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A friction identification approach based on dual-relay feedback configuration with application to an inertially stabilized platform

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ABSTRACT

The low-velocity motion performance of servo system tends to be deteriorated greatly by friction due to its relay nonlinearity and Stribeck effect in low-velocity state. To achieve better low-velocity motion performance, an effective friction identification approach based on the dual-relay feedback configuration is presented for a typical low-velocity servo system, an inertially stabilized platform (ISP). An improved piecewise friction model derived from Tustin model is used to describe the frictional behaviors including static, Coulomb, viscous friction and especially Stribeck effect, so that the frictional behaviors can be appropriately formulated through describing functions. The dual-relay feedback configuration is designed to excite limit cycles in a velocity feedback loop, rather than in the position loop, so as to avoid noise arisen from the derivative of the position response signal. Properties of the limit cycle oscillations are analyzed for proper selections of dual-relay gains, based on which a systematic procedure for friction identification is proposed. Simulations and experiments on a servomechanism of ISP verify the effectiveness of the proposed approach.

1. Introduction

In design of controllers for high performance servo-mechanical systems, friction is always recognized to be the dominant nonlinear factor that should be deliberately considered [1–4]. Friction causes adverse effects on the systems, such as stick–slip motion [1,2], which may incur unacceptable tracking errors and even instability of the systems, especially in the low-velocity region and with frequent motion reversals. Let us take an inertially stabilized platform (ISP) as an example. ISPs are widely utilized on the electro-optical detection systems (EODSs) in a variety of moving vehicles to isolate vibration-sensitive optical sensors from their environments, so as to make the line-of-sights (LOSs) of optical sensors equipped in the platforms stably point to the targets located in inertial space [5]. The ISP usually works at low velocity and in small stroke. Friction existing in the gimbal bearings of the ISP degrades the performance of the servomechanism when the ISP runs bidirectionally in a low angular rate around its setpoint as a vibration-isolator [4].

To compensate for the friction in a servomechanism effectively, it is essential to identify the frictional characteristics as the first step. So far, many friction models have been built, among them, Tustin model [1,3], LuGre model [2,6], and Generalized Maxwell-Slip (GMS) model [7,8] are pervasively applied. LuGre and GMS model, regarded as the dynamic friction models, can completely describe the frictional behaviors even in a minimal displacement, i.e. the frictional memory, presliding displacement, etc. In view of that, dynamic-friction-model-based control methods are prevalent for friction compensation in high precision servo systems [6–9]. In spite of the effectiveness of these advanced control methods, they are always complex due to the nonlinear frictional dynamics related to both presliding displacement and sliding velocity as represented in the dynamic friction models. On the other hand, the static model, e.g., Tustin model or its variants, despite static mapping between friction and velocity [3], is well-recognized to be able to approximate the real friction with high confidence [10]. It is also proved that the switching static model and the dynamic model predict almost the same limit cycles generated by friction in control system [11]. Hence, the static friction model such as the Tustin type, still has great significance in practical applications for friction identification and compensation.

Relay feedback approaches [12–22] are frequently applied for friction identification, because the system dynamics including parameters to be identified can be explicitly and conveniently described in frequency or time domain by the approaches. The relay-based friction identification methods can be generally categorized into two kinds, they are, the frequency-domain or called

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describing function (DF)-based method [12–17] and time-domain method [18–22]. The basic concepts of the two types of methods were both discussed by Besançon-Voda et al. [12,18] in the early years. Tan et al. [13] modeled the friction as a combination of Coulomb and viscous components and firstly applied a dual-relay feedback structure to identify the parameters of servo system with friction. Chen et al. [14,15] extended Tan's method and respectively utilized the dual-relay and hysteretic relay feedback structures in a position feedback loop to estimate the friction by a systematic identification procedure. Kim and Chung [16] and Wu et al. [17] used a similar relay-based method for friction identification and improved the servo performance. On the other hand, Olsson and Åström [19] discussed the properties of limit cycle oscillations in time domain generated due to friction, based on which Chen et al. [20] proposed a triple-relay feedback identification approach by fully using the switching conditions of a stable limit cycle. Liu et al. [21] developed the time-domain method to identify the direction-dependent characteristics of friction. Recently, a combination of time-domain identification method and the disturbance observer was applied to a high acceleration motion control [22]. Unlike the method by Liu et al. [21] done in velocity loop, Liu et al. [22] identified the system by measuring the position signals when limit cycles are extracted.

Generally, we can estimate a more accurate value of friction by time-domain methods than DF methods, because the exact time-domain response and the switching property of relay feedback system are fully applied in time-domain methods so as to avoid identification error arisen from DF approximation in DF methods. However, time-domain methods have a common drawback that all of them treat the friction as a relay nonlinearity ignoring frictional negative viscous behavior in low-velocity state such as Stribeck effect, which may still cause inaccuracy especially for precise control of the system operated at low velocity. On the other hand, despite completely friction identification including various components such as static, Coulomb, viscous friction and boundary lubrication velocity [14,16] by DF methods, some improvements are still left, i.e., Chen's method [14] is not suitable for a low-velocity servo system due to the simplified friction model in that research, and the simultaneous multi-parameter optimization process without any initial conditions in Kim's method [16] may lead to identification error due to the local minimum. In conclusion, to improve the low-velocity motion performance of the system, a more effective friction identification approach is still required.

This paper proposes an improved friction identification approach based on dual-relay feedback configuration with application to a low-velocity servo system, an ISP. In order to describe the frictional behavior in low-velocity state more effectively, a piecewise friction model derived from Tustin model is presented. Based on the proposed properties of limit cycles induced by dual relays, a systematic friction identification procedure is presented to fully identify the frictional behavior both in high- and low-velocity states. Simulation and experiment results show that the proposed method can make a further improvement on the low-velocity motion performance of servo system than other relay-based methods.

The paper is organized into 5 sections. In Section 2, the velocity-loop dual-relay feedback identification approach is discussed, in which DF formulation for the frictional nonlinearity and the properties of limit cycles are emphasized. In Section 3, the systematic four-phase identification procedure to derive all the parameters of the friction model is proposed. In Section 4, the servo mechanical system of ISP is introduced, and the simulations and experiments in this servomechanism are conducted to verify the effectiveness of the proposed friction identification. Finally, conclusion is drawn in Section 5.

2. Dual-relay feedback identification approach

2.1. Dual-relay feedback configuration

A velocity-loop dual-relay feedback configuration is designed as shown in Fig. 1a, by which a limit cycle representing the velocity response of the system can be extracted and measured, so as to avoid the noise arisen from derivative of the position signal in position-loop relay feedback system. In Fig. 1a, the dual-relay module $N_r(A)$ is set to identify the inherent frictional nonlinearity $N_f(A_v)$ by limit cycle analysis. Note that $G_g(s)$ is an intentional low-pass digital filter cascaded in the dynamic model of the servomechanism $G_m(s)$ to increase the order of servo system so as to insure the availability of DF approximation as much as possible. In practical relay tests, the velocity response signal $\dot{\theta}_o$ is smoothed into $\dot{\theta}_f$ by $G_g(s)$. That is, when the system moves sinusoidally, the amplitude of a limit cycle after filtered, denoted as A, can be derived as the product of the amplitude of velocity sinusoidal response A_v and the norm of the frequency response function of the low-pass filter $|G_g(j\omega)|$, and the relationship among them is $A = A_v|G_g(j\omega)|$.

Provided that the input reference signal $\dot{\theta}_i$ is set to 0, Fig. 1a can be transferred equivalently into Fig. 1b, as shown from which the dual-relay feedback system is composed of an equivalent linear model $G_{eq}(s) = G_m(s)G_g(s)$ and an equivalent nonlinearity $N_{eq}(A, \omega)$ including the inherent frictional nonlinearity $N_f(A_v)$ and the intentional dual-relay module $N_r(A)$.

The existence of the sustained limit cycle arisen from dual relays of the system can be predicted by Nyquist curve of the frequency response function of the linear model $G_{eq}(j\omega)$ and the negative inverse DF of the equivalent nonlinear element $N_{eq}(A, \omega)$ described in polar coordinates [23]. If limit cycles exist, harmonic balance condition [24] is satisfied, which is expressed as

$$1 + G_{eq}(j\omega)N_{eq}(A, \omega) = 0. \tag{1}$$

Through analyzing the amplitudes A and frequencies ω of the limit cycles combined with harmonic balance condition, the friction included in the DF of the equivalent nonlinear element $N_{eq}(A, \omega)$ can be identified.

2.2. DF approximation of nonlinear element

Tustin type of friction model describing frictional velocity-dependent behavior has been widely utilized to compensate for friction in servo system [3,10], which is expressed as

$$T_f = (T_c + (T_s - T_c)e^{-|\dot{\theta}|/\delta} + f_v|\dot{\theta}|)\mathrm{sgn}(\dot{\theta}) \tag{2}$$

where T_c is Coulomb friction, T_s is static friction, f_v is viscosity constant and δ is Stribeck velocity. Stribeck term in Eq. (2) can be expanded by Taylor series as

$$(T_s - T_c)e^{-|\dot{\theta}|/\delta} = (T_s - T_c)\sum_{n=0}^{\infty}\frac{(-1)^n}{n!}\left(\frac{|\dot{\theta}|}{\delta}\right)^n. \tag{3}$$

In this paper, the order n of Taylor expansion in Eq. (3) is chosen to 1 due to the tradeoff between the reasonability and simplification in calculation, in that case, Tustin model can be further rewritten by piecewise function respectively in low- and high-velocity states as

$$T_f = \begin{cases} (T_s - f_0|\dot{\theta}|)\mathrm{sgn}(\dot{\theta}) & |\dot{\theta}| < \delta \\ [T_c + f_v(|\dot{\theta}| - \delta)]\mathrm{sgn}(\dot{\theta}) & |\dot{\theta}| \geqslant \delta \end{cases}, \tag{4}$$

where $f_0 = (T_s - T_c)/\delta$. The corresponding friction curve related to velocity is shown in Fig. 2, from which we can see that there are four parameters T_c, T_s, f_v and δ to be identified. Compared to the four-parameter friction model used by Chen et al. [14] and Kim