Modeling and bonding-free fabrication of flexible fluidic microactuators with a bending motion

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ABSTRACT

Flexible fluidic actuators recently attracted the interest of the microsystem community, especially for soft robotic applications including minimally invasive surgery. These actuators, based on a well-known actuator design where a void is surrounded by an asymmetric elastic structure, can achieve large bending strokes when pressurized. Miniaturized versions of these actuators typically fail due to poor bonding of constituting components, and further, there is little theoretical understanding of these devices. This paper presents a new actuator design which does not require any bonding, and provides new insights in the modeling of these actuators. The newly developed production process of the actuators is based on out-of-plane high aspect ratio micromolding, which enables high-throughput bonding-free fabrication. Furthermore, a mathematical model based on Euler-Bernoulli's beam equation with a deformable cross section is developed that shows good agreement with validation experiments on prototypes. These theoretical insights greatly facilitate the design and optimization of flexible bending actuators.

1. Introduction

Microsystem technology has fostered the development of a wide variety of innovative actuation principles, ranging from electrostatic comb drives to thermal bimorph actuators [1, 2]. More recently, fluidic actuators attracted the attention of the microrobotics community because of their ability to generate large strokes, as well as high force densities [3, 4]. Developments in this field include microrobotic hands [5, 6], tools for minimally invasive surgery [7, 8, 9], actuator arrays for braille displays as well as systems for cell differentiation studies [10].

This paper focuses on a particular type of fluidic actuators, introduced by Suzumori et al. [11] and Konishi et al [12]. These actuators essentially comprise an asymmetric flexible structure around an internal void, which shows a large bending deformation when the void is pressurized (see Fig. 1). This asymmetry can be created using materials that have different elastic properties on each side of the void, or by using an asymmetric geometry of the cross section. Examples of asymmetric cross-section designs found in literature are depicted in Fig. 2. Miniaturized versions of these actuators are typically fabricated by molding two pieces of PDMS, which define the actuator geometry. These two components are then bonded together using partially cured PDMS or O₂ plasma activation to form the pressurizable void [5, 17]. Unfortunately, this bond is prone to failure at high pressures, leading to robustness

issues. Furthermore, little theoretical knowledge for supporting the design of these actuators is available [17, 18, 19].

The actuators developed in this paper use a geometrical asymmetry that is created by an eccentric cylindrical void in a PDMS cylinder (see Fig. 3). These actuators are made in a newly-developed single-piece micromolding process, without requiring a bonding step to create a pressurizable void. This greatly simplifies the production process, which is beneficial for further miniaturization. Besides developing a production process, this paper presents a model that describes the bending motion of flexible fluidic bending actuators and that captures the physics behind this deformation. This model is used to determine optimal actuator dimensions for a certain bending stroke, limiting stresses in the structure in order to maximize robustness.

2. Theoretical analysis

2.1 Model

An analytical model is developed, which captures the deformation of all bending actuators that are composed of an asymmetric structure around a pressurizable void (material or geometrical asymmetry). As an example a PDMS circular beam with diameter D of 1 mm and length L of 10 mm (see Figs. 3 and 4) will be analyzed. The actuation behavior of other flexible bending actuators with different geometries can be deduced analogously. This PDMS tube has an internal eccentric cylindrical void with radius a and eccentricity e (see Fig. 4), and will be modeled as a beam. This beam is subjected to a bending moment M caused by the offset d between the neutral axis of the beam and the line of action of the force F , which is caused by the pressure on the end cap of the actuator. The beam equation of Euler-Bernoulli calculates the bending deformation of the actuator:

$$\frac{d\theta(s)}{ds} = \frac{M(s)}{EI}$$

Where $\theta(s)$ is the angle between the horizontal line and the local tangent of the beam, s is the natural coordinate along the beam with a range from 0 to L , $M(s)$ is the bending moment along the beam, E is the Young's modulus of the material and I is the second moment of area. The model considers the cross section to be constant across the whole length of the actuator; end effects are thus neglected. The bending moment $M(s)$ equals $F \cdot d$ and is constant across the length of the actuator. As a consequence, the beam equation can be simplified as follows:

$$\theta(s) = \frac{Ms}{EI} \quad , \quad M = F \cdot d \quad , \quad 0 \leq s \leq L$$

The overall deformation of the actuator can be calculated using the following formulas:

$$x(s) = \int_0^s \cos(\theta(s)) ds$$

$$y(s) = \int_0^s \sin(\theta(s)) ds$$

The amount of deflection is determined by the ratio of M to EI . Whereas E is a material property, the bending moment M and the second moment of area I depend on the cross section and its deformation. In the next sections, the deformation of the cross section will be determined by using analytic elastic deformation theory and by using 2D finite element modeling.

2.1.1 Analytic deformation

As described by Hrylyts'kyi et al [20], the cross-sectional deformation of an eccentric tube subjected to internal pressure, is equivalent to finding solutions of the biharmonic equation for the stress function $\Phi(r, \varphi)$:

$$\Delta^2 \Phi(r, \varphi) = 0,$$

where Δ is the Laplace operator.

This equation is solved using polar coordinates for convenience, with axes defined as shown in Fig. 4. When the stress function is determined, the following stresses can be calculated:

$$\sigma_r = \frac{1}{r} \frac{\partial \Phi}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \Phi}{\partial \varphi^2}, \quad \tau_{r\varphi} = -\frac{1}{r} \frac{\partial^2 \Phi}{\partial r \partial \varphi} + \frac{1}{r^2} \frac{\partial \Phi}{\partial \varphi}$$

$$\sigma_\varphi = \frac{\partial^2 \Phi}{\partial r^2}, \quad \sigma_z = \nu(\sigma_r + \sigma_\varphi),$$

where σ_r , σ_φ and $\tau_{r\varphi}$ are stresses in the polar coordinate system.

The mechanical boundary conditions on the inner and outer surfaces are:

$$\text{Inner surface:} \quad \sigma_r = -p, \quad \tau_{r\varphi} = 0,$$

$$\text{Outer surface:} \quad \sigma_n = 0, \quad \tau_n = 0,$$

where index n indicates the direction normal to the outer contour.

Radial u_r and tangential u_φ displacements can be determined using following relationships:

$$\frac{1}{r} \frac{\partial u_\varphi}{\partial \varphi} + \frac{u_r}{r} = \frac{1+\nu}{E} [(1-\nu)\sigma_\varphi - \nu\sigma_r], \quad \frac{\partial u_r}{\partial r} = \frac{1+\nu}{E} [(1-\nu)\sigma_r - \nu\sigma_\varphi],$$

$$\frac{1}{r} \frac{\partial u_r}{\partial \varphi} + \frac{\partial u_\varphi}{\partial r} - \frac{u_\varphi}{r} = \frac{2(1+\nu)}{E} \tau_{r\varphi}.$$

An analytical expression for $\Phi(r, \varphi)$ is suggested that is only valid when the eccentricity e is small compared to the inner radius a , in the following form:

$$\Phi(r, \varphi) = A_0 \ln r + \frac{C_0}{2} r^2 + D_0 + \left(\frac{A_1}{r} + C_1 r + D_1 r^3 \right) \cos \varphi.$$

Together with the boundary conditions and conditions of uniqueness of displacement, the deformations can be expressed as follows:

$$u_r = \frac{1+\nu}{E} \left\{ -\frac{A_0}{r} + C_0(1-2\nu)r + \left[\frac{A_1}{r^2} + (1-4\nu)D_1 r^2 \right] \cos \varphi \right\},$$

$$u_\varphi = \frac{1 + \nu}{E} \left[\frac{A_1}{r^2} + (5 - 4\nu) D_1 r^2 \right] \sin \varphi,$$

with

$$A_0 = -p \cdot \frac{a^2 \left(\frac{D}{2}\right)^2}{\left(\frac{D}{2}\right)^2 - a^2}, A_1 = \frac{-e \cdot A_0 \cdot a^4}{\left(\frac{D}{2}\right)^4 - a^4}, C_0 = p \cdot \frac{a^2}{\left(\frac{D}{2}\right)^2 - a^2}, D_1 = a^{-4} \cdot A_1.$$

The cross-sectional deformation and von Mises stresses of an actuator with eccentricity e of 0.15 mm, inner radius a of 0.25 mm and Young's modulus E of 1.4 MPa [17] at a pressure of 0.1 MPa modeled with this analytical model can be seen in Fig. 5 (left).

2.1.2 FEM deformation

Cross-sectional deformation has also been calculated using a finite element software package (MSC Marc), which enables modeling large deformations with non-linear material properties. In contrast to the analytical model, where a constant Young's modulus is assumed, PDMS is represented in this model as a non-linear material using a second order Ogden material model with parameters: $\mu_1 = 0.193$ MPa, $\mu_2 = 2.678$ MPa, $\alpha_1 = 5.269$, $\alpha_2 = -0.051$ (parameters fitted from prior experiments with the same material). The cross-sectional deformation and von Mises stresses of the actuator with eccentricity e 0.15 mm and inner radius a 0.25 mm at a pressure of 0.1 MPa, are depicted in Fig. 5 (right).

When comparing cross-sectional deformations of both models for different combinations of inner radius and eccentricity, these combinations must obey certain limits, namely:

$$a < D/2 - e$$

$$\sigma_{von} \leq \sigma_y.$$

The first limit indicates that the two circles composing the cross section are not intersecting. While the second limit states the von Mises stress criterion prohibiting yielding, with σ_{von} being the largest von Mises stress in the structure and σ_y the yield stress of PDMS (2.24 MPa). Combinations of inner radius and eccentricity must lie below these limit lines (Figs. 6--8 and Fig. 13).

From this cross-sectional comparison (Fig. 6) it can be seen that the analytical deformation is in good agreement with FEM, but only for small eccentricities (lower left corner on Fig. 6). This is not surprising since the analytical model makes the assumption that the normal on the outer surface is parallel to the normal on the inner surface for a certain value of φ , which is only valid for small values of e (See Fig. 4). This discrepancy between models can also be seen in Fig. 5, where the analytical model shows stresses that maintain a circular pattern at the thin wall (right hand side of the cross sections in Fig. 5 (left)), where in reality this thin wall would behave as a membrane yielding a uniform membrane stress (cf. stress distribution in the thin section of FEM in Fig. 5 (right)).

Next, the analytic and FEM simulations of the cross-sectional deformation are substituted into Euler-Bernoulli's beam equation to compute the overall deformation of the actuators. Based on the cross-sectional deformation, the bending moment M , the second moment of area I , and the neutral axis can be calculated, with

$$M = F \cdot d = p \cdot A_F \cdot d,$$

where A_F is the area enclosed by the inner contour, and d is the distance between neutral axis and the centroid of A_F (see Fig. 6).

Using the Euler-Bernoulli beam equation, the actuator deformation can be calculated, see Fig. 7. This figure shows that when cross-sectional deformations are calculated using FEM, the overall deformation is larger than if they were calculated using elastic deformation theory. As will be seen later, tests on prototypes are in agreement with the larger deformations predicted by FEM. Moreover, Fig. 7 shows that a large bending motion can be expected near the yield limit and with large inner radii.

Because a theory is needed that is valid for all actuators, including those with deformations close to the yield limit, and because the analytical model proves to be inadequate in this region, further discussions will be based on the FEM model. In other words, a hybrid model is proposed where non-linear deformations are simulated by 2D FEM while the rest of the behavior is calculated by the linear beam equation of Euler-Bernoulli.

2.2 Optimization

The actuation stroke is optimized as a function of inner radius a and eccentricity e for a fixed outer diameter and for fixed maximal von Mises stress. Lines of fixed maximal von Mises stress with values $\sigma_{von} = 0.85\sigma_y$, $\sigma_{von} = 0.65\sigma_y$ and $\sigma_{von} = 0.55\sigma_y$ are lines parallel to the yield limit $\sigma_{von} = \sigma_y$ on the inner radius vs. eccentricity graph (see Fig. 8 (top)). Along these lines, actuators can be found that for a given pressure exhibit analogous loading conditions. Actuator deformations along these analogous loading lines can be expressed in terms of the curvature $\kappa (= \frac{d\theta(s)}{ds})$, which can be computed, according to the theory of Euler-Bernoulli, taking the ratio of M to El , where E has a value of 2.5 MPa as will be discussed later on. Along these deformation lines of analogous loading, clear optima in curvature can be seen (see Fig. 8 (bottom)). These optima lie along a line of optimal actuator configurations, with equation " $\kappa = 1.316 \text{ mm}^{-2} \cdot a - 0.339 \text{ mm}^{-1}$ ", where a is expressed in mm (dotted line in Fig. 8 (bottom)). It can be seen that for these optimal actuators, combinations of eccentricity and inner radius all lie within close proximity of the line defined by " $e = -0.438 a + 0.233 \text{ mm}$ " (dotted line in Fig. 8 (top)).

Thus, in order to design an actuator that experiences minimal stresses when pressurized, the desired bending curvature and the corresponding optimal inner radius are selected according to the dotted line of the bottom section of Fig. 8. The eccentricity is defined along the dotted line from the top section of Fig. 8. It can also be seen that it is theoretically impossible to construct actuators with outer diameter of 1 mm and a curvature larger than

0.132 mm⁻¹, this is the optimum lying on the yield limit, assuming a pressure load of 0.1 MPa.

To validate the deformation theory and the existence of the optima, actuator prototypes are developed, using a new production process, which will be discussed in the next paragraph.

3. Production process

In the existing literature on flexible fluidic bending microactuators, a pressurizable void is created by bonding two layers at their edges (see Fig. 1). This bonding can be done using an oxygen plasma [17], vacuum ultraviolet (VUV) radiation [21] or using partially cured PDMS [22]. In this paper a more robust bonding-free production process is developed. In this process, depicted in Fig. 9, flexible fluidic bending microactuators are made in one piece by means of out-of-plane high aspect ratio micromolding. The mold consists of two halves made by micromilling of aluminum. To produce the long eccentric cylindrical void, a microrod (tungsten carbide) with radius a is placed in the bottom half of the mold. This microrod is positioned eccentricly relative to a hollow cylinder, in the top half of the mold. Precise alignment of both halves of the mold is achieved by means of dowel pins. The mold is first treated with a demolding agent (devcon liquid release agent), and then filled with PDMS (Dow Corning Sylgard 184 mixed in a 10:1 ratio). The mold is then closed, and PDMS is cured at 90°C until fully cross-linked. Because of the high aspect ratios (ratio of diameter of microrod to cavity length is 13.2 to 16.7 depending on the microrod diameter) ethanol is used as a lubricant during demolding. After demolding, the actuators are connected to a pressure supply using a PMMA connection device that uses a reversible press fitting to ensure pressure sealing. Figure 10 shows a prototype microactuator that is made using this production process, before and after actuation. Note that this process does not involve any bonding in the actuator or interconnection.

4. Test results

To validate the theoretical analysis discussed above, microactuators with different inner radii a and eccentricities e are tested using a static pressure test setup where actuator deformation is recorded using a digital optical microscope with measuring software. A series of deformations with increasing pressure is shown in Fig. 11. This figure shows that there is a good agreement between modeled actuator deformation and measured prototype deformation for this actuator ($a = 0.305$ mm, $e = 0.14$ mm). The Young's modulus value that is used in the Euler-Bernoulli beam equation is 2.5 MPa, which is significantly higher than values for PDMS Young's modulus that normally can be found in literature. This can be explained by the fact that cross-sectional deformations cause a stiffening of the beam structure, due to Poisson's effect. Because PDMS is a nearly incompressible material ($\nu \approx 0.5$) this effect is substantial, and cannot be neglected. This effect can be seen in Fig. 12 ($a = 0.36$

mm and $e = 0.06$ mm), where at high pressures the model predicts deformations that are too large. For low pressures (≤ 0.2 MPa), a constant Young's modulus of 2.5 MPa yields results that correspond well with experiments (Figs. 11 and 12), while at larger pressures, larger Young's moduli, which vary with pressure, are needed to match theory with practice. Note also that actuators fabricated by this process are able to operate at much higher pressures than previously achieved with similar actuators fabricated by bonding, which typically failed at a pressure of about 0.1 MPa [17].

As discussed above, the theoretical analysis shows the existence of an optimal combination of inner radius and eccentricity where bending deformation attain a maximum. The prototype actuators with different combinations of inner radius and eccentricity were subjected to a pressure of 0.1 MPa and the resulting curvatures are plotted in Fig. 13 on top of the modeled solutions ($p \leq 0.2$ MPa, so Young's modulus was fixed at 2.5 MPa). This figure shows test results that are in good agreement with the proposed theory. Although actuators where produced that theoretically should yield at a pressure of 0.1 MPa (line of experimental test results crossing the line of yield limit), no such behavior was observed, implying that the yield stress of PDMS is larger than the assumed 2.24 MPa.

Because measurements on prototypes validate the proposed theory, this theory is considered a good starting point to design flexible fluidic bending microactuators.

5. Conclusion

In this paper, a theoretical analysis of flexible fluidic bending microactuators is proposed, where cross-sectional deformations are calculated using both analytical and finite element modeling. This analysis showed that large bending deformations are expected when both the eccentricity and the inner void radius are large: e.g. a curvature of about 0.14 mm^{-1} can be achieved for an actuator with inner radius of 0.35 mm and eccentricity of 0.09 mm at a pressure of 0.1 MPa. Unfortunately, in this large-stroke region, analytical predictions of the cross-sectional deformations proved not to be adequate. This problem was overcome by combining FEM simulations of the cross-sectional deformation with the analytical beam equation of Euler-Bernoulli, resulting in a quantitatively correct model.

Furthermore, a novel production process based on micromilling and high aspect ratio molding of PDMS enables producing flexible microactuators in one piece and with high throughput. These actuators show large bending deformations and can operate at higher pressures than similar actuators fabricated by bonding two components.

Both theory and experiments indicate the presence of optimal actuator configurations for a certain bending stroke, which minimizes stresses in the structure. For an outer diameter of 1 mm, and a pressure of 0.1 MPa, these optimal actuators can be designed in function of the desired bending curvature κ . The optimal inner radius a (in mm) is defined by: $\kappa = 1.316 \text{ mm}^{-2} \cdot a - 0.339 \text{ mm}^{-1}$, while the relation between eccentricity e and inner radius a is defined by " $e = -0.438 a + 0.233 \text{ mm}$ ".

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Fig. 1. Actuation of a flexible fluidic bending actuator.

Actuator	Cross section	material asymmetry	geometrical asymmetry
[Suzumori 1991]	rubber		X
[Lazeroms 1996]	silicone rubber		X
[Hirai 2001]	silicone rubber		X
[Ikeuchi 2009]	poly lacticacid		X
[Konishi 2011]	PDMS	X	X
[Mori 2012]	aramid fiber silicone rubber	X	

Fig. 2. Comparison of asymmetric cross section designs for flexible actuators found in literature: Suzumori [11], Lazeroms [13], Hirai [14], Ikeuchi [15], Konishi [7], Mori [16].

Fig. 3. Geometry of the developed actuator, showing cross section.

Fig. 4. Modeled structure: cross sections.

Fig. 5. Comparison of cross-sectional deformation and von Mises stresses: analytical model (left) and FEM (right). Unloaded cross section below the cross-sectional symmetry line (hatched), loaded cross section above symmetry line where deformation scaling is set to 1. ($a = 0.25\text{mm}$, $e = 0.15\text{mm}$, pressure = 0.1 MPa).

Fig. 6. Comparison of cross-sectional deformation: analytical model (blue) and FEM (red).
(pressure = 0.1 MPa).

Fig. 7. Comparison of actuator bending deformation, cross-sectional deformations calculated by means of FEM (purple), analytical model (red), no cross-sectional deformations (blue).
(pressure = 0.1 MPa).

Fig. 8. Modeled bending deformations for actuators with different combinations of eccentricity and inner radius. (pressure = 0.1 MPa).

Fig. 9. Production process: overview.

Fig. 10. Actuator prototype, before (left) and after pressurization (right).

Fig. 11. Comparison between experimental (top) and modeled (bottom) actuator deformation. ($a = 0.305$ mm, $e = 0.14$ mm).

Fig. 12. Comparison between experimental (top) and modeled (bottom) actuator deformation. ($a = 0.36$ mm, $e = 0.06$ mm).

Fig. 13. Bending deformations for actuators with different combinations of eccentricity and inner radius: comparison between modeled and prototype actuators. (pressure = 0.1 MPa).