Applied Thermal Engineering 31 (2011) 1082–1090

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

# Analysis of a magnetic refrigerator

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ARTICLE INFO

*Article history:*
Received 5 May 2010
Accepted 1 December 2010
Available online 8 December 2010

*Keywords:*
Magnetic refrigeration
Thermo-fluid analysis
Coefficient of performance

ABSTRACT

A magnetic refrigerator consists of heat exchangers and beds of magnetic materials. The analysis considered a system that operates near room temperature in a magnetic field between 1 and 7.5 T and uses 3 kg of gadolinium (Gd) spheres, packed in two magnetocaloric beds. Different heat transfer fluids (water, ammonia, and R-134a) were used. The beds were periodically magnetized and demagnetized and the fluid flows were arranged to meet the cycle requirements. A sensitivity analysis was performed. Findings indicate that the higher the magnetic field, the higher the cooling power for the same temperature span. It was also observed that the cooling power decreases with increase in the temperature span for various magnetic fields. The ratio of $COP_{actual}$ to the $COP_{Carnot}$ decreases with an increase in the temperature span. These trends as well as magnitudes of cooling power and coefficient performance were compared with experimental measurements and found to be in reasonably good agreement.



## 1. Introduction

Magnetic refrigeration profits from the fact that the temperature of certain materials increases when placed in a magnetic field, and likewise decreases when the magnetic field is removed. This phenomenon is known as the “magnetocaloric effect, MCE”. Most of the research on the MCE has been associated with materials for very low temperature applications such as helium or hydrogen liquefaction, or materials near room temperature for applications such as conventional air conditioning and refrigeration.

Barclay [1] analyzed magnetic refrigeration cycles and concluded that magnetic refrigeration has a good potential for air conditioning applications. Barclay et al. [2] presented experimental data for a reciprocating magnetic refrigerator. A theoretical model of the active magnetic regenerative refrigerator (AMRR) has been proposed by DeGregoria [3]. DeGregoria et al. [4] tested an experimental magnetocaloric refrigerator designed to operate within the range of 4–80 K. Helium gas was used as the heat transfer fluid. A single magnet was used to charge and discharge two in-line beds of magnetic material. Hagmann and Richards [5] proposed a two stage magnetic refrigerator for astronomical applications. Johnson and Zimm [6] tested the AMRR model against a cryogenic experimental device.

Pecharsky and Gschneidner [7] presented a review on magnetic refrigeration and commented on the development of new magnetic refrigeration technology as an energy efficient and environmentally safe alternative to existing vapor compression refrigeration. Zimm et al. [8] investigated magnetic refrigeration for near room temperature cooling. Water was used as the heat transfer fluid. A porous bed of magnetocaloric material was used in the experiment. It was found that using a 5 T magnetic field, a refrigerator reliably produces cooling powers exceeding 500 W, at coefficient of performance values of 6 or more. It should be noted that the COP of 6 does not include cooling power necessary to maintain the superconducting magnet. Yayama et al. [9] proposed a new hybrid cryogenic refrigerator by combining the magnetic and gas cooling systems. Their results showed a large refrigeration power in contrast to the conventional gas refrigeration. Shirron et al. [10] presented test results of an adiabatic demagnetization refrigerator that produced continuous cooling at sub-Kelvin temperatures.

Yu et al. [11] offered a review on the development of the magnetic material, magnetic refrigeration cycle, magnetic field and the regenerator for room temperature magnetic refrigeration. They concluded that although the development of room temperature magnetic refrigeration has not yet matured, this technology seems to be a viable alternative to the traditional vapor compression technology. Bruck et al. [12] studied a new class of magnetic materials for room temperature applications. These materials are manganese-iron-phosphorus-arsenic (MnFE(P,As)) compounds. This new material exhibits a larger MCE than that of Gd metal. They reported further improvement in the performance of the material by increasing the Mn content. Shir et al. [13] developed a model for temperature in magnetization and demagnetization processes in

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doi:10.1016/j.applthermaleng.2010.12.002

magnetic regenerative refrigeration. This model can be used to simulate optimum operating conditions and duty cycle of magnetic regenerative refrigeration.

Lin et al. [14] presented a model for the irreversible ferromagnetic Stirling refrigeration cycle. The general performance characteristics of this model provided some theoretical guidance for the optimal design of magnetic refrigerators. Yang et al. [15] investigated the performance of an irreversible regenerative magnetic Brayton refrigeration cycle. They evaluated the performance of this cycle by discussing several special cases in detail. Shir et al. [16] developed a time and spatially dependent model to evaluate the performance of a magnetic regenerative refrigeration system. They showed a good agreement between the measured temperature profile and calculated results from the model. Allab et al. [17] presented and validated a one-dimensional time-dependent model of the active magnetic regenerative refrigeration cycle. This model is a useful tool to study transient and steady behavior of the active magnetic regenerative cycle and to determine optimal conditions for achievement of the highest temperature span.

Magnetic refrigeration is an environmentally-clean refrigeration technology. Because magnetic refrigeration employs magnetic materials as refrigeration media, this technology does not have any ozone-depleting or greenhouse effect. Engelbrecht et al. [18] described a numerical model capable of predicting the practical limits of performance of the magnetic refrigeration technology applied to space-conditioning applications. System performance was compared to current vapor compression technology for space-conditioning and refrigeration applications. Engelbrecht et al. [19] presented a review of recent developments in the field of room temperature magnetic refrigeration and discussed some design issues that may affect practical systems. Their conclusion was that although there have been some promising advances in the field of magnetic refrigeration, there are still many practical issues that must be addressed before magnetic refrigeration systems become competitive with vapor compression systems for residential applications. Russek and Zimm [20] concluded that in order for magnetic refrigeration to attain widespread acceptance, its costs must be competitive with vapor cycle technology. They presented some transition magnetocaloric materials that offer good potential for cost reduction.

Yao et al. [21] carried out an experimental study on the performance of a room temperature magnetic refrigerator using permanent magnets. They used helium gas as the heat transfer fluid and a magnetic field of 1.5 T. Their results provide useful data for design and development of room temperature magnetic refrigerators. Li et al. [22] developed a numerical model for predicting the performance and efficiency of active magnetic regenerative refrigerators. They predicted system performance using different heat transfer fluids. They concluded that liquid is more favorable than gas for use as a heat transfer fluid in magnetic refrigeration cycles. Allab et al. [23] presented a methodology to represent the interaction of a magnetic field source system for magnetic refrigeration with magnetocaloric material.

Dieckmann et al. [24] commented that much recent and ongoing research has focused on enhancing the performance of magnetic refrigeration prototype systems to achieve similar or superior efficiency at similar or lower cost than conventional vapor compression equipment. Tishin [25] presented a perspective view of current technology and predicted trends for use of the magnetocaloric effect in applications for magnetic cooling technology. He listed materials that have superior magnetocaloric properties. Kitanovski and Egolf [26] presented an evaluation of magnetic refrigeration applied to existing refrigeration technologies. They concluded that magnetic refrigeration is an environmentally benign alternative to conventional refrigeration and air conditioning technologies.

The above review of literature shows that all previous studies analyzed cycle performance based mainly on the thermal behavior of the magnetic material. A very small number of studies have been published with regard to the effects of the heat transfer fluid on the performance of the cycle. The working fluid should be considered in the same way as it is done for other refrigeration cycles, using key parameters such as the coefficient of performance (*COP*). The model developed in this investigation demonstrates that the thermodynamic cycle can be analyzed using the working fluid processes in a simplified way. Flow control can be realized by using valves among the cycle components. An eight-valve configuration is used to accomplish the working processes of a magnetic refrigerator.

## 2. Analysis

Fig. 1 is a schematic of the bed and heat exchanger assembly. The cooling system consists of two porous medium beds, two heat exchangers, and a pump. The heat transfer fluid is water. The beds are periodically magnetized and demagnetized and the fluid flows are arranged to meet the cycle requirements. The system fluid passes through the hot heat exchanger, which uses air to transfer heat $Q_h$ to the atmosphere. The fluid then passes through the demagnetized cooler magnetocaloric bed and losses heat. This cold fluid cools the conditioned space by exchanging heat with the cooling load $Q_c$. The heat transfer fluid is then heated by the magnetized magnetocaloric bed, where it continues the cycle around. The magnetocaloric material beds are alternately magnetized and demagnetized, and fluid flow is channeled accordingly to continuously run the cooling operation.

The magnetic refrigeration material used in this study is gadolinium metal. Its molecular weight is $M = 157.25$ kg/kmol and has a density of$\rho = 7898$ kg/m$^3$. The magnetocaloric effect in a material can be measured by three different techniques: the direct

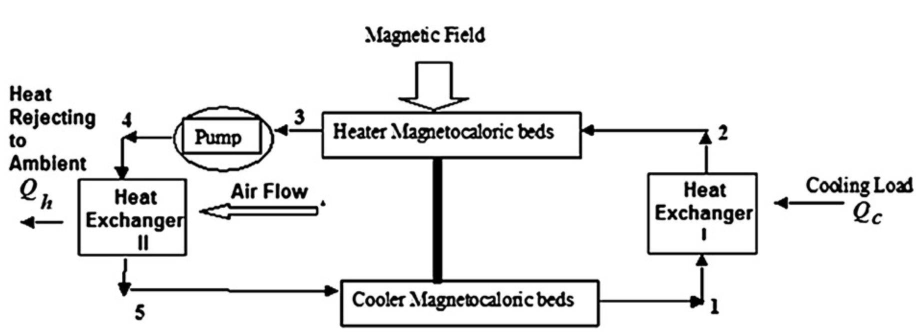


**Fig. 1.** Schematic of the bed-heat exchanger assembly.

**Table 1**
Temperature rise in the magnetic material as a function of the magnetic field [27].

| Magnetic Field Change, $\Delta B$ [T] | Adiabatic Temperature Rise, $\Delta c$ [K] |
|---|---|
| 1 | 2.0 |
| 2 | 4.0 |
| 3 | 6.0 |
| 5 | 11.6 |
| 7.5 | 16.0 |
| 10 | 19.3 |

measurement of the temperature rise (or drop) as the magnetic material is put into or pulled out of a magnetic field; from the field and temperature dependence of magnetization; and from the heat capacity measured at various applied magnetic fields. Table 1 shows the adiabatic temperature rise under a magnetic field representing the magnetocaloric effect of gadolinium, as presented by Tishin et al. [27]. These values correspond to the maximum adiabatic temperature change from this reference. As can be seen, the higher the magnetic field is the higher the temperature change.

The potential refrigeration capacity can be evaluated from Eq. (1) as presented by Barclay [1].

$$Q_{pc} = T_c \Delta s\, f\, m_m \tag{1}$$

where $Q_{pc}$ is the potential cooling capacity, $T_c$ is the cold source temperature, $\Delta s$ is the entropy change at $T_c$, $m_m$ is the mass of the magnetic material, and $f$ is the cycle frequency.

For the gadolinium, the average $\Delta s$ is proportional to the magnetic field change $\Delta B$. Because the heat generated within the magnetic material will be fully absorbed by the material and will result in its temperature change we can write

$$T_c \Delta s = c_{pm} \Delta c \tag{2}$$

where

$$c_{pm} = \frac{\partial s}{\partial T} \overline{T} \tag{3}$$

Here $\Delta c$ is the adiabatic temperature rise in the magnetic material and $\overline{T}$ is an average temperature over $T_c$ and $T_c \pm \Delta c$ (Magnetization or demagnetization).

If it is assumed that the s-T curves are approximately parallel, then $\frac{\partial s}{\partial T}$ is the same for low field and high field. This is a reasonable approximation over limited temperature ranges.Therefore,

$$Q_{pc} = \Delta c\, m_m \frac{\partial s}{\partial T}(T_c \pm \Delta c/2) f \tag{4}$$

From the actual gadolinium T-s curves, $\frac{\partial s}{\partial T}$ can be approximated to be 0.986 J/kg $K^2$ over a magnetic field of 1–10 T and temperature range of 280–308 K [27]. $\Delta c$ can also be evaluated from actual gadolinium s-T curves. Table 1 shows an average $\Delta c$ as a function of the field change$\Delta B$.

## 3. System configuration

The system configuration with eight valves needed for operation, is illustrated schematically in Fig. 2. The beds are periodically magnetized and demagnetized and the fluid flows are arranged to meet the cycle requirements. For the first stage, valves 1, 2, 3, and 4 are opened, and valves 5, 6, 7, and 8 are closed. In this case, the magnetic field is applied to the heater magnetocaloric bed while the cooler magnetocaloric bed has been demagnetized. Cold fluid exiting the cooler magnetocaloric bed moves to cold heat exchanger where it absorbs heat $Q_c$ from the refrigerated space and proceeds to the heater magnetocaloric bed. As the fluid passes through the heater magnetocaloric bed, it picks up the heat generated by the magnetic field and proceeds to the hot heat exchanger. In the hot heat exchanger, the working fluid is cooled by heat rejection to the ambient ($Q_h$) and it completes the cycle. When the magnetic field is released from the top bed and applied to the bottom bed, the cooler bed becomes the heater bed, and vice versa by exchanging roles. In this case, valves 5, 6, 7, and 8 are opened and valves 1, 2, 3, and 4 are closed. The cold fluid from the top magnetocaloric bed now proceeds through valve 7 to the cold heat exchanger. The fluid leaving the cold heat exchanger proceeds through valve 8 to the bottom magnetocaloric bed which is now acting as the heater magnetocaloric bed. The hot fluid flows to the hot heat exchanger via valve 6 where heat is rejected to the ambient and cooler fluid moves to top magnetocaloric bed through valve 5 to complete the cycle. So the cycle meets the scheme planned in Fig. 1. The pump is used to compensate for the pressure drop encountered by the flow loop. Although the set of eight valves may seem excessive from the viewpoint of system reliability, it has features that are very useful such as simplicity of control of fluid flow path to maximize the heat transfer effectiveness in magnetic material beds during magnetization and demagnetization. Moreover, these valves are simple two position (ON-OFF) valves and therefore should be quite robust and reliable. The on/off operation can be easily programmed using an electronic control board and can be automatically controlled in sequence with magnetization and demagnetization. Fig. 3 shows the variation of the

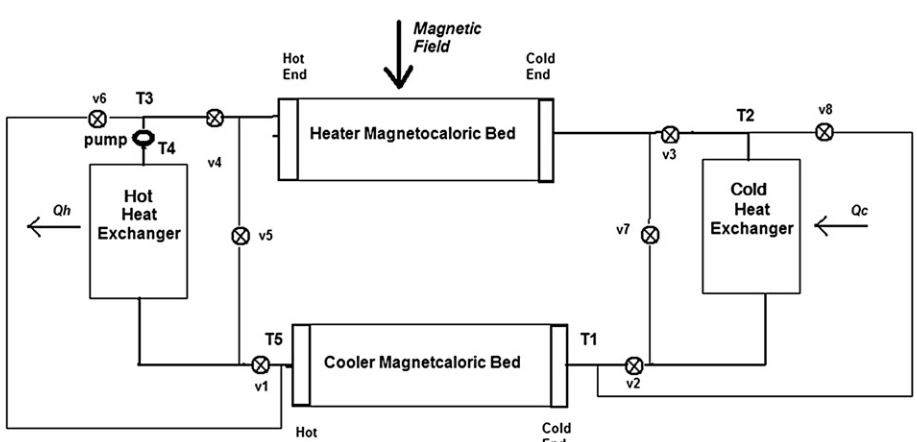


**Fig. 2.** Configuration of eight valve magnetic refrigerator.

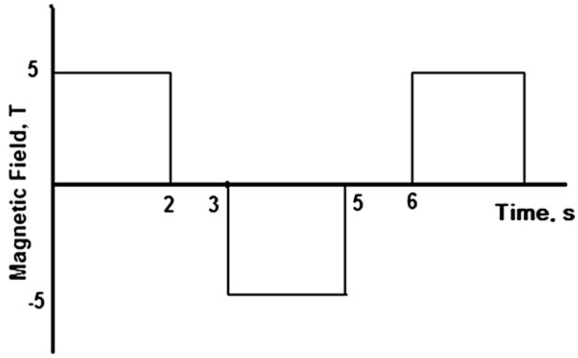


**Fig. 3.** Variation of magnetic field with time.

magnetic field with time used in the present investigation. The cycle then repeats itself every 6 s, after the beds have switched functions (cooling or heating).

## 4. Cycle analysis

The thermodynamic cycle of the eight valve magnetic refrigerator can be described by considering each element within the refrigerator considering the position (closed or opened) for each valve. To simplify the analysis, the following assumptions are made:

- Negligible temperature oscillations in the connecting pipes
- Pressures across the magnetocaloric beds and heat exchangers remain constant.
- The pump is used only to overcome the friction loss encountered in the system.

The heat transfer fluid at approximately $T_5$ gets cooled to $T_1$ by the demagnetized cold magnetic material bed. This cooled fluid is then sent to the cold heat exchanger, where it absorbs the cooling load. This fluid leaves the cold heat exchanger at $T_2$. The warm fluid then flows through the opposite magnetized magnetic material bed, where it is heated to $T_3$. The fluid is then pumped to $T_4$. This hot stream is now cooled in the hot heat exchanger to $T_5$. The cycle then repeats itself at every time period.

Thermodynamic analysis of each component was done by considering a control volume around that component. The process in the cold heat exchanger can be expressed by the relation:

$$Q_C = \dot{m} \;\; c_{pf}(T_2 - T_1) \tag{5}$$

where $Q_C$ is the cooling load, $\dot{m}$ is the mass flow rate of the working fluid, and $c_{pf}$ is its specific heat. The control volume around the heater magnetocaloric bed represents the process of the fluid being heated by the effect of the magnetized bed. This process can be represented as:

$$Q_{pc} = \dot{m} \;\; c_{pf}(T_3 - T_2) \tag{6}$$

The pump is used just to overcome the friction in the piping and in the heat exchangers, therefore it is considered to have equal temperature between the pump entrance and exit.

$$T_4 = T_3 \tag{7}$$

The process in the hot heat exchanger can be expressed through the relation:

$$Q_h = \dot{m} \;\; c_{pf}(T_4 - T_5) \tag{8}$$

where $Q_h$ is the heat dissipated to the ambient. The control volume around the cooler magnetocaloric bed represents the process of the fluid being cooled by the effect of the demagnetized bed. This process can be represented as:

$$Q_{pc} = \dot{m} \;\; c_{pf}(T_5 - T_1) \tag{9}$$

In this case $Q_{pc}$ corresponds to the cooling effect by demagnetizing the bed material.

The *COP* is a dimensionless quantity that describes the performance of a refrigeration cycle. The *COP* is calculated from the cooling load $Q_c$, and the heat rejected, $Q_h$, by

$$COP = \frac{Q_c}{(Q_h - Q_c)} \tag{10}$$

The Carnot cycle, which is completely reversible, is a perfect model for a refrigeration cycle operating between two fixed temperatures. The Carnot limit to the *COP* of a refrigerator is

$$COP_{Carnot} = \frac{T_c}{(T_h - T_c)} \tag{11}$$

Real cycle operation differs from ideal cycle operation by having heat transfer across connecting pipes, and pressure drops due to friction in these pipes, which would lower the thermal *COP* described above. In addition, there will be losses associated with the mechanical and electrical work required to run the system. These losses can be accounted for by modifying the ideal work ($Q_h$-$Q_c$) used in Eq. (10) with actual work $W_{actual}$ needed for the operation of the system. The actual coefficient of performance ($COP_{actual}$) of the refrigeration system can then be calculated as

$$COP_{actual} = \frac{Q_c}{W_{actual}} \tag{12}$$

The actual work is the sum of the magnetization and pumping work.

$$W_{actual} = W_{MAG} + W_{pump} \tag{13}$$

Pump work is usually negligible compared to the total work. A mechanical work is needed to drive the magnetic refrigerator. This work usually comes from an electric motor with an efficiency $\eta_{motor}$. Eddy currents, induced by the magnetic field, lead to further energy loss which can be incorporated by $\eta_{eddy}$. The magnetocaloric material also exhibits a hysteresis effect that can be accounted for by using $\eta_{hysteresis}$. These three efficiencies are assumed with realistic values, carefully selected for the same magneto-material as presented by Kitanovski and Egolf [26].

$$\eta_{motor} = 0.90, \eta_{eddy} = 0.95, \eta_{hysteresis} = 0.97$$

The magnetization work takes into account the system irreversibility as

$$W_{MAG} = \frac{W_{ideal}}{\xi} \tag{14}$$

where the total efficiency$\xi$ is represented by

$$\xi = \eta_{motor}\eta_{eddy}\eta_{hysteresis} \tag{15}$$

Eqs. (5) through (15) represent the thermodynamic model, and Eqs. (1) through (4) are used to calculate the potential heating or cooling of the magnetic materials.

## 5. Results and discussion

The above model allows a quantitative analysis of the thermodynamic cycle for a magnetic refrigerator. Sensitivity analysis has

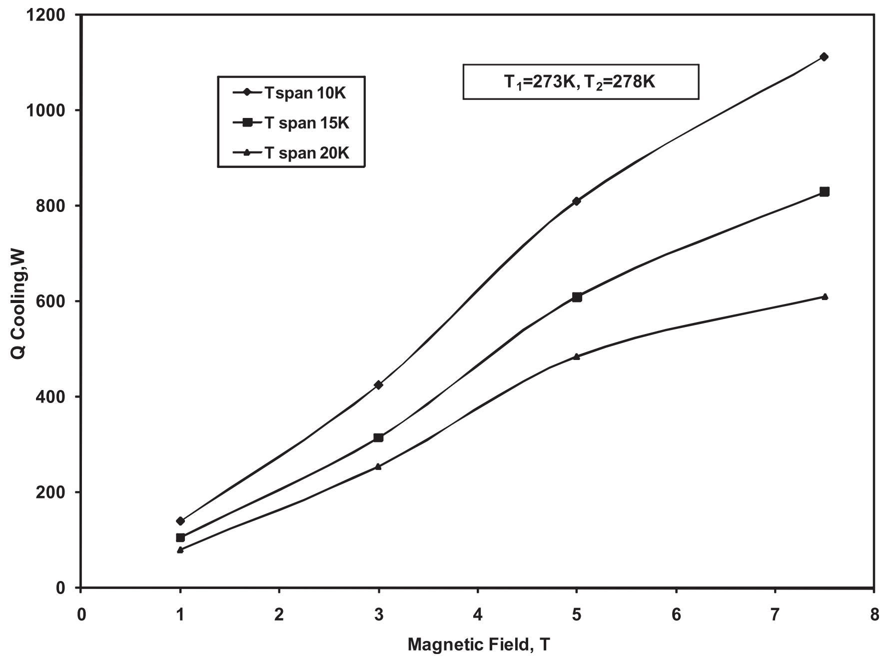


**Fig. 4.** Cooling power vs. magnetic field for various temperature spans.

been performed using gadolinium metal as the magnetic material and water as the primary working fluid. Cooling power, temperature span, and coefficient of performance are analyzed by simulations. Calculations have been made for the proposed model operating at near room temperature in a magnetic field between 1 and 7.5 T and using 3 kg of Gd spheres packed in two identical magnetocaloric beds. The lowest working fluid temperature was set at 273 K. The temperature at the hot heat exchanger is limited by the ambient temperature. The temperature span, defined as the working fluid temperature difference between exits of two heat exchangers, was varied over a range. The primary objective of these calculations was to observe the effect of the variation of the temperature span on the system performance.

Fig. 4 shows the cooling power as a function of the magnetic field. The cooling power was evaluated from the magnetization data. The temperature spans of 10 K, 15 K, and 20 K were considered in this case. It can be seen that for an increase in the magnetic field, there is an increase in the cooling capacity. When the applied field is increased, the cooling load increases because of the increased adiabatic temperature change. It is also observed that the cooling power decreases with an increase in the temperature span. This result is expected in any refrigeration system. It is also noticed that the sensitivity of the temperature span is higher at larger values of the magnetic field.

Fig. 5 shows the coefficient of performance (*COP*) dependence on temperature span for various magnetic fields. The *COP* ranges from

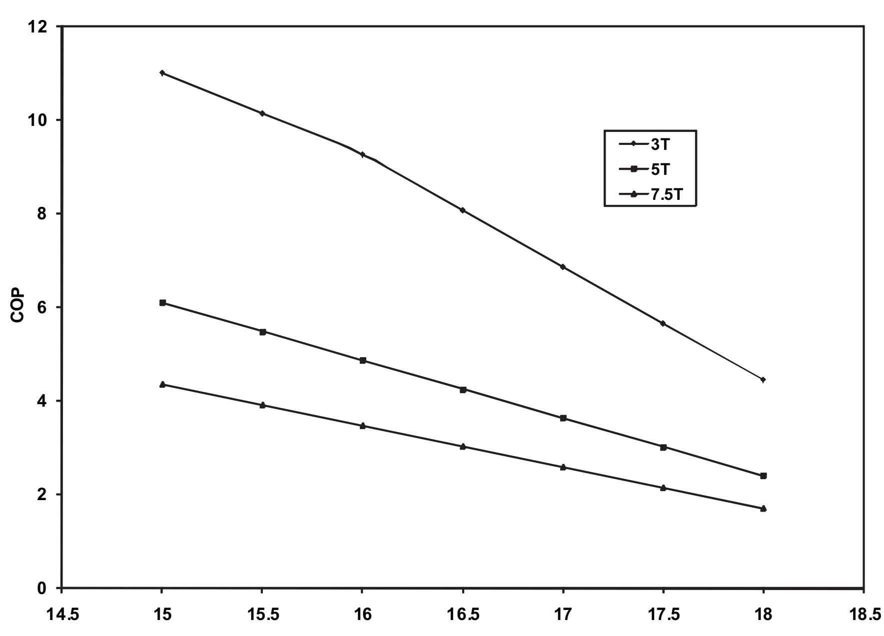


**Fig. 5.** *COP* vs. temperature span for different magnetic fields.

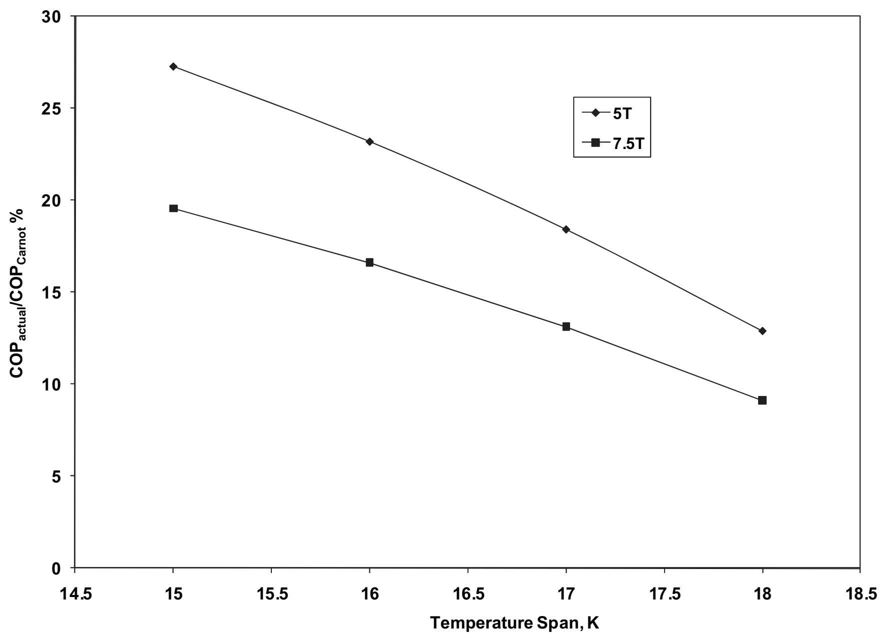


**Fig. 6.** $COP_{actual}/COP_{Carnot}$ vs. temperature span for various magnetic fields.

about two to eleven, which compares favorably with commercial vapor cycle refrigerators. It may be noted that the *COP* decreases with temperature span which is expected in any refrigerating machine, since a higher temperature lift requires a higher amount of work for the same cooling capacity. In this plot it can also be seen that *COP* decreases with magnetic field strength which may be somewhat counter-intuitive. As seen in Fig. 4, for any given temperature span, cooling load $Q_c$ increases with magnetic field strength. However, the net work ($Q_h$-$Q_c$) also increases with magnetic field strength. Since the rate of increase of ($Q_h$-$Q_c$) is larger than $Q_c$, the net result is a decrease in *COP* with magnetic field strength.

The Carnot cycle, which is completely reversible, is a perfect model for a refrigeration cycle operating between two fixed temperatures. The Carnot limit to the *COP* of a refrigerator is the maximum coefficient of performance that may be attained by a refrigerating machine. Fig. 6 presents the ratio of $COP_{actual}/COP_{Carnot}$ vs. temperature span for 5 T and 7.5 T magnetic fields. $COP_{actual}$ is the coefficient of performance predicted by the present model. The ratio $\mathrm{COP}_{\mathrm{actual}}/COP_{Carnot}$ is presented as a percentage of the Carnot cycle performing in the same temperature range. As expected, the ratio of $COP_{actual}/COP_{Carnot}$ drops with increasing temperature span. When the magnetic field is increased, the

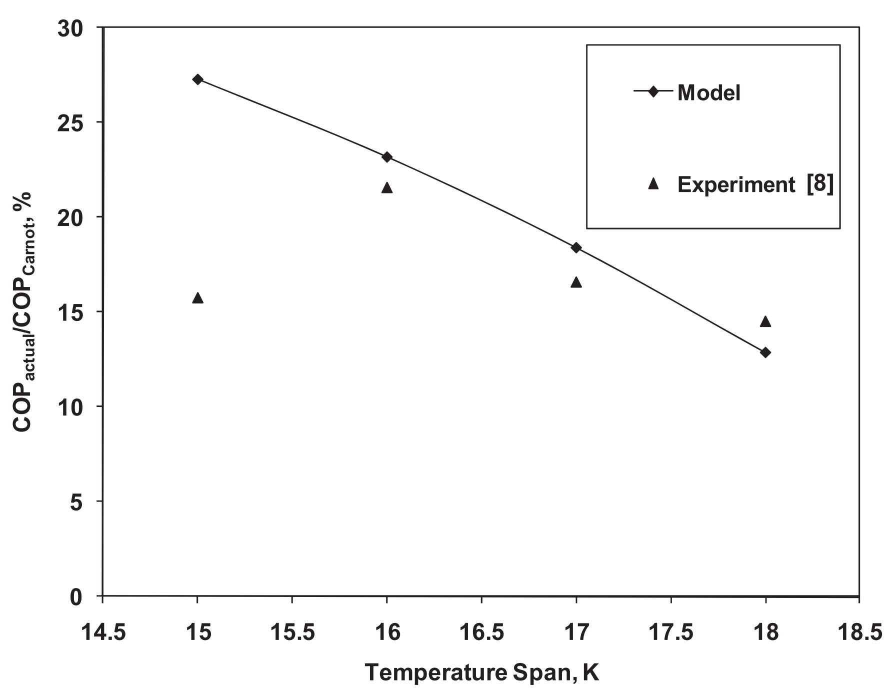


**Fig. 7.** Predicted and experimental $COP_{actual}/COP_{Carnot}$ vs. temperature span for 5 T magnetic field.

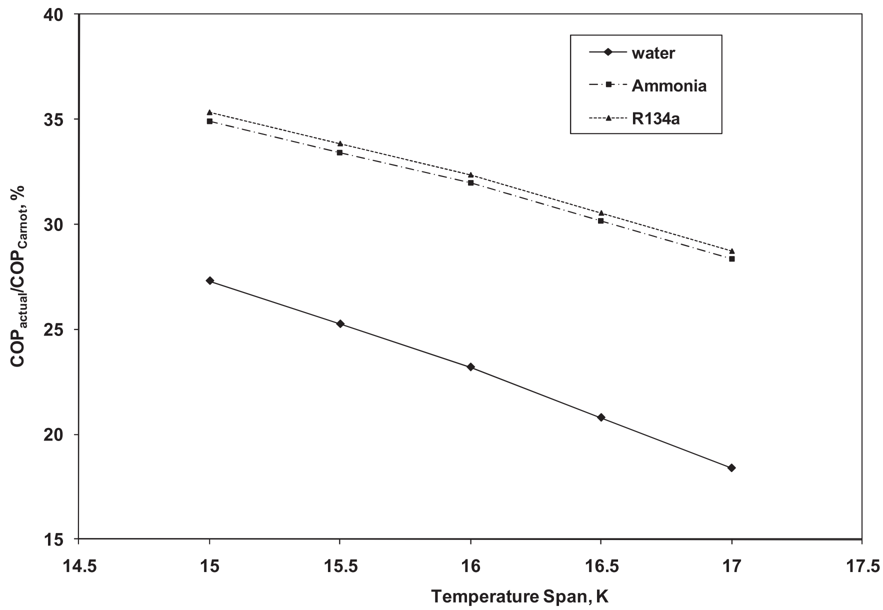


**Fig. 8.** $COP_{actual}/COP_{Carnot}$ vs. temperature span for various fluids with a 5 T magnetic field.

tendency of the ratio decreases. These trends agreed with those shown by experimental data.

Fig. 7 presents predicted and experimental ratios of $COP_{actual}/COP_{Carnot}$ vs. Temperature span for 5 T magnetic field. The experimental data used, was presented by Zimm et al. [8]. This comparison has been performed based on experimental data that takes into account magnetic effects similar to those used in the present model. Model predictions exhibit higher coefficient of performance than experimental data for most of the temperature span. This is expected due to the fact that the model used some simplifying assumptions which may not hold quite accurately in a real device. The large deviation seen at 15 K temperature span is possibly due to larger percentage error associated with measurements at smaller temperature span. Keeping in mind these limitations, the agreement between test data and model predictions may be considered to be quite reasonable. It can be noted that the present model is based only on analysis of the refrigeration cycle, from the working fluid standpoint.

Fig. 8 shows the behavior of the ratio of $COP_{actual}/COP_{Carnot}$ with the temperature span for various working fluids under a 5 T magnetic field. As seen, R-134a and Ammonia exhibit better performance than water as the working fluid. This result was expected because R-134a and Ammonia exhibit more cooling potential, due to higher enthalpy differences for the same

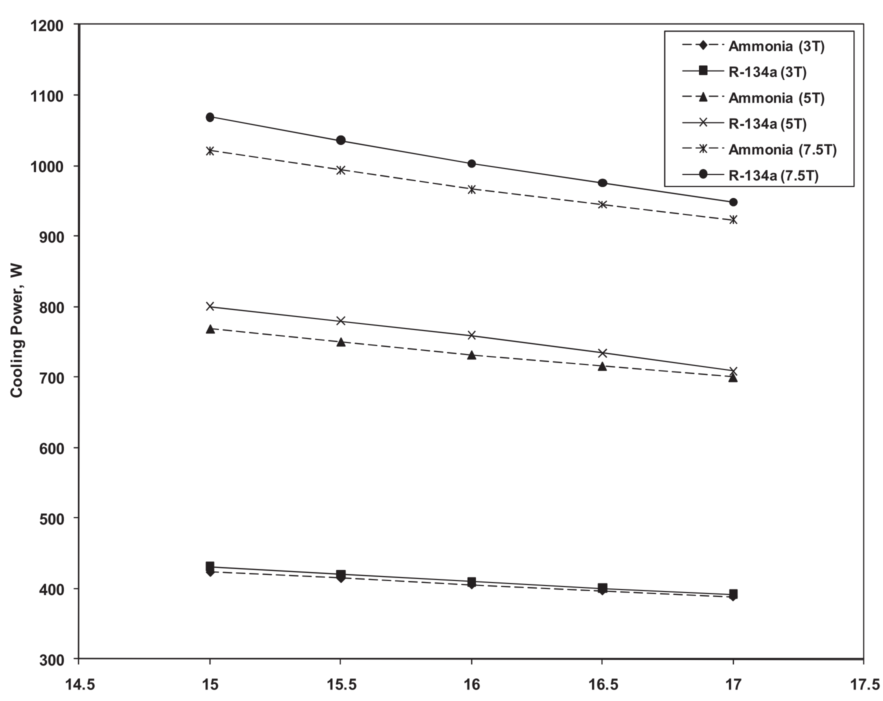


**Fig. 9.** Cooling power vs. temperature span for different magnetic fields with R-134a and ammonia.

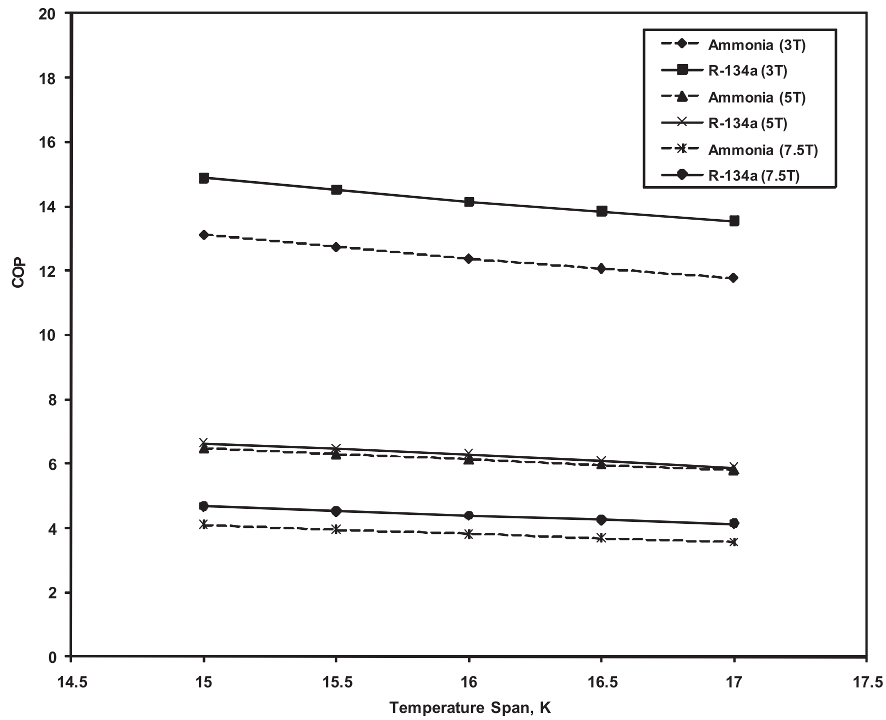


**Fig. 10.** *COP* vs. temperature span for different magnetic fields with R-134a and ammonia.

**Table 2**
Comparison of coefficient of performance between present model and commercial refrigerators.

| Coefficient of Performance (Typical 18 $ft^3$ refrigerator) | | |
|---|---|---|
| Magnetic Refrigerator $(COP)_{actual}$ | Commercial Vapor Cycle Refrigerators [28] | |
| Water | Refrigerant R-134a | Refrigerant R22 |
| 5.06 | 2.26 | 2.29 |

**Table 3**
Comparison of cooling capacity between present model and experimental measurements.

| Q Cooling (W) at 1.5 T Magnetic Field | |
|---|---|
| Present Model | Experimental Measurements [21] |
| Water | He |
| 189 | 51.3 |

temperature conditions. The ratio is higher than 30% over the entire temperature span. Fig. 9 presents cooling power dependence on temperature span for R-134a and ammonia. Fig. 10 presents the corresponding *COP* curves for these two fluids.

Vineyard [28] presented a complete study of alternative refrigerants for a household refrigerator. He tested various viable refrigerants including pure and mixture refrigerants. Table 2 compares results for coefficient of performance between present model and commercial refrigerators as presented by Vineyard [28].

**Table 4**
Comparison of coefficient of performance between present model and experimental measurements.

| $(COP)_{actual}$ at 1.5 T Magnetic Field | |
|---|---|
| Present Model | Experimental Measurements [22] |
| Water | Water |
| 17.4 | 14 |

This comparison was done based on the same temperature span and a magnetic field of 5 T with the same cooling capacity. It is shown that a magnetic refrigerator provides significantly higher coefficient of performance, compared to a vapor compression refrigerator of the same capacity. Therefore, magnetic refrigeration has a great potential to provide cooling with lower power, when compared to the conventional method of compressor based refrigeration. Table 3 shows a comparison of the cooling capacity of present model and experimental data as presented by Yao et al. [21]. This comparison was done based on a magnetic field of 1.5 T and the same magnetic material. It is shown that present model predicts significantly higher cooling capacity compared to experimental data with the same magnetic field. Our model uses water as working fluid, while experimental data presented by Yao et al. [21] used Helium as the working fluid. As expected, gas is not preferable for use as a heat transfer fluid, mainly due to its low volumetric specific heat capacity and low thermal conductivity, despite having low viscosity. Table 4 compares results for coefficient of performance of present model and experimental model as presented by Li et al. [22]. This comparison was made based on the same magnetic material, and a magnetic field of 1.5 T with the same heat transfer fluid. It can be seen that our magnetic refrigerator model predicts slightly higher coefficient of performance compared to experimental model. This may be due to somewhat simplifying assumptions. However, our proposed simplified model is robust enough to predict potential for magnetic refrigeration use.

## 6. Conclusions

A model for a magnetic refrigerator for continuous refrigeration has been developed. The thermodynamic characteristics of the magnetic refrigerator have been studied. A method has been presented for evaluating each process in the cycle, the corresponding working fluid temperatures, and the net cooling power. Sensitivity analysis was performed to study the behavior of the cycle. For this cycle, two important performance parameters were evaluated:

the coefficient of performance and the refrigeration capacity. It was shown that the ratio of $COP_{actual}/COP_{Carnot}$ decreases with an increase in the temperature span. The cooling capacity increases with increase in the magnetic field. The model showed good agreement with experimental data. Various working fluids were studied, finding that R-134a exhibits better performance. Magnetic refrigeration exhibits a great potential by showing higher coefficient of performance when compared to commercial vapor cycle refrigerators.

## Acknowledgements

This research was supported by National Aeronautics and Space Administration (NASA) under grant number NAG3-2751. The authors would also like to express their sincere thanks for the editorial corrections done on the manuscript by Dr. Frank Pyrtle III in the Department of Mechanical Engineering at the University of South Florida.

## Nomenclature

| | |
|---|---|
| B | Magnetic field strength, T |
| $c_p$ | Specific heat, J/kg K |
| f | Frequency, $s^{-1}$ |
| $m$ | Mass, kg |
| $\dot{m}$ | Mass flow rate, kg/s |
| M | Molecular weight, kg/kmol |
| Q | Heat transfer rate, W |
| $s$ | Entropy, J/kg K |
| t | Time, s |
| $T$ | Temperature, K |

*Greek symbols*

| | |
|---|---|
| ρ | Density, kg/m$^3$ |

*Subscripts*

| | |
|---|---|
| actual | Correspond to this actual model |
| Carnot | Corresponds to Carnot cycle |
| *c* | Cold heat exchanger |
| *f* | Fluid |
| *h* | Hot heat exchanger |
| *pc* | Potential cooling |
| *m* | Magnetic material |
| MAG | Magnetic |

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