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Variable load control strategy for room-temperature magnetocaloric cooling applications

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ABSTRACT

Room temperature magnetocaloric cooling is more environmental friendly but not yet more energy efficient than the state-of-the-art vapor compression technology. In this paper, we provide one more argument to support magnetic cooling technology, which is the superior energy saving potential under part load conditions. Therefore, magnetic cooling system may not compete with vapor compression under full load nominal condition, but its seasonal or annual overall efficiency could be better when part load characteristics are taken into account. To show the operation feasibility under part load condition, a feedback control strategy is proposed and incorporated into a magnetic cooling system model in Simulink first. The robust control quality is then revealed by numerical simulation studies for five different variable part load profiles. Furthermore, the transient accumulated energy performances are compared with those estimated based on the quasi-steady state condition to simplify the calculation on the overall energy efficiency benefit. Finally, a case study is carried out for unitary air-conditioning application, revealing that the overall energy efficiency is almost twice of the energy efficiency evaluated under full load condition.

1. Introduction

Room temperature magnetocaloric cooling technology is a fast developing field in the past 20 years [1–4]. It is based on the magnetocaloric cooling effect (MCE) that has been found in various magnetocaloric materials (MCMs), including the first-order materials such as LaFeSiH [5], GdSiGe [6], MnFePAs [7] and their derivatives, as well as the second-order materials such as Gd [8]. The modern magnetic cooling starts with the classic and most-widely applied gadolinium as refrigerant and using super-conducting magnetic to validate the applicability of this technology in 1976 [9]. Since then, many prototypes have been developed based on various driving and heat exchange designs [10–15]. In 2014, Astronautics demonstrated a rotary system with 12 beds of layered LaFeSiH materials, and achieved more than 3 kW cooling under zero temperature span condition, corresponding to 2 J/g specific cooling power [16]. Prototypes from TU Denmark used layered Gd and GdY materials as refrigerant, which performed a coefficient of performance (COP) of 3.6 under 15.5 K temperature span condition, corresponding to a maximum second-law efficiency of 18% [17,18]. Recently, attractive new ideas are proposed and under investigations, including 3D printing for future MCMs with micro-channel or other structures [19], fully solid-state heat exchange mechanism [20], and thermal diodes design [21], revealing bright prospects of this technology in the future. Following the successful stories from academia, industrial partners including Astronautics, CoolTech, Haier and Camfridge are endeavors to commercialize this technology for residential, commercial, medical and industrial applications.

The modern magnetic cooling systems are based on the concept of active magnetocaloric regenerator (AMR) [22]. The principle of AMR requires cyclic reciprocating flow of a heat transfer fluid (HTF) inside a bed filled with MCMs to form a temperature difference, which is much larger than the adiabatic temperature change of the MCMs. Fig. 1 (a) shows a magnetic cooling system with a pair of two AMRs, in which the two AMRs experience 180° phase difference. Usually there are multiple pairs of AMRs in one magnetic cooling system to best utilize the magnetic assembly. There are two heat exchangers (HEXs) in the system, one with low temperature T_c and the other with high temperature T_h. The AMRs and two HEXs are connected by uni-directional pipes in the configuration to minimize dead volume effect [16]. The valves can be integrated into one

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Nomenclature

AMR	active magnetocaloric regenerator
Bi	Biot number [−]
c	specific heat [J·kg^{-1}·K^{-1}]
COP	coefficient of performance [−]
d	diameter [m]
EER	energy efficiency ratio [Btu·W^{-1}·h^{-1}]
fr	cycling frequency [s^{-1}]
f	friction factor [−]
HEX	heat exchanger
HTF	heat transfer fluid
h	convective heat transfer coefficient [W·m^{-2}·K^{-1}]
j	bin number [−]
k	thermal conductivity [W·m^{-1}·K^{-1}]
MCE	magnetocaloric effect term [W·m^{-3}]
MCM	magnetocaloric cooling materials
$\dot{m}$	mass flow rate [kg·m^{-3}]
Nu	Nusselt number [−]
N	number of hours within each temperature bin [−]
L	length of AMR [m]
p	fluid pressure [Pa]
Pe	Peclet number [−]
Pr	Prandtl number [−]
Q_c	accumulated cooling capacity [J]
$\dot{Q}_c$	cooling power [W]
Re	Reynolds number [−]
SEER	seasonal energy efficiency ratio [Btu·W^{-1}·h^{-1}]
SS	(cyclic) steady state
s	total specific magnetic entropy [J·kg^{-1}·K^{-1}]
T_c	heat source temperature (low temperature HEX) [K]
T_f	heat transfer fluid temperature [K]
T_h	heat sink temperature (high temperature HEX) [K]
ΔT_{lift}	temperature span of the system [K]
ΔT_{ad}	material adiabatic temperature span [K]
t	time [s]
t_f	fluid flowing time in each half cycle [s]
t_{mag}	magnetization time in each half cycle [s]
V^*	utilization factor [−]
u	heat transfer fluid velocity [m·s^{-1}]
W	accumulated power consumption [J]
$\dot{W}$	power consumption rate [W]
x	spatial coordinate [m]
β	specific heat transfer area [m^2·m^{-3}]
ε	porosity [−]
ρ	density [kg·m^{-3}]
$\mu_0 H$	magnetic field intensity [T]

Subscripts

amb	ambient
eff	effective
f	heat transfer fluid
in	inlet
M	magnetocaloric materials

rotary valve or multiple pieces, which serve to distribute the HTF into each AMRs based on the cycle specification. For example, Brayton cycle requires the valve to control the HTF flow in each AMR according to the magnetic field as illustrated in Fig. 1 (b). Note that the plotted magnetic field characteristic is for illustration purpose, and realistic profiles may vary based on specific designs.

In a system operated under Brayton cycle, when the magnetic assembly approaches AMR#1, the field intensity increases before it reaches a plateau, a.k.a. magnetization process. Meanwhile, the AMR#2 experiences demagnetization process in which the field intensity gradually reduces to minimum. There is no flow in either AMR during this process so that the temperature of MCM in AMR#1 increases adiabatically and vice versa for AMR#2. The (de)magnetization is followed by heat exchange, in which the HTF flows from cold side of AMR#1 to hot side, absorbing heat and generating or maintaining the temperature gradient in the AMR. The HTF from hot side of AMR#1 then flows through pipes, valves and pump to reject the absorbed heat in the T_h HEX, corresponding to negative flow rate as plotted in Fig. 1 (b). The AMR#2 undergoes reverse flow from hot side to cold side to cool the load in T_c HEX. The HTF flow terminates at the end of the field plateau, and then the two AMRs switch roles, proceeding to the second half cycle.

One of the main driver of this technology is being environmental friendly without utilization of conventional refrigerants with high global warming potential. However, since direct emission impact of refrigerants is much less than that of the indirect emission, i.e. emission due to energy consumption [23], the aforementioned environmental argument now relies on energy efficiency. Unfortunately, the highest reported experimental second-law efficiency is 18% for magnetic cooling, and theoretical projections to include pump, fan power consumption and other losses yield second-law efficiency of about 30% [24,25]. These values are still less than those of vapor compression systems, which is 40%–50% for air-conditioning applications [26].

One potential way to compensate is to account for part load efficiency. The above numbers are all evaluated under the nominal conditions, i.e. full load conditions, however, most of the air-conditioners and refrigerators are operated under part load for most of the time. In principle, magnetic cooling systems show significant COP enhancement when cooling load reduces [4,27], since the operating frequency and flow rate drops accordingly, leading to less heat transfer irreversibility and less pump power consumption. On the other hand, the efficiency of variable speed compressor deviates from nominal maximum value in vapor compression systems when the cooling load reduces. This is partially due to the intrinsic efficiency characteristics of motor with variable speed drive, which is more than 10% when motor speed reduces from full load to 50% load [28]. Most importantly, inverter efficiency degrades significantly when compressor frequency is away from basic frequency, which could lead to more than 30% efficiency drop as indicated by both simulation and experiments [29,30]. In addition, the volumetric efficiency and isentropic efficiency of compressor deviates from the nominal maximum efficiency when the pressure ratio or temperature lift shifts from nominal design conditions [31]. Therefore, when considering the overall energy efficiency, magnetic cooling technology may become more competitive than the state-of-the-art vapor compression technology, though the latter performs better under nominal condition. This is the motivation to conduct this study.

We set unitary air-conditioners as a target application to demonstrate the energy efficiency benefit under part load operation, since the seasonal cooling load variation could spread over one magnitude. It should be noted that the values reported for cooling capacity in this study do not represent real numbers in such air-conditioner products and are for the sake of case studies only. The cooling capacity for a magnetic cooling system can be scaled by filling more MCMs.

To fulfill the aforementioned goals, the paper is organized as