

Effect of Friction Torque on Electromechanical Brake System Dynamics

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ABSTRACT

Actuator and roller screw mechanism are key components of electromechanical brake (EMB) system in automotive and aerospace industry. The inverted planetary roller screw mechanism (IPRSM) is particularly competitive due to its high load-carrying capacity and small assembly size. For such systems, friction characteristic and friction torque generated from rolling/sliding contacts can be an important factor that affects the dynamic performance as well as vibration behavior. This paper investigates the modeling and simulation of the EMB system in early design stage with special attention to friction torque modelling of IPRSM. Firstly, a step-by-step system model development is established, which includes the controller, servo motor, planetary gear train and roller screw mechanism to describe the dynamic behavior of the EMB system. Secondly, detailed analytical formulations are established to calculate the friction torque in the time domain for evaluating its influence on the EMB displacement dynamic response. Finally, the dynamic performance for the EMB system is simulated under various driving conditions. Simulation results show that the friction torque has a significant influence on system brake performance. Larger friction torque tends to increase the vibration level. A force control strategy is proposed to decrease the oscillatory movement caused by the friction.

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INTRODUCTION

With the rapid development of future vehicle concepts, the automotive industry is progressing towards electromechanical brake-by-wire (BBW) systems as it allows compact component structures, improvement in dynamic response times, and potential ability of adding safety features through individual brake control against skidding [1]. Currently, there are two implementations of BBW systems: 1) electrohydraulic brake (EHB) systems and 2) electromechanical brake (EMB) systems. In a conventional electrohydraulic brake or EHB system, brake torque is generated by the hydraulic piston when pressing the brake pads onto the brake disc. Since the EHB applies hydraulic power, it has some weaknesses such as hydraulic oil leakage, the need for additional power sources, the lack of longevity of hydraulic components, and high maintenance cost [2].

In contrast, this brake behavior can be implemented by applying a brushless DC motor driving screw mechanisms in the EMB system, which provides an ideal basis to convert electric quantities into

clamping forces [3]. The rotor is integrated with the nut, so when the stator is powered, the rotor/nut rotates and pushes the screw spindle forward or backward through the planetary rollers. Here, the motion conversion assembly is called inverted planetary roller screw mechanism (IPRSM). There already exists EMB development in which the IPRSM is applied to the disk brake [4]. It is particularly suitable due to its high load-carrying capacity, small building volume and high rotating speed capacity, which makes the EMB space saving and environmentally friendly because of the elimination of fluid. Furthermore, without the liquid filling into the hydraulic cylinder, the brake response to pedal commands is faster than that of traditional brake system, especially in a low brake torque situation.

However, the introduced mechanical device will inevitably change or affect the brake performance of EMB system in despite of the above advantage. When generating a consistent brake torque, the large axial clamping force is immediately absorbed by the nut-roller-screw mechanism without transferring down to the motor and gear train [5] and thus leads to a braking pressure hysteresis or fluctuation because

of the heavy mechanical friction. Hence, the friction torque may lead to brake vibration and noise issues and should not be neglected in the dynamic modeling of roller screw mechanism. But there are not many references discussing the influence of friction torque on the EMB system dynamic response. As a result, detailed mathematical modeling plays a critical role in designing and developing the EMB system including brushless DC motor, planetary gear train, roller screw mechanism, and floating brake caliper.

There are many research studies of EMB systems reported in the previous literature that focus on the friction force/torque analysis and the system dynamic response simulation and testing. Analytical ball bearing torque calculations and friction losses estimation in ball screw system were conducted by Houpert [6] and Olaru [7], which allow to design and optimize a bearing or ball screw for a specific application. Karam et al [8] developed an accurate model of mechanical power transmission within roller-screw electromechanical actuators with special focus on friction effect. The friction representation model was suggested and validated on a standard actuator test stand. A frictional model between the motor speed and torque was proposed by consideration of the Coulomb and viscous friction [9]. These friction forces were compensated by proposed feedforward method. Jaeho et al [5] developed a detailed analytical model including essential nonlinear characteristics for electromechanical brake. The linearized model was obtained for its modal properties to understand the modes that are critical to the low frequency dynamics. A performance simulator and dynamic models were developed [10] for a hybrid electric vehicle [11]. The control performance based on cascade control loops was evaluated via the simulation of the regenerative braking during various conditions. A clamping-force control algorithm was proposed [12] for a planetary reduction gear-type EMB considering the friction characteristics and the clamping force estimation. The experiments showed that the friction of the screw thread plays a major role in the overall friction.

The aim of this paper is to develop a dynamic model with special emphasis on friction torque modelling of inverted planetary roller screw mechanism in early design stage. In the first part of this paper, electromechanical brake components are introduced that consists of control unit, servo motor, planetary gear train and roller screw mechanism. Then, an analytical friction torque model of the inverted planetary roller screw mechanism is presented for integrating in the whole EMB model. Finally, the dynamic performance simulation for the roller screw-type EMB is performed [13], and the influence of friction torque on brake system performance is investigated.

MODEL DEVELOPMENT

The dynamic model of electromechanical brake [3, 5] consists of an electric control unit, a brushless DC motor, a mechanical transmission system (planetary gear train and roller screw mechanism) and a disk brake caliper. The motor and transmission system constitute the so-called actuator which generates the brake forces in the brake caliper. Each of the actuators is in a position to deliver forces of up to several tons. The rotational motion of the motor is transformed into a linear

motion of the screw through the planetary gear and the screw thread. During the operation of the brake, the linear motion of the screw generates a clamping force between the pad and the disk, which provides the braking torque. This type of brake (Figure 1) is made structurally stiff by fixing the nut/carrier to the caliper housing and applying roller screw mechanism instead of ball screw mechanism.

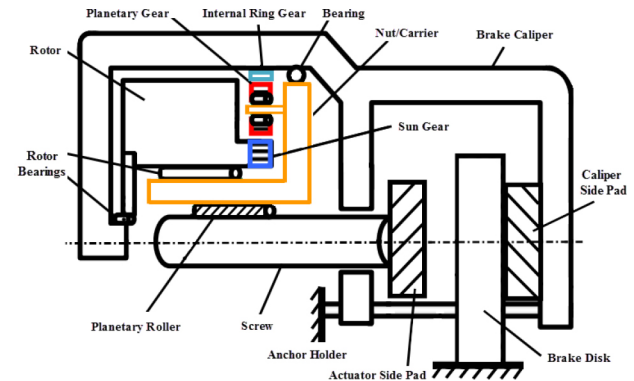


Figure 1. Structure for electromechanical disk brake.

To evaluate the performance of the EMB shown in Figure 1, a dynamic model of the EMB is developed as shown in the following sections.

Controller Model

The controller is one of the most important components in the EMB system. In order to prevent armature current overload and maintain dynamic response, the PI control is constructed based on cascade control loops composed of the current and velocity, while the position control loop is taken with the proportional control for design flexibility. As shown in Figure 2, the controller model also includes signal sampling, voltage and current amplitude limiting, and force control loop.

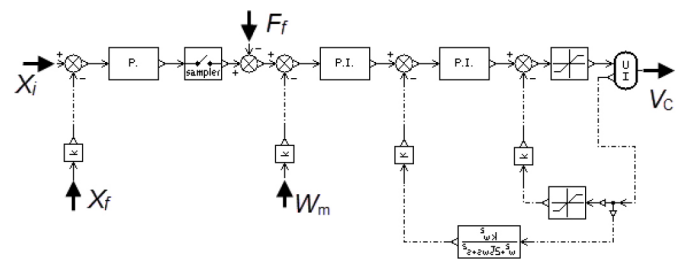


Figure 2. The controller model.

Motor Model

Brushless DC motor (Figure 3) is applied for the limited space and high reliability in this study. The relationship between the armature current I_c and input voltage V_c can be expressed by a first order transfer function G_e :

$$I_c = G_e(V_c - K_o \omega_m) \quad (1)$$

$$G_e = \frac{1/R_c}{L_c/R_c s + 1}$$

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where ω_m is motor speed, K_ω is electromotive force of the motor, R_c is electric resistance of the motor, L_c is electric inductance of the motor. All the relative values of the motor model are listed in [Table 1](#).

Table 1. Parameters of motor model.

EMF of the motor	0.25 V/(rad/s)
Electric resistance of the motor	1 Ω
Electric inductance of the motor	0.01 H
Electromagnetic torque constant	0.179 Nm/A
Motor rotor inertia	0.001 kg·m ²
Motor friction coefficient	0.002 Nm/rpm

Assuming that the field current is constant, the motor torque T_m can be represented as:

$$T_{\text{m}} = K_{\text{c}} I_{\text{c}} \quad (3)$$

where K_c is electromagnetic torque constant. When the current reaches saturation, the Equation (3) changes to the following expression:

$$T_{\text{peak}} = K_{\text{c}} I_{\text{lim}} \quad (4)$$

where T_{peak} is peak voltage and I_{lim} is maximum current.

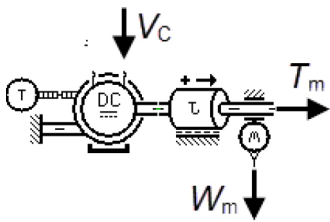


Figure 3. The motor model.

Reduction Gear Model

Currently, the power density of motor is not large enough that alone providing the required braking torque because of limitations such as the installation space. Therefore, a torque-amplified device such as a reduction gear set is commonly needed in the EMB system. In this study, a planetary gear train is employed as shown in [Figure 4](#). The motor power is transmitted through the sun gear to the carrier while the ring gear is fixed as shown in [Figure 1](#). The torque and speed ratio of the planetary gear train are obtained as:

$$T_{\text{nut}} = T_{\text{carrier}} = \frac{Z_{\text{sun}} + Z_{\text{ring}}}{Z_{\text{sun}}} T_{\text{sun}} = i_{\text{gear}} T_{\text{sun}} = i_{\text{gear}} T_{\text{in}} \quad (5)$$

$$\omega_{\text{nut}} = \omega_{\text{carrier}} = \frac{Z_{\text{sun}}}{Z_{\text{sun}} + Z_{\text{ring}}} \omega_{\text{sun}} = \frac{1}{i_{\text{gear}}} \omega_{\text{sun}} = \frac{1}{i_{\text{gear}}} \omega_{\text{m}} \quad (6)$$

where i_{gear} is gear ratio, Z_{sun} and Z_{ring} are the teeth number of sun gear and ring gear, T_{nut} , T_{carrier} , T_{sun} , ω_{nut} , ω_{carrier} and ω_{sun} are the torque and speed of nut, carrier gear and sun gear respectively.

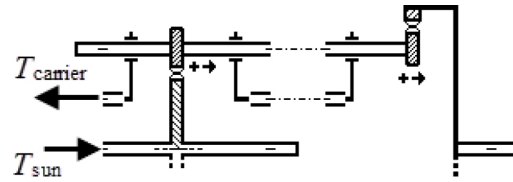


Figure 4. The planetary roller train model.

Roller Screw Mechanism Model

In order to convert the rotation of the planetary gear into a linear movement for the necessary clamping force of the brake caliper assembly, an inverted planetary roller screw mechanism is employed. The modelling of IPRSM takes the effects of nonlinearities associated with friction and clearance into consideration by integrating the components' rotation and translation. The friction characteristic is a key factor to the system dynamic and vibration performance. In this study, the total friction torque M_{IPRSM} of IPRSM is calculated by an analytical approach described in the following sub-sections. The detailed structure and definition of IPRSM can be found in [14-15]. All the friction torque related parameters and complete nomenclature of IPRSM are listed in Table 3 shown in Appendix.

Curvature Friction Torque

For thread teeth of the rollers subjected to a normal load from the nut or the screw, a single contact pure rolling can only occur at two lines. Outside of two lines, sliding occurs in one direction; between the lines sliding occurs in the opposite direction. This contrary sliding generates the curvature friction torque M_C . Essentially, it is due to the local shear stresses τ that developed in the slip region. The curvature friction torque is defined by Equation [16]:

$$M_C = \int \tau \cdot z \cdot dA = \int f_C \cdot \sigma \cdot z \cdot dA \quad (7)$$

Considering Hertzian distribution of the contact pressure and load distribution of nut-roller contacts and screw-roller contacts respectively [17] in Figure 5, an analytical relation of M_c can be further derived. Using the appropriate sign for sliding friction force, i.e., negative for $X_1 < X < X_2$ ($X = \frac{x}{a}$, $X_1 = \frac{x}{a_1}$, $X_2 = \frac{x}{a_2}$) and positive elsewhere, the final friction torque due to the contact race curvature on i th equivalent arc of different contact positions M_{c-} is obtained as:

$$M_{C_{\bar{x}i}^-} = 0.1 f_c Q_{\bar{x}i}^- \frac{a_{\bar{x}i}^{-2}}{2R_a} \left(\int_{-1}^1 (X^2 - X^4) dX - 2 \int_{X_1}^{X_2} (X^2 - X^4) dX \right) \quad (8)$$

where the subscript $\bar{x}$ denoted as n or s to represent the nut-roller contacts or the screw-roller contacts respectively and the second subscript i corresponds to the i th equivalent arc along the roller axis.