

Research paper

Dynamic-mesh modeling of low-cost polymer microchannel regenerators for elastocaloric cooling

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ABSTRACT

Elastocaloric cooling is a low-global-warming-potential alternative to vapor-compression refrigeration. However, the high cost of shape memory alloys and high-force actuators limits its large-scale deployment. Polymer-based materials such as natural rubber (NR) and thermoplastic polyurethanes (TPU) provide cost-effective alternatives but undergo large operational strains (up to 600%), which challenge conventional regenerator designs and numerical modeling approaches. In this study, a transient one-dimensional numerical model with a dynamic-mesh formulation is developed to capture coordinate transformations and coupled thermo-hydraulic behavior in large-deformation elastocaloric regenerators. Four microchannel geometries (parallel plate, square, triangular, and circular) are evaluated. Using NR as the working material, the triangular configuration achieves the highest specific cooling power (SCP) due to enhanced heat transfer and reduced viscous dissipation. Parametric optimization identifies an optimal pore size of $\sim 400\ \mu\text{m}$ at 50% porosity. Replacing NR with high elastocaloric response TPU further improves cooling performance. At an operating frequency of 1 Hz, the optimized triangular TPU regenerator is numerically predicted an SCP of $17.4\ \text{Wg}^{-1}$, which is three times higher than that of NR. Polymer-based regenerators offer an active-material cost per unit cooling power of $4.3 - 5.0 \times 10^{-4}\ \$/\text{W}$, highlighting the potential for cost-effective elastocaloric cooling. This work provides quantitative design guidelines for scalable polymer-based elastocaloric cooling systems.

1. Introduction

The global demand for space cooling is projected to triple by 2050, imposing significant pressure on energy infrastructure and climate mitigation efforts due to the continued reliance on Vapor-Compression (VC) refrigeration [1–3]. Conventional Hydrofluorocarbon (HFC) refrigerants, including R-134a and R-410 A, exhibit Global Warming Potentials (GWP) several thousand times higher than CO_2 and are subject to increasingly strict international phase-down regulations. Solid-state cooling technologies have therefore attracted growing attention as sustainable alternatives. Among them, elastocaloric cooling utilizes the latent heat associated with stress-induced phase transformations to achieve refrigeration without gaseous working fluids. It has been identified by both the U.S. Department of Energy and the European Commission as a promising non-vapor-compression technology [4,5].

An Active Elastocaloric Regenerator (AER) combines cyclic loading of a porous elastocaloric matrix with alternating fluid blows, where regeneration enables the device temperature span to exceed the adiabatic temperature change of the material. Recent years have witnessed rapid

advancements in AER prototypes for large-scale cooling applications, achieving cooling capacities ranging from hundreds of watts to the kilowatt scale [6–13]. Most elastocaloric systems reported to date rely on metallic Shape Memory Alloys (SMAs), typically configured as NiTi-based tube bundles. Substantial efforts have been devoted to improving heat transfer performance and mechanical durability [10–14]. The first NiTi plate-based AER produced a 15.3 K temperature span and an SCP of $0.8\ \text{Wg}^{-1}$ [6]. Qian et al. utilized compression-driven NiTi tubes to achieve a 22.5 K temperature span and Specific Cooling Power (SCP) of $0.35\ \text{Wg}^{-1}$ [9], with a maximum cooling power of 260 W obtained under a different operating mode. Similarly, Ahčin et al. reached a span exceeding 30 K with a notable SCP of $4.4\ \text{Wg}^{-1}$ using a related NiTi tube design [10]. A spiral-tube AER subsequently achieved a 50.6 K water-side temperature span and 203 W of cooling power while demonstrating more than 10^6 compression cycles [15]. Zhou et al. further demonstrated that multi-stage cascaded configurations combined with optimized high surface-to-volume regenerator geometries can deliver temperature lifts above 70 K and kilowatt-scale cooling capacities,

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Nomenclature**Abbreviations**

1D	One-Dimensional
AER	Active Elastocaloric Regenerator
ALE	Arbitrary Lagrangian–Eulerian
CFL	Courant–Friedrichs–Lewy
COP	Coefficient of Performance
CR	Cold Reservoir
CSS	Cyclic Steady State
eCM	Elastocaloric Material
FDM	Finite Difference Method
GWP	Global Warming Potential
HFC	Hydrofluorocarbon
HMDI	4,4'-Dicyclohexylmethane diisocyanate
HM-PTMEG	TPU synthesized from HMDI and PTMEG
HR	Hot Reservoir
HX	Heat Exchanger
MDI	4,4'-Diphenylmethane diisocyanate
M-PCL	TPU synthesized from MDI and PCL
M-PPG	TPU synthesized from MDI and PPG
M-PTMEG	TPU synthesized from MDI and PTMEG
NR	Natural Rubber
PCL	Polycaprolactone diol
PP	Parallel Plate
PPG	Polypropylene glycol
PTMEG	Poly(tetramethylene ether) glycol
PVT	Poly(vinylidene fluoride–trifluoroethylene)
PVTC	Poly(vinylidene fluoride–trifluoroethylene–chlorotrifluoroethylene)
RMSE	Root-Mean-Square Error
SCP	Specific Cooling Power
SEBS	Styrene–ethylene–butylene–styrene
SIC	Strain-Induced Crystallization
SMA	Shape Memory Alloy
TE	Thermoelectric
TPU	Thermoplastic Polyurethane
VC	Vapor Compression

Symbols

a_s	Specific surface area [m^{-1}]
a_0	Biot-number correction coefficient [–]
A	Channel flow area [m^2]
$\mathbf{A}$	Global coefficient matrix
A_c	Cross-sectional area [m^2]
A_{surf}	Surface area [m^2]
Bi	Biot number [–]
$\mathbf{B}$	Right-hand-side vector
c	Specific heat capacity [$\text{J kg}^{-1} \text{K}^{-1}$]
d	Side length (square) or diameter (circular) [m]
D_h	Hydraulic diameter [m]
D_p	Axial dispersion coefficient [–]
E_{tol}	Energy conservation residual tolerance [–]
f	Operating frequency [Hz]
f_F	Darcy friction factor [–]

Gz	Graetz number [–]
h	Height of isosceles triangle [m]
H	Regenerator overall height [m]
H_1	Spacing of parallel plates [m]
H_2	Wall thickness of parallel plates [m]
J	Jacobian determinant [–]
k	Thermal conductivity [$\text{W m}^{-1} \text{K}^{-1}$]
L	Regenerator length [m]
m	Mass [kg]
$\dot{m}$	Mass flow rate [kg s^{-1}]
N_t	Number of temporal nodes [–]
Nu	Nusselt number [–]
Nu_0	Uncorrected Nusselt number [–]
N_x	Number of spatial nodes [–]
P	Pressure [Pa] or wetted perimeter [m]
P_{ext}	External regenerator perimeter [m]
Pe	Peclet number [–]
Pr	Prandtl number [–]
Q	Cooling energy [J]
$\dot{Q}$	Cooling power [W]
$\dot{q}$	Volumetric heat generation rate [W m^{-3}]
Re	Reynolds number [–]
t	Time [s]
T	Temperature [K]
$\mathbf{T}$	Global temperature vector [K]
ΔT_{ad}	Adiabatic temperature change [K]
u	Interstitial fluid velocity [m s^{-1}]
u_{mesh}	Mesh velocity [m s^{-1}]
u_{rel}	Fluid velocity relative to the mesh [m s^{-1}]
U_{loss}	Overall heat-loss coefficient [$\text{W m}^{-2} \text{K}^{-1}$]
v_s	Superficial velocity [m s^{-1}]
V	Volume [m^3]
V^*	Dimensionless volume ratio [–]
w	Base width of isosceles triangle [m]
W	Regenerator width [m]
W_{mech}	Mechanical work [J]
W_{net}	Net work input [J]
W_{pump}	Pumping work [J]
x	Reference coordinate (undeformed configuration) [m]
X	Spatial coordinate (deformed configuration) [m]

Greek letters

α	Elastocaloric coefficient [K]
Δ	Difference
ε	Strain [–]
ϵ	Porosity [–]
η	Hysteresis ratio [–]
λ	Stretch ratio [–]
μ	Dynamic viscosity [Pa s]
ν	Poisson's ratio [–]
ρ	Density [kg m^{-3}]
σ	Stress [Pa]
τ	Period time [s]
ν_H	Hencky–Poisson ratio [–]
ψ	Deformation factor [–]

Subscripts and superscripts

0	Initial state
*	Deformed state or deformation-dependent parameter
ad	Adiabatic
amb	Ambient
blow	Heat transfer blow phase
c	Cold side or cooling
cold	Cold reservoir
eff	Effective
ext	External
f	Fluid phase
f	Final (e.g., λ_f)
fluid	Displaced fluid
H	Hencky
h	Hot side
hot	Hot reservoir
hys	Hysteresis
i	Initial
ideal	Idealized
in	Inlet
max	Maximum
mesh	Computational mesh
min	Minimum
mech	Mechanical
net	Net
out	Outlet
pp	Pore
pump	Pump
reg	Regenerator
rel	Relative to the moving mesh
rev	Reversible
s	Solid
source	Heat source
span	Span
stat	Static
surf	Surface
total	Total
trans	Transverse
void	Void
HX	Heat exchanger
n	Time-step index

translating to a highly competitive SCP of 12.3 W g^{-1} [11,12]. More recently, a compressive multi-cell NiTi architecture delivered a specific cooling power of 1.43 W g^{-1} and enabled sub-zero cooling down to -20°C [13]. Additive manufacturing has enabled 3D-printed NiTi-tube prototypes, with a recent device achieving a 20°C temperature span and 50 W of cooling power [16]. Table 1 summarizes the mean active-material price, cyclic durability, and key performance of recent NiTi-based elastocaloric prototypes and polymer-based systems.

These advances show that AER technology has moved beyond proof-of-concept operation toward application-relevant temperature spans and cooling capacities. Nevertheless, a large temperature span is generally obtained near zero cooling load and maximum cooling power is measured near zero temperature span, which needed to be considered together for the practical use. Moreover, scaling SMA-based AERs requires increased active NiTi materials, compression force, fluid-distribution complexity, and actuator size in the system. Cost

effectiveness consequently depends not only on SCP but also on active-material price, manufacturing, driver and work-recovery efficiency, pumps, durability, and system operation cost. For example, a recent cam-disc drive demonstrated approximately 70% mechanical-level work recovery at an actuation force of 40 kN , indicating the importance of the drive subsystem [25].

Fig. 1 compares active-material prices rather than a complete life-cycle cost assessment. Although NiTi exhibits large adiabatic temperature changes (ΔT_{ad}) exceeding 25 K [26,27], its material cost ($> 185 \text{ \$/kg}$) limits scalability for widespread deployment. Shape memory alloys, together with rare-earth magnetocaloric materials (e.g., Gd) and electrocaloric polymers (e.g., P(VDF-TrFE-CFE)), are significantly more expensive than standard gaseous vapor-compression refrigerants. In contrast, polymers such as Thermoplastic Polyurethanes (TPU) and Natural Rubber (NR) are comparable in cost to the industrial benchmark R134a ($\sim 7 \text{ \$/kg}$). Notably, NR has a material cost of $\sim 2 \text{ \$/kg}$ while delivering a substantial elastocaloric effect ($\Delta T_{ad} \sim 10 \text{ K}$) [19,28]. More recently, high-modulus TPU (HM-PTMEG) has demonstrated an adiabatic temperature change exceeding 20 K [29]. The available durability comparison between polymer- and alloy-based systems remains asymmetric. Long-duration NiTi devices have been studied extensively, including system-level operation beyond 10^7 cycles, whereas only a few polymer-based cooling prototypes report comparable device-level testing [8,17,19,20]. This difference reflects the present maturity of prototype validation but not indicates the lifetime disadvantage of polymers. Polymer material-level studies demonstrate 1.7×10^5 elastocaloric cycles for NR without failure [22]. Prestrained natural-rubber fatigue data report 10^7 cycles for a 200%–400% strain window and up to 8×10^7 cycles for a narrower 250%–300% window [23,30]. A TPU tensile-fatigue study reached 10^6 cycles under 13 MPa [24]. These results indicate the potential for developing fatigue-resistant polymer-based regenerators.

Moreover, the actuator and auxiliary components can be major cost contributors for an elastocaloric system: a recent NiTi cam-disc system operated at an actuation force of approximately 40 kN , whereas reported polymer devices generally require forces of hundreds of newtons, which can significant reduce the cost from actuation [19,21,25,31,32]. These properties motivate polymer AERs as a route to reduce active-material cost and transformation force, while a full techno-economic comparison requires experimentally validated device lifetime, manufacturing, actuator, and auxiliary-power data.

To leverage these low-cost and low-force materials, recent studies have explored polymer-based elastocaloric cooling devices. Sebald et al. demonstrated the first regenerative elastocaloric cooler using NR tube bundles, establishing the feasibility of Strain-Induced Crystallization (SIC) in regenerative cycles [18]. Zhang et al. developed a roller-based prototype using rolled SEBS sheets to achieve large strains, reporting a maximum SCP of 1.9 W g^{-1} and a temperature span of 5.2 K [32]. Miniature-scale elastocaloric devices based on thin NR foils and solid-to-solid heat transfer have also been demonstrated, enabling compact heat exchange configurations [19]. More recently, Wang et al. fabricated three-dimensional soft elastomeric regenerators via additive manufacturing, highlighting the potential of polymer architectures with complex internal channels [21].

Despite these advances, polymer-based elastocaloric cooling remains at an early stage of development. Achieving high-performance cooling requires efficient AER designs. Numerical modeling is essential for guiding regenerator geometry design and optimizing operating parameters because of the large parameter space. One-Dimensional (1D) numerical models are widely used for regenerative caloric systems because they offer a good balance between computational efficiency and physical accuracy, enabling extensive parametric studies [48–52].

However, most existing 1D AER models assume a fixed regenerator volume with constant regenerator length L [49,51,52], as shape memory alloys typically operate under small strains ($\sim 3\%$ – 8%). In contrast, polymer-based regenerators require large uniaxial strains (300% to

Table 1

Reported cooling performance, indicative active-material price, and cyclic durability of recent elastocaloric cooling prototypes. Polymer material-level fatigue tests are listed in a separate section.

Reference	Prototype/material evidence	Material price (\$/kg)	Reported cyclic durability (cycles)	Key performance
<i>Elastocaloric cooling prototypes</i>				
Bachmann et al. (2021) [17]	NiTi active elastocaloric heat-pipe system	185.6	$> 10^7$ (system-level runout; no material failure)	SCP 6.27 W g ⁻¹ ; 5.6 K; 7.9 W
Greibich et al. (2021) [8]	NR-membrane snap-through heat pump	2.70	$\sim 10^3$ (device test)	SCP 20.9 W g ⁻¹ ; 7.9 K; COP 4.7
Ahčin et al. (2022) [10]	NiTi-tube AER	185.6	$> 3 \times 10^5$ (regenerator cycles)	SCP 4.4 W g ⁻¹ ; 31.3 K
Qian et al. (2023) [9]	Multimode NiTi-tube system	185.6	$> 7.5 \times 10^4$	SCP 0.35 W g ⁻¹ ; 22.5 K; 260 W
Zhou et al. (2023) [15]	Spiral-tube NiTi AER	185.6	$> 10^6$ (device cycles)	Cooling power 203 W; 50.6 K
Sebald et al. (2023) [18]	NR-tube-based AER	2.70	n.r. (device)	SCP 0.14 W g ⁻¹ (1.5 W/10.6 g); > 8 K; COP 6
Zhou et al. (2024) [11]	Multi-material NiTi cascade	185.6	$> 10^6$ (device cycles)	75 K temperature lift
Ludwig et al. (2024) [19]	Bistable NR-foil cooling device	2.70	10^4 (device test; no significant performance loss)	SCP 3.4 W g ⁻¹ ; 4.2 K; COP 5.7
Zhang et al. (2025) [20]	Roller-cam NiTi-tube device	185.6	$> 2 \times 10^7$ (device runout; no failure of finned NiTi-tube)	40.6 W; 27.4 K
Zhou et al. (2025) [12]	Multi-cell NiTi AER	185.6	5×10^5 (device cycles)	SCP 12.3 W g ⁻¹ at 3.5 Hz; 1284 W at zero temperature lift
Wang et al. (2025) [21]	3D-printed TPU AER	7.55	$> 10^5$ (3D-printed TPU; not fatigue)	SCP 1.85 W g ⁻¹ ; 4.7 K
Zhou et al. (2026) [13]	Sub-zero NiTi-based prototype	185.6	n.r.	SCP 1.43 W g ⁻¹ ; -20 K
Zhong et al. (2026) [16]	3D-printed NiTi prototype	185.6	3.15×10^4 (prototype); $> 3 \times 10^6$ (material)	50 W; 20 K
<i>Polymer material-level fatigue (not device lifetime)</i>				
Sebald et al. (2016) [22]	NR elastocaloric material specimen	2.70	1.7×10^5 runout at engineering strain 2–5; no crack	Material ΔT decreased 12% and then stabilized
Chang et al. (2021) [23]	Prestrained natural-gum-rubber specimens	2.70	10^7 at 200%–400% strain; up to 8×10^7 at 250%–300%	n.r.
Wang et al. (2023) [24]	TPU plate fatigue specimen	7.55	10^6 at 13.08 MPa	n.r.

Note: Prices are the mean material price from Fig. 1 (NiTi: 185.583; TPU: 7.55; NR: 2.702 \$/kg); fabrication and balance-of-system costs are excluded. Reported cycle numbers are not directly comparable because loading mode, strain or stress range, frequency, failure criterion may differ. Polymer material-specimen results provide supporting fatigue evidence only and are not device-lifetime predictions. n.r., not reported.

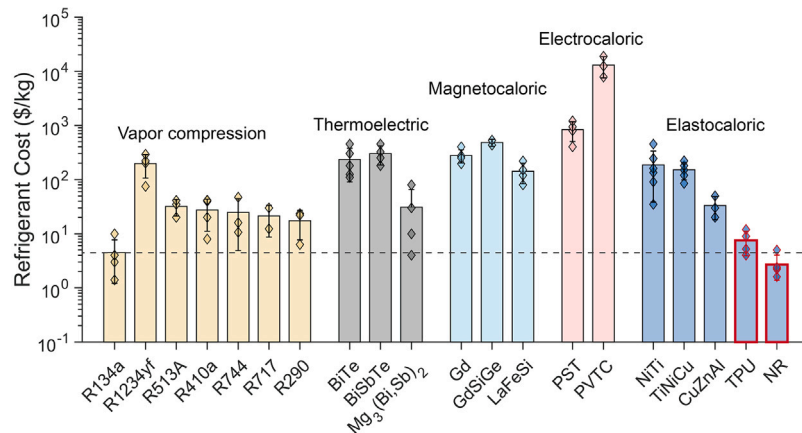


Fig. 1. Refrigerant price comparison across conventional vapor-compression refrigerants and representative solid-state caloric materials. Polymer-based elastocaloric materials (TPU and NR) are highlighted with red markers to emphasize their competitive cost relative to other solid-state refrigerants. The horizontal dashed line represents the price benchmark of R134a, serving as a reference for industrially established vapor-compression refrigerants. Cost data are compiled from Refs. [33–47]. The figure compares raw active-material prices only and does not represent a ranking of technical priority or a life-cycle-cost assessment. Abbreviations: NR — Natural Rubber; TPU — Thermoplastic Polyurethane; PVT — P(VDF-TrFE); PVTC — P(VDF-TrFE-CFE).

600%) to activate SIC and achieve significant temperature changes. Consequently, the regenerator length varies dynamically during operation ($L(t)$), altering the heat transfer area and temperature distribution,

which directly affects regenerator performance. Current modeling approaches rely mainly on simplified analytical methods, such as lumped-element models for thin foils or analytical models for simple tube

geometries [19,31,53]. These approaches cannot adequately capture the transient thermo-hydraulic coupling in regenerators with complex internal architectures such as microchannels.

To address this challenge, we develop a transient one-dimensional numerical model incorporating a dynamic mesh formulation to capture coordinate transformations and thermo-hydraulic behavior in large-deformation polymer-based AERs. The model enables systematic evaluation of different microchannel architectures and operating conditions. Using this framework, four representative microchannel geometries are analyzed to identify optimal structural parameters for high-performance polymer-based regenerators.

2. Numerical model of polymer-based elastocaloric regenerators

2.1. Operation of polymer-based elastocaloric regenerators

The operating principle follows the conventional four-step AER cycle. The temperature evolution of the fluid (T_f) and solid (T_s) under steady-state conditions is illustrated in Fig. 2:

- i. During adiabatic loading, the regenerator temperature (T_s) increases while the regenerator is stretched, leading to significant elongation of the fluid channels.
- ii. After loading, the piston drives the heat transfer fluid from the Cold Reservoir (CR) to the Hot Reservoir (HR). During this process, heat is transferred from the regenerator to the fluid, and the fluid temperature gradually approaches the solid temperature.
- iii. During adiabatic unloading, the regenerator temperature (T_s) decreases as the regenerator returns toward its original configuration.
- iv. The piston then drives the heat transfer fluid from HR back to CR, during which the fluid temperature decreases toward the solid temperature.

Polymer-based active elastocaloric regenerators operate under much larger strains than conventional SMA-based systems (typically 300%–600% compared with 3%–8% for SMAs), for which the deformation of spatial nodes is usually negligible in numerical simulations. In contrast, the large deformation in polymer-based regenerators significantly alters the regenerator geometry during operation and needs to be explicitly considered in the numerical model. During processes i and iii, the regenerator experiences substantial stretching, which modifies key geometric parameters such as channel length, cross-sectional area, hydraulic diameter, and heat transfer surface area. On one hand, the increased channel length and reduced cross section lead to higher pressure drops and increased pumping powers. On the other hand, these geometric changes increase the surface-to-volume ratio, which can enhance heat transfer performance. Consequently, the large deformation of polymer-based regenerators strongly influences overall system performance and is considered in both numerical modeling and device design.

2.2. Governing equations for elastocaloric regenerators

In this study, the regenerator is modeled as a one-dimensional porous medium consisting of a solid elastocaloric refrigerant and a heat-transfer fluid phase [48,52]. To account for the time-dependent variation of the regenerator geometry during large deformation, the governing equations are formulated in a transient form. In particular, the cross-sectional area A_c and the regenerator length L vary during operation and are therefore explicitly included in the model.

The energy conservation equations for the fluid and solid phases are expressed as follows:

$$\underbrace{A_c \epsilon \rho_f c_f \frac{\partial T_f}{\partial t}}_{\text{Heat storage}} \bigg|_x = \underbrace{\frac{Nuk_f}{D_h} a_s A_c (T_s - T_f)}_{\text{Convection}} + \underbrace{\frac{\partial}{\partial x} \left(k_{eff} A_c \frac{\partial T_f}{\partial x} \right)}_{\text{Axial conduction}} \quad (1)$$

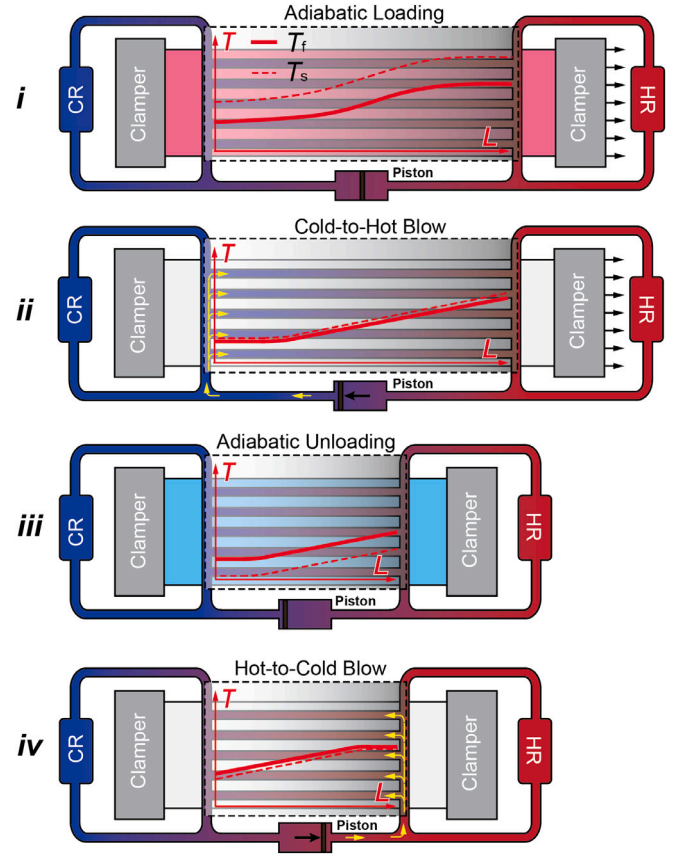


Fig. 2. Schematic illustration of the operating cycle of a large-deformation elastocaloric regenerator, with the evolution of the axial temperature gradient during cyclic operation. The solid and fluid temperatures are denoted by T_s and T_f , respectively. The temperature profiles correspond to the steady-state condition under cyclic operation.

$$\underbrace{-\dot{m}_f c_f \frac{\partial T_f}{\partial x}}_{\text{Advection}} + \underbrace{\left[\frac{\partial P}{\partial x} \frac{\dot{m}_f}{\rho_f} \right]}_{\text{Viscous dissipation}} = \underbrace{A_c (1 - \epsilon) \rho_s c_s \frac{\partial T_s}{\partial t}}_{\text{Heat storage}} \bigg|_x = \underbrace{A_c (1 - \epsilon) \dot{q}_{source}}_{\text{Elastocaloric source}} + \underbrace{\frac{Nuk_f}{D_h} A_c a_s (T_f - T_s)}_{\text{Convection}} + \underbrace{\frac{\partial}{\partial x} \left(k_{stat} A_c \frac{\partial T_s}{\partial x} \right)}_{\text{Conduction}} - \underbrace{U_{loss} P_{ext} (T_s - T_{amb})}_{\text{Ambient heat exchange}} \quad (2)$$

In these equations, the subscripts f and s denote the fluid and solid phases, respectively. T represents temperature and ϵ denotes the porosity of the regenerator. The thermophysical properties are characterized by the density ρ and specific heat capacity c of each phase. The regenerator geometry is described by the cross-sectional area A_c , hydraulic diameter D_h , and specific surface area a_s . U_{loss} is an effective overall heat-loss coefficient and P_{ext} is the external regenerator perimeter per unit axial length, calculated by $P_{ext} = 4\sqrt{A_c}$. Heat transfer is governed by the fluid thermal conductivity k_f , the static thermal conductivity k_{stat} , and the total effective conductivity k_{eff} . The total effective fluid conductivity accounts for axial dispersion effects, expressed as:

$$k_{eff} = k_{stat} + k_f D_p (Pe) \quad (3)$$

where D_p is dispersion coefficient, which is a function of Peclet number (Pe) and associated with the bed geometry. Pe number is calculated by $Pe = PrRe/\epsilon$. Correlations for corresponding dispersion coefficients can be found from [54,55]. And the static thermal conduction for microchannel regenerator is calculated by [56]:

$$k_{stat} = k_s(1 - \epsilon) + k_f\epsilon \quad (4)$$

where k_s is the thermal conductivity of solid; k_f is the thermal conductivity of fluid. Convective heat transfer between the fluid and solid phases is characterized by the Nusselt number (Nu). The flow dynamics are described by the mass flow rate $\dot{m}_f$ and the pressure gradient $\partial P/\partial x$.

The elastocaloric heat generation in the solid phase is represented by the source term $\dot{q}_{source}$, which accounts for both reversible elastocaloric heating and hysteresis losses during cyclic loading and unloading. The source term is expressed as

$$\dot{q}_{source} = \rho_s c_s \alpha \frac{\partial \epsilon}{\partial t} + \rho_s c_s \left| \alpha \frac{\partial \epsilon}{\partial t} \right| \frac{\eta_{hys}}{2} \quad (5)$$

The first term represents the reversible elastocaloric heat associated with the strain-rate-driven temperature change of the material. The second term accounts for hysteresis losses resulting from the asymmetry between the adiabatic temperature changes during loading and unloading. The hysteresis contribution is introduced through the hysteresis ratio (η_{hys}), which is commonly used to represent irreversible entropy generation in regenerative caloric simulations [52,57]. In this formulation, hysteresis losses are assumed to be proportional to the reversible elastocaloric heat and equally distributed between the loading and unloading processes.

The hysteresis ratio is defined as

$$\eta_{hys} = \frac{|\Delta T_{ad,loading}| - |\Delta T_{ad,unloading}|}{0.5 \cdot (|\Delta T_{ad,loading}| + |\Delta T_{ad,unloading}|)} \quad (6)$$

The elastocaloric coefficient α is determined from strain-dependent adiabatic temperature changes:

$$\alpha = \frac{\Delta T_{ad}(\epsilon_f) - \Delta T_{ad}(\epsilon_i)}{\epsilon_f - \epsilon_i} \quad (7)$$

where ϵ_i and ϵ_f denote the initial and final strains of the regenerator. Here, the baseline ΔT_{ad} data is extracted from the loading (heating) process, and corresponding ΔT_{ad} at the initial strain based on experimental strain-dependent ΔT_{ad} results in Fig. 11a.

In the present model, the elastocaloric coefficient α is derived solely from strain-dependent data. The temperature and path dependencies of the hysteresis are neglected in the model. The temperature dependence of ΔT_{ad} is neglected due to limited experimental information on $\Delta T_{ad}(T, \epsilon)$. Since the simulations in this work focus on a relatively small system temperature span of 8 K, the influence of ambient temperature on the elastocaloric response is assumed to be negligible.

2.3. Microchannel geometries and dynamic mesh transformation

Four microchannel regenerator geometries are investigated in this study: Parallel Plates (PP), square channels, triangular channels, and circular channels. These channel configurations have been widely used in magnetocaloric regenerators due to their favorable heat transfer characteristics [48,58,59]. The modeled regenerators have a rectangular cross section with width W and height H , within which the microchannels are uniformly distributed with constant wall thickness, as illustrated in Fig. 3a.

The characteristic geometric parameters for each channel type are defined as follows: spacing H_1 and wall thickness H_2 for the parallel-plate geometry; side length d for square channels; base width w and height h for isosceles triangular channels; and diameter d for circular channels. The corresponding hydraulic diameter, porosity, and specific surface area used in the simulations are summarized in Table 2.

Accurate modeling of porous elastocaloric regenerators requires appropriate correlations for fluid flow and heat transfer within the microchannels. In particular, two key parameters must be determined: the pressure drop across the regenerator and the convective heat transfer between the fluid and solid phases. The pressure drop is evaluated using the Darcy friction factor f_F , while the convective heat transfer is characterized by the Nusselt number Nu . Table 3 summarizes the friction factor and Nusselt number correlations adopted in this work for different channel geometries under laminar flow conditions.

To numerically solve the governing equations, the regenerator domain is discretized along the axial direction into N_x spatial nodes. The fluid and solid domains are solved simultaneously using a central difference scheme for spatial derivatives and implicit time discretization. The large deformation of polymer-based regenerators modifies the channel length and spatial distribution of computational nodes during operation, as illustrated in Fig. 3b-c. The governing equations are therefore solved on a dynamic mesh that evolves with the regenerator deformation.

The relationship between the reference coordinate x (undeformed state) and the spatial coordinate X (current deformed state) is defined by

$$X(t) = \lambda(t)x, \quad u_{mesh} = \frac{\partial X}{\partial t} \Big|_x = \dot{\lambda}x. \quad (8)$$

The arbitrary-Lagrangian-Eulerian (ALE) derivative and spatial transformation are [65–67]

$$\frac{\partial T}{\partial t} \Big|_X = \frac{\partial T}{\partial t} \Big|_x - u_{mesh} \frac{\partial T}{\partial X}, \quad \frac{\partial}{\partial X} = \frac{1}{\lambda} \frac{\partial}{\partial x}. \quad (9)$$

Consequently, the transport velocity and mass flow used in the moving-mesh control volume are

$$u_{rel} = u_f - u_{mesh}, \quad \dot{m}_{rel} = \rho_f \epsilon A_c u_{rel}. \quad (10)$$

During loading and unloading, the fluid follows the mesh so that $u_{rel} = 0$ and $\dot{m}_{rel} = 0$. During each fluid blow, $\dot{\lambda} = 0$, $u_{mesh} = 0$, and $\dot{m}_{rel} = \dot{m}_f$. This proportional mapping moves every node consistently with the regenerator.

2.4. Deformation-dependent parameters and modified governing equations

The stretching of the regenerator affects not only its geometry but also the fluid dynamics and heat transfer characteristics. To capture these effects, key geometric and flow parameters must be updated during deformation. In the following formulations, the superscript (*) denotes parameters modified by deformation.

a. Deformation factor

For an elastomer undergoing uniaxial tension with stretch ratio ($\lambda = L(t)/L_0$), the transverse deformation is governed by the Poisson effect. In large-deformation mechanics, the transverse stretch ratio λ_{trans} can be expressed as a power-law relationship with the longitudinal stretch [68]:

$$\lambda_{trans} = \lambda^{-\nu} \quad (11)$$

where ν is the Poisson's ratio. We define the transverse stretch as the deformation factor

$$\psi \equiv \lambda_{trans} = \lambda^{-\nu} \quad (12)$$

which describes the scaling of all transverse geometric dimensions during deformation.

The simulations use a constant Hencky–Poisson ratio (ν_H), for which $\lambda_{trans} = \lambda^{-\nu_H}$ and $J = \lambda^{1-2\nu_H}$, with $\nu_H = 0.49$ for NR and 0.45 for TPU [69–71]. At every time step, the selected volume response updates $A_c = A_{c,0} \lambda_{trans}^2$, $D_h = D_{h,0} \lambda_{trans}$, $a_s = a_{s,0}/\lambda_{trans}$, and $\rho_s = \rho_{s,0}/J$ (see Supplementary Section S4), while the pore geometry and porosity are held fixed.

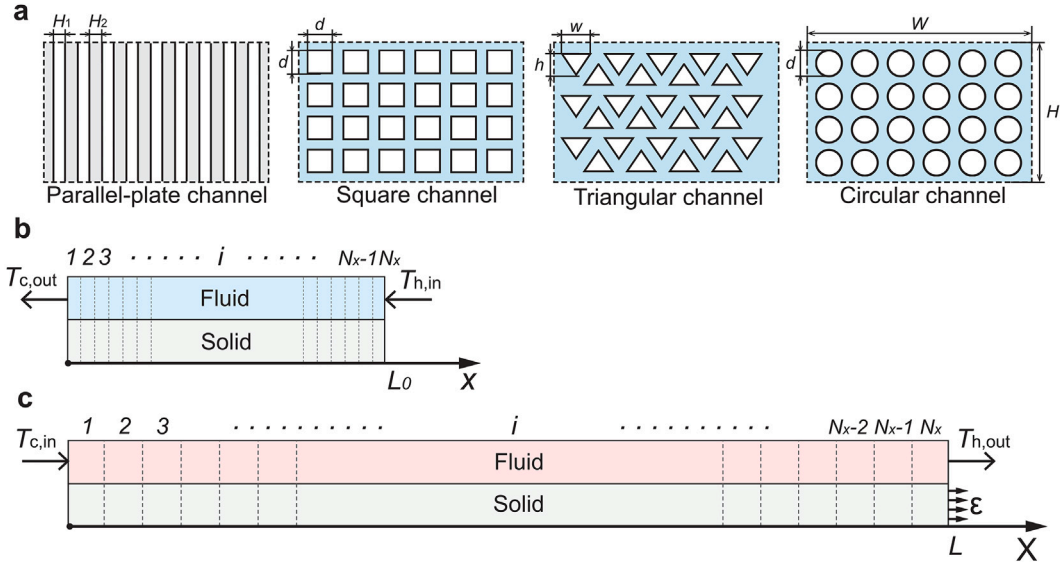


Fig. 3. Microchannel geometries and spatial discretization of large-deformation polymer regenerators. (a) Cross-sectional configurations of microchannel regenerators with parallel-plate, square, triangular, and circular architectures uniformly distributed within a rectangular domain of width W and height H . The geometric parameters are defined as spacing H_1 and wall thickness H_2 (parallel plate), side length d (square), base width w and height h (isosceles triangle), and diameter d (circular). (b) and (c) Spatial discretization of the fluid and solid domains in the undeformed (L_0) and stretched (L) states. The domains are discretized into N_x nodes along the axial direction, and the computational grid deforms dynamically with the regenerator according to the stretch ratio $\lambda = L/L_0$.

Table 2

Hydraulic diameter, porosity, and specific surface area of the microchannel regenerators used in the simulations [48,60].

Bed type	Dimension	Hydraulic diameter, D_h	Porosity, ϵ	Specific surface area, a_s
Parallel plate	H_1, H_2	$2H_1$	$\frac{H_1}{H_1+H_2}$	$\frac{2}{H_1+H_2}$
Square	d	d	ϵ	$\frac{4\epsilon}{D_h}$
Triangular	w, h	$\frac{2wh}{w + \sqrt{w^2 + 4h^2}}$	ϵ	$\frac{4\epsilon}{D_h}$
Circular	d	d	ϵ	$\frac{4\epsilon}{D_h}$

Table 3

Thermo-hydraulic correlations used for different microchannel regenerator geometries.

Bed types	Ref.	Friction factor (f_F)	Pressure drop (ΔP)	Nusselt number (Nu_0)	Remarks
Parallel plate	[61]	$\frac{96}{Re_{pp}}$	$f_F \frac{L}{D_h} \frac{\rho_f (v_s/\epsilon)^2}{2}$	$\left(7.541^{3.59} + (1.841 Gz^{\frac{1}{3}})^{3.59}\right)^{\frac{1}{5.59}}$ $Gz = Re_{pp} Pr \frac{D_h}{L}, Re_{pp} = Re/\epsilon$	Laminar flow, $Gz < 10^5$
Square	[61–63]	$\frac{56.91}{Re_{pp}}$	$f_F \frac{L}{D_h} \frac{\rho_f (v_s/\epsilon)^2}{2}$	$3.09 + \frac{0.0668 Gz}{1 + 0.04 Gz^{2/3}}$ $Gz = Re_{pp} Pr \frac{D_h}{L}, Re_{pp} = Re/\epsilon$	Laminar flow
Triangular	[58,62]	$\frac{203.36}{Re_{pp}} + 5.6 \times 10^{-8}$	$f_F \frac{L}{D_h} \frac{\rho_f (v_s/\epsilon)^2}{2}$	$2.28 + 1.1667 Gz^{\frac{0.8046}{L}}$ $Gz = Re_{pp} Pr \frac{D_h}{L}, Re_{pp} = Re/\epsilon$	Laminar flow
Circular	[48,64]	$\frac{64}{Re_{pp}}$	$f_F \frac{L}{D_h} \frac{\rho_f (v_s/\epsilon)^2}{2}$	$3.657 + \frac{0.19 Gz^{0.8}}{1 + 0.117 Gz^{0.467}}$ $Gz = Re_{pp} Pr \frac{D_h}{L}, Re_{pp} = Re/\epsilon$	Laminar flow, constant wall temperature

Note: For evaluated channel geometries, the effective Nusselt number used in the interfacial heat transfer calculations includes the Biot number correction ($Nu = Nu_0/(1 + Bi/a_0)$), as discussed in Supplementary Information Note 1. The correlations in this table were derived from rigid laminar channels and are used as quasi-steady local correlations evaluated with the instantaneous deformed hydraulic diameter, Reynolds number, Graetz number, and channel length. They do not include soft-wall motion, transient entrance effects, wall compliance, or flow-structure interaction. Therefore, the predicted performance of triangular, square, and circular channels should be interpreted as a model-based design trend that requires future experimental calibration in the dynamically deforming polymer-based regenerators.

Recent 3D-printed TPU studies report small volumetric variations and asymmetric fluid displacement [21,72]. Here we did a sensitivity analysis for a TPU-based triangular channel regenerator with varying v_H from 0.40 to 0.49 around the baseline value of 0.45. Across 0.1–1.0 Hz, the resulting optimized predictions differed from the baseline by –6.00% to +4.71% for maximum SCP and by –2.15% to +2.95% for maximum COP, detailed in Supplementary Section S4 and Supplementary Figure S13. The present model does not resolve pressure-induced channel deformation and fluid–structure interaction. To address these effects will require the future development of a highly

coupled 3D multiphysics model that integrates large-deformation solid mechanics, fluid dynamics, and heat transfer.

b. Deformation-dependent geometric parameters

Based on the deformation factor ψ , the instantaneous geometric parameters of the regenerator can be derived.

• Cross-sectional area

Assuming isotropic deformation in the transverse plane, both width and thickness scale with ψ . The cross-sectional area therefore becomes:

$$A_c^* = \psi W_0 \cdot \psi H_0 = A_{c,0} \psi^2 = A_{c,0} \lambda^{-2\nu} \quad (13)$$

• Hydraulic diameter

The hydraulic diameter is defined as

$$D_h = \frac{4A}{P} \quad (14)$$

where P is the wetted perimeter. Since both area and perimeter scale with the transverse dimensions, the hydraulic diameter scales linearly with ψ :

$$D_h^* = D_{h,0} \psi = D_{h,0} \lambda^{-\nu} \quad (15)$$

• Specific surface area

The specific surface area is defined as the heat transfer surface area per unit volume

$$a_s = \frac{A_{surf}}{V} \quad (16)$$

For a unit channel length, $A_{surf} \propto PL$ and $V \propto A_c L$. Therefore,

$$a_s^* \propto \frac{PL}{A_c L} \propto \frac{P_0 \psi}{A_{c,0} \psi^2} = a_{s,0} \frac{1}{\psi} = a_{s,0} \lambda^\nu \quad (17)$$

• Porosity

In the present model, the porosity is assumed to remain constant during deformation

$$\epsilon \approx \epsilon_0 \quad (18)$$

This assumption is valid for solid elastomers where the fluid channels deform synchronously with the surrounding material, without pore collapse and wall buckling. For compressible porous structures, such as foams or certain printed elastomers, the porosity may vary with deformation and would require additional modeling based on finite-element simulation or in-situ measurements [72].

c. Deformation-dependent fluid parameters and governing equations

The reduction in cross-sectional flow area leads to an increase in the interstitial fluid velocity, which modifies the Reynolds number and heat transfer characteristics.

The relative interstitial velocity used by the ALE control volume is

$$u_{rel}^* = \frac{\dot{m}_{rel}}{\rho_f \epsilon A_c^*} \quad (19)$$

The Reynolds number during a fluid blow therefore becomes

$$Re^* = \frac{\rho_f |u_{rel}^*| D_h^*}{\mu_f} \quad (20)$$

The corresponding Nusselt number is obtained from geometry-dependent correlations (Table 3),

$$Nu^* = f(Re^*, Pr, D_h^*/L(t)) \quad (21)$$

Since the static thermal conductivity k_{stat} is a function of porosity (Eq. (4)), with the constant ϵ assumption, k_{stat} will be a constant and deformation-independent. The total effective conductivity is then expressed as:

$$k_{eff}^* = k_{stat} + k_f D_p(Pe^*) \quad (22)$$

By incorporating the coordinate transformation and deformation-dependent parameters into the governing equations, the final form of

the energy equations for large-deformation elastocaloric regenerators can be obtained:

$$A_c^* \epsilon \rho_f c_f \left. \frac{\partial T_f}{\partial t} \right|_x = \frac{Nu^* k_f}{D_h^*} a_s^* A_c^* (T_s - T_f) + \frac{\partial}{\partial X} \left(k_{eff}^* A_c^* \frac{\partial T_f}{\partial X} \right) - \dot{m}_{rel} c_f \frac{\partial T_f}{\partial X} + \left| \frac{\partial P^*}{\partial X} \frac{\dot{m}_{rel}}{\rho_f} \right| \quad (23)$$

$$A_c^* (1 - \epsilon) \rho_s c_s \left. \frac{\partial T_s}{\partial t} \right|_x = A_c^* (1 - \epsilon) \dot{q}_{source} + \frac{Nu^* k_f}{D_h^*} a_s^* A_c^* (T_f - T_s) + \frac{\partial}{\partial X} \left(k_{stat} A_c^* \frac{\partial T_s}{\partial X} \right) - U_{loss} P_{ext}^* (T_s - T_{amb}) \quad (24)$$

For the deforming square external envelope, $P_{ext}^* = 4\sqrt{A_c^*}$ is updated from the instantaneous cross-sectional area. All storage, convection, conduction, advection, viscous-source, elastocaloric-source, and ambient-loss terms therefore use the same instantaneous geometry. $U_{loss} = 15 \text{ W m}^{-2} \text{ K}^{-1}$ was adopted for all microchannel simulations at $T_{amb} = 298 \text{ K}$, obtained from the validation of rubber-tube based regenerators discussed in the next section. Their stretch scaling depends on the selected J model.

2.5. Performance metrics of the elastocaloric regenerator

The cooling performance of the AER depends on both geometric and operational parameters, including the regenerator porosity (ϵ), characteristic channel size, mass flow rate ($\dot{m}_f$), heat sink temperature (T_H), operating frequency (f), and applied stretch ratio (λ). A dimensionless volume ratio (V^*), defined as the ratio between the displaced fluid volume and the total void volume of the regenerator during one heat transfer blow, is used to regulate the mass flow rate. This parameter has been widely used in experimental studies of active caloric regenerators [11,18] and is defined as

$$V^* = \frac{V_{fluid}}{V_{void}} \quad (25)$$

where V_{fluid} represents the displaced fluid volume and V_{void} is the total void volume inside the regenerator.

Accordingly, the mass flow rate of the heat transfer fluid is calculated as

$$\dot{m}_f = \frac{V^* \epsilon (L_0 A_{c,0}) \rho_f}{t_{blow}} \quad (26)$$

where L_0 and $A_{c,0}$ denote the initial length and cross-sectional area of the regenerator, respectively, and t_{blow} represents the duration of a single heat transfer blow (hot or cold).

When the AER reaches steady state, its cooling performance is evaluated using four key performance metrics: temperature span, cooling power, SCP, and Coefficient of Performance (COP).

The temperature span represents the temperature difference between the hot-side and cold-side outlet temperatures and is defined as:

$$T_{span} = T_h - T_c \quad (27)$$

The cycle-averaged cooling power ($\dot{Q}_c$) is defined as the heat absorbed from the cold-side heat source by a control volume around the cold-side heat exchanger during the hot-to-cold blow (Phase iv):

$$\begin{aligned} \dot{Q}_c &= f \int_0^{t_{blow}} |\dot{m}_{rel}(t)| c_f \left[T_{f,c,out}^{HX}(t) - T_{f,c,in}^{HX}(t) \right] dt \\ &= f \int_0^{t_{blow}} |\dot{m}_{rel}(t)| c_f \left[T_c - T_f(0, t) \right] dt, \end{aligned} \quad (28)$$

where $t = 0$ denotes the start of Phase iv, t_{blow} is its duration, and the cold-side heat-exchanger inlet and outlet temperatures satisfy

Table 4

Key parameters used in the numerical modeling of microchannel elastocaloric regenerators.

Parameters	Value
Regenerator geometries	Parallel plate Square channel Triangular channel Circular channel
Regenerator material	Natural rubber TPU (HM-PTMEG)
Heat transfer fluid	Water
Thermal conductivity of solid, k_s [W m ⁻¹ K ⁻¹]	0.15 (NR), 0.26 (TPU)
Thermal conductivity of fluid, k_f [W m ⁻¹ K ⁻¹]	0.56
Applied stretch ranges, λ [-]	3 – 6, 3 – 7
Specific heat of solid, c_s [J kg ⁻¹ K ⁻¹]	1800 (NR) [73], 1800 (TPU) [74]
Specific heat of fluid, c_f [J kg ⁻¹ K ⁻¹]	4180
Poisson's ratio, ν [-]	0.49 (NR) [70], 0.45 (TPU) [71]
Density of solid, ρ_s [kg m ⁻³]	930 (NR), 1030 (TPU)
Density of liquid, ρ_f [kg m ⁻³]	998
Total regenerator length, L [mm]	30
Regenerator width, W [mm]	50
Regenerator height, H [mm]	50
Characteristic pore size, [mm]	$H_1 = H_2 = 0.3, 0.15$ (parallel plate) $d = 0.3$ (square) $w = h = 0.3, 0.485$ (triangular) $d = 0.3$ (circular)
Porosity of the regenerator, ϵ [-]	50%
Hot-side temperature, T_h [K]	298
Ambient temperature, T_{amb} [K]	298
Ambient heat-loss coefficient, U_{loss} [W m ⁻² K ⁻¹]	15
Temperature span, T_{span} [K]	8
Operating frequency, f [Hz]	0.1 – 1
Dimensionless volume ratio, V^* [-]	0.2 – 2
Hysteresis ratio, η_{hys} [-]	0.07 (NR) [75], 0.062 (TPU) [29]

Note: For the geometric optimization study discussed in Section 4.2, the porosity is varied from 30% to 70%, and the characteristic microchannel size ranges from 100 to 1000 μm .

$T_{f,c,in}^{HX}(t) = T_f(0, t)$ and $T_{f,c,out}^{HX}(t) = T_c$, respectively. The specific cooling power is therefore

$$SCP = \frac{\dot{Q}_c}{\dot{m}_s}. \quad (29)$$

Here, COP denotes the idealized regenerator-level coefficient of performance, $COP_{reg,ideal}$. It includes the modeled material hysteresis work and ideal pumping work, but excludes actuator inefficiency, pump inefficiency, valves, manifolds, external heat exchangers, controls, electronics not represented by the adopted boundary model, and other auxiliaries, which is not a system COP.

The idealized regenerator-level COP is defined as

$$COP = \frac{\dot{Q}_c}{\dot{W}_{net}} \quad (30)$$

The total net work input includes both the mechanical work required to operate the elastocaloric material ($\dot{W}_{mech}$) and the pumping work required to circulate the heat-transfer fluid ($\dot{W}_{pump}$):

$$\dot{W}_{net} = \dot{W}_{mech} + \dot{W}_{pump}, \quad (31)$$

$$\dot{W}_{mech} = \frac{\dot{m}_s}{\rho_s} \oint \sigma d\epsilon, \quad (32)$$

$$\dot{W}_{pump} = \int_{t_{blow}} \Delta P(t) \frac{\dot{m}_f(t)}{\rho_f} dt. \quad (33)$$

where σ is the stress and the closed-loop integral represents the net mechanical input when mechanical work recovery is assumed. The pumping work is evaluated from the pressure-drop during the two fluid-blow phases in one cycle.

The main parameters used in the numerical simulation are summarized in Table 4. The four regenerators share identical rectangular dimensions, porosity, and characteristic pore sizes to enable a direct comparison between different microchannel architectures. The operating frequency and dimensionless volume ratio (which determines

$\dot{m}_f$) are selected as the primary optimization parameters. The applied stretch ratio in the simulations ranges from 3 to 7, while an additional comparison with a lower stretch range (3–6) is presented in the final section of the study. Furthermore, a recently developed high-performance thermoplastic polyurethane (TPU, HM-PTMEG) is incorporated into the simulations and compared with a conventional natural rubber regenerator.

2.6. Numerical solution procedure

The governing energy equations are discretized in both space and time using a Finite Difference Method (FDM). A first-order upwind scheme is applied to the advection terms, while axial thermal conduction in the solid and fluid phases is discretized using a second-order central difference scheme.

Temporal integration is performed using a fully implicit backward Euler method, which is unconditionally stable. This implicit formulation allows the use of relatively large time steps without violating the Courant–Friedrichs–Lewy (CFL) stability condition, thereby reducing the computational cost of long cyclic simulations. Based on grid independence tests, the regenerator length is discretized into $N_x = 1000$ uniform spatial nodes. The time domain of a single elastocaloric cycle is divided into $N_t = 1000$ time steps, which are distributed according to the durations of the four operational phases, a discretization scheme proven to guarantee numerical convergence shown in Supplementary Figure S1. The time step is therefore given by

$$\tau^n = \frac{\tau}{N_t} \quad (34)$$

where τ is the cycle period.

The implicit FDM formulation converts the continuous partial differential equations into a block-tridiagonal system of linear equations of the form $\mathbf{A}\mathbf{T} = \mathbf{B}$. Here, $\mathbf{A}$ is the sparse global coefficient matrix representing thermo-hydraulic transport processes, and $\mathbf{B}$ is the

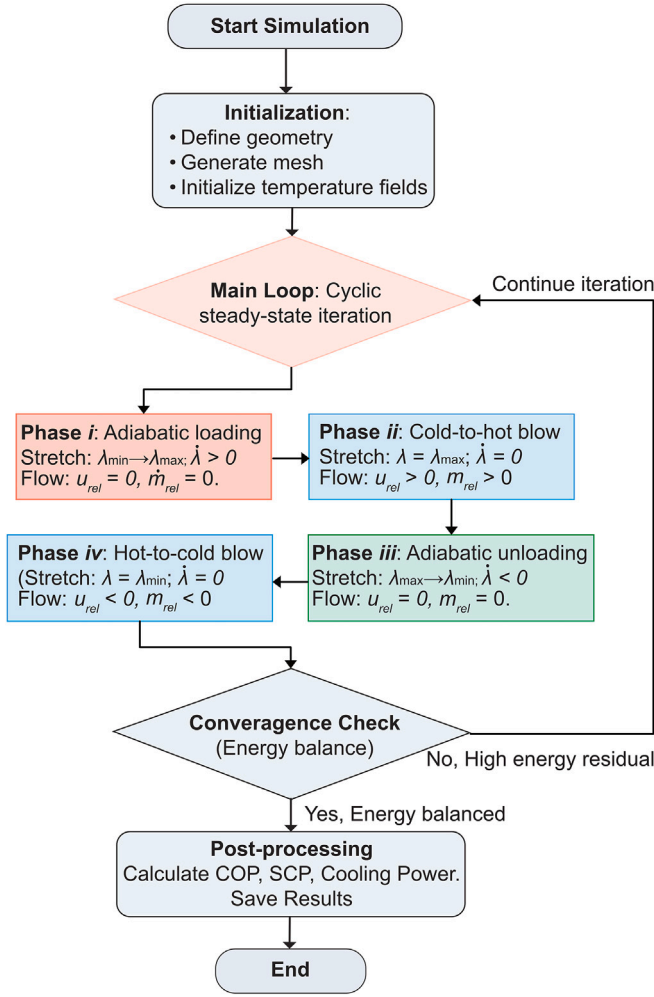


Fig. 4. Numerical solution procedure for the polymer-based elastocaloric regenerator model. The one-dimensional transient model simulates the active elastocaloric regenerator through a four-step thermodynamic cycle. After initializing the geometry, spatial mesh, and temperature field, the solver iteratively updates strain-dependent geometric and hydraulic parameters together with the elastocaloric heat source. The coupled solid–fluid governing equations are solved implicitly using the sparse linear solve $T = A \backslash B$ during alternating adiabatic deformation phases (Phases *i* and *iii*) and fluid flow phases (Phases *ii* and *iv*). The transient loop continues until the global energy balance convergence criterion is satisfied, yielding the steady-state performance metrics of the regenerator, including the COP and specific cooling power SCP.

right-hand-side vector containing the temperature distribution from the previous time step. The resulting system is solved using the sparse MATLAB backslash operator, $T = A \backslash B$. Accurate implementation of boundary conditions is essential for maintaining global energy conservation. At the fluid inlet, a Dirichlet boundary condition ($T_f = T_{in}$) is imposed, while a zero-gradient Neumann boundary condition ($\partial T_f / \partial x = 0$) is applied at the outlet.

The 1D simulation follows the thermodynamic cycle illustrated in Fig. 4. During adiabatic loading (Phase *i*), the regenerator is stretched from the minimum stretch ratio λ_{min} to the maximum stretch ratio λ_{max} . The geometric solver dynamically updates the deformation-dependent parameters, including the hydraulic diameter and specific surface area, according to the instantaneous stretch ratio. The elastocaloric source term, which accounts for both reversible elastocaloric heating and hysteresis losses, is then applied to the solid energy equation. During this phase, the relative fluid velocity and relative mass flow rate are zero. During cold-to-hot blow (Phase *ii*), the heat transfer fluid flows

from the cold end toward the hot end ($\dot{m}_{rel} > 0$). Convective heat transfer transports the elastocaloric heat generated during stretching to the hot-side heat exchanger. During adiabatic unloading (Phase *iii*), the elastomeric regenerator relaxes from λ_{max} back to λ_{min} . The geometric parameters are correspondingly updated, and an endothermic elastocaloric cooling source term is applied to the solid energy equation. During hot-to-cold blow (Phase *iv*), the fluid flow reverses direction ($\dot{m}_{rel} < 0$), moving from the hot end toward the cold end and absorbing heat from the cooling load.

To initiate the simulation, a linear temperature distribution between T_{cold} and T_{hot} is prescribed along the regenerator nodes. The four-step algorithm is then iterated until the system reaches a Cyclic Steady State (CSS). The CSS condition is evaluated by examining the global energy balance over one complete cycle [56]. The iterative solver terminates when the global energy conservation residual falls below the prescribed tolerance $E_{tol} = 10^{-6}$. Once CSS is achieved, the performance metrics of the active elastocaloric regenerator, including the SCP and COP, are calculated and exported.

The present framework is a periodic regenerator model with prescribed reservoir inlet temperatures rather than a complete-device start-up model. During the two fluid-blow phases, T_h and T_c are imposed as inlet boundary conditions, and the initial linear temperature distribution is used as a numerical starting guess to accelerate convergence to CSS. The model therefore resolves the intra-cycle evolution of $T_f(x, t)$ and $T_s(x, t)$ at a specified temperature span, but it does not predict an AER developing from an initially isothermal state to a final no-load temperature span.

3. Validation of the numerical model

Model validation is performed by comparing the numerical predictions with experimental data obtained from a rubber-based regenerative elastocaloric cooler developed by Sebald et al. [18]. In their work, the cooling performance of natural rubber tube bundles operating under active regenerative cycles was experimentally characterized.

Table 5 summarizes the geometric configurations, material properties, and operating parameters of the two experimental test configurations (Device#1 and Device#2), together with the corresponding parameters used in the numerical simulations. Device#1 consists of 19 tubes, resulting in a relatively high bed porosity, whereas Device#2 employs 55 tubes with smaller diameters to achieve a denser packing fraction. The simulations are performed using the nominal geometric and operating parameters (e.g., stretch ratio, frequency, volume ratio, and reservoir temperatures) reported in the experiments in Table 5. Additional simulation parameters for natural rubber and heat transfer fluids (e.g. k_s , k_f , c_s , c_f , ν) not reported in the experiment are obtained from the literature in Table 4. The numerical model represents an idealization of the physical elastocaloric system, which assumes uniform fluid flow distribution through the tube bundles, perfect adiabatic boundaries to the surroundings, and idealized convective heat transfer, whereas the actual experimental conditions are inherently more complex. We validated the simulation results across two different volume ratio cases: $V^* = 1.0$ and $V^* = 1.5$, and the simulation uncertainty is analyzed with $\pm 5\%$ parameter perturbations, including U_{loss} and k_s , and ± 0.5 K variation for T_{amb} . The $V^* = 1.0$ data are used for the primary validation in Figs. 5 and 6, and another $V^* = 1.5$ validation case is reported in Supplementary Figures S2 and S3. A common $U_{loss} = 15.237$ W m⁻² K⁻¹ was obtained under the constrained fit. Therefore, the 15 W m⁻² K⁻¹ value is used throughout the microchannel simulations. Full calibration details are provided in Supplementary Note S5.

Fig. 5 compares the $V^* = 1.0$ predictions with the experiments and their reported uncertainties. For Device #1, the RMSE values of cooling power and COP are 0.078 W and 0.431. For Device #2, they are 0.094 W and 0.239, respectively, which indicates a good agreement of the predictions and experiments. At the smallest Device #2 temperature span, the measured and predicted COP values are 6.031 and 6.073,

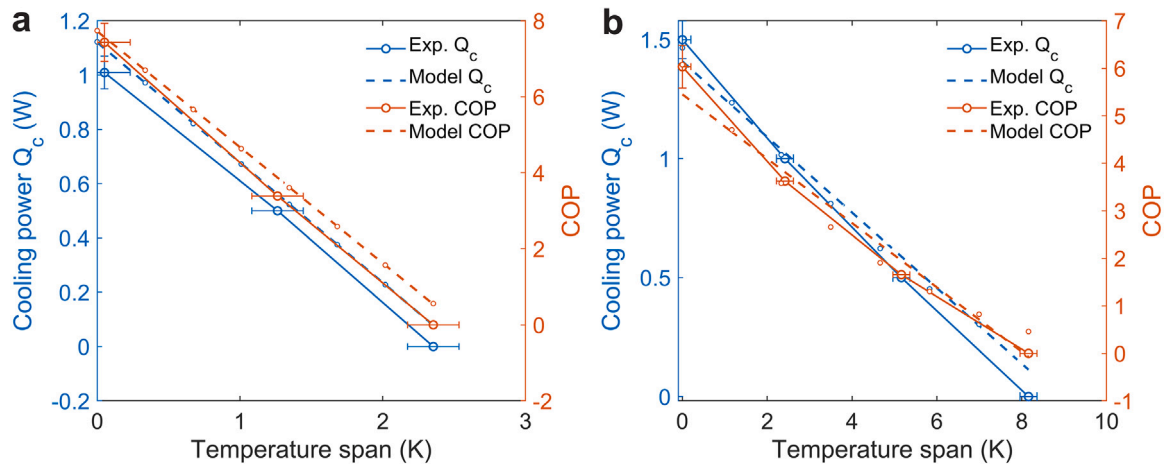


Fig. 5. Validation of the numerical model against the natural-rubber tube-bundle experiments of Sebald et al. [18] at 0.1 Hz, $V^* = 1.0$, and stretch ratio 3.5–5.5. Comparison of Cooling Power (Q_c) and Coefficient of Performance (COP) as functions of temperature span for (a) Device #1 and (b) Device #2. Solid lines with markers and error bars indicate experimental data, whereas dashed lines represent numerical predictions.

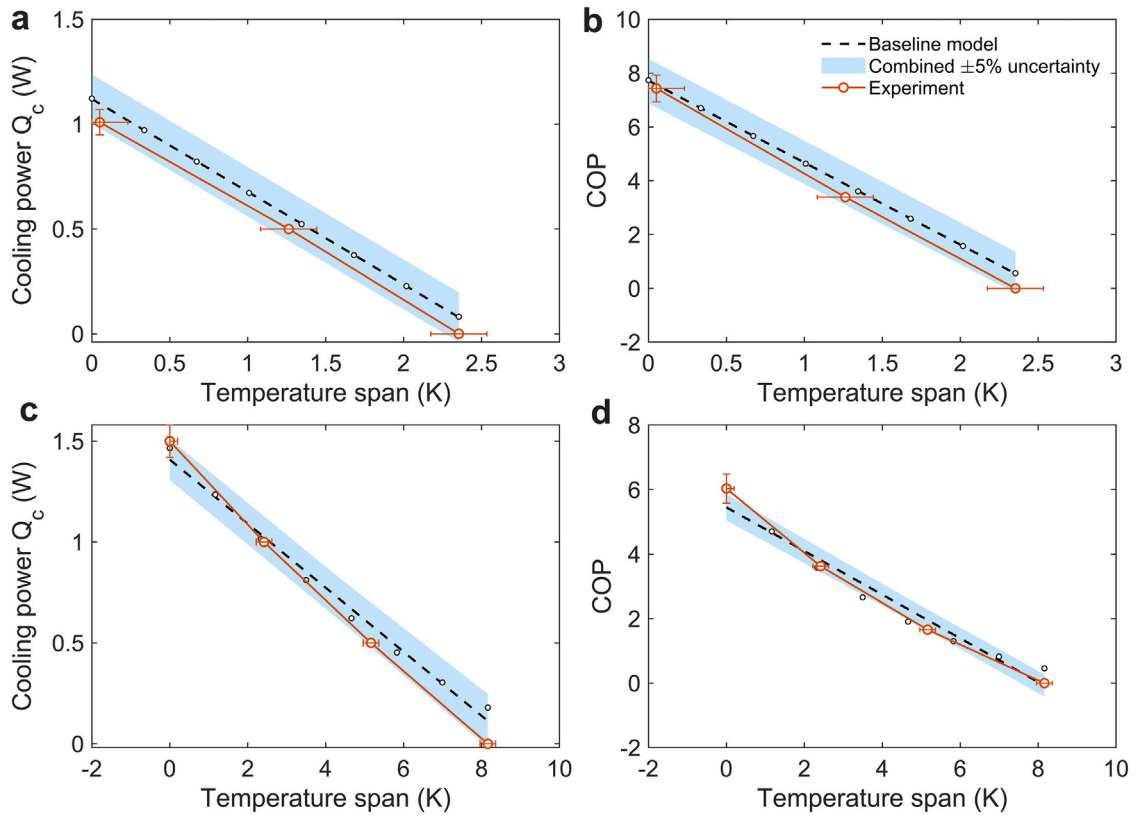


Fig. 6. Uncertainty analysis for the $V^* = 1.0$ validation case from Fig. 5. Dashed curves denote the baseline model, the blue error bands are the combined envelope of the $\pm 5\%$ parameter variations (with T_{amb} varied by ± 0.5 K), and orange markers with horizontal and vertical error bars denote the experiments.

Table 5

Experimental parameters of the rubber-tube elastocaloric regenerators reported by Sebald et al. [18] and used in the corresponding numerical simulations.

Parameters	Device#1 (Experiment & Simulation)	Device#2 (Experiment & Simulation)
Material	Natural rubber	Natural rubber
Geometry	Rubber tube bundles	Rubber tube bundles
Hot reservoir temperature (K)	299	299
Ambient temperature, T_{amb} (K)	298	298
Mass of the material (g)	4.9	10.6
Outer tube diameter (mm)	4.76	3.18
Inner tube diameter (mm)	3.18	1.59
Porosity, ϵ	44.6%	25%
Number of tubes	19	55
Regenerator length, L (mm)	30	30
Heat transfer fluid	Water	Water
Stretch ratio range, λ	3.5–5.5	3.5–5.5
Operating frequency, f (Hz)	0.1	0.1
Volume ratio, V^*	1.0, 1.5	1.0, 1.5
U_{loss} ($\text{W m}^{-2} \text{K}^{-1}$)	15.237 (Sim.)	15.237 (Sim.)

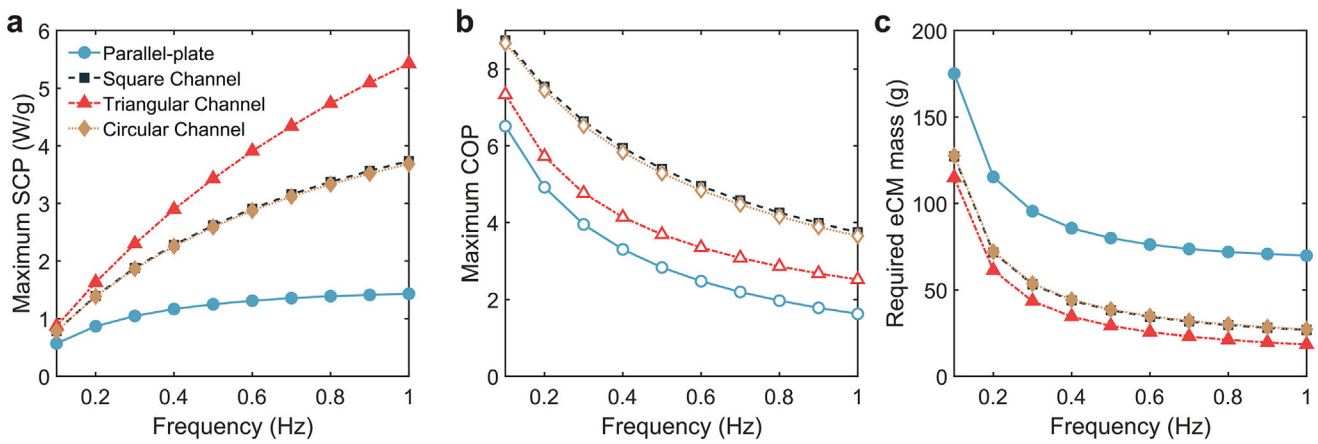


Fig. 7. Geometry-dependent cooling performance of NR-based elastocaloric microchannel regenerators under a temperature span of 8 K. (a) Maximum SCP as a function of operating frequency for different channel geometries. (b) Maximum COP as a function of frequency. (c) Estimated elastocaloric material (eCM) mass required to achieve a cooling capacity of 100 W.

respectively. The eight simulated points reproduce the approximately linear decrease of cooling power. Fig. 6 shows the corresponding the uncertainty analysis of the predicted results with the error band. The blue band is the envelope of individual $\pm 5\%$ perturbations of the applicable heat-transfer, heat loss parameters, and T_{amb} is varied by ± 0.5 K. Considering the horizontal and vertical experimental uncertainties, all $V^* = 1.0$ cooling-power and COP interval intersects the corresponding model envelope.

4. Results and discussion

4.1. Performance of natural rubber-based microchannel regenerators

4.1.1. Effect of microchannel geometry on cooling performance

Fig. 7 presents the simulated cooling performance of four NR-based microchannel regenerators with different channel geometries (see Fig. 4a). The parameters for the individual regenerators are taken from Table 4. The maximum SCP and COP shown in Fig. 7a and b were obtained by optimizing the dimensionless volume ratio (V^*), defined as the ratio of the pumped heat-transfer fluid volume to the regenerator void volume per blow, under a fixed temperature span of 8 K across different operating frequencies (see Supplementary Figures S4–S8).

Among the evaluated geometries, the triangular-channel regenerator delivers the highest SCP, whereas the square and circular channels

exhibit superior COP. For all microchannel geometries, SCP initially exhibits an increasing trend with operating frequency due to the enhanced heat transfer enabled by more frequent regeneration cycles. At 1 Hz, the triangular channel reaches a maximum SCP of 5.4 W/g, while the square and circular channels achieve values close to 3.7 W/g. Compared with conventional parallel-plate regenerators (Supplementary Figures S11–S13), these microchannel architectures significantly improve cooling performance owing to their larger specific surface area, which enhances solid–fluid heat exchange at the microscale. However, the sharp corner regions in the triangular channels increase hydraulic resistance, resulting in a higher pressure drop. The associated pump work penalty leads to a slightly lower COP compared with the square and circular channels.

To evaluate the engineering implications of these performance differences, Fig. 7c estimates the required mass of Elastocaloric Material (eCM) to achieve a cooling capacity of 100 W at an 8 K temperature span based on the maximum SCP values. Owing to its superior SCP, the triangular-channel regenerator requires the least material, ranging from 115 g to 18.4 g of NR as the operating frequency increases from 0.1 Hz to 1 Hz. In comparison, the square and circular channels require approximately 27.1 g of NR at 1 Hz. In practical elastocaloric cooling systems, high system cost and bulky configurations are largely associated with expensive shape memory alloys and the heavy actuators required to drive them. Reducing the required eCM mass therefore directly lowers material cost and system weight.

4.1.2. Influence of porosity and channel size on cooling performance

To investigate the influence of regenerator porosity and microchannel dimensions on cooling performance, the triangular-channel regenerator is taken as a representative case. The porosity is varied from 30% to 70%, and the characteristic pore size is varied from 100 to 1000 μm to optimize the geometric parameters.

Fig. 8a and b compare the maximum SCP and COP as functions of operating frequency for different porosities at a fixed pore size of 300 μm (i.e., the characteristic triangular channel size, $w = h = 300 \mu\text{m}$). Regenerators with higher porosity (above 50%) exhibit larger maximum SCP, showing approximately a 1.5-fold improvement compared with the 30% porosity case, while the SCP values of the 50% and 70% regenerators are nearly identical. However, the COP trends in Fig. 8b show that the regenerator with 30% porosity achieves a higher COP than the higher-porosity ones. Although higher porosity generally reduces hydraulic resistance by enlarging the flow channels, it shifts the optimal dimensionless volume ratio (V^*). However, since V^* is defined relative to the internal volume of the regenerator, it does not directly reflect the absolute mass flow rate. Under optimized operating conditions, the fluid velocity and the associated pressure drop (ΔP) may still increase. To clarify this effect, the pressure drop and the contribution of pump work (W_{pump}) to the total net work (W_{net}) were evaluated at the optimal V^* for each regenerator. The results show that the pressure drop in the 70% porosity regenerator at its optimal V^* (9.0 bar at 1 Hz and 2.1 bar at 0.1 Hz) is higher than that in the 30% porosity regenerator (8.6 bar at 1 Hz and 1.8 bar at 0.1 Hz). Correspondingly, the pump work accounts for approximately 51% of W_{net} in the 70% AER at 1 Hz, whereas it contributes only about 30% pump work fraction in the 30% AER (see Supplementary Figures S4–S8). This increased pumping work ratio leads to the lower COP in high-porosity regenerators. Therefore, the triangular-channel regenerator with 50% porosity provides a balanced performance, maintaining an SCP comparable to that of the 70% regenerator while exhibiting a COP only about 10% lower than that of the 30% case. For comparison, circular-channel regenerators with varying porosities were also evaluated (Supplementary Figure S15), which similarly indicate that a porosity of approximately 50% yields near-optimal performance.

In Figs. 8c and 8d, the regenerator geometry is further parameterized by varying the characteristic pore size ($h = w$) from 100 to 1000 μm across operating frequencies from 0.1 to 1.0 Hz at the optimized porosity of 50%. As shown in Fig. 8c, the maximum SCP increases first and decreases with increasing pore size at the evaluated frequency range. The highest SCP of 6.2 W/g is obtained at a 200 μm pore size under an operating frequency of 1.0 Hz. This behavior results from the larger specific surface area of smaller microchannels (Supplementary Figure S11), which enhances solid–fluid heat transfer. In contrast, the performance enhancement associated with increasing operating frequency becomes less pronounced at larger pore sizes (e.g., 1000 μm), indicating that insufficient heat transfer surface area eventually limits the cooling capacity.

The COP exhibits a different trend, as shown in Fig. 8d, increasing initially and then decreasing with increasing pore size due to the trade-off between heat transfer effectiveness and viscous dissipation. For smaller pores ($< 400 \mu\text{m}$), although high SCP values are achieved, the narrow channels introduce higher viscous resistance and pumping losses, which reduce the overall efficiency. As the pore size increases toward $\sim 500 \mu\text{m}$, the flow resistance decreases substantially, resulting in an improved balance between pumping work and cooling capacity, where the COP reaches its maxima across all operating frequencies. However, further increasing the pore size ($> 500 \mu\text{m}$) weakens the thermal interaction between the oscillating fluid and the solid matrix, leading to greater irreversible heat transfer losses and the decrease in COP. Combining the results from Figs. 8c and 8d, a characteristic pore size of approximately 400 μm emerges as the optimal design for the triangular-channel NR regenerator, maximizing COP while maintaining

a high SCP. A similar analysis for circular-channel regenerators indicates an optimal pore size of approximately 200 μm (see Supplementary Figure S16).

To show the importance of the dynamic-mesh model for large-deformation regenerators, a controlled comparison with the fixed-geometry model based on a 50% NR triangular-channel with 300 μm channel size is presented in Supplementary Note S6 and Figures S20–S21. Across 0.1–1.0 Hz, the dynamic-mech model predicts 20.9%–84.3% higher maximum SCP, whereas the fixed-geometry mode predicts 11.6%–39.2% higher maximum COP. The dynamic-mech model should be prioritized for the large-deformation elastocaloric regenerators (strain $> 10\%$), nearly incompressible elastomers, and strongly deformable fluidic channels. The fixed-geometry mode may be sufficient only for small deformation regenerators (strain $< 10\%$) or nearly invariant flow geometry.

4.2. Performance of HM-PTMEG TPU-based microchannel regenerators

Although the above-mentioned results demonstrate the feasibility of polymer-based elastocaloric cooling using NR microchannel regenerators, the relatively small adiabatic temperature change ($\Delta T_{\text{ad}} \sim 8 \text{ K}$) of NR, compared with conventional NiTi-based shape memory alloys ($\Delta T_{\text{ad}} \sim 25 \text{ K}$), limits the achievable cooling capacity. Recently, a novel thermoplastic polyurethane (TPU, HM-PTMEG) has been reported to exhibit a large ΔT_{ad} of around 20.1 K, which is comparable to NiTi, while maintaining low mechanical hysteresis [29]. This material therefore presents a promising opportunity for high-performance polymer regenerators. To evaluate its potential, Fig. 9 illustrates the simulated cooling performance of four regenerators utilizing TPU (HM-PTMEG) under a temperature span of 8 K with optimized V^* . The parameters for the individual regenerators are taken from Table 4.

As shown in Fig. 9a, replacing NR with TPU (HM-PTMEG) substantially improves the SCP across all frequencies, with all microchannel regenerators consistently outperforming the parallel-plate regenerator. The triangular TPU regenerator is numerically predicted to reach 17.4 W/g at 1 Hz, which is more than three times the 5.4 W/g achieved in the NR-based triangular regenerator. The square and circular channels also deliver considerable specific cooling power, both approaching 13.1 W/g at 1 Hz. However, the efficiency comparison in Fig. 9b shows that despite achieving larger SCP values, the narrow corners of the triangular channel produce higher flow resistance during operation (Supplementary Figure S14). This increases the fraction of pump work in the total input energy, leading to a slightly lower COP compared with the circular and square channel regenerators. Benefiting from more favorable hydrodynamics, the square and circular channels maintain relatively high COP values between 6.4 and 11.7 while still delivering substantial cooling power.

Based on the maximum SCP results, Fig. 9c estimates the TPU mass required for a 100 W cooling device. Owing to the high cooling power density of TPU, the triangular-channel regenerator requires only 5.7 g of active material at 1 Hz, and 43 g even when operated at a low frequency of 0.1 Hz. In contrast, the NR counterpart requires 18.4 g and 115 g under identical operating frequencies. The square- and circular-channel regenerators require only about 7.6 g of TPU at 1 Hz. This substantially reduced material requirement ($< 10 \text{ g}$ for a 100 W device) indicates a preliminary numerical reduction in active-material demand, but system compactness, manufacturing cost, and experimental performance require future validation.

4.3. Performance comparison of different microchannels at an equal hydraulic diameter

To separate the influence of channel shape from the hydraulic diameter, the four channel geometries were further compared at an equal undeformed hydraulic diameter of $D_{h,0} = 300 \mu\text{m}$. The square-channel side length and circular-channel diameter were therefore both

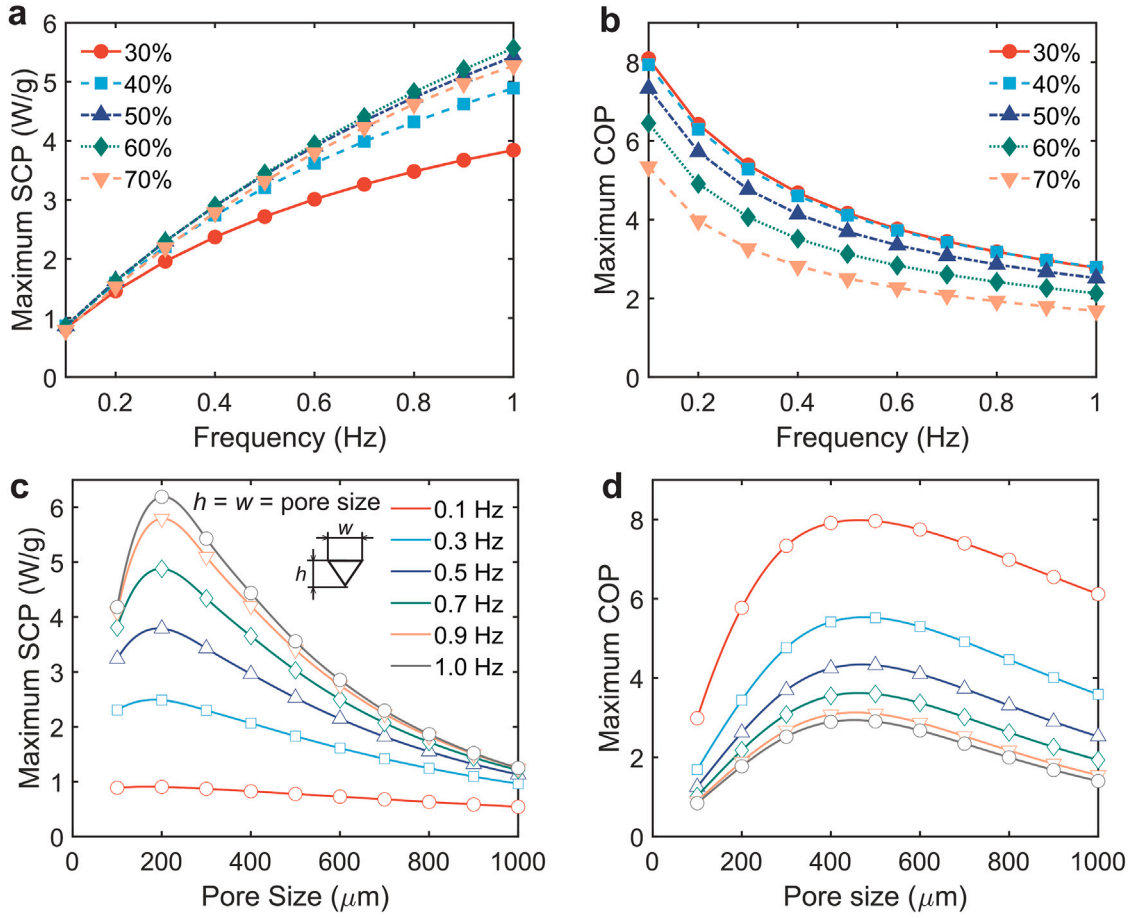


Fig. 8. Influence of geometric parameters on the cooling performance of the triangular-channel regenerator. (a) Maximum SCP and (b) maximum COP as functions of operating frequency for regenerators with different porosities at a fixed pore size of 300 μm . (c) Maximum SCP and (d) maximum COP as functions of pore size ($h = w$) at different operating frequencies for the optimized porosity of 50%.

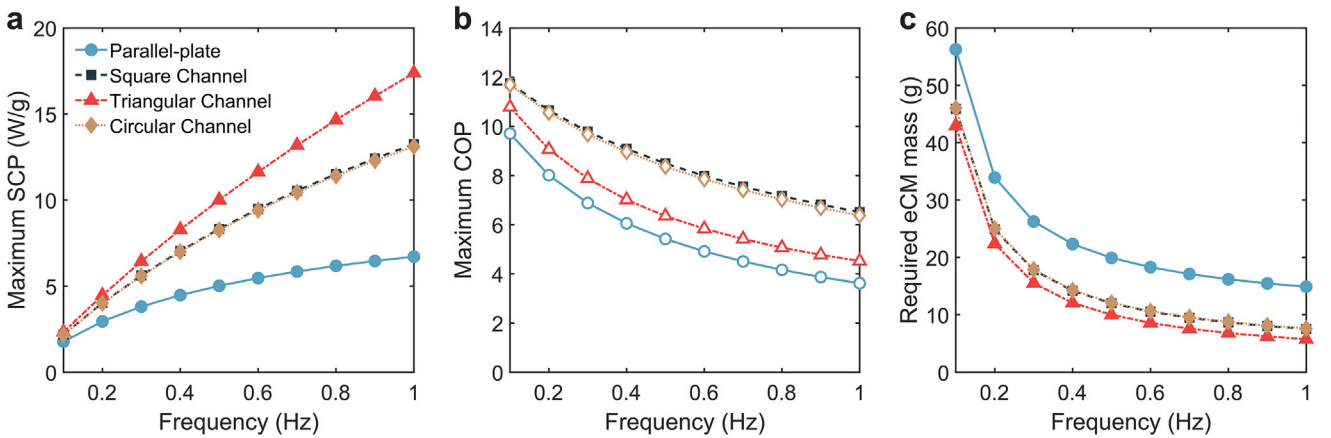


Fig. 9. Geometry-dependent cooling performance of TPU (HM-PTMEG) elastocaloric microchannel regenerators under a temperature span of 8 K. (a) Maximum SCP as a function of operating frequency for different channel geometries. (b) Maximum COP as a function of operating frequency for different channel geometries. (c) Estimated elastocaloric material mass required for each regenerator to deliver a cooling capacity of 100 W.

set to 300 μm . Based on the definitions in Table 2, the parallel-plate spacing and wall thickness were set to $H_1 = H_2 = 150 \mu\text{m}$, while the isosceles triangular channel used $w = h = D_{h,0}(1 + \sqrt{5})/2 = 485.41 \mu\text{m}$. All regenerators had a porosity of 50%, a stretch range of $\lambda = 3-7$, and a temperature span of 8 K. All other material and operating parameters were retained from Table 4. At each frequency, the maximum SCP and maximum COP were selected independently by optimizing V^* using the same procedure as in Supplementary Figures S4–S8.

As shown in Fig. 10, equalizing $D_{h,0}$ substantially narrows the geometry-dependent variation in SCP. At 1 Hz, the parallel-plate geometry gives the highest SCP, reaching 3.9 W g^{-1} for NR and 13.6 W g^{-1} for TPU. The corresponding values for the other geometries are 3.68–3.74 W g^{-1} for NR and 13.11–13.25 W g^{-1} for TPU. The stronger triangular-channel advantage observed in Figs. 7 and 9 therefore arises primarily from the smaller hydraulic diameter and larger specific surface area produced when the same nominal characteristic dimension

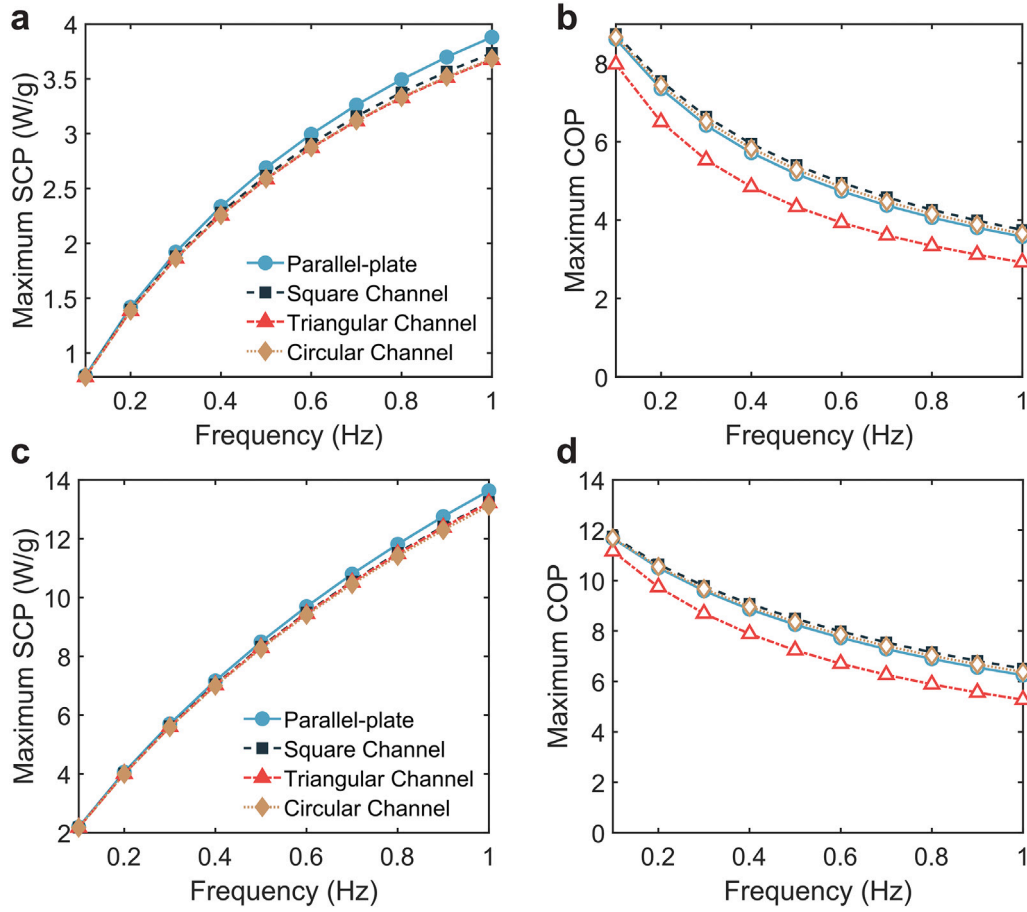


Fig. 10. Predicted cooling performance of the four microchannel geometries at an equal undeformed hydraulic diameter of $D_{h,0} = 300 \mu\text{m}$. (a) Maximum SCP and (b) maximum COP of the NR regenerators. (c) Maximum SCP and (d) maximum COP of the TPU (HM-PTMEG) regenerators. The porosity was fixed at 50%, the stretch range was $\lambda = 3-7$, the temperature span was 8 K, and V^* was independently optimized for SCP and COP at each operating frequency.

is assigned to all geometries. Once the hydraulic diameter is held constant, the effect of channel shape on SCP is comparatively modest.

The COP remains more sensitive to channel shape because the friction-factor and heat-transfer correlations retain their geometry dependence. At 1 Hz, the square channel achieves the highest COP, with values of 3.75 for NR and 6.51 for TPU, followed by the circular channel (3.65 and 6.38) and the parallel plate (3.58 and 6.25). The triangular channel shows the lowest COP (2.92 for NR and 5.28 for TPU), corresponding to reductions of 22.1% and 19.0%, respectively, compared to the square channel. This separation becomes more pronounced with increasing frequency because the pumping-work contribution increases with flow rate. Thus, under an equal- D_h comparison, the parallel-plate geometry provides a small SCP advantage, whereas the square and circular channels provide the more favorable efficiency balance.

4.4. Comparison of NR and TPU elastocaloric regenerators

To assess the application potential of the novel TPU (HM-PTMEG)-based microchannel regenerators for high-performance elastocaloric cooling, Fig. 11 compares NR- and TPU-based regenerators from three perspectives: intrinsic material properties, cooling performance, and economic viability.

Regarding the intrinsic material properties, Fig. 11a compares the experimentally measured adiabatic temperature changes (ΔT_{ad}) of NR and four representative TPU materials as functions of stretch ratio λ [29,75]. Fig. 11b further compares their effective adiabatic temperature changes over the practical stretch ranges of $3 \rightarrow 6$ and $3 \rightarrow 7$, calculated as $\Delta T_{ad}(\lambda_f) - \Delta T_{ad}(\lambda_i)$. The comparison shows substantial differences among the TPU formulations, with HM-PTMEG providing the

largest effective ΔT_{ad} over both stretch intervals. For the regenerator-level simulations in Figs. 11c and d, NR and HM-PTMEG were selected as representative conventional and high-performance polymer refrigerants, respectively. Fig. 11c investigates the influence of stretch range on the maximum SCP using the triangular microchannel regenerator as a representative configuration. Under identical stretch intervals (e.g., $3 \rightarrow 7$), the TPU regenerator exhibits substantially higher SCP than the NR counterpart, reflecting the large ΔT_{ad} advantage shown in Fig. 11b. Furthermore, increasing the stretch range from $3 \rightarrow 6$ to $3 \rightarrow 7$ enhances SCP for both materials. This effect is particularly pronounced for NR, where the SCP increases by a factor of approximately 3.4 (from 1.6 W/g to 5.4 W/g), consistent with the strong dependence of ΔT_{ad} on stretch ratio shown in Fig. 11a.

Fig. 11d summarizes the maximum SCP for both materials across different channel geometries. TPU-based regenerators consistently outperform NR-based counterparts across all geometries, with performance improvements of several-fold. Notably, the parallel-plate regenerator shows the most pronounced enhancement when utilizing TPU, achieving nearly a 4.8-fold increase in SCP (from 1.4 W/g to 6.7 W/g).

From the perspective of commercialization and engineering implementation, active-material demand is a useful but incomplete screening metric. Fig. 11e presents the active-material cost per unit cooling power (\$/W), defined as active-material price (\$/g) divided by model-predicted SCP (W/g), using material prices summarized in Fig. 1. The two red markers represent the optimized NR- and TPU-based microchannel regenerators simulated in this work, achieving values of 5.0×10^{-4} and 4.3×10^{-4} \$/W, respectively. This performance is lower than that of typical thermoelectric (TE) materials (< 3.7

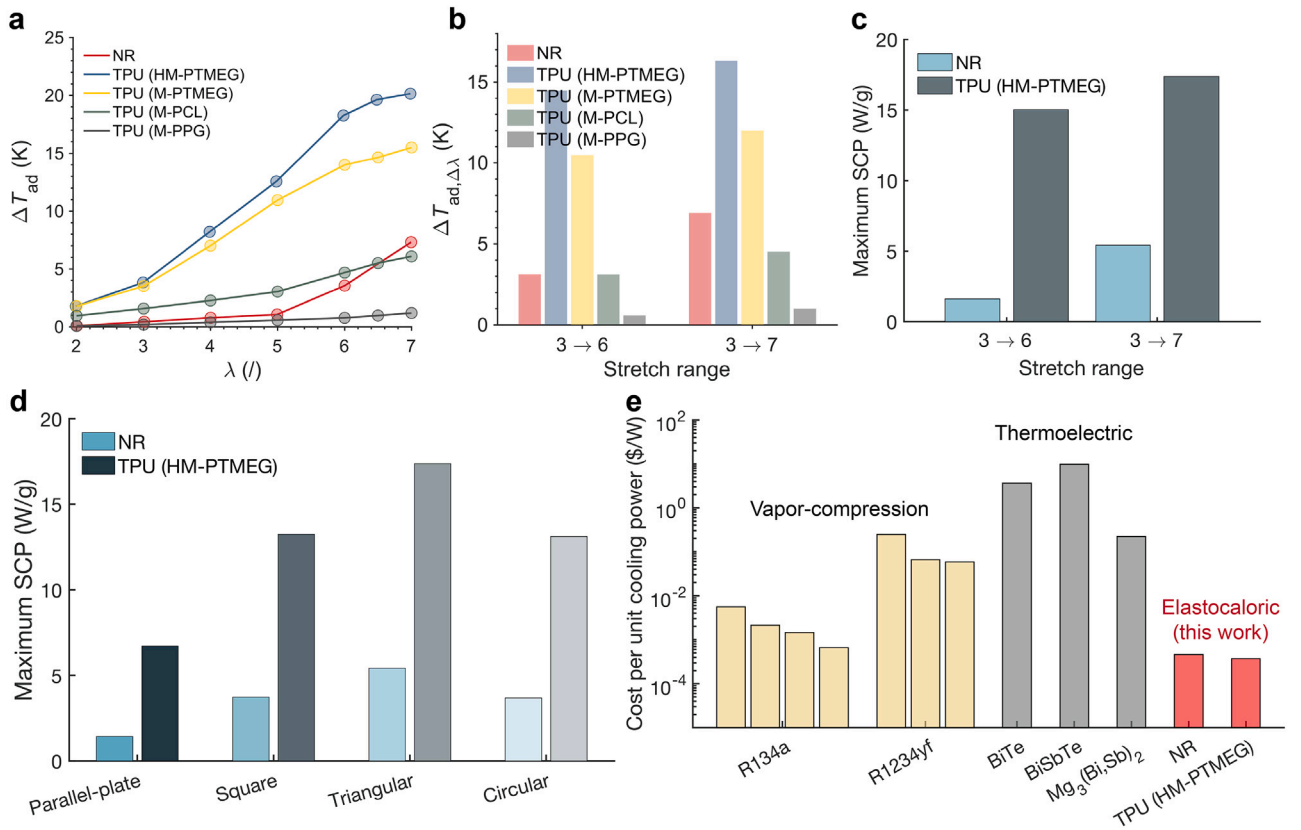


Fig. 11. Comparison of elastocaloric properties, cooling performance, and cost effectiveness of Natural Rubber (NR) and TPU microchannel regenerators. (a) Experimental adiabatic temperature change (ΔT_{ad}) as a function of stretch ratio (λ) for NR and TPU. (b) Effective ΔT_{ad} span ($\Delta T_{ad}(\lambda_f) - \Delta T_{ad}(\lambda_i)$) for two practical stretch ranges (3 → 6 and 3 → 7), where λ_f and λ_i denote the final and initial stretch ratios. (c) Maximum SCP of the triangular-channel regenerator for NR and TPU under different stretch ranges. (d) Maximum SCP of NR and TPU regenerators across different microchannel geometries. (e) Active-material cost per unit cooling power (\$/W) comparison, comparing the NR and TPU regenerators developed in this work with representative conventional cooling technologies reported in the literature [76–79].

\$/W) and conventional vapor-compression refrigerants, such as R134a and R1234yf. These values indicate a potential active-material cost advantage only, where the metric excludes fabrication, microchannel manufacturing, actuation, pumps, valves, sealing, heat exchangers, controls, maintenance, fatigue replacement, and system integration. A second cost-related advantage may arise at the actuator level. Conventional SMA-based elastocaloric systems require high-force actuation because NiTi is loaded at high transformation stresses. Depending on the active cross section and cooling capacity, the required actuator force can reach the kN range and may scale to tens of kN in large-capacity modules [10,15,25], requiring hydraulic, electromagnetic, or other robust high-force actuators. In contrast, polymer elastocaloric regenerators operate at much lower stresses. Although they require much larger displacement, polymer-based regenerators can be driven by compact linear, rotary, cam, or belt-based actuators with force requirements on the order of hundreds of newtons [21,31,32]. Therefore, polymer systems may reduce system cost not only through lower active-material price but also through smaller and simpler actuation hardware, which shows strong potential for cost-effective elastocaloric regenerators.

5. Outlook and future research

5.1. Future research priorities

To further advance polymer elastocaloric cooling systems and refine current numerical models, several research directions should be pursued.

First, improvements in numerical modeling are required to better capture the complex thermo-hydraulic behavior of large-deformation

regenerators. Current one-dimensional models rely on empirical correlations for heat transfer and pressure drop (e.g., Nusselt number and friction factor) that were originally derived for rigid channels. These correlations may deviate from the actual flow and heat transfer behavior in dynamically deforming microchannels. Therefore, future studies should incorporate experimental measurements from real rubber-based regenerators. In-situ measurements of pressure drop and convective heat transfer coefficients during cyclic stretching and fluid flow would enable more accurate calibration of model parameters. In addition, the current model accounts volume changes of the regenerator using a constant Hencky-Poisson's ratio during large deformation and assumed a fixed porosity. Since elastomers could exhibit variable Poisson's ratios and compressibility, channel volumes may change during operation [21,72]. Future models should incorporate these deformation-induced volume changes and dynamic fluid-volume compensation, potentially by integrating geometric information obtained from finite-element simulations [72].

Second, further progress in elastocaloric polymer materials is required. Although TPU significantly improves cooling performance, polymer elastocaloric materials still face intrinsic limitations such as low thermal conductivity and limited long-term fatigue stability. Addressing these challenges may require developing composite elastocaloric polymers containing highly conductive fillers (e.g., liquid metals, graphene, or carbon nanotubes) while simultaneously optimizing polymer network structures to improve mechanical durability and fatigue resistance.

Finally, practical implementation of polymer elastocaloric regenerators requires scalable fabrication of microchannel structures. Producing flexible microchannels with characteristic sizes of several hundred

micrometers remains challenging but feasible. Previous studies have demonstrated that advanced manufacturing techniques such as fused deposition modeling [80], hot embossing [81], and injection molding [82] can produce flexible TPU microchannels with dimensions below 100 μm . Future work should therefore focus on fabricating regenerator prototypes with characteristic pore sizes on the order of several hundred micrometers (e.g., 300–500 μm) and performing cyclic experimental testing. In addition, the rigid-channel correlations are applied quasi-steadily; pressure-induced deformation and fluid–structure interaction are not resolved, and high-pressure cases are idealized numerical operating points. The highly coupled 3D multiphysics model that integrates large-deformation solid mechanics, fluid dynamics, and heat transfer is intended for future channel-volume and stress-concentration calibration, while local stress concentration and fatigue failure mechanisms in rubber-based regenerators under large deformation should be experimentally investigated.

5.2. Potential applications of polymer-based elastocaloric regenerators

At system level, a practical polymer AER unit would require two or more cascaded/parallel regenerators to achieve practical temperature spans and cooling capacities, as well as mechanical drivers, fluidic units, valves and manifolds, cold- and hot-side heat exchangers, and control system. Each regenerator would follow the four-step loading–blow–unloading–blow cycle described in Section 2.1. A first application is local liquid cooling of electronics or compact equipment at moderate temperature lift. The cold-side loop could be coupled to a cold plate, while the hot-side loop rejects heat through a liquid-to-air or liquid-to-liquid heat exchanger. The 100 W design and 8 K temperature span evaluated in this study are representative of the regenerator scale required for such local cooling. At 1 Hz, the model estimates that approximately 5.0 g of triangular-channel TPU is required to provide 100 W. A second application is a modular liquid chiller for small space-cooling. Multiple cascaded regenerators could be connected in parallel through common cold- and hot-side manifolds, allowing enhanced cooling capacity. Alloy-based AERs have already demonstrated a 75 K water-side lift and kilowatt-scale cooling capacity [11,12]. Compared with these NiTi systems, polymer regenerators offer lower active-material price and lower operating stress, but require much larger displacement and must overcome low thermal conductivity, large-deformation sealing, fluid-volume compensation, and long-term fatigue. These trade-offs should be evaluated experimentally before selecting either material class for a given application.

6. Conclusion

In summary, this study develops a transient one-dimensional numerical model with dynamic meshing to analyze large-deformation polymer elastocaloric cooling systems. By introducing a deformation factor, the model captures the evolution of thermo-hydraulic parameters under high stretch ratios and is benchmarked against reported NR-tube-based regenerator data. The results demonstrate that microchannel regenerators significantly outperform conventional parallel-plate designs under a same channel size. Among the investigated geometries, triangular channels achieve the highest SCP due to their larger specific surface area at identical characteristic pore sizes. At an operating frequency of 1 Hz, triangular regenerators fabricated from NR and thermoplastic polyurethane (TPU, HM-PTMEG) are model-predicted to achieve maximum SCP values of 5.4 W/g and 17.4 W/g, respectively. In contrast, square and circular channels exhibit higher COP because of their lower flow resistance, with TPU-based systems maintaining COP values between 7 and 13. Geometric optimization further identifies a porosity of 50% and a characteristic pore size of approximately 400 μm as the optimal configuration that balances heat transfer performance and pumping losses. In addition, introducing the high-performance TPU material substantially enhances cooling capability. For example,

delivering a cooling capacity of 100 W requires only 5.7 g of TPU compared with 18.4 g of NR, enabling lightweight system designs. When eliminating the influence of channel shape on hydraulic diameter and simulating 4 regenerator under an equal D_h , their cooling power is almost similar and the square and circular channels provide the more favorable COP. Finally, a regenerator-level economic analysis shows that the active-material cost per unit cooling power of NR and TPU microchannel regenerators reaches 5.0×10^{-4} and 4.3×10^{-4} \$/W, respectively, indicating strong potential for cost-effective elastocaloric regenerators.

CRedit authorship contribution statement

Kun Wang: Writing – review & editing, Writing – original draft, Validation, Software, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Qing Wang:** Writing – review & editing, Resources, Methodology. **Manfred Kohl:** Writing – review & editing, Supervision, Resources, Methodology. **Rasmus Bjørk:** Writing – review & editing, Supervision, Methodology. **Jingyuan Xu:** Writing – review & editing, Supervision, Resources, Project administration, Methodology, Funding acquisition, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix A. Supplementary data

Supplementary material related to this article can be found online at <https://doi.org/10.1016/j.enconman.2026.122234>.

Data availability

Data will be made available on request.

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