Design of a Piezoelectric Impact Mechanism for Tap Centering

By
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And
Master of Science

Abstract

Many manufacturing processes require the centering of a round part on a rotating axis. Traditionally, this centering is done manually by an experienced machinist. This thesis investigates automated centering that mimics the manual technique using a novel piezoelectric impact mechanism. A prototype is built to solve a centering problem on a metrology machine used by The Timken Company. The device positions parts up to 90 Kg to micron precision with less than 3% error. The piezoelectric impact mechanism can reduce centering time to half that of conventional techniques and is suited for metrology and grind applications.

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Forward

This thesis was done as part of the Engineering Internship Program (EIP) in cooperation with the Timken Company in North Canton, Ohio. The work was conducted over a six month period beginning in June and ending in December 2002.

Several people at Timken contributed to my work and understanding of the material. Dave Flanders was my mentor who’s knowledge of and experience with the Universal Measuring Machine was invaluable. Mike Thomson, a consultant in the group, helped me understand the previous work done on centering on the UMM. Our brainstorming sessions were productive and enjoyable because our creativity and enthusiasm fed off each other. Joe Pack, the group Manager provided technical guidance and encouragement and helped me take advantage of the resources within the group. His leadership and extrapolation of ideas to broader applications was inspiring. My hope is that the people I had contact with at Timken enjoyed my time there as much as I did, and that the work summarized here proves useful to Timken.
# Table of Contents

1. Introduction................................................................. 6
2. Universal Measuring Machine and Grind Applications................. 6
3. The Benchmark: Manual Centering...................................... 7
4. Prior Art of Centering and Impact Mechanisms....................... 8
   4.1. Previous Centering Work........................................... 8
   4.2. Previous Piezoelectric Impact Mechanism Work............... 9
5. Four Different Approaches to Automated Centering................ 10
   5.1. Positioning Table.................................................. 11
   5.2. Robotic Placement.................................................. 11
   5.3. Push Centering.................................................... 11
   5.4. Tap Centering.................................................... 16
6. Impulse-Displacement Theory and Experiment........................ 17
   6.1. Impulse-Displacement Theory.................................... 17
   6.2. Impulse-Displacement Experiment................................ 19
   6.3. Results of Displacement-Impulse Experiment................... 20
   6.4. Discussion of Displacement-Impulse Experiment............... 21
7. Impact Mechanisms........................................................ 23
   7.1. Spring Mechanism.................................................. 24
   7.2. Pneumatic Mechanism.............................................. 24
   7.3. Electromagnetic Mechanism...................................... 24
   7.4. Piezoelectric Mechanism........................................ 24
8. Piezoelectric Impact Mechanism Theory................................ 25
9. Piezoelectric Impact Mechanism Design and Construction........... 27
   9.1. Critically Damped Linear Bearing Design....................... 28
   9.2. Un-damped Pendulum Design..................................... 31
10. Electronic Control Options: Three Inexpensive Approaches........ 33
    10.1. Pulse Width Modulation........................................ 33
    10.2. Series Resistance Control..................................... 35
    10.3. Voltage-Charge Control........................................ 36
11. Electronic Controller Construction and Performance................ 38
12. System Performance and Positioning Ability........................ 39
13. Scaling Piezoelectric Impact Mechanisms to Grinders................ 40
14. Further Work..................................................................... 42
References........................................................................... 44
Appendix A: Piezoelectric Impact Mechanism Matlab Simulation....... 45
Appendix B: PIC16F877 Microcontroller Code............................. 48
Table of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Photograph of the benchmark: Vince</td>
<td>7</td>
</tr>
<tr>
<td>2</td>
<td>Off center part and LVDT</td>
<td>12</td>
</tr>
<tr>
<td>3</td>
<td>LVDT output versus angular position</td>
<td>13</td>
</tr>
<tr>
<td>4</td>
<td>Flow chart of iterative centering operation</td>
<td>14</td>
</tr>
<tr>
<td>5</td>
<td>Polar plot of UMM centering operation, putt-putt</td>
<td>15</td>
</tr>
<tr>
<td>6</td>
<td>UMM actual displacement versus optimum displacement</td>
<td>16</td>
</tr>
<tr>
<td>7</td>
<td>Theoretical displacement-impulse curves</td>
<td>19</td>
</tr>
<tr>
<td>8</td>
<td>Displacement-impulse experimental results</td>
<td>21</td>
</tr>
<tr>
<td>9</td>
<td>Plot of part position following steel on steel impact</td>
<td>22</td>
</tr>
<tr>
<td>10</td>
<td>Plot of part position following vinyl on steel impact</td>
<td>22</td>
</tr>
<tr>
<td>11</td>
<td>Comparison chart of mechanism concepts</td>
<td>23</td>
</tr>
<tr>
<td>12</td>
<td>Force-displacement plot for piezoelectric actuator</td>
<td>26</td>
</tr>
<tr>
<td>13</td>
<td>Energy flow chart for piezoelectric impact mechanism</td>
<td>26</td>
</tr>
<tr>
<td>14</td>
<td>Plot of Matlab simulation results</td>
<td>27</td>
</tr>
<tr>
<td>15</td>
<td>Plot of critically damped free mass position</td>
<td>29</td>
</tr>
<tr>
<td>16</td>
<td>Solid model of critically damped prototype mechanism</td>
<td>30</td>
</tr>
<tr>
<td>17</td>
<td>Solid model of un-damped pendulum prototype mechanism</td>
<td>32</td>
</tr>
<tr>
<td>18</td>
<td>Photograph of un-damped pendulum prototype mechanism</td>
<td>32</td>
</tr>
<tr>
<td>19</td>
<td>Schematic of PWM control circuit</td>
<td>34</td>
</tr>
<tr>
<td>20</td>
<td>Schematic of series resistance controller</td>
<td>35</td>
</tr>
<tr>
<td>21</td>
<td>Schematic of voltage-charge controller</td>
<td>36</td>
</tr>
<tr>
<td>22</td>
<td>Drawing of hydraulic circuit analog</td>
<td>37</td>
</tr>
<tr>
<td>23</td>
<td>Photograph of voltage-charge controller</td>
<td>38</td>
</tr>
<tr>
<td>24</td>
<td>Oscilloscope trace of voltage-charge control</td>
<td>39</td>
</tr>
<tr>
<td>25</td>
<td>Plot of economic scalability of piezoelectric impact mechanism</td>
<td>41</td>
</tr>
</tbody>
</table>
1. Introduction

Many manufacturing techniques rely on a rotating axis to shape axially symmetric parts to precise tolerances. For example, Timken uses grind machines to finish cups and cones for tapered roller bearings. Some metrology machines also rely on a rotating axis for gaging round parts. These two applications have a need to accurately center the parts on the rotating axis before they can be machined or gaged. If the part is off center for a grind operation, form error will result. If the part is off center for gaging, measurement error will result. This paper gives background on the two centering applications: grinding and metrology, and explores traditional centering techniques which are then applied to an automated approach of tap centering. A prototype tap-centering machine designed to fit the requirements of Timken’s Universal Measuring Machine (UMM) is built and tested using a digital control system. The results show that when coupled with an intelligent control algorithm, automated tap centering will reduce cycle times by half and improve centering accuracy.

2. Universal Measuring Machine and Grind Applications

The UMM is a gaging device that can verify machine setup, complete machine capability studies, perform process control, and conduct quality audits of cup, cone, and roller manufacturing processes. Timken has nine UMM’s, six of which are used for production gaging in a plant in Asheboro, NC, and are a bottleneck in the manufacturing line. Reducing the UMM’s cycle time leads to large savings and centering occupies a significant portion of the cycle time. The average cycle time of a UMM is about four minutes. Of this time, 45 to 90 seconds are spent centering the part. A realizable reduction in centering time of 20 seconds leads to an annual savings of $21,000 at the Asheboro plant.

The value of automated centering for grind operations is much greater than that for the UMM, but more difficult to quantify. Centering is non-value added but can consume a large portion of overall cycle time. Currently, centering on production grind
machines is done manually by an experienced operator. Dramatic cost savings will come from automation of the grind machines and reduction in cycle time. The Timken plant in Asheboro, NC, for example, has 6 grind cells. Based on having one operator per cell being paid an average yearly salary of $50,000, a yearly savings of $300,000 is realized.

3. The Benchmark: Manual Centering

Manual centering is by far the most dominant method of centering parts. Until I observed a manual centering operation first hand, I leaned toward centering techniques that use a motorized table to move parts into position. However, watching an experienced operator was a near religious experience that inspired me to want to build a machine to do the same.

Joe Pack, my manager at Timken, suggested instrumenting an operator to establish a benchmark for centering. Vince Lombarzzo, a machinist who has been with Timken for over thirty years agreed to demonstrate the technique while an oscilloscope recorded his actions. Figure 1 shows Vince using an instrumented impact hammer to center a part on a vertical axis magnetic chuck.

![Figure 1: The Benchmark: Vince, a machinist with thirty years experience at Timken, centers a part using a hammer and dial gage.](image)
To center a part, Vince positions a dial gage on the surface of the part. He watches the dial as the part is rotated and taps the part into position with carefully timed hammer strikes. Using these two tools, the experienced machinist can center a part to within a few microns in about thirty seconds. This does not include time taken to setup the dial gage, which presumably is a permanent fixture at a production grinding cell.

For this experiment, the oscilloscope traces the load on the hammer and the position of the part. The use of a Yokogawa storage scope allows a clear view of the machinist’s technique represented by a sinusoidal output from the LVDT and the sharp impulses from the hammer. The data provide a benchmark for the level of repeatability needed to center parts, and also gives rough limits for the impact strengths needed to move the parts. Perhaps the most useful observation is that the first few hammer strikes are not the most forceful, but are test hits to give the machinist a feel for how the part moves. Similar exploratory taps can be applied to an algorithm for automated centering.

4. Prior Art of centering and impact mechanisms

The manual technique of centering is broadly recognized and used. Because of the value in automating and improving centering techniques, there is a lot of prior work that has been done. This section presents a review of the literature that pertains both to centering and to piezoelectric impact mechanisms—the centering technique pursued in this thesis.

4.1. Previous Centering Work

Much of the background work that led to this thesis was done at Timken to address centering problems in metrology machines. Since the early 1990’s, Bruce Berner, Tom Ballas, and Mike Thomson have done work with both tap and push centering techniques. The push centering mechanism on the UMM, although not optimized, demonstrates how a DFT can be used to determine the off center vector with just one additional axis. During my six months at Timken, Bruce demonstrated a push centering mechanism with a learning algorithm that adjusted its actuations based on previous pushes. It still suffered from some of the shortcomings of push centering at
small displacements, and Bruce encouraged me to explore tap positioning. An imaginative device is currently being used at Timken to position rollers on a gaging machine. The device uses speaker cones driven at several tens of Hertz to tap parts into position. The impacts are not precisely controlled, but the mechanism works with very inexpensive parts.

Many patents for centering technologies are held by the Bryant Grinder Corporation. Their work is on high load centering mechanisms for production grind machines. The patents describe using actuated shoes to position parts, and since the shoes are sized for specific parts, the mechanisms are not easily adapted to different size parts. They own a lot of the intellectual property on learning algorithms applied to centering.

Apart from large manufacturing companies, several researchers have focused on part centering for its value in gaging and for its potential value in semiconductor manufacturing. Hatsuzawa describes a push centering mechanism that used three pusher arms for actuation and a separate probe for measurement (1988). The performance of the device was poor compared to what the theory predicted, mostly because of compliance in the pusher arms. Centering took six minutes, and the author’s discussion of the problems with the device fit well with the discussion presented in this thesis about push centering challenges. A paper by Sata describes a push centering mechanism based on a stepper motor actuator and eddy current sensor (1983). He mentions the usefulness of the device for measuring work piece roundness during centering.

4.2. Previous Piezoelectric Impact Mechanism Work

Piezoelectric actuators see relatively limited use because of their high cost, small displacements, and perceived shortcomings such as creep and hysteresis. However, for some applications, such as impact mechanisms, these disadvantages are offset by the high stiffness and simplicity of piezoelectric actuators when used dynamically. This section describes previous work on piezoelectric impact mechanisms, especially that done by Higuchi, and how it is applied to this work.

A lot of work as been done by Higuchi to explore piezoelectric impact mechanisms. He has a patent from 1990 for a fine motion device that relies on the impact
force of a piezoelectric element and inertial mass (1990). The device, described in more detail in a later issue of The International Journal of the Japan Society of Precision Engineering, uses high frequency asymmetric waveforms to drive small masses in precise steps (1999). His work evolves into a three DOF positioning table that uses three impact mechanisms as actuators. By driving short stroke actuators and friction elements at high frequency, around 20 KHz, precision displacements of a few millimeters are achieved.

A larger impact device is described by Higuchi and others in a January 2003 publication of Precision Engineering (2003). Regrettably, this thesis work is carried out during the six month period before Higuchi’s publication, so it does not have the benefit of his research on this device. It is interesting to note the similarities and differences between the work described in Precision Engineering and the work described here, and how those differences affect their applicability to impact centering.

Both devices rely on a rapidly expanding piezoelectric element constrained by inertial forces. However, Higuchi presents his impact device in the context of a positioning table. This application allows for higher normal forces on the contracted actuator than are acceptable for tap centering applications. This problem of high normal forces is elaborated on in Section 9.1. Higuchi also presents a useful analysis of the contact forces and deformation at the interface between the actuator and mass. I assume the resilience and stiffness of the steel on steel interface is high enough that little energy is dissipated. This assumption could explain why my analysis predicted larger impulses than were measured experimentally.

5. Four Different Approaches to Automated Centering

There are four main approaches to automated centering: robotic placement, positioning table, push centering, and tap centering. The first two involve position feedback and placement in which the part is moved to its centered position. The last two involve an iterative sequence of measuring and actuation to progressively move the part into position.
5.1. Positioning Table

The positioning table was the first approach considered. An x-y positioning table would rely on an initial position measurement to determine how far off center the part is and then a single x-y movement would place the part in the center of the table. Two considerations rule out the use of a positioning table; high cost and loss of measurement reliability.

The cost of a positioning table is driven up by the high load-90 Kg, high precision-2 micron requirements. The cost of positioning stages skyrockets with increasing load and precision. Quotes are in excess of $60,000 to have the stage commercially built. An additional factor that increases the cost of the table is the space limitation. The stage would have to fit on the existing metrology machine and the thickness of the stage directly subtracts from the measuring volume of the UMM.

The other consideration that rules out the use of a positioning stage is the error that results from having the part supported by a stage during measurement. When the UMM was designed, great care was put into the design of a stiff platen. Without considerable cost, it is difficult to approach this stiffness and that hurts the reliability of the machine.

5.2. Robotic Placement

Robotic placement as a centering technique is an extension of the idea of using a robotic loading arm to automate loading of the UMM. With sufficient measurement capability, a robotic arm can approach the 2 micron accuracy needed for UMM centering, and would be able to place parts on center. However, with high load and high precision comes an enormous price tag. A precision robotic arm will cost more than the UMM itself, so it is quickly ruled out.

5.3. Push Centering

Push centering is the approach currently used to center parts on the UMM. This section describes the theory of push centering, and presents actual performance data from the UMM followed by a discussion of why push centering is problematic.
Push centering is an iterative operation in which the part is progressively moved toward the center position by discrete pushes. In between pushes, the part is gaged to determine its off-center vector. Since the UMM is already equipped with a precision rotary stage, this gaging and actuation is accomplished with a single radial arm. Figure 2 shows a solid model of the radially positioned Linear Variable Differential Transformer (LVDT).

**Figure 2:** A simplified diagram of the radially positioned LVDT and off center part

The red arrow connects the geometric center of the part with the rotation axis and is the off center vector of the part. Figure 3 shows typical output from the LVDT over one full rotation.
The signal is a mixture of the sinusoid that corresponds to off center error, higher frequency sinusoids that correspond to form error in the part, and noise from the amplification electronics. Because the rotational speed of the table is known, a Discrete Fourier Transform (DFT) corresponding to the rotational frequency tells the amplitude and phase of the off center vector. The high point is marked by a red circle.

With the off center vector known, the push centering mechanism waits until the angular position of the table matches the phase of the off center vector, and then actuates the part a distance equal to the off center amplitude. This process is repeated until the off center error is measured to be less than the centering tolerance. Figure 4 shows a flow chart for the push centering process.
Push centering has a number of advantages over positioning tables or robotic placement because it is implemented with relatively low cost components, and is able to take advantage of an existing axis of motion. It is also able to center parts that have a large form error because it calculates a DFT. Form error results primarily from vibration during grinding and can be a problem for other centering techniques.

However, push centering gave unsatisfactory results on the UMM. The poor performance stemmed from algorithm problems and from some fundamental difficulties of push centering. The fundamental difficulty arises from the dependence of the operation on precise positioning during actuation. Because the period of actuation is brief, high forces are involved making system compliance, friction, and inertia critical factors.

System compliance is a problem because the compliance loop runs all the way through the UMM. The compliance of the actuation arm, supporting structure, and air
A surprising source of compliance arises from deformation at the interface between the probe tip and the part. Unfortunately, this interface is at best a line contact, and at worst a point contact because the parts are cylindrical. A Hertz contact stress analysis is done to show that elastic deformation of up to 8 microns is possible with heavy parts (Johnson). This is disastrous when trying to position to an accuracy of 2 microns. Friction and inertia play their part by making actuation force difficult to predict, so system compliance cannot be compensated for.

These problems are evident in the performance of the UMM. Centering times vary from 30 to 90 seconds per part and are a significant portion of the overall cycle time. Frequently, the UMM push centering mechanism makes up to thirty push attempts before successfully centering the part. Figure 5 shows the UMM Putt-Putt approach to centering. The blue points represent the position of the part following a push attempt. Following the first push, the part is much closer to centered. However, there are over twenty overlapping points that represent unsuccessful pushes at small distances.

Figure 5: The poor performance of the UMM push centering is evident in this plot showing part position after successive pushes. It resembles a game of Putt-Putt.
The problems of compliance, friction, and inertia discussed earlier are responsible for the unsuccessful pushes near center. Figure 6 shows data from more than twenty centering runs in which the actuation distance is compared to the ideal actuation distance. Perfect actuation would lie along the 100% line demarcated in green. Below distances of about a centimeter, the push algorithm becomes unreliable and frequently overcorrects.

![Graph showing the performance of the UMM push centering mechanism over twenty centering operations.](image)

**Figure 6:** Performance of the UMM push centering mechanism over twenty centering operations. Accuracy, which would ideally lie along the green 100% line, breaks down below about a centimeter.

5.4. Tap Centering

Tap centering relies on an iterative operation similar to the one used for push centering. The difference is that tap centering moves the part by delivering a precise impulse instead of a position command. Because the impulse is at a frequency higher than the resonant frequency of the structure, the negative effects of compliance associated
with push centering are not a problem. The following section introduces the requirements for successful tap positioning, and supports theory with experimental data.

6. Impulse-Displacement Theory and Experiment

6.1. Impulse-Displacement Theory

There are three requirements for successful tap positioning: there needs to be a relationship between an applied impulse and the distance a part moves, there needs to be a device that can deliver precise impulses, and there needs to be an intelligent control algorithm to determine the timing and magnitude of the impulses. This section explores the first requirement: the relationship between impulse and displacement both theoretically and experimentally.

The success of tap positioning hinges on the repeatable relationship between part motion and applied impulse. To examine this relationship, a theoretical model is proposed and experiments are conducted to validate the model. The model proves to be accurate, and additional insight is gained into the role of resonance in the repeatability of tap positioning. This insight is summarized by a "rule of thumb for impact positioning" which is: do not excite resonance in the part and the repeatability will greatly increase.

The theoretical model is derived for a part sliding across a horizontal surface in the direction of an applied impulse. An impulse, \( P \), applied to a mass, \( m \), leads to a change of momentum of the part given by

\[
P = mv_1 - mv_0
\]

(1)

Since the part is initially at rest, the velocity of the part immediately after the impact is

\[
v_1 = \frac{P}{m}
\]

(2)

The part will move until its kinetic energy is dissipated by friction. Assuming Coulomb friction, the work done by friction is the product of force and distance, \( d \), where force is
the product of gravitational acceleration, \( g \), the kinetic coefficient of friction, \( \mu_k \), and mass.

\[ W = mg\mu_k d \]  

(3)

The kinetic energy that must be dissipated by friction is

\[ E_k = \frac{1}{2} \frac{P^2}{m^2} \]  

(4)

where \( P/m \) is from Equation (2). Rearranging the work-energy relationship gives

\[ d = \frac{P^2}{2m^2 \mu_k g} \]  

(5)

Figure 7 shows the result of Equation (5) in the form of displacement-impulse curves for a coefficient of friction equal to 0.1. The plot shows the usefulness of the theory for tap positioning because knowing the part mass and the desired displacement clearly determines the impulse that will get it there.
Figure 7: Theoretical displacement-impulse curves for different masses at a coefficient of friction of 0.1

6.2. Impulse-Displacement Experiment

Experiments are conducted to prove the validity of the impulse-displacement model. The test apparatus supports the part on a surface with a consistent coefficient of friction, allows the position of the part to be measured, and allows the impulse to be measured.

Carbide is chosen as the surface for the test apparatus. Carbide's hardness makes it a popular choice in metrology, and its coefficient of friction against steel is fairly constant at about 0.1 (Wang, 1991). Three polished carbide rails are arranged radially and separated by 120 degrees. This arrangement is seen in the UMM and gives reliable three point contact for the part. Cylindrical steel rings were used as the test parts because they rest nicely on the three carbide rails and are similar to the steel cups and cones Timken frequently places in the UMM.
The position of the part needs to be measured with micron accuracy because the distances that the parts are moving are small, typically less than 100 microns. Two different position measurement techniques are used: capacitance probe and LVDT. The capacitance probe gives very good precision, but a range of only tens of microns. The LVDT provides good precision over a range of a few millimeters.

The impulse is recorded using two different size instrumented impact hammers that were originally intended for modal analysis. A small instrumented impact hammer gives good precision over a small range. It is used for small impulses with light parts. A larger hammer is used for heavier parts that require larger impulses. Both hammers have two different types of tips: steel and vinyl, this option of tips proves to be very important, as is discussed in the next section. The output from the impulse hammer is recorded using a Yokogawa data recorder. Voltage values are sampled at 500 kHz and stored as .dat files. A MATLAB script integrates the voltage-time curves and uses the voltage-load conversion factor to determine the total applied impulse (Appendix A).

6.3. Results of Displacement-Impulse Experiment

The initial results are shown in Figure 8. The displacements using the steel-tipped hammer are not predictable and do not support the theory. However, the results using the vinyl-tipped hammer were much more predictable and agreed well with the theory, represented by the red line. Reasons for these results are discussed in the following section and their importance for tap positioning is shown with a “rule of thumb for tap positioning.”
Figure 8: Displacement-impulse experimental results using instrumented impact hammer with steel tip and with vinyl-tip. The points corresponding to the vinyl tip lie closely along the theoretical line.

6.4. Discussion of Displacement-Impulse Experiment

The results from the displacement-impulse experiment show that the theory is promising, but clearly there are some effects that are not taken into account in the theory. The good correlation that exists between theory and experiment for the vinyl tipped hammer is promising, but the unpredictability seen with the steel tipped hammer is disappointing. This section discusses the role of resonant frequency excitation and presents a rule of thumb for tap positioning: Do not excite resonance in the part.

To investigate why impulse displacement data is so much more predictable for the vinyl tip than for the steel tip, the time plots of the capacitive displacement sensors are examined. Figures (9) and (10) show part motion following a steel on steel impact and a vinyl on steel impact, respectively. A plot of the kinetic coefficient of friction corresponding to the second derivative of position and Newton’s Second Law is shown as well.
Figure 9: Part position following steel on steel impact, measured using capacitive displacement sensor. The kinetic coefficient of friction is calculated from the second derivative of position (on a smaller time window applicable to part motion).

Figure 10: Part position following vinyl on steel impact. The kinetic coefficient of friction is shown for a condensed time window applicable the time the part is in motion.
These plots show obvious differences between the steel on steel and vinyl on steel impacts. The steel on steel position plot shows steady high frequency ringing and large variations in the kinetic coefficient of friction. No vibrations are seen following the vinyl impact, and the kinetic coefficient of friction varies little.

It is concluded that the steel on steel impact is much higher frequency than the vinyl on steel impact, and the sharper impact excites resonant frequencies in the part. The output from the load cell shows that the frequency of steel on steel impacts is about 20KHz and the frequency of vinyl on steel impacts is about 1.7 KHz. An analysis of the first mode of vibration of a steel ring predicted resonant frequencies of about 6 KHz (Harris, 1961). This analysis leads to a rule of thumb for tap positioning, which is worth repeating: **Do not excite resonance in the part.** This rule is reflected in my final design in the form of a damping element that reduces the sharpness of the impact.

7. Impact Mechanisms

The experienced machinist uses different sized hammers and various strokes to achieve precision over a broad range. This research focuses on the design and construction of a mechanism that can replace the machinist’s hammer by delivering precise impulses over a broad range. A comparison of spring mechanisms, pneumatic mechanisms, electromagnetic mechanisms, and piezoelectric mechanisms is made before selecting the piezoelectric mechanism.

The following figure shows a qualitative comparison of these mechanisms where pneumatic is considered a baseline.

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**Figure 11:** A chart comparing four mechanism designs
7.1. Spring Mechanism

A spring mechanism is similar to the plunger found on pinball machines. Varying the distance the spring is compressed before it accelerates a mass varies the strength of the impact. Wesley Huang at RPI successfully employed a spring mechanism for three axis positioning of polygons with centimeter precision (2002). Disadvantages of spring mechanisms include a limited range of impacts, usually around one decade, and the need for a device that can quickly cock and release the mass.

7.2. Pneumatic Mechanism

A pneumatic impact mechanism was tested at Timken when they first investigated tap positioning in the early 1990’s. The device functioned by accelerating a low-friction piston with controlled air pressure. The major disadvantage of the pneumatic mechanism is its inability to deliver small impulses due to friction of the seals. This limited the range of control to about one decade.

7.3. Electromagnetic Mechanism

Electromagnetic mechanisms can take the form of solenoids, voice coils, or linear motors. The simplest embodiment is an audio speaker that incorporates both a voice coil and the guiding flexures in the form of a speaker cone. Their main disadvantage is heat generation because much of the energy in an electromagnet is dissipated as heat. They also require robust power supplies to achieve high forces. Some automated modal analysis hammers use electromagnetic actuation (The Modal Shop, 2002).

7.4. Piezoelectric Mechanism

A piezoelectric mechanism in which the impact is generated by the rapid expansion of a piezoelectric element is favorable because it allows precise impact control over a broad range through a fairly simple control scheme. The piezoelectric element behaves like a capacitor electrically and can convert a large amount of electric power into
mechanical power over a brief period of time. This is well suited for the low duty cycle the hammer would see in operation.

8. Piezoelectric Impact Mechanism Theory

The heart of the impact mechanism is the piezoelectric actuator and free mass. These components are sized correctly by first estimating the desired range of the mechanism and then running a MATLAB simulation to examine how different actuator-mass combinations affect the range. The desired range is determined from the smallest and largest impulses needed to move the lightest and heaviest parts respectively. The smallest impulse the mechanism must provide needs to position the lightest parts to a resolution less that twice the centering tolerance. Such an impulse would move a part that is barely outside of centering tolerance to a position just within tolerance on the opposite side of center. With a minimum part mass of 1 Kg and a centering tolerance of 2.5 microns, this lightest impulse is predicted to be 2.2 mN-s. This estimate is made from Equation (5), derived in Section 6.1, for a mass sliding on a horizontal surface following an impact.

The largest impulse the mechanism must provide needs to be able to move the largest part some reasonable distance. The largest possible part is determined from the load capacity of the UMM which is 90 Kg. Ten microns is a reasonable distance to be able to move very large parts because it will still be an improvement over existing centering ability without placing too high demands on the mechanism. Equation (5) is used to estimate the maximum desired impulse of 282 mN-s.

Selecting an actuator that can deliver this range of impulses is the next step. A MATLAB script is used along with product specifications from different actuator manufacturers. Different product specifications are simulated with different sizes of free masses to determine the minimum and maximum impulses the actuator-mass pair will produce.

A piezoelectric actuator behaves like a capacitor electrically and responds to a charge with a stiff (typically in the tens of KN/micron) mechanical expansion over a short distance, typically tens of microns. Figure (12) shows the force-displacement
relationship for the actuator chosen for this design, an American Piezo Company Model Number PST 150/10/20.

![Graph showing force-displacement relationship for the actuator](image)

**Figure 12:** Force-displacement relationship for piezoelectric actuator chosen for this design, Model Number PST 150/10/20 manufactured by American Piezo Company.

In order to take advantage of this high stiffness, the actuator’s small displacement must be resisted by a large inertial mass. The resulting high force generated by the acceleration of the inertial mass results in useful work, which is stored in the form of kinetic energy in the mass. Figure (13) shows the flow of energy from electric to kinetic energy.

![Energy flow diagram](image)

**Figure 13:** The flow of energy from electric energy, $E_e$ in the capacitor, to potential energy, $E_{pzt}$ in the compressed piezoelectric actuator, to kinetic energy $E_m$ in the free mass. Losses are also shown at each conversion.

The impulse results when the mass collides with the part and transfers momentum to it.
The system experiences losses that are difficult to quantify, so estimating impulse based on energy flow into the capacitor is not accurate. A more accurate approach is to develop a simulation in MATLAB. The script is developed to show how an input voltage waveform translates into an impulse. The MATLAB simulation calculates the change in length of the piezoelectric actuator for sequential points along the input voltage waveform (Appendix A). The piezoelectric actuator's stiffness is used to determine the expansion force and the position, velocity, and acceleration of the free mass, for each point based on the values of the previous points. Figure (14) shows the graphical output from the MATLAB script. The predicted final velocity of the free mass is used to predict the resulting impulse.

![Graphical Output from MATLAB Script](image)

**Figure 14:** Graphical output from the MATLAB script used to predict the impulse resulting from the free mass velocity. The input waveform, shown in red, is supplied by a controller, and is related to actuator displacement and stiffness, Figure (13), by the blue force curve.

9. **Piezoelectric Impact Mechanism Design and Construction**

The theory behind impact positioning agrees well with experimental results, and MATLAB simulations predict that the piezoelectric impact mechanism can deliver
precise impulses over an appropriate range, but a test device is needed to prove the concept. Two different prototypes are constructed. The first design, a critically damped linear bearing design, does not operate reliably, so it is adapted to a second design based on an un-damped pendulum. Both designs have an integrated LVDT for position measurement and are sized to fit the UMM.

9.1. Critically Damped Linear Bearing Design

As the Matlab simulations show, a piezoelectric impact mechanism will only work if there is some inertial mass for the actuator to expand against. The first embodiment of the piezoelectric impact mechanism sandwiches the piezoelectric actuator between a free mass and the part to be gaged. This arrangement has some advantages and disadvantages. It proved not to be as good of an arrangement as the second design, but important lessons are learned from its operation, so it is described here.

The two advantages of sandwiching the piezoelectric actuator between a free mass and the part are simplicity and the dependence of impulse strength on part mass. Larger part masses result in larger impulses for a given excitement voltage. This effect is advantageous because it increases the range of impacts the device can produce. However, there are several disadvantages to this design: the piezoelectric actuator must be in good contact with the part, it is difficult to prevent secondary impacts, and the impacts are sharp.

The first disadvantage arises because the compliance between the piezoelectric actuator and the part is critical. As the Matlab simulations show, slop between the actuator and the inertial mass wastes the actuator’s short, but forceful expansion. Here, the part mass is one of the inertial masses. It is difficult to get good solid contact between the actuator tip and the back of the probe, and between the probe and the part. This is a variable that is unpredictable and has a huge effect on the strength of the impulse.

Another disadvantage of this design is that after an impulse, the free mass and actuator must come back into contact with the part without causing a second impact. Secondary impulses can move the part farther than intended, and are unpredictable because it is difficult to know how much energy has been dissipated through friction. An
elegant way to prevent secondary impacts is to critically damp the system. By controlling the mass, spring force, and damping, the system can be critically damped so that after the impulse, the free mass will gently settle back onto the part without an impact.

Figure (15) shows a plot of free mass position following an impulse.

![Plot of free mass position following an impulse.](image)

**Figure 15:** Plot of free mass position following an impulse. With \( m = 0.5 \) Kg, \( k = 79 \) N/m, and \( b = 12.6 \)N-s/m, the system is critically damped and there is no unwanted secondary impact.

Designing a critically damped system requires choosing values for the mass, \( m \), and spring constant, \( k \); these two parameters define the appropriate damping ratio. The mass is already sized by using the Matlab scripts to estimate the impulse range. The spring stiffness is important because it determines the gaging force. The upper limit on the spring stiffness is defined by the need to keep from moving the part during gaging. The lower limit is defined by the need to keep the free mass against the part during gaging. This means the natural frequency of the mass-spring-damper system must be greater than the rotation rate of the part.
Figure (16) shows a solid model of the critically damped linear bearing design. The dashpot and mass are shown, but the spring is not. It is a compression spring positioned behind the free mass to push it forward.

Figure 16: Solid model of the critically damped linear bearing piezoelectric impact mechanism. The spring is not shown, but goes behind the free mass to push it forward.

The front of the assembly is used for gaging to determine off-center error. It consists of a carbide probe tip attached to a precision ground steel shaft that rides on a ceramic linear bearing. The shaft is spring loaded to maintain gaging force with the part. The back end of the probe extends beyond the LVDT body and is used to transmit the impact force from the piezoelectric actuator.

The piezoelectric actuator is held against the back of the probe by a spring, and a spherical contact is used to decouple side loads. The shaft connecting the free mass to the dashpot allows for axial misalignment by flexing at both ends. Axial misalignment is
held to a minimum by supporting all of the cylindrical elements on an aluminum block with machined vee blocks.

9.2. Un-damped Pendulum Design

A second prototype is built to fix some of the problems with the critically damped linear bearing design. The second prototype suspends the free mass from thin flexures that let it swing like a pendulum. This allows the mass to rest against the rigidly fixed piezoelectric actuator instead of following the part during gaging. That reduces the compliance in the system to just the interface between the spherical carbide steel actuator tip and flat steel face of the free mass.

The free mass acts as an energy storage element, so instead of a sharp impulse being generated directly against the part as in the previous design, the sharp impulse is generated against the free mass. The free mass then swings into the back of the probe which transmits the impact force to the part. The advantage to this arrangement is that compliance can be built into the impact mechanism to avoid exciting resonance in the part. A close look at Figures (17) and (18) reveals a blue viscoelastic pad on the front face of the free mass. The pad deforms to lengthen the duration of the impact, thereby preventing the excitation of resonance in the part through a sharp impact. The viscoelastic pad greatly increases displacement repeatability.
Figure 17: Solid model of un-damped pendulum design

Figure 18: The piezoelectric impact test device machined at Timken. It has an integral spring loaded LVDT (Linear Variable Differential Transformer) for gaging.
The piezoelectric actuator is Model Number PST 150/10/20 from American Piezo Company. It is paired with a 0.2 Kg free mass, which moves on spring steel flexures. The probe passes through an Schaevitz LVDT so that its position can be read from a BNC connector on the back of the device. The probe moves freely through the LVDT and rests on an oiled ceramic linear bearing.

10. Electronic Control Options: Three Inexpensive Approaches

For the piezoelectric impact mechanism to be useful, it must have a control system that can deliver measured impacts at precise times. The controller must fulfill two requirements: interface to an intelligent control algorithm, and handle the high power requirements of the actuator. The first requirement is satisfied by a design that interfaces to an intelligent control algorithm with two inputs: an 8-bit parallel signal that corresponds to impulse strength, and a trigger signal that determines impact time. The second requirement is more demanding because of the piezoelectric actuator’s high current draw.

Since the actuator looks like a capacitor electrically, it draws large currents during rapid changes in voltage. With the impact mechanism, currents in excess of 100 Amps can flow into the 4 μF actuator. A commercial actuator that can handle this much current is very expensive, most have slow response times, and they would not produce rapid enough actuator expansion. Commercial amplifiers designed for piezoelectric actuators cost thousands of dollars, are designed for continuous moderate frequency operation, and can only supply a few amps. Although the piezoelectric impact mechanism requires high currents, it is only for very short durations: less than a millisecond. During tap centering, these actuations are separated by about a second. The control schemes described here all take advantage of the low duty cycle operation of the impact centering mechanism by using a large storage capacitor to smooth out the power demand. Three approaches are considered: Pulse Width Modulation (PWM), series resistance control, and primary capacitor charge control.

10.1. Pulse Width Modulation
PWM is a popular method for digital control of DC motors. Figure (19) shows a PWM circuit that interfaces with a microcontroller. Varying the duty cycle of the square wave output from the microcontroller effects the current flowing through Q1, a Power MOSFET. Normally, PWM operates at a frequency of a few Kilohertz. However, control of the rapid charging of the piezoelectric actuator requires frequencies around one Megahertz. This high frequency requirement ruled out the choice of PWM due to limitations of both the microcontroller and the MOSFET.

![Figure 19: A Pulse Width Modulation (PWM) circuit to control current going from a primary capacitor to the piezoelectric actuator.](image)

Not only is high frequency a problem for the microcontroller which operates at a maximum frequency of 20 MHz, leaving little time for command execution in between toggling of Q1, but is also a problem because it pushes the limits of the MOSFET. Although MOSFET's can operate in the MHz range, the power is limited by higher resistance during turn ON and turn OFF times. Ideally the ON resistance of the MOSFET would be zero and the switch from OFF to ON would be instantaneous. In reality, ON resistance is around half an Ohm even for a good power MOSFET. More importantly, turn ON and turn OFF is not instantaneous, and the increased resistance during these times leads to heat generation. These delays are seen even with a perfect voltage source at the gate, and are made worse by source and lead resistances.
An additional challenge with high frequency PWM is supplying the power needed to drive the MOSFET. The gate of a power MOSFET has some capacitance, usually around 700pF. This gate capacitor must be repeatedly charged to around 15 volts and then discharged to ground to turn the MOSFET ON and OFF. Power is the product of the driving frequency, the gate capacitance, and the square of the driving voltage. At one MHz, the power draw is 158 mW. This is manageable with an amplifier, but should not be ignored because the addition of an amplifier increases transient times.

10.2. Series Resistance Control

Series resistance control is another method that is explored, but not used in the final design. This approach relies on changing the time constant of the system by varying the series resistance between the power source and the piezoelectric actuator. Figure (20) shows a circuit that translates an 8-bit number into a resistance value.

\[ R_{eq} = \sum R_i \cdot \left(\frac{1}{2^n-1}\right) \]

A nice feature of the series resistance approach is that the equivalent resistance corresponds directly to the 8-bit parallel input which is controlled by a microcontroller. The parallel input controls relays that bypass corresponding resistors. A power MOSFET is once again used as the trigger and prevents the flow of current through the relays.
during actuation. This reduces the wear on the relays because arcing is not an issue. However, sometimes the high currents did destroy the relays by welding the contacts closed.

10.3. Voltage-Charge Control

The method I chose is a voltage-charge control scheme. It doesn’t suffer from many of the drawbacks of the previously described control methods because it stays within the operating ranges of basic off the shelf components. The voltage-charge control method involves adjusting the charge on a very large capacitor by measuring and adjusting its voltage. The charge is then dumped into the piezoelectric actuator so that it is in parallel with the large capacitor and they share a common voltage. Figure (21) shows a simplified schematic of the circuit that is used to control the piezoelectric impact mechanism, represented by $C_p$. Figure (22) represents the hydraulic analog of the voltage-charge control circuit. The large tub clearly shows that sufficient charge can be stored to quickly bring the smaller storage element up to the same potential. The following paragraphs reference components in both the circuit diagram and the hydraulic analog.

**Figure 21:** The simplified schematic for a voltage-charge control scheme.
The circuit is controlled by a Microchip PICF877 microcontroller. Because it operates at TTL voltage levels, a voltage divider is used to bring the 150 V high voltage down to the 5 V level that the on-chip A/D Converter can handle. Additionally, an Op-Amp voltage follower is used to interface the voltage divider to the A/D Converter so that excessive power is not dissipated in the voltage divider as it tries to supply the necessary power to the Analog input.

![Figure 22: The hydraulic analog of the voltage-charge control circuit shows how the level of charge in a large capacitor can be regulated and then quickly dumped into a smaller capacitor, Cp, representing the piezoelectric actuator.](image)

The PICF877 monitors the 8-bit A/D Converter and compares it to the desired 8-bit voltage level. If the measured voltage is too high, it turns on Q2 to dump charge through R2. If the voltage is too low, it turns on Q3 to charge the capacitor through R3. R2 and R3 were chosen to limit the current through the optocoupled transistors, Q2 and Q3. Fairchild Semiconductor makes an H11D1 optocoupled transistor that can handle 100 mA. The measure and adjust cycle occurs at about 50 Hz and effectively maintains the voltage to within a bit of the desired voltage. For 8-bit control, this corresponds to an accuracy of about 0.6V over the 150 V range. The response time is defined by the RC time constant of R2 and R3 and C1. This time constant means that the controller can reach any voltage in about 5 Tau, or in this case, 750 ms, fast enough for impact centering. If faster settling times are desired R2 and R3 can be reduced through the use of a more robust optocoupled transistor.
11. Electronic Controller Construction and Performance

The voltage-charge control technique is implemented to control the piezoelectric impact mechanism. Its performance is tested independently from the piezoelectric impact mechanism to verify that it reliably holds the target voltage and can discharge that voltage across the piezoelectric actuator on command.

Figure (23) shows the completed voltage-charge controller. The low voltage microchip control circuit is bread boarded to allow easy adjustment of component values and the high voltage is housed under Plexiglas to help prevent shocks. The temptation of convenience to work on it without the Plexiglas cover in place is dangerous and I quickly gained a respect for the charge held in high voltage capacitors (orange cylinders).

![Figure 23: The completed voltage-charge controller including the microchip processor and high voltage circuit, shielded by Plexiglas.](image)

After some adjustment, the circuit works beautifully and as planned. The code presented in Appendix B reads an 8-bit user input from DIP switches and adjusts the
voltage on C1 based on an input of “0” being 0 V and an input of “255” being 150 V (Appendix B). Typically the controller reached the programmed voltage in less than half a second, and held it there within the 1-bit error. Figure (24) shows a trace of the voltage on the capacitor. The frequency of the control cycle is shown to be 44 Hz, and magnitude of the voltage error is only 168 mV. The shape of the wave results from the discharge of C1 through the voltage measurement circuit followed by rapid charging through Q3 that brings the voltage back up to the target.

![Figure 24: A trace of the voltage on the storage capacitor, C2, as it is actively controlled by the microcontroller. The measure and adjust cycle occurs at 44 Hz and the magnitude of the error is 168 mV.](image)

12. System Performance and Positioning Ability

The impact mechanism is used to determine the relationship and repeatability of displacements that result for different 8-bit inputs. The 8-bit inputs are simulated using DIP switches so the impulse can be consistently controlled.

The repeatability of the displacements of a 1.6 Kg part with an input of 128 is within 3% for twenty different trials. This corresponds to an excitation voltage of 75
Volts, and a mean displacement of 180 microns. The prototype achieves remarkably consistent and predictable displacements corresponding to commanded inputs. Several experimental runs using different masses ranging from less than 1 Kg to 8 Kg are conducted. Although the individual runs vary some, it is most important that the displacements be consistent within a run. This allows an intelligent algorithm to control the impact mechanism and learn from past attempts.

A possible cause of the variation between different runs is ringing between the carbide and the steel. Ringing occurs when a thin film of oil greatly increases the coefficient of friction between the two surfaces. This is prevented by carefully cleaning the surfaces, but that is not a practical solution for production. However, the effect is consistent within each trial, so an intelligent algorithm can correct for changing coefficients of friction by learning from previous taps within an individual run.

Within an individual run, displacements are predictable to less than 3% over the entire range. This means that a part that is initially 5 mm off center would be less than a micron off center after just three taps. If the intelligent algorithm requires one initial tap for learning, the total would be four taps. A reasonable rate of rotation would allow four taps to be made in less than ten seconds which is less than half the time required for current centering techniques.

13. Scaling Piezoelectric Impact Mechanisms to Grinders

The largest potential savings comes from automation of centering on grind machines. Grind machines operate in harsher environments and require higher loads than metrology applications. However, piezoelectric impact mechanisms scale well and both of these obstacles can be overcome.

The harsher environments of grind machines require shielding of the device from coolant, oil, and debris. This is especially important if the impact mechanism has an integral LVDT, which gives inaccurate readings when contaminated by metal chips. A welded steel bellows is one solution that would seal out contaminants and could also provide spring loading for the probe.
The higher load requirements are a result of the magnetic chucks the grind machines use. Magnetic chucks increase the normal force on the part to counteract the high side loads encountered during grinding. Traditionally, manual centering is done by reducing the force of the magnetic chuck, tapping the part into place, and then increasing the force of the magnetic chuck. Many grind machines have a horizontal axis, so just keeping the part on the face of the chuck requires a normal force ten times higher than the force of gravity, assuming a $\mu$ of 0.1.

Higher normal forces require larger impulses. Increasing the size of the piezoelectric actuator and the size of the free mass will increase the impulse the impact mechanism can deliver. Figure (26) shows the favorable scaling of piezoelectric impact mechanisms based on potential energy versus price. The available energy from an impact mechanism increases faster than the cost. This decreasing marginal cost of the impact mechanism and the low-cost design of the controller make the application of piezoelectric impact centering to grind applications attractive.

Figure 25: Plot shows the potential energy versus the cost of the piezoelectric actuator based on the American Piezo Company product line. The potential energy of the piezoelectric actuator is estimated from the spring energy resulting from rapid expansion.
14. Further Work

There are several avenues of further work that follow this thesis. They include the development of an intelligent control algorithm for tap centering, the formulation of a better understanding of the fundamentals of tap positioning, and the application of tap positioning to two and three dimensional applications outside of centering. Before the piezoelectric impact device can be useful for tap positioning, an intelligent control algorithm must be developed. The skilled machinist who uses a mallet and dial gage is a very sophisticated control system that takes input from such things as the size, shape, mass, and feel of the part, and the look and feel of the platen, and countless other qualitative variables that are learned from experience. It is impossible to quantify and observe all of those factors with a computer, but the computer does have the advantage of computational power. By designing an intelligent algorithm that takes advantage of the precision and repeatability of the piezoelectric impact mechanism by learning from past actions, faster automated centering is accomplished.

A learning algorithm is more powerful with a better understanding of the physics of tap positioning. The simple model of impulse and displacement presented in this paper is a good start, but does not describe the effects of vibration, tribology, or very small displacements. Vibration of the part has a large effect on distance moved and repeatability of positioning. Understanding the effects of resonant vibration of the part could change it from an undesirable, unrepeatable variable to a controlled variable that allows larger displacements. Examining the tribology between the part and the platen might explain some of the error in positioning accuracy; it is suspected that ringing plays a role when the displacements are much smaller than expected. Very small displacements are also hard to model. At scales less than a few microns, surface finish effects become important because the height of surface features is of the same order as the desired displacements. Expanding the impulse-displacement model to include some of these effects would lead to design insight and improvements in the control algorithm.

Another exciting avenue of additional work is looking at applications of tap positioning to existing positioning needs. Precise positioning is an expensive and essential part of many manufacturing techniques. During a conversation with Wesley
Huang from RPI, he described his work on three dimensional positioning of polygonal parts (2002). His work revolved around impact planning, but could benefit from a more precise impact delivery method. Impact positioning could be used for orienting and positioning parts on assembly lines or for aligning wafers in clean rooms.
References


Huang, W. H. Personal interview. 1 November 2002


The Modal Shop, Inc. *Electric Impact Hammer Family, Product Reference*


Appendix Section

Section A. MATLAB script modeling piezoelectric expansion against free mass

%Andrew Wallace
%Timken
%October 4, 2002
%this is a second attempt at modeling the piezo element which ignores
%the mass of the piezo element and treats it as a spring
cler all
close all
ml = .2; %kg,mass of free weight
m2 = 1; %kg,mass of object
b = 1; %believe it or not, damping
L = .028; %length of piezo
time = .08; %ms,total duration of simulation
Vmax = 150; %V,maximum applied voltage to piezo
Dmax = .000020; %m,displacement at Vmax
k = 120000000; %N/m,axial stiffness of material
dt = .0001; %ms,time increment
s1 = zeros(1,(time/dt));
s2 = zeros(1,(time/dt));
v1 = zeros(1,(time/dt));
v2 = zeros(1,(time/dt));
fs = zeros(1,(time/dt));
fd = zeros(1,(time/dt));
s1(1) = 0;
s2(1) = L;
v1(1) = 0;
v2(1) = 0;
volts = zeros(1,(time/dt));
tvec = 0:dt:(time-dt);
C = .0000037; %F, piezoactuator capacitance
R = .5; %ohms, series resistance in dump circuit
tau = R*C; %s,voltage form time constant
Vapp = 150; %applied voltage
for i = 1:time/dt
    volts(i) = Vapp-(Vapp*(exp((-i*dt/1000)/tau)));
    %if (volts(i)>=50)
    %    volts(i) = 50;
    %end
end
dd = volts*(Dmax/Vmax);
for i = 2:(time/dt)
    fs(i) = (((dd(i)+L)-(s2(i-1)-s1(i-1))))*k;
    v1(i) = v1(i-1)-((fs(i)/m1)*(dt/1000));
    v2(i) = v2(i-1)+((fs(i)/m2)*(dt/1000));
    s1(i) = s1(i-1)+(v1(i)*((dt/1000)));  
    s2(i) = s2(i-1)+(v2(i)*((dt/1000)));  
    (time/dt)-i;
end

disp1 = s1;
disp2 = s2-s2(1);
impulse = 0;
for i = 1:length(fs)
    if (fs(i) > 0)
        impulse = impulse + (fs(i)*dt/1000);  
    end
end

figure(1)
subplot(2,1,1);plot(tvec,volts)
title('Driving Voltage and Force versus time')
ylabel('Voltage')
axis([0 time -10 160])
subplot(2,1,2);plot(tvec,fs)
xlabel('time, ms')
ylabel('Force, N')
figure(2)
subplot(2,1,1);plot(tvec,v1)
title('Velocity versus time')
ylabel('vel of free weight')
subplot(2,1,2);plot(tvec,v2)
xlabel('time, ms')
ylabel('vel of object')
figure(3)
subplot(2,1,1);plot(tvec,disp1)
title('Positions versus time')
ylabel('Pos of free weight')
subplot(2,1,2);plot(tvec,disp2)
xlabel('time, ms')
ylabel('Pos of object')
for i = 1:length(fs)
    if (fs(i) < 0)
        fs(i) = 0;
    end
end
for i = 2:length(v2)
    if (v2(i) < v2(i-1))
        v2(i) = max(v2);
    end
end
end
end
tvecmicro = tvec*1000;
fs = fs/1000;
v2 = v2*100;
figure(4)
subplot(3,1,1);plot(tvecmicro,volts)
ylabel('voltage, V')
subplot(3,1,2);plot(tvecmicro,fs)
ylabel('force, kN')
subplot(3,1,3);plot(tvecmicro,v2)
xlabel('time, micro seconds')
ylabel('velocity, cm/s')
Section B. PIC F877 Microchip Code For Voltage-Charge Control

;*******************************************************************************
;* Andrew Wallace
;* Timken
;* PIC16F877
;* October 28, 2002
;* fire control for piezo actuator
;*******************************************************************************

list p=16F877

include <p16f877.inc>

;*******************************************************************************
;* port definitions
;*******************************************************************************

;********port A

capan  equ 00h  ;capacitor voltage
lvdtan  equ 01h  ;lvdt voltage

;********port B

switch  equ 00h  ;momentary switch
msb  equ 05h
smsb  equ 04h
tmsb  equ 03h
lsb  equ 02h

;********port C

tap  equ 00h  ;turn on tap
drain  equ 01h  ;drain capacitor
fire  equ 02h  ;fire FETs
outer  equ 03h  ;outer for debug purposes
pulse  equ 07h  ;encoder pulse

;********port D
;PORTE is configured as 8bit output

;********** port E

;**********************************************************************
;*      memory symbols
;**********************************************************************

FLAG  equ  21h ;FLAG BITS
vcom   equ  22h ;8bit power input
voltage equ 23h ;adconversion result
pcnt   equ  24h ;pulse count
mshi   equ  25h ;boolean input
lshi   equ  26h ;boolean input
mslow  equ  27h ;boolean input
lslow  equ  28h ;boolean input
hinumms equ 29h ;hinum most sig from calc
hinumls equ 2Ah ;hinum least sig from calc
himark equ 2Bh ;EEPROM address of hi point
tester equ 2Ch ;EEPROM value being tested
slicehi equ 2Dh ;hi point in terms of slice
slicelo equ 2 Eh ;diametric opposite of hi
delay equ 2Fh ;delay counter for AD conversion
unaddr equ 71h ;universal address for eeprom
undat  equ 72h ;universal data for eeprom
unread equ 73h ;universal eeprom data read
slice  equ 74h ;slice count 0-63
adchi  equ 75h ;hi result of AD conversion
adclo  equ 76h ;low result of AD conversion
;   equ 77h
;   equ 78h
;   equ 79h

;************************************************************************
;*      flag symbols
;************************************************************************

hun    equ  00h
trudat equ 01h ;boolean was true
kickit equ 02h ;go into actuation mode
meas equ 03h ;measuring rotation
act equ 04h ;actuation rotation

******* Switch Symbols *******

******* subroutines *******

org 0x000
nop
clrwdt
clrf PCLATH
goto starter

looper
; call getdat ;get and organize lvdt data
; rrf adclo,1
; rrf adclo,1
; bcf adclo,7
; bcf adclo,6
; btfsc adchi,1
; bsf adclo,7
; btfsc adchi,0
; bsf adclo,6
; movf adclo,W
; movwf PORTD

; btfss PORTB,switch
; bsf PORTC,fire

movlw B'00000000' ;raw impact strength
movwf vcom
; btfss PORTB,msb
bsf vcom,7
btfss PORTB,smsb
bsf vcom,6
btfss PORTB,tmsb
bsf vcom,5
btfss PORTB,lsb
bsf vcom,4
call adgo ;reads cap voltage

50
call  compare ; compares cap v to vcom

; call  pwait
; call  swread
; call  getdat ; get and organize lvdt data
; rrf  adclo,1
; rrf  adclo,1
; bcf  adclo,7
; bcf  adclo,6
; btfsc adchi,1
; bsf  adclo,7
; btfsc adchi,0
; bsf  adclo,6
; movf  adclo,W
; movwf PORTD

goto looper

measin ; measure initiate
movlw B'00000000'
movwf slice
movwf hinumms ; highest so far
movwf hinumls
movwf slicehi
call inlize

measl ; measure loop

call  pwait
call  getdat
call  comper
bsf  PORTC,outer
movf  slice,1 ; *
btfsc STATUS,2 ; *
goto actin
goto measl

actin
call  pwait
call  calc

bcf  PORTC,outer
movf  slicehi,W
subwf slice,0
btfsc STATUS,2
bsf PORTC,outer

movf slice, 1 ; *
btfss STATUS, 2 ; *
goto actin
movlw B'00000000'
movwf hnumms ; highest so far
movwf hnummls
movwf slicehi
goto measl

 *******************************************
,* Subroutine title
 *******************************************

adgo

banksel ADCON1
movlw B'00000100' ; Left justify, 3 channel
movwf ADCON1

banksel ADCON0
bcf ADCON0, 3
bsf ADCON0, GO

adnotd

btfss PIR1, ADIF ; wait for ad conversion
goto adnotd
movf ADRESH, W
movwf voltage
retlw 00h

; compare maintains proper cap voltage

compare

movf vcom, W ; test if voltage low
subwf voltage, W ; voltage-vcom
btfss STATUS, 0 ; if carry goto tooiow
goto tooiow

movf voltage, W
subwf vcom, W ; vcom-voltage
btfss STATUS, 0
goto toohii
bcf       PORTC,tap
bcf       PORTC,drain
retlw     00h
toolow
bcf       PORTC,drain
bsf       PORTC,tap
retlw     00h
toohi
bcf       PORTC,tap
bsf       PORTC,drain
retlw     00h

.......................
.*                      
.*                    pwait (pulse wait)
.*
.*
.*
.*
.*
.......................

pwait       movlw B'00010000'
movwf pcnt
ndone
stilhi
btfsc      PORTC,pulse
goto      stilhi
stillo
btfss      PORTC,pulse
goto      stillo
decfsz    pcnt,1
goto      ndone
incf      slice,1
movlw B'01000000'
subwf     slice,0
btfsc     STATUS,2
clrf      slice
retlw     00h

.............
.*    boolean comparison
.*
.*

53
boolcom
movf mslow,0
subwf mshi,0
btfss STATUS,0 ;if lo > hi, don't skip
goto notso
movf mshi,0
subwf mslow,0
btfss STATUS,0 ;if hi > lo, don't skip
goto trunuf

movf lslow,0
subwf lshi,0
btfss STATUS,0 ;if lo > hi, don't skip
goto notso

trunuf
bsf FLAG, trudat
retlw 00h

notso
bcf FLAG, trudat
retlw 00h

getdat
banksel ADCON1
movlw B'10000100' ;Right justify, 3 channel
movwf ADCON1

banksel ADCON0 ;
bsf ADCON0,0
bsf ADCON0,3 ;select AN1
movlw B'01000000'  
movwf delayer

adddel
decfsz delayer, 1
goto adddel
bsf ADCON0,GO

adnodo
btfss PIR1, ADIF ;wait for ad conversion
goto adnodo
; bcf  ADCON0,0
movf  ADRESH,W
movwf  adchi
movwf  undat
movf  slice,W
movwf  unaddr
movlw  B'00010000'
addwf  unaddr,1
; call  writer   ***********debug
banksel  ADRESL
movf  ADRESL,W
movwf  adelco
movwf  undat
movf  slice,W
movwf  unaddr
movlw  B'01010000'
addwf  unaddr,1
; call  writer   ***********debug
banksel  PORTA
movlw  B'00100000'
movwf  delayer
adddel2
decfsz  delayer,1
goto  adddel2
retlw  00h

;*************************************************************************
;*
writer
;*************************************************************************

writer
banksel  EECON1
prev
btfsc  EECON1,WR
goto  prev
banksel  EEADR
movf  unaddr,W  ;address to be written to
movwf  EEADR
movf  undat,W   ;data value
movwf  EEDATA
banksel  EECON1
bcf  EECON1,EEPGD
bsf  EECON1,WREN
movlw  0x55

movwf EECON2
movlw 0xAA
movwf EECON2
bsf EECON1,WR
bcf EECON1,WREN
banksel PORTA
retlw 00h

;**************************************************************************
;* reader
;**************************************************************************

reader
banksel EEADR
movf unaddr,W
movwf EEADR
banksel EECON1
bcf EECON1,EEPGD
bsf EECON1,RD
banksel EEDATA
movf EEDATA,W
movwf unread
banksel PORTA
retlw 00h

;**************************************************************************
;* inlize
;**************************************************************************

inlize

inhi
btfsc PORTC,pulse
goto inhi

inlow
btfss PORTC,pulse
goto inlow

call getdat
retlw 00h

;**************************************************************************
;* onerot
;**************************************************************************
onerot
    bsf    PORTC,outer
    goto   onerot

******************************************************
*  swread
******************************************************

swread
    btfsc  PORTB,switch
    retlw  00h
    bsf    FLAG,kickit
    goto   measin
    retlw  00h

******************************************************
*  calc
******************************************************

calc
    movlw  B'00000000'
    movwf  hinumms       ;highest so far
    movwf  hinumls
    movlw  B'00000000'
    movwf  himark        ;address of highest
    movlw  B'00010000'
    movwf  tester

iter
    movf   tester,W
    movwf  unaddr
    call   reader
    movf   unread,W
    movwf  mshi
    movf   tester,W
    movwf  unaddr
    movlw  B'01000000'
    addwf  unaddr,1
    call   reader
    movf   unread,W
movwf lshi

movf hinumms,W
movwf mslow
movf hinumls,W
movwf lslow

call boolcom

btfss FLAG, trudat
goto nothier

; it is higher

movf mshi,W
movwf hinumms
movf lshi,W
movwf hinumls
movf tester,W
movwf himark

nothier

incf tester
movlw B'01010000'
subwf tester,0
btfsc STATUS, 2
goto doncyc
goto iter

doncyc

movf himark,W
movwf slicehi
movlw B'00010000' ; put other stuff in here
subwf slicehi,1
retlw 00h

;******************************************************************************
; comper
;******************************************************************************

comper

movf adchi,W
movwf mshi
movf adclo,W
movwf lshi

movf hinumms,W
movwf mslow
movf hinumls,W
movwf lslow
call boolcom
btfss FLAG, trudat
retlw 00h
movf slice, W
movwf slicehi
movwf slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo
incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
crfl slicelo

incf slicelo, 1 ; increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clr slicelo
incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
clrf slicelo
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo, 0
btfsc STATUS, 2
clrf slicelo

incf slicelo, 1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
cclf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
cclf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
cclf slicelo

incf slicelo,1 ;increment
movlw B'01000000'
subwf slicelo,0
btfsc STATUS,2
cclf slicelo

reqlw 00h

***t**e*********** **** *****

starter

banksel TRISC ;sets port C, 0,1,2,3 to output
movlw B'111110000'
movwf TRISC

banksel TRISD ;sets port D to output
movlw B'00000000'
movwf TRISD
movlw B'111111111' ;sets port b to input
movwf TRISB

banksel ADCON0
movlw B'100000001' ;Fosc/32, A/D enabled
movwf ADCON0

banksel ADCON1
movlw B'00000100' ; Left justify, 3 channel
movwf ADCON1

banksel PORTD
movlw B'00000000'
movwf PORTC
movwf PORTD

clf Flags
movlw B'00000000'
movwf slice

call inlize reinstage ***************

goto looper

END