Influence of Actuator Dynamics on Passive Haptic Interface Performance

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Abstract— **A dynamic simulation of a passive haptic interface is enhanced by the addition of an actuator model. The actuator of this haptic display (robot) is a controlled friction device. The robot utilizes four friction clutches as passive actuators. Stick-slip effects within the clutches and dynamic response are critical characteristics that are modeled. Experiments presented justify the model. The simulation is then used to evaluate possible system modifications with the goal of improving the path-following performance of the robot. An actuator modification is studied, as well as the addition of torque feedback into the system controller.**

Keywords— **haptics, passive robotics, friction modeling, dynamic simulation.**

I. Introduction

THE amount of research involving haptic interfaces—devices that provide force feedback to a human user—has bloomed in recent years. The haptic display has many applications in a wide range of fields, including teleoperation, physical rehabilitation, design visualization, and virtual reality. One area that has not been widely studied, however, is that of passive haptic devices. These are devices which contain actuators that may only remove energy from the system. Such a device would be useful in situations where safety has high priority.

In order to study the performance of this class of devices, researchers at Georgia Tech have developed PTER— the Passive Trajectory Enhancing Robot. PTER is a planar robotic arm in a five-bar parallel linkage configuration (see Figure 1.) Four electromagnetic friction clutches comprise PTER's set of actuators, yielding an overactuated system. Energy is input to the system by the user, who applies force to the handle on the tip of the arm. PTER in turn applies reaction forces to the user's hand.

The construction of PTER allows controlled frictional coupling of the two axes of the device. Two clutches couple each axis independently to ground and two others couple the axes to each other, either directly or through gearing which inverts the relative axis velocity. A more complete description of PTER's construction and operation can be found in [3].

The range of haptic sensations is inherently restricted when one chooses to use a passive haptic display. The primary focus of previous research involving PTER has been tip trajectory guidance. A secondary objective is to minimize the magnitude of the generated clutch torques in order to provide a smoother feel to the user. It is felt that PTER's performance in both of these areas is greatly influenced by the clutches themselves— specifically by nonlinear friction properties, i.e. stiction and stick-slip effects. In order to improve the performance of the system, a simulation of these effects would be desirable. The relative simplicity of a single brake model belies the complexity involved when four brakes interact.

This paper focuses on the design and implementation of a physical clutch model for use in a preexisting dynamic simulation, and the application of the resulting simulation to identify possible system modifications that could improve the path-following performance of PTER. Changes in friction materials as well as a modified control strategy are considered.

II. Definition of the Clutch Model

The original Davis simulation [2] contains a basic physical model for the clutches. In short, when the controller desires a certain torque from a certain clutch, the simulation assumes that the desired torque will immediately be

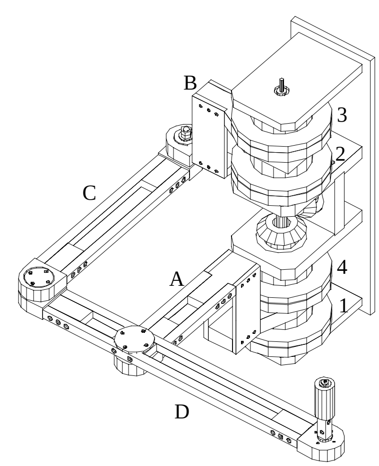

Fig. 1. *PTER with clutch numbers and link letters*

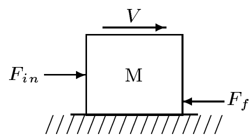

Fig. 2. *Simple One-Mass Friction System*

generated by the clutch as long as the magnitude is within physical limits and the sign does not violate the passive constraint (i.e., energy is not being added to the system.)

This kind of model is sufficient to study control characteristics for an idealized system, but considering that friction effects and dynamic response have significant influence on PTER's performance, it was felt that a better clutch model would improve comparisons to the experimental results.

A. Friction Model

After a review of existing numerical friction models, the Karnopp model [5] was selected. Stick-slip friction is a discontinuous phenomenon. It consists, however, of two separate modes, each of which is piecewise continuous for a specific system variable. The applicable mode at any point in time depends on whether or not there is relative velocity between the two friction surfaces. The Karnopp model utilizes different sets of governing equations for each of these modes. This provides a practical means of dealing with the friction force discontinuity at zero relative velocity. For a simple single-mass system such as that in Figure 2, the Karnopp model yields the following equations for frictional force F_f:

$$F_f = \begin{cases} g(V) & : \quad |V| \geq \delta V \\ F_{in} & : \quad |V| < \delta V, F_{in} \leq F_B \\ F_B & : \quad |V| < \delta V, F_{in} > F_B \end{cases} \tag{1}$$

where F_B is the breakaway force— the force required to overcome static friction and initiate movement.

When the magnitude of the relative velocity V between the two surfaces is greater than or equal to a very small value δV, the system is said to be in the *slip* mode. Even though the system is physically slipping whenever V is not equal to zero, the nonzero region around $V = 0$ is defined in order to account for close-to-zero errors. In the slip mode the friction force is dependent only on the relative velocity, and is determined by an arbitrary function $g(V)$.

When the magnitude of the relative velocity V between the two surfaces is less than δV, the system is said to be in the *stick* mode. In the stick mode the system is static, and the friction force F_f exactly cancels the driving force F_{in}, unless F_{in} exceeds the breakaway force F_B. In the latter case F_f is equal to F_B and the body will experience nonzero acceleration. After a short interval the magnitude of V will exceed δV and the model will transition from the stick mode to the slip mode.

B. Implementation of Clutch Dynamics

A first-order transfer function closely relates clutch input voltage to applied normal force. This model was implemented in the simulation. The gain and time constant were obtained through component tests and manufacturer's data. The Karnopp model was modified specifically to simulate stick-slip friction in PTER's clutches. The equations were transformed from linear to angular coordinates. Variable normal force was introduced, dependent on the input voltage of each clutch. Also, the model was duplicated four times in order to represent each of PTER's four clutches. After these modifications the actual governing equations for PTER's clutches were:

$$\tau_{f,x} = \begin{cases} g_x(\omega, v) & : \quad |\omega| \geq \delta\omega \\ \tau_{in,x} & : \quad |\omega| < \delta\omega, \tau_{in,x} \leq \tau_{B,x}(v) \\ \tau_{B,x}(v) & : \quad |\omega| < \delta\omega, \tau_{in,x} > \tau_{B,x}(v) \end{cases} \tag{2}$$

where $x = 1 \ldots 4$ represent the four clutches, and v is the input voltage to each clutch.

III. Enhancement of the Dynamic Simulator

A. Model Implementation

The simulation utilizes a position error impedance controller to attempt tracking control of the desired path. The distance between the robot endpoint and the desired path is used as the error signal. Desired endpoint forces are computed by simulating a spring and a damper between the endpoint and the desired position normal to the desired path, and a damper tangent to the path. It is important to note that these computed forces are desired forces, and may or may not be achieveable due to the passive constraints on the actuators. The desired endpoint forces are then transformed into a set of achievable desired clutch torques which match the desired values as closely as possible. A piecewise linear model of torque vs. input voltage for the clutches similar to the model that is used in PTER's actual control system was added to the controller. This allows the controller to output a control signal that could then be utilized by the new dynamic clutch model to determine generated clutch torques.

The complexity of the new clutch model code requires a large case statement which determines in which of the six possible dynamic modes the system is in. The dynamic modes comprise a two degree-of-freedom mode (where there is nonzero relative surface velocity in all four clutches), four separate one degree-of-freedom modes (where there is zero relative surface velocity in a single clutch), and a zero degree-of-freedom mode (the entire system is static.) Depending on the current dynamic mode and the voltage applied to each brake, the case statement determines generated clutch torques based on the predefined models for $\tau_{f,x}$ as shown in Equation 2.

Computing generated clutch torques for clutches in the slip mode is straightforward— the torque is merely a function of relative plate velocity and clutch input voltage. However, the calculations for stuck clutches are more complex, as the generated torque in this case depends on the input torque. For single degree-of-freedom situations and zero degree-of-freedom situations where no more than two clutches are applied, the generated torques are derived from the torques necessary to keep stuck clutches in the stuck state. To implement this concept, any torques generated by clutches in the slip state are first computed, and then the equations of motion are solved with appropriate constraints on the angular accelerations of links A and B.

The equations of motion of the system are

$$\begin{aligned} M_{11}\ddot{\theta}_A + M_{12}\ddot{\theta}_B + V_1 &= \tau_1 + \tau_3 + \tau_4 + \tau_{A,ext} \quad (3) \\ M_{21}\ddot{\theta}_A + M_{22}\ddot{\theta}_B + V_2 &= \tau_2 - \tau_3 + \tau_4 + \tau_{B,ext} \quad (4) \end{aligned}$$

where the M_{xy} values represent PTER's inertial matrix, $\ddot{\theta}_x$ is the angular acceleration for link x, the V_x values account for velocity-dependent effects, τ_x is the torque generated by clutch x, and $\tau_{x,ext}$ is the resultant torque on link x due to the tip input force.

This strategy does not work when more than two clutches are applied in zero degree-of-freedom situations, for the equations of motion become statically indeterminate. In this case, a strategy termed the *lumped actuator approach* is utilized. The four clutches are considered as an entire actuation system, and rather than solving for each individual generated clutch torque, the model determines whether or not the system of clutches is capable of keeping the system fully static. If a configuration of clutch torques exists that will both keep the system fully static and are below the stick-slip transition levels, then it is assumed that the system will remain fully static. If no such configuration exists, the weakest clutch in the given configuration is assumed to transition to the slip mode.

B. *Performance of New Clutch Model*

In order to validate the operation of the new clutch model, two separate tests were carried out. The first was a series of single degree-of-freedom tests. In these simulations, a ramped input torque is applied to a specific arm. Test results for clutch 1 are shown in Figures 3 and 4. In this test, a ramped input torque is applied to link A and no torque is applied link B. A small input signal is applied to clutch 1 and a large input signal to clutch 2. The dashed line in Figure 3 is a single degree-of-freedom line— it represents the full range of motion of the tip of the robot when clutch 2 is locked.

Two effects are to be noted. The first is the proper functioning of the stick-slip friction model. The ramped input torque is effectively 'absorbed' by clutch 1 up to the breakaway level (63.5 in·lb in this case). Once the breakaway level is reached and the non-zero net torque on link A causes the link velocity to rise above the Karnopp limit $\delta\omega$, the link starts to move and the torque generated by clutch 1 reverts to the dynamic value, which is lower than the static value. The second effect is the fact that the robot endpoint does indeed follow the single degree-of-freedom line. It can also be seen that the torque applied by clutch 2 increases as the velocity of the endpoint increases. Even though no torque is externally applied to link B, the V_2 term in Equation 4 exerts torque on link B through the non-zero velocity of link A.

The implementation of the clutch dynamics was tested by feeding a constant tip force into the simulation, along with a step input to a single clutch. This configuration behaved as expected, generating a first-order step response for the generated clutch torque.

IV. Evaluation of System Configurations

A. *Test Definition*

A standard test was defined in order to compare different system configurations. A straight path spanning the width of the workspace was defined to which the controller tries to constrain the tip of the robot. The input force consists of two components. The first is a constant force that is always tangent to the desired path. The second is normal to the desired path with a value F_n as defined in Equation 5.

$$F_n = \frac{F_{max}}{5} \sum_{n=0}^{4} \sin(2^n \pi t) \quad (5)$$

The normal force F_n models an input disturbance to the controller. Several frequencies were combined so that cross-effects between a specific frequency and the clutch time

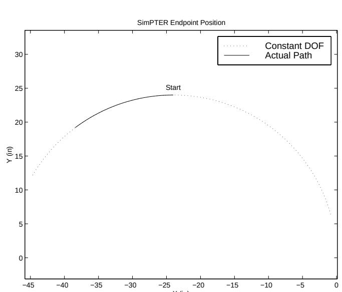

Fig. 3. *Clutch 1 Single DOF Test— Endpoint Path*

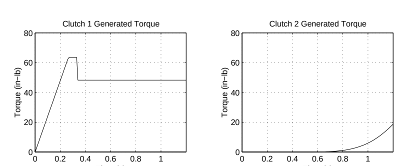

Fig. 4. *Clutch 1 Single DOF Test— Clutch Torques*

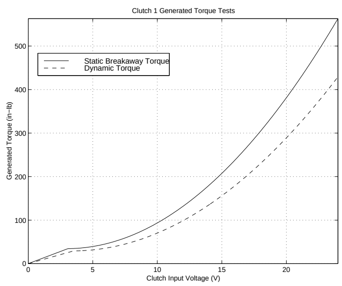

Fig. 5. *Experimental Torque Models— Clutch 1*

constants in the simulation would be reduced. It was desired that the test results reflect on the overall performance of the system— not only on the performance given a particular set of input conditions. A simple input force was chosen even though a model of the human user would be more representative of the final application. It is possible, however, to run repeatable experiments with a force input for eventual comparison to system simulations.

B. Simulation Configurations

B.1 Baseline Tests

After implementing the new dynamic actuator model, the simulation was used to evaluate the effects of two separate system modifications. One of these involved physical modification of the actuators, while the other was a change in controller software. In order to evaluate changes in system performance due to these proposed modifications, a baseline simulation run was defined.

The baseline run is based on the original configuration of PTER. An experimental testbed was built in order to measure generated clutch torque versus clutch input voltage. An aluminum beam was fitted with strain gauges and attached to clutch number 1. A constant input voltage was applied and force was manually applied to the end of the beam until the brake slipped. The tester then continued to apply force to the beam in an attempt to move it at a constant velocity. During these tests, a PC with a data acquisition card running LabVIEW took strain gauge measurements.

These tests allowed the construction of models for both breakaway torque and dynamic torque versus clutch input voltage. In order to convert the experimental results into a form suitable for simulation, a curve fit was made. A combination linear and quadratic segment was used as appears in Figure 5. This function is efficient and accurately represents the data. Accurate tests could not be performed below an input voltage of 4 volts, and the static residual breakaway torques and dynamic torques in the system with no clutches engaged are assumed to be zero since the actual values are unmeasureably small for this setup.

B.2 Modified Dynamic Model Tests

It was felt that by replacing or modifying the clutches to improve stick-slip behavior or response time, the performance of the system could be augmented. A possible modification to this end is the replacement of the clutches' friction material with a more suitable one. Delrin on steel was identified as an alternative friction interface. Even though Delrin has lower friction coefficients than the present clutches, its friction behavior would seem better suited in a device where stick-slip effects predominate. This is because Delrin has a higher dynamic coefficient of friction than static. Based on frictional coefficients published for both Delrin ($\mu_s = 0.20$ and $\mu_k = 0.35$ for Delrin 100) and PTER's current clutches ($\mu_s = 0.45$), friction models were constructed for a hypothetical Delrin clutch by scaling the models developed for the existing clutches.

Research is currently considering a new Delrin based brake. The above-reported friction parameters for Delrin have not been observed in preliminary experiments, it should be noted. Both the experimentally determined values and values from the literature have been used in the simulation.

B.3 Torque Control Feedback Tests

The original impedance controller as described in section III-A uses a look-up table to determine the voltage commands to issue to each clutch. Unfortunately, it is difficult to accurately model the torque versus input voltage profile of friction clutches. A wide array of environmental conditions including ambient temperature and relative humidity influence the clutch response. In order to improve the performance of the controller, it was suggested that a torque feedback loop be added to the existing controller. Although torque measurement hardware did not originally exist on PTER, a project to retrofit the clutches with torque sensors is underway.

A separate proportional feedback loop was added to the controller for each clutch. The error signals are the differences between the desired torques and the generated torques. Figure 6 is a diagram of the torque controller. The system variables are as follows: $x(t)$ is the complete dynamic state of the system, $y(t)$ and $\bar{y}(t)$ are clutch command signals, τ_d is the desired clutch torque vector, and τ_a

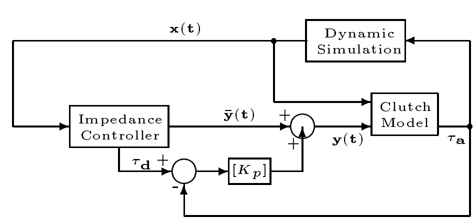

Fig. 6. *Controller with Torque Feedback*

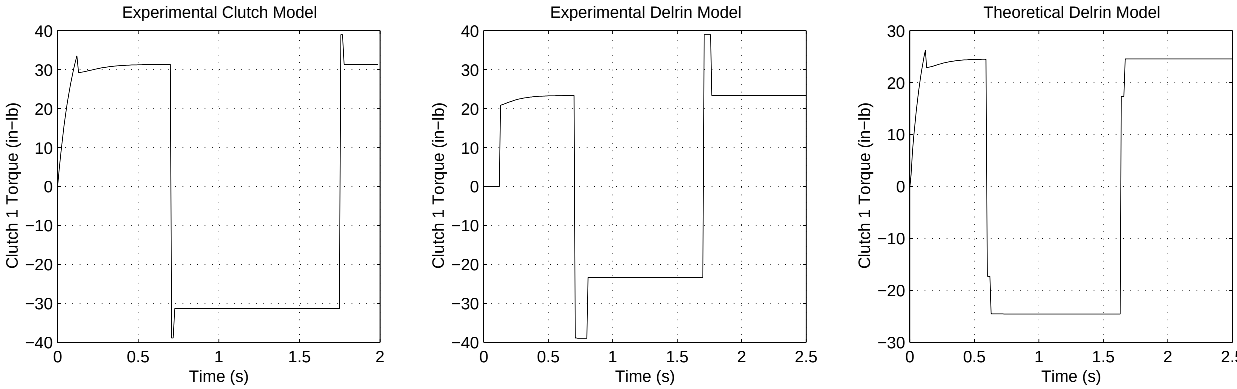

Fig. 9. *Acceleration Performance of the Different Clutch Models*

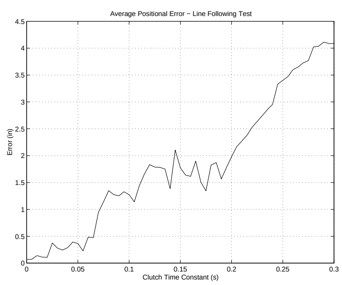

Fig. 7. *Baseline Runs - Average Error*

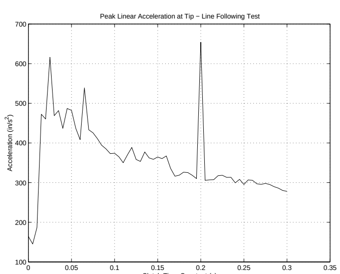

Fig. 8. *Baseline Runs - Peak Linear Acceleration*

is the actual clutch torque vector.

C. Analysis

C.1 Baseline Tests - Results

The experimentally derived baseline model was inserted into the simulation and runs were made for differing values of the clutch time constant. Several trends were observed. The first was the expected behavior of the path-average position error tending to increase with a higher brake time constant (see Figure 7.) Another expected trend is shown in Figure 8; the peak linear acceleration of the tip of the robot decreases with increasing time constant. Other notable trends include increasing total path length and increasing average tip velocity, both with increasing time constant. These trends verify the successful operation of the clutch dynamics in the simulation.

Model	Avg Position Error (in)	Avg Tip Velocity ($\frac{in}{s}$)	Avg Tip Accel ($\frac{in}{s^2}$)
Baseline	1.794	19.69	64.74
Delrin– Exp	1.796	33.02	137.2
Delrin– Theory	1.498	33.4	144.2

TABLE I

Clutch Model Comparison

C.2 Modified Dynamic Model - Results

Two more sets of simulation runs were performed, again with differing time constants, for the two new Delrin models. Table I summarizes the average values of several variables over all of the test runs.

As stated above, the experimental data gathered from the Delrin brake testbed did not match expectations, exhibiting stick-slip behavior similar to the baseline clutches. Because of this, the similar average position errors for the baseline and experimental Delrin models are not surprising. The experimental Delrin model's higher average tip velocity and acceleration is likely due to the fact that the Delrin brake models in the simulation exhibit lower peak friction capability than the baseline model, due to the lower coefficients of friction possessed by Delrin compared to the baseline friction material.

The theoretical Delrin model (exhibiting higher dynamic friction than static friction) performed better in path following over the range of time constants, despite its lower peak friction capability. It shares the higher average tip velocities and accelerations of the experimental Delrin model for the same reasons stated above.

It was expected that the theoretical Delrin model would exhibit lower tip acceleration performance, which would improve the tactile "feel" of our device (the secondary goal stated in the Introduction.) As has been stated, the higher average accelerations exhibited in the above tests was expected to be due to lower maximum torque capability. Fur-

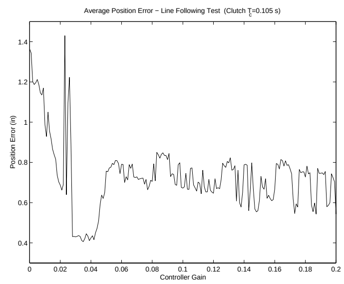

Fig. 10. *Effect of Proportional Feedback Gain on Position Error*

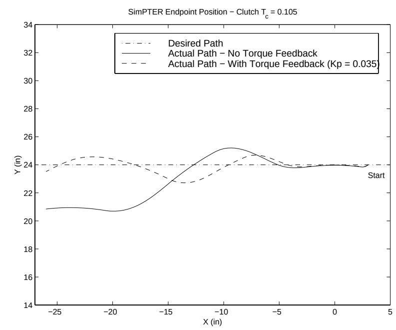

Fig. 11. *Line Following Test*

ther tests were performed to clarify the acceleration performance of the theoretical Delrin model. Figure 9 shows the generated torque of clutch 1 with a constant input voltage when a step force is applied to the tip of the robot. The experimental clutch (baseline) model and experimental Delrin models exhibit a similar jump in generated torque when the direction of motion changes (a slip-stick-slip transition.) The theoretical Delrin model, however, exhibits no such jump. This indicates that a Delrin brake exhibiting frictional characteristics similar to those contained in the literature would indeed exhibit lower jerk, and as a result have a smoother tactile feel to the user.

C.3 Torque Control Feedback - Results

The simulation was again run for several values of the controller gain, K_p. The time constant for each run was set at 0.105 seconds, which is the time constant listed by the manufacturer for PTER's clutches. Figure 10 is a plot of average positional error versus controller gain.

The simulation indicates clearly that a significant gain in path-following performance can be achieved with this type of controller. Performance remains relatively constant for gains above 0.1, as this is the point at which the controller begins to saturate the actuators. Figure 11 illustrates the increased line-following performance of the system when the torque controller is utilized with a gain of 0.06 (the optimum gain for path error minimization before significant actuator saturation occurs.)

V. Conclusion and Future Work

After the simulation was enhanced by adding an actuator model, the effects of two possible system modifications were examined. Preliminary tests provide data indicating that a Delrin based clutch could increase the path following performance of PTER if a device could be designed that exhibits a higher dynamic than static coefficient of friction. Although simulated path following tests showed high average tip acceleration, further tests have shown that stick-slip-stick transitions in a Delrin clutch should have a smoother tactile feel than the present clutches. The discrepancy between these two results likely lies in the fact that the Delrin model has lower maximum torque capabilities than the baseline model, which results in higher velocities and accelerations when the system attempts to compensate for large tracking errors. It appears that implementing torque feedback control in PTER's controller would be a way to improve the path following performance of the device while utilizing the present clutches.

Future plans include the construction of a motorized testbed to perform system identification on one of PTER's clutches in hopes of better modeling its friction characteristics and time constant. Such a testbed would also allow direct comparison of experimental and simulated clutch behavior. Another clutch will be retrofitted with Delrin as its friction material and tested in the testbed. These tests should provide better modeling data for the simulation. A modified clutch with integrated torque sensing capability will also be used, and feedback torque control will be attempted. These tests will be performed with the aim of enhancing the performance of PTER through the installation of improved clutches, improvement the accuracy of the simulation for the purpose of controller and clutch evaluation, and implementing improved controllers on PTER.

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