A NOVEL SUPER-COOLED HUMIDIFICATION-DEHUMIDIFICATION SYSTEM DRIVEN BY THERMAL VAPOR COMPRESSION UNIT (HDDTVC) FOR SEAWATER DESALINATION

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ABSTRACT

The current study presents the concept of a novel seawater desalination system that is configured by a humidification-dehumidification unit based on the vapor-compression process (HDDTVC). The coupled refrigeration unit employs the thermal compression of vapor by a jet-ejector. The technical new idea of proposed desalination system depends mainly upon creation of a super-cooled temperature under the ambient condition (*near 0 °C*) for the dehumidification process instead of using the heat sink in cooling process. Subsequently, the significant development in the unit performance is achieved by increasing its availability due to condensation and collection of the dew from the atmospheric vapor as an additional source of water production in addition to the product from the main source of seawater. A computer program based on a simulation mathematical model is constructed and a comprehensive analysis is discussed for the new system. The alternate novel process is characterized with the ability to recuperate the retrogressive water production in traditional process, where the productivity of freshwater may equal 8 folds compared with the conventional humidification-dehumidification process applied for the countries of hot and tropical climatological conditions. The amount of extracted and retrieved water from the dew in most humid regions can reach around 51% and 37% of the new unit production.

In this investigation, enhancement of the unit performance is the main objective .The influence of changing the characteristics of the jet-ejector (the cornerstone of the thermal refrigeration vapor-compression unit) on the system performance is discussed and analyzed so as to clarify the controlling parameters. Also, the applicability of different working fluids is tested for the vapor compression process.

Based on the current work, the new HDDTVC is suggested as a promising unit of medium-scale commercial production for provisions the arid and isolated communities with their requirements of potable water.

Keywords: Humidification - dehumidification process, heat transfer, refrigeration, jet-ejectors.

1. INTRODUCTION

Freshwater is currently considered an important strategic issue in most arid and semiarid countries of the globe. The great shortage of water retards the economic development, creates generic poverty and degrades the healthy standard, subsequently increasing diseases in these countries. Due to the escalating shortage of potable water, two out of three people will lack sufficient freshwater by the year 2025. Recently, desalination of seawater is regarded as one of the serious solutions for increasing water supply. Competitiveness of desalination with other polices of water conservation returns to the rapid economical and technological development in the field.

Semiarid countries in Middle East and Arabian Gulf regions become nowadays wholly dependent on desalination processes. Saudi Arabia and Bahrain obtain about 95% of their water demand by desalting seawater plants. Each day, about $23*10^6$ m^3 of freshwater is produced from seawater by many commercial thermal and membrane techniques, which are driven by fossil fuels. Most of large capacity plants involve: multi-stage flash (MSF), multi-effect distillation (MED) and reverse osmosis RO. But, these techniques are of high cost for production of small amounts of potable water. Additionally, it is impractical to apply these units in isolated and remote areas due to the limited maintenance facilities and energy supply.

Consequently, the small capacity plants that use nonconventional and renewable energy are currently urgent and suitable for these locations. Humidification–dehumidification distillation process (HDD) is highly recommended for small processes. The design configuration of HDD process could be applicable in industrial regions as well as isolated communities for brackish and seawater desalination. Such process is highly valued by several researches owing to their simple technological requirements (Kalil [1], Rajvanshi [2], Kudish et al. [3]). Majority of the previous literature studies focused on the performance evaluation, modeling, parametric studies and empirical correlation developed for those units combined with solar energy use.

Table 1 summarizes the previous investigations on HDD design and performance [2-28]. Careful review in this table shows the main conclusion that all these investigations concern preliminary prototypes and pilot scale units of few cubic meters production per day. Currently, no real commercial plant is in operation by the HDD system worldwide. The reason for delay of the commercial implementation of this technology in desalination market may be attributed to some drawbacks. These disadvantages are restricted in: (i) low-scale water production, (ii) high power consumption, (iii) high unit cost. Thus, this process is currently in the development phase to enhance the practical and economical unit performance. But on the other hand, HDD process has some characterized features such as: simple construction and technology, it can be used in isolated areas of small population, has no negative impact on the environment, and it can employ the low grade renewable thermal energy such as solar energy, waste industrial heat and geothermal energy. However, it is recommended that research attempts must continue to prove its commercial use on the short range. Development issues may involve new techniques, efficient energy use by heat recovery, utilization of renewable and sustainable energy, usage of nonconventional materials and introducing the enhanced extended surfaces to improve evaporation/condensation processes.

Table 1 Previous investigations on HDD design and performance

Ref. No.	Reheating Process for	Max. allowable temp.	Energy source	Flow Pattern Water/air	Type of study	Humidifier type	Water stream	Air stream	Combination between humidifier & dehumidifier	Remarks
[2] 1981		-	-	Forced/ Natural	Theoretical	-	-	23.9 - 26.6 °C	Dehumidifier Only	• Seawater is pumped by 3-wind machines of total energy 600 kW. • dew collection plant. • Cold seawater for condensation (5 °C) from 500 m depth and 5 km from the shore. • collector area is 18.6 m^2 • No. of collectors are 6950 with area $1.29*10^5$.
[3] 1998	air + water	77 °C	Solar 650 W/m^2	Forced / Forced	Theoretical / experimental	Pad (wick)	-	2.5 m^3/h	Combined unit with open cycle	• Double-glazed solar still, double-effect. • Thermal energy recycle. • Solar still area 1 m^2 (.54x1.85m)
[4] 1999	Water	90 °C	Solar	Forced/ Natural	Theoretical	-	-	-	Combined tower with closed air cycle	• Heat recovery condenser for preheating feed water. • Results were compared to pilot units. • Performance study.
[5] 2000	Water	85 °C	Solar	Forced / Forced	Experimental	Spray + pad (honey comb packing)	-	-	Separate units	• Performance study. • Optimization of air speed.
[6] 2001				Forced / Forced	Theoretical / experimental	Pad (membrane)	19-54 kg/h	30-80 m^3/kg	Separate units	• Used a hollow poly propylene capillaries. • Water and L_iCl saturated solution are the working fluids. • Compact prototype humidifier (593 m^2/m^3)
[7] 2002	Water	70 °C	Solar	Natural/ Natural	Theoretical / experimental	Pad	42-70 °C		Combined tower with closed air cycle	• Heat recovery system. • Bench unit. • Parametric study.
[8-10, , 17] 2002	Air		Solar						Separate unit	• Greenhouse produces fresh water and crops. • Shallow greenhouse area 10000 m^2.
[11] 2003	Air + water	60-80 °C	Hot water 90-100 °C	Forced/ Forced	Experimental	Spray + pad	360 kg/h 41-60 °C		Separate unit	• Pad humidifier has 3 cassettes in series. Made of corrugated cellulosic material – Water flow, downward, air passes in cross flow. • Spray humidifier is a 30 cm diameter and 4 cm length tube, has U-shape- 4 nozzles made of poly propylene.
[12] 2003	Water			Natural	Theoretical	Pad				• Performance study • Different gas carriers: hydrogen- helium- neon- nitrogen- oxygen- air- carbon dioxide
[13] 2003	Air + water	70 °C	Solar	Forced	Theoretical		Basin water 12-20 kg 43 °C	Up to 2 kg/s Up to 57 °C	Combined still with closed air cycle	• Multi-basins solar still. • Parametric study. • Transient performance work.

[14-16] 2002-2003	Air + water	80 °C	Solar	Forced/ Forced	Theoretical/ Experimental	Pad	600 kg/h 43 °C	300-1700 kg/h up to 80 °C	Separate units	• Pilot plant • Stepwise loading of air by vapor, 4-stages for heating and humidification processes. • New solar collectors for heating and humidifier are developed. • Conceptual design for 10 m^3/d.
[18] 2004	Air + water	60 °C	Solar		Theoretical/ Experimental	Pad			Separate units	• Heat recovery in condenser. • Performance study. • Steady-state model solves the differential equations.
[19] 2004	Air + water	60 °C	Solar	Natural	Theoretical		Basin water 12-20 kg/s		Combined still with closed air cycle	• Numerical study. • Productivity of natural circulation still gives the same productivity of forced circulation (5.1 kg/m^2.d). • The unit cost is 9 \$/m^3 fresh water.
[20] 2004	Air + water		Solar	Natural / Natural	Theoretical	Spray	.005-.06 kg/s	.005-.04 kg/s	Separate units	• Performance study. • The model is validated against experimental work of other investigators.
[21] 2004	Air + water	46 °C	Solar	Forced / Forced	Experimental	Spray pad	.005-.07 kg/s	.005-.045 kg/s	Separate units	• Bench plant. • Transient and steady state work.
[22] 2004	Air		Electric energy	Forced	Experimental	Spray + pad (baked clay)	1-15 kg/min 15 - 35 °C	150-185 kg/h 65-95 °C	Separate units	• Bench unit. • Parametric study.
[23] 2005	Air + water	90 °C	Electric heating	Forced / Forced	Experimental + mathematical model	Pad (plastic material)	.33–3 kg/min 35 - 90 °C	1.2 - $1.44x10^9$ kg/h	Separate units	• Bench unit. • Characteristic study for the unit
[24] 2005	Air + water	88 °C	steam + electric energy	Forced / Forced	Experimental	Falling water film	2-20 kg/h 67 - 88 °C Steam .05 - .36 kg/h 85 °C	4.75-18 kg/h	Combined unit shell-tube unit	• Bench unit. • Steam increases the temperature and humidity of humid air exit from humidifier. • New baffled shell-tube HD unit, with 73 Cu tubes and poly propylene shell. • The unit process operates at high temperature.
[25] 2005	Air	90 °C	Solar	Forced	Experimental		60-140 kg/h 25-70 °C air Water content 20-55 g/kgair			• Prototype unit. • The efficiency of a new solar design plate collector was studied experimentally. • Parametric study for the solar collector was developed by E. Chafik.
[26] 2005					Theoretical					• Four layouts for HD desalination unit were discussed. And analyzed.
[27] 2005	Water	80 °C	Solar	Forced / Forced	Theoretical	Spray	Up to 80 °C		Combined unit with closed air cycle	• Pinch technology is used for optimization of solar HD desalination process. • Mass flow ratio, air temperature and rejected water temperature were optimized.
[28] 2006	Air or water	70 °C	Solar	Forced / Forced	Theoretical	pad	26.7-45 G/min Up to 70 °C	56-85 kmole/h	Separated column	• Process design and optimization were performed for minimum solar energy.

Because the atmospheric air contains nearly about 12,650 km^3 of vapor water evaporated from land, seas and oceanic surfaces, a new design concept of humidification- dehumidification process is proposed in the current work to recover partially these amounts and produce additional freshwater from seawater. Enhancement of unit performance by increasing freshwater productivity to reach the feasible medium scale is the main aim of the new technique to overcome the first reason of HDD drawbacks.

2. NEW HUMIDIFICATION-DEHUMIDIFICATION SYSTEM

2.1 Idea of Proposed Technology

Conventional HDD process is based on the use of a carrier gas (usually air) as a working fluid in closed/open cycle. Humidification-dehumidification cycle is shown in **Fig. 1** on T-ϕ diagram (psychrometric chart). The process is substantially formed of three main units: heater for heating the air from ambient (1) to a higher temperature (2) with conserving the absolute humidity constant. Second is the humidifier (evaporator), where the air humidity is increased to nearly saturation condition by evaporating a portion of seawater, while the moist air temperature decreases (3). The third is a dehumidifier (condenser), which is used for removing the majority of air humidity by a condensation process for water vapor to the state (4). The exhaust air temperature (4) stays always higher than the ambient temperature (1) by a difference of 3-5 °C because the coolant is used at the ambient temperature for condensation process in the humidifier. Then, exhaust air has an elevated water content level compared to that of intake air, so it seems that as we support atmospheric air humidity. Thus, in the conventional process, it is impossible to lower the downstream state ($\varphi 4$) to the ambient state ($\varphi 1$ or $\varphi 5$). The rest humidity will eventually be discarded with air, i.e. the portion which is unavailable for retrieval ($\varphi 4$ - $\varphi 5$) is considered lost water. This means that the extracted water is less than the supplied atomized seawater. Therefore, partial recovery of water from the created humidity occurred, which asserts the reduction of water availability. Thus, the humidification-dehumidification process is considered an irreversible process because it can not revert to the initial state 1.

The relative humidity of air entering $\varphi 3$ and leaving $\varphi 4$ the dehumidifier often is in the saturation condition. But, the temperature of the two states of air is different. In fact, the amount of freshwater produced is driven largely by the difference between the terminals temperature of air in the dehumidifier: initial state 3 and final state 4. All the previous research efforts have been dedicated only on rising the upstream air temperature (2) as a trial for increasing air temperature difference, subsequently water production. Therefore, introduction of feed heaters in HDD process was the purpose for rising the temperature of one or both of intake air and feed seawater.

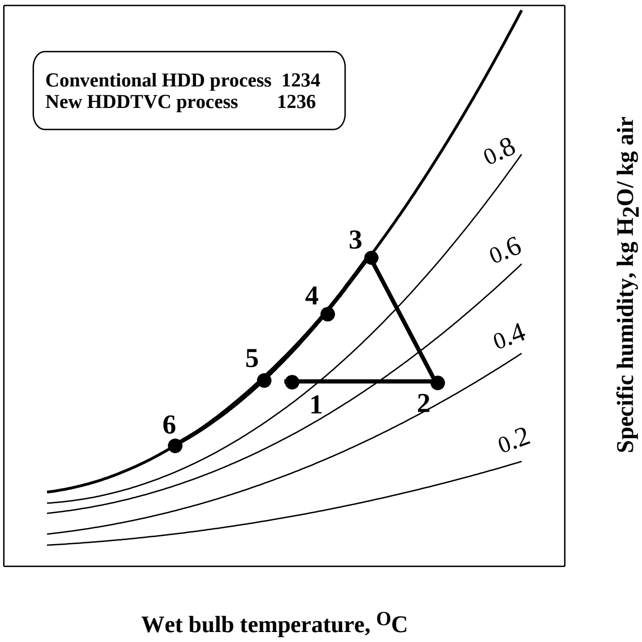


Fig. 1 Humidification- dehumidification cycles.

The condensation process inside the dehumidifier is practically performed by using seawater or air for cooling. Thus, the heat sink conditions actually restrain the HDD unit performance as the dehumidifier recovers partially the sprayed water by a value of about 60-80% only. Therefore, the new idea of the current investigation aims to augment the dehumidifier performance and water production by eliminating the dependency on the state of heat sink. The main aim in this study is the development of the traditional HDD by maximizing the terminals temperature difference range by lowering the outgoing air (state (4) in the conventional unit) to a considerable temperature beyond the ambient temperature to become state 6 (in the new unit). The technical achievement of this concept can be principally performed by the aid of refrigeration thermal vapor compression system which could improve the new process performance by creating a super cooled temperature in dehumidifier for the purpose of rising freshwater rating. The provided high cooling action usually brings the dew temperature of humid air nearer to the zero temperature.

Therefore, in the new dehumidification process (3-6) the essential benefit is recovering all amount of water sprayed and evaporated as a created humidity in humidifier (φ3- φ5), and also catching an additional part of water accompanied with intake humid air (φ5- φ6) , which may increases water productivity several times than that produced in conventional unit. For that proposed new system, the amount of water produced is higher than that initially supplied by seawater and the surplus amounts consist of the recovered lost water (φ4- φ5) in addition to the collected dew (φ5- φ6).

2.2 Vapor Compression – Humidification Dehumidification (HDDTVC) Technology

The new idea of the proposed humidification–dehumidification technology for seawater desalination aims to produce a cooling effect by suitable refrigerant. The basic design of the novel plant (*vapor compression- humidification dehumidification HDDTVC process)* is schematically illustrated in **Fig. 2**. For the purposes of reducing the unit cost and power consumption, the well known technology of thermal vapor compression can be utilized instead of mechanical compressor. As is clearly shown, the proposed desalination system is divided into two main parts: refrigeration vapor compression unit and humidification-dehumidification unit. The HDD circuit depends mainly on the open air cycle and closed water cycle. The HDD circuit is composed of preheaters (RC-PH), main heaters (MH), air humidifier (AH) and air dehumidifiers (RE-DH). While, the jet-ejector refrigeration system or which is well known as thermal vapor compression employs four main components: a vapor generator (G), a jet-ejector (EJ), a refrigerant condenser (RC-PH) and a refrigerant evaporator (RE-DH). The two units RC-PH and RE-DH have double functions and combine the two main circuits of air and refrigerant. The characteristics of this arrangement representing the combined cycle of HDDTVC are plotted on the T-S diagram in **Fig. 3.**

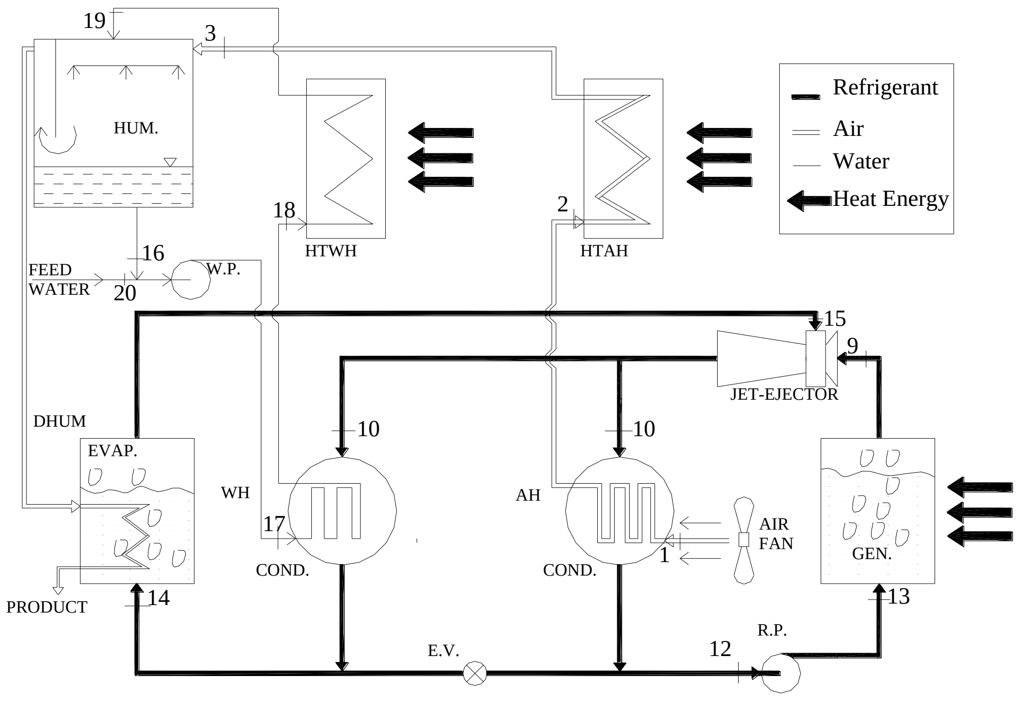


Fig. 2 Schematic Diagram of New Humidification-Dehumidification System.

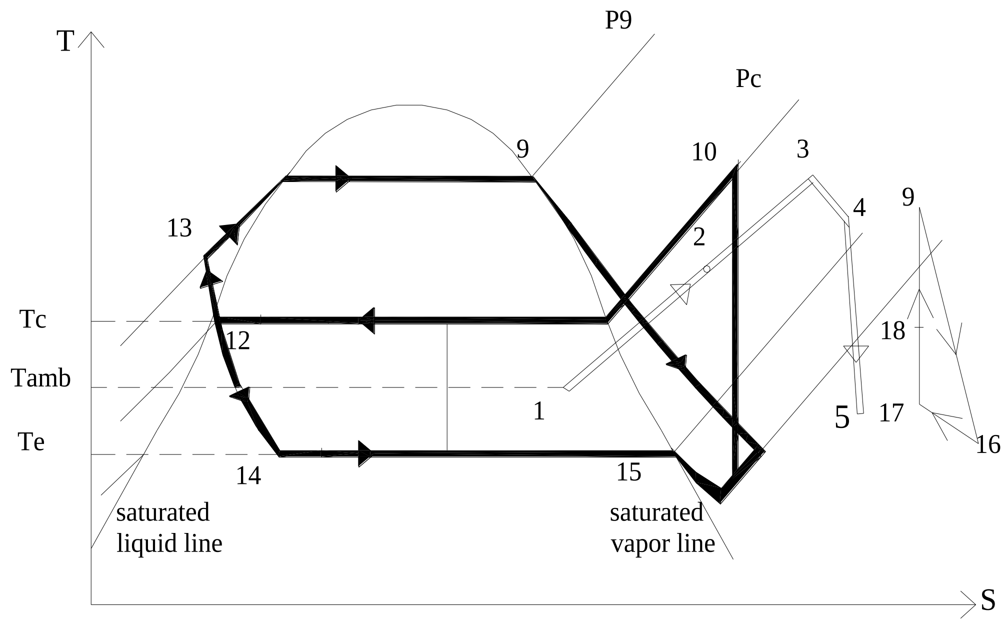


Fig. 3 T-S Diagram for All Processes of New Humidification-Dehumidification System.

Usually, the refrigerant vapor is generated at relatively high pressure and temperature in the generator by the assistance of an external low grade heat source. This vapor is called the primary vapor and it is used mainly as the actuating fluid in the vapor compression system. Then, the refrigerant vapor enters the jet-ejector and expands inside a supersonic nozzle forming a supersonic flow at low pressure at the nozzle exit. This action causes a suction of the secondary vapor that coming from the evaporator. The two streams mix inside the mixing chamber of the ejector and then the mixture is slightly compressed in the diffuser. The vapor leaving the ejector enters the two condensers at a pressure and temperature lower than that of the generator state. The condensate exits from the condenser as a saturated liquid refrigerant. A portion of that liquid returns again to the generator (primary quantity) by the aid of liquid pump. While, the other liquid portion (secondary quantity) expands to the evaporator by a suitable expansion valve causing the refrigerant fluid to be relatively at low pressure and temperature. Finally, the cycle starts again and repeated over and over. As is evident from the refrigeration cycle, three levels of temperature are categorized:

- The higher temperature level in the vapor generator (G).
- The medium temperature level in the vapor condenser (RC-PH).
- The lower temperature level in the evaporator (RE-DH).

The jet-ejector always plays an important role as a thermal compressor driven by an external low-grade heat source.

For the humid air cycle (humidification-dehumidification process), the intake air and feed seawater are primarily heated in the preheaters (RC-PH), because the temperature of the refrigerant inside the condenser tubes is usually higher than that of ambient temperature. Therefore the condensers are used as heat recovery section for heating air and water on the outer surface of the condensers tubes. Then, air and water leave the preheaters to enter the main heater (MH) for heating both the two fluids to the design temperatures via external heat energy. Ultimately, the two hot streams go to the air humidifier device (AH). A portion of seawater vaporizes causing the air relative humidity to be increased nearly to the saturation condition, where its temperature decreases. Then, the humid air directly enters to the dehumidifier (RE-DH) to reduce air humidity by condensation of water vapor on the outer surface of evaporator tubes. The condensation process continues to a temperature under the ambient temperature. Additionally, the exit air from the dehumidifier is at a low temperature and it can be utilized commercially for the purposes of cold stores, refrigerators and air conditioning systems. The cold water leaving the humidifier returns back to water preheater. The condensate from the dehumidification process represents the product freshwater.

Actually, the vapor compression circuit is characterized by three main functions in HDDTVC process:

- The main function is cooling the humid air to a temperature lower than ambient temperature to wholly increase the amount of product.
- Recovery of heat energy dissipated from refrigerant condenser (RC-PH) by preheating the intake air and feed seawater.
- Condensation of the moistness in air to supply the required potable water in refrigerant evaporator (RE-DH).

Obviously, this project presents a new method to improve conventional HDD performance via overcoming one of its disadvantages. It is an approach towards designing of the moderately scale seawater desalination plants by maximization of the HDD unit capacity.

3. SIMULATION OF SYSTEM MODELING

3.1 Refrigeration Unit and Jet-ejector Relationships

Constant- pressure ejector model is taken into consideration with chocked flow. The flow is assumed as one-dimensional and steady-state condition.

Motive vapor flow rate $\dot{m}_p$ is calculated from isentropic expansion in the nozzle as:

$$\dot{m}_p = P_p \ A_1 \sqrt{\frac{\gamma \ \eta_n}{R \ (T_p + 273)} \left(\frac{2}{\gamma + 1} \right)^{(\gamma + 1)/(\gamma - 1)}} \tag{1}$$

Mach number of primary vapor M_{pII} can be presented at nozzle outlet from the area ratio of the nozzle throat and the nozzle outlet section as following:

$$\frac{A_{II}}{A_I} = \sqrt{\frac{1}{M_{pII}^2} \left((\frac{2}{\gamma + 1}) (1 + \frac{\gamma - 1}{2} M_{pII}^2 \right)^{(\gamma + 1)/(\gamma - 1)}} \tag{2}$$

Pressure at exit from nozzle P_{II} is formed by:

$$P_{II} = \mathrm{P_p}\left[\mathrm{M}_{\mathrm{pII}}^2 \frac{\gamma - 1}{2\,\eta_{\mathrm{n}}} + 1\right]^{\gamma/(1-\gamma)} \tag{3}$$

Mach number of secondary stream M_{sII} is calculated at nozzle exit as:

$$M_{s\,II} = \sqrt{\frac{2}{\gamma - 1}\left(\frac{\mathrm{P_s}}{\mathrm{P_{II}}}\right)^{(\gamma-1)/\gamma} - 1} \tag{4}$$

Critical Mach numbers of primary M^*_{pII} and secondary M^*_{sII} streams :

$$M^*_{pII} = \sqrt{\frac{\mathrm{M}_{\mathrm{pII}}^2\ (\gamma + 1)}{\mathrm{M}_{\mathrm{pII}}^2\ (\gamma - 1)\ + 2}} \quad : \quad M^*_{sII} = \sqrt{\frac{\mathrm{M}_{\mathrm{sII}}^2\ (\gamma + 1)}{\mathrm{M}_{\mathrm{sII}}^2\ (\gamma - 1)\ + 2}} \tag{5}$$

Critical Mach numbers of the mixture M^*_{IV} upstream of shock wave (before shock) state:

$$M^*_{IV} = \frac{\mathrm{M}^*_{\mathrm{pII}} + \mathrm{w}\ \mathrm{M}^*_{\mathrm{sII}} \sqrt{\dfrac{T_s + 273}{T_p + 273}}}{\left[(1 + \mathrm{w})\ (1 + \mathrm{w}\ \dfrac{T_s + 273}{T_p + 273}\right]^{\frac{1}{2}}} \tag{6}$$

Mach number at state 4 before shock M_{IV} can be calculated as:

$$M_{IV} = \mathrm{M}^*_{\mathrm{IV}} \sqrt{\frac{2}{(\gamma + 1) - (\gamma - 1)\,\mathrm{M}^{*2}_{\mathrm{IV}}}} \tag{7}$$

Mach number of the mixed flow downstream (after the shock wave) state is:

$$M_V = \frac{\mathrm{M}_{\mathrm{IV}}^2 + \dfrac{2}{\gamma - 1}}{\mathrm{M}_{\mathrm{IV}}^2\ \dfrac{2\gamma}{\gamma - 1} - 1} \tag{8}$$

Pressure increase across the shock wave is calculated with the assumption of $P_{IV} = P_{II}$ as:

$$P_V = \mathrm{P_{IV}}\ \frac{1 + \gamma\,\mathrm{M}_{\mathrm{IV}}^2}{1 + \gamma\,\mathrm{M}_{\mathrm{V}}^2} \tag{9}$$

Pressure lift in the diffuser (condenser pressure) P_c;

$$P_c = \mathrm{P_V}\left[\frac{\eta_{\mathrm{d}}\,(\gamma - 1)}{2}\ \mathrm{M}_{\mathrm{V}}^2 + 1\right]^{\frac{\gamma}{\gamma - 1}} \tag{10}$$

The area ratio of the nozzle throat and diffuser in constant area is:

$$\frac{A_I}{A_{III}} = \frac{P_c}{P_p}\left[\frac{1}{(1+w)\left(1+w\dfrac{T_s+273}{T_p+273}\right)}\right]^{\frac{1}{2}} * \frac{\left(\dfrac{P_{II}}{P_c}\right)^{\frac{1}{\gamma}}\left[1-\left(\dfrac{P_{II}}{P_c}\right)^{\frac{\gamma-1}{\gamma}}\right]^{0.5}}{\left(\dfrac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left[1-\dfrac{2}{\gamma+1}\right]^{0.5}} \tag{11}$$

3.2 Humid Air Relations

Specific humidity φ of air can be presented by:

$$\phi = \frac{.62198 \quad P_d}{(P_{atm} - P_d)} \quad : \quad P_v = RH \,.\, P_s \tag{12}$$

Air specific enthalpy is:

$$h_a = 1.005\, T_a + \phi \; h_g \tag{13}$$

Saturation enthalpy of water vapor h_g and saturation pressure P_s, are calculated from the following formulas:

$$h_g = 2500.7843 + 1.90865\, T_a - 1.59097 * 10^{-3} \; T^2 \tag{14}$$

$$P_s = 0.722677 - 0.8538 * 10^{-3} \; T + 45.65 * 10^{-4} \; T^2 - 5.091673 * 10^{-5} \; T^3 \tag{15}$$

Partial pressure of water vapor is:

$$P_d = RH * P_s \tag{16}$$

3.3 Controlling Parameters

Enthalpy of exit mixture from jet-ejector h_{10} can be calculated by applying the energy balance equation on jet-ejector:

$$h_c = \frac{h_9 - w\, h_{15}}{1 + w} \tag{17}$$

Energy balance equation for humidifier is applied to find water flow rate $\dot{m}_w$ as:

$$\dot{m}_w = \frac{\dot{m}_a \,(h_4 - h_3)}{c_{pw}\,(T_{wi} - T_{wo})} \tag{18}$$

Energy balance equation of dehumidifier is applied to find the daily water production M_d and air temperature exit from humidifier and entering the dehumidifier T_4:

$$M_d = 86.4 \; \dot{m}_a \,(\phi_4 - \phi_5) \tag{19}$$

$$h_4 = h_5 + \frac{\dot{m}_s \,(h_{15} - h_{14})}{\dot{m}_a} = h_3 + \frac{\dot{m}_w \; c_{pw} \,(T_{wi} - T_{wo})}{\dot{m}_a} \tag{20}$$

The performance of HDD unit is defined as the refrigeration load of the evaporator divided by heat input in both vapor generated and high temperature heater plus all work required for pumps and air fan. Also, actual coefficient of performance of VC unit can be defined by the desired evaporator load divided by the required heat of generator and pump work:

$$COP_{hd} = \mathrm{Q_{ev}} \big/ (\mathrm{Q_g} + \mathrm{Q_{hth}} + \mathrm{W_{pr}} + \mathrm{W_{pw}} + \mathrm{W_{pa}}) \tag{21}$$

$$COP_{vc} = \mathrm{Q_{ev}} / (\mathrm{Q_g} + \mathrm{W_{pr}}) \tag{22}$$

The maximum coefficient of performance of ideal reversible refrigerant cycle is given as:

$$COP_{vc,\,\mathrm{id}} = \frac{\mathrm{T_e} + 273}{\mathrm{T_c} - \mathrm{T_e}} * \frac{\mathrm{T_g} - \mathrm{T_c}}{\mathrm{T_g} + 273} \tag{23}$$

Thermal loads of each refrigerant evaporator, generator, high temperature heaters and refrigerant condenser are presented as:

$$Q_{ev} = \mathrm{\dot{m}_a}\,(\mathrm{h_4} - \mathrm{h_5}) = \mathrm{\dot{m}_s}\,(\mathrm{h_{15}} - \mathrm{h_{14}}) \tag{24}$$

$$Q_g = \mathrm{\dot{m}_p}\,(\mathrm{h_9} - \mathrm{h_{13}}) \tag{25}$$

$$Q_{hth} = \mathrm{\dot{m}_a}\,(\mathrm{h_3} - \mathrm{h_2}) + \mathrm{\dot{m}_w}\,(\mathrm{h_{18}} - \mathrm{h_{188}}) \tag{26}$$

$$Q_c = \mathrm{\dot{m}_a}\,(\mathrm{h_2} - \mathrm{h_1}) + \mathrm{\dot{m}_w}\,(\mathrm{h_{18}} - \mathrm{h_{17}}) = \mathrm{\dot{m}_c}\,(\mathrm{h_{10}} - \mathrm{h_{12}}) \tag{27}$$

Mechanical work done of each: refrigerant pump, water pump and air fan are:

$$W_{pr} = \mathrm{\dot{m}_p}\,(\mathrm{h_{13}} - \mathrm{h_{12}}) / \eta_{\mathrm{pw}} \; : W_{pr} = \mathrm{\dot{m}_w}\,(\mathrm{h_{17}} - \mathrm{h_{16}}) / \eta_{\mathrm{pw}} \; : W_{fa} = \mathrm{\dot{m}_a}\,\Delta \mathrm{P_{fa}} / \rho_{\mathrm{a}}\ \eta_{\mathrm{fa}} \tag{28}$$

Total refrigerant flow rate upstream jet-ejector is found as:

$$\dot{m}_c = \dot{m}_p + \dot{m}_s \tag{29}$$

The performance parameters for jet-ejector are compression ratio CR and expansion ratio ER, and they can be defined as:

$$CR = \frac{\mathrm{P_c}}{\mathrm{P_s}} = \frac{\mathrm{P_{10}}}{\mathrm{P_{15}}} \tag{30}$$

$$ER = \frac{\mathrm{P_g}}{\mathrm{P_s}} = \frac{\mathrm{P_9}}{\mathrm{P_{15}}} \tag{31}$$

The present procedure required many iterative calculations. For this purpose, a computer program is constructed based on the above simulation. Iterations are made to determine the humid air state exit from the humidifier T_4 which determines the dehumidifier capacity, and entrainment ratio W that defines the ejector capacity.

4. ANALYSIS AND DISCUSSION

The thermal performance study of the new desalination system is performed with the aid of refrigeration compression process. To evaluate the unit performance, the two important indices of the water productivity and thermal efficiency are taken into consideration.

4.1 Verification of HDDTVC Benefits

Verification of the proposed technology's benefit can be examined by a technical comparison with conventional HDD system. **Fig. 4-a** indicates the unit performance for the two systems at similar climatic ranges of temperatures. Owing to the creative super cooling effect of HDDTVC unit, the amount of freshwater that could potentially be produced is relatively higher than that may be produced from the conventional HDD unit as is shown from the figure, particularly at high temperature level of heat sink. As the ambient temperature increases, as in summer season, the production of conventional HDD unit retrogresses, which indicates the great significance of HDDTVC system and its ability to recuperate the retrograded production and keeping the unit capacity constant and independent on the heat sink condition. The enhancement factor (EF) can be proposed as a measure for the degree of modification in production when the new system is utilized instead of conventional one. ED is defined as the following:

$$EF = \frac{\dot{M}_{d,HDDTVC} - \dot{M}_{d,c}}{\dot{M}_{d,c}} = \frac{\dot{M}_{d,r} + \dot{M}_{d,cd}}{\dot{M}_{d,c}} \qquad (32)$$

It is well known that the ambient temperature shows a drastic change allover the year in the region of Middle East and Arabian Gulf. In winter, the temperature can vary from 10-20 °C, consequently, the modification in the pertinent productivity of HDDTVC is about 8 - 27 % over conventional HDD. While, in summer the variation in temperature is in the range of 30-42 °C, therefore, the system becomes more efficient and the improvement in water production approaches 91- 705 % with respect to the conventional unit. Furthermore, the unit coefficient of performance COP is enhanced at high temperatures than low temperatures, where its value increased in the studied temperature range from 0.395 - 0.521.

The generic amount of fresh water produced by the dehumidifier of the new system actually comes from two sources: the retrieved water which is provided from the sprayed and evaporated in the humidifier, in addition to the extracted water that may be caught from the atmospheric humid air, due to the high cooling effect. Regarding the water extracted rate, its amount is noticeable to be intensively increased with temperature, although the amount of total water produced stays unchangeable. Where, the ratio of that extracted water reaches about 51 % of the composite water produced at 42 °C as shown in **Fig. 4-b**. Accordingly, the rate of circulated water in the HDDTVC unit decreases, which saves the potential pumping power of water circuit. In addition, the HDDTVC unit size becomes generally compact, so low water cost is expected in these conditions. The same trend can be noticed for the relative humidity of the intake air, where the participation of extracted water in the total water produced becomes much bigger at higher relative humidity.

From this, the new humidification-dehumidification process is viewed as a promising technique for economically duplicating the amount of freshwater produced many times especially for those countries of hot weather.

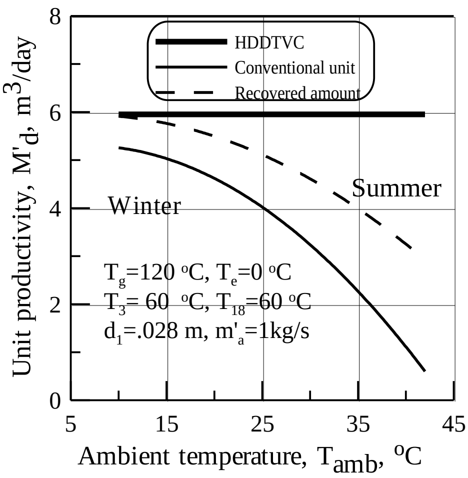


Fig. 4-a Comparison of water productivity between the new HDTVC unit and conventional unit.

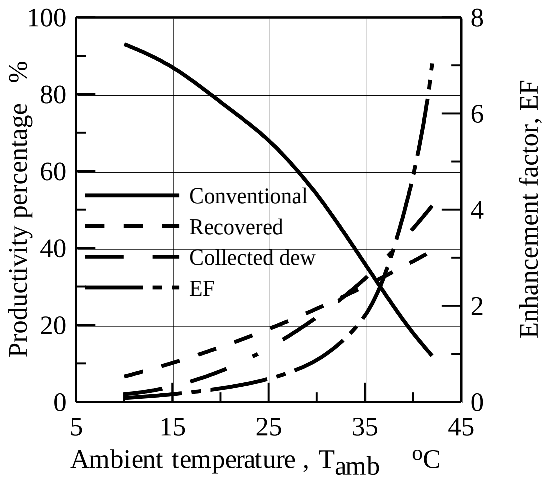


Fig. 4-b Comparison of water productivity between the new HDTVC unit and conventional unit.

4.2 Performance Study

Different operating parameters were studied, including vapor generator temperature T_g, evaporator temperature T_e, air flow rate $m^{\cdot}_a$, maximum air temperature T_3 and maximum water temperature T_{18}. Monitoring the unit performance with changing these operating parameters is hypothetically performed for specified design geometry of constant pressure jet-ejector (rating study). Also, the effect of changing the design throat diameter of nozzle d_1 is taken into consideration.

Often, two main states govern the HDDTVC's performance: state 4 of the input humid air to dehumidifier and state 188 of the output hot water from the condenser. These states are calculated via applying energy balance for air, water and refrigerant circuits. So, the resultant of changing any parameter usually depends on the range of its influence on these two states, subsequently the unit performance.

4.2.1 Vapor Generator Temperature, T_g

HDDTVC system can be operated at different temperatures and pressures which are achievable using different low grade heat. The results of variation of the heat source circumstance are plotted in **Fig. 5**. The graph represents the unit performance against specified operating conditions. It is shown that by changing the generated vapor temperature in the range of 70-140 °C, freshwater production (Eqn. 19) increases sharply due to rising the upstream air temperature of dehumidifier T_4 as indicated from Fig. 3 (T-S diagram). While, doubling the temperature increases strongly the production by more than six folds. Therefore, the generator vapor temperature is typically the driving force for HDDTVC unit.

As is indicated from the figure, the HDD unit's performance linearly decreases with the temperature of vapor generation owing to the further rate of increase of Q_g than the rate of Q_{ev} as formed in Eqn. (20).

4.2.2 Evaporator Temperature, T_e

Fig. 6 illustrates the influence of variation of the evaporator temperature on the unit performance at a temperature range of 2-40 °C.

From Eqn. (20), it is clear that water productivity depends mainly upon the two states of refrigerant (14 & 15) in evaporator (RE-DH) as well as the secondary refrigerant flow rate. Therefore, the amount of water produced might be increased as the evaporator temperature and its pertinent heat content (h15) increased while keeping the condenser temperature nearly constant. The considerable augmentation in water production may reach 68% for the studied range of variation of T_e, leaving the other operating conditions constant. Thus, the evaporator heat load Q_{ev} increase with evaporator temperature, which in turn rises the value of unit performance COP_{hdd}. From the other hand, the vapor compression unit's thermal efficiency usually increases with evaporator temperature, as concluded from Eqn. (23), which rises the HDDTVC unit efficiency as a whole, which potentially can lower energy consumption and the accompanied water cost.

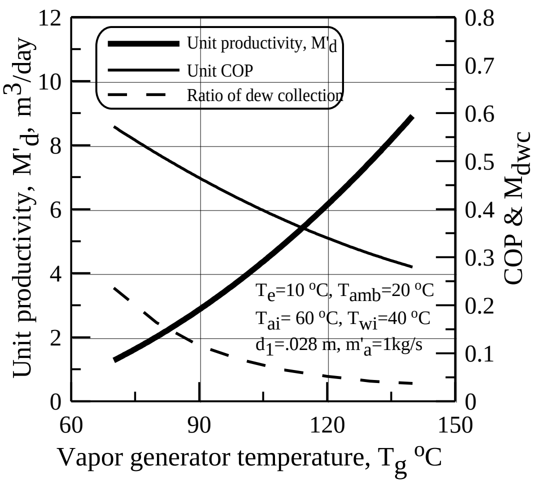


Fig. 5 Variation of unit performance M'_d, COP) as a function of vapor generator temperature T_g.

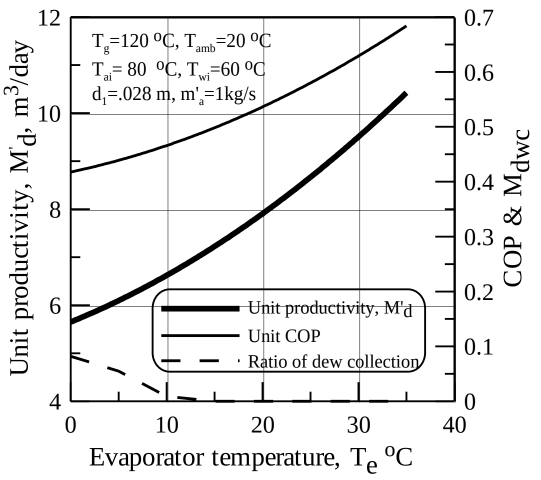


Fig. 6 Variation of unit performance M'_d, COP) as a function of evaporator temperature T_e.

4.2.3 Air Flow Rate, $\dot{m}_a$

Influence of air flow rate on unit performance is depicted in **Fig. 7** for various values of air flow rate ranging from 1-6 kg/s. It is clear that increasing the amount of delivered dry air to the unit leads to a slight reduction in the amount of product water, where, water productivity decreases by about 9.3% only for a high rise of air flow rate by six fold. Usually, the rise in air flow rate is accompanied by a reduction of the required seawater flow rate $\dot{m}_w$ to the humidifier (Eqn. 18). Therefore, the reduction in amount of freshwater is attributed to the reduction of state 4 and its properties (T_4, h_4, φ_4) as is shown from Eqns. (19) and (20) while keeping other states and parameters constant. On the contrary, the performance ratio of the HDDTVC system remains constant with air flow rate, because the required evaporator load Q_{ev}, generator load Q_g and heating load Q_{hth} of the high temperature heater are constant. Therefore, coefficient of performance COP remains unchanged and equals the value of 0.694.

4.2.4 Maximum Air temperature, T_3

From the two Eqns. (19) and (20), it is shown that the maximum allowable air temperature has no effect on each of the air state 4 and the daily water produced. Therefore, the amount of water remains constant as it can be seen from **Fig. 8**; although the air temperature changed from 31-75 °C. Water flowing throughout water circuit always decreases with the maximum air temperature T_3 as concluded from Eqn. (18). Thereby, the required heating load Q_{hth} for the main heater, evaporator load Q_{ev} and generator load Q_g, are constant, which substantially keeps the unit performance COP_{hdd} also constant at 0.213.

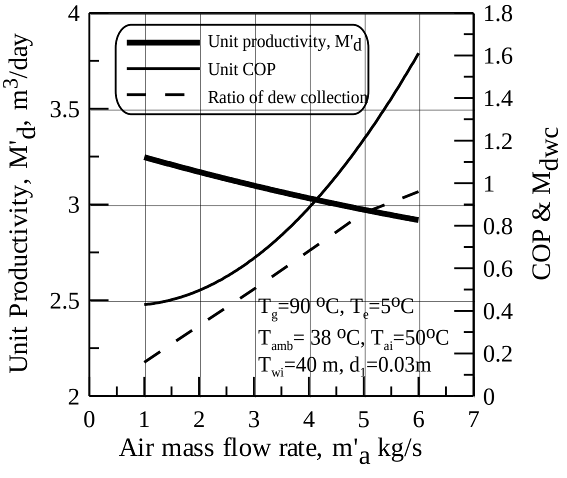


Fig. 7 Variation of unit performance M'_d, COP) as a function air flow rate M'_a.

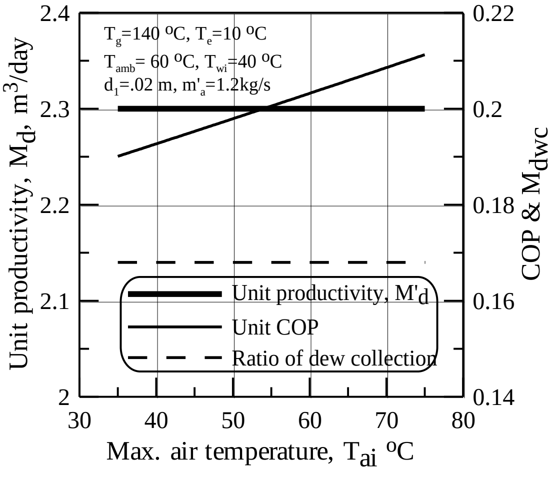


Fig. 8 Variation of unit performance M'_d, COP) as a function of max. air temperature T_a.

4.2.5 Maximum Water Temperature, T_{18}

Fig. 9 indicates the effect of changing the maximum water temperature entering the humidifier, while keeping other parameters constant at specified condition of: vapor generator temperature T_g = 140 °C, evaporator temperature T_e = 10 °C, ambient temperature T_{amb} = 20 °C and air flow rate $\dot{m}_a$ = 1.2 kg/s.

By increasing the maximum water temperature from 25-80 °C, freshwater productivity and unit performance stay constant, because the maximum water temperature has no effect on all values of Q_{hth}, Q_{ev} and Q_g. The values of water productivity and COP_{hd} are 2.34 and 0.216 m^3/D respectively.

4.3 Design Study of Nozzle Diameter, d_1

For the design and construction purposes, it is convenient to study and discuss the influence of changing the nozzle configuration on unit performance. This is important to qualitatively determine the appropriate relative nozzle diameter to get the best performance. The relationship between the nozzle throat diameter and the unit performance (water productivity and COP) is plotted in **Fig. 10** in the range of d_1 = 3-5.5 cm, keeping the operating condition constant at: generator temperature T_g = 160 °C, evaporator temperature T_e = 90 °C, maximum air temperature to humidifier T_3 = 90 °C, maximum water temperature to humidifier T_{18} = 60 °C, ambient temperature T_{amb} = 20 °C, air flow rate m_a = 10 °C and relative humidity RH = 75%.

The daily water produced is found to rise considerably with the nozzle diameter at the same condition as is obvious from the figure. This high increase in productivity may be attributed to the rise in circulated refrigerant (eqn.1) and water (Eqn.18), which in turn rises heat loads for evaporator (Eqn.20 & 24) and generator (Eqn.25). This should increase the heat transfer area for all equipments and their sizes, which make studying the economical cost of HDDTVC unit is essential. Normally, water productivity is found to proportional with the square power of diameter d_1^2. Therefore as the nozzle diameter increases from 1 to 5 cm, the promising amount of freshwater can be increased by 25 fold. The issue is different for the unit efficiency as is seen from the values of the COP that stay constant without change.

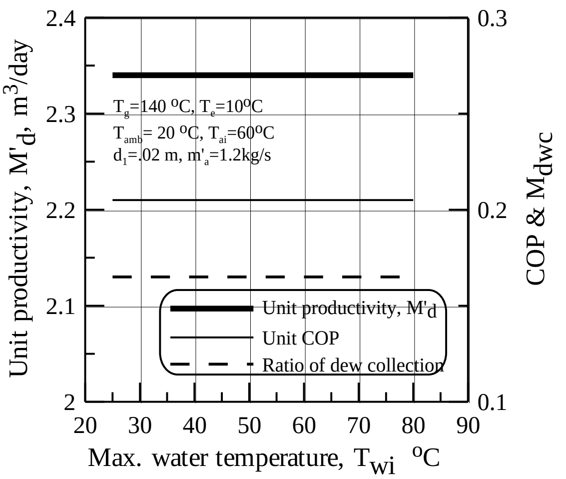


Fig. 9 Variation of unit performance M'_d, COP) as a function of max. water temperature $T_{wi.}$

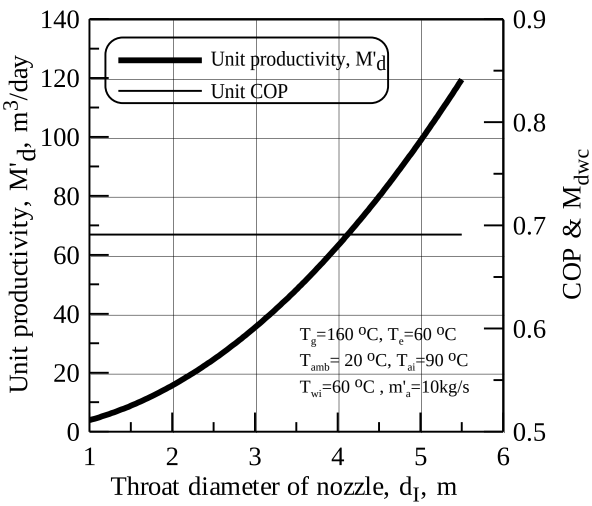


Fig. 10 Variation of unit performance M'_d, COP) as a function of throat nozzle diameter d_I.

4.4 Performance map of HDDTVC unit

It is desirable to determine the available range of operating conditions of the HDDTVC unit for a specified ejector of constant geometry for the behalf of the operators. **Fig. 11** illustrates the operating characteristic curves or performance map for a steam jet-ejector configuration of A_3/A_1 = 125, A_2/A_1 = 45 and nozzle throat diameter d_1 = 0.03 m. Any point on the map depicts the required HDDTVC system's water production when certain operating parameters are applied for that ejector. The practical significance of the current unit's performance map may be attributed to that it gives the operators a degree of freedom to modify or change the operational conditions according to change water requirements.

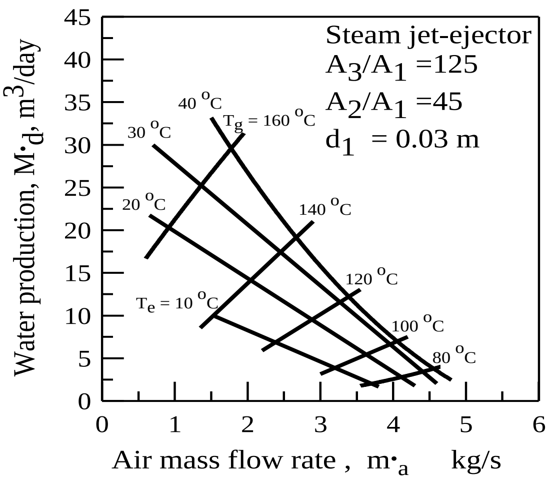
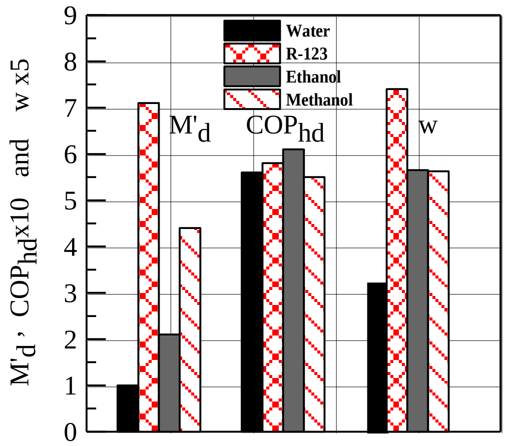


Fig. 11 Performance map of HDDTVC unit based on operating conditions.

4.5 Working Fluids

In the previous analysis, water is used as the main working fluid in vapor compression refrigeration unit for its plentiful availability and advantageous features. But, system performance depends to a great extent on thermodynamic properties of the working fluids. Therefore, various fluids which possibly be employed in the unit are nominated in order to examine the system efficiency. Of these refrigerants are the environmental friendly hydrofluorocarbon cryogens R-134a (CH_2FCF_3) and R-123 ($CHCl_2lCF_3$) are selected, in addition to the organic fluids such as ethanol ($C_2\ H_5\ OH$) and methanol ($CH_3\ OH$).

Characteristics of the proposed humidification-dehumidification unit are based on: entrainment ratio ω, unit productivity $M^{\cdot}_d$, thermal efficiency of both VC and HDD units, and jet-ejector features represented in expansion ratio ER and compression ratio CR. These features are normally used for direct comparison among working fluids to clarify if there are other fluids can be operated at the same conditions. The results are plotted in **Fig. 12** for the combined operating conditions: generator temperature T_g = 70 °C, evaporator temperature T_{ev} =10 °C, ambient temperature T_{amb} = 15 °C, maximum air and water temperatures $T_3 = T_{18}$ = 40 °C and air flow rate $\dot{m}_a$ **=** 1 kg/s. The ejector characteristics are: throat diameter of nozzle d_1 = 2 cm, area ratio A_{II}/A_I = 8 and A_{III}/A_I = 30.



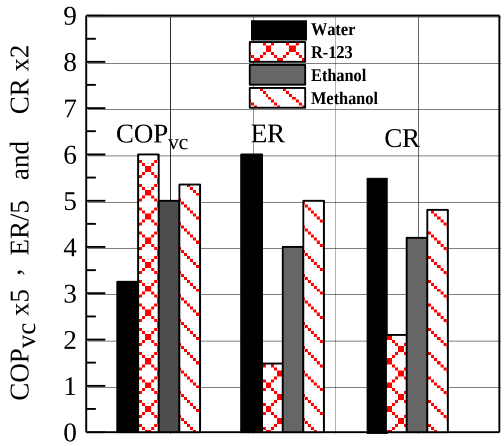


Fig. 12. Variation of unit characteristics with different working fluids (water, R-123, Ethanol and Methanol)

From the figure, the high system capacity demonstrated by entrainment ratio and water productivity is given by halocarbon-based cryogens, then methanol. Also, the two types of working fluids gives higher system thermal efficiency represented by COP of both VC and HDD units, because the overall unit performance is directly dependent on the accompanied VC unit performance. On the other hand, water shows higher values of CR and ER of jet-ejector than other fluids which induce high temperature at which the heat can be rejected (condenser temperature). This in turn improves the heat that may be recovered from the condenser section and well consequently rise the system performance, as it is indicated from the relatively high value of COP_{hd} which equal 0.56 (near the highest value 0.61), although the lower values of water production.

Finally, it is concluded that HDD system using working fluids of higher molecular mass such as halocarbon-based refrigerants (R-123 & R-134a) seems to be more practical than systems using water because they increase the system capacity and overall performance at the studied condition. This conclusion well agrees with previous study of Chunnanond and Aphornratana [29]. Although, water as working fluid of smaller molecular mass gives a lower system capacity and performance, but it has some advantageous features. Normally, it needs the lowest mechanical power for the pumps and smallest piping size because its extremely high latent of vaporization causes a low circulation rate. Moreover, it has minimal environment impact such as zero ozone depletion potential (ODP) and global warming potential (GWP), in addition to its plentiful availability and cheaper worldwide price.

Therefore, working fluid characteristics should have the following requirements:

- The high cooling effect to mainly create an evaporator temperature T_e at 0 °C or negative values.
- The evaporation pressure in generator P_g should be just above atmospheric pressure to avoid operation at high pressures which need heavy construction and cost.
- The condenser pressure must be too high to avoid operation under high vacuum conditions, in order to lower the leakage problems and the cost of heat transfer area.
- The capability of fluid to provide a high condenser temperature at a certain compression ratio, to enhance the heat recovery in the system.
- The large latent heat of vaporization to lower circulation rate and minimizing the required pumping power.

4.6 System Improvements

From the previous analysis and discussion, obviously the unit efficiency of HDDTVC is relatively small. Thus, several approaches can be applied to technically improve the system performance by:

1- Optimum utilization of heat source via maximization of the generator and evaporator temperatures to the most possible limit, using the largest throat area of ejector nozzle and lowering the heat losses, to finally get the maximum possible productivity.

2- Minimization the pumping power by using the natural forces as alternative to the mechanical driven pumps. This idea can be achieved by elevating the condenser level above the generator level, where the gravitational head difference can easily return the refrigerant liquid to the generator.

3- Utilization of the heat pipe heat exchanger technology instead of conventional heat exchangers. The HDDTVC system includes many exchangers as: vapor generator, condensers, evaporator and heaters. Normally, heat pipe heat exchanger has enhanced heat transfer coefficients, higher thermal conductance, compact construction and low economical cost.

4- Increasing the unit thermal efficiency by introducing a pre-heater unit for the return liquid refrigerant from the condenser just before entering the generator.
   The temperature of refrigerant is slightly increased which considerably reduces the required heat input to the generator.

5- Lowering the temperature of the input refrigerant to the evaporator by using a pre-cooler in HDDTVC unit in order to improve the system efficiency. This shall increase the cooling load of evaporator which is used in the condensation process (dehumidification process).

6- The principal of heat pipe can be applied to the ejector (heat pipe-ejector) to enhance the unit coefficient of performance by eliminating the refrigerant pump. The condensed liquid refrigerant can return from the condenser to the generator unit through the wick structure of heat pipe by the capillary action.

7- Enhanced features should be met for the dehumidifier, heat exchangers and heat recovery sections by using: finned tubes, CuNi 90/10 heat transfer material and 304l SS for unit frames to withstand seawater corrosion.

8- The principle of staging can be implemented in the HDD plant by using several humidifier sections in series to increase the water content in the moist air at moderate operating temperatures.

5. CONCLUSIONS

The principal difference of the current new desalination system from the prior designs is the capability for continuous cooling to a highly super-cooled temperature beyond ambient condition during all seasons. The concept of humidification-dehumidification unit HDDTVC is substantially based on using a refrigeration cycle with thermal vapor-compression unit so as to improve the condensation system. This novel system provides the following results:

1- The novel system is considered as an alternative technology for the conventional humidification-dehumidification system. The current new system has the ability to recuperate the retrogressive product in traditional unit and produce an amount of water equal to 8 folds with respect to old system for countries of high temperature and tropical climates.

2- The idea of creation a super-cooled temperature for the dehumidification process of the HDDTVC unit for the production of freshwater depends substantially on retrieving the unavailable quantities and collecting the dew from the atmospheric air as an additional source, besides of the seawater as a main source. The ratio of retrieved and extracted water from the dew may reach about 51% and 37% of the potential composite water produced for the humid regions, which improves the economical cost of water.

3- The proposed system is developed to provide potential amounts of hundreds cubic meters per day to be classified as a promising unit of medium-scale commercial production using the low-grade heat resources.

4- The jet-ejector is considered the cornerstone in the thermal refrigeration vapor-compression unit, and its operational and design characteristics affect the performance of the humidification-dehumidification system as follow:
    - It depends to a great extend on the generator temperature, evaporator temperature and throat diameter of the nozzle.
    - Insignificant influence on the rate of air flow.
    - Independent on the maximum possible temperature of heated air and water.

5- For the operational purposes, a typical performance map is designed for the supplied jet ejector of constant configuration at different operating circumstances.

6- Working fluids of higher molecular mass halocarbon-based refrigerants (for the vapor compression unit) such as R-123 are more efficient than other fluids as:

water, ethanol and methanol, due to its increase of the HDDTVC overall performance. But, water has some advantageous features.

7- Further proposals are discussed and analyzed so as to optimize the operation and performance of HDDTVC system from a thermodynamic and economic point of view.

Based on these conclusions, the new proposed HDDTVC seems to be the promising unit for provision the arid and isolated communities with the required potable water.

Nomenclature

A	cross-sectional area, m^2.	M	Mach number.
C_p	specific heat, kJ/kg.C.	$M^{\cdot}$	distillate water production, m^3/d.
COP	coefficient of performance.	$M^{\cdot *}$	critical Mach number.
CR	compression ratio.	$m^{\cdot}$	mass flow rate, kg/s.
d	diameter, m.	p	pressure, Pa.
ER	expansion ratio.	Q	heat rate, kW.
HDD	humidification-dehumidification.	R	gas constant, kJ/kg.°C.
HDDTVC	humidification-dehumidification by thermal vapor compression.	RH	relative humidity, %.
		W	work done, kJ.
h	specific enthalpy, kJ/kg.	W	entrainment ratio.

Greek letters

γ	specific heat ratio.	φ	specific humidity, kg H_2O/ kg air
γ	efficiency		

Subscripts

a	air	n	nozzle
atm	atmospheric	p	primary stream, pump
c	condenser	r	refrigerant
d	distillate	s	saturated condition, secondary stream
ev	evaporator		
f	air fan or compressor	v	vapor
g	saturated vapor, generator	w	water
hth	high temperature heater	I-V	locations inside the jet-ejector

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