An Ultrasonic Piezoelectric Motor Utilizing Axial-Torsional Coupling in a Pretwisted Non-Circular Cross-Sectioned Prismatic Beam

David Wajchman, Kuang-Chen Liu, James Friend, Member, IEEE, and Leslie Yeo

Abstract—A rotary piezoelectric motor design with simple structural components and the potential for miniaturization using a pretwisted beam stator is demonstrated in this paper. The beam acts as a vibration converter to transform axial vibration input from a piezoelectric element into combined axial-torsional vibration. The axial vibration of the stator modulates the torsional friction forces transmitted to the rotor. Prototype stators measuring \(6.5 \times 6.5 \times 67.5 \text{ mm} \) were constructed using aluminum \((2024-T6)\) twisted beams with rectangular cross-section and multilayer piezoelectric actuators. The stall torque and no-load speed attained for a rectangular beam with an aspect ratio of 1.44 and pretwist helix angle of 17.7° were 0.17 mNm and 840 rpm with inputs of 184.4 kHz and 149 mW, respectively. Operation in both clockwise and counterclockwise directions was obtained by choosing either 70.37 or 184.4 kHz for the operating frequency. The effects of rotor preload and power input on motor performance were investigated experimentally. The results suggest that motor efficiency is higher at low power input, and that efficiency increases with preload to a maximum beyond which it begins to drop.

I. INTRODUCTION

The continued research on motor miniaturization stems from the possibilities that would be enabled by practical motors in the submillimeter to micrometer scale. In particular, motors at the submillimeter length scale would enable a wide variety of biomedical applications. For example, micro-robots small enough to swim within the circulatory or digestive systems of the human body could perform tasks such as monitoring, drug delivery, and surgical procedures, expanding the repertoire of tools available for minimally invasive and noninvasive surgery [1].

Conventional electromagnetic motors down to \(\phi 2 \text{ mm}\) in diameter with a stall torque of the order of 10 \(\mu\text{Nm}\) are commercially available [2], [3]. However, further miniaturization requires technology not currently available for the fabrication of its coil components, and the scaling of magnetic fields is not favorable for their use at smaller scales in any case. In the late 1980s, electrostatic micromotors of a few hundred micrometers in diameter were made possible through the use of silicon micro-fabrication techniques [4]. However, these generate very low torque at high speed (in the order of 10 pNm and 10,000 rpm) [5], requiring reduction gears accompanied by the problem of friction and wear, especially relevant at these scales.

In comparison to electrostatic motors, piezoelectric motors offer promising alternatives with the advantage of high torque-to-speed ratio (eschewing the need for gears), a holding torque without power expenditure, and low electromagnetic interference. The chief problem hindering their miniaturization, as with electromagnetic motors, lies with the difficulty of fabrication. However, with improvements in the machining and film deposition techniques of piezoelectric material, and the addition of some ingenuity in the design of the driving mechanism [6], [7], reduction in size to the few to submillimeter scale is not beyond reach [8], [9].

The class of piezoelectric motor targeted for miniaturization in this study is the hybrid ultrasonic motor [10]. These motors employ a combination of torsional and axial vibration in the stator such that the frictional force transmitted to the rotor is greater during the “upstroke” than the “downstroke,” imparting a non-zero net torque on the rotor. The direction of rotation can be changed by reversing the phase difference between the axial and the torsional vibration. Conventional hybrid ultrasonic motors use both axially and circumferentially poled piezoelectric disks to generate the desired stator motion. However, the complex structure of the torsional piezoelectric element makes it very difficult to fabricate at the submillimeter scale.

Tsujino et al. [11] demonstrated that the torsional piezoelectric element can be eliminated through the use of a stator with coupled axial-torsional deformation, acting as a vibration converter to generate the desired stator motion from only axially poled piezoelectric elements. However, the helically slotted cylinders used as vibration converters in their centimeter-scale motors may be difficult to fabricate for micromotors.

Friend et al. [12] proposed the use of pretwisted beams with non-circular cross sections as vibration converters that are simpler to fabricate on the microscale. As a first step toward a submillimeter-scale micromotor, the proposed hybrid ultrasonic motor is constructed and tested in this study at the centimeter-scale to investigate its performance characteristics (e.g., efficiency, torque-speed curve) and how they are influenced by rotor preload and input power.

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TABLE I
FINITE ELEMENT ANALYSIS RESULTS OF THE VARIATION OF THE
FUNDAMENTAL AXIAL RESONANCE \(f_{A,1}\) AND THE SECOND
TORSIONAL HARMONICS \(f_{T,2}\) FOR RECTANGULAR PRISMATIC BEAMS
AS ASPECT RATIO (h/w) IS VARIED (\(\nu = 0.33, L = 20\ m, h = 1\ m\)).

<table>
<thead>
<tr>
<th>Aspect ratio</th>
<th>(f_{A,1}) (Hz)</th>
<th>(f_{T,2}) (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.00</td>
<td>127.8</td>
<td>144.0</td>
</tr>
<tr>
<td>1.50</td>
<td>127.7</td>
<td>133.3</td>
</tr>
<tr>
<td>1.66</td>
<td>127.7</td>
<td>127.7</td>
</tr>
<tr>
<td>2.00</td>
<td>127.7</td>
<td>116.3</td>
</tr>
</tbody>
</table>

II. DESIGN AND FABRICATION

In this preliminary study, only the core components of the micromotor were constructed; the prototype consists of an axially poled multilayer piezoelectric actuator (MLPA; AE050510; NEC/Tokin, Tokyo, Japan) bonded to a non-circular pretwisted beam with the rotor (a steel ball) balanced on top of the pretwisted beam.

The operation of the motor relies on the coupling between the axial and torsional deformation of pretwisted beams. Various models and techniques have been used by researchers to explain the phenomenon, from the helical-fiber assumption and the shell theory to finite element analysis [13], [14]. Since the late 1970s, most beam theories attributed the main source of the coupling to the St. Venant warping function of non-circular cross sections [14]–[16]. However, the validity of the theories involving the warping function appears limited to a small amount of pretwist [17]. Thus finite element analysis (FEA) was used in this study.

The elimination of the torsional piezoelectric element means that the direct control over the phase difference between the axial and torsional vibration is lost. The phase reversal required for bidirectional operation now depends on the beam having different harmonics with suitable vibration mode shapes and phase lags [18].

For an efficient conversion of the axial vibration input from the MLPA into coupled axial-torsional motion at the stator-rotor interface, the motor needs to operate in a resonance mode (ideally with matched torsional \(f_T\) and axial \(f_A\) resonance frequencies). Preliminary FEA was carried out on the pretwisted beam to investigate the effect of beam geometry on the resonance frequency and the mode shape.

The effect of changing the cross-section aspect ratio (\(AR = \text{height/width}\)) of the rectangular prismatic (non-twisted) beams on \(f_T\) and \(f_A\) is shown in Table I. The FEA-based result showed that \(f_A\) is independent of \(AR\), while \(f_T\) is maximized when \(AR = 1\) (square beams) and decreases monotonically as \(AR\) increases. Most notably, it was shown that if \(AR = 1.66\), the fundamental axial resonance is matched with the second torsional harmonic under free-free boundary conditions. Since pretwist was shown to increase \(f_T\) and decrease \(f_A\) [14], [17], the aspect ratio necessary to match the resonance frequencies of the two desired modes is expected to be greater than 1.66 in the presence of pretwist.

As shown in Fig. 1, the motor features a piece of axially poled MLPA (6.5 × 6.5 × 12 mm) bonded to a pretwisted aluminium beam using cyanoacrylate glue. Four beams measuring 47.4 ± 0.3 mm in length, with different aspect ratios and pretwist, were tested for efficiency comparison (geometric details are shown in Table II). Note that a countersink of 135° included angle was machined into the beam tip to support the rotor—a 20-mm-diameter steel ball. The beam was formed by twisting a machined long and straight aluminium beam in a jeweller’s lathe until the required amount of pretwist was achieved. The beam was then cut to the correct length. The rate of twist along the length of the beam is approximately constant using this method.

![Fig. 1. Motor dimensions and the dotted line along which vibration measurements were made for (a) torsional/flexural vibration and (b) axial vibration. Note that the figure is not to scale, and the axis of vibration detected by the MSA 400 is in and out of the page.](image)
### III. Experimental Methods

#### A. Vibration Modes and Resonance Frequencies

The resonance frequencies of the beam and the associated mode shapes were determined through the combined use of an impedance analyzer (4294A; Agilent Technologies, Inc., Santa Clara, CA), a scanning 3D laser Doppler vibrometer (MSA-400; Polytec GmbH, Waldbronn, Germany) and visual inspection.

Among the resonances, either a clockwise or a counterclockwise rotor rotation (defined in terms of the top-down view) is generated, depending on the phase difference between the axial and torsional vibration. The MSA-400 was thus used to visualize the beam vibration. The vibrometer was used to measure the out-of-plane vibration velocity along the dotted lines shown in Fig. 1; the “axial line” allowed the detection of axial and $y$-axis bending vibration while the “torsional line” detected torsional and $x$-axis bending vibration. An accurate determination of the torsional resonance frequency can also be performed using the continuous measurement function of the vibrometer by adjusting the frequency of the signal generator and noting the frequencies that produce the highest torsional vibration velocity.

#### B. Torque-Speed Measurements

To investigate the effect of rotor preload and input power on the performance characteristics of the motor, an experimental method similar to that proposed by Friend et al. [19] was used. The motor was set up as shown in Fig. 2, with the MLPA end of the stator assembly loosely attached to the surface of an electronic scale using doublesided tape, and the steel ball balanced on the tip of the pretwisted beam as the rotor. Note that beam 2 was selected for this experiment due to its lower $AR$, which makes it easier to balance the rotor. With the electronic scale tared before the steel ball was placed upon the stator, a direct measurement of the preload was made.

### IV. Results and Discussion

#### A. Vibrometer Results

Frequency scans up to the 230-kHz range were performed on the four motor-beam configurations; Table III lists the frequencies where significant rotation is generated. A comparison of beam 2’s vibration results at the clockwise frequency (70.3 kHz) and counterclockwise frequency (183.3 kHz) is shown in Fig. 3. The measurements were made along the dotted lines in Fig. 1. The nearly odd-symmetry seen along the torsional line shows that torsional
motion dominates over \( x \)-axis bending vibration; however, the slight imbalance in vibration velocity along the axial line (most likely caused by inaccuracies in the fabrication of the beams) suggests the presence of a small amount of \( y \)-axis bending vibration, which may explain the rotor’s tendency to wobble under very low preloads.

The results of the two frequencies are aligned so that the torsional modes are shown in phase with each other. The phase lag \( \phi \) of the axial velocity relative to the torsional velocity is seen by inspection to be \( \phi \approx 45^\circ \) at 70.37 kHz and \( \phi \approx -45^\circ \) at 184.4 kHz. The torque transmitted to the rotor is the largest over the part of the cycle where the axial acceleration is positive upwards, which occurs for 184.4 kHz when \(-160^\circ < \theta < 20^\circ\), and for 70.4 kHz when \(-90^\circ < \theta < 90^\circ\). This explains why the ball spins in opposite directions from 70.4 kHz to 184.4 kHz.

### B. Stall Torque and Steady-State Speed Estimate

A stator-rotor model with sufficient complexity to predict the motor performance is beyond the scope of this paper. However, simple estimates of the stall torque and the steady-state rotor speed can be made based on the vibration velocity of the stator.

With an input frequency in the order of 100 kHz, and axial vibration amplitude in the order of 1 \( \mu \)m, the stator’s axial acceleration would be in the order of \( 10^6 \) m/s\(^2\), compared to the 10 m/s\(^2\) gravitational acceleration. When the axial acceleration of the stator is greater than the gravitational acceleration holding the rotor and stator together, the rotor trajectory will be one of repeated separation and collision with the stator. Frictional impulse then becomes the chief torque transmission mechanism between the stator and the rotor.

**Table IV**

<table>
<thead>
<tr>
<th>Power input (mW)</th>
<th>Stator speed (rad/s)</th>
<th>Rotor speed (rad/s)</th>
<th>Speed ratio</th>
<th>Stall torque (mNm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>60.6</td>
<td>151</td>
<td>97</td>
<td>0.64</td>
<td>0.21</td>
</tr>
<tr>
<td>84.8</td>
<td>187</td>
<td>110</td>
<td>0.59</td>
<td>0.25</td>
</tr>
<tr>
<td>96.8</td>
<td>176</td>
<td>120</td>
<td>0.68</td>
<td>0.28</td>
</tr>
</tbody>
</table>

Assuming uniform pressure is present over the stator-rotor contact, the frictional torque impulse \( \hat{T} \) is related to the axial impulse \( \hat{F} \) by the following equation:

\[
\hat{T} = r_c \mu_k \hat{F}, \quad (2)
\]

where \( r_c \approx 1 \) mm is the radius of the countersink on the beam tip and represents the radius of the contact line, and \( \mu_k \approx 0.5 \) is the kinetic coefficient of friction between steel and aluminum [21]. Assuming that the stator mass is much greater than the rotor mass, and that the coefficient of restitution is zero, then the separation and landing speed of the rotor would be equal to the stator’s axial vibration speed \( \hat{u}_s \). Thus the axial impulse \( \hat{F} \) is

\[
\hat{F} = 2m_r \hat{u}_s, \quad (3)
\]

where \( m_r = 32 \) g is the mass of the rotor. The rotational speed of the rotor before \( \omega^+ \) and after \( \omega^- \) each impact is related to the torque impulse by

\[
\hat{T} = J_b (\omega^+ - \omega^-), \quad (4)
\]

where \( J_b \) is the moment of inertia of the rotor. For a parabolic rotor trajectory, the time between collisions with the stator \( \Delta t_c \) is estimated to be

\[
\Delta t_c = 2\hat{u}_s / g. \quad (5)
\]

The average torque \( T \) over the time between each collision can thus be estimated by combining (2) through (5):

\[
T = J_b \frac{\omega^+ - \omega^-}{\Delta t_c} = r_c \mu_k m_r g. \quad (6)
\]

Eq. (6) gives a torque estimate of 0.16 mNm, which is of the same order of magnitude as the measured stall torque shown in Table IV.

The maximum possible rotational rotor speed is the peak torsional vibration speed of the stator (from here on denoted as the “stator speed”). Depending on the portion of the stator vibration over which torque is transmitted, the rotor speed would approach some fraction of the stator speed. The measured results for beam 4 at 174.1 kHz (shown in Table IV) indicate that the ratio of the rotor speed over the stator speed (or simply the “speed ratio”) is \( 0.64 \pm 0.05 \). The speed ratios are consistently less than one, and this could be expected since the stator speed in Table IV is measured without the rotor, while the rotor...
**Fig. 4.** Motor efficiency of the four beam geometries as input power is varied. Note that beam 4 with $AR = 1.60$ has the highest efficiency.

**speed** is measured with the rotor which acts to suppress the stator vibration. As the rotor is decreased in size from the crude example used here, the speed ratio may increase since the stator vibration would be less affected by the rotor.

**C. Beam Geometry and Motor Efficiency**

Experimental results summarized in Fig. 4 show that out of the four beam geometries, beam 4 has the highest efficiency over the input power range of 32–97 mW. Beam 4’s aspect ratio of 1.60 is 3.6% lower than the “optimal” ratio of 1.66 estimated from the prismatic beam results in Table II. While close, the result is in contrast to the vibration analysis of pretwisted beams discussed in Section II: $AR$ should be higher than 1.66 to cancel the increase in $f_T$ due to pretwist. The discrepancy may be caused by the fact that the beam’s interaction with the piezoelectric element and the rotor is ignored in the preliminary analysis here.

**D. Motor Performance Characteristics**

Velocity-time data for beam 2 was obtained from the displacement sensor output via (1). A curve of the form

$$\omega(t) = \omega_0(1 - e^{-t/\tau_R})$$  \hspace{1cm} (7)

was then fitted to the velocity data to produce a set of charts similar to that of Fig. 5, where $\omega_0$ is the steady-state speed of the motor and $\tau_R$ is the rise time of the response. The measured data (represented by the dotted points) can be seen to have a wide distribution. This may be partly explained by the wobbling of the rotor due to the presence of weak bending vibrations, as discussed in Section IV-A. Since the wobble is approximately symmetrical about the measurement axis, the average rotation velocity is assumed to lie in the middle of the spread. The curves are fitted by choosing two values: the steady-state speed $\omega_0$, and the rise time $\tau_R$. The resulting linear torque-speed relationship and the quadratic power-speed relationship is shown in parts (a) and (b) of Figs. 6 to 9. Part (c) of these figures shows the trend in stall torque and steady-state speed as preload and input power are increased. Note that multiple trials at the same configuration are performed; averages are therefore used in parts (a) and (b), while results from individual trials are presented in part (c).

1. **Effects of Preload on Performance:** Figs. 6 and 7 show that, between 100 and 280 mN, increasing the preload increases the stall torque without significantly affecting the steady-state speed (averaging at around 200 rpm). A possible explanation for this result is that higher preload increases the frictional torque transmitted to the rotor; the steady-state speed, on the other hand, depends largely on the torsional vibration velocity of the beam, which is not affected at moderate preloads.

As preload is increased at the clockwise driving frequency (70.37 kHz) of beam 2, the peak output power increases to a maximum of approximately 0.3 mW, and subsequently falls. A similar relationship between the peak power output and preload might be expected from at the counterclockwise driving frequency (184.4 kHz); however, the maximum preload applied in the experiment was limited to 320 mN (corresponding to the total weight of the steel ball) at which point the peak power output still appears to be rising. The trends described above can be seen from the relationship between the peak power efficiency and the preload (see Figs. 6 and 10), which is closely related to the relationship between peak power and preload.

2. **Effects of Input Power on Performance:** Both the steady-state speed and the stall torque are increased by the input power (see Figs. 8 and 9). This suggests that the higher stator vibration velocity caused by the increase in the input power affects both the starting torque and the final rotation speed of the rotor.

The trend in Fig. 11 indicates that efficiency decreases as input power is increased. One explanation for this trend is that the increased vibration velocity results in increased slip and frictional loss. Another hypothesis attributes the decreased efficiency to the presence of weak bending modes, which has little effect on the movement of the ball at low power, but at higher power, causes the ball to lose traction with the beam tip. Another reason is the large hysteresis loss in the soft PZT used in the MLPA [22].
Fig. 6. Clockwise rotation characteristics of the motor at 70.37 kHz versus applied preload.

Fig. 7. Counterclockwise rotation characteristics of the motor at 184.4 kHz versus applied preload.

Fig. 8. Clockwise rotation characteristics of the motor at 70.37 kHz versus input power.

Fig. 9. Counterclockwise rotation characteristics of the motor at 184.4 kHz versus input power.
Fig. 10. Beam 2’s motor efficiency versus applied preload at (a) the clockwise and (b) the counterclockwise frequencies with average input power of 100 and 45 mW, respectively.

Fig. 11. Beam 2’s motor efficiency versus input power at (a) the clockwise and (b) the counterclockwise frequencies with preload fixed at 320 mN.

E. Comparisons with Other Motors

The measured efficiency of the motor varied considerably, depending on the beam geometry, input power, and rotor preload. It ranged from 3% for beam 2 to 8% for beam 4. The maximum torque attained was approximately 0.28 mN.m (beam 4 with 96.8 mW input power at 174.1 kHz), and the maximum speed was 1360 rpm (beam 2 with 240 mW input power at 184.4 kHz). This is fairly moderate performance for a motor of this size. The in-plane shearing motor reported in [19], for example, produces a comparable peak torque of 0.27 mN.m and a peak rotation speed of 800 rpm, but is only one-tenth the volume of the motor in this study.

Compared to other motors, this motor does not have great efficiency. The hybrid ultrasonic motor [20] and the in-plane shearing motor mentioned above have an efficiency of up to 40%; even electromagnetic motors with 2-mm diameter have efficiency in the 25–35% range. However, the large performance variation observed in this paper suggests that further improvements in the efficiency is possible. An important point to note is that the efficiency quoted here was determined using a fixed-frequency drive, implying that if the highest efficiency operating frequency of the motor shifts due to heating or hysteresis in the piezoelectric material, the results would have been affected. This is always a difficult decision—either include a drive circuit that tracks the operating frequency and argue it does not affect the efficiency, or exclude it and suffer the effects. Ultimately, the advantage of the present motor is that it would be easier to miniaturize than other motor designs.

V. Conclusions

An ultrasonic motor utilizing a rectangular cross-sectioned pretwisted beam and an axially poled MLPA was presented. It was shown that, as a result of the beam’s geometry, a coupled axial-torsion vibration can be realized at the tip. This vibration can be used to spin a rotor. Due to the presence of different resonance torsional modes in the beam, different phase lags between the axial and torsional modes allow bidirectional operation.

Four beam geometries were tested, and the best efficiency was achieved by beam 4, which has an aspect ratio of 1.6 and helix angle of 33.1°. This aspect ratio is very close to the optimal value predicted from results for a prismatic beam. The maximum efficiency achieved by beam 4 was 8.5% at 174.1 kHz with an input power of 96.8 mW. Tests on beam 2 showed that preload increased the stall torque without significant effects on the steady-state speed, though efficiency appears to first rise and then fall. The effect of raising input power was to increase both the stall torque and the steady-state speed at the expense of efficiency.
The generation of weak bending modes is a concern here, particularly as miniaturization is considered. These modes are believed to occur from inaccuracies in fabrication of the beams, particularly at the ends. Fortunately, techniques such as LIGA technology and laser machining for microfabrication offer accuracies superior to the methods used in this preliminary study.

While the efficiency was low in this study, improvement may be possible if coupling is optimized by matching the torsional and axial resonance frequencies. Additionally, the motor’s strength lies with its simple structure, which paves the way for further miniaturization.

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References

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