Computational study of friction models' parameters in a heavily loaded ball and socket coupling

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EXTENDED ABSTRACT

1 Introduction

This paper investigates the effect of various friction models' parameters used in a heavily loaded ball and socket (BS) coupling. The case study is performed on the BS coupling used in a tilting pad journal bearing (TPJB), which can be heavily loaded by both static (rotor weight) and dynamic (rotor imbalance) loads. The TPJB contains movable pads whose motion is enabled by using flexible supports, most often rocker pivots or spherical pivots (which have the BS construction). The main advantage of TPJBs is their high stability which is given by their ability to adapt bearing geometry to various operating conditions. As shown by several experiments, e.g. [1], the friction in the BS coupling has a negative effect on a TPJB dynamic performance and should be included in the computational models. Two types of friction models are studied in this work – Bengisu-Akay model [2] as a representative of static friction models, and the amended LuGre model [3] as a representative of dynamic friction models. The mathematical model of the whole TPJB system with BS couplings is formulated, and the effect of friction models' parameters is investigated.

2 Mathematical model and simulation results

The Jeffcott rotor of mass $2m_J$ with two lateral degrees of freedom y_J and z_J is considered in the mathematical model. The rotor is supported by two TPJBs with four pads. Each pad has two degrees of freedom - tilting angle δ and radial displacement η. The scheme of all applied forces on the rotor and i-th pad is shown in Fig. 1. The whole model contains five rigid bodies, from which the rotor interacts with pads through hydrodynamic forces. The resultant equations of motion for the rotor and i-th pad are

$$m_J\,\ddot{y}_J = -m_J g + \Delta m E\,\omega^2 \cos(\omega t) + \sum_{i=1}^{N} F_{hd,i}^{y}, \tag{1}$$

$$m_J\,\ddot{z}_J = \Delta m E\,\omega^2 \sin(\omega t) + \sum_{i=1}^{N} F_{hd,i}^{z}, \tag{2}$$

$$I_{P,i}\,\ddot{\delta}_i + m_{s,i} C_{\xi,i}\,\ddot{\eta}_i = -F_{hd,i}^{y'}(R+\kappa_i) - M_{f,i} - m_{s,i}\,g\left[C_{\xi,i}\sin(\vartheta_i-\delta_i) - C_{\eta,i}\cos(\vartheta_i-\delta_i)\right], \tag{3}$$

$$m_{s,i}\,\ddot{\eta}_i + m_{s,i} C_{\xi,i}\,\ddot{\delta}_i - m_{s,i} C_{\eta,i}\,\dot{\delta}_i^2 = -F_{hd,i}^{z'} - F_{bs,i} - m_{s,i}\,g\sin(\vartheta_i-\delta_i). \tag{4}$$

For more details on geometry, forces and used parameters, see [4]. In general, the equations of motion contains gravitational force, unbalance force, inertial forces, hydrodynamic forces, radial reaction in the BS coupling and friction moment.

Since the ball in the BS coupling has a slightly smaller diameter than the socket, the contact between the ball and the socket is not local compared to the BS parts dimensions. Therefore, normal contact forces in the BS coupling are evaluated using the conformal contact theory proposed by Fang et al. [5]. The scheme of the BS coupling is shown in Fig. 1. The ball can move inside the socket and friction appears, which generates a friction moment acting on the pads. The Bengisu-Akay friction model was used at first. To better catch various friction phenomena in the model, the amended LuGre friction was also investigated.

Both studied friction models use similarly described Stribeck curves, using analogical shaping parameters. The usual value of the critical velocity v_c for the Bengisu-Akay model in multibody system dynamic problems is 0.001 m·s^{-1} [2]. This parameter defines an area around zero slip velocity, where the friction force is smoothened to prevent discontinuity. Based on the problem analysis, it was found that the usual value is still too high to properly catch the friction force generated around the zero BS relative velocities. Thus, this parameter is set to $v_c = 10^{-5}$ m·s^{-1}, which is closer to a non-smooth Coulomb model. A significant change in the pad tilting motion is documented in Fig. 2 for rotor speed 6500 rpm. For $v_c = 0.001$ m·s^{-1}, only sliding friction occurs, while with lower values, the friction sticking is more apparent.

The bristle average stiffness from the amended LuGre model depends on the normal contact force. In many works related to friction modelling, such as [2], the bristle stiffness is considered as 10^5 N·m^{-1}. Unfortunately, this value can be valid only for lightly loaded contacts and there is a lack of references regarding the LuGre friction model behaviour in case of very high loads. In the presented case, the BS coupling is loaded by force, whose amplitude can reach tens of thousands of newtons. The significant influence of the bristle stiffness parameter σ_0^A on the pad tilting motion is shown in Fig. 2. For lower σ_0^A almost no sticking occurs, while with higher σ_0^A the sticking phenomena together with presliding behaviour appear.

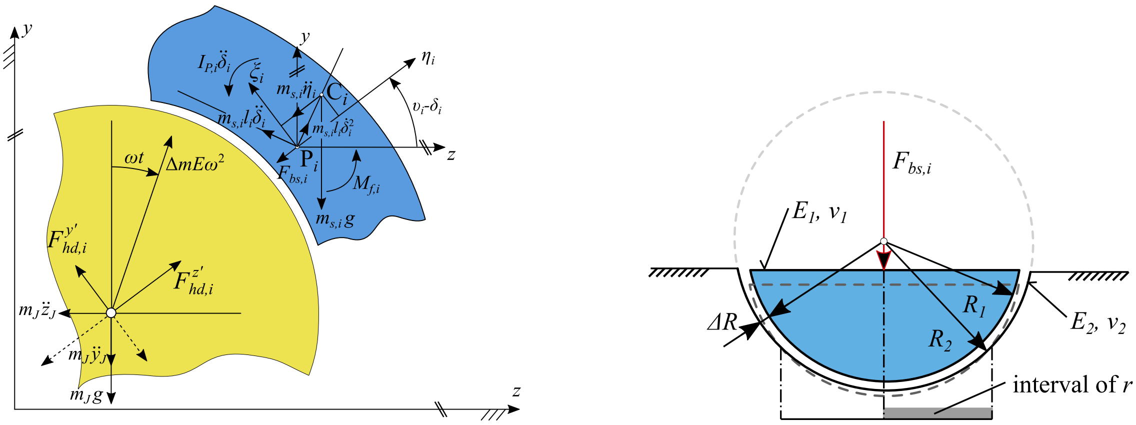

Figure 1: Mathematical model of the TPJB and the BS coupling: acting forces on the rotor and i-th pad (left), and the scheme of the BS coupling conformal contact (right).

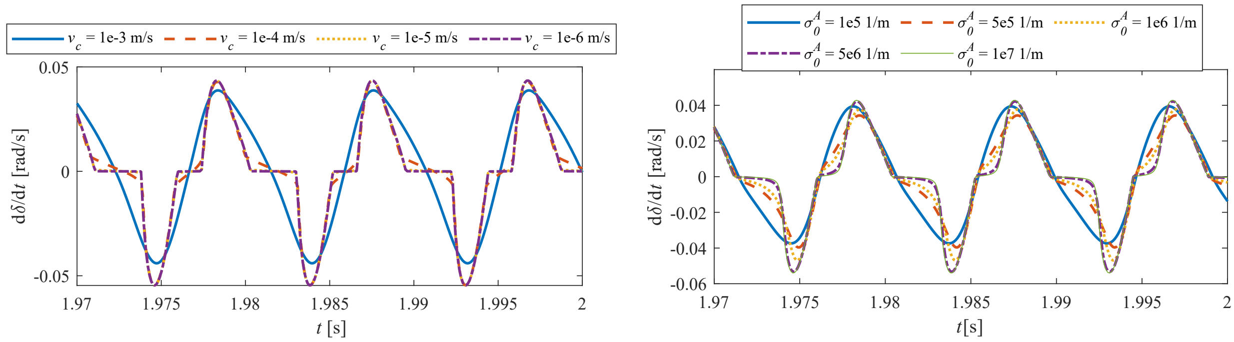

Figure 2: Influence of critical velocity v_c of the Bengisu-Akay model (left) and influence of LuGre bristle stiffness parameter (right) on pad tilting velocity.

3 Conclusions

Since the tested friction models lead to qualitatively different results in the pad tilting motion in case of steady-state TPJB behaviour, as it is shown in Fig. 2, it is appropriate to perform computational studies regarding the effects of various friction models' parameters. It has been shown that the generic and usually used values of some parameters can give misleading results. Therefore, they should be chosen carefully concerning the problem being solved. The work demonstrates that precise BS friction modelling can play an essential role in the computational modelling of whole rotor systems. The detailed models allow a more profound analysis of various rotor nonlinear phenomena, such as fretting wear in contact areas, pivot durability and clearance analysis.

Acknowledgments

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