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Applying Two-phase Zeotropic Heat Transfer Fluids to Solid-state Cooling Cycle and Comparison with Baseline using Single Phase Water

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Applying two-phase zeotropic heat transfer fluids to solid-state cooling cycle and comparison with baseline using single phase water

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ABSTRACT

Solid-state caloric cooling cycles are considered to be a promising alternative to conventional cooling technology. Theoretically, they can compete with the conventional vapor compression cycle and are much more efficient. However, this technology does have one problem: its performance decreases rapidly if the temperature difference between the condenser and evaporator gets larger. The environment where cooling technology is used often has large temperature differences between condenser and evaporator. For instance, in a typical automotive system, the condenser operates at 45°C and the evaporator at 5°C. Thus, the temperature difference between the condenser and evaporator reaches 40°C. The cooling load of the caloric cycle decreases by 86% when the temperature difference increases from 5°C to 20°C. This weakness prevents the magnetic refrigeration cycle from becoming commercialized and competing with vapor compression cycle.
To solve this issue, the authors have applied two-phase zeotropic blends as heat transfer fluid in the caloric cycles. There are two reasons. First, using two-phase flow can increase the Nusselt number. The flow is laminar when the system uses single phase water, so the Nusselt number cannot be improved in the regenerator. Second, two-phase zeotropic mixture can generate a temperature gradient in the regenerator to overcome the large temperature difference. Therefore, an ethane/isobutane mixture has been chosen due to its high temperature glide (~ 40°C). This selection is different from that of the vapor compression cycle which normally prefers small glide zeotropic mixture.
This paper shows that using two-phase zeotropic mixture through the entire regenerator has much better performance than using single and two-phase flow together in the caloric cooling cycle. Optimized zeotropic refrigerant system was chosen to compare with a water system. As a result, the system using two-phase zeotropic blend can increase temperature difference between cold and hot side of the regenerator because Nusselt number increases and lower mass flow rate is required to reach the optimal displacement ratio. Therefore, the amount of heat transfer in the cold heat exchanger significantly increases and the system's efficiency ultimately gets much better.
This is first time to use two-phase flow zeotropic mixture in the caloric refrigeration system and this paper expands the application of two-phase zeotropic blends to caloric cooling systems.

1. INTRODUCTION

Solid-state caloric cycles are the most promising technologies to replace current cooling cycles because this advanced technology can overcome the various environmental issues air conditioning systems cause. The technology does not use the refrigerant that causes ozone depletion and global warming effects, which are the biggest problems occurred in the vapor compression cycle. Furthermore, it started getting recognitions by many researchers because of its high efficiency. For the past few decades, numerous researchers and companies have been actively developing solid-state cooling technology. However, the study on heat transfer fluids used in caloric regenerators is somehow neglected, even though heat transfer phenomena in the regenerator is important to determine system performance. Heat transfer fluid has a great effect on the performance of solid-state caloric cycles. In general, high thermal conductivity and thermal diffusivity and low viscosity of the fluid generates good cooling performance (Kitanovski, 2015).
Most of the magnetic regenerator prototypes has used water or water with alcohol additives as a heat transfer fluid because it has good thermohydraulic properties and characteristic of nontoxicity and simplicity of use. Petersen (2008) numerically and experimentally compared water, ethanol, glycerin and mercury. Mercury has the highest temperature span in regenerator because of its thermal conductivity. Wu *et al.* (2014) analyzed the impact different heat transfer

fluids in active magnetic refrigeration. This paper shows mercury can increase the cooling capacity by 600% compared to water and a fluid with high conductivity, high density and low specific heat is suitable as heat transfer fluid in active magnetic refrigeration. Kitanovski *et al.* (2012) showed that Galinstan is more efficient coolant fluid compared to water. These papers showed that liquid metals are most suitable as a heat transfer fluid but it is highly toxic. Among other fluids pure water is the best fluid for solid-state caloric cycles.

This paper has applied different working fluids to a solid-state caloric cooling system and compares the performance achieved to a baseline system using water. When single phase water is used as heat transfer fluid in the solid-state caloric cooling system, the flow is found to be laminar. Therefore, the heat transfer coefficient cannot be improved for a fixed regenerator hydraulic diameter, because the Nusselt number is constant for fully developed laminar flow. However, when a zeotropic blend is used as the heat transfer fluid, it is expected that Nusselt number increases and temperature gradient appears in the regenerator. Therefore, an ethane/isobutane mixture (50/50% by mass) has been chosen as heat transfer fluid due to its high temperature glide during condensation and evaporation. Unlike in conventional vapor compression systems that aim to use blends with low temperature glides, the higher glides in caloric system can be helpful in increasing the temperature lift of the system

2. NUMERICAL MODEL

2.1 Schematic of solid-state caloric cooling cycle

Figure 1 represents the schematic of the model for solid-state caloric cooling cycles. The system comprises a permanent magnet, a regenerator and two heat exchangers. The regenerator is filled with circular type of MCM through which heat transfer fluid (water or ethane/isobutane mixture) 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.

2.1 Correlations of heat transfer coefficient and pressure drop in circular tube

Extended Gnielinski's correlation has been used to calculate heat transfer coefficient for single phase flow (Abraham *et al.*, 2011). Chen's correlation (1966) and Cavallini's correlation (2006) have been applied into model to calculate the heat transfer coefficient of evaporation and condensation, respectively, in this paper. Bell and Ghaly developed below approximate correlation of condensation heat transfer for mixture to consider mass transfer resistance due to zeotropic refrigerant(1973). However, this correlation has been used for evaporation heat transfer for mixture. Therefore, Bell & Ghaly correlation has been used in the model to calculate the heat transfer coefficient of evaporation and condensation of mixture for simplification of model and computational speed. Homogeneous model (McAdams *et al.*, 1942) has calculated the pressure drop of mixture in the model.

$$\frac{1}{\alpha} = \frac{1}{\alpha_f} + \frac{Z_G}{\alpha_G} \tag{1}$$

α_f: Coefficient of the condensation layer film
α_G: Vapor heat transfer coefficient
$Z_G = \frac{q_{sen,G}}{q}$: Ratio of the local vapor sensible heat flux to the local total heat flux

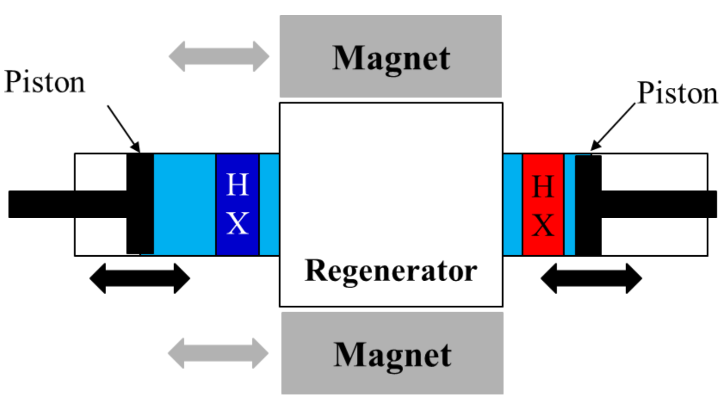

Figure 1: Schematic of modeled solid-state caloric cooling cycle

3. RESULTS AND DISCUSSION

3.1 Mass flow rate

Table 1 shows operating conditions. Figure 2 shows COP and cooling and heating capacity along mass flux. General trend of the cooling capacity, heating capacity and COP are similar to those of AMRRC using water. The COP decreases with the increase in mass flux. The cooling and heating capacity increase with the increase in mass flux. However, when mass flux is about 800 kg s^{-1} m^{-2}, cooling and heating capacity is maximum and after mass flux reaches 800 kg s^{-1} m^{-2}, cooling and heating capacity decrease. This is mainly because when mass flux is low, cold fluid near the cold side of the regenerator or hot fluid near the hot side of the regenerator only enter the cold heat exchanger or hot heat exchanger, respectively. When mass flux increases, colder fluid or hotter fluid enter the cold heat exchanger or hot heat exchanger, respectively, with high velocity, so the cooling and heating capacity increase. The other reason is the increase in heat transfer coefficient as mass flux increases. However, if mass flux more increases, relatively hot fluid finally enters the cold heat exchanger and cold fluid enters the hot heat exchanger. Therefore, the cooling and heating capacity decrease. 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. In this graph, heating capacity increases faster than cooling capacity when mass flux is low. This is because refrigerant in hot side is two-phase so its heat transfer coefficient linearly increases with mass flux, but refrigerant in cold side is single phase so its heat transfer coefficient stays constant with mass flux, when mass flux is low as shown in Figure 3.

Figure 3 represents average heat transfer coefficients of single phase and two-phase flow versus mass flux. The heat transfer coefficient of two-phase flow increases with the increase in mass flux, but the slope gradually decreases. The heat transfer coefficient of single-phase flow is almost constant when the Reynolds number is less than 2300. The heat transfer coefficient significantly increases after the Reynolds number reaches 2300 because the flow turns from laminar flow into transition flow. Average heat transfer coefficient and pressure drop per unit length versus mass flux calculated from the entire regenerator are shown in figure 4. In this graph, the pressure drop per unit length exponentially increases. However, the slope linearly increases the final region (800 ~ 1000 kg s^{-1} m^{-2}). It is because the temperature of heat transfer fluid in the hot side of regenerator decreases compared to that of other regions (100 ~ 800 kg s^{-1} m^{-2}). The changing trend of the average heat transfer coefficient of two-phase flow in Figure 3 is similar to that of the average heat transfer coefficients in Figure 4 because over 90% of heat transfer fluid in regenerator is two-phase flow. The average heat transfer coefficient flow slightly more increases when mass flux is between 700 ~ 800 kg s^{-1} m^{-2} compared to other regions. This is because of the effect of the increase in heat transfer coefficient in single flow region.

Table 1: Operating conditions

Total volume	3.3L
Temperature span	20°C (5 – 25 °C)
frequency	5Hz
Porosity	0.5
Hydraulic diameter	0.0003m

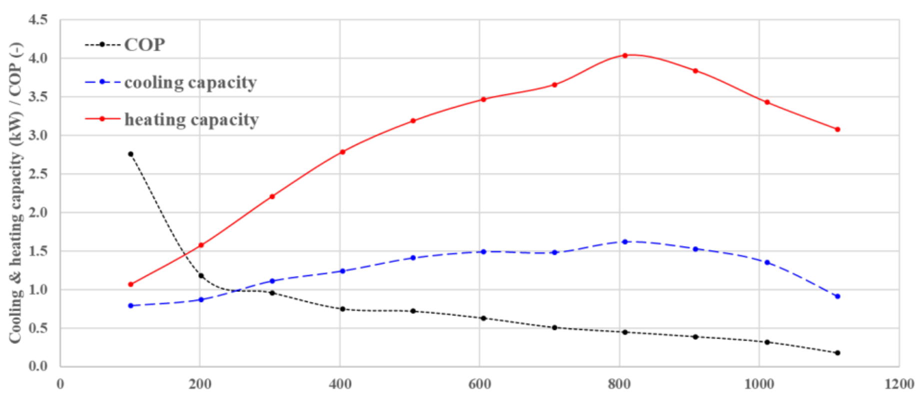

Figure 2: COP, cooling & heating capacity versus mass flux

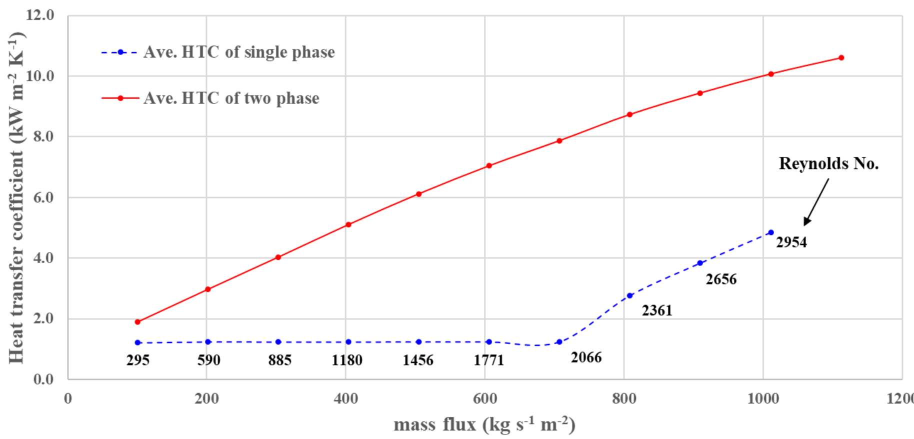

Figure 3: Average heat transfer coefficients of single phase and two-phase flow versus mass flux

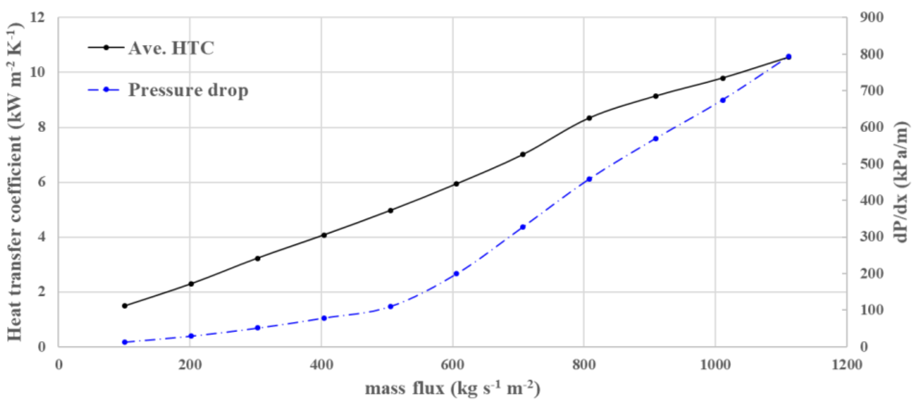

Figure 4: Average heat transfer coefficients and pressure drop per unit length versus mass flux

3.2 Aspect ratio and frequency

Figure 5 represents the effect of aspect ratio on cooling capacity. In the graph, as aspect ratio increases maximum cooling capacity increases. This is because the regenerator with higher aspect ratio needs higher mass flux to reach the optimal displacement ratio, which causes an increase in heat transfer coefficient. Therefore, the regenerator with higher aspect ratio can generate higher cooling capacity compared to the regenerator with lower aspect ratio. Figure 6 shows the COP versus cooling capacity for various aspect ratio. There is the optimal aspect ratio which can generate certain cooling capacity. (e.g. Aspect ratio of 4 is optimal to generator cooling capacity of 0.87 ~ 1.17 kW.) In this condition, aspect ratio of 8 is optimal to generate the cooling capacity of 1.3kW (100W/1kg of MCM).

Figure 7 and Figure 8 show cooling capacity versus mass flux and COP versus cooling capacity for various frequency,

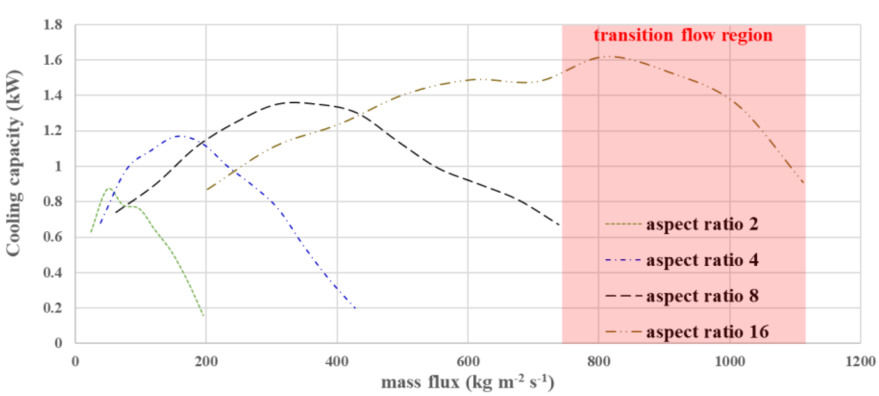

Figure 5: Cooling capacity versus mass flux for various aspect ratio

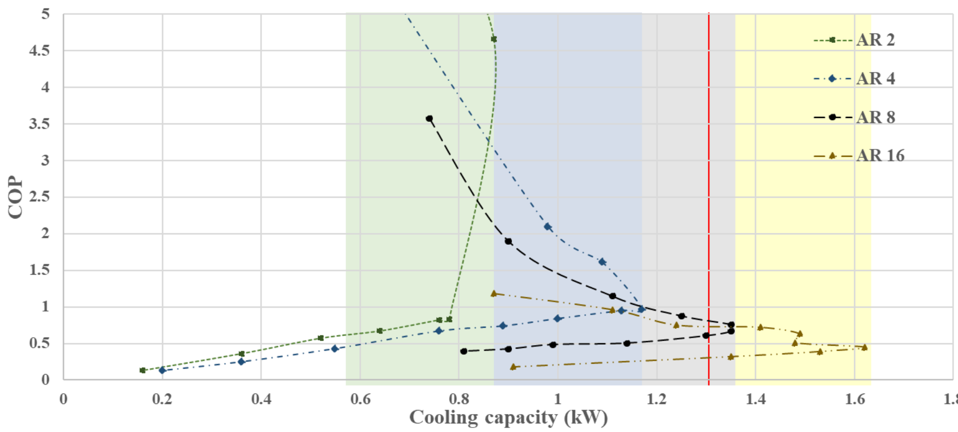

Figure 6: COP versus cooling capacity for various aspect ratio

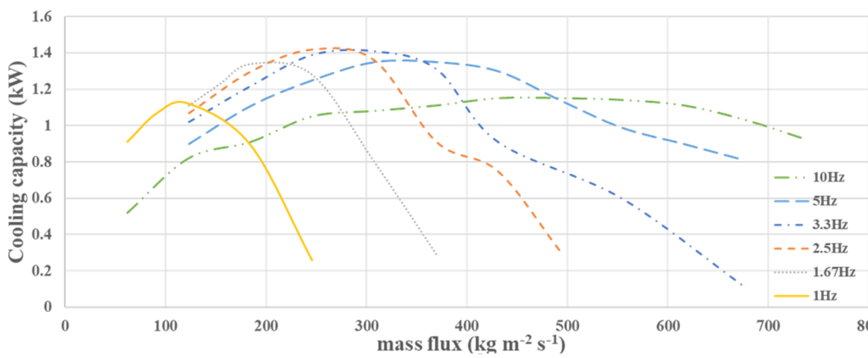

Figure 7: Cooling capacity versus mass flux for various frequency

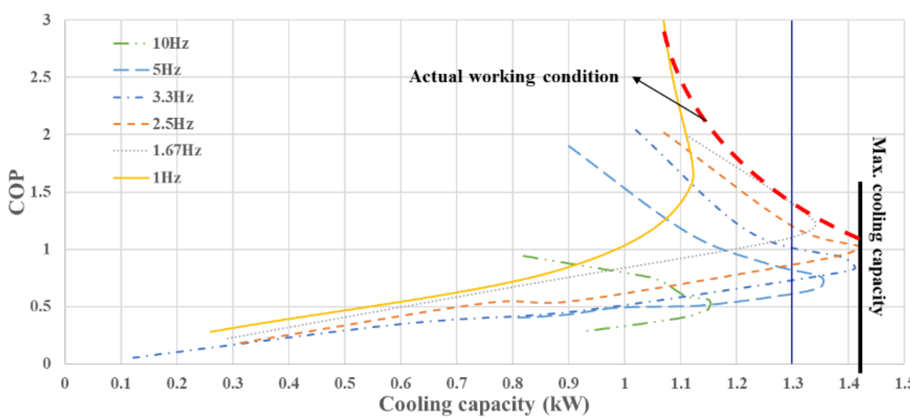

Figure 8: COP versus cooling capacity for various frequency

respectively. When frequency increases (cycle time decreases), the optimal mass flux to generate maximum cooling capacity increases as shown in figure 7, 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 1.67Hz can generate 1.3kW cooling capacity with COP of 1.4.

3.3 Effect of pressure and composition on system performance

Table 2 represents boiling point, dew point and temperature glide as a function of operating pressure and mass fraction of ethane calculated by REFPROP program. Combination of operating pressure and composition of mixture

Table 2: Boiling point, dew point and temperature glide as a function of operating pressure and mass fraction of ethane

Pressure	1300 kPa			1500 kPa			1700 kPa		
mass fraction of ethane	boiling point (°C)	dew point (°C)	temperature glide (°C)	boiling point (°C)	dew point (°C)	temperature glide (°C)	boiling point (°C)	dew point (°C)	temperature glide (°C)
0.1	47.4	69	21.6	55	75.5	20.5	61.9	81.4	19.5
0.2	25.9	60.4	34.5	33.3	66.5	33.2	40	72	32
0.3	**11.6**	**52.1**	**40.5**	18.5	57.9	39.4	24.9	63	38.1
0.4	1.7	44	42.3	8.2	49.5	41.3	**14.2**	**54.4**	**40.2**
0.5	-5.3	35.9	41.2	**0.8**	**41.1**	**40.3**	6.5	45.7	39.2
0.6	-10.6	27.5	38.1	-4.8	32.4	37.2	0.6	36.8	36.2
0.7	-14.8	18.4	33.2	-9.1	23.1	32.2	-4	27.3	31.3
0.8	-18.1	8.2	26.3	-12.7	12.7	25.4	-7.7	16.8	24.5
0.9	-20.9	-4.5	16.4	-15.6	0	15.6	-10.8	4	14.8

determines boiling point, dew point and temperature glide. Based on these results, we can get zeotropic mixtures with different boiling points and same temperature glide or zeotropic mixtures with different temperature glide and same boiling point by controlling the combination. These characteristics are very important in zeotropic magnetocaloric refrigeration system because they can change the ratio of single phase and 2-phase of mixture and quality of 2-phase flow in regenerator as shown in Figure 9. The change could affect heat transfer coefficient and pressure drop of 2-phase flow as well as the amount of energy that must be added to the heat transfer fluid in order to increase 1°C. Therefore, it has a significant effect on system performance.

Firstly, to simplify model and see the effect of changing boiling temperature, boiling temperature of zeotropic mixture has been changed at the fixed temperature glide. Also, properties, such as internal energy, enthalpy, density, specific heat, etc, change together with boiling temperature. For example, if boiling temperature increases by 5°C, properties move to right side by 5°C as shown in Figure 10. By doing this, the only effect of boiling point on system performance can be investigated.

Figure 11 shows cooling capacity versus mass flux and Figure 12 represents COP versus cooling capacity for different boiling point. As boiling temperature decreases, cooling capacity and COP increases. When boiling temperature is -5 °C, the COP of system reaches almost 5.9 to generate 1.3kW cooling capacity. This is because an increase in 2-phase region causes an increase in average HTC and using higher quality region causes an increase in average HTC. Lastly, higher heat capacity of 2-phase causes higher heat transfer between HTC and MCM during blow periods.

The initial and final temperatures of HTF and MCM during hot to cold blow period for 0°C and -5°C boiling points are shown in Figure 13 (mass flow rate = 0.2kg s^{-1}). In case of the system using the zeotropic mixture with 0°C boiling point, single and 2-phase flow of mixture exist together in regenerator. There are single phase flow in cold side of regenerator and 2-phase flow in hot side of regenerator. MCM and HTF in cold side of regenerator are in thermal equilibrium state at the final of hot to cold blow period, which means that there is no heat transfer between MCM and HTF and MCM cannot cool down HTF further over long period of time. On the other hand, in case of the system using the zeotropic mixture with -5°C boiling point, only 2-phase flow of mixture exists in regenerator. Therefore, average heat transfer coefficient of HTF with -5°C boiling point is higher than average heat transfer coefficient of

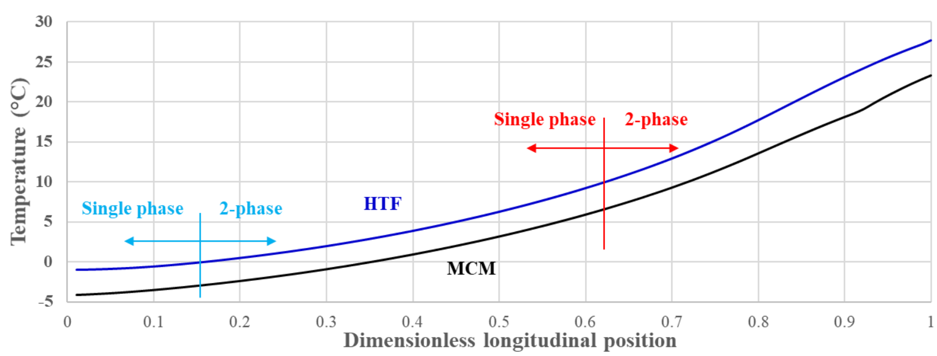

Figure 9: Ratio change of single phase to 2-phase of zeotropic mixture in regenerator under different boiling point (0°C (Green), 5°C (Red))

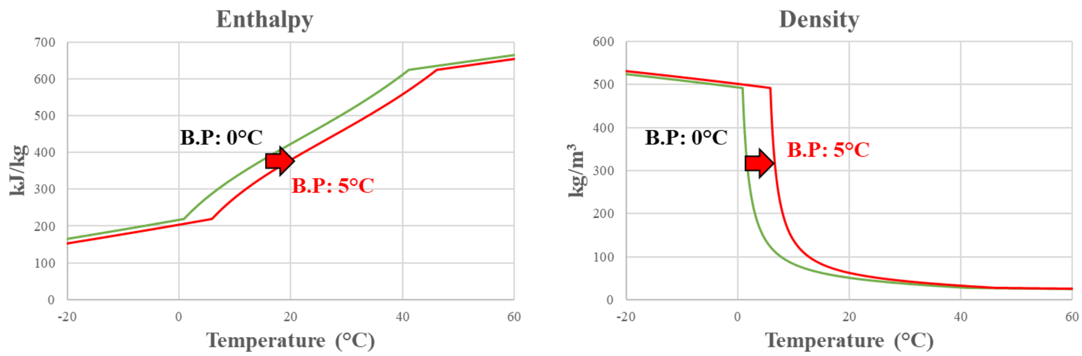

Figure 10: Enthalpy and density change under different boiling point (0°C (Green), 5°C (Red))

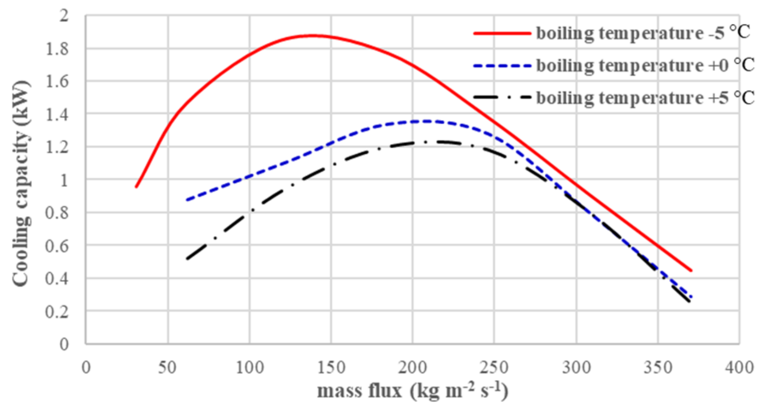

Figure 11: Cooling capacity versus mass flux for different boiling point

HTF with 0°C boiling point. In addition, for -5°C boiling point, temperatures of MCM and HTF in cold side of regenerator is just about to meet at the final of hot to cold blow period, which means that heat transfer between MCM and HTF occurs during the whole period of hot to cold blow so heat transfer rate increases. This is mainly because 2-phase of zeotropic needs more energy to increase its temperature.

The initial and final temperatures of HTF and MCM during cold to hot blow period for 0°C and -5°C boiling points are shown in Figure 14 (mass flow rate = 0.2kg s^{-1}). In case of the system using the zeotropic mixture with 0°C boiling point, the temperature change of HTF is similar to that of MCM during cold to hot blow period. On the other hand, the temperature change of HTF in cold side of regenerator is less than that of MCM during cold to hot blow period in the system using the zeotropic mixture with -5°C boiling point, because more energy is needed to increase temperature of 2-phase flow due to latent heat. Therefore, MCM and HTF are in thermal equilibrium at lower temperature at the final of cold to hot blow period and MCM has lower temperature after demagnetization process than that of MCM in system using the zeotropic mixture with 0°C boiling point.

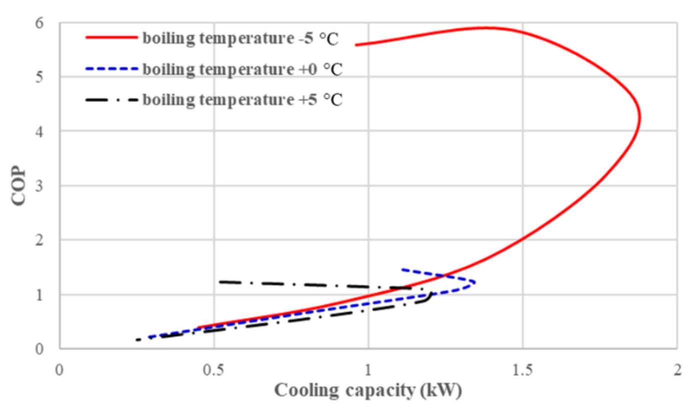

Figure 12: COP versus cooling capacity for different boiling point

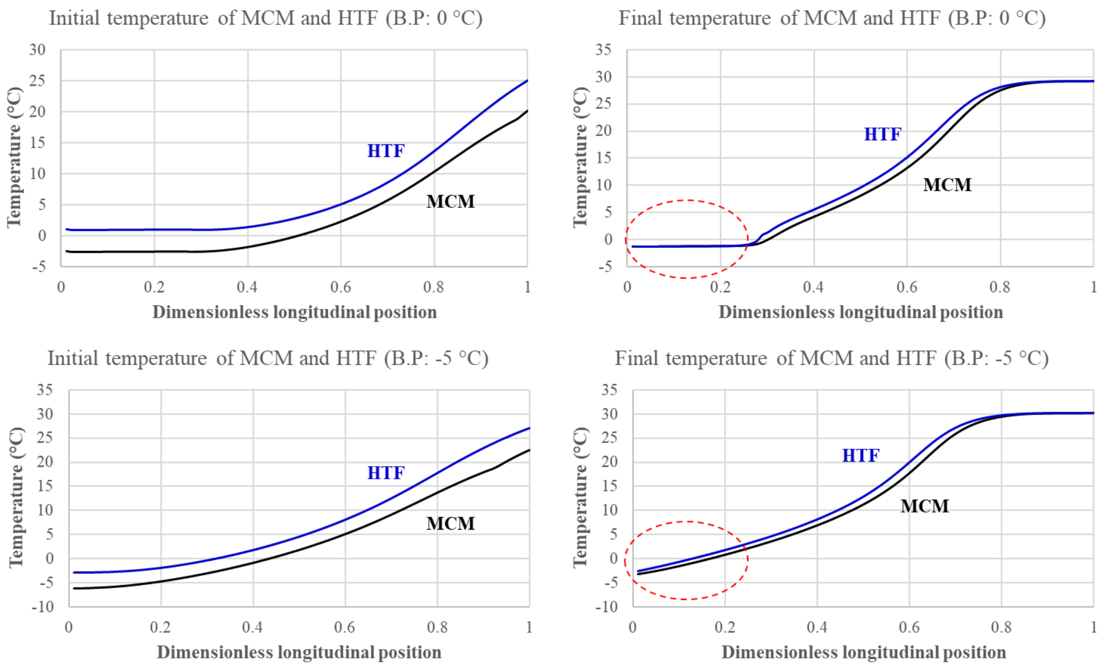

Figure 13: Initial and final temperatures of HTF and MCM during hot to cold blow period for 0°C and -5°C boiling points (mass flow rate = 0.2kg s^{-1})

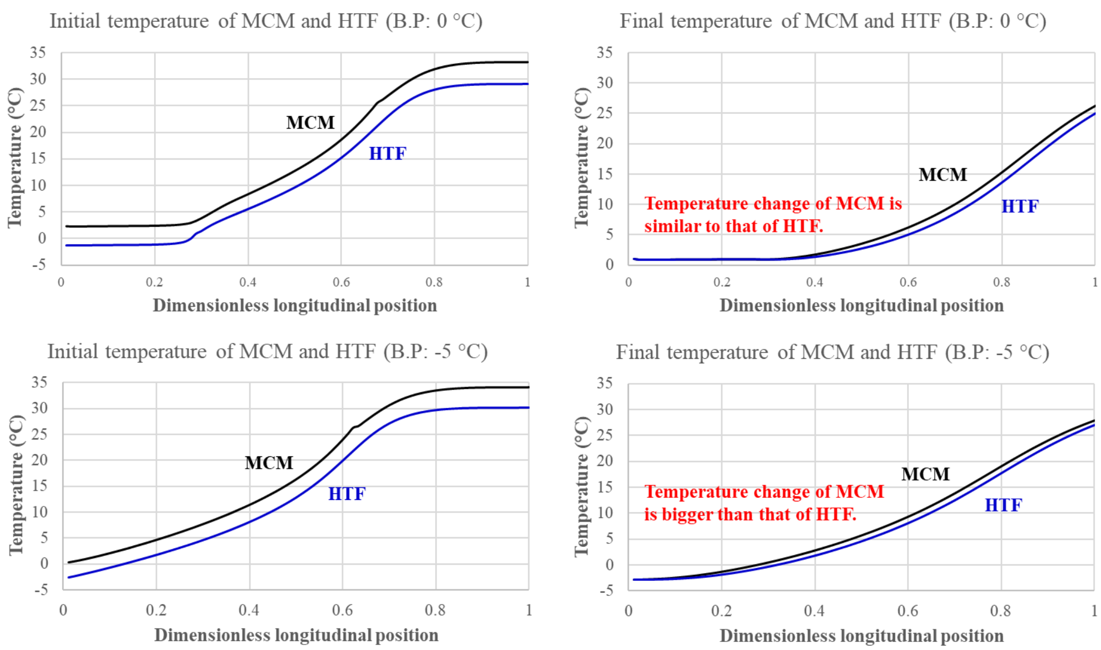

Figure 14: Initial and final temperatures of HTF and MCM during cold to hot blow period for 0°C and -5°C boiling points (mass flow rate = 0.2kg s^{-1})

In conclusion, using only two-phase flow in regenerator causes an increase in heat transfer between MCM and HTF in cold side of regenerator during hot to cold blow period and a decrease in the temperature of MCM at the final of cold to hot blow period. In particular, this effect mostly concentrates on the cold side of regenerator which leads to a decrease in temperature of cold reservoir tank by 2.5°C though the temperature of hot reservoir is increased by 1.1°C. This is the reason why COP significantly increases when using the zeotropic mixture with -5°C boiling point.

Table 3: Optimized conditions and system performance for the AMRRC using zeotropic mixture and water

	Water system	Zeotropic system
Aspect ratio (-)	2	8
Frequency (Hz)	1	1.67
Mass flow rate (kg s^{-1})	0.75	0.15
Nu (-)	3.6	8.6
Mass flux (kg m^{-2} s^{-1})	184	92
Cooling capacity (kW)	1.43	1.75
COP (-)	4.7	6.3
Pressure drop (kPa)	8.3	12.5
Temperature of cold / hot reservoirs (°C)	-1.5 / 29.6	-3.2 / 30.9

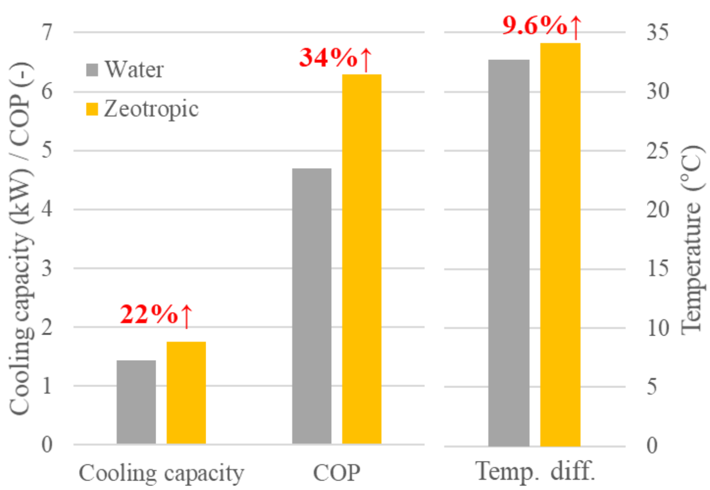

Figure 15: System performance comparison of zeotropic system with water system

3.4 Comparison of the AMRRC using zeotropic mixture with the AMRRC using single phase water

To compare the AMRRC using zeotropic mixture with the AMRRC using single phase water, optimized conditions are chosen for both cases. Optimized conditions and system performance for both cases are shown in Table 3. Figure 15 represents system performance comparison of zeotropic system with water system.

Using zeotropic mixture in caloric system as HTF increases Nu 2.4 times and temperature difference between hot and cold reservoir tanks by 9.6%. Cooling capacity and COP are increased by 22 and 34%, simultaneously. This is mainly because zeotropic system needs lower mass flow rate compared to water system to reach the optimal displacement ratio due to the difference of density between both HTFs and optimized aspect ratio. So, smaller heat capacity of HTF enters and exits regenerator during one blow period in zeotropic system. Total heat transfer between MCM and HTF can further increase and decrease the temperature of HTF during hot and cold blow periods, respectively, in zeotropic system. Therefore, this effect can increase temperature difference of HTF between cold and hot side in regenerator, cooling capacity and COP.

4. CONCLUSIONS

An ethane/isobutane zeotropic mixture (50/50% by mass) has been chosen as heat transfer fluid due to its high temperature glide during condensation and evaporation. Unlike in conventional vapor compression systems that aim to use blends with low temperature glides, the higher glides in caloric system can be helpful in increasing the temperature lift of the system.

The COP of the optimized system with 8 aspect ratio, 1.67 Hz and -5°C boiling point is almost 5.9 to generate 1.3kW cooling capacity. This optimized zeotropic system has been compared with the optimized water system. In conclusion, using the zeotropic mixture increases the Nusselt number 2.4 times and the temperature difference between hot and cold reservoir tanks by 9.6%. In return, cooling capacity and COP increased by simultaneously by 22 and 34%.

No prior study has described the application of two-phase zeotropic blends to a caloric refrigeration system before. The results are intriguing and demonstrate that it is possible to expand the application of caloric cooling systems due to the use of high-glide zeotropic mixture.

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