

DOI: 10.1002/ (No. smll.201802931)

Article type: **(Full Paper)**

Bio-inspired 3D Printed Locomotion Devices Based on Anisotropic Friction

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This manuscript is in memory of Prof. Jacob Israelachvili, who has devoted his career to the study of various interactions between molecules and surfaces with the aim to determine the theoretical basis and factors that contribute to friction, fluidity, adhesion, and repulsion in a wide range of applications.

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Keywords: *Anisotropic friction; locomotion; wheat own; bio-inspired; microbots*

Abstract: Anisotropic friction plays a key role in natural systems, particularly for realizing the purpose of locomotion and strong attachment for the survival of organisms. Of particular interest, here, is the observation that the friction anisotropy has been promoted numerous times by Nature, for example by wild wheat awn for its targeted and successful seed anchorage and dispersal. Such feature has, however, not been fully exploited in man-made systems, such as microbots, due to technical limitations and lack of full understanding of the mechanisms. To unravel the complex dynamics occurring in the sliding interaction between anisotropic micro-structured surfaces, we first systematically investigate the friction induced by the asymmetric plant micro-structures. Inspired by this, we then prepare anisotropic polymer micro-actuators with 3D printed micro-relieves. By varying geometric parameters, we study how the micro-actuators generate strong friction anisotropy and controllable motion in cylindrical tubes. Advanced theoretical models are proposed to understand and predict the dynamics behaviour of these synthetic systems and to shed light on the mechanisms governing their behaviour. Finally, we develop a simple microbot prototype and successfully realize cargo transportation functions. This research provides both in-depth understanding of anisotropic friction in nature and new avenues for developing intelligent actuators and microbots.

1. Introduction

Friction is everywhere in nature and is an essential factor for the evolution of highly-specialised surfaces in organisms having specific interactions with environment.^[1] A common phenomenon encountered in many biological systems is frictional anisotropy, which is based on the directional asymmetry in the friction response to sliding. It results from specific structural features at the contact surfaces. Numerous anisotropic surfaces are encountered both in biological,^[2] and non-biological systems,^[3] and the features responsible for frictional anisotropy range from the molecular scale up to the macro scale. In nature, such typical structural features are widespread both in animals and plants,^[4] providing evolutionary optimized interactions (against the specific substrate) for the purpose of locomotion and transportation. For example, the snake utilizes this natural feature to execute crawling locomotion (such as serpentine, sidewinding and concertina locomotion),^[5] whereas the gecko's agile climbing on ceilings is linked to the directional friction/adhesion characteristics originating from the feet hierarchical microstructuring⁴. Interestingly, whilst many plants species show adaptations to attachment against various substrate surfaces due to anisotropic surface microstructures,^[6] the latter are not uncommon in the human body too; muscles,^[7] articular cartilage^[8] and hair^[9] are all characterized by anisotropic features, which endow them special functionalities. Furthermore, anisotropic structural features are also extremely common in man-made materials and systems, such as solid surfaces obtained from cleaving crystals in certain orientations,^[10] 1D or 2D materials surfaces,^[11] synthesized soft matter^[12] and bio-inspired materials surfaces.^[13] Over the past 20 years, research reports on investigations of anisotropic surface features and their effect on friction have become an important focus of the scientific community.^{[10] [14]}

Friction anisotropy in both biological or non-biological systems has been reported to be generated by non-symmetric and/or non-homogeneous surface features. To date, the link between

inhomogeneous surface topography and friction has been extensively studied both in natural and synthetic materials,^[15] and a common purpose for these investigations is to develop novel bio-inspired actuation devices,^[16] because natural friction-anisotropy-based systems show typical structures and topographies deployed to produce different actuation mechanisms. Theoretical and numerical models to study these phenomena have also been developed and have investigated key parametric dependences of anisotropic friction efficiency both in biological^[17] and artificial systems.^[18] For example, the anisotropic movement responsible for both attachment and detachment of gecko's feet has been successfully applied to the preparation of climbing robots,^[19] and the study of anisotropic friction locomotion mechanism of snakes has enabled the development of snake-like robots.^[20]

Although current research reports indicate the concrete prospects to apply anisotropic friction for the development of new devices, it still remains an open question, whether scientists can develop artificial actuation systems and microbots with precise scale-matching as those encountered in nature. We observe that a microbot equipped with friction-based locomotion could be designed to provide both high manoeuvrability and accurate targeting in confined three dimensional spaces, as well as stress compatibility with living soft matter due to the capability to limit interface stresses.^[21] Moreover, with an increasing requirement for the implementation of microbots for robotic assembly, drug delivery, lab-on-chip and sensing applications, microsurgery and cancer treatment,^[22] there is now a practical need to build artificial prototypes that can serve as models for understanding and predicting the dynamics of future industrial microactuators. As an example, one of the most challenging task for medical microbots is controlling their motion in complex biological environments;^[22] in this respect, the physical propulsion mode,^[23] together with anisotropic friction-based locomotion might be a promising enabling technology for their development.

Here, we report an in-depth theoretical and experimental study leading to new insight of asymmetric features in plants and their possible role in the dynamics of locomotion driven by friction anisotropy. Starting from the observation of anisotropic friction in nature, we first looked at the most promising features adopted in plants and animals to promote anisotropic responses in interacting surfaces. Although it has been shown that, for example, insects adopt anisotropic features for oviposition, skin penetration, and to modulate adhesion and control frictional response,^[24] the structures resulting from evolution of these species are usually not regular and makes it difficult to systematically study their behavior. By contrast, the investigation of some plants and their regular asymmetric features can provide important insight and inspiration to enable synthetic solutions. In particular, we fabricated three-dimensional (3D) anisotropic polymer constructs using a UV-irradiated 3D printing technology. The printed polymer constructs are characterized by two dimensional arrays of a micro-structure unit, the thorn, with controllable thorn size, tilt angles and area density, exhibiting friction anisotropy when slid against a poly(dimethylsiloxane) (PDMS) counter-surface. One of the most interesting aspects of this study is the demonstration that the artificial 3D anisotropic constructs can generate accurate directional propulsion in rubber tubes subjected to cyclic stretching, the latter being also successfully employed for the execution of cargo delivery. Theoretical models are also proposed, whose results are in good agreement with the cargo and locomotion experiments, which can be implemented as physical models for the development of microbot control algorithms; they offer a full explanation for the mechanisms responsible for the motion induced by the anisotropic surface features, hence providing analytical tools for their design and paving the new avenues for the truly development of friction-induced microbot devices.

2. Results and Discussion

2.1. Anisotropic micro-structural geometries: Inspiration from natural plants

Literature shows that anisotropic micro-structural geometries are the key surface feature in numerous of plants species: the wheat awn (**Figure 1a**), the stem of *Humulus japonicus* (**Figure 1e**), the pine leaves (Figure S1, Supporting Information) and the stems of wheatgrass (Figure S2, Supporting Information). The anisotropic friction, induced by the presence of oriented thorns on plants surfaces, can be easily experienced by touching these features with finger pads. In a prototype friction experiment, the wheat awn was tightly fixed onto a testing platform and subjected to relative sliding motion against a hemispherical PDMS countersurface. Typical curves of the friction force vs. time are presented in Figure 1b, Figure 1f, Figure S3a and Figure S3b in Supporting Information for the wheat awn (*Triticum aestivum*), stems of *Humulus japonicus*, pine leaves (*Pinus bungeana* Zucc.) and wheatgrass (*Agropyron desertorum* (Fisch.) Schult), respectively. Black curves show the case when the sliding direction follows the orientation of thorns (*positive sliding direction*, see also the schematic insets in Figure 1), while the green curves show the tangential forces measured when the sliding occurs in the opposite direction (*negative sliding direction*). Obvious differences are observed in the frictional response of different natural systems, whereas all the results show a typical force fluctuation, indicating that the tribopair is passing over the thorn at the friction discontinuity. The slope of the friction curve (red dashed lines in Figure 1f) indicates the rate of elastic energy which is stored during the elastic slip just before the occurrence of the frictional instability (and the corresponding elastic energy release). Such energy storage is typically lower for the positive sliding direction with respect to the negative direction, as could be directly related to the local, relative, thorn angular alignment with respect to the PDMS surface. Among the investigated natural systems, it was found that the wheat awn

showed the largest friction anisotropy. As a consequence, in this research, the wheat awn was chosen as the candidate for further systematic investigation.

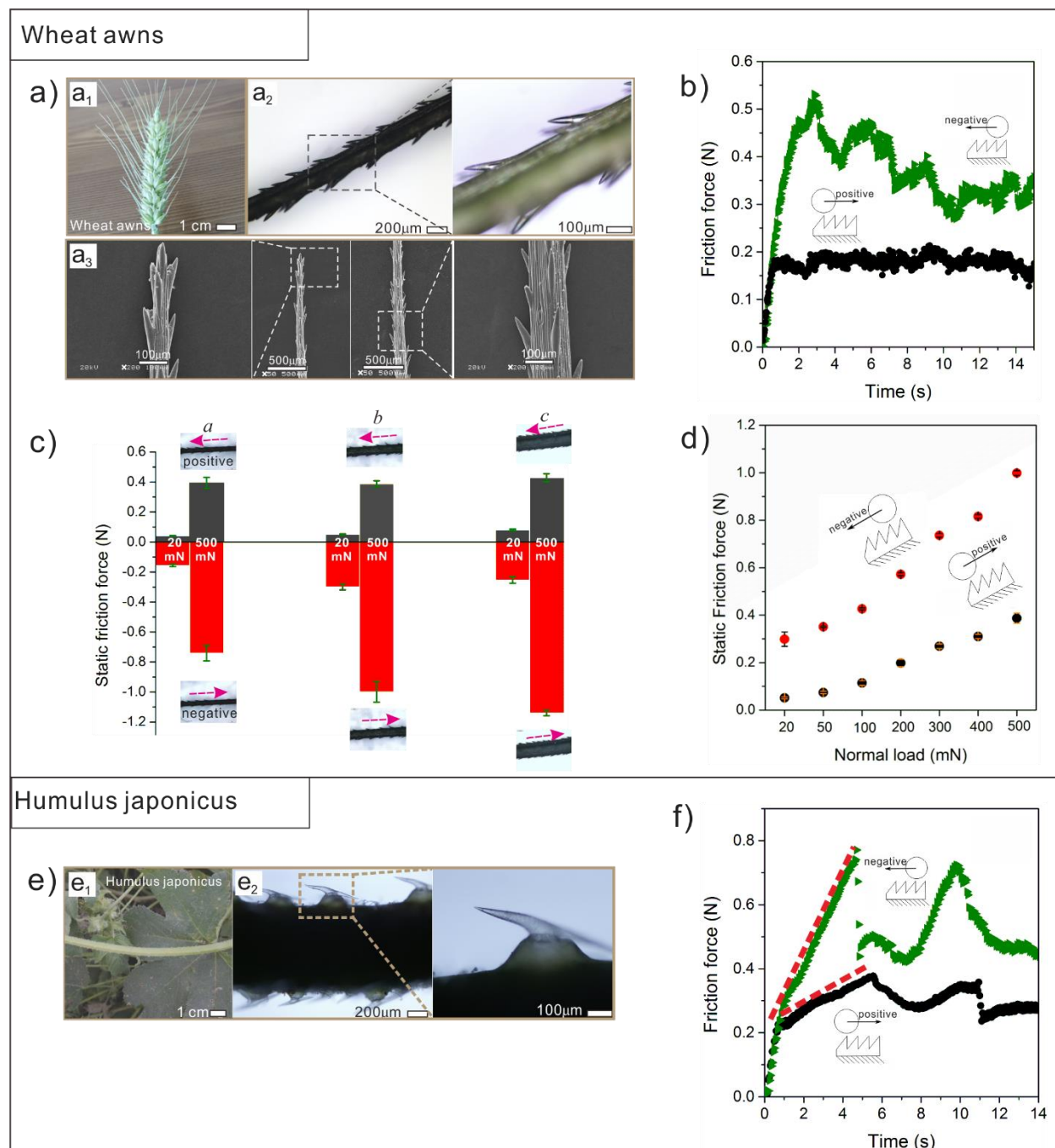


Figure 1. Geometrical and frictional properties of the anisotropic plant surfaces analyzed in this study. The most interesting micro-structural features of the plants can be identified by using optical and scanning electron microscopy (SEM). In the figure, the top and bottom panels are for the wheat awn and *Humulus japonicus* stem, respectively. (a): (a_1 - a_2) Optical microscopy and (a_3) SEM images of the wheat awn, at different positions along the awn ideal axis of symmetry; (b) Measured friction force as a function of sliding time, under load of 200 mN and sliding velocity

of 15 mm min^{-1} . (c) Breakloose (or static) friction force obtained by three different parts of the same awn (thin part: close to the seed; middle part and thick part: close to the top, see a₃) in positive and negative sliding directions, and for different loads. The sliding velocity is set to 15 mm min^{-1} . (d) Break loose friction force as a function of the normal load, in positive and negative sliding directions. The insets show the direction of the motion of the ball with respect to the awn. (e): (e₁-e₂) Optical microscopy of the *Humulus japonicus*. (f) Measured friction force as a function of sliding time for stem of *Humulus japonicus*, under load of 200 mN and sliding velocity of 15 mm min^{-1} .

Furthermore, we found that the friction force of wheat awn exhibits large differences, both in the positive and negative sliding direction, when different parts of the plant are probed. Indeed, when measuring over the thin part (close to the seed), the middle part and the thick part (close to the top), friction measurements show dissimilar results, which do not depend on the loading conditions (Figure 1c). These differences originate from the variable microstructure area density, resulting in a variable contact stiffness of the awn/PDMS interface.

Typical load-dependencies of the static or break-loose friction force are presented in Figure 1d. The break-loose force exhibits a nearly-linear increase with load both at the positive and negative sliding directions. Here the maximum load is limited to 500 mN, to avoid the peeling off and rupture of the thorns by the counterpart surface (observed above this load).

Inspired by the wheat awn microgeometry, we fabricated artificial polymer constructs with a 3D printing technique available through a commercial printer (Figure S4, Supporting Information). In Figure 2a, we summarize the geometrical characteristics of the repeated printed microstructure. In particular, in Figure 2a₂ we report the axial section of the artificial thorn: θ_1 and θ_2 correspond the tilt angles, L_0 is the lattice axial constant, R_i (R_o) is the awn internal (external) radius. θ_3 is the thorn breath angle. In this study, R_o/R_i as well as θ_3 are kept constant, whereas R_i , L_0 and (θ_1, θ_2) are varied during the printing process, see Figure 2b₃, Figure 2b₄, Figure 2b₅ and Figure S5 in Supporting Information respectively. The virtual prototype is shown in Figure 2b₁, together with the corresponding printed device in Figure 2b₂, both characterized by four

equally-spaced angular thorns. The nano-mechanical characterization of the printed architecture is reported in Figure S6 in Supporting Information, with a measured elastic modulus of 3.15 GPa.

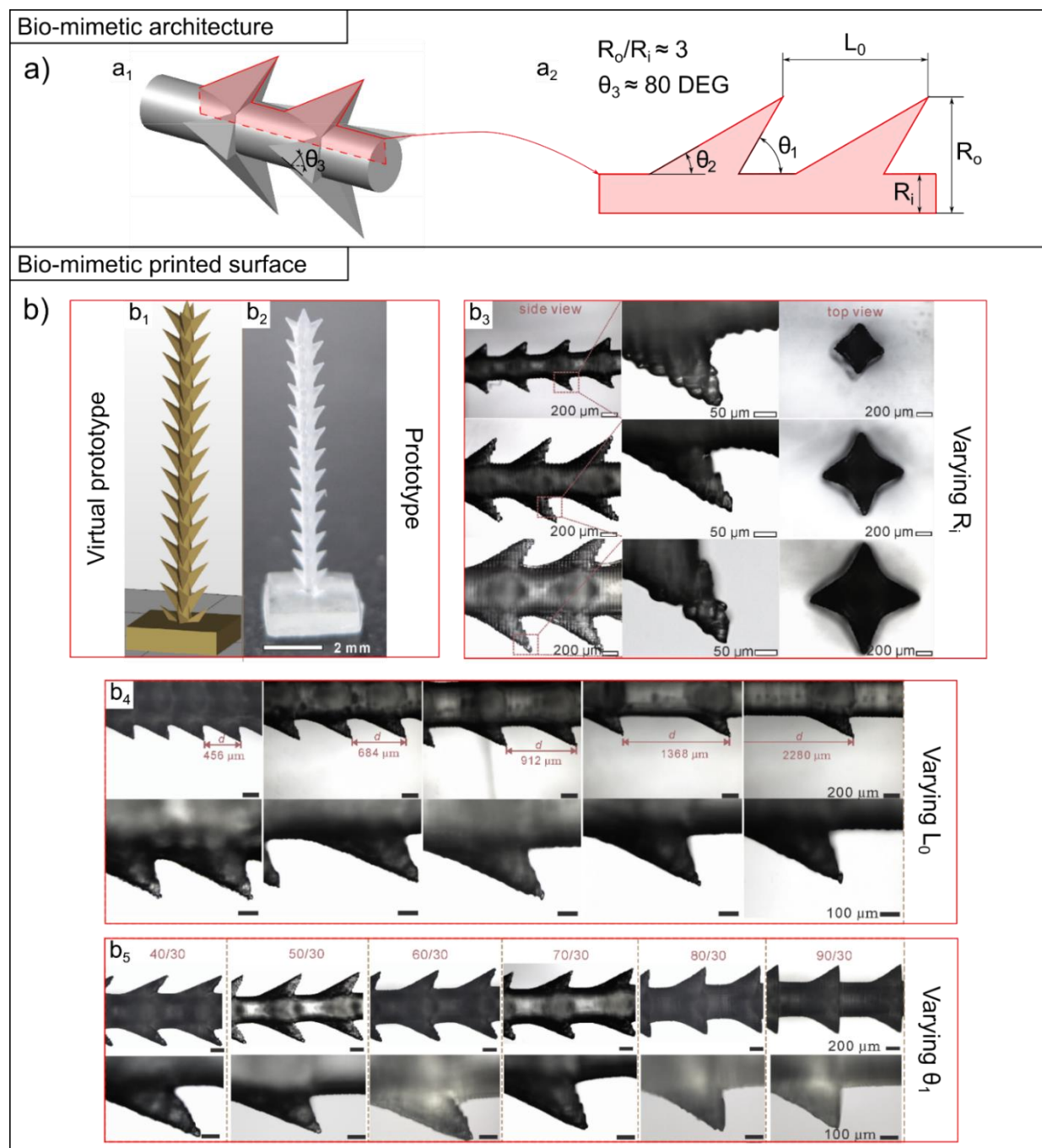


Figure 2. Microstructural characteristics of the bio-inspired surface. (a): Representative unit of the microstructured virtual prototype (a₁) and related axial section (a₂), with indication of the microstructure parameters varied in this study. (b): Virtual prototype (b₁) and photo (b₂) of the corresponding printed replica. (b₃) to (b₅) Optical micrographs of the printed micro-geometries at varying internal radius R_i , axial length density ($\propto L_0^{-1}$) and pitch angle θ_1 , respectively. The printing resolution is $20 \pm 10 \mu\text{m}$.

2.2 Locomotion dynamics model

In this work, the anisotropic rod locomotion and payload operation is obtained by inserting the micro-printed awn in a cylindrical stretchable tube, the latter corresponding to the actuation driver. Thus, we have built the locomotion dynamics model in the framework of continuum mechanics (see Section S2 for the theory in Supporting information), assuming a homogeneously rough cylinder of length L_r and nominal radius R in axial symmetric contact with the inner side of an ideally smooth cylindrical crown, with inner and outer radius, respectively, R_i and R_o . In particular, the roughness has been assumed homogeneous as if the representative size of the thorn, say h_t , were much smaller than the contact radius R . Such separation of length scales, a common assumption in the roughness contact mechanics research,^[25] allows indeed to separate the micro-contact dynamics from the rod/tube macro-contact dynamics, in the sense that the micro-contact dynamics is effectively integrated out in term of effective rod/tube friction coefficients. Furthermore, in this work the cylindrical tube corresponds to the actuation driver, and, as such, the model is developed considering the driving constraint applied to the tube. However, we stress that the resulting contact mechanisms (see below) are expected to be qualitatively the same to case when the awn is designed to be the actuation driver, the latter case being the subject of a follow-up article.

In **Figure 3** we show the dynamics results for the case of a bio-mimetic actuator characterized by a neat advancing ratcheting locomotion, under relatively small tube stretching excitation frequency (negligible inertia effects). We stress that whilst the dynamics key-features are of general applicability, the specific results are obtained by adopting an awn/tube contact interface with maximum and minimum effective friction coefficient $f_{\max} = 0.06$ and $f_{\min} = 0.02$, respectively, tube internal and external diameters $D_i (= 2R_i) = 1.68$ mm and $D_o (= 2R_o) = 3.68$ mm, rod length $L_r = 50$ mm and actuation stretch $\varepsilon_0 = 0.073$. We also use a geometric

tube/awn misfit $u_0 = 50\mu\text{m}$. In particular, the top-right graph shows the dimensionless rod displacement u_r/L_r as a function of the actual stretch ratio $\varepsilon_p/\varepsilon_0$. The grey curve reports the awn displacement during the first tube stretching motion (up to (B) in the graph), and it is followed by black and red curves (from (B) to (I) in the graph). The latter correspond to the displacement law during a single actuation cycle, and are given by a tube release (black lines) and stretching stage (red lines). Successive actuation cycles are simply a repetition of the (B)-(I) curve, and as such they are not shown here for simplicity. We remark that the physical properties of the system at state (I) is equal to the state (B), unless for the shifted rod position by Δx_r .

The encircled sequence of labels, reported as alphabetical list, provides an easy-on-the-eye timeline of the actuation process; furthermore, each encircled label corresponds to a specific 3D (three-dimensional) contact map reported in the figure panels. We remark that the generic contact map is a **true-scaled representation of numerical results**. In the initial stage panel, corresponding to the first tube stretching motion, the rod sliding motion is given by the solid grey line in the graph. Note that the initial injection of the awn in the tube is not modelled, instead we assume the contact to begin with a frictionless state (as if the rod were inserted in the tube without the occurrence of wall friction). However, we observe that the exact knowledge of the initial contact condition is not relevant to the stationary ratcheting dynamics, since the motion reaches a well-defined periodic dynamic loop already from the first actuation cycle. In other words, the actuator **shows no stress- memory behavior after the initial actuation loop**.

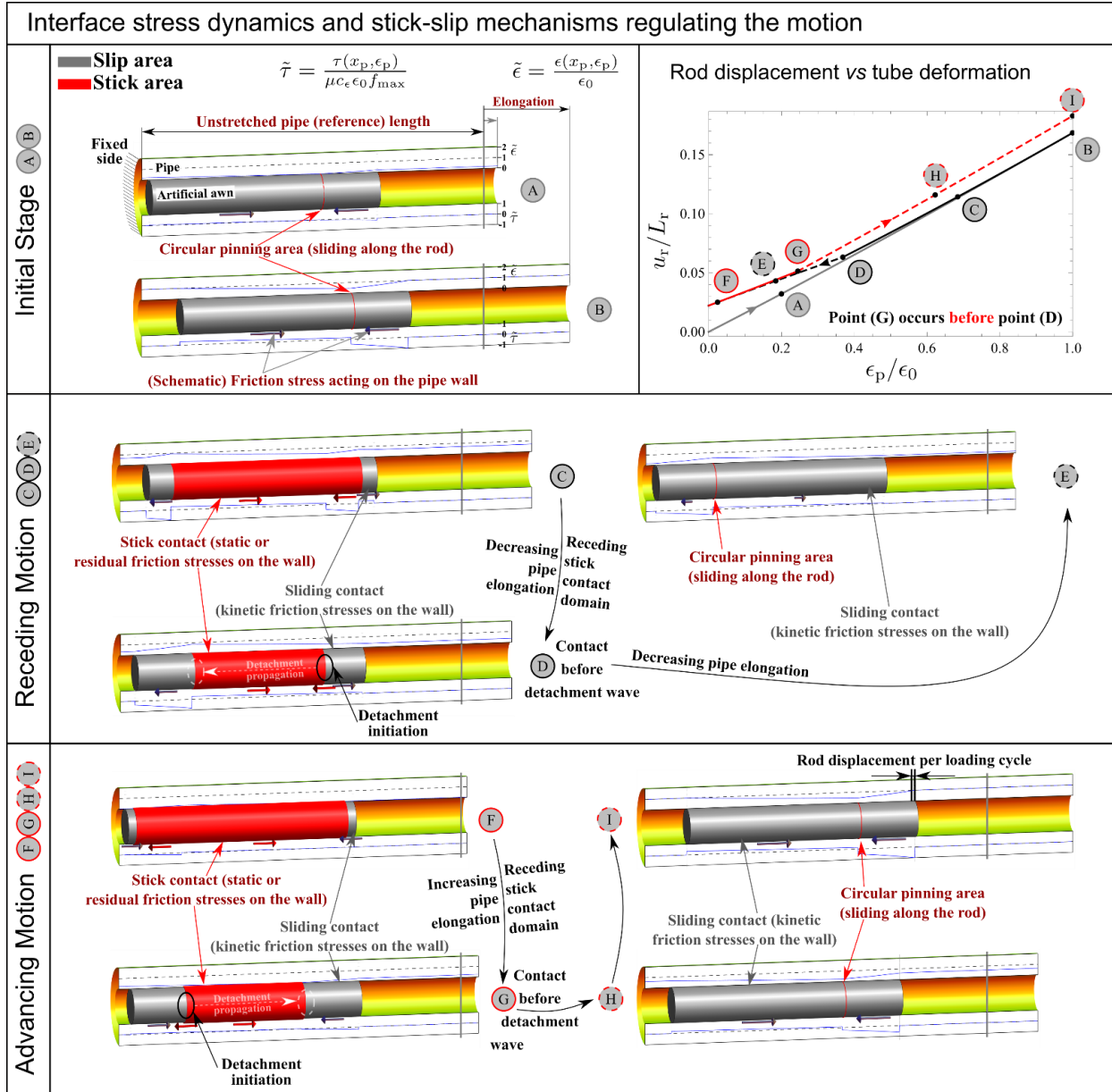


Figure 3. Theoretical results. The interface stress dynamics and the related stick-slip motion occurring during one actuation cycle at small excitation frequencies. (Top-right panel): Dimensionless rod displacement u_r/L_r as a function of the normalized tube stretch ϵ_p/ϵ_0 , where ϵ_0 is the maximum tube actuation deformation. The gray solid line represents the micro-actuator displacement during the initial stretching stage (initial stage panel), ending with the state (B). The black and red lines constitute instead the generic actuation cycle, where the black is for the tube release stage (receding motion panel) and the red for the stretch elongation stage (advancing motion panel). The latter stage ends with the state (I), which is characterized by the same physical parameters of (B) but a different axial position of the bio-mimetic awn. Furthermore, the black and red solid lines indicate the occurrence of stick-slip stage in the printed awn-tube contact, whereas the dashed lines show that full sliding contact is occurring (apart from a circle of stick contact). **(Top-left and remaining panels):** field numerical results of printed-awn tube contact dynamics. The initial stage (top left panel) is followed by the simulation results describing the generic actuation cycle (receding and advancing motion panels). On the printed awn, the red areas indicate

the occurrence of stick contact and residual frictional stresses, whereas the gray domains are for the sliding interaction. On the tube, we have adapted the normalized interface (frictional) shear stress $\tilde{\tau}$ and axial deformation $\tilde{\varepsilon}$ as a function of the contact position. The encircled letters link the generic simulation result to the rod displacement curve on the top-right panel.

Thus, by increasing the tube stretch from 0 to, say, $\varepsilon_{(A)}$, the rod undergoes a relative (to the tube) sliding motion along the whole cylindrical surface, apart from a red annular region (of infinitesimal axial size) where a stick or pinning contact occurs, see the 3D contact map in (A). At both sides of such pinning circle, the rod relative sliding speeds have opposite sign, as imposed by the global mechanical equilibrium of the rod. In (A) we also schematically report, with solid arrows, the frictional shear stress acting on the tube wall (showing opposite sign), as well as the local tube stretch $\tilde{\varepsilon}$ and shear stress $\tilde{\tau}$ curves as a function of the axial position ($\tilde{\varepsilon}$ and $\tilde{\tau}$ defined in the figure legend). When the tube stretch is increased up to $\varepsilon_{(B)} = \varepsilon_0$ (B), we observe that the wall frictional stresses available on the tube determine a stretch softening in the fraction of tube under interaction with the rod. In particular, the minimum stretch is located in correspondence of the pinning circle, the latter also characterized by a discontinuity in the frictional stresses.

At point (B), stretch reversal is imposed to the tube. During reversal, the interaction changes from full sliding (apart from the pinning circle) to full stick contact, characterized by residual frictional interface stresses. By further stretch release (see the receding motion panel in Figure 3) up to, e.g., point (C), the contact interface is characterized by an inner domain with residual frictional stresses, and by the outer area with sliding frictional stresses. Interestingly, local maxima in the stretch values occur now in correspondence of the stick-slip borders. **By further decreasing the external stretch**, such borders become **closer to each other** until a limiting stick area is reached, see (D); **this is characterized by the disappearance of one of the two local stretch maxima of above**. At such point, an infinitesimal further decrease in the external stretch will induce an interface

detachment wave, leading to the downsizing of the stick area from a finite extension (see (D)) up to an infinitesimal annular region (see e.g. (E)).

When further relaxing the externally applied stretch down to zero, a new stick contact configuration is reached, characterized by residual frictional stresses over the whole contact interface. Such a point (not shown in the figure, for simplicity) is thus a reversal configuration, as before. Increasing now the external stretch (see the advancing motion panel in Figure 3) determines the new initial stick contact interface, as well as the establishment of a sliding interaction in the outer contact domain. However, differently from the receding motion, the advancing motion shows local minima in the stretch values in correspondence of the stick-slip borders, see (F). By further increasing the external stretch, the stick domain is reduced until a limiting area is reached, see (G), characterized by the disappearance of one of the local stretch minima. Afterwards, an infinitesimal further increase in the external stretch will induce the formation of a detachment wave, propagating in the opposite direction with respect to the receding case. Such wave will re-establish the contact to occur under full sliding, apart in an infinitesimal annular stick region, see e.g. (H) and (I). This infinitesimal annular pinning region, which exists in the sliding regime during the actuation stretch, is typically located at a different rod axial position with respect to the case occurring during the actuation release. Therefore, the neat ratchet motion is physically the result of these different rod/tube clamping positions, which allow the rod to have a neat advancing motion per cycle.

Note in the dimensionless rod displacement (u_r/L_r) graph, that the receding motion is depicted with black line (solid and dashed), whereas the advancing motion with red line (solid and dashed). Both stages show an initial stick regime (solid line) followed by a slip regime (dashed line). We observe that the sliding stage in the receding motion (black dashed line) and the stick stage in the advancing motion (red solid line) are almost parallel straight curves. Similar consideration applies for the stick stage in the receding motion (black solid line) and the sliding

stage in the advancing motion (red dashed line). As a consequence, the relative extension, along the stretch axis, of such regimes determines the total (neat) advancing or even (neat) receding capability of the actuator. Indeed, if point (G-advancing detachment wave) occurs for external stretches lower than for point (D-receding detachment wave), the rod would show a neat advancing (or positive) displacement per actuation cycle, as shown in the contact image (I). The opposite case, thus for point (G-advancing detachment wave) occurring for external stretches larger than for point (D-receding detachment wave), is shown in Figure S15 in Supporting Information.

When the actuation frequency increases, inertia effects cannot be neglected anymore and in **Figure 4** (top-left panel) we show the numerical results and, in particular, we provide the ratcheting amplification factor $[\Delta x_r]_{N\text{LinM}}/[\Delta x_r]_{\text{LinM}}$ as a function of the dimensionless square frequency number $\tilde{m}$, at varying system parameters, see Section 2 in Supporting information for details. Here the implemented actuation stretch law is triangular. The ratcheting amplification factor provides the ratio between the cumulated displacements calculated by including ($[\Delta x_r]_{N\text{LinM}}$) or not ($[\Delta x_r]_{\text{LinM}}$) the inertia effects into the locomotion dynamics. The black (red) [red dashed] curve is calculated for the same parameters of Figure 6a₂, but for a minimum and maximum friction coefficient 0.1 and 1.0 (2.0) [0.2], respectively, and for 20 tube actuation cycles. The ratcheting amplification factor shows the occurrence of three typical features, in particular a linear regime for $\tilde{m} \rightarrow 0$, which is characteristic of the inertialess response. Secondly, a sublinear regime with anti-resonance (markedly for small friction values) appears for increasing dimensionless square frequencies $\tilde{m}$. By further increasing $\tilde{m}$, the resonance and post-resonance inertial regimes occur. Among the physical parameters affecting the dynamics, the friction coefficient anisotropy has the largest effect on the ratcheting motion: a ten-fold variation of the maximum friction value leads to a ten-fold variation of the resonance amplification factor and its frequency as well.

In the lower panel, three ratcheting trajectories (in dimensionless units) are reported in the phase diagram and correspond to the three system actuation regimes A) linear, B) resonance, and C) inertial regime, see insets with encircled letters. Furthermore, in the phase diagrams, the solid lines are representative of the motion occurring in the stick-sliding stages, whereas the dotted lines are for the ballistic stage. The latter is defined as the stage where the rod has no stick contact with the tube, i.e. full sliding occurs at the interface. Interestingly, we note that, during the linear regime, the ratcheting occurs with the rod advancing (receding) motions which match the tube elongation (release) stages. As such, the rod speed is constant and positive during tube elongation, and it reverses to a negative but constant value during tube release (apart from the time instants where motion inversions occur, see dotted lines in Figure 4A)). This is due to the occurrence of the stick-sliding regime (see solid lines in Figure 3) over the whole rod motion, and leading to a rod velocity which matches the tube velocity in the stick contact section (note that the tube actuation velocity driver adopted in this study is triangular-based, leading to time-constant tube actuation velocities).

When the actuation frequency is increased, see Figure 4B, the ratcheting motion shows again a stick-sliding regime but only for the very first actuation cycles (inset purple in Figure 4B), followed by a ballistic motion, where the rod is always in sliding contact with the tube.

Furthermore, the rod trajectory is never receding, apart from the very first few cycles. Finally, when the actuation frequency is further increased, see Figure 4C, the ratcheting occurs in the inertial regime from the beginning, with ballistic trajectory. In this latter case, the dynamics of the anisotropic rod is no further characterized by ratcheting, since the rod velocity never shows negative values (thus receding motion) in the phase diagram.

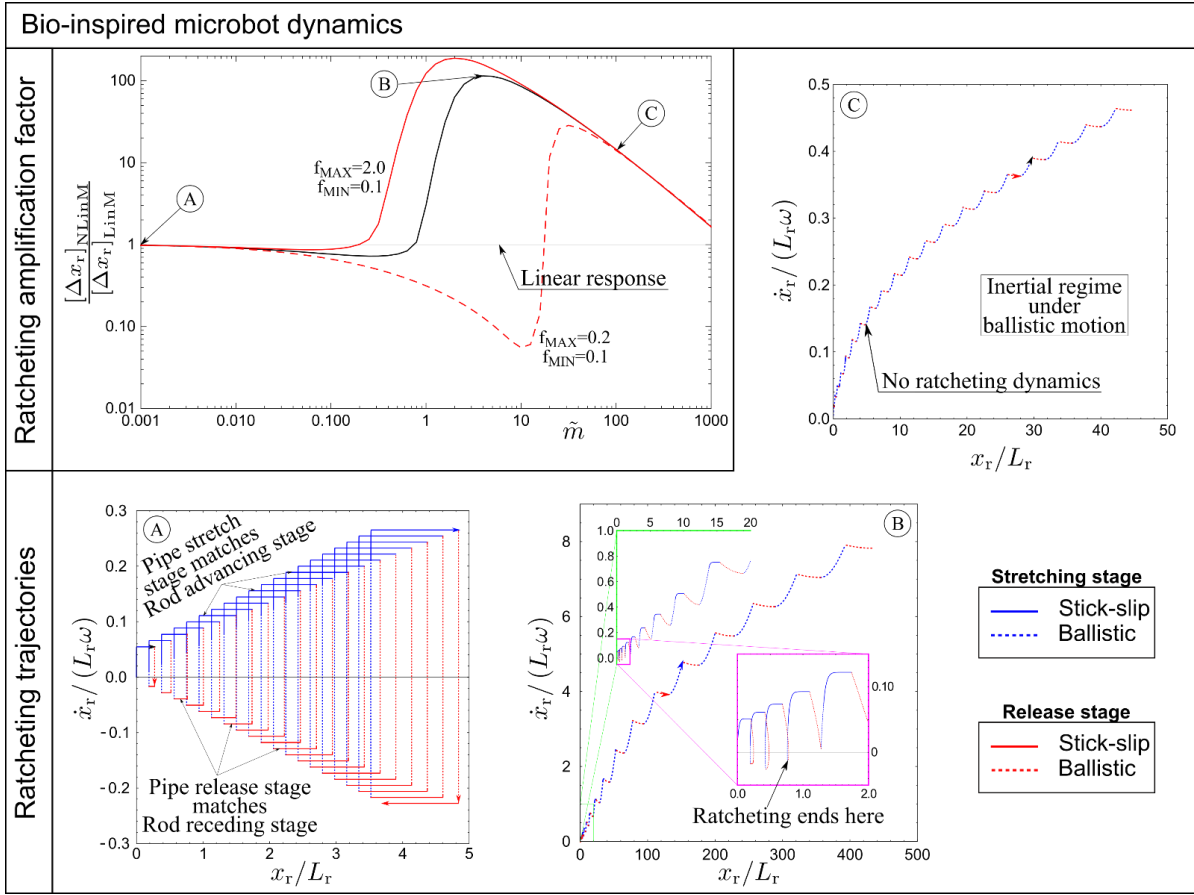


Figure 4. Theoretical results. Dynamic response of the bio-mimetic printed microbot and phase diagrams. (*Top panel*): Ratcheting amplification factor $[\Delta x_r]_{\text{NLinM}}/[\Delta x_r]_{\text{LinM}}$ as a function of the dimensionless actuation frequency number $\tilde{m}$, where $[\Delta x_r]_{\text{NLinM}}$ is the microbot displacement and $[\Delta x_r]_{\text{LinM}}$ is the same displacement as predicted in the inertialess regime. The black (red) [dashed] curve is calculated for the same parameters of Fig. 6a₂, but for a minimum and maximum friction coefficient 0.1 and 1.0 (0.1 and 2.0) [0.1 and 0.2], respectively, and for 20 tube actuation cycles. (*Lower panel*): Microbot ratcheting trajectories in the phase diagram, corresponding to the three identified system actuation regimes (A) linear, (B) resonance, and (C) inertial regime. In the phase diagrams, x_r and $\dot{x}_r$ are the printed awn displacement and velocity, respectively, whereas ω is the actuation frequency. Furthermore, the solid lines are for the stick-sliding stages, whereas the dotted lines are for the ballistic stages (meaning that the rod has no annular stick contact with the tube, see e.g. the red circles in Fig. 3). Finally, the tube elongation stages are indicated in blue color, whereas red is used for the tube release stages – see also the insets in the panel.

2.3 Frictional anisotropy of 3D printed plant-inspired architectures

In order to select the prototyped micro-geometrical parameters exhibiting the largest frictional anisotropy, we performed friction measurements by employing a soft PDMS ball as a counter-surface against the micro-printed awn architecture, see **Figure 5**. In particular, In Figure 5a₁ we report the friction force as a function of sliding time, for a squeezing load 0.5 N, and in Figure 5a₂ the break-loose friction force at different squeezing loads. The effect of anisotropy is conspicuous from low loads. When a normal load of 1.8 N is employed, the friction difference between the two directions of sliding can be marked as a factor of 2.5 between the recorded peak forces.

In the panel b, we provide the break-loose friction force measured at different tilting angles, obtained by varying (Figure 5b₂) θ_1 and (Figure 5b₃) both θ_1 and θ_2 , at constant normal load of 0.5 N, see the schematic of Figure 5b₁ for the parameters description. It is found that the tilt angle of the polymer thorns can generate significant impact on the friction results, Figure 5b₂ and 5b₃. The maximum frictional anisotropy is achieved with the tilting angle set 60/30. Moreover, the packing (measured by the axial distance) of polymer thorns highly affects the results, see Figure 5c. Enlarging the spacing between polymer thorns induces strong fluctuations of the friction values along the *negative sliding direction*, which are strongly mitigated in the positive sliding direction. Very high micro-structure area density leads to low values of both minimum and maximum friction, resulting in a lower friction anisotropy and thus, according to the theory, lower locomotion capabilities. On the other hand, low arrangement density gives rise to low (high) values of friction along the *negative (positive) sliding direction*.

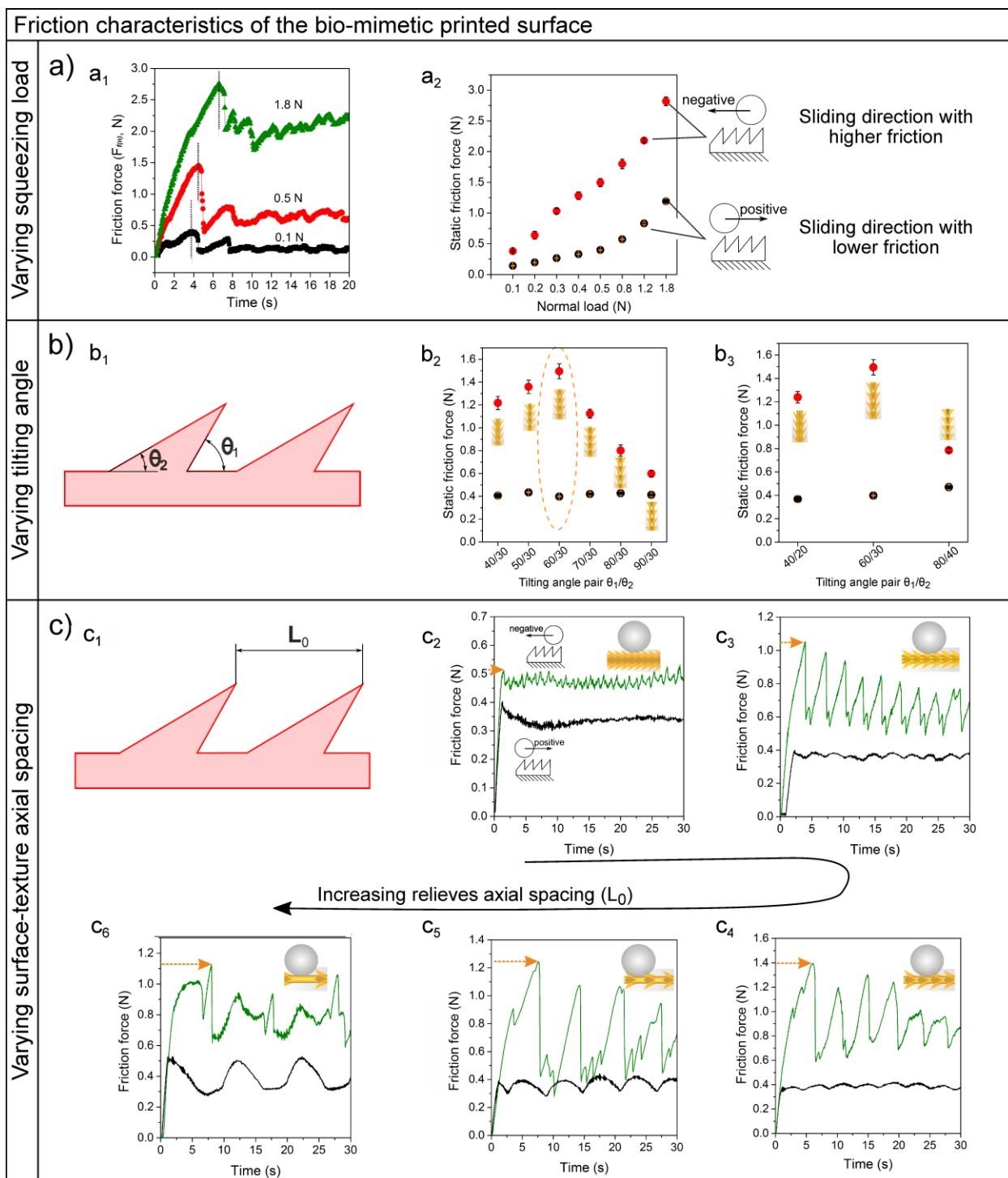


Figure 5. Frictional properties of the bio-mimetic printed anisotropic surfaces analyzed in this study. (a): (a₁) Measured friction force as a function of sliding time, for a squeezing load 0.5 N, and (a₂) breakloose friction force at different squeezing loads. (b): Breakloose friction force measured at different tilting angles, obtained by varying (b₂) θ_1 and (b₃) both θ_1 and θ_2 , at constant normal load of 0.5 N. (c): Breakloose friction force measured at increasing texture axial density, at constant normal load of 0.5 N. The sliding velocity is set to 15 mm min⁻¹ in all tests.

The maximum friction difference is thus achieved for an intermediate value of texture area density, in particular when the spacing between the thorns is $\sim 200\text{ }\mu\text{m}$, see Figure 5c₄. The latter is used in the locomotion and cargo experiments in the following.

2.4 Bio-inspiration for locomotion and cargo operations

The insight into the friction anisotropy obtained through the tribological tests helped us to characterize the interfacial response of the printed architectures when in contact with a PDMS counter-surface. Therefore, the printed structures were placed into the tube (Figure S7, Supporting Information), which was then subjected to periodic stretching, see the schematic of **Figure 6a₁**. Using this testing and actuation design, we find that the moving speed of printed architecture increases monotonically with the tensile frequency of the tube (Figure 6a₂, points). The moving speed of the 3D architecture can achieve up to $63.7\text{ mm/min}^{\pm}$ at a stretching speed of 13500 mm/min . The experimental results are compared with the theoretical prediction (blue curve), and with the prediction of a simplified linear ratcheting model (see also Section 2 in Supporting Information).^[26] Here, it is clearly shown that a simplified linear model is not able to predict the response of the system under investigation and that the non-linearity, including inertia effects, described in the previous section must be accounted for to predict the response of the synthetic construct and the motion of the rod within the flexible tube. The very good agreement with the experimental results provides us with the confidence that the proposed theoretical model can capture the complex dynamics governing the system under investigation.

We now show the capability of the bio-inspired micro-actuator as microbot for cargo operations. In order to assess the transportation capacity of such actuation system, we develop a test platform to measure the maximum payload mass which can be displaced by the synthetic awn, see Figure 6b₁ for the schematic of the setup. In the typical experiment, one end of the tube was fixed, whereas the other was constrained to the dynamical stretcher. Thus, the printed architecture

was first inserted into the tube, and then linked to an electronic tensile meter. The maximum cargo load, *i.e.* the load preventing the system to exhibit an advancing motion, can be therefore recorded, as reported in Figure 6b₂ in term of cargo weight (points). In particular, the maximum cargo weight increases nonlinearly with the strain. The result indicates that such actuation system may generate larger driving forces at higher strains, which can be expected, given that at higher strains the motion ability is increasingly reduced (see e.g. Figure S17 and Figure S8a₃ in Supporting Information), in order to cope with the overall conservation of the mechanical energy provided by the actuation driver. In Figure 6b₂ we also show the theoretical cargo prediction (see Section 2 in Supporting Information for the model extension to locomotion with cargo). The very good agreement between theoretical and experimental results reported in Figure 6b₂ again witnesses the accuracy of the model. The model is easily scalable and can be applied to design actuation and cargo transportation devices for a number of different applications. Furthermore, it provides fundamental insight into different regimes that can be experienced by such systems and, as such, can lead to a great advance with respect to the current state-of-the-art in the study of complex motion devices, whose actuation is governed by the microscopic-induced transitions perceived at the macroscopic scale.

In Figure 6b₃ we report a demonstrative experiment where the anisotropic rod is linked to a payload constituted by a mass, with vertical actuation axis. We observe that the anisotropic interaction can be successfully used to vertically move the 3D architecture and, consequently, to carry the payload along the ratcheting motion (Figure 6b₃). We find that the maximum weight that can be set in motion is approximately 200 folds larger than that of the printed architecture in our experiment.

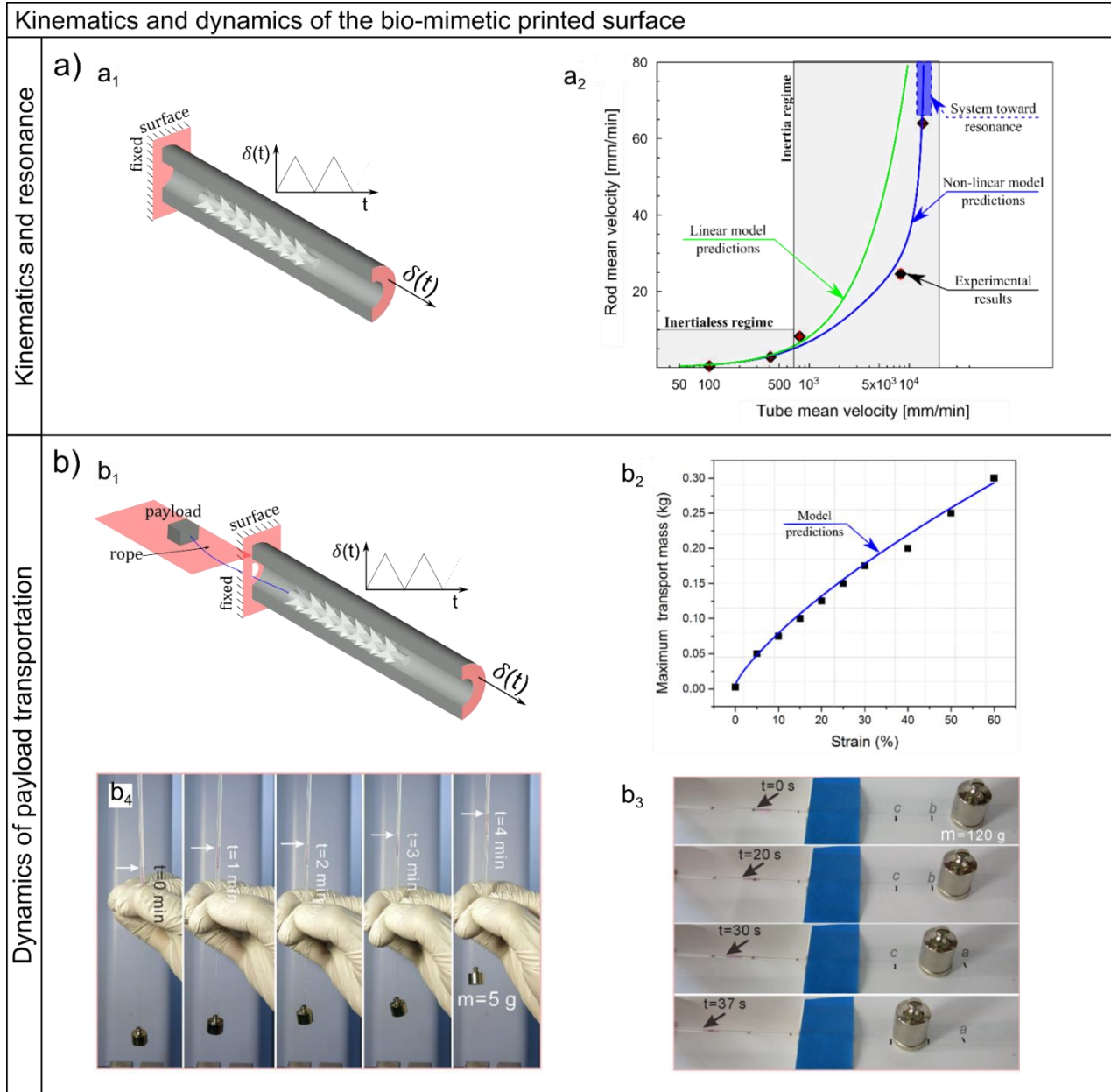


Figure 6. Measurements and theoretical predictions of the microbot displacement and payload dynamics. (a): Displacement dynamics results. (a₁) Schematic of the system setup and actuation diagram; (a₂) Measured rod mean velocity as a function of the tube elongation speed (dots, printed polymer architecture with $\theta_1/\theta_2=60/30$ DEG at a tube strain of 30%) compared with the theoretical predictions (blue curve). The measurements are interrupted close to the occurrence of the resonance regime, predicted by the theory. The theoretical results are calculated on the basis of a rod length 6 mm, rod mass 5 g, tube length 150 mm, tube stretch of 30%, measurements time window 5 min., internal contact tube/rod diameter 1.7 mm, initial average tube/rod contact pressure 50 Pa, and friction coefficients as those showed in Fig. 5a₂. The effective awn/tube interface stiffness has been estimated on the bases of a 1% effective contact area, for the case of (θ_1/θ_2 : 60/30) and distance 0.5 mm between the awn anisotropic relieves. (b): Payload dynamics results. (b₁) Schematic of the system setup and actuation diagram; (b₂) measured maximum payload mass as a function of the tube maximum deformation (dots) compared with the theoretical

predictions (blue curve). The model predictions are obtained with the same parameters of Figure 6 but for an initial average tube/rod contact pressure 65 Pa. (b₃) Demonstration experiment for cargo transportation (friction payload) of a 120 g weight under a stretching frequency of 12.5 Hz (amplitude: 22.5 mm). The outer diameter of the rubber tuber is $2780 \pm 5 \mu\text{m}$, while the inner diameter $1680 \pm 5 \mu\text{m}$. (b₄) Demonstration experiment for cargo transportation (gravity payload) of a 5.0 g weight under a stretching speed of $400 \text{ mm} \cdot \text{min}^{-1}$ at tube strain: 30%. In all the tests, the printed polymer architecture is with $\theta_1/\theta_2=60/30$ DEG tilting angles.

Such remarkable cargo-carrying property, which is extensively found in natural systems, has not been however thoroughly studied or applied in artificial devices, and can become very important for bio-inspired applications, such as for cargo transportation in micro-channels, as micro-vessels, or for micro-manufacturing purposes in confined environment. When the payload is, instead, intermittent, such as when the load is constituted by a frictional force (see the schematic of Figure 6b₁), the system can easily transport 120 g weight under a high oscillating frequency of 25 Hz, as reported in Figure 6b₄ (Movie S4, Supporting Information).

2.5 Discussion and Theoretical Insight

In nature, anisotropic micro-structural geometries are the key surface feature in numerous plants species and animals for their locomotion on a variety of substrates, thanks to frictional anisotropy. Bormuth and co-workers (2007) firstly explored how anisotropic friction is related to the motion of the wild wheat awn into soil in response to changes of e.g. humidity^[6a] and other environmental parameters. Shelley and co-workers (2009) revealed that snakes propulsion on flat ground possibly relies critically on the frictional anisotropy of their skins.^[20] We have also performed experiments on the nature-made awn, see Figure S8 in Supporting Information. In particular, we have adopted the same setup shown in Figure 6a₁ to execute motion experiments by placing the natural awn inside of the rubber tube, the latter driven by a cyclic mechanical uniaxial stretcher. As expected, both the stem of *Humulus japonicus* (Figure S8b₁ and Movie S1 in Supporting Information) and wheat awn (Figure S8b₂ and Movie S2 in Supporting Information) can generate ratchet movements

under mechanical cyclic stretching very similar to the synthetic awn. In the experiments, whose typical results are shown in Figure 8a₂ in Supporting Information, the wheat awn travels along the tube by ~38 mm after 2 min, and ~75 mm after 4 min, under a constant actuating velocity of 400 mm min⁻¹ (the tube is elongated/retracted according to a triangular time-displacement driving cycle). It is observed that the moving speed of the awn increases approximately linearly with the actuation speed (Figure S8a₂, Supporting Information), in the range of probed speeds. It was also found that gravity has negligible effect on the moving speed of the awn (see black filled squares and red filled circles results in Figure S8a₂ in Supporting Information). In the control experiment (when the awn is set in the direction opposite to the one for which self-propulsion is expected), no self-moving phenomenon is observed (the black non-filled squares, the red non-filled circles in Figure S8a₂ in Supporting Information). A very interesting observation in this experiment is that the moving speed of the awn reduces largely, when increasing the strain applied to the rubber tube (Figure 3d). In the typical case, moving speed of the awn is ~15.4 mm min⁻¹ at 3.75% stretch strain and reduces to ~9 mm min⁻¹ at 60% strain. This is most likely due to the fact that high axial strains correspond to large changes in the diameter of the rubber hose. As a result, the awn undergoes large circumferential compressive strains and large normal forces act on the thorns, so that residual frictional stresses might characterize the whole tube/awn contact domain for most of the actuation cycle, the latter occurring therefore mostly under stick conditions. This is in perfect agreement with the simulation results (Figure S17, Supporting Information) for the case of negligible pipe/awn misfit (see the related discussion). As a consequence of this, the moving speed of the awn decreases with increasing axial strain, potentially leading to a null- or near-receding motion beyond a certain stretch threshold is applied, in perfect agreement with the theory reported in Section S2.3 (Figure S15 and Figure S16, Supporting Information).

We observe that, to understand this non-trivial actuation mechanism, a more advanced ratchet model (which accounts for inertial effects) is derived and then extended to include the coupling between longitudinal and radial deformation dynamics of the average tube/awn interface, where Coulomb friction is assumed, in Section S2 in Supporting Information. Interestingly, three different motion regimes are predicted, corresponding to an advancing (Figure 3), receding (Figure S15, Supporting Information) and sticky (Figure S16, Supporting Information) ratcheting dynamics. Furthermore, their relations with the tube stretch, tube size and contact friction properties are unraveled in the “ratcheting maps” reported in Figure S17 and Figure S18 in Supporting Information, to provide a complete study of the system in various scenarios.

To explore the role of friction anisotropy which, as suggested by the theory, has a strong influence on the locomotion capabilities, we have performed locomotion experiments by employing printed micro-geometries with different geometrical characteristics (linked to different friction anisotropy, see Figure 5). A typical real-time displacement snapshots of the moving architecture (marked by red dye) in the tube relative to the initial position is shown in Figure S9a₂ and Movie S3 in Supporting Information. We investigated the effect of printed sawtooth area density on moving speed of the architecture, Figure S9b₁ in Supporting Information. Lowering the sawtooth area density, the moving speed noticeably decreases; however, motion can still be induced. Furthermore, we find that the moving speed exhibits correlation with the tilt angle of the sawtooth (Figure S9b₂, black dots, Supporting Information). Indeed, the fastest response is captured at the tilt angle of θ_1/θ_2 : 60/30, where the largest frictional anisotropy occurs (see Figure 5c), in perfect agreement with the theoretical arguments (see Figure 4 – top panel).

We finally contemplate the effect that the applied normal stress has on the directional motion of the architecture. To do this, we examine how the moving speed of the printed architecture depends on the diameter of the tube (Figure S9b₂, red dots, Supporting Information), while keeping

the same tube wall thickness. It is observed that the moving speed remarkably reduces in thin tubes. This is again in agreement with the theoretical predictions presented in Figure S18 in Supporting Information (left results – the case with misfit, given that the same printed awn is used for both diameters). This confirms that when the diameter of the tube is below a certain value, the excessive normal contact stresses lead the contact to be dominated by stick conditions over most of the actuation cycle, with consequent reduction of the neat-advancing motion capabilities.

3. Conclusion

In summary, in order to reveal and exploit with man-made systems the motion/attachment mechanisms of plants, micro-thorn polymer architectures with controllable thorn size, tilt angle, and area density, were prepared by 3D printing technology. As expected and observed from the study of natural systems, the prepared 3D architectures show significant friction anisotropy and stretching-assisted directional self-motion within rubber tubes. This actuation mechanism is successfully exploited for the development of remotely assisted self-propelling systems and devices, whose potential use for cargo delivery is also experimentally demonstrated. Theoretical models of increasing order of complexity were also developed, which provide very valuable deep understanding of the mechanisms behind the functionality of such devices. The **behavioral** maps obtained through investigation of various parameters were validated using the experimental results that deliver important design guidelines. To the best of our knowledge, these results are the first ones that systematically combine experimental study, theoretical understanding and design strategies, and shed light on the mechanisms governing anisotropic friction induced motion. The results also propose routes for their exploitation in synthetic systems. They can be used not only to elucidate mechanisms behind friction-induced actuation systems in nature, but also to guide future studies on the design of intelligent robots and the use of self-propelling systems to operate in harsh environments. The theoretical modelling presented in this work provides further

motivation for exploring and mimicking the mechanisms of anisotropic friction in nature and engineering.

4. Experimental Section

Materials

Properties of silicone rubber tubes: Tensile strength 7 MPa; elongation to rupture 300 ± 100 %; density 1.25 g/cm^3 ; elastic modulus $1.8\pm 5\%$ MPa (10%-30%). Properties of 3D printed architectures: Elastic modulus 1.15 GPa; Hardness 134 MPa. The Elastomeric PDMS hemispheres were prepared according to classical procedures.^[27] Properties of PDMS ball: Elastic modulus 2.0 MPa; diameter 6 mm.

3D printing polymer constructs

A commercial 3D printer (America ASIGA Pico 27) was employed to build the anisotropic rod through a layer-by-layer polymerization with spatial resolution of $10\text{ }\mu\text{m}$. An ultraviolet (UV) curable resin bought from ASIGA was employed. The solid 3D models with a variety of thorn size, tilt angles and area density were designed using a free CAD tool.

Friction test

In the typical experiment, the friction was measured with a 14-FW Static-dynamic tribometer (HEIDON Co., Ltd.) with single-direction sliding mode. The friction tests of the natural and printed samples were performed at a sliding velocity of 15 mm/min , within a sliding distance of 15 mm , under different loads using an elastic PDMS hemisphere (diameter of 6 mm) as a counter-surface.

Demonstrations for the stretching assisted self-motion

The mechanical stretching of the rubber tube was performed using an electrical universal material testing machine with a 500 N load cell (EZ-Test, SHIMADZU). The stretching velocity and strain of the tube can be changed to generate a loading–unloading cycle. The tube length (gauge length) in all tests is kept constant at 150 mm , so the tensile elongation of the tube at different strain can

be easily calculated. The average moving speed (v_m) of the printed architecture in the rubber tube can be calculated as below, e.g. in consideration of 40 cycles:

$$v_m = \frac{s_m}{t}$$

Here, s_m is the moving distance of the printed architecture at the end of the repeated stretching cycles, while t is the total time.

Supporting Information

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

Author Contributions

F. Zhou proposed the idea. S. H. Ma performed the entire set of experiments, C. Y. Yan assisted the manufacturing of the devices. S. H. Ma drafted the paper. X. L. Wang, S. N. Gorb and F. Zhou discussed the experimental results. M. Scaraggi and D. Dini derived the theoretical formulation and M. Scaraggi performed the simulations. S. H. Ma, M. Scaraggi, D. Dini, S. N. Gorb and F. Zhou. discussed the results and revised the paper.

Acknowledgements

The authors gratefully acknowledge the financial support from the National Natural Science Foundation of China (51775538, 51705507) and the Young Elite Scientists Sponsor Ship Program by CAST (2017QNRC001). D.D. acknowledges the support received from the EPSRC under the Established Career Fellowship Grant EP/N025954/1. M.S. acknowledges the support received from the COST Action CA15216 under grant STSM-CA15216-37369. S.N.G. acknowledges the support Leverhulme Trust [project CARBTRIB].

Received: ((will be filled in by the editorial staff))

Revised: ((will be filled in by the editorial staff))

Published online: ((will be filled in by the editorial staff))

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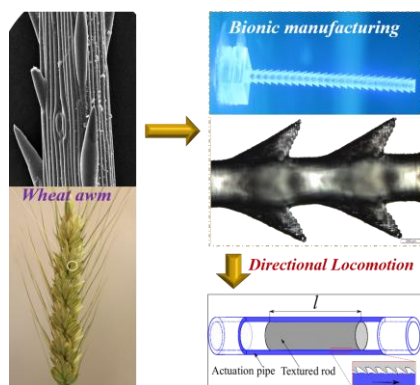
The table of contents entry: **Inspired by anisotropic friction feature in nature**, here we prepare anisotropic polymer micro-actuators with precise scale-matching as those encountered in nature by 3D printing technology. The as-prepared micro-actuators with controllable thorn size, tilt angles and area density exhibit typical friction anisotropy feature and can successfully realize cargo transportation functions. This work provides new avenues for developing intelligent actuators and microbots.

Keyword: *Anisotropic friction; locomotion; wheat awn; bio-inspired; microbots*

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Bio-inspired 3D Printed Locomotion Devices Based on Anisotropic Friction

ToC figure



Supplementary Material

To

Bio-inspired 3D Printed Locomotion Devices Based on Anisotropic Friction

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The SI is outlined as follows:

- Section S1 contains supplementary characterization data regarding both natural and artificial ratchet units.
- Section S2 covers the derivation of the ratchet models adopted in this work, as well as their exploitation to unravel the fundamental physics regulating the different ratchet scenarios responsible for the motion of the systems under investigation.
- Section S3 discusses the physical parameters (reported in form of maps) affecting the ratchet motion, with particular focus on their link with the generation of a receding, null and advancing ratchet motion.

Section S1 – Supplementary information on natural and man-made ratcheting units

In this section, we provide supplementary data regarding the experimental study and the prototyping of nature- and man-made anisotropic friction surfaces. In particular, we show:

- **Supplementary Fig. 1** - Optical and SEM images of pine leaves (*Pinus bungeana* Zucc) (from left to right);
- **Supplementary Fig. 2** - Optical acquisitions of fresh wheatgrass leaves (*Agropyron desertorum* (Fisch.) Schult));
- **Supplementary Fig. 3** - Characteristic time dependency of the sliding friction force for pine leaves under 100 mN load (left), and for fresh wheatgrass leaves under 50 mN load (right);
- **Supplementary Fig. 4** - Optical image of the 3D printer (FREEFORM PICO 27, AISGA, USA) adopted in this study for generating the polymer anisotropic architectures;
- **Supplementary Fig. 5** - Optical acquisition of the printed polymer architectures with different tilt angles (left), and its schematic representation (right);
- **Supplementary Fig. 6** - Characteristic nano-indentation force-displacement curve of the polymer adopted for the architecture printing process (the elastic modulus is calculated by using the slope of unloading). A diamond tip was used as probe, with E_d (elastic modulus)=1194 GPa, ν_d (Poisson's ratio)=0.07, returning a combined interface reduced elastic modulus E_r =3.59 GPa. Given that the printed polymer shows a ν_p (Poisson's ratio)=0.35, the elastic modulus of the printed architecture E_p can then be easily calculated from

$$E_r^{-1} = E_p^{-1}(1 - \nu_p^2) + E_d^{-1}(1 - \nu_d^2);$$

- **Supplementary Fig. 7**. Optical image of the rubber tube employed for the stretching actuation (left), and of the printed polymer architecture (right). Note that a small interference fit is introduced in this study between the rubber tube and the anisotropic rod.
- **Supplementary Fig. 8**. Experimental locomotion results of *Humulus japonicus* stem and wheat awn in a rubber tube under cyclic mechanical stretching.

Top panel: (a₁) Schematics of the actuation setup; (a₂) dependence of the locomotion speed of the awn with the tube stretching speed; (a₃) dependence of the locomotion speed of the awn with the stretch strain of the rubber tube.

Bottom panel: (b₁) Snapshot of the motion generated by a fresh stem of *Humulus japonicus* moving upward in a rubber hose with constant stretching speed of 400 mm min⁻¹ at a 30% strain; (b₂) snapshot of the motion generated by a wheat awn moving upward in a rubber tube (outer diameter 1.78 mm, inner diameter 0.66 mm) with constant stretching speed of 400 mm min⁻¹ at a 30% strain (gauge length 245 mm).

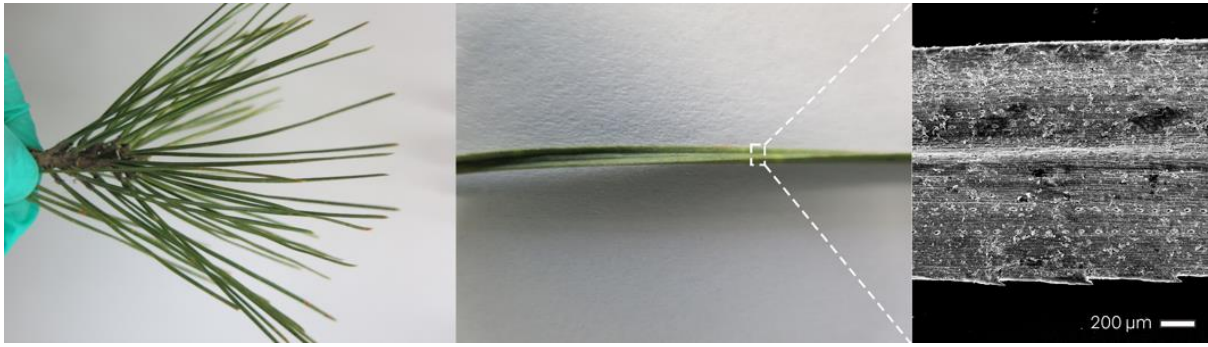
- **Supplementary Fig. 9**. Experimental locomotion results of the printed polymer architecture in the rubber tube.

Top panel: (a₁) Schematics of the actuation setup; (a₂) typical snapshots captured at different time (entire stretching time: 5 min) illustrating the relative motion of the printed polymer architecture (θ_1/θ_2 : 60/30) in the rubber tube under stretching speed of 400 mm·min⁻¹(tube strain: 30%).

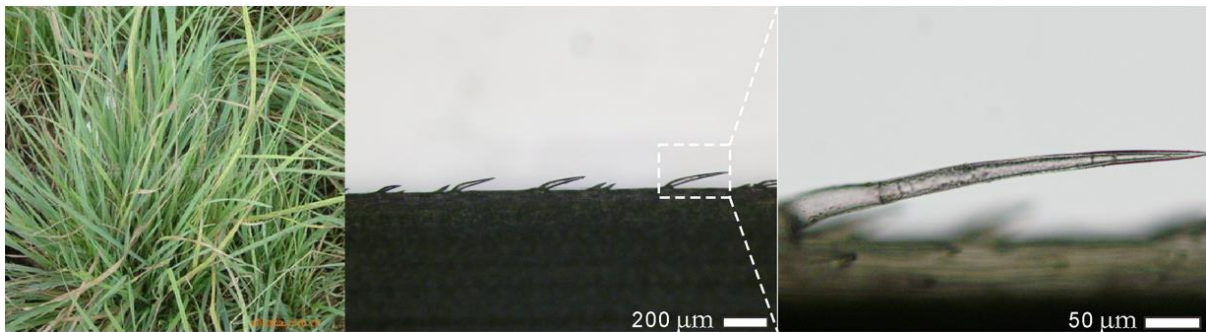
Bottom panel: (b₁) Dependence of the arrangement density of the polymer thorns in the printed architectures (θ_1/θ_2 : 60/30) on the moving speed at a tube strain of 30%. The distance between two adjacent thorns increases from 456 µm to 912µm, and then to

2280 μm , as reported also in Figure 4d; (b_2) dependence of the tilt angle of the polymer thorns (θ_1/θ_2 : 40/30, 60/30 and 80/30), and the employed tube diameter (inner diameter: 1.68 mm, 1.33 mm) on the locomotion speed at a tube strain of 30%.

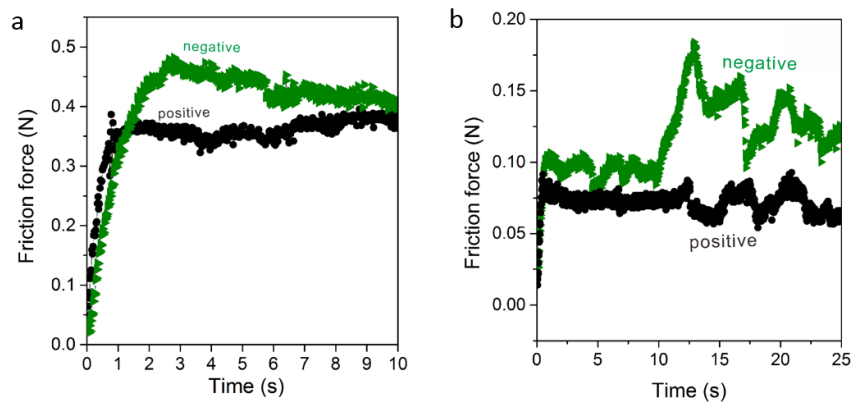
- **Supplementary Fig. 10.** Schematic representation of the actuation mechanism. The inset shows a magnified view of the contact interface.
- **Supplementary Fig. 11.** Numerical results. Ratcheting amplification factor $[\Delta x_r]_{\text{NLinM}}/[\Delta x_r]_{\text{LinM}}$ as a function of the dimensionless frequency number $\tilde{m}$, at varying system parameters. The black reference curve is calculated for the same parameters of Figure 6a, but for a minimum and maximum friction coefficient 0.1 and 1.0, respectively, and for a 20 tube actuation cycles. The red solid and dashed curves are for a maximum friction coefficient 2.0 and 0.2, respectively; the blue solid and dashed curves are for a tube stretch 0.5 and 0.1, respectively; the green solid and dashed curves are for a number of tube actuation 30 and 10, respectively.
- **Supplementary Fig. 12.** Numerical results. Ratcheting normalized displacement $[\Delta x_r(\varepsilon)]_{\text{NLinM}}/[\Delta x_r(\varepsilon)]_{\text{LinM}}$ (black lines) and velocity $[\Delta \dot{x}_r(\varepsilon)]_{\text{NLinM}}/[2\varepsilon_0\omega x_{r0}]$ (blue lines) as a function of the actual tube stretch ε (in log scale). The results are for the same parameters of Figure 6a, but for a minimum and maximum friction coefficient 0.1 and 1.0, respectively, and for a single tube actuation cycle starting with an initial rod position $x_{r0}/L_r = 50$. Eight values of dimensionless square frequency numbers $\tilde{m}$ are adopted (see figures): (A) and (B) are in the pre-antiresonance regime, (C) in the post-antiresonance, (D) and (E) in the pre-resonance, (F) is close to resonance, and (G) and (H) are in the post-resonance regime. The arrows over the curves show the time flow along the triangular actuation cycle, from $\varepsilon = 0$ to ε_0 (tube stretch) and back to $\varepsilon = 0$ (tube release). The generic solid line is adopted when the rod-tube contact occurs under stick-slip, whereas the dashed line is for the ballistic case (full sliding).
- **Supplementary Fig. 13.** Numerical results. The rod stick-contact position $2\bar{s}_r/L_r$ (black lines), as a function of the actual tube stretch ε (in log scale). The results are calculated for the same parameters of Figure S11.
- **Supplementary Fig. 14.** Schematic representation of the actuation mechanism under frictional cargo.
- **Supplementary Fig. 15.** Numerical results. Ratchet mechanism leading to a receding motion.
- **Supplementary Fig. 16.** Numerical results. Ratchet mechanism not leading to a clear motion.
- **Supplementary Fig. 17.** Actuation cycle response for a ratchet system as a function of the friction anisotropy, at varying actuation tube stretch ε_0 . The system geometry is kept constant. The left and right side is with and without initial rod/tube mechanical interference, respectively.
- **Supplementary Fig. 18.** Actuation cycle response for a ratchet system as a function of the friction anisotropy, at varying tube cross sections (but constant rod length). The actuation tube stretch ε_0 is kept constant. The left and right side is with and without initial rod/tube mechanical interference, respectively.



Supplementary Figure 1 | Optical images and SEM image of pine leaves (*Pinus bungeana* Zucc) (from left to right).



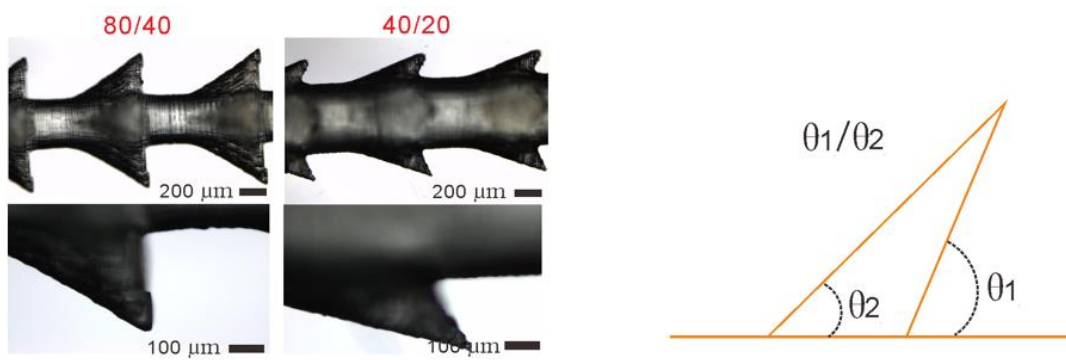
Supplementary Figure 2 | Optical acquisitions of fresh wheatgrass leaves (*Agropyron desertorum* (Fisch.) Schult).



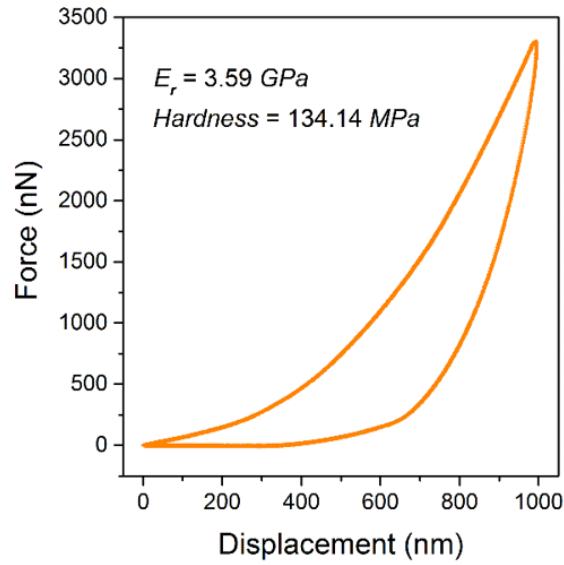
Supplementary Figure 3 | Characteristic time dependency of the sliding friction force for pine leaves under 100 mN load (**a**), and for fresh wheatgrass leaves under 50 mN load (**b**).



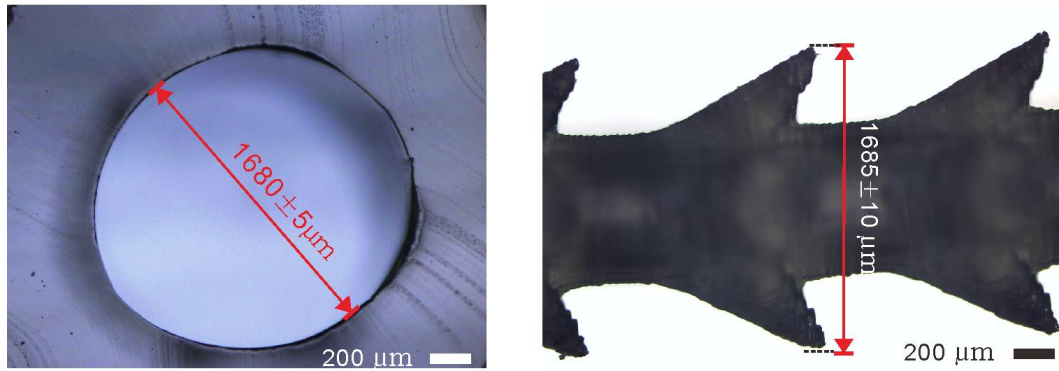
Supplementary Figure 4 | Optical image of the 3D printer (FREEFORM PICO 27, AISGA, USA) adopted in this study for generating the anisotropic polymer architectures.



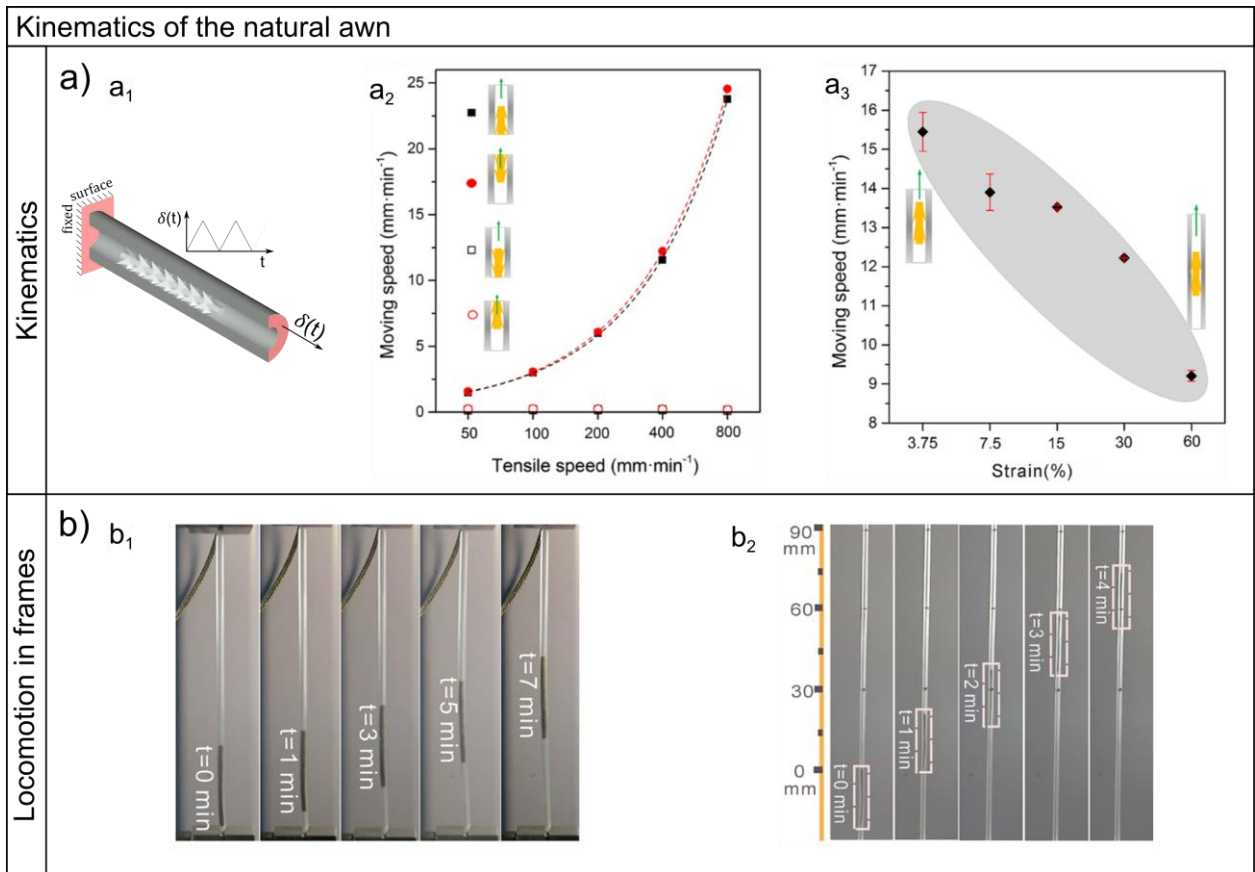
Supplementary Figure 5 | Optical acquisition of the printed polymer architectures with different tilt angles (left), and its schematic representation (right).



Supplementary Figure 6 | Characteristic nano-indentation force-displacement curve of the polymer adopted for the architecture printing process.



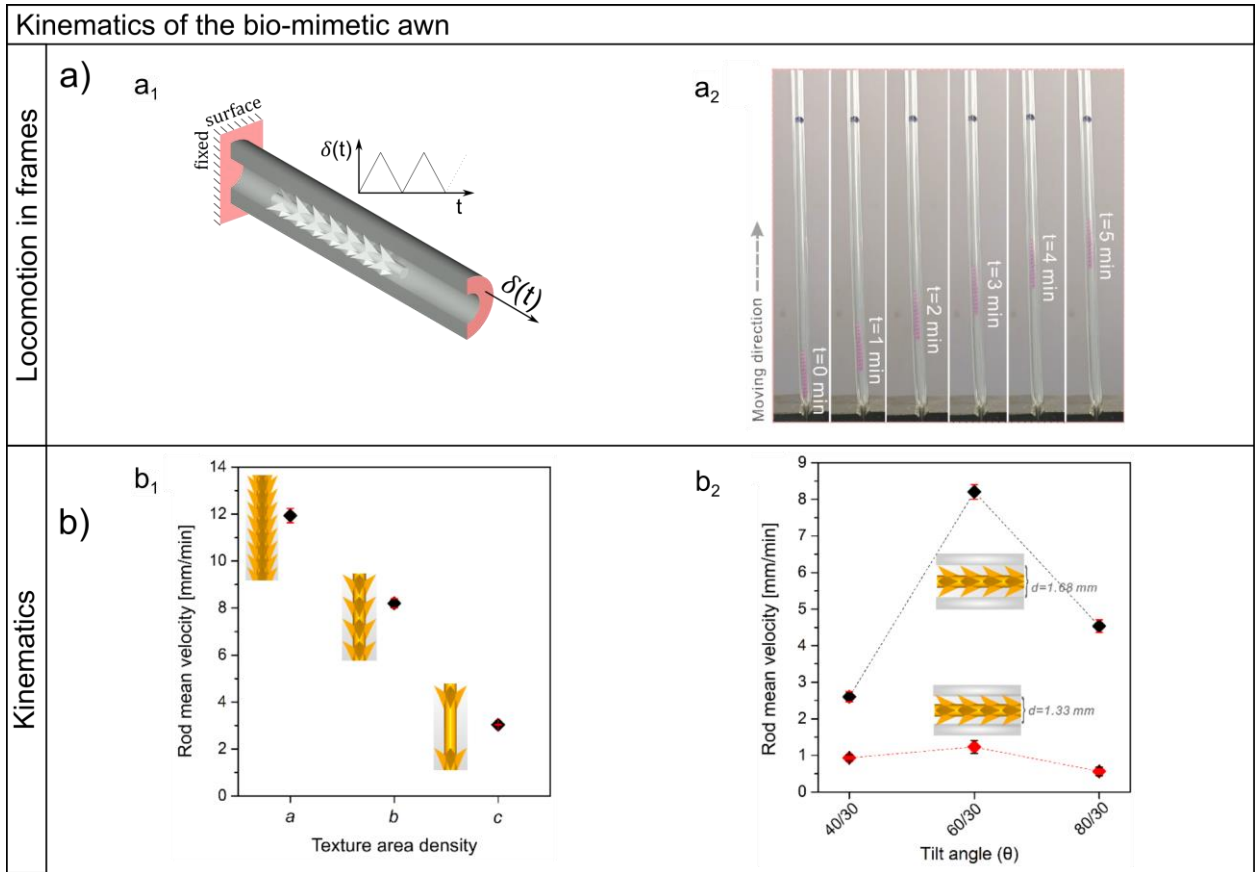
Supplementary Figure 7 | Optical image of the rubber tube employed for the stretching actuation (left), and of the printed polymer architecture (right). Note that a small interference fit is introduced in this study between the rubber tube and the anisotropic rod.



Supplementary Figure 8 | Experimental locomotion results of *Humulus japonicus* stem and wheat awn in a rubber tube under cyclic mechanical stretching.

Top panel: (a_1) Schematics of the actuation setup; (a_2) dependence of the locomotion speed of the awn with the tube stretching speed; (a_3) dependence of the locomotion speed of the awn with the stretch strain of the rubber tube.

Bottom panel: (b_1) Snapshot of the motion generated by a fresh stem of *Humulus japonicus* moving upward in a rubber hose with constant stretching speed of 400 mm min^{-1} at a 30% strain; (b_2) snapshot of the motion generated by a wheat awn moving upward in a rubber tube (outer diameter 1.78 mm, inner diameter 0.66 mm) with constant stretching speed of 400 mm min^{-1} at a 30% strain (gauge length 245 mm).



Supplementary Figure 9 | Experimental locomotion results of the printed polymer architecture in the rubber tube.

Top panel: (a_1) Schematics of the actuation setup; (a_2) typical snapshots captured at different time (entire stretching time: 5 min) illustrating the relative motion of the printed polymer architecture (θ_1/θ_2 : 60/30) in the rubber tube under stretching speed of $400 \text{ mm} \cdot \text{min}^{-1}$ (tube strain: 30%).

Bottom panel: (b_1) Dependence of the arrangement density of the polymer thorns in the printed architectures (θ_1/θ_2 : 60/30) on the moving speed at a tube strain of 30%. The distance between two adjacent thorns increases from $456 \mu\text{m}$ to $912 \mu\text{m}$, and then to $2280 \mu\text{m}$, as reported also in Figure 4d; (b_2) dependence of the tilt angle of the polymer thorns (θ_1/θ_2 : 40/30, 60/30 and 80/30), and the employed tube diameter (inner diameter: 1.68 mm, 1.33 mm) on the locomotion speed at a tube strain of 30%.

Section S2 - Simplified ratchet model

Model \ Approximation	Rod with inertia	Simple radial tube deformation	Axial dependence of the radial tube deformation
Sec. S2.1: Linear model	No	No	No
Sec. S2.2: Non-linear	Yes	Yes	No
Sec. S2.3: Advanced	Yes	Yes	Yes

Supplementary Table 1| Ratcheting models and corresponding degree of approximation

In this section, we derive ratchet models at increasing degree of refinement of the theory, as schematically reported in Supplementary Table 1:

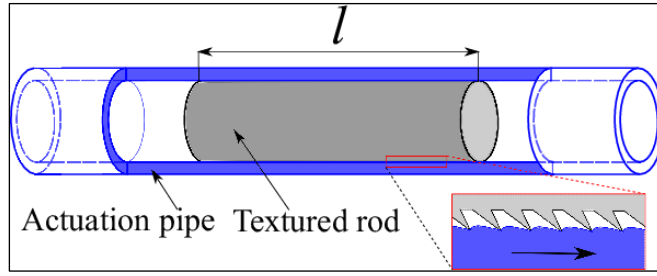
- 1) In Sec. S2.1, the linear ratchet model (LinM) is derived by considering the case, where the rod inertia can be neglected, as well as considering the tube elastic and characterized by only axial displacement fields (during the actuation cycle). As such, the model is strictly applicable to quasi-static systems only (no inertia effects are introduced), with low coupling between the longitudinal and radial displacement fields. This is *e.g.* the case where the tube/rod interface is characterized by a relatively low sliding friction (under quasi-static motion).
- 2) In Sec. S2.2, the non-linear ratchet model (NLinM) is derived, as an extension of LinM by including the rod inertia and other ballistic effects in the ratchet dynamics, as well as by considering a simple model for the tube radial deformation.
- 3) Finally, in Sec. S2.3 the advanced ratchet model (AdvM) is built based on the NLinM and by taking into account the axial dependence of the radial tube deformation.

In all the theories reported in this study, the rod is assumed to be much stiffer than the tube (and hence modelled as rigid). This assumption is well supported by the threefold difference in the elastic moduli of the two materials, see the Materials and Method Section in the main manuscript. Furthermore, the micro-contacts occurring between the rod micro-relieves and the internal wall of the tube are averaged out in term of effective contact pressure and friction coefficient, therefore not accounting for local effects and inhomogeneity.

S2.1 - Simplified theory of ratchet actuation mechanism - linear ratchet model (LinM)

We assume a homogeneously rough cylinder of length L_r and nominal radius R in axial symmetric contact with the inner side of an ideally smooth cylindrical crown, with inner and outer radius, respectively, R_i and R_o . In this work the cylindrical crown (the tube) corresponds to the actuation driver, and, as such, the model is developed considering the driving constraint applied to the cylindrical crown. However, we observe that the model below can be easily extended to the case, where the driving constraint is applied to the biomimetic rod (in such a case, the latter corresponds to the actuation driver). Since $R \geq R_i$, a non negligible interference fit pressure is generated at the contact interface. Furthermore, we consider $R \ll L_r$, as well as the roughness (covering the rod) to be characterized by a nominal wavelength $\lambda_0 = 2\pi/q_0 \ll L_r$, where q_0 is a nominal large scale roughness frequency. Finally, we assume that the representative size of the roughness (which represents the thorn of the awn in the synthetic construct), say h_{rms} , is much smaller than the contact radius R . Such separation of length scales allows to separate the micro-contact dynamics from the rod/tube macro-contact dynamics. The schematic of the macro-contact geometry is

reported in Supplementary Fig. 10.



Supplementary Figure 10 | Schematic representation of the actuation mechanism. The inset shows a magnified view of the contact interface.

Given the rotational symmetry of the macro-geometry, we can accordingly consider the law of motion of the generic cylindrical surface point to be provided by a simple scalar function, e.g. $x_r(s_r, t)$ [$x_p(s_p, t)$ for the tube]. $x_r(s_r, t)$ [$x_p(s_p, t)$ for the tube] describes the position of the rod material point s_r [s_p for the tube] as a function of time. We consider $s_r \in [-L_r/2, L_r/2]$, so that $s_r = 0$ corresponds to the center of mass of the rod. Furthermore, the tube can be considered fixed at one end, thus $s_p \in [0, \infty[$, with $s_p = 0$ corresponding to the fixed tube end. We can write

$$\frac{\partial x_r(s_r, t)}{\partial s_r} = 1 + \varepsilon_r(s_r, t)$$

and

$$\frac{\partial x_p(s_p, t)}{\partial s_p} = 1 + \varepsilon_p(s_p, t),$$

where $\varepsilon_r(s_r, t)$ and $\varepsilon_p(s_p, t)$ is the local deformation of the rod and tube, respectively. Thus

$$x_r(s_r, t) = x_r(t) + s_r + \int_0^{s_r} ds'_r \varepsilon_r(s'_r, t)$$

and

$$x_p(s_p, t) = s_p + \int_0^{s_p} ds'_p \varepsilon_p(s'_p, t).$$

$x_r(t)$ is the position of the undeformed rod center of mass. Since the rod-tube (effective conformal) contact occurs along the rod material coordinates $s_r \in [-L_r/2, L_r/2]$, we can write $x_r(s_r, t) = x_p(s_p, t)$ for $s_r \in [-L_r/2, L_r/2]$. Thus, we obtain the range of tube material points in contact as a function of time

$$s_p + \int_0^{s_p} ds'_p \varepsilon_p(s'_p, t) = x_r(t) + s_r + \int_0^{s_r} ds'_r \varepsilon_r(s'_r, t).$$

We note again that the Young's modulus of the rod is in order of GPa, whereas the tube is made by a rubber compound of elastic modulus in the order of MPa. As such, we can neglect the deformation of the bio-mimicking awn with respect to the tube deformation dynamics (i.e. $\varepsilon_r(s_r, t) \approx 0$), resulting in a range of tube material points involved with the contact to be well approximated by:

$$s_p + \int_0^{s_p} ds'_p \varepsilon_p(s'_p, t) = x_r(t) + s_r. \quad (\text{S2.1})$$

The sliding speeds are easily calculated from

$$\frac{\partial x_r(s_r, t)}{\partial t} = \frac{\partial x_r(t)}{\partial t}$$

$$\frac{\partial x_p(s_p, t)}{\partial t} = \int_0^{s_p} ds_p' \frac{\partial \varepsilon_p(s_p', t)}{\partial t}.$$

We now write the momentum equation for the rod, along the sliding direction. Since $\partial x_r / \partial s_r = 1$ we have

$$\int ds_r \sigma_T(s_r, t, v_r, \sigma_0) = m_r \frac{\partial^2 x_r(t)}{\partial t^2} - F(t), \quad (\text{S2.2})$$

where $\sigma_T(s_r, t, v_r, \sigma_0)$ is a friction force per unit length, given by the local micro-contact mechanics, where

$$v_r(s_r, t) = \frac{\partial x_r(s_r, t)}{\partial t} - \frac{\partial x_p(s_p, t)}{\partial t} = \frac{\partial x_r}{\partial t} - \frac{\partial \varepsilon_p}{\partial t} \frac{x_r + s_r}{1 + \varepsilon_p} \quad (\text{S2.3})$$

is the relative sliding speed of the rod material point s_r with respect to the tube, σ_0 the local contact pressure, and $F(t)$ a generic tangential –in the direction of motion- force carried by the rod. The force $F(t)$ is also named *cargo force* in the main manuscript.

The situations in which inertia (inertialess) and cargo (cargoless) are not considered correspond to Eq. (S2.2) when $m_r = F = 0$. In such a case the motion equation is simply given by

$$\int ds_r \sigma_T(s_r, t, v_r) \approx 0. \quad (\text{S2.4})$$

From (S2.4) it follows that the friction forces acting on the rod have a null total resultant, implying that the rod contact surface is split in two domains ($[-1/2, \bar{s}_r]$ and $[\bar{s}_r, 1/2]$), characterized by opposite relative sliding speeds. A direct consequence of this is that, in the inertialess and cargoless cases, the only possible motion stage is the stick-sliding (i.e. no ballistic regime occurs). To determine $\bar{s}_r$ one simply integrates (S2.2) between $[-1/2, \bar{s}_r]$ and $[\bar{s}_r, 1/2]$ with different sliding friction coefficients and same contact pressure along the contact domain, leading to the stick rod material point

$$\bar{s}_r = \pm \frac{f_{\max} + f_{\min}}{f_{\max} - f_{\min}} \frac{L_r}{2}, \quad (\text{S2.5})$$

where the top (bottom) sign is for tube elongation (release). $f_{\max}$ ($f_{\min}$) is the maximum (minimum) friction coefficient in the tube/rod contact. The physical reason leading to $\bar{s}_r > 0$ ($\bar{s}_r < 0$) during tube elongation (release) is that the sticking rod point during elongation must be located at larger s_r with respect to the release stage, in order for the rod to move forward with a larger distance (during tube stretching) than during the backward (release) motion. Then, by substituting Eq. (S2.5) in Eq. (S2.3) with the condition $v_r(\bar{s}_r, t) = 0$, after integration over one actuation cycle, one easily gets the clear ratcheting step after one actuation cycle Δx_r , which reads

$$\Delta x_r = L_r \frac{f_{\max} - f_{\min}}{f_{\max} + f_{\min}} \frac{\varepsilon_0}{1 + \varepsilon_0}. \quad (\text{S2.6})$$

Equation (S2.6) is employed in the green curve of Fig. 6a₂ (main manuscript). Note that Δx_r only depends on the degree of friction anisotropy ($\frac{f_{\max} - f_{\min}}{f_{\max} + f_{\min}}$), as well as on the rod length (L_r) and tube maximum stretch (ε_0). Note that Eq. (S2.6) is similar to the analytical result reported in ref.¹ and is derived assuming uncoupling between the longitudinal and radial deformation dynamics of the tube, as well as with constant tube deformation along the contact interface. Furthermore, no inertial

effects are included. As a result, $\Delta x_r > 0$ independently of the amount of tube elongation and of friction anisotropy, leading to an advancing motion as the only predictable ratcheting state. Nevertheless, we show in the following section that depending on the amount of tube stretching as well as of friction anisotropy and actuation speed, the ratcheting exhibits non-trivial dynamics, which in turn may lead to *e.g.* neat receding or stick motions.

S2.2 – Non-linear theory of ratchet actuation mechanism – non linear ratchet model (NLinM)

We start observing that the rod inertia m_r is ~ 10 g in our application. One might wonder whether it is relevant to study inertial effects for micro actuation mechanisms, where an actuation force between 1 to 10 N is required to stretch the mm-sized tube to a strain value of ~ 0.3 , whereas the inertial (apparent force) term can be estimated to be ≈ 0.03 N for the highest actuation speed reported in the Fig. 6a₂. However, we do observe that in order to provide a full picture of the system and an accurate analysis, the inertial force term has to be compared with the resultant frictional forces acting at the interface of the tube/rod contact rather than the actuation force. The entity of such frictional term is extremely sensitive to the amount of interference fit between the anisotropic rod and the internal tube wall. In this preliminary study, we have adopted relatively small values of tube/rod interference fit, as shown in Supplementary Fig. 7. Therefore, in order to make such estimation, we first observe that for an initial interference fit of ~ 2.5 μm (see Supplementary Fig. 7), assuming the classical contact mechanics relations, to apply for the topmost rod relief/tube contact relations, we get (with a representative effective width ~ 200 μm) a contact radius ~ 22 μm . Thus, the local micro-contact pressure can be estimated to be ~ 0.1 MPa, and the local micro-contact force ~ 0.2 mN. By considering ~ 100 contacts per rod length (~ 6 mm), and the friction coefficient ~ 2 (average friction, see friction experiments in Fig. 5a₂), we get a total rod frictional force ~ 0.04 N, which is of the same order as the inertial force. Consequently, at least in the initial stage of the actuation cycle, where the tube lateral contraction does not play an important role (tube stretch is zero), the inertia might dominate the ratcheting dynamics. Nevertheless, given the small true contact area between tube and rod (the normalized value with respect to the nominal contact area is $\alpha \sim 0.005$ in the initial conditions), even at non-zero tube stretch the average tube/rod contact pressure will be of similar order. This is due to the high compliance of the interface stiffness (the interface stiffness can be shown to be proportional to the normalized true contact area for rough interactions).

To take into account the inertial effect, we solve Eq. (S2.1) - (S2.3) derived above, with the friction force σ_r independent of the sliding speed, but characterized by a linear dependence with the local contact pressure σ , which we show in the following (see Section S2.3) to be, for a thin-walled tube, a function of the form:

$$\sigma \approx \mu c_\varepsilon \varepsilon_p + \sigma_0,$$

where $c_\varepsilon = 8\alpha/(2 + \varepsilon_R)$ is an order α parameter (α is the normalized true contact area). Moreover, the local contact pressure σ and the tube deformation are assumed to be constant in the contact domain, i.e. $\varepsilon_p(s_p, t) \approx \varepsilon_p(t)$ (this latter assumption will be removed in the advanced ratchet model, see in the following). After integration of Eq. (S2.2), we obtain:

$$\bar{s}_r(t) = -\frac{\partial^2 x_r(t)}{\partial t^2} \frac{m_r}{2\pi R \sigma_0 \tilde{\sigma}(\pm 2f_0)} + \frac{F(t)}{2\pi R \sigma_0 \tilde{\sigma}(\pm 2f_0)} \pm \frac{L_r}{2} \frac{f_{\max} + f_{\min}}{f_{\max} - f_{\min}}, \quad (\text{S2.6})$$

where $f_0 = (f_{\max} + f_{\min})/2$, $\tilde{\sigma} = (1 + \mu c_\varepsilon \varepsilon_p / \sigma_0)$ and where the top (bottom) sign is for the tube elongation (release) stage.

In the cases where $|\bar{s}_r| < 1/2$, $\bar{s}_r$ is the rod material point at which the inversion of the interface friction forces occurs, i.e. where a rod circle is in sticking contact with the tube. As such, at $\bar{s}_r$ the relative sliding speed $v_r = 0$, leading from Eq. (S2.3)

$$\frac{\partial x_r(t)}{\partial t} = \frac{x_r(t) + \bar{s}_r(t)}{1 + \varepsilon_p(t)} \frac{\partial \varepsilon_p(t)}{\partial t}. \quad (\text{S2.7})$$

By embedding Eq. (S2.7) in Eq. (S2.2) we have

$$v_r = -\frac{s_r - \bar{s}_r}{1 + \varepsilon_p} \frac{\partial \varepsilon_p}{\partial t},$$

showing that rod points are characterized by opposite relative sliding speeds depending on their position with respect to the circle of stick material points $\bar{s}_r$.

In those other cases, where the sticking circle does not exist, e.g. for large inertial terms or cargo forces (see Eq. (S2.6)), the rod motion occurs under ballistic conditions. In this case, Eq. (S2.7) is replaced by:

$$\bar{s}_r(t) = -L_r/2. \quad (\text{S2.8})$$

To summarise:

- the anisotropic rod equation of motion is given by Eqs. (S2.6) and (S2.7) if $|\bar{s}_r| \leq 1/2$ in the stick-sliding regime,
- and by Eqs. (S2.6) and (S2.8) if $|v_r(\bar{s}_r)| > 0$ in the ballistic regime.

The transition between the two motion regimes is *trimmed* when $v_r(\bar{s}_r) = 0$.

In the following, Eqs. (S2.6), (S2.7) and (S2.8) are made dimensionless adopting L_r as a reference measure of length and the inverse of the tube stretching frequency ω^{-1} as a reference measure of time. Furthermore, the same equations are much easily solvable through adopting the variable change $\beta = \log(1 + \varepsilon_p)$. In this case Eqs. (S2.6), (S2.7) and (S2.8) become:

$$\tilde{\tilde{s}}_r(\beta) = \mp \frac{\tilde{m}}{\tilde{\sigma}} \frac{\partial^2 \tilde{x}_r(\tilde{t})}{\partial \tilde{t}^2} \pm \frac{\tilde{F}(\beta)}{\tilde{\sigma}} \pm \frac{\varepsilon_f}{2} \quad (\text{S2.9})$$

$$\frac{\partial \tilde{x}_r}{\partial \beta} = \tilde{x}_r(\beta) + \tilde{\tilde{s}}_r(\beta), \quad (\text{S2.10})$$

$$\tilde{\tilde{s}}_r(\beta) = \pm 1/2, \quad (\text{S2.11})$$

where $\tilde{m} = m_r \omega^2 / (4\pi R \sigma_0 f_0)$, $\tilde{F} = F / (4\pi R \sigma_0 f_0 L_r)$, $\varepsilon_f = \frac{f_{\max} + f_{\min}}{f_{\max} - f_{\min}}$, $f_0 = \frac{f_{\max} + f_{\min}}{2}$ and $\tilde{\sigma} =$

$1 + \mu c_\varepsilon \sigma_0^{-1} (e^\beta - 1)$. As before, the top (bottom) sign is for the tube elongation (release) stage.

The solution of Eq. (S2.10), needed for the stick-sliding regime, can be obtained in standard way as:

$$\frac{\partial \varepsilon_p(t)}{\partial t} > 0: \quad \tilde{x}_r(\beta) e^{-\beta} = x_{r0} + \int_0^\beta d\alpha e^{-\alpha} \tilde{s}_r(\alpha), \quad (\text{S2.12.1})$$

$$\frac{\partial \varepsilon_p(t)}{\partial t} < 0: \quad \tilde{x}_r(\beta) e^{-\beta} = x_{r,max} e^{-\beta_{max}} - \int_{\beta}^{\beta_{max}} d\alpha e^{-\alpha} \tilde{s}_r(\alpha), \quad (\text{S2.12.2})$$

where $x_{r,max} = \tilde{x}_r(\beta_{max})$ and with $\beta_{max} = \log(1 + \varepsilon_0)$.

Dynamic response

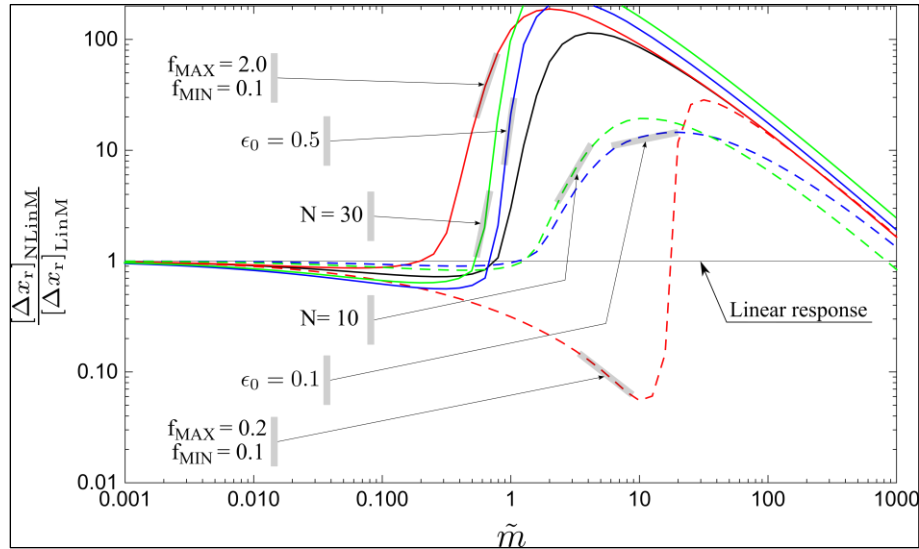
The ratcheting dynamics is analysed here by solving the system of differential Eqs. (S2.9) - (S2.11). For the actuation law, we have used a triangular-based actuation cycle for $\varepsilon_p(\tilde{t})$, in particular $\varepsilon_p(\tilde{t} \in [0, 1/2]) = 2\varepsilon_0\tilde{t}$, whereas $\varepsilon_p(\tilde{t} \in [1/2, 1]) = 2\varepsilon_0(1 - \tilde{t})$. We also consider zero cargo force, i.e. $\tilde{F} = 0$. In this case, no analytical solution can be found, so Eqs (S2.9) - (S2.11) are solved numerically, in their appropriate regimes of validity. With the triangular-based actuation cycle, Eq. (S2.9) can be rewritten as

$$\tilde{s}_r(\beta) = \mp \tilde{m} \left(\frac{\partial^2 \tilde{x}_r}{\partial \beta^2} \dot{\beta}^2 + \frac{\partial \tilde{x}_r}{\partial \beta} \ddot{\beta} \right) \pm \frac{\varepsilon_f}{2}, \quad (\text{S2.13.1})$$

$$\dot{\beta}^2 = -\ddot{\beta} = 4\varepsilon_0^2 e^{-2\beta}. \quad (\text{S2.13.2})$$

In Supplementary Fig. 11 we show the numerical results and, in particular, we provide the ratcheting amplification factor $[\Delta x_r]_{NLinM}/[\Delta x_r]_{LinM}$ as a function of the dimensionless square frequency number $\tilde{m}$, at varying system parameters. In particular, the black curve is calculated for the same parameters of Fig. 6a₂, but for a minimum and maximum friction coefficient 0.1 and 1.0, respectively, and for a 20 tube actuation cycles. Moreover:

- the red solid and dashed curves are for a maximum friction coefficient 2.0 and 0.2, respectively;
- the blue solid and dashed curves are for a tube maximum strain of 0.5 and 0.1, respectively;
- the green solid and dashed curves are for a number of tube actuation 30 and 10, respectively.



Supplementary Figure 11 | Numerical results. Ratcheting amplification factor $[\Delta x_r]_{NLinM}/[\Delta x_r]_{LinM}$ as a function of the dimensionless frequency number $\tilde{m}$, at varying system parameters. The black reference curve is calculated for the same parameters of Figure 6a₂, but for a minimum and maximum friction coefficient 0.1 and 1.0, respectively, and for a 20 tube actuation cycles. The

red solid and dashed curves are for a maximum friction coefficient 2.0 and 0.2, respectively; the blue solid and dashed curves are for a tube stretch 0.5 and 0.1, respectively; the green solid and dashed curves are for a number of tube actuation 30 and 10, respectively.

The ratcheting amplification factor shows the occurrence of three typical features, in particular a linear regime for $\tilde{m} \rightarrow 0$, which is characteristic of the inertialess response. Secondly, a sublinear regime with anti-resonance (markedly for small friction values) appears for increasing dimensionless square frequencies $\tilde{m}$. By further increasing $\tilde{m}$, the resonance and post-resonance inertial regimes occur. Among the physical parameters affecting the dynamics, the friction coefficient has the largest effect on the ratcheting motion: a ten-fold variation of the maximum friction value leads to a ten-fold variation of the resonance amplification factor and its frequency as well. As such, the friction (or better, the traction coefficient) needs to be an extremely controlled parameter for this micro-actuators.

In Supplementary Fig. 12 and Supplementary Fig. 13 we show, respectively, the ratcheting normalized displacement $[\Delta x_r(\varepsilon)]_{NLinM}/[\Delta x_r(\varepsilon)]_{LinM}$ (black lines) and velocity $[\Delta \dot{x}_r(\varepsilon)]_{NLinM}/[2\varepsilon_0\omega x_{r0}]$ (blue lines), and the rod stick-contact position $2\bar{s}_r/L_r$ (black lines), as a function of the actual tube stretch ε (in log scale). The results are calculated for the same parameters of Fig. 6a2, but for a minimum and maximum friction coefficient 0.1 and 1.0, respectively, and for a single tube actuation cycle starting with an initial rod position $x_{r0}/L_r = 50$. Eight values of dimensionless square frequency numbers $\tilde{m}$ are adopted (see figures): (A) and (B) are in the pre-antiresonance regime, (C) in the post-antiresonance, (D) and (E) in the pre-resonance, (F) is close to resonance, and (G) and (H) are in the post-resonance regime. The arrows over the curves show the time flow along the triangular actuation cycle, from $\varepsilon = 0$ to ε_0 (tube stretch) and back to $\varepsilon = 0$ (tube release). The generic solid line is adopted when the rod-tube contact occurs under stick-slip, whereas the dashed line is for the ballistic case (full sliding). We stress that $[\Delta \dot{x}_r(\varepsilon)]_{NLinM}/[2\varepsilon_0\omega x_{r0}]$ is not proportional to the time derivative of $[\Delta x_r(\varepsilon)]_{NLinM}/[\Delta x_r(\varepsilon)]_{LinM}$.

We observe that the ballistic stage, where full sliding contact occurs between rod and tube, is generically occurring during the beginning of both tube elongation and release stages. In particular the actuation-stretch extension over which the full sliding occurs is negligible for relatively small values of excitation frequencies (case (A)), whereas it expands (predominantly in the release stage) at increasing frequencies (see e.g. (E)) up to dominate the whole actuation release-stage (F) and both elongation- and release-stage in the post-resonance regime (H).

Supplementary Figure 12 | Numerical results. Ratcheting normalized displacement $[\Delta x_r(\varepsilon)]_{NLinM}/[\Delta x_r(\varepsilon)]_{LinM}$ (black lines) and velocity $[\Delta \dot{x}_r(\varepsilon)]_{NLinM}/[2\varepsilon_0\omega x_{r0}]$ (blue lines) as a function of the actual tube stretch ε (in log scale). The results are for the same parameters of Fig. 6a₂, but for a minimum and maximum friction coefficient 0.1 and 1.0, respectively, and for a single tube actuation cycle starting with an initial rod position $x_{r0}/L_r = 50$. Eight values of

dimensionless square frequency numbers $\tilde{m}$ are adopted (see figures): (A) and (B) are in the pre-antiresonance regime, (C) in the post-antiresonance, (D) and (E) in the pre-resonance, (F) is close to resonance, and (G) and (H) are in the post-resonance regime. The arrows over the curves show the time flow along the triangular actuation cycle, from $\varepsilon = 0$ to ε_0 (tube stretch) and back to $\varepsilon = 0$ (tube release). The generic solid line is adopted when the rod-tube contact occurs under stick-slip, whereas the dashed line is for the ballistic case (full sliding).

The resonant behavior is physically obtained when no stick contact occurs during the tube release stage (so that the rod can slide forward despite of the tube receding motion), whereas a certain amount of stick-contact is preserved during the final stage of the tube elongation. The latter is needed to charge the rod in term of kinetic energy, which will then be converted in dissipation during the tube release to perform the displacement. This phenomenon can thus be expected to be intensified by increasing the degree of frictional anisotropy, confirming the results of Supplementary Fig. 11.

In Supplementary Fig. 13 the rod stick-contact position $2\bar{s}_r/L_r$ is shown as a function of the actual tube stretch ε (in log scale), for the same parameters of Supplementary Fig. 12. The full sliding (ballistic) regimes occurs for $2\bar{s}_r/L_r = -1$, whereas the stick-sliding contact occurs otherwise, with a rod stick contact position which differs depending of the tube actuation stage. We note that $2\bar{s}_r/L_r$ is always negative during the ballistic regime, independently of the tube actuation stage. This can be understood as follows. During the very beginning of the tube elongation stage, the rod undergoes negative (with respect to the reference system) f_{min} -friction forces for $s_r < \bar{s}_r$, and positive f_{max} -friction forces for $s_r > \bar{s}_r$, leading for equilibrium to $\bar{s}_r > 0$ (if $f_{max} > f_{min}$). If the inertia term is added to the previous equilibrium scheme (equivalent to sum up negative apparent inertial forces), the $\bar{s}_r$ point must decrease, limiting to the value $2\bar{s}_r/L_r = -1$. During the very beginning of the tube release stage, the rod undergoes positive (with respect to the reference system) f_{max} -friction forces for $s_r < \bar{s}_r$, and negative f_{min} -friction forces for $s_r > \bar{s}_r$, leading for equilibrium to $\bar{s}_r < 0$ (again if $f_{max} > f_{min}$). If the inertia term is added to the previous equilibrium scheme, considering that now the rod has to decelerate from its initial velocity conserved from the tube actuation-cycle reversal point (equivalent to sum up positive apparent inertial forces), the $\bar{s}_r$ point must again decrease, limiting to the value $2\bar{s}_r/L_r = -1$.

Supplementary Figure 13 | Numerical results. The rod stick-contact position $2\bar{s}_r/L_r$ (black lines), as a function of the actual tube stretch ε (in log scale). The results are calculated for the same parameters of Supplementary Fig. 12.

Cargo with low-frequency ratchet motion

We now analyse the inertialess ratchet motion during cargo operation. We first consider the case where the cargo is a constant load, see e.g. the experiment shown in Fig. 6b₄. In this case, we use again (S2.9), (S2.10.1) and (S2.10.2). We define here $b = \mu c_\varepsilon \sigma_0^{-1}$. From (S2.12.1) we get:

$$\frac{\partial \varepsilon_p(t)}{\partial t} > 0: \quad \tilde{x}_r(\beta) e^{-\beta} = x_{r0} + \int_{e^{-\beta}}^1 dx \left(\frac{\tilde{F}}{(1-b)} \frac{x}{x + \frac{b}{1-b}} + \frac{\varepsilon_f}{2} \right), \quad (\text{S2.14.1})$$

which, upon integration

$$\tilde{x}_r(\beta) e^{-\beta} = x_{r0} + \tilde{F} \left(\frac{1-e^{-\beta}}{(1-b)} + \frac{b \log(b+e^{-\beta}(1-b))}{(1-b)^2} \right) + \frac{\varepsilon_f}{2} (1 - e^{-\beta}). \quad (\text{S2.14.2})$$

From (S2.12.2):

$$\frac{\partial \varepsilon_p(t)}{\partial t} < 0: \quad \tilde{x}_r(\beta) e^{-\beta} = x_{r,max} e^{-\beta_{max}} + \int_{e^{-\beta_{max}}}^{e^{-\beta}} dx \left(\frac{\tilde{F}}{(1-b)} \frac{x}{x + \frac{b}{1-b}} + \frac{\varepsilon_f}{2} \right), \quad (\text{S2.14.3})$$

which, upon integration

$$\begin{aligned} \tilde{x}_r(\beta) e^{-\beta} &= x_{r,max} e^{-\beta_{max}} \\ &+ \tilde{F} \frac{(-1+b)(e^{-\beta_{max}} - e^{-\beta}) + b \log \left[\frac{b+e^{-\beta_{max}}(1-b)}{b+e^{-\beta}(1-b)} \right]}{(-1+b)^2} + \frac{\varepsilon_f}{2} (e^{-\beta} - e^{-\beta_{max}}). \end{aligned} \quad (\text{S2.14.4})$$

From (S2.14.2) and (S2.14.4) we get the ratcheting step

$$\Delta \tilde{x}_r = \frac{\varepsilon_0}{1+\varepsilon_0} \left(\varepsilon_f - \frac{F}{2\pi R \sigma_0 f_0 L_r} \frac{(1-b) + b \frac{1+\varepsilon_0}{\varepsilon_0} \log \left(b + \frac{1}{1+\varepsilon_0} (1-b) \right)}{(1-b)^2} \right). \quad (\text{S2.15})$$

We observe that, by imposing in (S2.9) $\tilde{s}_r(\beta) \geq -1/2$, the ratchet motion under quasi-static loading will be allowed for

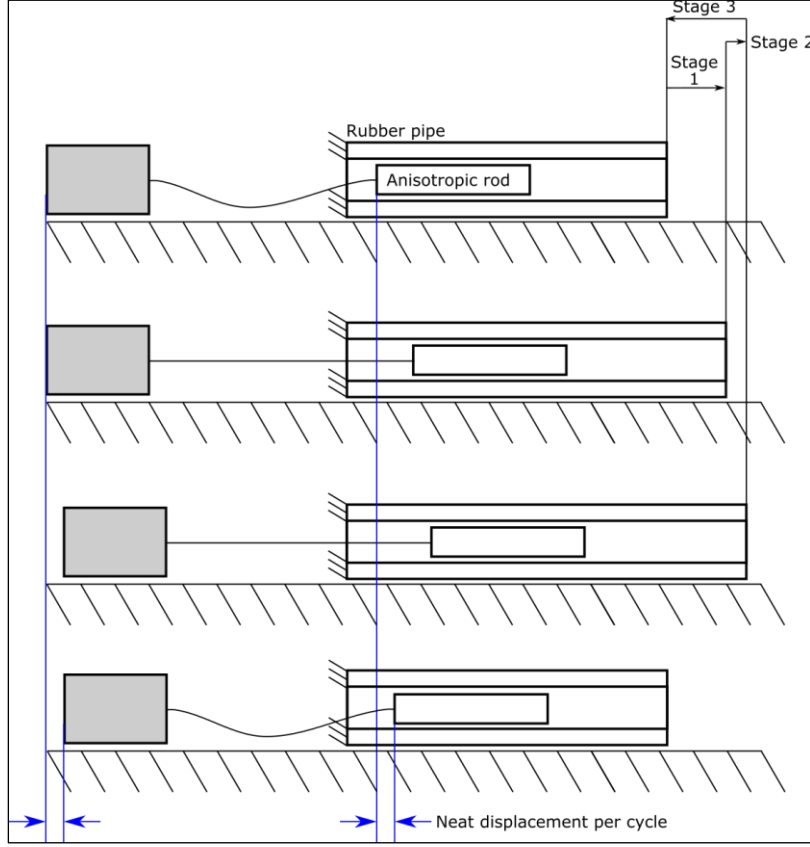
$$f_{\max} \geq \frac{F}{2\pi R \sigma_0 L_r}, \quad (\text{S2.16})$$

which coincides, as must be expected, with the static equilibrium condition. Observe that we can get (S2.16) from (S2.12.1), (S2.12.2) and (S2.9), by simply considering a stretch independent $\tilde{\sigma}$ given by

$$\tilde{\sigma} \cong \frac{1-b}{1 + \frac{b}{1-b} \frac{1+\varepsilon_0}{\varepsilon_0} \log \left(b + \frac{1}{1+\varepsilon_0} (1-b) \right)}. \quad (\text{S2.17})$$

Another application of the theory is for ratchet motion with stepwise-varying cargo force, such as in the case where the cargo is given by a frictional force $\tilde{F}_T$, see the experiment reported in Fig. 6b₂ and Fig. 6b₃, and the schematic of Fig. S14.

In particular, in Supplementary Fig. S14 we show the three stages characterizing the actuation cycle. Stage 1 and 2 occur under tube elongation, and in particular during stage 1 no cargo takes place (the rope is only straightened), whereas during the stage 2 the payload is displaced and the cargo motion occurs. During stage 3, the tube is released together with the connecting rope. Here we assume that, for the generic actuation cycle, the final configuration is approximately equal to the initial configuration, i.e. the rod and the payload are at the same distance in the initial and final configuration.



Supplementary Figure 14 | Schematic representation of the actuation mechanism under frictional cargo.

This is actually a good approximation if the ratcheting step does not vary abruptly between two consecutive cycles, as is the case in all the calculations we have reported in this research paper. With this assumption, we solve (S2.12.1), (S2.12.2) and (S2.9) by adopting $\tilde{F} = 0$ during stage 1, $\tilde{F} = \tilde{F}_T$ during stage 2 and again $\tilde{F} = 0$ during stage 3, as well as we impose the motion step during stage 2 to be equal to the neat ratcheting step. After some algebra, we find that the transition between stage 1 and stage 2 occurs at $\bar{\beta}$

$$\bar{\beta} = \log \left(\frac{2\tilde{F}_T(e^{\beta_{\max}} - 1)}{a_1 a_2} \right), \quad (\text{S2.18})$$

where

$$a_1 = (1 - e^{-\beta_{\max}})(\varepsilon_f - \tilde{F}_T) - e^{\beta_{\max}} \left(\frac{\varepsilon_f}{2} + x_{r0} \right)$$

$$a_2 = \sqrt{1 - \frac{4\tilde{F}_T(e^{\beta_{\max}} - 1) \left(\frac{\varepsilon_f}{2} + x_{r0} \right)}{\left((1 - e^{-\beta_{\max}})(\varepsilon_f - \tilde{F}_T) - e^{\beta_{\max}} \left(\frac{\varepsilon_f}{2} + x_{r0} \right) \right)^2} - 1}.$$

The ratcheting steps is

$$\Delta x_r / L_r = \frac{\varepsilon_0}{1 + \varepsilon_0} \varepsilon_f + \left(\frac{1}{1 + \varepsilon_0} - e^{-\bar{\beta}} \right) \tilde{F}_T. \quad (\text{S2.19})$$

Here $F_T = \mu m g / \tilde{\sigma}$, where μ is the sliding friction coefficient between block and substrate and the stretch-independent $\tilde{\sigma}$ is given by (S2.17).

S2.3 – Advanced theory of ratchet actuation mechanism (AdvM)

In this section we assume the friction force independent of the sliding speed, and linearly dependent with the normal load (Coulomb friction model), whereas we relax the assumptions of uncoupling between longitudinal and radial deformation of the actuation tube. The geometric details of the contact are reported in Section S1, whereas the schematic of the macro-contact geometry is given in Supplementary Fig. 1. In the following, we derive the tube deformation model.

We assume the tube deformation to fall within the linear elasticity theory. We further simplify the model by assuming the tube material to be incompressible, leading to $\nabla \cdot \mathbf{u} = 0$, where $\mathbf{u}$ is the tube displacement field. We consider cylindrical coordinates (r , θ and z , where z coincides with the rod and tube axis), and the axial-symmetric case to hold, leading to $\partial \cdot / \partial \theta = u_\theta = \sigma_{\theta r} = \sigma_{\theta z} = 0$. Here σ is the stress tensor. Furthermore, we consider the tube to be thin-walled, and subjected to an external driving longitudinal deformation. The thin wall assumption can be adopted to expand the longitudinal deformation field ε_{zz} (and σ_{zz}) at linear order in the tube cross thickness, i.e. $\varepsilon_{zz}(r, z) \approx \varepsilon_{zz}^0(z) + \varepsilon_{zz}^1(z)(r - R_m)$, where R_m is the average tube radius. Hence, considering that under isochoric deformation, the following Lamé relation applies (μ is the shear modulus)

$$\sigma_{rr} \approx 2\mu \partial u_r / \partial r,$$

by integrating $\nabla \cdot \mathbf{u} = 0$ with boundary conditions (BCs) $\sigma_{rr}(r = R_o, z) = 0$ (stress-free outer surface), and by imposing $u_r(r = R_i, z) = u_0(z)$ (interference fit displacement at inner surface) one finds the following radial displacement field

$$u_r(\rho, z) = \varepsilon_{zz}^0(z) R_m \frac{(16\rho^3(\varepsilon_R^2 + 4) - 4\rho^2(\varepsilon_R(\varepsilon_R + 18) + 12) + 24) + (3\varepsilon_R + 2)(\varepsilon_R^2 - 4)^2)}{8\rho\varepsilon_R(\varepsilon_R(\varepsilon_R + 12) + 12)} \\ + u_0(z) \frac{4(2 - \varepsilon_R)(8\rho^3 - 12\rho^2 + (\varepsilon_R + 2)^2(2\varepsilon_R + 1))}{8\rho\varepsilon_R(\varepsilon_R(\varepsilon_R + 12) + 12)},$$

where $\rho = \frac{r}{R_m}$, $2R_m = R_i + R_o$, and $R_m \varepsilon_R = R_o - R_i$.

We now calculate the radial stress on the inner side of the cylindrical crown $\sigma_{rr}(R_i, z)$, which we define $\sigma_{rr}(z)$ for simplicity of notation

$$\sigma_{rr}(z) \approx -\mu \varepsilon_{zz}^0(z) \frac{8(\varepsilon_R^3 + \varepsilon_R^2 - 12)}{(\varepsilon_R - 2)(\varepsilon_R(\varepsilon_R + 12) + 12)} - \mu \frac{u_0(z)}{R_m} \frac{16(\varepsilon_R^2 + 6)}{(2 - \varepsilon_R)(\varepsilon_R(\varepsilon_R + 12) + 12)}. \quad (\text{S2.20})$$

By defining $\varepsilon_{zz}(z) = \varepsilon_{zz}^0(z)$ and the following geometrical parameters

$$c_\varepsilon = \frac{8(\varepsilon_R^3 + \varepsilon_R^2 - 12)\alpha}{(\varepsilon_R - 2)(\varepsilon_R(\varepsilon_R + 12) + 12)} \\ c_u = \frac{16(\varepsilon_R^2 + 6)}{(2 - \varepsilon_R)(\varepsilon_R(\varepsilon_R + 12) + 12)} \\ e_0(z) = \frac{u_0(z)}{R_m} \frac{c_u}{c_\varepsilon},$$

(S2.20) becomes

$$\sigma_{rr}(z) = -\mu c_\varepsilon [\varepsilon_{zz}(z) + e_0(z)]. \quad (\text{S2.21})$$

In c_ε , the parameter α is the normalized true contact area, which we include in (S2.21) in order to take into account the tube/rod true contact stiffness in the calculation of the tube/rod contact stresses $\sigma_{rr}(z)$.

We now apply the momentum equation along the z -axis. We consider $\sigma_{zz} \approx 2\mu \partial u_z / \partial z$ as well as

the friction shear stress given by $\sigma_{rz}(z) = -f(z)\sigma_{rr}(z)$. Note that we have not used the Heaviside step function in the friction shear stress (no adhesion is considered here), since $\sigma_{rr}(z)$ is always compressive in this study, given that $\varepsilon_{zz}(z)$ and $e_0(z)$ are positive valued. Thus, the momentum equation reads

$$2\mu \frac{\partial \varepsilon_{zz}(z)}{\partial z} dz d\theta \Delta_R R_m - f(z)\sigma_{rr}(z) d\theta R_i dz = 0. \quad (\text{S2.22})$$

Substituting (S2.21) in (S2.22)

$$0 = \frac{\partial \varepsilon_{zz}(z)}{\partial z} + f(z)c[\varepsilon_{zz}(z) + e_0(z)] \quad (\text{S2.23})$$

where

$$c = \frac{c_\varepsilon R_i}{2\varepsilon_R R_m} \frac{1}{R_m}.$$

We now consider $e_0(z)$ not to be a function of the cylinder axis coordinate z , i.e. the interference fit is assumed constant along the contact interface. In this case, (S2.23) admits the following solution

$$\bar{\varepsilon}_p(s_p) = \bar{\varepsilon}_0 \exp \left[-c \int_{s_{p0}}^{s_p} ds f(s) \right], \quad (\text{S2.24})$$

where we have defined $\bar{\varepsilon}_p(s_p) = \varepsilon_p(s_p) + e_0$ (thus, $\bar{\varepsilon}_0 = \varepsilon_0 + e_0$). Note in (S2.24) that we have changed the variable z with the more familiar material point (or section, given the axial-symmetry of the formulation) variable s_p . s_{p0} is simply the tube material section where the contact with the internal rod starts from. In such section, the tube stretch is equal to the externally applied stretch ε_0 .

Assuming Coulomb friction, by integrating (S2.24) and by applying the condition $\bar{\varepsilon}_p(s_{p1}) = \bar{\varepsilon}_0$, where s_{p1} is the tube material section where the contact with the internal rod ends, one has the following tube longitudinal deformation laws

$$\frac{\bar{\varepsilon}_p(s_p)}{\bar{\varepsilon}_0} = \begin{cases} e^{-cf_-(s_p-s_{p0})} & s_p \leq \bar{s}_p \\ e^{-cf_+(s_p-s_{p1})} & s_p \geq \bar{s}_p \end{cases}. \quad (\text{S2.25})$$

In (S2.25) $\bar{s}_p$ is the tube material section at which the inversion of the sign of the friction coefficient occurs, whereas f_- and f_+ are the friction coefficients for $s_p \leq \bar{s}_p$ and $s_p > \bar{s}_p$, respectively. We rephrase (S2.25) in the following

$$\frac{\bar{\varepsilon}_p(\Delta s_p, s_{p0})}{\bar{\varepsilon}_0} = \begin{cases} e^{-cf_-(\Delta s_p)} & \Delta s_p \leq \Delta \bar{s}_p \\ e^{-cf_+(\Delta s_p - \Delta s_{p1})} & \Delta s_p > \Delta \bar{s}_p \end{cases}. \quad (\text{S2.26})$$

where we have simply defined relative material section coordinates $\Delta s_p = s_p - s_{p0}$, $\Delta s_{p1} = s_{p1} - s_{p0}$ and $\Delta \bar{s}_p = \bar{s}_p - s_{p0}$. In (S2.26) s_{p0} , $\Delta \bar{s}_p$ and Δs_{p1} are unknowns. $\Delta \bar{s}_p$ can be calculated imposing the C^0 continuity of $\bar{\varepsilon}_p$ at $\Delta s_p = \Delta \bar{s}_p$, leading to

$$\Delta \bar{s}_p = \frac{f_+}{f_+ - f_-} \Delta s_{p1}, \quad (\text{S2.27})$$

We need a further equation to calculate Δs_{p1} . We note that the deformed tube length must be equal to the rod length L_r . We recall from Sec. S2.1 that the generic axial position of the tube material section s_p reads

$$x_p(s_p, t) = s_p + \int_0^{s_p} ds'_p \varepsilon_p(s'_p, t), \quad (\text{S2.28})$$

leading to $L_r = x_p(s_{p1}, t) - x_p(s_{p0}, t)$. Upon integration of (S2.28) one gets

$$\begin{aligned}
L_r &= s_{p1} - s_{p0} + \int_{s_{p0}}^{s_{p1}} ds \varepsilon_p(s, t) = \Delta s_{p1} + \int_{s_{p0}}^{s_{p1}} ds [\bar{\varepsilon}_p(s, t) - e_0] \\
&= \Delta s_{p1}(1 - e_0) + \int_0^{\Delta s_{p1}} d\Delta s \bar{\varepsilon}_p(\Delta s, t) \\
&= \Delta s_{p1}(1 - e_0) + \bar{\varepsilon}_0 \frac{e^{-cf_- \Delta s_{p1}} - 1}{-cf_-} + \bar{\varepsilon}_0 \frac{1 - e^{-cf_+(\Delta s_{p1} - \Delta s_{p1})}}{-cf_+} \\
&= \Delta s_{p1}(1 - e_0) + \bar{\varepsilon}_0 \frac{e^{-c \frac{f_+ + f_-}{f_+ - f_-} \Delta s_{p1}} - 1}{-cf_-} + \bar{\varepsilon}_0 \frac{e^{-c \frac{f_+ + f_-}{f_+ - f_-} \Delta s_{p1}} - 1}{cf_+}
\end{aligned}$$

resulting, after manipulation, in the following elongation constraint

$$\frac{L_r - \Delta s_{p1}(1 - e_0)}{\varepsilon_0 + e_0} = \left(\frac{1}{cf_+} - \frac{1}{cf_-} \right) \left(e^{-c \frac{f_+ + f_-}{f_+ - f_-} \Delta s_{p1}} - 1 \right). \quad (S2.29)$$

Finally, s_{p0} can be determined by cinematically linking the tube with the rod. To do so, we impose the following

$$x_r(s_r = -L_r/2, t) = x_p(s_{p0}, t), \quad (S2.30)$$

i.e. we say that the tube material section position at s_{p0} coincides with the rod material section position at $s_r = -L_r/2$. We recall again that the generic axial position of the rod material section s_r reads

$$x_r(s_r, t) = x_r(t) + s_r; \quad (S2.31)$$

by using (S2.30) with (S2.28) and (S2.31) one obtains

$$s_{p0} = \frac{x_r(t) - \frac{L_r}{2}}{(1 + \varepsilon_0)}. \quad (S2.32)$$

The motion law can now be determined by assuming the no-slip condition occurring in $\bar{s}_p$ (where the inversion in the friction forces occurs, e.g. we neglect rod inertia in the following)

$$\left[\frac{\partial x_p(s, t)}{\partial t} \right]_{s=\bar{s}_p} = \frac{\partial x_r(t)}{\partial t}. \quad (S2.33)$$

However, from (S2.28) one has that the sliding speed of the generic tube material section s_p reads

$$\frac{\partial x_p(s_p, t)}{\partial t} = \int_0^{s_p} ds_p' \frac{\partial \varepsilon_p(s_p', t)}{\partial t} = \dot{\varepsilon}_0(t) \left(s_{p0} + \int_{s_{p0}}^{s_p} ds \frac{\partial \bar{\varepsilon}_p(s, t)}{\partial \bar{\varepsilon}_0} \right). \quad (S2.34)$$

By substituting (S2.34) in (S2.33), and by using (S2.26) and (S2.32), the following motion law is obtained

$$\frac{\partial x_r(t)}{\dot{\varepsilon}_0(t) \partial t} = \frac{\partial x_r(t)}{\partial \varepsilon_0(t)} = \frac{x_r(t) - \frac{L_r}{2}}{1 + \varepsilon_0(t)} + \frac{1 - e^{-c \frac{f_+ + f_-}{f_+ - f_-} \Delta s_{p1}}}{cf_-}, \quad (S2.35)$$

which has to be solved with (S2.27) and (S2.29) in the variables $x_r(t)$, $\Delta \bar{s}_p(t)$ and $\Delta s_{p1}(t)$.

Observe first that for small value of interface friction, which results in a relatively small degree of coupling between tube radial deformation and rod kinematics, we can expand at first order in

$\frac{f_+ + f_-}{f_+ - f_-}$ the following term

$$\frac{1 - e^{-c \frac{f_+ + f_-}{f_+ - f_-} \Delta s_{p1}}}{cf_-} \approx \frac{f_+}{f_+ - f_-} \Delta s_{p1} + o \left[\left(\frac{f_+ + f_-}{f_+ - f_-} \right)^2 \right],$$

so that (S2.35) becomes

$$\frac{\partial x_r(t)}{\dot{\varepsilon}_0(t) \partial t} \approx \frac{x_r(t) - \frac{L_r}{2}}{1 + \varepsilon_0(t)} + \frac{f_+}{f_+ - f_-} \Delta s_{p1}. \quad (\text{S2.36})$$

Again at first order in $\frac{f_+ f_-}{f_+ - f_-}$ (S2.29) becomes

$$L_r \approx \Delta s_{p1}(t)(1 + \varepsilon_0(t)) + o\left[\left(\frac{f_+ f_-}{f_+ - f_-}\right)^2\right]. \quad (\text{S2.37})$$

Substituting Δs_{p1} of (S2.37) in (S2.36) we get

$$\frac{\partial x_r(t)}{\dot{\varepsilon}_0(t) \partial t} \approx \frac{x_r(t) + \frac{f_+ + f_- - L_r}{f_+ - f_-}}{1 + \varepsilon_0(t)},$$

which is exactly (S2.7).

The ratcheting dynamics occurring during the stretching/un-stretching actuation can be thus determined by solving the algebraic-differential system of Eqs. (S2.27), (S2.29) and (S2.35) in the variables $x_r(t)$, $\Delta s_{p1}(t)$ and $\Delta \bar{s}_p(t)$.

In the main paper we have experimentally investigated the performances (ratcheting motion and cargo properties) of the bio-mimetic actuator and compared those with the results predicted by the theory. The latter, in the previous sections, has been applied to an extensive investigation about how the ratchet motion is affected by the actuation and contact parameters, such as tube stretch, tube/rod friction and degree of frictional anisotropy, as well as and tube cross-section. We have also described how ratchet is affected by the actuation frequency, the latter being responsible for the actuation dynamics to run in the linear, anti-resonant, resonant and inertial regimes (see e.g. Supplementary Fig. 11). However, in the following we show that, even by neglecting the inertia contributions (thus, for low frequency actuation cycles), very many interesting phenomena still occur during the ratchet dynamics.

We start from observing that, by neglecting inertia, (S2.6) shows that the ratchet step per cycle is simply given by (also derived in Ref. [1])

$$\Delta x_r = L_r \frac{f_{\max} - f_{\min}}{f_{\max} + f_{\min}} \frac{\varepsilon_0}{1 + \varepsilon_0} \quad (\text{S2.6})$$

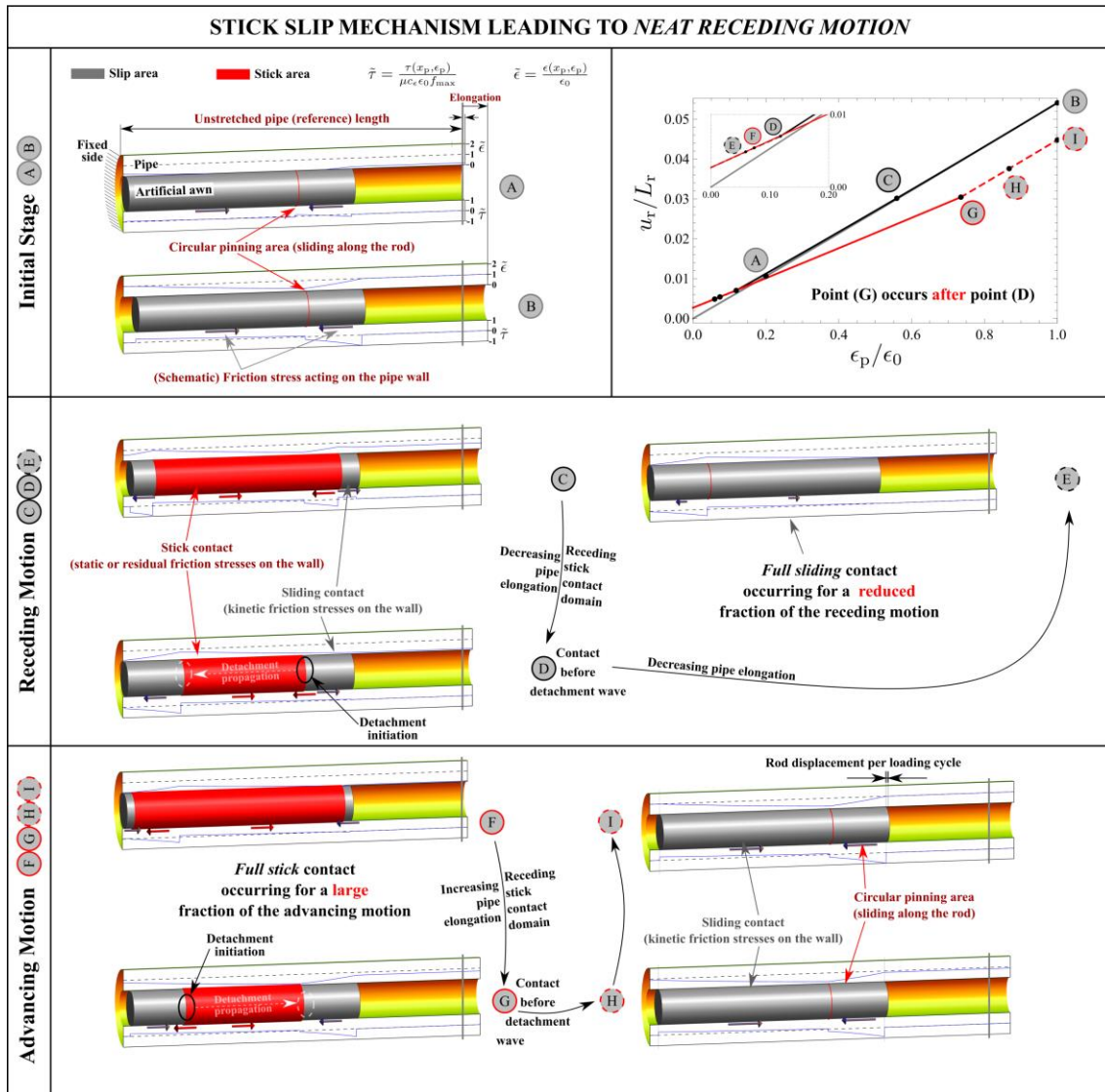
where Δx_r is the sliding step per actuation cycle, L_r is the awn rod length, ε_0 is the actuation stretch and $f_{\max}$ ($f_{\min}$) is the maximum (minimum) tube/awn wall friction. (S2.6) is thus valid in the inertialess regime and with no cargo. It shows no dependence on the tube radius and on the initial geometrical interference fit between tube internal wall and average external cylindrical surface of the awn, as well as no availability of receding ratchet motion. This is a consequence of the model simplifications, such as the uncoupling between radial and axial interface deformation dynamics. In the following, instead, we show the existence of three different motion regimes, corresponding to an advancing (see Fig. 3, described in the main manuscript), receding (Supplementary Fig. 15) and sticky (Supplementary Fig. 16) ratchet dynamics.

Neat receding ratchet motion

In Supplementary Fig. 15 we show the case opposite to Fig. 3, characterized by a point (G-advancing detachment wave) occurring for an external stretch larger than for point (D-receding detachment wave). This results in a ratchet motion characterized by a neat receding (or negative) displacement per actuation cycle, as shown in the displacement graph. The numerical results have

been obtained with the same physical parameter adopted for the case of Fig. 3, but for a larger applied actuation stretch $\epsilon_0 = 0.1$. Therefore, for the adopted set of contact parameters, increasing the actuation stretch determines a reduction of the displacement step per cycle up to a reversal of the ratcheting motion, in qualitative agreement with the experimental results provided in Supplementary Fig. 8a3. In Section S3, we provide a more comprehensive numerical study on the role of contact parameters on the neat ratchet motion. In particular, we report H -maps ($H = \Delta x_r / (\epsilon_0 L_r)$) as a function of the friction anisotropy, for a several parameters, such as the actuation stretch, geometrical interference fit and tube cross section.

We note in Supplementary Fig. 15 that the same contact mechanisms is reported for the case of advancing ratchet motion (Fig. 3), however, occurring with a different magnitude in term of stretch dependence and residual stresses. Indeed, the residual stresses occurring in Supplementary Fig. 15-(F) are smaller in magnitude than the corresponding stresses in Fig. 3-(F). This is a result of the larger actuation stretch ϵ_0 adopted in the case of Supplementary Fig. 15.



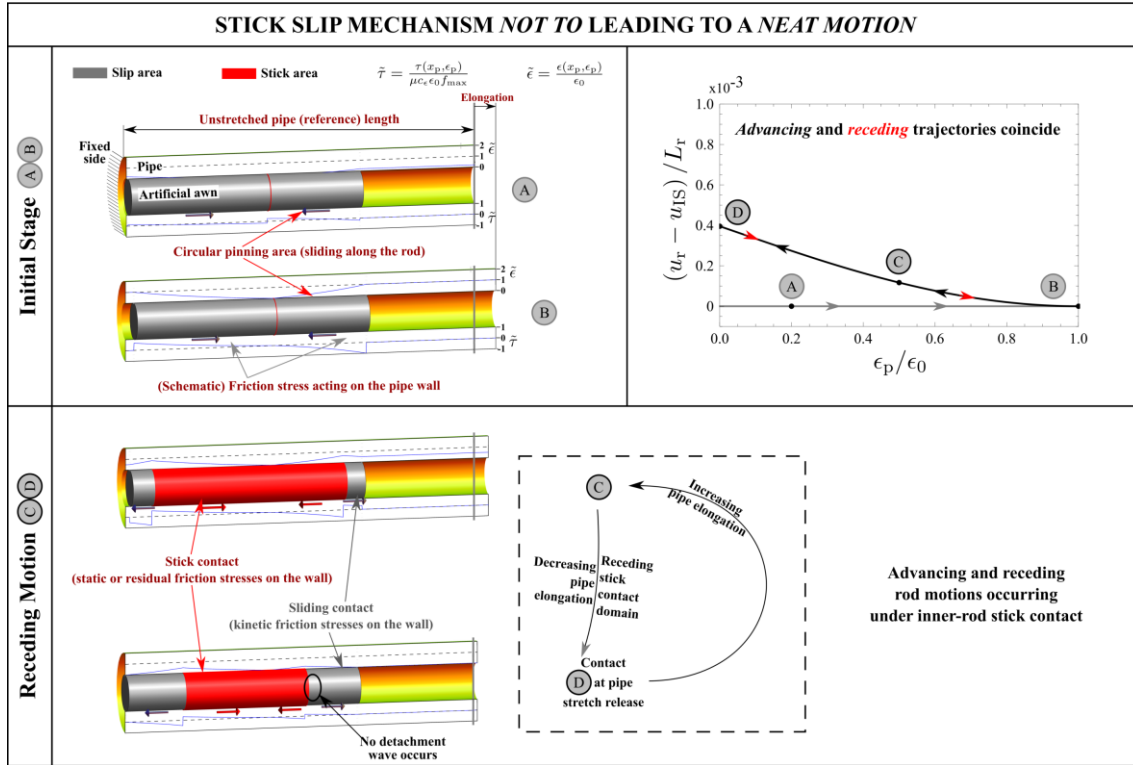
Supplementary Figure 15 | Numerical results. Ratchet mechanism leading to a receding motion.

A further consequence of the larger residual stresses is that, during the generic actuation cycle (point

(B) to (I) in the figure), the tube stretching stage occurring under stick contact (corresponding, approximately, to points (F) to (G) in the figure) occurs for a larger portion of stretching actuation before the occurrence of the detachment wave (G), causing the neat receding motion.

Sticky motion leading to no ratcheting

In Fig. 3 and Supplementary Fig. 15 we have shown two possible cases of ratchet motion, leading to advancing and receding displacement per cycle, respectively. In both cases, despite of the different ratcheting outcomes, both tube stretch and release actuation are typically characterized by stick and sliding sub-stages. Indeed, after any actuation reversal, a sticky contact stage occurs followed by a sliding contact stage. In particular, a detachment wave allows the sticky contact to become sliding under an infinitesimal variation of actuation stretch, and leading to the formation of a new tube/rod pinning circle. We now keep the same parameters of Supplementary Fig. 15, but with a different minimum friction coefficient, $f_{\min} = 0.04$ (doubled with respect to the value in Supplementary Fig. 15).



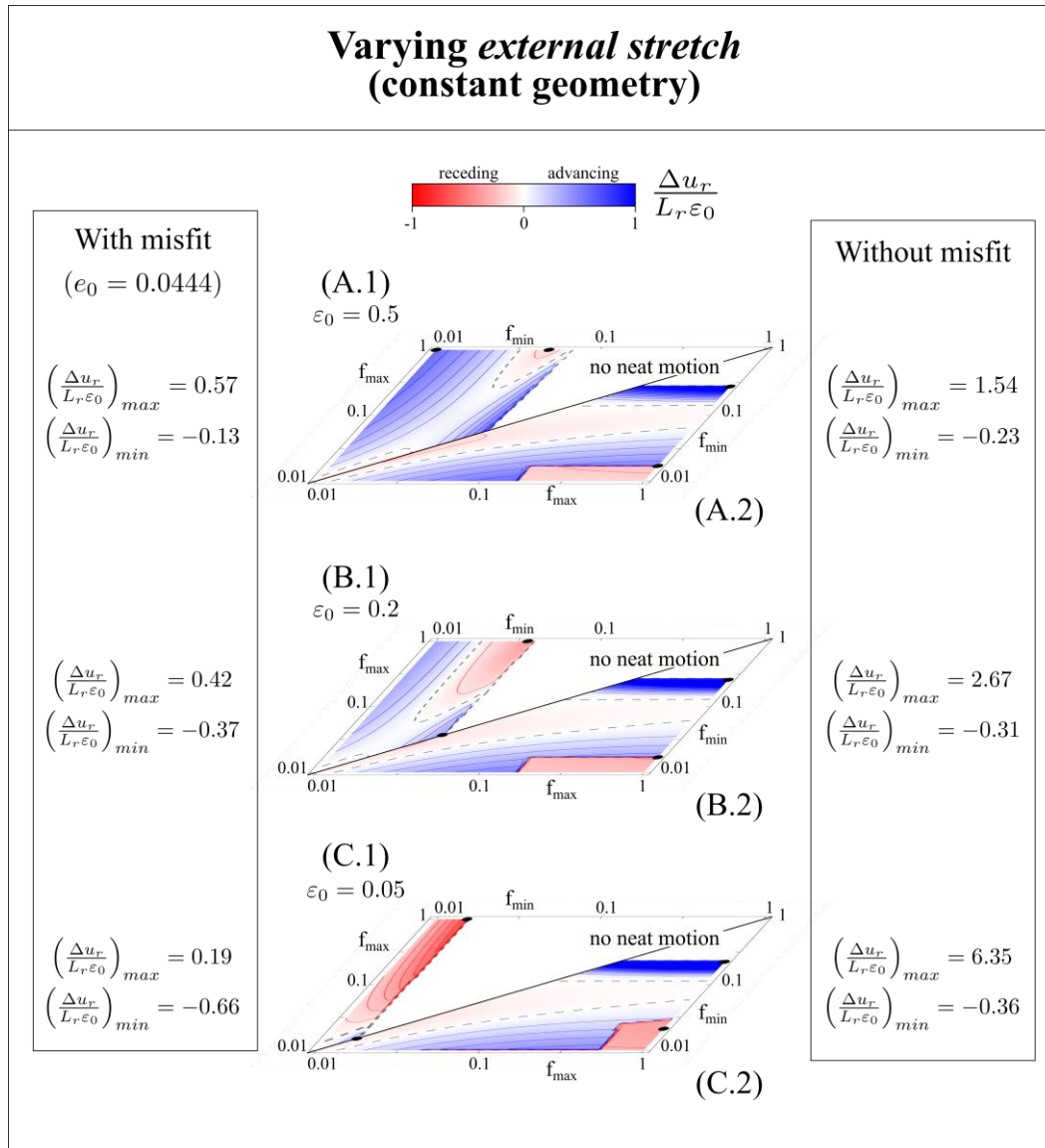
Supplementary Figure 16 | Numerical results. Ratcheting mechanism not leading to a clear motion.

Differently from before, for the sake of readability, the top-right graph of Supplementary Fig. 16 shows the dimensionless rod displacement u_r/L_r relative to the dimensionless rod displacement in the initial stage u_{IS}/L_r , as a function of the actual stretch ratio ϵ_p/ϵ_0 . Thus, the grey curve is a straight horizontal line, ending with the stretch reversal point (B). At point (B), the generic release/stretch actuation cycle begins. Interestingly, the model predicts the occurrence of purely sticky contact stages, i.e. no detachment wave occurs and, consequently, no subsequent sliding contact stage. Indeed, the inner bio-mimetic awn is always sticking along a fraction of the internal tube contact surface. The same considerations apply after the actuation reversal point (D) is reached,

where the tube stretch actuation begins. As a consequence, both release and stretch displacement curves are superposed, so that no neat ratchet motion can be generated. This is shown in terms of black and red arrows in the rod displacement graph.

S3 Ratcheting maps: A parametric study

A parameter of interest describing the efficiency of a ratchet system is the transmission ratio $H = \Delta x_r / (L_r \varepsilon_0)$. As before, Δx_r is the neat rod displacement per actuation cycle, whereas L_r is the rod length and ε_0 is the tube longitudinal deformation. In the following we provide H -maps as a function of the degree of friction anisotropy, at varying actuation and contact conditions. The base system is characterized by $\varepsilon_R = 0.75$ and $R_m/L_r = 0.027$ (similar to the experimental system reported in the main paper). In particular, in Supplementary Fig. 17 we show the transmission ratio H for three values of actuation stretch, $\varepsilon_0 = 0.05, 0.2$ and 0.5 . The left map corresponds to the case where an initial interference fit $u_0 \neq 0$ is applied between rod and tube (in particular, $e_0 = 0.0444$ is adopted), whereas the right side is for an ideal $u_0 = 0$. The related left and right panels summarize the minimum and maximum H values encountered in the parametric study, whereas their location is shown with black circles in the H -maps. Note interestingly that the largest H ($H = 6.35$) is obtained for the smallest actuation stretch, in the case where no interference fit is applied (case C2), whereas the smallest positive H is obtained for the same conditions but with interference fit (case C1). Furthermore, we note that, for the case with interference fit (C1), most of the H -map shows a zero or even a neat receding ratcheting motion, and the largest positive H is predicted in absence of friction anisotropy. Again, for the case with interference fit, an increasing tube stretching determines, instead, the maximum H to occur in correspondence of the largest friction anisotropy.



Supplementary Figure 17 | Actuation cycle response for a ratchet system as a function of the friction anisotropy, at varying actuation tube stretch ε_0 . The system geometry is kept constant. The left and right side is with and without initial rod/tube mechanical interference, respectively.

The physical origin of such strongly non-linear behaviour has to be ascribed to the interplay between average friction, frictional anisotropy and initial interference fit in the determination of the sliding and residual interface frictional stresses. Indeed, the initial interference fit provides an initial stress field which can strongly alter the distribution of interfacial stresses, and thus, of stick and slip areas, depending on the amount of tube stretching and frictional anisotropy.

Thus, **for small actuation strain**, the initial interference fit regulates the distribution of interface residual stresses and thus the ratchet dynamics. In particular, the case with interference fit (C.1) shows that residual stresses occur during the whole actuation cycle, leading to sticky contacts for the majority of friction combinations. Nevertheless, for small average friction and small friction anisotropy, the geometric interference works in line with the stretching dynamics to provide a neat advancing ratchet motion, whereas for large friction anisotropy the opposite occurs. In the latter case, indeed, there is always a sticky contact during tube elongation, due to the large residuals

stresses, whereas a partial sticky/sliding contact occurs during tube retraction, finally leading to the neat receding ratchet motion. However, when interference fit is not applied (C.2), the H -map shows that larger friction anisotropy provides larger advancing neat motion, unless for the cases with high average friction, where again sticky ratcheting motion occurs. At **increasing actuation strain**, the H -map obtained with interference fit shows a larger advancing motion area than at small actuation strain, with higher H values. In particular, the maximum H is obtained in correspondence of the larger friction anisotropy (see (A.1)). The case without interference fit (A.2), instead, is not qualitatively affected by the amount of tube stretch, however, the maximum H decreases at increasing actuation stretch.

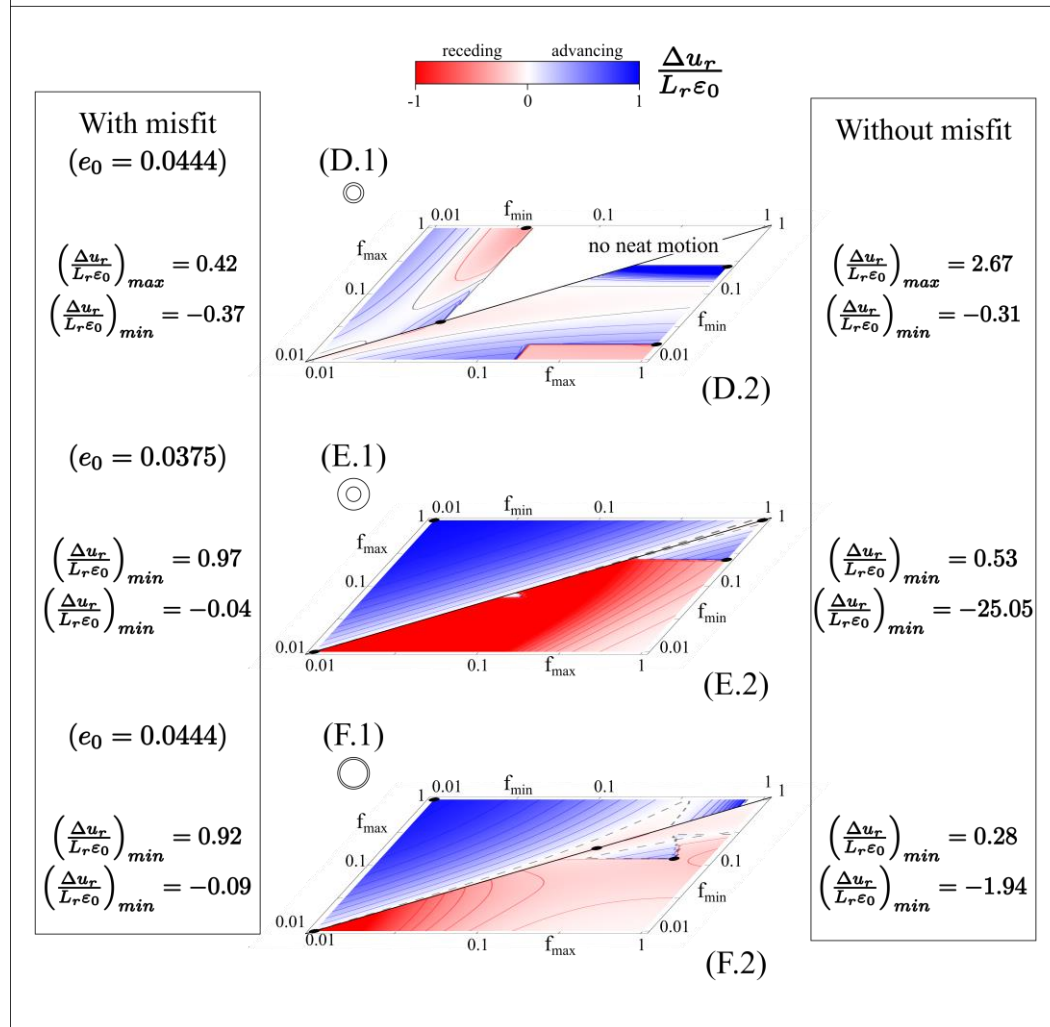
The parametric study above suggests that the dynamics of the bio-mimetic actuator is strongly affected by the initial interference fit between the components. This is due to the set of an initial residual stress field which, provided by the geometrical interference fit, affects the formation of stick/slip domains during the tube stretch/release and thus the ratchet motion itself. We note, however, that since a interference fit cannot be avoided for practical reasons, a large stretch is necessary to provide stable ratcheting H -maps, despite at the cost of a reduced transmission ratio (with respect to the zero interference fit case). Here stability is intended as the capability to keep the (designed) optimum advancing ratchet motion under small variations of the friction coefficients. We suggest, however, that the largest performances would be obtained in the case where both interference fit and tube stretch were controlled during actuation, such as in the case where the anchor were subjected to actuation stretch instead of the tube. This more effective actuation mechanism will be reported in a follow-up paper.

It is also very interesting to investigate the role of the macro-geometry of the system. In particular, in the following we vary the internal and external tube radius, while adopting the same dimensional values of interference fit u_0 . The actuation stretch is fixed at $\varepsilon_0 = 0.2$ (similar to case B.1 and B.2 of Supplementary Fig. 17). In Supplementary Fig. 18 we show the transmission ratio H for the three following geometries

- case D: $\varepsilon_R = 0.746$ and $R_m/L_r = 0.027$ (corresponding to case B.1 and B.2 of Supplementary Fig. 17);
- case E: $\varepsilon_R = 1.82$ and $R_m/L_r = 0.192$; such values correspond to the case where $R_i = [R_i]_{\text{case A}}$ and $R_o = 10[R_o]_{\text{case A}}$;
- case F: $\varepsilon_R = 0.746$ and $R_m/L_r = 0.268$; such values correspond to the case where $R_i = 10[R_i]_{\text{case A}}$ and $R_o = 10[R_o]_{\text{case A}}$.

We observe that the tube geometry has a strong impact on the ratchet dynamics, and in particular increasing the anchor ratio R_m/L_r highly stabilize the ratcheting H -map, when interference fit is applied (see E.1 and F.1), in perfect qualitative agreement with the experimental results of Supplementary Fig. 9b2. On the other side, the case with zero interference fit shows the occurrence of neat receding motion for most of the actuation map, with larger values of receding H for the case of thick walled tube.

Varying the *pipe cross section* (constant external stretch)



Supplementary Figure 18 | Actuation cycle response for a ratchet system as a function of the friction anisotropy, at varying tube cross sections (but constant rod length). The actuation tube stretch ε_0 is kept constant. The left and right side is with and without initial rod/tube mechanical interference, respectively.

References

- (1) Kulić, I. M.; Mani, M.; Mohrbach, H.; Thaokar, R.; Mahadevan, L. Botanical Ratchets. *Proc. R. Soc. B* **2009**, 276, 2243-2247.