Performance investigation of a high-field active magnetic regenerator

Reed Teyber
Lawrence Berkeley National Laboratory

Jamelyn Holladay
Pacific Northwest National Laboratory

Kerry Meinhardt
Pacific Northwest National Laboratory

Evgueni Polikarpov
Pacific Northwest National Laboratory

Edwin Thomsen
Pacific Northwest National Laboratory

See next page for additional authors

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Abstract
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Keywords
Active magnetic regenerator, Magnetocaloric effect, Superconducting magnet, Liquefaction, Optimization

Disciplines
Materials Science and Engineering

Authors
Reed Teyber, Jamelyn Holladay, Kerry Meinhardt, Evgueni Polikarpov, Edwin Thomsen, Jun Cui, Andrew Rowe, and John Barclay

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Performance investigation of a high-field active magnetic regenerator

Reed Teyber\textsuperscript{a,b,e,*}, Jamelyn Holladay\textsuperscript{b}, Kerry Meinhardt\textsuperscript{b}, Evgueni Polikarpov\textsuperscript{b}, Edwin Thomsen\textsuperscript{b}, Jun Cui\textsuperscript{d}, Andrew Rowe\textsuperscript{e}, John Barclay\textsuperscript{c}

\textsuperscript{a}Lawrence Berkeley National Laboratory, Berkeley, CA 94720
\textsuperscript{b}Pacific Northwest National Laboratory, Richland, WA 99354
\textsuperscript{c}Emerald Energy NW LLC, Bothell, WA 98012
\textsuperscript{d}Ames National Laboratory and Iowa State University, Ames, IA 50010
\textsuperscript{e}Institute for Integrated Energy Systems, University of Victoria, Victoria, B.C. V8W 2Y2

Abstract

Regenerative magnetic cycles are of interest for small-scale, high-efficiency cryogen liquefiers; however, commercially relevant performance has yet to be demonstrated. To develop improved engineering prototypes, an efficient modeling tool is required to screen the multi-parameter design space. In this work, we describe an active magnetic regenerative refrigerator prototype using a high-field superconducting magnet that produces a 100 K temperature span. Using the experimental data, a semi-analytic AMR element model is validated and enhanced system performance is simulated using liquid propane as a heat transfer fluid. In addition, the regenerator composition and fluid flow are simultaneously optimized using a differential evolution algorithm. Simulation results indicate that a natural gas liquefier with a 160 K temperature span and a second-law efficiency exceeding 20\% is achievable.

Keywords: Active magnetic regenerator, magnetocaloric effect,

*Corresponding author. E-mail: rteyber@lbl.gov

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Nomenclature

**Roman**

- $A$ area \([m^2]\)
- $B$ magnetic flux density \([T]\)
- $c$ specific heat \([J \text{ kg}^{-1} \text{ K}^{-1}]\)
- $D$ diameter \([\text{mm}]\)
- $F_e$ combined emissivity \([-]\)
- $f$ operating frequency \([\text{Hz}]\)
- $\dot{h}$ average convection coefficient \([W \text{ m}^{-2} \text{ K}^{-1}]\)
- $H$ magnetic field strength \([A \text{ m}^{-1}]\)
- $k$ thermal conductivity \([W \text{ m}^{-1} \text{ K}^{-1}]\)
- $L$ length \([\text{mm}]\)
- $m$ mass \([\text{kg}]\)
- $n$ number of regenerators \([-]\)
- $P$ system charge pressure \([\text{MPa}]\)
- $\dot{Q}$ heat transfer \([W]\)
- $R$ thermal mass ratio \([-]\)
- $T$ temperature \([\text{K}]\)
- $V_D$ displaced volume \([\text{cm}^3]\)
- $\dot{W}$ work \([W]\)
- $z$ centerline axis of solenoid \([\text{mm}]\)
**Greek**

κ  effective thermal conductivity [-]

η\text{II}  second law efficiency [-]

Φ  utilization [-]

ρ  density [kg m\(^{-3}\)]

μ  viscosity [\(\mu\text{Pa-s}\)] or magnetic permeability [T m A\(^{-1}\)]

ζ  reduced magnetocaloric effect [-]
Subscripts and Superscripts

- ad, adiabatic
- AMR, Active Magnetic Regenerator
- app, applied
- C, cold reservoir or cold side
- CHEX, cold side heat exchanger
- Curie, Curie or transition temperature
- csg, regenerator casing
- f, fluid
- H, hot reservoir or high field
- I, interface
- i, layer number
- int, internal field
- L, low field
- mag, magnetic
- net, net cooling power
- o, free space
- parasitic, parasitic heat leak
- p, constant pressure
- pump, pump work
- reg, regenerator
- s, solid
- span, temperature span
1. Introduction

Distributed-scale liquefaction technologies play an important role in energy storage [1], stranded natural gas recovery [2] and hydrogen fuel cells [3]. Efficient magnetic liquefaction technologies have been of interest since the active magnetic regenerator concept was patented in 1982 [4]. In an active magnetic regenerator (AMR), one or more ferromagnetic refrigerants with sequentially lower Curie temperatures are layered to create compact, porous, high-performance regenerators. The AMR is periodically magnetized and demagnetized so as to execute a cascade of Brayton refrigeration cycles.

Decades of research and development have confirmed the potential for regenerative cycles utilizing active solid magnetic refrigerants. While rejecting heat to liquid nitrogen, Zimm et al. (1996) [5] measured a 35 K temperature span using a 7 T superconducting solenoid and 4 kg of GdNi$_2$ magnetocaloric material. With 4.4 kg of Gd and a 5 T field strength, Zimm et al. (1998) [6] later presented a 33% second law efficiency while cooling 600 W at a 10 K temperature span using water as a heat transfer fluid.

Rowe (2002) [7] developed a reciprocating Active Magnetic Regenerative Refrigerator (AMRR) device with a superconducting magnetic field generator and helium as a heat transfer fluid. The device performance was found to increase with charge pressure [8], suggesting a need for higher density heat transfer fluids. With a 5 T field strength and 0.27 kg of magnetocaloric material (Gd-Gd$_{0.85}$Er$_{0.15}$-Tb), an 85 K temperature span was measured from room temperature; the highest reported with an AMRR device [9].

Numazawa et al. (2014) [10] investigated an Active Magnetic Regenerative Liquefier (AMRL) with a daily liquid hydrogen production of 10 kg.
The authors simulated liquid propane as a heat transfer fluid in the stages between the freezing temperatures of propane and glycol (100-235 K). In an alternative configuration, a figure of merit (FOM) of 0.47 was simulated while rejecting heat to LN2. For the final stage of the liquefier prototype, the authors built and demonstrated a thermo-siphon Carnot Magnetic Refrigerator (CMR) that condensed hydrogen on plates of magnetocaloric material.

Kim et al. (2013) [11] measured a 56 K temperature span while rejecting heat to LN2 in a AMRL apparatus for hydrogen liquefaction. The device used a 4 T superconducting magnet and 0.08 kg of magnetocaloric material spread across two stages to allow different helium flow rates in the warm and cold stages. An optimized layering composition was proposed [12], and more recently, the device was retrofitted with a GdBCO high temperature superconducting solenoid [13]. While the charge-discharge magnet enables a stationary system, AC winding losses limit the device operating frequency [14].

These works describe continued progress toward efficient liquefaction, however ongoing experimental and numerical efforts are required to increase performance and decrease cost. In this work, we describe the development and test results of a large-scale AMRR device that produces the largest temperature span reported in literature. After quantifying loss mechanisms, an AMR model is validated and system performance maps are presented for a number of heat transfer fluids. Finally, a four-material regenerator composition is optimized using a differential evolution algorithm to identify the potential of the bespoke apparatus to liquefy natural gas for transportation and stranded gas recovery applications.
The manuscript is organized as follows. The AMRR apparatus is introduced before describing the regenerator configuration and superconducting magnet system. The demagnetizing fields and parasitic thermal losses are then quantified before introducing the AMR model with a brief description of the thermophysical and magnetocaloric properties. The experimental results are presented with a validation of the model, followed by simulated performance maps and an optimization of the regenerator composition. The implications are then discussed with recommendations for future work.

2. Methods

2.1. AMRR apparatus

The AMRR prototype is shown schematically in Fig. 1. Dual-reciprocating regenerators are displaced inside a stationary superconducting solenoid. The configuration allows each differential regenerator section to undergo independent Brayton refrigeration cycles of: (1) adiabatic magnetization, (2) isofield heat rejection, (3) adiabatic demagnetization and (4) isofield heat absorption. In operation, warm fluid is pumped to the hot end at $T_H$ where heat is released ($\dot{Q}_H$) and cold fluid is pumped to the cold end at $T_C$ where heat is absorbed ($\dot{Q}_C$).

Cylindrical regenerators are constructed from gadolinium spherical particles with a particle diameter range of 150-300 $\mu$m. Particles are epoxied into monolithic structures with an approximate porosity of 0.37. Each regenerator has a diameter and length of 2.5 inches (63.5 mm), yielding a total refrigerant mass of 2.1 kg (1.05 kg per regenerator). As shown in Fig. 2, the regenerator housings are constructed from G10 composite and are mounted
Figure 1: Schematic of reciprocating AMRR device.

axially opposite onto a common cold heat exchanger (CHEX). To emulate an externally applied load, the CHEX contains thin-film heaters with an uncertainty of 2%. Gas manifolds on the regenerator faces reduce entrance effects and flow maldistribution [15].
Figure 2: Schematic of regenerator assembly inside NbTi superconducting magnet.
Regenerators are moved in and out of the stationary magnetic field with a LabVIEW-controlled reciprocating drive actuator, and a second actuator controls a double-acting piston to displace heat transfer fluid. The piston and regenerators are driven out of phase with equal displacement times of 1 second, yielding a trapezoidal waveform with a fixed operating frequency of 0.25 Hz.

The system is charged with 1.45 MPa of helium to safely operate within the maximum 1.73 MPa pressure rating of the double-acting piston. The maximum displaced fluid volume is 2520 cm$^3$ at a stroke of 20 cm. In addition to a double-acting piston, the heat transfer fluid (HTF) subsystem contains two brazed-plate counterflow heat exchangers with temperature controlled recirculators, allowing the heat rejection temperature ($T_H$) to be controlled in experiments. Omega E-type thermocouples measure $T_H$ and $T_C$ with an approximate uncertainty of 0.5 K. Measurements are recorded with a National Instruments CompactDAQ.

2.2. Superconducting magnet system

A persistent-mode, conduction-cooled NbTi Cryomagnetics superconducting solenoid is used to generate the static magnetic field as shown in Fig. 3. The solenoid consists of two composite windings, as described in Ref. [16], and has a maximum magnetic flux density of 7 T. The superconducting solenoid is thermally connected to a two-stage Gifford-McMahon (GM) cryocooler with cooling capacities of 50 W at 40 K and 1.5 W at 4.2 K. The first stage of the GM cold head cools a 40 K conductive heat shield, shown in Fig. 3, that is also a 40 K radiation barrier to the NbTi superconducting solenoid. The cold box is a super-insulated double-wall dewar and is evacu-
ated to eliminate convective heat leak. The remaining heat leaks come from infrared radiation and thermal conduction through the structural supports, instrumentation leads, and current leads to the magnet windings. In the quiescent cold state, the measured heat leaks into the 2nd stage of the GM cryocooler are 360 mW.

![Figure 3: NbTi superconducting magnet assembly.](image)

Although the superconducting solenoid is capable of 7 T, the translating 1.05 kg regenerators create a heating effect that limits the attainable magnetic field as described in Teyber et al. (2018) [16]. While passive force balancing is an area of ongoing investigation, in the present work the maximum applied field and frequency are limited to 3.3 T and 0.25 Hz to avoid a magnet quench.
2.3. Demagnetizing fields

The magnetic field distribution is altered by the presence of a magnetic material. The electromagnetic finite-element model of Ref. [16] is used to simulate the internal magnetic field along the solenoidal axis as shown in Fig. 4. The black curve shows the magnetic field distribution with an air-bore (3.3 T), and the remaining curves show the internal field distribution with regenerators at $T_H=285$ K and temperature spans of 0 K, 30 K, 60 K and 90 K.

There are two mechanisms altering the magnetic field distribution. The first of which is the beneficial concentration of bore field lines at some locations in the regenerators, first explored by Rowe and Tura (2008) [17]. The second mechanism is the demagnetizing field [18], where the magnetized regenerators create an internal field that opposes the applied field [19]. As the temperature span increases, the regenerators become increasingly ferromagnetic and the demagnetizing field increases. This reduces the average internal field, used to evaluate the adiabatic temperature change and specific heat [20]. The AMR simulations described here use the average internal fields of $\mu_0H_{H,\text{int}} = 3.1$ T and $\mu_0H_{L,\text{int}} = 0.2$ T.

2.4. Parasitic heat leaks

As the AMR device is an imperfect tool for measuring the regenerator performance, loss mechanisms must be quantified to compare experiments with simulations [21]. Here we simplify the thermal design problem into a steady-state, non-interacting system of conduction through the G10 regenerator housings, electromagnetic radiation from the superconducting magnet bore into the CHEX and convection from the surrounding air into the CHEX.
Figure 4: Impact of magnetocaloric material and temperature span on internal field along the solenoidal axis (Fig. 3). Vertical dashed lines indicate regenerator positions. Average high and low internal fields ($\mu_0 H$) are 3.1 and 0.2 T, respectively.

For surfaces completely enclosed by a warmer surface, the electromagnetic radiation is given by [22]:

$$\dot{Q}_{\text{radiation}} = F_{\epsilon} \sigma A_{\text{CHEX}} (T_H^4 - T_C^4)$$

(1)

where $F_{\epsilon}$ is the combined emissivity of the two surfaces, $\sigma$ is the Stefan-Boltzmann constant and $A_{\text{CHEX}}$ is the surface area of the cold heat exchanger assembly. Radiation heat transfer occurs between the cylindrical stainless steel magnet bore ($\epsilon = 0.34$, 12.7 cm diameter), shown in Fig. 3, and the outer diameter of the cylindrical G10 regenerator assembly ($\epsilon = 0.7$, 8.25 cm diameter). The average steady-state bore temperature is taken as the heat
rejection temperature and the CHEX length is 19 cm.

The CHEX is subject to convective heat leaks as regenerators are displaced 25.4 cm in the stationary superconducting magnet. The reciprocating assembly is simplified as steady-state, fully developed flow allowing the average convection coefficient of a concentric tube annulus to be considered. This evaluates to 3 W/m²-K for laminar flow and the geometry described above [23]. The convective heat leak is reduced by a factor of 2 to account for the regenerators being stationary for half of a cycle.

\[ \dot{Q}_{\text{convection}} = \bar{h} A_{\text{CHEX}} (T_H - T_C) / 2 \]  

(2)

The last parasitic heat transfer mode stems from conduction through the G10 regenerator housings; it should be emphasized that the static and dynamic regenerator conductivity are implicit in the AMR model described below. Due to the relative contributions of the terms, thermal interactions between the magnetocaloric material and G10 housing are neglected along the regenerator length. Although the thermal conductivity of G10 varies with weave orientation and supplier, we consider \( k = 0.55 \) W/m-K. The inner and outer diameters of the regenerator housing are 6.35 cm and 8.25 cm, respectively, and the regenerator length is 6.35 cm. The conduction heat leak is then multiplied by the number of regenerators in the system (\( n_{\text{reg}} = 2 \)). Figure 5 summarizes the radiation, convection and conduction heat leaks for \( T_H = 285 \) K.

\[ \dot{Q}_{\text{conduction}} = n_{\text{reg}} k A_{\text{csg}} \frac{T_H - T_C}{L_{\text{reg}}} \]  

(3)
Figure 5: Impact of cold side temperature on parasitic heat transfer modes with $T_h=285$ K. Total parasitic heat leak is sum of radiation, convection and conduction modes.
2.5. AMR model

State-of-the-art AMR models numerically solve the coupled energy equations for the solid and fluid phases [24]. Degregoria (1992) [25] developed an early AMR modeling tool at Astronautics Corporation of America that solved a simplified set of governing equations in the case of negligible entrained fluid capacity, axial conduction and viscous dissipation. In recent years, a number of AMR models have been developed with increased sophistication, as described in Engelbrecht et al. (2006) [26], Aprea and Maiorino (2010) [27] and Tusek et al. (2011) [28]. Instead of prescribing a fluid velocity waveform \textit{a priori}, the models of Barclay et al. (2014) [29], Aprea et al. (2015) [30], Park et al. (2015) [12] and Trevizoli et al. (2016) [31] solved the coupled momentum balance and energy conservation equations. This can be important when modeling AMR devices with a compressible heat transfer fluid.

AMR modeling tools play a critical role in the design of regenerators layered with multiple magnetocaloric materials. Several models have been used to investigate multi-material regenerators with rare-earth alloys of equal layer length, including Aprea et al. (2011) [32], Lei et al. (2016) [33] and Teyber et al. (2016) [34]. Park et al. (2015) [12] took an additional step of simulating the individual layer lengths that maximize the performance of a hydrogen liquefier. While multilayering has improved AMR performance, the nonlinear interactions of individual layers make optimal regenerator compositions sensitive to changes in device configurations and operating conditions [35].

In this work, the AMR device is numerically investigated with the computationally efficient semi-analytic AMR element model [36]. Rowe (2012)
[37, 38] proposed the use of analytical expressions to describe the magnetic work and cooling power in an AMR. These expressions were obtained under the assumption of local thermal equilibrium between the solid and fluid phases. An effective conductivity incorporating finite convection accounts for non-equilibrium between solid and fluid at the macroscopic level. Using the formulation presented by Burdyny et al (2014) [39], the cooling power of a material undergoing an AMR cycle is:

\[
\dot{Q}_{C}^{AMR} = m_s c_s f \zeta T_C \left( \frac{\Phi}{R} \right) \left[ 1 - \left( \frac{\Phi}{2R} + \left( \frac{\Phi \zeta}{R \kappa} \right)^{-1} \right) \left( \frac{T_H}{T_C} - 1 \right) \right]
\]  

(4)

where \( m_s \) is the mass of magnetocaloric material, \( c_s \) is the average high-field solid specific heat of a layer, \( f \) is the operating frequency, \( T_H \) and \( T_C \) are the hot and cold temperatures on the boundaries of the regenerator and \( \zeta \) is defined as the minimum reduced adiabatic temperature change \( (\Delta T_{ad}/T) \) along a layer. \( \kappa \) is the effective thermal conductivity which contains a contribution from thermal diffusion and a degradation factor to account for finite convective heat transfer [39]. Magnetocaloric material properties such as specific heat and adiabatic temperature change are determined via mean field theory (MFT) [40].

Utilization (\( \Phi \)) is a measure of the displaced fluid volume in a regenerative blow and is defined as the ratio of the fluid to solid thermal mass:

\[
\Phi = \frac{\rho_f V_d c_p}{m_s c_s}
\]  

(5)

where \( \rho_f \) is the fluid density, \( V_d \) is the displaced fluid volume and \( c_p \) is the average fluid specific heat. The term \textit{semi-analytic} stems from a modification proposed by Burdyny et al (2014) to include the thermal mass ratio \( (R) \):
\[ R = 1 + \frac{m_f c_p}{m_s c_s} \]  \tag{6}

where \( m_f \) is the entrained fluid mass in the pores.

The net work consists of magnetic and pump work, where the cycle-averaged pump work is defined as the product of volumetric flow rate and pressure drop. The pressure drop is numerically estimated using Ergun’s relation (1952) \cite{Ergun1952}. The semi-analytic expression for magnetic work is given by:

\[
\dot{W}_{\text{mag}} = m_s c_s f \left( \frac{\Delta T}{T} \right) \left[ \frac{R - 1}{R} \Delta T_{\text{ad}} + \frac{\Phi}{R} (T_H - T_C) \right] \]  \tag{7}

The multilayer AMR element model \cite{Hartanto2019} divides a single regenerator into a number of AMR elements, and here we extend the methodology to four elements as shown in Fig. 6. The heat rejection from each AMR element, \( \dot{Q}_{\text{AMR},i}^{\text{H}} \), is solved from a layer energy balance:

\[
\dot{Q}_{\text{AMR},i}^{\text{H}} = \dot{Q}_{\text{AMR},i}^{\text{C}} + \dot{W}_{\text{mag}} + \dot{W}_{\text{pump}} \]  \tag{8}

and an optimization routine determines the interface temperatures \((T_1, T_2, T_3)\) that satisfy an energy balance between layers (i.e. \( \dot{Q}_{\text{AMR},i}^{\text{H}} = \dot{Q}_{\text{AMR},i+1}^{\text{C}} \)). This formulation allows spatially varying material properties and heat fluxes (i.e. viscous dissipation) to be resolved. Further implementation details and a validation can found in Ref. \cite{Hartanto2019}.

The net cooling power is obtained from an energy balance at the cold node and multiplied by the number of regenerators in the system. The useful refrigeration effect is reduced by parasitic heat leaks.
The second law efficiency is then defined as:

\[ \eta_{II} = \frac{\dot{Q}_C^{\text{net}} T_{\text{span}}}{T_C (\dot{W}_{\text{pump}} + \dot{W}_{\text{mag}})} \] (10)

however it should emphasized that systems with superconducting magnets have an additional work term from the cryocooler that is not considered here.
2.6. Fluid properties

Experiments are performed with 1.45 MPa helium, however elevated charge pressures of 3 MPa and 6 MPa are numerically investigated along with liquid propane at 3 MPa. Table 1 summarizes the key thermophysical properties; considering that pump work is the product of pressure drop and volumetric flow rate, the viscosity is relatively insensitive to charge pressure and fluid density. Although the high specific heat and low entrained fluid mass make helium favorable for high temperature spans and low cooling capacities, more work is expended to drive a large volume of low density heat transfer fluid at a constant utilization (Eq. 5).

Table 1: Overview of thermophysical fluid properties from NIST. Although water is not investigated due to its high freezing temperature, the properties are shown for reference. Values here are evaluated at a reference temperature of $T_H=280$ K, however temperature dependent properties are simulated.

<table>
<thead>
<tr>
<th>fluid</th>
<th>phase</th>
<th>P [MPa]</th>
<th>$\rho_f$ [kg/m$^3$]</th>
<th>$c_p$ [kJ/kg-K]</th>
<th>$\mu_f$ [$\mu$Pa-s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>helium vapor</td>
<td>1.45</td>
<td>2.50</td>
<td>5.19</td>
<td>19.07</td>
<td></td>
</tr>
<tr>
<td>helium vapor</td>
<td>3</td>
<td>5.08</td>
<td>5.19</td>
<td>19.133</td>
<td></td>
</tr>
<tr>
<td>helium vapor</td>
<td>6</td>
<td>10.01</td>
<td>5.19</td>
<td>19.253</td>
<td></td>
</tr>
<tr>
<td>$C_3H_8$ liquid</td>
<td>3</td>
<td>524</td>
<td>2.52</td>
<td>121.63</td>
<td></td>
</tr>
<tr>
<td>$H_2O$ liquid</td>
<td>1.45</td>
<td>1000.6</td>
<td>4.19</td>
<td>1431</td>
<td></td>
</tr>
</tbody>
</table>

2.7. Magnetocaloric properties

After presenting the experimental results with a single material gadolinium regenerator, an optimization is formulated to maximize the efficiency of a four-material regenerator with temperature spans pertaining to natural gas liquefaction or the first stage of a hydrogen liquefier ($T_C=120$ K). The
four magnetocaloric materials are representative of Gd$_{0.27}$Ho$_{0.73}$, Gd$_{0.3}$Dy$_{0.7}$, Gd$_{0.65}$Dy$_{0.35}$ and Gd alloys with Curie temperatures of 173 K, 213 K, 253 K and 293 K, respectively, as shown in Fig. 7.

Figure 7: Adiabatic temperature change (A) and specific heat (B) of rare-earth alloys generated with molecular mean field theory (MFT). Dashed lines show adiabatic temperature change with internal field change from 0.2 to 3.1 T and solid lines show field change from 0.2 to 6 T.
Fig. 7 (B) shows an important property of second order ferromagnetic refrigerants, where below the Curie temperature, the low field specific heat is on the order of 10% larger than the high field specific heat. As described in Holladay et al. (2018) [42], this difference in solid thermal mass allows more heat transfer fluid to be displaced in the low-field blow (hot-to-cold) than the high-field blow (cold-to-hot). The resulting flow imbalance allows several percent of the cold heat transfer fluid to bypass the magnetized regenerator, pre-cooling a process stream initially at $T_H$ to the cold temperature of the stage. While bypass flow is not explicitly investigated here, we focus on configurations where the hot side temperature remains below the Curie temperature. The operating parameters are summarized in Table 2.

Table 2: Summary of experimental operating conditions.

<table>
<thead>
<tr>
<th>$T_H$ [K]</th>
<th>$f$ [Hz]</th>
<th>$\mu_0 H_{H,\text{int}}$ [T]</th>
<th>$\mu_0 H_{L,\text{int}}$ [T]</th>
<th>$m_{s,\text{reg}}$ [kg]</th>
<th>$D_{\text{reg}}$ [mm]</th>
<th>$L_{\text{reg}}$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>285</td>
<td>0.25</td>
<td>3.1</td>
<td>0.2</td>
<td>1.05</td>
<td>63.5</td>
<td>63.5</td>
</tr>
</tbody>
</table>
3. Results

3.1. Validation with experiments

The heat rejection temperature was set at $T_H = 318$ K, where a temperature span of 100 K was measured; the largest reported in literature. The heat rejection temperature was then lowered to $T_H = 285$ K, where bypass flow is possible, and the device reached an ultimate temperature span of 67 K. The applied load was then increased incrementally to form the experimental load curve shown in Fig. 8.

Fig. 8 also shows the impact of loss mechanisms on the AMR simulation. The red curve, $\dot{Q}_{\text{AMR}}^C$, shows the raw regenerator performance in the absence of parasitic heat leaks and with the pump work excluded from the layer energy balance. The impact of parasitic heat leaks is shown in blue, and the simulated load curve with both parasitic heat leaks and pump work is shown in black.
Figure 8: Experimental results using Gd and the simulated impact of loss mechanisms on AMR performance with $T_H = 285$ K and the parameters in Table 2.
3.2. Gadolinium performance maps

The impact of the fluid properties and applied field strength on AMR cooling power and second law efficiency are simulated in Fig. 9 and Fig. 10. Each window shows the impact of internal field strength (x axis) and reference utilization (y axis) on the cooling power (Fig. 9) and second law efficiency (Fig. 10). Recall that $\Phi_{\text{ref}}$ is a normalized metric for the amount of fluid displaced in a regenerative blow (Eq. 5 evaluated at the peak zero-field specific heat of 290 J/kg-K); low utilizations are conducive to high temperature spans, while the cooling power and pump work tend to increase with utilization. The three columns correspond to increasing temperature span (left column; $T_{\text{span}} = 30$ K, middle column; $T_{\text{span}} = 60$ K, right column; $T_{\text{span}} = 90$ K). The rows show the impact of the heat transfer fluid on the performance map. The top row, with Helium at 1.45 MPa, corresponds to the device described in this manuscript, and the experimentally measured points in Fig. 8 are indicated with red circles. The second, third and fourth rows show the implications of a 3 MPa charge pressure, a 6 MPa charge pressure and liquid propane ($C_3H_8$) at 3 MPa as a heat transfer fluid, as summarized in Table 1. Note that the y axis (utilization) scales from 0-1 for propane and 0-0.35 for helium.

It is seen that for the ranges considered, the cooling power and efficiency decrease with temperature span and increase strongly with applied field. Increasing the applied field to the design limit of 7 T is a priority and is an area of ongoing research [16]. With regards to the fluid properties, consider the top left window corresponding to 1.45 MPa helium and $T_{\text{span}} = 30$ K. Although the utilization of 0.1 is low, increasing the displaced fluid volume (i.e. moving
Figure 9: Cooling power performance maps. Columns show increasing temperature span and rows show heat transfer fluids with increasing density. Although the color scales are limited to 300 W, a maximum cooling capacity of 750 W is simulated with propane at $T_{\text{span}} = 30$ K.
upwards from the red circle) causes a decrease in cooling power. This decrease in performance at low utilizations is due to the associated pump work,
which is the product of pressure drop and volumetric flow rate. Although the low pressure drop of helium-based AMR devices can be deceiving, the large volumetric displacements increase the interstitial shear stress resulting in significant viscous dissipation.

To improve the performance of the described AMR device, for any field strength, the heat transfer fluid density must be increased (moving down in rows). The second and third rows from the top show that both the cooling power and second law efficiency are noticeably improved by increasing the helium charge pressure to 3 MPa and 6 MPa, respectively. Furthermore, the highest cooling powers and second law efficiencies for each temperature span are obtained with liquid propane.

3.3. Multilayer optimization

An optimization is formulated in Eq. 11 to maximize efficiency with the thermal reservoirs of a natural gas liquefier \((T_H = 280 \text{ K}, T_C = 120 \text{ K})\) with the four ferromagnetic refrigerants shown in Fig. 7.

\[
\max \eta_H(m_{s,1}, \ldots, m_{s,4}, V_{D,1-2}, V_{D,3-4})
\]

\[
s.t. \quad T_{H,i} < T_{\text{Curie},i}
\]

\[
\sum m_{s,i} = m_{s,\text{reg}}
\]

\[
m_{s,i} \leq m_{s,i+1}
\]

\[
V_{D,1-2} \leq V_{D,3-4}
\]

Based on the results above, we consider 3 MPa liquid propane and due to the larger temperature span, we assume here that the multilayer regenerator assembly is evacuated to eliminate convective heat leak \((Q_{\text{parasitic}} = \)

28
\( \dot{Q}_{\text{radiation}} + \dot{Q}_{\text{conduction}} \). The design variables are the refrigerant masses in each layer, \( m_{s,i} \), the volume of fluid displaced through the coldest two layers, \( V_{D,1-2} \), and the volume of fluid displaced through the warmest two layers, \( V_{D,3-4} \). The motivation of a variable displaced fluid volume is discussed below. The first constraint (\( T_{H,i} < T_{\text{Curie},i} \)) forces the temperatures of each layer to remain below the Curie temperature, allowing bypass flow, and the second constraint (\( \sum m_{s,i} = m_{s,\text{reg}} \)) prescribes the total regenerator mass. A fixed regenerator mass of \( m_{s,\text{reg}} = 1.5 \) kg is considered, the limit of what can be accommodated in our experimental device, while keeping the regenerator diameter fixed at 63.5 mm. The last two constraints require the regenerator mass and displaced volume to increase towards the warm end of the regenerator; while this imposes designer intuition on the optimized result, this greatly reduces the number of non-feasible designs and optimization convergence time. The Python SciPY differential evolution algorithm is used \[43\] with a population size of 5000. The optimized configuration is summarized in Table 3 and Fig. 11. With a 160 K temperature span, the optimized configuration yields a 41.6 W cooling power and a 21 % second law efficiency.

With the interface temperatures constrained to remain below the Curie temperature, no feasible solutions were found with a single displaced volume (i.e. \( \text{Max } \eta_{II}(m_{s,1}, ..., m_{s,4}, V_{D,1-4}) \)). This highlights the increased complexity of the multilayering problem with bypass flow. Varying displaced fluid volumes can be accomplished with multiple stages, as in Kim et al. (2013) \[11\], however this can also be accomplished with a diversion flow as proposed by Holladay et al. (2017) \[44\]. The diverted fluid volume of 78 cm\(^3\), shown in Fig. 11, can be obtained with a variable flow resistance in the diversion line.

29
Although the low pump work of 2.06 W is counter-intuitive, it is explained by the low viscosity (Table 1), the low superficial fluid velocity through the large diameter regenerators and the constraint on the interface temperatures. The magnetic work, on the other hand, is strongly dependent on the temperature span and magnetic field which are fixed in the optimization. Finally, note that $T_1$ and $T_3$ in Fig. 11 are at the constraints of 160 K and 240 K, respectively, while $T_2$ is below the 200 K limit. This suggests that further performance improvements can be obtained by implementing diversion flows between each layer.

Table 3: Optimization results with liquid propane heat transfer fluid (3 MPa), 1.5 kg regenerator mass ($m_{s,\text{reg}}$) and $T_{\text{span}}= 160$ K. Work and cooling power are shown for both regenerators.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>C$_3$H$_8$ (3 MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_{D,1-2}$ [cm$^3$]</td>
<td>47</td>
</tr>
<tr>
<td>$V_{D,3-4}$ [cm$^3$]</td>
<td>125</td>
</tr>
<tr>
<td>$m_{s,1}$ [kg]</td>
<td>0.305</td>
</tr>
<tr>
<td>$m_{s,2}$ [kg]</td>
<td>0.315</td>
</tr>
<tr>
<td>$m_{s,3}$ [kg]</td>
<td>0.370</td>
</tr>
<tr>
<td>$m_{s,4}$ [kg]</td>
<td>0.510</td>
</tr>
<tr>
<td>$\eta_{II}$ [-]</td>
<td>0.21</td>
</tr>
<tr>
<td>$\dot{Q}_C$ [W]</td>
<td>41.6</td>
</tr>
<tr>
<td>$\dot{W}_{\text{pump}}$ [W]</td>
<td>2.06</td>
</tr>
<tr>
<td>$\dot{W}_{\text{mag}}$ [W]</td>
<td>262.1</td>
</tr>
</tbody>
</table>
Figure 11: Optimized layering configuration with $T_H = 280$ K and $T_C = 120$ K. More fluid is displaced through layers 3-4 than layers 1-2. Difference is diverted through blue stream to other regenerator, not shown. Temperature nodes correspond to Fig. 6 and each layer has corresponding material properties shown in Fig. 7.
4. Discussion

Although the helium experiments are performed at a utilization of 0.1, simulations indicate that larger displaced fluid volumes do not improve performance. Rather than displacing a greater volume of heat transfer fluid, the performance maps highlight the necessity of a high-density heat transfer fluid and liquid propane appears promising for natural gas liquefaction or the first stage of a hydrogen liquefier. Due to flammability concerns, however, a pressurized liquid propane system must be carefully engineered.

With liquid propane, the simulated four-layer regenerator composition cools a 41.6 W load at a 160 K temperature span with a 21 % second law efficiency. The requirement of variable displaced fluid volumes, however, suggests that the complexity of the multilayer design problem is increased when the interface temperatures are constrained to allow bypass flow. Furthermore, additional performance improvements are expected by utilizing the sensible energy of a bypassed fluid stream [42].

Preliminary modeling results suggest that higher efficiencies are possible with increased system scales. Future works will focus on minimizing the combined capital and operating costs [45] of an increased-capacity magnetocaloric liquefier. This approach allows the magnet design and cryocooler requirements to be coupled in the optimization [46]. Advanced regenerator matrices with variable porosities are also a promising area of research [47], as lower porosities reduce the required high-field volume and associated magnet cost at the expense of increased pump work. Finally, the optimized configurations will be investigated using a multiphase numerical AMR model that allows the effects of bypass flow to be investigated explicitly.
5. Conclusion

The design of a large-scale active magnetic regenerator apparatus with a conduction-cooled superconducting magnet is described, and with 2.1 kg of gadolinium, a 100 K temperature span is measured; the largest reported in literature. The device loss mechanisms are quantified to facilitate the simulation of system performance maps using the semi-analytic active magnetic regenerator element model. The efficiency is found to increase strongly with magnetic field strength and heat transfer fluid density, motivating liquid propane in magnetocaloric natural gas liquefiers. For the first time, the material composition and device operating parameters are simultaneously optimized. With a regenerator mass of 1.5 kg (3 kg total), the optimized configuration allows a 160 K temperature span to be produced with a second-law efficiency exceeding 20 % for small-scale natural gas liquefaction.

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appreciated. Additionally, Dr. Andrew Rowe would like to acknowledge the support of the Natural Sciences and Engineering Research Council of Canada.
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