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Dual-mode pneumatic artificial muscle integrating shape memory polymer constraints: multiphysics modeling and experimental validation

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Abstract

This paper presents a dual-mode pneumatic artificial muscle integrated with three circumferentially arranged S-shaped shape memory polymer (SMP) constraints. By selectively heating the SMPs, the actuator achieves reversible switching between axial elongation and multi-directional bending, enabling four distinct deformation modes (3C, 2C1H, 1C2H, and 3H). A thermo-mechanically coupled, energy-based theoretical framework is developed to model this behavior. The static equilibrium equations are derived from the total potential energy functional—which includes the hyperelastic strain energy of the bladder, the temperature-dependent stiffness of the SMPs, and the work done by pneumatic pressure—using the principle of minimum potential energy. A high-fidelity finite element model employing a sequentially coupled thermo-pneumatic-solid strategy is also established. A prototype was fabricated and systematically tested on a custom platform integrating pneumatic, thermal, and mechanical measurement systems. The results demonstrate good agreement among the theoretical model, finite element simulation, and experimental data, with the majority of relative errors falling below 10%. The experimentally measured maximum bending curvature reaches 0.0174 mm^{-1} , and the maximum axial elongation ratio is about 18.4%. An error analysis identifies residual SMP stiffness, contact friction, and braided sleeve slippage as primary sources of discrepancy. This study provides a systematic theoretical and technical basis for designing, modeling, and controlling multi-modal soft actuators.

Nomenclature

Geometry and kinematics

Symbol	Unit	Description
L_0	mm	Initial length of actuator central axis
L	mm	Current length of actuator central axis
r_0	mm	Initial average cross-sectional radius of inner bladder
r	mm	Current average cross-sectional radius
t	mm	Wall thickness of inner bladder
b	mm	Cross-sectional width of SMP constraint
h	mm	Cross-sectional thickness of SMP constraint
θ_0	°	Initial braid angle of fiber sleeve
s	mm	Arc-length coordinate along actuator axis
R	mm	Radius of curvature of bending arc
ϕ	rad	Central angle / tip bending angle
κ	mm^{-1}	Bending curvature, $\kappa = 1/R$
κ_0	mm^{-1}	Intrinsic pre-curvature of SMP constraint after shape-setting
λ	—	Axial stretch ratio, $\lambda = L/L_0$
$\lambda_1, \lambda_2, \lambda_3$	—	Principal stretches (axial, circumferential, radial)
ε	—	Axial strain, $\varepsilon = \Delta L/L_0$

Volume and area

Symbol	Unit	Description
A	mm^2	Cross-sectional area of inner bladder
V	mm^3	Current internal volume
V_0	mm^3	Initial internal volume

Energy and work

Symbol	Unit	Description
Π	mJ	Total potential energy of the system
U_{bladder}	mJ	Strain energy of inner bladder
U_{braid}	mJ	Strain energy of braided sleeve
U_{SMP}	mJ	Strain energy of SMP constraint structures
W	MPa	Strain energy density function (Mooney-Rivlin)

Forces, moments, and pressure

Symbol	Unit	Description
P	MPa	Internal pneumatic pressure
F_{axial}	N	Axial output force (theoretical)
F / F_a	N	Axial output force / measured axial force
M_{bend}	N·mm	Bending output moment
M	N·mm	Bending moment
σ_{max}	MPa	Maximum von Mises stress
θ_{max}	°	Maximum bending angle

Hyperelastic material parameters (mooney-rivlin)

Symbol	Unit	Description
C_{10}, C_{01}	MPa	Mooney-Rivlin material parameters
I_1, I_2	—	First and second invariants of the right Cauchy-Green tensor

SMP stiffness parameters

Symbol	Unit	Description
T	$^{\circ}\text{C}$	Temperature of SMP
T_g	$^{\circ}\text{C}$	Glass transition temperature of SMP
ΔT	$^{\circ}\text{C}$	Glass transition width parameter
$K_{\text{SMP}}(T)$	$\text{N}\cdot\text{mm}/\text{rad}$	Temperature-dependent equivalent bending stiffness of SMP
K_{high}	$\text{N}\cdot\text{mm}/\text{rad}$	SMP bending stiffness in glassy state ($T \ll T_g$)
K_{low}	$\text{N}\cdot\text{mm}/\text{rad}$	SMP bending stiffness in rubbery state ($T \gg T_g$)
K_{braid}	$\text{N}\cdot\text{mm}/\text{rad}$	Equivalent bending stiffness of braided sleeve
K_{tensile}	N mm^{-1}	Equivalent tensile stiffness of SMP constraint
K_{bending}	N mm^{-1}	Equivalent bending stiffness from three-point test (F/δ)

SMP viscoelastic parameters (finite element model)

Symbol	Unit	Description
$E(T, t)$	MPa	Temperature- and time-dependent relaxation modulus
E_g	GPa	Glassy modulus ($T \ll T_g$)
E_r	MPa	Rubbery modulus ($T \gg T_g$)
E_{∞}	MPa	Equilibrium modulus at infinite time
E_i	MPa	Relaxation modulus of i -th Maxwell element
$\tau_i(T)$	s	Temperature-dependent relaxation time of i -th Maxwell element
a_T	—	Time-temperature shift factor (WLF)
C_1, C_2	—, K	WLF equation coefficients
n	—	Number of Maxwell elements in Prony series

Braided sleeve material parameters

Symbol	Unit	Description
E_1	GPa	Elastic modulus in fiber direction
E_2, E_3	MPa	Elastic moduli in transverse directions
G_{12}, G_{13}	MPa	In-plane shear moduli
ν_{12}, ν_{13}	—	Poisson's ratios

Sensors and kinematic measurement

Symbol	Unit	Description
V_i	V	Real-time output voltage of i -th flexible displacement sensor ($i = 1, 2, 3$)
$V_{0,i}$	V	Zero voltage of i -th sensor in initial Free State
k_i	$\text{mm}\cdot\text{V}^{-1}$	Calibration sensitivity coefficient of i -th sensor
Δl_i	mm	Length change of i -th sensor
$\Delta \bar{l}$	mm	Average length change of three sensors
ΔL	mm	Axial displacement measured by laser displacement sensor
s_0	mm	Initial length of sensor coverage region on actuator surface
s_i	mm	Deformed length of local region covered by i -th sensor
ϕ_i	rad	Circumferential angular position of i -th sensor
y_i	mm	Distance from i -th sensor to neutral axis
α	rad	Central angle corresponding to local sensor region
β	—	Ratio of local region length to total arc length (s_0/L_0)
a_i	mm	i -th element of coefficient matrix A
A	—	Coefficient matrix for least-squares bending angle reconstruction
b	mm	Measurement vector
c_0, c_1, c_2	—	Polynomial fitting coefficients for axial displacement calibration
δ	mm	Mid-span deflection in three-point bending test
R^2	—	Correlation coefficient of curve fitting

1. Introduction

Soft robotics represents a cutting-edge field with broad application prospects in areas such as medical rehabilitation, bionic exploration, and intelligent manufacturing [1]. This potential stems from their inherent compliance, adaptability to unstructured environments, and capacity for safe human-robot

interaction. Unlike traditional rigid ones, soft robots are constructed from flexible materials, enabling continuous, smooth deformations and complex motions [2]. These capabilities make them exceptionally well-suited for tasks in unstructured environments [3]. Consequently, the performance of soft actuators, as the core components, directly determines the capabilities and potential applications of the entire robotic system [4].

Pneumatic artificial muscles (PAMs) are among the most widely adopted actuators in soft robotics, valued for their high power-to-weight ratio, structural simplicity, and biomimetic motion. The classic McKibben-type PAM consists of a hyperelastic inner bladder constrained by a braided fiber sleeve [5]. Inflation causes the bladder to expand radially, while the braided sleeve converts it into axial contraction. However, conventional PAMs were limited to a single axial contraction motion, which is insufficient for multi-degree-of-freedom and multi-modal actuation in complex tasks [6]. To enrich the motion repertoire of PAMs, researchers have developed various modified configurations, such as asymmetric constraints for directional bending or multi-chamber designs for multi-directional motion [7]. Nevertheless, these approaches often result in increased structural complexity, fixed deformation modes, and a lack of active tunability.

Shape memory polymers (SMPs), a representative class of smart materials, can exhibit significant stiffness changes and recover a pre-programmed shape under external stimuli such as temperature, light, or electricity [8]. The integration of SMPs with PAMs enables active regulation of actuator stiffness and reversible switching between deformation modes, paving the way for next-generation intelligent soft actuators [9]. However, the incorporation of SMPs also introduces complex multi-physics coupling issues, including pneumatic actuation, hyper-elastic large deformation, thermally induced phase transition [10], and nonlinear structural contact [11–13]. These coupled behaviors present significant challenges for theoretical modeling, numerical simulation, and experimental characterization.

Expanding the functional capabilities of PAMs by integrating them with SMPs is a key area of contemporary research. Pioneering this area, Takashima *et al* introduced a configuration where an SMP sheet, embedded with heating wires, was externally attached to a PAM [14, 15]. This design significantly improved the bending range of motion and fabrication repeatability. Subsequently, Jiang *et al* from Chongqing University developed a reconfigurable soft helical actuator with a variable-stiffness skeleton. An SMP-based skeleton is wrapped around the bottom of a fiber-reinforced chamber; heating releases this constraint, enabling a real-time switch between bending and helical motions [16]. Meanwhile, Gong *et al* from the University of Science and Technology of China combined the thermally induced shape memory property of liquid crystal elastomers with the magnetic guidance capability of magnetorheological fluids to develop a soft composite actuator with multi-stimulus response [17]. By synergistically combining photothermal effects and magnetic fields, they achieved complex motions including bidirectional crawling and in-situ steering [13, 18].

In addition, Pagoli *et al* systematically reviewed seven influential design types of soft fluidic actuators, providing an important reference for subsequent innovative designs [19]. Collectively, the research outcomes from the aforementioned teams offer rich configuration references and methodological inspirations for the dual-mode PAM with integrated S-shaped SMP constraint proposed in this paper. A quantitative comparison of these designs with the present work is summarized in table 1.

Building on the foregoing discussion, this paper proposes a novel dual-mode PAM integrated with three circumferentially equally spaced S-shaped SMP constraints. By independently controlling the thermal state of each SMP constraint, the actuator can achieve reversible switching between axial contraction and axial elongation modes at low and high temperatures, respectively. Furthermore, by selectively heating a subset of the constraints, it can generate directional bending deformations toward different directions, thereby realizing four programmable deformation modes, including axial motion and multi-directional bending. This paper presents systematic mechanical modeling, multiphysics simulation analysis, and experimental validation of the actuator. The objectives are to reveal its thermo-mechanical coupling mechanism and to establish a complete methodology for performance prediction and structural optimization, thereby providing theoretical support and a technical foundation for the development of intelligent soft actuators capable of multi-modal and multi-directional deformation.

Compared with the aforementioned SMP-PAM designs, the actuator proposed in this work offers three distinguishing capabilities that have not been simultaneously demonstrated in prior studies. First, the three circumferentially distributed S-shaped SMP constraints, each independently addressable via its embedded heating film, enable the actuator to switch among four distinct deformation modes (3C, 2C1H, 1C2H, and 3H) within a single device, whereas previous designs are limited to at most two modes. Second, the selective activation of different constraint subsets produces multi-directional bending toward the rigid constraints, a capability absent from single-constraint or single-chamber SMP-PAM configurations. Third, this work provides a complete and internally consistent methodology chain spanning

Table 1. Comparison with prior SMP-integrated PAM designs.

Feature	Takashima <i>et al</i> (2025) [14]	Jiang <i>et al</i> (2024) [16]	This Work
Actuation modes	1 (bending)	2 (bending + helical)	4 (3C, 2C1H, 1C2H, 3H)
Multi-directional bending	No	No	Yes (3 directions, 120°)
SMP configuration	Segmented sheet, SMA wire	Single skeleton, chamber bottom	3 S-shaped, 120° equally spaced
Max. bending curvature / angle	~6.2°–9.5°	Not reported	0.0174 mm ^{−1}
Max. blocking force / moment	~3.9–8.6 N	Not reported	24.9 N / 156 N-mm
Max. elongation ratio	Not reported	Not reported	18.4%
Stiffness ratio (glassy / rubbery)	~300	Not reported	~80
Theoretical model	No	Yes (minimum potential energy)	Yes (energy-based)
Multiphysics FEM simulation	No	No	Yes (thermo-pneumatic-solid coupled)
Integrated sensing	No	No (external camera only)	Yes (FDS kinematic reconstruction)
Operating pressure	~0.25 MPa	Not reported	0.1–0.3 MPa

an energy-based theoretical model, a high-fidelity sequentially coupled finite element framework, and systematic experimental validation with integrated flexible sensing for in-situ kinematic reconstruction. These three capabilities, taken together, distinguish the present actuator from prior SMP-PAM designs and establish a new baseline for multi-modal soft actuation with integrated sensing.

2. Design and manufacturing

2.1. Design approach

The basic structure of a bending-type McKibben PAM is fundamentally composed of an elastic inner tube, an outer constraining braided sleeve, and sealing end caps. During inflation, the inner tube expands radially, and the sleeve's anisotropic properties convert this expansion into a bending motion. Conventional designs achieve this directional bending by attaching a rigid constraint, such as a beam, along one side of the sleeve to induce asymmetric deformation [20–22].

This improved design replaces the single constraint beam with three S-shaped elements made of SMP. These elements are fixed at 120° intervals around the circumference of the braided sleeve. The geometry of these S-shaped structures creates a non-uniform circumferential constraint, which induces controllable bending. Moreover, by exploiting the thermal responsiveness of SMP, they can change their stiffness and shape under thermal stimuli, thereby dynamically adjusting the bending angle and force [23, 24]. This enhances the compliance and environmental adaptability of the actuator. The choice of three constraints arranged at 120° intervals represents the minimum configuration capable of generating multi directional bending in a plane: two constraints would restrict bending to a single axis, while four or more constraints would increase fabrication complexity and wiring requirements without providing additional deformation modes beyond those already achievable with three. The 120° spacing ensures symmetric circumferential coverage and balanced force distribution among the constraints [20].

At ambient temperatures (below the glass transition temperature T_g of the SMP), the three rigid S-shaped SMP constraint structures retain their initially designed curved shape, providing circumferentially symmetric constraint on the braided sleeve. As a result, the radial expansion of the muscle upon inflation is significantly restricted. However, when heated above T_g by an external heat source, the SMP undergoes a sharp drop in elastic modulus, transitioning from a rigid glass state to soft rubbery state. Consequently, the S-shaped structures are easily straightened by the internal stresses and pneumatic pressure, effectively neutralizing their constraining capability and allowing the actuator to move freely.

Directional bending is achieved by selectively activating the SMP constraint structures. When two of the three circumferentially distributed structures are heated, the unheated beam that remains rigid provides a strong asymmetric constraint on one side of the braided sleeve. Consequently, radial expansion of the pneumatic muscle on that side is significantly restricted. Upon inflation, the expansion energy of the inner bladder is converted into directional bending deformation. Conversely, heating a single structure causes the actuator to bend towards the midpoint of the two remaining rigid constraints.

If all three SMP constraint beams are heated simultaneously, the circumferential constraint on the braided sleeve becomes relatively uniform again. The pneumatic actuation then primarily drives the PAM

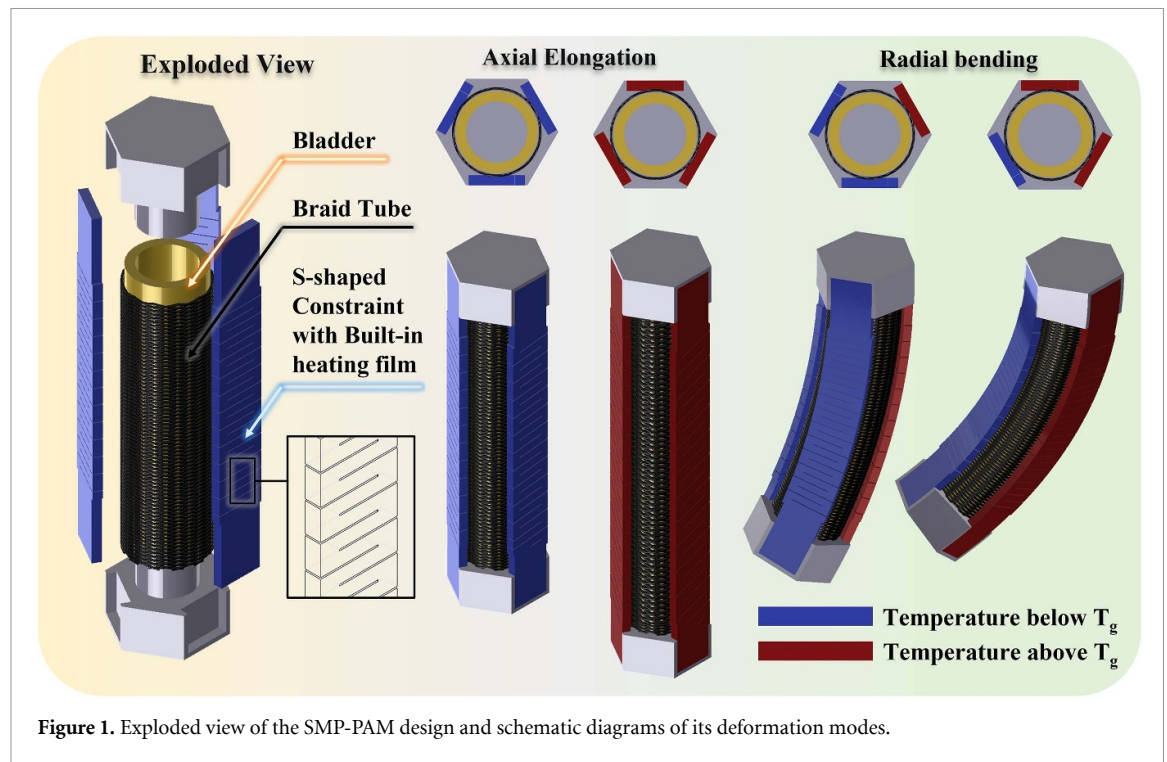


Figure 1. Exploded view of the SMP-PAM design and schematic diagrams of its deformation modes.

to produce axial elongation motion, similar to a conventional McKibben muscle. Upon removing the heat source, the SMP cools and solidifies, regaining its original rigidity and S-shaped geometry. The deformation mode of the PAM immediately switches back to directional bending. Consequently, selective and reversible thermal stimulation enables dynamic switching between axial and bending modes, as well as precise control over the bending direction within a single integrated device.

In summary, by independently controlling the temperature of its three S-shaped SMP constraints, the proposed dual-mode PAM achieves four distinct deformation modes. These modes are categorized by the number of constraints in the glassy (C) versus rubbery (H) states: 3C, 2C1H, 1C2H, and 3H. The symmetric configurations (3C and 3H) facilitate axial motion, representing high-stiffness and low-stiffness states, respectively. Conversely, the asymmetric configurations (2C1H and 1C2H) induce directional bending away from the thermally activated constraints. This selective modulation framework enables the actuator to switch dynamically between axial extension and multi-directional bending, providing a versatile platform for multi-modal soft actuation. The exploded view of the proposed SMP-PAM design and the schematic diagrams of its four deformation modes are presented in figure 1.

2.2. Material selection

The inner bladder material must possess high elasticity, low hysteresis, fatigue resistance, excellent airtightness, and compatibility with the operating temperature range of the SMP. Based on a comprehensive comparison, silicone rubber was selected as the base material. It exhibits excellent mechanical properties and thermal stability: tensile strength >2.5 MPa, elongation at break $>800\%$, and an operating temperature range from -45 °C– 230 °C, fully covering the glass transition temperature range of the SMP. The hyper-elastic constitutive parameters of the material will be calibrated through subsequent uniaxial tensile tests.

The braided sleeve is made of a high-strength polyethylene terephthalate (PET) fiber braided mesh with a fiber diameter of 0.2 mm, a braid density of 30 meshes per inch, and an initial braid angle $\theta_0 = 25^\circ$. PET fibers exhibit a high tensile modulus (>10 GPa), low creep, and excellent fatigue resistance, effectively fulfilling the functional requirements of radial constraint and axial guidance. Consequently, the deformation mode of the actuator remains controllable during pressurization.

The SMP material is required to meet the following conditions: a glass transition temperature T_g within a readily controllable range (50 °C– 90 °C), a large modulus variation between the glassy and rubbery states (glassy modulus >1 GPa, rubbery modulus <50 MPa), good 3D printability, and compatibility with the actuator integration process. Based on these requirements, a commercial thermosetting SMP filament was selected as the raw material [25]. The thermomechanical parameters of the material will be systematically characterized through subsequent dynamic mechanical analysis (DMA) tests [16].

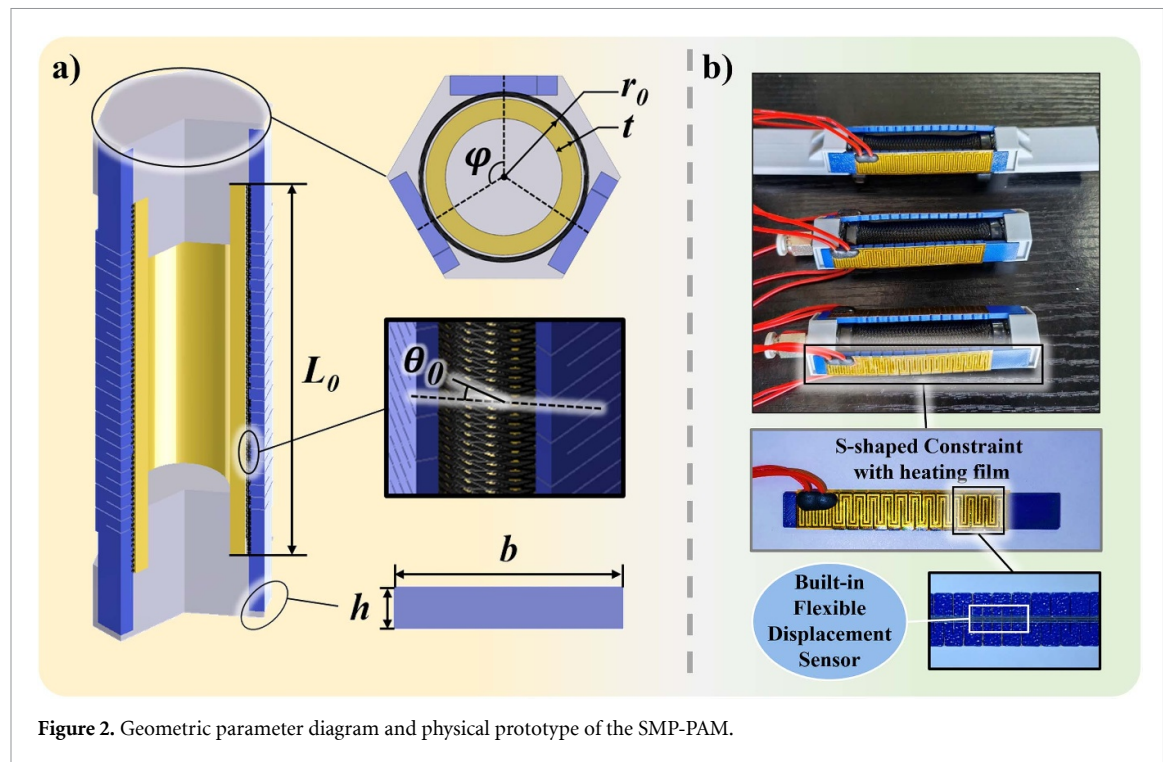


Figure 2. Geometric parameter diagram and physical prototype of the SMP-PAM.

2.3. Manufacturing process

The fabrication of this improved PAM begins with the preparation and assembly of its core components. First, a high-elasticity silicone rubber tube is selected as the inner bladder. Its ends are hermetically bonded and fixed with 3D-printed rigid end caps that incorporate an inflation port. Subsequently, a braided sleeve made of high-strength fibers is tightly fitted over the inner bladder. The sleeve length is slightly shorter than the free expanded length of the inner bladder upon inflation, ensuring the generation of axial force during expansion.

Three identical S-shaped curved beam-like structures were precisely fabricated via fused deposition modeling 3D printing using an SMP filament with a specific glass transition temperature (T_g). These SMP structures were shaped above T_g and then cooled to retain the preset S-configuration. After printing, the samples were annealed in an 80 °C oven for 1 h to eliminate residual stresses induced during printing [26]. The annealed samples were then surface-polished. A miniature polyimide heating film was attached to the middle of the bottom surface of each SMP structure, with leads drawn out and secured [27]. Finally, at room temperature, these structures were longitudinally affixed to the outer surface of the braided sleeve, equally spaced at 120° intervals along the circumference, with their length matching the working length of the braided sleeve.

After fabrication, an initial state inspection was performed on the pneumatic muscle prototype. Geometric dimensions were measured using a vernier caliper to obtain the actual length and outer diameter, which were then compared with the design values. The prototype was pressurized to 0.2 MPa and immersed in water to check for bubble leakage, thereby verifying its airtightness. The initial bending angle of the actuator was measured under zero pressure to ensure that the pre-bending induced by the SMP constraint structures remained within the expected range. Finally, the heating films were energized, and the temperature distribution was monitored using an infrared thermal imager to ensure heating uniformity. The finished actuator prototype is shown in figure 2. A total of three sets of prototypes were fabricated for subsequent experimental studies.

3. Modeling

This section constructs a mechanical model for an improved PAM with SMP constraints, aiming to describe its mechanical behavior in two modes: bending mode and axial mode. Due to the presence of SMP constraints, the model must account for the ‘switching’ effect caused by the rigid/compliant state transition of SMP constraints, as well as the bending resulting from the resulting asymmetric constraints. For computational convenience, the following assumptions are introduced in this section.

- (1) The pneumatic muscle cross-section remains circular throughout deformation, with radius uniformly distributed along the arc length;
- (2) The pneumatic muscle exhibits uniform bending curvature, and fibers maintain an equi-pitch helix after bending;
- (3) Fiber length in the woven tube remains constant during bending, though the weave angle varies circumferentially;
- (4) The inner liner is an isotropic, incompressible material undergoing uniform deformation.

For the actuator geometry and loading conditions considered in this study, the above assumptions and the two additional simplifications are justified as follows. The circular cross-section assumption (1) is standard in PAM analytical modeling, as the high-modulus braided sleeve (>10 GPa) effectively resists local ovalization within the pressure range employed here (0.1–0.3 MPa) [7]. The constant curvature assumption (2) follows from the longitudinal uniformity of the SMP constraints, which yields an approximately constant bending moment along the actuator axis; this has been validated for fiber-reinforced pneumatic actuators of comparable scale [20], and the FEM results in section 4.4 confirm that curvature varies by less than 5% along the actuator length. The uniform radius condition is a direct consequence of assumptions (1) and (2) [7]. The braid strain energy is negligible because the PET fiber modulus exceeds the silicone rubber modulus by four orders of magnitude; the sleeve acts primarily as a kinematic constraint rather than an energy storage element [5]. Finally, the uniform pressure condition is justified under the quasi-static loading protocol (stepped pressurization with 10 s holds), as pressure equilibration occurs far faster than structural deformation in the small bladder volume ($\sim 10^4$ mm³) [6].

Based on these assumptions, this section establishes a total potential energy function using the principle of virtual work. This function takes temperature and air pressure as inputs and generalized coordinates (axial strain or bending curvature) as variables. The equilibrium state is determined through energy minimization, and the temperature dependence of the SMP modulus enables a unified description of both deformation modes.

3.1. Defining geometric relationships and generalized coordinates

The proposed dual-mode PAM integrated with SMP constraints consists of three main components. (1) The inner bladder is made of silicone rubber and forms a hollow thin-walled cylinder with initial length L_0 , initial radius r_0 , and wall thickness t . (2) A fiber braided sleeve is tightly wrapped around the outer surface of the inner bladder, with an initial braid angle θ_0 . (3) Three S-shaped SMP constraint structures are equally spaced circumferentially at 120° intervals and attached to the outer surface of the braided sleeve, each characterized by a cross-sectional width b and thickness h .

To facilitate mechanical modeling, two coordinate systems are established in this paper: a global coordinate system and a local coordinate system. The global coordinate system $OXYZ$ is defined with its origin at the centroid of the fixed end cross-section of the actuator. The Z -axis is aligned with the initial axis of the actuator, and the X -axis points toward the bending direction (i.e. the side opposite the three SMP constraint structures). The local coordinate system is established along the actuator axis with an arc-length coordinate $s \in [0, L]$, which is used to describe the deformed geometric configuration.

To establish an energy-based mechanical model, the kinematics of the actuator must be described first. This model assumes that, under quasi-static deformation, the central axis of the PAM can be represented as a circular arc while the cross-section remains circular. Based on the above assumptions, three generalized coordinates—namely, the axial stretch ratio, the bending curvature, and the average cross-sectional radius—are defined to characterize the deformation state of the system.

The axial stretch ratio is defined as $\lambda = L/L_0$, where L_0 and L denote the initial length and current length of the central axis, respectively. This ratio describes axial deformation. The bending curvature is defined as $\kappa = 1/R$, with R being the radius of curvature of the bending arc. The corresponding central angle is $\phi = \kappa L$. This curvature parameter quantifies the degree of bending. The average cross-sectional radius r is determined by the coupling between the geometric constraint of the braided sleeve and the generalized coordinates.

Based on the fundamental assumption of fiber inextensibility, the fiber length remains constant during deformation. The constraint relationship between the axial stretch ratio λ and the radius ratio r/r_0 is established as follows [7]:

$$\lambda^2 + \left(\frac{r}{r_0}\right)^2 \tan^2 \theta_0 = \sec^2 \theta_0. \quad (1)$$

For a pneumatic muscle in a bent state, the internal volume during operation must be obtained by path integration along the circular arc. Assuming that the cross-sectional area $A = \pi r^2$ is uniformly distributed along the axis, the volume can be precisely calculated as follows [28]:

$$V = \pi r^2 L \cdot \frac{\sin(\kappa L/2)}{\kappa L/2}. \quad (2)$$

When the curvature is small ($\kappa L \ll 1$), a Taylor series expansion can be employed as an approximation:

$$V \approx \pi r^2 L \left[1 - \frac{(\kappa L)^2}{24} \right]. \quad (3)$$

Then the initial value of the volume can be expressed as:

$$V_0 = \pi r_0^2 L_0. \quad (4)$$

3.2. Establishing system energy relations

With the kinematic description and generalized coordinates defined, the energy-based modeling centers on formulating the system's total potential energy Π of the system. This total potential energy consists of the elastic strain energy of each internal component and the work done by the external air pressure. Specifically, the total potential energy of the system can be expressed as the sum of the following four terms [29]:

$$\Pi(\lambda, \kappa; P, T) = U_{\text{bladder}} + U_{\text{braid}} + U_{\text{SMP}}(T) - P(V - V_0) \quad (5)$$

where P is the internal surface pressure, T is the temperature of the SMP constraint structures, and V_0 is the initial volume of the inner bladder.

The silicone rubber inner bladder is modeled as an incompressible hyperelastic material. The classical Mooney–Rivlin constitutive model is employed to describe its strain energy density function. Under the assumption of uniform deformation, the principal stretches experienced by the inner bladder are the axial stretch ratio, the circumferential stretch ratio, and the radial stretch ratio, which can be expressed as follows:

$$\begin{aligned} \lambda_1 &= \lambda \\ \lambda_2 &= \frac{r}{r_0} \\ \lambda_3 &= (\lambda_1 \lambda_2)^{-1}. \end{aligned} \quad (6)$$

The overall strain energy density function can be expressed as follows:

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) \quad (7)$$

where I_1 and I_2 are the first and second stress invariants, respectively, C_{10} and C_{01} are the material parameters of the hyperelastic model,

$$\begin{aligned} I_1 &= \lambda_1^2 + \lambda_2^2 + \lambda_3^2 \\ I_2 &= \lambda_1^2 \lambda_2^2 + \lambda_2^2 \lambda_3^2 + \lambda_3^2 \lambda_1^2. \end{aligned} \quad (8)$$

The total strain energy is the product of the strain energy density and the volume, representing the elastic potential energy stored in the inner bladder material during expansion and bending, and is given by:

$$U_{\text{bladder}} = WV. \quad (9)$$

The high-strength braided sleeve primarily provides kinematic constraints. Its fibers possess extremely high axial stiffness, so the strain energy due to fiber elongation is generally negligible, i.e. $U_{\text{braid}} = 0$. However, during bending deformation, the fibers may rearrange and develop slight bending strain. Therefore, to simplify the model, the equivalent effects of this energy and the friction dissipation between the braided sleeve and the inner bladder are incorporated into an empirical bending stiffness term K_{braid} . Its influence is then combined with that of U_{SMP} to jointly account for bending deformation.

The strain energy U_{SMP} of the SMP constraint structures is key to achieving multi-mode deformation prediction in the model. For modeling purposes, the three S-shaped SMP structures are treated as a continuous, distributed linear torsional spring on the outer (convex) side of the bending actuator. The restoring moment is proportional to the bending curvature κ and the equivalent bending stiffness K_{SMP} of the SMP structures. K_{SMP} is a strong function of temperature T , characterizing the dramatic variation of the SMP material modulus with temperature [30]:

$$U_{\text{SMP}}(T) = \frac{1}{2} K_{\text{SMP}}(T) \cdot L_0 \cdot (\kappa - \kappa_0)^2 \quad (10)$$

where κ_0 is the intrinsic pre-curvature of the SMP constraint structure after cooling and setting (introduced by its S-shaped geometry). $K_{\text{SMP}}(T)$, a strong function of temperature T , is determined by the modulus and structural geometry of the SMP material. A hyperbolic tangent function is used to describe its temperature dependence as follows [31]:

$$K_{\text{SMP}}(T) = K_{\text{low}} + \frac{K_{\text{high}} - K_{\text{low}}}{2} \cdot \left[1 + \tanh \left(-\frac{T - T_g}{\Delta T} \right) \right] \quad (11)$$

where K_{high} and K_{low} are the equivalent bending stiffness in the low-temperature glassy state ($T \ll T_g$) and the high-temperature rubbery state ($T \gg T_g$), respectively, T_g is the glass transition temperature, and ΔT is the transition width parameter. The above parameters are calibrated by DMA tests and three-point bending tests of the SMP material.

The parameters K_{high} , K_{low} , T_g , and ΔT in equation (11) were extracted from DMA storage modulus curves and three point bending tests conducted on the SMP material used in this study (section 5.2). The tanh function serves as a phenomenological fit to the equilibrium stiffness temperature profile and is intended for quasi static analysis where rate dependent effects are not activated. For the finite element simulations (section 4.2), a generalized Maxwell viscoelastic model with Williams–Landel–Ferry (WLF) time temperature shifting is employed to capture the rate dependent relaxation behavior of the SMP that becomes relevant under transient thermal loading. Both representations describe the same SMP material and are calibrated to the same DMA data set. The theoretical formulation neglects viscoelasticity because it solves for equilibrium configurations under steady state thermal and pressure conditions; under such quasi static conditions, the relaxation modulus approaches the equilibrium value, and the tanh representation in equation (11) provides a consistent first order description of the temperature dependent stiffness.

Substituting the given geometric constraint of the braided sleeve, the radius r can be expressed as a function of the axial stretch ratio λ :

$$r(\lambda) = r_0 \sqrt{\frac{\sec^2 \theta_0 - \lambda^2}{\tan^2 \theta_0}}. \quad (12)$$

Consequently, the volume V and the stress invariants I_1 , I_2 can all be expressed as functions of (λ, κ) . After substituting the various energy terms, the total potential energy Π can be expressed as a functional that depends only on the generalized coordinates (λ, κ) and the control parameters (P, T) . This lays the foundation for the subsequent static equilibrium analysis.

$$\Pi(\lambda, \kappa; P, T) = U_{\text{bladder}}(\lambda, \kappa) + U_{\text{SMP}}(\kappa; T) - P \cdot [V(\lambda, \kappa) - V_0]. \quad (13)$$

3.3. Derivation of equilibrium equations

After formulating the total potential energy functional $\Pi(\lambda, \kappa; P, T)$ of the system in terms of the generalized coordinates (λ, κ) , the static equilibrium state is governed by the principle of minimum potential energy. This principle states that, under given external loads (air pressure P) and constraints (temperature T), the stable equilibrium configuration corresponds to a stationary value of the total potential energy. Therefore, the equilibrium equations are derived by setting the first variation of the total potential energy with respect to each independent generalized coordinate to zero.

The axial equilibrium equation can be obtained, which describes the balance among the axial expansion force generated by the internal pressure P , the elastic restoring force of the inner bladder, and the axial component of the constraint force from the braided sleeve. This equation determines the axial stretch ratio λ of the actuator under a given pressure [5],

$$\frac{\partial \Pi}{\partial \lambda} = \frac{\partial U_{\text{bladder}}}{\partial \lambda} - P \cdot \frac{\partial V}{\partial \lambda} = 0. \quad (14)$$

The bending moment equilibrium equation can be obtained. It primarily describes the balance among the net driving moment generated by the air pressure P under asymmetric constraints, the restoring moment provided by the SMP constraint structures (together with the equivalent braided sleeve stiffness), and the elastic resisting moment resulting from the bending deformation of the inner bladder. This equation determines the equilibrium curvature κ of the actuator,

$$\frac{\partial \Pi}{\partial \kappa} = \frac{\partial U_{\text{bladder}}}{\partial \kappa} + \frac{\partial U_{\text{SMP}}}{\partial \kappa} - P \cdot \frac{\partial V}{\partial \kappa} = 0. \quad (15)$$

Expanding the partial derivatives of each term in the above equation:

The partial derivatives of $\partial U_{\text{bladder}}$ with respect to the strain invariants involve the Mooney–Rivlin model and can be expressed as:

$$\frac{\partial U_{\text{bladder}}}{\partial \lambda} = V_0 \left(\frac{\partial W}{\partial I_1} \frac{\partial I_1}{\partial \lambda} + \frac{\partial W}{\partial I_2} \frac{\partial I_2}{\partial \lambda} \right) \quad (16)$$

$$\frac{\partial W}{\partial I_1} = C_{10}, \quad \frac{\partial W}{\partial I_2} = C_{01}. \quad (17)$$

The partial derivatives of I_1 and I_2 with respect to λ are:

$$\begin{aligned} \frac{\partial I_1}{\partial \lambda} &= 2\lambda_1 \frac{\partial \lambda_1}{\partial \lambda} + 2\lambda_2 \frac{\partial \lambda_2}{\partial \lambda} + 2\lambda_3 \frac{\partial \lambda_3}{\partial \lambda} \\ \frac{\partial I_2}{\partial \lambda} &= 2\lambda_1 \lambda_2^2 \frac{\partial \lambda_1}{\partial \lambda} + 2\lambda_1^2 \lambda_2 \frac{\partial \lambda_2}{\partial \lambda} + 2\lambda_2 \lambda_3^2 \frac{\partial \lambda_2}{\partial \lambda} + 2\lambda_2^2 \lambda_3 \frac{\partial \lambda_3}{\partial \lambda} + 2\lambda_3 \lambda_1^2 \frac{\partial \lambda_3}{\partial \lambda} + 2\lambda_3^2 \lambda_1 \frac{\partial \lambda_1}{\partial \lambda}. \end{aligned} \quad (18)$$

The partial derivatives of V with respect to λ and κ are:

$$\frac{\partial V}{\partial \lambda} = \pi L_0 \left\{ \left(r^2 - \frac{2\lambda r_0}{\tan^2 \theta_0} \right) \left[1 - \frac{(\kappa \lambda L_0)^2}{24} \right] - \frac{r^2 \lambda^2 \kappa^2 L_0^2}{12} \right\} \quad (19)$$

$$\frac{\partial V}{\partial \kappa} = \pi r^2 \lambda L_0 \left[-\frac{2\kappa (\lambda L_0)^2}{24} \right] = -\frac{\pi r^2 \lambda^3 L_0^3}{12} \kappa. \quad (20)$$

The partial derivative of U_{SMP} with respect to κ can be obtained directly by differentiation as follows:

$$\frac{\partial U_{\text{SMP}}}{\partial \kappa} = K_{\text{SMP}}(T) L_0 (\kappa - \kappa_0). \quad (21)$$

3.4. Model simplification and parameter calibration

When the temperature is below the glass transition temperature of the SMP, $K_{\text{SMP}}(T) \gg 0$, and the SMP structures provide strong constraints; therefore, the actuator predominantly exhibits directional bending. In this mode, axial elongation is significantly suppressed, and it can be assumed that $\lambda = 1$ (i.e. the centerline length remains constant). The bending moment equilibrium equation in terms of κ alone can then be obtained as follows:

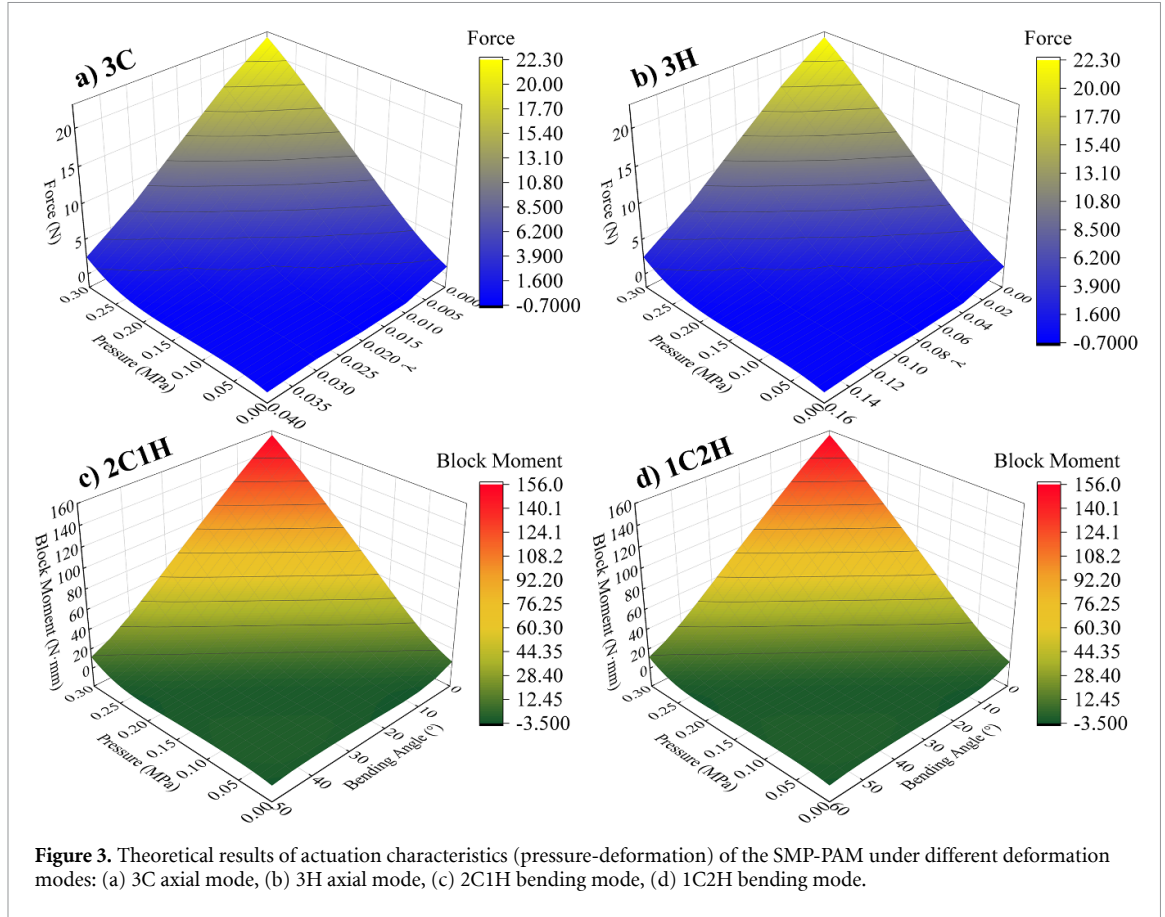
$$\left. \frac{\partial U_{\text{bladder}}}{\partial \kappa} \right|_{\lambda=1} + K_{\text{high}} L_0 (\kappa - \kappa_0) - P \left. \frac{\partial V}{\partial \kappa} \right|_{\lambda=1} = 0. \quad (22)$$

This is a univariate nonlinear equation in κ and can be solved numerically. Once κ is obtained, the tip bending angle is given by:

$$\phi = \kappa L_0. \quad (23)$$

The bending output moment M_{bend} can be obtained by the principle of virtual work as follows:

$$M_{\text{bend}} = -\frac{\partial \Pi}{\partial \phi} = -\frac{1}{L_0} \cdot \frac{\partial \Pi}{\partial \kappa}. \quad (24)$$



In the equilibrium state, $\partial\Pi/\partial\kappa = 0$; therefore, the bending output moment can be obtained either by evaluating the potential energy change between adjacent states, or directly from the restoring moment provided by the SMP constraints as follows:

$$M_{\text{bend}} = K_{\text{high}} L_0 (\kappa - \kappa_0) \quad (25)$$

When the temperature is above the glass transition temperature of the SMP, $K_{\text{SMP}}(T) \approx 0$, and the SMP constraints become largely ineffective. The actuator then degenerates into a quasi-axisymmetric McKibben muscle, predominantly exhibiting axial contraction/elongation. In this case, the bending curvature $\kappa \approx 0$, and the equilibrium is governed by the axial force equilibrium equation, yielding:

$$\frac{\partial U_{\text{bladder}}}{\partial \lambda} - P \cdot \frac{\partial V}{\partial \lambda} = 0. \quad (26)$$

The axial output force F_{axial} can be obtained by the principle of virtual work as follows:

$$F_{\text{axial}} = -\frac{\partial \Pi}{\partial (\Delta L)} = -\frac{1}{L_0} \cdot \frac{\partial \Pi}{\partial \lambda}. \quad (27)$$

The theoretical pressure-deformation relationships predicted by the model for all four deformation modes are summarized in figure 3.

4. Simulation

This section aims to establish a high-fidelity multi-physics finite element model of the dual-mode PAM integrated with SMP constraints, and to systematically investigate its deformation behavior and mechanical response under thermo-mechanical coupling. First, based on previous material and structural test data, a parametric geometric model is developed and accurate material constitutive relations are defined. Second, a rigorous mesh independence verification is performed to ensure numerical convergence of the simulation results. Third, using a sequentially coupled analysis framework, the quasi-static responses of

the actuator in both bending and axial modes are simulated and then compared with theoretical models and experimental results for validation.

4.1. Parametric geometric model and meshing

To perform simulation analysis on this dual-mode PAM integrated with SMP constraints, a complex multiphysics model capable of accurately capturing its thermo-mechanical coupling and large-deformation geometric nonlinearity (NLGEOM) must be established. To facilitate structural optimization and parametric studies, a parametric three-dimensional geometric model is first constructed, precisely describing the spatial arrangement and assembly relationships among the inner bladder, the braided sleeve, and the SMP constraint structures.

To balance computational efficiency and simulation accuracy, different modeling strategies are adopted for different components. The silicone rubber inner bladder is modeled as a thin-walled hollow cylinder with uniform wall thickness. It is meshed using hexahedral solid elements (C3D8H) with a swept mesh technique, and three integration points are set through the thickness to accommodate large-deformation analysis [22]. The material is modeled as a hyperelastic solid, whose constitutive behavior is described by the Mooney–Rivlin model.

Given the difficulty of directly modeling the complex fibrous network structure of the fiber braided sleeve, a ‘homogenized shell’ approach is adopted. The sleeve is simplified as a thin-layer shell element (S4R) with a structured mesh wrapped around the outer surface of the inner bladder. Anisotropic mechanical properties are assigned by defining local material orientations, endowing the sleeve with high tensile modulus in the fiber direction and low modulus in the shear deformation direction.

The three S-shaped constraint structures are generated using parametric curves. They are meshed with structured grids using solid elements (C3D8H), with local refinement applied to their curved arc segments and the contact regions with the braided sleeve. Their material properties are defined as temperature-dependent functions. A thermo-mechanical coupled constitutive model is adopted for the SMP material, implemented through a user-defined material subroutine (Abaqus UMAT), to describe the sharp variation of its elastic modulus near the glass transition temperature

4.2. Multiphase material definition and multiphysics coupling

Based on previous experimental results, the material constitutive relations of each component can be defined as follows. The hyperelastic mechanical behavior of the silicone rubber inner bladder is described using an incompressible Mooney–Rivlin model. The material parameters $C_{10} = 0.124 \text{ MPa}$ and $C_{01} = 0.06 \text{ MPa}$ are determined by least-squares fitting of the uniaxial tensile test data described in Chapter 5. The correlation coefficient between the fitted curve and the experimental results is $R^2 > 0.98$, indicating that the model can accurately capture the mechanical response of the inner bladder material within the working strain range.

The braided sleeve is modeled using an orthotropic linear elastic constitutive model, with the material principal directions defined in a local cylindrical coordinate system. The modulus in the fiber direction (1-direction) is taken as $E_1 = 20 \text{ GPa}$, while the transverse moduli are $E_2 = E_3 = 200 \text{ MPa}$. The in-plane shear moduli are $G_{12} = G_{13} = 80 \text{ MPa}$, and the Poisson’s ratios are $\nu_{12} = \nu_{13} = 0.3$. The above parameters are estimated based on the braid density and fiber volume fraction, and are determined in conjunction with classical braided composite material theory [32].

The SMP material is modeled using a generalized Maxwell viscoelastic constitutive model combined with the time-temperature superposition principle to describe its thermo-mechanical behavior. The relaxation modulus is expressed as:

$$E(T, t) = E_{\infty} + \sum_{i=1}^n E_i e^{-t/(\tau_i(T))} \quad (28)$$

where the temperature dependence of the relaxation time $\tau_i(T)$ is described by the WLF equation as follows:

$$\log a_T = -\frac{C_1 (T - T_g)}{C_2 + (T - T_g)} \quad (29)$$

where a_T is the shift factor, and $T_g = 86^\circ \text{C}$ is the glass transition temperature of the material. Based on DMA tests, the glassy modulus is determined as $E_g = 1.2 \text{ GPa}$ and the rubbery modulus as $E_r = 15 \text{ MPa}$. The coefficients of the WLF equation are $C_1 = 17.44$ and $C_2 = 51.6 \text{ K}$ (with T_g as the reference temperature). The above constitutive model is implemented through a UMAT, enabling the material stiffness matrix to be dynamically updated according to the real-time temperature field at each integration point.

Table 2. Mesh independence verification results.

Mesh model	Total elements	$\theta_{\max}$ (°)	RE(%)	$\sigma_{\max}$ (MPa)	RE(%)
G1	9583	47.4	—	4.57	—
G2	18 570	45.3	4.59	5.20	12.1
G3	40 854	44.7	1.40	5.41	3.95
G4	91 106	44.5	0.43	5.49	1.46

The simulation is performed within a multiphysics coupling framework to capture the ‘thermal-pneumatic-solid’ interaction. A sequentially coupled analysis strategy is adopted. First, a transient heat transfer analysis is conducted to compute the temperature field of the SMP structures and the entire actuator under external heat sources. The resulting temperature field is then imported as a predefined field into a nonlinear static/dynamic analysis to drive the change in stiffness properties of the SMP material. This one way coupling approach is justified for the quasi static conditions considered here, where the SMP heating is dominated by the embedded Joule heating films rather than by deformation induced changes in convective boundaries, and the bending angles are moderate (below 50°) such that the change in surface geometry exposed to ambient air is small. Under conditions involving large cyclic deformations or rapid transients, where structural deformation appreciably alters the convective heat transfer area or the local airflow around the actuator, a fully coupled two way thermo mechanical analysis would be required.

For computational efficiency, fluid inertial effects are neglected, and a uniform pressure is applied directly to the bladder’s inner surface. A linearly increasing uniform pressure load (0.1–0.3 MPa) is applied to the inner surface of the inner bladder. The NLGEOM option is enabled, and the Newton–Raphson iteration method is used for solution. The contact property is defined as hard contact, with a friction coefficient of 0.15.

4.3. Mesh independence verification

In finite element simulation analysis, mesh density directly affects the accuracy and reliability of the computational results. An overly coarse mesh may lead to distorted numerical solutions that fail to capture the stress gradients and deformation characteristics in critical regions. Conversely, an overly fine mesh, while improving accuracy, significantly increases computational cost and may even cause convergence difficulties due to accumulated rounding errors. Therefore, to eliminate the interference of mesh discretization on simulation results and ensure that the obtained numerical solution approximates the true continuous body response, a systematic mesh independence verification must be performed prior to the formal analysis [12].

In this section, a hierarchical verification strategy combining global uniform refinement with local refinement in critical regions is adopted. First, based on the initial geometric characteristics and empirical estimation, a relatively coarse baseline mesh model (G1) is established. Subsequently, a series of progressively refined mesh models (G2, G3, G4) are generated by uniformly reducing the global element size with a refinement factor of 1.5. At the same time, to ensure that the stress gradients in the curved arc segments of the S-shaped SMP constraint structures, the contact interfaces between the inner bladder and the braided sleeve, and the regions expected to undergo severe bending deformation are adequately resolved, local mesh refinement is simultaneously applied to the aforementioned regions.

Mesh convergence is evaluated based on two metrics: the maximum bending angle and the maximum von Mises stress. Convergence is quantified by calculating the relative error (RE) of the above metrics between two consecutive mesh refinement levels as follows:

$$RE = \frac{|V_{i+1} - V_i|}{|V_{i+1}|} \times 100\% \quad (30)$$

Once the REs of all key metrics fall below 2%, the simulation results can be deemed mesh-independent.

Based on the geometric model and material parameters, a typical operating condition is selected as the unified computational case for mesh independence verification. In this condition, the SMP is in the glassy state ($T = 25^\circ\text{C}$) and a nominal pressure of 0.2 MPa is applied. The boundary conditions, loading settings, and solver control parameters are kept strictly consistent across all cases. The results of the mesh independence verification are presented in table 2.

From G3 to G4, the REs of all three key metrics have dropped below 1.5%, meeting the convergence criteria. Further mesh refinement would lead to a significant increase in computational cost with only

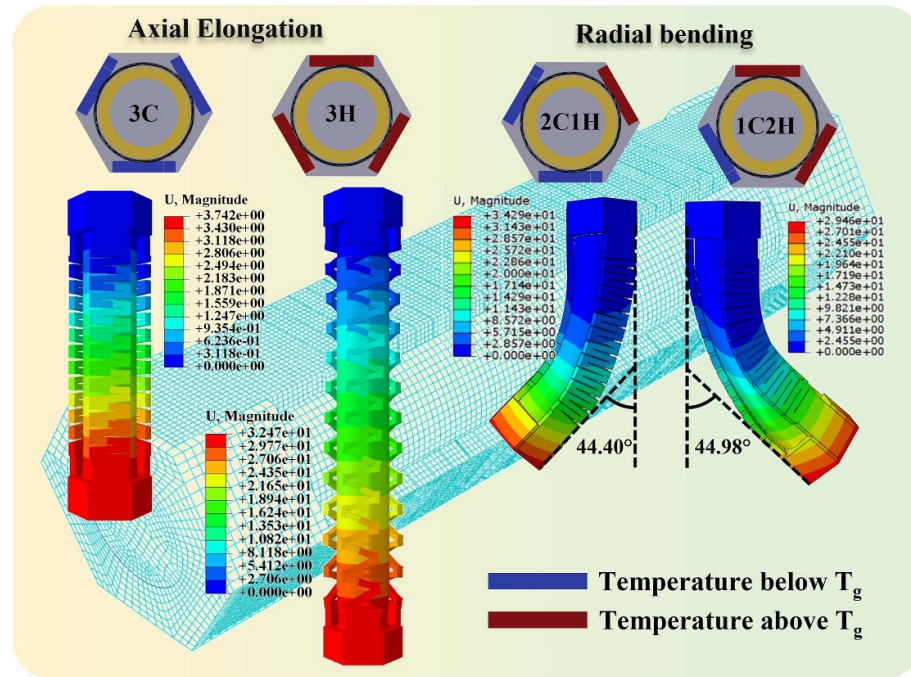


Figure 4. Displacement contours of the SMP-PAM under different deformation modes and the global mesh.

limited improvement in accuracy. Therefore, a mesh density equivalent to that of G4 is adopted for all subsequent simulations, i.e. the 'optimal mesh' configuration.

4.4. Simulation results and analysis

4.4.1. Deformation and stress distribution in bending mode

At a low temperature ($T = 25^{\circ}\text{C}$, with the SMP in the glassy state), a gradually increasing pressure load from 0.1 MPa – 0.3 MPa is applied to the inner wall of the actuator. The overall displacement contour obtained from the simulation (figure 4) shows that the actuator produces significant directional bending toward the side opposite the three SMP constraint structures, with its central axis approximating a circular arc, thereby validating the reasonableness of the pure bending kinematic assumption. The maximum bending curvature obtained from the simulation is 0.0157 mm^{-1} (1C2H mode; 0.0155 mm^{-1} for 2C1H), which is in good agreement with the experimental value of 0.0174 mm^{-1} .

The maximum stress concentration occurs on the inner side of the curved arc of the S-shaped SMP constraint structures, where the combined tensile and bending loads induced by bending deformation are significant. At a pressure of 0.3 MPa, the maximum von Mises stress is well below the tensile strength of the SMP material, indicating a sufficient safety margin. The stress distribution in the inner bladder is relatively uniform, with the maximum stress appearing in the outer bending region, which is consistent with the scenario of the inner bladder undergoing circumferential tension. The maximum stress in the fiber direction of the braided sleeve remains within the elastic range of the PET material.

4.4.2. Deformation and stress distribution in axial mode

At a high temperature ($T = 90^{\circ}\text{C}$, with the SMP in the rubbery state), the internal pressure of the actuator is also gradually increased from 0.1 MPa – 0.3 MPa. The overall displacement contour obtained from the simulation (figure 4) shows that the actuator predominantly exhibits axisymmetric axial elongation, with the bending angle approaching zero, thereby validating the assumption that the actuator degenerates into a conventional McKibben muscle when the SMP constraints become ineffective. At a pressure of 0.3 MPa, the pressure-axial displacement relationship exhibits a good linear trend, which is consistent with the theoretical prediction of a classic McKibben muscle, further confirming that the reduction in SMP stiffness at high temperatures eliminates the constraint function.

At the high temperature, the maximum von Mises stress appears at the connection between the inner bladder and the end cap, far below the peak stress in the bending mode. This is attributed to the softening of the SMP constraints, which leads to uniform deformation of the actuator and eliminates local stress concentrations. Almost all of the total strain energy of the system is stored in the inner bladder. This indicates that the SMP constraint structures barely participate in load bearing in the axial mode, a result that is consistent with the design expectation.

In figure 4, ‘displacement’ refers to the distance from the initial (unpressurized) position. For the axial deformation modes (3C and 3H), it denotes the axial displacement along the actuator axis; for the bending deformation modes (2C1H and 1C2H), it denotes the radial displacement in the bending direction.

5. Experiment and results

5.1. Experimental platform and testing methods

5.1.1. Experimental platform setup

A multifunctional experimental platform, shown in figure 5, was constructed to characterize the static and dynamic performance of the dual-mode, SMP-constrained PAM comprehensively. The platform consists of four subsystems: pneumatic control, thermal control, motion measurement, and force measurement. The pneumatic control system is supplied with stable compressed air by an oil-free air compressor. The air pressure is regulated by a precision electro-pneumatic proportional valve and then supplied to the PAM, while the internal pressure is monitored in real time by a digital pressure sensor with a measurement range of 0–0.3 MPa [33].

The thermal control system uses a programmable DC power supply to apply precise voltages to the miniature polyimide heating films (dimensions: 10 mm × 70mm, resistance: 10Ω) embedded in the SMP constraint structures, thereby achieving Joule heating. An infrared thermal imager is used to monitor the surface temperature distribution of the three SMP structures in real time, ensuring that they uniformly reach the target temperature.

In the force measurement system, the free end of the PAM is connected via a fixture to an electronic universal testing machine (EUT4203, range: ±200N, accuracy: 0.1%). The sensor signals are conditioned by a dynamic strain amplifier and then synchronously acquired with the pressure and temperature signals, enabling the measurement of the axial output force and the bending normal force. The bending moment is obtained using a four-point bending fixture and calculated as the product of the measured force and a fixed moment arm length.

The kinematic measurement system uses flexible displacement sensors (FDSs) attached to the surfaces of the three S-shaped constraints. These conductive-polymer strain sensors are thin, flexible, and responsive. They can perceive local length changes of the SMP structures during deformation in real time. By using a pre-calibrated relationship curve between the sensor output voltage and length, and after multi-channel data acquisition and post-processing, the bending angle, axial displacement, and motion trajectory of the PAM tip can be reconstructed. This method avoids the occlusion issues of traditional optical systems and enables continuous in-situ monitoring of actuator deformation during heating [34].

5.1.2. Kinematic measurement method

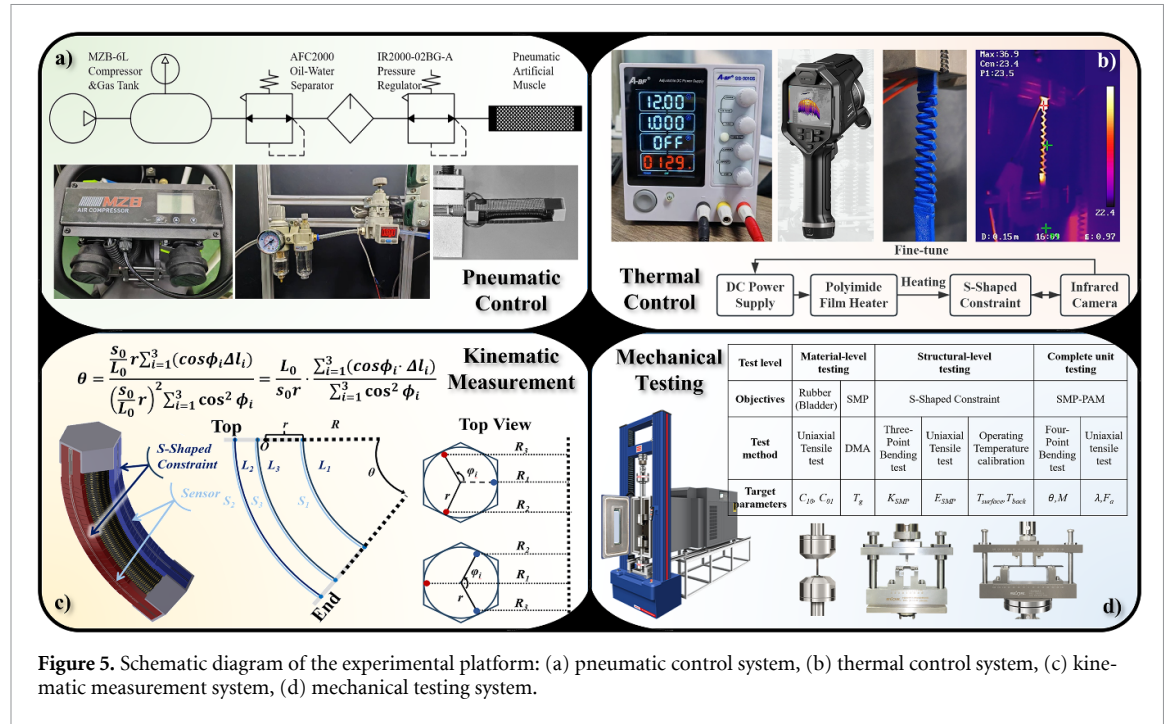
The kinematic parameters of the actuator are derived using FDSs attached to the surfaces of the three S-shaped constraint structures. Since flexible sensors inherently measure the length change of the region they cover, rather than directly measuring curvature or angle, a geometric mapping from local length changes to the tip bending angle must be established. The following sections detail the calculation principle and formula derivation for converting the raw sensor length signals into the tip bending angle [20].

Each FDS was independently statically calibrated before being attached. The sensor was stretched to a known length, and the relationship between its output voltage V_i and the corresponding elongation Δl_i was recorded. Within the working range of the sensor, the two exhibited a good linear relationship as follows:

$$\Delta l_i = k_i (V_i - V_{0,i}) \quad (i = 1, 2, 3) \quad (31)$$

where Δl_i is the length change of the i th sensor, V_i is the real-time output voltage of the sensor, $V_{0,i}$ is the zero voltage of the sensor in the initial Free State, and k_i is the calibration sensitivity coefficient (unit: mm·V⁻¹), obtained by linearly fitting the calibration data. The three sensors are respectively attached to the middle arc segments of the three S-shaped constraint structures, arranged along the longitudinal centerline of each constraint structure. Their length changes reflect the tensile or compressive deformation of the PAM surface in the longitudinal direction at the corresponding locations.

To infer the tip bending angle of the actuator from the local length changes measured by the sensors, a geometric model of the actuator’s bending deformation is established. It is assumed that the actuator deforms as a pure circular arc in the bending mode, i.e. the central axis bends into an arc of radius R



and central angle θ . Under the pure bending assumption, the length of the neutral axis is generally considered to remain constant, thus:

$$L_0 = R\theta \quad (32)$$

where θ is the tip bending angle (in radians), and R is the radius of curvature.

The three S-shaped constraint structures are attached to the outer surface of the actuator, and their distance to the neutral axis depends on their circumferential positions. Let the outer radius of the actuator cross-section be r , and the circumferential angle of the i th sensor be ϕ_i (with respect to the bending direction, where the outer vertex of bending is defined as $\phi = 0$ and the inner vertex as $\phi = 180^\circ$). For pure circular arc bending, the length L_i of a longitudinal fiber at a distance y_i from the neutral axis satisfies:

$$L_i = (R + y_i)\theta \quad (33)$$

where y_i is the distance from the i th sensor to the neutral axis. For a circular cross-section, assuming that the neutral axis passes through the centroid of the cross-section, we have:

$$y_i = r \cos \phi_i, \quad L_i = (R + r \cos \phi_i)\theta. \quad (34)$$

The length change Δl_i of the i th sensor is the difference between its deformed length L_i and its initial length $L_{0,i}$. Since the sensor is attached to the actuator surface, its initial length corresponds to the arc length in the initial straight state of the actuator. In the initial straight state, $R \rightarrow \infty$, $\theta \rightarrow 0$, and all longitudinal fibers have the same length L_0 . However, the sensor only covers a local region on the actuator surface; let the initial length of this region be s_0 (which is usually equal to the effective measuring length of the sensor itself). After deformation, the length of this local region becomes:

$$s_i = (R + r \cos \phi_i)\alpha \quad (35)$$

where α is the central angle corresponding to this local region.

Since the sensor is attached to the middle of the constraint structure, it is assumed that the central angle of this local region is proportional to the overall central angle θ , with a proportionality coefficient $\beta = s_0/L_0$ (i.e. the ratio of the local region length to the total arc length). Then:

$$\alpha = \beta\theta = \frac{s_0}{L_0}\theta \quad (36)$$

$$s_i = (R + r \cos \phi_i) \frac{s_0}{L_0} \theta. \quad (37)$$

Substituting $R = L_0/\theta$ yields:

$$s_i = \left(\frac{L_0}{\theta} + r \cos \phi_i \right) \frac{s_0}{L_0} \theta = s_0 + \frac{s_0}{L_0} r \cos \phi_i \theta. \quad (38)$$

Therefore, the length change of the i th sensor is:

$$\Delta l_i = s_i - s_0 = \frac{s_0}{L_0} r \cos \phi_i \theta \quad (i = 1, 2, 3) \quad (39)$$

From the above equation, it can be seen that the length change Δl_i of each sensor is linearly related to the tip bending angle θ , with the proportionality coefficient depending on the sensor position ϕ_i , the geometric parameters of the actuator (r, L_0), and the sensor coverage length s_0 . Substituting the known parameters yields:

$$\begin{bmatrix} \frac{s_0}{L_0} r \cos \phi_1 \\ \frac{s_0}{L_0} r \cos \phi_2 \\ \frac{s_0}{L_0} r \cos \phi_3 \end{bmatrix} \theta = \begin{bmatrix} \Delta l_1 \\ \Delta l_2 \\ \Delta l_3 \end{bmatrix}. \quad (40)$$

In theory, the three equations should yield the same value of θ . However, due to sensor measurement errors, positioning deviations during attachment, and deviations of the actual deformation from the ideal pure circular arc, the θ values obtained from the three equations may differ. To address this, the optimal θ value is determined by the least-squares method after multiple measurements.

Let the coefficient matrix be $\mathbf{A} = [a_1, a_2, a_3]^T$ and the measurement vector be $\mathbf{b}$. Then the least-squares solution is:

$$\theta = (\mathbf{A}^T \mathbf{A})^{-1} \mathbf{A}^T \mathbf{b} = \frac{\sum_{i=1}^3 a_i \Delta l_i}{\sum_{i=1}^3 a_i^2}. \quad (41)$$

Substituting $a_i = \frac{s_0}{L_0} r \cos \phi_i$ gives:

$$\theta = \frac{\frac{s_0}{L_0} r \sum_{i=1}^3 (\cos \phi_i \Delta l_i)}{\left(\frac{s_0}{L_0} r \right)^2 \sum_{i=1}^3 \cos^2 \phi_i} = \frac{L_0}{s_0 r} \cdot \frac{\sum_{i=1}^3 (\cos \phi_i \cdot \Delta l_i)}{\sum_{i=1}^3 \cos^2 \phi_i}. \quad (42)$$

When the actuator is in the axial mode (SMP heated to the rubbery state), the bending angle $\theta \approx 0$. However, the three sensors may still measure length changes, which mainly arise from the circumferential elongation of the surface due to cross-sectional expansion. In this axial mode, the average length change from the three sensors serves as an indirect measure of the axial strain. Through independent calibration experiments, with the axial displacement ΔL synchronously measured using a laser displacement sensor and the length changes Δl_i of the three FDSs recorded simultaneously, an empirical relationship is established as follows:

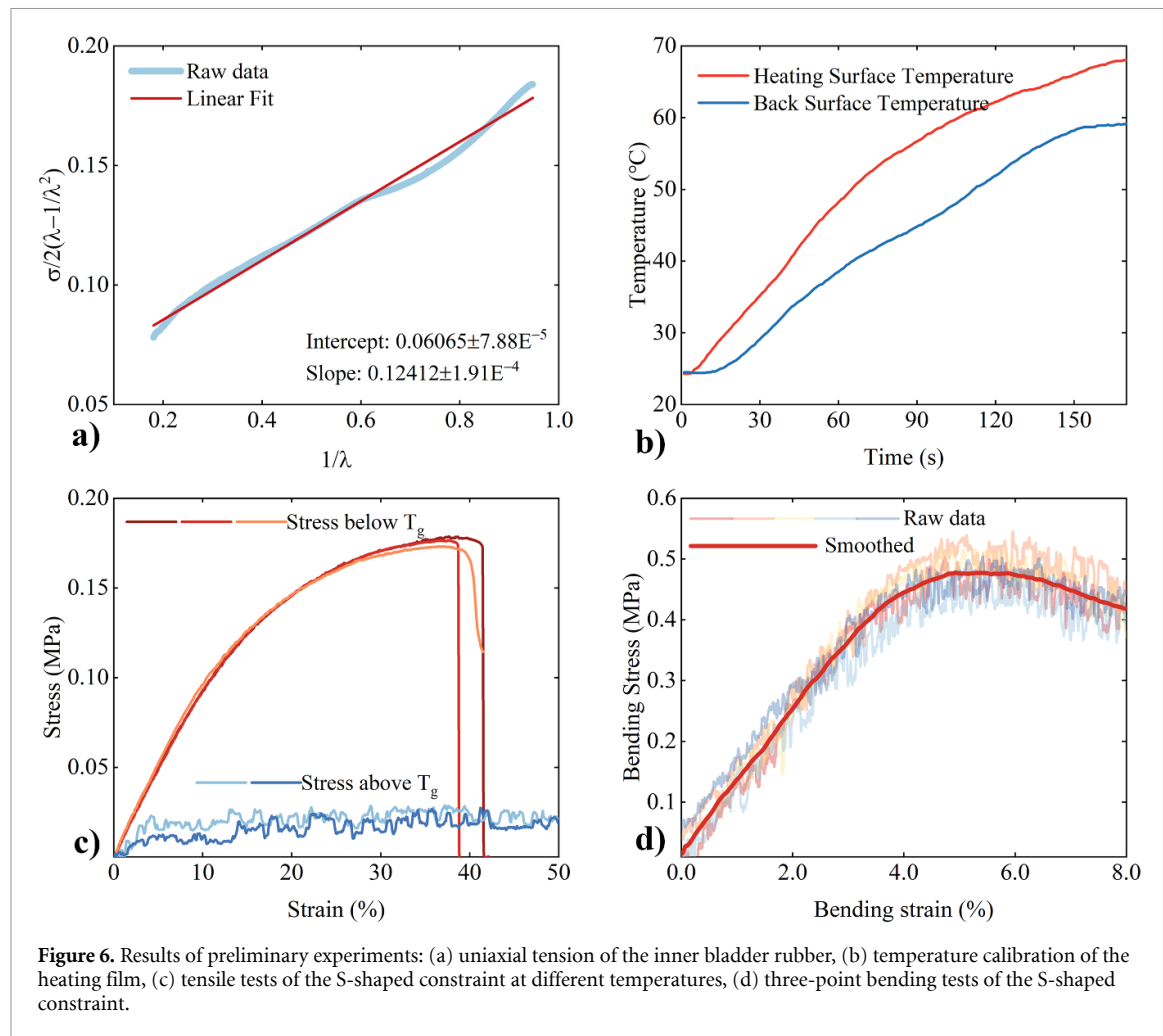
$$\Delta L = c_0 + c_1 \overline{\Delta l} + c_2 (\overline{\Delta l})^2 \quad (43)$$

$$\overline{\Delta l} = \frac{1}{3} \sum_{i=1}^3 \Delta l_i. \quad (44)$$

where c_0 , c_1 , and c_2 are the polynomial fitting coefficients.

5.2. Preliminary experiments at the material and structural levels

This section covers the preliminary tests required for preparing SMP-PAM. Material-level tests include tensile testing of rubber hoses and DMA analysis of SMP. Uniaxial tensile testing of rubber determines hyperelastic parameters for the Mooney–Rivlin model; DMA testing of SMP determines the material's glass transition temperature (T_g) and storage modulus. Structural-level tests involve mechanical testing of SMP-printed constrained structures to determine equivalent tensile stiffness and equivalent bending stiffness under various temperature control conditions. The complete set of preliminary experimental results is presented in figure 6 and discussed in the following subsections.



5.2.1 Hyperelastic parameter calibration of the silicone rubber inner bladder

According to the ASTM D412 standard, dumbbell-shaped specimens were fabricated from the silicone rubber tube material, and quasi-static uniaxial tensile tests were conducted using a universal testing machine. The crosshead speed was set to 50 mm min^{-1} , and the tests were carried out until specimen failure, with the complete engineering stress–strain curves recorded. The obtained stress–strain curves exhibit typical hyperelastic characteristics, namely a low initial modulus, significant stiffening with increasing strain, and no obvious yield point. To support subsequent finite element analysis, the widely used two-parameter Mooney–Rivlin model was employed for fitting. By performing a least-squares fit to the experimental data within the moderate strain range, the material parameters were determined as $C_{10} = 0.124 \text{ MPa}$ and $C_{01} = 0.06 \text{ MPa}$ [7].

5.2.2 DMA of the SMP material

DMA was performed on standard specimens made of the SMP raw material using a dynamic mechanical analyzer in single-cantilever beam mode. The temperature scan covered the phase transition range of the material, with a heating rate of 2°C min^{-1} , and tests were conducted at a fixed frequency of 1 Hz and a small strain amplitude. A significant $\tan\delta$ peak was observed during the heating process, with the peak temperature at 86°C . This temperature is identified as the glass transition temperature of the SMP, which will serve as the reference temperature for the subsequent thermal control strategy [30].

5.2.3. Characterization of equivalent stiffness of SMP constraint

To directly obtain the equivalent mechanical properties of the SMP constraint structures as used in the PAM, beam-shaped specimens with S-shaped geometry were fabricated using the same 3D printing process as the final product. The specimens were heated using heating films and tested on a universal testing machine. For the equivalent tensile stiffness test, both ends of the specimen were clamped, uniaxial tension was applied, and force–displacement curves were recorded at different constant temperatures (covering the range around T_g). The equivalent tensile stiffness K_{tensile} was calculated from the slope of

the linear region. For the equivalent bending stiffness test, a three-point bending fixture was used. Force-deflection curves at the central loading point were recorded at different temperatures. The equivalent bending stiffness was calculated using the formula $K_{\text{bending}} = F/\delta$, where F is the load and δ is the mid-span deflection.

The test results in this section directly verify the feasibility of the design from the material level. In PAM operation, the high stiffness of the SMP constraints at low temperatures effectively restricts radial expansion on the side to which they are attached, forcing the PAM to produce directional bending. At high temperatures, the extremely low constraint stiffness renders that side almost ineffective, allowing the PAM to recover nearly axisymmetric expansion and thus predominantly generate axial displacement. The order-of-magnitude difference in equivalent stiffness is precisely the physical core that enables the reversible switching between the ‘bending mode’ and the ‘axial mode’ of the actuator.

5.3. Movement testing

This section will conduct testing and validation of the deformation behavior of the SMP-PAM under pneumatic actuation. The PAM designed in this paper can switch freely between two deformation modes—elongation and bending—by adjusting the stiffness of the constrained structure through temperature control. Bending manifests two different curvature configurations. This section will provide detailed characterization of these four deformation modes.

First, perform elongation deformation testing. When all three circumferential restraint beams maintain identical stiffness, the PAM will undergo axial elongation deformation. The internal braided mesh of the PAM interacts with the elastic tube, exhibiting a tendency for axial elongation under pneumatic pressure. When the confinement beams are maintained at low temperatures with high stiffness, the PAM exhibits a small maximum elongation rate. When the confinement beams are heated above T_g , their stiffness significantly decreases, resulting in a large maximum elongation rate for the PAM.

The specific testing procedure involves applying a gradually increasing pneumatic pressure stimulus to the PAM at room temperature and recording its axial length until elongation ceases. This yields the relationship between PAM elongation and pneumatic pressure under high-stiffness conditions of the constrained structure. Subsequently, using heating films, all three constrained structures are heated above their glass transition temperatures. The PAM is again subjected to pneumatic pressure stimulation, and its axial length is recorded until elongation ceases, yielding the elongation rate under low-stiffness constrained structure conditions.

Next, bending deformation testing is conducted. When the three circumferential restraint beams possess different stiffnesses, the PAM will undergo bending deformation due to the uneven constraints, bending toward the side with higher stiffness. With temperature control, the three restraint beams can be divided into two scenarios: one hard and two soft, or one soft and two hard. In both cases, due to the differing overall stiffness of the restraint beams and the varying distances from the neutral plane, the PAM will exhibit bending deformation with two distinct curvatures.

The specific test procedure involves heating a single constrained beam above T_g and maintaining this temperature. A gradually increasing pneumatic pressure stimulus is applied to the PAM tube, with the bending angle and deflection recorded until maximum deflection is reached. After removing the pressure stimulus to allow the PAM tube to vent to atmospheric pressure, the test waits for the PAM to return to its initial state. Then, another constrained beam is heated above T_g and held at that temperature. The pressurization process is repeated, recording the bending angle and deflection until the maximum deflection is reached. This yields the pressure-bending angle relationship for two distinct bending modes.

The deformation behavior of the SMP-PAM prototypes under applied pressure of 0.1–0.3 MPa in all four modes is presented in figure 7.

5.4. Actuation performance testing

To comprehensively characterize the static and dynamic response characteristics of the dual-mode PAM integrated with SMP constraints under the two operating modes, this section systematically conducts quasi-static performance tests for both the bending mode and the axial mode, as well as dynamic response tests during mode switching. All tests are performed on the experimental platform described in section 5.1, and each test is repeated three times with the results averaged to ensure data reliability and repeatability.

5.4.1. Static performance testing in bending mode

To characterize the directional bending deformation capability and output torque characteristics of the actuator under the condition where part of the SMP constraints are in the glassy state (low temperature), and to obtain the baseline static performance curves in the bending mode, the ambient

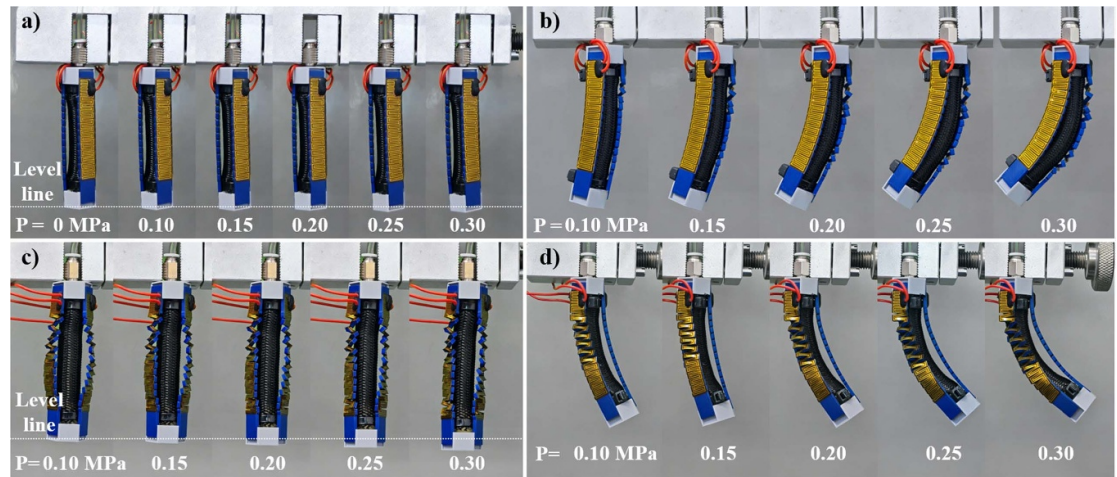


Figure 7. Deformation behavior test of SMP-PAM prototypes under applied pressure of 0.1–0.3 MPa in different modes: (a) 3C axial mode, (b) 2C1H bending mode, (c) 3H axial mode, (d) 1C2H bending mode.

temperature is kept constant at $T = 25^\circ\text{C}$, which is far below the glass transition temperature of the SMP material ($T_g = 86^\circ\text{C}$), thereby ensuring that one or two of the S-shaped SMP constraint structures remain in the high-stiffness glassy state. Prior to the test, the surface temperature of the SMP structures is monitored in real time using an infrared thermal imager to confirm that the temperature distribution is uniform and stable, thereby eliminating the potential influence of local temperature fluctuations on the constraint stiffness.

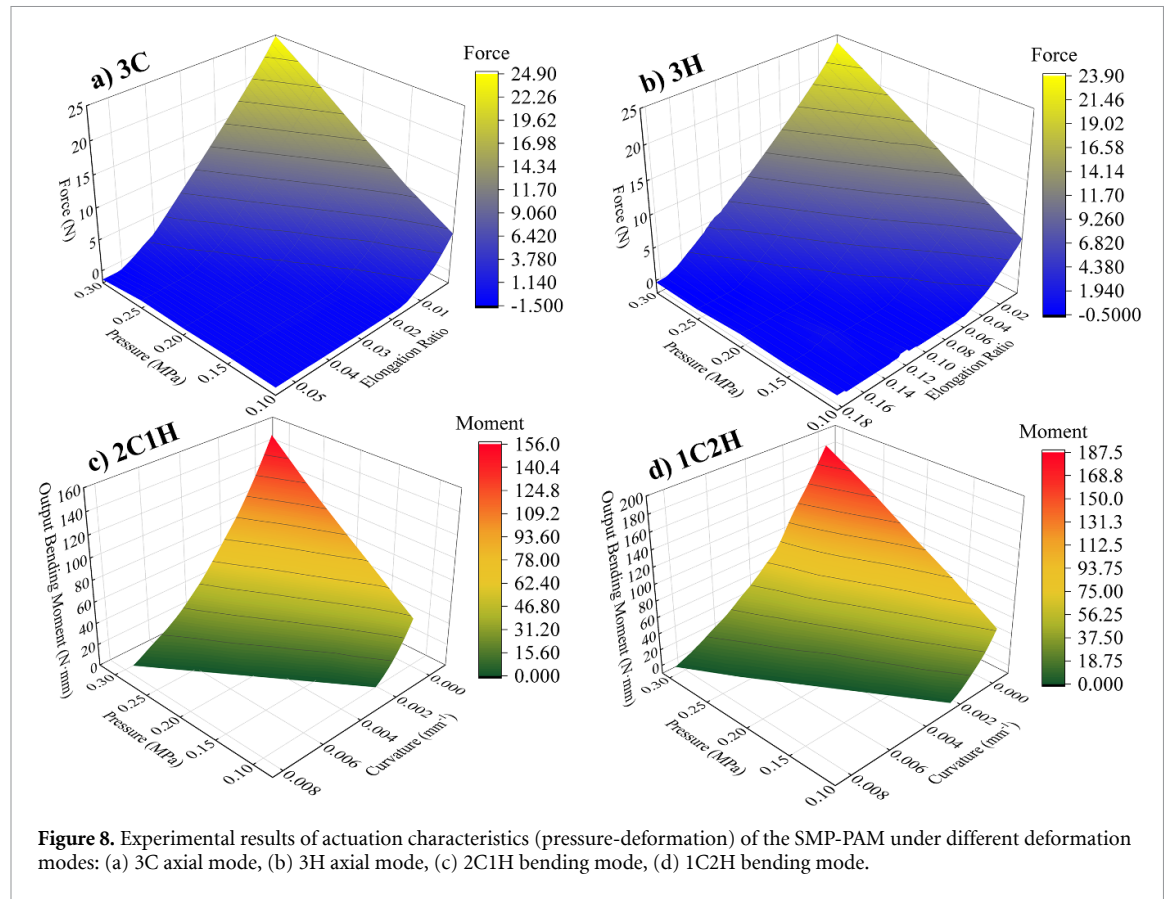
The tests were conducted using a stepped pressure loading method, with a step size of $\Delta P = 10\text{ kPa}$, increasing the internal pressure of the actuator gradually from $P = 0.1\text{ MPa}$ to 0.3 MPa . At each pressure step, the pressure was held constant for 10 s to ensure that the system reached a quasi-static equilibrium state. Subsequently, the following three types of data were synchronously acquired: (1) the current tip bending angle θ , calculated using the method described in section 5.1.2 based on the FDSs attached to the surfaces of the three S-shaped constraint structures; (2) the load-deflection curve, recorded by loading the actuator in a four-point bending fixture using a universal testing machine, from which the bending moment M was calculated according to the four-point bending principle; and (3) the current pressure value, recorded by a digital pressure sensor.

Using the air pressure P as the abscissa and the tip bending angle θ and bending moment M as the ordinates, the pressure-bending angle characteristic curve and the pressure-bending moment characteristic curve are plotted. Meanwhile, the maximum tip bending angle $\theta_{\max}$ and the maximum bending moment $M_{\max}$ are calculated. Each test condition is repeated three times, and the results are averaged to reduce random errors.

5.4.2. Static performance testing in axial mode

To characterize the axial contraction/elongation deformation capability and axial output force characteristics of the actuator under the condition where the SMP is in the rubbery state (high temperature), and to obtain the baseline static performance curves in the axial mode. A DC voltage (typically 12 V) is applied to the miniature polyimide heating films embedded in the SMP constraint structures, uniformly heating the three SMP structures to $T = 90^\circ\text{C}$, which is approximately 5°C above the glass transition temperature of the material ($T_g = 86^\circ\text{C}$). An infrared thermal imager is used to monitor the temperature distribution in real time, ensuring uniform heating and that the temperature fluctuation is controlled within $\pm 1^\circ\text{C}$. After the temperature stabilizes, the test is carried out under constant temperature conditions to eliminate the dynamic interference of temperature variation on the SMP stiffness.

The tests were conducted using the same stepped pressure loading method as in the bending mode, with a step size of $\Delta P = 10\text{ kPa}$, increasing the internal pressure of the actuator gradually from $P = 0.1\text{ MPa}$ to 0.3 MPa . At each pressure step, the pressure was held constant for 10 s to ensure that the system reached a quasi-static equilibrium state. Subsequently, the following data were synchronously acquired: (1) the axial displacement ΔL , calculated using the method described in section 5.1.2 based on the FDSs attached to the surfaces of the three S-shaped constraint structures; (2) the axial output force F_a , measured by a tension-compression load cell connected to the free end of the actuator; and (3)



the current pressure value, recorded by a digital pressure sensor. Each test condition was repeated three times and the results were averaged.

Using the air pressure P as the abscissa and the axial displacement ΔL and axial output force F_a as the ordinates, the pressure-axial displacement characteristic curve and the pressure-axial output force characteristic curve are plotted. The axial strain is calculated as $\varepsilon = \Delta L/L_0$. The experimental curves are then compared with the theoretical predictions of a classic McKibben muscle to verify that, after the SMP constraints become ineffective at high temperatures, the actuator degenerates into a quasi-axisymmetric PAM. The cycle-to-cycle consistency under combined thermal and pneumatic loading is further examined in section 5.5, where a 50 cycle test under the 1C2H mode confirmed that the bending curvature at the 50th cycle retained 98% of the first cycle value.

The complete set of experimental actuation characteristics (pressure-deformation) under all four deformation modes is shown in figure 8.

5.5. Thermal-pneumatic cyclic testing

To provide a preliminary assessment of the actuator performance stability under repeated thermal and pneumatic loading, a 50 cycle test was conducted using the 1C2H bending mode as a representative configuration. One SMP constraint was kept in the glassy state (unheated) to serve as the rigid constraint, while the remaining two constraints were subjected to cyclic thermal loading. The temperature of the heating films was cycled between 55 °C (below T_g) and 100 °C (above T_g), and the internal pneumatic pressure was simultaneously cycled between 0 and 0.2 MPa. A complete cycle consisted of heating the two constraints above T_g while pressurizing to 0.2 MPa, holding for 10 s to record the steady state bending deformation, then venting to atmospheric pressure and allowing the SMP to cool passively below T_g before the next cycle commenced. The surface temperatures of both the heated side and the back side of one SMP constraint were monitored throughout using the infrared thermal imager, and the bending curvature was derived from the FDS readings using the method described in section 5.1.2.

The recorded temperature and curvature histories are presented in figure 9. Figure 9(a) shows the temperature profiles of the heated surface and the back surface of one SMP constraint over the 50 cycles, together with the T_g reference line. The heated surface temperature consistently reached the target range of 90 °C–100 °C during each heating phase, while the back surface temperature lagged by approximately 10 °C–15 °C, reflecting the through thickness thermal gradient inherent to the thin plate geometry. The

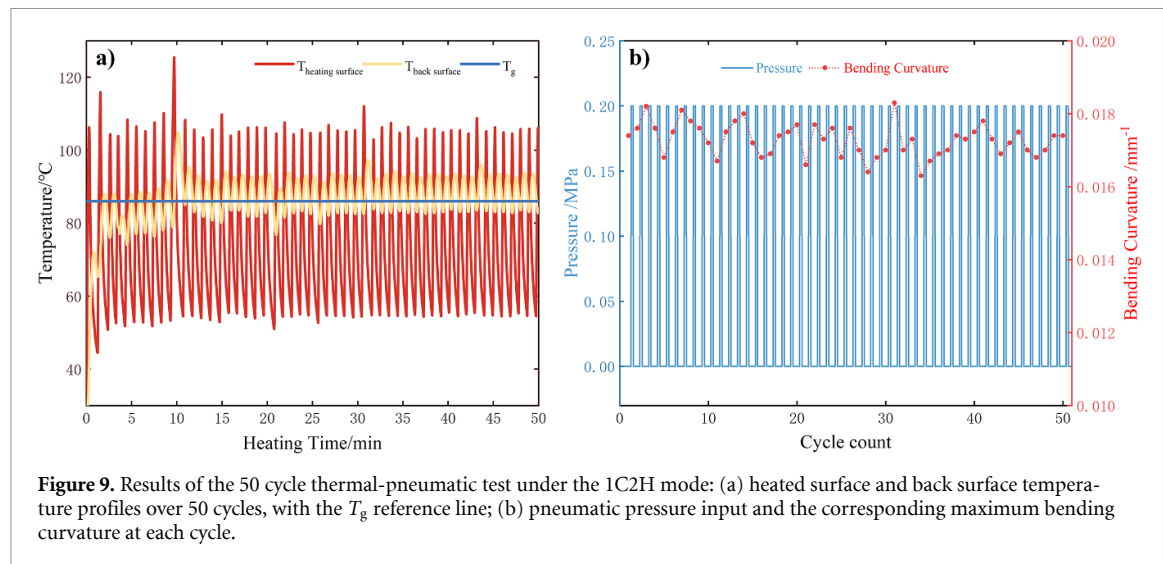


Figure 9. Results of the 50 cycle thermal-pneumatic test under the 1C2H mode: (a) heated surface and back surface temperature profiles over 50 cycles, with the T_g reference line; (b) pneumatic pressure input and the corresponding maximum bending curvature at each cycle.

temperature profiles exhibited good repeatability across all 50 cycles, with no discernible drift in the peak or baseline temperatures.

Figure 9(b) shows the corresponding pneumatic pressure input and the maximum bending curvature measured at the pressurized state of each cycle. The curvature remained stable throughout the 50 cycles, with the 50th cycle curvature reaching 98% of the first cycle value. The average maximum curvature across all cycles was consistent with the quasi static experimental results reported for the 1C2H mode in section 5.4 (approximately 0.0174 mm^{-1}). No systematic decreasing or increasing trend was observed, indicating that the SMP constraints maintained their stiffness contrast and geometric integrity over the tested duration.

These results provide initial evidence that the actuator can sustain short term cyclic operation without significant performance degradation. The 50 cycle test demonstrates that the SMP constraint structures, the heating films, and the bladder sleeve assembly retain functional stability under combined thermal and pneumatic cycling. Extended cycling tests over hundreds to thousands of cycles are required to fully characterize the long term fatigue behavior and have been included in the future work plan (section 6.3).

6. Results and discussion

6.1. Comprehensive comparison of results

To systematically evaluate the prediction accuracy of the theoretical model and finite element simulation method established in this paper under different deformation modes, this section compares the theoretical values, simulation values, and experimental averages of the key performance indicators—including the maximum elongation ratio λ , the maximum curvature κ , the maximum axial force F , and the maximum bending moment M —for the four deformation modes (3C, 2C1H, 1C2H, and 3H), and calculates the corresponding REs. The results are summarized in table 3.

In the 3C mode, where all three SMP constraints remain in the high-stiffness glassy state, axial elongation is theoretically expected to be completely suppressed. Consequently, the theoretical model predicts a maximum elongation ratio of 0%. However, both the simulation and experimental results show a certain degree of axial elongation (7.48% and 5.57%, respectively), indicating that the radial constraint of the braided sleeve in the actual actuator is not perfectly rigid, and the expansion of the inner bladder can still be converted into a small amount of axial displacement through changes in the fiber braid angle. The RE between the simulation and the experiment is 34.3%, suggesting that the simulation model may either underestimate the friction between the braided sleeve and the inner bladder or overestimate the radial compliance of the braided sleeve. In contrast, the theoretical value (22.3 N), simulation value (25.5 N), and experimental value (24.9 N) of the maximum axial force agree well, with a simulation-experiment error of only 2.41%, indicating high reliability of the force prediction.

In the 2C1H mode, where two glassy constraints and one rubbery constraint act together, directional bending is generated. The theoretical value of the maximum curvature is 0.0184 mm^{-1} , the simulation

Table 3. Comparison of theoretical, simulation, and experimental results for different deformation modes.

Deformation mode	Performance metric	Theoretical value	Simulation value	Experimental average $\pm$ SD	Theoretical-experimental relative error	Simulation-experimental relative error
3C (Axial)	Maximum elongation ratio λ (%)	0	7.48	5.57 ± 0.35	—	34.3%
	Maximum axial force F (N)	22.3	25.5	24.9 ± 2.17	10.4%	2.41%
2C1H (Bending)	Maximum curvature κ (mm^{-1})	0.0184	0.0155	0.0172 ± 0.00101	7.34%	15.6%
	Maximum bending moment M (N·mm)	158	169	156 ± 14.66	1.47%	8.33%
1C2H (Bending)	Maximum curvature κ (mm^{-1})	0.0184	0.0157	0.0174 ± 0.00125	17.3%	9.98%
	Maximum bending moment M (N·mm)	158	169	188 ± 12.22	15.9%	6.60%
3H (Axial)	Maximum elongation ratio λ (%)	23.8	19.7	18.4 ± 1.49	29.3%	7.07%
	Maximum axial force F (N)	22.3	25.5	23.9 ± 1.86	6.70%	6.69%

value is 0.0155 mm^{-1} , and the experimental average is 0.0172 mm^{-1} . The theoretical-experimental error is 7.34%, and the simulation-experimental error is 15.6%, indicating that the theoretical prediction is more accurate. Regarding the maximum bending moment, the theoretical value (158 N·mm), simulation value (169 N·mm), and experimental value (156 N·mm) are close to each other, with a theoretical-experimental error of only 1.47% and a simulation-experimental error of 8.33%. This demonstrates that the theoretical model has excellent predictive capability for the bending output under this asymmetric heating condition, while the simulation model slightly overestimates the result, which may be related to the setting of the rubbery-state stiffness parameters of the SMP or the choice of the contact friction coefficient [6].

The 1C2H mode is geometrically symmetric to the 2C1H mode, but with a different number of heated constraints. The theoretical value of the maximum curvature remains 0.0184 mm^{-1} , the simulation value is 0.0157 mm^{-1} , and the experimental value is 0.0174 mm^{-1} . The theoretical-experimental error increases to 17.3%, while the simulation-experimental error is 9.98%, indicating that the theoretical value is significantly higher than the experimental one. This discrepancy may arise from the fact that the theoretical model does not handle the asymmetric ‘cold-hot’ distribution with sufficient precision; the two rubbery constraints may retain a certain bending stiffness at high temperature, thereby partially counteracting the guiding effect of the cold-side constraints. Regarding the maximum bending moment, the experimental value (188 N·mm) is significantly higher than the theoretical value (158 N·mm) and the simulation value (169 N·mm), with a theoretical-experimental error of 15.9% and a simulation-experimental error of 6.60%. The higher experimental moment suggests that the two heated constraints are not completely softened and still provide additional resistance to bending, whereas the theoretical model assumes zero stiffness in the rubbery state, leading to an underestimation of the output moment.

In the 3H mode, where all SMP constraints are in the rubbery state, the actuator degenerates into a quasi-McKibben muscle. The theoretical value of the maximum elongation ratio is 23.8%, which is much higher than the experimental value of 18.4%, resulting in a RE of 29.3%; in contrast, the simulation value of 19.7% gives an error of 7.07% relative to the experiment. The overestimation by the theoretical model indicates that using an ideal step function to describe the stiffness change of the SMP (i.e. taking the rubbery-state modulus as zero) is overly simplistic. In reality, the material still retains a modulus of approximately 15 MPa at $T = 90^\circ\text{C}$, which imposes a non-negligible constraint on axial elongation. Because the simulation model incorporates the measured rubbery-state modulus, its prediction accuracy is significantly improved. Regarding the maximum axial force, the theoretical value (22.3 N), simulation value (25.5 N), and experimental value (23.9 N) are in good agreement, with a theoretical-experimental error of 6.70% and a simulation-experimental error of 6.69%, confirming the reliability of both methods for force prediction.

6.2. Error analysis and discussion

To thoroughly discuss the sources of deviation between the theoretical model and the simulation method in predicting the multi-mode deformation behavior of the actuator, this section presents a systematic error analysis of the key performance indicators under the four deformation modes based on the data listed in table 3, and discusses the causes of the errors from the perspectives of material constitutive models, geometric assumptions, boundary conditions, and experimental measurements.

6.2.1. Overall error characteristics

As can be seen from table 3, the simulation model achieves better prediction accuracy than the theoretical model for most indicators. Among the eight comparison indicators, six (75%) have a simulation-experimental RE of less than 10%, while only four (50%) of the theoretical-experimental REs are below 10%. The largest errors of the theoretical model occur in the elongation ratio in the 3H mode (29.3%) and the curvature in the 1C2H mode (17.3%), whereas the largest errors of the simulation model are observed in the elongation ratio in the 3C mode (34.3%) and the curvature in the 2C1H mode (15.6%). The error distribution correlates strongly with the deformation mode: for axially dominated modes (3C and 3H), the largest errors are in the elongation ratio; for bending-dominated modes (2C1H and 1C2H), they are in the curvature.

6.2.2. Analysis of error sources in the theoretical model

The theoretical model is based on the energy method and the principle of minimum potential energy, and incorporates several idealized assumptions, which are the primary sources of error.

- (1) Limitation of the step-change assumption for SMP stiffness: The theoretical model uses a hyperbolic tangent function to describe how the SMP's equivalent bending stiffness varies with temperature, assuming it approaches zero when $T > T_g$ (i.e. $K_{low} \approx 0$). However, the actual SMP material still retains a storage modulus of approximately 15 MPa at 90 °C (the test temperature for the 3H mode), corresponding to a non-zero residual stiffness. This leads to an overestimation of the axial elongation ratio in the 3H mode (29.3% error) and an overestimation of the curvature in the 1C2H mode (17.3% error), because the residual stiffness of the two heated constraints is not taken into account. This simplification is adopted for analytical tractability rather than physical fidelity: the DMA data in section 5.2.2 confirm a rubbery state storage modulus of approximately 15 MPa, which is non negligible. The resulting overestimation is systematic and its magnitude is reported in table 3, enabling users of the analytical model to apply a correction factor if higher prediction accuracy is required in the high temperature regime.
- (2) Deviation from the pure circular bending assumption: The theoretical model assumes that during bending, the central axis of the actuator forms a pure circular arc and that the length of the neutral axis remains constant. Under asymmetric heating conditions (e.g. 2C1H and 1C2H), the actual bending shape may deviate from the ideal circular arc, particularly with local reverse bending or curvature jumps at the interface between heated and cooled constraints. This results in a deviation of 7%–17% between the theoretical curvature prediction and the experimental values.
- (3) Simplification of the braided sleeve as ideally inextensible: The theoretical model assumes that the fibers of the braided sleeve are absolutely inextensible and that there is no relative slip between the sleeve and the inner bladder. The axial elongation of 5.57% observed experimentally in the 3C mode directly contradicts this assumption, indicating that even when all three constraints are in the glassy state, the braided sleeve still allows limited radial-to-axial conversion. The theoretical value of zero results in an infinite RE.
- (4) Calibration uncertainty of the hyperelastic parameters of the inner bladder: The Mooney–Rivlin parameters are obtained from uniaxial tensile tests, but during actuator operation the inner bladder is subjected to biaxial or triaxial stress states. The uniaxial calibration of the constitutive model introduces systematic errors. This affects the force and displacement predictions for all modes, and has a more significant impact on the bending modes.

The influence of these idealized assumptions can be assessed by comparing the theoretical predictions against the FEM results, since the FEM framework does not impose the circular cross section, constant curvature, or uniform radius constraints. As shown in table 3, the theoretical experimental REs for curvature and bending moment in the 2C1H mode remain below 10% (7.34% and 1.47%, respectively), indicating that the geometric assumptions introduce limited error under moderate bending conditions.

The larger deviations observed in the 1C2H mode (17.3%) and the 3H mode (29.3%) are driven primarily by the step change simplification of the SMP stiffness rather than by the kinematic assumptions, as the simulation model, which relaxes these geometric constraints while retaining the same SMP constitutive formulation, yields markedly smaller experimental errors in both modes (9.98% and 7.07%). This comparison confirms that the idealized kinematic framework is adequate for the quasi static operating regime considered in this study, and that future model refinement should prioritize the SMP constitutive representation over geometric fidelity.

6.2.3. Analysis of error sources in the simulation model

Although the simulation model can partially overcome the limitations of the theoretical assumptions, the following sources of error still exist.

- (1) Deviation in contact parameter settings: The friction coefficient between the inner bladder and the braided sleeve is set to 0.15 in the simulation, a value based on empirical estimation rather than direct measurement. In the 3C mode, the simulated elongation (7.48%) is significantly higher than the experimental value (5.57%), with an error of 34.3%, indicating that the simulation may underestimate the frictional constraint, leading to excessive axial slip. In the 2C1H mode, the simulated curvature is lower (error of 15.6%), which may also be related to the contact friction overestimating the bending resistance [28].
- (2) Simplification of the SMP viscoelastic constitutive model: The simulation employs a generalized Maxwell model combined with the WLF equation, but the parameters are based on linear viscoelastic assumptions from DMA tests, without fully accounting for the nonlinear viscoplastic response under large deformation. In the 1C2H mode, the simulated bending moment (169 N·mm) is lower than the experimental value (188 N·mm), with an error of 6.60%, which may be attributed to an underestimation of the residual stiffness of the two heated constraints [6].
- (3) Mesh discretization and solver errors: Although mesh independence verification has been performed, the finite element solution itself is subject to numerical integration errors and depends on iteration convergence tolerances. Local stress concentrations in the bending modes are highly sensitive to mesh quality, which may cause deviations in curvature prediction.
- (4) Approximation of the sequentially coupled thermo-mechanical analysis: The simulation adopts a sequentially coupled strategy, where the temperature field is first computed and then the material properties are updated, neglecting the reverse effect of deformation on the temperature distribution (e.g. bending changes the convection conditions around the heating films). This one-way coupling may introduce thermal lag errors during rapid heating stages.

6.2.4. Experimental measurement uncertainty

Some of the errors can also be attributed to the experimental measurement process. The temperature compensation calibration of the FDSs has a residual error of $\pm 3\%$, which may affect the elongation measurement in the high-temperature mode (3H). The temperature measurement accuracy of the infrared thermal imager is $\pm 2^\circ\text{C}$, causing the actual SMP temperature to fluctuate in the range of 88°C – 92°C , leading to uncertainty in stiffness. Fluctuations in the pressure control loop (± 0.005 MPa) have a significant influence on the curvature sensitivity at low pressure levels. In addition, manufacturing variations among prototypes (e.g. deviations in the attachment position of the SMP structures and non-uniform wall thickness of the inner bladder) introduce repeatability errors, which have been mitigated in the experiments by averaging three repeated measurements.

6.2.5. Assessment of error impact

Based on the above analysis, the main errors of the theoretical model stem from the neglect of the residual stiffness of the SMP and the pure bending assumption, leading to large deviations in the prediction of elongation ratio and curvature under asymmetric heating and high-temperature modes. In contrast, the main errors of the simulation model originate from the calibration uncertainties of the contact parameters and the viscoelastic parameters of the SMP, resulting in significant deviations in the elongation ratio in the 3C mode and the curvature in the 2C1H mode. Nevertheless, both methods demonstrate high reliability in predicting key mechanical quantities like axial force and bending moment, as the vast majority of these errors remain below 10%. This confirms the validity of the models' core physical framework [34].

6.2.6. Trade-off between deformation and switching speed

The geometric and material parameters that enhance deformation performance inevitably penalize switching speed. Increasing the SMP constraint thickness h amplifies the stiffness ratio between the glassy and rubbery states, yielding larger bending curvature and output moment; yet the accompanying rise in thermal mass prolongs both heating and cooling across the glass transition. Likewise, a higher T_g broadens the temperature window over which the actuator retains high stiffness, but demands greater heating power and longer ramp times to reach the softened state. Conversely, thinner SMP elements or lower- T_g compositions accelerate thermal response at the cost of constraint rigidity and deformation output. These competing trends imply a design envelope that must be navigated by application: rapid cyclic actuation favors thin, low- T_g constraints, whereas high-force, large-curvature tasks benefit from thicker, high- T_g configurations despite their slower switching dynamics.

6.2.7. Remarks on dynamic and thermal response

The foregoing analyses are confined to quasi-static loading conditions. In practical soft robotic applications, the dynamic response, particularly the thermal time constants that govern mode switching, is equally critical. Although transient thermal data were not systematically acquired in the present study, representative measurements and first-order estimates are provided here to indicate the relevant time scales.

Each polyimide heating film (10 Ω) driven at 12 V delivers approximately 14.4 W of Joule heating power. For a single SMP constraint of thickness h , width b , and length L_0 , the sensible heating time from ambient (25 °C) to above T_g (86 °C) can be estimated as

$$t_{\text{heat}} \sim \frac{\rho c_p (bhL_0) \Delta T}{\eta P}, \quad (45)$$

where ρ and c_p are the density and specific heat capacity of the SMP, $\Delta T = T_g - T_{\text{amb}}$, P is the electrical heating power, and η accounts for thermal contact efficiency and convective losses. Substituting the geometric and material parameters of the present system into this expression yields a characteristic heating time on the order of 5–10 s, which is consistent with direct observations during testing. Passive cooling under natural convection returned the SMP below T_g in approximately 30 s, and this cooling phase constitutes the rate-limiting step for cyclic mode switching. The corresponding total energy consumption is approximately 43 W when all three constraints are heated simultaneously (3H mode). Owing to the thin-plate geometry of the SMP structures, thermal imaging confirmed that the temperature gradient through the thickness was negligible and that the in-plane temperature distribution exhibited good uniformity.

The surface temperature of the SMP structures was monitored in real time via an infrared thermal imager throughout all experiments (section 5.1.1), and the steady-state temperature-voltage calibration is provided in figure 6b. Quantitative characterization of transient heating and cooling curves, internal thermal gradients, and cycle-by-cycle energy metering lies beyond the scope of the present work, and their systematic acquisition has been incorporated into the future experimental program outlined in section 6.3. Based on the approximately 30 s cooling limited cycle time, the actuator is suited for applications requiring switching frequencies below 0.03 Hz. Dynamic stability under cyclic loading, actuation bandwidth under varying payloads, and the potential for resonance induced instabilities at higher operating frequencies remain open questions that warrant dedicated experimental investigation. Hysteresis arising from the viscoelastic response of the SMP and friction at the bladder sleeve interface, which would become significant under continuous cyclic actuation, has not been characterized in the present quasi static study.

6.2.8. Discussion of limitations

Several aspects of the present study warrant acknowledgment as limitations. The thermal response speed and energy consumption, while characterized in sections 6.2.6 and 6.2.7, impose practical constraints on cyclic operation: the approximately 30 s passive cooling time restricts the switching frequency to the sub Hertz range, and the 43 W heating power for simultaneous three constraint operation may hinder deployment in untethered systems with limited onboard power.

Preliminary 50 cycle thermal pneumatic testing under the 1C2H mode (section 5.5) showed that the bending curvature at the 50th cycle retained 98% of the first cycle value with no systematic degradation trend, indicating acceptable short term stability of the SMP constraints, heating films, and bladder sleeve assembly under combined thermal and mechanical cycling. The long term fatigue behavior over hundreds or thousands of cycles, including potential progressive degradation of SMP stiffness, shape fixity,

and recovery ratio, remains an open question. Extended cyclic testing is required before deployment in applications demanding sustained operation.

Scaling the actuator to substantially different dimensions introduces coupled challenges. Miniaturization reduces thermal mass and may improve switching speed, but also decreases the absolute bending moment and axial force output. Conversely, scaling up amplifies force capacity at the expense of thermal management, as the increased cross section prolongs heating and cooling times and may introduce larger through thickness thermal gradients. The geometric and material trade offs discussed in section 6.2.6 become increasingly critical at both extremes of the size spectrum.

The actuator performance is sensitive to environmental conditions. Elevated ambient temperatures reduce the thermal gradient available for SMP actuation and may partially soften the glassy state, while low ambient temperatures increase the heating power required to reach T_g . Humidity and prolonged moisture exposure may degrade the silicone rubber bladder and compromise the electrical insulation of the heating films. Operation in dusty or high vibration environments, common in many target applications, remains untested. Systematic environmental robustness characterization is necessary before the actuator can be considered for field deployment.

6.3. Future improvement

Based on the above research results and error analysis, future work can be carried out in the following aspects. First, at the material level, the hyperelastic constitutive parameters of the inner bladder should be calibrated through biaxial tensile tests, and the master curve of the relaxation modulus of the SMP over a wide temperature range should be obtained via DMA to more accurately describe its non-ideal step-change behavior and viscoelastic dissipation. Second, in terms of modeling and simulation, a two-way thermo-mechanical coupling model can be developed to account for the reverse effect of deformation on the temperature field distribution. At the same time, an adaptive contact algorithm can be introduced, with the dynamic friction coefficient at the inner bladder-braided sleeve interface calibrated through in-situ friction tests. Third, regarding experimental characterization, the number of prototypes and test cycles should be increased to establish statistically significant performance tolerance bands, and higher-precision non-contact three-dimensional topography measurement techniques should be employed to verify the spatial configuration of bending deformation. Finally, based on the multi-mode controllable actuation concept established in this paper, independent addressable heating and closed-loop temperature control of the SMP constraints can be further explored to achieve real-time adjustment of arbitrary bending directions and curvatures of the actuator in three-dimensional space. This can then be integrated into a soft robotic system to accomplish more complex biomimetic motions and adaptive manipulation tasks.

7. Conclusions

This paper presented a dual-mode PAM integrating three circumferentially arranged S-shaped SMP constraints. By independently controlling the thermal state of each constraint, the actuator achieves four distinct deformation modes (3C, 2C1H, 1C2H, and 3H) within a single device, encompassing axial elongation and multi-directional bending. An energy-based theoretical model with temperature-dependent SMP stiffness and a sequentially coupled thermo-pneumatic-solid finite element model were developed. A prototype was fabricated and characterized on a custom experimental platform integrating pneumatic, thermal, and flexible displacement sensing subsystems.

The actuator delivers an experimentally measured maximum bending curvature of 0.0174 mm^{-1} (theoretical 0.0184 mm^{-1} ; FEM 0.0157 mm^{-1}) and a maximum elongation ratio of 18.4% (theoretical 23.8%; FEM 19.7%) under 0.1–0.3 MPa. The theoretical model predicts bending moment and axial force with REs below 10% in the symmetric configurations, while the FEM results, which incorporate the measured SMP rubbery state modulus, yield errors below 10% in six of the eight comparison metrics. A 50 cycle thermal pneumatic test under the 1C2H mode confirmed that the bending curvature retained 98% of its initial value without systematic degradation. The residual SMP stiffness at elevated temperatures and the contact friction at the bladder sleeve interface were identified as the dominant sources of model-experiment discrepancy.

The contributions of this work are threefold: a multi-constraint architecture that enables four deformation modes and multi-directional bending in a single device; a complete methodology chain spanning theory, multiphysics simulation, and experiment with integrated flexible sensing; and systematic performance data, including cyclic testing, that establish benchmarks for this class of actuator. These results

provide a foundation for the design and modeling of multi-modal soft actuators, while the identified limitations indicate clear priorities for future work.

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Data availability statement

The data that support the findings of this study are available upon reasonable request from the authors.

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