

Thermodynamic Performance Assessment and Comparison of Active Magnetic Regenerative and Conventional Refrigeration Systems



Hadi Ganjehsarabi, Ibrahim Dincer, and Ali Gungor

1 Introduction

Refrigeration is a significant process in various industrial sectors, including food industry, and progress on developing efficient and environmentally benign systems and applications has accelerated. Recently, magnetic refrigeration at room temperature has attracted great interest, in part due to its being environmentally friendly than a domestic refrigerator. The concept of magnetic refrigeration near room temperature was proposed well before the advent of electrically driven refrigerators. Magnetic refrigeration near room temperature was proposed by Brown (1976). This technology was further developed by Steyert (1978) who was assessing the Stirling cycle for magnetic refrigerator (MR). Many new magnetocaloric material and system designs are proposed as the technology developed. Numerous experimental studies on magnetic refrigerator in the literature have been reported (Okamura et al. 2007; Zimm et al. 2006; Aprea et al. 2014). Bjørk et al. (2011) examined the effects of various design parameters on the capital costs of a refrigerator.

H. Ganjehsarabi (✉)

Erzincan University, Faculty of Engineering, Mechanical Engineering Department, Erzincan 24100, Turkey

e-mail: hadi.ganjehsarabi@gmail.com

I. Dincer

Faculty of Engineering and Applied Science, University of Ontario Institute of Technology, 2000 Simcoe Street North, Oshawa, ON L1H 7K4, Canada

A. Gungor

Department of Mechanical Engineering, Faculty of Engineering, Ege University, TR-35100, Bornova, Izmir, Turkey

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F. Aloui, I. Dincer (eds.), *Exergy for A Better Environment and Improved*

Sustainability 1, Green Energy and Technology,

https://doi.org/10.1007/978-3-319-62572-0_85

Ganjehsarabi et al. (2014) performed the energy and exergy analyses of AMR. Their study provides some useful information about the effects of various design parameters on the performance of the system.

Monfared et al. (2014) recently investigated the environmental impacts of active magnetic refrigerator by performing a life cycle assessment. Their results show that the magnetic refrigeration has higher environmental impacts mainly due to the use of rare earth metals in the magnet material. A study of a magnetic refrigerator and a conventional refrigerator is, therefore, required in order to determine whether a magnetic refrigerator is a more efficient choice than a traditional refrigerator.

In this paper, we thermodynamically study active magnetic and conventional refrigerators for analysis and assessment. A comparative performance assessment is undertaken to study and compare the COP and exergy efficiency values obtained and the effects of using these systems on the sustainable index. The effect of variation of hot source temperature on the cost per unit of cooling and sustainability index (SI) is investigated to provide some useful economic information.

Nomenclature

| | |
|----------------------|--|
| A_c | Cross section area (m^2) |
| c | Specific heat ($N \cdot m^{-2}$) |
| D | Diameter of the regenerator section (m) |
| dp | Diameter of the particles (μm) |
| Ex | Exergy flow rate (W) |
| h | Convection coefficient ($W \cdot m^{-2} \cdot K^{-1}$) |
| L | Length of the regenerator, m |
| \dot{m} | Mass flow rate, $kg \cdot s^{-1}$ |
| Nu | Nusselt number |
| Pr | Prandtl number |
| t_1 | Magnetization time step (s) |
| t_2 | Isofield cooling time step (s) |
| t_3 | Demagnetization time step (s) |
| t_4 | Isofield heating time step (s) |
| T | Temperature, K |
| \dot{W} | Work, $kJ \cdot s^{-1}$ |
| ΔP | Pressure drop, Pa |
| <i>Greek letters</i> | |
| ϵ | Porosity of the regenerator bed |
| ρ | Density $kg \cdot m^{-3}$ |
| μ | Efficiency (–) |
| <i>Subscripts</i> | |
| des | Destruction |
| inf | Undisturbed flow |
| f | Fluid |
| p | Pump |
| s | Solid |

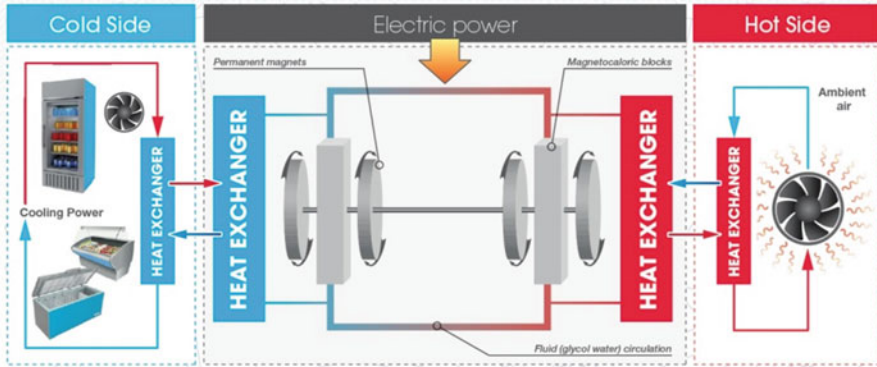


Fig. 1 General functioning of the magnetic refrigeration system [cooltech-applications.com]

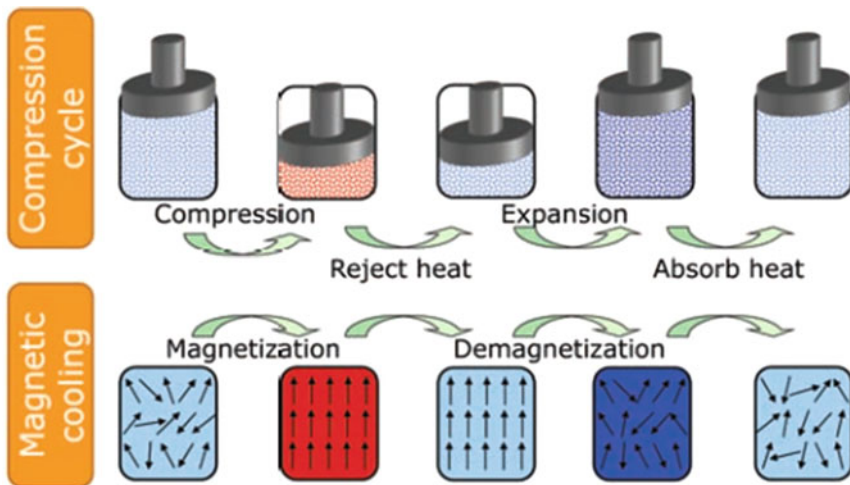


Fig. 2 Illustration of the analogy between a conventional refrigerator and the AMR cycle (Nielsen 2010)

2 Active Magnetic Regenerative Refrigerator

The schematic of active magnetic refrigerator is provided in Fig. 1. It consists of four main processes, namely, two constant field and blow processes.

To better understand of the working mechanism of regenerative refrigerators, the analogy between the magnetic refrigerator and a conventional refrigerator is given in Fig. 2.

During the magnetization process, the magnetic material is magnetized which causes to increase the temperature that corresponds to the compression of a gas. In

the next process, heat is then rejected to the surrounding bringing the system back to the temperature it had before magnetization/compression. During the demagnetization process, the magnetic material is then demagnetized corresponding to the expanding gas. In this manner a temperature below the initial temperature is reached. Finally, a heat load is absorbed and the cycle restarts.

3 Analysis

3.1 Heat Transfer Model

The mechanisms of heat and mass transfer in an AMR bed are complex, so specific assumptions have to be made in order to pose the governing equations. The following assumptions are made: (1) the axial heat conduction in the regenerator is neglected, (2) heat transfer to the surrounding within the AMR bed is negligible, (3) the working fluid is incompressible, (4) the temperature and velocity profile of heat transfer fluid are uniform during the period of flow blowing, (5) the effect of viscous dissipation within the AMR bed is negligible, and (6) the properties of magnetocaloric material (except for the specific heat) are constant in the regenerator. Under these assumptions, energy equation of working fluid (f) and the magnetic material (s) for the AMR bed becomes

$$\varepsilon\rho_f c_f A_c \frac{\partial T_f}{\partial t} + \rho_f c_f u A_c \frac{\partial T_f}{\partial x} = h a_{sf} A_c (T_s - T_f) \quad (1)$$

$$(1 - \varepsilon)\rho_s c_s A_c \frac{\partial T_s}{\partial t} = h A_c a_{sf} (T_f - T_s) \quad (2)$$

where, ε is porosity, A_c is the cross-sectional area of the AMR bed, c is the specific heat, and a_{sf} is the specific surface area. The following correlation for Nusselt number is given by Rohsenow et al. (1985):

$$Nu_f = 2 + 1.1 Re_p^{0.6} Pr_f^{1/3} \quad (3)$$

where Re_p is the particle Reynolds number and Pr is the Prandtl number of the heat transfer fluid.

3.2 Solution Method

The energy equations for working fluid and magnetocaloric material were solved to yield the temperature distributions throughout the AMR bed. Before the time step

was increased, an inner iteration was performed until the following convergence criterion was satisfied:

$$\delta = \text{Max} \left\{ \left| \frac{T_{sI}(0, x) - T_{sI}(t_1 + t_2 + t_3 + t_4, x)}{T_{sII}(0, x) - T_{sII}(t_1 + t_2 + t_3 + t_4, x)} \right|, \right\} < 10^{-6} \tag{4}$$

3.3 Thermodynamic Modeling

3.3.1 Energy Analysis

The refrigeration capacity, rejection heat to the surrounding and magnetic work were calculated by performing the values of the temperature fields coupled with the properties of magnetocaloric material and prescribed flow rate of working fluid. the refrigeration capacity, rejection heat to the surrounding, and magnetic work can be written as follows:

$$\dot{Q}_C = \int_{t_1+t_2+t_3}^{t_1+t_2+t_3+t_4} \dot{m}(t) c_f (T_C - T_{f_i}(t, 0)) dt \tag{5}$$

$$\dot{Q}_H = \int_{t_1}^{t_1+t_2} \dot{m}_f(t) c_f (T_{f_{II}}(t, L) - T_H) dt \tag{6}$$

$$\dot{W}_M = \dot{Q}_H - \dot{Q}_C \tag{7}$$

for the present problem, the Ergun correlation for pressure drop in the working fluid flow is given by (Kaviany, 1985) as

$$\frac{\partial P}{\partial x} = 180 \left(\frac{1 - \epsilon}{\epsilon} \right)^2 \frac{\mu_f}{d_p} w_{inf} + 1.8 \left(\frac{1 - \epsilon}{\epsilon} \right) \frac{\rho_f}{d_p} w_{inf}^2 \tag{8}$$

The pump work rate is a function of pressure drop, the diameter of the particles d_p , fluid viscosity, and the velocity of fluid and can be expressed as follows (Aprea et al. 2011):

$$\dot{W}_p = \frac{\dot{m}(t)(\Delta P_{t_2} + \Delta P_{t_4})}{\eta_p \rho_f} (t_2 + t_4) \tag{9}$$

The coefficient of performance (COP) of the system is defined as follows:

$$\text{COP} = \frac{\dot{Q}_c}{\dot{W}_P + \dot{W}_M} \quad (10)$$

3.3.2 Exergy Analysis

The objective of the exergy analysis is to evaluate the system irreversibility by computing the exergy destruction rate in the entire system and to calculate the associated exergy efficiency. The general exergy balance equation can be represented as the total exergy input equal to the total energy. Including all exergy terms, the general exergy balance becomes (Dincer and Rosen 2007)

$$\sum_{\text{in}} \dot{E} x_{\text{mass, in}} - \sum_{\text{out}} \dot{E} x_{\text{mass, out}} + \dot{E} x_{\text{work}} - \dot{E} x_{\text{heat}} - \dot{E} x_{\text{des}} = 0 \quad (11)$$

which can also be written as

$$\sum_{\text{in}} \dot{m}_{\text{in}} e x_{\text{in}} - \sum_{\text{out}} \dot{m}_{\text{out}} e x_{\text{out}} + \dot{E} x_{\text{work}} - \dot{E} x_{\text{heat}} - \dot{E} x_{\text{des}} = 0 \quad (12)$$

The exergy efficiency of the refrigeration system is evaluated by using

$$\eta_{\text{ex}} = \frac{\dot{Q}_c}{\dot{W}_P + \dot{W}_M} \left(\frac{T_H}{T_C} - 1 \right) \quad (13)$$

3.3.3 Exergoeconomic Analysis

Specific exergy costing (SPECO) is a methodology which is introduced by Lazzaretto and Tsatsaronis (2006) to calculate exergy-related cost in thermal system. In this manner, a link between the fuel and product terms of the entire system and corresponding costs is formed. The general balance equation which is needed for exergoeconomic analysis yields the following:

$$\sum_e (c_e \dot{E} x_e)_k + c_w \dot{W}_k = c_q \dot{E} x_{q,k} + \sum_{\text{in}} (c_{\text{in}} \dot{E} x_{\text{in}})_k + \dot{Z}_k \quad (14)$$

Here c_i , c_e , c_w , and c_q represent average costs per unit of exergy in (\$/kW). \dot{Z}_k is computed by utilizing the cost of an investment and operation and maintenance costs of k_{th} component.

In this problem, a cost balance for the entire AMR cycle can be written as (Rowe, 2011) :

$$c_q \dot{E} x_q = c_{\text{elect}} \dot{W}_M + c_{\text{elect}} \dot{W}_P + \dot{Z} \tag{15}$$

where C_q is the cost per unit of cooling and C_{elect} is electricity selling price, which is taken as 0.075 \$/kWh.

4 Results and Discussion

A consistent comparison of conventional refrigeration system with the active magnetic refrigeration system configurations is accomplished by performing an exergy analysis. In this context, the main objective is to compare the performances of conventional refrigerator and active magnetic refrigerator and to investigate the effect of variations of heat source temperature, T_H , on the performance of an AMR cycle and conventional refrigeration system. To simulate the performance of the conventional refrigerator, a simulation code developed with EES software is used. The simulation of conventional refrigeration system is done by utilizing an actual data reported in the literature (Hepbasli 2007). The working fluid of the conventional refrigerator is R134a. The simulation conditions are given as follows: the cold side temperature is set to $T_C = 255$ K.

In the AMR cycle, under the initial and boundary conditions, the equations governing the cycle were solved. Table 1 summarizes the parameters for model inputs that were used in order to perform the analysis of two configurations.

Figure 3 illustrates variation in the operating efficiency (COP) as a function of heat source temperature for the two configurations. The conventional refrigeration system has higher system efficiency than magnetic refrigeration. It is found that the COP of both systems decrease with increasing heat source temperature.

The effect of heat source temperature on exergy efficiency for the two configurations is shown in Figs. 3 and 4. As the heat source temperature increases from

Table 1 Parameters for the evaluation of active magnetic and conventional refrigerators

| Property | Active magnetic refrigerator | Conventional refrigerator |
|--|---------------------------------------|---------------------------|
| T_H | 293 K | 293 K |
| T_C | 255 K | 255 K |
| Magnetocaloric material of first-stage bed | Gd _{0.85} Dy _{0.15} | – |
| Magnetocaloric material of second-stage bed | Gd _{0.94} Dy _{0.06} | – |
| Applied magnetic field | 1.5 Tesla | – |
| Working fluid | Water–glycol mixture | R134a |
| Mass flow rate | 0.15 kg/s | 0.0009 kg/s |
| Mass of Gd _{0.85} Dy _{0.15} in the first-stage bed | 156 gr | – |
| Mass of Gd _{0.94} Dy _{0.06} in the first-stage bed | 155 gr | – |

Fig. 3 Variation of COP with heat source temperature $T_H(K)$ for active magnetic and conventional refrigerators

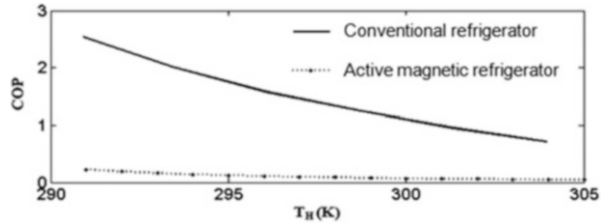


Fig. 4 Variation of exergy efficiency with heat source temperature $T_H(K)$ for active magnetic and conventional refrigerators

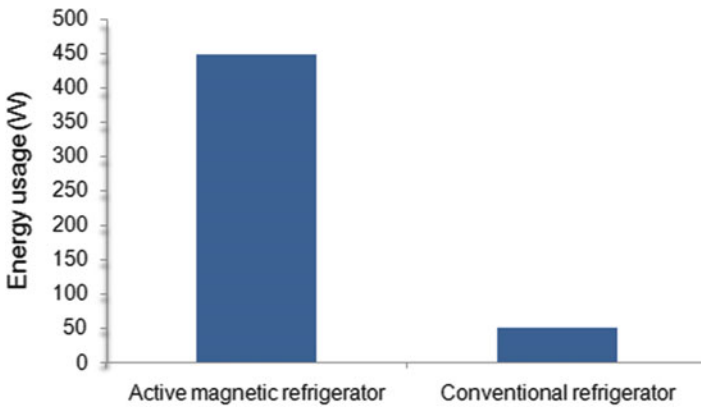
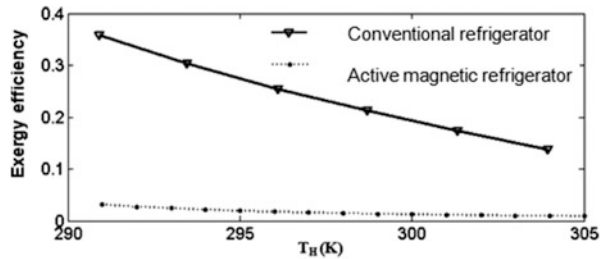


Fig. 5 Energy usage range for active magnetic and conventional refrigeration systems

290 to 305 K, the exergy efficiencies of active magnetic refrigeration and conventional refrigeration system decrease from 3.2% to 1% and 36.62% to 14%, respectively. The reason for this decrease is that as the heat source temperature rises, the cooling capacity decreases.

The conventional refrigeration system has a higher COP; this trend is also reflected in the energy usage. Figure 5 illustrates the energy usage of the conventional refrigeration system and active magnetic refrigeration system. Active magnetic refrigeration system has the highest energy usage due to its high exergy destruction rate in comparison with conventional refrigerator.

The effect of heat source temperature on cost per unit of cooling for both configurations is shown in Fig. 6. As the heat source temperature increases from

Fig. 6 Variation of cost per unit of cooling with heat source temperature $T_H(K)$ for active magnetic and conventional refrigerators

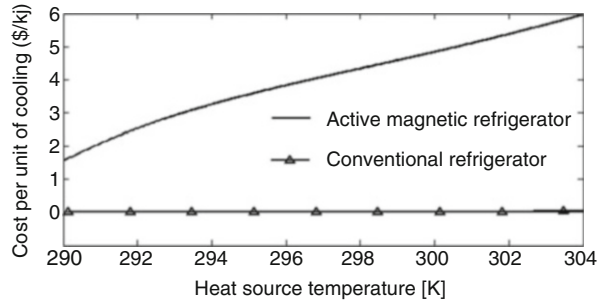
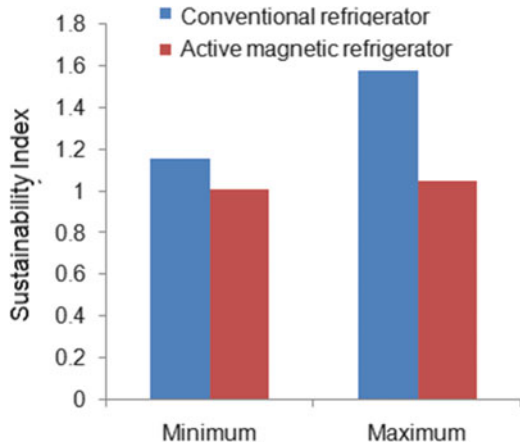


Fig. 7 Sustainability index range for active magnetic and conventional refrigeration systems



290 to 305 K, the cost per unit of cooling of active magnetic refrigeration and conventional refrigeration system increases. The reason for this increase is that as the heat source temperature rises, the COP decreases as a result of a decrease in cooling capacity.

A comparison of SI of both systems is illustrated in Fig. 7. As can be seen in this figure, the SI of conventional refrigeration system appears to be higher as compared to the active magnetic refrigeration system due to the fact that conventional refrigeration system has higher exergy efficiency.

5 Conclusions

In the present study, two configurations of refrigeration system, namely active magnetic and conventional refrigeration systems are examined based on exergy and exergoeconomic analyses as well as SI. Some closing remarks are listed as follows:

- The COP and exergy efficiency of the conventional refrigerator decrease with the hot source temperature while they decrease slightly for the active magnetic refrigerator.
- The cost per unit of cooling of the active magnetic refrigerator increases with the hot source temperature while it decreases slightly for the conventional refrigerator.
- Conventional refrigeration system has higher sustainability index because of higher exergy efficiency as compared to the active magnetic refrigeration system.

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