



Negative-Pressure Soft Linear Actuator with a Mechanical Advantage

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Negative-Pressure Soft Linear Actuator with a Mechanical Advantage

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Soft, linear actuators (those that generate motions in a straight line—for example, the muscles of animals^[1]) have emerged from evolution as the best solution for moving limbed organisms such as vertebrates and arthropods (as well as many organisms without limbs, such as mollusks, annelids, and jellyfish) in an unstructured environment. Their compliance enables adaptive interactions with the environment, and non-damaging (if required) contact with one another. At the same time, this compliance reduces the cost to the organism of precise controls and feedback loops.^[2, 3] Linear actuation is also particularly compatible with a limbed body-plan in its geometric adaptability since this body-plan is often based on rigid or semi-rigid structural elements (the skeleton), which move relative to one another by linear contraction of muscles around fulcra (e.g. joints). Hard, human-engineered machines now often use the rotary motion of electric motors, although pneumatic and hydraulic pistons (based on force generated by expansion of hot gas, pressurized air, vacuum, or pressurized liquid), other actuators (e.g. magnetic solenoids), and transducers of biomechanical forces (e.g. screw drivers) are also important. A key characteristic of soft machines is that they can be made collaborative (e.g. intrinsically safe in close proximity to humans).

Devices or systems that generate a mechanical advantage—levers, gears, chain drives, block and tackles, and others—are useful in amplifying either force or displacement in hard machines. Humans and other vertebrates also use hard levers—system of bones, articulated joints, and tendons—to amplify the displacement that muscles generate (albeit at the cost of reduced force)^[1]. A device that generates a mechanical advantage—i.e. amplifies force—in a soft system would expand the capabilities of soft robots and machines. This paper reports the first of such devices. The compliance of pneumatic soft actuators allows them to distribute their stress over large areas,^[2] but it also means the pressure output of these systems is often limited by their

pressure input, and thus also by the mechanical characteristics—especially the Young’s Modulus—of the material of which they are made.^[4] A soft pneumatic actuator designed to generate a mechanical advantage would help to overcome this limitation.

Our goal was to introduce mechanical advantage into a soft pneumatic actuator. Figure 1A is a conceptual picture summarizing this goal. Compared to lifting a weight (generating $mg\Delta h$ work) directly using a pneumatic piston (supplying $P\Delta V$ work), pulling a weight on a slope increases the weight m pulled per applied pressure P , while decreasing the height Δh per volume change ΔV . The objective here is to realize this mechanical advantage in a soft actuator without using an external slope.

This paper demonstrates a new design of a soft linear actuator that generates a tunable mechanical advantage. For simplicity, we call these structures “shear-mode vacuum-actuated machines,” and abbreviate them with the acronym “shear-VAMs”. A shear-VAM comprises two flexible but inextensible strips bridged by tilted parallel beams. These beams can be fabricated of an elastomer, a composite of soft and rigid material, or rigid structures (trusses) that can pivot on their ends; the system we have explained are of the first (elastomeric) class. The spaces between the beams are sealed pneumatically with two thin elastomeric membranes, and thus form void chambers within. These chambers are connected to a single external vacuum source (or, in more complex devices, multiple independently controllable sources). Through a network of channels embedded in the structure, a shear-VAM operates by reducing the pressure of void chambers in an elastomeric structure to below that of atmospheric pressure (that is, to negative pressure, or partial vacuum). When the chambers are evacuated, ambient atmospheric pressure compresses the two inextensible strips together. The design—in which the beams bend more easily than they

compress—causes these beams to tilt further; this increase in tilt, in turn, causes the strips to translate parallel to one another, and to generate force (see Figure 1B, Movie S1).

These structures, together with soft actuators that we (in the form of pneu-nets^[5-9], buckling actuators^[10], and vacuum-actuated muscle-inspired pneumatic structures^[4]) and others (flexible microactuators^[11], jamming grippers^[12], dielectric elastomer-driven actuators^[13, 14], shape memory and cable-driven soft arms^[15], etc.) have described, belong to a new class of machines—soft machines^[2, 16]—which are more collaborative, often more adaptive to irregular targets^[16], and sometimes simpler to control^[16] than more familiar hard machines.

Among these soft pneumatic actuators, we refer to those powered by negative pressure (vacuum) rather than positive pressure as vacuum-actuated machines (VAMs). These devices allow a range of functions that can sometimes be difficult to achieve by their conventional pressure-driven counterparts. Examples of VAMs include rotary actuators that combine vacuum and reversible buckling of elastomeric beams as their mechanism of action (rotary-VAMs),^[10] and linear actuators (using the same mechanism) that mimic the performance—and many useful functions—of human muscle (linear-VAMs).^[4]

Vacuum-actuated machines (VAMs) such as shear-VAMs are safer around humans than many “hard” actuators, and even ostensibly soft actuators that operate under high positive pressure (e.g. McKibben actuators and many of their relatives^[17]). Actuators powered by pneumatics are also safer and less likely to fail than those powered by high voltages (e.g. actuators made with dielectric elastomer—such as those explored by SRI^[13, 14]—which can fail by dielectric breakdown).

Experimental design of shear-VAMs. A design representative of a simple shear-VAM consists of two flexible but inextensible strips (1.5 mm thick, 17 mm wide, and 10 mm apart)

bridged by four tilted parallel elastomeric beams (1.8 mm wide, 8.9 mm long, at 63° angle to the strips, and spaced in 10 mm-intervals along the strips, in Figure 1C). We used a Nylon mesh embedded in an elastomer—Ecoflex 00-30 (Young’s modulus $E = 43$ kPa), Dragon Skin 10 Slow ($E = 153$ kPa), or Elastosil M4601 ($E = 520$ kPa)—for the inextensible layer. We used the same elastomer in the elastomeric beams in shear-VAMs. The empty spaces in between the beams were converted into enclosed chambers by sealing (front and back) with thin membranes made of the same elastomer (1 mm thick). The final structure comprises a pneumatic structure with several chambers connected to a common source of negative pressure (e.g. vacuum, or pressure less than the ambient pressure). The beams each had an opening in the middle (a 3 mm-wide slit) such that the chambers were pneumatically connected. The chambers were further connected to an external source of vacuum through a piece of tubing that pierces one of the strips. (Figure S1 and the SI summarize details of fabrication.)

Characterizing the maximum force of actuation of shear-VAMs. A shear-VAM is similar to a pneumatic or hydraulic piston, in that it works by converting an applied pneumatic pressure ΔP to an output force F . As we apply an increasing difference of pressure ΔP (as defined in Equation 1) between that of the atmosphere external to the shear-VAM (P_{ext}), and that of the partial vacuum inside it (P_{int}), its void chambers deflate, and the two inextensible strips translate relative to each other (as shown in Figure 1B, Movie S1).

$$\Delta P = P_{ext} - P_{int} \quad (1)$$

The two inextensible strips move until, at a critical difference of pressure ΔP_{crit} , the void chambers collapse completely (or as completely as they can within the limits of the design) and bring the actuator to a stop. The actuation of a shear-VAM results in a decrease in its length Δh

(also indicated in Figure 1). We defined this change in length Δh —effectively the relative distance of translation between the two inextensible strips—to be the distance of actuation of a shear-VAM. The actuation of a shear-VAM also applies a force F , as indicated in Figure 1B and defined in Equation 2, where m is the mass of a test object, and a is the acceleration of that object. We defined this force—the force that lifts and accelerates a load—to be the force of actuation of a shear-VAM.

$$F=mg+ma \tag{2}$$

The distance of actuation Δh (upon application of a difference of pressure $\Delta P > \Delta P_{crit}$) is determined primarily by the geometry of the shear-VAM. Figure 2A shows that this distance Δh stays roughly constant when various loading forces F (in N, given by a hanging weight) are applied to the shear-VAM while it actuates, as long as the loading force is less than a certain maximum value F_{max} . We define F_{max} to be the maximum force a shear-VAM of this particular design can generate. For a load F greater than F_{max} , the beams in the shear-VAM will tilt in the opposite direction when the pressure ΔP is increased. In other words, the shear-VAM *lifts* the weight for a distance of Δh while $F < F_{max}$ (i.e. produces a contraction), and it *lowers* the weight for a distance of $\Delta h'$ while $F > F_{max}$ (i.e. produces an elongation). Figure S2 and Movie S2 demonstrate this effect. The distance of elongation $\Delta h'$ is again roughly constant under various constant loads F greater than F_{max} , as $\Delta h'$ is also determined primarily by the geometry of the shear-VAM.

The value of F_{max} is dependent on various characteristics (geometry and materials parameters) of the actuator. For example, for shear-VAMs that have the same geometry, ones that are stiffer (i.e. made of elastomers with higher Young's modulus) generate a higher force upon actuation than those that are less stiff. Figure 2B shows that the maximum force of actuation F_{max} (in N) of a shear-VAM is proportional to the Young's modulus E (in Pa) of the material of which it is fabricated (Equation 3):

$$F_{max}=kEL^2, \quad (3)$$

where L is the length scale (in this case the length) of the shear-VAM, and k is a dimensionless constant. Figure S3 and Movies S3, S4 show that shear-VAMs with indistinguishable geometries, but made in different materials, lift different weights.

This linear relationship (Equation 3) is confirmed theoretically through dimensional considerations (a detailed theoretical analysis is in the SI). This scaling property (Equation 3) allows one to construct shear-VAMs capable of generating a high force simply by choosing a stiff elastomer during fabrication.

Characterizing the thermodynamic efficiency of shear-VAMs. The thermodynamic efficiency of transduction of the pressure-volume work required to actuate shear-VAM into mechanical (force \times distance) work (e.g., lifting a weight) is primarily governed by the work required to compress the elastomer. (Here, we only account for the pressure-volume work and not the electrical energy required to operate the vacuum pump). The loss of energy due to hysteresis is small relative to the work done to compress the elastomer (details are in the SI, Figure S6). Our experimental data yield a thermodynamic efficiency of 35% for a distance of actuation of ~ 4.7 mm at 200 g loading for the shear-VAM shown in Figure 1B. (For comparison,

the corresponding value of a human skeletal muscle is ~40%).^[18] Repeated cycles of actuation can lead to higher efficiency because the energy stored in the deformed, elastomeric components can, in principle, be at least partially recovered during unloading. Another similar VAM—a linear VAM^[4]—has a similar potential for recovery of energy (the ability of soft actuators to store and recover energy over repeated cycles of actuation has been reported in a number of designs^[18]).

Characterizing the mechanical advantage of shear-VAMs. We define the geometrical parameters that characterize a shear-VAM (Figure 3A), where: L is the length of the elastomeric body of the actuator (i.e. the long dimension of the parallelepiped), a is the length of the tilted beams, b is the width of the actuator (i.e. the third dimension of the parallelepiped), α is the angle between the strips and the beams, and $A=L*b$ is the “lateral area” of the shear-VAM (the shaded area in Figure 3A). The force of actuation is approximately given by Equation 4 (the SI includes a theoretical derivation, and a comparison to conventional pneumatic/hydraulic systems).

$$F=\eta\alpha A\Delta P/\tan\alpha, \quad (4)$$

where $\eta\alpha$ is the thermodynamic efficiency of the shear-VAM for an infinitesimal movement near angle α . This value is approximately equal to the total thermodynamic efficiency of the shear-VAM η .

Equation 4 indicates that we can increase the force of actuation of a shear-VAM by increasing the lateral area of the shear-VAM $A=Lb$ (Figure 3A). Assuming the pneumatic source (not shown in Figure 3A) has a fixed working area (e.g. area of the diaphragm of a pump, or the

area of the plunger of a syringe) of A_0 , it generates a driving force of $F_{in}=A_0\Delta P$. The shear-VAM demonstrates a net mechanical advantage (MA) given by Equation 5.

$$MA=F/F_{in}=\eta A/(A_0\tan\alpha) \quad (5)$$

We note that the mechanical advantage of a shear-VAM can be, in principle, increased indefinitely as we increase the lateral area A (although the MA is, of course, limited by the tensile strength of the strips). Figure 3B, C illustrate two cases where we increase either the length L or the width b to increase the lateral area A , and consequently, to boost the force of actuation F .

In particular, increasing the length L allows a shear-VAM to increase its force of actuation F without increasing the apparent cross-sectional area $basin\alpha$. This feature allows shear-VAMs to have a mechanical advantage not only for force, but also for pressure. This mechanical advantage (MA_p) is defined as the ratio of the pressure that performs useful work to the pressure that is applied (Equation 6).

$$MA_p=P_{out}/P_{in}=\eta L/(asin\alpha), \quad (6)$$

where $P_{out}=F/(basin\alpha)$, and $P_{in}=\Delta P$.

Equation 5 also indicates another source of mechanical advantage, which comes by changing the angle α . Due to the inverse relationship between mechanical advantage MA and $\tan\alpha$, one can increase the mechanical advantage of any shear-VAM simply by reducing the

angle α . When the angle α is small, the distance of actuation Δh per unit of change in angle $\Delta\alpha$ is also smaller—a tradeoff of smaller distance for larger force of actuation F .

Figure 3D shows the relationship between the force of actuation F of shear-VAMs of two different lengths $L=62\text{ mm}$ and 32 mm (each connected to a fixed strain gauge), and the difference of pressure ΔP (in kPa) applied across the inside and outside (the ambient atmosphere) of the void chambers of these shear-VAMs (see the SI and Figure S6 for details of this measurement). At the same ΔP , the curves show a near doubling of force of actuation F , when the length L is doubled—a result consistent with Equation 4. Figure 3E shows that the maximum force of actuation of shear-VAMs F_{max} also increases with their length L , consistent with Equation 4. The plot verifies that the relationship is linear. Error bars in Figure 3D, E were measured from seven replicate measurements of the same sample (standard deviation of measurements on different devices are larger than those from repeated measurements of the same device, as shown in Figure S4, but can in principle be greatly reduced in machine-made devices as opposed to handmade ones).

Parallel actuation and stackability of the shear-VAMs. Multiple shear-VAM units can be positioned in parallel or in series and actuated together to generate more force or more distance of actuation (see Figure 4 and Movies S5, S6). Figure 4A shows two shear-VAMs working in parallel in a mirror configuration. This configuration generates about twice as much force ($\sim 2F$) as a single shear-VAM of the same length L , but has about the same distance of actuation ($\sim \Delta h$). Figure 4B shows two shear-VAMs working in series. This configuration has about twice the

distance of actuation ($\sim 2\Delta h$) as a single shear-VAM of the same geometry, but generates about the same force ($\sim F$). These force or distance scaling relationships are universal to parallelizing or stacking of any linear actuator. Shear-VAMs, in particular, are naturally fit for parallelization, as their lateral areas remain flat during actuation (as shown in Figure 4A; the same is not true for other pneumatic linear actuators such as McKibben actuators).

Using shear-VAMs in robots that locomote. The agonist-antagonist arrangement is useful in the muscle of animals in enabling more effective movements. Since shear-VAMs resemble biological muscle in that they are soft linear actuators, this arrangement can be borrowed in making devices with shear-VAMs that move or locomote. Figure 5 and Movies S7, S8 show a swimming device that uses a pair of shear-VAMs in an agonist-antagonist arrangement to drive its paddle. The paddle moves either forward or backward when the corresponding shear-VAMs actuate and pull the lever that is connected to the paddle. The paddle can pivot around its connection backwards but not forward—this design helps to generate a hysteresis that is required to propel the swimmer forward in water.

Conclusions. Shear-VAMs have three characteristics that are useful for making soft machines. i) They provide a tunable mechanical advantage. ii) They can be easily used in series or in parallel. iii) They contract rather than expand in volume on actuation. Shear-VAMs and other soft pneumatic actuators are useful in supplementing more familiar hard machines with the following advantages: i) increased safety in use around humans or animals, and non-damaging interactions with delicate objects; ii) low cost of fabrication; iii) light weight and low density (the actuating fluid is air, and the elastomers we use have densities around $\sim 1 \text{ g/cm}^3$).

A shear-VAM is a soft linear actuator that works by converting the pneumatic pressure applied perpendicular to its inextensible lateral surfaces to a force parallel to them via tilted

elastomeric beams. It provides a mechanical advantage (that is, magnification) in terms of both force and pressure relative to the input. It does so by increasing its length (for both force and pressure) or width (for only force). The design of shear-VAM provides a new tool for making biomimetic and/or functional soft machines. Shear-VAMs could, in particular, be useful for generating high forces or generating reasonable forces with a small input pressure in a soft structure.

Supporting Information

Supporting Information is available from the Wiley Online Library or from the author.

Acknowledgments

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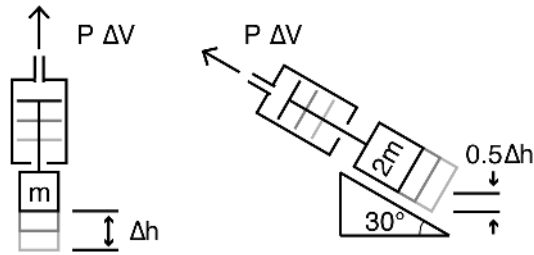
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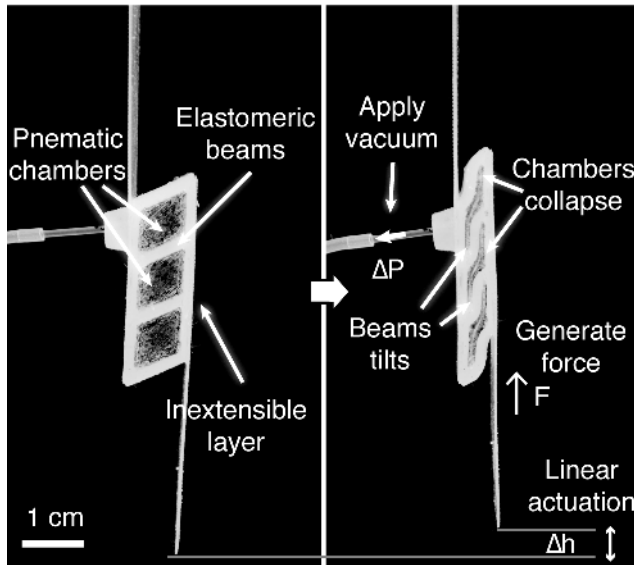
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A An example of mechanical advantage



B Mechanism of motion of a shear-VAM



C Schematic representation of a shear-VAM

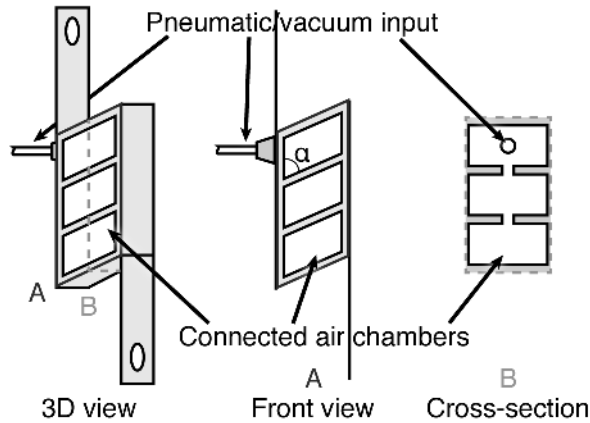


Figure 1. Schematic description of a shear-mode vacuum-actuated machine (shear-VAM). A) An example of mechanical advantage in pneumatic actuation (ignoring effects of friction). B) Mechanism of motion of a shear-VAM. A shear-VAM consists of two flexible but inextensible

strips bridged by tilted parallel elastomeric beams, with the closed chambers connected pneumatically to a source of vacuum. When the chambers are evacuated, the two strips come together, while the beams tilt further and push the strips to move parallel to one another, and generate a distance of actuation Δh and/or a force of actuation F (depending on the loading condition). C) Schematic representation of a shear-VAM.

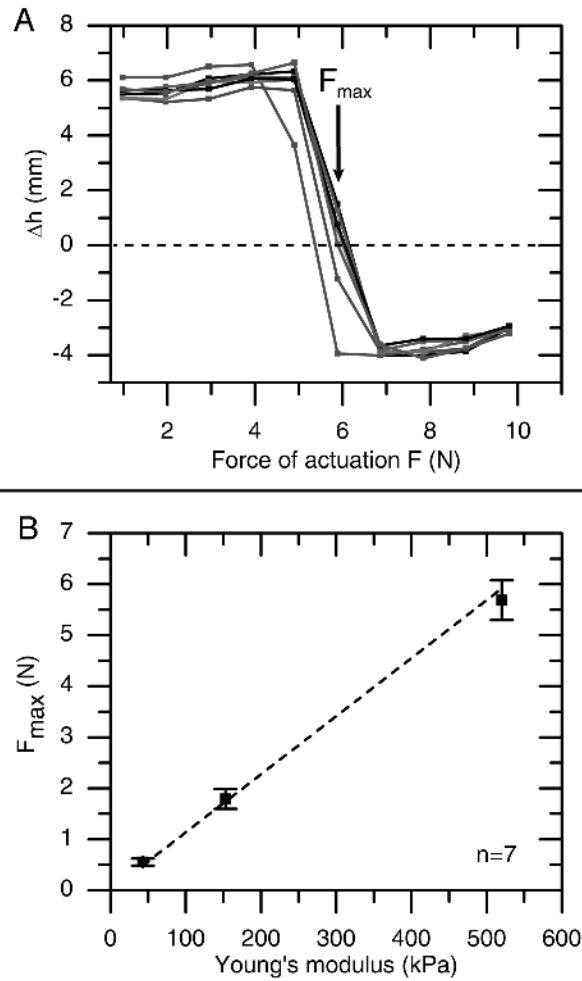


Figure 2. Characterizing the maximum force of actuation of shear-VAMs. A) The relationship between the distance of actuation Δh (in mm) and the force of actuation F (in N) on shear-VAMs, measured on seven different samples made of Elastosil ($E = 520$ kPa). A sufficient difference of pressure $\Delta P = 90$ kPa $>$ ΔP_{crit} is applied to collapse the void chambers completely. B) The relationship between the maximum force of actuation F_{max} of shear-VAMs (in N) and the Young's Modulus E (in kPa) of the elastomer used in fabricating the shear-VAMs.

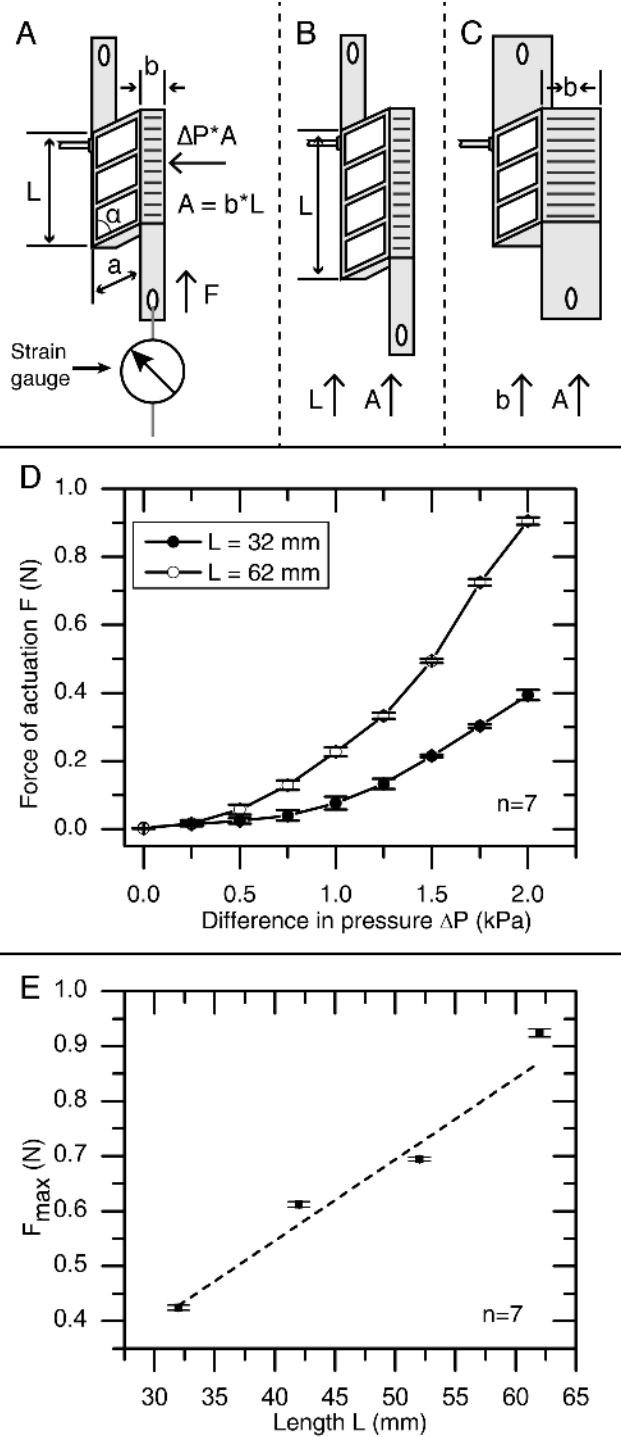


Figure 3. Characterizing the mechanical advantage of shear-VAMs. A) A schematic diagram marks different dimensions of a shear-VAM (length L , beam length a , width b , beam angle α ,

and lateral area A). B, C) Schematic drawings illustrate increasing either the length L or width b of a shear-VAM increases its lateral area A , and thus increases the force of actuation F (Equation 4). D) The relationship between the force of actuation F of shear-VAMs of two different lengths $L=62\text{ mm}$ and 32 mm (each connected to a fixed strain gauge), and the difference of pressure ΔP (in kPa) applied across these shear-VAMs (see the SI and Figure S6 for details of this measurement). Data shown are mean \pm S.D. ($n = 7$ repeated measurements). E) The maximum force of actuation of shear-VAMs F_{max} (in N) made of Ecoflex ($E = 43\text{ kPa}$) vs. their length L .

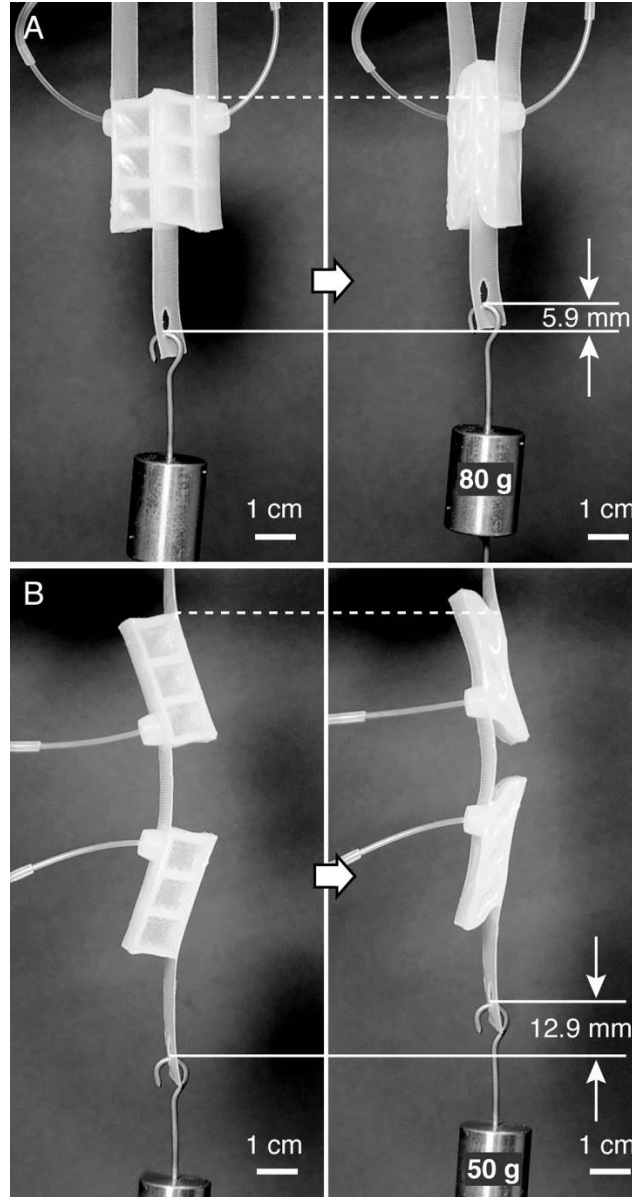


Figure 4. Multiple shear-VAMs working in combination. A) Two shear-VAMs working in parallel in a mirror configuration. This configuration generates about twice as much force compared to a single shear-VAM of the same geometry, but has about the same distance of actuation. B) Two shear-VAMs working in series. This configuration has about twice as much distance of actuation as a single shear-VAM of the same geometry, but generates about the same force.

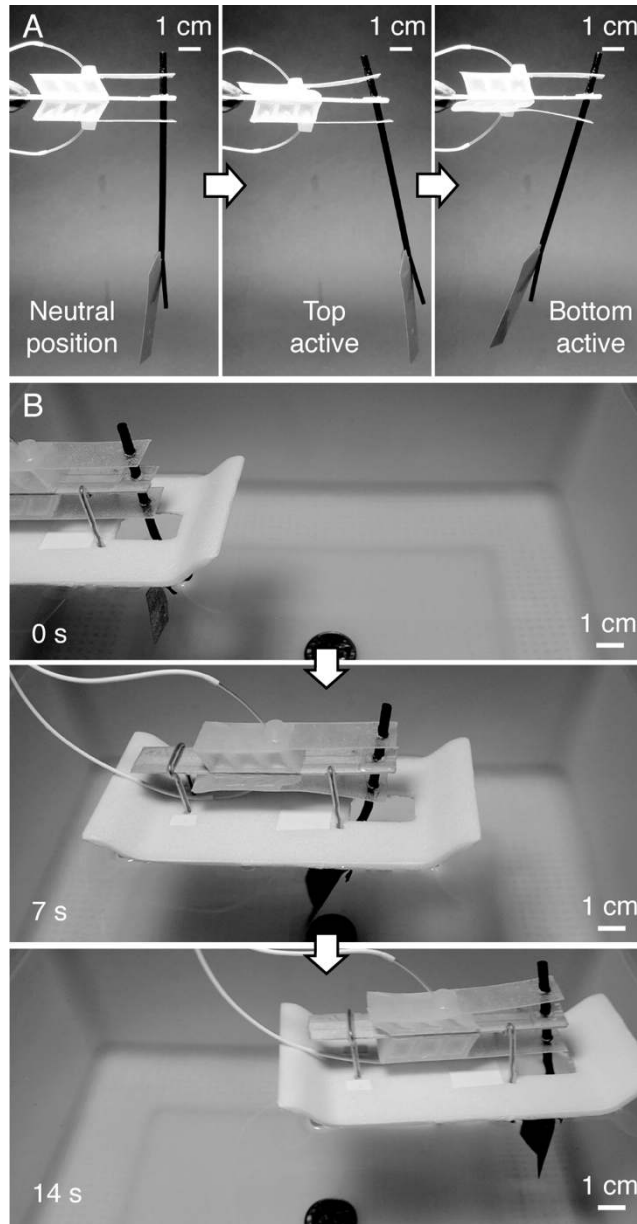


Figure 5. Soft robot actuated with shear-VAMs. A) Two shear-VAMs in an agonist-antagonist arrangement can drive a paddle back and forth. The paddle can pivot around its connection backwards but not forward. This mechanism can be used in a soft machine that paddles. B) A soft robotic swimmer with a paddle powered by two shear-VAMs in an agonist-antagonist arrangement. Scale bars are 1 cm-long. A quarter coin in the water also marks the scale.

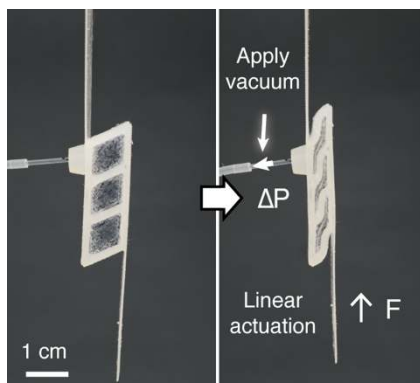
Soft linear actuators are useful for making collaborative machines (e.g. machines that interact safely with people in continuous, transitory, intentional, and incidental contact) and in designing biomimetic robots based on animals over a range of levels of evolutionary development. This study demonstrates a new type of soft linear actuator that uses negative pressure (vacuum) for its actuation. This actuator also acts as a device designed to generate a tunable mechanical advantage—the ratio of the force that performs useful work to the force that is applied—in soft materials. It expands the capabilities of soft robots and machines, and in particular, overcomes one of the limitations of current soft pneumatic actuators—that is, the force they apply is limited by the pressure used to actuate them.

Keywords

vacuum, soft actuators, pneumatic actuation, linear actuator, mechanical advantage

D. Yang, M. S. Verma, E. Lossner, D. Stothers, and G. M. Whitesides*

Negative-Pressure Soft Linear Actuator with a Mechanical Advantage



Supporting Information

Negative-Pressure Soft Linear Actuator with a Mechanical Advantage

Dian Yang, Mohit S. Verma, Ju-Hee So, Elton Lossner, Duncan Stothers, and George M.

*Whitesides**

Fabrication of shear-VAMs

The elastomeric parts of shear-VAMs were created by replica molding (Figure S1). We designed the molds using computer-aided design (CAD) (Solidworks) and fabricated them with acrylonitrile butadiene styrene (ABS) plastic using a 3D printer (StrataSys Fortus 250mc). Curing a silicone-based elastomer (Ecoflex 00-30, dragon skin 10 slow, or Elastosil M4601) against the molds at room temperature (4 hours for Ecoflex 00-30, and 6 hours for dragon skin 10 slow and Elastosil M4601) produced two halves of the shear-VAMs. These two halves were aligned and bonded together by applying uncured elastomer at their interface, prior to curing once again at room temperature for the same amount of time.

The inextensible strips were fabricated by placing pieces of nylon mesh on top of an acrylic plastic board, then pouring the corresponding elastomer over the mesh. A wooden stick was used to smooth the top surface of the elastomer. The composite strip is then formed after

curing the elastomer in a 60 °C oven for 15 min. Two holes 6-mm in diameter are punched at the end of each strip for fixation purpose. Another smaller 3.5-mm diameter hole is punched on one of the two strips on a shear-VAM so that tubing can go through and transduce pneumatic pressure. The body of shear-VAM and the two strips are assembled by applying the same elastomer that they are made of as glue, and curing them at room temperature for 6 hours.

A conically shaped piece of the elastomer was bonded to the side of the actuator to provide additional material that allowed tubing (Intramedic polyethylene tubing, ID 0.76mm) to be securely attached to the structure. The conical piece was first pierced by a cannula. The tubing was fed through the cannula, which was then removed to leave the tubing embedded in the shear-VAM. The tubing was secured by elastic deformation of the elastomer, which, as the tubing displaced some of its volume, reacted by applying pressure to close the hole surrounding the tubing.

Why the Young's modulus E of the elastomeric material used to fabricate a shear-VAM is proportional to its maximum force of actuation F_{max}

A similar analysis was described in our paper on linear-VAMs,^[1] we have adapted the analysis here. Consider the body of the shear-VAM. Since the inextensible strips are not under substantial deformation during actuation, the two side-surfaces of the shear-VAM that touch the inextensible strip can be considered to be each in a fixed boundary condition relative to itself. The two surfaces are, however, not fixed relative to each other. Due to the C2 symmetry of the shear-VAM, the two surfaces must move parallel to each other during actuation, and a shear stress is applied during this actuation. Let τ be the shear stress due to the hanging weight—that

is, the load F divided by the lateral-area of the undeformed actuator A . And let s be the shear strain between the two surfaces—that is, the relative movement of the two strips Δh divided by the length of the actuator L (Equation S1). We regard an actuator as a thermodynamic system of two independent variables that can independently change the state of the actuator—the difference of pressure ΔP and the loading stress τ . To a good approximation, the elastomer is incompressible. Hence the state of the actuator depends on difference of pressure ΔP , but not on the absolute pressures inside and outside the actuator. Since the experiment is insensitive to small change in temperature, we do not list temperature as a variable. Using the neo-Hookean model, we may obtain Equation S2 on the basis of dimensional considerations:

$$s = \Delta h / L \quad (\text{S1})$$

$$s = g(\Delta P / E, \tau / E), \quad (\text{S2})$$

where g is a function of two variables and E is the Young's modulus of the elastomeric material.

If we make two actuators with indistinguishable geometric features, but of materials with different Young's moduli, we can plot s as a function of $\Delta P / E$ and τ / E . The two surfaces will fall on top of each other.

Notice that according to Equation S2, if one increases E , ΔP , and τ by a common factor k , one obtains the same strain s . Assuming E , ΔP , and τ are increased to $E' = kE$, $\Delta P' = k\Delta P$, and $\tau' = k\tau$, we have relationship S3, where if Equation S2 describes a shear-VAM made of an elastomer of

modulus E then Equation S4 must describe a shear-VAM of indistinguishable geometry, but made of an elastomer of modulus E' .

$$\tau/E = \tau'/E' \quad (\text{S3})$$

$$s = g(\Delta P/E', \tau'/E'), \quad (\text{S4})$$

We note that a shear-VAM of modulus E can lift weight $F = \tau A$ only if the absolute value of $s(\Delta P)$ monotonically increases with ΔP ; note also that Equation S2 and S4 as a function of ΔP must be simultaneously monotonically increasing, if one of them is so. Therefore a shear-VAM of modulus E can lift weight $F = \tau A$ if and only if a shear-VAM of modulus E' can lift weight $F' = \tau' A$. In other words, the Young's modulus E of the elastomeric material used to fabricate a shear-VAM is proportional to the maximum load it can lift.

Measurement of thermodynamic efficiency

We generated the pressure-volume hysteresis curves by pumping water (an incompressible fluid) in and out of the shear-VAMs. The actuator was fully submerged in a 1-gallon container of water. The hydraulic actuation, and measurement of volume was performed with a syringe pump (Harvard Apparatus, PHD 2000), and the pressure measurement was performed with a pressure sensor (Transducers Direct, TDH31) connected to the syringe pump and the pressure transfer line (Figure S5). We fixated the actuator in a position that was submerged fully in water. We filled the actuators with water by submerging them in the container of water and deflating them several times until bubbles no longer emerged.

Within each test, we switched from deflation to inflation when the actuator had achieved approximately complete contraction (about 3 mL change in volume). We chose the rate of deflation and inflation to be 1 mL/min, which was sufficiently slow to achieve quasistatic conditions. We repeated the deflation-inflation cycle seven times.

The fluid used for inflation/deflation (water) is effectively incompressible that we could equate the volume decrease/increase of fluid in the syringe to that of the increase/decrease in the volume of the channels in the shear-VAMs. The shear-VAMs required removal of $V_0=3$ mL of water to achieve an actuation distance of $\Delta h \sim 4.7$ mm, while lifting a 200 g test weight, and while the applied differential pressure ramped up from 0 kPa to 15 kPa. We calculated the thermodynamic efficiency η by dividing “energy out” E_{out} by “energy in” E_{in} (Equation S5). E_{out} was obtained by calculating the potential energy gain of lifting the weight ($m = 200$ g was the weight we used, and g is the acceleration due to gravity) (Equation S6). E_{in} was obtained by integrating the differential pressure with respect to the change in volume (Equation S7). This value is represented by the area under the P-V curve (Figure S5B).

$$\eta = E_{out}/E_{in}. \quad (S5)$$

$$E_{out} = mg\Delta h. \quad (S6)$$

$$E_{in} = \int_{V_0}^0 P(V) dV. \quad (S7)$$

Over a total of six runs, we obtained an efficiency of $\eta = 35\% \pm 1\%$. Note that the loss of energy due to hysteresis was small compared to the work done by the syringe pump during actuation

(Figure S5B). The loss of efficiency was mainly due to the storage of elastic energy in the elastomer, with a small contribution from hysteresis.

Approximate theoretical derivation of the force of actuation F

Due to conservation of energy, the generation of a higher force of actuation F of shear-VAM (or of any other pneumatic actuators) requires the supply of a higher difference of pressure ΔP . Although ΔP is limited to 1 atm under atmospheric pressure, we will show that the force of actuation F of shear-VAMs increases linearly to the “lateral-area” of the strip A under fixed ΔP , limited only by the tensile strength of the strips. We will show that, unique to shear-VAM, the effective cross-sectional area of a shear-VAM does not necessarily increase as we increase A , allowing the shear-VAM to generate a mechanical advantage.

We can derive (approximately) the force of actuation F a shear-VAM produces for a given difference of pressure ΔP (that is less than the critical difference of pressure of a shear-VAM ΔP_{crit}) through an analysis of virtual work.^[2] When a difference of pressure ΔP is applied, the volume of the void chambers decreases, and the two strips move towards each other. Assume an infinitesimal reduction of angle α to $\alpha - \delta\alpha$ (in radian), while the “lateral-area” A moves under force $\Delta P \times A$, and the shear-VAM reduces its volume from $A \times a \times \sin\alpha$ to $A \times a \times \sin\alpha - \delta\alpha$. The pneumatics virtual work (pressure-volume work) done to the system is:

$$\begin{aligned} \delta W_{in} &= \Delta P \times \delta V = \Delta P \times A \times a \times \sin\alpha - a \times \sin\alpha - \delta\alpha \\ &= \Delta P \times A \times a \times \cos\alpha \times \delta\alpha \end{aligned} \quad (S8)$$

where δV is the change of overall volume of the elastomeric part of shear-VAM. Since the elastomer is incompressible, δV is also equal to the volume of air pumped out of the void chambers of the shear-VAM. In the meantime, the shear-VAM exerts force F over a distance of $\delta h = (a \times \cos \alpha - \delta a - a \times \cos \alpha)$. The virtual work output (force-distance work) by the system to lift the load is:

$$\begin{aligned} \delta W_{out} &= F \times \delta h = F \times (a \times \cos \alpha - \delta a - a \times \cos \alpha) \\ &= F \times a \times \sin \alpha \times \delta \alpha \end{aligned} \quad (S9)$$

Dividing Equation S9 by Equation S8, we obtain:

$$\eta \alpha = \delta W_{out} / \delta W_{in} = F \times \tan \alpha / (\Delta P \times A) \quad (S10)$$

where $\eta \alpha$ is the efficiency of the shear-VAM near angle α , defined as the ratio between work out δW_{out} and work in δW_{in} (Equation S10). Since we know $\delta W_{in} = \delta W_{out} + \delta W_{lost}$, where δW_{lost} is the elastic and inelastic energy lost in compressing the elastomers, we know $\eta \alpha < 1$.

Equivalently, Equation S10 can be written as a formula for the force of actuation F :

$$F = \eta \alpha \times \Delta P \times A / \tan \alpha \quad (S11)$$

The efficiency $\eta \alpha$ comes primarily from the elastic loss in collapsing the chambers, and should be, in principle, constant (for the same angle α) as long as the beam length a and the beam spacing (10 mm in this design) are fixed. This efficiency $\eta \alpha$ is approximately equal to the thermodynamic efficiency of the shear-VAM η . Using the data plotted in Figure 3D, the

efficiency $\eta\alpha$ in two shear-VAMs of different length $L=62\text{ mm}, 32\text{ mm}$ can be calculated using Equation S10: the efficiencies are about the same for the two different lengths L , at $\sim 40\%$ (at $\Delta P \approx 2\text{ kPa}, \alpha \approx 45^\circ$), and they are similar to the total thermodynamic efficiency of the shear-VAM ($\sim 35\%$).

In this derivation, we assumed the contribution of out-of-plane deformation of the inextensible strips to Δh is negligible—this assumption is especially true if the strips are made of materials with high stiffness. For more flexible strips, the Δh will be larger than our estimation due to tilting of the elastomeric body at un-actuated state (Figure 4B shows an example of this tilting), resulting in an under-estimation of δh . We also assume that the deformation of the elastomeric membranes is negligible. This assumption results in an over-estimation of change in volume δV as a function of angle α . Overall, these assumptions results in an overestimation of F , since:

$$F \times \delta h = \eta \times \Delta P \times \delta V \quad (\text{S12})$$

This overestimation, however, doesn't change the qualitative behavior of F . Thus we can still use Equation S11 to study the scaling properties of F , which is the purpose of this derivation.

Comparison of shear-VAMs to conventional pneumatic/hydraulic systems

Mechanical advantage is often present in ordinary pneumatic/hydraulic systems. In a simple system with two cylinders (input cylinder with area A_0 and output cylinder with area A), the mechanical advantage $MA = A/A_0$. There are two key differences between an ordinary

pneumatic system and shear-VAMs. i) The direction of force applied is *perpendicular* to the surface in the ordinary system, while it is *parallel* for shear-VAMs. This difference in direction implies that pressure-based mechanical advantage (Equation 6) can be achieved, which is not possible in the ordinary system. ii) The mechanical advantage for shear-VAMs, in terms of force, can be higher than that for the ordinary systems. Force generated by the ordinary system is $\Delta P \times A$, whereas force generated by shear-VAMs is given by Equation 4. Thus, for the same area A , and pressure differential ΔP , the ratio of the force generated by shear-VAMs to the force generated by ordinary system is $\eta/\tan(\alpha)$. For the current design (with $\eta \approx 35\%$), shear-VAMs can achieve greater mechanical advantage than ordinary systems for $\alpha < 20^\circ$.

Experimental procedure for measuring the relationship between the force of actuation F and the difference of pressure ΔP

Figure S6 shows a schematic diagram for the testing setup that measures relationship between the force of actuation F of shear-VAMs and the difference of pressure ΔP applied across the inside and outside of the shear-VAMs. The bottom end of each shear-VAM is tied to a weight placed on a scale to measure the force of actuation of the shear-VAM. The forces did not exceed the weight and thus, the weight did not lift off the scale. The scale together with the weight acts as a strain gauge to measure the force of actuation F of the shear-VAMs.

We generated the difference of pressure ΔP (i.e. a partial vacuum) by pumping air out of the actuator with a syringe connected to a syringe pump (Harvard Apparatus, PHD 2000). The pressure measurement was performed with a pressure sensor (Honeywell ASDX005D44R) connected to the syringe pump and the pressure transfer line (Figure S6). The voltage signal from the pressure sensor is received through a DAQ (NI USB-6210) and read with National

InstrumentsTM LabVIEW. We slowly extracted air using the syringe pump at a rate of 2 mL/min and recorded the force readings at predetermined values of pressure. We repeated the deflation-inflation cycle seven times to obtain error bars for Figure 3D.

“Shear-VAMs” actuated with positive pressure

We can vary the design of a shear-VAM to generate an actuation stress by inflating the structure rather than deflating it. Figure S8 and Movie S8 show a variant of shear-VAM made of Ecoflex ($E = 43$ kPa) with four beams that is driven by positive pressure to lift a 50 g-weight. This actuator is made based on a shear-VAM but the elastomeric body is glued in the opposite direction to the strips so that the angle between the beams and the strips $\alpha < 90^\circ$ is now $\alpha' = 180^\circ - \alpha > 90^\circ$. This configuration generates a smaller distance of actuation Δh than a normal shear-VAM.

SI References

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- [2] C. L. Dym, I. H. Shames, *Solid Mechanics: A Variational Approach*, Springer, 1973.

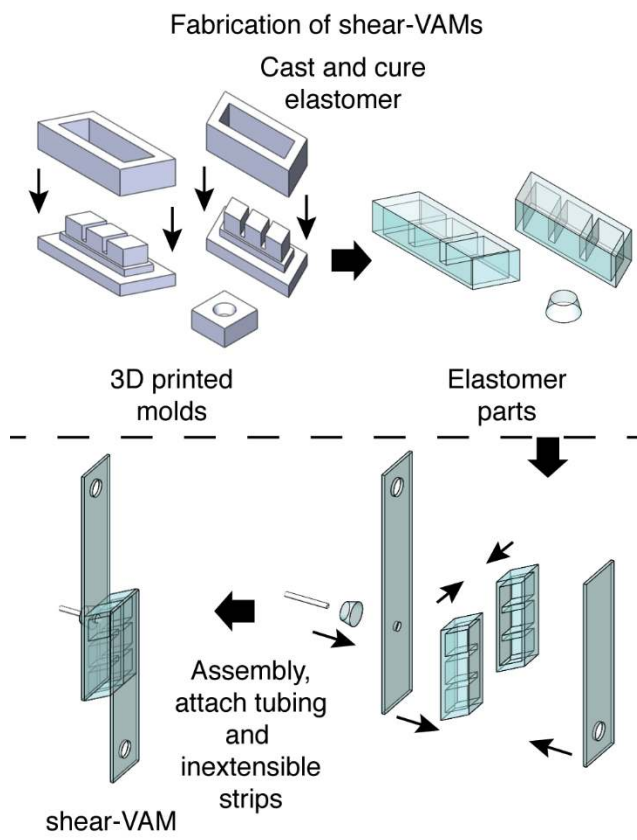
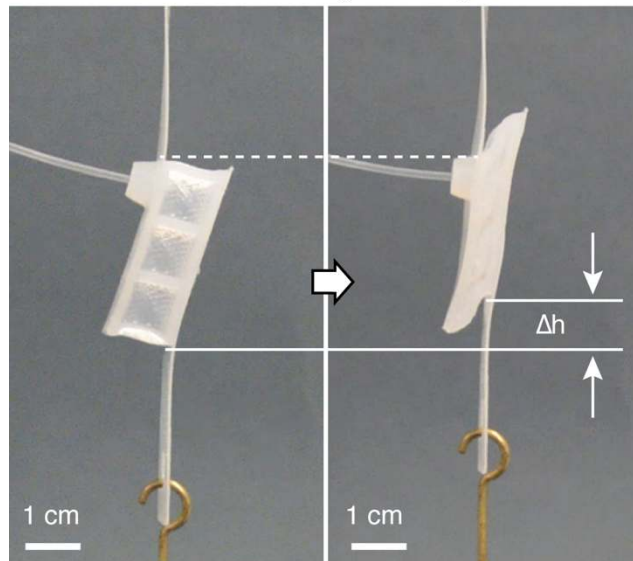


Figure S1. Fabrication of shear-VAMs.

A shear-VAM lifts a 40-gram weight



B shear-VAM lowers a 50-gram weight

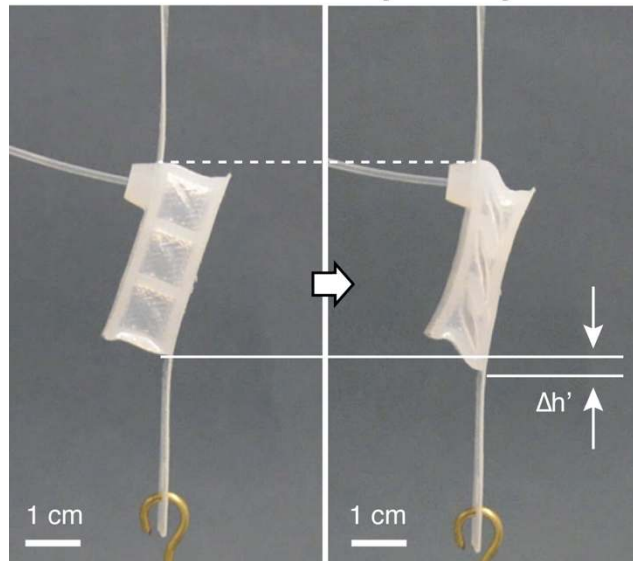


Figure S2. A shear-VAM contracts or extends on actuation depending on the load. A) The shear-VAM *lifts* the weight for a distance of Δh upon actuation while $F < F_{max}$. B) The shear-VAM *lowers* the weight for a distance of $\Delta h'$ upon actuation while $F > F_{max}$. The difference of pressure applied is $\Delta P = 90$ kPa.

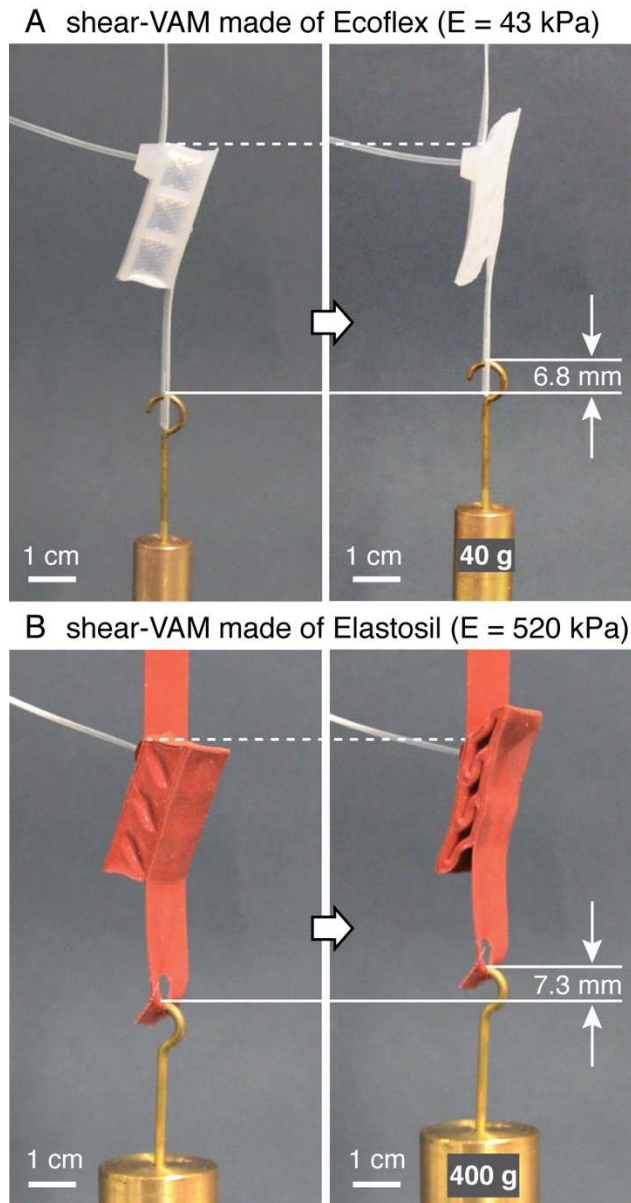


Figure S3. A shear-VAM fabricated from a stiffer elastomer can lift a heavier weight. A) A shear-VAM made of Ecoflex ($E = 43$ kPa) with four beams lifts a 40 g-weight ($\Delta P = 90$ kPa). B) A shear-VAM of the same geometry but made of Elastosil ($E = 520$ kPa) lifts a 400 g-weight ($\Delta P = 90$ kPa).

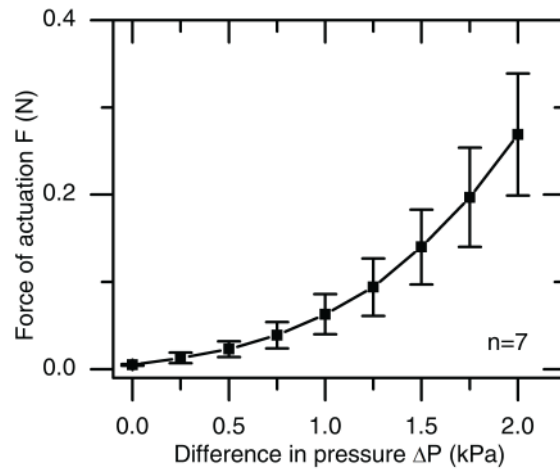
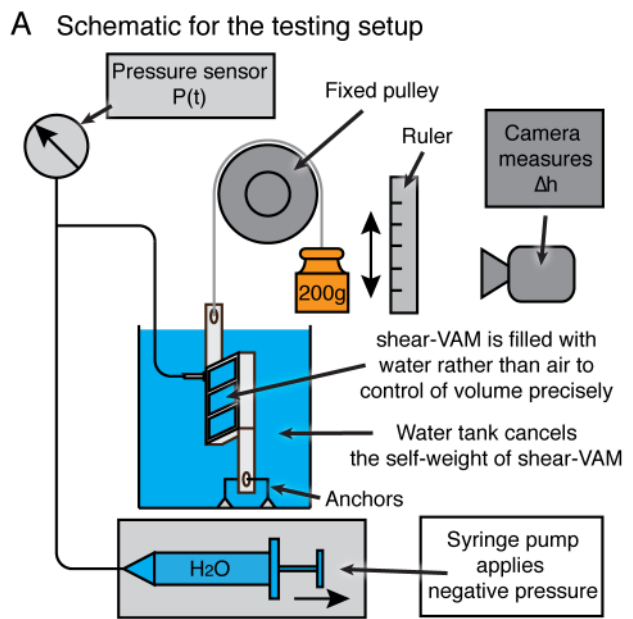


Figure S4. The force of actuation F of seven different shear-VAMs of length $L=32 \text{ mm}$ (connected to a fixed strain gauge) vs. the difference of pressure ΔP (in kPa) applied across the inside and outside of the shear-VAMs.



B P-V curves of the shear-VAM lifting a 200 g weight

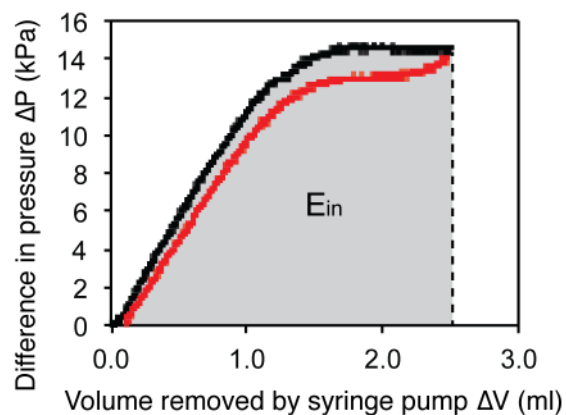


Figure S5. Experiment used to determine the thermodynamic efficiency of operation of a shear-VAM. **(A)** Schematic describing the setup used for testing. **(B)** P-V curves of a shear-VAM fabricated in Elastosil lifting a 500 g weight. The actuation curve is marked in black, and the return curve is marked in red. The shaded area E_{in} represents the fluidic energy input via the syringe pump.

Schematics for the static F vs. ΔP testing setup

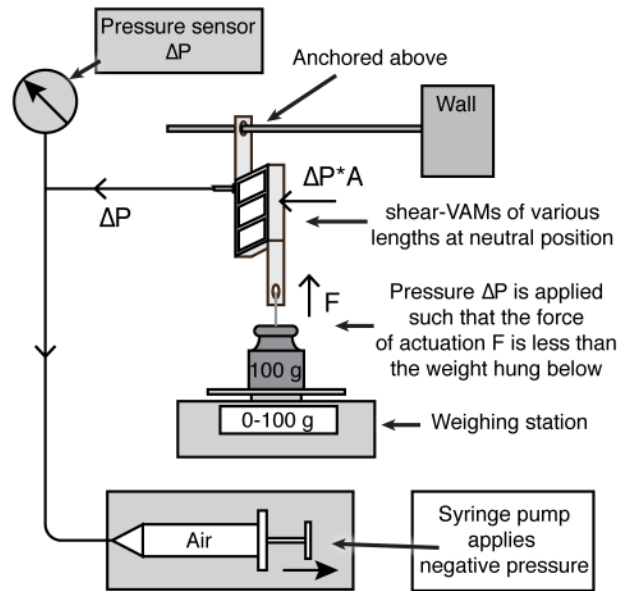


Figure S6. Schematic diagram for the testing setup that measures relationship between the force of actuation F of shear-VAMs and the difference of pressure ΔP applied across the inside and outside of the shear-VAMs. The bottom end of the shear-VAM is tied to a weight placed on a scale to measure the force of actuation of the shear-VAM. The forces did not exceed the weight and thus, the weight did not lift off the scale. The scale together with the weight acts as a strain gauge to measure the force of actuation F of the shear-VAMs.

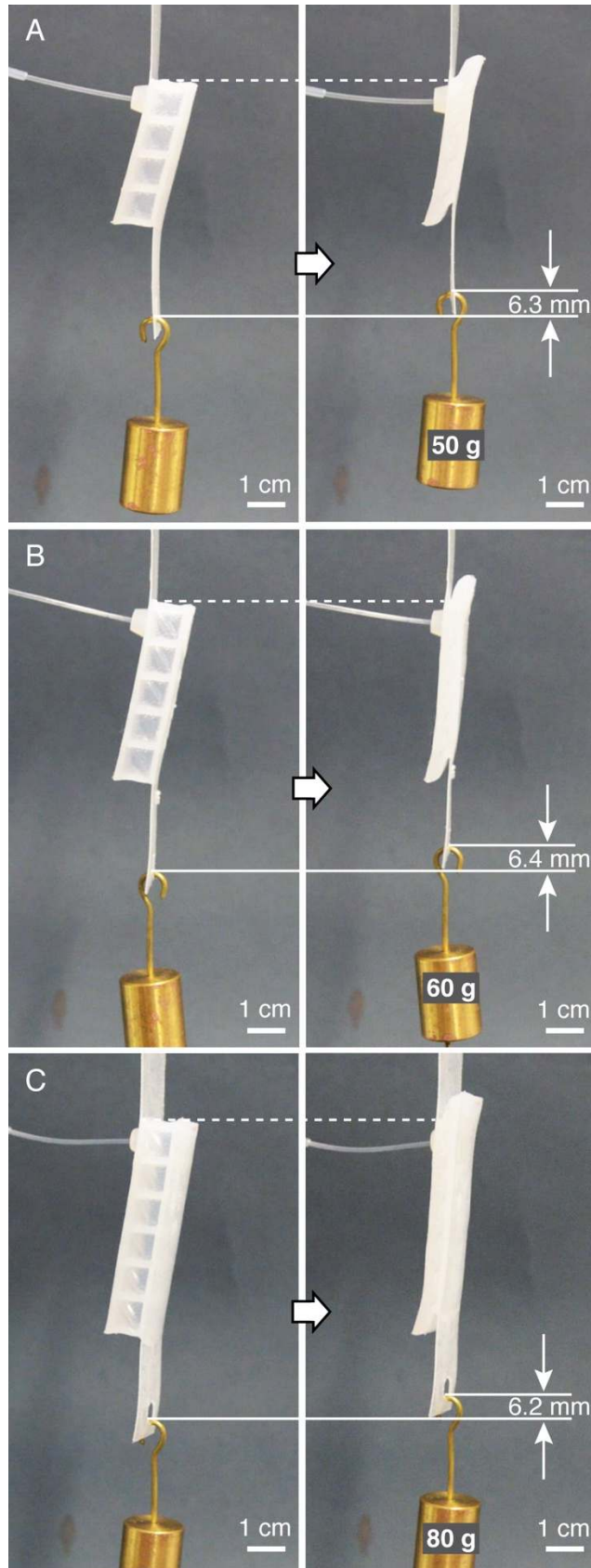


Figure S7. The maximum force of actuation of a shear-VAM F_{max} (in Newtons) increases with its length L . A series of shear-VAMs of different lengths is tested to lift an increasing series of weights in 10 g-intervals (i.e. 10 g, 20 g, 30 g, etc.). The figure shows each of them lifting their highest weight in the series. A) A shear-VAM made of Ecoflex ($E = 43$ kPa) with five beams and length $L=42$ mm lifts a 50 g-weight. B) A shear-VAM made of the same material but with six beams and length $L=52$ mm lifts a 60 g-weight. C) A shear-VAM of the made of the same material but with seven beams and length $L=62$ mm lifts an 80 g-weight.

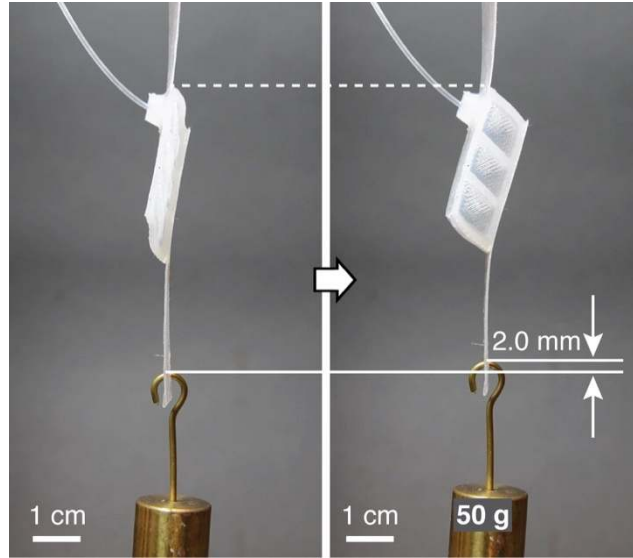


Figure S8. A variant of shear-VAM made of Ecoflex ($E = 43$ kPa) with four beams and length $L=32$ mm that is driven by positive pressure lifts a 50-g weight. The distance of actuation Δh is less than that of a normal shear-VAM operated by vacuum.

Supporting Movie Legends

Movie S1. A shear-VAM fabricated in Ecoflex ($E = 43$ kPa) performing an actuation. The inside of the membrane is painted with a black marker to reveal the shape of the chamber more clearly. A black outline is visible after the actuator contracts.

Movie S2. When a shear-VAM is actuated, it could either perform a contraction or elongation depending on the load.

Movie S3. A shear-VAM fabricated in Ecoflex ($E = 43$ kPa) lifting a 40-g weight.

Movie S4. A shear-VAM fabricated in Elastosil ($E = 520$ kPa) lifting a 400-g weight.

Movie S5. Two shear-VAMs fabricated in Ecoflex ($E = 43$ kPa) in a parallel configuration lifting about twice as much as one such shear-VAM, but the same distance.

Movie S6. Two shear-VAMs fabricated in Ecoflex ($E = 43$ kPa) in series configuration lifting about the same weight as one such shear-VAM, but about twice the distance.

Movie S7. Two shear-VAMs in an agonist-antagonist configuration are able to drive a paddle used for a toy boat.

Movie S8. A paddle driven by two shear-VAMs in an agonist-antagonist configuration is able to move a toy boat in a water tank.

Movie S9. A shear-VAM where the beams are tilted in the opposite direction can be actuated with positive pressure.