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Parametric studies on modified configurations of ball-type passive balancers for improved transient and steadystate responses

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ABSTRACT

A passive balancing device is a bearing with a set of masses that are free to move about a shaft axis of rotation. Beyond the first critical speed of the shaft, the masses assume positions that reduce vibrations due to imbalance. The conventional design of passive balancers is a dual-ball bearing. This type of balancer only performs its function at supercritical speeds and when the ball/track contact is nearly frictionless. In this work, additional experimental verification of a previously formulated mathematical model is conducted using published experimental data. The model was then used to investigate passive balancing performance numerically. Conventional and non-conventional bearing configurations were tested with consideration of rolling resistance and ball collisions. Results suggest that when rolling resistance is considered, a 1-track bearing configuration with any number of balls yields better performance than multi-track bearings. The 1-track configuration improved performance by 57% when compared to a 3-track configuration at supercritical steadystate. It is also shown that a multi-partition balancer improves performance significantly during shaft speed up – 69% improvement compared to a balancer without partitions. With configuration adjustments, the bearing remains entirely passive while vibration suppression performance is improved.

1. Introduction

1.1. Background

Rotary machines are a foundational component of industrial and military technology. They are used in a wide array of applications ranging from optical disk drives to wind turbine generators. As oscillatory systems, they are susceptible to unwanted vibrations. Reducing or completely eliminating those vibrations alleviates their adverse effects which vary by application. In aircraft engines, structural vibrations affect fatigue, performance, and structural integrity of system [1]. Undesirable vibrations demand additional maintenance, reducing operational time. In wind turbines, vibrations degrade efficiency and decrease system lifespan [2].

In rotorcraft and propelled aircraft, reducing vibrations due to rotor imbalances is critical in vehicle design. The AH-64 tail rotor is one example of systems requiring complex balancing procedures due to the asymmetrical blade configuration [3]. This introduces additional cost to the design, operation, and maintenance of the aircrafts. In addition to cost increases, vibration in

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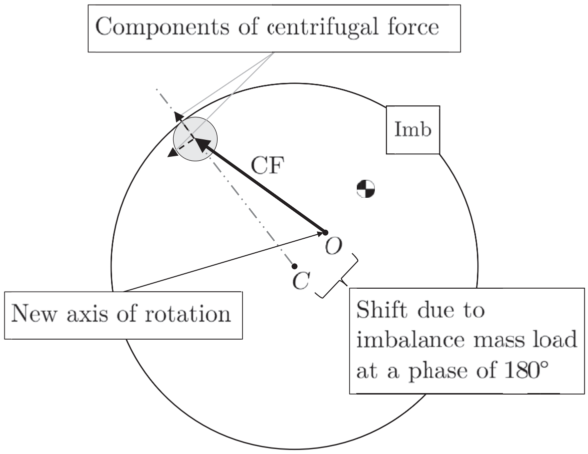

Fig. 1. Balancer system response with 180° phase.

rotorcraft affects the passengers and on-board systems. In medical evacuation helicopters, excessive vibrations cause medical instruments to malfunction [4]. In military rotorcraft, excessive vibrations degrade performance of on-board weapon systems due to difficulty in using sights [5].

Passive balancing devices have been studied to suppress vibrations of imbalanced shafts. The ball-type passive balancer was invented in 1932 by Thearle [6]. It was recognized that such device is only operable at supercritical shaft speeds. In 1967, Inoue et al. [7] studied the dynamics of the balancer suggested by Thearle and generated equations of motion for an ideal system consisting of a balancer on spring/dashpot supports. The stability of a linearized system was studied analytically. It was shown that ideal steadystate balancer mass positions can be predicted analytically.

In 1999, Chung and Ro [8] generated nonlinear equations of a rotating rigid track with two balancing balls embedded. The perturbation method was used to derive linear variational equations. In 2009, DeSmidt [9] provided a theoretical model that can predict and numerically simulate vibrations along the length of a flexible shaft with a mounted passive balancing system. In 2015, Haidar and Palacios [10,11] provided an experimentally verified model of a flexible shaft system with a single passive balancer.

1.2. The passive balancing method

When a rotary system passes its first flexural resonance frequency, it is said to be operating at supercritical speeds. In this region, the phase of the system response with respect to its centrifugal loads becomes 180°. The supercritical system response is illustrated in Fig. 1. An eccentric imbalance mass (*Imb*) exists on a rotor disk which induces a centrifugal load (CF) on the rotating shaft.

The forces acting on a free balancing mass as a result are shown. *C* is the geometric center of the shaft and disk, and *O* is the new axis of rotation. At this phase, the eccentricity grows in a direction 180° from the centrifugal load vector of the imbalance mass. The tangential component of the balancing mass centrifugal load causes the mass to move away from the imbalance mass restoring the rotation axis to the shaft geometric axis and suppressing vibration.

1.3. Performance studies

While passive balancing bearings lead to vibration suppression in ideal conditions, they still possess flaws preventing operation in all conditions. In 1988, Tadeusz [12] recognized that balancer mass resistive forces including friction could result in residual imbalance and therefore residual vibrations in the system. In 2016, Haidar and Palacios [11] experimentally demonstrated the detrimental effects of rolling resistance on the performance of a ball-type passive balancer at steady supercritical shaft speeds.

Passive balancers are also known to become unstable near the shaft critical speed. In 2008, Lu and Hung analytically examined the stability of a three-ball balancer [13]. It was demonstrated that a three-ball configuration resulted in a larger stable region of operation. In 2012, Ishida et al. [14,15] experimentally tested a multi-track design for ball-type passive balancers which improved vibration suppression at resonance and reduced the negative influence of track/ball friction.

There is a need to comprehensively determine the effect of bearing configuration on passive balancing performance. Thus far, the primary focus of balancer bearing performance studies has been on steadystate supercritical operation. This is the ideal operating condition of such devices. However, in many practical application, the transient response of the balancer bearing while transitioning through the critical speed is just as crucial since the bearing can and does severely increase shaft displacement