Accurate Friction Compensation for a High Precision Stage using a Synchronous Piezoelectric Device Driver

Truong Ngoc Minh  Student Member (Nagaoka University of Technology)
Kiyoshi Ohishi  Senior Member (Nagaoka University of Technology)
Masasuke Takata  Non-member (Nagaoka University of Technology)
Seiji Hashimoto  Member (Gunma University)
Kouji Kosaka  Non-member (Tech-concierge Kumamoto Inc.)
Hiroshi Kubota  Non-member (Kumamoto University)
Tadahiro Ohmi  Member (Tohoku University)

Keywords: SPIDER, high positioning systems, robust control, friction compensation

The scope of this paper is to deal with accurate friction compensation for a high precision stage using a synchronous piezoelectric device driver (SPIDER) by proposing a feed-forward friction compensator (FF) based on the LuGre friction model.

Firstly, a model of SPIDER system is identified because the structure of SPIDER system is hard to model mathematically. We use the auto-regressive with exogenous input (ARX) and the state-space model to describe the dynamics of SPIDER. From experimental results, it is seen that the identified state-space model gives more accurate results than the identified ARX model. Then, we choose the state-space model to design a robust feedback controller (Cz) based on the doubly coprime factorization and the disturbance observer for satisfying such requirements as high accuracy, fast response with small overshoot, and robustness.

The LuGre friction model is known that can represent accurately friction phenomena such as stick-slip motions, and pre-sliding displacements. To identify the LuGre friction model parameters, we have carried out a series of experiments in different friction regimes, i.e., 1) in the sliding phase and 2) in the stick phase. Steady state slip experiments are used to identify viscous friction coefficient $\alpha_2$, Coulomb friction $\alpha_0$, and Stribeck velocity $v_0$. Experiments in stick regime are used to identify the stiffness of the microscopic displacement $\sigma_0$ and the corresponding damping coefficient $\sigma_1$.

The structure of the proposed model-based feed-forward friction compensator (FF) is shown in Fig. 1. This FF uses the reference position to generate an appropriate control contribution $u_{ff}$ that leads to enhance the tracking performance of SPIDER system. Denoting reference velocity and friction state by $\dot{x}_r$, and $z_{cr}$, respectively, then a feed-forward control signal $u_{ff}$ is given by equation (1).

$$u_{ff} = \sigma_0 \dot{x}_{cr} + \alpha_1 \left( \frac{\dot{x}_r - |\dot{x}_r|}{g(\dot{x}_r)} \right) + \alpha_2 \dot{x}_r$$

where $\sigma_0$, $\alpha_1$, $\alpha_2$ are parameters of LuGre model.

The performance of the proposed FF controller is evaluated through comparison of experimental results with two other friction compensators. The first one using the bang-bang control (BB) is based on the well-known static friction model combined of Coulomb and static friction compensation. The second one is a friction state observer (FO) based on LuGre model that requires only position measurement. From Table 1, it is obvious that excellent tracking performance is achieved with the FF algorithms. FF algorithm also gives more smooth control inputs than these of BB and FO. Moreover, the feed-forward approach does not raise the question of closed-loop stability, which should be incorporated in the controller design. Experiments also confirm that SPIDER system is suitable for the next generation of high precision positioning systems.

Fig. 1. Block diagram of the model-based feed-forward friction compensation

Fig. 2. Tracking performance of the proposed feed-forward friction compensators with sinus inputs

Table 1. Comparisons of tracking errors

<table>
<thead>
<tr>
<th>Types of controller</th>
<th>Ramp inputs Mean error [mm]</th>
<th>Ramp inputs Max. error [mm]</th>
<th>Sinus inputs Mean error [mm]</th>
<th>Sinus inputs Max. error [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C(z) + BB$</td>
<td>0.0011</td>
<td>0.0177</td>
<td>0.001</td>
<td>0.062</td>
</tr>
<tr>
<td>$C(z) + FO$</td>
<td>0.0009</td>
<td>0.0064</td>
<td>0.0005</td>
<td>0.050</td>
</tr>
<tr>
<td>$C(z) + FF$</td>
<td>0.0004</td>
<td>0.0064</td>
<td>0.0002</td>
<td>0.025</td>
</tr>
</tbody>
</table>
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Kiyoshi Ohishi† Senior Member
Masasuke Takata† Non-member
Seiji Hashimoto‡‡ Member
Kouji Kosaka*** Non-member
Hiroshi Kubota**** Non-member
Tadahiro Ohmi†† Member

It is well-known that one of the major limitations to achieve a good performance in mechanical systems is the presence of friction. High resolution positioning systems operating with accuracies in nanometer region usually exhibit relatively large steady-state tracking errors or even oscillations if controllers are designed without considering friction. Consequently, this paper aims at improving the position control of a high precision stage using a synchronous piezoelectric device driver (SPIDER) by comparing the performances of three friction compensators. These friction compensators detect friction in the system and use this information to modify the control input. The first using bang-bang control is based on the well-known static friction model. The second is a friction state observer based on the dynamic friction LuGre model, and the third is a feed-forward compensator based on the LuGre model. In order to effect a fair comparison, three friction compensators uses the same identified friction parameters in controller synthesis. The performance comparisons are presented by means of experimental results on the proposed high precision stage using SPIDER.

Keywords: SPIDER, high positioning systems, robust control, friction compensation

1. Introduction

High precision positioning systems are always required because of the needs of machining and processing of, for example, semiconductors manufacturing. Nowadays, ultrasonic motors have become a standard option for sub-nanometer accuracy and resolution positioning systems. Ultrasonic motors using resonant vibrations of the piezoelectric elements can achieve high precision, high speed and high power. However, the resonant vibrations reduce the nanometer-range positioning and increase the dead time consuming the electric power. To overcome this problem, we have developed a high precision stage using a synchronous piezoelectric device driver (SPIDER) that can be achieved readily the nanometer regime positioning. Basic characteristics of SPIDER system have been reported in.

Due to the driving method, SPIDER system is a friction-driven system. In high precision motion systems, friction can severely deteriorate the performance and can introduce negative side effects such as tracking errors, large settling times or limit cycles. Then, accurate friction compensation is needed to increase tracking performance considerably, represented for example by smaller tracking errors and/or lower settling times.

From the literature study, a clear distinction can be made between two friction compensation classes, i.e., the non-model-based and the model-based compensation. Compensation of friction can be difficult for several controller tasks that encounter the stiction or low velocities because of its nonlinear and discontinuous characteristics. Thus, the non-model-based compensation methods are less general and also less in use for controller design in applications with high precision position. The most commonly used model-based compensation technique is a fixed compensation based on a good identified friction model. On the other hand, model-based methods compensate the friction force by applying an equivalent control force based on the friction model in the opposite direction.

To date, several kinds of friction model have been discussed. They can be classified into static and dynamic models depending on the inclusion of frictional memory. For static friction models, this frictional memory is omitted,
whereas for dynamic friction models this memory behavior is described with additional dynamics between velocity and friction force. These dynamic friction models, however, require the estimation of certain fictitious state variables. Which model is preferred depends on the purpose of it, but the model that accurately describes friction is in general to be preferred.

The previous work in this SPIDER system use a classical static friction model to describe the friction force between the actuator and the stage. Then, the scope of this paper is to deal with accurate friction compensation methods for the high precision stage using SPIDER by comparing performances of three friction compensators. The criterion for comparison is based on the tracking errors, both at the steady state and during transients. First, driving principles and characteristics of the proposed high precision stage are introduced. Next, a brief discussion on the control synthesis of three friction compensation methods is represented. Herein, we use both a static friction model, and a dynamic LuGre model to design friction compensators. The first one using the bang-bang control is based on the well-known static friction model. This friction model is a static map between the velocity of the stage and friction force combined the Coulomb, and static friction. The second one is a friction state observer based on the LuGre model and the last one is a feed-forward compensator based on the LuGre model. Finally, experimental results are presented to give the realism.

2. High Precision Stage Control System

2.1 System configuration The high precision stage shown in Fig. 1 has two main parts (1) the plant; and (2) the control system. The plant consists of SPIDER, a stage with an additional load of 20 kg, a linear scale and a linear encoder as a position sensor with 100 nm resolution. SPIDER is a piezoelectric actuator using the non-resonant characteristic of an ultrasonic motor. It has eight legs, and each leg consists of eight piezoelectric plates as shown in Fig. 2. The upper four plates are used for the expand motion and the bottom plates are used for the shear motion. The friction tip of SPIDER contacts with the side of the stage. The control system is implemented by using a Pentium IV PC with a servo driver. The weight of the moving part of the stage is approximately 1.2 kg. The longitudinal feed of stage is 100 mm.

2.2 Operating Principle Walking drive is a method to feed the stage over a linear guide by utilizing micro-deformations of SPIDER. Its principle is similar to walking motions of such animals as human beings, horses. Fig. 2 illustrates one complete cycle consisting of eight steps of the feed mechanism. First, the stack A shrinks, while the stack B clamps and feeds the stage. The stack A corresponds to one leg of animals, while the stack B corresponds to the other leg. In order to perform the stage feed, it is necessary a leg raising that exceeds the roughness of the osculating plane. It is impossible a displacement of the stage if the traction force required exceeds the limit imposed by the friction. The precision stage under study has a driving rail made of ultra-flat alumina ceramic with an average roughness of 0.2 µm. For the next movement, the function of stacks are reversed. A continuous displacement can be performed by repeating this sequence periodically. It should be noted that the strokes of legs are limited but the stage can move infinitely. The operating sequence becomes possible by applying a sinusoidal wave voltage for both expand and shear motion. The phase difference between two motions is π/2 rad, and that of two legs is π rad.

2.3 System Identification SPIDER system is a complicated system, where the structure is hard to model mathematically, and intermediate state variables are extremely difficult to measure. Therefore, we use the system identification method to determine SPIDER system’s model. Input data is the applied voltage to the actuator, and output data is the position signal of the stage.

The auto-regressive with exogenous input (ARX) and the state-space model are used to describe the dynamics of SPIDER. The state-space model provides a more complete representation of the system than polynomial models because state-space models are similar to the mathematical model of the identified system. Moreover, the parameter settings for the state-space model is only the order, or the number of states of the model. While using the ARX model method, the user need to specify system orders and delays that may suffer potential problems with numerical instability and excessive computation time. The model order is often set higher than the actual model order to minimize the equation error.

From Fig. 3, it is seen that the identified state-space model gives more accurate results than the identified ARX model. Then, we choose the state-space model to design control elements. The corresponding transfer function is given by

![Fig. 1. Experimental configuration](image1.png)

![Fig. 2. Operating sequence of SPIDER](image2.png)
3. Robust Feedback Controller

SPIDER system requires controllers to satisfy such requirements as high accuracy, fast response with small overshoot, and robustness. In addition, simple controller structure, ease of controller design and adjustment are important in practical applications. We have proposed a robust feedback controller based on the doubly coprime factorization and the disturbance observer(6). In Fig. 4, the inner loop system is the closed-loop system based on state feedback and state observer. The outer loop system is equivalent to the closed-loop system based on disturbance observer. It is seen that $g(z)$ is equivalent to a low-pass filter used to make the inverse of the nominal transfer function realizable.

$$P(z) = \frac{N(z)}{D(z)}$$  \hspace{1cm} (2)

$$N(z)X(z) + D(z)Y(z) = 1$$ \hspace{1cm} (3)

$$C(z) = \frac{X(z)}{Y(z)} + \frac{1}{Y(z)} \frac{g(z)}{1 - g(z)} \frac{1}{N(z)}$$ \hspace{1cm} (4)

Poles of state feedback and state observer are chosen by the coefficient diagram method(6). In this work, the equivalent time constant and the stability index are chosen as $\tau = 0.1$ and $\gamma_1 = [2.5 \ 2.2]$, respectively. Parameters of $C(z)$ are specified in Table 1.

4. Design of Friction Compensator

Consider the SPIDER system with friction given by equation (5).

$$m \ddot{x} = u - F$$ \hspace{1cm} (5)

where $m$ is the mass, and $x$ is the position of the stage, $u$ is the control input representing the effect of all applied forces except the friction, $F$ is friction force. In order to compensate effective friction force, major effort is made in the development of friction models, suitable for analysis and controller synthesis, which have limited complexity but a rich similarity to practically observed friction properties. Friction models can be subdivided with respect to their detail in describing surface contact properties occurring on a microscopic and macroscopic level(3). In this paper, the well-known static friction model, and the dynamic LuGre model are used to design friction compensators.

4.1 Friction Models

4.1.1 Static Model

The observed friction phenomena during early days of scientific study of friction have led to models of Coulomb, viscous and static friction and its possible combinations, which are often referred to as classical models of friction. In this work, we use the static friction model combined Coulomb, and static friction. It should be noted that one of the main problems with the classical model is the discontinuity at the zero velocity. Then, it may lead to
non-uniqueness of the solutions to the equations of motion for the system.

4.1.2 Dynamic Model  
Canudas et al. has proposed a new friction model that can represent friction phenomena such as stick-slip motions, frictional lag, and pre-sliding displacements \(^{(a)}\). The friction interface is modeled as elastic bristles. Deflection of the bristles caused by a relative motion of two surfaces gives rise to the friction force. The standard form of the LuGre model is given by the following equation \((6)\).

\[
\dot{z} = v - \frac{|v|}{g(v)} z \tag{6}
\]

where \(z\) is the friction internal state, \(v\) is the relative velocity between the two surfaces.

The function \(g(v)\) defining the Striebeck curve has been proposed to be in equation \((7)\).

\[
g(v) = \alpha_0 + \alpha_1 e^{-(v/v_0)^2} \tag{7}
\]

where: the sum \(\alpha_0 + \alpha_1\) corresponds to stiction force, and \(\alpha_0\) to the Coulomb friction, \(v_0\) influences when transition from stick to slip is to appear.

The friction force generated from this bending of the bristles is described as following equation \((8)\).

\[
F = \sigma_0 z + \sigma_1 \dot{z} + \sigma_2 v \tag{8}
\]

where \(\sigma_0\), \(\sigma_1\), and \(\sigma_2\) are the stiffness of the microscopic displacement and the corresponding damping coefficient, respectively, \(\sigma_2 v\) is the linear viscous friction.

Parameters \(\alpha_0\), \(\alpha_1\), \(v_0\) are referred to as static parameters.

Parameters \(\sigma_0\), \(\sigma_1\) are referred to as dynamic parameters.

4.1.3 Friction Parameters Identification  
One aspect when dealing with the characterization of the friction force is the actual identification of friction. However, it is not possible to measure the friction force directly. Therefore, the estimation of the friction model parameters is important for obtaining quantitatively accurate friction models, which can be used as a mathematical representation of the friction. Experiments for the identification procedure are performed by sensing quantities that are influenced indirectly by the friction force, such as displacements, velocities of the stage connected to the frictional contact surface.

To identify friction parameters of the LuGre model, a series of experiments has been outlined Canudas de Wit et al. \(^{(a)}\). The idea is to estimate the model parameters in different friction regimes, i.e., 1) in the sliding phase and 2) in the stick phase. Steady state slip experiments are used to identify viscous friction coefficient \(\alpha_2\), Coulomb friction \(\alpha_0\), and Striebeck velocity \(v_0\). The velocity of the stage is kept constant for a considerable time to measure one point of the steady state relationship between friction force and velocity. Polynomial fits are used on the data to find stiction force, \(\alpha_0\), and \(\alpha_2\). The sum \(\alpha_0 + \alpha_1\) corresponds to the stiction force. Then, it is possible to estimate \(\alpha_1\) if \(\alpha_0\) is known. \(v_0\) defines at which velocity a decrease of the friction force present in the motion from stick to slip. As a result, by starting in stick, and then ramp up an applied control input until slip is reached, it is possible to find \(v_0\).

Experiments in stick regime are used to identify \(\sigma_0\) and \(\sigma_1\). In the stiction regime, where there is no apparent motion \((z = x, v = dx/dt)\) the friction in equation \((8)\) can be approximated by

\[
\dot{z} = v \quad F = \sigma_0 x + \sigma_1 v \tag{9}
\]

Therefore, this linearized equation can be used as a regression model for (least square) identification if \(F, x, v\) are known when making very small motions near \(x = 0\). This identification is difficult to carry out since it is experimentally difficult to measure \(F, x, v\). To cope with this problem, the control input to the actuator was chosen as a slowly varying sine wave resulting in velocities varying from zero in the static regime to velocities in the viscous regime. This input allows to maximize the amount of data collected near zero velocity while minimizing the amount of data in the viscous region.

It should be noted that friction is history dependant and experimental identification may often exhibit different friction values. Therefore, identification tests should be performed close together. With this in mind, we performed ten times. The friction parameters were calculated for each time and the average of these values are used in the controller synthesis. A summary of all identified friction parameters is given in Table 2.

### Table 2. Identified Friction Parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Positive velocity</th>
<th>Negative velocity</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>(F_{Static})</td>
<td>28.04</td>
<td>-25.62</td>
<td>V</td>
</tr>
<tr>
<td>(F_{Coulomb})</td>
<td>22.893</td>
<td>-17.48</td>
<td>V</td>
</tr>
<tr>
<td>(\alpha_0)</td>
<td>5.15</td>
<td>-8.14</td>
<td>V</td>
</tr>
<tr>
<td>(v_0)</td>
<td>15</td>
<td>-12</td>
<td>mm/s</td>
</tr>
<tr>
<td>(\sigma_0)</td>
<td>-0.023</td>
<td>-0.023</td>
<td>V/mm</td>
</tr>
<tr>
<td>(\sigma_1)</td>
<td>1.588</td>
<td>-1.588</td>
<td>V/mm</td>
</tr>
<tr>
<td>(\sigma_2)</td>
<td>0.28</td>
<td>-0.24</td>
<td>V/mm</td>
</tr>
</tbody>
</table>

**Fig. 6.** Block diagram of friction compensator based on bang-bang control

**4.2 Friction Compensator Based on Bang-bang Control**  
Fig. 6 illustrates a block diagram of the proposed friction compensator based on bang-bang control for reducing the effect of friction \((BB)\). This controller contains two elements. The first one is the Coulomb friction compensation that checks the sign of the stage velocity, and adds a compensation voltage to the control input. The other one is the static friction compensation that considers the sign of the tracking error, and adds another compensation voltage to the control input. The compensation voltages of static friction and Coulomb friction are showed in Table 2.

**4.3 Model-based Friction State Observer**  
With the LuGre friction model, it is not possible to measure the friction state \(z\) since this signal is not a physical quantity. Then, a friction state observer \((FO)\) that requires only position measurement has been proposed.
where \( L \) is the observer state vector. Denoting reference velocity and friction force in SPIDER system.

The observation error \( e = w - \hat{w} \) is in equation (13).

The term \( \hat{f}(w) - \tilde{f}(\hat{w}) \) can be approximated after a first-order Taylor expansion of \( \hat{f}(w) \) around the point \( \hat{w} \) in equation (14).

Because the estimated state vector is used for control, then generally, the observer poles specified by matrix \( A \) append to the closed-loop poles. However, the complex structure of (14) implies that it is not possible to assign any observer error dynamics with an appropriate choice of the observer gains. A large \( l_1 \) leads to a fast response of the velocity estimation error and therefore to a good perturbation rejection. In order to have a considerable effect on damping, the gain \( l_1 \) should be chosen such that \( l_1 > \frac{1}{m} \) from (14). Moreover, experiments have shown that \( l_2 \) does not improve performance. Then, observer gains are chosen as \( l_1 = 2.5 \) and \( l_2 = 0.7 \).

Using the estimated friction internal state, it is possible to calculate friction force from equation (8). Then, the estimated friction force is added to the output of \( C(z) \) controller for compensating friction force in SPIDER system.

**4.4 Model-based Feed-forward Friction Compensator**

The structure of the proposed model-based feed-forward friction compensator (FF) is shown in Fig. 8. This FF uses the reference position to generate an appropriate control contribution \( u_{ff} \) that leads to enhance the tracking performance of SPIDER system. Denoting reference velocity and friction state by \( z_{xr} \) and \( z_{xf} \), respectively, then a feed-forward control signal \( u_{ff} \) is given by equation (15).

In\(^{[6]}\), Canudas provides the boundedness of \( z_{xf} \) for any velocity \( x_r \). This leads to the control signal \( u_{ff} \) is bounded for any sufficient smooth reference \( x_r \). Thus, the model-based feed-forward friction compensator has robust stability because it leads to a bounded output of the closed-loop system.

**5. Experimental Results and Analysis**

In order to illustrate the effectiveness of the control strategy in handling friction compensation, actual positioning experiments were conducted. Digital implementation is assumed in experimental setup. The sampling time of experiments is 100 [\( \mu \text{s} \)]. Three different controllers, \( C(z) + BB \), \( C(z) + FO \), and \( C(z) + FF \) were implemented to evaluate their performance. Parameters of \( C(z) \) controller are designed in Table 1. Fig. 9 shows the comparison of ramp responses of SPIDER system with and without BB approach. It presents that using
C(z) only exhibits satisfactory control performance, but the adverse effect of friction force on the tracking performance of C(z) is evident. The controller C(z) can not compensate position errors due to the friction force around the zero velocity. But C(z) + BB controller further enhances the tracking performance and the settling time is reduced.

Figs. 10 and 11 present experimental responses of SPIDER with three friction compensators. The comparison of position control is summarized in Table 3. With ramp responses, C(z) + FO reduces the tracking error in comparison with C(z) + BB. In addition, SPIDER system with C(z) + BB causes a greater maximum error that is about 2.77 times than for the C(z) + FO solution. With sinus responses, C(z) + FO also display better tracking capability than that of using C(z) + BB along the entire reference signal. These results prove that C(z) + FO has a better performance than C(z) + BB.

But the proposed C(z) + FF displays more precise positioning performance than that with C(z) + FO. C(z) + FF greatly

<table>
<thead>
<tr>
<th>Types of controller</th>
<th>Mean error [mm]</th>
<th>Max. error [mm]</th>
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<td>C(z) + FO</td>
<td>0.0069</td>
<td>0.0064</td>
</tr>
<tr>
<td>C(z) + FF</td>
<td>0.0004</td>
<td>0.0004</td>
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</table>
improves the tracking performance since the friction are effectively compensated by FF. The mean errors of this control approach are 0.4 [µm] with ramp responses, and 0.2 [µm] with sinus responses. These steady-state errors are approximately the resolution of the encoder that is frequently defined as the smallest positional increment which can be commanded of SPIDER system. Compared with the response in Figs. 11(a) and 11(b), the tracking response of Fig. 11(c) shows better tracking results at each velocity reversal. In addition, observe that control input signals of $C(z) + BB$, and $C(z) + FO$ are not smooth, which is not the case for $C(z) + FF$ solution. These observations lead to the conclusion that SPIDER system achieves excellent tracking performance with FF algorithms based on the LuGre model.

6. Conclusion

This paper experimentally compares the position control performances obtained with three different friction compensation design methods for the proposed high precision stage using SPIDER. It is evident from Table 3 that the proposed feed-forward friction compensator based on the LuGre model is the best solution. SPIDER system using this friction compensation algorithm has small tracking errors, and smooth control inputs. Moreover, the feed-forward approach does not raise the question of closed-loop stability, which should be incorporated in the controller design. While the increased complexity of the algorithm implemented, and the difficult tuning procedure (compared to the feed-forward solution) are the main disadvantages of the friction state observer compensation approach. Experiments also confirm that SPIDER system is suitable for the next generation of high precision positioning systems. Excellent tracking, however, requires accurate friction parameters. Further investigations are planned regarding robustness with respect to exogenous variables such as changing load that might limit the applicability of an estimated friction model considerably.

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References


Truong Ngoc Minh (Student Member) was born in Hanoi, Vietnam, in 1980. He received his B.S., M.S. degree, all in electrical engineering, from Hanoi University of Technology, Vietnam, in 2002, 2004, respectively. From 2004, he is currently a Ph.D. student at Nagaoka University of Technology. Mr. Minh is a member of the Institute of Electrical Engineers of Japan.

Kiyoshi Ohishi (Senior Member) received the B.S., M.S., and Ph.D. degrees, all in electrical engineering, from Keio University, Japan, in 1981, 1983, and 1986, respectively. From 1986 to 1993, he was an Associate Professor with Osaka Institute of Technology, Osaka, Japan. From 1993 to 2003, he was an Associate Professor with Nagaoka University of Technology, Nagaoka, Japan, where he is currently a Professor. Dr. Ohishi is a member of the Institute of Electrical Engineers of Japan, Society of Instrument and Control Engineers, the Japan Society of Mechanical Engineers, and the Robotics Society of Japan.

Masasuke Takata (Non-member) received the B.S., M.S., and Ph.D. degrees, all in the department of industrial chemistry, from the University of Tokyo, Japan, in 1971, 1973, and 1976, respectively. From 1976 to 1979, he was an Research Associate with the University of Tokyo, Japan. From 1980 to 1990, he was an Associate Professor with Nagaoka University of Technology, Nagaoka, Japan. From 1990, he is currently a Professor with Nagaoka University of Technology. Dr. Takata is a member of the American Ceramic Society, Ceramic Society of Japan, and Japan Society of Applied Physics.

Seiji Hashimoto (Member) was born in Aomori Prefecture, Japan, on December 19, 1971. He received the B.E. M.E. and Ph.D. degrees in Electrical and Electronic Engineering from Utsunomiya University, Tochigi, Japan, in 1994, 1996 and 1999, respectively. He joined the Department of Mechanical Engineering at Oyama National College of Technology, Tochigi, Japan in 2000 as a Research Associate. He joined the Department of Electronic Engineering at Gunma University, Gunma, Japan in 2002 as a Research Associate, and in 2005, he became Associate Professor. His research interests include system identification, motion control, and its application to industrial fields. He received the IEEE IES best presentation award in 2001 and 2003, and the IEEJ Paper Presentation Award in 1996 and 2005. Dr. Hashimoto is a member of IEEE IAS, CSS, the Society of Instrument and Control Engineers, and the Japan Society of Mechanical Engineers.
Accurate Friction Compensation of a High Precision Stage

Kouji Kosaka (Non-member) was born in Yamanashi, Japan, on February 4, 1963. He graduated from the Tokyo Metropolitan Institute of Technology in 1984. In 1984, he joined Elionix Inc., where he was engaged in the development of electron-beam lithography systems. In 1987, he moved to APCO Ltd., Tokyo, Japan, where he has been working on the development of electron-beam lithography systems. In 1992, he established Tokyo Technology Inc., where he was engaged in a comprehensive study on high precision feeding system by using PZT-actuator in concurrence with the development of electron-beam lithography systems. Those research and the development was supported by JST (Japan Science and Technology Agency) from 1999 to 2004, and research achievements have already been put to practical use in the semiconductor manufacturing fields. In 2005, he received Ph.D degree in Electronic Engineering from Kumamoto University, Kumamoto, Japan. Currently, he established Tech-Concierge KUMAMOTO Inc. at Kumamoto Prefecture and has been working on researching for the new application of PZT-actuator.

Hiroshi Kubota (Non-member) was born in Yokohama, Japan, on August 20, 1955. He received the B.S., M.S., and Dr. of Sci. degrees in pure and applied sciences from the University of Tokyo, Tokyo, Japan, in 1979, 1981, and 1986, respectively. His Ph.D. thesis involved the study on Non-linear transport phenomena in the Quasi-1 dimensional conductors. He joined Kumamoto University, Kumamoto, Japan, in 1984, where he has been engaged in the research on the sub-0.1 micrometer formation of low dimensional materials. From 1991 to 1992, he was a Visiting Associate of California Institute of Technology. Since returning to Japan, he has been also engaged in the National projects on sub-0.1 micron ULSI evaluation. He is a member of the Japan Society of Applied Physics and American society of Physics.

Tadahiro Ohmi (Member) was born in Tokyo, Japan, in 1939. He received the B.S., the M.S., and Ph.D. degrees in electrical engineering from Tokyo Institute of Technology, Tokyo, in 1961, 1963, and 1966, respectively. Prior to 1972, he served as a Research Associate in the Department of Electronics of Tokyo Institute of Technology. In 1972, he moved to Tohoku University and he is presently a Professor at the New Industry Creation Hatchery Center (NICHe), Tohoku University. He received the Ichimura Award in 1979, the Inoue Harushige Award in 1989, the best paper award of IEEE Transaction on Semiconductor Manufacturing in 1989, the Ichimura Prizes in Industry-Meritorious Achievement Prize in 1990, the Okouchi Memorial Technology Prize in 1991, The Minister of state for Science and Technology Award for the promotion of invention 1993, the Invention Prize and 4th International Conference on Soft Computing (IIZKA’96) Best Paper Award in 1996, the IEICE Achievement Award in 1997, the Werner Kern Award in 2001, and ECS Electronics Division Award, the Medal with Purple Ribbon from Government of Japan and the Best Collaboration Award (the Prime Minister’s Award) in 2003. Dr. Ohmi is a member of the Institute of Electronics, Information and Communication Engineers of Japan (IEICE, Fellow), the Japan Society of Applied Physics (JSAP), the Electrochemical Society (ECS), and the Institute of Electrical and Electronics Engineers (IEEE, Fellow).