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Effects of eccentric phase difference between two discs on oil-film instability in a rotor–bearing system

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ABSTRACT

The operating speed of the rotating machinery often exceeds the second order or even higher order critical speeds to pursue higher efficiency. Thus, the second mode whirl/whip can appear when the operating speeds approach or exceed twice the second order critical speed according to the reference A. Muszynska (2005) [1]. In this study, we investigate how the eccentric phase difference between two discs influence the oil-film instability (the first/second mode whirl/whip) in a rotor–bearing system. Firstly, a lumped mass model with 20 degrees of freedom (DOFs) of a rotor system with two discs considering the gyroscopic effect is developed. The graphite bearing and sliding bearing are simulated by a spring-damping model and a nonlinear oil-film force model based on the assumption of short bearings, respectively. The research focuses on the effect of eccentric phase differences of two discs on the onset of instability and nonlinear responses of the rotor–bearing system by using the bifurcation diagrams, spectrum cascades, vibration waveforms, orbits and Poincaré maps. The results show that the instability speed increases when the eccentric phase difference becomes larger and it increases almost linearly when the eccentric phase difference is greater than 20°. Moreover, complicated combination frequency components related to the first and second mode whirl/whip frequencies are also excited and the transfer of the self-excited vibration energy between the first and second mode whips can be observed under a larger eccentric phase difference. The study may contribute to a further understanding of the oil-film instability of such a rotor–bearing system.

1. Introduction

Modern rotating machines, such as turbines, compressors and generators, are designed for high speed, high flexibility and high efficiency. In order to avoid unstable vibrations at higher operating speeds, more and more attention has been paid to self-excited vibration. The occurrence of rotor lateral self-excited vibration known as “whirl”, “whip” or simply “instability” arises from the presence of nonlinear fluid forces as the threshold speed is exceeded by the rotating speed, which is the phenomenon of dynamic instability resulted from the interaction between the rotor and the sliding bearing. The instability is typically subsynchronous because it would induce excessive vibration at the first or second mode whirl/whip frequency, and it would contribute to unstable operation of the system, high-level vibration, eventual rubbing between rotor and stator, and potential damage of the rotating machinery [1,2].

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A great number of theoretical researches have been carried out in order to reveal the effect of the self-excited vibration due to seals or bearings. In earlier studies, the rotor–bearing system was usually linearized about a stationary equilibrium position and the stability (threshold speed of instability) of the linearized system was analyzed by a classical eigenvalue analysis, such as logarithmic decrement, etc [3–5]. The parameters of the rotor–bearing system and operating conditions (e.g. rotating speed) were usually chosen in such a way that the rotor operated below the threshold speed of instability. However, the observed phenomena indicate that bearing fluid force has strong nonlinearity, and the linear model will fail to analyze the nonlinear dynamic behavior of the rotor bearing system when the perturbed motion of journal is no longer small.

Many literatures reported the typical nonlinear vibration phenomena caused by oil-film instability, such as subharmonic, superharmonic vibration, jump etc. Ehrich [6] noticed and analyzed the appearance of a pseudo-critical peak in the response amplitude for a rotating machine operating at twice its fundamental critical rotational speed. He referred to the phenomenon as subharmonic vibration. Bently [7] did systematic experimental work on the second and third order responses of an experimental rotor, observing pseudo-critical peaks at respectively twice and three times the fundamental critical speed. Childs [8] analyzed the problem, expanding on Bently's model of the problem, and referred to the problem as fractional frequency rotor motion. Botman [9] observed non-synchronous vibrations at speeds above twice the system critical speed in a high-speed rigid rotor-damper system. Ehrich [10] stated that rotor dynamic instabilities and self-excited vibrations generally took the form of lateral flexural vibrations at the rotor natural or critical frequency, and most often below the running speed. Nikolajsen et al. [11] observed non-synchronous vibrations in a flexible, symmetric rotor on two identical plain journal bearings supported by centralized squeeze film dampers. Zhang et al. [12] presented a mathematical model and a computational methodology to simulate the complicated flow behaviors of the journal microbearing in the slip regime, and their investigation showed that the rotor motion was stable with half-frequency whirling when the system located in the lower stability region, and the rotor had high-frequency whirling when the system located in the upper stability region.

In order to better simulate the nonlinearity of sliding bearing, many researches developed some nonlinear oil-film force models. On journal bearing impedance descriptions, Childs et al. [13] proposed a nonlinear hydrodynamic force model, which could be used to analyze stability but failed to simulate fluid-induced instability. Muszynska et al. [14] presented a simple model of nonlinear fluid dynamic forces generated in bearings based on the results of a series of experiments. A parameter called the fluid average circumferential velocity ratio was used to describe the characteristic of the fluid motion as a whole. The fluid film radial stiffness, damping, and inertia effects were described by nonlinear functions of the rotor eccentricity ratio inside the bearing. The model could be used to analyze anti-swirl to prevent rotor instability correctly and effectively. Capone [15,16] developed numerical simulations to study cylindrical hydrodynamic bearings, obtaining the orbits of the shaft in the bearings based on a nonlinear hydrodynamic force model. Zhang et al. [17] proposed an effective model for unsteady oil-film force to express time-varying boundaries of the film that whirled rapidly around the journal center.

Based on Capone model, Adiletta et al. [18] analyzed the possible chaotic motions stemming from the nonlinear response of the bearings; Jing et al. [19,20] studied the nonlinear dynamic behavior of bearing considering the oil whip phenomenon; de Castro et al. [21] researched the system instability threshold influenced by the amount of unbalance, rotor arrangement form and bearing parameters; Ding et al. [22] analyzed the non-stationary dynamic responses of the system during speed-up with a constant angular acceleration for a multi-bearing rotor; Cheng et al. [23] investigated nonlinear dynamic behaviors of a rotor–bearing–seal coupled system. Rao et al. [24] presented an analytical approach based on the long bearing model and the short bearing model for nonlinear transient analysis. Chen et al. [25] studied chaotic behavior of a flexible rotor supported by oil-film bearings with nonlinear suspension by using the long bearing approximation [26]. Schweizer et al. [27] studied nonlinear oscillations of an automotive turbocharger rotor supported by full-floating ring bearings, and explained occurring nonlinear effects: self-excited vibrations, oil whirl/whip phenomena, subharmonics, superharmonics, combination frequencies and jump phenomena.

In order to verify and revise the theory model, a lot of experiment works have been performed by using different forms of test rigs. Newkirk et al. [28] reported experimental cases in which the rotating speed reached five or six times the first critical speed before the instability occurred. Pinkus et al. [29] reported cases in which whipping disappeared and resumed again, and cases of stable and unstable states separated by regions of transient whip. For identifying the nonlinear aspects of the dynamics of a rigid unbalanced rotor on lubricated journal bearings, Adiletta et al. [30] performed an experiment to confirm the theoretical results. Through experiment, Fan et al. [31] investigated the phenomena in start-up vibration responses and presented a method for predicting instabilities of rotor systems in the coexistence of oil whip and dry whip. By a test rig, EI-Shafei et al. [32] studied the onset of instability on a flexible rotor mounted on two plain cylindrical journal bearings and analyzed the influence of rotor imbalance, oil pressure and misalignment to the initial instability speed.

It should be noted that in all the above researches, only the first mode whirl/whip is concerned. In fact, the operating speed of the rotating machinery often exceeds the second order or even higher order critical speeds to pursue higher efficiency. Thus, the second mode whirl/whip can appear when the operating speeds approach or exceed twice the second order critical speed according to the literatures [1,32]. The researches on the second mode whip phenomenon, the relation between the first and second mode whip and the effect of unbalance on the whip forms are not sufficient. The presented study in this paper is designed to investigate the effects of eccentric phase differences of two discs on the first and the second mode whips. A nonlinear oil-film force under short bearing assumption [15,16] is adopted. Numerical integrations