Numerical Study on Natural Vacuum Solar Desalination System with Varying Heat Source Temperature

H Ambarita*
Sustainable Energy Research Centre, Faculty of Engineering University of Sumatera Utara, Jl. Almamater Kampus USU Medan 20155

*himsar@usu.ac.id

Abstract. A natural vacuum desalination unit with varying low grade heat source temperature is investigated numerically. The objective is to explore the effects of the variable temperature of the low grade heat source on performances and characteristics of the desalination unit. The specifications of the desalination unit are naturally vacuumed with surface area of seawater in evaporator and heating coil are 0.2 m² and 0.188 m², respectively. Temperature of the heating coil is simulated based on the solar radiation in the Medan city. A program to solve the governing equations in forward time step marching technique is developed. Temperature of the evaporator, fresh water production rate, and thermal efficiency of the desalination unit are analysed. Simulation is performed for 9 hours, it starts from 8.00 and finishes at 17.00 of local time. The results show that, the desalination unit with operation time of 9 hours can produce 5.705 L of fresh water and thermal efficiency is 81.8 %. This reveals that varying temperature of the heat source of natural vacuum desalination unit shows better performance in comparison with constant temperature of the heat source.

1. Introduction
Saline water desalination has been known by human being since long time ago in order to fill the need for freshwater. Recently, many regions such as Middle East, Arabic countries, some Asians, Europe, Africa, North and South America, and Australia are using desalination system to meet their need for freshwater [1]. In Indonesia, some areas also use desalination to produce freshwater for their living. There are many methods can be used in producing freshwater from seawater. Some are multi-stage flash, multi effects distillation, vapor compression, reversal osmosis, electro-dialysis, etc. As energy source, the conventional desalinations use fossil fuel to power the system. Based on a study, around 10,000 ton of fossil fuel was burnt to produce freshwater from the seawater [2]. Thus, desalination system is a sector that emits Green Houses Gases (GHGs) significantly. This fact strives researchers to power desalination using renewable energy resources [3].

Solar energy is the most used renewable source for producing freshwater from the seawater, it is estimated around 57%. Solar-powered saline water desalination (or solar desalination) is predicted will be increase significantly in the future [4]. There are several methods of solar desalination found in the literature such as solar thermal, photovoltaic, hybrid, direct, and indirect systems. This paper focuses in natural vacuum solar desalination. Several studies on natural vacuum solar desalination have been found in literature. Kharabsheh and Goswani [5,6] reported a theoretical analysis on water desalination system using low-grade solar heat in vacuum. The vacuum, under which water can be rapidly
evaporated, is resulted naturally by gravity and atmospheric pressure. In the study a preliminary experimental work also carried out to validate the theoretical results. The effects of withdrawal rate, depth of the water body in evaporator, temperature of heat sources, and condenser temperature, on the performance were studied. The results showed that the proposed system can produce 6.5 kg/day.m² evaporator area which is better than flat-basin solar still. The main advantage of natural vacuum system is it can be powered with low-grade of heat source. This can be supplied with solar collector with high efficiency. However, the vacuum solar desalination still showed a low performance. In order to increase the performance, many modifications have been proposed by researchers. Maroo and Goswani [7] proposed a theoretical analysis on single-stage and two-stage solar driven flash desalination system based on passive vacuum generation. The main components of their proposed system are evaporator(s), condenser(s), collection tanks, heat source, and seawater circulation pump. Partial heat recovery is attained by first passing the feed water through the condenser(s), followed by the heat source. Additional distillate output is obtained in the second stage of the two-stage system without any extra heat addition, since high temperature brine from the first stage is passed and flashed in the second stage. The results showed that when the system is coupled with solar collector of 1 m² area, a single-stage system produces 5.54 kg of fresh water in 7.83 hours, while the two-stage system produces 8.66 kg in 7.7 hours. Gude et al [8] performed a theoretical study on low temperature desalination using solar collectors augmented by thermal energy storage. The objective of installing a thermal energy storage tank is to design a continuous flow low temperature desalination powered by solar collector system. The results show that a low temperature desalination system designed to produce 100 Liter/day of fresh water require a solar collector area of 15 m² with 1 m³ of Thermal Energy Storage (TES) volume. The additional area of solar collector helps the TES to accumulate excess energy which can be stored and supplied on cloudy day or low solar insolation day. Gude et al [9] reported a feasibility study on two-stage low temperature desalination. The principle of the proposed system is similar to multi effect boiling desalination process. Theoretical analysis and preliminary field tests in Puget Sound bay area were carried out. The results proved that the two-stage desalination process has potential for stands alone small to large scale applications in water and energy scarce rural areas with specific energy consumption of 1500 kJ/kg of freshwater. Recently Ambarita [10] reported a study on the natural vacuum solar desalination in a lab scale unit. In order to simulate the heat source an electric heater with constant temperature is used. The characteristics and performance of the system are presented.

Those reviewed studies showed that attention of researchers on the natural vacuum solar desalination is increasing. However, the performance is still low. Thus several modifications are extremely needed. To propose a good modification mechanism and characteristics of the natural vacuum desalination should be explored more. In this work a lab scale natural vacuum desalination with varying heat source temperature will be investigated numerically. The main objective is to explore the characteristic and performance of the natural vacuum desalination system when the temperature of the heat source is varied based on the solar radiation in the solar collector. The comparisons with constant heat source temperature will be made. The results of the present study are expected to supply the necessary information on development high performance solar desalination system.

2. Method
The schematic diagram of the natural vacuum solar desalination unit is shown in Figure 1. The main components of the unit are seawater, brine, and freshwater tanks, evaporator, condenser, and solar collector. The solar collector is used to heat the hot water and a pump which is powered by a solar cell. The natural vacuum in the unit is resulted by placing the unit in a location at10.3 m above the water level of seawater tank. The solar energy will heat the transfer fluid in the solar collector and a pump will be used to send it to evaporator. The transfer fluid will supply an amount of heat into the system. Thus, the seawater temperature will raise. It results in evaporation of water with a rate of $V_e$ [m³/s]. The vapor will be condensed in the condenser and becoming freshwater. Since the water content
decreasing, salinity of seawater in the evaporator will increase. The increasing salinity of seawater will decrease the evaporation rate in the evaporator. To avoid this, brine water in the evaporator will be withdrawn with a rate of $\dot{V}_w$ [m$^3$/s]. Thus the new seawater will be injected to the evaporator with a volume rate of $\dot{V}_i$ [m$^3$/s]. Here, ambient temperature and solar radiation are recorded by using HOBO micro station data logger. The measured radiation will be used to simulate temperature of the heat source in the evaporator and measured ambient temperature will be used to set boundary condition in the condenser.

![Figure 1](image_url)

**Figure 1.** Schematic diagram of the natural vacuum solar desalination unit [10]

### 2.1. Governing Equations

In order to perform the simulation, governing equations upon the system are developed. By implementing the mass conservation in the evaporator will give:

$$\frac{d}{dt}(\rho V)_s = \rho_i \dot{V}_i - \rho_w \dot{V}_w - \rho_s \dot{V}_e$$

(1)

Where $V$ [m$^3$] and $\rho$ [kg/m$^3$] is volume and density of the seawater in the evaporator, respectively. The subscripts of $s$, $i$, and $e$ represent seawater, injected, and evaporation, respectively. Solute conservation in the evaporator gives the following equations:

$$\frac{d}{dt}(\rho CV)_s = (\rho C)_i \dot{V}_i - (\rho C)_s \dot{V}_w$$

(2)
Where $C$ [%] is the solute concentration. It is assumed that no stratification of the seawater temperature in the evaporator. Thus, conservation of energy in the seawater gives the following equation:

$$ \frac{d}{dt}(\rho c_p V T)_x = \dot{Q}_m + (\rho c_p T)V_i - (\rho c_p T)V_a - \dot{Q}_w - \dot{Q}_{\text{wall}} $$  

(3)

Where, $\dot{Q}_m$ [Watt] and $c_p$ [J/kgK] is the heat transfer rate from heater and specific heat capacity, respectively. Here the heat transfer rate from the heater to seawater in the evaporator is calculated by the following equation.

$$ \dot{Q}_m = hA(T_{sc} - T_i) $$  

(4)

Where $T_{sc}$ [°C] is solar collector temperature which is modelled by using measured solar radiation.

In the present study, the evaporator material temperature is divided into bottom, side wall and top wall. The heat capacity of the evaporator material is taken into account. The energy conservation in the bottom plate of the evaporator gives:

$$ \frac{\partial}{\partial t}(\rho c_p T)_b = h_s (T_s - T_b) - h_a (T_s - T_a) $$  

(5)

Where the subscript $b$ and $a$ refer to bottom of the evaporator and ambient air, respectively.

In the present study, solar collector will be used to heat the transfer fluid and it will be sent to the heater inside the evaporator. The heater is made of cylindrical tube and it is immersed horizontally into the seawater in the evaporator. The evaporation rate between two chambers, one contains seawater and the other freshwater, which is connected via a vacuum channel, can be calculated by the following equation[11].

$$ V_e = A_{\text{surf}} \alpha_m \left( \frac{P(T_i)}{(T_i + 273)^{0.5}} - \frac{P(T_f)}{(T_f + 273)^{0.5}} \right) $$  

(6)

Where $V_e$ [m$^3$/s] and $A_{\text{surf}}$ [m$^2$] are volume evaporation rate and evaporation surface, respectively. The parameter $\alpha_m$ [kg/m$^2$.Pa.s.K$^{0.5}$] is an empirical coefficient which is developed from experiments. Al-Kharabsheh and Goswani [5] suggested the following value.

$$ 10^{-7} \leq \alpha_m \leq 10^{-6} $$  

(7)

In the previous study [10], several calculations and experimental data are used to develop a suitable value of $\alpha_m$. The value of $\alpha_m = 9 \times 10^{-6}$ is suitable for the present simulation. Thus this value will be used for all simulations. In equation (5), $P$ [Pa] is vapor pressure, it is as a function of temperature inside the evaporator.

As a note the objective of natural vacuum desalination is to produce freshwater by using energy efficiently. Based on this objective, two parameters to show the performance of the system will be used. The first parameter is amount of freshwater produced which is calculated using the following equation.

$$ V_{\text{tot}} = \int_{0}^{t_{\text{end}}} V_e \, dt $$  

(8)

Where $V_{\text{tot}}$ [m$^3$] and $t_{\text{end}}$ [s] is the total volume of the produced freshwater and time at the end of experiment/calculation, respectively. The second parameter is thermal efficiency which is defined as ratio of useful energy to energy input.

$$ \eta_{\text{th}} = \frac{Q_{\text{useful}}}{\dot{Q}_m} = \frac{\int_0^{t_{\text{end}}} \rho V_s h_f \, dt}{\int_0^{t_{\text{end}}} \dot{Q}_m \, dt} $$  

(9)
The equations to calculate other parameters such as heat transfer coefficient, vapor pressure, and thermal properties of seawater are not shown here and can be found in Ambarita [10].

2.2. Numerical solution
All of the governing equations are converted from differential equation form into linear equation system by using discretization technique. The used technique is forward time step marching technique. Thus the governing equations, equation (1), (2), (3), (5), and equation (6), will be converted into the following equations, respectively.

\[ C^{j+1} = C^j + \frac{\Delta t}{\rho V} \left[ (\rho C)_i V_i - (\rho C)_w \right] \]

\[ T_s^{j+1} = T_s^j + \frac{\Delta t}{\rho c_p V_s} \left[ Q_m + (\rho c_p T)_i V_i - (\rho c_p T)_w V_w - Q_e - Q_{wall} \right] \]

\[ T_b^{j+1} = T_b^j + \frac{\Delta t}{(\rho c_p V)_b} \left[ \frac{N_{ub,b}}{L_b} k_b (T_s - T_b) - \frac{N_{ub,c}}{L_b} k_b (T_{wall} - T_b) \right] \]

\[ T_d^{j+1} = T_d^j + \frac{\Delta t}{(\rho c_p V)_d} \left[ \frac{N_{ud,b}}{L_d} k_d (T_s - T_d) - \frac{N_{ud,c}}{L_d} k_d (T_{wall} - T_d) \right] \]

\[ T_t^{j+1} = T_t^j + \frac{\Delta t}{(\rho c_p V T)} \left[ \frac{N_{ut,b}}{L_t} k_t (T_s - T_t) - \frac{N_{ut,c}}{L_t} k_t (T_{wall} - T_t) \right] \]

In the above equations, the superscript \( j \) refers to the present value and \( j + 1 \) is the value at the next time step. Due to stability consideration, the values of \( \Delta t \) must be relatively low. In this work, the value of \( \Delta t = \) 1sec is used. This value shows stability. Here, a Fortran code program is written to solve the transient governing equations which are coupled with all of the heat transfer equations and thermal properties equations. The Fortran, stand for Formula Translation, is a general purpose, imperative programming language that is especially suited to numeric computation and scientific computing.

3. Results and Discussions
In this work the weather condition (radiation and ambient temperature) will be measured using HOBO micro station data logger. By using the measured radiation, heat source temperature in the evaporator will be modeled. On the other hand, the measured ambient temperature will be used as boundary condition in the condenser. Here, the characteristic of the system with varying heat source temperature will be investigated. In the previous literature, the experiments and simulations have been performed with constant heat source temperature [10]. Evaporator of the system is a hollow cylinder with conical cover on the top made of Stainless steel 304 with a thickness of 5.8 mm. The diameter and the height of the evaporator are 500 mm and 150 mm, respectively. The height of the conical top is 120 mm. In all experiments the depth of seawater in the evaporator is 80 mm. The condenser is a horizontal circular tube with circular fins and also made of Stainless steel 304 with a thickness of 2.54 mm. The inside diameter and the length of the condenser is 100 mm and 500 mm, respectively. The number of fin is 10 fins with a diameter and thickness of 254 mm and 0.6 mm, respectively. The distance between fins is 40 mm. The condenser and evaporator are connected by using a flange with a diameter and thickness of 128 mm and 15 mm, respectively. The results of simulations with constant and varying heat source temperatures will be compared.

3.1. Numerical validations
In order to make sure the developed numerical solution is free of error, a numerical validation has been performed. Experiments using the experimental apparatus [10] are carried out for 6 days. In the experiments, electric heater is used as heat source with a constant temperature at 50°C. At the same day, simulations are also carried out. The produced freshwater resulted from numerical and experimental works are presented in Figure 2. The figure shows that, in the experimental works, the produced freshwater varies from 0.95 L to 1.2 L with an average value of 1.14 L. On the other hand, the simulation works show that the produced freshwater varies from 1.268 L to 1.301 L with an average value of 1.2675 L. Since the heat source temperature is constant, the variations in the results are strongly affected by ambient temperature. The produced freshwater from numerical results are relatively higher than numerical one. This is because, in the numerical work some parameters are modelled. The difference of experimental and numerical results, in average, is only 9.9%. The results are also compared with work of Al-Kharabsheh [3,4]. It shows only small discrepancy which is less than 10%. This fact reveals that numerical and experimental results shows a good agreement. The developed numerical method will be used to simulated the system.

![Figure 2. Comparison of numerical and experimental results](image-url)
3.2. Characteristics of the system

Figure 3 shows heat source temperature, evaporator chamber, and condenser temperatures. It can be seen that heat source temperature varies from 30°C at 8.00 local time [WIB] and it reaches the maximum value of 85°C at 13.00 WIB. After reaching the maximum value it decreases gradually and reach minimum temperature of 28°C at 17.00 WIB. Evaporator temperature shows the same trend as heat source temperature. As a note, heat source will heat the evaporator. In general, heat source temperature is bigger than evaporator temperature. However, after 15.45 WIB the heat source temperature is lower than evaporator temperature. This is because the solar radiation decreases significantly. Since the condenser made of stainless steel, its heat capacity still can release the heat to keep the temperature higher than the heat source temperature. The evaporation rate [gram/sec] of the system is also shown in Figure 3. The figure reveals that evaporation rate and heat source temperature show the same trend. It will reach its maximum value when heat source temperature reaching its maximum value. After 14.45 WIB, when the heat source temperature is lower than evaporator temperature, the desalination unit still can produce freshwater. This is because evaporator temperature is higher than temperature saturation of the freshwater in the evaporator.
Heat transfer rate and transient thermal efficiency of the system are shown in Figure 4. The heat transfer rate injected to the system, heat evaporation, and heat loss are shown by red, blue, and brown lines, respectively. The figure shows that, from 8.00 to 12.00 WIB the heat injected to the system increases as time increases. After this, the heat injected decreases as time increases. It can be seen, after 15.45 WIB the heat injected to the system will be negative. In other words, evaporator will supply energy to the heat source. This is because solar radiation not sufficient enough to keep the heat source temperature higher than evaporator. This fact reveals that, in the real case of natural vacuum desalination system, there is a high possibility that the heat source temperature will be lower than evaporator temperature. This is bad for the performance of the system in producing the freshwater. In order to avoid this condition, the flow of the transfer fluid from solar collector unit to evaporator should be stopped. Thus, evaporation heat will be supplied by heat capacity of the evaporation material. In Figure 4 transient efficiency of the unit is shown by black line. It can be seen clearly that transient thermal efficiency increases in all time. However, after 13.00 WIB, transient thermal efficiency will increase significantly and towards infinite number. This is because the heat input to the system decrease significantly and it close to zero and finally reach negative value.

3.3. Comparison with constant heat source temperature
The characteristics of the natural vacuum solar desalination system have discussed in the previous subsection. In order to explore the different of the system when it is operated with constant heat source temperature and varying heat source temperature, here a simulation is also carried out to provide the comparison. In the simulation with constant heat source temperature, the temperature is fixed at 60°C. Figure 5 shows useful heat and cumulative thermal efficiency of the system. As a note, energy from the heat source goes to useful heat and heat loss. The useful heat is defined as the cumulated heat that is used to evaporate the freshwater from the seawater. The figure shows that the useful heat for case with constant heat source temperature increase linearly. This is because the rate of heat evaporation is almost constant as the heat source temperature is constant. On the other hand, for case with varying temperature, in the beginning the useful heat increases significantly. After 14.00 WIB the gradient of useful heat decrease. This is because the heat input from the source to the system decrease.
The comparison shows that the useful heat of the system with varying heat source temperature is better than the constant temperature.

Figure 5 also shows the comparison of total efficiency of both systems. The efficiency of the system with constant heat source temperature shown by solid red line. It can be seen that for all simulation time, the efficiency increases as time increase. In the beginning, from 8.00 WIB to 10.00 WIB, the efficiency increases significantly. This is because the heat source already constant at high temperature while the condenser still at a lower temperature. Thus the heat input to the system is used efficiently. On the other hand, after 12.00 WIB the gradient of efficiency decreases as time increase. This is because the temperature difference between evaporator and condenser already decrease. For system with variable heat source temperature, the efficiency increases for almost all time. Only in the beginning the efficiency is close to zero. This is because the heat source temperature is still very low in the beginning. The comparison shows that the total efficiency of system with varying and constant temperatures are 61.7% and 81.8%, respectively.

![Figure 5. Comparison of useful heat and thermal efficiency with constant and varying heat source temperatures of the system](image)

Freshwater produced and its production rate are also compared for both systems. Figure 6 shows the freshwater production rate and fresh water produced for both systems. The figure shows that for case with constant heat source temperature, freshwater produced increases linearly as time increase, shown by blue solid line. This is because the production rate is almost constant during simulation, shown by red solid line. As expected the constant heat source temperature will produce constant production rate. Even though, condenser temperature is not constant the effect to the production rate is not significant. This suggest that effect of the condenser temperature to the production rate is not very significant.

For the case with varying heat source temperature, fresh water produced increase significantly in the beginning and almost constant at the end, shown by blue dashed line. This is because the production rate shows similar trend with heat source temperature, shown by red dashed line. The curve shows sinusoidal pattern. The produced freshwater for systems with constant and varying heat source temperatures are 5.709 L and 2.81 L, respectively. This fact reveals that the natural vacuum solar desalination with varying heat source temperature is better than constant heat source temperature.
Figure 6. Comparison of produced freshwater and production rate with constant and varying heat sources temperatures

4. Conclusions
A natural vacuum solar desalination system has been studied numerically. The developed governing equations have been solved using forward time step marching technique. Measurement of ambient temperature and solar radiation are used to develop boundary conditions. The simulations show the different characteristics of the desalination system with constant and variable heat source temperatures. The results show that the desalination unit with varying heat source temperature for operation time of 9 hours (8.00 to 17.00 WIB) can produce 5.705 L of freshwater and thermal efficiency of 81.8 %. On the other hand, the same system with constant heat source temperature can only produce 2.81 L of freshwater and thermal efficiency of 61.7%. This reveals that varying temperature of the heat source of natural vacuum desalination unit shows better performance in comparison with constant temperature of the heat source. The simulation results suggest that in the real case of natural vacuum desalination system, there is a high possibility that the heat source temperature will be lower than evaporator temperature. This is bad for the performance of the system in producing the freshwater. In order to avoid this condition, the flow of the transfer fluid from solar collector unit to evaporator should be stopped.

Acknowledgments
The author gratefully acknowledges that the present research is supported by Ministry of Research and Technology and Higher Education Republic of Indonesia. The support was under the research grant PUPT USU of Year 2016.
References
[1] M.A. Eltawil, Z. Zhengming, L. Yuan, A review of renewable technologies integrated with desalination systems, Renewable Sustainable Energy Reviews 13 (2009) 2245–62.
[2] S. Kalogirou, Seawater desalination using renewable energy sources. Prog Energy Combust Sci 31 (2005) 242–81.
[3] A. Subramani, M. Badruzzaman, J. Oppenheimer, J.G. Jacangelo, Energy minimization strategies and renewable energy utilization for desalination: a review, Water Res 45 (2011) 1907–20.
[4] S. Al-Kharabsheh, D.Y. Goswami, Analysis of an innovative water desalination system using low-grade solar heat. Desalination 156 (2003) 323–32.
[5] S. Al-Kharabsheh, D.Y. Goswami, Theoretical analysis of a water desalination system using low grade solar heat, Journal of Solar Energy Engineering 126 (2004) 774-780.
[6] V.G. Gude, N. Nirmalakandan, S. Deng, A. Maganti, Low temperature desalination using solar collectors augmented by thermal energy storage, Appl Energy 91-1 (2012) 466–74.
[7] V.G. Gude, N. Nirmalakandan, Combined desalination and solar-assisted air-conditioning system, Energy Convers Manag 49 (2008) 3326–30.
[8] 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) 635–46.
[9] T. Ayhan, H. Al-Madani, Feasibility study of renewable energy powered seawater desalination technology using natural vacuum technique, Renewable Energy 35 (2010) 506–14.
[10] H. Ambarita, Study on the performance of natural vacuum desalination system using low grade heat source, Case Studies in Thermal Engineering 8(2016) 346-358.
[11] GA Bemporad, Basic hydrodynamics aspects of solar energy based desalination process, Desalination 54 (1995) 125 -134.