Status of humidification dehumidification desalination technology

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Abstract

Using a carrier gas to desalinate seawater has many disadvantages. Existing carrier gas systems called humidification dehumidification (HDH) desalination systems suffer from low energy recovery and the presence of non-condensable gases in the dehumidifier which is reflected in low performance and high cost of water production (up to 22 €/m^3) for these systems. In this paper, we propose and review various potential solutions to these issues. It has been found that the solutions proposed by our team and other researchers have improved the energy efficiency rather drastically. It is concluded that some of the versions of the technology which can use low grade heat are applicable for use in decentralized and small scale desalination systems. It is also noted that it is not yet possible to clearly identify the niche for some of the new versions of the technology, that use higher grade of energy, in the absence of cost details.

Nomenclature

Acronyms

GOR	Gained Output Ratio
HDH	Humidification Dehumidification
MED	Multi-effect Distillation
MSF	Multi-stage Flash
RO	Reverse Osmosis
MVC	Mechanical vapor compression
SW	Specific Work
TVC	Thermal Vapor Compressor/ Thermocompressor

Symbols

$\dot{H}$	total enthalpy rate (W)
h	specific enthalpy (J/kg)
h_{fg}	specific enthalpy of vaporization (J/kg)
HCR	heat capacity rate ratio (-)
$\dot{m}$	mass flow rate (kg/s)
P	absolute pressure (Pa)
Q	heat transfer rate (W)
T	temperature (°C)
$\dot{W}$	work rate (W)

Greek

Δ	difference or change
ε	energy based effectiveness (-)
η_{pp}	power production efficiency (-)
φ	relative humidity (-)
ω	absolute humidity (kg water vapor per kg dry air)

Subscripts

a	humid air
act	actual
cg	carrier gas
c	cold stream
D	dehumidifier
da	dry air
ht	heater
H	humidifier
i	inlet
in	entering
max	maximum
o	outlet
out	leaving
pw	product water
sat	saturated
sw	seawater
v	vapor

I INTRODUCTION

Humidification dehumidification (HDH) technology is a distillation technology which operates using air as a carrier gas. The simplest version of this technology has a humidifier, a dehumidifier and a heater to heat either the air or the seawater stream. We had previously described and analyzed various embodiments of this system [1, 2]. In general, the energy consumption of these systems is high. Also, because of presence of air in the system the dehumidifier size is very large. In this paper, we will describe the various efforts by the present authors and researchers elsewhere to solve these problems.

I.1 Performance metric

The figure of merit that defines energy performance for HDH and other thermal desalination systems is called the Gained Output Ratio (GOR). GOR is the ratio of the latent heat of evaporation of the water produced to the net heat input to the cycle. This parameter is, essentially, the effectiveness of water production, which is defined as an index of the amount of the heat recovery affected in the system.

$$\mathrm{GOR} = \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{Q}_{in}} \tag{1}$$

Latent heat is calculated at the average partial pressure of water vapor (in the moist carrier gas mixture) in the dehumidifier. We will use this parameter in all the subsequent sections to define the performance of the HDH system. The value of this parameter along with the quality of the energy input to the system and the daily water production in the system will be used to determine the niche markets for the various HDH systems.

GOR is a thermal energy based parameter, and a power production efficiency (η_{PP}) needs to be assumed to use GOR for work driven systems. This approach is useful when performance of thermal energy based desalination systems are being compared to work driven systems. However, for comparing the performance of work driven systems, specific work consumption (SW) is used. SW is the amount of electrical energy (in kJ_e) consumed to produce one kilogram of fresh water.

$$\mathrm{SW} = \frac{\dot{W}_{net}}{\dot{m}_{pw}} \tag{2}$$

$$\mathrm{GOR} = \frac{h_{fg} \cdot \eta_{pp}}{\mathrm{SW}} \tag{3}$$

I.2 Research methodology

While experimental analysis of HDH systems is perhaps the best way to investigate their thermodynamics, it is not possible to evaluate innovative changes to the system by this method alone. Hence, the present authors formulated on-design models that were used to reliably simulate the thermodynamic behavior of the HDH systems. This includes defining an energy effectiveness to quantify the performance of the humidifier and the dehumidifier. The various details of the models have been dealt with in great detail in a previous publication [3].

The energy effectiveness for the humidifier and the de-humidifier is defined as follows

$$\varepsilon = \frac{\Delta\dot{H}}{\Delta\dot{H}_{\max}} \tag{4}$$

This definition is based on the maximum change in total enthalpy rate that can be achieved in an adiabatic heat and mass exchanger. It is defined as the ratio of change in total enthalpy rate ($\Delta\dot{H}$) to the maximum possible change in total enthalpy rate ($\Delta\dot{H}_{\max}$). The maximum possible change in total enthalpy rate can be of either the cold or the hot stream, depending on the heat capacity rate ratio of the two streams. The stream with the minimum heat capacity rate dictates the thermodynamic maximum amount of heat transfer between the fluids that can be attained.

II. INNOVATIONS FOR REDUCING ENERGY CONSUMPTION

II.1 Multi stage air heated cycle

Chafik [4] proposed a novel scheme to maximize the humidity ratio at the exit of the humidifier. The air in this cycle is heated in a solar collector and sent to a humidifier where it is saturated. It is then further heated and humidified again. This is repeated in four or five stages. The idea behind this scheme was to increase the exit humidity of the air so that water production can be increased. Chafik was able to increase the exit humidity from 4.5% (by weight) for a single stage system to 9.3% for a four stage system. However, thermodynamic simulations carried out by the present authors showed that the despite the two fold increase in the humidity, the GOR of the cycle rises by only 9% [2]. This can be understood using a psychrometeric representation in Figure 1) and by rewriting GOR as a function of system parameters: vapor productivity ratio (VPR) and specific net heat input (SNH). The physical significance of these parameters is explained in a previous publication [5]. In Fig. 1, paths 1-2, 3-4 and 5-6 correspond to humid air heating and paths 2-3, 4-5, and 6-7 correspond to the humidification process. Path 7-1 corresponds to the dehumidification process.

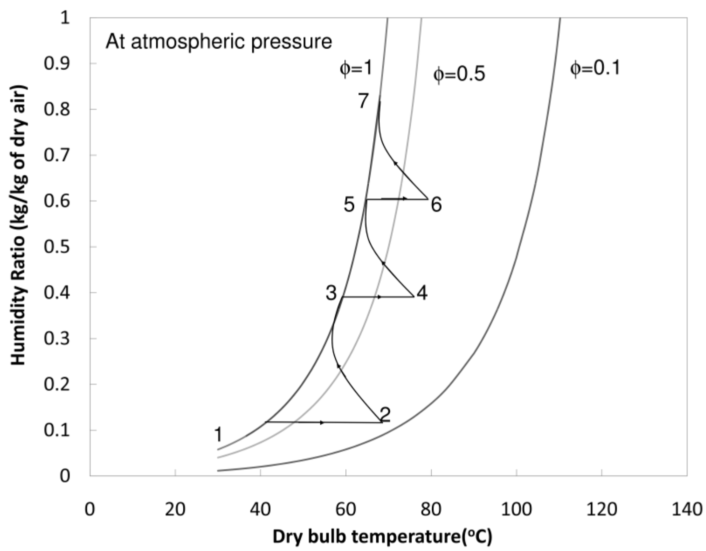

Fig. 1. Psychrometeric representation of multi stage air heated HDH system

The Gained Output Ratio can be rewritten as follows

$$\mathrm{GOR} = \underbrace{\left\{\frac{\dot{m}_{pw}}{\dot{m}_{cg}\cdot \boldsymbol{\omega}_{H,o}}\right\}}_{VPR} \underbrace{\left\{\frac{\dot{m}_{cg}\cdot \boldsymbol{\omega}_{H,o}}{\dot{Q}_{in}}\right\}}_{1/\mathrm{SNH}} h_{fg} \tag{5}$$

$$=\mathrm{VPR}\cdot\frac{1}{\mathrm{SNH}}\cdot h_{fg} \tag{6}$$

VPR and SNH, in turn, can be expanded and written as follows

$$\mathrm{VPR} = \frac{\omega_{D,0}}{\omega_{H,0}} \tag{7}$$

$$\mathrm{SNH} = \frac{\Sigma(h_{a,ht})}{\omega_{H,0}} \tag{8}$$

Here, $\omega_{H,o}$ refers to the humidity from the outlet of the final humidification stage. Designing the HDH system with multiple stages of the heating and humidification processes increases the VPR because $\omega_{H,o}$ is increased. However, SNH is also increased because for each humidification stage heat is input to the moist air stream. Hence, the GOR is not increased greatly by this innovation.

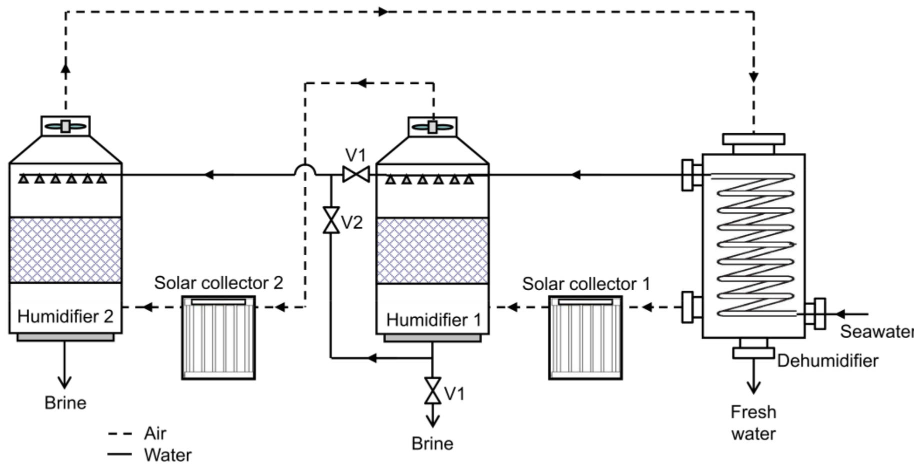

Fig. 2. Two stage air heated HDH system

A two stage HDH air heated system similar to the above mentioned system [4] is designed and being built at King Fahd University, Dhahran where various options of both air and water cycles are considered. In Chafik's system, seawater is preheated in the dehumidifier then distributed in parallel to the humidifiers. However, it was found (theoretically) that using series configuration (where water leaving the first humidifier is fed to the next humidifier), increases the GOR by about 5% compared to the parallel configuration. Figure 2 represents the two stage air heated HDH system. It is designed to have both options of parallel or series water flow.

II.2 Mechanical compression driven HDH

HDH systems have traditionally been designed to operate at atmospheric pressure. However, to increase the vapor content of moist air the systems need to be operated at subatmospheric pressures. For example, at a dry bulb temperature of 65 °C the humidity ratio of moist air is increased two fold when the operating pressure is reduced from 100 kPa to 50 kPa. However, if the entire HDH system is operated under this reduced pressure, the increase in thermal performance is relatively low. As we reduce the pressure of the entire system, the humidity ratio throughout the system is increased. Thus, VPR is only very slightly increased and SNH essentially remains the same as in atmospheric pressure systems. Hence, we find the increase in GOR to be very small. If we are to increase the humidity ratio only in the humidifier and not in the dehumidifier then, the VPR can be maximized. This can be done by the varied pressure HDH system invented by the present authors [5].

The proposed cycle operates the humidifier and dehumidifier at different pressures. As shown in Fig. 3, the pressure differential is maintained using a compressor and an expander. The humidified carrier gas leaving the humidification chamber is compressed in a mechanical compressor and then dehumidified in the condenser or the dehumidifier. The dehumidified carrier gas is then expanded to recover energy in form of work. The expanded carrier gas is then sent to the humidification chamber. The carrier gas is thus operated in a closed loop. The feed seawater is preheated in the dehumidifier before it is sent to the humidification chamber thus recovering some of the work input to the compressor in form of thermal energy which is given back to the carrier gas stream during the humidification process. The brine from the humidification chamber is then disposed of.

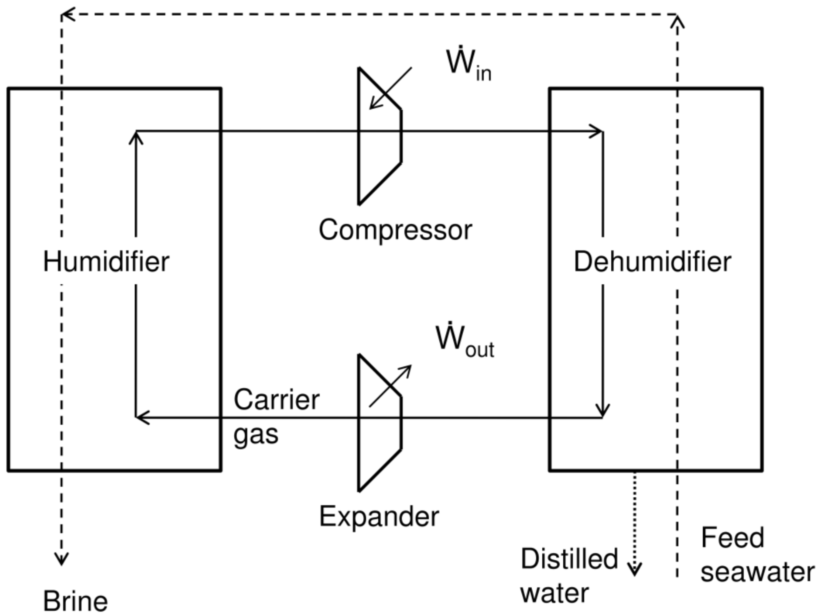

Fig. 3. Schematic diagram of mechanical compression driven HDH system

From a previous study we understand the following about the mechanical compression driven HDH system:

1. A parametric study explaining the influence of various system and component variables on system performance showed that important design parameters include the expander and compressor efficiencies, air side pressure drops in the humidifier and the dehumidifier, and the pressure ratio provided by the compressor.
2. The possibility of using a throttle instead of a mechanical expander was examined and it was found that the cycle with the throttle has a much higher energy requirement because of high irreversibility in the throttling process.
3. The mechanical compression driven HDH cycle has much higher performance compared to existing HDH cycles. This was made possible because of the increase in VPR from a value of 0.2-0.4 for single pressure cycles to 0.5-0.8 for compression driven cycle.
4. However, in spite of the increase in performance, the new cycle remains less efficient than RO and MVC for seawater desalination.

Further, research is being pursued to bring the performance closer to MVC and RO by reducing the gas side pressure drop and also by designing the mechanical compression cycles with thermodynamic balancing. Such concepts are explained in the succeeding sections.

II.3 HDH with thermodynamic balancing

-Holst pioneered a novel scheme to balance the stream-to-stream temperature difference in the humidifier and the dehumidifier in HDH systems: circulating air by natural convection [6]. He reported that the balanced system had a low thermal energy consumption of up to 120 kWh/m^3. More recently, Terra Water GmbH has patented a similar technology for HDH systems with air circulated by forced convection [7]. In this section, we develop an understanding of these systems by comparison with MED.

Multiple-effect distillation (MED) consists of several consecutive chambers (effects) in which seawater is vaporized and subsequently cooled to form pure liquid water per Fig. 4. In a single effect system, the input heat required to vaporize the water is equivalent to the latent heat of evaporation, h_{fg}, of the mass of water transformed from a liquid to gaseous state. All of the heat input into the system is used once to evaporate the water and there is no heat recovery. This cycle corresponds to a gained output ratio (GOR) of about 1.

To improve GOR, heat can be recycled from this first effect to evaporate an additional mass of water in subsequent effects. In a typical arrangement such as Figure 4, no additional heat supplements the downstream effects, and the heat from the previous effect is recovered in the condenser to power the boiler of the new effect. Thus, each effect will be at a lower temperature than the preceding, and therefore the pressure must be reduced at each stage in order to induce vaporization. Ideally, this process could be repeated in an infinite number of effects, but in application the number of effects is limited by practical heat exchanger sizes as well as the inability to design for low leakage losses to the ambient at lower pressures; therefore, there is a finite temperature drop from each stage to the next. If the system is sized to produce an equivalent mass of water in each stage, the GOR is approximately equal to the number of stages. For instance, for the system mapped in Figure 5, the GOR is 12.

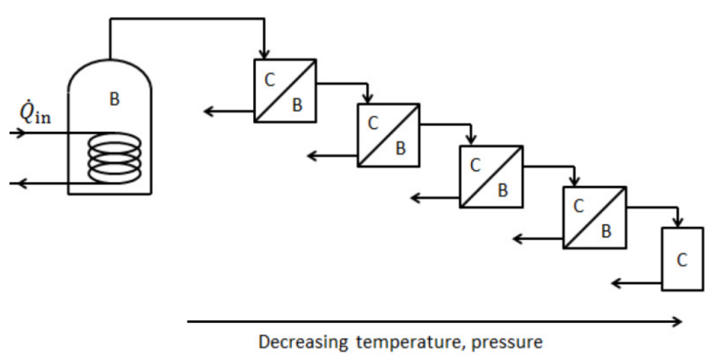

Fig. 4. Typical MED System

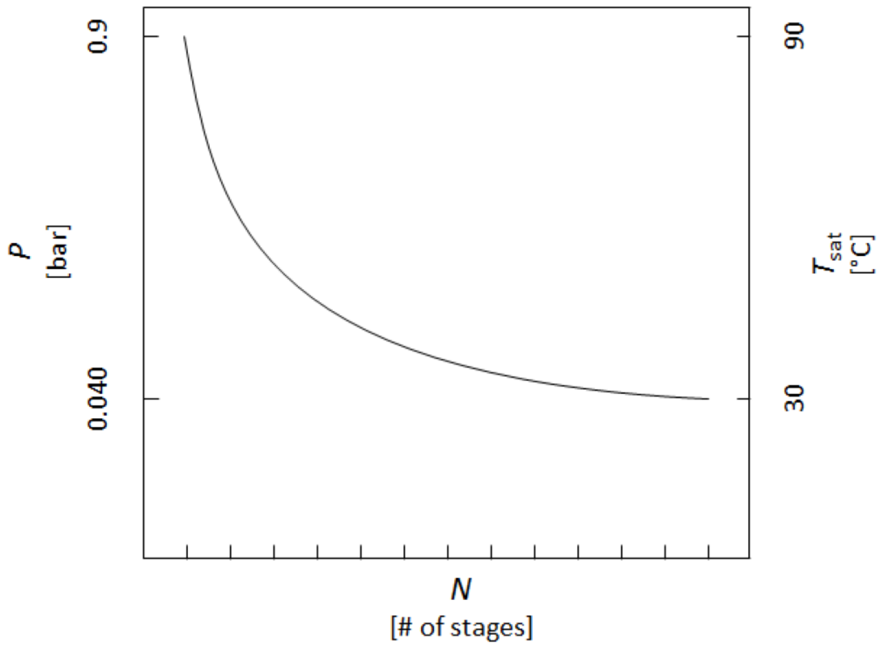

Fig. 5. MED Pressure and Temperature Dependency on Number of Stages

For a comparable HDH system, the cycle is not broken up into separate effects because the evaporation and condensation does not occur at a constant temperature. An example closed air, open water, air-heated HDH system is illustrated in Figure 6. Using the same boundary conditions as with the MED system above, the HDH system initially has a much lower value of GOR of 3-4. As seen in Figure 7, the air and water streams of the system are not balanced. Wherein the MED system, the water temperature ran the full range of 90 C to 30 C, in the HDH dehumidifier the water stream cannot maintain the same ΔT to the air stream along the length of the dehumidifier. The temperature range for the water is about 1/3 - 1/4 of the MED system. The GOR sees a similar reduction down to 3-4. A similar case exists in the humidifier where the ΔT between the two streams widens away from the inlet and outlet.

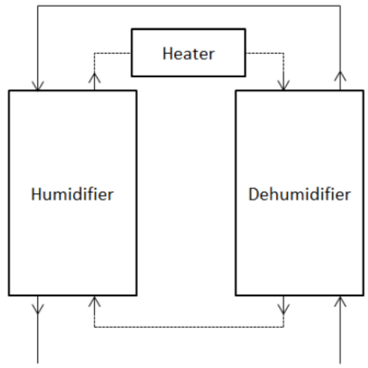

Fig. 6. Closed Air Open Water, Air Heated, Humidification Dehumidification System

In a heat exchanger where the driving force is limited to temperature difference alone, a system may be thermally balanced by minimizing the maximum temperature difference between the two streams. In a simultaneous heat and mass exchanger (such as the humidifier and dehumidifier in an HDH system), the driving force is a combination of the temperature difference and the mass concentration difference. A thermally balanced HDH system maintains a constant driving force by maintaining a constant difference in temperature and mass concentration between the air and water streams of both the humidifier and the dehumidifier.

In order to balance the humidifier and dehumidifier, the air and water streams of each must transfer an ideal amount of heat and mass between the two which can be accomplished only when the driving force within the system is kept constant. In such a case, the temperature profiles of the two streams along the length of the humidifier and dehumidifier are parallel. In an unbalanced case, the heat flow from one stream to the other is far from ideal because the heat capacity of one stream is dissimilar to the other. If one stream is unable to transfer enough heat to the other, the temperature gap widens and the system becomes less efficient. Conversely, when the heat capacity rate of each stream is equivalent, the heat flow between the streams is optimized, temperature divergences are eliminated.

It has been proposed to balance the humidifier and dehumidifier by extracting fluid from either the air or water stream in one component and injecting it into the other. [8]. By changing the heat capacity rate of the streams at any point inside the component, the two streams may be forced closer to each other to reduce entropy generation at a given point within the component. For example, extracting water from the dehumidifier (Figure 8) will adjust the temperature profile of the water stream and bring it closer to a parallel arrangement with the air stream temperature profile, thus reducing the entropy generation in the component. If this combination of extraction and injection is performed multiple times, in an arrangement known as multi-extraction (ME), the difference in the capacity rate between the two streams in both components is reduced along the entire length of the device as seen in Figure 9.

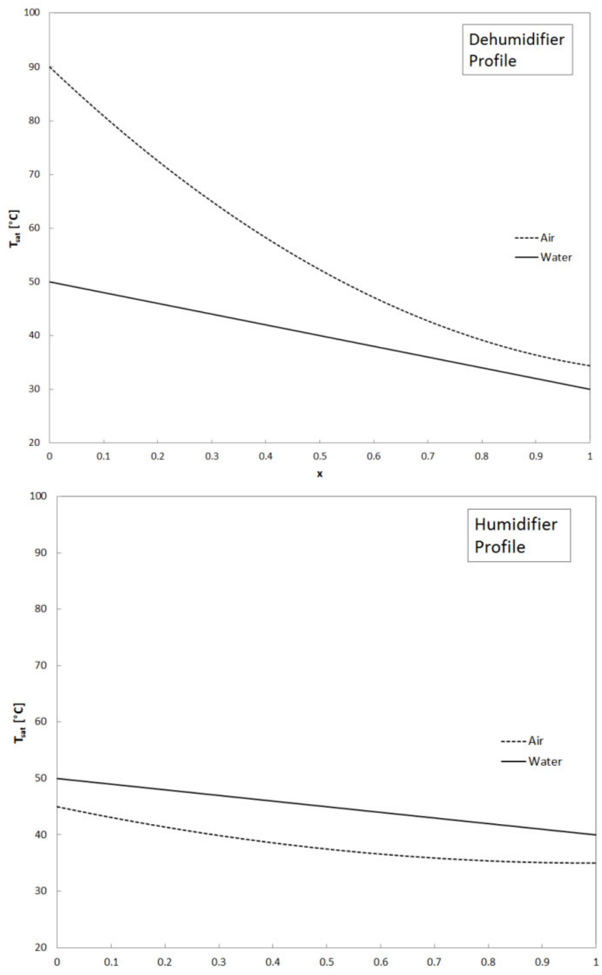

Fig. 7. Unbalanced Dehumidifier and Humidifier Temperature Profiles

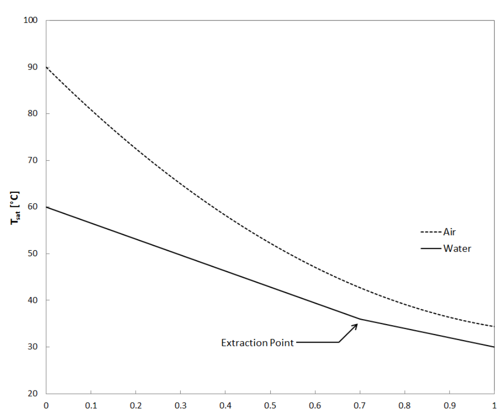

Fig. 8. Dehumidifier Temperature Profiles: Single Extraction

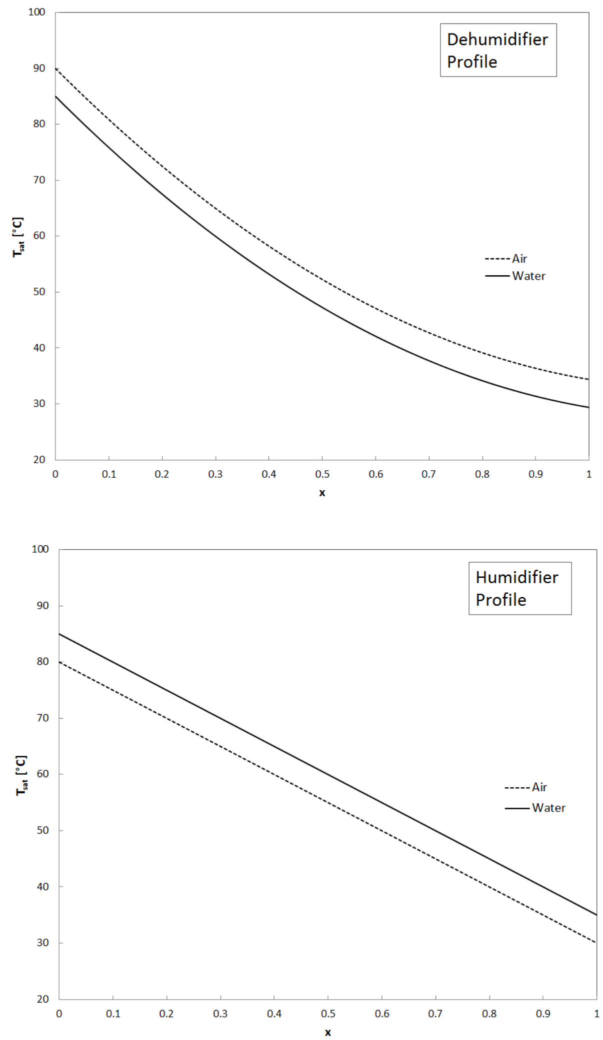

Fig. 9. Dehumidifier and Humidifier Temperature Profiles: Multi-Extraction, Fully Balanced

In the fully balanced system (as described in [7]), the maximum water temperature (before heating) has been increased from 50 to 85 C by reducing the terminal temperature difference in the dehumidifier. It is possible to increase the temperature range with a single or small number of extraction points, and such balancing would improve GOR to be close to that of a MED system with a similar temperature/vapor pressure range. However, this method will not have a value of GOR which matches the equivalent MED system because there is a finite stream to stream temperature difference to drive the rate processes. With a single extraction, the slope of the temperature profile within a component can be improved to better match the adjacent stream, but there will still be a large temperature divergence of the two streams away from the system inlet, outlet, and extraction point. Only when there are multiple extractions can the stream profiles be forced together along the entire length of the component causing the mismatch between the streams to reach a minimum. In this fully balanced case, the system has a similar temperature range as compared to the MED system and, importantly, has reduced entropy generation due to small mismatch between the profiles of each stream. The balanced system then approaches the performance of an MED system with identical boundary conditions. In Figure 9, this system may have a GOR of ~11 as compared to the MED GOR of 12. In this case, the act of balancing has improved GOR dramatically by optimizing the heat flux between the streams of both humidifier and dehumidifier and minimizing entropy generation.

II.4 HDH with common heat transfer wall

Several works [9-11] have been published on the subject of so-called dewvaporation, a variant of humidification-dehumidification desalination that combines both major components into one unit separated by a thin wall. The inventors of the technology contend that such a common wall enhances heat recovery between the humidifier and dehumidifier, resulting in a net performance improvement and an increased GOR. A basic schematic of the process is shown in Figure 10. The carrier gas, air, enters the evaporation side of the unit, where it contacts a film of warm feed water sprayed along the side of the common wall. The vaporization of water from the film into the air is enhanced by heat transfer through the thin wall from the condenser side. Upon exiting the evaporator, the warm, moist air is further heated and humidified with an injection of steam; this mixture then enters the condenser. The higher temperatures on the condenser side yield a net heat transfer through the wall to the evaporator side, causing the water vapor in the humid air to cool and condense on the surface of the common wall. The entire process occurs at atmospheric pressure.

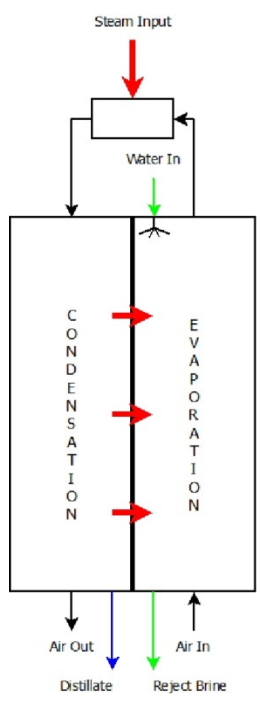

Fig. 10. Schematic of HDH system with common heat transfer wall.

Hamieh and Beckman [9] report theoretical values of GOR as high as 16.8 for brackish water (defined as a TDS of 800 ppm) and 9.5 for seawater (TDS = 42,000 ppm), which imply the potential for the technology to compete on an energy basis with large scale thermal technologies such as MSF and MED. The authors' first experimental values of GOR are considerably lower (11 for brackish water), however, and are explained by incomplete wetting of and poor air distribution along the common heat transfer surface. Experimental results are also given for high recovery ratios of up to 85% with no scaling issues. Because the evaporation process occurs at the liquid-vapor interface and not directly on the common heat transfer wall, all salt precipitation is reported to have flowed down along the liquid film and collected at the bottom of the unit, leaving the heat transfer surface unaffected by scale formation. With appropriate measures for controlling the falling precipitates, these results affirm an advantage of HDH and perhaps thermally-driven desalination process in general–the ability to desalt high salinity input waters at high recovery ratios with minimal fouling and performance impacts.

The same authors reported experimental performance in a later publication [11], with best salt rejection from a seawater (TDS = 33,000 ppm) system with a production rate of 1.8 kg/h and a heat transfer area of 7.9 m^2 achieving a GOR of 4.57. Such figures yield a production of 5.6 kg/m^2d. Cost data are also given for a 3.79 m^3/d (1000 gal/d) unit, and range from US $1648 to US $2814 in capital expenses, using the same materials used in the experimental tower.

II.5 Hybridization of HDH system with RO

We had, previously, shown that the thermal energy consumption of steam driven thermal desalination systems can be decreased significantly by reducing the total entropy rate of steam entering the system [12]. We proposed a novel carrier gas based desalination cycle which can use steam at a high temperature (>200 C) without causing formation of hard scales. This system is based on the principle of humidification dehumidification (HDH) desalination. This system was found to have a high GOR when hybridized with RO.

In this proposed system we operate the humidifier and dehumidifier at different pressures. As shown in Fig. 11, the pressure differential is maintained using a thermal vapor compressor (TVC) and an expander. Conceptually, this is similar to the variable pressure HDH system using a mechanical compressor that we described in section 3 except for the use of a TVC instead of a mechanical compressor. The humidified carrier gas leaving the humidification chamber is compressed in a TVC using a steam supply and then dehumidified in the condenser (or the dehumidifier). The dehumidified carrier gas is then expanded to recover energy in form of work. The recovered work is used in a reverse osmosis unit to desalinate the brine from the humidifier. The expanded carrier gas is send to the humidification chamber. The carrier gas is thus operated in a closed loop. The feed seawater is preheated in the dehumidifier before it is sent to the humidification chamber thus recovering some of the energy input to the compressor in form of thermal energy which is given back to the carrier gas stream during the humidification process. As shown in Fig. 11, both the thermo-compression and the expansion process can lead to condensation of a small amount of water out of the carrier gas.

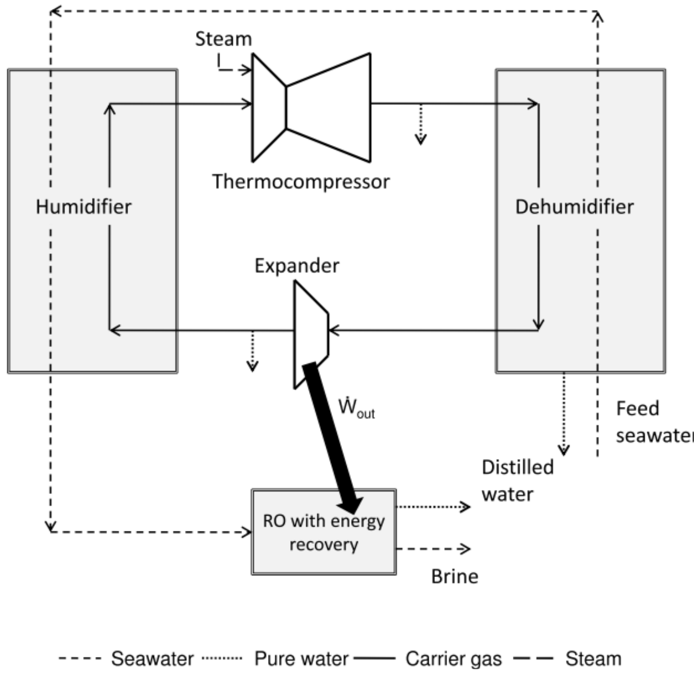

Fig. 11. Schematic diagram of thermal vapor compression driven HDH-TVC-RO system.

Figure 12 illustrates an example of the new desalination cycle on a psychrometric chart. Path 1-2 is the air humidification process that is approximated to follow the saturation line. Path 2-3 is the thermo-compression process in which the humidified air is compressed to a higher pressure and temperature. Path 3-4 is the dehumidification process which is also approximated to follow the saturation line at a higher pressure, P_D. Path 4-1 is the air expansion process where some of the energy input in the compressor is recovered.

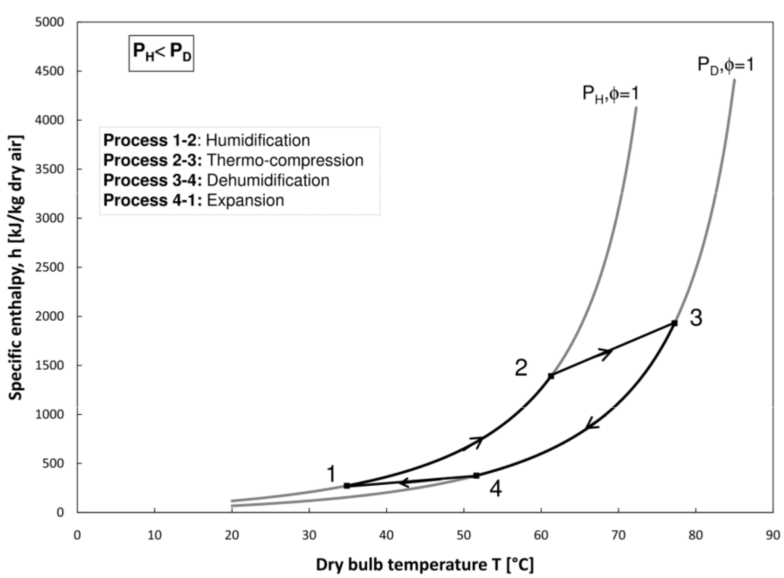

Fig. 12. Psychrometeric representation of thermal vapor compression driven HDH system.

From a previous study we understand the following about TVC driven HDH system hybridized with RO [12]:

1. It has been found that recovering energy given by steam in the TVC as a work in an expansion process leads to a more efficient system than using a isenthalpic throttling process. However, use of an efficient expander leads to majority of the water being produced in the RO unit coupled to it. In these scenarios, the HDH system itself operates as a power and water co-production unit where all of the power produced is used for further desalination in a RO unit.
2. For optimal performance of the HDH-TVC-RO system, use of an efficient TVC and a high pressure (P_{st} = 10 to 30 bar) steam are most important.
3. It is also pivotal to design the system for as low a pressure ratio as is possible. The air side pressure drops in the HME devices also play a significant role.
4. The humidifier pressure has been found to have little effect on the system performance, it is the pressure of the dehumidifier relative to the humidifier that is of importance.
5. The HDH-TVC-RO system is appropriate for medium scale, stand alone, decentralized seawater desalination using medium pressure steam. In these situations a GOR of 20 and an equivalent electricity consumption of 9.5 kWh/m^3 can be attained.

II.6 Merits and demerits of the new systems

Each of the technology discussed in this section have merits and demerits which are briefly described in Table 1.

Table 1. Merits and Demerits of innovations for reducing energy consumption of HDH systems

Technology	Merits	Demerits
Multi stage air heated HDH system	None	Low energy recovery resulted in a high system cost [4]
Mechanical compression driven HDH system	High energy recovery	Use of electricity instead of low grade heat
HDH with thermodynamic balancing	High energy recovery	Performance is a strong function of feed seawater and ambient conditions. Variations lead to drastic reduction in performance.
HDH with common heat transfer wall	High energy recovery	Performance is depended on wettability of the dehumidifying surface. Also, large surface area required for the heat transfer wall.

Thermal compression driven HDH	High energy recovery. When hybridized with RO, performance is comparable to MSF and MED.	Use of RO increases need for maintenance. Needs medium pressure steam (>10 bar) to run efficiently.

III INNOVATIONS TO REDUCE DEHUMIDIFIER SIZE

III.1 Helium as carrier gas for the HDH system

The primary advantage afforded by carrier gas as opposed to pure vapor desalination technologies is the ability to operate at close to atmospheric pressure whilst maintaining a low water saturation temperature. Unfortunately, the presence of a carrier gas impacts negatively upon other aspects of system design and performance. Typically, HDH processes use air as a carrier gas, primarily due to it being readily available at all locations. However, where air operates within a closed loop, the question must be asked as to whether there exists a carrier gas that gives superior system performance. Advantages offered by using alternative carrier gases may be categorized as those that reduce the system size, and thus capital cost, and those that improve the energetic performance of the system. These categories are discussed in turn.

III.1.1 Effect of Thermophysical Properties upon System Size

The presence of any carrier gas within a HDH system will always degrade heat transfer coefficients, since a further thermal resistance is introduced that would not be present in a pure vapor environment. The extent to which the carrier gas impedes heat transfer depends upon its thermophysical properties such as its thermal conductivity, specific heat capacity and density. Table 2 lists all important thermophysical properties of air, helium, nitrogen, argon and carbon dioxide at $T = 25°C$ and $P = 1$ atm. Since these gases are to be used in a mixture with steam, the properties of dry saturated steam at a pressure of one atmosphere are also shown.

Table 2. Thermophysical properties of different carrier gases at STP and dry saturated steam at atmospheric pressure

	M [g/mol]	k [W/m.K]	c_p [J/kg.K]	ρ [kg/m^3]
Air	28.97	0.02551	1.005	1.169
He	4.003	0.1502	5.193	0.1615
N_2	28.01	0.02568	1.038	1.13
Ar	39.95	0.01796	0.5203	1.611
CO_2	44.01	0.01657	0.8415	1.775
Steam	18.02	0.02503	2.043	0.5897

Typically, the size and cost of the dehumidifier required in HDH systems is of significant concern due to the high thermal resistance of the air carrier gas. The use of helium as a carrier gas is interesting, as due to its superior thermal conductivity, a helium carrier gas provides greater heat transfer coefficients and smaller dehumidifier sizes. From a previous study conducted by the authors [13], the heat transfer coefficient achieved during the dehumidification of moist helium was estimated to be 5-6 times that achieved during the dehumidification of moist air.

III.1.2 Effect of Carrier Gas upon System Performance

Since, for a given rate of water production, the dehumidifier is smaller using helium rather than air as a carrier gas, it follows that the pressure drop would also be reduced. In previous work by the present authors, it was shown that the gas side pressure drop in the dehumidifier using moist helium could be 1/5 to 1/8 times that when using air as the carrier gas. As we had previously shown [5,12], the pressure drop in the dehumidifier (or humidifier for that matter) significantly affects the overall performance of HDH systems.
Figure 13 highlights the benefits in terms of energetic efficiency of reduced dehumidifier pressure drops. Other than the reduced dehumidifier pressure drop in the case of helium, it avers that there is little difference between the thermodynamic performance of helium and air carrier gas cycles. The thermodynamic performance of a HDH cycle is dictated by the mole fraction rather than the mass fraction (humidity) of vapor in the carrier gas.

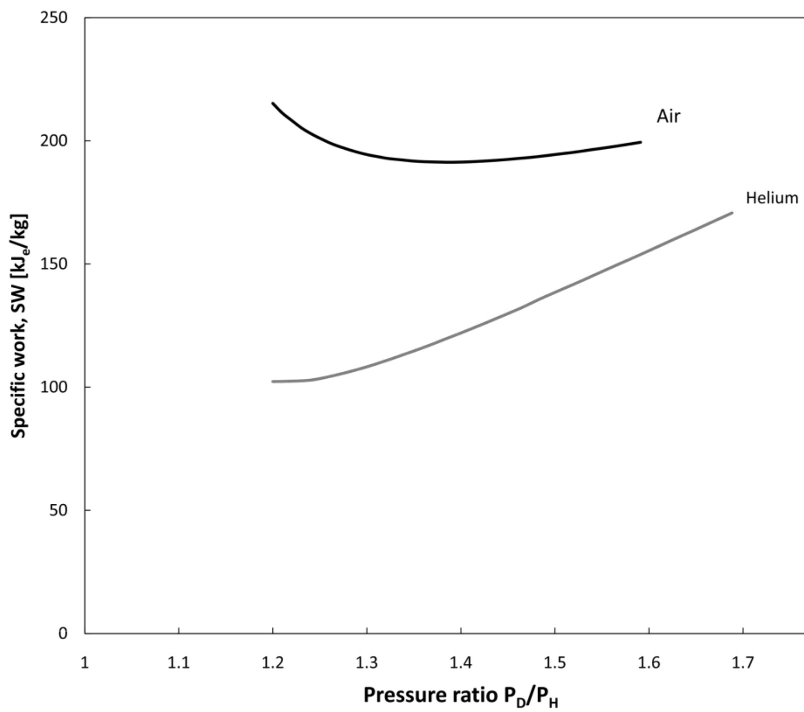

Fig. 13. The effect of reduction in gas side pressure drop in the dehumidifier on the relative performance of mechanical compression driven HDH cycle with helium and air as carrier gas.

$\Delta P_{H,cgm} = 0.5$ kPa; $\Delta P_{D,a} = 2$ kPa; $\Delta P_{D,he} = 0.4$ kPa; $T_{sw,in} = 30°C$; $\varepsilon_H = \varepsilon_D = 80\%$; $\eta_{com} = \eta_e = 100\%$; $P_{D,in}\ /\ P_{H,out} = 1.2$; $HCR_{\mathrm{H}} = 1$.

III.2 HDH system with direct-contact dehumidification

A schematic of a HDH system with a direct contact dehumidifier is shown in Fig. 14. In this system, the air is humidified in the humidification chamber in counter flow with hot brine. The humidified air is then dehumidified in a direct contact heat exchanger using distilled water. In the process the distilled water is heated up. The heated pure water is used to preheat the feed seawater in a separate liquid-to-liquid heat exchanger. Klausner and Mei [14] pioneered such a system.

The heat transfer coefficients are higher for the direct contact dehumidifier than for the other HDH systems. Hence, the heat exchanger size and cost are relatively low. But, because the heat from the humidified air is not recovered directly to preheat seawater, the system heat recovery is low and the energy consumption is high. This can be understood using Eq. 6. VPR does not change when a direct contact dehumidifier instead of a non-direct contact heat exchanger is used, but the SNH increases because the heat input in the water heater is more since the energy recovery in the dehumidifier is reduced. For this reason, these systems have not become commercially viable.

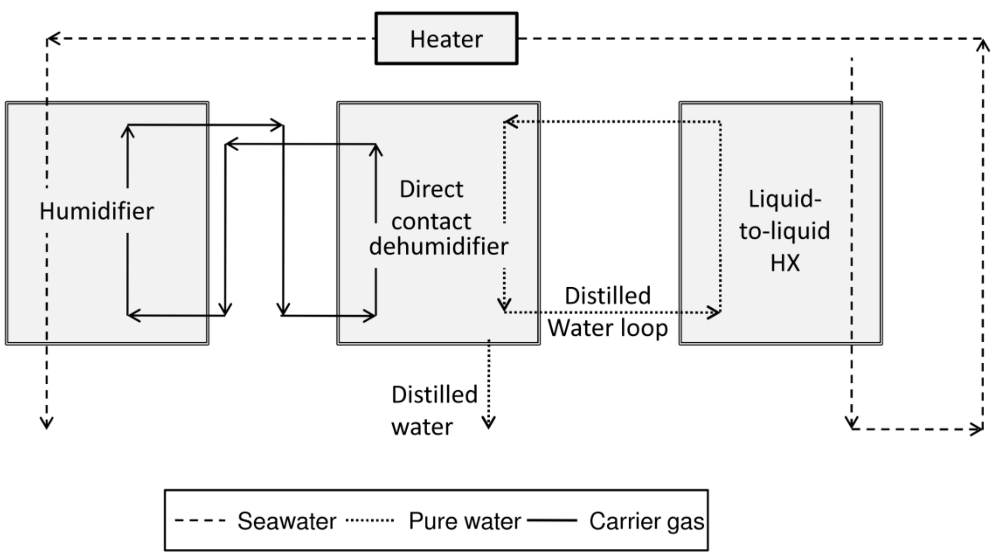

Fig. 14. HDH system with direct contact dehumidifier

IV. Potential Niche for new HDH technologies

1. Mechanical compression driven HDH has great potential because the capital cost of the system itself is expected to be very low. However, balancing and other methods to reduce energy consumption need to be implemented to reduce the energy cost. The niche could be

very small scale, PV driven, decentralized water production. It has a distinct advantage in that the performance (like other thermal desalination systems) is independent of salinity and no pretreatment is required. In comparison to MVC systems which require expensive turbomachinery (produced by only a select few companies around the world), mechanical compression driven HDH system is expected to be simpler and to be made up off inexpensive off-the-shelf components. Research work is in progress in this regard.
2. Currently, two companies from Germany (Magewater Management GmbH and Terra Water GmbH) have commercialized (single pressure) HDH systems with thermodynamic balancing. The niche for this technology is small scale, renewable energy based, decentralized sea and brackish water desalination.
3. The HDH system with a common heat transfer wall is also a commercial product which is patented and produced by two companies - Altela Inc in the US and Three Millenium Water Technologies in France. The niche for these systems is similar to the HDH systems with thermal balancing. Three Millenium Water Technologies is also marketing the product as modular and applicable for larger scales. Also they do not necessarily describe their product as solar energy driven.
4. The HDH system driven by a TVC and hybridized with RO has great potential if an appropriate and cost effective way to produce medium pressure steam is identified. The present authors are pioneering this effort.

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