Design of a Piezoelectric Poly-Actuated Linear Motor

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Design and analysis for an efficient and force dense piezoelectric poly-actuated linear motor is presented. A linear motor is constructed with multiple piezoelectric actuator units engaging a rod having gear teeth. The multiple piezo-units are placed along the geared rod with a particular phase difference such that a near constant force is generated regardless of the rod position by coordinating the multiple piezo-units. Rolling contact buckling mechanisms within the piezo-units provide large displacement amplification with high energy transmission and low loss properties from the piezo-units to the geared rod. This piezo-based motor has capacitive actuator characteristics which allow it to bear static loads efficiently. Furthermore, the poly-actuator architecture presented provides for scalability through modular design.

First, the basic design principle describing the engagement of buckling amplification mechanisms to a phased array-shaped gear rod is presented, and the resulting force and displacement characteristics are analyzed. Design methods for creating a piezoelectric poly-actuated linear motor are then summarized. A prototype design is presented for which a maximum mean force of 213 N, a maximum velocity of 1.125 m/s, and a force density of 41 N/kg is calculated.

INTRODUCTION

A primary objective in actuator design is to provide adequate force, displacement, and velocity for a given application. In addition, to enhance the system performance, power/force density, response and energy consumption properties are considered. These implicit characteristics typically dominate an actuator design. However, limiting energy consumption in an actuator can reduce the dependence on large energy sources and provides the potential for many actuators to be used cooperatively without drawing additional energy. Balancing these two goals is particularly relevant to mechatronic and robotic systems where specific force and displacement outputs are desired, but power sources may have limited energy supply, such as in the case of mobile machines. Historically, meeting these actuator design goals simultaneously has been difficult.

Many conventional actuators have been developed for robotic and mechatronic systems which maximize density of force and displacement, yet fail to combine high response and low power consumption simultaneously. These conventional actuators are generally considered to be either inductive or capacitive actuators. In inductive actuators, such as magnetic motors with typically high bandwidths, torque output is a function of electrical current resulting in high power consumption when large torque is provided at any actuator velocity. Alternatively, capacitive actuators, such as pneumatic and hydraulic actuators, operate at lower bandwidths but provide force output as a function of an effort variable such as pressure. Thus these actuators consume limited energy when providing force at small or zero displacements [1]. This can lead to energy efficiency, for example, in ground based robots which must support significant gravity or other static loads for long periods of time during operation.

Piezoelectric actuators represent one class of capacitive actuator which can provide high operational bandwidths, above 10 kHz, as well as power densities of approximately $10^8$ W/m$^3$ [2]. These important performance features have promoted widespread usage of piezo-actuators in small scale applications for acoustics, vibration control and precision
positioning [3]-[4]. Yet, piezoelectric actuators remain severely limited in output stroke. Unamplified piezoelectric stack actuators, such as Lead Zirconate Titanate (PZT), achieve free strain on the order of 0.1% [1]. This small strain leads to displacements on the order of 4×10^{-2} mm which are not sufficient for typical robotic or mechatronic applications and require that displacement amplification devices be used.

Piezoelectric displacement amplification is generally categorized into internally leveraged, externally leveraged, and frequency leveraged designs. Internally leveraged actuators such as stack, bender, and unimorph actuators make use of carefully selected piezoelectric material geometries to amplify displacement [1]. Stroke output is limited by the internal stiffness of the actuators. Stack actuators are simple and effective internally leveraged actuators that maintain high force density and are used as building blocks in both externally leveraged and frequency leveraged mechanisms [6-8].

In frequency leveraged mechanisms, also known as ultrasonic motors (USM), piezoelectric actuators are driven at resonant frequencies to impart motion through a friction drive to the actuator output [5]. For rotary USM devices, this output is in the form of unlimited rotation while linear devices amplify displacement using “inchworm” motion to slide along a track [1]. USM devices are characterized by slow speed and high torque since they operate as direct drive motors. However, driving force and torque may be adversely affected by friction drive material wear.

Several types of externally leveraged, or mechanically amplified, actuators have been researched. These include the flexensional, lever, Moonie and Cymbal mechanisms [1]. These devices typically produce displacement amplification one order of magnitude greater than the stack actuator building blocks. Recent external leveraging research has been performed on amplification mechanisms that exploit non-linear structural buckling. These are of particular interest because they achieve displacement amplification of two orders of magnitude in a single actuator stage [6-8].

The basic design of these buckling actuators is shown in Fig. 1. As described in [6-8], buckling amplification mechanisms produce “mono-polar” or “bi-polar” stroke of the output “keystone” which produces a non-linear force shown later. Control of output displacement direction is not trivial, and can only be managed by additional constraints also described later. Methods of tuning force-displacement output, controlling displacement direction, and improving actuator efficiency have been central to research of these actuators [6].

A review of historical and recent research into piezoelectric actuators yields similar limitations as those of traditional actuators. However, piezoelectric stacks connected to buckling amplification mechanisms show an additional level of performance gain. A remaining area of research is the transformation of limited-stroke, repetitive motion into an unlimited or sufficiently long stroke without significantly degrading performance. Furthermore, the energy efficiency and force control to be gained by using multiple actuator units working as phased arrays is of interest.

This paper presents the development and design methodology of a practical and implementable linear motor based on piezoelectric actuators with buckling mechanism displacement amplification. Prior research on these individual unit actuators has shown high performance but undesirable non-linear force characteristics. Through the use of several novel design elements and proposed actuation techniques, the poly-actuated linear motor design provides improved force-output linearity over previously researched piezo-actuators. Furthermore, this design represents the first implementation of a linear poly-actuator using the highly efficient rolling contact mechanisms described in [6]. The resulting design yields a linear actuator with capacitive properties and performance sufficient for use in robotic or mechatronic applications.

**DESIGN CONCEPT**

The concept of a piezoelectric poly-actuated linear motor is to construct a motor, driven by power dense piezo stack actuators, which achieves standard robotic and mechatronic system force requirements, significant displacement, and low energy consumption during static load holding. The design should achieve the following functional requirements:

i) **Long stroke**: achieve a stroke output of 500mm or more

ii) **High force density**: achieve a force output greater than 100 N with minimized motor mass

iii) **Energy efficiency**: Support static loads at any actuator output position without consuming electrical power

iv) **Force control**: Achieve force controllability beyond that of typical frequency leveraged actuators

v) **Fault Tolerance**: Provide levels of fault tolerance such that failure of individual mechanical work elements do not prevent actuator function

vi) **Backdriveability**: Minimize motor impedance when back-driving

**Basic Architecture**

The basic operation of the linear motor functions with the phased bipolar actuation of rolling contact buckling mechanisms coupled to a shaped gear output rod. Reciprocating force inputs from the stack actuators through the buckling mechanisms applies force input perpendicular to the wavy groove of the linear output rod as shown in Fig. 2.
Buckling Mechanism
A previously designed rolling contact, buckling, displacement amplification mechanism has been used as the building block for the poly-actuator design. As discussed in previous publications, these buckling amplification mechanisms transmit a high percentage of mechanical work [6]. This combined with the shaped gear rod allows for higher linear motor output efficiency during motion, while the capacitive properties provide low energy consumption during static holding.

Energy Transfer through Rigid Structures
Prior work on the rolling contact buckling mechanism describes the detrimental effects of structural compliances at the rolling contact joints and within the buckling mechanism frame [6]. In this design, revisions are made to improve the rolling contact stiffness in an effort to boost buckling unit force output. Several rolling contact geometries have been considered prior to a final selection.

The structural compliance arising from the mechanism frame is improved by using anisotropic materials with increased elastic modulus along the load direction. Prior research describes the potential for careful material selection, in particular the choice of composite materials, for improving structural load efficiency [6]. This linear motor design expands upon such research by creating the primary load structure with a high modulus carbon fiber.

Poly-Actuator Modularity
The modular architecture of the poly-actuator arrangement in the linear motor provides several functions. Primarily, the use of multiple buckling actuators in parallel provides the ability for the motor force output to be controlled with high resolution. Any force ripples or non-linearities transmitted to the gear rod from the buckling units can be mitigated by phase control of the other buckling units. In essence, nodes of zero force transmission, and regions of varying force output can be combined in a constructive/destructive interference methodology to achieve a smooth net output force. Additionally, multiple actuators working in parallel can boost instantaneous force and provide redundancy and fault tolerance should PZT, buckling mechanism, or force transmission components fail.

Shaped Gear Rod
Exploiting the mono-polar and bi-polar motions of the buckling amplification mechanism requires addressing a reciprocating displacement at the keystone and a non-linear force application to the gear output rod. To minimize force ripples to the gear rod, and provide a smooth force input to a driven load, the parameters of the phase shaping of the gear are controlled. Additionally, while “gearing” occurs in the buckling mechanism between stack actuator force output and buckling mechanism force output, an additional level of gearing is achievable by tuning the gear output shaft “pitch” length and amplitude.

Since bi-polar motion of the buckling mechanism is used to drive the designed gear rod, the buckling output must be controlled to actuate in the proper direction at the appropriate rod position. Buckling motion at the kinematic singularity is not deterministic and must be controlled externally. In this motor design this is performed via a continuous contact engagement between each buckling mechanism output and the gear rod shaped surface. In regions away from a buckling unit’s singularity, the buckling motion is deterministic and drives the gear rod. At a singular point, the gear rod’s motion, driven by buckling units not near their own singularity, forces the buckling mechanism with nearly zero impedance across its singularity. This buckling mechanism output direction then becomes deterministic again.

ANALYSIS
The main components of the proposed motor are the buckling PZT actuators using rolling joints as unit actuators and the phased array shaped (PAS) mechanism.

Essential force property of Phased Array Actuators
In the poly-actuator design shown in Fig. 2, the general expression of the static output force \( F_x \) can be given by Eq.(1).
\[ F_x = \sum_{i=1}^{N} F_{xi}(\psi) = \sum_{i=1}^{N} R_{PAS}(\psi + \phi_i) F_{yi}(G(\psi + \phi_i), u_i) \]

where \( x = \frac{\Delta \psi}{2a} \) \( R_{PAS}(\psi + \phi_i) = \frac{d}{dx} G(\psi + \phi_i) \)

In the above equation, \( i \) indicates the \( i \)th unit actuator, \( N \) is the total number of unit actuators used in the poly-actuator, \( \psi \) is the output position of the poly-actuator described by the phase angle of PAS, \( \phi_i \) is the layout position of the \( i \)th unit actuator in the phase angle of PAS, \( F_{xi} \) is the contribution to the output force of the poly-actuator by the \( i \)th unit actuator, \( G \) is the trajectory of the unit actuator inputs along the profile of the PAS, \( R \) is the slope of the actual output position of the poly-actuator \( x \), \( F_{yi} \) is the output force of the \( i \)th unit actuator in the \( y \)-direction, \( u_i \) is the input to the \( i \)th unit actuator, and \( \lambda \) is the “lead” length of trajectory \( G \) along the output direction, which corresponds to the rod output displacement per one buckling cycle.

Due to the unit actuators being connected in parallel through the PAS, all output forces of the unit actuators are summed after the transformation by the PAS. The equation indicates that there are four design parameters in this type of poly-actuator:

a) the properties of a unit actuator
b) the trajectory of the PAS
c) the layout of unit actuators along the PAS
d) the drive input to each unit actuator

Equation 1 also shows that all unit actuators interact through the PAS. If the quantity \( N \) is small, we need to consider the interaction between buckling units via the PAS. Conversely, a large quantity \( N \) allows focusing only on the interaction between the PAS and each buckling unit because the cloud of unit actuators can have much larger impedance than each single unit actuator.

**BUCKLING PZT ACTUATORS AS UNIT ACTUATORS**

**Geometric Properties of Buckling Actuators**

The essential analysis method for the buckling actuators is described in papers [6-8]. Referring to the conceptual configuration of the buckling displacement amplification mechanism shown in Fig. 1 and assuming ideal solid base structures and ideal joints, the amplification ratio \( R_{\theta} \) with respect to the joint angle is obtained by Eq. (2) and is shown in Fig. 3.

\[ R_{\theta} = \frac{dy}{dz} = \frac{dL}{d\theta} \left( \frac{dz}{d\theta} \right) \left( \frac{L \tan \theta}{\cos \theta} \right)^{-1} = \frac{1}{\sin \theta} \]

Due to the singularity, infinite instantaneous amplification occurs at the zero angle/displacement configuration. In the proximity of the singular point, much higher amplification ratios compared to other one-step leverage mechanisms [9] can be obtained by this configuration.

**Slip-less Roller Joints.** An important feature to realize the ideal property mentioned in the above section is the rolling contact joint between the keystone and piezo actuators and between the piezo actuators and side blocks. The analysis method for the buckling actuators which use the rolling joints is described in [6]. A conceptual configuration of the mechanism is described in Fig. 4. This poly-actuator employs cylindrical rolling contacts at all joints. The side blocks and keystone share one radius value, and all caps share another radius value as indicated in Fig. 5. Fig. 5 shows one half of the roller buckling mechanism. Due to the symmetric architecture, the keystone can translate without any rotation. In Fig. 5, \( d \) represents the distance between the centers of the cap radii, \( z \) represents the displacement of the piezo actuators, and \( b \) represents the perpendicular distance between misaligned load axes from the rolling contact points to the centers of the cap radii.
If the summation of \( d \) and \( z \) equal zero, \( b \) also becomes zero. However, \( z \) changes during actuation and \( d \) is a constant length such that it is impossible to keep \( b \) equal to zero throughout the entire operation condition. If \( b \) is not equal to zero and the interaction forces are not zero within the system, frictional forces arise at the contact points between the caps and side blocks and the keystone to maintain a force balance with the moment due to the normal force at the contact points and \( b \). Assuming the resultant interaction force is \( F_z \), the distance between the contact points is \( L_c \), and the angle between the contact point, the center of the side block, and the horizontal axis is \( \theta \), and the friction force \( F_{Frict} \) can be obtained by Eq. (3)

\[
F_{Frict} = \frac{b}{L_c} F_z = \frac{y \cos \theta - L \sin \theta}{L_c} \frac{F_z}{L_c} \\
= \frac{(L - d) \sin \theta + d \sin \frac{r_1 + r_2}{r_2} \cos \theta - L \sin \theta}{L_c} \frac{F_z}{L_c} \tag{3}
\]

The friction can cancel the moment which can cause slip at the contact points. However, it is known that micro slip occurs in the rolling contact and should accumulate during each cycle. To prevent slip which causes instability to the buckling motion, we set \( d \) equal to zero which reduces the friction to an almost negligible amount in the small \( \theta \) rotation regime due to \( z \ll L \). This condition means that the cap radii are concentric which can be confirmed by Eq. (4) which is obtained by approximating trigonometric functions as a function of small \( \theta \) rotation.

\[
F_{Frict} \mid d=0 = \frac{L}{L_c} F_z \sin \theta (\cos \theta - 1) \approx -\frac{L}{2L_c} F_z \theta^2 \tag{4}
\]

**Preload Compensation Spring.** There are several reasons for preloading the buckling mechanism [6]. The first is to maximize the PZT output. PZT stack actuators are created by stacking thin layers of piezoelectric ceramics and electrodes. As a result of this structure, PZT stack actuators cannot bear high tensile forces as compared to the high allowable compressive forces. However, if a large enough preload force is applied, the resultant force of the piezo actuator and the preload can achieve both tensile and compressive forces, which allows the buckling mechanism to output both compressive and tensile forces. The second reason is to provide a constant force throughout the entire piezo actuator displacement. The compliance of the preload mechanism makes it difficult to harvest the maximum energy from piezo actuators. A third reason is to keep compressive forces on all contact surfaces. Compression maintains all bodies in position through frictional force. The final reason is to reduce the compliance at the rolling contact surfaces which is explained in more detail later.

To satisfy all of these functions simply, we attach a linear spring, \( k_{PCS} \), called the preload compensation spring (PCS) to the keystone and apply a preload force \( F_{PL} \) between the side blocks horizontally as shown in Fig. 6. The preload is produced by shortening the distance between the keystone and side blocks. Since the axial displacement of the PZTs is very small compared to the distance between the side blocks and the keystone, we can describe the displacement of the caps as a change in diameter of the concentric radii of the caps. Based on this assumption, the force balance between \( F_{PL} \) and the force produced by the PCS is described by Eq. (5) and the relation between the \( F_{PL} \) and \( k_{PCS} \) can be obtained through approximations of trigonometric functions as in Eq. (6).

\[
2F_{PL} \cos \theta = \frac{1}{\sin \theta} k_{PCS} y \tag{5}
\]

\[
F_{PL} = \frac{L}{2} k_{PCS} \tag{6}
\]

where \( \cos \theta \approx 1 \), \( \sin \theta \approx y/L \)

**Output Property.** To calculate the static output property of the roller buckling mechanism driven by piezo actuators, the compliant model shown in Fig. 6 is used. The four compliance properties which exist in the PCS, PZTs, roller joints, and the frame are considered. From the basic relations shown in Eq. (7), the output force property, \( F_z \), is obtained for the roller buckling actuators as shown in Eq. (8).
Based on Eq. (8), the output force-displacement property of a buckling actuator is described as shown in Fig. 7. One unique property of the buckling PZT actuator, the so called "bi-directional motion" is that the output motion occurs in both directions about the central position where all roller joints are aligned, otherwise described as the "zero output position". Depending on the singularity, the buckling mechanism can easily realize roughly 50 times the displacement amplification. The bi-directional motion doubles this stroke, creating a displacement amplification of around 100. The maximum output displacement is defined as the distance between positions where the output force becomes zero with the minimum and maximum electrical inputs to the PZTs. This is obtained via Eq. (9).

\[
\gamma_{\text{max}} = y|_{F_y=0,F_{\text{PZT}}=F_{\text{max}}} - y|_{F_y=0,F_{\text{PZT}}=0} = \frac{2L}{k_{\text{PZT}}} F_{\text{PZTmax}}
\]  

Referring to Hertzian contact theory, the rolling joint stiffness \( k_j \) is obtained by Eq. (10) as the contact stiffness between cylindrical surfaces where \( F_j \) is the contact force, \( \delta \) is the deformation mainly occurring at the contact line, \( L \) is the length of the cylinder, \( E \) is the Young’s modulus of the cylinder material, and \( \nu \) is Poisson's ratio.

\[
k_j = \frac{dF_j}{d\delta} = \frac{\pi L_j E}{2 (1 - \nu^2) \left(-\frac{1}{3} + \ln \left(\frac{2\pi L_j E (r_1 + r_2)}{F_j (1 - \nu^2)}\right)\right)}
\]  

As shown in the above equation, \( k_j \) varies depending on \( F_j \) which is the resultant summed force of the PZT and the preload \( F_c \). However, due to the preload application through the buckling mechanism, the variation of \( k_j \) in the operational range becomes significantly smaller, approaching roughly 10% of the full stiffness range, as is shown in Fig. 8.

**FIGURE 7. OUTPUT PROPERTY OF ROLLER BUCKLING PZT ACTUATORS**

**FIGURE 8. CONTACT STIFFNESS VARIATION**

**Sinusoidal Trajectory PAS**

Along with the general property of poly-actuators, Eq. (1), one can use any type of PAS. In this paper, we focus on one specific PAS design based on the cam and roller follower type transmission shown in Fig. 2. Similar to common cam design methods, the minimum radius of curvature of the follower trajectory needs to be larger than the radius of the follower to enable a continuous transition at the point when the directions of both the cam and the follower motions change. Also, this design focuses on a PAS which makes the trajectories of the follower roller axes sinusoidal enabling smooth energy transmission between the cam, follower, and the reciprocating motion of the followers. The trajectory of the follower roller axes is defined in Eq. (11):

\[
y_{FC} = y_{FC_{\text{max}}} \sin \frac{2\pi x}{\lambda}
\]  

where \( y_{FC} \) is the position of the follower axis along the PAS toward the output direction of the unit actuator, \( y_{FC_{\text{max}}} \) is the maximum displacement of the follower central axis, \( \lambda \) is the wave length of the PAS, and \( x \) is the relative displacement.
between a unit actuator and the PAS toward the output
direction of the poly-actuator.

Considering Eq. (8), and (11), the property of the
transformed force of each unit actuator by the PAS is given by
Eq. (12) and is shown in Fig. 9.

\[
F_{su}(x) = A_{Fx} \left( A_{F_{PZT}} F_{PZT} - 2 \right) \sin \left( 2 \left( \frac{2\pi}{\lambda} x \right) \right) + \sin \left( 4 \left( \frac{2\pi}{\lambda} x \right) \right) \]

(12)

where

\[
A_{Fx} = \frac{\pi k_{PZT} y_{FCmax}^2}{\lambda L^2} \frac{k_s}{k_s + k_{PZT}}, \quad A_{F_{PZT}} = \frac{8L}{k_{PZT} y_{FCmax}^2} \]

Considering Eq. (9), the above constants are also
described as

\[
A_{Fx} = \frac{\pi R_x z_{max}^2}{\lambda} \frac{k_s}{k_s + k_{PZT}}, \quad A_{F_{PZT}} = \frac{4}{R_y F_{PZTmax}} \]

(13)

If \( y_{FCmax} \) is equal to \( y_{max} \), the coefficient for \( F_{PZT} \) becomes
4 when \( F_{PZT} \) is equal to \( F_{PZTmax} \), which gives the symmetric \( F_{sw} \)
property between ON and OFF conditions for PZTs as shown
in Fig. 9. The coefficient of \( F_{PZT} \) to \( F_s \) increases with the
square of \( R_x \), but simultaneously, the effect of the compliances
to \( F_s \) also increase with the fourth power of \( R_x \). This suggests
that higher \( R_x \) makes control of \( F_s \) more difficult.

\[
F_s(x) = A_{Fx} \sum_{i=1}^{N} \left\{ \sin \left[ 4 \left( \frac{2\pi}{\lambda} x + \phi_i \right) \right] + \left( A_{F_{PZT}} F_{PZT,i} - 2 \right) \sin \left[ 2 \left( \frac{2\pi}{\lambda} x + \phi_i \right) \right] \right\} \]

(14)

From Eq. (14), we can find that \( F_s \) has components of two
and four cycles in one PAS wavelength. For this property, if \( \phi_i \)
only consists of the phase set of \( \{0, \pi/8, 2\pi/8, 3\pi/8\} \) with
respect to the PAS wave cycle, the terms other than containing
\( F_{PZT} \) cancel between the unit actuators. In such a case, \( F_s \)
becomes Eq. (15).

\[
F_s = A_{Fx} A_{F_{PZT}} \sum_{i=1}^{N} \{ F_{PZT,i} \sin \left[ 2 \left( \frac{2\pi}{\lambda} x + \phi_i \right) \right] \} \]

(15)

In Eq. (15), if all PZTs are kept at the same output
condition, \( F_s \) always becomes zero.

In another case of the phase set of \( \{0, \pi/12, 2\pi/12, 3\pi/12, \pi/12, 5\pi/12\} \), the component of the 4 times of the PAS wave
cycle remains in \( F_s \) with the \( F_{PZT} \) component similar to Eq.
(15).

**Switching Drive**

From Fig. 9, each unit actuator can always output both
positive and negative force for the poly-actuator output by
switching the input to the PZT from ON to OFF and from OFF
to ON at every multiple of \( \lambda/4 \) position with respect to the
phase of the PAS wave. In addition, the force transitions
become the same between the conditions of “PZT ON by
moving from zero to the top of the wave” and “PZT OFF by
moving from the top of the wave to zero” because of the
cancellation between the efforts of the buckling actuator force
output, the preload force, and the PCS force.

The result of all unit actuator output forces being kept
positive for the poly-actuator is the maximum output force of
the poly-actuator. Figure 10 shows the property of the
maximum output force of the poly-actuator which consists of
4 and 6 unit actuators which have the phase layout mentioned
above. The larger number of unit actuators produces a higher
mean and a lower variation of maximum force for the poly-
actuator.

![FIGURE 9. TRANSFORMED FORCE PROPERTY OF UNIT ACTUATORS BY A PAS](image-url)
IMPLEMENTATION

Figure 11 shows a designed model of an experimental poly-actuator and Tab. 1 shows the main design parameters.

The maximum velocity is set by considering the thermal property of the PZTs. The voltage and frequency affect the amount of energy loss in the PZT actuators and the thermal excitation. PZT materials lose their piezoelectricity above their Curie temperatures. For this reason, the practical restriction on the operating condition is determined by considering the temperature.

The wavelength of the PAS is determined by considering the contact stress between the PAS and follower rollers driven by the unit actuators. Since the maximum contact stress occurs at the tip of the PAS tooth, the curvature radius of the tip has been made larger than 0.5mm and with a hardened tool steel.

Unit Actuator

Figure 12 shows the section view of the unit actuator used in the poly-actuator. Essentially, the fundamental mechanism shown in Fig. 6 has been adopted in the buckling unit. Practical revisions include the radii of the keystone, caps and side frames, and a spatially condensed PCS structure. The triangle shaped and thickness varied leaf springs used in the PCS satisfy both tuned stiffness requirements in the buckling direction and stability against undesired degrees of freedom. The main requirements for the PCS are the following:

1) A specified and tuned stiffness in the unit actuator output direction defined by Eq. (8)
2) As high stiffness as possible in the poly-actuator output direction
3) Spatial constrains to fit into the space between the coupler and the rest of the motor frame
4) Low stress

Another important aspect of the unit actuator is potential misalignment between rolling contact components which can cause a rotation of the keystone. Due to the high force of both the PZT output and the preload, a resultant moment can be more than 1Nm with only 50µm of misalignment, which is hard constraint through the compliant PCS. To provide a restoring moment to the keystone, the distance between the keystone radii is reduced below the concentric condition. This reduced distance between rolling contact radii stabilizes the keystone and prevents rotation. Without this configuration, the keystone is unstable.
TABLE 2. DESIGN PARAMETERS FOR UNIT ACTUATORS

<table>
<thead>
<tr>
<th>Item</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
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</thead>
<tbody>
<tr>
<td>Blocking force of PZT</td>
<td>$F_{PZT\text{Block}}$</td>
<td>5,250</td>
<td>N</td>
</tr>
<tr>
<td>Free displacement of PZT</td>
<td>$z_{\text{max}}$</td>
<td>42</td>
<td>µm</td>
</tr>
<tr>
<td>Preload force</td>
<td>$F_{PL}$</td>
<td>5,250</td>
<td>N</td>
</tr>
<tr>
<td>Arc radius</td>
<td>$r_1$</td>
<td>14.5</td>
<td>mm</td>
</tr>
<tr>
<td>Cap</td>
<td>$r_2$</td>
<td>23.0</td>
<td>mm</td>
</tr>
<tr>
<td>Stiffness (PZT)</td>
<td>$k_{pzt}$</td>
<td>125</td>
<td>N/µm</td>
</tr>
<tr>
<td>Stiffness (Compression)</td>
<td>$k_J$</td>
<td>447</td>
<td>N/µm</td>
</tr>
<tr>
<td>Frame (Tensile)</td>
<td>$k_F$</td>
<td>800</td>
<td>N/µm</td>
</tr>
<tr>
<td>Combined stiffness</td>
<td>$k_S$</td>
<td>196</td>
<td>N/µm</td>
</tr>
</tbody>
</table>

CFRP Frame
The frame for each buckling mechanism is designed using a high modulus carbon fiber reinforced plastic (CFRP). As mentioned above, the frame compliance is critical because if too compliant it results in reduced energy transmission from the PZTs to the unit actuator output. Additionally, the frame bears only tensile load in one load direction due to the preloading force and stack actuator force, and thus stiffness and strength requirements in other directions are significantly lower. The buckling units also do not require the frame material to contain magnetic flux as is the case in the design of inductive motors. For these reasons, the use of a high elastic modulus unidirectional fiber is appropriate. The important mechanical properties for this frame are listed and compared to a typical high strength structural steel in Tab. 2. Comparing the two material properties, the weight of the frame made from CFRP is only 1/8 of the weight of a steel frame with the same stiffness, which contributes to improved force density of the poly-actuator.

Thus, a change of the frame structure material to one with higher elastic modulus allows the same geometry or a reduced cross-section geometry to be used while limiting structural compliance. In the design, high modulus carbon fiber is aligned with the primary load axis allowing a reduction of ground structure (buckling unit frame) weight. As a result, energy efficiency can be met or improved while force density is also improved. The carbon fiber laminate used in the design more than doubles the frame material elastic modulus from 210 GPa of steel to nearly 460 GPa.

TABLE 3. COMPARISON BETWEEN STEEL AND HIGH ELASTIC CFRP

<table>
<thead>
<tr>
<th>Item</th>
<th>CFRP prepreg MPI K1392U</th>
<th>Steel SS440C</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2.15</td>
<td>7.8</td>
<td>$10^3$ kg/m³</td>
</tr>
<tr>
<td>Tensile modulus</td>
<td>460</td>
<td>210</td>
<td>GPa</td>
</tr>
<tr>
<td>Tensile modulus density</td>
<td>214</td>
<td>25.6</td>
<td>MPa/( kg/m³)</td>
</tr>
<tr>
<td>Ultimate tensile strength</td>
<td>2.10</td>
<td>1.75</td>
<td>GPa</td>
</tr>
<tr>
<td>Tensile stress density</td>
<td>976</td>
<td>164</td>
<td>MPa/( kg/m³)</td>
</tr>
</tbody>
</table>

CONCLUSION
We have presented a design and analysis method for a new type of linear actuator consisting of buckling PZT actuators built into a poly-actuator architecture. By controlling the phase distribution of buckling units, the actuation timing, and appropriately shaping an output gear, we have developed a system which transforms individual, non-linear, and undesirable force characteristics into a practical linear motor design. Additionally, previously researched buckling mechanisms have undergone revision to improve upon already high force transmission and efficiency characteristics and have been included. New materials and structural geometries have been incorporated to maintain the dynamic performance requirements while minimizing the size and weight of the linear actuator. The analysis calculations indicate that the linear motor characteristics are within the range of typical robotic and mechatronic system requirements.

An experimental motor is under construction and will be used to verify the static analysis results shown in this paper. In addition, we will analyze and optimize the dynamic properties of the motor, and develop suitable control methods and drive electronics to derive the maximum potential of the motor.
REFERENCES


