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Comprehensive Study of Heat Transfer and Pressure Drop in Regenerator and Optimization of Solid-state Caloric Cooling Cycles Using Realistic Hydraulic Diameter of Regenerator

Minwoong Kang
University of Illinois, mk16@illinois.edu

Stefan Elbel
University of Illinois

Comprehensive study of heat transfer and pressure drop in regenerator and optimization of solid-state caloric cooling cycles using realistic hydraulic diameter of regenerator

Minwoong Kang[1], Stefan Elbel[1,2,]*

[1] Air Conditioning and Refrigeration Center, Department of Mechanical Science and Engineering, University of Illinois at Urbana-Champaign, 1206 West Green Street, Urbana, IL 61801, USA

[2] Creative Thermal Solutions, Inc., 2209 North Willow Road, Urbana, IL 61802, USA
* Corresponding Author
Email: elbel@illinois.edu

ABSTRACT

Many researchers have used very small hydraulic diameters in regenerators of solid-state caloric cooling cycles, because smaller diameters can generate higher cooling capacity and system COP. However, using very small diameters hardly represent the real performance of the caloric cooling cycle, because they cannot be manufactured for a real system. Therefore, this paper has used more realistic hydraulic diameter which is used in commercial heat exchangers. To get the more accurate heat transfer coefficient and friction factor, the model used in this paper incorporates hydrodynamic and thermal developing regions which are usually neglected in other papers. This paper shows that COP is a function of heat transfer coefficient, pressure drop and displacement ratio. Higher heat transfer coefficient, lower pressure drop and the optimal displacement ratio can generate higher COP in magnetocaloric refrigeration cycle. From simple analytical comparison, it is expected that plate type regenerators can generate the highest cooling capacity and COP. The regenerator with smaller hydraulic diameter has higher performance, but 0.3mm hydraulic diameter has been chosen in this paper due to manufacturing limitation.
In addition, this paper has investigated the effect of the length of the regenerator, the cycle frequency and the regenerator's porosity on heat transfer phenomena in the regenerator. The cycle frequency has a large effect on cooling capacity and system efficiency while the effects of length and porosity of the regenerator are marginal. For the plate-type regenerator with 0.3mm hydraulic diameter, the system with 0.2m length of the regenerator, 0.45Hz cycle frequency and 0.5 porosity has the highest COP of 2.3 to generate 100W kg^{-1} cooling capacity for a temperature lift of 20°C. This paper provides detailed information of heat transfer phenomena in the solid-state cooling cycles, which need to be understood thoroughly in order to efficiently utilize caloric effects.

1. INTRODUCTION

Magnetocaloric refrigeration has been emphasized as alternative with the best experimentally achieved exergy efficiency among 20 different technologies (Brown and Domanski, 2014). Elastocaloric refrigeration represents the most promising alternative, and magnetocaloric refrigeration is a very promising alternative for future applications (Goetzler et al., 2014). Also, many companies have developed magnetic cooling systems. The reasons are that the technology has theoretically high efficiency and does not cause any environmental problem, such as global warming effect due to conventional vapor compression cycles.
However, the regenerator in the solid-state cooling system undergoes cyclic temperature changes. The cyclic temperature change causes cyclic losses because the regenerator needs to recover its original temperature. This characteristic is different from vapor compression cycle. In the vapor compression cycle, hot spot always stays hot and cold spot always stays cold. That's the reason why different heat exchanger designs are required for solid-state cooling system. Also, there is another difference. In the vapor compression cycle, higher flow rate can make higher cooling or heating effect due to the higher heat transfer coefficient. However, in solid-state cooling, higher mass flow rate does not guarantee higher cooling or heating effects because flow is laminar in regenerator. Moreover, too high flow rate decreases cooling and heating capacity, because high flow rate can decrease temperature gradient in the regenerator by mixing cold and hot side fluid. This is another reason why very different heat exchanger design is required for solid-state cooling system. Therefore, this paper has investigated the heat transfer phenomena between solid and liquid in the regenerator and the optimized regenerator design for solid-state caloric cycles.

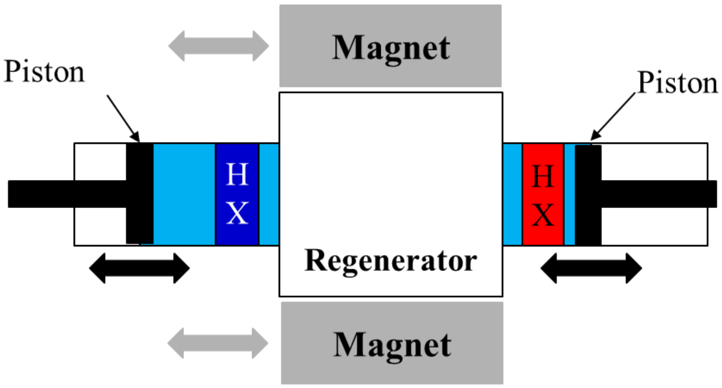


Figure 1: Schematic of modeled solid-state caloric cooling cycle

2. NUMERICAL MODEL

2.1 Schematic of solid-state caloric cooling cycle

Figure 1 shows the schematic of the modeled solid-state caloric cooling cycles. The system is made up of a regenerator, two heat exchangers (cooling heat exchanger and heating heat exchanger) and a permanent magnet. The regenerator is filled with plate type of magnetocaloric material (MCM) through which heat transfer fluid (HTF) passes. The regenerator is magnetized and demagnetized repeatedly by entering and exiting a magnetic field caused by a moving permanent magnet. The performance of this cycle using 2 Tesla of magnetic field is investigated using 1D model.

2.2 Correlations of heat transfer coefficient and pressure drop in parallel plate

This paper has used Nickolay and Martin correlation (2002) for heat transfer coefficient given in Equation (1) and Shah correlation (1974) for friction factor given in Equation (2). They are chosen because they include both developing and developed regions and are known as more accurate than any other correlation.

- Nickolay and Martin (2002)

$$\overline{Nu}_D = \left[7.541^{3.592} + \left(1.841 \cdot Gz^{1/3}\right)^{3.592}\right]^{1/3.592} \tag{1}$$

- Shah (1978)

$$f \cdot Re = 4 \times \left(\frac{3.44}{(x^+)^{1/2}} + \frac{24 + 0.674/(4x^+) - 3.44/(x^+)^{1/2}}{1 + 0.000029 \cdot (x^+)^{-2}}\right) \tag{2}$$

2.3 Selection of hydraulic diameter

Many researches have used very small hydraulic diameter of regenerator, such as 0.05mm or 0.1mm, in their models for solid-state caloric cooling cycles, because a decrease in hydraulic diameter increases heat transfer coefficient and specific surface area, which cause an increase in cooling capacity and cycle efficiency as shown in Figure 2. Figure 2

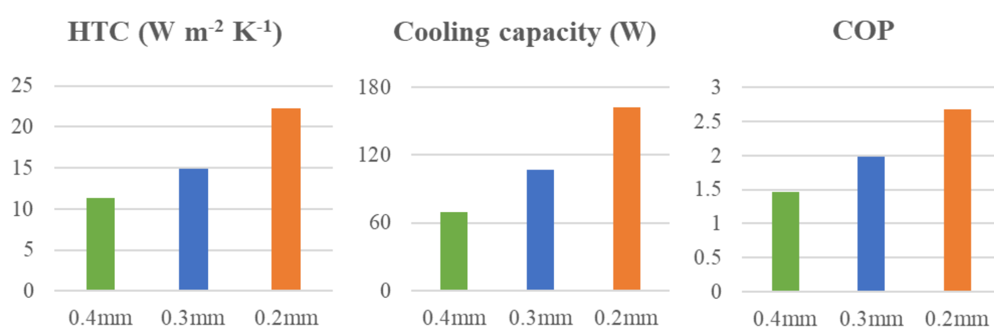


Figure 2: Heat transfer coefficient, cooling capacity and COP for the hydraulic diameter of 0.2, 0.3 and 0.4mm

Table 1: Nusselt number and friction factor for different type of regenerator (Laminar & L >> d_h)

	Friction factor	**Compared w/ base**	**Nu**	**Compared w/ base**
Circular (Base)	64/Re	-	3.66	-
Rectangular	57/Re	89 %	3.55	97 %
Plate	96/Re	150 %	7.54	206 %

shows the heat transfer coefficient, cooling capacity and COP for the hydraulic diameter of 0.2, 0.3 and 0.4mm when using same volume of regenerator. However, there is the limitation of hydraulic diameter because of the manufacturing limitation.
Tušek et al. (2012) used 0.2 mm hydraulic diameter for his parallel-plate type of regenerator which is manufactured by laser welding technology and Kirol and Dacus (1987) used the parallel-plate regenerator with 0.25mm hydraulic diameter which made by the photochemical machining. The hydraulic diameter of 0.3 ~ 0.4mm is known as the minimum value in the industrial field. Therefore, this paper has used the hydraulic diameter of 0.3mm.

3. RESULTS AND DISCUSSION

3.1 Comparison of difference type of regenerator

Table 1 shows Nusselt number and friction factor for different micro-channel type of the regenerator when flow is laminar, and the length of regenerator is much the larger hydraulic diameter of channel. The geometric type of the regenerator is directly related to the heat transfer coefficient and friction factor as shown in Table 1. Therefore, the effect of type of regenerator on the system performance can be simply anticipated by checking their Nusselt number and friction factor and the effect of Nusselt number and friction factor on the system performance when other parameters, such as mass flow rate, cycle frequency, specific surface area, volume of regenerator and porosity which are not related to the geometry of regenerator are fixed. It is known that the effect of heat transfer coefficient on the system performance is 1.5 to 2.0 times higher than the effect of friction factor. The plate type of regenerator is expected to have the highest system efficiency among three different type of regenerator because it has almost two times higher Nusselt number although it also has the highest friction factor. Therefore, this paper has used the plate type of regenerator for solid-state caloric cooling cycle.

3.2 Consideration of developing regions

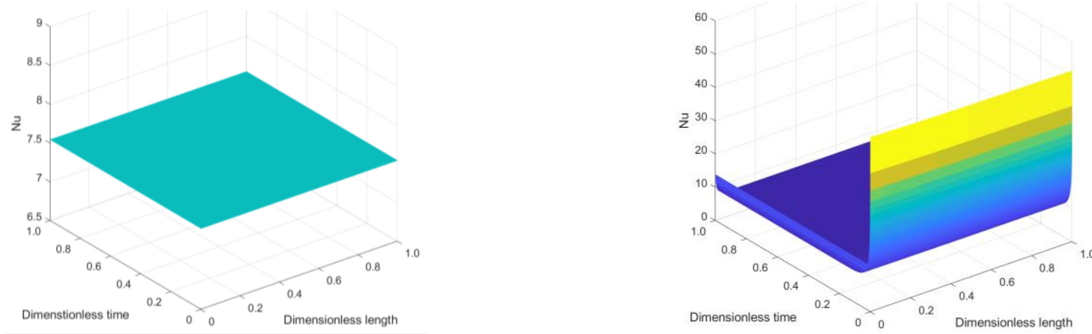
Table 2 and Table 3 represent operating conditions and performance results to compare the model including developing regions with the model neglecting developing regions. When cycle frequency is 1Hz, two model are similar to each other but when cycle frequency is 5Hz, the results of two model are very different as shown in Table 2. This is because developing regions comprise larger part of blow period at 5Hz than at 1Hz. An increase rate of Nu is bigger

Table 2: Operating conditions

Length (m)	0.2	
Porosity (-)	0.6	
Cycle frequency (Hz)	1	5
Mass flux (kg m^{-2} s^{-1})	150	250

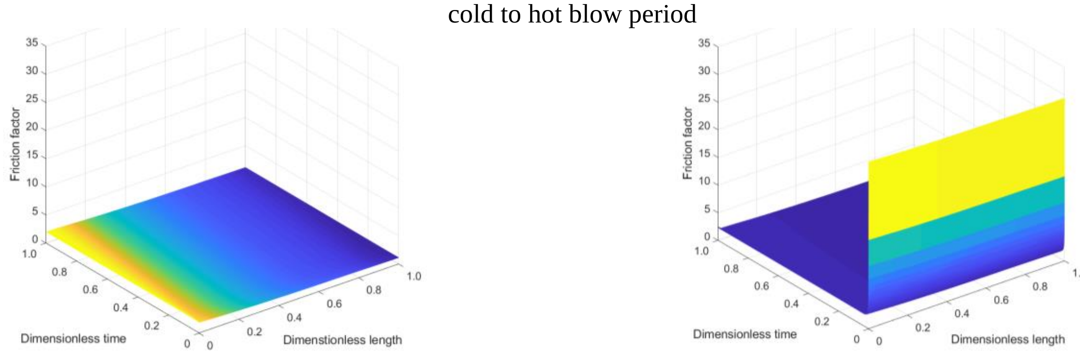
Table 3: Performance results of the model neglecting developing regions and the model including developing regions (1Hz and 5Hz)

	1Hz		5Hz	
	Neglecting developing regions	Including developing regions	Neglecting developing regions	Including developing regions
COP (-)	1.37	1.38	0.22	0.35
Qc/MCM (W kg^{-1})	123	125	49	78
Nu (-)	7.5	7.7	7.5	8.2
Pressure drop (Pa)	18	19	29	31



(a) (b)

Figure 3: Nu for (a) model neglecting developing regions and (b) model including developing regions during cold to hot blow period



(a) (b)

Figure 4: Friction factor for (a) model neglecting developing regions and (b) model including developing regions during cold to hot blow period

than increase rate of pressure drop (Nu: 9.3%, pressure drop: 5.4%) because thermal entrance length is longer than hydrodynamic entry length due to relatively large Prandtl number of water. Therefore, when cycle frequency is high, developing regions should be considered in model. Figure 3 and Figure 4 show Nu and friction factor for (a) model neglecting developing regions and (b) model including developing regions during cold to hot blow period. These graphs represent developing regions of entrance of regenerator as well as after flow direction is changed.

3.3 Mass flux (Displacement ratio)

Table 4 shows operating conditions and Figure 5 shows COP and cooling and heating capacity along mass flux. When other conditions are fixed and only mass flux changes, mass flux is directly proportional to the displacement ratio ($\tilde{V}$). The displacement ratio is given in Equation (3), which is the ratio of the volume of the displaced fluid to the total volume of fluid in the regenerator as shown in Figure 6.

$$\tilde{V} = \frac{V_{f,displaced}}{V_f} \tag{3}$$

Thus, the performance change graph of mass flux is similar to that of the displacements. When mass flux is low and increased, the temperature difference between MCM and HTF is maintained and heat transfer is increased as new fluids are introduced rapidly. Therefore, cooling and heating capacity increases. However, further increase of mass flux eventually leads to very cold fluid flowing into the hot side of generator and very warm fluid entering the cold

Table 4: Operating conditions

Temperature span	20°C (5 – 25°C)
Frequency	0.45Hz
Porosity	0.6
Hydraulic diameter	0.0003m

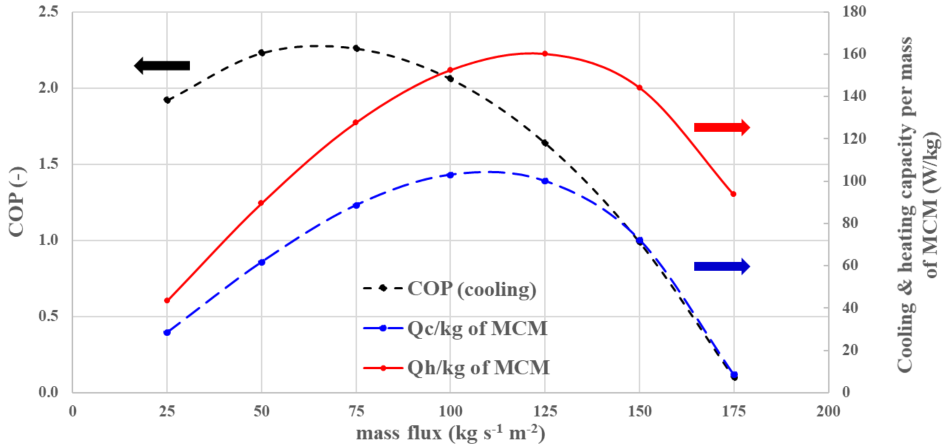


Figure 5: COP, cooling & heating capacity versus mass flux

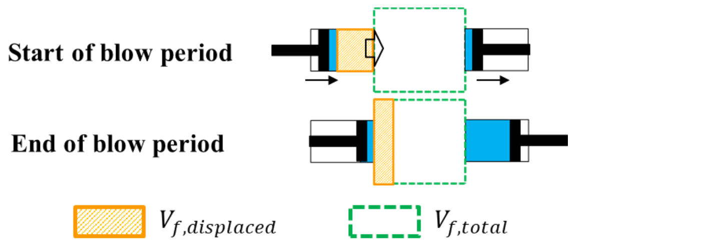


Figure 6: Volume of the displaced fluid and total volume of fluid in the regenerator

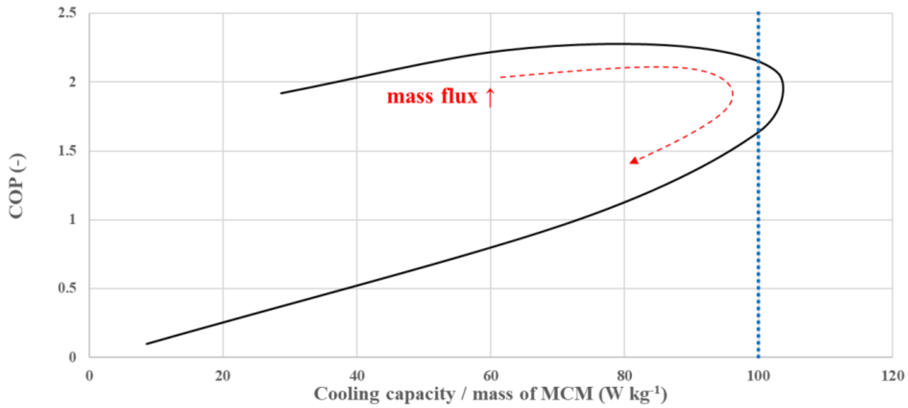


Figure 7: COP versus cooling capacity / mass of MCM

side of generator, reducing the temperature inside the register and eventually reducing cooling and heating capacity. COP also increases at the beginning of mass flow increase. It tends to decrease in the latter half. This is because the thermal conductivity change rate of the water is greater when the temperature is lower than when the temperature is high. In other words, when the thermal conductivity is low, the change rate of HTC is high and when the change rate of the thermal conductivity is high the change rate of HTC is low. As a result, when the water temperature entering the CHX decreases and the water temperature entering the HHX increases, the former shows a greater variation than the latter, indicating increasing the COP. However, when the water temperature entering the CHX increases and the water temperature entering the HHX decreases, the COP decreases. In addition, higher mass flux causes higher pressure drop. Therefore, COP is a function of heat transfer coefficient, pressure drop and displacement ratio. Higher heat transfer coefficient, lower pressure drop and the optimal displacement ratio can generate higher COP in magnetocaloric refrigeration cycle.
Figure 7 represents COP versus cooling capacity per mass of MCM. This paper has often used this type of graph because it can represent COP and cooling capacity together, making it easy to compare graphs under different conditions. In this graph, the x-axis is cooling capacity per mass of magnetocaloric material. Therefore, it is easy to

calculate how much mass of MCM is needed to generate certain cooling capacity under different conditions. Also, this paper has used 100 W kg^{-1} of MCM as standard to compare different conditions, because it is known as the moderate and subsequently cooling capacity and COP decreases. This graph features show that there are two mass flux points that can generate a constant cooling capacity and therefore two COPs. In general, the only upper graphs with higher efficiency have been used to compare the efficiency of other conditions.
cooling capacity. As Figure 7 shows, as mass flux increases, the graph heads to the upper right, to the lower right and finally to the lower left. This is because COP and cooling capacity increases at the beginning and then COP decreases.

3.4 Length of regenerator

Many papers have shown the effect of aspect ratio on the system performance. However, if the hydraulic diameter is fixed in the channel type of the regenerator, the effect of height and width of the regenerator on system performance is just superposed. Therefore, it is reasonable to look at the effects of length than aspect ratio of regenerator rather. Figure 8 shows the COP versus cooling capacity for various length. Figure 9 represents the optimal mass flux to generate maximum cooling capacity, dissipation loss, conduction loss and total loss for different length. Loss effect in the Figure 9 means the percentage of loss to total heat transfer between MCM and HTF during one blow period. The regenerator with higher length needs higher mass flux to reach the optimal displacement ratio, which causes an increase in pressure per unit length. Therefore, a dissipation loss increases as length of regenerator increases. It can also be seen that the temperature gradient ratio decreases as the length increases and so the conductive loss per length decreases until they are all at 200mm. However, after that, the effect of the increase in mass flux cancels out the decrease in the temperature gradient ratio. Therefore, total loss is the minimum when the length is about 200mm, and the performance is then the highest. However, both are pretty small compared to heat transfer between MCM and HTF. Length of regenerator has little effect on system performance.

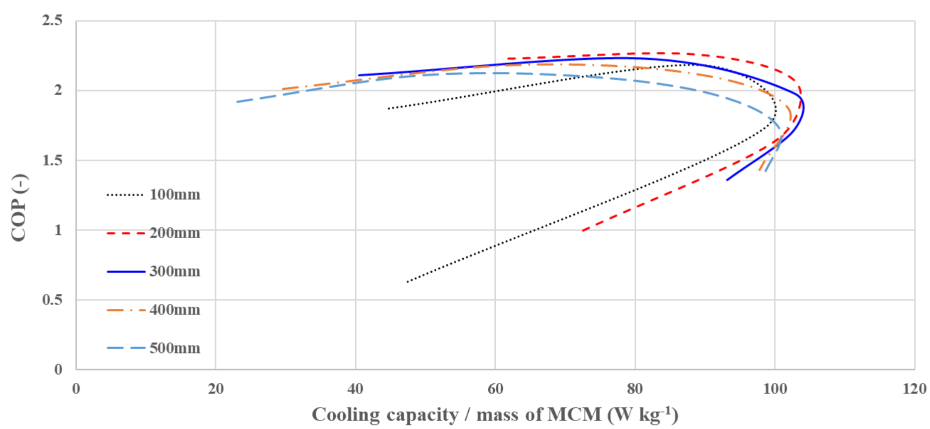


Figure 8: COP versus cooling capacity for various length

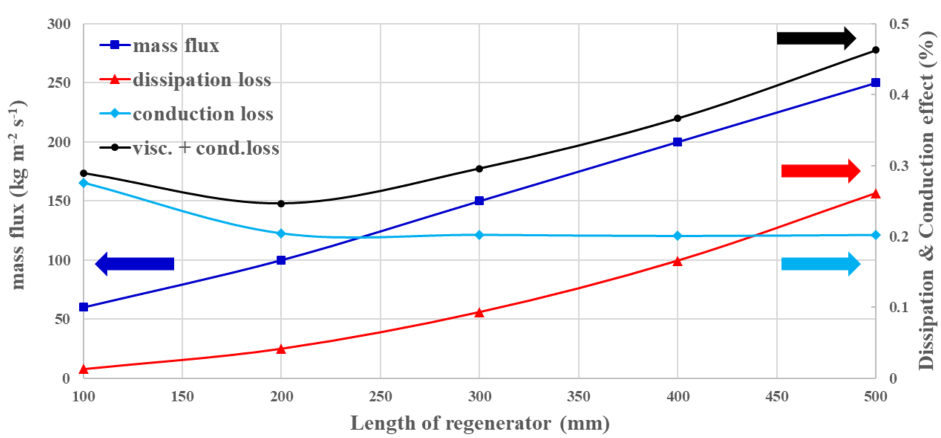


Figure 9: Optimal mass flux, dissipation loss, conduction loss and total loss for various length of regenerator

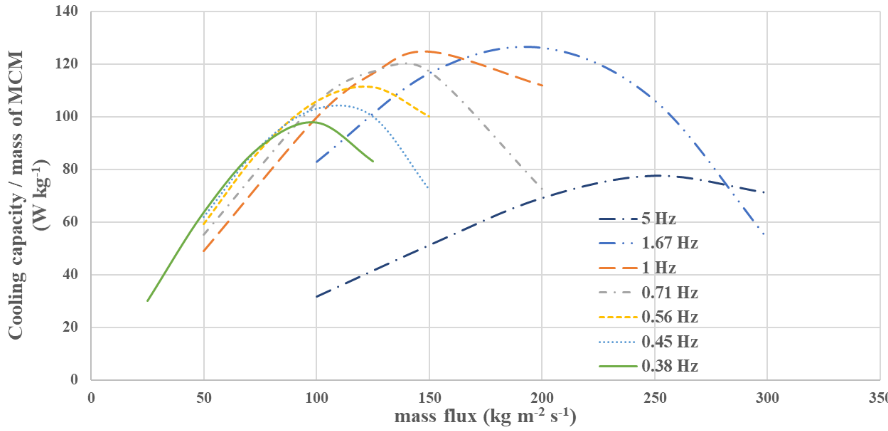


Figure 10: Cooling capacity versus mass flux for various frequency

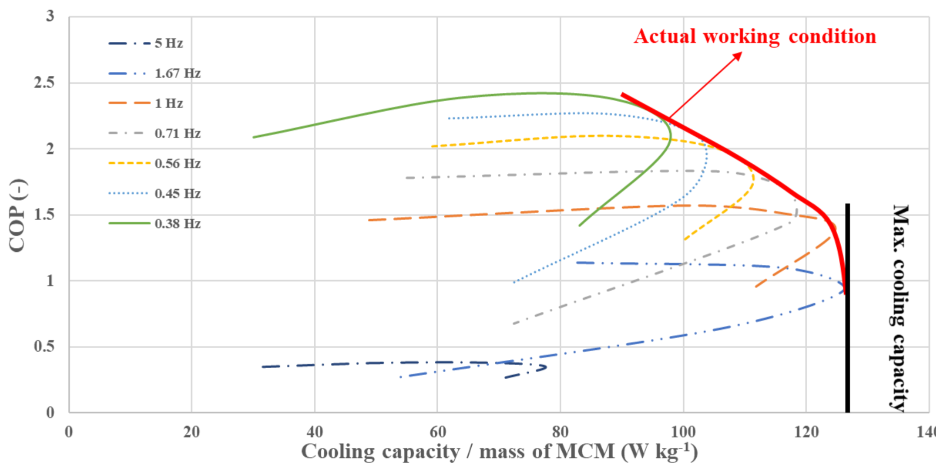
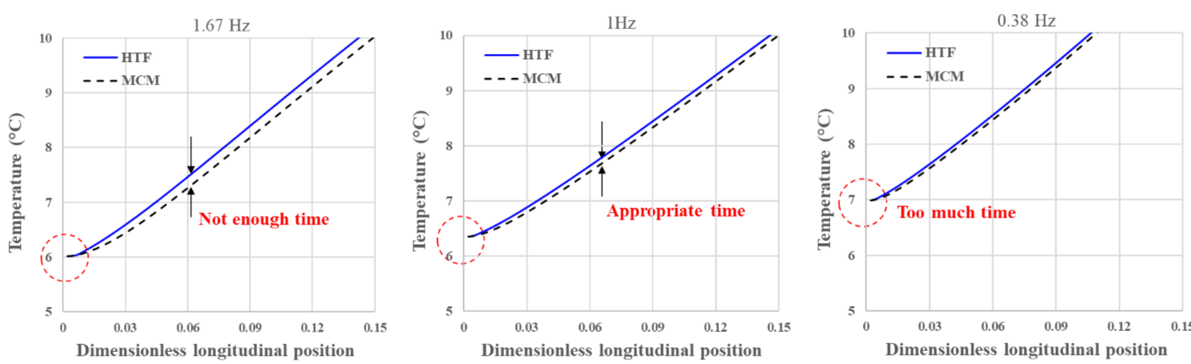


Figure 11: COP versus cooling capacity for various frequency



(a) (b) (c)

Figure 12: Temperature gradient of MCM and HTF at the end of hot to cold blow of (a) 1.67Hz (b) 1Hz (c) 0.38Hz (using different mass flux for each frequency to generate maximum cooling capacity)

3.5 Cycle frequency

Figure 10 and figure 11 show cooling capacity versus mass flux and COP versus cooling capacity for various frequency, respectively. The frequency sensitively affects cooling capacity and system efficiency. If cycle time is too long, MCM and HTF already are in thermal equilibrium at the end of blow period and system uselessly spends pump power, because there is no heat transfer between MCM and HTF. Also, hot fluid enters cold side of regenerator and decreases temperature of cold side during hot to cold blow, thereby decreasing temperature difference between cold and hot side of regenerator and cooling capacity, and vice versa. If cycle time is too short, there is no enough time for heat transfer between MCM and HTF. Therefore, it could decrease temperature difference between cold and hot side of regenerator and cooling capacity. Also, COP could decrease because magnetic power cannot be used efficiently.

When frequency increases (cycle time decreases), the optimal mass flux to generate maximum cooling capacity increases as shown in Figure 10, which is related to displacement ratio. When frequency increases, maximum cooling capacity increases because only coldest fluid can enter into the cold side heat exchanger. If frequency further increases, time is limited for heat transfer and temperature of HTF cannot be decreased enough. The COP to generate maximum cooling capacity increases with an increase in cycle time. The system with frequency of 0.45Hz can generate 100W kg^{-1} cooling capacity with COP of 2.25. Based on figure 11, the AMRRC cycle can be controlled using system frequency. It is difficult to change length of the regenerator while system is operating, but the AMRRC system can easily change the cycle frequency to get optimized COP at the required cooling capacity.

Figure 12 represents temperature profiles of MCM and HTF in the regenerator at the end of hot to cold blow for cycle frequency of 1.67Hz, 1Hz and 0.38Hz. From these graphs, it can be noticed that 1.67Hz cycle frequency does not have enough time to transfer heat between HTF and MCM so they are not in thermal equilibrium at at the end of hot to cold blow and only very cold fluid neat the cold side of the regenerator can enter the CHX. Therefore, the system with 1.67Hz can generate higher cooling capacity but lower COP, because it cannot efficiently use caloric effect. However, vice versa for 0.38Hz cycle frequency because it has too much time for heat transfer between MCM and HTF.

3.6 Porosity

Figure 13 and 14 show the temperature gradient of MCM and HTF at the end of hot to cold blow and total heat transfer between MCM and HTF per mass of MCM during one blow period, respectively. Porosity affects specific surface area (=heat transfer area / volume) and magnetocaloric effect. As porosity increases, specific surface area increases

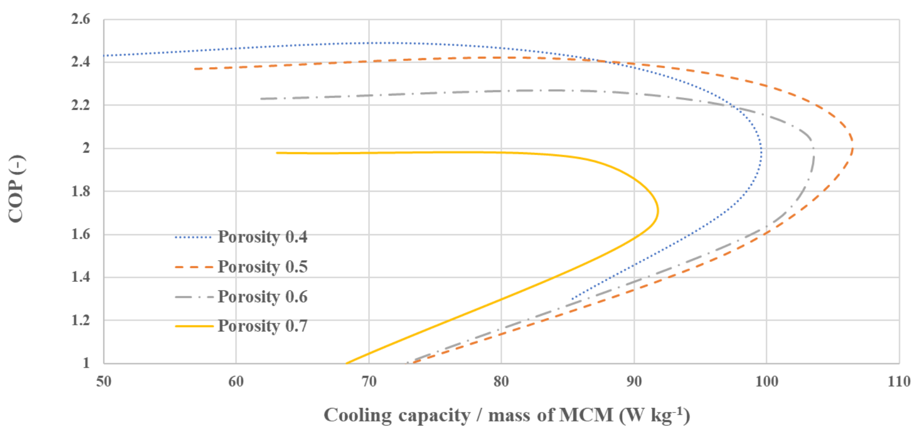


Figure 13: Temperature gradient of MCM and HTF at the end of hot to cold blow

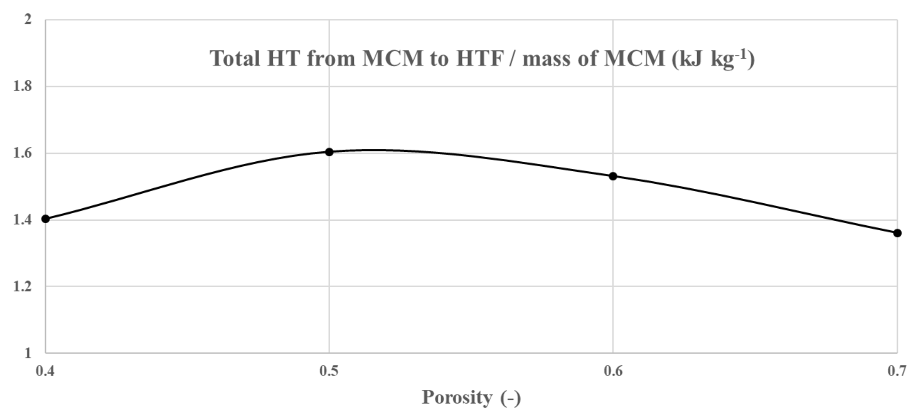


Figure 14: Total heat transfer between MCM and HTF per mass of MCM during one blow period

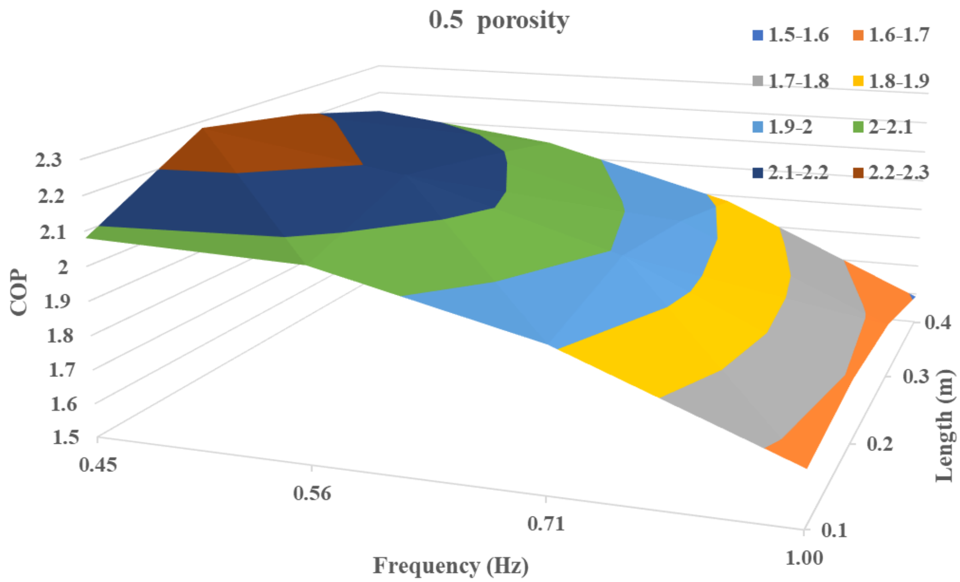


Figure 15: COP along cycle frequency and length of regenerator at 0.5 porosity (100 W kg^{-1})

but caloric effect decreases due to a decrease in mass of MCM. As porosity increases total heat transfer between MCM and HTF per mass of MCM increases due to the increase in specific area. However, total heat transfer decrease after 0.5 porosity because the caloric effect decreases. Therefore, Regenerator with porosity of 0.5 has biggest heat transfer from MCM to HTF per mass of MCM and the highest COP to generate 100W kg^{-1}.

Figure 15 shows COPs as a function of cycle frequency and length of regenerator at 0.5 porosity and the regenerator with 200mm length, 0.45Hz cycle frequency and 0.5 porosity has the highest COP of 2.28 to generate 100W kg^{-1} cooling capacity under the temperature span of 20˚C (5 ~ 25).

4. CONCLUSIONS

This paper has incorporated heat transfer coefficient of Nickolay and Martin correlation (2002) and friction factor of Shah correlation (1974) to take into account developing region. This paper has used these equations for not only entry developing region but also the developing region after changing flow direction. There is no paper which consider both developing regions. The model including the developing regions has similar heat transfer coefficient and friction factor to those of the model neglecting them when cycle frequency is low. However, as frequency increases the difference between both models increases. When cycle frequency is 5Hz, the Nu of former model is 9.3% higher than the Nu of latter model. Pressure drop increases by 5.4% when using new model. Therefore, when system has high cycle frequency, both developing regions should be considered. This is meaning result because the performance of solid-state caloric cooling system is dependent on heat transfer coefficient. Moreover, as heat transfer coefficient increases, optimal cycle frequency increases.

As mass flux increases, cooling capacity, heating capacity and COP increase and then decrease again. This is because when mass flux increases at the first time, heat transfer increases. Therefore, cooling and heating capacity increase. However, when mass flux increases further, cold fluid enters into hot side and hot fluid enters into cold side. Therefore, cooling and heating capacity decrease again. Furthermore, the reason of COP change trend is the change rate of thermal conductivity of water. As length of the regenerator increase, optimal mass flux to reach to the optimal displacement ratio increases and temperature gradient rate decreases. Therefore, conduction loss decreases and dissipation loss increases. Finally, the regenerator with 200mm has the smallest total loss and the most efficient to generate 100W kg^{-1} of MCM. However, the length of the regenerator has little effect on system performance. It is noticed that the cycle frequency has significant effect on cooling capacity and COP. This is because the cycle frequency affects the amount of heat transfer fluid entering the regenerator, heat transfer between fluid and solid and outlet temperature of fluid. Especially, high frequency does not allow enough time for heat transfer and temperature gradient in the regenerator is decreased at low frequency. Therefore, it is necessary to find appropriate cycle frequency to increase system efficiency. It is also noticed that cycle frequency can be easily controlled to get the required cooling capacity and optimized COP. As porosity increases specific heat transfer area increases but total magnetocaloric effect decreases. Therefore, porosity should be optimized to increase system efficiency. For plate type of regenerator with 0.3mm hydraulic diameter, the regenerator with 200mm length, 0.45Hz cycle frequency and 0.5 porosity has highest COP of 2.3 to generate 100W kg^{-1} cooling capacity under the temperature span of 20˚C. Finally, by applying univariate method this regenerator has been verified as optimal point.

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