A compact elastocaloric refrigerator

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GRAPHICAL ABSTRACT

PUBLIC SUMMARY

- Nickel-titanium shape-memory wires are green solid-state refrigerants
- A single motor pulls the NiTi wires to reject heat before unloading the wires for cooling
- First fully integrated elastocaloric refrigerator prototype in the world
- Next generation refrigerators demanding zero GWP, low vibration, and high efficiency
Elastocaloric cooling is regarded as one of the most promising cutting-edge alternatives to conventional vapor compression refrigeration systems. This technology is based on the temperature change of materials when being subjected to uniaxial stress, which has been observed in polymers, alloys, and ceramics. However, the existing elastocaloric prototypes have a bottleneck problem of an excessive mass ratio between the actuator and the solid-state refrigerant. To overcome this challenge, this study proposes an elastocaloric refrigerator using a single actuator with an inclined angle to produce a vertical tensile force to nickel-titanium (NiTi) shape-memory wires and a lateral motion to translate the NiTi wires between the hot and cold sides. The refrigerator can achieve a 90% improvement in the mass ratio between the solid-state refrigerant and actuator compared to the currently best-reported elastocaloric cooling prototype. The NiTi wires exhibit an adiabatic temperature change of 6.6 K during unloading at the strain of 4.8%. The proposed refrigerator can achieve a 9.2 K temperature span when the heat source and sink are insulated from ambient and has a cooling power up to 3.1 W under zero-temperature-span condition. By using thinner NiTi wires or NiTi plates, the developed elastocaloric refrigerator could be a starting point to promote applications of this technology in the future.

INTRODUCTION

According to the Intergovernmental Panel on Climate Change’s report from 2018, if climate warming continues progressing at the current rate between 2030 and 2052, the global average temperature will be 1.5°C higher than the value before industrialization.1 Such a significant increase in the global average temperature can cause a series of negative geological effects, including the melting of glaciers, a sea level increase, disturbance to the hydrological cycle, and the acidification of seawater.2 Therefore, it is crucial to reduce emissions of greenhouse gases. Refrigeration and air conditioning equipment contribute to more than 7.8% of the global annual emission of greenhouse gases,3 which is slightly less than the emission of the transportation sector of 12.1%. Currently, about 70% of emissions from refrigeration systems are due to the power required to operate these systems. However, as renewable power sources, such as solar and wind energy, have been continuously growing and are expected to dominate in the near future, the significance of the emissions from operating the refrigeration systems will be reduced. Nevertheless, the remaining 30% of emissions from the refrigeration systems are due to the hydrofluorocarbon (HFC) refrigerants used in the vapor compression cooling equipment, and each HFC molecule is thousands of times more devastating than a CO₂ molecule in terms of global warming potential (GWP). Due to the increasing impacts of high-GWP refrigerants, caloric cooling technologies have attracted substantial attention worldwide in recent years because the solid-state caloric materials cannot vaporize and therefore have zero GWP.4

Elastocaloric cooling exploits the change in temperature or entropy when the material is subjected to uniaxial stress. Shape-memory alloys (SMAs) with the martensitic phase transition have been the most popular elastocaloric cooling materials.5 Although not thermodynamically reversible, the martensitic phase transition in SMAs is “nominally reversible,” which means that the strain can be fully recovered to the initial state after each cycle of application and removal of the stress.6 The adiabatic temperature change of Ni₅₀Mn₃₁.₅Ti₁₈.₅B₀.₂ alloy can be as high as 31.5 K, which is equivalent to the isothermal entropy change of 0.29 J cm⁻³ K⁻¹, which is one magnitude larger than the 0.035 J cm⁻³ K⁻¹ isothermal entropy change of HFC-32 for vapor compression air conditioners. Due to this unique advantage of the colossal cooling power density, elastocaloric cooling has been regarded as the most promising technology among the caloric cooling technologies, i.e., barocaloric cooling,8 magnetocaloric cooling,9 and electrocaloric cooling,10 by the US Department of Energy.11

Since 2010, many elastocaloric cooling prototypes have been reported.12 The single-stage elastocaloric cooling prototypes were developed first, and they include prototypes based on water-cooled nickel-titanium (NiTi) shape-memory tubes13 and NiTi-based foils using solid-solid contact heat exchange design,14–17 which were cyclically operated. A continuously operated elastocaloric cooling device that uses two co-rotating rings with NiTi wires, which could continuously cool the supply air, was reported.18 To improve system temperature span for practical applications, active regeneration19,20 and cascading multiple NiTi plates21 were successfully applied to elastocaloric cooling prototypes. As shown in Table S1, the temperature span and cooling power of these elastocaloric prototypes are sufficient for practical applications, but their compactness measured by the volumetric specific cooling power has been a bottleneck for their commercialization. Namely, dual-loaded motors used for symmetric loading or separate control for loading and heat exchange22 significantly increase the footprint of the motors. In recent years, different approaches have been proposed to address this challenge. For instance, bending can be used instead of tension or compression of NiTi strips, which can reduce the loading force by more than 50%.23 Another approach is to use SMA heat actuators to replace mechanical actuators, but this approach requires using low-grade heat sources such as solar thermal collectors.24

To tackle the challenge of compactness, this study develops a mechanical architecture that increases the mass ratio between the SMA and actuator by more than 90% compared to the state-of-the-art prototypes. The key improvement in the proposed design that leads to such a drastic reduction in the motor mass is an inclined architecture that exploits the simplicity of a direct drive motor, as shown in Figure 1A. In the proposed architecture, the properly sized motor without any transmission is fixed on the steel frame (4) at an inclined angle, which can provide a vertical tensile force to generate the elastocaloric effect and a lateral motion to translate the NiTi wires between the hot and cold sides. It should be noted that linear actuators of the existing prototypes are oversized because they contain built-in transmissions since they are not specifically designed for driving the SMA. The proposed refrigerator’s volume is 4.5 L, which is equivalent to a specific volume of 1.5 L W⁻¹, which is significantly better compared to the other prototypes.

To load and unload the NiTi wires, the output shaft of the motor rotates forward and backward periodically; the output shaft of the motor is connected to a screw shaft (9) through coupling. Mounted on the screw shaft (9), a screw slider (8) transforms rotation into linear displacement, which is connected to the primary clamp (7) located at the bottom of the eighteen NiTi wires, following a similar clamping design as predecessor prototypes, as shown in Figure 1B.22,25 The top side of the NiTi wires is connected to the second clamp (3), which is mounted to a rail (1) and rail slider (2) assembly so that the horizontal movement of the two clamps can be synchronized using the two vertical linear bearings. A hot-side radiator as a heat sink (5) is fixed on one side of the NiTi wires with a cooling fan on top of it to reject the heat to the ambient. Similarly, a cold-side radiator as a heat source (6) is placed on the other side of the NiTi wires to cool down the insulated cabinet. A photo of the prototype refrigerator with an exterior casing is shown in Figure 1C.

The operating principle of the proposed elastocaloric refrigerator is as follows. During loading, the primary clamp (7) provides tensile stress to the NiTi wires in the vertical direction. Meanwhile, the inclined angle of the motor forces the NiTi wires to move toward the hot-side radiator (5), which is followed by the contact heat exchange with the hot-side radiator (5) that releases the latent heat, as shown in Figure 1D. During unloading, the reverse rotation of the motor drives...
the primary clamp (7) to move upward along the screw shaft, while the secondary clamp (3) moves along the sliding rail (2) in the same direction. Driven by the clamps on both sides, the NiTi wires move toward and eventually touch the cold-side radiator (6). As shown in Figure 1E, the NiTi wires are unloaded and can absorb the heat from the cold-side radiator (6) to cool the stored products in the insulated cabinet.

RESULTS

In addition to the novel mechanical design that can reduce the system volume, a successful and competitive elastocaloric cooling system also requires decent performances of NiTi wires. As presented in Figure S1A, the microstructure consists of the matrix phase and secondary phase. Energy-dispersive X-ray spectroscopy analysis has revealed that the chemical compositions of the matrix and secondary phases are Ni$_{55.76}$±$0.28$ (wt.%) and Ni$_{38.74}$±$0.34$ (wt.%), respectively. The Ti:Ni atomic ratio of the secondary phase is approximately 2:1, conforming to the stoichiometric Ti$_2$Ni phase. Combined with the X-ray diffraction analysis presented in Figure S1B, the matrix phase can be determined as a B2-type parent phase, and the Ti$_2$Ni phase cannot be detected from the X-ray diffraction pattern due to its low volumetric fraction. The martensitic transformation (MT) starts at 266 K and finishes at 226 K, while the inverse MT starts at 246 K and finishes at 274 K, which is lower than room temperature, so the NiTi wire can be used as an ambient cooling material, as shown in Figure S1C.

To stabilize the super-elastic behavior of the NiTi wires, each wire was subjected to 200 loading and unloading training cycles at the strain rate of 10$^{-4}$ s$^{-1}$. The trained wires were further tested at different strain rates, as shown in Figure 2A. The stress was held for 60 s to allow the wires to reject heat and reach ambient temperature before unloading to the stress-free state at the same strain rate. The corresponding temperature changes were directly measured by an infrared thermometer. As shown in Figure 2B, the faster strain rate led to a better adiabatic temperature change during loading and unloading could be achieved. In situ measurement of the polycrystalline NiTi wires assembly in the prototype demonstrated a 7.8 K repeatable adiabatic temperature change during loading and unloading at 4.8% strain, as shown in Figure 2E. The average temperature change of the NiTi wires assembly was slightly less because of the inhomogeneous temperature distribution during the loading and unloading processes, as shown in Figure 2F. The inhomogeneous temperature along the NiTi wires was due to the local nucleation and propagation of Lüders band-like martensite fronts during incomplete phase transition at 4.8% strain. This incomplete and inhomogeneous temperature distribution should be considered when evaluating system performance, as presented by the simulation results in Figure S3, which were obtained using the method described in Figure S2.

When the strain rate was maintained at 0.06 s$^{-1}$, the optimal operating frequency of the system was determined by setting different contact heat exchange durations of the NiTi wires with the two heat exchangers, as shown in Figure 3A. After 90 cycles, Figure 3A shows that the temperature span ($\Delta T_{\text{span}}$), defined as the temperature difference between the hot-side and cold-side radiators, first increased and then decreased with the frequency. When the frequency of the system was 0.152 Hz (2.5 s for heat exchange with the hot-side radiator and cold-side radiator and 0.8 s for loading and unloading), the temperature span reached its maximum at 8.6 K. This trend has been in line with the previous studies, namely, the optimum frequency is a result of the competition between faster
cycling and more efficient heat exchange per cycle. The time-dependent changing trends of the temperature span for different frequencies are presented in Figure S4.

The temperature changing trends of the two heat exchangers under the optimal system operating frequency of 0.152 Hz are shown in Figure 3B, where it can be seen that simulated temperature values (see supplemental materials and methods for more details) well match the experimental results. Although the temperature span became visually steady after 90 cycles, as shown in Figure 3A, it continued to increase slightly and reached 9.2 K after 170 cycles under the same strain of 4.8%, as presented in Figure 3B. As listed in Table S1, the 9.2-K temperature span has been superior to most of the single-stage devices. The temperature of the hot-side radiator (heat sink) increased monotonically with time, whereas the temperature of the cold-side radiator (heat source) first decreased and then increased. This was because other than the trivial heat leaks, the system was considered adiabatic, and the input mechanical power gradually dissipated into heat via the hysteresis of the NiTi wires.

Since the goal is to deliver cooling power at a temperature lower than the ambient temperature, the cooling performance of the proposed system was also measured when the hot-side radiator was actively rejecting heat to the ambient, using the identical frequency from the adiabatic tests. Figure 3C shows the temperature changing trends of the two heat exchangers when the hot-side radiator was cooled by the fan while the cold-side radiator was still adiabatic. The temperature of the hot-side heat exchanger remained almost constant at 23.4 °C, while the cold-side radiator’s temperature reached 17.6 °C after the system operated for 1,500 s, corresponding to a 5.8-K temperature span. This temperature span was less than that of 9.2 K in Figure 3B because the heat sink was constantly rejecting heat to the ambient, so the rejected heat could not contribute to the accumulation of the temperature span. Furthermore, rejecting heat would require a heat transfer temperature difference between the NiTi wires and the heat sink, which approached zero for the adiabatic case, as shown in Figure 3B, but was about 2 °C in this set of tests, thus resulting in a reduced temperature span.

When the polymer sheet heater adjacent to the cold-side radiator was turned on, the cold side was no longer adiabatic. Under such conditions, the system temperature span became lower than that in Figure 3C. By setting different input power of the heating film, the relationship between the temperature span and the

Figure 2. Performance of the NiTi wires (A) Stress-strain curves at different strain rates obtained by MTS test machine. (B) Temperature response curves at different strain rates obtained by the MTS test machine. (C) Infrared images of the loading and unloading states of the NiTi wires at the strain rate of 0.06 s⁻¹ obtained by the MTS test machine. (D) Stress-strain curve at 4.8% strain. (E) In situ temperature measurement results of the NiTi wires at the strain of 4.8% in the elastocaloric refrigerator. (F) In situ infrared image of the loading and unloading states of the NiTi wires at the strain of 4.8%.
cooling power could be obtained when the system reached steady operation. Figure 3D shows the system temperature span at the heater’s input powers of zero, 0.8, 1.25, 1.8, 2.45, and 3 W, corresponding to the input voltages of zero, 2, 2.5, 3, 3.5, and 3.9 V, respectively. The demonstrated linear relation between the cooling power and system temperature span is in line with previous literature. By extrapolating the linear fitting, the maximum cooling power of the system could reach 3.1 W when the temperature span was zero, which was superior to the results of the fluid-free elastocaloric cooling prototypes displayed in Table S1. In addition, the maximum temperature span was 5.8 K at the zero cooling power.

DISCUSSION
Compared with the existing elastocaloric cooling devices, the proposed prototype is to our knowledge the first standalone air-cooled elastocaloric...
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power is 3.1 W. For instance, a thermoelectric refrigerator of a similar size re-
elicient of the cooling power at a zero temperature span given byEquation 1, the
Figure 3E.
power, also outperforms the state-of-the-art design by 20%, as shown in
from the best-reported value of 1.33 \times 10^{-3} to 2.56 \times 10^{-3} in this study, as shown in Table S1. Furthermore, the system compact-
ness of the proposed design, measured by the volumetric specific cooling power, also outperforms the state-of-the-art design by 20%, as shown in
Figure 3E.

For the 0.9-L refrigerating compartment in this study, the maximum cooling power is 3.1 W. For instance, a thermoelectric refrigerator of a similar size re-
quires a cooling power of 1.1 W\(^2\) to generate sufficient cold energy for the refriger-
ated cabinet to cool beverages. This indicates that the proposed elastocaloric refrigerator is practical and particularly suitable for wine coolers since only a 10-K
temperature span is needed.

However, the 3.1-W cooling power is achieved at the cost of 9 g NiTi wires, corresponding to 0.34 W g\(^{-1}\) specific cooling power. Such cooling power density is less than that of the water-based prototypes because the liquid has a potentially higher heat exchange coefficient,\(^{19}\) nor is it superior to the thin-plate-based pro-
totypes that rely on the solid-solid contact heat exchange.\(^{15}\) According to the de-
nition of the cooling power at a zero temperature span given by Equation 1, the
relatively insufficient specific cooling power is due to limited operating frequency f. In the current system layout, further increasing frequency beyond 0.152 Hz will
lead to the reduction in the heat exchange time \(\tau\) and thus cooling capacity per cycle \(Q_{\text{cycle}}\), as a result of limited line-contact-based heat transfer area \(A\) between the NiTi wires and the two heat exchangers.

\[
\dot{Q} = Q_{\text{cycle}}f, \quad \text{(Equation 1)}
\]

where \(\dot{Q}\) is the cooling power [W], \(f\) is the frequency [Hz] that is defined by Equation 2; \(Q_{\text{cycle}}\) is the cooling capacity per cycle [J], and it is defined by Equation 3.

\[
f = \frac{1}{2\tau + \tau_{\text{loading}} + \tau_{\text{unloading}}}, \quad \text{(Equation 2)}
\]

where \(\tau\) denotes the contact time between the two heat exchangers and NiTi wires [s]; \(\tau_{\text{loading}}\) is the loading time [s]; \(\tau_{\text{unloading}}\) is the unloading time [s].

\[
Q_{\text{cycle}} = \frac{m_{\text{NiTi}} c_p_{\text{NiTi}} \Delta T_{\text{ad}}}{t_{\text{steady}}} \left(1 - e^{-\frac{t_{\text{unloading}}}{t_{\text{steady}}}}\right), \quad \text{(Equation 3)}
\]

where \(m_{\text{NiTi}}\) is the mass of the NiTi wire [kg]; \(c_p_{\text{NiTi}}\) is the specific heat of the NiTi wire \((J kg^{-1} K^{-1})\); \(h_{\text{NiTi, sink/source}}\) is the contact heat transfer coefficient \([W m^{-2} K^{-1}]\); \(A\) is the contact heat transfer area [m\(^2\)]. \(\Delta T_{\text{ad}}\) is the adiabatic temperature change of the NiTi wire [K].

The aforementioned challenge of limited cooling power density could be mitigated by increasing the heat transfer area \(A\), which will result in constant \(Q_{\text{cycle}}\) at reduced heat transfer time \(\tau\), and thus will facilitate higher cycling frequency \(f\) and cooling power. Increasing the heat transfer area can be achieved by replacing the baseline 0.7-mm NiTi wires with wires of a smaller diameter or NiTi thin plates, which is possible since a NiTi wire with a diameter of less than 0.2 mm\(^2\) and NiTi plate with a thickness of 0.2 mm\(^2\) have been reported. In fact, using the 0.2-mm NiTi wire, 0.1-mm NiTi wire, 0.2-mm NiTi plate, and 0.1-mm NiTi plate can increase the heat transfer area by 3.5, 7.0, 8.8, and 17.5 times, respectively.

As expected, the increment in the heat transfer area directly resulted in a drastic improvement in the operating frequency and specific cooling power, which was consistent with the scaling law reported by Brueuder et al.\(^{30}\) When the total mass of NiTi materials was constant as a result of a fixed driven motor, an increase in the specific cooling power led to an increment in the absolute cooling power. Using the experimentally validated numerical model and the parameters listed in Tables S2 and S3, the performances of the 0.2-mm NiTi wire, 0.1-mm NiTi wire, 0.2-mm NiTi plate, and 0.1-mm NiTi plate were predicted. The results showed that the system temperature span was doubled, from the baseline 5.2 K to 10.4 K, when the heat sink radiator was actively rejecting heat to the ambient, as shown in Figure 4A. The cooling power at zero temperature span could be enhanced by a factor of six, from the baseline 3.1 W to 17.9 W, as shown in Figures 4B and 4C, while the optimum frequency could be increased from 0.152 Hz to 0.385 Hz. Meanwhile, larger cooling power could pull down the temperature of the heat source at a faster rate, from the baseline result of 700 [s] to 370 [s], as shown in Figure S5A. The pulldown rate, defined by Equation 4, can be increased by more than four times, from 0.007 K s\(^{-1}\) to 0.028 K s\(^{-1}\), as shown in Figure S5B.

\[
V = \frac{\Delta T_{\text{span}}}{t_{\text{steady}}}, \quad \text{(Equation 4)}
\]

where \(t_{\text{steady}}\) denotes the time needed for the heat source radiator to reach equi-
librium [s].

Future development of advanced elastocaloric materials with a larger latent heat and better fatigue performance would also benefit the system performance. As pointed out earlier in Figure 2, the adiabatic temperature change of the NiTi wire assembly used in the proposed prototype is limited since only part of the available elastocaloric effect is exploited. If the fatigue could be improved so that the elastocaloric refrigerator could leverage the full potential of the caloric effect, the system temperature span, cooling power, and pull-
down rate could be further improved by 60.4%, 39.5%, and 75.6%, respectively, compared to the incomplete transformation case with a 0.1-mm thin plate, as shown in Figure S6. If cutting-edge materials such as all-d-metal-Heusler alloys

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**Figure 4. Simulated performance of the elastocaloric refrigerator with different shapes and geometries of the SMA**

(A) Effect of the cycling frequency on the temperature span of the system when both heat exchangers are adiabatic. (B) Effect of the cycling frequency on the cooling power when the temperature span is zero. (C) The cooling power versus temperature span at the optimal frequency.

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[Diagram showing simulated performance of the elastocaloric refrigerator with different shapes and geometries of the SMA, including graphs for temperature span and cooling power versus frequency and temperature span.]

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6. \(A\) is the contact heat transfer area [m\(^2\)]. \(\Delta T_{\text{ad}}\) is the adiabatic temperature change of the NiTi wire [K].
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CONCLUSIONS

This paper presents a compact standalone elastocaloric refrigerator based on the inclined single-motor scheme and solid-solid contact design using polycrystalline NiTi shape-memory wires as the refrigerant. In this way, the mass ratio between the NiTi refrigerant and the actuator can be increased to 2.56 × 10⁻³, which demonstrates an improvement of over 90% compared to the best-reported value, leading to 20% improvement in the volumetric specific cooling power. In situ measurements indicate that polycrystalline NiTi wires produce 7.8 K adiabatic temperature change during loading and 6.6 K during unloading at a strain of 4.8%. Under the same strain, the proposed elastocaloric refrigerator system exhibits a maximum 9.2 K temperature span and a maximum 3.1-W cooling power at the optimal system operating frequency of 0.152 Hz. The 3.1-W cooling power is sufficient to refrigerate a 0.9-L compartment in the refrigerator prototype. The cooling power can be further increased to 17.9 W by substituting 0.1-mm NiTi plate for NiTi wires as predicted by simulation. These cooling performances of the proposed refrigerator prototype at a significantly reduced footprint demonstrate the feasibility of applying elastocaloric cooling technology to small home appliances, such as wine coolers, and the compact system architecture proposed in this study could inspire more innovative designs in the future.

MATERIALS AND METHODS

Materials and methods are provided in the supplemental information.

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AUTHOR CONTRIBUTIONS

The authors declare no competing interests.

SUPPLEMENTAL INFORMATION

Supplemental information can be found online at https://doi.org/10.1016/j.xinn.2022.100205.