Fuzzy Inchworm Motors For Improved Power and Area Efficiency

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ABSTRACT

An adjustable gap GCA-driven inchworm motor is proposed. A spring-loaded Pawl based on the vernier is described to permit smaller step sizes and fundamental design tradeoffs are explored. A static gap adjuster is also described, and power/area efficiency benefits analyzed. Finally, a specific design in a single mask SOI process is presented which is predicted to show considerable performance improvements over standard inchworm designs.

I. INTRODUCTION

A common problem for linear actuators is that they provide significant force only over a limited displacement. This limitation creates a problem in applications where both large force and large displacement are desirable. Inchworm motors overcome this limitation by mechanically accumulating motion over many small cycles to produce large displacement.

Inchworm motors have seen a great deal of use associated with piezoelectric actuators, [1] and more recently, in association with electro static gap closing actuators (GCA) [2]. Initial designs typically made use of a simple straight shuttle gripped by a moving clutch using simple static friction. Such a design requires that the friction forces, and so clutch forces, be exceed the load forces to move the shuttle without slipping. This requires a large clutch to driver ratio and so is inefficient.

Several attempts have been made to increase the effective coefficient of friction, primarily by adding gear-tooth like roughness to the shuttle and pawl [1][4]. This approach works, but it also places some restraints on the minimum step-size of a given inchworm cycle. Since pawl and shuttle teeth must align to mesh, a given actuation cycle must displace the combined length of a gear tooth and gap, at best 2λ, twice the minimum feature width of the process being used.

When using GCAs, the minimum step constraint can become a real problem because the force output of a GCA is proportional to the inverse square of its initial gap. This problem is somewhat overcome by using two drivers, driven anti-phase to each other, each displacing by λ. Nonetheless, since only one of the two drivers is being used at a time, such a system is inherently area inefficient. To maximize force, it would be desirable to be able to engage the shuttle at arbitrarily small increments and so actuate over an arbitrarily small gap.

Generating an arbitrarily small gap requires both a new technique for gripping the shuttle and a mechanism for adjusting the gap to be smaller than process constraints allow. Such a gap adjuster effectively creates a transmission like structure that can improve efficiency by trading force for displacement. Such a structure could also be designed to allow for reversibility.

Here we present designs for structures that we believe will accomplish both of the functions described above. A pawl structure based on a vernier permits step sizes less than the minimum gear length. A variable gap-stop transmission that allows adjustment of the gap to smaller sizes is also presented.

II. SHUTTLE ENGAGEMENT MECHANISM

MEMS verniers have been used to measure displacement to a precision of much less than the minimum feature size in a given process [5][6]. This precision is possible because photolithography permits minimum distinguishable differences in dimension that are an order of magnitude more accurate than the minimum feature size possible.

A structure similar to a vernier can be used to build a pawl that engages a studded shuttle at increments considerably smaller than λ. The basic concept is the same; the periodicity of the teeth on the shuttle is either slightly longer or slightly shorter than that of the teeth on the pawl (see figure 1). This guarantees that one of the teeth of the pawl will always be aligned to the teeth of the shuttle to an accuracy set by difference between the periodicities of the shuttle and pawl. This, in turn permits the main shuttle driver to actuate over a distance of:

$$Step = 2\lambda/N \qquad (1)$$

where N is the number of teeth on the pawl.

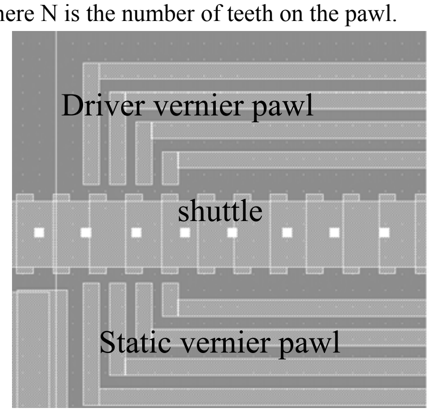

Figure 1. Close-up vernier pawls. Note that while the shuttle gear structure repeats every 11um, only a 3um step is necessary to ensure that one of the vernier fingers will engage

A problem with this approach is that only one of the teeth is guaranteed to engage, while the others will butt against the teeth of the shuttle. If the teeth of the pawl are rigidly coupled, the unaligned teeth will prevent the aligned teeth from engaging. It is therefore necessary to individually spring-load each of the pawl's teeth so that those teeth not aligned to the pawl can deflect out of the way. A relatively simple structure to accomplish this is shown in figure 2.

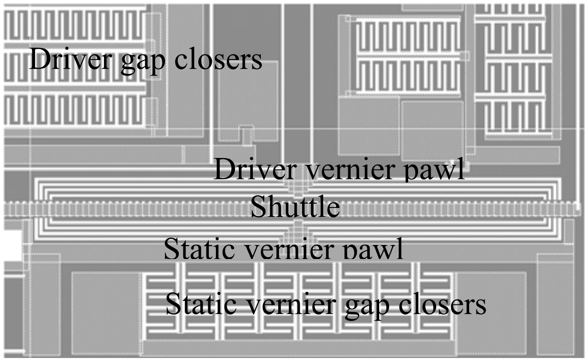

Figure 2. Close-up vernier spring clutch pawls. Two clutches are used; one to drive the shuttle, the other to hold the shuttle in place while the driver resets.

Spring-loaded teeth require that, for the pawl to engage, the engagement actuator must overcome not just the stiffness of its own suspension, but also the stiffness of the beams suspending each of the unaligned teeth. This stiffness can be described by the equation:

$$K = 2IE(N\text{-}1)/L^3 \qquad (2)$$

N is the number of teeth, L is the length of the beams on which the teeth are mounted, I is the cross-sectional moment of inertia of the beams and E is Young's modulus.

Pull-in of the clutch GCA, requires that

$$F_{cl} > 16IE(N\text{-}1)(h_t+g_s)/(\,9L^3\,) \qquad (3)$$

h_t is the height of shuttle gear teeth, g_s is the separation left by the clutch's gap stop and F_{cl} is the force the shuttle exerts when it first contacts the shuttle. In addition, the clutch will have to close a gap before it first contacts the shuttle, and thus will also have to overcome the stiffness of its own suspension. The result is a force-displacement relationship such as that shown in figure 3 below. To guarantee pull-in, the total force on the shuttle must be positive across all displacements.

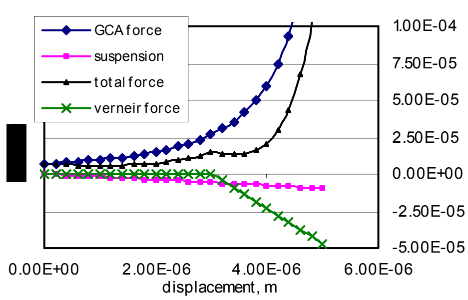

Figure 3. Shuttle forces. In order to ensure pull-in, the shuttle gap closers must overcome not only their suspension but also the spring constants of the vernier springs.

A second requirement on the spring-loaded pawl teeth is that their tips not deflect excessively when lateral drive is applied. This deflection can be described by:

$$\Delta_x = F_x L_t^2 L\,/(IE) \qquad (4)$$

Δ_x being the lateral deflection of the pawl tooth, F_x the lateral force, L_t the length of the vertical portion of the tooth, and L, E, and I are as before. It can be seen that lateral deflection will be worst when the outermost tooth is engaged. In this case, L_t is at least:

$$L_t > 2(N\text{-}1)\lambda \qquad (5)$$

Finally, if we set a tolerance on how much lateral deflection is permissible, (call this δ_x) we can say:

$$F_x < \delta_x IE/(4\lambda^2 L(N\text{-}1)^2) \qquad (6)$$

Combining (3) and (6) gives us the relation:

$$F_x/F_{cl} < 9\delta_x L^2/[16\lambda^2 * (h_t+g_s)(N\text{-}2)^3] \qquad (7)$$

This inequality sets the maximum ratio of driver –to-clutch force and so directly impacts the energy and area efficiency of the motor. Note that while maximum output force (from (5)) is dependent on I, the efficiency (from (6)) is not, implying that the two inequalities can be decoupled. It should be possible to optimize efficiency by increasing L while maintaining a large F_x by changing I. This solution, however, cannot be exploited indefinitely. If made too long, the beams will buckle.

Buckling occurs when narrow beams are subjected to high axial loads. Although buckling is often characterized by sudden catastrophic failure, milder failures occur at lower loads if the axial load is off-center [8]. In the case of spring-loaded vernier teeth, Δ_x will increase nonlinearly at high F_x, deviating from the predictions of equation (4). This behavior can be characterized by the much nastier set of equations:

$$\Delta_x = L_t^2[\sin(pL)+B\cos(pL)+BpL\sin(pL)-B] \qquad (8)$$

$$B = p[\cos(pL)-1]/[\sin(pL)+pL\cos(pL)] \qquad (9)$$

$$P=[F_x/(IE)]^{1/2} \qquad (10)$$

Hence, as illustrated in figure 4, there is an upper bound on L set by P.

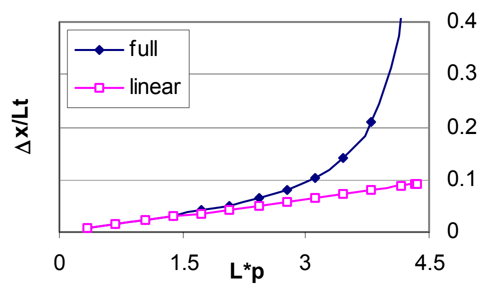

Figure 4. Full (8) and linear (4) approximations of finger deflection with onset of buckling.

III. Design and Operation of Variable Gap Transmission

Force in a GCA is proportional to the inverse square of the initial gap (G_o) between the parallel plates. This G_o dependence arises from the assumption that a constant load (F_L) must be initially overcome by the $1/(G_o)^2$ dependent electrostatic force. A variable gap transmission would allow adjustment of the initial gap and thus, of the displacement and force of individual actuation cycles. In an array of GCAs, such a variable gap could be used to permit actuation in two directions; a feature absent in previous inchworm designs. It can also be shown that both Force/Area efficiency and Energy efficiency can be drastically improved with a variable gap design.

Energy Efficiency Improvements

Because autonomous MEMS devices such as micro-robots must either scavenge power from the environment or carry a micro-battery, energy efficiency is critical. The energy input to an inchworm motor can be approximated by the energy stored capacitively in its GCAs at the end of a cycle:

$$E_{in} = \tfrac{1}{2}CV^2 = \tfrac{1}{2}\varepsilon_o V^2 A_c(1/(G_f) \qquad (11)$$

A_c is the plate-to-plate capacitor area and G_f is set by an anchored minimum gap stop, which prevents the plates of the gap closers from coming into contact. The mechanical energy output E_{out} is the product of the required pull-in force F_p and the distance closed by the GCA (G_o-G_f):

$$E_{out} = F_p(G_o-G_f) = (\tfrac{1}{2}\varepsilon_o V^2 A_c(1/(G_o)^2))(G_o-G_f) \qquad (12)$$

Combining (10) and (11) to estimate the GCA efficiency:

$$E_{out}/E_{in} = (G_f/G_o)(1-G_f/G_o) \qquad (13)$$

This expression implies that a maximum efficiency of 25% is achieved only when G_f/G_o=2. Since G_o=stepsize+G_f, G_f should be made as small as possible to lower G_o and thus increase force. The lower bound on G_f is set by a minimum feature resolution (λ_R) of the process, which is approximately an order of magnitude smaller than the minimum feature size (λ). Thus, if step length is equal to the minimum feature size (as in previous, fixed-step designs) and G_f is minimized, the energy efficiency is a factor of $(\lambda/\lambda_R)^{1/2}$ less than optimum. Hence, an adjustable gap allows the motor to simultaneously increase efficiency and force output.

Force/Area Efficiency

Assuming force is given as a design constraint and fabrication imposes a minimum feature size (λ), a logical step would be to optimize the motor design in terms of energy efficiency. This optimum energy efficiency of 25% is possible only when $G_o = 2G_f$.

With a fixed design, the minimum step size cannot dip below λ because each step must displace at least one gear tooth of size λ. Taking G_o=$2G_f$ to maximize efficiency, and setting the step size Go-Gf = λ requires that $G_f = \lambda$ and G_o=2λ. If finger length is L_f and finger width is λ, and we leave an optimal gap between GCAs of $2.7G_0$ [2] the minimum die area and force output of a fixed gap GCA is:

$$A_{fixed} = 2(9.4\lambda L) \qquad (14)$$

$$F_{fixed}=\tfrac{1}{2}\varepsilon_o V^2 A_c/(4\lambda^2) \qquad (15)$$

The factor of 2 on A_{fixed} is due to the fact that a fixed design has two anti-phase drive GCA arrays.

From (1), a vernier pawl allows a minimum step size G_o-G_f= $2\lambda/N$. With G_o=$2G_f$, the optimal dimensions are G_o=$4\lambda/N$ and G_f= $2\lambda/N$. Therefore, the die area and force output of a unit variable step GCA is

$$A_{var} = (2+14.8/N)\lambda L_f \qquad (16)$$

$$F_{var}= \tfrac{1}{2}\varepsilon_o N^2V^2A_c/(16\lambda^2) \qquad (17)$$

For any N>1 this leads to significant area efficiency improvements. For example, if N=4, we find:

$$((F/A)_{var})/((F/A)_{Fixed})=13.2 \qquad (18)$$

Transmission Operation

To gain the benefits described above, the shuttle driver GCA must be shifted from its initial fabricated position to a smaller G_o. This may be accomplished by applying a sufficient voltage to pull the plates in to their minimum plate separation G_f and inserting a gap stop behind the closed GCA. When the voltage difference is removed, the GCA will recoil under the spring force of its own suspension beams. By inserting a variable gap-stop that prevents the GCA from completely recoiling, the variable gap transmission can reduce the starting gap for the next cycle, setting G_o+G_f<λ. Furthermore, since the variable gap mechanism must only be able to resist the suspension spring forces (as opposed to the full load of the shuttle, $F_{L)}$ it need not be completely rigid.

Figure 5 is an image of the gap biasing mechanism. The comb drive is used to "shift gears" by properly aligning one of the gap stops with the contact point of the driver GCA. The top two gap stops set G_o for rightward actuation and the bottom two set the gap for driving left.

Although the driver GCA is preset to pull to the right, the rightward gears cannot be engaged until the GCA is closed once. While the GCA is closed, the transmission comb drives can move the appropriate gap-stop into place. This motor can reverse directions if the shuttle has been displaced at least G_o right of center. Reversing directions proceeds as follows: shift the transmission to neutral (past the rightward gap stops), engage the clutch, allow the shuttle's suspension beams to pull the driver gap closing array a distance G_o left of center, apply a sufficient voltage difference to close the gaps to the left, and align one of the bottom two gap stops with the contact point to select a leftward drive gear.

IV. PROCESS PARAMETERS

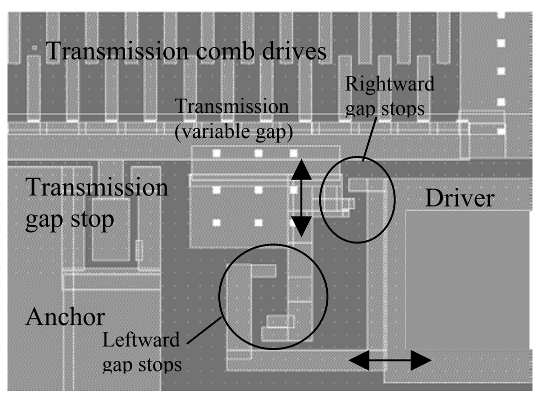

Figure 5. Transmission. A variable gap-stop is created by moving a set of stops up and down with a comb drive.

The structures described above were designed and will be fabricated in a single mask SOI process with the following design rules:

Minimum feature separation: λ_s=3µm
Minimum beam width: λ =5 µm
Maximum beam width: 15 µm
Maximum allowed design area: 2 mm^2
SOI Layer thickness: Silicon-50µm Oxide-2µm

V. Summary of Fuzzy Inchworm Motor Design

The overall design of an inchworm motor incorporating the vernier pawls and variable-gap transmission described above is shown in figure 7. Two clutches are used; one is attached to the main shuttle driver and transfers force and displacement from the driver GCA to the shuttle, the other holds the shuttle in place while the driver resets. A set of comb drives adjusts the transmission setting. The design is sufficiently modular as to permit individual testing of the static clutch, the driver, and the transmission. In addition, a frictional pawl plus GCA permit loading of the shuttle and several suspended GCAs of dimensions similar to those used in the design permit independent characterization. The design is intended to be driven with 30V signals

In this process, λ is 5um, so the periodicity of the shuttle teeth is 11µm, with a tooth height of 2µm. This in turn sets Δ_x to be 0.5µm (to account for deflection in both static and driver clutch pawls). A G_f value of 1µm was found to be sufficient for preventing finger pull-in on 40µm fingers.

Given a desired output force of Fx=500uN, and minimum width beams with I=520μm^4, and assuming E= 160Gpa, the buckling length on the vernier finger is 1.8mm. With L=400µm, there is an effective safety factor of 4. Keeping the F_x/F_{cl} ratio greater than 1 forces N=4. This permits a ratio of Fx/F_{cl} =11, but the design as implemented is somewhat more conservative, using a ratio of about 4, to allow for uncertainties in the process.

Given these values of N and l, the shuttle gap closers must displace by at least 3µm; 3.5µm was used to be safe. In the interest of achieving high force to area efficiency, a gap stop G_f of 1µm was used (this degrades energy efficiency from the ideal case). The transmission is set up then to permit forward and backward motion in 3.5 µm steps. Two additional gears were included using a much smaller step (1.5µm as drawn) this permits some additional testing of the transmission idea, (and may turn out to be usable under sufficiently high over-etch conditions).

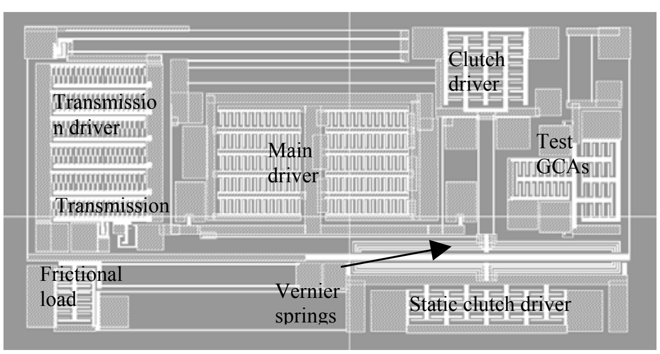

Figure 7. Layout for prototype fuzzy inchworm motor. This design incorporates a main drive gap-closing actuator, a variable gap transmission, and two clutch drivers with associated vernier spring pawls.

As it stands, in the large gap transmission setting we expect an energy efficiency of 11% (accounting for energy consumed by clutch drivers), a force per-area efficiency in the driver of 1250$\mu N/mm^2$ and an over all force/area efficiency of 300$\mu N/mm^2$. The force/area is degraded partially by the two clutch drivers and a great deal by the transmission comb drive. A rough calculation of area taken by an inchworm motor exerting similar force in the same process (using two antiphase drivers) but without use of a transmission, requires about six times as much area for the driver GCAs and twice the area total.

VI. Future Work –Step Size Reduction

One potentially viable method of further relaxing minimum step constraints is to eliminate teeth from the shuttle. However, a smooth shuttle is generally unpopular because the required normal force between pawl and shuttle must exceed the desired driving force by as much as a factor of 2.5 to account for a coefficient of static friction of 0.4[3]. Generating such a high force from the clutch actuator is inherently inefficient.

One approach to overcome this is to design a pawl that tightens its grip on the shuttle as more drive force is applied. Such self-locking mechanisms come in many forms in the macro scale and may turn out to be feasible. Such shuttle ideas, however, require multiple structural layers and thus could not be attempted in the current process.

A smooth design facilitates replacement of discrete G_o values by a continuously variable transmission. The discrete gap transmission could easily be made continuous by simply replacing the stair step "gears" with a continuously sloped edge. A continuous design would allow drastic force/area improvements by effectively removing the minimum step size constraint.

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