Positive, negative or close to zero
Tuning the stiffness of compliant ortho-planar mechanisms.

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POSITIVE, NEGATIVE OR CLOSE TO ZERO

Compliant ortho-planar mechanisms can be fabricated from a single sheet of material and they can move out of the plane in which they are fabricated. In this work, a design of such a mechanism is proposed in which stiffness compensation is applied. With a finite-element model, the force-deflection relation of the buckled mechanism was modelled and it was shown that by changing the design parameters, the stiffness of the mechanism can be tuned such that it can be positive, negative or even close to zero. The mechanism was prototyped and the force-deflection relation was measured to validate the simulations.

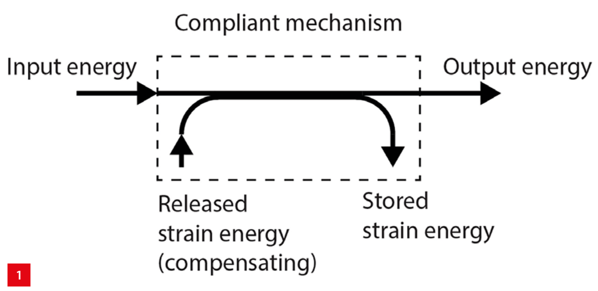
THIJS BLAD, RON VAN OSTAYEN AND NIMA TOLOU

Introduction

Compliant mechanisms are mechanisms in which motion is the result of a deflection of flexible members. These mechanisms have the advantage over their conventional counterparts that they are easier to fabricate, can be more compact and are free of friction, play and wear. Currently, these mechanisms are already applied in for example accelerometers and microphones, and in the future they can be used in applications such as energy harvesters [1].

However, a drawback of compliant mechanisms is their inherent stiffness, a property which describes the extent to which the mechanism resists deformation. Due to this stiffness, a significant part of the input energy is stored as strain energy in the deforming flexible members, and is thus not used for the intended function of the mechanism. As a result, the mechanism may have a poor mechanical efficiency, low range of motion, and high natural frequencies.

In order to overcome this problem, the stored strain energy may be compensated by releasing strain energy in another part of the mechanism [2]. Preloading is a simple way to introduce such compensating energy in the mechanism. During motion, the energy will flow from preloaded parts to deforming parts of the mechanism. This principle is known as stiffness compensation (see Figure 1) and can be used to achieve zero-stiffness behaviour.



1

Energy flow in a compliant mechanism where stiffness compensation is applied.

AUTHORS' NOTE

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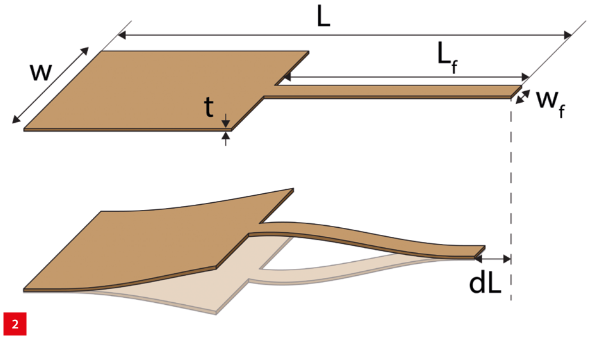
In recent work [3] [4], the principle of stiffness compensation has been applied to a special type of compliant mechanisms called compliant ortho-planar mechanisms (COMs). COMs can be fabricated using planar manufacturing methods and they can move out of the plane in which they are fabricated. Compared to mechanisms that move in-plane, this has the benefit that these mechanisms allow for larger motions while remaining compact, as their parts do not collide.

Design and analysis

In order to apply the principle of stiffness compensation, the mechanism shown in Figure 2 was designed. The mechanism consists of a wide and a narrow section and can be fabricated from a single sheet of material. The energy that is necessary for the stiffness compensation is embedded in the mechanism by preloading, which is done by compressing the mechanism such that it buckles out-of-plane.

Table 1
Relevant design parameters of the design shown in Figure 2.

Parameter	Value (mm)
L	20
L_f	12.1
dL	0.4
w	2
w_f	0.2
t	0.1



2

Design of the compliant ortho-planar mechanism before and after preloading.

The design parameters of the mechanism are given in Table 1. The mechanism has a length, L, and a thickness, t, and is compressed over a distance dL. Moreover, the wide and narrow sections have widths of w and w_f, respectively. The narrow section is called the flexure, and the parameter L_f is introduced to identify its length.

When the mechanism is buckled, it takes the shape of a bridge extending out of plane. Two states are shown in Figure 2, which can be identified as 'knee-up' and 'knee-down', and the mechanism can be moved between these states. The stiffness of the mechanism during this motion can be found by analysing its force-deflection relation.

For most compliant mechanisms, the force required to move the mechanism increases with increasing deflection and the mechanism is said to have a positive stiffness. However, in this design the shape of the force-deflection relation can be tuned by the choice of L_f. As a result, the stiffness that can be found for this mechanism can be positive, negative or even (close to) zero.

To analyse mechanical behaviour of the mechanism, a finite-element model was built in ANSYS. In this model, the mechanism was first buckled by constraining one end of the mechanism in all directions, and displacing the other end. Small imperfections were incorporated in the model to prevent the simulation to crash due to singularities in this preloading step. After the buckled shape was achieved, a displacement was applied at the interface of the wide section and the flexure. Over the range of motion of the mechanism, the reaction forces were recorded at regular intervals to determine the force-deflection behaviour. The force-deflection relations were simulated for different variations of the flexure length L_f and the results of this analysis are shown in Figure 3.

From the figure, it can be seen that very interesting force-deflection relations are found for this mechanism. First of all, the force-deflection relations are clearly nonlinear and show a different stiffness at different parts of the curve. It can be observed that for small deflections the mechanism has a low stiffness, as an almost flat force-deflection relation is found. The force-deflection relation rapidly steepens for larger displacements, which is the result of the mechanism being straightened and loaded in tension.

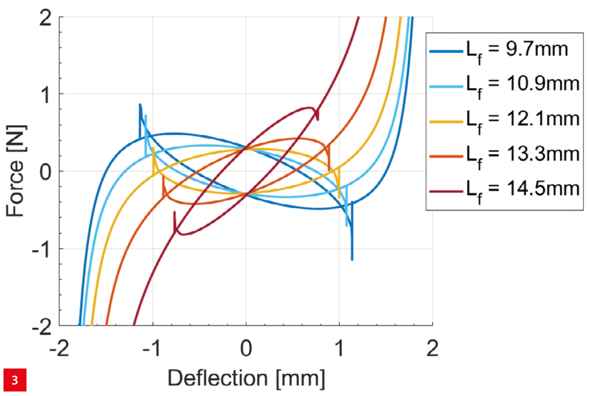
Furthermore, it can be found that the force-deflection relations do not follow a single curve, but make an oval shape. This oval shape actually consists of two load paths, which correspond to the 'knee-up' and 'knee-down' states of the mechanism. When the mechanism is moved from one side to the other, there is a point where the mechanism rapidly changes between these states, a behaviour which is called snap-through.

When looking at the orientation of the oval-shaped part of the force-deflection relation, this part can slope upwards, slope downwards or can be relatively flat. It can be seen that in this mechanism, the slope is dependent on the value of the design parameter L_f. When this parameter is increased, an increasingly upwards-sloping force-deflection relation is found. This means that the level of stiffness compensation can be tuned by changing this parameter. For example, a value of L_f = 12.1 mm was found to result in a mechanism with the flattest force-deflection relation.

Fabrication and experiments

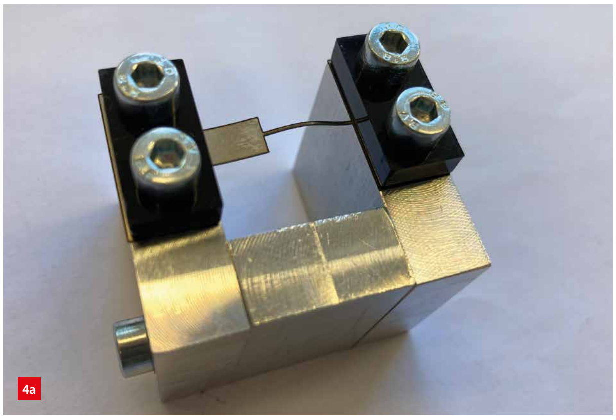
The mechanism was fabricated from a sheet of 0.1 mm thick spring steel (E = 190 GPa) using laser micromachining. For this, a Spectra-Physics Talon 355-15 diode pumped solid-state (DPSS) UV laser system with a wavelength of 355 nm and maximum power of 15 W at 50 kHz was used.

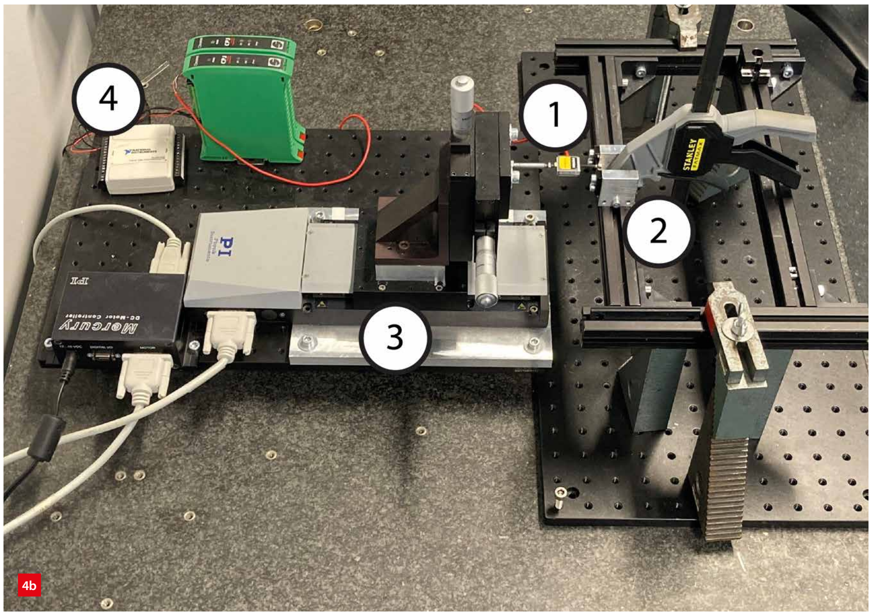
The mechanism was clamped in an aluminium frame consisting of a base plate and two sliding sides. These sides are aligned using dowel pins and can be clamped to the base



3

Results of the finite-element simulations showing the force-deflection relations of the mechanism for different variations of the design parameter L_f.


4a

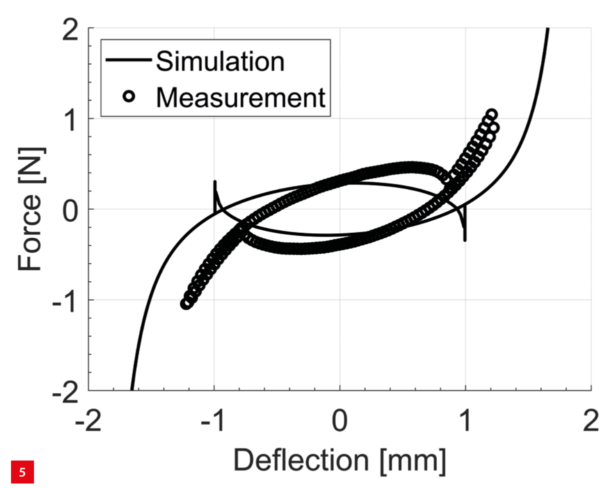


4b

Experimental realisation.
(a) Assembled prototype of the mechanism.
(b) Set-up used to measure the force-deflection relation.
See text for explanation of the various elements.

by tightening the bolts on the sides. The assembly process was as follows. First, a spacer with a thickness of 0.4 mm (i.e., compression distance dL) was placed between the sides and the base of the frame and the bolts were tightened. Next, the beam was mounted to the frame by clamping it between the aluminium and a PMMA bracket. In this state the beam is stress-free and therefore not buckled. Then, the bolt on the side was released such that the spacer could be removed and was subsequently tightened again such that the sides of the frame had moved exactly the thickness of the spacer compared to the stress-free configuration. This introduced an axial preload and the mechanism could buckle out-of-plane to one of its stable positions. The assembled structure is shown in Figure 4a.

The force-deflection relation of the mechanism was evaluated experimentally with the set-up shown in Figure 4b. For this a FUTEK LRM200 force sensor (1) was connected to the mechanism (2) and displaced by a PI M-505 motion stage (3), from which the internal encoder captured position data. Data was recorded using an NI USB-6008 (4) in 100 steps with a resolution of 20 μm. The probe was fixed to the beam at the interface of the flexure using a rolling contact and a magnet. This ensured that the probe would remain in contact with the mechanism at all times. In Figure 5, the measured and simulated force-deflection relations are compared.



5

Simulated and measured force-deflection relations of the compliant ortho-planar mechanism.

Conclusion

From Figure 5, it can be seen that the effects found during the simulations were also observed in the measurements, and that a good correspondence was found. This validates that the used model is able to accurately simulate the mechanics of the buckled mechanism. Moreover, with the proposed design it has been shown that stiffness compensation can be applied in compliant ortho-planar mechanisms and that close to zero-stiffness behaviour can be achieved.

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