronous instability. The FPTPBs have demonstrated excellent performance in industrial applications and also provide a benefit of eliminating bearing stack-up tolerance.

The increase of the oil, bearing and rotor temperatures due to viscous shearing of the lubricant has always been a design concern. This action decreases lubricant viscosity and also causes bearing and rotor thermal expansions. Accurate prediction of the reduced lubricant viscosity and hot bearing clearance is critical in evaluating the bearing steady performance. This can be achieved by adopting the recent thermo-elasto-hydrodynamic (TEHD) analysis to account for the elastic/thermal deformation of the bearing and rotor on the basis of THD analysis [6–8]. Researchers have found that under certain conditions the Morton effect (ME), which is a thermally induced rotor instability problem, can occur in overhung machines supported by oil film bearings [9–13]. Significant temperature difference (ΔT) may develop across the journal circumference due to synchronous vibration of the journal and the resulting non-uniform viscous shearing of the lubricant. The temperature difference may bend the shaft and induce thermal imbalance. In some cases, the rotor vibration and circumferential ΔT can form a positive feedback, triggering large rotor vibration and ensuing trip of the machinery.

Compared with the oil FPTPBs, the gas FPTPBs generally experience less heat generation due to the lower gas viscosity. However, the advantage of lower viscosity may be lost by the fact that gas bearings typically operate with larger velocity gradient which results from smaller bearing clearances and higher rotating speeds. Thus frictional heating, and the resulting deformations in gas bearings should always be considered. Salehi [14] reported that the maximum bearing temperature could increase by 50°C in a gas foil bearing running at 30,000 rpm.

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Nomenclature

ME	Morton effect
Ω	Rotor spin speed in rad/s
R	Rotor outer radius
h	Film thickness
k	Heat conductivity
C_{pad}	Pad radial clearance
C_{brg}	Bearing radial clearance
h_{sft}	Shaft thermal expansion in radial direction
h_{pad}	Pad thermal expansion in radial direction
T_{pad}	Pad temperature
T_{air}	Air temperature
k_{pad}	Pad thermal conductivity
k_{air}	Air thermal conductively
$M_{rot}, C_{rot}, K_{rot}$	Rotor mass, damping and stiffness matrix
F_{mew2}	Mechanical imbalance force
F_{gyro}	Gyroscopic force
F_{brg}	Bearing force
F_{ext}	External force
$I_\delta, C_\delta, K_\delta, M_{tilt}$	Pad tilting mass, damping, stiffness coefficients and moments
I_p, C_p, K_p, F_p	Pad translation mass, damping, stiffness coefficients and force

Howard [15] found that the gas bearing stiffness decreased with temperature by as much as a factor of two from 25°C to 538°C, indicating that the bearing behavior was heavily temperature dependent. Radil [16] reported thermally induced gas bearing failures including seizure when the bearing was operated with small radial clearance.

Peng and Khonsari [17], Kim and San Andres [18] introduced the THD model to predict the steady state performance of gas foil bearings with improved accuracy relative to isoviscous models. For gas FPTPBs, Sim and Kim [19] presented a TEHD model accounting for the shaft thermal expansion and centrifugal growth. The rotor circumferential temperature distribution was assumed to be uniform so that the circumferential ΔT was always neglected in these previous analyses. This assumption is accurate for bearing steady state analyses such as dynamic coefficient prediction, however, it is not generally valid for ME instability studies, which require more detailed rotor thermal modeling. Prior literature only addressed the ME occurring in overhung machines supported by oil lubricated bearings. The occurrence of the ME in gas bearing machines is becoming of growing importance with increasing industrial applications of gas bearings, especially in turbochargers with overhung masses. The compressibility of the film heightens the level of modeling sophistication needed in prediction models for the ME, especially at the design stage. This paper aims to extend the high-fidelity ME prediction model from oil lubricated bearings to gas bearings, especially gas FPTPBs. The paper's main sections are: Section 2 introduces the theories related to the ME modeling, Section 3 discusses the prediction algorithms, Section 4 verifies the TEHD analysis with measured bearing performance by Ref. [20], Section 5 demonstrates a case study of ME instability in the gas FPTPB system, and Section 6 is the conclusion.

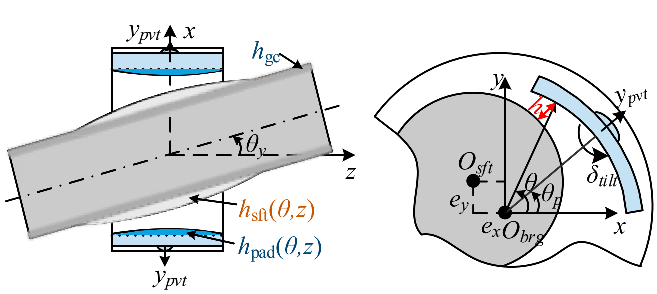

Fig. 1. Film thickness diagram.

2. High fidelity model of gas tilting pad journal bearing

2.1. Reynolds equation

For the ideal gas, the density and pressure are related by $\rho = P/\Re_g T$ with $\Re_g$ and T representing the gas constant and operating temperature, respectively. The Reynolds equation for the ideal gas film is listed in Eq. (1) for the purely hydrodynamic lubrication. The gas viscosity μ is temperature dependent and should be updated at each time step according to the film temperature distribution predicted by the energy equation. Note that x, y, z are the circumferential, radial and axial direction, respectively.

$$\nabla\cdot\left(\frac{-h^3P}{12\mu}\nabla P\right)+\frac{\Omega R}{2}\frac{\partial(Ph)}{\partial x}+\frac{\partial(Ph)}{\partial t}=0 \tag{1}$$

The ambient pressure P_a is imposed on both axial boundaries ($z = 0, L$) of each bearing pad. Distinct from the cavitation model of a fluid film bearing, sub-ambient pressure may exist in the diverging film thickness area of the gas bearing. Hybrid gas bearings operate with both hydrostatic and hydrodynamic lift, the former utilizing externally pressurized gas injected into the bearing through an orifice. The Reynolds equation for the hybrid gas bearing is listed below in Eq. (2) for the orifice area. Note that $\dot{m}_{OR}$ is the mass flow rate through the orifice, A is the effective orifice area πa^2 or πdh, and the mass flow rate $\dot{m}_{OR}$ is detailed in the appendix.

$$\nabla\cdot\left(\frac{-h^3P}{12\mu}\nabla P\right)+\frac{\Omega R}{2}\frac{\partial(Ph)}{\partial x}+\frac{\partial(Ph)}{\partial t}=\frac{\mathscr{R}_g T.\dot{m}_{OR}}{A} \tag{2}$$

2.2. Energy equation

The 3D energy equation takes into account the film temperature in all directions and thus is more precise than the 2D version, which neglects the temperature gradient in the axial direction and may over predict the thermal bow caused by the ME [21]. The 3D energy equation for the gas bearing is shown in Eq. (3), where ρ is the gas density, c_p is the gas specific heat, u, w are the velocity components in the circumferential and axial direction, and μ is the viscosity.

$$\rho c_p\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+w\frac{\partial T}{\partial z}\right)=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)+\left(u\frac{\partial P}{\partial x}+w\frac{\partial P}{\partial z}\right)+\Phi \tag{3}$$

where, $\Phi=\mu\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right]$

The velocity field u and w are acquired from the pressure gradient predicted by the Reynolds equation. Note that μ is temperature dependent and thus should be updated accordingly as the temperature field is updated from the energy equation solution. The thermo-viscosity relationship is modeled with a linear function as $\mu(T) = \mu_0 + \alpha T$, where T is temperature in °C, $\mu_0 = 1.835 \times 10^{-5} Pa{\cdot}s$, $\alpha = 4E-8$ [14]. Note that the gas viscosity will increase with temperature while the oil viscosity will decrease, and this also contributes to the necessity to perform the thermal analysis for gas bearings at high operating temperature.

2.3. Film thickness formula

During operation, both (1) the rotor centrifugal growth and (2) the thermal growth of the rotor & bearing will reduce the bearing clearance and thus should be included in the film thickness formula. The former is