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Heat pump seawater distillation system using passive vacuum generation system

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HIGHLIGHTS

- Passive vacuum system helps seawater distillation using traditional refrigerant heat pumps.
- Passive vacuum system reducing the saturation temperature of seawater.
- The system is suitable for remote areas and could be designed and installed in different sizes.
- The system gives promising competitive energy consumption with other techniques.

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ABSTRACT

The passive system that generate vacuum is a reliable systems and could allow heat pumps that uses traditional refrigerant to be used in seawater distillation process. This could be done by reducing the saturation temperature of seawater to be matched with the operating temperature ranges of these refrigerants. A proposed system uses heat pump for seawater evaporation and condensation. The suggested system is suitable for remote areas and could be designed and installed in the various sizes. Also the compressor could be run with solar PV panels. The followings are a detailed description of system with thermal analysis and energy consumption. It also represents an energy comparison of the proposed system with the other desalination methods. The comparison shows that the system gives promising competitive energy consumption.

1. Introduction

One of the main advantages of heat pump system is that a large amount heat added or removed versus natural process of heat transfer by temperature difference. This process is performed by doing work on working fluids (almost refrigerant). This work almost is several times lower than the quantity of heat removed or added. That means the quantity of heat removed or added by the heat pump system is not actually considered as a cost indicator but the work done through the compressor is the indicator. It is well known that the thermal distillation process of sea water to produce fresh water consumes a large quantity of energy in the form of heat [25]. The combination of heat pump and distillation systems to desalinate seawater may be promising and simple economic method compared with traditional desalination techniques.

V.V. Slesarenko Ref. [1,2], used a heat pump system for waste heat for reducing power consumption in small capacity thermal desalination plant to produce fresh water. An analysis of a vacuum desalination system was provided by R. Senthil Kumar, et al. [3]. Validation was made for experimental data available in the literature. A heat pump work with absorption system instead of vapor compression system was used experimentally for water purification by J. Siqueiros [4]. The main advantage of these units is the ability to be designed as small scale or mobile units. This will allow using it in disaster areas. The cost is competitive compared to the reverse osmosis and the electro dialysis technologies. Warren Rice, [5] used a hydraulic refrigerant compressor to eliminate problems that have been experienced with conventional vapor compression refrigerant compressors used on freeze desalination plants. J.R. Lara, et al. [6], represented a detail economics analysis of MVC. Also Yasu Zhou et al., [8] represents a comprehensive design model of single-effect mechanical vapor recompression (MVR) system. Hisham Ettouney [9] represented a detailed model of the MVC process, including several new design features. D. Yogi Goswami [10] uses the natural forces of gravity and atmospheric pressure to create a vacuum in the solar driven flash desalination system. S. Al-Kharabsheh, [11,13] experimentally studied the operating parameters for solar desalination system based on an innovative passive vacuum concept and compared

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results with theoretical results. The system utilizes natural means (gravity and atmospheric pressure) to create a vacuum. A solar-powered barometric desalination system consists of an airtight barometric chamber positioned to the altitude experimentally studied by I.W. Eames, [12]. W.M. Yang et al., [14] introduced a various methods of enhancing the performance of heat pumps which followed by a review of major hybrid heat pump systems suitable for application with various heat sources. Nominee H., [15], concluded that the production rate increases by increasing the operating temperature, evaporator design temperature for MVC system.

S. Iniyan, [16], supplied a review for solar thermal technologies. Performance analyses of existing designs (study), mathematical simulation (design) and fabrication of innovative designs with suggested improvements (development) have been discussed. Kyaw Thu, et al., [17], developed a system operates at sub-atmospheric pressures and temperature. The system is a combination from a Multi-Effect Distillation (MED) and an adsorption (AD) cycle. Chennan Li, et al. [18] proposed a new combined power and desalination system. This system combines a supercritical organic Rankine cycle (SORC), an educator and a multi-effect distillation (MED) desalination system. Jingwei Hou, et al. [19], introduce a novel process to optimize the utilization of energies. The distillation unit was driven by the waste heat of the spray system. K. Sampathkumar, [20], provide a detailed review of different studies on the active solar distillation system over the years. Mario Reali [21], concerns on design aspects for the recently proposed solar barometric distillation technology for seawater desalting via underground barometric layout. The proposed desalting technology has good energy efficiency and promising technical economic features. Also Zakaria [23,24] uses heat pump to supply a sea water desalination system with low grade heat energy. Heat pump uses R-134a as working fluid. The heat pump C.O.P. reached to be 8 and production rate 1.38 kg/h.

Finally, it could be concluded that seawater desalination techniques still need to be more simple and sustainable and less in energy usage. The main problem, which prevents using heat pumps that uses traditional refrigerants, in seawater distillation, is that the refrigerant maximum operating temperature is much lower than the seawater saturation temperature at atmospheric conditions. V.V. Slesarenako, [1] tried to solve this problem by using pure water as working medium in the heat pump.

In the following a trial to solve this problem by reducing the pressure applied on seawater surface. This will consequently reduce the saturation temperature of seawater. This reduction will make the operating temperature of used heat pump consistent with a heat pump that uses a traditional refrigerant range. The techniques of vacuum pressure

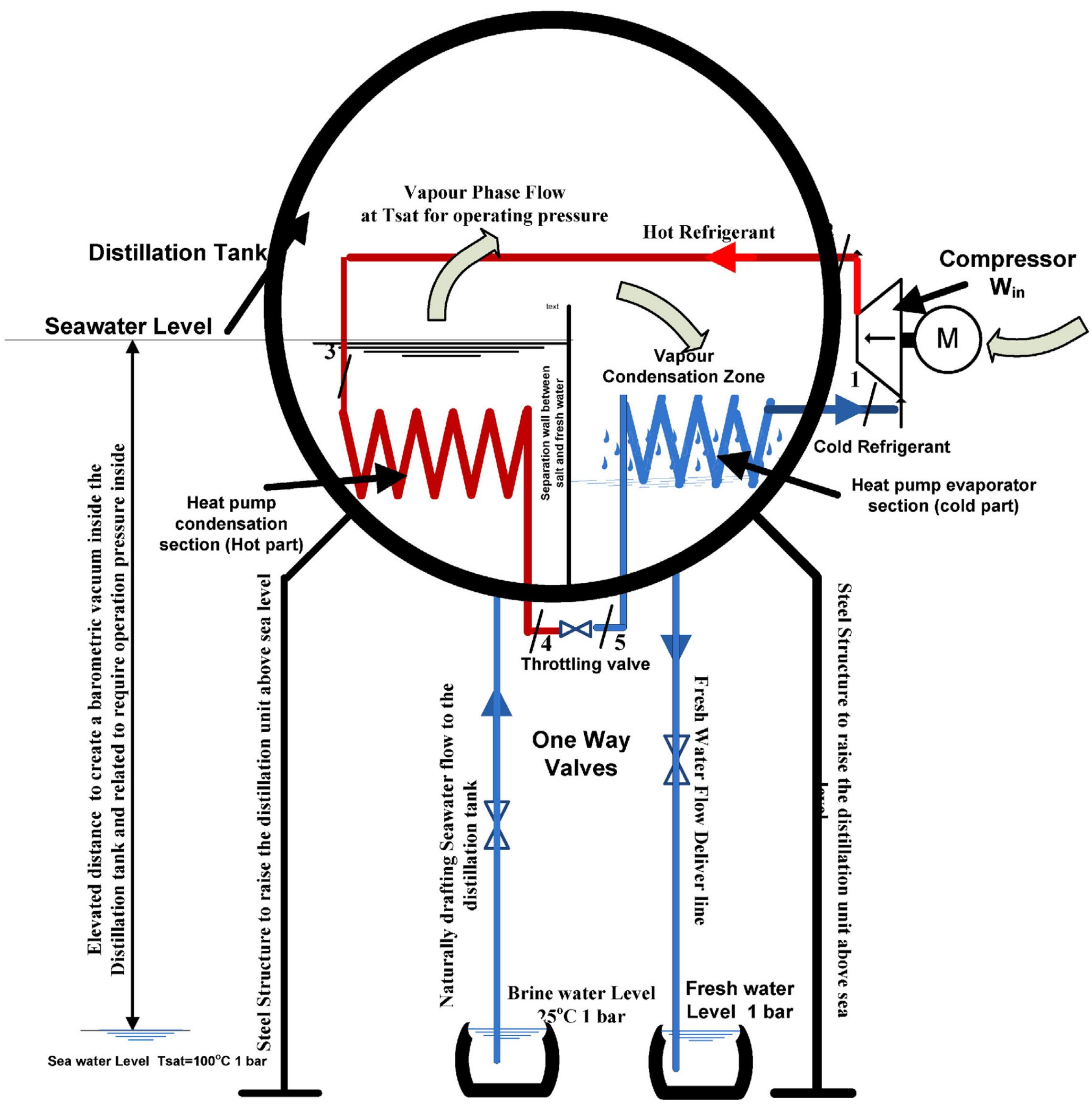

Fig. 1. Layout of the proposed water distillation system.

that is generated by barometric evacuation allows the traditional heat pump working refrigerant to be used in seawater distillation process by reducing the saturation temperature to a proper value suitable for heat pumps that works with traditional refrigerants. This will reduce the cost of using energy in the distillation process. This reduction mainly depends on the type of refrigerant that is used as a working fluid in the heat pump.

2. System description

The proposed system could be described as a single unit of the system that represented by V.V. Slesarenako, [1]. The unit is raised with a steel structure to be above the seawater level by elevation distance to generate a barometric vacuum inside the distillation tank. The system consists of two main parts as shown in Fig. 1. The first part is a heat pump system. The second part is the distillation tank which mainly consists of two main sections. The first section is used in the sea water evaporation process (Heat rejection or condensation section of the heat pump) and the second one is used for sea water condensation (Heat absorption or evaporation section of the heat pump). The distillation tank should be maintained at a vertically elevated distance higher than the level of the source tank of sea water to accomplish a barometric vacuum pressure in the tank. The value of vacuum pressure that was generated on the surface of seawater is mainly depends on the value of vertically elevated distance and affect physically on seawater saturation temperature. In this case, the saturation temperature of sea water will match with operating temperature for the most refrigerants that used in heat pumps. So it effects on the heat pump cycle performance.

The condensation part of the heat pump cycle (hottest part) is used in the evaporation process for seawater at an evaporation process section. The evaporator of heat pump (coldest part) is used in the condensation process of seawater which produces fresh water. The compressor work removes heat from the cold part (fresh water condensation zone) and delivers it into the hottest part of the system (evaporation zone of seawater). This work that done by the compressor is the only type of energy used and there is no need for any other type of energy to be added by any other source of energy in the seawater distillation process except of the seawater and the produced freshwater transportation. The main factors that effect on the operation and performance of the system are saturation temperature of seawater (Actually, it is the vacuum pressure that generated by an vertically elevated distance of sea water level in distillation tank), temperature of seawater and coefficient of performance of the vapor compression cycle (mainly depend on the refrigerant gas type that is used in the heat pump).

The generated barometric vacuum pressure inside the distillation tank for each vertically elevated distance of the distillation tank (sea water saturation temperature) could be estimated from the Eqs. (1) and (2).

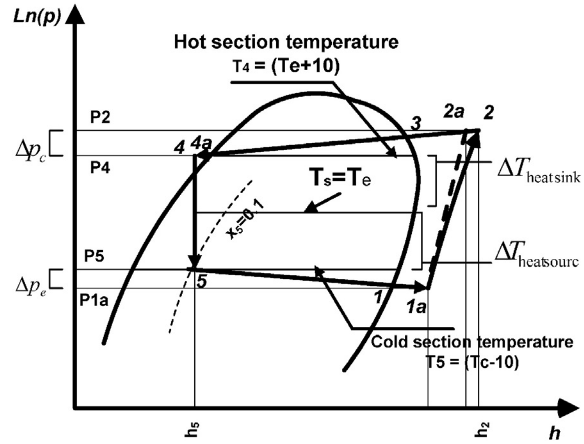

Fig. 3. Realistic vapor compression cycle $Ln(P) - h$ diagram.

$$P_{vac} = P_{atm} - \rho g h_{ele} \quad (\mathrm{N/m^2}) \tag{1}$$

$$h_{ele} = \frac{P_{atm} - P_{vac}}{\rho g} \quad (\mathrm{m}) \tag{2}$$

According to Eq. (2) and water properties [7], the maximum vertically elevated distance that could be achieved at approximately zero value of absolute pressure and its value is not more than 10 m. That means the vertically elevated distance of the distillation tank could be varied from seawater level up to 10 m above sea water level. When the distillation tank was on the seawater level (zero level), the sea water saturation temperature is 100 °C. This saturation temperature decreases with the increase in a vertically elevated distance of the distillation tank (i.e. increasing barometric vacuum pressure inside the distillation tank).

Fig. 2 shows the relation between saturation temperature for seawater and latent heat of vaporization of sea water due to vacuum pressure that generated in the distillation tank versus the vertically elevated distance from seawater level to the level of water inside the distillation tank. The more vertically elevated distance increases the more saturation temperature decrease. The latent heat of vaporization increase with the decrease in saturation temperature consequently with the distillation tank vertically elevated distances Ref. [7].

Fig. 3 shows a diagram for vapor compression cycle on Ln(p)-h diagram. At the evaporator outlet the refrigerant is super-heated which has contributed to the cooling capacity and compressor live time. Due to the pressure drop in the evaporator and the superheating at the evaporator

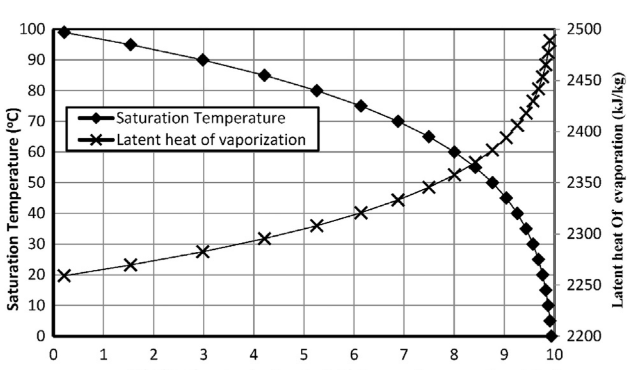

Fig. 2. Saturation temperature of seawater and latent heat of vaporization via elevation of the distillation tank above sea water level, Ref. [7].

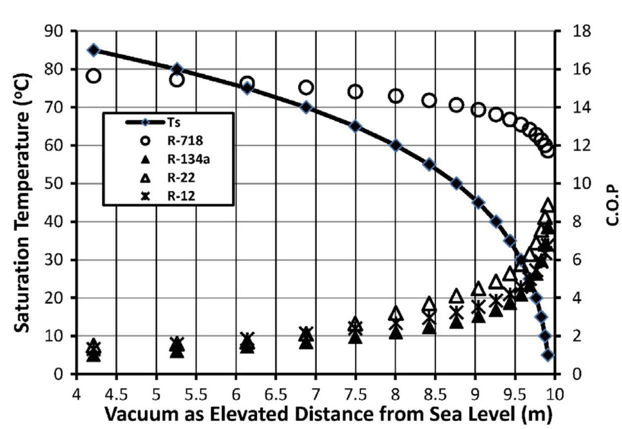

Fig. 4. C.O.P. Heat pump for tested working fluids via vertical elevated distances of distillation tank (h_{ele}).

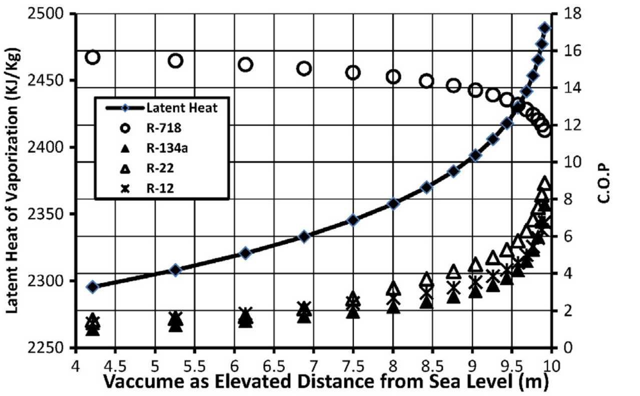

Fig. 5. C.O.P. of heat pump at different working fluids at different vertical elevated distance of the system with latent heat of vaporization at each elevated distance (h_{ele}).

outlet, the compression process starts at lower pressure and further with superheated vapor. The condenser is referred as hot part of the cycle (this is opposite in the physical meaning for the word "condenser") and evaporator is referred as cold part of the cycle (this is opposite in the physical meaning for the word "evaporator").

From all of the above it could be notice that the performance of system depends mainly on the distillation tank vertically elevated distances above the sea level that generates a vacuum pressure inside it. This value of work - the quantity of energy that needed to produce one kilogram of fresh water using the proposed system - changed according to the value of pressure vacuum inside the distillation tank. It also changes the latent heat of evaporation, which is critical for fresh water production rate of the suggested system. Finally the vertically elevated distances of the distillation tank changes both the saturation temperature of sea water and latent heat of vaporization.

3. Calculation procedure

For the points 1a, 2, and 5 on Fig. 3 could be considered as follows

$$T_5 = T_c - \Delta T_{\text{heatsink}} \tag{3}$$

$$T_4 = T_e + \Delta T_{\text{heatsource}} \tag{4}$$

With assumption of $T_4 \cong T_{4a}$

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

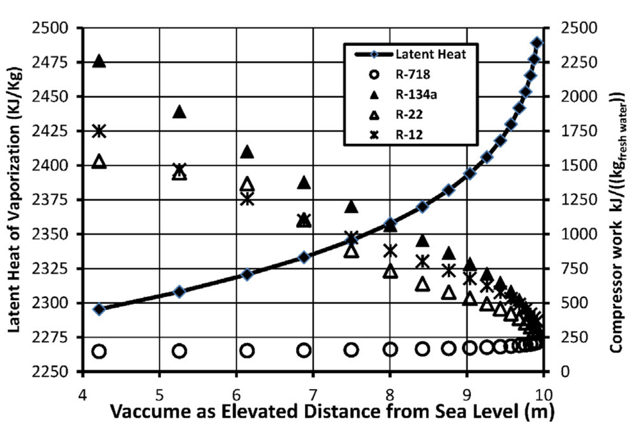

Fig. 6. Compressor work kJ/((kg$_{\text{fresh water}}$)) at different heat pump working fluids at different vertical elevated distance from sea level (h_{ele}).

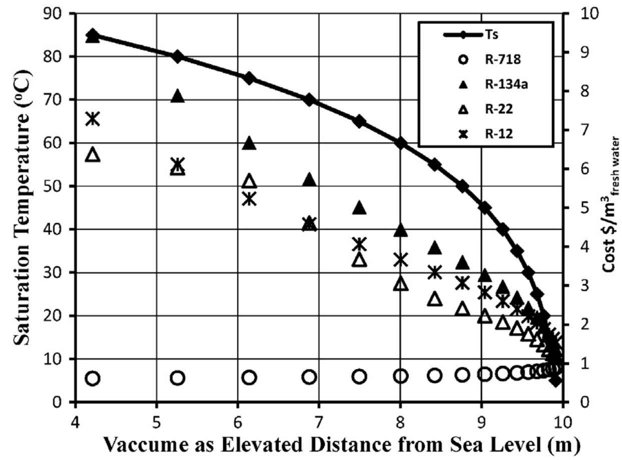

Fig. 7. Expected cost of the system with different heat pump working fluids (electricity price 1.5 \$/kW·h).

From practical consideration

$$\Delta p_e = p_5 - p_{1a} \approx 5 \text{ psi} \tag{6}$$

$$S_{1a} = S_{2a} \tag{7}$$

$$P_2 \approx P_{2a} \tag{8}$$

$$\Delta P_c = P_2 - P_4 = P_{2a} - P_4 \approx 5 \text{ psi} \tag{9}$$

At evaporator exit point the value of C_p and C_V should be estimated to calculate the value of γ

$$\gamma = \left(\frac{C_p}{C_V}\right)_{at\ T_1} \tag{10}$$

$$T_2 = T_1 \left(\frac{P_2}{P_{1a}}\right)^{\frac{\gamma-1}{\gamma}} \tag{11}$$

At T_2 and P_2 get the value of h_2 and then calculate the compressor work

$$W_{comp} = h_2 - h_{1a} \tag{12}$$

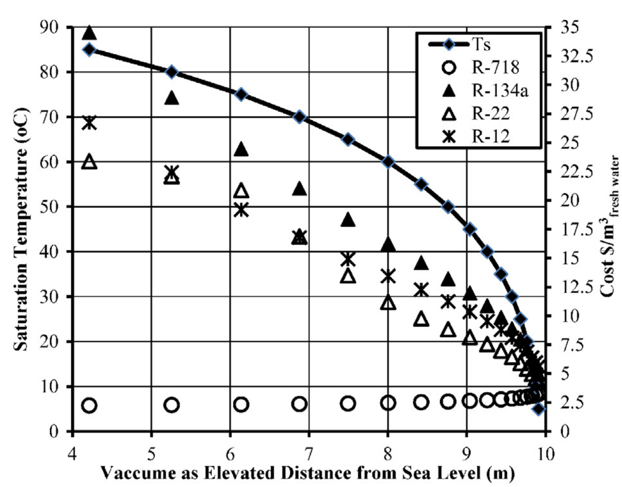

Fig. 8. Expected cost of the system with different heat pump working fluids (electricity price 5.5 \$/kW·h).

Table 1
Comparison between energy requirements for different water desalination methods stated in Ref. [6], Ref. [22] and that required for suggested system.

Energy	MSF	RO	Conventional vapor compression	Theoretical high temperature vapor compression		Present system			
				Case C \$5.00/GJ 5.5 Cent/kW·h	Case A \$0.50/GJ 1.5 Cent/kW·h	$T_{sat}=80$ $h_{ele}=9.88$ m R-718 working fluid		$T_{sat}=10$ $h_{ele}=4.25$ m R-718 working fluid	
						5.5 Cent/kW·h	1.5 Cent/kW·h	5.5 Cent/kW·h	1.5 Cent/kW·h
Water cost (\$/m^3)	0.77–1.84	0.64–1.98	0.46–2.5	0.49	0.38	2.27	0.62	4.27	1.16
Heat (MJ/m^3)	145–290			30.8	54.3	–		–	
Work (MJ/m^3)	14.4	21.6–36.0	21.6–36.0	15.7	27.7	146		280	

$$\text{C.O.P.} = \frac{h_1 - h_4}{W_{comp}} \tag{13}$$

The cost to produce one kilogram of fresh water is estimated according to Eqs. (14) and (15).

$$\left(\begin{matrix}\text{Water}\\ \text{productivity}\end{matrix}\right) = \left(\frac{60*60*\text{Compressor Work(kW)}}{\text{Latent heat of vaprization}(h_{fl}))(\text{kJ/kg}_{\text{fresh water}})}\right) \quad \text{kg/h r} \tag{14}$$

$$\text{Cost} = \left(\text{price}\left(\frac{/\text{kW hr})*\text{Compressor Work(kW)}}{\text{Water Productivity}(\text{kg}_{\text{fresh water}}/\text{hr}))}\right)/\text{kg}\right. \tag{15}$$

By assuming values for vertically elevated distances of the distillation tank above sea level (vacuum pressure inside the distillation tank increases with the increase of elevated vertical distance) which changes the sea water saturation temperature T_{sat}, the value of compressor work could be determined.

The software was designed to calculate the properties of R-12, R-22, R-134a, and R-718 (water) at different points of operation for heat pump cycle that shown in Fig. 3. The program is used to evaluate the coefficient of performance for heat pump cycle operated with above mentioned refrigerants at different operating conditions.

Figs. 4 and 5 show the variation of C.O.P. for different heat pump working fluids with variation of vacuum as elevated distance from sea level for the system. These plots are accompanied with variation of saturation temperature of sea water and variation of latent heat of vaporization both with the variation of vacuum as elevated distance. The major note is that R-22, R-12 and R-134a have a trend in operation and R-718 (pure water) has another trend.

The C.O.P. value for heat pump that operates with working fluid R-22, R-12, R-143a, increases with the increase of vacuum pressure (vertically elevated distances). This is due to decrease in saturation temperature to be closer to working fluid best operating temperature range. The C.O.P. may reach to be about 8, 9, and 7.5 at the best operating points of saturation temperature 5 °C. This will happened at high vacuum elevated distance 9.8 m above sea level. For pure water as a working fluid the direction of C.O.P. value reversed i.e. the C.O.P. value decreases with the increase of saturation temperature of sea water due to that more decrease of it the more increase in compressor work. This happens due to shifting far away about the best operating temperature range.

Fig. 6 shows the variation of expected compressor work for the different heat pump working fluid at different elevation from the sea level. The compressor work tends to decrease with the increase of elevated vertical distance for most of working fluid except R-718 (pure water). This due to that the most of refrigerants best operating temperature range is lower than that of evaporation of sea water at different elevation distance. For heat pump working fluid R-718 the compressor work increases from (146 kJ/kg$_{\text{fresh water}}$) to be (212.5 kJ/kg$_{\text{fresh water}}$) with increase of elevated vertical distance from 4.25 m to 9.88 m.

Figs. 7 and 8 show the variation of cost for each meter cubic of fresh water at electricity cost 1.5 \$/kW and 5.5 \$/kW respectively according to Eq. (15). The cost is comparable with the other cost of desalination technology, see Table 1.

J.R. Lara [6] represented an economic analysis for different desalination systems. A comparison was made between the proposed system and other suggested system (Table 1). It is clear that the energy cost is competitive with other techniques at the illustrated operating conditions.

Using water as refrigerant is recommended in recent times more over than other last times due to that it is environmentally friend than other refrigerant, and also the economic costs and safety properties of refrigerants are heavily started to taken into consideration in last decade [24]. So the especially compressors - especially multi-stage turbo compressors with intercoolers between stages - expected to be overcome its technical problems that face using water as a working fluid in heat pumps.

4. Conclusion

Sea water distillation using heat pump combined with passive vacuum system to distillate sea water is very simple and easy for construction and fast start up. It doesn't need a special care or pretreatment and high technical stuff to work on so that the system covers a lot of disadvantages of other desalination techniques. The cost for each meter cubic of fresh water is competitive with other techniques of sea water desalinations. The system is could be installed in different scale and suitable for rear areas. Using water as a heat pump working fluid is much beneficial than other type of refrigerant.

Nomenclature

C.O.P.	Coefficient of performance for heat pump cycle	
T	Temperature	K
ρ	Density	kg/m^3
S	Entropy	kJ/kg·K
W	Compressor Work	kJ
C_P,C_V	Specific heat constants	kJ/kg·K
P	Pressure	N/m^2
h	Enthalpy	kJ/kg
γ	Specific gas constant $\gamma = \left(\frac{C_p}{C_v}\right)_{\text{at specified temperature}}$	–
g	Gravity acceleration	m/s^2
hr	Hour	–

Abbreviation

comp	Compressor
vac	Vacuum
atm	Atmospheric
ele	Elevation
sat	Saturation

e Evaporator
c Condenser
psi Pressure unit (pound per square inch)
$\Delta T_{heatsink}$ Temperature difference between refrigerant and seawater at evaporator section
$\Delta T_{heatsource}$ Temperature difference between refrigerant and seawater at condenser section

References

[1] V.V. Slaresnko, Heat pump as a source of heat energy for desalination of sea water, Desalination 139 (2001) 405–410.
[2] V.V. Slaresnko, Desalination plant with absorption heat pump for power station, Desalination 126 (1999) 281–285.
[3] R. Senthil Kumar, et al., Analysis of jet pump assisted vacuum desalination system using power plant waste heat, Desalination 179 (2005) 345–354.
[4] J. Siqyereiros, F.A. Hollond, Water desalination using heat pumps, Energy 25 (2000) 717–729.
[5] W. Rice, D.S.C. Chau, Freeze desalination using hydraulic refrigerant compressors, Desalination 109 (1997) 157–164.
[6] J.R. Lara, G. Noyes, M.T. Holtzapple, An investigation of high operating temperature in mechanical vapor-compression desalination, Desalination 227 (2008) 217–232.
[7] W. Wagner, H.-J. Kretzschmar, International steam tables properties of water and steam based on the industrial formulation, IAPWS-IF97ISBN 978-3-540-21419-9, Springer-Verlag, Berlin Heidelberg, ISBN: 978-3-540-74234-0, 1998 and 2008, http://dx.doi.org/10.1007/978-3-540-74234-0 (©).
[8] Y. Zhou, C. Shi, G. Dong, Analysis of a mechanical vapor recompression wastewater distillation system, Desalination 353 (17) (November 2014) 91–97.
[9] H. Ettouney, Design of single-effect mechanical vapor compression, Desalination 190 (2006) 1–15.
[10] S.C. Maroo, D.Y. Goswami, Theoretical analysis of a single-stage and two-stage solar driven flash desalination system based on passive vacuum generation, Desalination 249 (2009) 635–646.
[11] S. Al-Kharabsheh, D. Yogi Goswami, Experimental study of innovative solar water desalination system utilizing a passive vacuum technique, Sol. Energy 75 (2003) 395–401.
[12] I.W. Eames, G.G. Maidment, A.K. Lalzad, A theoretical and experimental investigation of a small-scale solar-powered barometric desalination system, Appl. Therm. Eng. 27 (2007) 1951–1959.
[13] S. Al-Kharabsheh, D. Yogi Goswami, Analysis of an innovative water desalination system using low-grade solar heat, Desalination 156 (2003) 323–332.
[14] K.J. Chua, S.K. Chou, W.M. Yang, Advances in heat pump systems: a review, Appl. Energy 87 (2010) 3611–3624.
[15] N.H. Aly, A.K. El-Fiqi, Mechanical vapor compression desalination systems - a case study, Desalination 158 (2003) 143–150.
[16] M. Thirugnanasambandam, S. Iniyan, R. Goic, A review of solar thermal technologies, Renew. Sust. Energ. Rev. 14 (2010) 312–322.
[17] K. Thu, Y.-D. Kim, G. Amy, W.G. Chun, K. Choon, A hybrid multi-effect distillation and adsorption cycle, Appl. Energy 104 (2013) 810–821.
[18] L. Chennan, D. Yogi Goswami, A. Shapiro, E.K. Stefanakos, G. Demirkaya, A new combined power and desalination system driven by low grade heat for concentrated brine, Energy 46 (2012) 582–595.
[19] J. Hou, H. Cheng, D. Wang, X. Gao, C. Gao, Experimental investigation of low temperature distillation coupled with spray evaporation, Desalination 258 (2010) 5–11.
[20] K. Sampathkumar, T.V. Arjunan, P. Pitchandi, P. Senthilkumar, Active solar distillation—a detailed review, Renew. Sust. Energ. Rev. 14 (2010) 1503–1526.
[21] M. Reali, Solar barometric distillation for seawater desalting part III: analyses of one-stage and two-stage solar vapor thermo-compression distillation technologies, Desalination 207 (2007) 304–323.
[22] M. Khayet, Solar desalination by membrane distillation: dispersion in energy consumption analysis and water production costs (a review), Desalination 308 (2013) 89–101.
[23] Z.M. Amin, M.N.A. Hawlader, Analysis of solar desalination system using heat pump, Renew. Energy 74 (2015) 116e123.
[24] A. Kilicarslan, N. Muller, A comparative study of water as a refrigerant with some current refrigerants, Int. J. Energy Res. 29 (2005) 947–959.
[25] H.T. El-Dessouky, H.M. Ettouney, Fundamentals of Salt Water Desalination, Elsevier, 2002.