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Optimized flexural hinge shapes for microsystems and high-precision applications

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ABSTRACT

Positioning devices based on flexural hinges are often used in microsystems and precision mechanisms. In this work a nonlinear parametric optimization of flexural hinge shapes is performed. Predefined and freeform parametric shapes are compared in terms of compliance, strength, stress concentration factors and parasitic shifts. Transversal and axial compliances and stress concentration factors are also calculated, permitting design guidelines to be established. It is shown that optimized hinge shapes lead to far better performances with respect to conventional circular notches.

1. Introduction

The design of mechanical devices for application in information technology, metrology, robotics, in the field of microsystems (cantilevers, sensors, metrology devices, etc.), but also in particle accelerators (e.g. optics manipulation units), in precision machine tools, as well as in other applications, often makes use of compliant mechanisms relying on the elastic properties of matter [1–6]. These devices are often based on the employment of flexural hinges [7,8] where the primary (also referred to as sensitive) degree of freedom (DOF) is rotation ϕ_z around a single bending axis (Fig. 1). High off-axis stiffness minimizes deformations along these DOFs. Flexures-based design makes possible the attainment of monolithic microdevices (i.e. where no assembly is needed) characterized by compatibility with the respective working environments (e.g. clean room micro-fabrication facilities, vacuum or radiation environments), as well as the attainment of frictionless ultra-high precision positioning on the meso- and macro-scale [3,5,7–15].

Until recently the choice of notch shapes for flexural hinges was determined by the available production technologies. In fact, the notches were mainly produced by conventional rotating machine tools and therefore were limited to circular shapes. The availability of high-precision milling and especially of electro-discharge machining (EDM), as well as of lithography-based micro-manufacturing technologies, has allowed these limitations to be overcome. The shapes of the notches can therefore nowadays be chosen based on the design requirements for specific applications. A strong improvement of the flexural hinge main features can thus be achieved. In particular, the main drawback of compliant mechanisms based on flexures – their limited travel ranges – can be tackled [7].

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Nomenclature

a	length of the hinge fillet region along the y axis; semi-axis of the elliptic hinge [m]
b	height of the hinge fillet region along the x axis; semi-axis of the elliptic hinge [m]
BS	Baud's hinge shape
C_{nrm}	normalized compliance [rad] as defined in Eq. (7)
e	eccentricity, i.e. deviation of the flexural kinematics from that of an ideal hinge [m]
E	Young's modulus [Pa]
EB	elliptic hinge having the same fillet height as the BS hinge
F	objective function in the optimization process
FO	freeform optimized hinge shape
GR	Grodzinski's hinge shape
i	index that indicates the various hinge fillet shapes
K_{C}	compliance ratio as defined in Eq. (10)
K_{δ}	relative parasitic shift amplitude as defined in Eq. (11)
K_{σ}	ratio of normalized (i.e. maximal) stresses as defined in Eq. (9)
K_{σ}^{*}	relative stress concentration factor as defined in Eq. (12)
L	hinge length [m]
L_{p}	length of the prismatic section of the hinge [m]
M	couple loading the hinge [Nm]
OC	optimized circular hinge shape
OE	optimized elliptical hinge shape
PC	pure circular hinge shape
PP'	parasitic shift [m]
PR	prismatic beam hinge
r	fillet radius in the case of circular hinge fillets [m]
r_i	radii of the middle points of the spline functions defining the FO fillet shape [m]
s	length of the hinge contour [m]
$t(y)$	hinge thickness varying along its length [m]
t_{min}	minimal hinge thickness [m]
TB	Thum's and Bautz's hinge shape
w	width of the hinge [m]
x, y	coordinate system in hinge plane
α_{k}	stress concentration factor
$\gamma = L/t_{\mathrm{min}}$	hinge aspect ratio
δ_{nrm}	normalized parasitic shift as defined in Eq. (8)
θ	slope of the tangent to the hinge fillet with respect to the x axis [rad]
σ	normal stress [Pa]
σ_{n}	nominal stress [Pa]
σ_{nrm}	normalized stress [rad^{-1}] as defined in Eq. (6)
σ_{max}	maximum stress in the hinge [Pa]
τ	tangential stress [Pa]
ϕ_{z}	primary rotation (sensitive) degree of freedom [rad]

Starting from the closed form Euler–Bernoulli beam model developed by Paros and Weisbord [16] for conventional circular hinges, several authors have hence dealt with the compliance of flexural hinges of different shapes. In particular, Smith and coauthors [17] calculated analytically and numerically the stiffness and stress concentration factors of elliptical flexural hinges with different major-to-minor axis ratios and verified experimentally the obtained results. A generalized extension of this approach by employing energy considerations was given by Lobontiu [7], where a wide range of single- and multiple-axes flexural hinge shapes (elliptic, parabolic and hyperbolic) were considered both analytically and numerically in terms of their in-plane and out-of-plane compliances; the actual hinge kinematics was also analyzed.

A compliance optimization along the sensitive DOF, based on a parametric finite element model (FEM), was recently proposed by De Bona and Munteanu [18], allowing freeform optimized hinge shapes to be obtained. Kinematics and strength constraints are considered in the optimization process. In fact, extension of the hinge range of motion along the primary DOF is coupled with the necessity to consider its behavior in the geometrically nonlinear field where parasitic deflections become clearly evident. The calculation of nonlinear effects, as well as the necessity to consider stress-concentration effects, fully justifies the employment of a numerical (FEM) approach.

From the above analysis of the state of the art it is evident that an integrated approach to the optimized design of flexural hinges, which would allow determining advantages and drawbacks of predefined and freeform hinge shapes, is still missing. In this work, therefore, the optimization procedure developed by De Bona and Munteanu [18] is revisited so as to take