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A new ideal evaporative freezing cycle

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Abstract

A new ideal evaporative freezing cycle for freezing of water is proposed and analyzed by using the conservation of energy and the conservation of mass principles. The proposed cycle utilizes low temperature heat sources such as solar energy, geothermal energy, and waste heat, and consists of a freezing chamber, an air-to-air heat exchanger, a desiccant chamber, an air-to-water heat exchanger, and a fan through which air circulates at atmospheric pressure. The operating principles of the cycle is based on the fact that as dry air picks up moisture from water, the water vapor absorbs heat primarily from the remaining body of the water, and thus the water is cooled and frozen. It is shown that the proposed system can produce 28.4 g ice/kg dry air circulated at most and have a thermal coefficient of performance up to 0.47. The proposed evaporative freezing cycle offers a viable alternative to the conventional refrigeration methods and provides refrigeration by using the inexpensive source of thermal energy source. Also, various aspects of the cycle proposed is discussed.

Keywords: Freezing; Water; Refrigeration; Evaporative; Desiccant; Cycle

1. Introduction

It is well-known that water can be frozen by evacuating the chamber it is in, and thus lowering the vapor pressure below the saturation pressure of water at the freezing temperature. Although this is one of the earliest methods used to produce ice, it requires careful sealing, the use of electric power to drive the vacuum pump, and a container that can withstand the force caused by the pressure difference between inside and outside the chamber. All of these problems can be avoided by lowering the vapor pressure in the chamber by utilizing evaporative cooling coupled with a desiccant section.

Evaporative coolers are commonly used in areas with dry climates in place of air-conditioners because of their lower installation and energy costs [1,2]. An evaporative cooler uses about one-fourth of the electricity that a vapor-compression air-conditioning system uses. The use of evaporative coolers has been extended to humid climates by the incorporation of a desiccant chamber to dehumidify the air before passing it through the evaporation section. It has been shown that such systems can reduce the air-conditioning costs significantly, especially when solar energy is used to regenerate the desiccants [3–5]. In this work we propose to extend the concept of evaporative cooling in conjunction with desiccants to the freezing of water. It is expected to realize similar energy efficiency during freezing of water [6,7].

There is a one-to-one correspondence between the saturation temperature and saturation pressure of water, and thus water can be brought to a saturation state by simply lowering its saturation pressure. For example, the saturation pressure of water is 0.87 kPa at 5 °C, and it drops to 0.40 kPa at −5 °C [8]. Therefore, water can be cooled to 5 °C by lowering its vapor pressure to 0.87 kPa, and it can even be frozen to −5 °C by lowering its vapor pressure further to 0.40 kPa. One way of reducing the vapor pressure in a water container is to evacuate the container using a vacuum pump. Vacuum cooling has long been used to cool leafy vegetables soon after harvesting. Making ice by using a vacuum pump is nothing

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Nomenclature

$\dot{m}$	flow rate, kg/s
h	enthalpy, kJ/kg
Q	heat, kJ/kg
COP	coefficient of performance

Subscripts

a	dry air
v	water vapor
i	inlet
e	exit
L	low temperature
f	liquid
g	gas
if	latent heat of fusion
w	water
reg	regeneration
fg	latent heat of vaporization

Greek symbols

ω	specific humidity, kg H_2O/kg dry air
ϕ	relative humidity

new, either. Dr. William Cullen has made ice in Scotland in 1775 by evacuating the air in a water tank.

The heat of vaporization of water is 2501 kJ/kg at 0 °C, and 2442 kJ/kg at 25 °C whereas the heat of fusion (or freezing) of water is 334 kJ/kg [9]. Therefore, about 7 kg of water can be frozen by evaporating only 1 kg of water.

Although technologically sound, vacuum freezing is not energy efficient, and thus it is not a viable method of freezing of water. An alternative way to reduce the vapor pressure and thus to freeze water is to expose the water to air whose vapor pressure is below 0.61 kPa, the saturation pressure of water at 0 °C. This can be done by passing the air first through a desiccant chamber and drying it.

The evaporative freezing cycle proposed in this study can eliminate the disadvantages associated with vacuum freezing systems. First of all, the cycle operates at atmospheric pressure, and thus all the problems related to maintaining a vacuum are avoided. An exhaustive survey of the pertinent literature has failed to indicate any prior work on evaporative freezing systems. Therefore, it is believed that the cycle is a novel concept.

2. Conceptual design

The operating principle of the proposed evaporative freezing system is similar to the operating principles of evaporative or swamp coolers commonly used in dry climates, and the desiccant-coupled evaporative cooling systems used in humid climates. But in the case of evaporative freezing, the objective is to cool the water instead of the air. The driving force for evaporation is the difference between the saturation pressure of water at the water temperature and the vapor pressure of air. Therefore, the drier the air, the higher the rate of evaporation of water. Also, the lower the temperature of air, the higher the fraction of heat that will come from the water.

A schematic of the proposed evaporative freezing system is given in Fig. 1. The proposed system consists of an insulated freezing chamber where water is placed, an air-to-air heat exchanger (called the regenerator) to cool the incoming air, a desiccant chamber to remove the moisture in the air, an air-to-water heat exchanger to cool the air heated in the desiccant chamber, a fan to move the air in the system, and the connections between the components. As can be seen from the figure, this is an open system and air at atmospheric pressure is recirculated through the different components until the desired result is achieved.

The water to be frozen is placed in a freezing chamber that is well-insulated. The heavy insulation is to ensure that heat transmission to the chamber through its envelope is minimal so that the heat of vaporization originates primarily from the water, resulting in a drop of water temperature. Of course part of the heat of vaporization comes from the air in the system, and air temperature also drops as it passes through the chamber. A revolving wetted honeycomb wick structure can be used to maximize the exposed surface area of water and thus the rate of evaporation. Ideally, air will leave the chamber as saturated (or close to being saturated) at a low temperature.

The cool and humid air is then routed from the freezing chamber to an air-to-air heat exchanger to cool the air entering the chamber. In the ideal case of a heat exchanger with an effectiveness of 1, the incoming air will be cooled to the temperature of the air leaving the chamber since the flow rates are the same, and the effect of vapor is negligible. In reality, such heat exchangers have an effectiveness of about 0.8, and thus the incoming air temperature will be somewhat higher. Of course the lower the temperature of the incoming air, the smaller the fraction of the heat of vaporization that is absorbed from air.

The air exiting the air-to-air heat exchanger is then routed to a desiccant chamber in which it is dehumidified by means of formation of hydrates. The desiccant

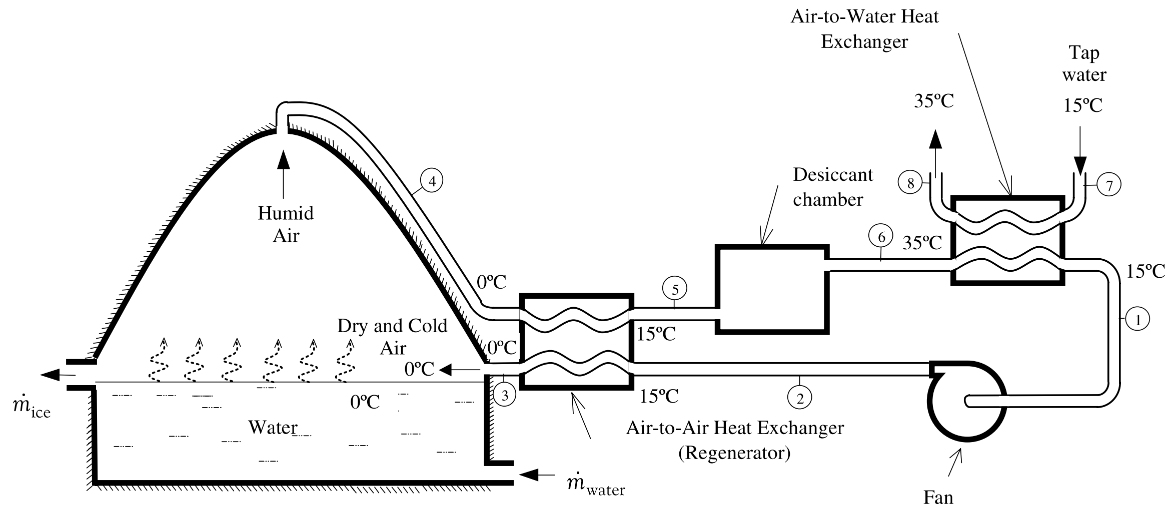

Fig. 1. A new evaporative freezing cycle for freezing of water.

instantly attracts moisture when the vapor pressure at the desiccant surface is less than that of air. The formation of hydrates is an exothermic reaction, and thus heat is released as the desiccant removes moisture from the air. Consequently, the air temperature rises, and it may reach up to 70 °C, depending on the moisture content of the air. The air leaves the desiccant chamber at a high temperature and a low humidity. The desiccant eventually becomes saturated with moisture, and it needs to be recharged. This is done by forcing hot air at temperatures between 50 and 300 °C through the desiccant chamber. The hot air can be obtained by burning fossil fuels such as natural gas, or by using a renewable energy such as solar energy or geothermal energy, or utilizing the hot exhaust gases from sea vessels. In the proposed system, a wheel-type revolving desiccant chamber can be used so that the desiccant is continually recharged. This is commonly done in desiccant-coupled evaporative cooling systems. The recharging system is not shown in the figure.

Next, the hot and dry air leaving the desiccant chamber is cooled in an air-to-water heat exchanger (such as a car radiator) by tap water. The cooled air is then passes through a fan which is used to circulate the air in the system. The air is further cooled in the air-to-air heat exchanger before entering the freezing chamber, and the process is repeated.

Each cycle reduces the temperatures of air and water, and the process continues until the temperature of water drops below the freezing point, and the nucleation of ice crystals begins. Once freezing starts and ice is removed periodically, the process continues almost steadily. Then the freezing chamber can be recharged, and the new water can be precooled by the cold water since we will be using a batch process. It is recognized that in a large commercial system, continuous operation is more desirable, and it can be accomplished by supplying fresh water into the chamber steadily while removing some of ice in the chamber.

3. Psychrometric chart analysis of the ideal evaporative freezing cycle

The ideal evaporative freezing cycle is shown schematically on a psychrometric chart in Fig. 2. The ideal cycle does not involve any internal and external irreversibilities, and consists of the following six processes:

- 1–2 circulation fan;
- 2–3 regeneration in the air-to-air heat exchanger (internal heat transfer from warm air to cold air);
- 3–4 isothermal humidification in the freezing chamber;
- 4–5 regeneration in the air-to-air heat exchanger (internal heat transfer from warm air to cold air);
- 5–6 dehumidification and heat generation in the desiccant chamber;
- 6–1 heat rejection in the air-to-water heat exchanger.

The cool and dry air enters the well-insulated freezing chamber at state 3. The water vapor is added to the air without the addition of heat. The direction of this process on the psychrometric chart may vary significantly depending on the temperature of water present in the chamber. There are four types of process that may take place in the chamber. If the water vapor at the interface is saturated at the dry bulb temperature of air, the process proceeds at a constant dry bulb temperature. If the enthalpy of water vapor at the interface is greater than the enthalpy of saturated vapor at the dry bulb temperature, the air is heated and humidified. If the water vapor enthalpy at the interface is less than the

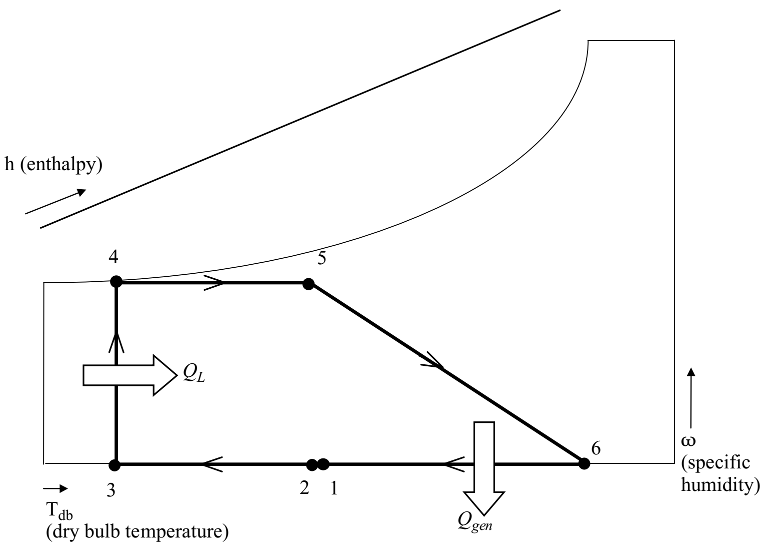

Fig. 2. Psychrometric analysis of evaporative freezing. The locations of state points are shown in Fig. 1.

enthalpy of saturated vapor at the dry bulb temperature, the air will be cooled and humidified. There is a special case process that is used in swamp coolers. When liquid water at the wet bulb temperature is sprayed, the process follows approximately a line of constant wet bulb temperature [10]. However, no matter which type of these processes occurs in the chamber, the dry bulb temperatures of the air exiting and entering ideally tend to be equal due to the air-to-air heat exchanger. Consequently, the isothermal humidification process must proceed at a constant dry bulb temperature. As the air is humidified along the constant dry bulb temperature, the latent heat of vaporization is transferred from the water to the air. If the evaporation of the water is continued, the water will cool down and eventually freeze at the freezing temperature.

The humid air at state 4 enters the air-to-air heat exchanger to cool the dry air entering the chamber, and leaves as warm air at state 5. As the temperature of the dry air entering the chamber decreases in the heat exchanger, the temperature of the humid air exiting the heat exchanger increases. Since there is no water vapor addition in the heat exchanger, the specific humidity of both streams remains constant. Thus, the air streams in the heat exchanger proceeds a line of constant specific humidity on the psychrometric chart.

The warm and humid air at state 5 enters the desiccant chamber in which it is dehumidified, and heated as a result of the heat released during the absorption process. While the dry bulb temperature of the air increases in the desiccant chamber, the specific humidity of the air decreases to the values at state 6, where the dry air enters the air-to-water heat exchanger.

The temperature of the dry and hot air at state 6 is reduced at constant specific humidity in the heat exchanger by rejecting heat to a cooling medium such as tap water. From state 6 to state 1, the latent heat of vaporization in the freezing chamber and the heat generated in the desiccant chamber are rejected through the air-to-water heat exchanger to the cooling medium. The cool and dry air enters the fan at state 1, and is pumped to state 2. The air temperature fairly remains constant during this process due to a slight decrease in the specific volume of the air. Then, the air passes through the air-to-air heat exchanger for further cooling at constant specific humidity, and enters the freezing chamber at state 3, completing the cycle.

4. Energy analysis of the ideal evaporative freezing cycle

The energy and mass equations in this section are developed by utilizing reversible components associated with the evaporative freezing cycle. These equations can be used to determine the minimum energy required and the maximum ice production rate by the evaporative freezing cycle. Also, the ideal energy and ice production values as well as the coefficient of performance provide a base criteria for the feasibility analysis.

In order to carry out the thermodynamic analysis of the evaporative freezing cycle, certain idealizations must be made the cycle.

1. All the components associated with the cycle (the air-to-air heat exchanger, desiccant chamber, air-to-water heat exchanger, fan, and freezing chamber)

are steady-flow devices, and thus all six processes that make up the evaporative freezing cycle can be analyzed as steady-flow processes. The operation of the freezing chamber appears to be unsteady. However, the mass of water is very large compared to the ice production rate, and thus the chamber can be thought to be continuously supplied with the water as the ice is produced.
2. The kinetic and potential energy changes of air and water are usually small relative to the heat transfer terms, and therefore are neglected.
3. Components and connections which have lower operating temperatures than the surrounding air are assumed to be well-insulated such that there is no heat gain. The other components which have higher temperatures than the surroundings such as the desiccant chamber are not insulated to allow heat loss to the environment.
4. The air at state 3 enters at a relative humidity of $\phi = 0\%$ and leaves the chamber as saturated air ($\phi = 100\%$). In other words, the chamber operates at a relative humidity difference of 100%. The humidity of 100% of the air can be approached by using exposed surface maximizers such as wetted honeycomb wicks, and high pressure spray nozzles. The humidity of 0% can be achieved by assuming no moisture leaks from the environment, and a 100% effective desiccant chamber.
5. Other parasitic energy inputs to circulate air and drive other components of the system to overcome pressure drops are neglected.

The thermodynamic analysis of moist air processes is based on the conservation of energy and the conservation of mass principles. By employing these two principles, the cooling capacity and ice production rate will be identified, and therefore the performance of the evaporative freezing cycle will be determined. The analysis is focused on the freezing chamber rather than the other components of the cycle, because the main function of the air-to-air heat exchanger, desiccant chamber, air-to-water heat exchanger, and fan is to bring back the air at state 3 to the chamber at a constant dry bulb temperature, but a lower specific humidity.

When moisture is added to the air without the addition of heat, the process yields a straight vertical line from state 3 to state 4 (as shown on the psychrometric chart in Fig. 2) since the dry bulb temperature remains constant. A control volume can be considered in the freezing chamber, as shown schematically in Fig. 3. Under steady-flow conditions, the conservation of mass gives

Dry air mass balance:

$$\dot{m}_{a3} = \dot{m}_{a4} = \dot{m}_a$$

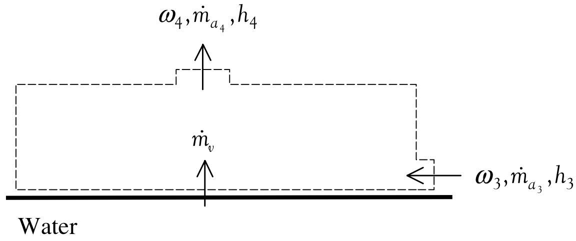

Fig. 3. Control volume above the water surface.

Water mass balance:

$$\dot{m}_a\omega_3 + \dot{m}_v = \dot{m}_a\omega_4 \rightarrow \dot{m}_v = \dot{m}_a(\omega_4 - \omega_3) \quad (1)$$

The energy balance on the control volume yields

Energy balance:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \rightarrow \dot{m}_a h_3 + \dot{m}_v h_v = \dot{m}_a h_4 \rightarrow h_4 - h_3 = \frac{\dot{m}_v}{\dot{m}_a} h_v \quad (2)$$

where h_v is the latent heat of vaporization at the range of freezing temperature and pressure ($h_v = h_g = 2504$ kJ/kg water vapor at 0 °C and 1 atm), and ω_4 and ω_3 are the specific humidities of the air ($\omega_4 = 0.00378$ kg water vapor/kg dry air, and $\omega_3 = 0$ kg water vapor/kg dry air at 0 °C and 1 atm). Substituting Eq. (1) into Eq. (2) gives

$$h_4 - h_3 = h_v(\omega_4 - \omega_3) \quad (3)$$

An interesting feature of this result is worth noting. The psychrometric analysis of the process from state 3 to state 4 reveals that the enthalpy of air at state 4 is greater than that of air at state 3 at constant dry bulb temperature. Not surprisingly, this is due to the fact that the latent heat of vaporization is transferred from the water to the air by evaporation. Consequently, the isothermal evaporation of the water provides a technique for cooling the liquid. If the process of evaporation is continued, the water will freeze and form ice crystals at constant temperature and pressure. The heat removed Q_L is the difference between the enthalpy of the air at state 4 and of the air at state 3. That is,

$$Q_L = h_4 - h_3 = h_v(\omega_4 - \omega_3) \quad (4)$$

Then, the cooling capacity can readily be calculated to be

$$Q_L = (2504 \text{ kJ/kg water vapor})(0.00378 - 0) = 9.5 \text{ kJ/kg dry air}$$

Therefore, the maximum possible cooling capacity for the evaporative freezing cycle operating at 0 °C and 1 atm is 9.5 kJ/kg dry air circulating. The cooling capacity of the cycle is dependent on both the atmospheric pressure and the flow rate of dry air. The cooling capacity at elevated locations tends to rise depending upon

Fig. 4. Control volume underneath of the water surface.

the atmospheric pressure. At 0.83 atm, for example, the cooling capacity becomes 11.3 kJ/kg dry air. As can be seen from the above result, the cooling capacity increases with increasing flow rate of dry air in the system.

While the air absorbs the moisture in the chamber, the water at 0 °C starts freezing gradually. A control volume underneath of the water surface in Fig. 4 can be considered to illustrate the mass balance. Applying the mass and energy balances on the control volume yields

Water mass balance:

$$\dot{m}_{\text{water}} = \dot{m}_{\text{vapor}} + \dot{m}_{\text{ice}} \tag{5}$$

Energy balance:

$$Q_{\text{L}} = \dot{m}_{\text{water}}(h_{\text{f}@T_{\text{w}}} - h_{\text{f}@T_{\text{f}}}) - \dot{m}_{\text{ice}}h_{\text{if}} \tag{6}$$

where h_{f} is the enthalpy of water, and h_{if} is the latent heat of fusion of water, T_{w} is the incoming water temperature, and T_{f} is the freezing temperature of water. Eq. (6) can be further simplified. Let us imagine the incoming water is cooled to about freezing temperature in a heat exchanger in which the ice produced are passed through it. The incoming water will not be cooled even ideally to the freezing temperature because $\dot{m}_{\text{water}}$ is slightly larger than the flow rate of $\dot{m}_{\text{ice}}$ due to the evaporation of the water. Then, we may assume that the incoming water always enters the freezing chamber at nearly freezing temperature. Approximately, Eq. (6) can be rewritten to be

$$Q_{\text{L}} \cong -\dot{m}_{\text{ice}}h_{\text{if}} \tag{7}$$

which means that all the cooling capacity is approximately spent to freeze the water rather than to cool the incoming water. Since h_{if} and Q_{L} are known to be 334 kJ/kg ice and 9.5 kJ/kg dry air, respectively, the ice production rate can simply be calculated to be

$$\dot{m}_{\text{ice}} \cong \frac{Q_{\text{L}}}{h_{\text{if}}} \cong \frac{9.5\ \text{kJ/kg dry air}}{334\ \text{kJ/kg ice}}$$
$$\cong 0.0284\ \text{kg ice/kg dry air} \cong 28.3\ \text{g ice/kg dry air}$$

Therefore, the amount of 28.4 g ice will be produced per kg dry air. The ice production rate varies considerably depending on the amount of air being circulated in the evaporative freezing cycle. If the air volume flow rate is 3 m^3/min, the ice production rate becomes (28.4 g ice)(1 kg/m^3)(3 m^3/min) = 85.2 g ice/min.

The thermal coefficient of performance (COP) is commonly used to compare the performance of refrigeration cycles. The definition of the thermal COP greatly varies for desiccant cooling systems. However, the majority of investigators define it as the ratio of the cooling capacity to the thermal energy required to regenerate the desiccant. Then, the thermal COP of the cycle is defined as

$$\text{COP} = \frac{Q_{\text{L}}}{Q_{\text{reg}}} = \frac{\text{cooling capacity}}{\text{regeneration heat}}$$

where Q_{reg} is the thermal energy required to regenerate the desiccant. During steady operation of the freezing chamber at 0 °C, the cooling capacity is equivalent to the enthalpy of vaporization of water h_{fg} at 0 °C. When a unit mass of moisture in the air is absorbed by the desiccants at a specified temperature, the amount of heat released is equal to the enthalpy of vaporization of water h_{fg} at that temperature. This is also equal to the amount of heat needed to dry the desiccant by vaporizing the moisture. Then the thermal COP of the ideal cycle can be expressed as

$$\text{COP} = \frac{Q_{\text{L}}}{Q_{\text{reg}}} = \frac{h_{\text{fg}@T_{\text{freezing}}}}{h_{\text{fg}@T_{\text{reg}}}}$$

This is a very significant result since it shows that a refrigeration system can produce as much cooling effect as the heat supplied to the system. We stated earlier that about 7 kg of water can be frozen by evaporating 1 kg of water. Thus we conclude that with the evaporative freezing system, about 7 kg of water can be frozen by supplying enough heat to vaporize 1 kg of water.

The actual evaporative freezing system will require more heat input that the h_{fg} since the desiccant material itself (as well as the matrix in system with a rotary desiccant wheel) will also be heated. But this effect can be minimized by using desiccants that absorb a large amount of water before becoming saturated. For example, the lithium chloride can absorb more than 1000 times its own weight in water as it turns from a solid to a liquid whereas silica gel can absorb only about 30% of its own weight in water.

The thermal COP can also be expressed in terms of the mass of the circulating air as

$$\text{COP} = \frac{Q_{\text{L}}}{Q_{\text{reg}}} = \frac{\text{cooling capacity per kg dry air}}{\text{regeneration heat per kg dry air}}$$

The cooling capacity Q_{L} remains relatively constant in steady operation, but the regeneration energy varies depending on type of desiccant used.

The thermal COP can be calculated for the solid desiccant DRIERITE which is made of gypsum (calcium sulfate, $CaSO_4$). The American National Bureau of Standards certified that DRIERITE has a water vapor absorption capacity of about 12% by weight, and it can dry the air up to the specific humidity of 5×10^{-6} g water vapor/kg dry air. It requires an average of 640 kJ of energy per kg saturated DRIERITE to regenerate [11].

Table 1
The coefficient of performance (COP) values at various heat exchanger effectiveness values

Cases	Effectiveness ε	Inlet temperature T_3 (°C)	Cooling capacity Q_L (kJ/kg dry air)	Ice production rate $\dot{m}_{ice}$ (g ice/kg dry air)	Coefficient of performance (COP)
Ideal case	1	0.0	9.44	28.3	0.47
Actual cases	0.9	1.5	7.93	23.8	0.39
	0.8	3.0	6.42	19.2	0.32
	0.7	4.5	4.91	14.7	0.24

The regeneration energy, Q_{reg}, per kg dry air can be calculated from the specific humidity of the air at state 4. Ideally, the air at state 4 holds a maximum amount of 0.00378 kg water vapor/kg dry air at 0 °C and 1 atm. The amount of DRIERITE to absorb 0.00378 kg water vapor is

$$(0.00378 \text{ kg water vapor/kg dry air}) \times (100 \text{ kg DRIERITE/12 kg watervapor}) = 0.0315 \text{ kg/kg dry air}$$

Then the needed regeneration energy, Q_{reg}, can be determined to be

$$(640 \text{ kJ/kg DRIERITE}) \times (0.0315 \text{ kg DRIERITE/kg dry air}) = 20.2 \text{ kJ/kg dry air}$$

Therefore, under ideal steady operating conditions, 20.2 kJ is required to regenerate 1 kg dry air being recirculated in the system. Then, the COP of the evaporative freezing cycle becomes

$$\text{COP} = \frac{Q_L}{Q_{reg}} = \frac{9.5 \text{ kJ/kg dry air}}{20.2 \text{ kJ/kg dry air}} = 0.47$$

The COP of 0.47 is the highest COP value that the evaporative freezing cycle can have, because it is calculated by assuming ideal devices and processes for the cycle. However, the analysis can be extended to actual cases in which effectiveness values of the air-to-air heat exchanger are lower than 1 in order to figure out how the COP of the cycle is affected with respect to the effectiveness of the heat exchanger. When the effectiveness becomes lower than 1, the air starts entering the freezing chamber at higher temperatures. The increase in the temperature can be determined by employing the effectiveness expression of the heat exchanger which is $\varepsilon = (T_2 - T_3)/(T_2 - T_4)$. By repeating the same procedure described above for the effectiveness values of 0.9, 0.8 and 0.7, the COP values of the system are calculated and presented in Table 1 along with the ideal case. The COP values are then plotted in Fig. 5 as a function of the effectiveness using values from Table 1. It is apparent from the figure and the table that the COP of the cycle drops about 50% from 0.47 to 0.24 while the effectiveness decreases from 1 to 0.7. This is expected, because the deviation from ideality always results in a decrease in performance. Along with the decrease in the COP, the ice production and cooling capacity rates also decrease by about 50% due to the lower effectiveness of the heat exchanger. The cooling capacity and thus the COP of the cycle also varies with the atmospheric pressure at which the cycle operates. At higher elevations, the evaporative freezing cycle will have a higher cooling capacity. For example, the cooling capacity increases from 9.5 to 11.25 kJ/kg dry air when the atmospheric pressure goes down from 1 to 0.83 atm.

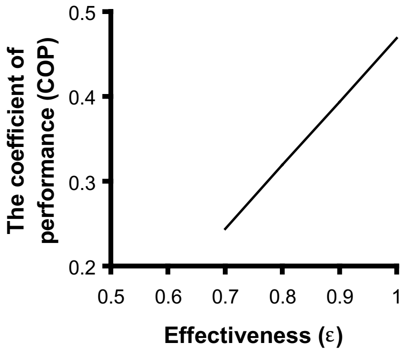

Fig. 5. The variation of the coefficient of performance with the heat exchanger effectiveness.

5. Conclusion

A novel method for evaporative freezing of water is presented. The evaporative freezing cycle offers a viable alternative to the existing refrigeration methods. It is easy to build, energy efficient, and environmentally friendly. It has the potential of being superior to any of the existing methods when an inexpensive source of thermal energy is available to recharge the desiccants. The technology developed can also be used in areas other than freezing such as, desalination, freeze separation in the cleanup of minewastes, saline ponds, tritiated water, and other waste streams [12–15].

The evaporative freezing cycle is an open cycle in which air is recirculated at the atmospheric pressure. The system is easy to build using off-the-shelf components such as an air-to-air heat exchanger, desiccant chamber, air-to-water heat exchanger, and connection components. However, the system must be constructed carefully to avoid any air leaks. Since any leak of warm and moist air from the environment into the system will have a detrimental effect on the performance of the system. Openings on the components and joints must be sealed off with proper sealants.

Solid desiccants require large power requirements (due to high pressure drop in the desiccant chamber) to move air through the desiccant chamber. Liquid desiccants, on the other hand, require less parasitic power and less heat input for regeneration. Liquid desiccants can be regenerated easily with low-temperature thermal energy. Therefore, the use of liquid desiccants can dramatically improve the COP of the system.

One of the interesting features of the evaporative freezing system is that it is a heat-driven system that primarily removes moisture from an airstream. The heat can come from any suitable source such as natural gas, solar energy, geothermal energy, or waste heat, etc. Therefore, the cost of energy consumption of the evaporative freezing systems can be very low.

References

[1] D.R. Anderson, A.A. Pesaran, Innovative solid desiccant substrates for desiccant dehumidifiers, ASHRE Transactions, New York, 1991, pp. 578–586.

[2] P.L. Dhar, S.C. Kaushik, S. Jain, D. Pahwa, R. Kumar, Thermodynamic analysis of desiccant-augmented evaporative cooling cycles for Indian conditions, ASHRE Transactions, New York, 1995, pp. 735–749.

[3] S.P. Jain, L. Dhar, S.C. Kaushik, Evaluation of solid-desiccant-based evaporative cooling cycles for typical hot and humid climates, Int. J. Refrigerat. 18 (5) (1995) 287–296.

[4] R.K. Jollier, R.S. Barlow, F.H. Arnold, An overview of open cycle cooling systems and materials, ASME J. Solar Energy Eng. 104 (1982) 28–34.

[5] T. Kravchik, E. Korin, I. Borde, Influence of material properties and heat removal on mass and heat transfer in a solid desiccant dehumidifier, Chem. Eng. Process 27 (1990) 19–25.

[6] S.M. Lu, R.J. Shyu, W.J. Yan, T.W. Chung, Development and experimental validation of two novel solar desiccant-dehumidification–regeneration systems, Energy 20 (8) (1995) 751–757.

[7] J.S. Nelson, W.A. Beckman, J.W. Mitchell, D.J. Close, Simulation of the performance of open cycle desiccant systems using solar energy, Solar Energy 21 (1978) 273–278.

[8] Y.A. Cengel, M.A. Boles, in: Thermodynamics: An Engineering Approach, third ed., McGraw-Hill, New York, 1998, p. 52.

[9] Y.A. Cengel, Heat Transfer: A Practical Approach, McGraw-Hill, New York, 1998, p. 904.

[10] F.C. McQuiston, J.D. Parker, Heating, Ventilating, and Air Conditioning: Analysis and Design, fourth ed., John Wiley & Sons Inc., 1994.

[11] DRIERITE Catalog, W A Hammod DRIERITE Co., P.O. Box 460, Xenia, OH 45385.

[12] H.M. Handrickson, R.W. Moulton, Research and development of processes for desalting water by freezing, Office of Saline Water, US Department of Commerce, Washington, DC, R&D Report 1956, No. 10.

[13] I. Krepchin, R. Torbin, Design of a solar power plant for freeze desalination, ASME Paper No. 86-WA/Sol-5, Winter Annual Meeting, Anaheim, CA, 1986.

[14] G. Karnofsky, P.F. Steinhoff, Saline water conversion by direct freezing with butane, Office of Saline Water, US Department of Commerce, Washington, DC, R&D Report No. 40, 1960.

[15] H.F. Wiegandt, P. Harriott, J.P. Leinroth, Desalting of seawater by freezing, Office of Saline Water, US Department of Interior, Report No. PB251906, Washington, DC, 1973.