Mechanical Systems and Signal Processing 107 (2018) 549–566

Contents lists available at ScienceDirect

Mechanical Systems and Signal Processing

journal homepage: www.elsevier.com/locate/ymssp

Nonlinear dynamic analysis of rigid rotor supported by gas foil bearings: Effects of gas film and foil structure on subsynchronous vibrations

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ARTICLE INFO

Article history:
Received 16 October 2017
Accepted 3 February 2018

Keywords:
Nonlinear rotordynamics
Gas foil bearing
Gas film
Foil structure
Subsynchronous vibrations

ABSTRACT

Highly nonlinear subsynchronous vibrations are the main causing factors of failure in gas foil bearing (GFB)-rotor systems. Thus, investigating the vibration generation mechanisms and the relationship between subsynchronous vibrations and GFBs is necessary to ensure the healthy operation of rotor systems. In this study, an integrated nonlinear dynamic model with the consideration of shaft motion, unsteady gas film, and deformations of foil structure is established to investigate the effect of gas film and foil structure on system subsynchronous response. One test rig of GFB-rotor system is developed for model comparison. High agreement is shown between the prediction and test data, especially in the frequency domain. The nonlinear dynamic response is analyzed using waterfall plots, operation deflection shapes, journal orbits, Poincaré maps, and fast Fourier transforms. The parameter studies reveal that subsynchronous vibrations are highly related to gas film and foil structure. Subsynchronous vibrations can be adjusted by parameters such as bump stiffness, nominal clearance, and static loads. Therefore, gas foil bearing parameters should be carefully adjusted by system manufacturers to achieve the best rotordynamic performance.

1. Introduction

Gas foil bearings (GFBs) are a type of self-acting gas bearing with flexible foil supporting structure. The pressured gas film separates the shaft surface from the top foil and removes heat with low viscous drag, as shown in Fig. 1. Owing to the flexibility of foil structure, GFBs are superior to conventional rigid gas bearings and present better stability and higher load capacity [1]. As advancements in coating materials and manufacturing techniques are achieved [2–5], GFBs show better endurance to misalignment, high and low temperature, and foreign particles than do traditional gas bearings. GFBs have been used to replace rolling element bearings on military and commercial aircraft air cycle machines since the 1980s [1,6]. GFBs have been widely used in turbomachinery such as micro turbo-expanders and turbojet engines [6–8], high speed oil-free blowers [9], highly efficient micro gas turbines [10], and many other applications. Given their advantages in high efficiency and system simplification, GFBs are the most promising candidates for oil-free micro turbomachinery and have elicited attention from research and industrial circles.

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https://doi.org/10.1016/j.ymssp.2018.02.005

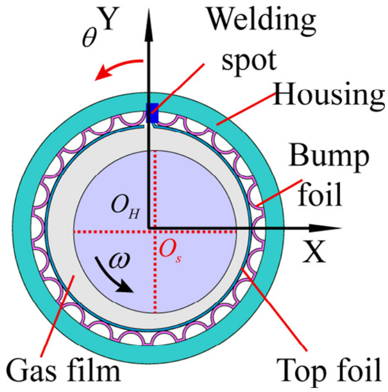

Fig. 1. Schematic view of GFB with labeled rotational direction.

Different analytical methods for predicting the dynamic force coefficients of GFBs have been developed since the first foil bearing topic paper was published in 1953 [11]. In GFBs, the gas film works in series with the foil structure to support the rotating shaft. As the most significant difference between GFBs and normal rigid housing gas bearings, the compliant bump foil structure elicits the attention of many researchers who model its behavior. Ku and Heshmat [12] presented a brief model for predicting bump foil structural stiffness and damping behavior. Peng and Carpino [13,14] developed an elastic bump foil model in which Coulomb friction damping effects were considered. Feng and Kaneko [15–17] derived a link-spring model to consider bump stiffness and Coulomb friction force between contact foil surfaces. Kim and San Andrés [18–20] discussed the effects of gas film on dynamic force coefficients of GFBs. Parameter analysis of preload, side feed pressurization, and rigid gas film are performed in their studies. However, the coefficients of dynamic bearing force are perturbation results of the steady static position of equilibrium. The limitations are clear that the method is only suitable when the rotor displacements and foil deflections are sufficiently small around the equilibrium position.

Many experimental tests have shown that vibration with large amplitudes at subsynchronous frequencies is common in GFB-rotor systems. Heshmat [4] achieved an advanced-design GFB-rotor system that reached up to 132,000 rpm (2200 Hz). The high subsynchronous vibrations at the turbine end show an amplitude of 23 μm at 350 Hz whereas the synchronous amplitudes are significantly small. Kim et al. [21] and Lee et al. [22] showed obvious subsynchronous limit circle motions in their GFB oil-free turbocharger experiments. Kim only provided a simple relationship between the subsynchronous vibration and natural frequencies of GFB-rotor system. Some nonlinear models have been established to investigate the dynamic performance of GFB-rotor systems. Compared with linear bearing model, Bou-Said et al. [23] found that the nonlinear analysis method can perform more accurately when the rotor eccentricity reaches a high value. Kim [24] parametrically studied different foil stiffness distributions on the dynamic performance of GFBs. Linear and nonlinear methods are compared in this reference, and the results show that the estimated onset speeds of subsynchronous motions differ. Le Lez et al. [25] derived nonlinear numerical predictions of GFB-rotor system stability limit and amplitude jump because of imbalance magnitudes. Bhore and Darpe [26] studied the nonlinear bifurcation phenomena of flexible rotor support on GFBs with parameters of rotor speed, imbalance mass, and so on. In these nonlinear models, the complex vibrations of GFB-rotor system is simulated. However, the relationship between the subsynchronous motions and gas film working in series with foil structure are not depicted clearly.

The subsynchronous vibrations of rotor-bearing system are commonly divided into two types, i.e., whirl and whip vibrations, depending on their distinguishing speed characteristics. Bently and Hatch [27] and Muszynska [28,29] discussed the whirl and whip phenomena in oil bearing rotor system by using the dynamic stiffness method. The fluid-induced subsynchronous vibration is the excitation of one natural mode related to system dynamic stiffness. The whirl vibration frequencies are determined by the weaker stiffness of oil film compared with shaft in the system dynamic model. As rotor speed increases, oil film stiffness increases but remains weaker than shaft stiffness. Thus, the frequencies of whirl vibration track the rotational speeds. However, when rotational speed is extremely high, shaft stiffness is weaker than oil film stiffness in series. Constant shaft bending stiffness makes the frequencies of whip vibration almost unchanged. In Ref. [30], they are called oil whirl and shaft whip to indicate the subsynchronous motion source. In the said work, the theory is adjusted and adopted to the analysis of GFB-rotor dynamic performance. The main difference is that the whip vibration is related to the foil structure in GFB-rotor system. However, the whip motion is the excitation of bending shaft mode in oil bearing lubricated rotor system. Furthermore, the whirl and whip vibrations are related to the stiffness relationship between gas film and foil structure in this rigid rotor system.

In the current study, an integrated nonlinear dynamic model of rigid rotor supported on GFBs is established to investigate the effect of GFBs on system subsynchronous response. One simple theory of gas film force in a series that possesses foil structure force is used to explain the complex whirl and whip vibrations. Transient bearing force is calculated with the combination of unsteady Reynolds equation and a simplified bump foil model. The performance of the GFB-rotor system is sim-