Actuator Design for a Passive Haptic Display

Lawrence Tognetti

Georgia Institute of Technology
July, 1999

Research Supported by National Science Foundation Grant IIS-9700528
Overview

- Background
- Description of existing actuators and their deficiencies
- Alternative clutches / brakes
- Redesign of PTER’s existing Dynacorp brakes / clutches
- Calibration of the new torque sensor
- Digital feedback control, experimental & simulation
- Conclusion
- Future work
- Questions
Background

Passive Trajectory Enhancing Robot (PTER)

- PTER uses friction brakes to constrict motion for simulation of haptic features.
- Over actuated
PTER’s Existing Brakes / Clutches

- Industrial electromagnetic friction units from Dynacorp
  - Model 310
  - Original max torque 300 ft-lbf (407 N-M)
  - Rated time constant 0.105 sec (coil build up)
- Modified to eliminate metal to metal contact
  - Reduced available torque
- No provisions of measuring actual applied torque for feedback control
Dynamic Response of PTER’s Actuators

- Open Loop Control
- Undesirable Dynamics
  - Non-linear electromagnet
  - Sliding on pins (binding)
  - First order response
    - R-L circuit
  - Pure time delay
    - Coil build up to attract armature plate
  - Steady state error
    - Each clutch’s output torque different
    - Max torque ranged from 15 to 125 Ft-Lbf, depending on unit

(Borrowed from Gomes, 1997)
Alternative Brakes / Clutches

- **Hysteresis**
  - Low torque ratings \(2.5\ \text{oz-in} (0.013\ \text{lbf-ft}) -- 3200\ \text{oz-in} (16.67\ \text{lbf-ft})\)
  - Cogging
  - Hysteresis in actuation curve

- **Eddy Current**
  - Torque dependent on both slip velocity and applied current
  - Large scale applications

- **Magnetic Particle**
  - Required to turn a full revolution for redistribution of magnetic particles (for consistent operation)

- **Electro-Rheological**
  - Not much information available
Electromagnetic Friction

- Metal to metal contact for completing the magnetic circuit
- Friction material is only used for stabilizing torque
- Max torque
  - 5 oz-in (0.026 lbf-ft) to 465 lbf-ft
- Response consists of both mechanical and electrical (applied voltage)
- Dynamic times depend on size of the unit (Kebco)
  - Pure time delay: $t_{11} = 0.004 - 0.055$ sec.
  - Rise time: $t_{12} = 0.006 - 0.240$ sec.
  - Engagement time: $t_1 = 0.01 - 0.295$ sec.
New Design Concept With Torque Sensor

- Spoke transmits torque from friction
- Spoke locates and supports armature plate
- Spoke deflects under vertical engagement and torque transmission
- Strain gauge measurement proportional to transmitted torque
Modified Clutch Layout (Top View)
Modified Clutch (Section View)
Spoke Def - Bod Model

Torque Bending

Axial Deflection
Integration Constants (Torque Bending)

\[
\begin{bmatrix}
0 & R & -1 & 0 & 0 & 0 & 0 \\
\frac{-1}{2} \left(2 \cdot a1 \cdot R + a1^2\right) (E2:\text{I2}) & \frac{1}{(E2:\text{I2})} & 0 & \frac{-1}{(E3:\text{I3})} & 0 & 0 & 0 \\
\frac{-1}{6} \left(3 a1^2 \cdot R + a1^3\right) (E2:\text{I2}) & \frac{a1}{(E2:\text{I2})} & \frac{1}{(E2:\text{I2})} & 0 & \frac{-1}{(E3:\text{I3})} & 0 & 0 \\
\frac{-1}{2} \left(2 b \cdot R + 2 \cdot b \cdot a1 + b^2\right) (E3:\text{I3}) & 0 & 0 & \frac{1}{(E3:\text{I3})} & 0 & \frac{-1}{(E4:\text{I4})} & 0 \\
\frac{-1}{6} \left(3 b^2 \cdot a1 + b^3 + 3 b^2 \cdot R\right) (E3:\text{I3}) & 0 & 0 & b & \frac{1}{(E3:\text{I3})} & 0 & \frac{-1}{(E4:\text{I4})} \\
\frac{-1}{2} \left(2 a2 \cdot L + 2 \cdot a2 \cdot R - a2^2\right) (E4:\text{I4}) & 0 & 0 & 0 & 0 & \frac{1}{(E4:\text{I4})} & 0 \\
\frac{-1}{6} \left(3 a2^2 \cdot R - 2 a2^3 + 3 a2^2 \cdot L\right) (E4:\text{I4}) & 0 & 0 & 0 & 0 & \frac{a2}{(E4:\text{I4})} & \frac{1}{(E4:\text{I4})}
\end{bmatrix}
\times
\begin{bmatrix}
F \\
\frac{-1}{2} a1^2 \cdot T (E2:\text{I2}) \\
C2 \\
\frac{-b \cdot T}{E3:\text{I3}} \\
D2 \\
\frac{-1}{2} b^2 \cdot T (E3:\text{I3}) \\
C3 \\
\frac{-1}{2} a^2 \cdot T (E4:\text{I4}) \\
D3 \\
\frac{-1}{2} a2^2 \cdot T (E4:\text{I4}) \\
D4
\end{bmatrix}
\]

12
**Electromagnet Model**

**Electric Current, i**

**Magnet Force**

**Coil**

**x**

---

\[ \text{Force} \times \delta X = \delta \text{Energy} \]

\[ \text{Force} = \frac{d(E)}{dx} = -\frac{1}{2} \frac{L_0}{a} \frac{1}{(1 + x/a)^2} i^2 \]

Using Principle of Virtual Work

\[ \text{Power} = \nu \times i \]

\[ L = L_1 + \frac{L_0}{1 + x/a} \]

\[ \text{Energy} = \frac{1}{2} Li^2 \]

\[ a \approx 0.0113, \ L_0 \approx 11.298 \]
Design Considerations

- Available magnetic force
- Cyclic fatigue
- Physical size constraints
- Sensor sensitivity
- Material selection
  - Spokes
    - Delrin 100P
    - Steel ASTM-A514
    - Aluminum 7075-T651
  - Armature Plate
    - Low carbon steel (under 1010)
- Alternative friction material
Torque Bar Calibration

- Strain measured with strain gauges in a half bridge
- Simple beam theory used to relate applied torque
- Spring scale used to apply approximate known torque
- Measurements compared, found to be sufficiently close
Clutch’s Sensor Calibration

- Strain in spokes measured by strain gauges in a full bridge
- First order equation fitting strain to measured torque from torque bar
- Model predicted a scaling factor of $3.361 \times 10^5$
  - 9.94% discrepancy from actual
- Model’s predicted spring constants
  - Axial -- $1.77 \times 10^3$ lbf/in
  - Radial -- 2,217 lbf-in/deg

\[ \tau = 3.057 \times 10^5 \varepsilon_{spoke} + 4.818 \text{ (in - lbf)} \]
Torque vs. Current Calibration

(Quadratic & Linear) \[ \tau = 181i^2 + 111i - 76 \]  \[ \text{MSE} = 148 \ (4.1) \]

(Quadratic) \[ \tau = 232i^2 - 20 \]  \[ \text{MSE} = 164 \ (23.4) \]

(Linear) \[ \tau = 502i - 269 \]  \[ \text{MSE} = 340 \ (207.6) \]

(Quadratic, No Offset) \[ \tau = 221i^2 \]  \[ \text{MSE} = 238 \ (126.5) \]
Digital Control

\[ i = f(\tau) \]

Inverse of Non Linear Torque Equation

\[ \tau = ai^2 + bi + c \Rightarrow k_{clutch} = \frac{d\tau}{di} = 2ai + b \]

\[ \tau = \tau_{eq} + \frac{k_p k_{clutch}}{1 + k_p k_{clutch}} (\tau_d - \tau_{eq}) + \frac{1}{1 + k_p k_{clutch}} \tau_{dist} \]

- Nonlinear feed-forward based on quadratic torque mapping
- Absolute value of torque fed back (direction insensitive)
- Proportional control based on error
- Larger gain equates to larger disturbance and error rejection
Experimental Results

• Labview processing controller at a non-deterministic 50 Hz ($t_s \approx 0.02$ sec)
• Small gains stabilized torque
• Large gains caused system to go unstable
Power Supply Model

- Second order response
  - $t_r \approx 0.056$ sec
  - $\%OS \approx 25\%$
  - $\omega_n = 61.25$ rad/sec
  - $\zeta = 0.40$

$PS(s) = \frac{I(s)}{I_c(s)} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} = \frac{3751}{s^2 + 49s + 3751}$

- Voltage applied to a first order system (RL Circuit)
  - $R \approx 11.68\Omega$
  - $L \approx 1.19\text{H}$

- Voltage saturation: 0 - 36 volts
Power Supply Data vs. Simulation

![Graphs showing commanded vs. actual current and resulting voltage for power supply and simulation](image)
Digital Stability -- Gain Margin

- \( i \approx 0.9 \) amps for unstable experiment
  - \( k_{\text{clutch}} = 435 \text{ in-lbf / amp} \)

- \( ts = 0.02 \) (50 Hz)
  - Gain margin = 5.011 db
  - Max system gain: \( k_p k_{\text{clutch}} = 1.78 \)
  - Max controller gain: \( k_p = 0.0041 \text{ amps / in-lbf} \)
  - Error & disturbance rejection to 36%

- \( ts = 0.002 \) (500Hz)
  - Gain margin = 22.475 db
  - Max system gain: \( k_p k_{\text{clutch}} = 13.30 \)
  - Max controller gain: \( k_p = 0.0306 \text{ amps / in-lbf} \)
  - Error & disturbance rejection to 7%
Simulations for Regulating @ 250 in-lbf

- Increased sampling rate allows increased proportional gain and increased error rejection
- Strong evidence that instability is caused by relatively slow controller and quick second order underdamped power supply dynamics

( $t_s = 0.02$ (50 Hz), $k_p = 0.0041$ )

( $t_s = 0.002$ (500 Hz), $k_p = 0.02$ )
Digital PD Control

\[ C(z) = k_p + k_d \frac{z-1}{z} = (k_p + k_d) \frac{z-\alpha}{z} = K \frac{z-\alpha}{z} \]
Digital PD Regulating @ 250 in-lbf

\[ \begin{array}{cccccc}
\alpha & K & k_p & k_d & \text{Adjusted } k_p & \text{Adjusted } k_d \\
0.5 & 16.25 & 8.125 & 8.125 & 0.0163 & 0.0163 \\
0.7 & 38.75 & 12 & 28 & 0.024 & 0.056 \\
\end{array} \]

(Simulations)

\((\alpha = 0.5)\)

\((\alpha = 0.7)\)
Reference Tracking (Digital PD)

- $\alpha = 0.5$
  - $k_p k_{clutch} = 8.125$
  - $k_d k_{clutch} = 8.125$

- Online gain normalization based on estimated clutch gain from feed-forward actuation.
  (Simulations)
Stick / Slip of the Armature Plate

- Determine if the armature plate comes to rest, sticking then slipping, causing the instability or limit cycle
- Compare angular velocity of the Shaft hub with the angular velocity of the armature plate
- Numerically differentiate and filter data
- Armature plate appears to not come to rest while hub is moving

\[ \tau = k(\theta_1 - \theta_2) \Rightarrow \dot{\tau} = k(\dot{\theta}_1 - \dot{\theta}_2) \]

(Data from unstable experiment displayed in slide 19)
Closing Remarks -- Conclusion

• Industrial available clutches / brakes were not found appropriate for our application.
• Redesigned electromagnetic clutch incorporating a torque sensor for feedback control was developed and one prototype was built.
• More thorough modeling & testing of the new clutch and power supply combination.
• Feasibility and benefits of closed loop control were demonstrated.
• Power Supply dynamics became a dominant factor.
• Faster digital sampling required for effective disturbance rejection
Closing Remarks -- Future Work -- I

- Faster Digital Controller
  - dSPACE & Real Time workshop
  - Hyperkernal
- Online Parameter Identification
- Analog Control
- Alternative Friction Materials
  - Delrin: $\mu_s = 0.20$, $\mu_d = 0.35$
- Power Supply Upgrade
  - Bipolar
  - Larger current and voltage ratings
- Design Modifications
Closing Remarks -- Future Work -- II

- Manufacture three more units
- Upgrade other components of PTER
  - Replace position potentiometers (Encoders, Resolvers)
  - Tachometers
  - Remount strain gauges in force sensors
  - Computer and Software
- Programming of old and new haptic algorithms
- Bilateral control of Hurbirt
- Possible improvements to the design of PTER
Questions

• What do you mean?
• How can you justify that?
• I don’t agree!
• Why didn’t you…….?  
• Did you try………….? 
• Etc?
Digital Control of a fast Power Supply

\[ PS(z) = Z[Zoh \ast PS(s)] = Z \left[ \frac{1 - e^{-Ts}}{s} \ast \frac{1}{s/a + 1} \right] = (1 - z^{-1}) \ast Z \left[ \frac{1}{s(s/a + 1)} \right] = \frac{1 - e^{-aT}}{z - e^{-aT}} \]

\[ \lim_{a \to \infty} \left[ \frac{1 - e^{-aT}}{z - e^{-aT}} \right] = \frac{1}{z} = z^{-1} \quad \text{Max open loop gain is 1} \]

- With an open loop gain of “1”, disturbance rejection is limited to only 50%
- Systems dynamics become much quicker then the digital controller
Tuning of the RL circuit

- **Pro**
  - Addition of resistors and capacitors can tune the circuit for desirable second order dynamics

- **Con**
  - Limited by achievable output

- Increasing “R” increases damping and required voltage

- Large “L” causes low damping and low natural frequency

- Small “C” increases natural frequency.

\[
\frac{I_{\text{clutch}}(s)}{I_{\text{supply}}(s)} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} = \frac{1/CL}{s^2 + \frac{R}{L} s + \frac{1}{CL}}
\]
Stick / Slip Dynamic Model

\[
\begin{bmatrix}
I_{\text{shaft+Hub}} & 0 \\
0 & I_{\text{Armature}}
\end{bmatrix}
\begin{bmatrix}
\ddot{\theta}_{\text{shaft+Hub}} \\
\dot{\theta}_{\text{Armature}}
\end{bmatrix}
+ \begin{bmatrix}
B_1 + B_2 & -B_2 \\
-B_2 & B_2
\end{bmatrix}
\begin{bmatrix}
\dot{\theta}_{\text{shaft+Hub}} \\
\dot{\theta}_{\text{Armature}}
\end{bmatrix}
+ \begin{bmatrix}
k & -k \\
-k & k
\end{bmatrix}
\begin{bmatrix}
\theta_{\text{shaft+Hub}} \\
\theta_{\text{Armature}}
\end{bmatrix}
= \begin{bmatrix}
\tau_{\text{in}} \\
\tau_{\text{friction}}
\end{bmatrix}
\]

- Account for braking force through the armature plate, compliant spoke, and to the hub.
- Karnopp model utilized for friction force.
- Damping added to account for viscous friction in the bearings & material damping in the spoke
- Stiff spoke and small armature plate results in a stiff system that Simulink can not successfully integrate.
- Fixed step simulations gave erroneous information
Industrial electromagnetic friction clutch / brakes (Dynacorp Model 310).

- Time constant for coil build up: \( \tau = 0.105 \text{sec} \) (given applied voltage)
- Original max rated torque = 300 Ft-Lbf (407 N-M)
- Max voltage: \( V = 24 \text{ volts} \)
- Max current: \( I = 2.376 \) amps
- Non asbestos friction material with coefficient of friction: \( \mu = 0.45 \)
- Modified to eliminate metal to metal contact (reduce max torque capabilities)
Hysteresis & Eddy Current

**Hysteresis**
- Max torque ratings
  - 2.5 oz-in (0.013 lbf-ft) -- 3200 oz-in (16.67 lbf-ft)
- No rubbing parts
- Hysteresis in the torque vs current mapping
- Cogging

**Eddy Current**
- Torque dependent on both slip velocity and applied current
- Large scale applications
Magnetic Particle

- Max torque
  - 5 oz-in (0.026 lbf-ft) to 738 lbf-ft
- Torque generated by shear and tensile stress between attracted magnetic particles
- Dynamic time constants ranging from 0.009 to 1.31 sec., depending on the size of the unit.
- Clean sealed operation
- Required to turn a full revolution to redistribute magnetic particles (for consistent operation)