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An Experimental Observation of Uncoupling of Multi-DOF PZT Actuators in a Compliant Mechanism

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Abstract

The control of a dynamic system with multiple degrees-of-freedom (DOF) is far simpler if the system is uncoupled. The property of an uncoupled system can be achieved with careful design of mechanical structures in the case of conventional mechanisms. The study reported in this paper will show via experiment the uncoupling property in a compliant mechanism, which is driven by three PZT actuators. This observation has the significance to motivate study on a new methodology for designing uncoupled multi DOF compliant mechanisms.

Key Words: Complaint mechanism, Micro motion, Control, Uncoupled systems.

1 Introduction

During the past decade considerable research has been conducted to develop micromanipulators to be used for purposes, such as biological cell manipulation in biotechnology or micro-component assembly in micro-technology. Many of these micromanipulators are based on the use of the piezo-ceramic actuator (PZT) and the compliant mechanism. PZT actuators can provide near linear motion with resolution of nanometers or even sub-nanometers. Compliant mechanisms, which move solely through deformation of flexures instead of bearings, provide smooth motion with no backlash or Coulomb friction. As there are no hard non-linearities in the compliant mechanism behavior there are no physical limitations on the resolution of position control. Therefore a manipulator based on these components is able to provide ultra-high precision positioning.

Parallel micromanipulators are commonly used in micromanipulation [1-5] as they possess the advantage of greater rigidity, which allows for more accurate motion and faster response. These attributes are particularly beneficial for ultra high precision positioning. In addition the actuators can be located in the base of the manipulator so that the link masses can be reduced.

The micromanipulator studied in this research is a 3 degree-of-freedom (DOF) parallel manipulator using a monolithic, flexure hinge, compliant mechanism, actuated by 3 PZT stack actuators, see Figure 1. Flexure hinges are thin sections of material that deform under load to provide rotational motion. These hinges act as joints, which are joined by thicker sections of material, which do not deform and act as rigid links. The end-effector platform is bolted to the ends of the 3 linkages, as indicated by the triangle in Figure 1. The end-effector has degrees of freedom in the x and y-axis and rotation about the z-axis. This compliant mechanism was developed by a collaborating research group [6,7].

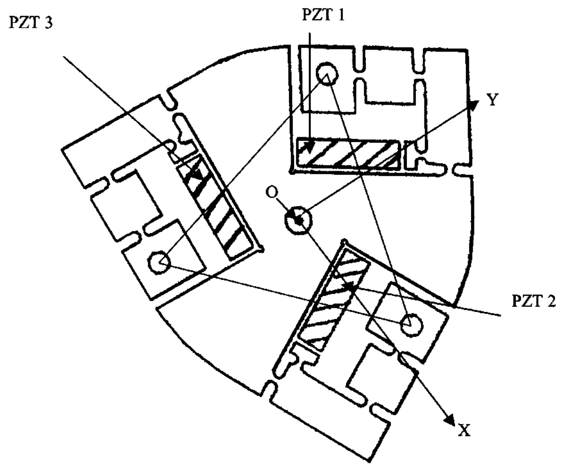

Figure 1 - Schematic of the compliant mechanism and PZTs

In order to design an effective controller for a manipulator, dynamic models are needed for control analysis. A model has been developed for this particular compliant mechanism by a previous researcher [8]. This model suggested that the mechanism be a coupled system. However, the experimental observations discussed in this paper suggest that the system be actually uncoupled. The objective of this paper is to develop an understanding of the uncoupling property in this compliant mechanism because such a property will improve control efficiency and effectiveness enormously.

2 Related Work

The original work conducted at the Advanced Engineering Design Laboratory (AEDL) of the University of Saskatchewan involved the modeling of the compliant mechanism. Kinematic and dynamic models of compliant mechanisms are commonly derived using a pseudo-rigid-body-model (PRBM) [9,10]. This assumes that the flexure hinges act as revolute joints with torsional springs attached, while the thick sections act as rigid links. A PRBM and corresponding kinematic and dynamic models were developed for this particular manipulator [8,11].

The dynamic model of a compliant mechanism differs from a standard joint mechanism as its joints have stiffness equivalent to the stiffness of the flexure hinge. The modeling of parallel manipulators is generally quite complex as the actuators are mechanically coupled and inertial variations are directly felt at each actuator. In the case of the parallel compliant manipulator the stiffness is also coupled adding extra complexity. The dynamic model developed in [8] represents the stiffness by a complex non-linear 3x3 matrix, but this model does not consider stiffness of the piezoelectric material itself. Consequently, the simulation which is based on this inaccurate dynamic model showed that there are strong couplings among three actuators in the mechanism that makes the conventional proportional and derivative control law fail [8].

Fite and Golfarb also developed a 3 DOF compliant manipulator for which they derived a dynamic model. They reported that the parallel compliant manipulator they had developed possessed highly coupled inertia and stiffness, which was also represented by a full 3x3 matrix [1].

Computed torque control algorithms have been widely used in the control of serial manipulators. However, they have been less commonly applied to the control of parallel manipulators as their coupled dynamics leads to complex dynamic equations and difficulty lies in developing a numerically simple model that can be calculated in real-time. Nonetheless, promising research has been conducted to apply the CTC algorithm to parallel macro-manipulators. A. Codourey successfully applied a CTC algorithm to the Delta robot using a computationally efficient model derived using the virtual work principle [12]. He reported considerable improvement in the control performance of the Delta robot by using this algorithm. Likewise CTC algorithms appear to be a promising method of control for parallel compliant micromanipulators. Feedforward control algorithms have been successfully applied to eliminate some of the non-linearities of the PZT actuators [13,14], while Fite and Goldfarb incorporated feedforward stiffness cancellation in the control of their micromanipulator [1]. A complete CTC algorithm requires a computationally efficient dynamic model to represent the actuators and the inertia, damping and stiffness of the compliant mechanism. The PRBM based dynamic model developed for this particular mechanism is too computationally demanding to be effective in a real-time CTC control algorithm [8]. Therefore a simpler model is required. Alternatively if the mechanism itself can be designed so that its dynamic behavior is simpler then the problem of control will be simplified.

It is interesting to note that there has been lack of study in the current literature on compliant mechanisms in terms of integrated design and control of such systems with particular attention to decoupling of active joint motions or actuations. Some studies on this direction in the case of conventional multi-DOF mechanisms have been performed [16].

3. EXPERIMENTAL STUDY

The research conducted here used an experimental set up to understand the dynamics of the compliant mechanism as shown in Figure 1. The work reported herein only considered the quasi-static response of the system. This means that motion frequency generated from PZT actuators is very low, ~1 Hz.

3.1 Experimental Set-up

The micromanipulation stage developed at the AEDL consists of 3 Tokin AE0505D16 PZT stack actuators assembled into a flexure hinge based compliant mechanism, as shown in Figure 1. This mechanism provides the end-effector with 3 DOF in a plane. Each unloaded actuator has a maximum displacement of approximately 18μm. These PZTs are each powered by a Piezosystem Jena ENT 400/20 power supply coupled to an ENV 400 amplifier module. Each power unit provides a 60W bi-polar output with voltage range of –10 to 150V. Strain gauges are fitted to the PZTs to determine their displacement. The PZT's are each wired to a Measurements Group 2120B strain gauge conditioner, which are coupled to a 2110B power supply. The amplifiers and strain gauge conditioning circuitry are connected to a dSPACE DS1102 DSP controller board via built-in DAC's and ADC's. A schematic of the experimental set-up is shown in Figure 2.

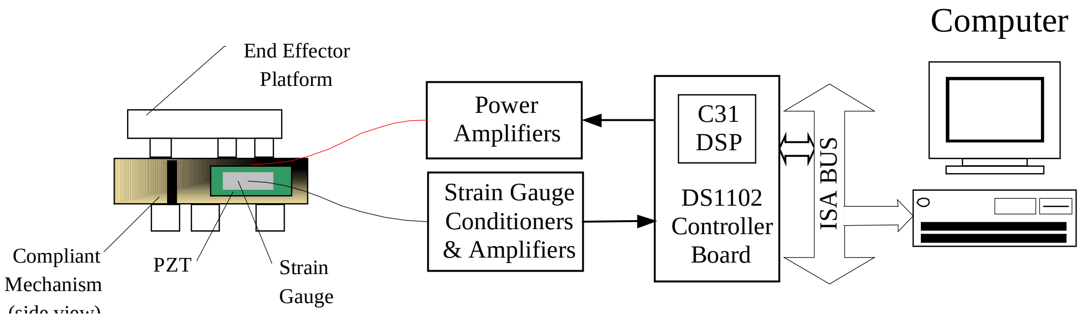

Figure 2 - Schematic of the experimental set-up

3.2 PZT Actuator Coupling Experiment - Method and Results

The quasi-static response of the amplifiers, PZTs and mechanism was of interest for this series of experiments. Therefore the stiffness coupling only will be considered. The input signal used throughout is a 1Hz triangular wave. The theoretical results suggest that if one PZT is displaced then static forces will be transmitted to the other PZTs by the stiffness of the mechanism. To observe the stiffness coupling of the compliant mechanism an input voltage was applied to PZT 1, while zero voltage was applied to the other two PZTs. The output displacement of all three PZTs was recorded. From this data a plot of displacement over time was obtained, see Figure 3. This procedure was then repeated, but while the input to PZT 2 remained zero the input to PZT 3 was 90V. The plot of displacements is also shown in Figure 3.

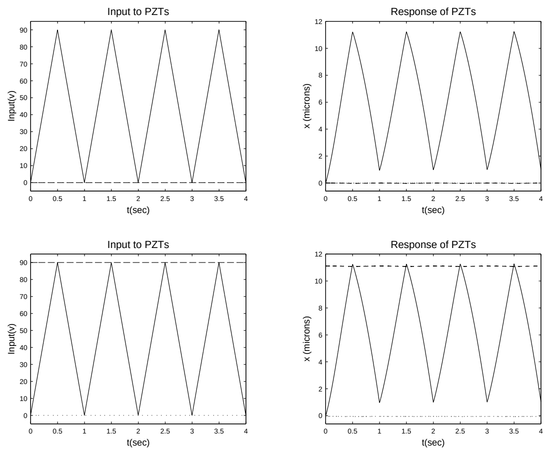

Figure 3 - Plot of input voltages and output responses of PZTs. Solid line is PZT 1, dotted line is PZT 2, dashed line is PZT 3

From the plots of Figure 3 it can be observed that when one PZT is displaced the resulting displacement of the others is negligible, regardless of their position. The maximum displacement of an un-actuated PZT was 0.03µm. This is 0.27% of the displacement of the actuated PZT. This indicates that the mechanism transmits minimal static forces from one PZT to another. Static forces acting on the PZTs are due to the stiffness of the mechanism. Thus it was concluded that the stiffness coupling of the mechanism is negligible. It was felt that for the sake of modeling the mechanism stiffness could be considered un-coupled. The magnitude of error introduced by making this assumption is minimal, while the computational efficiency of this simplified uncoupled model is far greater.

3.3 System Modeling

The theoretical compliant mechanism model [8] suggested that a non-linear, 3x3 matrix should represent the stiffness of the mechanism. However, the experimental results outlined above suggest that a 3 element diagonal matrix is more appropriate to describe the stiffness. It was then proposed that a simple linear relationship might effectively model the stiffness force generated by the mechanism at each PZT.

A Matlab model was developed to simulate the micromanipulation system. The amplifier was modeled by a first order transfer function, with experimentally determined parameters. The strain gauge conditioner, ADC and DAC were modeled as gains. To simulate the PZT a model developed by Goldfarb and Celanovic was used [15]. This uses a generalized Maxwell model, where the static hysteresis is identified as energy storage coupled to rate-independent dissipation and is represented by a generalized elasto-slip model. The hysteresis fitting parameters of this PZT model are determined from experimental data. The compliant mechanism stiffness was modeled as three independent linear springs with stiffness k_i (i=1,2,3).

3.4 Model Verification Experiment – Method and Results

The amplifier and PZT models first needed to be verified. PZT 1 was removed from the compliant mechanism and input waves with maximum voltages of 30, 60 and 90 V were applied. The output data was then used to determine the hysteresis fitting parameters of the PZT model. The experimental data obtained from PZT 1 and the corresponding simulation results are shown in Figure 4.

The compliant mechanism was then incorporated into the system model. The simulation was run several times and the value of the linear spring, k_1, varied until the simulation response matched that of the experimental data for PZT 1 shown in Figure 3. A final value of k_1 = 4.8e6 gave simulation results close to the experimental results. PZT 1 was replaced into the compliant mechanism. Input waves with maximum voltage of 30, 60 and 90 V were then applied to PZT 1. The experimental and simulation responses are shown in Figure 4.

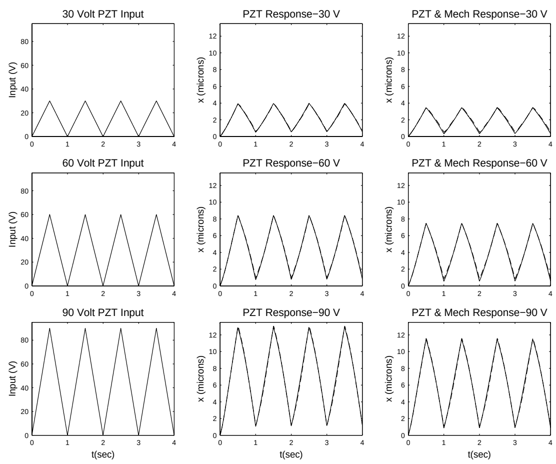

Figure 4 - Plot of input voltage and output displacement of PZT 1 when PZT is stand alone and when assembled into the compliant mechanism. The solid line is experimental data, dashed line is simulated results.

It can be seen from Figure 4 that the simulation results are close to the experimental results. This indicates that the un-coupled, linear spring model is appropriate to describe the stiffness behavior of this particular compliant mechanism.

4 Conclusions and Future Work

The experimental work conducted has revealed that the stiffness behavior of this particular parallel compliant mechanism is simpler than expected based on theoretical study or the findings of other researchers using different compliant mechanisms. It was found that the stiffness of the mechanism could be considered un-coupled, which is a surprising and useful result. Further, it was proposed that the un-coupled stiffness might be represented by linear springs acting independently on each PZT actuator. Through simulation and experimentation it was found that this representation was appropriate. These results lead to a computationally simple model of the stiffness of the mechanism. This model representation is suitable to be used as a term in a real-time computed torque controller.

The results of this experimental work also suggest that dynamic modeling without appropriate consideration of the physical property of the PZT actuators could lead to errors such that true physics of the system may not be presented [8].

Further work is required to determine simple inertia and damping terms for the compliant mechanism model so that a computed torque controller can be implemented.

Future work will also be conducted to determine what physical configurations of parallel compliant mechanisms result in un-coupled stiffness. It is intended that a design methodology should be developed that can be used to design parallel compliant mechanisms that posses simple un-coupled dynamics. Thereby controller design can be simplified and control performance improved through structural design.

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