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VARIABLE PRESSURE HUMIDIFICATION DEHUMIDIFICATION DESALINATION SYSTEM

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

Nature uses air as a carrier gas to desalinate seawater through evaporation and rain. Several investigators have previously studied desalination cycles based on carrier gas processes. However, single pressure carrier gas cycles suffer from low energy recovery and hence low performance. Here we discuss a novel carrier gas cycle which operates under varied pressure. This cycle operates the evaporation process under a reduced pressure and the condensation process at an elevated pressure to enhance energy recovery. The pressure is varied by using a mechanical compressor. This cycle has been found to be several times as efficient as the existing carrier gas cycles. In this paper, the salient features of this cycle are analyzed in an on-design sense by defining a component effectiveness for the simultaneous heat and mass exchange components and an isentropic efficiency for the compressor and the expander. Based on this study, ways to improve the cycle are proposed. The possibility of using a throttle valve instead of an expander and the effect this would have on the overall performance is reported. Comparison of the new desalination cycle with existing ones is also performed in terms of specific work consumption.

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

NOMENCLATURE

Acronyms

GOR	Gained Output Ratio
HDH	Humidification Dehumidification
HE	Heat Exchanger
HME	Heat and Mass Exchanger
RO	Reverse Osmosis
MVC	Mechanical Vapor Compression distillation

Symbols

c_p	specific heat capacity at constant pressure(J/kg-K)
$\dot{H}$	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)
$\dot{Q}$	heat transfer rate (W)
$\dot{S}_{gen}$	entropy generation rate (W/K)
SNW	specific net work (kJ_e/kg)
SW	specific work consumption (kJ_e/kg)
T	temperature (°C)
VPR	vapor productivity ratio (-)
$\dot{W}$	work transfer rate (W)

Greek

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

Subscripts

a humid air
act actual
c cold stream
com mechanical compressor
D dehumidifier
da dry air
e expander
h hot stream
H humidifier
i inlet
in entering
max maximum
min minimum
net net
o outlet
out leaving
pw product water
rev reversible
sat saturated
st steam
sw seawater
v vapor
w water

Superscripts

ideal ideal condition
rev reversible

1 Introduction

An alternative to conventional desalination systems is the humidification dehumidification (HDH) desalination system 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 [1, 2].

All existing HDH systems operate at a single pressure (normally at atmospheric pressure) and consist of three subsystems: (a) an air and/or water heater; (b) a humidifier or an evaporator; and (c) a dehumidifier or a condenser. These are simple systems and are relatively easy to design and fabricate. However, using a thermodynamic analysis [3], it was demonstrated that the thermal performance of these systems is very limited (a maximum Gained Output Ratio or GOR of 4.5). This is because the single pressure HDH system has three intrinsic disadvantages from a thermal performance perspective: (1) low water vapor content in air (low humidity ratio) at atmospheric pressure; (2) extra thermal resistance to heat transfer because of the presence of the carrier gas (air) in the condenser; and (3) lower energy recovery compared to MSF and MED systems. The third point is especially important because, unlike MSF and MED systems, multi-staging the HDH system does not yield any increase in performance [3]. In this manuscript, simple means to address the aforementioned demerits of the HDH system using the tools of classical thermodynamics are described.

1.1 Effect of operating pressure on the humidity ratio of moist air

All previous HDH systems in literature have been designed to operate at atmospheric pressure. However, to increase the vapor content of moist air the systems need to be operated at sub-atmospheric pressures. Figure 1 illustrates this concept in a psychrometric chart. 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.

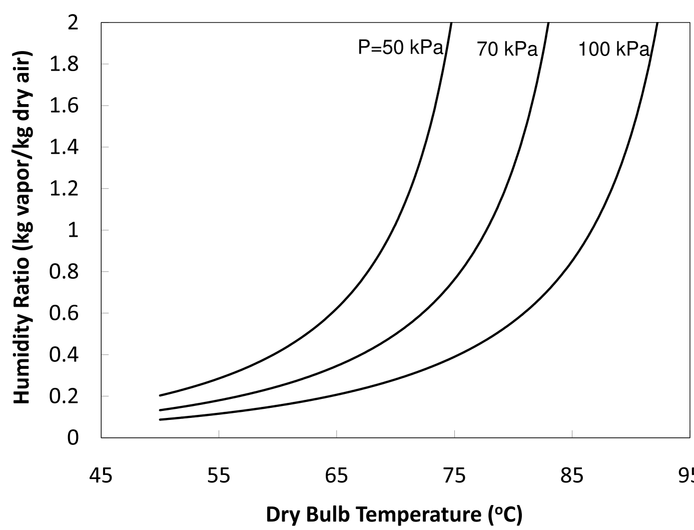

FIGURE 1. Effect of pressure on humidity ratio of saturated moist air.

However, if the entire HDH system is operated under this reduced pressure, the increase in thermal performance is relatively low. This is because: (1) the energy recovery is limited (for the same reasons as for the atmospheric pressure systems); and (2) the humidity ratio at the dehumidifier exit is also increased, limiting the water productivity [3].

1.2 Variable pressure HDH cycle

In this paper, a new HDH cycle to improve the energy efficiency of HDH is described. The proposed cycle operates the humidifier and dehumidifier at different pressures. As shown in Fig. 2, 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 a work transfer. The expanded carrier gas is then 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 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.

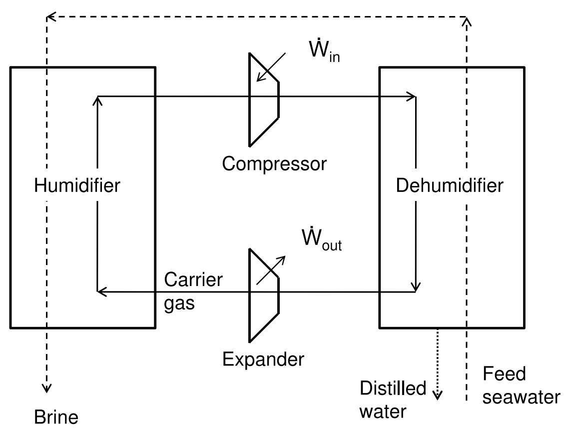

FIGURE 2. Schematic diagram of mechanical compression driven HDH system

Figure 3 illustrates an example of the cycle on a psychrometric chart. 1-2 is the air humidification process that is approximated to following the saturation line. 2-3 is the compression process in which the humidified air is compressed to a higher pressure and temperature. 3-4 is the dehumidification process. The state 4 is assumed to be saturated in this example. 4-1 is the air expansion process where some of the energy input in the compressor is recovered.

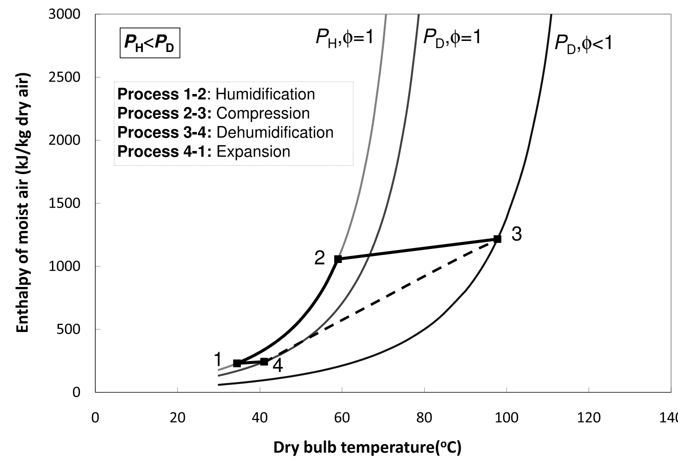

FIGURE 3. Mechanical compression driven HDH cycle represented in psychrometric coordinates.

2 Terminology used

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

2.1 Energy effectiveness

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

$$\varepsilon = \frac{\Delta\dot{H}}{\Delta\dot{H}_{\max}} \tag{1}$$

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 [4].

2.2 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 [5]. In other words, a heat exchanger is said to be thermally 'balanced' at a heat capacity rate ratio of one. This concept of thermodynamic balancing, even though very well known for heat exchangers, was only recently extended to HME devices [6]. It is important to establish a reliable definition for the heat capacity rate ratio for an HME in order to understand its influence on selecting the appropriate definition of effectiveness.

We define the heat capacity rate ratio as follows,

$$\mathrm{HCR} = \left(\frac{\Delta \dot{H}_{\max,c}}{\Delta \dot{H}_{\max,h}} \right) \tag{2}$$

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 [6].

2.3 Isentropic efficiency

The performance of the compressor and expander are defined by an isentropic efficiency. For a mechanical compressor, the isentropic efficiency is defined as the ratio of the reversible to actual work input.

$$\eta_{com} = \frac{\dot{W}_{rev}}{\dot{W}} \tag{3}$$

For an expander, the isentropic efficiency is defined as the ratio of the actual to reversible work output.

$$\eta_e = \frac{\dot{W}}{\dot{W}_{rev}} \tag{4}$$

2.4 System and performance parameters

As a first step for understanding the improved performance of the new HDH cycles the following system and performance parameters are defined.

1. Specific work consumption (SW): is the amount of electrical energy (in kJ$_e$) consumed to produce one kg of fresh water. This parameter is used commonly for defining the performance of work driven desalination systems.

$$\mathrm{SW} = \frac{\dot{W}_{in} - \dot{W}_{out}}{\dot{m}_{pw}} \tag{5}$$

The specific work consumption can be rewritten as follows

$$\begin{aligned} \mathrm{SW} &= \frac{\dot{W}_{in} - \dot{W}_{out}}{\dot{m}_{pw}} \\ &= \underbrace{\left\{ \frac{\dot{W}_{in} - \dot{W}_{out}}{\dot{m}_{da} \cdot \omega_{H,o}} \right\}}_{\mathrm{SNW}} \cdot \underbrace{\left\{ \frac{\dot{m}_{da} \cdot \omega_{H,o}}{\dot{m}_{pw}} \right\}}_{1/\mathrm{VPR}} \end{aligned} \tag{6}$$

Thus, SW is a function of two new system parameters - vapor productivity ratio (VPR) and specific net work (SNW).

2. Vapour productivity ratio (VPR): is defined as the ratio of the rate at which water is produced by the system to the rate at which water vapor is compressed in the system.

$$\mathrm{VPR} = \frac{\dot{m}_{pw}}{\dot{m}_{da} \cdot \omega_{H,o}} \tag{7}$$

VPR is a measure of how effective the humidifier and dehumidifier are at producing water given a fixed compression ratio, and expander and compressor efficiency. The value of VPR will always be less than 1, as water cannot be produced at a rate greater than that at which it flows into the dehumidifier. For example if the vapor productivity ratio is 0.25, this means for every four units of vapor that are compressed in the system, only one unit of water is produced. Evidently, VPR should be maximised to avoid water vapor from being needlessly compressed.

3. Specific net work (SNW) : is the net work input to the system per unit amount of vapor compressed.

$$\mathrm{SNW} = \frac{\dot{W}_{in} - \dot{W}_{out}}{\dot{m}_{da} \cdot \omega_{H,o}} \tag{8}$$

In the mechanical compression driven HDH system, compression of the carrier gas is an energetic loss which is only partially recovered as work in the expander and as heat in the dehumidifer. SNW is indicative of the work imparted to the useful component of the fluid mixture circulating in the system.

4. 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.

$$\mathrm{GOR} = \frac{\dot{m}_{pw} \cdot h_{fg}}{\dot{Q}_{in}} \tag{9}$$

Latent heat is calculated with the operating pressure assumed to be saturation pressure. GOR will be used to compare the new cycle to the existing HDH cycles.

3 Equations and modeling details

This section discusses the conservation equations for each of the four devices. Additionally, the fluid property packages and models used to solve the defined equations are described.

3.1 Humidifier

Consider a counterflow humidifier in which one fluid stream is pure water and the other stream is a mixture of air and water vapor. Mass balance dictates that the mass flow rate of dry air in the humidifier is constant:

$$\dot{m}_{da} = \dot{m}_{da,i} = \dot{m}_{da,o} \tag{10}$$

A mass balance on the water gives the mass flow rate of the water leaving the humidifier in the water stream:

$$\dot{m}_{w,o} = \dot{m}_{w,i} - \dot{m}_{da}\left(\omega_{a,o} - \omega_{a,i}\right) \tag{11}$$

Based on Eq. 1, the energy effectiveness, ε, may be written in terms of mass flow rates, temperatures, and humidity ratios [4].

Case I, $\Delta\dot{H}_{\max,w} < \Delta\dot{H}_{\max,a}$:

$$\varepsilon = \frac{\dot{m}_{w,i}h_{w,i} - \dot{m}_{w,o}h_{w,o}}{\dot{m}_{w,i}h_{w,i} - \dot{m}_{w,o}h_{w,o}^{ideal}} \tag{12}$$

Case II, $\Delta\dot{H}_{\max,w} > \Delta\dot{H}_{\max,a}$:

$$\varepsilon = \frac{\dot{m}_{da}(h_{a,o} - h_{a,i})}{\dot{m}_{da}(h_{a,o}^{ideal} - h_{a,i})} \tag{13}$$

Note that the First Law for the humidifier gives,

$$0 = \underbrace{\dot{m}_{da}\left(h_{a,i} - h_{a,o}\right)}_{\Delta\dot{H}_a} + \underbrace{\dot{m}_w h_{w,i} - \dot{m}_{w,o}h_{w,o}}_{\Delta\dot{H}_w} \tag{14}$$

where $\Delta\dot{H}_w$ is the change in total enthalpy rate for the feed water stream and $\Delta\dot{H}_a$ is the change in total enthalpy rate of the moist air stream.

3.2 Dehumidifier

Now consider a counterflow dehumidifier in which one fluid stream is pure water and the other stream is a mixture of air and water vapor. The air-vapor mixture is transferring heat to the water stream. In this process, some of the water vapor in the mixture condenses out and forms a separate condensate stream. Since all the dry air in the air stream and the water in the other fluid stream that enters the dehumidifier also leaves the device, the mass flow rate of dry air and mass flow rate of the water is constant.

$$\dot{m}_{da} = \dot{m}_{da,i} = \dot{m}_{da,o} \tag{15}$$

$$\dot{m}_{w,o} = \dot{m}_{w,i} \tag{16}$$

The mass flow rate of the condensed water can be calculated by using a simple mass balance:

$$\dot{m}_{pw} = \dot{m}_{da}\left(\omega_{a,i} - \omega_{a,o}\right) \tag{17}$$

The effectiveness definition of the dehumidifier is as follows:

Case I, $\Delta\dot{H}_{\max,w} < \Delta\dot{H}_{\max,a}$:

$$\varepsilon = \frac{h_{w,i} - h_{w,o}}{h_{w,i} - h_{w,o}^{ideal}} \tag{18}$$

Case II, $\Delta\dot{H}_{\max,w} > \Delta\dot{H}_{\max,a}$:

$$\varepsilon = \frac{\dot{m}_{da}(h_{a,o} - h_{a,i}) + \dot{m}_{pw}h_{pw}}{\dot{m}_{da}(h_{a,o}^{ideal} - h_{a,i}) + \dot{m}_{pw}^{ideal}h_{pw}^{ideal}} \tag{19}$$

Note that the First Law for the dehumidifier can be expressed as,

$$0 = \underbrace{\dot{m}_{da}\left(h_{a,i} - h_{a,o}\right) - \dot{m}_{pw}h_{pw}}_{\Delta\dot{H}_a} + \underbrace{\dot{m}_w\left(h_{w,i} - h_{w,o}\right)}_{\Delta\dot{H}_w} \tag{20}$$

where $\Delta\dot{H}_w$ is the change in total enthalpy rate for the feed water stream and $\Delta\dot{H}_a$ is the change in total enthalpy rate of the moist air stream.

3.3 Compressor

Consider a mechanical compressor which provides the driving pressure difference to the moist air stream by means of a work transfer ($\dot{W}_{in}$). The First Law for the compressor can be expressed as

$$\dot{W}_{in} = \dot{m}_{da}\left(h_{a,o} - h_{a,i}\right) \tag{21}$$

The isentropic efficiency for the compressor can be defined as:

$$\eta_{com} = \frac{h_{a,o}^{rev} - h_{a,i}}{h_{a,o} - h_{a,i}} \tag{22}$$

where the exit state from the compressor (which is at the dehumidifier inlet pressure) is calculated using the Second Law for the reversible case.

$$s_{a,o}^{rev} = s_{a,i} \tag{23}$$

3.4 Expander

The First Law for the expander can be expressed as

$$\dot{W}_{out} = \dot{m}_{da}\left(h_{a,i} - h_{a,o}\right) - \left(\dot{m}_w \cdot h_w\right)_{condensate} \tag{24}$$

The isentropic efficiency for the expander can be defined as:

$$\eta_e = \frac{\dot{m}_{da}\left(h_{a,i} - h_{a,0}\right) - \left(\dot{m}_w \cdot h_w\right)_{condensed}}{\dot{m}_{da}\left(h_{a,i} - h_{a,0}^{rev}\right) - \left(\dot{m}_w \cdot h_w\right)_{condensate}^{rev}} \tag{25}$$

where the exit state from the expander (which is at the humidifier inlet pressure) in the reversible case is calculated using the Second Law.

$$\dot{m}_{da}\left(s_{a,i} - s_{a,o}^{rev}\right) - \left(\dot{m}_w \cdot s_w\right)_{condensate}^{rev} = 0 \tag{26}$$

3.5 Solution technique

The solution of the governing equations was carried out using **Engineering Equation Solver (EES)** [9] which uses accurate equations of state to model the properties of moist air and water. EES evaluates water properties using the IAPWS (International Association for Properties of Water and Steam) 1995 Formulation [10]. Dry air properties are evaluated using the ideal gas formulations presented by Lemmon [11]. Moist air properties are evaluated assuming an ideal mixture of air and steam using the formulations presented by Hyland and Wexler [12]. Moist air properties from EES are in close agreement with the data presented in ASHRAE Fundamentals [13] and pure water properties are equivalent to those found in NIST's property package, REFPROP [14].

It was previously shown that the use of pure water properties instead of seawater properties does not significantly affect the performance of the HDH cycle at optimized mass flow rate ratios [15]. In the current manuscript this is especially true since all the data is plotted at optimized mass flow rate ratio (as described in the following section).

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' — 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 [16, 17].

The code written in EES was checked for correctness against various limiting cases. For example, when $\varepsilon_h = \varepsilon_d = 0$ the GOR was found to 0 for all values of top and bottom temperatures. When $\varepsilon_h = 1$, the minimum stream-to-stream terminal (at exit or inlet) temperature difference in the humidifier was identically equal to zero for all values of top and bottom temperatures. Several other simple cases where checked. Also, calculations were repeated several times to check for reproducibility.

4 Results and discussions

4.1 Parametric study

This section investigates the importance of various parameters on the overall performance of the variable pressure cycle driven by a mechanical compressor. Understanding the effect of these parameters is necessary to optimize the design of the cycle. The parameters studied include the mass flow rate of the air and water streams, the expander and compressor efficiencies, the humidifier and dehumidifier effectivenesses, the operating humidifier pressure, the air side pressure drops in the dehumidifier and humidifier, and the pressure ratio provided by the compressor.

Optimum Second Law performance. We have previously shown [3] that the performance of the HDH cycle depends on the mass flow rate ratio (ratio of mass flow rate of seawater at the inlet of the humidifier to the mass flow rate of dry air through the humidifier), rather than on individual mass flow rates. Moreover, we have also shown that there is an optimum performance at fixed input conditions and this occurs at a modified heat capacity rate ratio of unity (HCR=1) for either the humidifier or the dehumidifier [3, 6]. For mechanical compression driven HDH, the Second Law optimum occurs at a balanced condition for the humidifier. An example of this result is shown in Fig. 4.

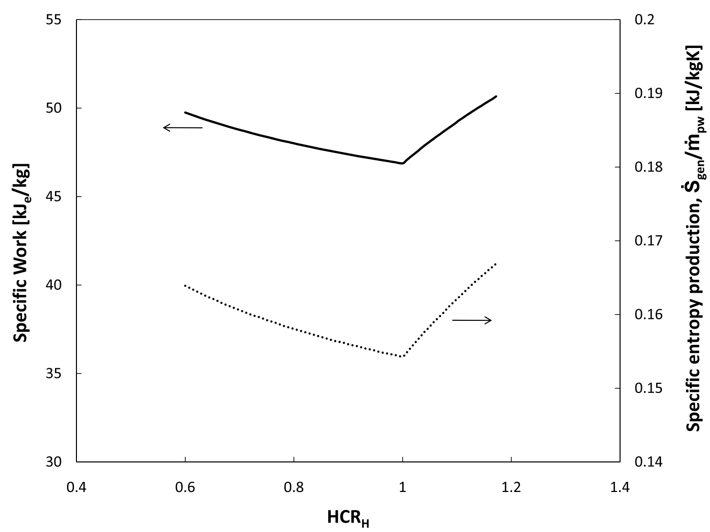

FIGURE 4. Effect of modified heat capacity ratio of humidifier on specific work and specific entropy generation. $T_{sw,in} = 30°C$; $\varepsilon_H = \varepsilon_D = 80\%$; $\eta_{com} = \eta_e = 100\%$; $P_H = 40$ kPa; $P_D = 48$ kPa.

Hence, in this and all the subsequent sections only the optimum performance values are reported.

Effect of component performance $(\eta_{com}, \eta_e, \varepsilon_H, \varepsilon_D)$**.** Figure 5 illustrates the variation in performance of the cycle at various values of isentropic efficiencies and HMX effectivenesses. In this figure, one of the effectivenesses or efficiencies is varied at a time while the others are fixed. The dehumidifier and humidifier effectiveness is fixed at 80% and the isentropic efficiencies are fixed at 100% except in the cases in which they are varied. The air side and water side pressure drop is assumed to be zero in both the humidifier and the dehumidifer, and seawater is assumed to enter the system at 30°C. The pressure ratio was fixed at 1.2.

It is observed that while a higher efficiency compressor and expander are vital for a low specific work consumption, the compressor efficiency is of greater relative importance. This general trend has also been observed for various other boundary conditions. It is important to note that, relatively, the performance of the cycle is less sensitive to the humidifier and dehumidifier performance.

Effect of pressure ratio (P_D/P_H) **and dehumidifier pressure** (P_d)**.** Figure 6 illustrates the effect of pressure ratio and humidifier pressure on cycle performance. Firstly, at a lower pressure ratio, the specific work is lower (indicating a higher system performance). The lower limit on pressure ratio required in the compressor is imposed by the dehumidifier minimum terminal temperature difference. For the present simulations the pressure ratio was varied from 1.2 to 2.4.

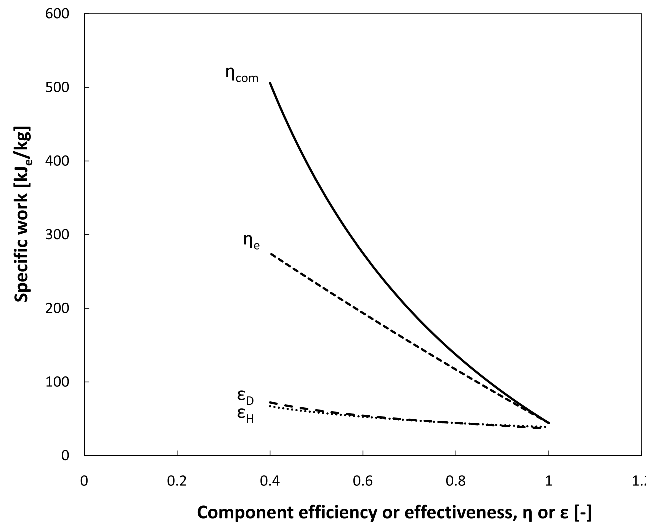

FIGURE 5. Effect of component effciency or effectiveness on cycle performance for $T_{sw,in} = 30°C$; $P_H = 33.33$ kPa; $P_D = 40$ kPa.

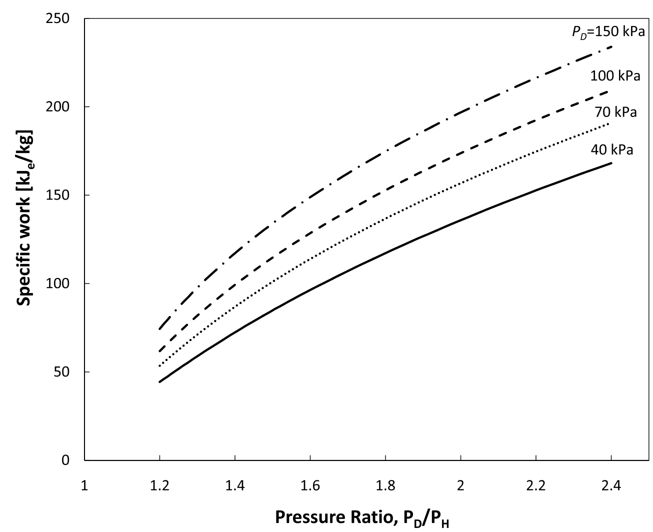

FIGURE 6. Effect of pressure ratio and dehumidifier pressure on cycle performance for $T_{sw,in} = 30°C$; $\varepsilon_H = \varepsilon_D = 80\%$; $\eta_{com} = \eta_e = 100\%$.

The reason for lower SW at lower pressure ratios can be explained using Fig. 7. At lower design pressure ratios, the vapor productivity ratio is lower. As already explained in Section 2.4, this is an expected trend. SNW increases with increasing pressure ratio and the slope with which the SNW increases is much greater than that for the increase in VPR. Specific work is the ratio of SNW to VPR (See Eqn. 6); and hence, at lower pressure ratios, we get a higher performance. In Fig. 6 it can also be

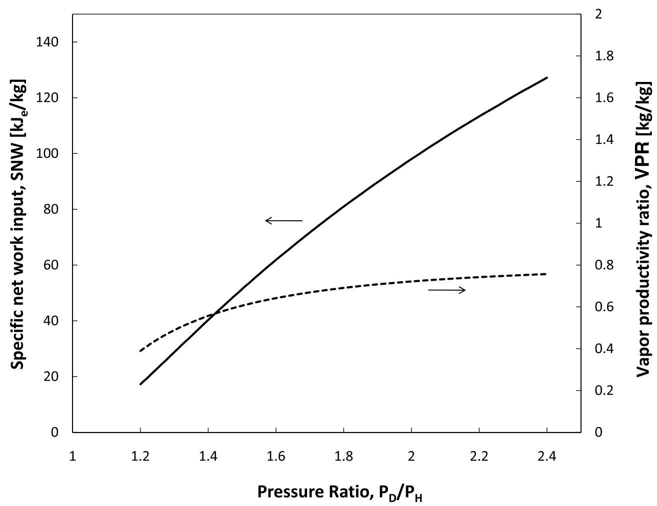

FIGURE 7. The effect of pressure ratio on specific net work and vapor productivity ratio to explain the trend in Fig. 6.

observed that a lower dehumidifier pressure gives a lower specific work. This is explained using the variation of SNW and VPR with dehumidifier pressure as shown in Fig. 8. Both SNW and VPR increase with increase in design dehumidifier pressure. VPR increases slowly compared to SNW and hence the specific work consumption decreases with lower dehumidifier pressures.

Effect of air side pressure drop $(\Delta P_H, \Delta P_D)$**.** The air side pressure drop can be substantial in heat and mass exchange (HME) devices if the design is not performed to optimize it. Figures 9 and 10 illustrate the effect of pressure drop of the air stream through the HME devices on the overall performance of the system. As the pressure drop increases, the specific work consumption increases rather drastically. Hence, it is vital to design the HME devices such that the pressure drop is minimal.

At higher values of pressure drop there is an optimum pressure ratio at which the specific work is minimum. The pressure drop in the dehumidifier and humidifier increase the specific work by a similar amount.

4.2 Selection of expansion device.

This section investigates the use of a throttle in place of a mechanical expansion device in the variable pressure system,

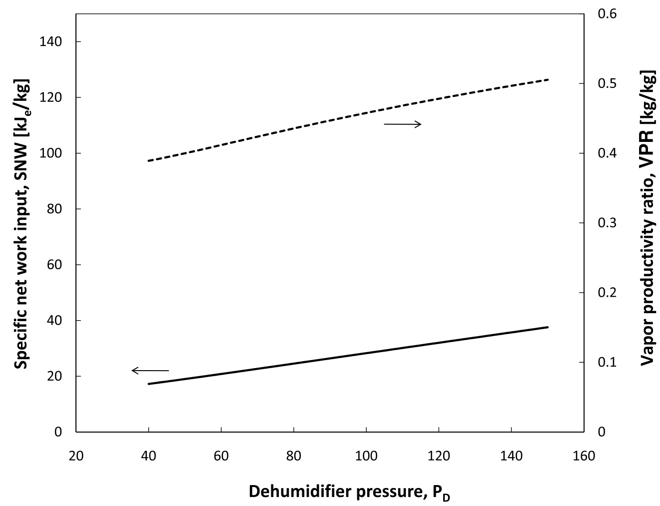

FIGURE 8. The effect of dehumidifier pressure on specific net work and vapor productivity ratio to explain the trend in Fig. 6.

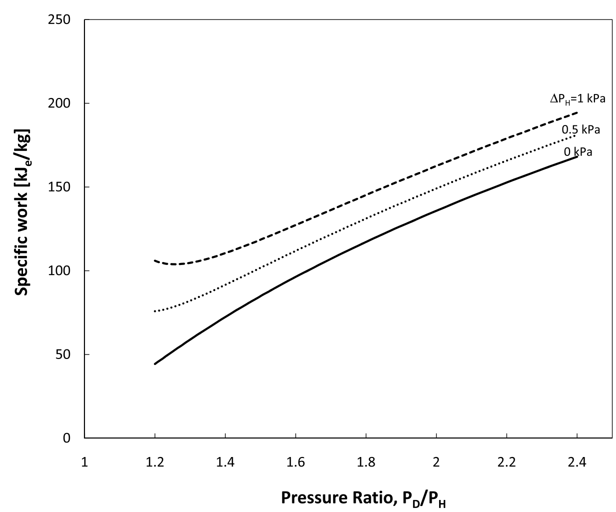

FIGURE 9. Effect of air-side pressure drop in the humidifier on cycle performance for $T_{sw,in} = 30°\text{C}$; $\varepsilon_{\text{H}} = \varepsilon_{\text{D}} = 80\%$; $\eta_{\text{com}} = \eta_{\text{e}} = 100\%$; $P_D = 40$ kPa.

downstream of the dehumidifier. Here, the throttle is modeled as an isenthalpic device. Figure 11 illustrates the performance loss because of using a throttle. It is clearly observed that, when using a throttle, the cycle has very high specific work consumption.

The reason for the low performance is shown in Fig. 12. This figure illustrates the entropy generation in each of the devices for certain boundary conditions. It can be immediately observed that for the cycle with the throttle, the entropy generation is very high because the process in the throttle is highly irreversible. We have previously proved [18] that the performance is inversely propor-

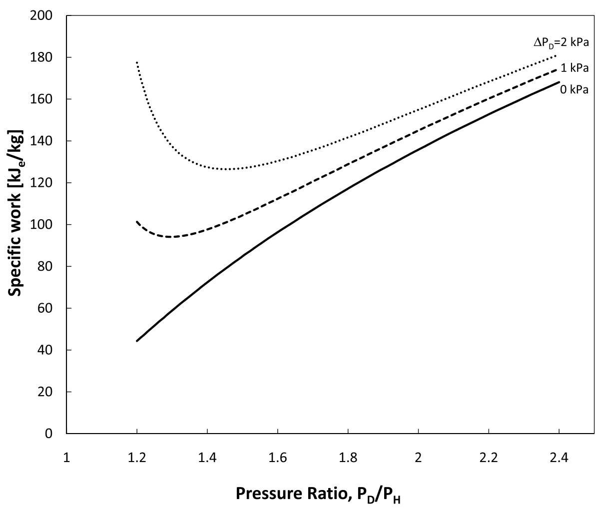

FIGURE 10. Effect of air-side pressure drop in dehumidifier on cycle performance for $T_{sw,in} = 30°\text{C}$; $\varepsilon_\text{H} = \varepsilon_\text{D} = 80\%$; $\eta_\text{com} = \eta_\text{e} = 100\%$; $P_{D,i} = 40$ kPa.

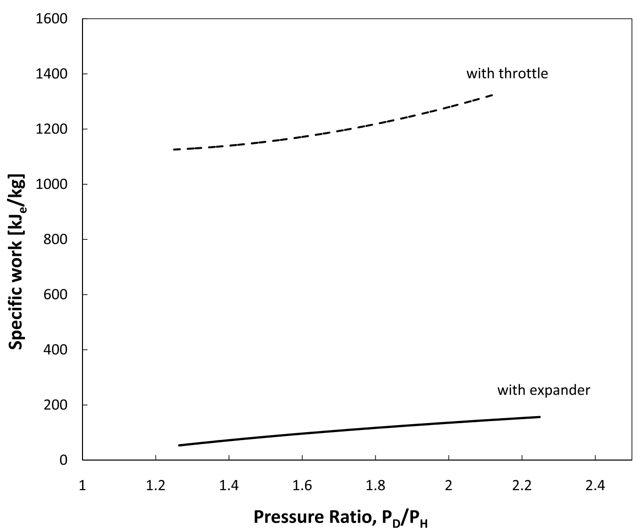

FIGURE 11. Effect of using a throttle versus using an air expander in the two pressure cycle for $T_{sw,in} = 30°\text{C}$; $\varepsilon_\text{H} = \varepsilon_\text{D} = 80\%$; $\eta_\text{com} = 100\%$; $\eta_\text{e} = 0$ or 100%; $P_D = 40$ kPa.

tional to total entropy generated in the system. Hence, the irreversiblity in the throttling process causes the system performance to drop significantly.

5 Comparison with other HDH cycles

In Table 1, the mechanical compression driven HDH systems are compared against exisiting designs including air heated and water heated HDH systems. A power production efficiency (η_{PP}) of 40% is used to convert the work consumed to heat and

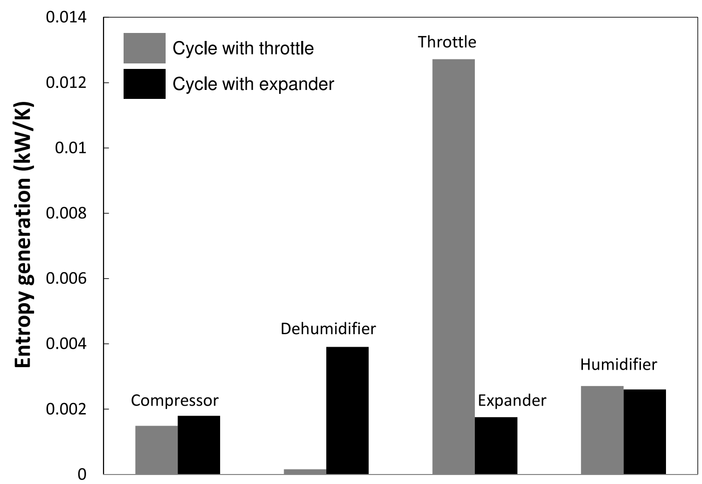

FIGURE 12. Entropy generation in the throttle and the air expander cycles for $T_{sw,in} = 30°\text{C}$; $\varepsilon_\text{H} = \varepsilon_\text{D} = \eta_\text{com} = \eta_\text{e} = 90\%$; $P_H = 40$ kPa; $P_D = 50$ kPa.

the comparison is done based on GOR.

$$\text{GOR} = \frac{\dot{m}_{pw} \cdot h_{fg} \cdot \eta_{PP}}{\left(\dot{W}_{in} - \dot{W}_{out}\right)} \tag{27}$$

TABLE 1. Comparison of mechanical compression HDH with other HDH desalination technologies

Technologies	GOR
Water heated HDH	2.5
Air heated HDH	3.5
Mechanical compression driven HDH	6

These values were calculated for a minimum terminal temperature difference of 5K in dehumidifier and 3K in humidifier. It is observed that the new cycle has a much higher energy efficiency than existing HDH systems.

6 Comparison with other work driven technologies

In Table 2, the mechanical compression driven HDH systems are compared to existing work driven seawater desalination technologies on a specific work consumption basis.

It is observed that the new cycle has a much higher energy consumption than RO and MVC. Further modifications need to

TABLE 2. Comparison of mechanical compression HDH with other work driven small scale desalination technologies

Technologies	**Energy consumed (kJ$_e$/kg)**
Reverse Osmosis (RO) with energy recovery	11–18
Mechanical vapor compression (MVC)	25–50
Mechanical compression HDH	200–260

be made to the cycle to make the performance comparable with MVC and RO. It is vital to identify compressors which can operate at a relatively high efficiency and at a low pressure ratio ($\leq$1.1). As we go to such a pressure ratio the performance of the system improves drastically.

7 Concluding remarks

1. A novel desalination cycle based on a variable pressure humidification dehumidification concept has been described in this manuscript. Various features of this cycle have been discussed in detail.
2. A parametric study explaining the influence of various system and component variables on system performance is described. It has been found 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.
3. 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 irreversiblity in the throttling process.
4. The mechanical compression driven HDH cycle has much higher performance compared to existing HDH cycles.
5. It is less efficient than RO and MVC for seawater desalination. More research needs to be done to bring the energy consumption of this cycle down further to the levels of RO and MVC.

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