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DESIGN OF MICROSYSTEMS BASED ON COMPLIANT STRUCTURES AND DEVICES

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1. Introduction

One of the fastest growing fields in engineering technology is that of microsystems, where devices whose characteristic dimensions and the thus resulting functionalities are of the order of micrometers, i.e. at least 1000 times smaller than that of conventional mechanical structures. Often a "scaled-down" approach to the mechanical design of microsystem devices does not allow suitable solutions to be obtained and therefore new design paradigms have to be adopted. In this work a presentation of the usage of compliant structures as a possible design solution is given. In fact, compliant mechanisms have been used in precision engineering for years [Smith 2000] and recently their usage has been extended also to micro and even nanotechnologies [Howell 2001, Lobontiu 2003]. The aim of this work is to present a broad overview of the main aspects of the design of compliant microsystems.
A compliant structure is a device that gains at least part of its mobility from the deflection of flexible members rather than from moveable joints (Fig. 1). From the mechanical design point of view, compliant devices can be classified as:

- compliant mechanisms based on leaf springs,
- compliant mechanisms employing flexural hinges (localised compliance) and
- continuum structures with distributed compliance.

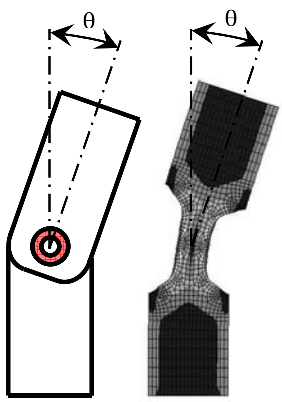


Figure 1. Rotation achieved by using a conventional mechanical and a compliant joint

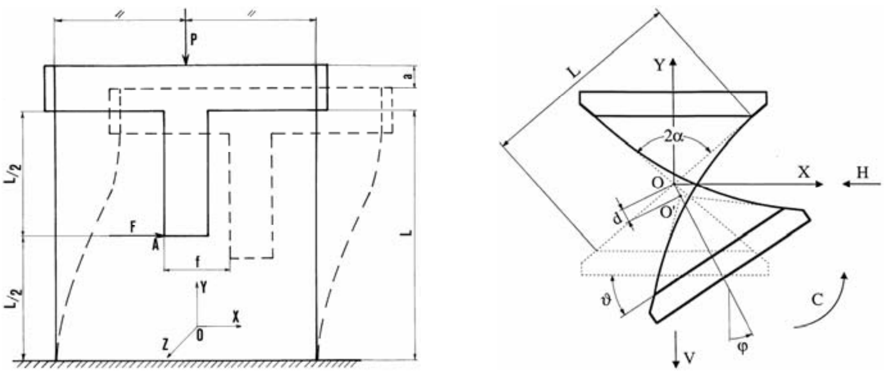
If compared with conventional mechanical bearings, flexible bearings are, in fact, characterised by: high precision, absence of friction (no need for lubrication), limited hysteresis, absence of wear, no risk of jamming, absence of backlash, as well as suitability for use in harsh or special environments

(e.g. clean-room microproduction facilities). Moreover, from the kinematics point of view, the main sources of errors are systematic and therefore simple control laws can be used. Finally, compliant structures can be manufactured monolithically and thus following a “design for no-assembly” approach, which often allows parallel instead of serial kinematics to be attained. All of these advantages imply also a significant reduction of costs of the obtained microsystems.
On the other hand, solutions based on compliant mechanisms evidence as drawbacks: limited strokes and load capacities, presence of restoring forces, a complex kinematics and the presence of parasitic motions (displacements along the secondary degrees of freedom). The effects of such limitations can be minimised if an accurate compliant device model comprising mechanical non-linearities is used.

2. Compliant micromechanisms based on leaf springs

Mechanisms based on leaf springs are devices in which motion is obtained by joining the movable member with the rigid one by means of beam-shaped elements characterized by a marked bending compliance and a high stiffness along the other degrees of freedom. Such devices can be:

1. Single leaf spring devices: a cantilever or a double clamped beam used in off-the-shelf ultra-high precision measurement devices such as the atomic force or the scanning tunnelling microscope (AFM & STM), the radio frequency micro-electromechanical systems (RF-MEMS) and the electrostatic projection displays, as well as in several custom developed nanotechnological solutions such as those for biological applications [Bhushan 2004].
2. Two-beam positioning devices (Fig. 2) used to achieve either a translation (parallel spring translators) or a rotation (cross-spring pivots).



(a) (b)

Figure 2. Parallel spring translator (a) and cross-spring pivot (b)

Usually all leaf spring based compliant mechanisms have working ranges such that parasitic displacements can arise. In the case of parallel spring translators (Fig. 2a) the main displacement f along the x-axis will thus be accompanied by the parasitic motion a along the y-axis, while in the case of cross-spring pivots (Fig. 2b) the main rotation movement ϑ will be coupled to a parasitic motion OO’ of the geometric centre O of the pivot. In order to evaluate these effects, models encompassing the non-linear mechanical behaviour of the beams must be adopted. It is thus possible to refer to the general case of a cantilever beam loaded at the free end with couples and inclined forces (Fig. 3).
This case can be appropriately studied using the well-known “Elastica” approach originally developed for the case of straight slender cantilevers loaded axially, which was recently extended to the generalized case shown on Fig. 3 [De Bona and Zelenika 1997]. A curvilinear coordinate system whose orientation is determined by the inclination of the force is introduced. The differential equation of the deflection curve, using the exact expression for the curvature, is then given by:

$$M_A = M - Fy = EI\left(\frac{d\theta}{ds} - \frac{1}{r}\right) \tag{1}$$

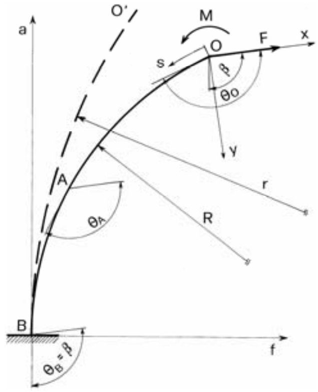


Figure 3. Cantilever beam with the general case of loading

According to the procedure extensively described in [De Bona and Zelenika 1997], the global Cartesian coordinates *a* and *f* of a generic point A of the elastic line of the spring-strip will then be:

$$\frac{a_A}{L} = \frac{x_B}{L}\cos\beta + \frac{y_B}{L}\sin\beta - \frac{s}{L}\left(\frac{x_A}{s}\cos\beta + \frac{y_A}{s}\sin\beta\right) \tag{2}$$

$$\frac{f_A}{L} = -\frac{x_B}{L}\sin\beta + \frac{y_B}{L}\cos\beta + \frac{s}{L}\left(\frac{x_A}{s}\sin\beta - \frac{y_A}{s}\cos\beta\right) \tag{3}$$

The following notation is adopted:
L – total length of the leaf spring
s – length of a section of the beam
x_B, y_B – Cartesian coordinates of point B of beam fixation in the reference frame linked to the inclination of force F
β – slope angle of force F with respect to axis a
Denoting then with $F(k, \varphi_i)$ and $E(k, \varphi_i)$ respectively the elliptic integrals of the first and second kind, with k the parameter of integration, and with φ the amplitude of the elliptic integrals (B and O are the respective points on the leaf spring):

$$\frac{x_B}{L} = 2\frac{E(k,\varphi_O) - E(k,\varphi_B)}{F(k,\varphi_O) - F(k,\varphi_B)} - 1 \tag{4}$$

$$\frac{y_B}{L} = \frac{2k(\cos\varphi_B - \cos\varphi_O)}{F(k,\varphi_O) - F(k,\varphi_B)} \tag{5}$$

The experimental verification of the above analytical approach performed by using a high precision laser interferometric measurement technique showed that the above analytical method allows a level of accuracy higher than the interval of uncertainty of the experimental measurements (of the order of 10^{-4} of the performed primary motion) to be achieved [De Bona and Zelenika 1993].
In order to minimize parasitic displacements, in literature [e.g. Smith 2000] are suggested "compensated" design configurations used recently also in microsystems technology for memory mass storage devices or for friction measurement equipment [Bhushan 2004]. In this case (Fig. 4) two simple translators are mounted onto each other thus causing the overall parasitic motion (sum of equal and opposite contributions from the two translators) to vanish. In reality this will be true only if the lateral (in plane and orthogonal to direction of motion) loads are negligible. If this is not the case, the

above outlined approach allows to "tune" the design parameters of the two single translators so as to reduce the resulting parasitic motion to negligible levels [De Bona and Zelenika 1993].

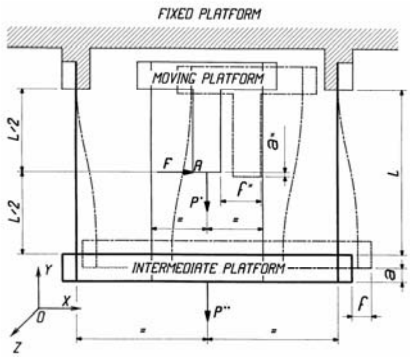


Figure 4. Compensated parallel spring translator

In the case when the overall mechanisms becomes complex, or the loads on the springs are different from those of Fig. 3, the Elastica approach cannot be used and a numerical methodology generally based on a finite element method has to be preferred. In this case, due to the fact that a non-linear large displacement analysis has to be performed, the computational effort by using the commercially available codes could be intensive. For this reason a special finite element type constituted by 3 nodes, each with a single angular DOF, has been proposed [Munteanu et al. 1996] and successfully implemented. As this method does not require an incremental approach to be followed, fast and accurate results have been achieved with very few elements.

3. Compliant mechanisms employing flexural hinges

Another widely used compliant mechanism configuration is that based on flexural hinges, where the compliance is localised in determined spots of the considered device [Lobontiu 2003, Smith 2000]. A conventional design solution is obtained by machining a circular cut-out on a rectangular cross section blank thus obtaining a marked increase of the flexural compliance in the plane of the notch, while retaining the stiffness along the other degrees of freedom [Paros and Weisbord 1965]. If the movable rigid member connected to the notch is much longer than the notch itself, the resulting motion can be determined by considering simple rigid body kinematical models.

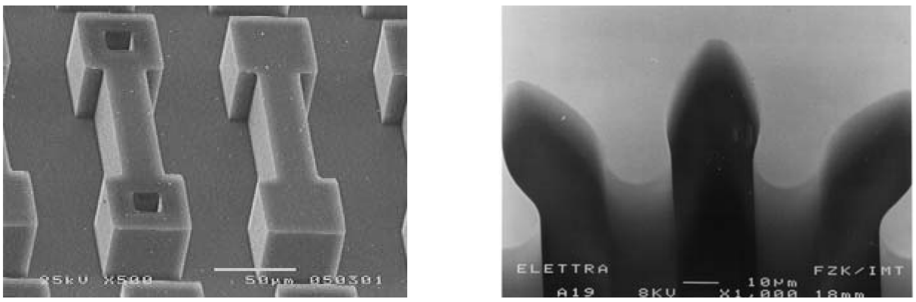


Figure 5. Complex in-plane geometries obtained by using deep X-ray lithography

Up to recently the choice of the notch shapes for flexural hinges was determined by the available production technologies, i.e. the notches were produced by conventional rotating machine tools and therefore limited to circular shapes. The availability of high-precision milling and wire electro-discharge machining (EDM), as well as micromanufacturing technologies, has allowed these limitations to be overcome. In fact, the most commonly used microfabrication processes are based on lithography (Fig. 5), a process that permits prismatic microstructures characterised by complex in-plane shapes, to be easily obtained. It should be noted that microsystem production techniques have been developed and optimised for in-plane structures, and therefore the geometrical complexity does not imply any additional effort, while the high-end production technologies which could be used to achieve the same results in the meso- and macro-fields imply technological difficulties and significant costs. In the case of microsystems, the shape of the notches (Fig. 6) can thus be chosen based on the design requirements for the specific application. This implies, however, that the mechanical model of the flexural hinge is required. This model has to comprise the accurate evaluation of the compliance, the stresses and the eventual parasitic motions of mechanisms based on the hinges. The evaluation of compliances is generally performed by means of the Euler-Bernoulli beam model [Paros and Weisbord 1965]. Even if strongly simplified, this approach gives reliable results also for non conventional hinge shapes, as was shown in [Lobontiu 2003] where a numerical validation of this approach by using the finite element method is performed.

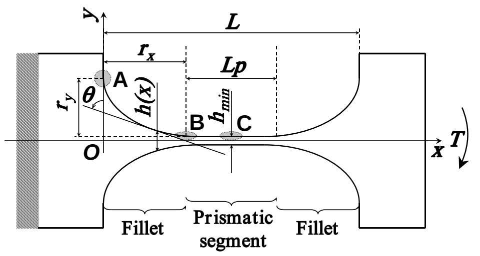


Figure 6. Example of a flexural hinge with a non-conventional shape

The main drawback of flexural hinges is that a high stress-concentration effect is present which limits their fatigue strength. It seems therefore appropriate to consider already in the design phase not only stiffness but also strength requirements. For this purpose it is often necessary to refer to a numerical model based on FEM. In fact, parametric solutions obtained empirically or analytically in closed form are available only for simple geometries. What is more, in the case of a conventional rotational joint, the relative rotation occurs about an axis passing through its geometrical centre. In the case of flexural hinges of other shapes, even for small displacements the actual centre of rotation does not coincide with the geometrical centre, thus producing a parasitic motion (Fig. 7).

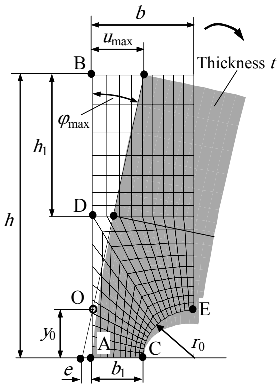
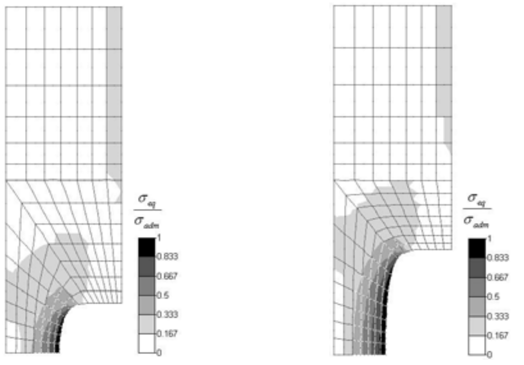


Figure 7. Parasitic deflection of a flexural hinge with semi-circular notches

In most cases this parasitic motion can be evaluated accurately only by following a numerical approach, which becomes absolutely necessary where high flexural rotation angles are aimed at and non-linear effects due to large deflections must be considered. The analytical solution given in [Lobontiu 2003] allows thus only the displacements of the geometrical centre of the there considered hinge shapes to be computed.

Fig. 7 shows the case of a conventional notch loaded with a pure couple. The conditions of symmetry and anti-symmetry enable to model numerically only a quarter of the hinge. The deflection of the notch induces a rotation of the movable member around point O instead of around the geometric centre A. For small displacements point O remains at a fixed position defined by y_0, which will be different for each hinge shape. In [De Bona and Munteanu 2005] it was shown that, if geometrical non-linearities are to be taken into account, a vertical displacement of point B of the movable rigid block will result, but this displacement is influenced very little by the shape of the hinge.



(a) (b)

Figure 8. Normalised stress distributions for optimised notch shapes for a small (a) and a larger (b) value of the admissible parasitic shift y_0

In any case the adaptation of the notch shape to the considered application makes necessary its optimisation. A possible approach to this is to couple a parametric FEM model to an optimization algorithm. The objective function to be minimised is then the flexural stiffness, while the constraints are given by strength (maximal stress) and kinematical (maximal parasitic motion) requirements; the design variables to be optimised are the geometric parameters defining the notch shape (in [De Bona and Munteanu 2005] defined via suitable spline functions). In Fig. 8 are shown the results obtained by employing this approach while maintaining constant the strength constraint and varying the kinematical constraint; it can be clearly seen that by allowing the parasitic shift to grow, the shape of the notch tends to become elongated.

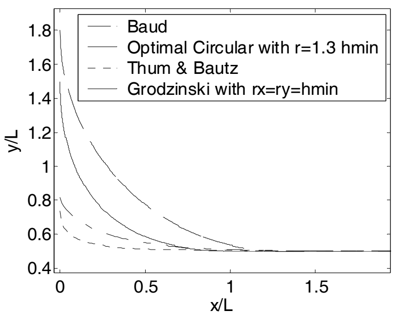


Figure 9. Shapes compared to leaf spring and circular notch in terms of stress and parasitic shift

In fact, in [Zelenika et al. 2004] are compared the ideal leaf spring and the conventional right circular notch shape with optimised notch fillet (Fig. 6) circular, elliptic and parabolic shapes, as well as with "streamline" fillet shapes obtained in classical mechanics via stress concentration minimisation criteria (Fig. 9). It could thus be shown that a compliance increase can be obtained only at the expense of an increase of the parasitic shift. Depending on the particular application, the optimal shape to be used will whence be based on a trade-off between the stress and the kinematical constraint. The streamline shapes are then to be used if the main concern is stress minimisation, while the optimised circular and elliptic shapes provide a good choice if aiming at a parasitic shift minimisation with far smaller stresses than with the circular hinges (Fig. 10).

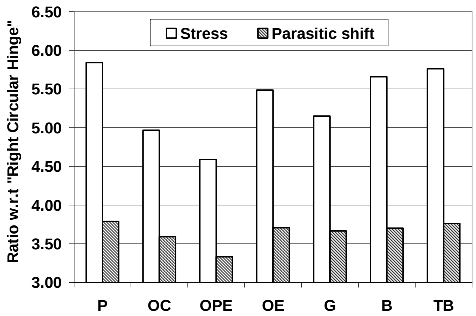


Figure 10. Ratio of the normalised stress of the right circular (RC) hinge vs. that of the other hinges (white bars) and ratio of the parasitic shift of the various hinges vs. that of the RC hinge (hachured bars). P – prismatic beam (leaf spring), OC – optimised circular hinge, OPE & OE – optimised elliptic hinges, G, B, TB – streamline hinges

4. Continuum structures with distributed compliance

Two approaches to synthesise compliant mechanisms are possible: kinematical synthesis and continuum synthesis approach. In the kinematical synthesis approach the compliant mechanism is obtained simply by introducing lumped compliance (leaf springs or flexural notches) in a traditional rigid link configuration. In the case of the continuum synthesis approach, a topology optimization based design is followed [Howell 2001]. A simple topology optimisation technique called "ground structure parameterisation" can then be used. The design space is here defined by a mesh of truss elements that are removed as the analysis determines that they are unnecessary (Fig. 11).

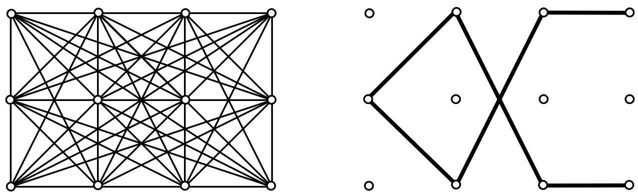


Figure 11. Topology optimisation based on truss elements removal

In the "continuous material density parameterisation" approach, the design space is a material modelled as a grid (Fig. 12). The applied analysis varies then the density of the material in each cell of the grid, leaving a rough image of the compliant mechanism that is finally refined performing classical shape optimization. This approach is widely used in the design of compliant micromechanisms. This approach was derived from a more general method called "homogenization method" where the design domain is treated as a composite made of a material and a void. For each element in the discretised domain, its void is defined using three parameters a, b and θ – Fig. 13.
The optimization algorithm determines the parameters' value for each element of the grid.
These approaches, even if quite promising, are still in the refinement stage and at present no commercial tools based on them are available. These methods also seem not to be able to take accurately into account, during the optimization phase, stress concentrations nor parasitic motions.

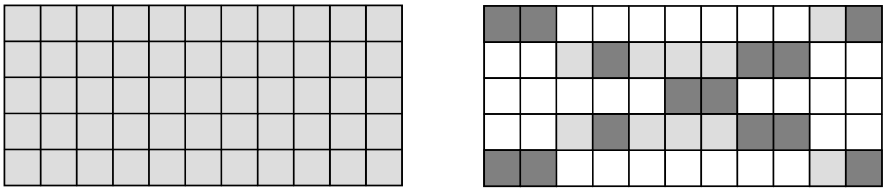


Figure 12. Continuous material density topology optimisation

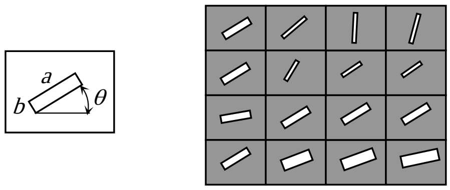


Figure 13. Homogenisation method approach

5. Conclusion

Depending on the application being considered, the illustrated design methodologies related to the usage of mechanisms based on leaf springs, on flexural hinges or of continuum structures with distributed compliance, will allow designers to profit from the usage of compliant structures in the development of microsystem devices.

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