Proceedings of ASME 2011 International Mechanical Engineering Congress & Exposition
IMECE 2011
November 11-17, 2011, Denver, USA

IMECE2011-62875

HELIUM AS A CARRIER GAS IN HUMIDIFICATION DEHUMIDIFICATION DESALINATION SYSTEMS

G. Prakash Narayan,
Ronan K. McGovern,
John H. Lienhard V*
Rohsenhow Kendall Heat Transfer Laboratory
Room 3-162, Department of Mechanical Engineering
Massachusetts Institute of Technology
Cambridge, MA 02139, USA.

Syed M. Zubair
Department of Mechanical Engineering
King Fahd University of Petroleum and Minerals
Dhahran, Saudi Arabia.

ABSTRACT

A promising technology for small scale seawater desalination is the humidification dehumidification (HDH) system. This technology has been widely investigated in recent years. Since existing HDH systems have very high specific energy consumption, the authors have previously invented several ways to increase the energy efficiency of these systems. Even for these relatively higher efficiency systems the dehumidifier is expected to be large, owing to the large thermal resistance associated with the presence of non-condensable carrier gas (air) in the system. In this manuscript, we demonstrate that changing the carrier gas from air to helium a potential solution to this problem. In addition, the energy performance of a brine heated HDH system using helium relative to those using air is analysed in detail through well established on-design models for the components in the system.

NOMENCLATURE

Acronyms

GOR Gained Output Ratio
HDH Humidification Dehumidification
HE Heat Exchanger
HME Heat and Mass Exchanger
MED Multi-effect Distillation
MSF Multi-stage Flash
RO Reverse Osmosis

Symbols

c_p specific heat capacity at constant pressure (J/kg·K)
D_h hydraulic diameter (m)
h specific enthalpy (J/kg)
h_{cgm} specific enthalpy of moist carrier gas (J/kg of dry carrier gas)
h_{fg} specific enthalpy of vaporization (J/kg)
HCR modified heat capacity rate ratio (-)
Ja Jakob number (-)
k thermal conductivity (W/m.K)
M molar mass (kg/kmol)
m_r mass flow rate ratio (-)
$\dot{m}$ mass flow rate (kg/s)
Nu Nusselt number (-)
p partial pressure (Pa)
P absolute pressure (Pa)
ΔP pressure drop (Pa)
$\dot{Q}$ heat transfer rate (W)
Re Reynolds number (-)
RR recovery ratio (%)
s specific entropy (J/kg·K)

*Address all correspondence to this author. Email:lienhard@mit.edu

s_{cgm} specific entropy of moist carrier gas (J/kg of dry carrier gas)
$\dot{S}_{gen}$ entropy generation rate (W/K)
SW specific work (J/kg)
T temperature (°C)
u heat transfer coefficient (W/m^2.K)
v_{mix} velocity of the carrier gas mixture (m/s)
$\dot{W}_{aux}$ auxiliary electricity consumption(kW)
x mole fraction (mol/mol)
z coefficient for experimental correlations (-)

Greek

Δ difference or change
ε energy based effectiveness (-)
ϕ relative humidity (-)
Φ ratio of Nusselt numbers (-)
ρ density (kg/m^3)
ω absolute humidity (kg water vapor per kg dry carrier gas)

Subscripts

a humid air
c cold stream
cg dry carrier gas
cgm humid carrier gas
D dehumidifier
e expander
h hot stream
H humidifier
he humid helium
ht heater
i inlet
in entering
max maximum
o outlet
out leaving
pw product water
rev reversible
sat saturated
w water

Superscripts

$ideal$ ideal condition
rev reversible

1 Introduction

Widely used thermal desalination technologies such as multi-stage flash (MSF) and multi-effect distillation (MED) are not suitable for small scale (1-100 m^3/day) applications. Reverse osmosis (RO) is suitable for these applications but it requires a continuous supply of electrical or mechanical energy. Many developing countries which suffer from water scarcity also lack in resources which can generate these sources of energy (fossil fuels). Some of these countries have an abundance of solar energy. Solar photovolatics can be used to operate reverse osmosis units for small scale applications in these countries. However, it may not be feasible due to the high cost of PV modules and maintenance of RO systems [1]. A much simpler option is to use the solar energy as a source of thermal energy. This requires the development of alternate thermal desalination technologies which can use this energy in an efficient way.

A promising alternative technology is the humidification dehumidification (HDH) desalination system, a small scale distillation technology which mimics nature's rain cycle. This technology has received ongoing attention in recent years, and several researchers have investigated and reviewed many realizations of this technology [2, 3]. These systems have very high specific energy consumption. However, several new ways to increase the energy efficiency of HDH systems have been invented and analysed by the present authors [4, 5, 6, 7]. For these relatively higher efficiency systems, a significant bottleneck for commercialisation remains the presence of a high percentage of non-condensable gas (air) in the condenser (or dehumidifier). This creates a large resistance to heat transfer in the condenser. Hence, the size of the heat exchangers required for condensation of water from the moist air mixture may be quiet large. Also, the gas side pressure drop and the associated auxiliary power consumption is high. In this paper, we propose to solve these problems by changing the carrier gas from air to helium. The energy performance of HDH systems using helium relative to air is analysed in detail using novel on-design models for the components described in a previous publication [8].

2 Rationale for selecting helium

In HDH systems, the properties of the carrier gas affect the overall cost of water production. While the psychrometeric properties affect the thermodynamics and the energy consumption of the system, the thermophysical properties, like thermal conductivity, affect the heat transfer area required in the heat and mass exchange devices. In this section, we consider these properties for various gases which could possibly be used in HDH systems and outline the rationale for using helium in HDH systems.

TABLE 1. Thermophysical properties of different carrier gases at STP and of 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

2.1 Thermophysical properties

Table 1 shows all the important thermophysical properties for air, helium, nitrogen, argon and carbon dioxide at T=25°C and P=1 atm. Since these gases are to be used as a mixture with steam in the HDH systems, the properties of dry saturated steam at P=1 atm are also shown.

It is observed that helium has a much higher thermal conductivity than the other gases in consideration, about six times that of air. The thermal conductivity is especially important because a higher value will help reduce the gas side thermal resistance in the dehumidifiers.

The specific heat capacity is also much higher for helium than for the other gases. Specific heat affects the ratio of mass flow rate of seawater entering the system to that of the carrier gas circulated in the system. The performance of a HDH system is optimal when operated at a particular ratio of mass flow rates [4]. Hence, it is important to identify this ratio for each of the carrier gases at a given operating condition; this is done as part of the thermodynamic study later in the paper.

As can be seen from density of the gases, helium is by far the lightest gas and this may cause some operational difficulties. Dynamic viscosity (not shown) is about the same for the gases under consideration.

2.2 Psychrometric properties

Using Dalton's law and approximating the carrier gas and water vapor mixture as an ideal mixture we can write down the humidity ratio of the carrier gas as a function of the mixture temperature (T), pressure (P), relative humidity (ϕ) and molar mass (M) of the carrier gas.

$$\omega(T,P,\phi) = \frac{M_w}{M_{cg}} \cdot \frac{\phi \cdot p_{sat}(T)}{P - \phi \cdot p_{sat}(T)} \tag{1}$$

From this equation, it may be seen that the humidity ratio is higher for low molar mass gases. We know that helium has low molar mass compared to air (see Tab. 1). Hence, the humidity ratio for helium is much higher than that for air (Fig. 1). At any given temperature, total pressure, and relative humidity, the humidity ratio for helium is 7.23 times that of air. As we would expect based on Eq. 1, this ratio is same as the ratio of the molar mass of helium to air.

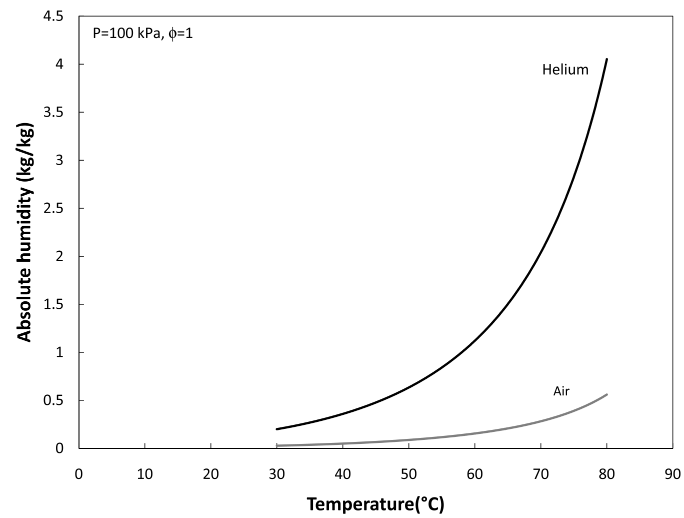

FIGURE 1. Psychrometeric chart for helium water vapor mixture and moist air.

The higher humidity ratio leads to a lower total mass flow of carrier gas per unit of water produced. This leads to a lower gas side pressure drop and a smaller auxiliary power consumption as explained in Sec. 2.4.

2.3 Estimate of gas side heat transfer coefficient in the dehumidifier

In the dehumidifier of a HDH system the thermal resistance to heat transfer associated with the presence of non-condensables (air) is much higher than thermal resistance of the liquid film formed as a result of the condensation of steam from the moist air mixture [9]. In this section, an estimate of this thermal resistance using wealth of experimental data in literature is made [10, 11]. In particular, researchers from BARC (Bhabha Atmoic Research Center) in India [12] reviewed various experimental correlations that give the condensation heat transfer performance of steam in the presence of non-condensables (including air and helium) for simple geometries (e.g., a vertical tube). The correlations are generally of the form

$$\mathrm{Nu_{cgm}} = \mathrm{z}_1 \cdot \mathrm{Re_{cgm}^{z_2}} \cdot x_{cg}^{z_3} \cdot \mathrm{Ja_{cgm}^{z_4}} \tag{2}$$

The Nusselt number is given as a function of the Reynolds number of the mixture, the bulk mole fraction of the carrier gas in the mixture and the Jakob number of the mixture.

$$\mathrm{Nu_{cgm}} = \frac{u \cdot D_h}{k_{cgm}} \tag{3}$$

$$\mathrm{Re_{cgm}} = \frac{\rho_{cgm} \cdot v_{mix} \cdot D_h}{\mu_{cgm}} \tag{4}$$

$$\mathrm{Ja_{cgm}} = \frac{c_{p,cgm} \cdot (T_{bulk} - T_{wall})}{h_{fg}} \tag{5}$$

The coefficients z_1 to z_4 will depend on the boundary conditions and the carrier gas. It is also common to normalize the Nusselt number for steam condensation in the presence of saturated helium by that in the presence of saturated air at a given temperature and absolute pressure (as shown in Eq. 6).

$$\Phi = \frac{\mathrm{Nu_{he}}}{\mathrm{Nu_a}} \tag{6}$$

To evaluate the Nusselt number and the heat transfer coefficient for gas side in the dehumidifier, the thermophysical properties of the carrier gas steam mixture are to be evaluated. The density, thermal conductivity and viscosity of gas mixtures at low pressures are evaluated using the following approximate expressions [13].

$$\rho_{cgm} = \frac{\sum_i \rho_i x_i}{\sum_i x_i} \tag{7}$$

$$k_{cgm} = \frac{\sum_i k_i x_i M_i^{1/3}}{\sum_i x_i M_i^{1/3}} \tag{8}$$

$$\mu_{cgm} = \frac{\sum_i \mu_i x_i M_i^{1/2}}{\sum_i x_i M_i^{1/2}} \tag{9}$$

TABLE 2. Estimated improvement in gas side dehumidification heat transfer coefficients when helium is used as the carrier gas.

T [°C]	x_{cg} [-]	k_{he}/k_a [-]	Φ [-]	u_{he}/u_a [-]
40	0.93	5.5	1.1	6.1
50	0.88	5.2	1.15	5.9
60	0.8	4.7	1.25	5.8
70	0.69	4.0	1.35	5.4

Using the appropriate correlations (internal flow with $\mathrm{Re_{cgm}} = 5000 - 10000$ and $\mathrm{Ja_{cgm}} = 0.038$) for Nusselt number [10, 12] and the property equations shown above, the ratio of heat transfer coefficients for condensation in the presence of helium and air are estimated in Table 2. From this estimate it may be seen that the heat transfer coefficient can be increased by 5-6 times. This is a major advantage of using helium as the carrier gas in HDH systems.

Arabi & Reddy [14] had investigated the natural convection heat transfer coefficient in humidification dehumidification systems for various carrier gases for certain specific geometries. They had also observed that helium has the highest heat transfer coefficient among the gases considered (2 times higher than air for the cases they considered). The increase is lower compared to our estimate because they studied natural convection systems and the estimate presented earlier in this section is for forced convection systems.

2.4 Gas side pressure drop in the dehumidifier

In this section, we estimate the gas side pressure drop for helium and air. The estimate is for a fixed flow geometry with a fixed hydraulic diameter and flow area. For these constraints, Shah [15] provides the following relationship of pressure drop on fluid properties and mass flow rate under turbulent flow conditions.

$$\Delta P_{cgm,D} \propto \frac{\mu_{cgm}^{0.2} \cdot \dot{m}_{cgm}^{1.8}}{\rho_{cgm}} \tag{10}$$

The mass flow rate of carrier gas to be used in these equations is for the case to produce a fixed amount of water in the HDH systems. This is explained in the following sections. Using the mixture property equations (Eqs. 7-9), the gas side pressure drop in the dehumidifier when using helium is found to be 1/5 to 1/8 times that when using air as a carrier gas. Along with the increase in heat transfer coefficient, the reduction in pressure drop clearly shows the potential of using helium as carrier gas. In Sec. 5, we analyse the effect of reduction in pressure drop on

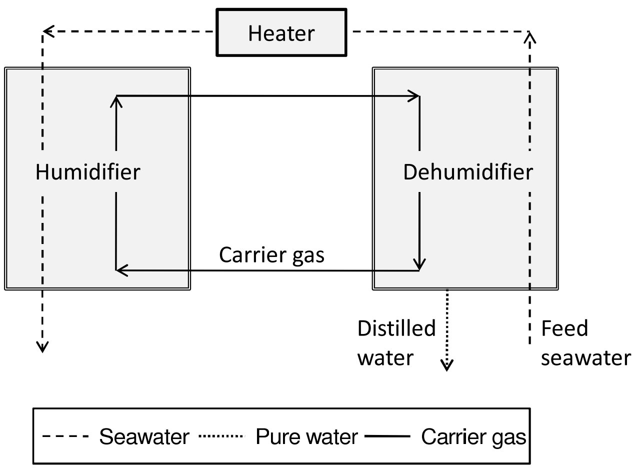

FIGURE 2. Schematic diagram of water heated HDH system.

the energy efficiency of the HDH systems.

3 Thermodynamic cycle for HDH desalination

The simplest HDH cycle is the water-heated, closed-air, open-water cycle. A schematic diagram of the cycle is shown in Fig. 2. In the humidifier, cold carrier gas is humidified in counterflow and direct contact with seawater which has been heated in a heater. In the dehumidifier, the water is condensed out of the humidified gas mixture by indirect contact and counterflow heat exchange with cold seawater, which in the process is preheated before it is sent to the heater. The dehumidified carrier gas is again sent to the humidifier, thus operating in a closed loop. This is vital to minimizing the amount of carrier gas required (especially because of the cost and regional scarcity of helium). The brine from the humidifier is disposed of.

4 Modeling details

4.1 Terminology used

In this section, the terminology used in the analysis is defined. This includes an energy-based effectiveness and a modified heat capacity rate ratio for the heat and mass exchange devices.

1. *Energy effectiveness:* An energy based effectiveness, analogous to the effectiveness defined for heat exchangers, is given as:

$$\varepsilon = \frac{\Delta\dot{H}}{\Delta\dot{H}_{\max}} \tag{11}$$

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 of the two streams. The stream with the minimum heat capacity rate dictates the thermodynamic maximum that can be attained. This concept is explained in detail in a previous publication [8].

2. *Heat capacity rate ratio*: In the limit of infinite heat transfer area for a pure heat exchanger, the entropy generation rate in the exchanger is entirely due to what is known as thermal imbalance or remanent irreversibility. This thermal imbalance is associated with conditions at which the heat capacity rate ratio is not equal to unity [16]. In other words, a heat exchanger is said to be thermally 'balanced' at a heat capacity rate ratio of one. The concept of thermodynamic balancing, even though very well known for heat exchangers, was only recently extended to HME devices [17]. It is important to establish a reliable definition for the heat capacity rate ratio for an HME. We define the heat capacity rate ratio as follows,

$$\mathrm{HCR} = \left(\frac{\Delta\dot{H}_{\max,c}}{\Delta\dot{H}_{\max,h}}\right) \tag{12}$$

The heat capacity rate ratio is essentially the ratio of maximum change in total enthalpy rate of cold to the hot streams in the heat and mass exchanger. This definition is derived by analogy to heat exchangers and the physics behind this derivation is explained in a previous publication [17].

4.2 Performance

As a first step for understanding the performance of the helium driven HDH cycles the following performance parameters are defined.

1. *Gained-Output-Ratio* (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 effected in the system.

$$\mathrm{GOR} = \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{Q}_{in}} \tag{13}$$

Latent heat is calculated at the average partial pressure of water vapor (in the moist carrier gas mixture) in the dehumidifier.

2. *Specific work* (SW): is the ratio of the auxiliary electricity consumed by blowers used for carrier gas circulation per

unit amount of water produced in the HDH system.

$$\mathrm{SW} = \frac{\dot{W}_{aux}}{\dot{m}_{pw}} \tag{14}$$

4.3 Property packages and solution technique

The solution of the governing equations was carried out using **Engineering Equation Solver (EES)** [20] which uses accurate equations of state to model the properties of moist air, helium and water. EES evaluates water properties using the IAPWS (International Association for Properties of Water and Steam) 1995 Formulation [22]. Dry air properties are evaluated using the ideal gas formulations presented by Lemmon [23] and dry helium properties are evaluated using the equations of state presented by Tillner-Roth [24]. Pure water properties are equivalent to those found in NIST's property package, REFPROP [26]. Moist air properties are evaluated assuming an ideal mixture of air and steam using the formulations presented by Hyland and Wexler [25]. The moist helium properties are also evaluated in the same manner. The humidity ratio is given by Eq. 1. Similar equations for specific enthalpy and specific entropy are:

$$h(T,P,\omega) = h_{cg}(T,p_{cg}) + \omega \cdot h_{st}(T,p_{st}) \;\; [\mathrm{kJ/kg_{cg}}] \tag{15}$$

$$s(T,P,\omega) = s_{cg}(T,p_{cg}) + \omega \cdot s_{st}(T,p_{st}) \;\; [\mathrm{kJ/kg_{cg}\cdot K}] \tag{16}$$

It is noted that in these equations the specific enthalpy and specific entropy of the components of the mixture are evaluated at the appropriate partial pressures and mixture temperatures. The entropy of mixing does not appear when it is thus calculated. The moist air properties presented using the above equations are in close agreement with the data presented in ASHRAE [27].

EES is a numerical solver, and it uses an iterative procedure to solve the equations. The convergence of the numerical solution is checked by using the following two variables: (1) 'Relative equation residual' — the difference between left-hand and right-hand sides of an equation divided by the magnitude of the left-hand side of the equation; and (2) 'Change in variables' — the change in the value of the variables within an iteration. The calculations converge if the relative equation residuals is lesser than 10^{-6} or if change in variables is less than 10^{-9}. These are standard values used to check convergence in EES. There are several publications which have previously used them for thermodynamic analysis [4, 5, 6, 7, 8, 17, 18, 28, 29].

The code written in EES was checked for correctness against various limiting cases. For example, when $\varepsilon_h = 1$, the minimum stream-to-stream terminal (at exit or inlet) temperature difference in the humidifier was identically equal to zero. Several other simple cases were checked. Also, calculations were repeated several times to check for reproducibility.

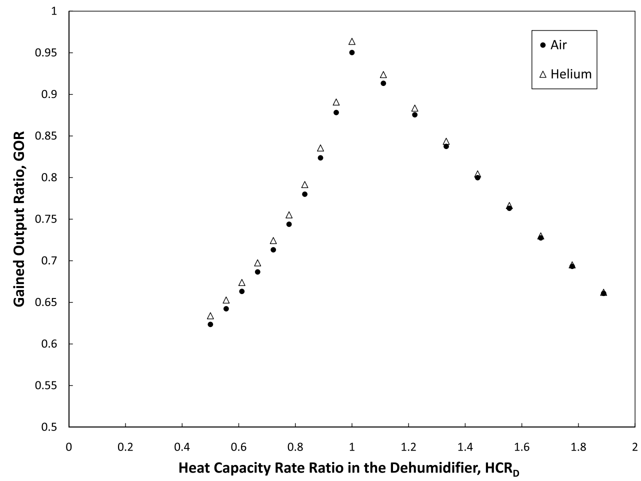

FIGURE 3. Relative performance of water heated HDH cycle with helium or air as carrier gas. $T_{sw,in} = 30°\mathrm{C}; \mathrm{T_{cg,H,out}} = 90°\mathrm{C}$; $\varepsilon_\mathrm{H} = 60\%$; $\varepsilon_\mathrm{D} = 90\%$; $P = 100$ kPa.

5 Relative thermodynamic performance of helium based cycles

The effect of changing the carrier gas from air to helium on the system performance is investigated in this section. HDH cycles are traditionally heat driven cycles run by using low grade energy to heat the seawater. The efficiency of the cycle itself is measured by the gained output ratio (GOR) defined in Eq. 13. The GOR for the water heated HDH cycle may be rewritten as follows

$$\mathrm{GOR} = \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{m}_w c_{p,w} \cdot (T_{w,ht,out} - T_{w,ht,in})} \tag{17}$$

From previous studies [4], we know that the modified heat capacity rate ratio (HCR) is the most important thermodynamic parameter for heat driven HDH cycles. Figure 3 illustrates the GOR of cycles using air and helium as the carrier gas against the modified heat capacity ratio of the dehumidifier. For this example, the component effectivenesses are fixed along with the operating pressures and feed seawater conditions. The seawater temperature at the exit of the heater is fixed at 90°C.

It is observed that the change in carrier gas has very little impact on the performance of the system. To explain this trend let us rewrite Eq. 17 as follows

TABLE 3. Various system parameters and temperatures for cases shown in Fig. 3

HCR_D	$\Delta T_{ht,he}$	$\Delta T_{ht,a}$	$\frac{m_{r,he}}{m_{r,a}}$	$\Delta T_{D,he}$	$\Delta T_{D,a}$
[-]	[°C]	[°C]	[-]	[°C]	[°C]
0.8889	31.73	31.81	7.041	20.3	20.49
1	29.56	29.65	7.049	26.59	26.8
1.333	31.87	31.97	7.074	30.66	30.94
1.667	34.26	34.35	7.084	32.67	32.98
2	36.55	36.63	7.087	33.23	33.52

$$\mathrm{GOR} = \frac{\Delta\omega \cdot h_{fg}}{\dot{m}_r c_{pw} \Delta T_{ht}} \tag{18}$$

$$= \frac{\frac{M_w}{M_{cg}} \Delta x \cdot h_{fg}}{\dot{m}_r c_{pw} \Delta T_{ht}} \tag{19}$$

where the mass flow rate ratio $m_r = \frac{\dot{m}_w}{\dot{m}_{da}}$.

So, the ratio of GOR for system with helium and air can be written as follows:

$$\frac{\mathrm{GOR_{he}}}{\mathrm{GOR_a}} = \left\{ \frac{M_a}{M_{he}} \frac{\Delta x_{D,he}}{\Delta x_{D,a}} \right\} \cdot \left\{ \frac{m_{r,he}}{m_{r,a}} \frac{\Delta T_{ht,he}}{\Delta T_{ht,a}} \right\}^{-1} \tag{20}$$

From Tab.3, it can be seen that

$$\frac{\Delta T_{ht,he}}{\Delta T_{ht,a}} \approx 1 \tag{21}$$

$$\frac{\Delta T_{D,he}}{\Delta T_{D,a}} \approx 1 \tag{22}$$

$$\frac{\Delta x_{D,he}}{\Delta x_{D,a}} \approx 1 \tag{23}$$

$$\frac{m_{r,he}}{m_{r,a}} \approx \frac{M_a}{M_{he}} \tag{24}$$

Hence,

$$\frac{\mathrm{GOR_{he}}}{\mathrm{GOR_a}} \approx 1 \tag{25}$$

Thus the GOR is approximately the same for helium and air systems.

Specific work consumption of the blower circulating the carrier gas through the system was also calculated. It was found that for a blower efficiency of 70%, a system using air as the carrier gas with a total air-side pressure drop of 1 kPa was found to need 3.5 kWh/m^3. For the same boundary conditions at a total pressure drop of 1/5 to 1/8 kPa (see section 2.4), a system with helium as the carrier gas will have an electricity consumption of 0.42 to 0.7 kWh/m^3. This is assuming all the pressure drop occurs in the dehumidifier and the humidifier is a low (zero) pressure drop device.

6 Concluding remarks

1. Large pressure drops and low heat transfer coefficients in the gas side of the dehumidifier are significant problems for HDH systems. A possible solution to these problems is to use helium as the carrier gas instead of air. Owing to its superior thermophysical and psychrometric properties, helium as a carrier gas is estimated to significantly improve the heat transfer coefficient and lower the pressure drop compared to systems using air as a carrier gas.
2. The thermodynamic performance (GOR) of the water heated HDH system is not affected by changing the carrier gas.
3. However, the reduction of pressure drop will reduce the auxiliary electricity consumption by 5 to 8 times and increase in dehumidifier heat transfer coefficient will reduce dehumidifier size greatly.
4. Hence, given the advantages of lower heat exchanger size requirement and lower electricity consumption, it is concluded that using helium as the carrier gas has promise for HDH desalination systems. The current crisis of lack of availability of helium, however, poses a challenge to the commercial realization of such a system.

ACKNOWLEDGMENTS

The authors would like to thank the King Fahd University of Petroleum and Minerals for funding the research reported in this paper through the Center for Clean Water and Clean Energy at MIT and KFUPM.

REFERENCES

[1] Bilton, A. M., Kelley, L. C., Dubowsky, S., Photovoltaic reverse osmosis - Feasibility and a pathway to develop technology, *Desalination and Water Treatment*, in press.

[2] Narayan, G.P., Sharqawy, M.H., Summers, E.K., Lienhard V, J.H., Zubair, S.M. and Antar, M.A., 2009. The potential of solar-driven humidification-dehumidification desalination for small-scale decentralized water production, *Renewable and Sustainable Energy Reviews*, **14**, No. 4, pp.1187-1201.

[3] Müller-Holst, H., 2007. Solar thermal desalination using the Multiple Effect Humidification (MEH) method in *Solar Desalination for the 21st Century*, Springer, Dordrecht.

[4] Narayan, G.P., Sharqawy, M.H., Lienhard V, J.H. and Zubair, S.M., 2010. Thermodynamic analysis of humidification dehumidification desalination cycles, *Desalination and Water Treatment*, **16**, pp 339-353.

[5] Narayan, G. P., McGovern, R. K., Zubair, S. M., and Lienhard V, J. H. 2011. Variable pressure humidification dehumidification desalination, Proceedings of *AJTEC 2011*, ASME/JSME 8th Thermal Engineering Joint conference, March 13-17 2011, Hawaii.

[6] Narayan, G.P., McGovern, R. K., Zubair, S. M., and Lienhard V, J. H. Carrier gas processes for seawater desalination, Proceedings of the IDA World Water congress 2011, Perth.

[7] Narayan, G.P., McGovern, R. K., Zubair, S. M., and Lienhard V, J. H. Steam driven humidification dehumidification desalination system, Under review.

[8] Narayan, G. P., Mistry, K. H., Sharqawy, M. H., Zubair, S. M., Lienhard V, J. H. 2010. Energy effectiveness of simultaneous heat and mass exchange devices, *Frontiers in Heat and Mass Transfer*, **1** (2), 1-13.

[9] Sievers, M., 2010. Design and optimization of a dehumidifier in a humidification-dehumidification(HDH) desalination system, Diploma Thesis, Technische Universtat Hamburg-Harburg.

[10] Hasanein, H. A., 1994. Steam condensation in the presence of noncondensable gases under forced convection conditions, Ph.D. dissertation, Massachusetts Institute of Technology.

[11] Kuhn S.Z., 1995. Investigation of heat transfer from condensing steam-gas mixture and turbulent films flowing downward inside a vertical tubes, Ph.D. dissertation, University of California at Berkeley.

[12] Maheshwari, N. K., Vijayan, P.K., and Saha, D., 2007. Effects of non-condensable gases on condensation heat transfer, 4th RCM on the IAEA CRP on Natural Circulation Phenomena, Vienna, Austria.

[13] Perry, R. H., Chilton, C.H., 1973. *Chemical engineers' handbook*, 5^{th} edition, McGraw-Hill.

[14] Abu Arabi, M. K., Reddy, K.V., 2003. Performance evaluation of desalination processes based on the humidification/dehumidification cycle with different carrier gases. *Desalination*, **156**, 281-293 .

[15] Shah, R. K., and Sekulic, D. P., 2003. Fundamentals of heat exchanger design, Wiley, NY.

[16] Bejan A., 1996. Entropy generation minimization: the method of thermodynamic optimization of finite size systems and finite time processes, CRC Press LCL, Florida.

[17] Narayan, G.P., Lienhard V, J.H., and Zubair, S.M., 2010. Entropy generation minimization of combined heat and mass exchange devices, *International Journal of Thermal Sciences*, **49** (10), pp 2057-2066.

[18] McGovern, R.K., Narayan, G.P., and Lienhard V, J. H. Second law characterisation of pressure exchanging device performance, Under review.

[19] Crisp, G.J., 2010. Actual Energy Consumption and Water Cost for the SWRO Systems at Perth Australia and Gold Coast - Case History, IDA energy conference, Huntington beach, USA.

[20] Klein, S.A. Engineering Equation Solver. Academic Professional, Version 8. URL http://www.fchart.com/ees/ees.shtml

[21] Miller, J.E., 2003. Review of Water Resources and Desalination Technologies. *Sandia National lab*, Report # SAND 2003-0800.

[22] Pruß, A. and Wagner, W., 2002. The IAPWS formulation 1995 for the thermodynamic properties of ordinary water substance for general and scientific use. *Journal of Physical and Chemical Reference Data* **31**(2), 387-535.

[23] Lemmon, E.W., Jacobsen, R.T., Penoncello, S.G., Daniel F., 2000. Thermodynamic Properties of Air and Mixtures of Nitrogen, Argon, and Oxygen from 60 to 2000 K at Pressures to 2000 MPa. *Journal of Physical and Chemical Reference Data* **29**(3), 331.

[24] Tillner-Roth, R., 1998. Fundamental Equations of State, *Shaker Verlag*, Aachen.

[25] Hyland, R. W., Wexler, A., 1983. Formulations for the Thermodynamic Properties of the Saturated Phases of H2O from 173.15 K to 473.15 K. *ASHRAE Transactions* Part 2A, Paper 2793.

[26] E. W. Lemmon, M. L. Huber, M. O. McLinden, NIST Standard Reference Database 23: Reference fluid thermodynamic and transport properties, REFPROP, Version 8.0 (2007).

[27] Wessel, D.J., 2001. ASHRAE Fundamentals Handbook 2001 (SI Edition). *American Society of Heating, Refrigerating, and Air-Conditioning Engineers*.

[28] Zmeureanu, R. and Wu, X.Y, 2007. Energy and exergy performance of residential heating systems with separate mechanical ventilation. *Energy* **32**(3), 187-195.

[29] Qureshi, B.A. and Zubair, S.M., 2006. A complete model of wet cooling towers with fouling in fills. *Applied Thermal Engineering* **26**(16), 1982-1989.