Clamp-Force Estimation for a Brake-by-Wire System: A Sensor-Fusion Approach

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Abstract—The elimination of a clamp-force sensor from brake-by-wire system designs is strongly demanded due to implementation difficulties and cost issues. In this paper, a new method is presented to estimate the clamp force based on other sensory information. This estimator fuses the outputs of two models to optimize the root-mean-square error (RMSE) of estimation. Experimental results show that the estimator can accurately track the true clamp force for high-speed cases as demanded by the antilock braking system controls. A training strategy has been used to ensure that the estimator can successfully adapt to parameter variations associated with wear. This paper is concluded with a discussion on the reliability of the developed clamp-force estimator.

Index Terms—Brake-by-wire, dynamic stiffness, optimization, sensor fusion, torque balance.

I. INTRODUCTION

ONE OF THE primary intentions for the introduction of drive-by-wire technologies is to ultimately develop intelligent vehicle control systems that improve performance by benefiting from the integration of electronic systems [1], [2]. Drive-by-wire is also intended to improve actuation response times by replacing mechanically actuated systems that are used in conventional vehicles. The design and implementation of electromechanical braking (EMB) systems for drive-by-wire have been focused upon by researchers and industry experts [3]–[6]. Fig. 1 shows a schematic diagram of a brake-by-wire system. The human–machine interface in a brake-by-wire system is provided by a pedal-feel emulator. Such a pedal is equipped with sensors that indicate the level of brake demand required by a driver. The output signals from these sensors are processed by an electronic control unit that appropriately controls the actuators. A high level of signal and hardware redundancy is employed in a brake-by-wire system to ensure fault-tolerant operations for this safety-critical application [7], [8].

There are two actuation designs that are preferred in the brake-by-wire systems. The first involves the use of electromechanical components that retain many of the hydraulic mechanisms adopted in conventional vehicles. Here, an electric-motor-driven pump, in conjunction with the proportioning valves, provides the method of brake control to each wheel. Due to the convenience of using the existing parts, this concept is the first proposed approach for the implementation of a brake-by-wire system in the automotive industry [9]. The second brake-by-wire approach reduces weight and is more environmentally friendly (due to the brake-fluid omission) than the electrohydraulic technologies. This scheme uses an electric motor drive that is coupled to a reduction-gear setup to provide brake control to each wheel. The motor is typically a three-phase brushless permanent-magnet dc type for the purpose of compactness and improved commutation efficiency. The reduction gearing generally consists of a planetary gear train connected to a ball screw that can generate clamp forces of up to 50 kN. Fig. 2 shows a sectioned view of a floating EMB caliper under development at the Pacifica Group Technologies (PGT).
EMB calipers generally utilize a clamp-force sensor to close
a loop for the purpose of controlling the caliper dynamic
performance. The control of an EMB with an internal clamp-
force sensor can be achieved using a standard motion control
architecture adopted for servo motors (cascaded position, ve-
locity, and current control loops), which is slightly modified to
suit the application at hand. Line et al. [10] replace the position
control loop with a force control loop to control an EMB. This
architecture is shown in Fig. 3.

A clamp-force sensor is a relatively expensive component
in an EMB caliper. The cost is derived from its high unit
value from a supplier, as well as marked production expenses
because of its inclusion. The latter emanates from the complex
assembly procedures dealing with small tolerances, as well as
online calibration for performance variability from one clamp-
force sensor to another. The successful use of a clamp-force
sensor in an EMB system poses a challenging engineering task.
If a clamp-force sensor is placed close to a brake pad, then
it will be subjected to severe temperature conditions reaching
up to 800 °C that will challenge its mechanical integrity. In
addition, temperature drifts may need to be compensated for.
This situation can be avoided by embedding a clamp-force
sensor deep within the caliper, i.e., at the near end of the ball-
screw (see Fig. 2). It has been shown that embedding this sensor
leads to hysteresis that is influenced by a friction between the
clamp-force sensor and the point of contact of an inner pad with
the rotor [11]. This hysteresis prevents a true clamp force to be
measured.

Due to the cost issues and engineering challenges involved
with including the clamp-force sensor, it is highly desirable to
eliminate this component from the EMB system. A potential
opportunity to achieve this presents itself in a sensor-fusion
approach at the signal level, i.e., to accurately estimate the
clamp force based on the alternative EMB-system sensory
measurements leading to the omission of a clamp-force sensor.

The organization of this paper is as follows. In Section II,
we briefly discuss the EMB sensors that can be used to create
a virtual clamp-force sensor. The advantages and disadvantages
of the previous schemes to exclude a clamp-force sensor are
mentioned. In the next three sections, we detail the develop-
mental path taken to attain a new clamp-force estimator. In
Section VI, we describe the test rig used to obtain the data
for analysis, which is then followed by experimental validation
of our final estimator. The next section discusses the relevant
issues concerning our findings within this paper. Finally, con-
clusions are drawn.

II. DEVELOPMENTAL BACKGROUND

Motor-current sensors are generally part of all EMB system
designs. There are three motor-current sensors for a three-phase
brushless dc motor that is typically used in the EMB designs.
Since a motor provides a torque input which, in turn, induces
the clamp force, the current and the clamp force are correlated.
In a simplified model, the torque produced by a permanent-
magnet dc motor is linearly proportional to the current passing
through its field coil such that

\[ T_m = K_m I_m \]

where \( T_m \), \( I_m \), and \( K_m \) are the motor torque, the motor current,
and the motor-torque constant, respectively. For a brushless
permanent-magnet dc motor, the current \( (I_m) \) is the quadrature
component of the resultant current space vector as determined
from the individual phases [12].

To calculate an induced clamp force in an EMB caliper using
a motor current, a torque-balance equation can be used. The
torque-balance equation states that the torque supplied by the
motor \( (T_m) \) equals the sum of the torques required to generate
the clamp force \( (T_{cl}) \)—also called the application torque—to
overcome the inertial effects \( (T_i) \) and to overcome the frictional
resistance \( (T_f) \). The application torque \( (T_a) \) is proportional to the
clamp force \( (F_{cl}) \), with the reduction-gearing gain \( (\gamma_{tot}) \)
determined by combining the load ratios of a series-connected
planetary gear train and a ball screw. The inertial torque \( (T_i) \)
is proportional to the motor angular acceleration \( (d^2\theta_m/dt^2) \)
with a lumped inertia gain \( (J_{tot}) \) that involves both rotational
and translational motions. The torque-balance equations are

\[ T_m = T_a + T_i + T_f \]

\[ I_m K_m = \gamma_{tot} F_{cl} + J_{tot} d^2\theta_m/dt^2 + T_f \]

\[ F_{cl} = (I_m K_m - J_{tot} d^2\theta_m/dt^2 - T_i) / \gamma_{tot}. \]
in the EMB system designs as it allows for the efficient and smooth control schemes to be implemented [12]. An alternative form of position sensing chosen for the EMB system designs is the use of the Hall sensors, which is a less expensive option. The drawbacks with this scheme are that the resolution is lower, which leads to inefficient commutation, and the torque ripples [12]. The work presented in this paper relies on the use of a resolver; however, the possibility of using the Hall sensors will be discussed later.

As described by Olsson et al. [13], an accurate friction model cannot be derived from first principles, and therefore, the friction torque ($T_f$) is undefined in (2). To improve accuracy, general friction models should be used in accordance with compensations for friction phenomena that occur in a particular system. Such frictional phenomena are identified from the empirical data. At present, the use of a friction model of any sort to estimate the clamp force for an EMB application tends to be avoided due to the difficulty in developing an accurate model that is robust to wear. Fig. 4 shows the clamp force plotted versus the motor torque for two cases: a relatively new caliper, and a caliper that has undergone durability testing. In both cases, the perturbation signals (motor angle) were identical, and the calipers were of the same mechanical construction. It can be seen that, for the same motor torque, a difference in the clamp force of up to 5 kN occurs. This is attributed to the frictional variation in the reduction gearing. We use an adaptive approach to overcome this problem for clamp-force-estimation purposes, which is detailed in later discussions.

Schwarz et al. [11] developed a clamp-force-estimation algorithm for an EMB caliper that is designed for a disk brake. Part of their algorithm involved the use of (2). To avoid a need for a friction model, they superimposed a high-frequency low-amplitude sinusoid on the gross angular motion from the motor. This served to force the motor to pass the same location in a short period of time between a clamping and a releasing action. Applying (2) to these instants and adding up the clamping and releasing equations, followed by some algebraic manipulation, yield the following expression for the estimated clamp force

$$F_{cl}^* = \left( \frac{T_{m,cl} + T_{m,rl} - J_{tot} d^2(\theta_{m,cl} + \theta_{m,rl})/dt^2}{2\gamma_{tot}} \right)$$  

(3)

where the cl and rl subscripts mean clamping and releasing, respectively. The friction term ($T_f$) has been cancelled out due to a sign change from clamping to releasing and vice versa. The major issue with this method is its limitations for high-speed applications. This is because, on geometrical grounds, capturing the clamping and releasing actions at the same motor angle is very difficult at high speed. Also, whether an EMB caliper has the dynamic control ability to reverse a direction at high speeds in a short interval of time is contentious. A means to cope with this deficiency is provided by Schwarz et al. [11] and is discussed as follows.

The characteristic curve of an EMB caliper is defined as the pseudo-static relationship that exists between the motor angle and the induced clamp force, as shown in Fig. 5. Schwarz et al. [11] propose to use a caliper characteristic curve solely to estimate the clamp force for feedback control in Fig. 3. In the instants where (3) can be applied, it is used to adapt the parameter variations in the characteristic curve associated with a pad wear.

Fig. 6 shows the clamp force versus the motor angle, where the latter is varied in a uniform random manner with a 100-ms sample time. It is apparent that there is a significant dynamic in
III. DYNAMIC-STIFFNESS MODEL

The dynamic relationship between the motor angle and the clamp force is attributed mainly to the viscoelastic effects exhibited by the aluminum caliper bridge [14] and, to some degree, the brake pads [15] (see Fig. 2 for the component locations). A step-by-step identification approach, as shown by Fig. 7, is used to attain an appropriate model structure of the relationship existing between the motor angle and the clamp force. An external clamp-force sensor is used throughout this identification process to give an accurate measure of the induced clamp force. Parameter adaptation is finally performed in the absence of any clamp-force sensor.

A. Frequency-Domain Identification

Frequencies ranging from 0 to 5 Hz are used for perturbing the motor angle from a caliper in a sinusoidal manner. It is ensured that the expected clamp-force operating range is covered. A fast Fourier transform is applied to the logged signals, the motor angle, and the induced clamp force for each frequency. The peak absolute value and the accompanying phase angle are extracted from each signal at each frequency. A decibel value is taken for the ratio of the absolute values at each frequency to give the continuous trace in the gain plot of Fig. 8(a). The difference in the phase angles for each frequency is taken to give the continuous trace in the phase shift plot of Fig. 8(b).

Since the roll-off from Fig. 8(a) is close to $-20$ dB per decade and the phase-shift range from Fig. 8(b) is near $-90^\circ$, Fig. 8 is indicative of a first-order system [16]. Based on this, it is determined that a relationship between the motor angle and the induced clamp force is of the first order. To determine the parameters of such a model, a least squares approach is
used as follows. For the sake of simplicity, the continuous-time notations will be shown. Consider a first-order transfer function expressed by the following equation:

\[ G(s) = \frac{F_{cl}^*(s)}{\Theta_m(s)} = \frac{K}{\tau s + 1} \tag{4} \]

where \( F_{cl}^* \) and \( \Theta_m \) denote the estimated clamp force and the motor angle in s-domain, respectively. The gain \( (K) \) and the time constant \( (\tau) \) are the parameters to be determined. In the frequency domain, (4) can be rewritten as follows:

\[ \frac{1}{|G(j2\pi f)|^2} = \frac{4\pi^2 \tau^2}{K^2} f^2 + \frac{1}{K^2} \tag{5} \]

where \( f \) is the frequency of the perturbation signal in hertz. Equation (5) is in a form of \( y = a_1 x + a_2 \), where the unknowns \( a_1 \) and \( a_2 \) contain the first-order gain \( (K) \) and the time constant \( (\tau) \) parameters. Applying the least squares to solve for \( a_1 \) and \( a_2 \) allows the first-order gain and the time constant to be subsequently evaluated. With a linear model defined, a frequency response is estimated, as shown by the dashed traces within Fig. 8(a) and (b), which is to be compared with the empirically determined frequency response, as shown by the continuous traces. Fig. 8(b) shows that the influence of the nonlinear effects is clear since there are trace deviations [16]. To illustrate the significance of this nonlinearity, a clamp-force response and its linear system estimate are shown in Fig. 9 for a sinusoid case. In both instances, the perturbing sinusoids were identical with frequencies of 1.1831 Hz. The error is obviously significant with a difference of over 10 kN that is shown in areas. Therefore, the introduction of compensation is required to improve the accuracy of the determined fundamental describing function.

**B. Compensation**

Converting (4) into a time-domain form yields

\[ F_{cl}^* = K\theta_m - \tau dF_{cl}^*/dt. \tag{6} \]

In a low-speed case, \( dF_{cl}^*/dt \approx 0 \); therefore, the estimated clamp force \( (F_{cl}^*) \) will nearly be linearly proportional to the motor angle \( (\theta_m) \). However, as shown in Fig. 5, a characteristic curve for a caliper is nonlinear and is accurately described by a third-order polynomial. This nonlinearity can be, at the very least, attributed to a variation in stiffness exhibited by the brake pads [17] and the caliper bridge [18]. Based on this, a more accurate variation of (6) is as follows:

\[ F_{cl}^*(k) = A_2\theta_m^3 + A_1\theta_m^2 + A_0\theta_m - \tau dF_{cl}^*/dt \tag{7} \]

where \( A_2, A_1, \) and \( A_0 \) are the characteristic-curve coefficients. The discrete form of (7) is of more practical use and is expressed as follows:

\[ F_{cl}^*(k) = \alpha_3\theta_m^3(k) + \alpha_2\theta_m^2(k) + \alpha_1\theta_m(k) + \alpha_0F_{cl}^*(k-1). \tag{8} \]

To determine the value of the coefficients in (8), a least squares approach is used with the uniform random data. The sample time of these data is 100 ms over an approximately 0–45-kN range. Applying the defined form of (8) to a new uniform random data set yields a root-mean-square error (rmse) of 0.28 kN. This error is attributed to the model structure of (8), being not entirely representative [19], as well as the sensory-noise effects.

**C. Parameter Adaptation**

Fig. 10 shows the characteristic-curve variation for the different pad thicknesses. It is clear since stiffness is an important part of (8); necessary parameter adjustments must be made in-service. Equation (3) provides a means to estimate the clamp force without the need for a friction model in a modified torque-balance approach. If an in-service EMB caliper automatically varies its motor angle in a cyclic manner, say for instance when the vehicle park brake is locked, (3) can then be used to estimate the clamp force. Fig. 11 shows such an input that we use, where a sinusoid of high frequency and low amplitude is superimposed on another sinusoid of lower frequency and higher amplitude. Two points in Fig. 11, clamp and release, are marked at the same motor angle. We apply (3) at these instances to attain an estimate of the clamp force for both points in time. We repeat this throughout the signal so that a series of clamp-force estimates is obtained. Keeping a log of these clamp-force estimates, along with the associated motor angles (using
adaptation process as well as the greater approximations used in the final model structure.

IV. TORQUE-BALANCE MODEL

It is shown that the viscous contribution to friction in an EMB caliper’s reduction gearing is small compared to the Coulomb friction component [11]. Hence, a simplified friction model [13], [20] is included in (2) as follows:

\[ T_m \approx T_a + T_i + (\mu F_{cl} + A) \text{sgn}(d\theta_m/dt) \]  \hspace{1cm} (9)

where \( \mu \) is the coefficient of the kinetic Coulomb friction, and \( A \) is a kinetic offset term. The offset term is required to take into account the frictional resistance prior to inducing a clamp. The sign function \( \text{sgn}(\cdot) \) (1 for positive and \(-1\) for negative arguments) is included to model the friction sign change that occurs between clamping and releasing. Fig. 4 shows the level of frictional variation occurring in the reduction gearing from an EMB caliper over time. To estimate the clamp force using a torque-balance approach, it is necessary that the friction model parameters be updated at timely intervals. Before discussing the methodology we use to cope with the friction variation, its characteristics are briefly reviewed for an EMB caliper.

The frictional variation occurring in an EMB caliper is not apparent after a limited number of cycles. Fig. 13 shows the motor torque versus time for a high-speed cyclic case where the motor angle is taken as the perturbation signal. The clamp force is approximately between 0 and 20 kN. It is observed that the motor torque has a nearly identical trace at the beginning and the end of the test. This shows that the frictional parameters remain constant during the short time intervals. Hence, the updating of the friction model is not required after every braking action; instead, it is required after a number of braking actions.

Employing the same methods to estimate the clamp force using (3) as previously described, we then determine the parameter values within a discrete version of (9) by performing a least squares fit. A discrete form of (9) is shown as follows:

\[ T_m(k) \approx \gamma_{tot} F_{cl}^*(k) + \frac{J_{tot}}{t_a^2} (\theta_m(k) - 2\theta_m(k-1) + \theta_m(k-2)) \]
\[ + (\mu F_{cl}^*(k) + A) \text{sgn}(\theta_m(k) - \theta_m(k-1)) \]  \hspace{1cm} (10)

where \( t_a \) is the sampling time. Adapting the friction model parameters in (10) can be performed at timely intervals throughout the service life of an EMB caliper.

After some algebraic manipulation of (10), a parametric expression to estimate the clamp force in real time is attained in (11), shown at the bottom of the page. Applying (11) to a uniform random test yields Fig. 14. The rmse is 0.61 kN.

\[ F_{cl}^*(k) = \frac{T_m(k) - J_{tot} \left( \frac{1}{t_a} (\theta_m(k) - 2\theta_m(k-1) + \theta_m(k-2)) - A \text{sgn}(\theta_m(k) - \theta_m(k-1)) \right)}{\gamma_{tot} + \mu t_a \text{sgn}(\theta_m(k) - \theta_m(k-1))} \]  \hspace{1cm} (11)
This error is attributed to the friction model, being not entirely representative [10], as well as the sensory-noise effects.

V. FUSION

If the statistics of the two independent estimates \( x_1 \) and \( x_2 \) of some desired quantity \( x \) are known, then an estimate \( \hat{x} \) will be optimal when the expected value of a loss function is minimized. A loss function is described as follows:

\[
Q = \mathbb{E}[(x - \hat{x})^2 | x_1, x_2] \tag{12}
\]

where \( Q \) is a measure of loss. Taking the partial derivative of the expectation of (12) and setting it equal to zero allows for an optimal estimate of \( \hat{x} \) to be described which is

\[
\hat{x} = \mathbb{E}[x| x_1, x_2]. \tag{13}
\]

To estimate the term \( \mathbb{E}[x| x_1, x_2] \), the conditional density function \( p(x| x_1, x_2) \) is required. Using Bayes’ theorem [21], this density function is given by

\[
p(x| x_1, x_2) = \frac{p(x_1, x_2 | x) p(x)}{\int_{-\infty}^{\infty} p(x_1, x_2 | x) p(x) dx}. \tag{14}
\]

The probability density functions on the right-hand side of (14) are all definable. If the statistics of \( x_1 \) and \( x_2 \) are zero-mean Gaussian, then \( p(x| x_1, x_2) \) can be found to be

\[
p(x| x_1, x_2) = \frac{\sqrt{\sigma_1^2 + \sigma_2^2}}{2\pi \sigma_1 \sigma_2} e^{-\frac{(x - (x_1 + \sigma_1^2 + x_2 - x_1))^2}{2\sigma_1^2 + \sigma_2^2}}. \tag{15}
\]

where \( \sigma_1 \) and \( \sigma_2 \) are the rmse’s of \( x_1 \) and \( x_2 \), respectively. The mean value of (15) is equal to the optimal estimate \( \hat{x} \) and is given as follows:

\[
\hat{x} = x_1 + \frac{\sigma_2^2}{\sigma_1^2 + \sigma_2^2} (x_1 - x_2). \tag{16}
\]

The rmse of \( \hat{x} \) is then given by

\[
\sigma_{\hat{x}} = \sqrt{\frac{\sigma_1^2 \sigma_2^2}{\sigma_1^2 + \sigma_2^2}}, \quad \sigma_{\hat{x}} \leq \min(\sigma_1, \sigma_2). \tag{17}
\]

The validation errors from both models that were previously given to estimate the clamp force were each found to have bell-shaped profiles when plotted in histograms. Based on this, (16) can be efficiently applied to fuse the two clamp-force estimates given by the two models. Writing (16) in terms that are relevant to this paper in discrete time notation yields

\[
\hat{F}_{cl}(k) = F_{ds}^*(k) + \frac{\sigma_{ds}^2}{\sigma_{ds}^2 + \sigma_{tb}^2} (F_{tb}(k) - F_{ds}^*(k)) \tag{18}
\]

where the subscripts ds and tb denote the dynamic stiffness and the torque balance, respectively.
VI. EXPERIMENTAL RESULTS

A. Experimental Setup

A test rig was set up for use on a prototype EMB caliper. An external servo motor is used to provide actuation by coupling it to the caliper reduction gearing, as shown in Fig. 15. The external motor is of the brushless permanent-magnet type, with ratings of 55.5 N·m and 5000 r/min, and ensures that the required clamp forces (up to 50 kN) can be achieved. MATLAB’s Simulink package, along with the xPC block set, provides a real-time operating system that is implemented to control the external motor angle.

The control of the external motor is achieved using the PID controllers within a standard motion control architecture: cascaded position, velocity, and current control loops. The logging of sensory measurements is attained via uploading the signal data to the host PC from the target PC, which are marked 1 and 2, respectively, in Fig. 15. The logged data are stamped at 100-µs time-step intervals. Both the host and target PCs have Pentium 4 processors operating at 2.4 GHz. To measure the caliper motor angle, an encoder output is taken from the 1 : 1 coupled external servo-motor resolver. The resolution of this encoder output provides 8192 counts per revolution. An external torque sensor is used to sense the torque input to the EMB caliper. An external force sensor is used to accurately measure the clamp force induced by the brake pads.

For the clarity purposes, the sensory information from this test rig is treated throughout this paper as being attained from an EMB caliper.

B. Results

Fig. 16 shows the performance of (18) in tracking high-speed clamp-force measurements from tests where the motor angle was taken as an input and varied in a uniform random manner. This result shows that an adaptation to antilock braking system (ABS) controls is possibly seen as the actuation speeds are comparable. A new rmse of 0.32 kN results, which is an approximately 10% improvement on the rmse from the adapted dynamic-stiffness model alone. This is larger than the optimal value given by (17). We attribute this to the deviations of error distribution from the assumed Gaussian distribution.

VII. DISCUSSION

The stiffness of brake pads has a reasonable influence on the overall stiffness of a caliper and disk system [18]. Since the brake pads have polymeric constituencies and, in practice, their temperature can increase up to 800 °C, significant variations in stiffness model parameters are likely. The developed clamp-force estimator within this paper is reliant on stiffness. In the static test rig by which our estimator was evaluated, the brake-pad temperature was constant. Here, the variation of the stiffness model parameters is an issue and should be investigated to determine its significance.

The control architecture used by Line et al. [10] for an EMB during clamping is given by Fig. 3. To handle the initial clearance existing between the pads and disk, they revert to a position control where the outer force control loop in Fig. 3 is replaced by a position control loop. The action of the clamp-force sensor indicates which control strategy is implemented at which time. It should be mentioned that efforts are made by EMB designers to keep the clearance length constant irrespective of the pad wear so that braking response times are kept consistent. The omission of a clamp-force sensor from an EMB caliper leads to a situation where an initial contact between the brake pads and disk cannot be sensed. The clamp-force estimator developed within this paper will not be effective unless knowledge of when the initial contact has occurred is known. To overcome this problem, schemes that rely on the caliper motor current or a combination of this signal and the resolver have been used to determine when a contact has been initiated [11]. Alternatively, a simple contact sensor can be used that is inexpensive, requires no calibration, and is temperature-insensitive.

Aside from a resolver, the Hall sensors are also opted for use in EMB calipers as the method of sensing the motor angle. The resolution this scheme offers is dependent on the number of Hall sensors and motor circuits used [12]. Typically, three Hall sensors are used with three motor circuits. This provides
a resolution for sensing a motor angle of 20°. At this resolution, and sampling at 4 ms (a practicable in-service rate), the dynamic-stiffness model (8) gives identical results. However, the torque-balance model (11) becomes erroneous at the same sampling rate. Therefore, the use of Hall sensors limits the use of a torque-balance approach to estimate the clamp force.

VIII. CONCLUSION

In this paper, a new design-friendly and cost-effective approach to control an automotive brake-by-wire actuator is presented. The idea that a clamp-force sensor can be omitted from an EMB system is strongly promoted by the findings within this paper. Using the signal from an internal resolver, a dynamic-stiffness relationship was used to estimate the clamp force. A second model was also used to estimate the clamp force, which was based on a torque-balance approach that relied on the use of the internal motor-current sensors and an internal resolver. An in-service adaptation technique was used to adapt wear-dependent parameters from both models. The outputs from these two independent models were fused using a maximum-likelihood estimator to give an optimized estimate of the clamp force. The experimental verification showed the ability of the developed estimator to successfully track highly dynamic situations. With further development, the potential cost savings involved with omitting a clamp-force sensor can be realized in future EMB designs.

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