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Friction modeling, identification, and compensation based on friction hysteresis and Dahl resonance

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ABSTRACT

This paper studies effects of friction on control systems and utilizes the observed frictional behavior to develop a parameter identification method for a friction model using frequency domain measurements. Friction exists in a wide range of drive systems due to physical contacts in bearing elements, transmissions, or motion guides. Friction in a control system can deteriorate performance by causing limit cycles or stick–slip, as well as larger tracking errors. Friction compensation can help to reduce following errors, but requires physical understanding and a reliable model of friction in both the gross- and the pre-sliding regimes. In this paper, we adopt the Generalized Maxwell-Slip (GMS) model and develop a frequency-domain method to identify the model parameters based on the frictional resonances, which occur due to the elastic behavior of friction at small amplitudes. With the experimentally identified parameters, the friction model is utilized to compensate the friction effects in a motion control system. The resulting system performance of a compensated and uncompensated control system is then compared in both the frequency and time domains to demonstrate the Dahl resonance identification method for a GMS model.

1. Introduction

Friction behavior can be divided into two regimes: gross-sliding and pre-sliding [1]. In the gross-sliding or simply sliding regime, friction is a function of the relative velocity of two sliding objects and is well established as the Stribeck curve containing the elements of static, Coulomb, and viscous friction [2]. Friction in the pre-sliding regime, on the other hand, is not directly a function of velocity but depends upon the history of displacement of objects in contact, acting as a nonlinear hysteretic spring [3].

In this paper, we experimentally explore friction behavior in the frequency domain, and use that perspective to identify parameters of the Generalized Maxwell-Slip (GMS) friction model [4]. A frequency domain identification was introduced by Hensen et al. [5] for an early dynamic friction model, called the LuGre model [6]. The identification approach developed in this paper extends the frequency-domain view to extract the multiple varying stiffnesses of the pre-sliding friction in the GMS model based on the frictional resonance, which is a frequency-domain reflection of the hysteretic nonlinear behavior of the pre-sliding friction. We experimentally validate the fidelity of our identification method in both the frequency and the time domains by comparing system performance with and without a model-based friction compensator.

Prior to an in-depth discussion of empirical friction models, friction model parameter identification, and friction compensation, we present a case study using a servomotor to help physically understand friction behaviors.

2. Case study: Motor spin free response

2.1. Experimental setup

Fig. 1 shows a friction experimental testbed including a servomotor and a high-resolution encoder. We use a slotless and brushless servomotor, BMS 60, by Aerotech so as to minimize unwanted nonlinear effects including brush friction and cogging. A high performance rotary encoder implemented at the back of the motor keeps track of the position displacement with an interpolated resolution of 10^6 counts per revolution. The position information goes through an FPGA encoder counter into a controller which is implemented on a National Instruments (NI) PXI 8110 real-time control-

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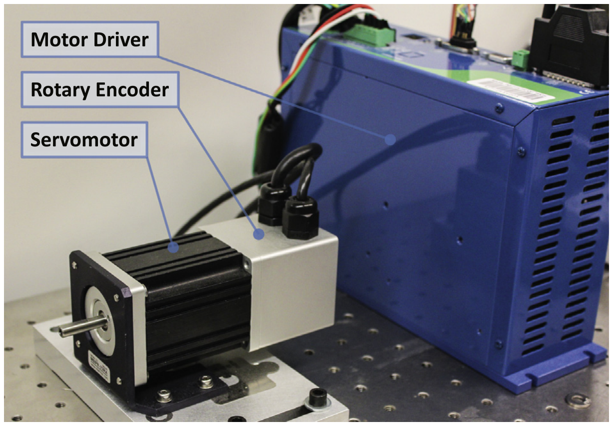

Fig. 1. Experimental setup including an Aerotech servomotor (BMS 60), encoder, and motor driver (Soloist Hpe 10).

ler connecting to the host computer via ethernet. We use this experimental setup for the case study, and for the rest of the experiments in this paper as well.

2.2. Free response of motor in rotation

Two simple experiments demonstrate the effects of bearing friction. In the first test, we manually spin the motor shaft by hand with the motor coils disconnected (open circuit) while reading motor position and velocity. The resultant time responses are shown in Fig. 2(a) and (b). The position response is measured at a sampling rate of 2 kHz, and the velocity is calculated in the servo controller by the backward difference of the measured position [7]. For the second test, the motor is excited by a current pulse with the amplitude of 0.05 A and a duration of 70 ms. Fig. 2(c) and (d) show the resultant responses of the motor driven open-loop test. The comparison between the manual and the open-loop tests indicates that the resulting oscillatory behavior is not caused by any electrical components of the system, but by friction from the direct physical contact of the mechanical parts, which is the bearing in this setup.

The spins illustrated in Fig. 2(a) and (c) are roughly 0.55 and 1.17 rad in amplitude with peak velocities of 7.6 and 15.4 rad/s respectively. For both spins, the friction of the motor ball bearing acts as a near-constant drag torque during continuous rotation. However, for the region where the velocity becomes low, the friction shows elastic behavior, which causes oscillations as seen in the magnified views. Note that the frequency of the oscillation is on the order of tens of Hertz, thereby implying that it cannot be due to mechanical eigenfrequencies such as due to motor flexibility.

The results of these tests show key aspects of the friction in both the gross- and pre-sliding regimes. In the gross-sliding regime, the friction opposes the relative velocity with nearly constant torque. In the pre-sliding regime as the rotor comes to rest, friction acts as an elastic element which resonates with the rotor inertia. Note that the oscillation period decreases as the shaft comes to the stop, as illustrated in Fig. 2(b), which indicates that the stiffness of the friction elasticity increases for smaller displacements. This observation agrees with the nonlinear hysteretic behavior of pre-sliding friction as reported in many references including [1,3] and with the GMS model adopted in this paper. Also note that the open-loop test shows a bit more lightly-damped response, but given that the two tests were conducted several months apart at different shaft positions with different environmental conditions including temperature and humidity, this discrepancy seems insignificant.

2.3. Paper outline

Both regimes should be taken into account in constructing a friction model and compensating the friction, especially the pre-sliding regime for high precision applications, since a positioning system enters this regime frequently and can be dominated by the pre-sliding friction. In Section 3, we discuss the classic and modern empirical friction models, which are efforts to mathematically represent the friction in both regimes. In this paper we use the Generalized Maxwell-Slip (GMS) model which has advantages of high fidelity and relatively simple implementation for real-time control purposes [4,8]. We develop and discuss a frequency-domain method to identify parameters of this model in Section 4. A model-based friction compensation technique is studied and the resultant performance is shown in both the frequency and the time domains in Section 5. Section 6 concludes the paper with final comments.

3. Prior art

In this section, we review representative aspects of friction together with several empirical friction models which are introduced in the literature and relevant to our study in this paper.

3.1. Classic friction model

The simplest model of the friction drag force is $F = \mu N$ where μ is the friction coefficient and N is the normal force. This model is often referred to as Coulomb friction. When the model is augmented with static friction and a linear viscous drag term, the force–velocity graph appears as plotted in Fig. 3(a). A model with continuous velocity dependency in the sliding regime was developed by Stribeck [9] in the form of

$$F(v) = s(v) + \sigma v = sgn(v)\left(F_c + (F_s - F_c)\exp\left|\frac{v}{V_s}\right|^{\delta}\right) + \sigma v,$$

and is illustrated in Fig. 3(b). This model includes viscous friction, where F_c, F_s, V_s, δ and σ represent the Coulomb friction, static friction, Stribeck velocity, shape factor, and viscous friction coefficient respectively. Note that in the Stribeck curve, $F(v)$ has both a velocity weakening curve, $s(v)$, and a velocity strengthening curve, σv [10].

The Stribeck curve is often called static since the curve equation is only a function of the relative velocity of the sliding objects, and can be experimentally obtained by measuring friction force over a range of constant velocities. The model's biggest drawback is the discontinuity at zero velocity, which is the pre-sliding regime. This limitation means that the Stribeck model cannot encompass dynamical phenomena including friction lag [11], rate-dependent breakaway force [12,13], and hysteresis with non-local memory [3,14,15].

3.2. More recent empirical friction models

There have been many efforts to develop friction models which effectively cover the sliding and pre-sliding friction regimes, and a variety of empirical models have been introduced in the literature attempting to represent the hysteretic elastic behavior of the pre-sliding regime depicted in Fig. 3(d) as well as the classic aspects of sliding friction.

Dahl [16,17] introduced a model describing the pre-sliding regime friction, in 1968, based on the speculation that the relationship of friction force and displacement resembles the stress–strain behavior of ductile materials. Introducing the LuGre model in 1995, Canudas de Wit et al. [6] constructed a state evolving equation