Thermal buckling of eccentric microfabricated nickel beams as temperature regulated nonlinear actuators for flow control

Matthew McCarthy\textsuperscript{a,}\textsuperscript{*}, Nicholas Tiliakos\textsuperscript{c,a}, Vijay Modi\textsuperscript{a}, Luc G. Fréchette\textsuperscript{b,a} \\
\textsuperscript{a} Columbia University, Department of Mechanical Engineering, New York, NY 10027, USA \\
\textsuperscript{b} Université de Sherbrooke, Department of Mechanical Engineering, Sherbrooke, Qué., J1K 2R1, Canada \\
\textsuperscript{c} ATK-GASL, Ronkonkoma, NY 11779, USA \\
Received 1 March 2006; received in revised form 12 May 2006; accepted 21 May 2006 \\
Available online 17 July 2006

Abstract

The design, fabrication and testing of micromachined nickel beams buckling under thermal loading is presented in this paper. The focus is on characterizing design parameters important to the implementation of electroplated nickel beams as the actuation mechanism in thermally adaptive microvalves for self-regulated MEMS cooling schemes. Nondimensional design curves of the thermal buckling phenomena have been analytically developed and validated with test results from electroplated nickel beams with slight eccentricities. Highly nonlinear deflection versus temperature curves were predicted by a closed form model and match well with experimental measurements. Buckling deflections of more than 50 μm were achieved at actuation temperatures under 100 °C. The fabrication process for suspended nickel beams is also presented, along with fabrication issues that impact the actuation capabilities of the beams.

Keywords: Buckling; Thermal; Actuator; Micro valve; MEMS; Cooling

1. Introduction

The current work is focused on developing a thermal actuation mechanism for flow control in a self-regulating MEMS cooling scheme. An array of adaptive valves would maintain a surface below some critical temperature by locally modulating flow rate through a cooling device in response to changes in heat load. An efficient adaptive microvalve would provide low valve actuation at moderate temperatures and relatively large valve actuation, hence large flow, at high temperatures. This requires highly nonlinear actuation capabilities over relatively small temperature ranges. Achieving this behavior without external electric control or power will result in a simple self-regulating valve, or valve array, that varies coolant flow rate nonlinearly with changes in local surface temperature. This creates a MEMS skin-cooling scheme similar in nature to its biological inspiration. Fig. 1 shows the schematic of a single device as well as an array of devices coating a heated surface.

The crucial element in this design is a microvalve that adapts to increases in temperature. By using the heat load to drive the thermal actuation, there is no need for control electronics or for resistive heating typically associated with electro-thermal actuation. This yields a simple, compact system that can be easily integrated into arrays of devices that need only fluidic connections for coolant.

The mechanism of buckling has been investigated as a means of achieving the nonlinear deflections needed for valve actuation. If a beam-column restrained axially is taken to elevated temperatures the difference in coefficient of thermal expansion between the beam and the material restraining it leads to an internal axial force. If the beam has a higher coefficient of thermal expansion it will be subject to compression. A slender beam-column in compression will exhibit negligible deflections at small loads and then buckle at some critical load leading to dramatic lateral deflections. The critical load for buckling can be geometrically tailored to a specific temperature, which makes thermal buckling of beam-columns an ideal actuation principle for a thermally adaptive microvalve.

Fig. 2 shows conceptually how buckling beam actuation can be used to increase flow rate. The doubly clamped planar beam...
fabricated over a small air gap on a silicon substrate will buckle as the beam approaches a critical temperature. An applied pressure difference will drive a flow through the thin air gap modeled as parallel plates. Accordingly, the mass flow rate will increase cubically with the parallel plate gap. This nonlinear mass flow versus deflection behavior, coupled with the nonlinear deflection versus temperature of thermally buckled beams, will result in an extremely nonlinear increase in mass flow rate in a designed temperature range. This behavior is crucial for the achievement of an efficient adaptive cooling scheme.

Thermally actuated devices transform the expansion of heated materials into a useful mechanical output. Techniques to amplify and manipulate the relatively small deflections associated with linear thermal expansion are typically needed. Previous work has been done with hot-arm type MEMS thermal actuators [1,2]. Thermal expansion is generated with resistive heating and amplified by a compliant arm structure. In Yan et al. [1] 150–250 μm long actuators produce tip deflections ≤10 μm when heated up to 1000 °C by voltages varying from 0 to 10 V. Mankame and Ananthasuresh [2] report deflections up to 50 μm for 2 mm long beams for similar temperatures and voltages. Thermal buckling offers not only the desired nonlinear behavior but also an effective amplification mechanism. The predicted lateral deflections of thermal buckling are more than an order of magnitude larger than the linear expansions associated with the same temperature rise. Where comparably sized electro-thermal actuators require temperatures greater than 1000 °C [2], similar deflections can be achieved through buckling at temperatures of less than 100 °C. This amplification allows relatively low temperature increases from the heat source to efficiently drive the actuation mechanism in the adaptive valve. The buckling instability of microfabricated beams has been previously investigated by McCarthy et al. [3], Carr and Wybourne [4] and Chiao and Lin [5] where beams were compressed and buckled under various loading mechanisms.

This work aims at integrating thermally buckling beams into temperature regulated nonlinear actuators. In this paper, modeling of the thermomechanical behavior of buckling clamped beams will be presented leading to nondimensional design curves. The focus will be on presenting the deflection and state of stress as a function of applied temperature rise, which is a notable difference from the existing electro-thermal actuation

---

**Fig. 1.** Novel MEMS adaptive cooling scheme: (a) a single device and (b) an array of devices cooling a surface subject to a spatially and temporally varying heat load.

**Fig. 2.** Visualization of thermally adaptive valve concept: (a) clamped beam fabricated over a slot through the substrate, with a pressure difference across it; (b) at low temperatures there is low mass flow through the thin air gap between the beam and substrate; (c) at high temperatures, the beam buckles, opening the air gap and allowing a dramatically larger coolant flow rate.
work [1,2,5] where the applied voltage was the controlling variable. Building the geometry shown in Fig. 2 requires the development of a fabrication process to produce planar clamped beams with a small stand off gap defining the desired parallel plate flow path. The achievement of this will be presented along with experimental results validating the thermal buckling model.

2. Theory

A long slender clamped–clamped beam in compression will exhibit a lateral deflection as the loading approaches a critical value and the beam becomes unstable. Due to symmetry, a clamped–clamped beam of length 2L buckling under a compressive force can be analyzed as a pinned–pinned beam of length L under the same loading. The pinned ends correspond to inflection points in the clamped beam, where the internal moment is null. The buckling analysis can be carried out for the simpler pinned–pinned case and the resulting deflections can be extended accordingly to the clamped–clamped case as shown in Fig. 3.

A perfectly prismatic beam of constant cross section will buckle in a discontinuous manner at the critical load; this however is a theoretical approximation. In reality, a compressive member will have some imperfection or asymmetry that leads to a continuous nonlinear deflection. To account for the imperfections of both the beam and the loading in the analysis, the compressive force can be applied at some distance from the neutral axis. The pinned beam-column with a compressive load, \( P \), applied at an eccentric distance, \( e/2 \), is statically equivalent to an axially loaded beam with an additional moment, \( M_0 = Pe/2 \), applied at the end points as shown here in Fig. 4. As \( e \to 0 \) the loading and the corresponding solution approach that of a theoretically perfect beam.

Incorporating this eccentricity is an analytical approach to introduce the parameter \( e \), a measure of the imperfections in both the beam and loading. In general \( e \) is not an actual dimension. However, when considering this modeling approach and extending the results to the clamped–clamped case the actual geometry being solved for is the one shown in Fig. 5.

Rather than treating \( e \) as an ambiguous parameter, it is advantageous to build beams with this actual eccentric clamped–clamped geometry. This ensures that the beam will buckle in the appropriate direction. It also provides a controllable design parameter, \( e \), and makes the modeling approach more accurate. Using this method, the buckling of compressed beams with a designed eccentricity has been investigated, focusing on the regime of small eccentricity ratios, \( e/h \to 0 \), for the specific geometry in Fig. 5. Of particular interest are closed form models for the beam shape, central deflection and maximum stress as a function of temperature rise, as developed next.

2.1. The elastic curve and the secant formulation

An elastic analysis of the equivalently loaded pinned–pinned case was carried out for thin beam-columns with curvature, \( dv/dx \), negligible compared to unity. The resulting elastic curve for the beam, \( v(x) \), as given by [6–8] and the free body diagram in Fig. 6 is determined from the governing equation:

\[
EI\frac{d^2v}{dx^2} = M(v) = -M_0 - Pv = -P \left(\frac{e}{2} + v\right)
\]

where \( v \) is the pinned–pinned deflection, \( I \) the beam moment of inertia and \( E \) is the beam material’s modulus of elasticity. The
resultant problem becomes:
\[
\frac{d^2 v}{dx^2} + \left(\frac{P}{EI}\right) v = -\frac{Pe}{2EI}, \quad v(0) = v(L) = 0
\]
which has the following solution:
\[
v(x) = \frac{e}{2} \left[ \tan \left( \frac{L}{2} \sqrt{\frac{P}{EI}} \right) \sin \left( \sqrt{\frac{P}{EI}} x \right) + \cos \left( \sqrt{\frac{P}{EI}} x \right) \right] - 1
\]
(3)
As seen in Fig. 3, the central deflection of the associated clamped–clamped problem, \(d_{\text{MAX}}\), is twice that of the central deflection of the pinned–pinned problem, \(v(x = L/2)\), hence:
\[
d_{\text{MAX}} = 2v \left( x = \frac{L}{2} \right) = e \left[ \sec \left( \frac{L}{2} \sqrt{\frac{P}{EI}} \right) - 1 \right]
\]
(4)

2.2. Maximum stress

A buckled beam under compressive loading is subjected to both axial and bending stress. The maximum stress is compressive and located at the midpoint on the lower surface of the beam, as drawn in Fig. 3. It can be written as the sum of the axial and bending components:
\[
\sigma_{\text{MAX}} = \sigma_A + \sigma_B = \frac{P}{bh} + \frac{h}{2L} \left| M \left( x = \frac{L}{2} \right) \right|
\]
(5)
Using the magnitude of the internal moment at the midpoint, as given by Eq. (1):
\[
\left| M \left( x = \frac{L}{2} \right) \right| = P \left( e + v \left( x = \frac{L}{2} \right) \right) = \left( \frac{Pe}{2} \right) \sec \left( \frac{L}{2} \sqrt{\frac{P}{EI}} \right)
\]
(6)
yields the maximum stress in the buckled beam:
\[
\sigma_{\text{MAX}} = \frac{P}{bh} \left[ 1 + \frac{3}{4} \left( \frac{e}{h} \right)^2 \right]\left[ 1 + 3 \left( \frac{e}{h} \right) \sec \left( \frac{L}{2} \sqrt{\frac{P}{EI}} \right) \right]
\]
(7)

Eqs. (4) and (7) define the beam central deflection and maximum stress as a function of axial load. An additional relation is needed to relate the axial force, \(P\), to the average beam temperature rise, \(\Delta T\).

2.3. Stress–strain–temperature relationship

Consider the stress–strain relationship of a heated beam restrained from expansion in the axial direction:
\[
\sigma_A = \frac{P}{bh} = E[\Delta \alpha \Delta T - \varepsilon']
\]
(8)
Here \(\Delta \alpha\) is the difference in the coefficient of thermal expansion between the beam and the substrate and \(\Delta T\) is the average temperature rise of the restrained beam. It is assumed that the beam and substrate are at equal isothermal temperatures. The axial stress is denoted by \(\sigma_A\), while \(\varepsilon'\) is the strain due to beam elongation:
\[
\varepsilon' = \frac{l - L}{L}
\]
(9)
where \(l\) is defined as the deformed beam length, given by:
\[
l = \int_0^L \sqrt{1 + \left( \frac{dv}{dx} \right)^2} \, dx
\]
(10)
The assumption of shallow beam curvatures, \(dv/dx \ll 1\), has already been asserted previously in this analysis. Accordingly the integrand in Eq. (10) can be simplified to:
\[
\sqrt{1 + \left( \frac{dv}{dx} \right)^2} \approx 1 + \frac{1}{2} \left( \frac{dv}{dx} \right)^2
\]
(11)
and the strain term in Eq. (8) can therefore be rewritten as:
\[
\varepsilon' \approx \frac{1}{2L} \int_0^L \left( \frac{dv}{dx} \right)^2 \, dx
\]
(12)
Knowing \(v(x)\) from Eq. (3) both the derivative and integral in Eq. (12) can be evaluated. Dropping the approximate equality, combining Eqs. (8) and (12) and rearranging terms yields Eq. (13):

This defines the relationship between applied axial load and the average temperature rise of the beam.
2.4. Nondimensional design curves

Collectively, Eqs. (4), (7) and (13) fully describe the thermomechanical behavior of doubly clamped eccentric beams. For convenience, several nondimensional parameters can be defined to simplify these equations. The critical load, \( P_{cr} \), is the force at which a theoretically perfect beam \((e=0)\) will buckle\([7]\):

\[
P_{cr} = \frac{\pi^2 EI}{L^2} = \frac{\pi^2 Ebh^3}{12L^2} \tag{14}
\]

A Critical Temperature Rise, \( \Delta T_{cr} \), can be defined by evaluating Eq. (8) at the critical load, noting that for a perfect beam prior to buckling there is no deflection and therefore no associated strain term, \( \varepsilon' \):

\[
\Delta T_{cr} = \frac{P_{cr}}{\Delta \alpha Ebh} = \frac{1}{12\Delta \alpha} \left( \frac{\pi h}{L} \right)^2 \tag{15}
\]

Utilizing these two relations and by simple examination of Eqs. (4), (7) and (13) nondimensional forms of deflection \((\delta)\), eccentricity \((e)\), axial load \((\eta)\), maximum compressive stress \((\Sigma)\) and temperature rise \((\theta)\) have been defined respectively as:

\[
\delta = \frac{d_{MAX}}{h} \tag{16}
\]

\[
e = \frac{e}{h} \tag{17}
\]

\[
\eta = \frac{\pi}{2} \sqrt{\frac{P}{P_{cr}}} = \frac{L}{2\sqrt{EI}} \tag{18}
\]

\[
\Sigma = \frac{\sigma_{MAX}}{E} \left( \frac{L}{h} \right)^2 \tag{19}
\]

\[
\theta = \frac{\Delta T}{\Delta T_{cr}} = \frac{12\Delta \alpha \Delta T (L/h)^2}{\pi^2} \tag{20}
\]

Nondimensional forms of the main relations Eqs. (4), (7) and (13) are obtained by rearranging and substituting in Eqs. (16)–(20), yielding

\[
\delta = \varepsilon [\sec \eta - 1] \tag{21}
\]

\[
\Sigma = \eta^2 \left[ 1 + \varepsilon \sec \eta \right] \tag{22}
\]

\[
\theta = \left( \frac{2\eta}{\pi} \right)^2 \left[ 1 + \frac{3}{4} \varepsilon^2 \left\{ \tan \frac{\eta \cos 4\eta}{2\eta} + \tan^2 \eta \left( 1 + \frac{\sin 4\eta}{4\eta} \right) \right\} \right] \tag{23}
\]

This set of nondimensional equations was solved numerically using MATLAB to eliminate the nondimensional axial load, \( \eta \). Curves for central beam deflection, \( \delta \), maximum compressive stress, \( \Sigma \), and its corresponding stress components are shown in Figs. 7–9, respectively, as a functional of temperature rise, \( \theta \).

Figs. 7–9 show that for a perfectly symmetric beam, \( \varepsilon = 0 \), there is zero deflection \((\delta=0)\) and linearly increasing stress up until buckling occurs at the critical temperature, \( \theta = 1 \). Deflection and stress then increase in a discontinuous manner with temperature. For imperfect beams, \( \varepsilon \neq 0 \), continuous nonlinear deflections and stresses are predicted. At low temperature rises, \( \theta \ll 1 \), the beam behavior is dominated by axial compression; the beam deflection and stress increase linearly with \( \theta \). At high temperatures, \( \theta > 1 \), bending begins to be appreciable, leading to larger deflections and therefore large strain. In this range, the strain term dominates, limiting the beam to finite deflections. At intermediate temperatures between these two regions, \( 0.5 < \theta < 1 \), the nonlinear behavior of the deflection and stress curves are extremely sensitive to \( \varepsilon \).

Figs. 7 and 8 provide succinct nondimensional design curves for thermally buckled eccentric beams in MEMS systems. These curves, along with the preceding analysis, capture the complex and highly nonlinear behavior exhibited in thermally buckled...
Fig. 9. Nondimensional axial, $\Sigma_A$, and bending, $\Sigma_B$, stress components for $\varepsilon = 0.0125$.

beams. The beam shape, central deflection and state of stress have all been modeled as they vary with temperature rise and eccentricity in the regime of small beam curvatures. Appropriately, the desired nonlinear behavior associated with the buckling phenomena dominates in this regime. These nondimensional design curves allow for easy implementation of thermal buckling in a desired temperature range and to a designed deflection.

3. Fabrication

In order to experimentally validate the theory presented, doubly clamped MEMS beams were fabricated using a through-mold electroplating process as shown in Fig. 10.

Eccentricities were defined by dry etching a silicon wafer with CF$_4$O$_2$ through a 5 μm SU-8 mask, Fig. 10(a and b). A sacrificial layer of AZ5214E photoresist was then spun and patterned to define the 1.5 μm air gap between the beams and the silicon surface, Fig. 10(c). An electroplating seed layer of 50 nm Cr/500 nm Au was thermally evaporated over the sacrificial layer, Fig. 10(d), contacting the substrate to create the clamped ends of the beams. The electroplating mold was defined by patterning AZ4620 thick photoresist and the beams were created by electrodeposition in a nickel sulfamate electroplating bath, Fig. 10(e and f). Finally the beams were released in acetone with ultrasonic agitation.

A similar process to make freestanding nickel structures using photoresist as a sacrificial layer was reported by Song and Ajmera [9]. In their work, SU-8 was used to create a high aspect ratio electroplating mold, rather than photoresist. It was observed that during the SU-8 patterning process the sacrificial photoresist layer would expand, leading to significant tearing of the metal seed layer. Accordingly, an extended hard baking time was employed to make the sacrificial photoresist chemically resistant to both the SU-8 and the SU-8 developer. This method was initially attempted for the current work. It was found that a sufficiently hard baked photoresist layer was difficult to release. This was likely due to the large under-etch distances of the wide beams studied for the current application. Alternatively, the process flow shown in Fig. 10 and discussed above was developed.

Using SU-8 for the electroplating mold [9] would allow for the creation of high aspect ratio structures. The planar nature of the beams investigated in the current work alleviates this requirement; thick photoresist (AZ4620) was used in place of SU-8. This eliminated the need for extreme hard baking of the sacrificial layer, thus making the release process trivial. A multiple step
spin coating process was used to create a thick low aspect ratio electroplating mold greater than 70 μm deep. To achieve these thicknesses AZ4620 photoresist was spun on then soft baked for 30 s at 115°C; this was repeated until the desired thickness was reached.

The gold seed layer thickness of 500 nm is larger than those typically used in MEMS electroplating processes [9–11]. Seed layer thicknesses of 100–200 nm are reported as adequate for the purpose of nickel electroplating. It was found however that after evaporating the seed layer over the sacrificial layer the subsequent baking steps of the photoresist mold caused dramatic local buckling of the seed layer around the free edges. This effect can be seen in Fig. 11. Increasing the electroplating seed layer thickness eliminated this phenomenon and resulted in a uniformly planar electroplating base.

For the purposes of validating the design curves presented in this work, several 300 μm wide beams were fabricated corresponding to two different eccentricity ratios, \( e/h \), and five different slenderness ratios, \( L/h \). Fig. 12 shows one such beam geometry at increasing magnification while Table 1 lists all the geometries fabricated for the current work.

Thickness, eccentricity and surface roughness measurements were taken using a Dektak Profilometer. The surface roughness was 1 μm, while the variation in thickness across a single beam was measured to be less than 2 μm. The nickel electroplating process was optimized to create a deposition with both low tensile residual stress as well as high yield strength, as per recommendations from Fritz et al. [11], Schlesinger and Paunovic [12] and Dunney [13]. The values used in the model for the modulus and the coefficient of thermal expansion, \( E = 200 \text{ GPa} \) and \( \Delta \alpha = 10^{-5} \text{ C}^{-1} \), were taken from [11] and [12], respectively. The plating conditions used in this work, as listed in Table 2, resulted in a deposition rate of approximately 7 μm/h.

<table>
<thead>
<tr>
<th>Beam</th>
<th>Thickness (h) (μm)</th>
<th>Length (L) (mm)</th>
<th>Eccentricity (e) (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>30</td>
<td>1</td>
<td>1.5</td>
</tr>
<tr>
<td>B</td>
<td>30</td>
<td>2</td>
<td>1.5</td>
</tr>
<tr>
<td>C</td>
<td>30</td>
<td>3</td>
<td>1.5</td>
</tr>
<tr>
<td>D</td>
<td>60</td>
<td>2</td>
<td>0.75</td>
</tr>
<tr>
<td>E</td>
<td>60</td>
<td>3</td>
<td>0.75</td>
</tr>
<tr>
<td>F</td>
<td>60</td>
<td>4</td>
<td>0.75</td>
</tr>
</tbody>
</table>

**Table 2**

<table>
<thead>
<tr>
<th>Composition</th>
<th>Operating conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>500 g/L</td>
<td>Ni(SO₄)₂ (NH₄)₂ pH 4–4.5</td>
</tr>
<tr>
<td>30 g/L</td>
<td>H₃BO₃ Temperature 35°C</td>
</tr>
<tr>
<td>3 g/L</td>
<td>Laurel sulfate Current density 10 mA/cm²</td>
</tr>
</tbody>
</table>

With a sulfur activated anode and mechanical agitation.
4. Experimental setup

The beam central deflection, \( d_{\text{MAX}} \), was measured experimentally with a Philtec Fiber Optic Displacement Probe for the six microfabricated nickel beams listed in Table 1. The beam temperature was controlled using a thin film heater and a thermocouple. A schematic of the test setup used is shown in Fig. 13.

5. Results and discussion

The beam central deflection versus temperature rise is plotted against the theoretical predictions for all six beams. Fig. 14 shows two graphs corresponding to the two eccentricity ratios considered in this work. Beams A, B and C were electroplated on the same wafer and have an eccentricity ratio of \( e/h = 0.05 \), similarly beams D, E and F were fabricated on a second wafer and have an eccentricity ratio of \( e/h = 0.0125 \). The theoretical results shown here are obtained by evaluating the nondimensional predictions for each beam’s specific geometry and adjusting to account for the residual stress. The designed tensile residual stress in the beams will lead to an actuation temperature offset. An initial increase in temperature will be required to overcome the tensile stress imparted during microfabrication. The measured temperature offsets, \( \Delta T_0 \), were used to determine the residual tensile stress, \( \sigma_0 \), in the nickel deposition for the two sets of beams fabricated, where:

\[
\sigma_0 = E \Delta \alpha \Delta T_0 \tag{24}
\]

Beams A, B and C were offset by a temperature difference of 27 °C, while beams D, E and F were offset by a temperature difference of 31 °C. The corresponding residual tensile stresses are 54 and 62 MPa, respectively for the two sets. These stresses are in good agreement with the values predicted by Schlesinger and Paunovic [12] and Dunney [13]. A high purity bath of the composition and operating conditions used in this work has been reported to produce depositions with residual tensile stresses of 55 MPa or less.

Accounting for the temperature offset caused by the residual tensile stress, the \( \Delta T \) used in the calculation of \( \theta \) in Eq. (20) can been defined as the temperature rise above the zero stress state, rather than ambient conditions. Accordingly the data for the three beams at a common eccentricity ratio, \( \varepsilon = e/h \), will collapse to a single nondimensional curve and can be compared against the predictions.

Fig. 15 shows relatively good agreement between the theoretical predictions and the measured deflections. When presented nondimensionally the 30 μm beams (A, B and C) show a larger amount of scatter in the data than that of the 60 μm beams (D, E and F). This is explained by the sensitivity of the optical probe measurements; the thinner beams have smaller deflections leading to higher percent errors.

It can also be seen that the predictions are less accurate in the temperature range of \( 0.5 < \theta < 1 \). As noted previously the shape of the deflection curve in this region is very sensitive to the asymmetry of the beam, modeled in this work by the eccentricity ratio \( \varepsilon = e/h \). This would suggest that modeling of the beams with a designed imperfection, in the form of an eccen-
electricity, has not completely captured the true imperfections of the beams tested. Two actual imperfections in the beams, the surface roughness and the thickness variation, were both measured to be on the order of the designed eccentricities. This seems to be a logical source of differences between the predictions and test results in this particular temperature range. Nonetheless, the predicted behavior qualitatively and quantitatively agrees with the data.

5.1. Hysteresis and yield

The beams tested in the current work were scanned with a profilometer before and after thermal actuation to examine the onset and effect of plastic deformation. The central deflection of the beams after returning back to ambient conditions is listed in Table 3 along with the calculated maximum stress experienced by each beam. A nickel electroplating bath of the composition and operating conditions used in this work has been reported to produce depositions with yield strengths of 400–600 MPa [11–13]. Table 3 shows the onset of an appreciable hysteresis effect occurring around 450 MPa for the beams tested in this work.

5.2. Repeatability

Two of the low stress beams exhibiting negligible hysteresis effects were additionally tested to examine the repeatability of the deflection measurements. Fig. 16 shows good repeatability of the test results for beams actuated with negligible plastic deformation.

6. Conclusion

Closed form models of the deflection and state of stress in thermally buckled clamped beams have been developed in the current work. Nondimensional characteristic curves of both the central beam deflection and the maximum compressive stress as a function of beam temperature rise have been generated. The effect of eccentricities on the thermomechanical behavior of the beams has also been investigated. A process flow has been developed for the fabrication of clamped nickel planar beams on a silicon substrate. Using a through-mold nickel electroplating process, eccentric beams were built over a thin air gap. Various beam geometries were fabricated and thermally actuated.

<table>
<thead>
<tr>
<th>Beam</th>
<th>Permanent deflection (µm)</th>
<th>Calculated maximum stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>0.50</td>
<td>75</td>
</tr>
<tr>
<td>B</td>
<td>1.00</td>
<td>150</td>
</tr>
<tr>
<td>A</td>
<td>1.25</td>
<td>400</td>
</tr>
<tr>
<td>F</td>
<td>2.00</td>
<td>150</td>
</tr>
<tr>
<td>E</td>
<td>4.00</td>
<td>250</td>
</tr>
<tr>
<td>D</td>
<td>10.25</td>
<td>450</td>
</tr>
</tbody>
</table>
Measurements of deflection versus temperature were taken and the results were used to validate the model. Central deflections over 50 μm were achieved with temperature rises of less than 100 °C. Good agreement was seen between the highly nonlinear theoretical predictions and the experimental results for all of the beams considered. Additionally, the calculated residual stress, yield strength and overall quality of the electroplated nickel shows general agreement with the findings reported from the pertinent literature. It has therefore been shown that the thermal buckling phenomena can be modeled and manipulated into temperature sensitive nonlinear actuators, with the desired parallel plate geometry of Fig. 2, to be implemented into a self-adaptive MEMS cooling scheme.

6.1. Ongoing work

The validated model for thermally buckling beams has been implemented in the design of several thermally adaptive microvalves. Using the fabrication results contained herein, geometries similar to the one shown in Fig. 2 have been constructed to make novel thermally adaptive microvalves. Full characterization of the thermally adaptive microvalves under various loading conditions, along with a modeling technique for the valve flow resistance and an expanded fabrication process will be reported in future publications.

Acknowledgements

The authors would like to thank Richard Harniman and Eric Holihan of the Shapiro Center for Engineering and Physical Science Research at Columbia University for their clean room and microfabrication assistance. This work has used the clean room supported by the Nanoscale Science and Engineering Initiative of the National Science Foundation under NSF Award Number CHE-0117752 and by the New York State Office of Science, Technology, and Academic Research. This work has also used the shared experimental facilities that are supported primarily by the MRSEC Program of the National Science Foundation under Award Number DMR-0213574 and by the New York State Office of Science, Technology and Academic Research (NYS-TAR). The authors would like to also thank Brandon Basso for his help in building the data acquisition system used in this work.

References


Biographies

Matthew McCarthy received his MS degree in mechanical engineering from Columbia University in 2004 and his BS in aerospace engineering from Syracuse University in 2002. Currently he is a PhD student in the Thermo-Fluid Sciences Laboratory at Columbia University. His doctoral research is focused on developing microfluidic sensors and actuators for aerospace and micro-cooling applications.

Nicholas Tiliakos is a senior scientist in the Advanced Concepts Group, within the Technology Development Department at ATK-GASL. His degrees are in mechanical engineering (BS 1990, Cornell University) and aerospace engineering (MS 1993, PhD 1997, University of Illinois at Urbana-Champaign) and his research experience has involved both analytical and experimental work in diverse technical areas including: space and hypersonic propulsion, plasma physics, combustion systems, thermal analysis for aerospace vehicle thermal management and Microelectromechanical Systems. Currently, Dr. Tiliakos is working on a MEMS adaptive cooling concept in the area of passive heat transfer thermal management. Dr. Tiliakos is also an adjunct professor in the Mechanical Engineering Department at Columbia University, since 2001, where he has taught advanced thermodynamics and advanced heat transfer.

Vijay Modi is currently professor of mechanical engineering. He joined Columbia University as an assistant professor in 1986. He received his PhD degree in mechanical engineering from Cornell University in 1984. Since then he held research positions at MIT (1985–1986) and visiting faculty positions at Cornell (1984–1985) and University of California at Berkeley (1996 and 2002). Prof. Modi’s research area is in the area of heat transfer and fluid mechanics. In the last 5 years he has also been involved with multidisciplinary projects involving design and fabrication of micro-sensors and microfluidic devices as well as modeling of fluid flows and heat/mass transfer in microsystems.

Luc G. Fréchette received his PhD and SM degrees from the Massachusetts Institute of Technology (MIT), Cambridge, in 2000 and 1997, respectively, following his BEng studies at the École Polytechnique de Montréal in Canada. He currently holds the Canada Research Chair in Microfluidics and Power MEMS, and is an associate professor in mechanical engineering at the Université de Sherbrooke, Canada. From 2000 to 2004, he was an Assistant Professor at Columbia University, New York, USA. His research expertise is in microengineering of miniature systems for energy conversion (Power MEMS), such as microfabricated heat engines and fuel cells, with activities ranging from integrated device development to more fundamental microfluidics, heat and mass transfer studies in such microsystems. Dr. Fréchette also enjoys developing MEMS sensors and actuators for aerospace applications, such as silicon carbide flow sensors and micro-cooling technology. He is a member of the ASME, IEEE, and AIAA societies.