

Study of the actuation losses in an electrocaloric cooling device

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ABSTRACT – Electrocaloric cooling is a promising technology that could replace existing gas-based refrigerators with more efficient solid state based devices. The key to improve electrocaloric coolers is to increase both the cooling power and efficiency, two goals that mostly conflict each other. Reaching high cooling power can be achieved by increasing the working frequency. The control of the heat exchange between the hot and the cold reservoirs produces viscous dissipation, a major hindrance to power improvement and to efficiency. Here we consider a cooler design that has been recently studied as a promising alternative to regenerator-based devices [1, 2], where a planar electrocaloric film is electrostatically actuated to move between the two reservoirs. Relying on a Poiseuille flow model, we examine the trade-off between the actuation time and the viscous losses, and its impact on the electrocaloric cooling power. Our model predicts that the cooling power can be increased up to 325 mW cm^{-2} .

Keywords – Electrocaloric, Cooling, Electrostatic, Actuator, Viscous losses.

1. INTRODUCTION

Electrocaloric materials (ECM) undergo entropy and temperature changes triggered by an electric field under isothermal and adiabatic conditions, respectively. Among emerging technologies, electrocaloric cooling is one of the most promising candidates to replace the traditional vapor compression refrigeration systems with solid-state working substances [3]. So far, devices using caloric materials as an active component have hardly been able to reach the specific cooling power and the coefficient of performance (COP) of gas-based refrigerators and heat pumps, so that designing a solid-based heat pump with a high throughput is nowadays one of the key technological challenges.

Here, we focus on a cooling device design that is similar to the one presented by Ma and coworkers [1]. In their device, they use an electrocaloric-polymer film switching between the hot and cold end ; a COP up to 13, a specific cooling power of 2.8 W g^{-1} and a working frequency of 0.8 Hz have been reported.

More precisely, in their device, the active film is electrostatically actuated to follow a zipping motion between two planar reservoirs, keeping an S-shape as detailed by Sato et al. [4]. Both faces of the film are covered with a conductive layer, so that when a voltage is applied between the top or bottom electrode and one of its faces, the film progressively zips to the electrode. Thus, the active substance is also the heat exchanger getting rid of the fluids and pumps needed in traditional regenerator designs, a configuration that is well-suited for light and compact actuators. The working cycle consists of four steps illustrated in Fig. 1: (a) the film heats because of the electric field and delivers the heat to the hot side, (b) the film is moved through electrostatic actuation to the cold side, (c) the film cools as the electric field is removed, and absorbs heat from the cold side, (d) the film is actuated back to the hot side.

The overall transferred heat over a cycle from the cold to the hot side Q_c depends on the intrinsic properties of the ECM and on the mass of the film. On the one hand, the cold power results from the product of the transferred heat and the working frequency, which depends on the heat exchange time and the

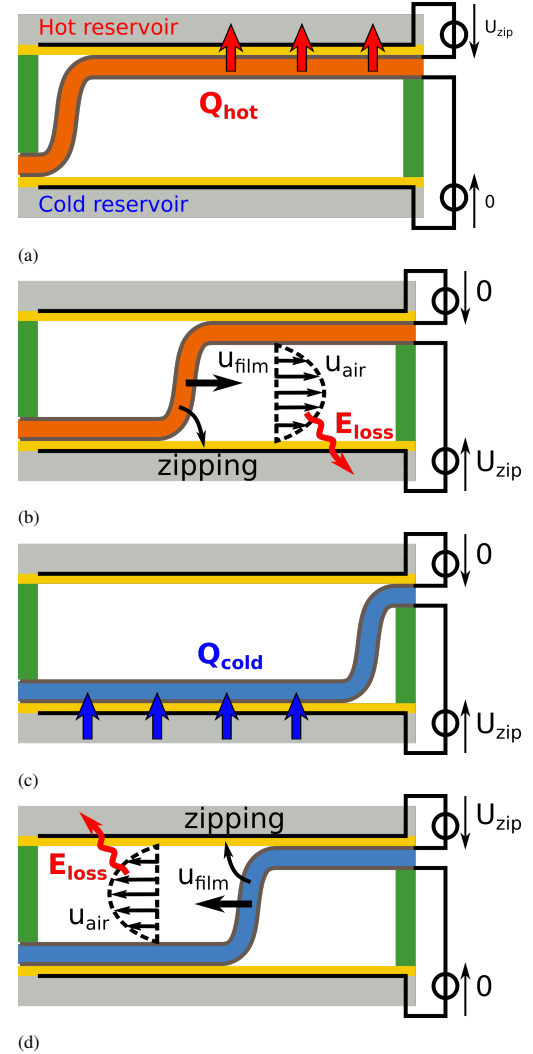


Figure 1. A full cycle of an electrocaloric cooler using electrostatic actuation. The electrocaloric cooler device is based on a thin electrocaloric film (green), with a layer of metallization on both faces (dark gray lines). These are also used to actuate the film towards the electrodes (black lines) on each reservoir (light gray), using electrostatic forces (U_{zip}), the speed of the film being u_{film} . On each end, the film is stuck to one of the two electrodes with an insulation layer in-between (yellow), and the film and the electrodes are separated with a spacer (green) of thickness h . When the film is colored in orange, an electric field is applied to trigger the electrocaloric film (the film is heated and transfers Q_{hot} to the hot reservoir), and when the film is blue, the field is removed (the film is cooled and receives Q_{cold} from the cold reservoir). u_{air} is the velocity field of the air, and E_{loss} is the energy lost through one actuation.

actuation time. On the other hand, the efficiency depends on the input work required to transfer heat and on the energy lost

during a cycle.

Our main goal here is to find the way towards increasing frequency (i.e. power) without giving up on efficiency. The duration of a full cycle corresponds to the two heat exchange stages (i.e. steps (a) and (c) described above) and the two actuations (i.e. the film switching time when moving from one reservoir to the other, steps (b) and (d)); hence the cycle frequency is

$$f = \frac{1}{2(t_{exch} + t_{act})}, \quad (1)$$

where for the sake of simplicity we assume that (b), and (d) stages actuation times t_{act} , as well as (a), and (b) exchange times t_{exch} , are equal two by two.

Using the same actuation principle, we have recently shown that the exchanged heat flux can be significantly enhanced [2]; therefore the exchange time t_{exch} can be reduced down in the 10 ms range. Here, we shall focus on the cost of reducing t_{act} in terms of actuation energy, and we will also examine the optimal balance between exchange and actuation times at a given frequency.

However, while increasing the force of actuation (i.e the voltage U_{zip} in Fig. 1) will allow to move the film faster, it will also generate additional losses, to the detriment of the efficiency. Accordingly, we need to study the energy-loss versus actuation-time trade-off, in order to determine by how much we can increase the frequency without significantly affecting the efficiency.

As our device is MEMS-sized (Micro-Electro-Mechanical-Systems), this makes the energy losses due to the viscous flow of air the predominant dissipative term; we will therefore study this term, neglecting other known loss sources (e.g. film deformation, Joule heating due to electrodes charging, finite time heat exchange, etc.).

We will start dealing with the problem by using the Hagen-Poiseuille flow model, a quite common first-approximation in MEMS literature [5, 6]. Using this model, we will estimate the energy lost due to actuation, and then compare it to heat values reported in the literature [3, 7]. As the viscous losses heat the device, they are detrimental to the cooling effect. Hence, we use a total cold flux that takes into account the negative contribution of losses. This allows us to illustrate how beneficial increasing the frequency is to the cold flux, and how the losses limit the optimal frequency of the device. Finally, we show that reducing the air density in the device would also reduce the losses, and therefore it would allow further increase to the cooling power.

2. FLOW MODELING

Here we shall model viscous flow of air between the surface of two parallel planar electrodes of length L , and width w , with an electrocaloric film of thickness d between them. The middle part of the film is elastically bent into an S-shape, the bent thickness being δs . The distance between the electrode surfaces is $h + d$ as shown in Fig. 2.

As $d, h, \delta s \ll L$ the S-bent can be approximated with a straight vertical interface located in $x = s$ (viz. the dashed magenta line in Fig. 2). This allows to skip all the interface details related to the bent shape focusing on the preponderant volume phenomena associated with the air flow, and considering the S-bent as a moving piston.

When a voltage is applied between one of the two pairs of electrodes, the piston is translated along the x axis with speed depending on the applied electric voltage U_{zip} [4].

When the film is zipping from one electrode to the other, the S-bent pushes the air in front of it over a distance L , and drags the air behind. In the $\delta s \ll L$ approximation where the S-bent behaves like a planar interface the gas moving in front and behind can be described as a plane viscous flow.

The most common way of dealing with such an issue is to use the parallel plane Hagen-Poiseuille flow model [6] assuming the

flow to be stationary, unidirectional (along x axis), isothermal, incompressible and laminar, the fluid to be Newtonian.

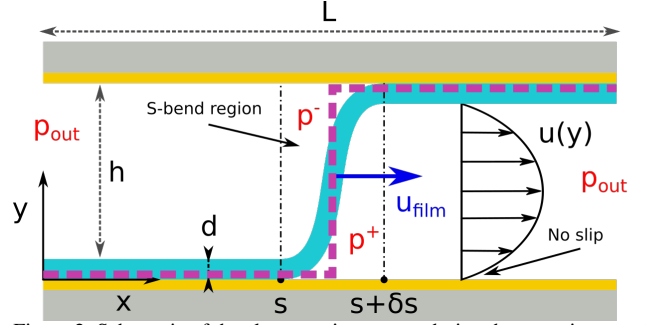


Figure 2. Schematic of the electrostatic actuator during the actuation step (b) in Fig. 1. The polymer film is in cyan, the electrical insulation layer is in yellow, u_{film} is the mean velocity of the film in the x direction. The S-bent region is located between s and $s + \delta s$, and the film is approximated by the dashed magenta line.

The mass conservation under the incompressible assumption gives

$$\frac{\partial u(x, y)}{\partial x} = 0 \quad (2)$$

where u is the axial velocity of the flow, and the force balance equation is

$$\frac{dp(x)}{dx} = \mu \frac{d^2 u(x)}{dy^2}, \quad (3)$$

where p is the pressure, and μ the viscosity of the air. In order to solve this equation, we need boundary conditions for velocity and pressure. The no-slip condition is generally used for channels; in this case $u = 0$ at the walls. By integrating equation (3) on y -axis, where the solution is a parabola for the velocity profile, we find the well-known Hagen-Poiseuille equation:

$$\frac{dp(x)}{dx} = \frac{12\mu}{wh^3} D_v \quad (4)$$

with the flow rate $D_v = wh u_{film}$. For instance, when the film moves forward (viz. in the positive x axis direction), it compresses the air in front (the air is pushed towards $x = L$) and generates a depression behind (the air is drained from $x = 0$). The pressure is p_f^+ in front of the film and p_f^- behind.

Therefore, we can consider two separate flows: the one in front of the film $s \leq x \leq L$ and the one behind $0 \leq x \leq s$, that are connected through the outside pressure p_{out} . By integrating Eq. 4 along x -axis, from s to L and for 0 to s , we have

$$\Delta p_{tot} = p^+ - p^- = \frac{12\mu L}{wh^3} D_v, \quad (5)$$

showing that the flow generated by the movement of the piston along a distance L is equivalent to the flow in a rectangular pipe of equivalent length. Now considering t_{act} as the time needed to the film to sweep the length L , namely $t_{act} = L/u_{film} = (whL)/(D_v)$, from (5) we get the following expression for the actuation time,

$$t_{act} = \frac{12\mu L^2}{h^2 \Delta p_{tot}} \quad (6)$$

The dissipated power can be written as $\Delta p D_v$, whence the viscous energy loss per actuation $\Delta p D_v t_{act}$ writes:

$$E_{visc} = \frac{12\mu w L^3}{h t_{act}}. \quad (7)$$

As the air is accelerated in the channel, the viscous losses is not the only energy that the actuator need to provide. It is worth noting that here we focus on the losses associated with the steady

state flow taking into account nothing but the air kinetic energy, namely:

$$K_{air} = \frac{1}{2}(\rho_{air}whL)u_{film}^2\chi = \frac{3}{5}\frac{\rho_{air}whL^3}{t_{act}^2}, \quad (8)$$

where ρ_{air} is the air density, and $\chi = 6/5$ is the integral of the square of the ratio of the velocity over the mean velocity.

3. LOSSES AND COOLING POWER

One of the the relevant quantities defining the performance of a cooling device is the heat flux through the exchange surface, which is why energies and works are commonly surface normalized. Consequently, from here on we shall switch to surface-energies using the plates area $A = wL$ as the normalization factor.

The surface energy lost through viscosity is therefore

$$e_{visc} = \frac{12\mu L^2}{ht_{act}}. \quad (9)$$

Let us now consider the the worst-case scenario where the inertia of the fluid is entirely dissipated in the air at each actuation. The surface kinetic energy of the air in the channel of a Poiseuille flow is:

$$k_{air} = \frac{3}{5}\frac{\rho_{air}hL^2}{t_{act}^2}, \quad (10)$$

the total energy loss is

$$e_{tot} = e_{visc} + k_{air} = \frac{12\mu L^2}{ht_{act}} + \frac{3}{5}\frac{\rho h L^2}{t_{act}^2} \quad (11)$$

Now we shall find the optimal thickness h_{opt} by minimizing $e_{tot}(h)$,

$$h_{opt} = \sqrt{\frac{20\mu t_{act}}{\rho_{air}}} \quad (12)$$

which corresponds to the thickness value where the kinetic energy equals the viscous loss. The minimized total energy is then

$$e_{tot,min} = \frac{12}{\sqrt{5}}\sqrt{\mu\rho_{air}}\frac{L^2}{t_{act}^{3/2}}. \quad (13)$$

3.1. Electrocaloric effect and cold flux

The two figures of merit of ECM are the adiabatic temperature change associated to the electric field change ΔT_{adia} , and the heat $T\Delta S$ that the material can exchange when the field is applied or removed along an isothermal transformation. Polyvinylidene fluoride polymers films are promising electrocaloric materials for which an adiabatic temperature change of 10 K and an isothermal heat exchange of 20 kJ kg⁻¹ have been reported [3]. Nevertheless, to give a reasonable estimate of the performance of an ECM mounted into a device under actual working condition, we shall use $\Delta T_{adia} = 1.1$ K and $T\Delta S = 1.2$ kJ kg⁻¹ as measured with a calorimetric setup by Basso and coworkers[7]. The electrocaloric material exchanges a fraction of $T\Delta S$ that depends on the thermodynamic working cycle. Here we assume that the exchanged heat is

$$Q_c = \rho_{ec}\frac{T\Delta S}{2}dwL, \quad (14)$$

where ρ_{ec} is the ECM density (1800 kg m⁻³), and dwL the volume of the film. The surface heat received from the cold source at each cycle is then

$$q_c = \rho_{ec}\frac{T\Delta S}{2}d. \quad (15)$$

Increasing the polymer film thickness is a possible strategy to improve the heat exchanged per cycle. However, even when assuming a perfect thermal interface, the heat flux will be limited by the thermal diffusivity of the film, α . Here we describe the role of ECM diffusivity introducing the phenomenological thermal time constant τ ,

$$\tau = \frac{d^2}{\alpha}, \quad (16)$$

which we will use to approximate t_{exch} . Neglecting the actuation losses and working with a vanishing actuation time we get the following cooling flux expression

$$\dot{q}_{ideal} = \frac{q_c}{2t_{exch}} = \frac{\rho_{ec}T\Delta S.\alpha}{4d} \quad (17)$$

where reducing the film thickness appears as a straightforward way to overcome the slowness of heat diffusion (decreasing t_{exch}), and thus to achieve a relevant power increase.

However, in an actual device, the increase of actuation losses associated with the reduction actuation time can hardly be neglected. On the contrary, as shown in Fig. 3 where the actuation losses are compared with the cold flux for different film thickness, they can dramatically increase negatively affecting the overall performance of the device. Actually, for actuation times below 1 ms, the actuation losses exceed the cold flux of the electrocaloric effect. Above 1 ms of actuation time, the cold flux is predominant, and for higher actuation time the viscous loss eventually becomes negligible with respect to the cold flux. Moreover, we see in Fig. 3 that the cold flux is modulated by the film thickness which is related to the exchange time.

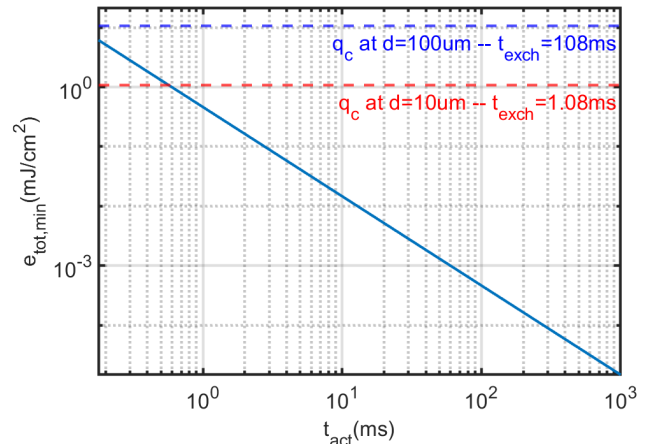


Figure 3. Losses due to actuation (blue line) depending on the actuation time compared to cold flux q_c due to the electrocaloric effect for different film thickness/exchange time.

3.2. Cooling power of the device

A simplified way to assess the detrimental impact of the actuation losses on the cooling device performance, is to estimate the overall power using a total exchanged heat, which is the difference between q_c and two times (two actuations per cycle) the aforementioned losses, namely

$$q_{tot} = q_c - 2e_{tot,min}, \quad (18)$$

whence using the cycle frequency we get,

$$\dot{q}_{tot} = \frac{\rho_{ec}\sqrt{\alpha}\frac{T\Delta S}{2}\sqrt{t_{exch}} - \frac{24}{\sqrt{5}}\sqrt{\mu\rho_{air}}\frac{L^2}{t_{act}^{3/2}}}{2(t_{exch} + t_{act})} \quad (19)$$

This new estimation of the cooling power takes into account both the reduction of heat exchanged with the cold reservoir (due to actuation losses) and the increase of the time period (due to the time of actuation) in a single criterion. Figures 4 and 5 show a color map of $\dot{q}_{tot}$, in which the parameters are t_{act} (related to the actuation losses) and t_{exch} (related to the cold flux). In both figures the dashed lines represent iso-frequency curves where $t_{exch} + t_{act} = f/2$. Only positive $\dot{q}_{tot}$ values are plotted while the white region corresponds to values of the (t_{act}, t_{exch}) plane where $\dot{q}_{tot} < 0$. The used parameters are given in Table 1. These graphs show an interesting trend: for $t_{act} < t_{exch}$ we see that increasing the frequency will also increase the power up to a maximum value. Once the maximum is reached, a further increase of the frequency produces significant losses up to the point where $\dot{q}_{tot} < 0$ (viz. white region).

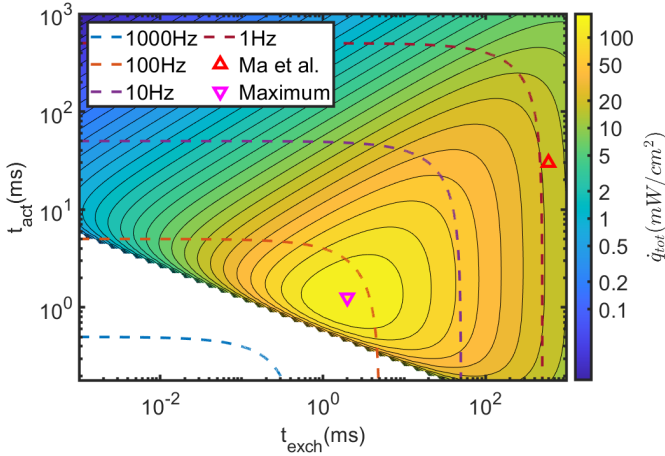


Figure 4. Color map of the cold power per unit of surface function of t_{exch} and t_{act} using a log scale. The losses were calculated using table 1. The white zone is when the total flux is negative, and thus it is 0 at the border.

In Fig. 4, the maximum of $\dot{q}_{tot}$, 183 mW cm^{-2} , is reached for an exchange time, t_{exch} , of 2 ms, for actuation time, t_{act} , of 1.3 ms. The resulting frequency is 154 Hz. The heat exchanged is 1.47 mJ cm^{-2} while the actuation losses are equal to 0.14 mJ cm^{-2} . At maximum power, the actuation time represents 40% of the time period, while the losses represents 19% of the heat transferred. Thus, to evaluate the performance of an electrocaloric cooling device, it is mandatory to study the actuation both in term of actuation time and of actuation losses.

Moreover, Fig. 5 shows that the reduction of air density by a factor 100 allows to increase the maximum to 325 mW cm^{-2} at a frequency of 486 Hz. In this case t_{exch} and t_{act} are respectively 0.63 ms and 0.40 ms, whereas q_c and $e_{tot,min}$ are respectively 0.83 mJ cm^{-2} and 0.079 mJ cm^{-2} . When comparing the two graphs, we see that the reduction of air density by a factor 100 shifted the values by more than two decades to the right for t_{exch} and less than one decade to the bottom for t_{act} , while the values of the effective flux were nearly doubled near the maximum.

In spite of the actuation losses, increasing the frequency proves to be effective to increase the cooling power. This optimum could be further improved either by enhancing the electrocaloric material, i.e the $T\Delta S$ and the diffusivity α , or by improving the actuation time versus losses trade-off.

To improve the actuation losses, two parameters can be tweaked : L^2 and $\rho^{1/2}$. While reducing the length L is very effective to improve the time versus energy loss trade-off, there is a limit at which we start loosing in effective area for the heat

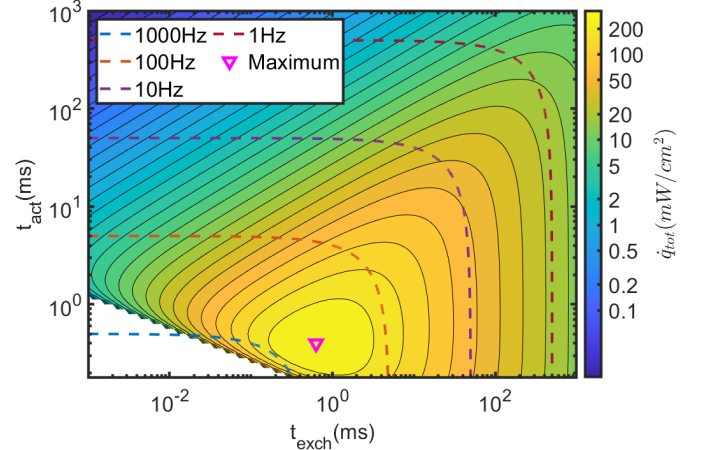


Figure 5. Colormap of the cold power per unit of surface for $p = 10 \text{ mbar}$

exchange because of the S-bent (i.e when $L \sim \delta s$). Whereas lowering the air density will only improve the trade-off as shown in Eq. 13; the better the vacuum, the lower the energy loss and the lower the actuation time.

Based on our model, the estimated heat flux in the configuration of Ma et al. is in the 10 mW cm^{-2} to 20 mW cm^{-2} , whereas they report 30 mW cm^{-2} (at 0.8 Hz). Our maximum heat flux is well above what has been reported by Ma et al. (183 mW cm^{-2} at 154 Hz), showing that optimizing the device geometry might significantly improve its performance. Furthermore, we show that reducing ρ_{air} is quite effective to reduce the losses of the actuator, and is thus a way to enhance the cooling power.

Parameter	Value/Unit	Description
L	5 cm	Length of the channel
ρ_{air}	1.2 kg m^{-3}	Air density at 1 bar
μ	$1.8 \times 10^{-5} \text{ Pa s}$	Viscosity of air
α	$9.3 \times 10^{-8} \text{ m}^2 \text{ s}^{-1}$	Diffusivity of PVDF

Table 1. Parameters used for the graphs. The geometric parameter L is taken from Ma et al. [1].

4. DISCUSSION

Although the film has a cyclic motion, we assumed that the flow was stationary, and we neglected the transient states where the air has to be accelerated or decelerated. From an energetic angle, this is equivalent to the following statement: $e_{visc} \gg k_{air}$. This approximation is valid for creeping flows where the Reynolds number is low and if the aspect ratio h/L is also low. However, we earlier sought the minimum of energy where the kinetic energy is balanced with the viscous losses; in this case the time derivative should be considered in the force balance equation.

Moreover, the flow was deemed incompressible. As mentioned by Gad-el-Hak [5], the general condition of incompressible flow is when the velocity of the flow is way less than the speed of sound. Since we want to actuate films of length of 5 cm in 1 ms, the velocity is 50 m s^{-1} which is sufficiently lower.

Furthermore, we considered that the kinetic energy is lost at each actuation. The actuation could be severely improved if we could store the inertia of the air during the exchange time and restore it for the next actuation, the same way we could recover the electrical charges from one side to the other when zipping.

Finally, to study the impact of the losses on the cooling power, we chose to model the losses as heat that is exchanged with the reservoirs. Without an experiment, it is however difficult to know how much goes to the cold or hot reservoirs. Thus, we

chose to provide an estimation in the worst case scenario, that is when the viscous dissipation ends up heating only the cold reservoir, limiting the cooling power of the device.

5. CONCLUSION

Increasing the frequency is one of the major goals to enhance electrocaloric coolers so that they can compete with typical gas-based refrigerators. The model describes the major limiting factor towards higher frequency: the trade-off between actuation losses and actuation time. A simple Poiseuille model was used to show which parameters can be used to minimize the energy while reducing the actuation time. With this model, the distance between the hot and cold plate h can be optimized to get the minimum amount of losses. Doing so, we found two key parameters to improve the trade-off: the length of the film L , and the density of the air ρ_{air} . While reducing the length is very effective because the losses diminish to L^2 , it is limited by the fact that the S-bent reduces the effective area for the heat exchange.

What is peculiar about this work lies in the combination of both the actuation and the heat exchange between the ECM and the reservoirs in a single figure of merit, which could be related to the cooling power depending on where the work of actuation will be dissipated.

Ultimately, lowering the pressure ρ_{air} can reduce drastically the losses by reducing the kinetic energy of air that is lost if nothing is done to store it. This would enable to actuate within 1 ms at minimal losses, and thus to increase the frequency and the heat flux of electrocaloric coolers up to 486 Hz and 325 mW cm⁻² respectively.

6. ACKNOWLEDGMENTS

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