
Supplementary information

**Kirigami-inspired parachutes with
programmable reconfiguration**

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Supplementary Materials

Kirigami-inspired parachutes with programmable reconfiguration

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Supplementary Methods

S1. Finite element simulations

To perform our fluid-structure interaction simulations, we use the FIRM framework developed to study flow-induced reconfiguration of flexible structures in a static context (1). The FIRM framework is developed on MATLAB and uses bilinear triangular plate elements based on the Discrete Kirchhoff Triangular plate formulation coupled with the Assumed Natural Deviatoric Strain membrane theory, which are meshed using GMSH. We perform both our traction simulations and our reconfiguration simulations using this framework.

A. Meshing. To mesh the disk with slits, we first model the external disk and add pockets to model the inner radius and the slits. Regarding the cuts, we consider that their ends are rounded with a diameter equal to the width of the cut. Here, we consider a cut width of 0.1 mm for all our simulations. This leads to the geometry illustrated in Fig. S1a for the Design B disk, for example. Rounding the slits allows us to represent reality better, but also limits some potential numerical difficulties due to distorted elements. However, due to the different scales of the structure and the slits' width, we were unable to have a sufficiently fine mesh without any distorted elements while keeping computation times low. For the same disk, by using an element size of 2 mm, we obtain the mesh illustrated in Fig. S1b,c,d. While some elements are very elongated compared to other more equilateral triangles, these distorted regions are mainly limited around the end of the slits, where we do not expect particularly important deformations for our applications and calculations. Moreover, as shown in Figs. S1c,d, the regions where we expect interesting and highly nonlinear behavior have more elements than found around the rounded ends. For example, where we expect the blades to deform similarly to beams circumferentially, we don't expect a particular radial behavior. Therefore, while we only use one element radially, or along the width of the blades, we use more elements along their lengths. This specific meshing configuration, which is reproduced for Design A as well, is a compromise between a sufficiently fine and undistorted mesh and acceptable computational times, as the whole geometry of the Design B kirigami disk comprises 10946 nodes and 22519 elements, which takes a long time to use in the finite element iterative process, especially considering that the corotational framework upon which the FIRM software is built is known to be computationally expensive (2).

B. Traction simulations. To perform the traction simulations, we clamp the disk on its external radius and apply a dead pressure on the nodes that are within r_1 , such as how it was performed experimentally. Since no instabilities or snapping behaviors are observed, we used the Newton-Raphson solver implemented within FIRM. The different mechanical properties of our disks are taken as defined in the Methods section of the main paper and we simulate both Designs A and B as defined in the main text. We use a residual tolerance $\epsilon = 10^{-9}$, along with a relaxation of 0.85 for Design A and 0.5 for Design B. The relaxation parameters were found through trial and error such that the simulations would not diverge. By applying a pressure load on our nodes, we do not know the exact load we apply beforehand, and instead compute it post-mortem from the applied load vector.

C. Reconfiguration simulations. Regarding the reconfiguration simulations, we instead clamp the nodes that were within r_1 , similarly to how the screw and nut applied the boundary condition experimentally in the wind tunnel, and applied the nominal pressure over the disk. Once deformed, the pressure distribution is recalculated until convergence through the momentum conservation equation used in Ref. (3). We perform our simulations in a continuation method by increasing the tested velocities incrementally. We use $\rho = 1.225 \text{ kg m}^{-3}$ for the density of the fluid and $C_D = 1.2613$ for the rigid drag coefficient of the disk,

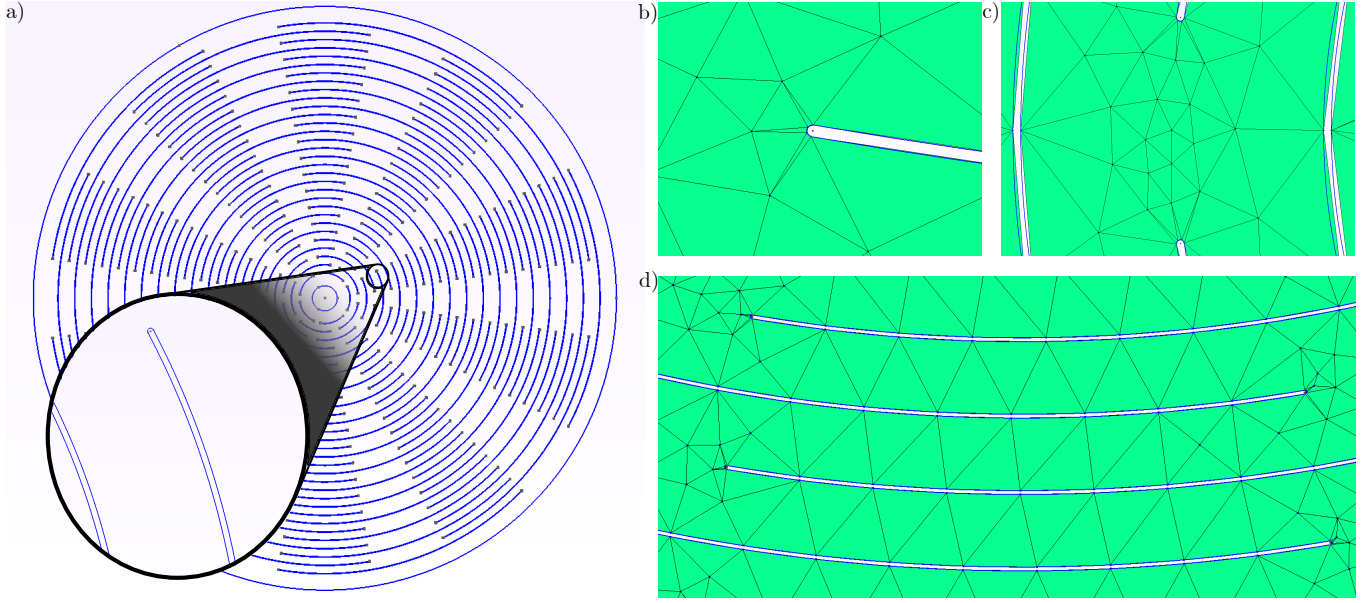


Fig. S1. Mesh of a kirigami inspired disk. a) Adapted geometry of the simulated disk, with the inset showing a closer image of the rounded end of a slit. The disk used here is the Design B kirigami disk. b) Meshed rounded end of a slit, showing distorted elements. c) Mesh of the region between two slits in the angular direction, showing a refined mesh as we expect non-planar behavior. d) Mesh of different blades along the radius of the disk. Multiple elements are used along the beams to observe a sufficiently refined deformation.

as is measured experimentally from Extended Data Fig. 6a. The different mechanical properties are again taken from Methods section of the main paper and we simulate both Designs A and B as defined in the main text. We use a residual tolerance $\epsilon = 10^{-9}$ using the implemented Newton-Raphson solver, along with a relaxation of 0.25 for Design A and 0.6 for Design B, again found through trial and error. For Design A, the simulation setup is done from here. However, Design B's deformation causes more complex manipulations.

D. Porosity model. The given implementation of the FIRM does not take into account the pores that open up as the structures elongates due to its simplified flow mechanics, leading to an overestimation of the drag. To solve this problem, we use an approximation of the fluid behavior through the pores. To do so, we compute the area that the pores open up in comparison to the frontal area of the structure. We define an approximate model of the porous structure's open area such that each pore is enclosed within beams that deform similarly to how a beam under small displacement deforms, considering the large displacement of each beam becomes complex to compute the adequate area. We consider that each beam has clamped-sliding boundary conditions, similarly to Fig. S2a, where we control the vertical displacement w . This leads to the following beam's displacement according to the applied force relationship when using Euler-Bernoulli beams along the curvilinear coordinate of the beam S

$$w(S) = \frac{FL^3}{6EI} \left(\frac{3}{2} \left(\frac{S}{L} \right)^2 - \left(\frac{S}{L} \right)^3 \right). \quad [1]$$

We assume the beams have this general shape, which is an hypothesis that decreases the validity of our solution as this shape is only valid in the case of small displacements and considers a punctual force rather than a distributed pressure due to the flow. Nevertheless, we expect the actual shape of the beam to be close to this hypothesis, but we do not necessarily know the applied force, while we know the applied displacement from our simulations. Indeed, we measure the deformed height of the

beginning and end of each beam, leading to the vertical displacement of these beams. Therefore, knowing $w_{S=L} = w_{end}$, we can find F and express the beam's deformed shape as

$$w(s) = 2w_{end}(s^2 - s^3), \quad [2]$$

with $s = S/L \in [0, 1]$. Due to all these simplifications, the porosity model we are developing is an approximation of reality rather than an exact model of what happens experimentally. By considering that the area of the pores is dictated as the area between the two beams, we find the area of half of one pore on ring j as

$$A_{pore,j} = \int_0^{L_j} w_{\text{top beam}}(S) - w_{\text{bottom beam}}(S) dS = \frac{L_j}{4} (w_{end}^{top} + w_{end}^{bottom}), \quad [3]$$

where $L_j = r_j \theta_\ell / 2$ the length of the half-slit made up by two sets of beams. We assume that each pore on a ring j is identical, leading to the total area of the pores A_p

$$A_p = \sum_{j=1}^{N_r} \frac{L_j N_\theta}{2} (w_{j,end}^{top} + w_{j,end}^{bottom}). \quad [4]$$

From this porous area, we can compute the porosity of the structure as the area of the pores compared to the total area the structure covers

$$\phi = \frac{A_p}{A_0 + A_p} \in [0, 1[, \quad [5]$$

where A_0 is the area of the underformed disk. The area $A_0 + A_p$ is therefore similar to the area of a bell that has a similar shape that our deformed kirigami disk, which has a porosity ϕ .

It was observed experimentally (4) that a porous structure has a different rigid drag coefficient than its non-porous equivalent following a linear relationship $C_d = C_D(1 - \sigma\phi)$, with C_d the drag coefficient of the porous structure, C_D that of the non-porous structure and σ a fitting parameter in the case of a cantilever plate filled with holes. We make the hypothesis that this relationship is maintained for our complex deformed structure such that the porosity is instead calculated from the opened up pores, thus changing the drag coefficient of the structure as it deforms. Moreover, Ref. (4) found $\sigma \approx 1$, thus we make the assumption that $\sigma = 1$ in our case. The reality will be slightly different, but this allows to obtain an approximation without using any computational fluid dynamics.

The applied nominal pressure on the structure is defined as $\rho U_\infty^2 C_d / 2$, but we simulated pressures defined as $\rho U_\infty^2 C_D / 2$. As discussed in Section S3, since the Cauchy and reconfiguration numbers scale with the non-porous drag coefficient $C_Y \sim C_D$ and $\mathcal{R} \sim 1/C_D$, we compute the porosity of the deformed structure at every velocity considered and compute the required non-porous drag coefficient to obtain the porous drag coefficient applied, thus modifying C_Y and $\mathcal{R}$ and leading to the porosity model presented in the main text. From the Cauchy and reconfiguration numbers obtained after applying the porosity model, we compute the new velocities and drag the structure generates.

S2. Theoretical beam model

A. Beam element definition. By considering the kirigami disk defined in Fig. 1a of the main text, we discretize this structure into an assembly of beams, as we observed experimentally that only the material between two slits deforms in a 2D manner, similarly to an Euler-Bernoulli beam. We neglect the in-plane curvature of the beams and assume they are equivalent to a straight beam of the same length. Since we consider that each beam is equivalent to its curved counterpart, we consider that the beams on ring j are at mean radius $\bar{r}_j = r_j + \Delta r_j/2$, where r_j is the radius of the previous slit, and, since the beams are only comprised of the overlap of two slits, their length is defined as $L = \bar{r}_j(\theta_\ell - \theta_d)/2$, as there are two beams per slit. We can define the length L and the width H of each beam as

$$L = \frac{\pi \bar{r}_j}{N_\theta} \frac{1 - \Theta}{1 + \Theta} \text{ and } H = \Delta r_j. \quad [6]$$

We can therefore define each beam that our specimens are comprised of, which allows to define their mechanical properties.

B. Stiffness model. We consider that each beam has clamped-sliding boundary conditions when the whole structure deforms in mode $\mathcal{K}$, such that

$$w(x=0) = 0, \theta(x=0) = 0; \quad [7]$$

$$\theta(x=L) = 0, V(x=L) = 0, \quad [8]$$

where w is the displacement of the beam, θ the rotation angle of the beam and V is the shear force of the beam. This model is illustrated in Fig. S2a.

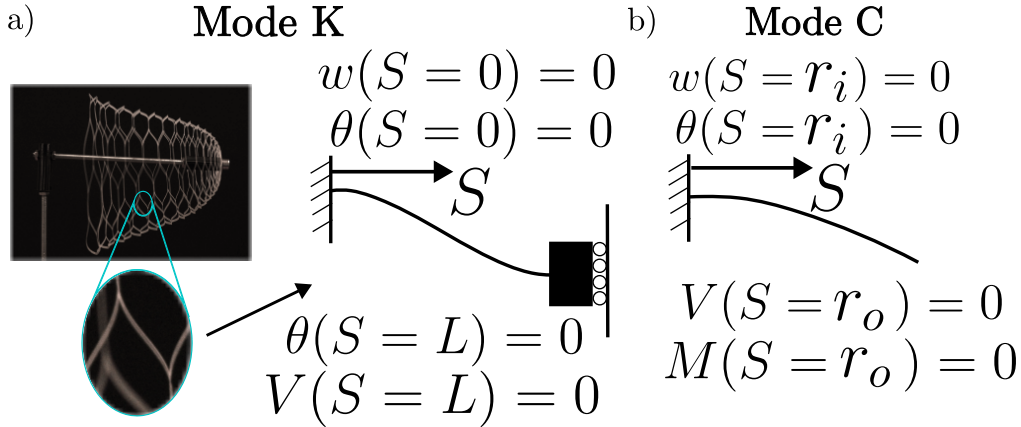


Fig. S2. Theoretical beam models. Illustration of the theoretical models for modes a) $\mathcal{K}$ and b) mode $\mathcal{C}$.

By using a Euler-Bernoulli beam formulation with these boundary conditions, we find that the force-end displacement relation of each beam is $F = (12EI/L^3)w_{end}$, with F the punctual force applied at the end of the beam, w_{end} the displacement at the end of the beam, E the beam's Young modulus and I its second area moment. By substituting the geometrical properties of our previously defined beams, we find the stiffness of the beam as

$$\tilde{K}_{1beam} = \frac{N_\theta^3 E \Delta r_j t^3}{\pi^3 \bar{r}_j^3} \left(\frac{1 + \Theta}{1 - \Theta} \right)^3. \quad [9]$$

We can consider that, for each ring, the beams are in parallel, which causes the force applied to one ring to be defined as $F_{1ring} = \left(\sum_{j=1}^{2N_\theta} \tilde{K}_{j,1beam} \right) \delta_{1ring}$, leading to $\tilde{K}_{1ring} = \sum_{j=1}^{2N_\theta} \tilde{K}_{j,1beam}$. Furthermore, each ring acts in series to one another, as the displacement of each ring contributes to the total displacement. Indeed, we define the total displacement as $\delta = \sum_{i=1}^{N_r} \delta_{i,1ring}$. Moreover, for each ring to deform in an independent manner, the force in each ring needs to be equal, as $F = F_{1,1ring} = F_{2,1ring} = \dots = F_{i,1ring}$. Therefore, we find that $\tilde{K} = \left(\sum_{j=1}^{N_r} \frac{1}{\tilde{K}_{j,1ring}} \right)^{-1}$. Putting this together defines the theoretical stiffness model for the angular kirigami disks. We also use the dimensionless terms $\Delta x_j = \Delta r_j / r_o$, $x_j = r_j / r_o$ and $\bar{x}_j = \bar{r}_j / r_o$, which leads to the structure's theoretical stiffness as

$$\tilde{K} = \left(\sum_{j=1}^{N_r} \frac{\pi^3 \bar{r}_j^3}{2N_\theta^4 E \Delta r_j t^3} \left(\frac{1-\Theta}{1+\Theta} \right)^3 \right)^{-1} = \frac{2N_\theta^4 E t^3}{\pi^3} \left(\frac{1+\Theta}{1-\Theta} \right)^3 \left(\sum_{j=1}^{N_r} \frac{\bar{r}_j^3}{\Delta r_j} \right)^{-1}, \quad [10]$$

which can be decomposed into the contributions from the material through the bending stiffness B , the disk's size through the outer radius r_o , and the kirigami pattern's added stiffness $\tilde{K}$ as

$$\tilde{K} = \left(\frac{E t^3}{12} \right) \left(\frac{1}{r_o^2} \right) \left(\frac{24N_\theta^4}{\pi^3} \left(\frac{1+\Theta}{1-\Theta} \right)^3 \left(\sum_{j=1}^{N_r} \frac{\bar{x}_j^3}{\Delta x_j} \right)^{-1} \right) = \frac{B \tilde{K}}{r_o^2}. \quad [11]$$

C. Extension length. In a similar manner to the stiffness model, we can estimate a theoretical maximum extension length such that each ring can be extended at a maximum of the length of one of its beams. Therefore, the maximum extension length of the kirigami disk is the sum of the length of each unique beam and can be defined in a dimensionless manner such that

$$L_e = \sum_{j=1}^{N_r} \frac{\pi \bar{r}_j}{N_\theta} \left(\frac{1-\Theta}{1+\Theta} \right) = r_o \sum_{j=1}^{N_r} \frac{\pi \bar{x}_j}{N_\theta} \left(\frac{1-\Theta}{1+\Theta} \right) = r_o \ell. \quad [12]$$

D. Transition criteria. To understand when the kirigami disks deform in a mode $\mathcal{C}$ or a mode $\mathcal{K}$, we aim to define an adequate transition criteria using our stiffness model.

D.1. Stiffness comparison. In the case of a disk that deforms in mode $\mathcal{K}$, we already know that an expression for its stiffness when a punctual load is applied in its center as equation Eq. (10). In the case of mode $\mathcal{C}$, we know that the stiffness for the displacement of the free end of a cantilevered beam follows $K \sim EI/L^3$, therefore, for the equivalent tapered beam, the stiffness will scale as

$$K_C \sim \frac{B}{r_o^2}. \quad [13]$$

When we compare the two modes, in the case where we had the exact expression of the stiffnesses, we would look for a transition where both stiffnesses become equal. However, since we did not compute the required constants, and due to our many assumptions, these constants would not allow for an exact representation of reality, we can instead look for $\tilde{K}/K_C = C_1$, with C_1 a constant. Therefore, we rather compute

$$\frac{\tilde{K}}{K_C} \sim \frac{B \tilde{K}}{r_o^2} \frac{r_o^2}{B} = \tilde{K} = C_2, \quad [14]$$

where C_2 is a constant.

E. Stiffness and elongation mapping. In the case of simplified patterns, such that $\Delta x_2 = 1/N_r$ and $n = 1$, we can simplify the equation for the pattern's stiffness $\bar{K}$ and the normalized maximum elongation ℓ such that

$$\bar{K} = \frac{24}{\pi^3} \frac{N_\theta^4}{N_r^6 (N_r + 1)^2} \left(\frac{1 + \Theta}{1 - \Theta} \right)^3 \quad [15]$$

$$\ell = \frac{\pi}{2} \frac{N_r (N_r + 1)}{N_\theta} \left(\frac{1 + \Theta}{1 - \Theta} \right). \quad [16]$$

Using these equations, we see that we can control a pattern's stiffness and elongation through two dimensionless terms, the slit length ratio Θ and the cut ratio N_θ/N_r^2 , as illustrated in Fig. S3a,b. Using these maps, which illustrate the cut ratio required for each Θ to obtain a certain $\bar{K}$ and ℓ , we can use them to design simplified parachutes that have a regular radial distribution of the cuts. When considering $n \neq 1$ and $\Delta x \neq 1/N_r$, this map becomes more complex as Δx_j are not constant anymore. With a combination of $\bar{K}$ and ℓ required, it is possible to find the cut and slit length ratio required as well, and since N_θ and N_r are integers, we can find how many radial and angular cuts are needed to obtain the required values. In the main text, we consider the case where Δx_j are not necessarily constant.

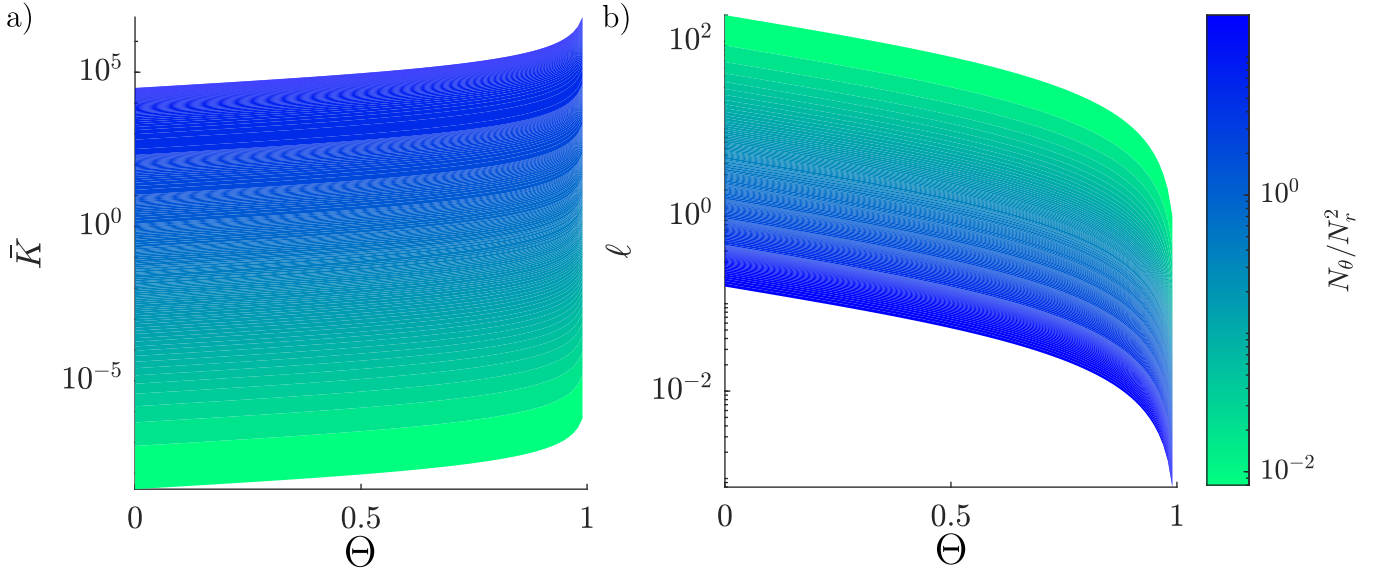


Fig. S3. Closed-loop kirigami parachutes design map. a) Design map of the pattern's stiffness of the parachute $\bar{K}$ with the slit length ratio Θ and cut ratio N_θ/N_r^2 as the control parameters. b) Design map of the maximum theoretical normalized elongation of the parachute ℓ with the slit length ratio Θ and cut ratio N_θ/N_r^2 as the control parameters.

S3. Dimensional analysis

A. Traction. We find the dimensionless form of our force-displacement curves to determine the expression of the typical elastic restoring forces. We normalize the displacement of the structure by the maximum extension length $w/L_e = w/r_o\ell$, while the reaction force F is normalized by the linear maximum expected force $F/KL_e = F/Kr_o\ell$. By considering all of the previously presented traction curves, we plot their dimensionless forms in Fig. S4. Due to the number of presented curves, as well as the number of data points making up the curves, we do not identify them as this would lead to an unclear plot. Nevertheless, the dimensional data is available in Extended Data Fig. 7 if necessary. All of the curves collapse on top of one another at small displacements, while they observe a large scatter at higher displacements, indicating that unconsidered physics might be at

play, such as plasticity or geometrically nonlinear variable stiffness. Nevertheless, we find that the typical restoring forces are adequately represented by the linear maximum expected force KL_e .

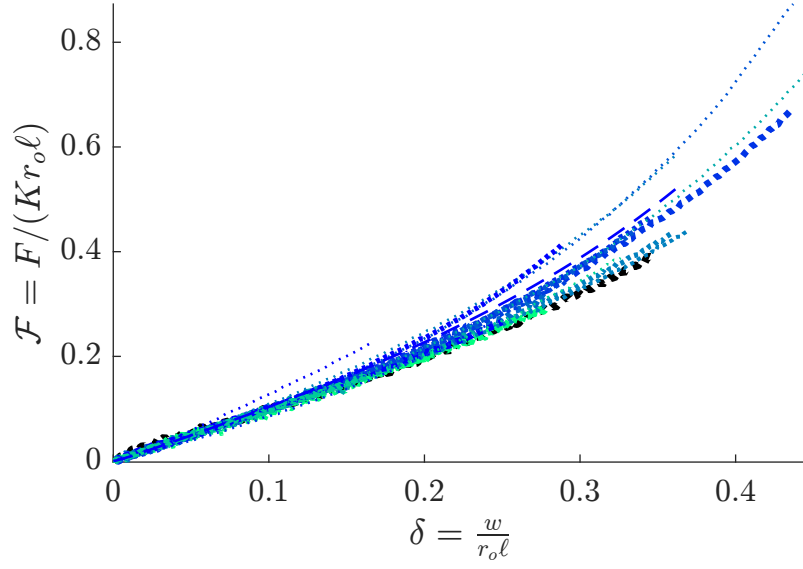


Fig. S4. Dimensionless force-displacement curves. Dimensionless form of the force-displacement curves of the kirigami disk, where F is the reaction force, w the vertical displacement of the center of the disk, K the structure's stiffness in the linear regime and L_e its maximum extension length.

B. Reconfiguration.

B.1. Drag. In the case of a disk that deforms in mode $\mathcal{K}$, we consider that the drag of the parachute follows

$$D[MLT^{-2}] = g\left(\rho[ML^{-3}], r_o[L], L_e[L], U_\infty[LT^{-1}], K[MT^{-2}], C_D[-]\right), \quad [17]$$

where the terms in brackets represent the dimensions of the variable, such that M is the mass dimension, L the length dimension and T the time dimension. Using Pi-Buckingham theorem with 7 parameters and 3 dimensions, we find 4 dimensionless numbers. We first have C_D as a given, then we choose

$$\ell = \frac{L_e}{r_o} \text{ The extended kirigami's aspect ratio;} \quad [18]$$

$$C_Y = \frac{\rho U_\infty^2 \pi r_o^2 C_D}{K L_e} \sim C_D \frac{\rho U_\infty^2 r_o}{B \bar{K} \ell} \text{ Cauchy number; Fluid force to elastic restoring forces;} \quad [19]$$

$$\mathcal{R} = \frac{D}{\frac{1}{2} \rho C_D U_\infty^2 A} = \frac{D}{D_{rigid}} \text{ Reconfiguration number; Structure's drag compared with its rigid counterpart.} \quad [20]$$

We note that we normalize the fluid forces by the maximum expected linear force, as it was shown that it is a good representation of elastic restoring forces.

B.2. Extension. We consider that the extension of the parachute follows

$$w[L] = h\left(\rho[ML^{-3}], U_\infty[LT^{-1}], r_o[L], L_e[L], K[MT^{-2}], C_D[-]\right). \quad [21]$$

Using Pi-Buckingham theorem again with 7 parameters and 3 dimensions, we find 4 dimensionless numbers again. For simplicity, we keep our previous definition of ℓ , C_Y and keep C_D . Then, we define a dimensionless extension of the structure

$$\delta = \frac{w}{L_e} = \frac{w}{r_o \ell}. \quad [22]$$

S4. Design of a larger parachute in realistic conditions

To design a larger parachute, we use the reconfiguration-Cauchy number curve that was obtained from the simulations performed when considering the applied porosity model. To design our parachute, we consider that, for a given velocity U_∞ , we want to maximize the weight used so that the parachute is as efficient as possible. We know that, when considering that the applied weight W is equivalent to its generated drag D at terminal velocity and by assuming that the drag of its rigid counterpart follows $D_{rigid} = \rho U_\infty^2 C_D A / 2$,

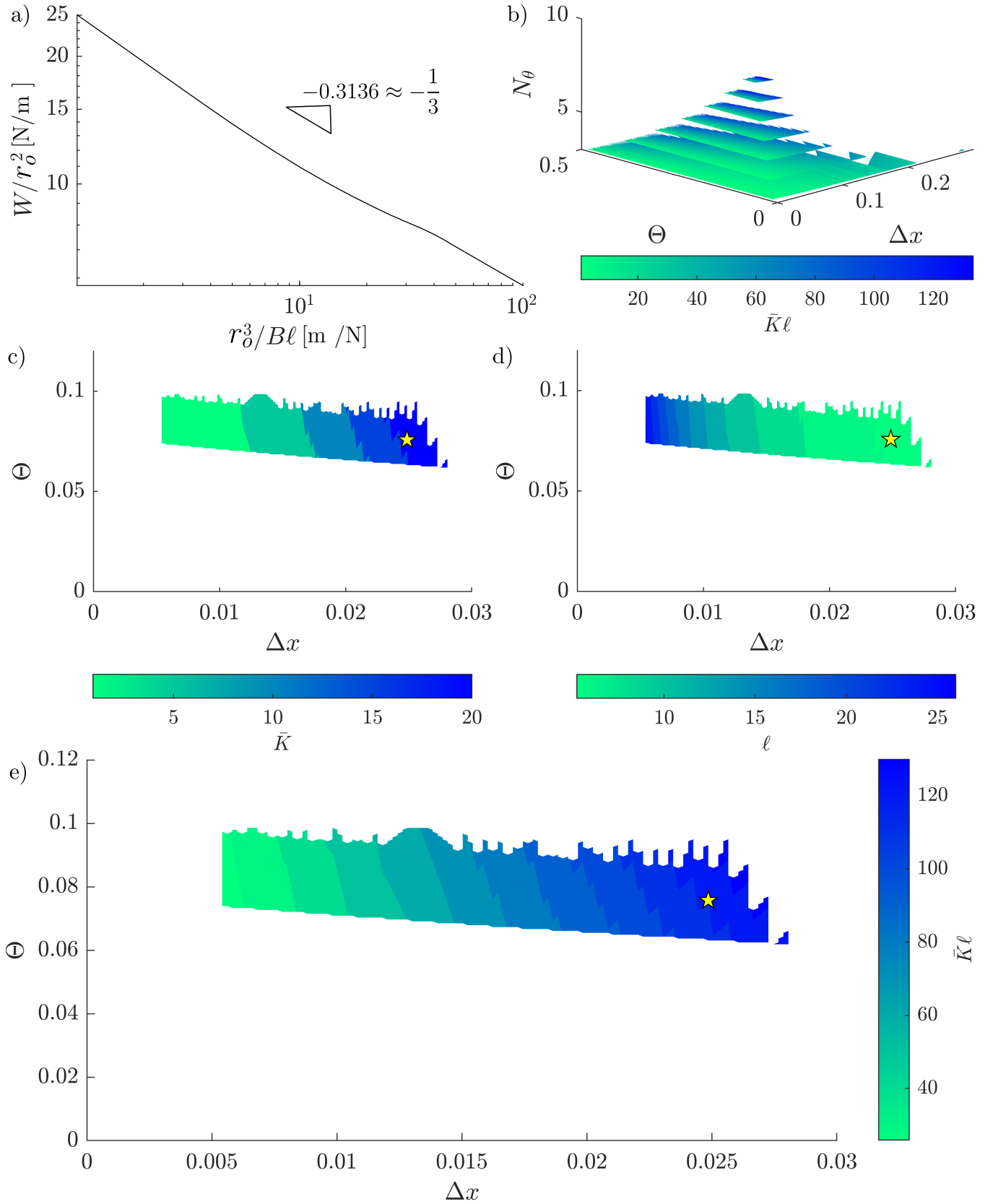
$$\mathcal{R} = \frac{W}{\frac{1}{2} \rho U_\infty^2 A C_D}, \quad [23]$$

therefore we want to maximize $\mathcal{R}$ at a given U_∞ , which requires that we minimize C_Y (see Fig. 3c from the main text), while keeping a certain elongation to allow for increased stability. One of the first things we can do to minimize the Cauchy number is to increase the pattern's added stiffness without deforming in a mode $\mathcal{C}$. Therefore, we choose $\bar{K} = 25$ so that we keep a certain margin of error with our transition criterion $\bar{K} = 28$. By considering that the parachute remains in air with the same rigid drag coefficient and terminal velocity, we find the relationship between the maximum allowed weight, the external radius,

the bending stiffness and the normalized elongation of the structure is approximately following the scaling law $W \sim B^{1/3} \ell^{1/3} r_o$, as shown in Fig. S5a. From this scaling law, we find that we want a parachute that is as big as possible and as stiff as possible in bending, while having a sufficiently long elongation. In order to manufacture a parachute that does not require assembly, we choose a radius of $r_o = 0.25$ m and we use the stiffest material in bending available to us, which are sheets of polyester plastic of thickness $t = 0.3175$ mm (MCMMASTER-CARR 9513K71).

In order to ensure that we only observe mode $\mathcal{K}$ from this parachute, we choose $n = 1$, and therefore $\Delta x_2 = \Delta x$, as those that behaved in degenerate modes in comparison with modes $\mathcal{C}$ and $\mathcal{K}$ typically used $n \neq 1$. We explore the complete design space $\Delta x \in]0, 1[$, $\Theta \in]0, 1[$ and $N_\theta \in \{3, \dots, 10\}$ and compute their respective $\bar{K}$ and ℓ . We consider that the inner center has a radius $r_i = 2 \times 10^{-2}$ m to attach the water bottle, with the first slits being cut at a spacing $\Delta r_1 = 10$ mm and that the minimum spacing between slits is $\Delta r_{min} = 1.35$ mm. We show the resulting pattern stiffness $\bar{K}\ell$ map across the design space in Fig. S5b, where we only show the configurations that respect the manufacturing and deformation constraints ($\bar{K} \leq 25$) and reduce the axes so that only regions where the configurations are feasible are shown.

We find that the design space that meets our requirements is below $N_\theta = 9$, which we choose to maximize the stiffness of the parachute. To choose a specific configuration within this design space, we know we want to minimize the Cauchy number at a given terminal velocity, which leads to maximizing the product $\bar{K}\ell$. As seen in Fig. S5b, the maximum $\bar{K}\ell$ product is also found at high N_θ . We plot the stiffness $\bar{K}$, elongation ℓ and product $\bar{K}\ell$ across this chosen design space in Fig. S5c,d,e, respectively. To maintain our margin of error ensuring we deform in mode $\mathcal{K}$ rather than mode $\mathcal{C}$, we do not go up to the border of the design space. Moreover, since we want a minimum elongation of the parachute to ensure stability, we do not want to use the smallest ℓ possible. Therefore, the exact configuration we choose is a compromise between our requirements, and we choose, following the presented plots, a configuration that belongs to the design space while having promising properties, such that $\Delta x = 0.0248497$, $n = 1$, $N_\theta = 9$ and $\Theta = 0.0757515$, which is identified in Figs. S5c,d,e by a star. While this specific design does not have the optimal $\bar{K}\ell$ value, it follows our design requirements and we ensure to have a sufficiently flexible parachute to deform in mode $\mathcal{K}$. Therefore, the manufactured parachute presented in the main text has an external radius $r_o = 0.25$ m, internal radius $r_i = 0.02$ m, thickness $t = 0.3175$ mm, first radial spacing $\Delta r_1 = 10$ mm, second radial spacing $\Delta r_2 = 6.21$ mm, $N_\theta = 9$ angular sectors and $\Theta = 0.0757515$. The results for this parachute are illustrated in Fig. 4 of the main text.



Supplementary Movie S1. Manufacturing and drop tests: Overview of the manufacturing process and drop testing of our kirigami-inspired parachutes.

Supplementary Movie S2. Wind tunnel testing and simulations: Kirigami disks deforming in mode $\mathcal{C}$ and $\mathcal{K}$ in the wind-tunnel and corresponding FIRM simulations.

Supplementary Movie S3. Manufacturing of large-scale parachutes: Time-lapse of the manufacturing and opening of a meter-scale parachute.

Supplementary Movie S4. Falling behavior of a large-scale kirigami-inspired parachute in realistic conditions: Dropping of a water bottle from a drone at 60 m elevation using our kirigami-inspired parachute.

Supplementary Movie S5. Mass dropping of two different populations of kirigami-inspired parachutes: Distinct falling behavior of kirigami parachutes mode $\mathcal{C}$ and $\mathcal{K}$ leads to self-sorting during a drop test.

SI Supplementary Data S1 (Supplementary_Data_windtunnel.xlsx)

Raw data from the experimental wind-tunnel test conducted on the kirigami specimens.

SI Supplementary Data S2 (Supplementary_Data_tensile.xlsx)

Raw data from the experimental tensile tests on Design A and Design B.

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