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Friction Compensation using Coulomb Friction Model with Zero Velocity Crossing Estimator for a Force Controlled Model in the Loop Suspension Test Rig

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Abstract

This paper presents a method of friction compensation for a linear electric motor in a model in the loop suspension test rig. The suspension consists of a numerically modelled spring and damper, with inputs of suspension motion. The linear motor is force controlled using a force sensor to track the output of the numerical model. The method uses a Coulomb friction model and applies a feedforward step signal when velocity zero crossing occurs. Velocity zero crossing estimation is achieved using an algorithm based on measured feedback velocity and force. Experimental results indicate reduction of force tracking error caused by Coulomb friction leading to improved test rig accuracy.

Keywords – force control; friction compensation; permanent-magnet linear motor; model in the loop (MIL); substructuring; hybrid experimental/numerical testing
1. Introduction

This paper presents a method of friction compensation for the force control of a model in the loop (MIL) suspension test rig. A MIL system emulates a physical system using a numerical model and applies the output physically via an actuator [1]. This type of MIL system is associated with hybrid experimental/numerical testing technique known as real-time dynamic substructuring [2]. However the test rig presented here does not qualify as a substructured test rig because there is no real physical test piece which is also called the substructure. In a typical substructured test rig the physical test piece interacts dynamically with the MIL system via a feedback loop. Instead this rig contains two dynamically interacting MIL systems but the concepts drawn from substructuring literature are still applicable.

Dynamic substructuring test rigs are typically limited by computation time constraints and the dynamic rating of actuators to achieve the desired outputs. The advantages of such a system are that the dynamic behavior of systems may be studied experimentally without the need for real components, and increased ease and flexibility of testing achieved by simply altering the numerical model. Furthermore the numerical model allows scaling with dynamical similitude, lending substructure systems to scaled experiments especially in situations where the dynamic ratings of actuators are constrained. This has been termed real-time pseudodynamic substructuring [3, 4].

Inaccuracies in substructuring test rigs are due to tracking delays and errors caused by actuators and sensing equipment. The test rig studied here uses a numerical model of a suspension unit whose output is the desired suspension force. An actuator attempts to track the output force in a closed loop force feedback system; this overall system is called the transfer system. A transfer system is defined to be a system which links the numerical model to the physical system [2]. The nature of delay and errors in transfer systems depend heavily on the nature of the output and the characteristics of the actuator. In a
typical position output substructuring application, transfer system lags are considered as the dominant performance limiter [2]. For force output applications, in addition to lag issues, actuator friction could have a major impact on errors, especially in suspension systems where velocity reversal is common.

This paper aims to improve the accuracy of a MIL suspension test rig by using a controller which compensates for the friction forces. Friction compensation is an extensively studied topic in engineering control literature [5]. A common class of friction compensation technique is feedforward and feedback model based compensation. Here friction is estimated using a motion based model where the input into the model is either the reference motion signal (feedforward) or a measured motion signal (feedback) [5]. Many applications of friction compensation that have been studied are for actuator positioning systems. The general lack of force sensors in typical position control systems has driven research to focus on implementing friction compensation without measuring the friction force directly. Instead friction forces are estimated using friction models based upon motion, also known as model based compensation [6]. In these systems the motion reference is known in advance so both feedforward and feedback motion based models can be used.

Classical models of friction include Coulomb and Stribeck friction models where the friction force is a simple function of velocity [5]. More complex dynamical models, such as the Lugre model incorporate rate dependent characteristics such as varying break away force and frictional lag [5]. Despite the comprehensiveness of dynamical friction models they present certain drawbacks in practical friction compensation. Complex friction models require well calibrated parameter identification and can be subject to variation from wear and tear or changes in alignment of the rig set up. Although complex dynamic models take into account the effect of small pre-sliding displacement, usually the effects are too small to be measured by typical position sensors [6]. Another difficulty with dynamic models such as
the Lugre model is the requirement of an internal friction state with fast dynamics. It has been reported that the delays in the dynamic internal friction state which results from digital control are poorest at velocity reversals which may also lead to limit cycles [6]. For these reasons, although the Lugre model receives widespread attention, there has been limited practical implementation due to realization difficulties [7]. In [7] feedback friction compensation was successfully implemented for position control in a table drive mechanism with linear motors using the Lugre model and a disturbance observer. To avoid difficulties of applying dynamic friction models to typical CNC machine tools, in addition to PID control, a double pulse compensation signal was devised based on analysis of the transient friction error at velocity reversal where friction effects were greatest [8].

Other than computational issues, the practical benefits of using a friction model depend largely on the quality of motion measurements available. Velocity information obtained from the differentiation of position signals can be inaccurate due to the noise in position sensors, with the errors more significant at low velocities. Methods of filtering or estimating a more precise value of velocity usually involve a trade off in terms of signal delay and accuracy. Some studies have concentrated efforts on improving velocity estimation, especially at low velocities, for the application of friction compensation, such as in [9] and [10]. In [9] velocity estimation was obtained by varying the sampling period of encoder counts, depending on the actuator velocity. In [10] an adaptive differential filter for velocity estimation is presented.

The task of force control and friction compensation for the MIL suspension test rig has a lot of similarities to force control in haptic interface applications, more so than force control in actuator positioning systems. Haptic interface devices with force feedback are required to transmit a force towards a user who is physically controlling the device motion, here precise force control in presence of
motion disturbance is needed for transparent force rendering [6, 11]. In these studies a force sensor is used for force feedback as it gives a more direct, reliable and accurate reading of force compared to the estimation of force from motor current. In both [6] and [11] force feedback is utilised in combination with model based friction compensation, and in [11] a comparison was conducted between different friction models. In [12] a hardware in the loop (HIL) test rig for an aircraft load simulator was achieved using force feedback in combination with a feedback Lugre friction compensation model; the experimental results verified high force tracking performance.

Figure 1 – Mechanical diagram of MIL test rig. (Side View)

Figure 2 – CAD drawing of test rig (Perspective View).
Similar to force feedback haptic interface applications, the test rig studied here (shown in Fig. 1 and Fig. 2), attempts to output actuator forces independent of actuator motion disturbance, prohibiting the use of feedforward model based friction compensation since suspension motion is not pre-determined. This leaves only the option of model based feedback friction compensation in conjunction with force feedback as a practical approach. Initial testing of the actuator showed that a proportional-integral force feedback controller is capable of compensating for friction except in velocity reversal regions. This suggests that performance levels similar to a complex dynamical model such as the Lugre model can be achieved using a simple Coulomb friction model. Research on friction compensation for a position controlled table drive in [13] demonstrated that friction characteristics are predominantly Coulomb during dynamic motion when examined using a dynamic friction model with elastic bristles.

In this paper a method of Coulomb friction compensation with force feedback control is presented. To deal with issues of motion sensor inadequacies, an algorithm which detects when velocity reversal occurs using information from both force and position sensors is utilised. The proposed method offers a simple and effective method of compensation for systems equipped with force sensors.
2. Experimental Rig

The friction compensation method described in this paper originated from practical issues encountered in building a quarter car suspension test rig. To allow testing of different suspension configurations, a MIL suspension system was proposed. Furthermore, in order to allow the incorporation of the structural dynamics of a flexible body, the usual sprung mass was replaced with another MIL system. The test rig becomes a multi-actuator MIL test rig with feedback and physical interaction between the two MIL systems as shown in the schematic Fig. 3.

![Schematic of suspension test rig with two active MIL substructures.](image)

Figure 3 – Control schematic of suspension test rig with two active MIL substructures.

In MIL test rigs the accuracy of the system depends on the errors and delays of the input to the numerical model and of the output of the transfer system. The inputs to the numerical model are the sensed outputs of the system according to the variables required for the calculation of the numerical model output. The transfer system in this context refers to the link between the numerical model and the physical system, namely the transfer function of the actuator which is close-looped controlled to
track the output of the numerical model. The real time nature of the system results in bi-directional coupling between the two MIL numerical model and the physical outputs through feedback loops.

In relation to the rig layout present in Fig. 3, there are two numerical models, one for the suspension force and one for the structural model. To investigate the accuracy of the substructure rig, the suspension numerical model is chosen as a passive linear spring and damper. The choice of suspension components to be emulated by the numerical model would affect the accuracy of the test rig if the profile of output forces cannot be satisfactorily achieved by the chosen actuator, or if the chosen pseudo mass of the structure is so small such that small force tracking errors lead to large mass accelerations.

The structural numerical model receives the input of suspension force, which is sensed using a load cell. The force is then used to calculate the acceleration of the sprung structure depending on the structure’s numerical model, then double integration of the internal acceleration state gives the output sprung mass position. The output mass position is tracked by a hydraulic actuator and the sensed position becomes the input into the force numerical model so that the complete system is a closed loop with the output of one numerical model becoming the input to the other, through the physical link of the transfer and sensing system. In typical MIL substructure systems the output from the numerical model is a displacement. In this case the test rig has one displacement output numerical model and one force output numerical model with bi-directional coupling.

The accuracy of the test rig may be validated by comparing the dynamics of the rig with the dynamics of the pure numerical model, namely the emulated system. The emulated system represents the dynamics that the rig would exhibit in the ideal case if there were no delay and errors in the transfer system. Since real components within the test rig would add to complications for comparison with the ideal emulated
system, no HIL suspension components are utilised and the wheel and tyre are ignored such that the input of suspension displacement into the suspension numerical model is considered to be between the ground and the sprung mass as shown in Fig. 4 and Fig. 5.

Figure 4 – Equivalent physical layout of rig for the single mass spring damper emulation test.

Figure 5 – Control schematic for the MIL suspension force only test rig.

Since this paper is concerned with friction compensation for a MIL suspension force tracking application, rig tests were performed assuming that the sprung mass transfer system and position measurement system has no error and delays. This was achieved by feeding back the sprung mass structure numerical model output of command position and velocity as inputs into the force numerical model as shown by
Fig. 4. With this setup all the errors in the system would be solely caused by errors in the force MIL suspension.

The performance of the MIL system can be assessed by comparison with an emulated system. The equation of the emulated system is a single mass spring damper given by,

\[ m\ddot{x}_s = -k(x_s - x_g) - c(\dot{x}_s - \dot{x}_g) - mg \]  

(1)

Here \( k \) is the spring stiffness, \( c \) is the damping coefficient, \( x_s \) is the sprung mass displacement, \( x_g \) is the ground displacement and \( m \) is the sprung mass inertia.

Performance indicators for the evaluation of the test rig accuracy are adopted from Wallace et al. [1]. In this case the emulated system is a pure numerical model of a spring mass damper whose force output is the idealised \( z^* \). Fig. 4 shows the demand force output from the numerical model, \( z \), and the actual force output from the transfer system, \( x \).

The numerical model accuracy \( e_1 \) is,

\[ e_1 = z^* - z \]  

(2)

The transfer system accuracy \( e_2 \) is,

\[ e_2 = z - x \]  

(3)

The overall accuracy of the test rig is the combination of numerical model and transfer system accuracy.
3. Controller Structure

The MIL suspension actuator used in the test rig is a Dunkermotoren linear motor STA38 which has a built in amplifier and current controller. The built in proportional, integral and derivative (PID) current control loop acts as the inner control loop. An outer control loop is designed to control the force of the actuator via force feedback and the control gains acting upon the force error provide a demand current input to the inner control loop.

The force controller has the structure of a standard PI feedback controller shown in Fig. 6. Furthermore a feed forward reference signal is incorporated to improve the performance of the actuator since motors force and current are typically proportional. Following the concept of Coulomb friction compensation the control architecture is designed to input a step command current which cancels out the Coulomb friction step change at the instance of velocity reversal.

![Figure 6 – Force Controller Structure.](image)
4. Friction Compensation For Deterministic Systems

The suspension test rig shown in Fig. 4 and Fig. 5 was subjected to a road input displacement signal as shown in Fig. 7(a). The disturbance ground input is a superposition of sine waves with random phases and amplitudes following the spectral density profile of a motorway road obtained from [14]. It corresponds to a vehicle travelling at 225 km/h from with frequencies limited between 0.1 and 5Hz. The period of the pseudo random input is 10 seconds and repeating cycles were used for experiments. Fig. 7(b) and Fig. 7(c) show the resulting suspension displacement and velocity respectively. Here the suspension displacement denotes the displacement between the ground and the sprung mass.

![Figure 7 – Road input disturbance and the resulting rig motion profile.](image)

Figure 7 – Road input disturbance and the resulting rig motion profile. a) Road displacement, b) suspension displacement, c) suspension velocity.
Fig. 8 presents the suspension force error ($e_2$) when subject to the road input in Fig 7. This shows that at higher velocities the force tracking error magnitude is relatively small, suggesting that most viscous friction is compensated for. This indicates that the error caused by Coulomb friction is much more severe than viscous friction as shown by the large force error at zero velocity. From Fig. 8 'no motion' represents the experimental case where the ground and sprung mass position are kept constant. Here the suspension velocity is actually zero and the presence of non-zero velocity in Fig. 8 is a result of noise from velocity readings.

These results suggest that major improvements in force tracking could be achieved by compensating for friction only at velocity reversal. The Coulomb friction model is chosen since it is the simplest method which captures the characteristic of friction at velocity sign changes. More complex static models and dynamic friction models are also avoided due to the difficulty of parameter identification and the requirement of very precise motion sensors [6]. This is in agreement with the finding in [13] that the
non-linear friction characteristic of a table drive is expected to behave as simple Coulomb friction when under dynamic table motion. Furthermore the friction characteristics of permanent magnet linear motors are commonly modeled using classical static friction models [15, 16, 17] as illustrated in Fig. 9.

Figure 9 – Diagram of static friction models relating friction force as a function of velocity: a) Coulomb, b) Coulomb and viscous, c) Coulomb, viscous and stiction d) Stribeck [5].

The Coulomb friction model chosen for this study is described by,

\[ f(v) = \begin{cases} F_c \times \text{sign}(v), & \text{if } v \neq 0 \\ -F_c \leq f(v) \leq F_c, & \text{if } v = 0 \end{cases} \]  

(4)

Where \( f(v) \) is the friction force as a function of velocity \( v \) and \( F_c \) is the Coulomb friction force.

The results of experimental friction compensation tests using a simple Coulomb friction model are shown in Fig. 10 where the control goal is constant force tracking while the suspension in Fig. 4 is subjected to a sinusoidal input motion disturbance. This demonstrates the principle of Coulomb model friction compensation assuming the zero velocity points are known in advance. Here the compensation signal is a step change of 0.15V, which results in a force equivalent to the step change in force due to Coulomb friction.
The results in Fig. 10(a) show that the simple Coulomb model is effective at reducing the friction force spikes when compensation occurs within +/- 0.005s of the velocity reversal. Fig. 10(b) and (c) show the effect of late and early compensation applied +/- 0.015s from the velocity reversal, which result in large force error spikes and no performance improvement. If the zero velocity estimation is determined too early then the compensation signal produces a force spike, or if too late the friction spike grows before compensation occurs.

Figure 10 – Comparison of friction compensation step triggering times on 1Hz suspension motion.

These experimental results suggest that the Coulomb friction model is effective at cancelling force spikes due to friction sign change at velocity reversal for permanent linear magnet motors, if velocity reversal times are known in advance (ie. the system is deterministic). This is possible for motion controlled systems where velocity reversal times are determined by the reference position profile. However for the case of a MIL suspension force control system the motion of the suspension is not determined by the suspension forces alone but in combination with the road disturbance input and
sprung mass dynamics. Since suspension motion is one of the unknowns to be investigated by the test rig, there is no easy model based method of estimating suspension velocity, especially if the road input was random. Furthermore the friction compensation for the test rig is intended to be generic and perform independently of disturbance. The only way of estimating or detecting zero velocity crossing for such non-deterministic systems is through investigation of the position signals from sensors and force sensors. The aim of friction compensation presented in this paper is to form a generic method for any force controlled system by utilising a simple Coulomb friction model. Despite the simplicity of the model the previous section highlights the possible benefits and also the importance of applying the compensation signal at the precise time. The challenge therefore becomes detecting when velocity reversal has occurred, or is about to occur.
5. Friction Compensation for Non-Deterministic Systems

Direct velocity measurement is difficult, so often velocity is estimated from position and or acceleration measurements. The major difficulty associated with this approach is that the friction spike occurs before the obtained velocity crosses zero. This is due to both the nature of Stribeck friction and lag associated position or acceleration signal filtering.

The method described in this paper attempts to maximize the effectiveness of detecting velocity reversal by using both motion and force information in a two stage method. The first stage is to create a window where velocity reversal is likely to occur. The second stage is then to look within the velocity window for a force profile which matches a known example friction spike which is experimentally obtained beforehand in a parameter identification process. The major benefit of using this force information to apply the compensation trigger is that it guarantees that the friction compensation signal is applied at the correct time provided that the detection is correct.

5.1 Zero Velocity Crossing Estimation

The velocity crossing window is estimated simply by using velocity and acceleration data at the current time step. Assuming constant acceleration, the predicted time to velocity zero crossing can be obtained from:

\[ T_0 = -\frac{v}{\dot{v}}, \]  

(5)

Where \( T_0 \) is the estimated time until velocity crossing and \( v \) and \( \dot{v} \) are respectively the velocity and acceleration of the linear electric actuator.
Tests conducted on the suspension rig for different frequencies and amplitudes of sine wave motions indicated that the start of a friction spike consistently occurred within 0.02 seconds before the zero crossing of the filtered velocity signal. Therefore a threshold of estimated velocity zero crossing time of 0.03 seconds could be used to ensure that a window is activated within a force spike. This difference in time between the force spike and the velocity crossing zero varies depending on the frequency and amplitude of the sine wave motions and also the bandwidth of the velocity filter. An appropriate time till velocity zero crossing threshold $T_T$ can be easily determined experimentally for any rig. Window activation rules are presented in Rule 1.

**Rule 1 - Window Activation**

\[ \text{Window} = 1 \text{ if } (T_0 > 0) \text{ AND } (T_0 < T_T) \text{ AND } (v > 0), \quad \text{Positive to negative crossing activation.} \]

\[ \text{Window} = -1 \text{ if } (T_0 > 0) \text{ AND } (T_0 < T_T) \text{ AND } (v < 0), \quad \text{Negative to positive crossing activation.} \]

\[ \text{Window} = 0 \text{ if } (T_0 < 0) \text{ OR } (T_0 > T_T) \quad \text{Window inactive.} \]

One implementation difficulty with this method is that, if the acceleration signal is lagging the velocity signal, then in certain cases velocity reversal can occur without the window being activated. This lag would be the present if acceleration is obtained by differentiating the velocity signal which would likely require filtering to reduce noise.

Another solution explored here is to forward predict the velocity signal using a polynomial extrapolation which is then differentiated. Although this reduces the accuracy of the acceleration signal the improved experimental results suggest that the window is sensitive to lag of acceleration signal. Although this method is effective in certain cases, there are also instances where because of future prediction, the
obtained acceleration could be leading the velocity signal such that acceleration switches sign prior to the actual velocity zero crossing. In these cases although a window is initiated it may be cut short before a friction force spike could be detected. To alleviate this problem two estimation windows were used in combination to ensure that both cases are dealt with, one based on a short time lag acceleration signal obtained from forward prediction and the other a low pass filtered acceleration signal with a longer time lag but less noise.

Another problem that could occur is that, due to the noise in the velocity signal, it is possible that at low velocities the signal could switch sign momentarily before the real zero crossing. This means that due to noise the window could change from active to inactive before a force spike occurs. The algorithm is adapted to deal with this issue by extending the window for a few time steps after zero velocity crossing as a preemptive action that the velocity zero crossing may be due to noise.

5.2 Force Spike Detection

The friction force spike detection algorithm is only active when the velocity zero crossing window is active. Since the force spikes are a result of the interaction between controller, sensors, friction, motion and force trajectories, the profile of the force error spike is not predictable. However there are still generalities that can be established, such as a minimum gradient of the rise, and a force error threshold which can be used to distinguish the friction spike from those caused by other errors. The aim is to detect the initial rise of the force error spike so that compensation is applied as quickly as possible for maximum effectiveness. Rule 2 demonstrates the concept and application of the force spike detection algorithm.
Rule 2 - Friction Force Spike Detection Algorithm

Conceptual Rule:

1. Window Condition: Velocity zero crossing window is active, the direction of the expected crossing determines the direction of the expected gradient change and force error.
2. Force Error Condition: The absolute force error value of the current time step must be equal or greater than the force threshold.
3. Force Error Rise Profile Envelope Condition: For a specified number of previous time steps, define a minimum absolute gradient between the force error for each subsequent time step.

Algorithm:

Force Detect = True,

If for n = 1: N (Window = 1) & (F_E(i) > F_T) & (|F_E(i-n+1) - F_E(i-n)| > G_T(n))

Else if for n = 1: N (Window = -1) & (F_E(i) < F_T) & (|F_E(i-n+1) - F_E(i-n)| < G_T(n))

Else

Force Detect = False,

For logics in Rule 2, i is the sample of the current time step, F_E is the force error, F_T the force error threshold, G_T the gradient threshold and N is the number of samples. For the test conducted in this study the specific thresholds and number of samples chosen for Condition 2 and Condition 3 were determined by examination of a friction force spike obtained experimentally. An example for this process is illustrated in Fig. 11.
Figure 11 – Example diagram of force spike profile envelope for Condition 2 and 3 used in this study for a force threshold of 4N.

Condition 1 defines the expected direction of the force error and gradient change which is determined by the direction of the expected zero velocity crossing. This constrains the direction of force error and gradient in Condition 2 and 3.

Condition 2 assigns a minimum force error threshold so that the friction force spike is detected only when a substantial force error has occurred. Choosing a low error threshold allows earlier detection of friction force spikes but also increases the likelihood of false detections due to noise in force readings. To ensure an improvement in force tracking performance the error threshold should be chosen so that it is within the bounds of force tracking error not caused by friction. For the example in Fig. 11, the force error threshold is chosen at 4N, this is below the no motion force tracking error which is between +/- 6N shown in Fig. 7. From experimental observation, a force error of 4N corresponds to an elapsed time of approximately +0.004s after the force spike rise. This ensures that if the detection is correct,
compensation will occur within 0.05s of the velocity reversal, leading to effective compensation as demonstrated in Section 4.

Condition 3 introduces a gradient threshold which is also obtained from observation of the friction force spike. After choosing a force error threshold, from the friction force spike diagram, the number of samples can be chosen so that gradient change from the beginning of the force error rise up to the threshold is constrained. This is simply creating a gradient change envelope constraining the shape of the friction force spike rise. Force error due to noise tends to be more oscillatory without such consecutive force changes in the same direction. The force profile envelope condition could be further defined to include not just the gradient but the complete force versus time profile of the friction force error spike rise. For this study, experimental results showed that good detection performance was achieved using a simple gradient envelope. The choice of the gradient envelope is illustrated by an example in Fig. 11; five gradient samples are chosen to cover the majority of the force error rise, the initial sample has a minimum gradient of zero and the subsequent samples a minimum gradient of 0.2. The minimum gradient threshold is chosen to be substantially lower than the gradient of the actual friction force spike, this is to improve robustness and demonstrates that the capability of the algorithm does not require tight force error envelope constraint.
6. Experimental Results

The friction compensation algorithm was first tested with constant force tracking and single frequency and amplitude sine wave suspension motions. Fig. 12 illustrates the performance of the detection algorithm for a 1Hz sine wave motion and Fig. 13 for 5Hz without actually implementing the compensation step signal. The force threshold was set to 2 to demonstrate the ability of the algorithm to correct for the friction spike early in the spike rise. Throughout a test of 25 seconds the algorithm successfully detected all friction force error spikes in early stages of the spike rise demonstrating good robustness.

Figure 12 – Zoomed-in view of 25s friction compensation test for 1Hz and 20mm amplitude sine wave motion.
The next step was to test the performance of the friction detection with the rig configuration emulating a single mass spring damper, using the pseudo random input disturbance as in Fig. 7. The resulting suspension motions are a result of the interaction of the disturbance and a virtual sprung mass damper with values of: \( m = 25 \text{ kg}, \ k = 350 \text{ N/m} \) and \( c = 50 \text{ Ns/m} \). This test would differ from previous tests in that now the electric actuator is tracking a varying force as defined by the numerical model unlike in earlier demonstrations whereby the force tracking was of a constant demand. The friction compensation algorithm is applied onwards after the first second and the force demand during this period is shown in Fig. 14.

Figure 13 – Zoomed-in view of 25s friction compensation test for 5Hz and 1mm amplitude sine wave motion.
Figure 14 – Demand and real force for rig single spring mass damper emulation test with no friction compensation.

With the introduction of random motion the algorithm will be far less successful for two primary reasons. The first is that motion of the suspension may indicate that the velocity is crossing zero but actually approach zero only to reverse and this would falsely trigger the zero crossing velocity windows. Secondly there are regions where suspension velocity remains low and close to zero with high levels of noise in the acceleration signal. In these conditions windows may easily be falsely activated or not activated.

The force tracking of the force profile in Fig. 14 is likely to increase tracking errors and worsen the performance of the friction detection algorithm when compared to the constant force tracking case. Increased force tracking errors lead to greater possibility of the triggering of false friction force spike detections.
Figure 15 – Friction detection and compensation demonstration for single mass spring damper emulation test subjected to 0.1-5.0 Hz random road input for algorithm with 4 N force threshold.

Figure 16 – Enlarged view of friction force spike detection and compensation from Fig. 15.
Fig. 15 demonstrates the typical performance of the detection algorithm on a section of the time response. The figure also compares the result on the force error of applying a 0.15V compensation step signal and the case of no compensation, showing that compensation is more effective when the correction signal is applied in the early stages of the force spike rise. Fig. 16 shows an enlarged view of a friction force spike compensation in Fig. 15. Further performance of the algorithm is quantified in terms of the root mean square (RMS) tracking error of the friction compensation algorithm compared with no compensation. The RMS force error is taken across the entire test run of 44 seconds and is presented in Table 1. To show that the window plays an important role in the performance, other algorithms without the window are also compared.

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<th>Synchronisation Error</th>
<th>Total Error</th>
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<td>Force Threshold Only 10</td>
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<td>0.255</td>
<td>3.877</td>
<td></td>
</tr>
<tr>
<td>Force Threshold Only 12</td>
<td>3.854</td>
<td>0.318</td>
<td>4.172</td>
<td></td>
</tr>
</tbody>
</table>

**Table 1** – Comparison of RMS error for full and partial friction compensation algorithms operating at different error force thresholds for friction force spike detection.
This capability to improve performance while using a lower force threshold compared to the no motion force tracking error is shown in Table 1 where the ‘force profile only’ algorithm outperforms the ‘force threshold only’ algorithm. From the table it is evident that ‘force profile only’ algorithms perform worse when the force threshold is lowered. This is as expected since false detections should increase. This is in contrast with the complete algorithm which maintains comparable performance irrespective of the force threshold since the velocity window helps to reduce the chance of false detections.

The results from the Table 1 clearly indicate that force error reduction is highest when using all elements of the designed friction compensation algorithm. All the algorithms utilize a 40ms buffer as the minimum time between compensation signals, meaning that two triggers will not occur within 40ms. This buffer is applied to prevent multiple triggering of force spike detections from a single force spike. There are two criteria for choosing the buffer time. First the buffer time period must be a lot smaller than the time period of the highest frequency in the system to prevent the buffer from missing detections. Here the test is subjected to a maximum frequency of 5 Hz or a period of 200ms. Secondly the buffer time must be greater than the time span of a single friction spike which is approximately 20ms.
7. Two Step Compensation

Results from Section 6 demonstrated the performance of friction compensation where the compensation step signal has amplitude equal to that of the Coulomb friction. It can be seen from Fig. 16 that the reduction of the friction force error spike does not commence immediately after triggering of the compensation step signal; the absolute force error continues to increase for a short time before decreasing. This is due to the step response dynamics of the linear electric motor and its controller. This means that increasing the amplitude of the step input compensation signal would result in a faster reduction of the force error. However, application of a step input larger than the Coulomb friction level would also result in a change of the equilibrium point of the integral term in order to maintain the same demand force.

Fig. 17 illustrates the effect of applying a compensation signal twice of that of the friction level. Although the friction spike error reduction occurs swiftly, the force errors after the friction spike are increased as the integral term drives towards a new equilibrium point.

![Figure 17](image)

Figure 17 – Experiment result comparing force error of 0.15V step compensation signal against 0.3V.
One way to enable a large compensation step to be used while not altering the integral term’s equilibrium point is to add a second step input in the opposite direction shortly after the first step so that the summation of the two step inputs equate to the friction level. In this case this means adding a second step with magnitude 0.15V opposing the first step.

To test the effect of the timing of the second compensation step signal a simulation was conducted and results are shown in Fig. 18. The simulation was simplified so that the outer loop force controller consisted of only an integral term. Since the controller consists of an outer force loop and inner current loop, there are two equilibrium conditions to be satisfied, namely force and current equilibrium. The current equilibrium is determined by the demand force and the friction force while the force equilibrium is at zero force error. The timing of the second step is crucial and should be triggered so that the motor current settles to the new equilibrium before the force error from the friction force spike returns to zero. This is to ensure that as the friction force error is driven to equilibrium at zero error, the current would already be close to the equilibrium current allowing the system to settle quickly. Under this arrangement, as the second step is applied it is opposing the action of the force integral controller and in doing so a gradual approach to zero force error can be achieved.

If the second step is applied late then the motor current is driven to the demand current defined by the first compensation step as the force error approaches zero. After the force error changes sign the integral controller will act in the opposite direction and drive the motor current to the equilibrium. If at this moment the second step is applied, both the force integral controller and the second step signal will be pushing the system in the same direction causing the force output to overshoot and oscillate as shown in Fig. 18. In effect the purpose of the two step compensation signals is to drive the motor current to its new equilibrium position such that action is done by the force controller integral term is
reduced. This is beneficial since high integral gains lead to oscillatory responses when correcting force errors.

Figure 18 - Simulation Result Illustrating the effect of the timing of Step 2 triggering. Step 1 compensation is applied at 5.004 second. a) Force error plot, b) Force controller integrator voltage, c) Current controller demand and actual current

This technique allows for any combination of step magnitudes of the two steps as long as their summation is equal and opposite to the Coulomb friction. This would mean that the larger the steps magnitudes the quicker the compensation signals will correct for the friction force spike as long as the current response to the second step settles before force error zero crossing. Fig. 19 compares the simulated response of two sets of step magnitudes showing superior performance when the step magnitudes are larger.
Figure 19 - Simulation Result Illustrating the effect of choosing steps with larger magnitude. a) Force error plot, b) Current controller demand and actual current

Figure 20 – Experiment result comparing 1 vs. 2 step compensation. Force error threshold for step 1 compensation is 4N. Step 2 is triggered when force error re-crosses 4N.
The two step compensation algorithm was applied to the same system as presented in Section 6, Fig. 15 and Table 1. The second step was triggered when the force error crossed the level of 4N after application of the first step. Experimental tests indicated that 4N was a sensible option as it ensured a safety margin such that the motor current would settle before force zero crossing. Fig. 20 shows a section of the result in a time series comparing one step compensation against two step compensation.

It is clear that for a friction cancellation step input of 0.3V, the two step compensation is better at reducing the subsequent force oscillations after the Coulomb friction spike. The RMS error from the complete 44s time response displayed in Table 2 confirms that the two step algorithm is superior compared to the one step algorithm. The two step compensation algorithm reduces the total error($e_1+e_2$) of the test rig by 37%. Fig. 21 demonstrates the two step algorithms ability to cancel out the friction spike compared to the no compensation case. The plot shows an approximate reduction in Coulomb friction force error spike of 50% from 20N to 10N.

Most existing state of the art friction compensation techniques use dynamic friction models such as Lugre and Dahl and have been applied to position control systems which makes comparison unsuitable. An indirect comparison can be made to force control systems such as haptic interface applications. In [6] a PID force control is used in combination with a Coulomb friction compensator, the results showed that across the velocity range the combined controller is more effective, there seems to be little if any reduction in peak force error at zero velocity crossing.
Figure 21 – Force velocity plot comparing no compensation and 2 step compensation.

<table>
<thead>
<tr>
<th>RMS error</th>
<th>No Compensation</th>
<th>1 Step Compensation</th>
<th>2 Step Compensation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force Tracking ($e_1$)</td>
<td>0.281</td>
<td>0.212</td>
<td>0.196</td>
</tr>
<tr>
<td>Synchronisation ($e_2$)</td>
<td>3.955</td>
<td>2.663</td>
<td>2.474</td>
</tr>
<tr>
<td>Overall ($e_1 + e_2$)</td>
<td>4.236</td>
<td>2.875</td>
<td>2.670</td>
</tr>
</tbody>
</table>

Table 2 – RMS tracking error of $e_1$ and $e_2$ and overall system error $e_1 + e_2$ for compensation using 4N force error threshold.
8. Conclusion

The friction compensation technique presented in this paper demonstrated the ability to improve the performance of MIL force control systems. The algorithm achieved approximately 50% reduction in friction force error spike for an actuator subjected to a pseudo random disturbance motion of 0.1-5Hz. In regards to the single mass spring damper MIL suspension test rig the two step friction compensation algorithm reduced the combined RMS force tracking and synchronisation error by 37% when compared with a situation in which no friction compensation algorithm was applied. The biggest advantage of this technique is that it could be implemented on any system without the need for modeling of the actuator, or friction characteristics except for a simple Coulomb model. Friction parameter identification, which can be troublesome for complex models is avoided. This algorithm only requires the determination of the integrator level change caused by Coulomb friction and observation of the friction force error spike in relation to velocity signal in order to choose the appropriate time till velocity zero crossing threshold, force error threshold and the friction spike rise envelope. A force sensor and position sensor for the actuator stroke are the only instruments required. This means that this technique could be implemented with relative ease on any force control system, especially for systems with non-deterministic actuator motion to achieve significant improvement in force tracking. The method is limited to smooth force profiles and actuator motion where the frequency must be lower than the buffer time constant between allowable compensation signals.
9. References


