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Performance Limits of Zero and Single Extraction Humidification Dehumidification Desalination Systems

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Abstract

Given simultaneous heat and mass transfer and a multiplicity of possible temperature and flow configurations, the optimisation of humidification-dehumidification desalination systems is complex. In literature, this optimisation has been tackled by considering moist air to follow the saturation curve in the humidifier and dehumidifier of a closed air water heated cycle. Under similar conditions and the same pinch point temperature differences, energy recovery was shown to improve with an increasing number of stages. In the present work, the limits upon the energy recovery and the water recovery (product water per unit of feed) of closed air water heated cycles are investigated. This is done by considering heat and mass exchangers to be sufficiently large to provide zero pinch point temperature and concentration differences with the humidifier and dehumidifier. For cycles operating with a feed temperature of 25\textdegree{}C and a top air temperature of 70\textdegree{}C, GOR is limited to approximately 3.5 without extractions (i.e. single stage system) and 14 with a single extraction (i.e. dual stage system) while RR is limited to approximately 7\% without extractions and 11\% with a single extraction. GOR increases and RR decreases as the temperature range of the cycle decreases, i.e. as the feed temperature increases or the top air temperature decreases. A single extraction is shown to be useful only when heat and mass exchangers are large in size. In addition, the effects of salinity and the validity of ideal gas assumptions upon the modelling of HDH systems are discussed.

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1. Motivation

Humidification-dehumidification (HDH) systems are commonly viewed as robust liquid purification systems, driven by low-grade heat and capable of operating on a small scale decentralized basis, [1, 2]. In desalination applications, HDH systems purify saline feed water. From another perspective, the role of HDH systems could be to concentrate industrial waste water streams.

The premise underlying HDH is that a volatile solvent containing a non-volatile solute, may be separated by humidifying a suitable carrier gas with that solvent and subsequently condensing it in pure form. Dew-vaporation systems [3] achieve this by transferring heat from the dehumidifier to the humidifier via a common heat transfer wall. HDH systems, by contrast thermally separate the humidification and dehumidification processes, as shown in Fig. 1.

The system described in Fig. 1 is a closed air water heated HDH process. Air enters the humidifier and is heated and humidified by a counterflow of feed. The moist air then flows to the dehumidifier, in which the air is cooled and water vapor is condensed due to heat transfer to a counterflow of liquid feed. This feed is heated in the dehumidifier and then in the heater, before being cooled and partially evaporated in the humidifier. The feed remaining at the bottom of the humidifier, known as the reject, leaves the system. Additionally, liquid (or air) may be extracted at intermediate points in the humidifier and injected into the dehumidifier. Figure 1 provides an example of a single liquid extraction, resulting in the formation of two humidifier and dehumidifier ‘stages’, top and bottom, each with a different liquid-to-air mass flow ratio.

Crucial to the energy efficiency of carrier gas based liquid purification processes is the recovery of heat during air cooling and dehumidification for use during air humidification and heating. The optimisation of an HDH system for energy recovery is not a simple task. Many parameters are involved, including top and bottom liquid and air temperatures and the liquid-to-air mass flow ratios throughout the system. In addition, humidifier and dehumidifier analysis is challenging, given the presence of simultaneous heat and mass transfer.
From the literature, it is evident that some progress has been made in improving energy recovery. Understanding the importance of the enthalpy-temperature characteristic of saturated moist air [4] has been central to this progress. As will be seen in Sect. 5, the mismatch in the capacity rates of the exponential moist air and the linear liquid operating lines result in the imposition of temperature and concentration differences between the streams. To reduce this mismatch, Younis et al. [5] formulated a mathematical model, using an NTU based method, of a closed air water heated cycle that allowed for bleed offs for air from the humidifier to the dehumidifier. The mass flow rate ratio of water to air in each section of the humidifier and dehumidifier was obtained by matching the slope of the operating lines of the moist air and brine on an enthalpy-temperature diagram. Later, this concept was modified by Brendel to allow for water extractions from the humidifier to the dehumidifier rather than air bleed-offs [6, 7].

Whilst the circulation of air in the systems considered by Younis et al. and Brendel were driven by forced convection, Müller-Holst et al. [8] developed a natural convection system. This system was designed such that an air flow pattern would establish itself in a manner that allowed for the matching of the capacity flow rates of moist air and liquid throughout the system. Thermal energy consumption of approximately 120 kWh/m$^3$ was reported for a system operating with feed and top brine temperatures of 25° and 85° respectively [4]. In further work, Müller-Holst [9] suggested that the mass flow rate ratio of moist air to liquid should be adjusted to achieve a constant temperature difference between the streams.

Narayan et al. [10] employed an enthalpy rate (product of enthalpy and mass flow rate) based effectiveness to quantify humidifier and dehumidifier performance in a parametric study of HDH systems. For fixed component effectivities and cycle temperature ranges it was shown that, by selecting the liquid-to-air mass flow ratio to provide a heat capacity rate ratio of unity in the dehumidifier, the entropy generation per unit water produced in the system was minimized and the GOR was maximised. Building upon these effectiveness methods, an enthalpy pinch based approach was proposed for the design of closed air water heated HDH systems with mass extractions and injections [11].

Rather than using a NTU, an effectiveness method or considering a humidifier and dehumidifier of fixed size, Hou et al. [12] employed pinch point temperature differences. For fixed pinch temperature differences, Hou demonstrated the existence of optimal values for the liquid-to-air mass flow
ratio of a closed air water heated cycle that maximise energy recovery. Later, the analysis was extended [13] to systems with two stages (one extraction). For the same pinch point temperature difference of 1°C, Hou demonstrated improved energy recovery with two stages. Zamen et al. [14] further built upon this analysis and demonstrated that there are diminishing returns in energy recovery for an increasing number of stages when the pinch point temperature difference is fixed.

In the present work, the limits upon the energy efficiency and the percentage of water recovered from the feed in a closed air water heated HDH system are investigated. Heat and mass exchangers are considered that are sufficiently large to provide zero pinch point temperature and concentration differences. Systems with zero extractions and with a single extraction are analysed over a range of feed and top air temperatures. Furthermore, employing a pinch point temperature difference to parametrise the size of heat and mass exchangers, the extent of benefits achieved in employing an extraction/injection are related to system size.

2. Performance Metrics

Before proceeding with our discussion of approximations, we define key performance parameters for the HDH process. The first parameter of interest concerns the energetic performance of the system, measured by considering the mass (or moles) of water produced per unit of heat input to the cycle. Since this quantity is dimensional, it is common to multiply by the latent heat of vaporisation to obtain the Gained Output Ratio, or the GOR, a dimensionless quantity.

\[
\text{GOR} = \frac{\dot{m}_p h_{fg}}{\dot{Q}} = \frac{\dot{N}_p \bar{h}_{fg}}{\bar{Q}}
\]

The GOR is a measure of the latent heat of water produced per unit of heat input. There is no consensus in the literature upon the temperature at which the latent heat of vaporization, \( h_{fg} \), should be computed. However, the current authors would suggest that it be evaluated at the ambient air temperature. We can consider a system that evaporates water at ambient air temperature (by operating at reduced pressure) and that subsequently condenses the water vapor without recovery of the heat of condensation. If GOR is defined such that \( h_{fg} \) is evaluated at ambient temperature, this system would have a GOR of 1. Thus the evaluation of the GOR of another system
would constitute benchmarking its heat requirements against this reference system.

Many HDH systems operate with a constant mass (or molar) flow rate of carrier gas. It is therefore useful to define all quantities on a carrier gas basis, *i.e.* per unit mass of carrier gas or per mole of carrier gas. Employing an asterisk to identify quantities measured on a carrier gas basis, the following expression for GOR is obtained:

\[
\text{GOR} = \frac{h_{fg}\Delta \omega}{\dot{Q}^*} = \frac{h_{fg}\Delta \chi}{\dot{Q}^*} 
\]

(2)

Then for a water heated system we have:

\[
\text{GOR} = \frac{h_{fg}\Delta \omega}{\Delta h_f^*} = \frac{\dot{h}_{fg}\Delta \chi}{\Delta h_f^*} 
\]

(3)

The second measure of interest concerns the fraction of feed water that the system is capable of recovering as product. The recovery ratio captures this fraction and is defined as the ratio of product water, \( \dot{m}_p \), to feed water, \( \dot{m}_{t,H_2O} \) either on a mass or molar basis:

\[
\text{RR} = \frac{\dot{m}_p}{\dot{m}_{t,H_2O}} = \frac{\dot{N}_p}{\dot{N}_{t,H_2O}} 
\]

(4)

3. Mass Balances, Enthalpy Balances and Saturation Curves

Saturation curves are extremely useful in visualising mass flows and energy flows within HDH. A plot of humidity ratio versus temperature allows us to visualise mass transfer, whilst a plot of enthalpy versus temperature illustrates energy transfer. These two plots share a common trait, in that their ordinates (humidity ratio and specific enthalpy) are both defined on a dry air basis. This is possible since the mass flow rate of air through the system is constant. In this section, we describe how the conservation of mass and energy within humidifiers and dehumidifiers can be graphically represented. We also describe the influence of salinity upon the saturation curves.

3.1. Mass Balance and the Saturation Curve

In Fig. 2, we consider a general control volume for humidification or dehumidification. A stream of liquid and a stream of moist air enter and exit
the control volume. Individual control volumes are also indicated for the gas mixture and the liquid. A portion of each individual control volume passes just inside the liquid on the liquid side of the liquid-vapor interface.

When the mass flow of air is constant throughout the system, we may specify all other mass flow rates and state variables on a unit dry air basis. Conducting a vapor mass balance upon the overall control volume and the individual control volumes yields the following relations:

\[ dm_l = -m_a d\omega = dm_v \]  

For dehumidification, Eq. (5) describes the relation between the condensate production and the change in humidity ratio of the air. For humidification, Eq. (5) relates the mass of feed evaporated to the change in humidity ratio of the air. If the carrier gas is ideal, then the saturation curve on a molar basis is independent of the choice of carrier gas. The ability of the ideal carrier gas to hold vapor increases rapidly with temperature due to the exponential dependence of vapor pressure on saturation temperature.

3.2. Energy Balance and the Saturation Curve

Returning to Fig. 2, and applying the First Law to the control volume of carrier gas, we may relate the heat transfer rate to the specific enthalpies and mass flow rates:

\[ -\delta \dot{Q} = m_a (dh_a + \omega dh_v) + m_a h_v d\omega + h_{li} dm_v \]  

Employing Eq. (5), we substitute for \( dm_v \):

\[ -\delta \dot{Q} = m_a \left[ dh_a + \omega dh_v + (h_v - h_{li})d\omega \right] \]  

\( \delta \dot{Q}_E \) refers to an external heat transfer to the liquid stream. In an externally adiabatic humidifier this heat transfer would be zero. In a dehumidifier \( \dot{Q}_E \) would be negative and would represent heat transfer to a coolant. Applying the First Law to the liquid control volume yields:

\[ \delta \dot{Q}_E + \delta \dot{Q} = m_l dh_l + h_l dm_l - h_{li} dm_l \]
\[
\frac{\delta Q_E + \delta Q}{\dot{m}_a} = \frac{\dot{m}_1}{\dot{m}_a} dh_1 + \frac{d\dot{m}_1}{\dot{m}_a} (h_1 - h_{i,1})
\]  
(9)

The above equations give rise to the definition of specific enthalpy on a dry air basis, denoted by \(h^*\), to be used in all further analyses. In the case of humidification, \(h^*_f\) would represent the specific enthalpy of the feed on a dry air basis. For dehumidification, \(h^*_c\) would represent the specific enthalpy of the condensate on a dry air basis. If the bulk to interface temperature difference within the air stream is much larger than within the liquid stream, the bulk enthalpy of the liquid, \(h_l\), will be very close to the enthalpy of the liquid at the interface, \(h_{l,i}\), allowing the final term of Eq. (9) to be neglected.

Equation (9) indicates that the process paths of liquid streams can be easily visualised on a diagram of \(h^*\) versus \(T\). In the humidifier the evaporation rate is small compared to the mass flow of liquid. Liquid lines appear straight if drawn on a plot of enthalpy per unit dry air versus temperature, since \(c_{p,l}\) is almost constant with temperature. The slope of such a line will be dictated by the mass flow ratio (or the molar flow ratio) of liquid to dry air.

We would like to obtain an expression describing the enthalpy per unit dry air of saturated air as a function of temperature. Returning to Eq. (7), we can consider the order of magnitudes of each of the terms in \(dh^*_a\). We begin by considering \(h_v - h_{l,i}\).

\[
h_v - h_{l,i} \approx c_{p,v} [T_v - T_{sat}(p_v)] + h_{fg}(p_v) + c_{p,l} [T_{sat}(p_v) - T_{l,i}]
\]  
(10)

\[
\approx h_{fg}(p_v)
\]  
(11)

The air and the vapor are at the same temperature, \(T_a = T_v\), and the temperature difference between the liquid and the air would typically be below 10 °C. Consequently, \(h_{fg}\) dominates Eq. (10). We now introduce this approximation into Eq. (7) to obtain Eq. (12)

\[
dh^*_a \approx \dot{m}_a \left(c_{p,a} + \omega c_{p,v} + h_{fg} \frac{d\omega}{dT_a}\right) dT_a
\]  
(12)

\[
dh^*_a \approx \dot{m}_a h_{fg} d\omega
\]  
(13)
The humidity ratio of saturated air at 50°C is approximately 0.1 kg vap/kg air, while the slope of the saturation curve $d\omega/dT_a$ is approximately 0.005. At moderate and high temperatures (approximately 40 to 100°C) the value of $dh_a^*$ is primarily dictated by the third term in (12). Hence, the shape of the saturation curve on a humidity-temperature and an enthalpy-temperature diagram are very similar in this region. This assertion will prove useful when evaluating the performance of HDH cycles in Sect. 5.

4. Application of the Saturation Curve Method

4.1. Modelling Approximations

To appreciate the most important factors influencing the design of HDH systems, the following approximations are employed within our analysis:

1. Constant Pressure
2. Zero Salinity
3. Ideal Gas Behavior
4. Saturation Curve Process Path

The pressure rise provided by fan in an HDH system would be expected to be on the order of a few kPa, compared to an absolute pressure of 100 kPa. Embodiments of the HDH cycle do exist within which the pressure of the gas mixture is varied in order to drive humidification and dehumidification, [15, 16]. The approximation of zero salinity eliminates the need to adjust the saturation pressure and heat capacity of the feed stream. The error involved in this approximation is dealt with in Sect. 6. The final two approximations are dealt with each in turn.

4.1.1. Ideal Gas Approximation

For HVAC analyses, the ideal gas approximation is very well accepted when modelling moist air mixtures. Since HDH systems operate over a higher range of temperatures, and may be designed employing gases other than air, it is worth verifying the accuracy of ideal gas approximations. HDH systems operate at temperatures between approximately 20°C and 80°C and at pressures of 1 bar or lower, [17]. The ideality of gases may be verified by considering the compressibility factor at typical pressures and temperatures of HDH systems. A compressibility factor close to unity indicates an ideal gas:
\[ Z = \frac{P_v}{RT} \quad (14) \]

Calculation of this compressibility factor for helium, air, carbon dioxide and vapor indicates that, for the range of temperatures and partial pressures experienced in HDH systems, the compressibility factor is always between 0.996 and 1. Whilst the saturation properties of steam at atmospheric pressure are known to deviate from ideal when close to saturation, the temperatures and pressures present in HDH systems are sufficiently low for this deviation to be negligible.

4.1.2. Saturation Curve Process Path

The saturation curve methodology for the analysis of HDH systems involves approximating the process path of the moist carrier gas during humidification and dehumidification as the saturation curve. At saturation, the temperature and vapor pressure are not independent. Consequently, the parametric space that the moist carrier gas may occupy is reduced to the saturation curve.

Making these approximations simplifies the analysis of HDH considerably. Leaving salinity aside (Sect. 6) and variations in system pressure aside, the state of the water at any point in the system is fully specified by its temperature. The state of the gas mixture is also fully specified by its temperature and the saturation curve relates the humidity ratio to temperature. Next we describe two types of water heated closed air cycles, the first without any extraction or injection and the second with one single extraction and injection of water (as in Fig. 1).

4.2. Zero Extraction Cycle

We begin by describing the process paths of the moist air, condensate and feed streams within the dehumidifier. We fix the temperature of the inlet feed stream at \( T_{\text{in},\text{D,B}} \), and also fix the top temperature of the moist air \( T_{\text{ma,T}} \). We begin by imagining a dehumidifier of infinite area such that the pinch point temperature differences at the top and bottom are zero. The enthalpy

\(^1\)Practically, it is more convenient to fix the brine top temperature, by means of adjusting the brine heater power. Theoretically however, it is convenient for water production in the cycle not to be affected by changes in the top air temperature. This will become clear in Sect. 5.1, where zero and single extraction systems are compared.
change in the feed stream exactly matches the enthalpy change in the moist air and condensate streams. The dehumidifier is large enough for the moist air and condensate streams to reach equilibrium at the feed inlet temperature and for the feed to reach equilibrium at the moist air inlet temperature. In other words \( T_{f,D,B} = T_{ma,B} = T_{C,D,B} \) and also \( T_{f,D,T} = T_{ma,T} \). By fixing the temperature of the moist air at any point, we are also fixing its humidity, since we are approximating the process path by the saturation curve. By making these specifications, the mass flow ratio of feed to dry air becomes fixed, in order for the First Law to be satisfied for the dehumidifier. This may be seen in Fig. 3.

At each point in the dehumidifier, the condensate is taken to be at the temperature of the feed that is in counterflow. The thermal resistance between the condensate and the liquid feed is considered negligible compared to the thermal resistance between the moist air and the condensate/moist air interface. The choice of whether the condensate is at the liquid feed or air temperature is insignificant since the effect of the change in enthalpy of the condensate upon the shape of the dashed line is very small. In Fig. 3 the enthalpy of the condensate and the feed streams in the dehumidifier are combined. Since the dehumidifier is externally adiabatic, the change in enthalpy of the feed water and condensate matches the change in enthalpy of the moist air exactly.

\[ h_{ma,T}^* - h_{ma,H,B}^* = h_{f,H,T}^* - h_{f,H,B}^* \] (15)

However, by setting a pinch point temperature difference of zero in the humidifier, we obtain a final relation that closes the system of equations. Note how, due to the shape of the saturation curve, there are pinch points at the inlet and outlet of the dehumidifier but within the humidifier. At the pinch point in the humidifier, the evaporating water line must be tangent to the saturation curve, since the pinch temperature difference is zero. The process path of the evaporating water is close to but not exactly a straight
The path is slightly convex due to the evaporation of water. Since the mass of water evaporated is small the curvature is not noticeable.

4.3. Single Extraction/Injection Cycle

The single extraction/injection cycle differs in that the mass flow rate ratio of liquid-to-air is not constant (or approximately constant) within the system. Instead, liquid is extracted from the humidifier and injected into the dehumidifier (Fig. 1). This has the effect of increasing the liquid-to-air mass flow ratio in the upper section of the humidifier and dehumidifier, and decreasing the liquid-to-air mass flow ratio in the lower sections. For this single extraction/injection cycle, we begin by considering the dehumidifier. We fix the inlet feed stream at \( T_{f,D,B} \) and the top temperature of the moist air, \( T_{ma,T} \). We set the pinch point temperature difference to be zero at the top and bottom of the dehumidifier but also at the point of injection, i.e. \( T_{f,D,M} = T_{ma,D,M} \), Fig. 4.

At this stage we have two equations, the First Law above and below the injection point. We also have three unknowns, the temperature of injection, \( T_{f,D,M} \), and the feed to dry air mass flow ratio above and below the point of injection.

Moving to the humidifier, we again fix the inlet air temperature, \( T_{ma,B} \). The feed temperature at the extraction point in the humidifier equals the feed temperature at the injection point in the dehumidifier. The mass flow ratio at the top of the humidifier is set by the mass flow ratio above the injection point in the dehumidifier. The mass flow ratio just below the point of extraction in the humidifier is set by the mass flow ratio below the injection point in the dehumidifier, Fig. 4. We are now left with four unknown temperatures, \( T_{ma,T} \), \( T_{r,H,B} \), \( T_{f,D,M} \) and the temperature of the moist air at the feed extraction point in the humidifier, \( T_{ma,H,M} \). The mass flow ratio above and below the injection point in the dehumidifier are also unknown. This leaves us with six unknowns. However, we can also find six equations. First, we have four equations from the application of the first law above and below the extraction point in the humidifier and above and below the injection point in the dehumidifier. Additionally, by setting pinch point temperature differences of zero in the humidifier, we have two further equations. As in the case with zero extractions/injections, the slope of the evaporating stream
must equal the slope of the saturation curve in the humidifier at the pinch points. This allows us to solve for the profiles obtained in Fig. 4.

5. Results: Explanation of the Performance Limits of HDH

This section describes the limits upon the RR and GOR of HDH systems. These limits serve to bound the performance achievable in real systems. To understand the basis for changes in RR and GOR, it is instructive to return to Fig. 3 for a zero extraction/injection system. The specific heat capacity and mass flow of feed water in the dehumidifier are constant. Strictly speaking, due to evaporation, the ratio of liquid to moist air flow rate in the humidifier is not constant. However, since the mass of water evaporated is small relative to the total mass flow rate of feed, the liquid path (that is slightly curved in Fig. 3) is well approximated by a straight line. The change in enthalpy associated with the sensible cooling of the condensate in the dehumidifier is small and therefore, the feed and condensate path appears almost as a straight line.

The RR (ratio of product to feed water) is proportional to the width, \( a \), of the saturation curve in Fig. 3. The saturation curve on an enthalpy-temperature diagram is approximately the same as the saturation curve on an humidity-temperature diagram, scaled by \( h_{fg} \). The height, \( b \), of the saturation curve in Fig. 3 is proportional to the quantity of water produced per unit of dry air. Meanwhile, the ratio of feed to dry air flow is given by the slope of the liquid lines in Fig. 3, which may be represented by the height of the curve divided by the width, \( b/a \).

The GOR may be related to the heat input per unit mass of air, Eq. (3), or equivalently, the difference in enthalpy between streams leaving and entering the system [Eq. (16) and (17)].

\[
h^*_{t,D,B} - h^*_{t,H,B} = \frac{\dot{m}_{f,H,B}}{\dot{m}_a} c_{p,t} [(1 - RR)T_{t,D,B} - T_{t,H,B}] \quad (16)
\]

\[
\text{GOR} = \frac{h_{fg} \Delta \omega}{\frac{\dot{m}_{f,H,B}}{\dot{m}_a} c_{p,t} [(1 - RR)T_{t,D,B} - T_{t,H,B}]} \quad (17)
\]

5.1. Impact of a Single Extraction/Injection upon GOR and RR

For a feed inlet temperature of 25°C, a top moist air temperature of 70°C and pinch point temperature differences of zero, the RR and GOR of a cycle
with a single extraction are 11.1% and 14.2 respectively compared to 6.9% and 3.5 respectively without extraction.

The water produced per unit mass (or mole) of carrier gas in the dehumidifier is the same for the zero and single extraction systems. This is the advantage of fixing the top moist air temperature as opposed to the top feed temperature. If instead we fixed the top feed temperature, the top moist air temperature would differ in the cases of zero and single extraction. The recovery ratio is closely given by the mass of product divided by the mass of feed, or in other terms, the mass of product per unit mass of dry air divided by the mass of feed per unit mass of dry air, Eq. (18). The mass flow rate of feed per unit mass of carrier gas at the inlet to the HDH systems in Fig. 3 and Fig. 4 is given by the slope of the feed line at the inlet. Clearly the slope is lower in the system with one single extraction and consequently the recovery ratio is higher.

\[
RR = \frac{\dot{m}_p}{\dot{m}_{f,H_2O}} = \frac{\dot{m}_p \dot{m}_{da}}{\dot{m}_{f,H_2O}}.
\]  

A single extraction affects the GOR in three ways, Eq. (17). The mass flow rate ratio of feed-to-air at the bottom of the humidifier decreases and the temperature difference between the reject and the feed inlet decreases, both causing the GOR to increase. The recovery ratio increases, but remains small, and its effect upon GOR is small.

From a second law perspective, both heat and mass transfer affect the irreversibility (or the entropy generation rate) within an HDH system. For fixed values of pinch point temperature or concentration differences, an extraction/injection can allow the average driving forces for heat and/or mass transfer to be reduced. Furthermore, extraction/injection can result in a more uniform distribution of driving forces across the system, further reducing irreversibility, [18].

Heat transfer is driven by local differences in temperature between the liquid and moist air streams. Comparing Fig. 4 to Fig. 3, it is clear that extraction reduces local stream to stream temperature differences. Mass transfer is driven by differences in vapor concentration between the bulk air and the air that is locally saturated at the moist-air-liquid interface. In Fig. 5 the dashed lines represent the saturation humidity of the bulk moist air, while the solid lines represent the saturation humidity at the temperature of the feed stream. An extraction/injection reduces the concentration differences
that drive mass transfer along the humidifier and the dehumidifier.

5.2. Effect of the Cycle Temperature Range

Figure 6 indicates the reduction in RR as the top moist air temperature is lowered. The RR decreases since the mass flow of feed per unit mass of air (dictated by the slope of the feed line at the dehumidifier inlet) decreases at a faster rate than the mass flow of product water per unit mass of air. Fig. 3 helps to make this intuitively obvious. As explained early in Section 5, the distance $\text{b}$ on Fig. 3 is proportional to the water produced per unit of air. As the top moist air temperature decreases, the length of $\text{b}$ decreases and the length of $\text{a}$ decreases.2 The rate of water production per unit mass of air, given by $\text{b}$, decreases at a faster rate than the ratio of feed to air flow (given by $\frac{\text{b}}{\text{a}}$). According to Eq. (18), the recovery ratio must decrease. In the case of a single extraction, the slopes of liquid lines also decrease with decreasing top moist air temperature, also leading to a decreasing recovery ratio.

The GOR increases with decreasing top moist air temperature for both zero and single extraction systems, Fig. 6. Simply said, this is because our piecewise linear approximation of the saturation curve becomes more accurate when considering a cycle that spans a smaller temperature range. Again, it is important to emphasise that the values obtained serve as bounds upon the performance achievable in real systems.

Figure 7 indicates that the RR decreases as the inlet feed temperature increases. Considering Fig. 3 and remarking that water production is affected, to only a small extent, by a changing bottom air temperature, we see that the slope of the feed line will increase with increasing inlet feed temperature, thus leading to a decreasing recovery ratio.

The GOR increases with increasing inlet feed temperature for zero and single extraction systems. Water production is weakly affected by increasing the feed temperature (as opposed to decreasing the top moist air temperature). With increasing temperature, the process path of the saturation curve

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2 Note that the feed line and the evaporating line must be almost parallel as the liquid to air mass flow ratios are equal in the humidifier and dehumidifier.
becomes closer to being linear, allowing a significant reduction in the heat input to the system.

[Figure 7 about here.]

5.3. Effect of Component Size

In a bi-fluid heat exchanger, the driving force for heat transfer is dictated by local temperature difference between the two streams. For a fixed duty, i.e. heat transfer rate, a smaller heat exchanger will require larger driving forces (assuming constant heat transfer coefficients). The pinch point temperature difference (PPTD) is defined as the minimum temperature difference between the two streams. It is somewhat characteristic of the driving force for heat transfer. To achieve a fixed rate of heat transfer within a heat exchanger, the pinch point temperature difference is somewhat indicative of the heat exchanger size required. As the PPTD increases, a smaller heat exchanger is required for the same total heat transfer rate and vice versa.

The above arguments may also be applied to a mass exchanger. The driving force for mass transfer is given by a local concentration difference. During humidification and dehumidification that driving force is to the difference between the bulk moist-air vapor concentration and the vapor concentration at the liquid interface. A pinch point concentration difference (PPCD) would be somewhat analogous to the PPTD in a heat exchanger. In a heat and mass exchanger (HMX), such as a humidifier or dehumidifier, there are two driving forces, temperature difference and concentration difference. The size of the heat and mass exchanger, for a fixed heat and mass transfer rate is therefore somewhat related to the PPTD and the PPCD.

When the saturation curve is followed by moist air, the temperature and humidity of the moist air are always linked. Hence, by specifying a PPTD between a fluid and moist air stream at one point in the system, we are in fact also specifying a PPCD between the fluid and the moist air stream at that same location. When the PPTD throughout is zero, the PPCD throughout would also be zero. Otherwise, the exact relationship between PPCD and PPTD at a point will depend upon temperature. If we were to specify the same PPCD at different points in the system, the PPTD would not be the same at each of those points. At low temperatures the PPTD would be be large and at high temperatures the PPTD would be small, due to the shape of the saturation curve. Conversely, if we were to specify the PPTD at various points in the system, the PPCD would not be the same throughout. What
we can say, is that for fixed top and bottom moist air temperatures, and a fixed value of PPTD throughout the system, we expect that by increasing the chosen value of PPTD we will reduce the HMX size required for a fixed water output.

In Fig. 8 the GOR is plotted versus PPTD, setting the temperature difference at each pinch point to be identical at all pinch points. The inlet feed temperature is 25°C and the top moist air temperature is 70°C. As the PPTD increases in the zero and single extraction cases, the GOR decreases due to an increasing temperature difference across the heater. The GOR of a system with a single extraction/injection is more sensitive to changes in PPTD than a system with zero extractions. In an HDH system (where streams the moist air stream varies in heat capacity) with zero extractions and a small PPTD, the PPTD (or indeed the PPCD) is not a very representative measure of average driving forces within the system. Rather, the average driving force is dictated by mismatch in shape between the saturation curve and the feed streams. Increasing the PPTD therefore has a weak effect upon the temperature difference obtained across the heater. By contrast, in Fig. 4a and Fig. 4b, the PPTD is much more representative of the mean temperature difference between the liquid and moist air streams (and the PPCD is much more representative of the mean concentration difference). This is because much of the mismatch in shape between the liquid and moist air paths has been eliminated by the single extraction/injection. Consequently, the temperature difference across the heater is much more sensitive to the PPTD when there is an extraction/injection.

[Figure 8 about here.]

There exists a critical value of PPTD beyond which there is no advantage in employing an extraction and injection of water from the humidifier to the dehumidifier. This effect may be explained by considering a plot of the extraction temperature and the feed temperatures at which the pinch points occur in the humidifier versus the PPTD, Fig. 9. As the PPTD increases, the pinch points and extraction/injection point occur at higher temperatures. Once the extraction temperature reaches the top moist air temperature, minus the value of the PPTD (i.e. 70°C - 3.1°C) in Fig. 9, there is no longer a benefit to extracting.

[Figure 9 about here.]
As with the GOR, the RR decreases with increasing PPTD in the cases of both zero and single extraction, Fig. 10. As PPTD increases, the moist air temperature at the outlet of the dehumidifier increases, as does its humidity. Consequently, the mass flow rate of water produced per unit mass of air decreases. The slope of the liquid lines must increase, meaning that the mass flow rate of feed per unit mass of air must increase. According to Eq (18), the mass of water produced per mass of feed must decrease.

One further consideration not addressed in this analysis is system productivity. It refers to the rate of water production for a system containing a humidifier and dehumidifier of fixed design and size. Consideration of the PPTD (or indeed PPCD) alone is insufficient to judge the productivity of an HDH system. Without calculation of heat and mass transfer coefficients, the absolute system size (and hence productivity) remains unknown. However, the use of PPTD allows us to consider the effects of increasing or decreasing system size on GOR and RR. From the perspective of the design engineer, the trade-off between decreased energy costs at higher values of GOR must be weighed carefully against increased capital costs due to lower system productivity.

6. Impact of Salinity upon the Optimisation of HDH Systems

The above analysis of an HDH cycle, done neglecting the effects of salinity, gives an intuitive insight into the coupling between the humidifier and dehumidifier. Here, we point out how the analysis would have differed had seawater properties been taken into account. Saline feed solutions carry a risk of scaling that can greatly affect system operation and limit the temperatures permissible in the system. Beyond the effects of scaling, salinity affects the energetic performance of HDH in two key ways:

1. Salinity lowers the saturation pressure of water and consequently the saturation humidity ratio and saturation enthalpy of moist air
2. Salinity increases the specific heat at constant pressure of the feed water

6.1. Effect of Salinity upon the Properties of Saturated Air

During dehumidification of a carrier gas-vapor mixture, the relevant saturation curve will always be that which is in equilibrium with pure water.
However, during humidification, salinity will influence the vapor pressure and thus the saturation humidity ratio and enthalpy of the carrier gas. In Fig. 11, saturated moist air in equilibrium with seawater (at 35 000 ppm) is compared with saturated moist air in equilibrium with pure water.

Pure water properties are evaluated using the IAPWS 1995 formulation [19]. The saturation pressure of seawater is computed using correlations developed by Sharqawy et al. [20]. The relative humidity and enthalpy of the moist air mixture in equilibrium with pure water are evaluated using the formulations presented by Hyland and Wexler [21], as is done throughout the paper. The moist air mixture in equilibrium with seawater is approximated as an ideal mixture and the enthalpy per unit of dry air is evaluated using the properties provided by Wagner and Pruss [19] and Lemmon et al. [22] for air.

The difference in saturation pressure (seawater to pure water) is approximately -2% over the temperature range of interest for HDH. This results in a difference in humidity ratio (seawater to pure water) ranging from about -2% to -3% for a temperature range from 25°C to 70°C. Consequently, for seawater HDH processes, the water vapor per unit of carrier gas in the cycle and the water produced per unit of carrier gas will both be lower.

The normalised difference in enthalpy is evaluated by comparing the difference in saturated enthalpies to the change in enthalpy of moist air in a seawater HDH cycle (Eq. 19). The enthalpy of saturated moist air at a fixed temperature in a seawater HDH system will be up to 3% lower than in a pure water system. The slope of the saturation curve within the humidifier of an HDH system will be lower than the saturation curve within the dehumidifier. If the same PPTD is to be maintained within the humidifier as in the system with pure feed, the evaporating feed line in Fig. 3 requires shifting to the right, resulting in a larger temperature difference across the heater.

\[
\delta h^* = \frac{h_{sat-sw}^*(T) - h_{sat-pure}^*(T)}{h_{sat-sw}^*(70^\circ C) - h_{sat-sw}^*(25^\circ C)}
\]  

6.2. Effect of Salinity upon Specific Heat

At a salinity of 35 000 ppm, the specific heat of seawater is about 4.5% smaller than that of pure water [20]. If the same feed to air mass flow ratio
was chosen for a system with such saline feed, the slope of the feed stream in the dehumidifier and the evaporating stream in the humidifier would decrease by about 4.5%. To counteract this, and to achieve the same matching of the process paths in the humidifier and dehumidifier, the mass flow ratio of feed to dry air would have to be increased by about 4.5%. This explains the findings of Mistry [23], who analysed the impact of salinity upon the optimal mass flow ratio and GOR of a closed air water heated cycle by fixing humidifier and dehumidifier effectiveness.

In a system with saline feed water, the properties of the evaporating stream within the humidifier will vary due to changing salinity. In a system with a single extraction/injection, the salinity of the brine introduced from the humidifier to the dehumidifier at the midpoint would be slightly different from the salinity of the feed just before the dehumidifier injection point. However, since the recovery ratio of HDH cycles is low (less than 12% of the feed water is evaporated and subsequently condensed), changes in salinity due to concentration of the brine in the humidifier will be very small. In other words, if we consider the effect of salinity upon specific heat to be a second order effect (4.5%), then the effect of changing salinity due to evaporation would be a third order effect.
7. Conclusions

The Gained Output Ratio and Recovery Ratio of closed air water heated HDH cycles were analysed, both with zero extractions and a single extraction. Conclusions regarding the modelling of HDH cycles are as follow:

- The use of an ideal gas model for water vapor and air is shown to be highly accurate for the modelling of HDH systems.

- The effect of salinity at 35 000 ppm is to reduce the change in moist air humidity ratio and enthalpy by approximately 1% and 3% respectively (for a feed temperature of 25°C and a top air temperature of 70°C).

Conclusions regarding HDH performance in the limit of heat and mass exchangers that are sufficiently large to provide zero pinch point temperature and concentration differences are as follow. For cycles operating with a feed temperature of 25°C and a top air temperature of 70°C:

- GOR is limited to approximately 3.5 without extractions and approximately 14 with a single extraction

- RR is limited to approximately 7% without extractions and approximately 11% with a single extraction

- GOR increases and RR decreases as the temperature range of the cycle decreases, i.e. as the feed temperature increases or the top air temperature decreases.

Regarding the benefits of a single extraction/injection, a pinch point temperature difference was employed to parametrise the size of heat and mass exchangers and the following conclusions were obtained:

- A single extraction is only useful when heat and mass exchangers are large relative to system productivity (water produced per day).

- There exists a critical pinch point temperature difference above which there is no advantage to extracting and injecting water from the humidifier into the dehumidifier.
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8. Bibliography


Nomenclature

Acronyms

HDH Humdification-Dehumidification
GOR Gained Output Ratio
PPTD Pinch Point Temperature Difference
PPCD Pinch Point Concentration Difference
RR Recovery Ratio

Symbols

A area [m$^2$]
c concentration [mol/m$^3$]
c$\text{p}$ specific heat capacity at constant pressure [kJ/kg$\cdot$K]
h specific enthalpy [kJ/kg]
h$_{\text{fg}}$ latent heat of vaporization [kJ/kg]
H enthalpy [kJ]
m mass flow rate [kg/s]
M molar mass [kg/kmol]
N moles
p partial pressure [bar or kPa]
P absolute pressure [bar or kPa]
Q heat input [kJ]
T temperature [°C or K]
w specific work [kJ/kg product]
Z compressibility factor
Greek
\( \delta \)  incremental amount/deviation
\( \Delta \)  change
\( \phi \)  relative humidity [-]
\( \chi \)  molar humidity [mol H\(_2\)O/mol carrier gas]
\( \omega \)  humidity ratio [kg H\(_2\)O/kg carrier gas]

Subscripts
a  dry air
B  bottom
cg  carrier gas
c  concentrate
da  dry air
D  dehumidifier
E  external
f  feed
H  humidifier
in  into system
l  liquid
ma  moist air
min  minimum
M  Middle (extraction/injection point)
p  product
r  reject
T  top
v  vapor

Superscripts
\bar{\cdot}  molar
\dot{\cdot}  rate [s\(^{-1}\)]
\ast  per unit dry carrier gas
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