Desalination 254 (2010) 179–184

Contents lists available at ScienceDirect

Desalination

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

New proposed system for freeze water desalination using auto reversed R-22 vapor compression heat pump

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ARTICLE INFO

Article history:
Received 7 October 2009
Received in revised form 18 November 2009
Accepted 18 November 2009
Available online 12 January 2010

Keywords:
Desalination
Freeze desalination
Water purification
Vapor compression desalination system

ABSTRACT

As natural resources are becoming limited and energy price dramatically increased, energy utilization with efficient systems is required to be used in desalination technologies. Freezing is a well known technique for water desalination. In the present paper, a new proposed system depends on optimization of utilizing the heat flow of heat pump system to increase the whole system efficiency is introduced. The suggested system overcomes a lot of disadvantages of traditional freezing methods of desalination like ice handling, special compressor types, etc. In the suggested system, the ice washing and melting process occur at the same place of formation by reversing refrigerant flow through the vapor compression cycle so there is no need for ice handing mechanical systems. A detailed description of the system and thermal analysis are represented with simple cost analysis. Cost comparing with other methods gives a promising lower cost for the suggested system with respect to other systems.

1. Introduction

There are many types of sea water desalination systems like multi-stage flash evaporator (MSF), reverse osmosis (RO), electro dialysis (ED), capacitive deionization technology (CDT) and freeze desalination. Thermal desalination plants such as MSF system require a large quantity of heat energy so they are generally accompanied by power plants. Reverse osmosis uses membrane as a nano-filter for removing salt particles from sea water stream and it consumes energy in the form of pumping work. RO system is a complicated technology and the membrane is very sensitive and costly during initial installation and sensitive in maintenance and needs for special regulation are considered by the manufacturer. Capacitive deionization technology (CDT) uses electrostatic charges on the surface of adjacent plates (serves as a capacitor) to collect the anion salt parts on one of the surface and cation part on the other surface according to surface charge. This technology used to produce ultra pure water that is used in medical industries, chemical labs, and boilers of steam turbines due to the limited quantity produced. Finally corrosive resistance and economic benefits are the major advantages of freeze desalination plants. Actually the cost for each method depends mainly on type of physical process of salt removal (i.e. evaporation, filtration, freezing or electrostatic potential difference). The efficiency of each type depends on the total energy required to remove the salt from the water which depends to some extent on the method of operation (evaporation, electrical dialysis and freezing, etc) and also on the purity of the required water.

One of the main advantages of freeze desalination systems is that it required removing only 420 kJ/kg to produce 1 kg fresh water, which is six times lower that MSF energy required to get 1 kg of fresh water moreover, the heat in the freezing method is removed by using vapor compression cycle principles, which means that the quantity of heat removed is not actually, considered as a cost parameter in freezing method but compression work is a major cost parameter.

In 1997 Rice and Chau [3] discussed the idea of using hydraulic refrigerant in freeze desalination plants. They stated that freeze desalination should be reconsidered and compared with other means of desalination with regards to energy efficiency, first cost, operating costs, ease of maintenance and useful life. The advent of the hydraulic refrigerant compressor is an "enabling" component to make freeze desalination much more attractive than it has been in the past.

In 1999 Slesarenko [2] studied using desalination plant with absorption heat pump for power station. The study shows that the application of thermal vacuum desalination plants for AHP makes it possible to lower the cost of desalinated water production by 2–2.5%.

In 2001 Slesarenko [1] studied using heat pump as a source of heat energy for desalination of seawater. The thermodynamic study of the system shows the benefits of using heat pump on using heat energy that was available especially at low temperature.

In 2008 Lara et al. [4] explored higher operating temperatures for seawater desalination. The operation at elevated temperature and pressure reduces the size of the latent heat exchangers and compressor which lower the capital costs. In addition, it reduces energy requirements which lower operating costs.

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0011-9164/$ – see front matter
doi:10.1016/j.desal.2009.11.030

In the following a detailed description of the new technique is introduced to overcome the difficulties of traditional freezing methods of desalination like ice handling, special compressor types, etc. Also thermal and simple cost analyses are represented to compare the new technique with traditional desalination methods.

2. System description

Due to its cost, complexity, maintenance, etc, the mechanical system used to remove the produced ice is considered as one of the most difficult in freeze desalination plants. The present technique treats this problem by using auto reversed vapor compression heat pump where the heat removed from the cycle is used to melt the produced ice from previous cycle of operation. A schematic diagram of the system is illustrated in Fig. 1. The system consists of two tanks and heat pump system. For the two tanks (A & B), each tank could be treated as heat source or as heat sink according to heat pump system switching sequence. This means that, when tank (A) is used as a heat sink for the vapor compression cycle, tank (B) serves as a heat source and then they switch the function automatically in the next cycle of operation.

The operation starts by filling tank (A) with salty water through solenoid S_{A1} to a desired level, then the heat pump system starts to operate. In this case, tank (A) is considered as a heat source for vapor compression cycle. Meanwhile a stream of salty water at supply temperature flows through tank (B) to remove heat reject in the condenser of the vapor compression cycle.

After a certain period of time – depending on the system, design and tank water-capacity – the water in tank (A) will be converted into slurry consisting of ice and brine water. At this point, the compressor of the heat pump system stops while the brine water in tank (A) is drained out through S_{A3} leaving ice crystals in the tank. After that, the remaining ice is washed through solenoids S_{A1} and S_{A4}. Meanwhile, the salty water is pumped to fill tank (B) through solenoid S_{B1}. After this process is finished, the heat pump system will be switched to use tank (A) – which contains remaining ice crystals from previous cycle – as a heat sink, and tank (B) which contains new salty water to function as a heat source. Then, the water in tank (B) starts to cool down to form another quantity of ice, while ice that was formed in tank (A) – from the previous cycle – starts to melt converting to fresh water. When the ice is completely melted and the temperature of the fresh water in tank (A) reaches the supply water temperature the fresh water in tank (A) starts to drain out through solenoid S_{A1} through fresh water line and the system used the salty supply water to remove heat from the condenser of the vapor compression cycle through the brine water line. The last step is repeated again with tank (A) as heat source and tank (B) as heat sink consequently.

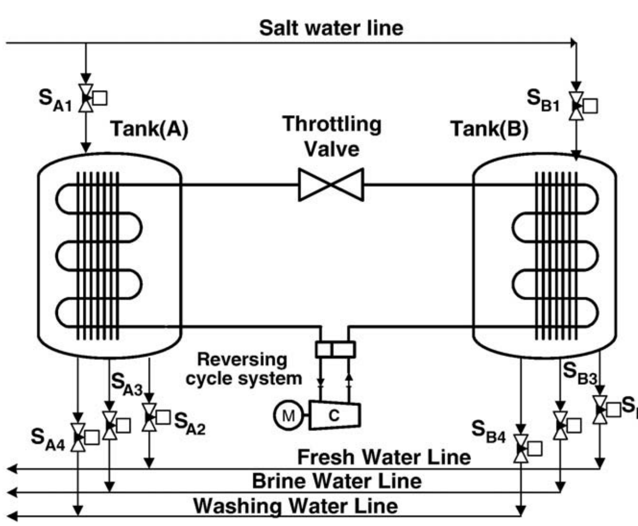

Fig. 1. Layout of the proposed freeze water desalination system using reversed vapor compression heat pump.

Two advantages can be easily recognized for this technique the first one is that there is no need to use complex mechanical systems to remove the ice. And the second one is higher efficiency can be achieved due to the optimum utilization of heat removing flow. Also maintenance of the system is just external washing of the heat exchanger fines and regular maintenance for solenoids and vapor compression systems which have no need for professional skills or other special regulation.

3. Vapor compression heat pump

Fig. 2 shows a realistic vapor compression cycle layout which mainly consists of four items: compressor, condenser, evaporator and throttling valve. The cycle has several operating conditions that significantly impact the cycle performance. Fig. 3 is a schematic of realistic vapor compression cycle on *Ln*(*P*)-h diagram. The main major notice is the slop in the process lines that used to be isobars, this is due to a certain pressure drop which is necessary to refrigerant flow through the heat exchangers and any connecting pipe. At the evaporator outlet the refrigerant is super heated which contributes to the cooling capacity and compressor live time. Due to the pressure drop in the evaporator and the superheating at the evaporator outlet, the compression process starts at lower pressure and further with superheated vapor.

For the heat pump system, there are two different systems that can be used in the present technique. In the first one the cycle works at constant temperature levels by using a constant pressure drop throttling valve. This system is considered as inexpensive, simple and easy in installation and control. The second system uses a variable pressure drop throttling valve to maximize the efficiency where the C.O.P. of the system varies depending on the change in heat sink temperature. This system needs higher restricted control system which increases the initial cost and also the system complicities.

4. System analysis

In the following section, an explanation for the performance variation between the two systems will be presented.

4.1. Using constant pressure drop throttling valve

Since we have a constant pressure drop for the heat pump system, both temperature levels and coefficient of performance will be constant. In this case and depending on practical experience we can assume that the temperature of the evaporator (in the heat pump system) is lower than the freezing temperature of the salt water by 5 °C, taken in consideration that the freezing temperature depends on the degree of saltiness of sea water. By assuming that, the condenser

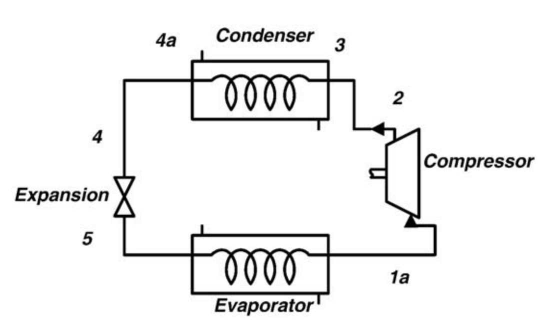

Fig. 2. Realistic vapor compression cycle layout.

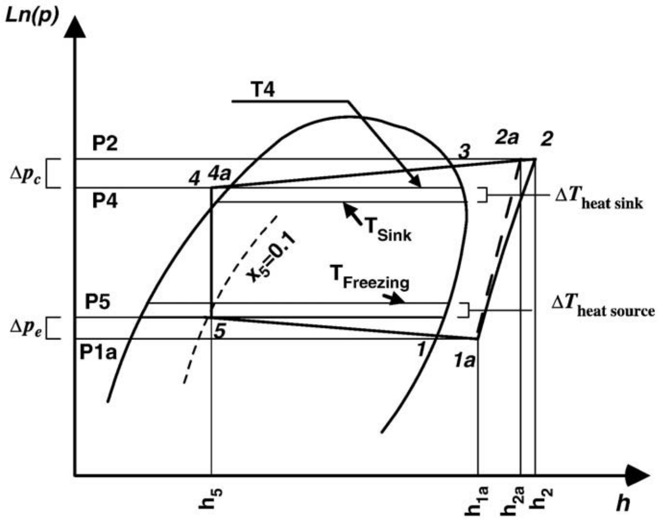

Fig. 3. Schematic diagram for realistic vapor compression cycle on *Ln*(*P*)-h diagram.

temperature (in the heat pump system) is higher than the heat sink final temperature by 5 °C. As mentioned earlier, the coefficient of performance will be constant through all stages of the heat pump system where the coefficient of performance can be stated as:

$$\text{C.O.P.} = \frac{h_{1a}-h_5}{h_2-h_{1a}} \tag{1}$$

From practical consideration the points 1a, 2, 5 on the Fig. 4 could be considered as follows:

$$T_5 = T_{\text{Freezing}} - 5, X_5 = 0.1 \tag{2}$$

$$T_4 = T_{\text{supply}} + 7 \tag{3}$$

With assumption of:

$$T_4 \cong T_{4a} \tag{4}$$

$$h_5 = h_4 \tag{5}$$

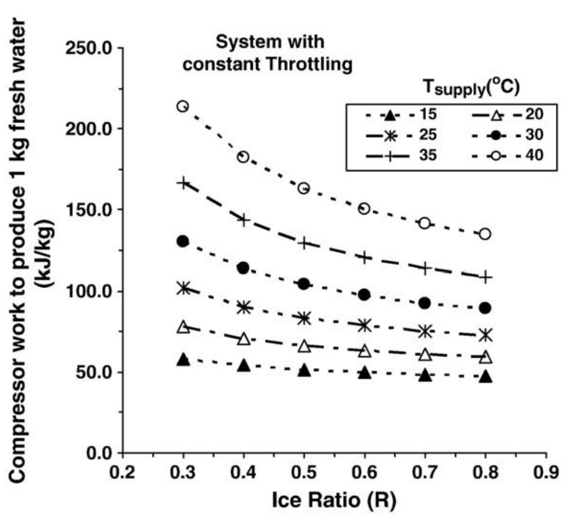

Fig. 4. Energy consumed to produce 1 kg of fresh water versus ice ratio at different supply salty water temperatures for system with variable throttling vapor compression cycle.

$$T_{1a} = T_5 - 5 \tag{6}$$

From practical consideration:

$$\Delta p_e = p_5 - p_{1a} \approx 5\,\text{psi} \tag{7}$$

$$S_{1a} = S_{2a} \tag{8}$$

$$P_2 \approx P_{2a} \tag{9}$$

$$\Delta P_c = P_2 - P_4 = P_{2a} - P_4 \approx 5\,\text{psi} \tag{10}$$

$$\eta_c = 0.9 = \frac{h_{2a}-h_{1a}}{h_2-h_{1a}} \tag{11}$$

The coefficient of performance of the cycle is:

$$\text{C.O.P.} = \frac{Q_L}{W} \tag{12}$$

Where

$$\left.\begin{aligned} Q_L. = &\underbrace{m^*C_P^*(T_{\text{Supply}}-T_{\text{Freezing}})}_{\substack{\text{Sensiable heat removed from the water}\\ \text{to reduce temperature from } T_{\text{supply}} \text{ to } T_{\text{Freezing}}}} \\ &+ \underbrace{R^*m^*\text{L.H.F.}}_{\substack{\text{Latent heat removed}\\ \text{to form ice in the tank}}} \end{aligned}\right\}\text{kJ} \tag{13}$$

Then

$$W = \frac{Q_L}{\text{C.O.P.}}\ \text{kJ} \tag{14}$$

From Eqs. (13) and (14) the value of work done to produce one kg of fresh water could be estimated from Eq. (15)

$$E = \frac{W}{R^*m}\ \text{kJ/kg} \tag{15}$$

From Eqs. from (7) up to (8), by assuming values of T_{Freezing} and T_{Supply}, at different ice ratios, the value of C.O.P. could be evaluated and consequently the compressor work which is the cost reflector can be determined at different supply temperatures and also for different percentages of ice formation. This value of work produces quantity of water equal to R^*m kg of fresh water.

4.2. Using variable pressure drop throttling valve

In this system, the C.O.P. for heat pump system will vary through time of operation due to change in throttling pressure drop. This pressure drop is linked by a control system to heat sink temperature which almost varies during the system operation. The same assumption mentioned in Section 4.1 will be used again where the temperature of the evaporator will be lower than the freezing temperature by 5 °C, and the condenser temperature will be higher than the sink temperature, which almost changes, by 5 °C. The work done at different ice ratios, T_{Supply} and T_{Freezing} will be calculated according to the following analysis.

$$dW = P^*\Delta\tau \tag{16}$$

$$\text{C.O.P.} = \frac{h_{1a}-h_5}{h_2-h_{1a}} \tag{17}$$

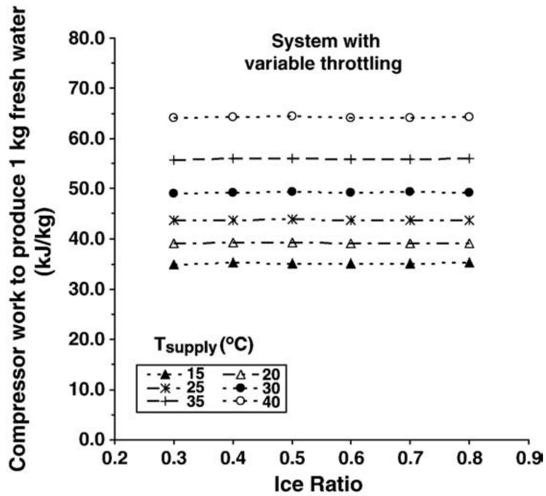

Fig. 5. Energy consumed to produce 1 kg of fresh water versus ice ratio at different supply salty water temperatures for system with variable throttling vaor compression cycle.

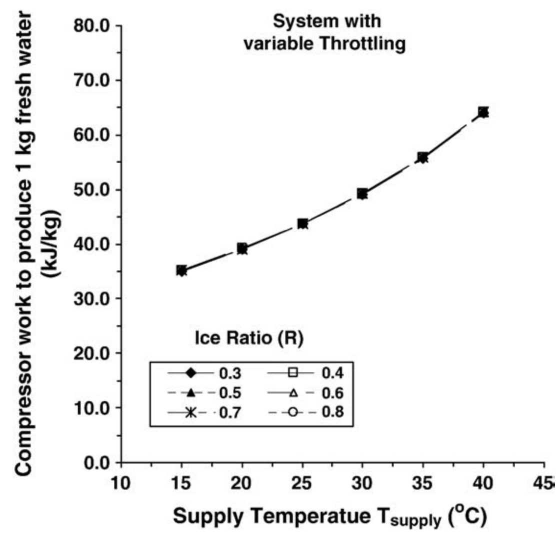

Fig. 7. Energy consumed to produce 1 kg of fresh water versus supply salty water temperature at different ice ratio for system with variable throttling.

$$T_{\mathrm{sink}} = \begin{Bmatrix} T_{\mathrm{Freezing}} \; if \; \sum(dQ_{\mathrm{L}} + dW) \leq m^{*}R^{*}\mathrm{L.H.F} \\ T_{\mathrm{supply}} \;\; if \; T_{\mathrm{sink}} \geq T_{\mathrm{supply}} \\ T_{\mathrm{Freezing}} + \dfrac{(\sum(dQ_{\mathrm{L}} + dW)) - R^{*}m^{*}\mathrm{L.H.F}}{R^{*}m^{*}Cp} \end{Bmatrix} \tag{18}$$

The values of h_5 and h_{1a} are constant through the operation of the system but the value of h_2 will vary related to the value of $T_{\mathrm{sin}k}$ which varying through system operation from $T_{Freezing}$ up to $T_{\mathrm{sup}\ ply}$

$$dQ_L = dW^{*}C.O.P \tag{19}$$

$$W = \begin{matrix} \sum dQ_{\mathrm{L}} = \begin{pmatrix} m^{*}C_{\mathrm{p}}{}^{*}(T_{\mathrm{supply}} - T_{\mathrm{Freezing}}) \\ + R^{*}m^{*}\mathrm{L.H.F.} \end{pmatrix} \\ \sum dW \\ \sum dQ_{\mathrm{L}} = 0 \end{matrix} \tag{20}$$

We have to notice that the heat sink temperature is equal to the freezing temperature ($T_{\mathrm{Sink}} = T_{\mathrm{Freezing}}$) at the beginning of system operation due to the existence of ice in the tank that is used as a heat sink. Then, the temperature starts to rise after the ice was melted. The solution of equations sequence from Eqs. (16)–(20) starts with a known compressor power, salty water supply temperature, ice ratio and total tank capacity. The time interval $\Delta\tau$ in Eq. (20) determines the accuracy of the solution.

5. Discussion

A computer program was built to study the above two systems. This program calculates working fluid properties at design points of vapor compression cycle that was shown in Fig. 3 then C.O.P. and finally the total power and consequently the consumed power to produce 1 kg of fresh water at different ice ratios and salty water supply temperatures. Ice ratio varies from 0.3 up to 0.8. Salty water supply temperature varies from 5 °C up to 40 °C with step 5 °C.

The effect of ice ratio on the consumed energy at different salty water supply temperatures for constant throttling is illustrated in Fig. 3. At certain salty water supply temperature, the more the ice ratio increases the consumed energy decreases because there is a fixed value of sensible heat that should be removed. The consumed power depends on the ratio of this value to the total amount of heat removed. This ratio decreases with the increase of the ice ratio. Also, the sensible heat value

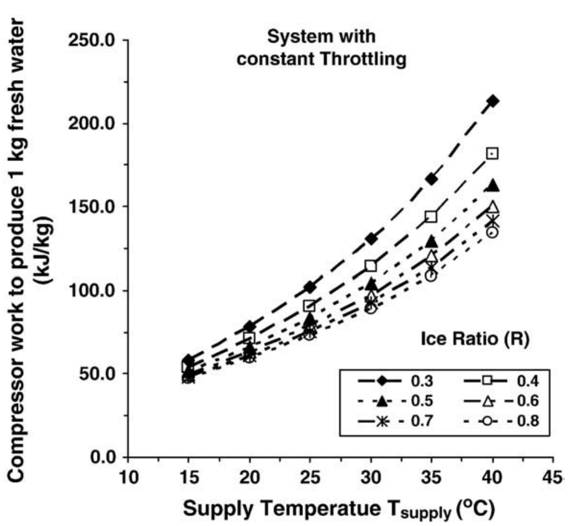

Fig. 6. Energy consumed to produce 1 kg of fresh water versus supply salty water temperature at different ice ratio for system with constant throttling.

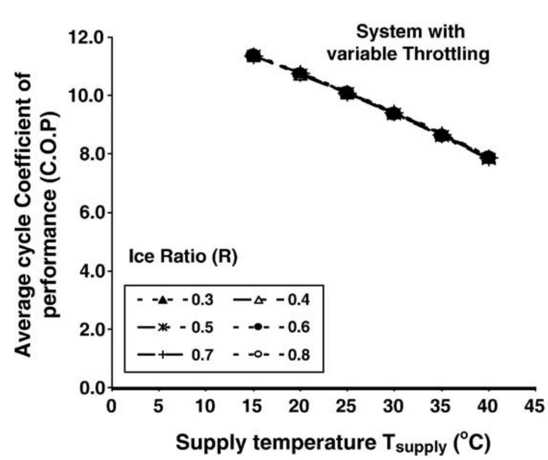

Fig. 8. C.O.P. of the cycle with variable pressure drop at different ice ratio and supply water temperature.

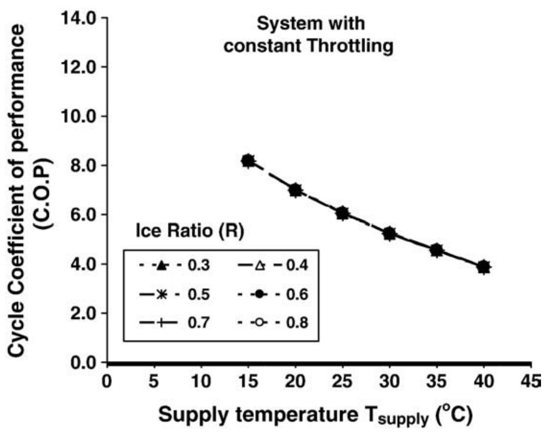

Fig. 9. C.O.P. of the cycle with variable pressure drop at different ice ratio and supply water temperature.

decreases with the decrease of salty water supply temperature. So, the difference in consumed power at low ice ratio and at high ice ratio decreases with the decrease of salty water supply temperature.

For example, at salty water supply temperature 40 °C and ice ratio 0.3, the energy consumed to produce 1 kg of fresh water is approximately 215 kJ/kg. For the same conditions but with ice ratio 0.4, the required energy is 180 kJ/kg with about 16% decrease in consumed energy. For the same temperature and with the increase of ice ratio to 0.5, the required energy is about 160 kJ/kg with a decrease of 11% lower than that for 0.4 ice ratio i.e. the value of enhancement in energy consumption decreases with the increase of ice ratio.

Fig. 5 shows the effect of varying ice ratio on consumed energy to produce 1 kg of fresh water at different water supply temperatures. The value of energy consumed is approximately constant with variation of ice ratio due to the nature of system where the operating pressure and temperature of the vapor compression cycle are linked to sink temperature which varies through the operation. These varieties that cause C.O.P. of vapor compression cycle, varies through the system operation which finishes the effect of existence of fixed sensible heat quantity.

Also, the effect of salty water supply temperature on the consumed energy at different ice ratio for constant throttling is presented in Fig. 6. It could be recognized that the difference between consumed power at high and low salty water supply temperature decreases with the increase of ice ratio. This is due to the increase of latent heat which depends mainly on the ice ratio.

At Fig. 7 the effect of salty water supply temperature on the consumed energy at different ice ratio for constant throttling is presented. It could be recognized that the consumed power is affected only by water supply temperature. This is due to the increase of quantity of sensible heat which does not depend mainly on the ice ratio.

It is clear from Figs. 4–7, that the variable throttling valve system consumes less energy compared to constant throttling valve system at the same salty water supply temperature and same ice ratio. Because the system using constant throttling has a constant value for C.O.P. but for the system using the variable throttling, C.O.P. starts with high value, then the value decreases with the increase in heat sink temperature until it reaches the same value of the C.O.P. for a constant throttling valve system under the same operating conditions (ice ratio and salty water supply temperature). This is one of the benefits of using the suggested system with variable throttling valve, where a system uses the existing ice in the heat sink tank to raise the C.O.P. value of the heat pump system.

Fig. 8 shows the effect of supply water temperature on the average C.O.P. of vapor compression cycle. The more the supply temperature increases the more the C.O.P. decreases. This is due to the effect of sensible heat quantity increasing. Ice ratio has no effect on C.O.P. of the cycle. It could recognize the same trend and the same behavior for constant throttling system at Fig. 9.

6. Cost analysis

In both systems above, the cost to produce 1 kg of fresh water is estimated according to Eqs. (21) and (22).

$$\left(\begin{matrix}\text{Water}\\ \text{productivity}\end{matrix}\right) = \left(\frac{60 * 60 * \text{Compressor power(kW)}}{\text{Energy consumed (kJ/kg) Fresh water}}\right)\text{kg/hr} \tag{21}$$

$$\text{Cost} = \frac{\text{price (\$/kW hr)*Compressor power (kW)}}{\text{Water productivity (kg/hr)}}\ /kg \tag{22}$$

Ref. [4] states an economical analysis for different desalination system like high temperature vapor compression heat pump, conventional vapor compression heat pump, reverse osmosis and multi-stage flash evaporator. A comparison was done between these systems and the suggested system with constant and variable throttling valve (See Table 1). It is clear that the cost is lower than the other techniques at the illustrated operating conditions. This cost is 50% less than most efficient methods in Ref. [4].

7. Conclusion

Freeze desalination using heat pump technique consumes less power than any other systems due to lower amount of energy needed to move (latent heat of freezing is lower than latent heat of

Table 1
Cost for different water desalination methods, see Ref. [4].

	Multi-stage flash	Reverse osmosis	Conventional vapor compression	Theoretical High temperature vapor compression		Present work 0.5 ice ratio — 25 °C input temperature		
Energy price				Case C $ 5.00/GJ 5.5 Cent/kW	Case A $ 0.50/GJ 1.5 Cent/kWh	8 Cent/kWh	5 Cent/kWh	2 Cent/kWh
Water cost ($/m3)	0.77–1.84	0.64–1.98	0.46–2.5	0.49	0.38	1.85^{c} 0.97^{v}	1.158^{c} 0.6^{v}	0.46^{c} 0.241^{v}
Capital cost (m3/day)	1598–2269	1035–1665	894–1322	884	620			
Heat (MJ/m3)	145–290			30.8	54.3			
Work (MJ/m3)	14.4	21.6–36.0	21.6–36.0	15.7	27.7			

C: constant.
V: variable.

evaporation). The suggested system overcomes a lot of disadvantages of traditional freezing methods of desalination like ice handling, special compressor types, etc. Cost comparing with other methods gives a promising lower cost for the suggested system more than any other systems. The value of heat pump C.O.P. could be improved by using variable throttling system linked to heat sink temperature.

Nomenclature

C.O.P.	Coefficient of performance for heat pump cycle	
$T_{Freezing}$	Water freezing temperature	°K
T_{Supply}	Salt water supply temperature	°K
T_{Sink}	Temperature of water in the tank that serves as heat sink	°K
Q_L	Heat absorbed from the tank that serves as heat source	J
W	Work done to remove heat from the tank that serves as heat source	kJ
m	Total mass of salt water at the beginning of freezing process	kg
C_P	Specific heat constant	kJ/kg °K
L.H.F.	Latent heat of freezing	
dW	Infinitesimal work done	J
P	Compressor power	W
$\Delta\tau$	Infinitesimal time interval	sec
ΔQ_L	Infinitesimal heat removed from the tank served as heat source	J
E	Energy consumed to produce 1 kg of fresh water	J
R	Ice ratio, mass of ice formed in the tank which serves as heat source to total mass of salt water supplied to the tank at start of operation.	
h	Enthalpy	kJ/kg

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