Design of a PZT-Actuated Proportional Drum Brake

Michael Gogola and Michael Goldfarb, Member, IEEE

Abstract—This paper presents the design of a piezoelectric-ceramic (PZT)-stack-actuated brake that is similar to a magnetic particle brake in dynamic range, size, weight, and cost, while providing a significantly larger bandwidth and requiring significantly less electrical power for a given continuous torque output. The device is essentially a single-pad drum brake that is actuated with a PZT-stack actuator. A significant component of the design is the compliant-mechanism-based transmission utilized to transmit the PZT-stack actuation into brake pad motion. Following the device description, experimental data is presented to characterize the performance of the brake. The performance characteristics are subsequently compared to those of a commercially available magnetic particle brake of comparable size and weight.

Index Terms—Compliant mechanism, magnetic particle brake, piezoelectric ceramic, proportional drum brake.

I. INTRODUCTION

Several control applications require the use of an electrically controlled proportional rotary brake (e.g., see [1]–[6]). Probably the most common and thoroughly developed example of such a device is the magnetic particle brake. Magnetic particle brakes produce a steady-state resistive torque roughly proportional to the input current. A sectioned view of a magnetic particle brake is shown in Fig. 1. DC current applied to the brake coil induces a magnetic field which links fine ferrite particles to the rotating brake shaft. The amount of current in the coil determines the strength of the magnetic field, which, in turn, determines the resistive torque imposed on the brake shaft. Compared with the closed-loop control of a high-performance dc torque motor, these devices provide a relatively low-power, lightweight, and highly stable means of exerting controlled dissipative mechanical torque. Due to these characteristics, magnetic particle brakes often find use in applications where minimal weight and power consumption are paramount.

Although electrical power consumption is low relative to a (velocity controlled) dc motor (for a given resistive torque), in many cases the electrical power required remains significant. In fact, in applications requiring minimal power consumption, the power required to impose a continuous resistive torque is probably the most significant limitation of a magnetic particle brake. Specifically, since a particle brake is electrically resistive in nature, the continuous current required to provide a resistive torque will result in continuous electrical power dissipation.

In applications requiring large control bandwidths, another significant limitation of the particle brake is the bandwidth of the dynamic between the coil current and the brake torque. Specifically, unlike a dc motor, the (instantaneous) output torque of the particle brake is not directly proportional to the input current. Rather, a dynamic exists between the input current and the brake torque (due to particle compaction), typically on the order of tens of milliseconds, which, in turn, limits the bandwidth of these devices. The current-to-torque dynamics for a small (and, therefore, relatively fast) magnetic particle brake are shown in Fig. 2. Specifically, the current-to-torque dynamic is the slower dynamic that follows the initial constant rate rise in torque. Note that the dynamic limitations of the servoamplifier response are those of the initial constant rate rise in torque, and not the (approximately) exponential rise that follows.

One means of providing low dc power consumption and large bandwidth in a rotary brake is to actuate the brake with a piezoelectric-ceramic (PZT)-stack actuator. Since PZT is essentially electrically capacitive in nature, a sustained output does not require a continuous current and, thus, does not require the associated electrical power dissipation. Also, a typical unloaded PZT-stack actuator exhibits a bandwidth on the order of tens of kilohertz, which is quite large relative to a typical mechanical control system. A stack also has an output force sufficient to create a braking torque that is comparable to current commercial brake designs. Further, a typical PZT stack is compact and should result in a relatively small brake size. Additionally, unlike a particle brake, a PZT-actuated brake need not be hermetically sealed and, therefore, will not entail
the frictional torque of sealed bearings which limits the low end of the dynamic range of particle brakes.

This paper presents the design of a piezoelectric-actuated brake that is similar to a magnetic particle brake in dynamic range, size, weight, and cost, while providing a significantly larger bandwidth and requiring significantly less electrical power for a given continuous torque output. The performance of the brake is characterized and compared with a commercially available magnetic particle brake of comparable size and weight.

II. PZT-ACTUATED BRAKE DESIGN

Despite the aforementioned advantages of piezoelectric actuation, use of a PZT stack as a brake actuator presents a significant design challenge. The primary challenge is that the strain-based deformations provided by the PZT actuator are limited to somewhat less than 0.1% strain, or 1/1000 of the actuator’s length. A commercially available stack 20 mm in length, for example, will provide approximately 15 μm of no-load displacement. Since the minimum distance between a brake pad and rotor countersurface is limited by machining and assembly tolerances to tens of micrometers, incorporating a PZT stack for brake actuation will require some form of mechanical transmission for displacement amplification. Additionally, since the brake pad must impose a significant normal force on the countersurface with limited input motion, the mechanical transmission must exhibit a large output impedance.

A. Brake Configuration

The general idea of the PZT-actuated brake design is to utilize a PZT stack to actuate a brake pad against a rotating countersurface. The authors considered drum brake and disc brake configurations, and opted for the drum configuration for two primary reasons. First, the authors were able to achieve significantly less dimensional variation in the precision machining of a drum surface relative to a disc surface, which is a significant advantage considering the limited displacement provided by a PZT stack. Specifically, the variation in the drum surface was on the order of 10 μm, while the variation in the disc surface at approximately the same radial distance was on the order of 100 μm. Second, in a drum brake configuration, the frictional forces between the pad and countersurface lie in a plane that is parallel to the drum, as opposed to a disc type, where the frictional forces lie in a plane that is perpendicular to the disc. This characteristic is significant when incorporating the use of a compliant mechanism actuation transmission (subsequently described), since flexure joints can exhibit significant compliance when subjected to out-of-plane loading. A compliant mechanism transmission for a disc configuration should, therefore, be oriented perpendicular to the disc, while a compliant mechanism transmission for a drum configuration should be oriented parallel to the drum, thus lending itself to a more compact design.

The PZT-actuated brake design is, therefore, based on a single-pad drum brake configuration, and is shown in cross-section in Fig. 3. The brake has six main components: the drum, the compliant-mechanism-based transmission (the actuation mechanism), the housing, the back plate, and two support bearings. The drum serves as the rotor and the output shaft. The two bearings that support the drum are in a cantilever-type arrangement, and are preloaded to eliminate backlash. The actuation mechanism is located inside the drum and the frictional contact occurs between the brake pad and the inner drum surface. The PZT actuation mechanism is attached to the back plate, which grounds the mechanism to the brake housing. Fig. 4 shows the assembled brake, which measures approximately 7.1 cm in diameter and 3.8 cm in length, and weighs approximately 0.56 kg.

B. Compliant Mechanism Transmission Design

A compliant mechanism is used to amplify the PZT displacement and, thus, provide a usable amount of brake pad travel. Compliant mechanisms are devices that attain motion by means of elastic deformation, most commonly from the combination of rigid links and lumped compliance at flexible
Compliant mechanisms are particularly well suited for use with PZT-stack actuators. Perhaps the most significant beneficial characteristic of compliant mechanisms is the complete absence of backlash, which is particularly important given the limited displacement provided by the PZT stack. The loss of even a few micrometers of motion would be a significant percentage of the stack input. A compliant mechanism is also devoid of Coulomb friction, which significantly enhances the resolution of the mechanism output. An additional benefit offered by a compliant mechanism is the inherent elasticity of the mechanism. A PZT stack can sustain compression, but not tension, and, therefore, can provide a forced expansion, but not a forced retraction. Retraction is, therefore, provided by the elasticity of the mechanism, which also acts to keep the stack in compression. In addition to these characteristics, compliant mechanisms are free of lubricants and can also decrease the cost of assembly in manufacture, since it is often possible to fabricate a compliant mechanism as a monolithic device (i.e., as a single piece).

The compliant mechanism transmission used in the PZT-actuated brake is shown in Fig. 6. The shaded regions of Fig. 6 represent the grounded portions of the mechanism, which are rigidly attached to the back plate. The mechanism is symmetric about the brake pad, so that the brake torque is independent of the direction of drum rotation. A kinematic half-model of the linkage is shown in Fig. 7. The mechanism is actuated by a single PZT 5 mm × 5 mm × 20 mm stack (Tokin model AE0505D16) with output characteristics as shown in Fig. 8. Functionally, actuation (elongation) of the stack pushes section 1 downward. section 2 acts as an amplification lever between the input (section 1) and the output (section 3), which increases the displacement provided by the PZT to approximately 100 µm of brake pad motion. When section 3 moves upward and the brake pad contacts the rotating drum, the normal force exerted produces a friction force in the tangential direction. To counteract this tangential load and stabilize the mechanism, section 3 is attached to the upper ground section through two long, thin sections that act as prismatic flexures. These prismatic flexures offer only a small amount of stiffness in the actuation direction (normal to the drum surface), and a large stiffness in the frictional load direction (tangential to the surface). The compliant mechanism transmission was electrically discharged machined (EDM) from a single piece of AISI 4142 heat-treated steel.
C. Design Optimization

Despite the advantages provided by compliant mechanisms, they have several deficiencies relative to a conventional mechanism. One of the most significant is that the elasticity that provides for retraction also decreases the amount of actuator force that is transmitted to the output. The amount of force required to displace the mechanism can be minimized by fabricating joints of minimal thickness, but this approach introduces another significant problem with compliant mechanisms. Specifically, the joints should ideally be highly compliant in the direction of kinematic motion (i.e., in bending), but essentially rigid in tension and compression (i.e., along the axial direction of the links). Both stiffness in bending and compliance in the axial direction result in lost force at the output, due, in essence, to elastic energy stored in the mechanism. Unfortunately, the geometric properties of a conventional flexure are such that, since both the bending and axial stiffness properties depend on essentially the same geometric variables, the two cannot be independently selected. The problem thus becomes a design tradeoff, which can be addressed with a design optimization. Specifically, the objective of the optimization problem in the design of the brake actuation mechanism is to maximize the brake output torque for a given PZT input by optimizing the flexure hinge dimensions in combination with the kinematic dimensions of the mechanism.

The analytical design optimization is based upon the quasi-static mechanism model illustrated in Fig. 9, which represents one-half of the compliant mechanism transmission. The lever represents section 2 (as shown in Fig. 6) and is assumed rigid (relative to flexure compliance). The geometry of each flexure hinge is given by $h_z$, $h_y$, and $L_z$, as illustrated in Fig. 5. Note that the hinge curvature is assumed to be of constant radius. The axial and bending stiffness of each hinge, $K_{a}$ and $K_{b}$, respectively, are given by expressions derived by Paros and Weisbord [9] and listed in the Appendix. The relative locations of the flexure hinges, given by $d_1$ and $d_2$, define the displacement amplification attained from the lever by

$$n = \frac{d_2}{d_1},$$

where $x_{rim}$ is the amount of deflection of the rim. Equation (2) is the objective function for the optimization problem, since the objective is to maximize the contact force for a given input. The output force is a function of the hinge dimensions of the three flexures and the amplification ratio, which are all subjected to physical constraints. The lower limits of the hinge thickness and length are constrained by the EDM machining process, which can produce a hinge of minimum thickness of 0.2 mm and a minimum length of 0.305 mm. The maximum amplification ratio is constrained by the mechanism outer dimension (18.75 mm) and the minimum dimension between two hinges (1.6 mm, determined by the geometry of the base of the hinges). The maximum possible displacement amplification is, therefore, 11.5. In addition to these geometric constraints, the hinges are constrained to not exceed a maximum allowable stress for the AISI 4142 steel.

The optimization was performed in MATLAB using an exhaustive search method to find the solution, which is listed in Table I. Notice that $h_1$, $L_1$, $L_2$, and $L_3$ all converged to the constrained lower limit, and the displacement amplification converged to the constrained maximum limit. The remaining parameters ($h_2$ and $h_3$) assumed intermediate values, as given in the table. The model produced an output force of 34 N. Since this model is a half-model representation of the system, the predicted total output would be 68 N. Approximating the
friction coefficient to be 0.45, this would result in an output torque of 770 N\(\text{mm}\) (6.9 in\(\text{lbs}\)).

### D. Finite-Element Analysis

A two-dimensional (2-D) quasi-static finite-element model (FEM) was used to verify the results of the optimization and to evaluate the levels of material stress. A 2-D model was used since the mechanism undergoes planar loading. Dynamic behavior was not considered in the analysis, since, typically, operation should be well below the first mode of the PZT and structure.

The PZT stack was modeled as a stiffness in parallel with a force source (as shown in Fig. 9), which will generate the output characteristics as depicted in Fig. 8. Specifically, the mechanism was extended in a section of equivalent stiffness to the PZT (shown in Fig. 10), and a distributed load of 1000 N (representing the maximum clamped output of the stack) was applied to the interface between the mechanism and PZT stack. The behavior of the sliding surfaces at the output was approximated using a two-step iterative process. First, the compliance of the drum was approximated analytically and represented as a 2-D cantilever beam attached to the mechanism at the output. The grounded ends of the beam were constrained to move in the actuation direction a distance equal to the initial gap distance at the output. The beam was unconstrained in the orthogonal direction to allow the mechanism to deform laterally under the frictional load. The mechanism was actuated to determine the normal load that resulted from the deflection of the beam. These data were then used to calculate a friction load, where the friction force was equal to the product of the normal force and friction coefficient, which was approximated as 0.45. This calculated friction load was then applied to the contact point in the appropriate direction. The analysis was then repeated by applying the calculated frictional load, until the applied frictional load matched the load predicted by the model.

The FEM of the actuation mechanism under maximum loading conditions is shown in Fig. 10. The model has a total of approximately 13,000 thin-shell linear quadrilateral elements and 14,000 nodes. The analysis predicted a normal force of 42.6 N and a frictional force of 19.2 N, which would result in an output torque of 480 N\(\text{mm}\) (4.3 in\(\text{lbs}\)). The FEM therefore predicted an output torque approximately 60\% of the analytical model utilized for optimization, which predicted 770 N\(\text{mm}\) (6.9 in\(\text{lbs}\)). This difference is most likely due to the fact that the analytical model assumed rigid links, which is typically a marginal assumption when designing compliant mechanisms for high output impedance environments [17]. The FEM predicted a maximum stress (in the pivot hinge) of 329 MPa, which is well below the stress limit of the heat-treated AISI 4142 material. Given the predicted maximum stress and the strength properties of the steel, the fatigue life was estimated to be on the order of a million cycles.

### E. Selection of the Brake Pad

The success of the brake design is also dependent on the material selection and design of the two contacting frictional surfaces. Since the output displacement is limited, material wear should be minimized. Additionally, since the maximum brake torque is directly proportional to the friction coefficient (for a given maximum normal force), the frictional coefficient of the two contacting surfaces was maximized. The contacting surfaces were, therefore, designed to minimize wear, while still retaining a large friction coefficient. A zirconium oxide ZY4Ce4 ceramic composite brake pad contacting a hardened steel drum was chosen as the design solution. This combination provides extremely high wear resistance and a coefficient of friction of approximately 0.45. For a detailed treatment of the issues involved in the tribology of these sliding surfaces, see [18].

### III. Performance Characterization

The flexure-based actuation mechanism was tested independently (i.e., without the drum) to determine its maximum attainable displacement using an optical displacement sensor.
A maximum displacement of 80 μm was measured, approximately 80% of the predicted FEM (no load) displacement. The discrepancy between the theoretical and experiment results is most likely due to inaccurate interface approximations in the FEM. Specifically, the grounded portions of the mechanism in the FEM were constrained to have no displacement. In the actual prototype, however, compliance exists in the grounded sections, and small deformations in these sections significantly affect the output.

The experimental setup used to test the brake torque is shown in Fig. 11. The brake shaft was driven at a constant rate with a dc motor (PMI model N12M4T), and the brake torque was measured with a torsional load cell mounted between the two. The dc motor was driven by a constant-velocity feedback loop designed for maximum disturbance rejection. Rotational speed was measured during the experiments and verified to be essentially constant and independent of brake torque. The load cell utilized for torque measurement was a custom-built strain-gage-based load cell.

Fig. 12 shows the maximum and minimum (low end) steady-state torques of the PZT brake, measured at a shaft speed of approximately 25 r/min. Additional tests demonstrated that this output torque was repeatable and independent of speed. The average maximum torque of the brake was 180 N-mm (1.6 in-lbs) with approximately 5% torque ripple, and required a voltage input of approximately 200 V. The maximum torque was lower than expected, approximately 40% of that predicted by the finite-element analysis. Several possible factors may have contributed to this difference. As was stated earlier, the most significant effect that was not considered in the FEM model was the compliance in the grounded sections of the actuation mechanism. Additionally, the drum/housing assembly may have been more compliant than analytically predicted. Further, due to limited output displacement of the actuation mechanism and machining and assembly tolerances, imprecise positioning of the brake pad relative to the drum surface may have also decreased performance (i.e., positioning the brake pad ~10 μm from the drum surface proved difficult).

Since the total motion of the brake pad is on the order of tens of micrometers, the actuation mechanism must have a high output impedance in order to exert a significant force on the drum. As a result, small variations in the concentricity of the drum generate noticeable torque ripple. It should be noted, however, that the average torque ripple in a high-performance dc motor is typically 7%–10% (e.g., typical Inland Motor dc motor).

Fig. 13 shows a rising and a falling step response of the PZT brake for a voltage input. Defining rise time as the time required for the response to rise from 10% to 90% of the initial steady-state low-end value to the final steady-state step value, and the fall time as the reverse, the PZT brake demonstrated a rise time of 2.6 ms and a fall time of 2.1 ms. Recall that the rise time and fall time are not necessarily the same for a PZT-actuated compliant mechanism, since one direction is forced by the actuator and the other is dictated by the elastic response of the mechanism.
To place the performance of the PZT-actuated brake in context, the performance characteristics were compared with those of a commercially available magnetic particle brake of comparable size. The particle brake utilized for the comparison was a Placid Industries model B2, which measures approximately 5.3 cm in diameter by 3.0 cm in length, as compared with the PZT brake which measures 7.1 cm in diameter and 3.8 cm in length. The two are also of comparable mass, 0.45 kg for the particle brake versus 0.56 kg for the PZT brake.

Fig. 14 shows the step responses of both the PZT brake and the magnetic particle brake for a voltage input. As seen in the figure, the PZT brake has a rise time of approximately 2.6 ms, as compared to the particle brake, which has a rise time of approximately 210 ms. Utilizing the rise time as an indicator of bandwidth, these results would indicate that the PZT brake has a bandwidth approximately 80 times the voltage-controlled magnetic particle brake. As mentioned previously, however, the speed of response of the particle brake can be increased by utilizing a current input. Fig. 14 also shows the step response of the particle brake to a current-controlled input. The input was generated by a Kepco model BOP36-6M amplifier with a maximum voltage output of ±36 V. Note that, in commanding a current step into an inductive load, the rate of change of the current is limited by the maximum output voltage of the source. This (constant) rate can be seen in the current step response of Fig. 14. Note that, after the current is at the source. This (constant) rate can be seen in the current step response of Fig. 14.

Table II summarizes the performance characteristics of the B2 magnetic particle brake and the PZT brake. As mentioned previously, however, the speed of response of the particle brake can be increased by utilizing a current input. Fig. 14 also shows the step response of the particle brake to a current-controlled input. The input was generated by a Kepco model BOP36-6M amplifier with a maximum voltage output of ±36 V. Note that, in commanding a current step into an inductive load, the rate of change of the current is limited by the maximum output voltage of the source. This (constant) rate can be seen in the current step response of Fig. 14. Note that, after the current is at the commanded level, the torque still exhibits the slow (particle) dynamic that was discussed in the introduction. As seen in the figure, the current-controlled particle brake has a rise time of approximately 100 ms and, thus, the PZT brake has a bandwidth approximately 40 times the current-controlled magnetic particle brake.

Another significant advantage of the PZT brake, as mentioned previously, is the fact that the piezoelectric ceramic is essentially capacitive in nature and, thus, requires no electrical power consumption for a steady-state torque output. In fact, in running the characterization experiments, the PZT brake would remain locked after the power was turned off, and would continue to remain locked until the lead was discharged to ground. Conversely, the magnetic particle brake requires continuous current and, thus, continuous electrical power consumption, in order to exert a steady-state torque. For example, the magnetic particle brake utilized in the performance comparison would require approximately 0.25 W of continuous electrical power to exert a steady-state torque of 110 N-mm (1.0 in-lbs). For applications requiring high duty-cycle torque and minimal electrical power consumption, the PZT brake, therefore, offers a significant advantage.

Despite these advantages, the PZT brake does not match the performance of the particle brake in maximum torque or amount of torque ripple. Specifically, the commercial particle brake utilized in the comparison exhibits a maximum rated torque of 280 N-mm (2.5 in-lbs), which is approximately 1.5 times that of the PZT brake and, additionally, exhibits negligible torque ripple. Another drawback of the PZT brake relative to the voltage-controlled particle brake is that the PZT brake requires voltages on the order of 100 V, while the particle brake requires voltages around 10 V. Regarding audible noise, both brakes offer the benefit of essentially silent operation. Table II summarizes the performance characteristics of the PZT brake versus those of the magnetic particle brake.

V. CONCLUSIONS AND RECOMMENDATIONS

The authors have presented a design for a PZT-actuated proportional rotary brake that offers some improved properties relative to a comparable magnetic particle brake. Specifically, the PZT brake exhibits a significantly larger bandwidth and requires significantly less electrical power for steady-state operation. Despite these advantages, the PZT brake does not match the performance of the particle brake in maximum torque or amount of torque ripple. Both characteristics, however, could be improved significantly in the PZT brake with relatively minor changes. Specifically, the maximum torque is directly related to the maximum force of the PZT stack, which, in turn, is directly proportional to the stack cross-sectional area. A 40% increase in the width of a square cross section stack (e.g., an increase from 5 to 7 mm) would, therefore, double the maximum torque of the PZT brake.

Table II: Comparison of the B2 Magnetic Particle Brake with the PZT-Actuated Brake

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>B2 MPB</th>
<th>PZT Brake</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max Torque [N-mm] (in-lbs.)</td>
<td>280 (2.5)</td>
<td>180 (1.6)</td>
</tr>
<tr>
<td>Min Torque [N-mm] (in-lbs.)</td>
<td>7.1 (0.063)</td>
<td>5.6 (0.05)</td>
</tr>
<tr>
<td>Rise Time [s], Voltage Control</td>
<td>0.21</td>
<td>0.0026</td>
</tr>
<tr>
<td>Current Control</td>
<td>0.10</td>
<td>-</td>
</tr>
<tr>
<td>Steady-State Power</td>
<td>0.25</td>
<td>-</td>
</tr>
<tr>
<td>Consumption @ 110 N-mm [W]</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Weight [kg] (lbs.)</td>
<td>0.45 (1.0)</td>
<td>0.56 (1.2)</td>
</tr>
<tr>
<td>Diameter [mm] (in.)</td>
<td>53 (2.1)</td>
<td>71 (2.8)</td>
</tr>
<tr>
<td>Length [mm] (in.)</td>
<td>30 (1.2)</td>
<td>38 (1.5)</td>
</tr>
</tbody>
</table>
Additionally, the torque ripple can be reduced with higher tolerance machining of the drum.

**APPENDIX**

The axial and bending stiffness of a flexure hinge, $K_a$ and $K_b$, respectively, are given by the following expressions derived by Paros and Weisbord [9]:

$$K_a = \frac{1}{C_a} \quad (A1)$$

where

$$C = \frac{1}{Eb} \left[ -2 \tan^{-1} \frac{\gamma - \beta}{\sqrt{1 - (1 + \beta - \gamma)^2}} + \frac{2(1 + \beta)}{\sqrt{2 \beta + \beta^2}} \times \tan^{-1} \left( \frac{2 + \beta}{\beta} \times \frac{\gamma - \beta}{\sqrt{1 - (1 + \beta - \gamma)^2}} \right) \right]$$

and

$$K_b = \frac{1}{C_b} \quad (A3)$$

where

$$C_b = \frac{3}{2EbR^2} \left[ \frac{1}{2 \beta + \beta^2} \left\{ \frac{1 + \beta}{\gamma^2} + \frac{3 + 2 \beta + \beta^2}{\gamma(2 \beta + \beta^2)} \right\} \times \left[ \tan^{-1} \left( \frac{2 + \beta}{\beta} \times \frac{\gamma - \beta}{\sqrt{1 - (1 + \beta - \gamma)^2}} \right) \right] \right]$$

$$R = \frac{L}{2} \quad (A5)$$

$$\beta = \frac{h}{L} \quad (A6)$$

$$\gamma = \frac{h + L}{L} \quad (A7)$$

**REFERENCES**


