Optimization of multi-pressure himidification-dehumidification desalination using thermal vapor compression and hybridization

Proceedings of the 21st National & 10th ISHMT-ASME Heat and Mass Transfer Conference

December 27–30, 2011, IIT Madras, India

ISHMT_USA_015

OPTIMIZATION OF MULTI-PRESSURE HUMIDIFICATION-DEHUMIDIFICATION DESALINATION USING THERMAL VAPOR COMPRESSION AND HYBRIDIZATION

Karan H. Mistry, G. Prakash Narayan, Alexander Mitsos, John H. Lienhard V*

Department of Mechanical Engineering
Massachusetts Institute of Technology
Cambridge, MA, USA

ABSTRACT

Humidification-dehumidification (HD or HDH) desalination, and specifically HD driven by a thermal vapor compressor (TVC), is a thermal desalination method that has the potential to produce potable water efficiently in order to address the growing demand for water. This article presents a numerical study and optimization of two HD-TVC cycle configurations in order to determine the best achievable thermal performance. Through the use of nonlinear programming, it is found that the simplest configuration of HD-TVC has performance comparable to a traditional single-stage, single-pressure HD cycle (GOR ≈ 0.8–2.0), while the hybridized HD-TVC cycle with reverse osmosis (RO) has thermal performance that is competitive with existing large scale desalination systems (GOR ≈ 11.8–28.3).

Keywords: desalination, humidification-dehumidification, thermal vapor compression, reverse osmosis, numerical optimization.

1 INTRODUCTION

Growing world population and increasing industrialization of the developing world is heavily taxing limited fresh water supplies. Desalination introduces a new source of potable water, but the cost of traditional fuel sources is growing, resulting in higher energy costs and higher costs of water production. As a result, there is a need for energy efficient methods of desalinating water.

Humidification-dehumidification (HD or HDH) desalination is a distillation method that closely mimics nature's water cycle and has the potential to operate either with solar heating (renewable energy) or with low-grade or waste heat from plant (cogeneration). A solar still is the most basic form of an HD cycle. In a still, solar heat evaporates water which then condenses on a cooler surface and the condensate is removed and used as product water. Unfortunately, as a result of their simple design, solar stills are very energy inefficient. When the water vapor condenses on the cold surface, most of the latent heat of vaporization is lost to the environment. By separating the evaporation and condensation processes, and by incorporating regenerative heating, the system's efficiency has the potential to be greatly improved. This is the foundation for HD desalination.

Due to the straightforward design and the poten-

*Corresponding author: lienhard@mit.edu

tial for production of potable water in remote areas with minimum need for electricity, HD desalination has received considerable attention over the past few years [Mistry et al., 2010, 2011a,b, Narayan et al., 2011b, 2009, Narayan et al., 2010b,c]. Through these various studies, it was found that HD cycles can be improved dramatically through changing the pressure of the humidifier and the dehumidifier and by using a thermal vapor compressor (TVC) [Narayan et al., 2011a].

The goals of this article are: to introduce the multi-pressure humidification-dehumidification desalination cycle with a thermal vapor compressor and to show the optimal performance of HD-TVC cycles.

2 VARIED PRESSURE HD CYCLE

A commonly used figure of merit for HD and other thermal desalination systems is the gained output ratio (GOR). GOR is the ratio of the latent heat of evaporation of the water produced to the net heat input to the cycle. This parameter is, essentially, the effectiveness of water production, and is defined as an index of the amount of the heat recovery achieved in the system.

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

The GOR of an HD system is a function of two system parameters: vapor productivity ratio (VPR) and specific net heat input (SNH). 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 leaving the humidifier (evaporator).

$$\mathrm{GOR} = \underbrace{\left[\frac{\dot{m}_p}{\dot{m}_{cg}\omega_H^{\mathrm{out}}}\right]}_{\mathrm{VPR}} \underbrace{\left[\frac{\dot{m}_{cg}\omega_H^{\mathrm{out}}}{\dot{Q}_{\mathrm{in}}}\right]}_{1/\mathrm{SNH}} h_{fg} = \frac{\mathrm{VPR}}{\mathrm{SNH}} h_{fg} \tag{2}$$

Evidently, at a constant value of SNH, VPR should be maximized to maximize GOR.

VPR is a system parameter which gives a measure of the loss in efficiency caused by using a carrier gas to desalinate impure water. The value of VPR is always 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 VPR is 0.25, then for every four units of vapor present in the carrier gas at the exit of the humidifier, only one unit of water is produced.

$$\mathrm{VPR} = \frac{\dot{m}_p}{\dot{m}_{cg}\omega_H^{\mathrm{out}}} = 1 - \frac{\omega_D^{\mathrm{out}}}{\omega_H^{\mathrm{out}}} \tag{3}$$

From the equation above, it is evident that to increase VPR of an HD system the exit humidity ratio from the dehumidifier should be decreased and/or the exit humidity ratio from the humidifier should be increased. To understand humidity ratio better, consider a simplified expression using Dalton's Law while approximating the carrier gas and water vapor mixture as an ideal gas mixture and assume that the carrier gas is insoluble in water.

$$\omega(T,p,\phi) = \frac{M_w}{M_{cg}} \frac{\phi p_{\mathrm{sat}}(T)}{p - \phi p_{\mathrm{sat}}(T)} \tag{4}$$

The humidity ratio of the carrier gas can thus be written as a function of the mixture temperature (T), pressure (p), relative humidity (ϕ), and molar mass (M) of the carrier gas. Thus, ω can be increased or decreased by modifying the pressure of the mixture at fixed temperature and relative humidity. For example, at a dry bulb temperature of 65 °C the humidity ratio of saturated moist air ($\phi = 1$) is approximately doubled when the operating pressure is reduced from 100 kPa to 50 kPa. This article describes a HD cycle which operates the humidifier at a lower pressure and the dehumidifier at a higher pressure to increase VPR in order to increase overall performance of the system [Narayan et al., 2011a,b, 2009].

The pressure differential in the proposed cycle is maintained using compression and expansion devices. In the cycle discussed in the present study, the humidified carrier gas exiting the humidification chamber is compressed using a TVC and then dehumidified in the dehumidifier (see Fig. 1). The dehumidified carrier gas is then expanded using either a throttle or an air ex-

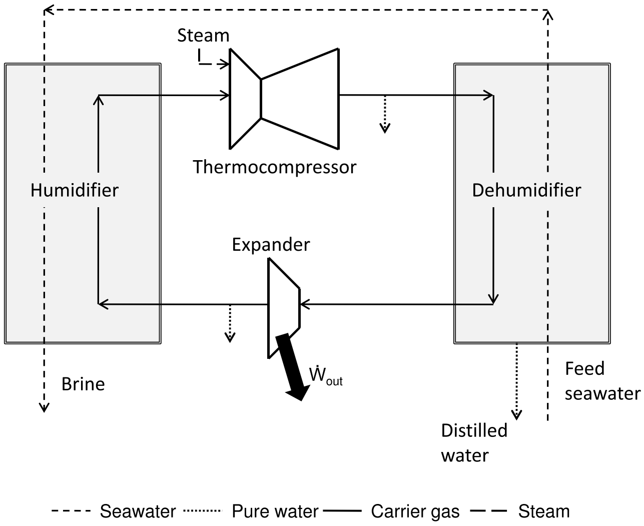

FIGURE 1. Schematic diagram of basic HD cycle with TVC and expansion device.

pander. The air expander allows for recovery of energy in the form of a work transfer. The expanded carrier gas is then sent to the humidification chamber. The carrier gas is thus operated in a closed loop. The feed seawater is preheated in the dehumidifier before it is sent to the humidification chamber, thus recovering some of the work input to the compressor in form of thermal energy which is given back to the carrier gas stream during the humidification process.

In a previous publication, a system using a mechanical compressor rather than a TVC was analyzed [Narayan et al., 2011a]. Such a system is powered using a supply of electricity. This article focuses on varied pressure systems powered by a TVC and highlights the optimal performance of this system using two different expansion devices (a throttle and an expander). The effect on optimal performance of coupling the expander to a reverse osmosis (RO) unit is also discussed.

3 MODELS AND SIMULATIONS

3.1 Modeling methodology

The modeling effort is based on the work by Mistry et al. [2010, 2011b]. JACOBIAN [Numerica Technology, 2009] is used for the system models since it simplifies the modeling of much larger problems through a modular method of model development. Complicated system models are built up by first modeling smaller components and then combining the component models to form complete systems. Modular development of systems allows for much cleaner models that are easier to modify while studying variations of a given system.

Separate simulation blocks are created to instantiate one or multiple system models and to fix the degrees of freedom. The advantage of this approach is that a single model (either for a component, or a cycle as a whole) can be used in multiple simulations. Therefore, simple variations of each of the models, including operating conditions and configurations, can be analyzed without having to duplicate code.

An existing in-house computational infrastructure linking various optimization solvers to JACOBIAN is used to optimize each of the HD-TVC cycles and configurations considered here. These optimization codes (discussed below) are sophisticated and allow for optimization over a large number of parameters.

3.2 Approximations

Calculations are performed for steady state while neglecting pumping power, kinetic, and potential energy effects. Components are approximated as well insulated (adiabatic with respect to the environment).

The moist air exiting from the humidifier and dehumidifier is approximated as saturated. Moist air exiting from a humidifier is always nearly saturated, provided there is sufficient contact area. Moist air exiting the dehumidifier may be below saturation, though calculations showed that dehumidifier exit humidity has a negligible effect on the system performance.

The dehumidifier condensate bulk temperature is evaluated as a function of the inlet and outlet wet-bulb temperatures of the moist air using a model developed by Mistry [2010], Mistry et al. [2010].

Finally, when the moist air exiting the TVC and the expander contains more moisture than saturated levels, it is assumed that the excess water is in the form of suspended liquid droplets.

3.3 Fluid properties

Dry air is modeled as an ideal gas (c_p is constant). Pure liquid water properties are evaluated using the International Association for the Properties of Water and Steam's 1997 Industrial Formulation [Cooper, 2007]. Dry air and pure water models are verified by comparing to REFPROP [Lemmon et al., 2007]. Seawater physical properties are evaluated using correlations presented in Sharqawy et al. [2010].

Moist air properties, evaluated per unit dry air, are based on the properties of the constituent substances and the ideal solution approximation [Mistry, 2010]. Moist air properties are verified by comparing to the values found in the ASHRAE Fundamentals Handbook [Wessel, ed., 2001].

3.4 Model components

The governing equations for each component are summarized below.

3.4.1 Humidifier and Dehumidifier Humidifiers and dehumidifiers are simultaneous heat and mass exchange devices. Provided that the states and mass flow rates of the two inlet streams are known, the outlet states are fully determined through the First Law of Thermodynamics, continuity, and a suitable definition of component effectiveness.

The energy-based effectiveness for simultaneous heat and mass transfer devices [Mistry et al., 2010, 2011b, Narayan et al., 2010a] is defined based on the change in enthalpy of the two streams:

$$\varepsilon = \max\left(\frac{\Delta\dot{H}_w}{\Delta\dot{H}_w^{\text{ideal}}}, \frac{\Delta\dot{H}_a}{\Delta\dot{H}_a^{\text{ideal}}}\right) \tag{5}$$

where the two ideal enthalpy changes are evaluated assuming a zero terminal temperature difference at the top (or bottom) of the exchanger: $T_{w,\text{out}}^{\text{ideal}} = T_{a,\text{in}}$ and $T_{a,\text{out}}^{\text{ideal}} = T_{w,\text{in}}$. Additionally, the moist air stream is assumed to be saturated at the exits.

3.4.2 TVC A thermal vapor compressor has four streams as seen in Fig. 1. The input steam (motive fluid), the moist air stream to be compressed (suction fluid), the compressed moist air stream (discharge fluid), and any possible condensate.

When the states of the inlet streams are known, the outlet states are determined using the First Law, continuity, and entrainment efficiency. The entrainment efficiency of a TVC is defined in terms of the stream enthalpy of the motive steam:

$$\eta_{\text{TVC}} = \frac{(\dot{m}h)_S^{\text{rev}}}{(\dot{m}h)_S} = \frac{\dot{m}_S^{\text{rev}}}{\dot{m}_S} \tag{6}$$

where $\dot{m}_S^{\text{rev}}$ is the mass flow rate of motive steam required to compress the suction fluid in a TVC operating under reversible conditions. Currently available TVCs tend to operate with low entrainment efficiencies with values ranging from 10–20%. There are several more advanced TVC designs that show promise at achieving higher entrainment efficiencies (about 40%).

3.4.3 Expander While there are many types of expansion devices, from a control volume point of view, all expanders can be characterized by an isentropic efficiency,

$$\eta_E = \frac{\dot{W}}{\dot{W}^{\text{rev}}} \tag{7}$$

where $\dot{W}^{\text{rev}}$ is the amount of work produced under reversible, adiabatic operation.

3.4.4 Throttle A throttle is the most basic form of an expansion device. Unfortunately, in its simplicity, it is also the most irreversible and is unable to produce any work. First Law, continuity, and backpressure are sufficient to determine the outlet state.

3.4.5 Reverse Osmosis A very simple approach is used for modeling RO behavior. Current RO systems can desalinate water using about 3.5 kWh/m^3 [Sommariva, 2010]. Therefore, the amount of water produced using the work from the expander is:

$$\dot{m}_{\text{RO}} = \frac{\dot{W}_{\text{expander}}}{3.5\,\text{kWh/m}^3} \cdot 1000\,\text{kg/m}^3 \cdot \frac{1\text{ hr}}{3600\text{ s}} \tag{8}$$

3.5 HD Cycles

HD cycle models are created by matching inlet and outlet streams of the component models and ensuring that all mass balances are maintained. For convenience, the ratio of the seawater flow rate in the dehumidifier to the flow rate of dry air is defined as:

$$\dot{m}_r \equiv \frac{\dot{m}_{w,D}}{\dot{m}_a} \tag{9}$$

where $\dot{m}_{w,D}$ is the mass flow rate of seawater in the dehumidifier and $\dot{m}_a$ is the mass flow rate of dry air in the moist air stream.

4 OPTIMIZATION METHODS

4.1 Methods

The general form of the optimization problem solved is

$$\min_{\mathbf{x}} f(\mathbf{x})$$
$$\text{such that } \mathbf{g}(\mathbf{x}) \leq \mathbf{0},\ \mathbf{x} \in [\mathbf{x^L}, \mathbf{x^U}],\ \mathbf{x^L}, \mathbf{x^U} \in \mathbb{R}^\mathbf{n}$$

where $\mathbf{x}$ are termed the optimization variables, $\mathbf{x}^L$ and $\mathbf{x}^U$ are the variable bounds, $f : [\mathbf{x}^L, \mathbf{x}^U] \to \mathbb{R}$ is the objective function and $\mathbf{g} : [\mathbf{x}^L, \mathbf{x}^U] \to \mathbb{R}^m$ are the inequality constraints.

The so-called sequential mode of optimization is used in which the optimization problem is separated from the simulation and only the degrees of freedom are used as optimization variables. This results in small-scale optimization problems with relatively expensive function and gradient evaluations.

First, the optimization algorithm selects values for these optimization variables. Then, it passes the optimization variable values as parameters to the simulator (JACOBIAN) which solves the model equations and evaluates the objective function and constraints. Additionally, gradients with respect to the optimization variables must be evaluated at each major iteration and JACOBIAN returns these gradients.

The gradient-based optimization solver, SNOPT [Gill et al., 2002], is used. SNOPT, a commercial code distributed as a set of Fortran 77 subroutines, is based on a sparse successive quadratic programming algorithm with limited-memory quasi-Newton approximations to the Hessian of the Lagrangian. Default values are used for the options and tolerances in JACOBIAN. For the optimization, the tolerances are set to 10^{-4}. Each local optimization run takes approximately 30 seconds when performed on a server consisting of PCs with two Intel Xeon E5405 quad core CPUs at 2.00 GHz (eight cores total, a single core is used per local optimization run) with 8 GB of RAM.

A limitation of gradient-based optimizers, including SNOPT, is that they generate local optima. Due to the non-convexity of the model equations, unfortunately local optimality does not imply global optimality and for the current case studies, the solution depends on the initial guess provided for each of the optimization variables. Therefore, the solutions reported cannot be rigorously guaranteed to be optimal. In order to overcome this limitation, a multi-start heuristic is used in which the initial guess is initialized randomly and 10,000 optimization runs are executed in a computer cluster.

4.2 Variable ranges and constraints

In order to optimize the system for maximum GOR, the main optimization variables considered are the mass flow rates, temperatures, pressures, and component effectivenesses. The simulator has to solve the model equations and calculate the cycle performance as a function of these variables. The problem constraints which the optimizer must satisfy are positive entropy generation and minimum terminal temperature difference. The optimization problems solved have 9 optimization variables with 4 constraints, and the embedded simulation problems include approximately 1200 state variables, depending on the particular cycle being considered. In most cases, a conservative and optimistic value for the variable range is selected. The ranges are discussed below and summarized in Table 1.

The mass flow rate ratio ($\dot{m}_r$) range is selected based on initial heuristic optimizations. Humidifier and dehumidifier efficiencies (ε_D, ε_H), TVC entrainment efficiency (η_{TVC}), and expander efficiency (η_e) are all given conservative and optimistic values based on ex-

TABLE 1. Variable ranges and constraints

Variable	Conservative Low	Conservative High	Optimistic Low	Optimistic High
VARIABLE RANGES				
$\dot{m}_r$	1	10	1	10
ε_D	0.01	0.8	0.01	0.9
ε_H	0.01	0.8	0.01	0.9
η_{TVC}	0.01	0.2	0.01	0.4
η_E	0.01	0.75	0.01	0.9
p_H [kPa]	20	100	20	100
p_r	1.2	5	1.01	5
T_S [K]	373.15	647	373.15	647
$p_{\text{sat},r}$	0.4	1	0.4	1
CONSTRAINTS				
$\dot{S}_{\text{gen}}$	0	∞	0	∞
TTD_H [K]	2.8	∞	2.8	∞
TTD_D [K]	4	∞	4	∞
p_S [kPa]	p_{sat}	22000	p_{sat}	22000
$T_w^{\max}$ [K]	0	343.15	0	358.15

TABLE 2. HD-TVC optimization results.
$T_{sw} = 303.15\,\text{K}$, $X_{sw} = 35,000\,\text{ppm}$
Cons: Conservative; Opt: Optimistic

Variable	with throttle Cons	with throttle Opt	with RO Cons	with RO Opt
$\dot{m}_r$	4.61	3.17	5.05	3.79
ε_D	0.80	0.90	0.80	0.76
ε_H	0.80	0.90	0.05	0.05
η_{TVC}	0.20	0.29	0.20	0.40
η_E	-	-	0.75	0.90
p_H [kPa]	55.06	82.82	45.32	68
p_r	1.20	1.13	1.20	1.20
T_S [K]	535.78	588.43	647	647
$p_{\text{sat},r}$	0.75	0.55	0.66	0.71
GOR	0.84	2.02	11.79	28.30
TTD_D [K]	9.30	4.64	5.12	4.13
TTD_H [K]	4.53	2.92	8.86	6.16
p_S [kPa]	3691	5834	14466	15538
$T_w^{\max}$ [K]	340.41	338.97	313.63	308.29

pectations of realistic hardware. Pressure in the humidifier (p_H) is limited to subatmospheric pressures based on previous studies. The pressure ratio of the dehumidifier to humidifier ($p_r = p_D/p_H$) is similarly chosen.

Motive steam temperature (T_S) and pressure (p_S) are limited to subcritical levels and the saturation pressure ratio, ($p_{\text{sat},r} \equiv p_S/p_{\text{sat}}$) is limited to be less than one to ensure that the state of water is always vapor.

Terminal temperature difference (TTD) constraints are selected based on the best performance of existing hardware and the maximum water temperature ($T_w^{\max}$) is determined based on solubility limits.

5 RESULTS AND CONCLUSIONS

Table 2 shows the various important design parameters at optimal performance of the HD-TVC-RO cycle and the HD-TVC cycle with a throttle. Through simulating and optimizing these cycles, it is observed that instead of a unique optimum (like was obtained for simple heated HD cycles [Mistry et al., 2011b]) there are numerous combinations (potentially on a manifold) of parametric values that resulted in very similar optimal GOR values. The values reported in the table are representative examples selected from these optimum values. While the above observation makes it difficult to come to sweeping generalizations regarding the importance of various design parameters based on the optimized results, there are still very clear conclusions that emerge from the current study.

5.1 HD-TVC cycle with throttle

The GOR for the HD-TVC cycle with the throttle is defined as below:

$$\text{GOR} = \frac{(\dot{m}_{\text{D}} + \dot{m}_{\text{TVC}} + \dot{m}_{\text{throttle}} - \dot{m}_S)\, h_{fg}}{\dot{m}_S (h_S^{\text{in}} - h_S^{\text{out}})} \tag{10}$$

From Table 2, it is seen that for the conservative case, the maximum performance as expected occurs at the highest possible component effectiveness/efficiency. It is also observed that there is an optimum value of humidifier pressure at which the GOR is maximized. This can be explained by considering GOR as a function of VPR and specific net heat [see Eq. (2)]. While VPR is increased at lower humidifier pressures, SNH input is also increased and the opposing effects cause GOR to be optimum at an intermediate value of humidifier pressure. This trend is found to be true for all four cases reported in this article.

It is also found that at lower pressure ratios the performance is generally better. Hence, it is vital to develop designs for low pressure ratio TVC and air expander. Work is in progress by the present authors in this regard. For the optimistic case, the trends are the same except for the effectiveness of the TVC. Interestingly it can be seen that the performance of the current cycle is optimum when the TVC effectiveness is not optimum. This is found to be true for all of the cluster of points around the optimum point.

The effect of steam conditions is important to design of these cycles [Narayan et al., 2011]. There is an optimum steam condition at which the entrainment ratio for a given pressure ratio is maximum. This is because of the way the thermodynamic properties of steam affect the entrainment ratio in the TVC [McGovern et al., 2011].

Finally, for both of these cases of the HD-TVC cycle with a throttle, the performance is fairly low. Hence, these cycles will not have much application unlike the cycle reported in the next section.

5.2 HD-TVC with expander and RO (HD-TVC-RO)

GOR for the HD-TVC cycle with an expander and RO is defined as below:

$$\mathrm{GOR} = \frac{\left(\dot{m}_{\mathrm{D}} + \dot{m}_{\mathrm{TVC}} + \dot{m}_{\mathrm{expander}} + \dot{m}_{\mathrm{RO}} - \dot{m}_S\right) h_{fg}}{\dot{m}_S (h_S^{\mathrm{in}} - h_S^{\mathrm{out}})} \tag{11}$$

The system performance for both the optimistic and the conservative cases is maximum at highest possible TVC and expander performance but at very low values of humidifier effectiveness. This shows that the cycle works best when most of the water is produced from RO rather than from the HD cycle. Moreover, the dehumidifier effectiveness is also not maximum for the optimum cases. High dehumidifier effectiveness results in high exit seawater temperature. Since limits are set on the maximum seawater temperature (to avoid scale formation), the dehumidifier effectiveness cannot always be at maximum values.

The observation made about the effect of the pressure ratio, the humidifier pressure and the steam conditions for the HD-TVC cycle with throttle also holds true for the HD-TVC-RO cycle. However, due to the effective recovery of energy in the expander, the HD-TVC-RO cycle is several times more efficient as other HD cycles. For the conservative case, GOR is above 10 and for the optimistic case, GOR is approaching 30. This shows the promise of this technology as a water desalination cycle.

5.3 Comparison to existing technologies

To evaluate and compare the performance of various thermal and electricity driven desalination technologies with that of the new system, GOR and equivalent electricity consumption ($\dot{E}_c$) are used [Narayan et al., 2011]. Equivalent electricity consumption, for a steam driven system, is defined as:

$$\dot{E}_c = \frac{\dot{m}_S (h_S^{\mathrm{in}} - h_S^{\mathrm{turbine,\,out}}) \eta_{\mathrm{gen}}}{\dot{m}_p \cdot 3.6} \left[\frac{\mathrm{kWh_e}}{\mathrm{m}^3}\right] \tag{12}$$

Where η_{gen} is efficiency of the electrical generator and is assumed to be 95%. In order to calculate $h_S^{\mathrm{turbine,\,out}}$, an isentropic efficiency of 85% and an exit temperature of 35 °C is assumed.

In order to calculate GOR, a thermal energy based performance parameter (see Eq. 1), for electricity driven systems, a power production efficiency (η_{PP}) of 40% is assumed in order to convert electricity consumed to thermal energy. Hence, for electrical energy driven systems,

$$\mathrm{GOR} = \frac{h_{fg} \eta_{pp}}{3.6 \dot{E}_c} \tag{13}$$

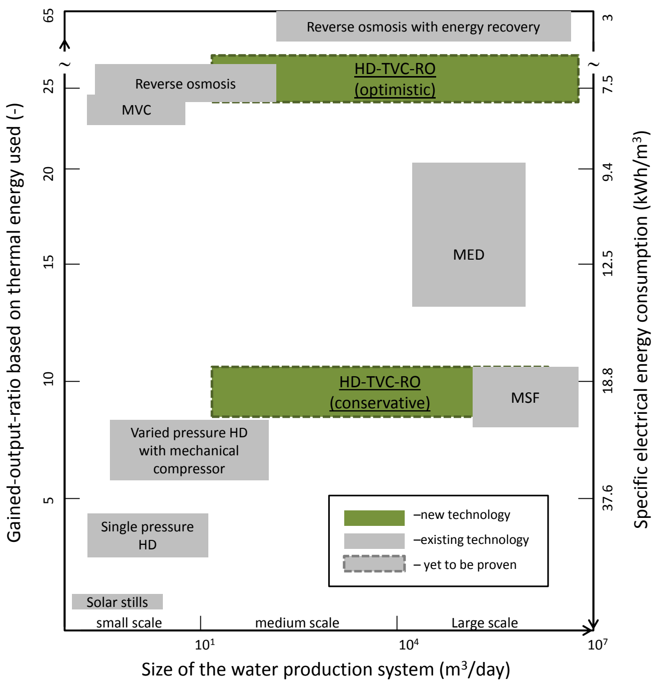

FIGURE 2. Benchmarking of new HD techniques against existing desalination systems.

From Fig. 2 it is observed that HD-TVC-RO system can potentially outperform MSF and MED in terms of GOR and equivalent electricity consumption. This performance is at a higher heating steam pressure (and temperature) than MSF and MED. MSF can run using saturated steam at 80–120 °C and MED at 60–80 °C [Morin, 1993].

The niche for the HD-TVC-RO technology is medium scale, stand alone, decentralized seawater desalination using medium pressure steam. It would be desirable to produce the steam using solar energy. Coastal communities which have high solar insolation, high water scarcity and unavailability of fossil fuels are a target for this technology.

ACKNOWLEDGMENTS

The authors would like to thank the King Fahd University of Petroleum and Minerals for funding the research reported in this article through the Center for Clean Water and Clean Energy at MIT and KFUPM and Numerica Technology for providing access to the JACOBIAN software for this research.

NOMENCLATURE

c_p	specific heat and constant pressure [kJ/kg-K]
$\dot{E}_c$	specific electricity consumption [kWh/m^3]
$\dot{H}$	enthalpy flow rate [kW]
h	specific enthalpy [kJ/kg]
h_{fg}	heat of vaporization [kJ/kg]
M	molecular weight [kg/kmol]
$\dot{m}$	mass flow rate [kg/s]
p	pressure [kPa]
$\dot{Q}_{\mathrm{in}}$	heat input [kW]
$\dot{S}_{\mathrm{gen}}$	entropy generation rate [kW/K]
$\dot{W}$	power [kW]
X	salinity [ppm]
T	temperature [K]
ε	heat and mass exchanger effectiveness [-]
η	efficiency [-]
ϕ	relative humidity [-]
ω	humidity ratio [kg/kg]
$(\cdot)_a$	air
$(\cdot)_{cg}$	carrier gas
$(\cdot)_D$	dehumidifier
$(\cdot)_E$	expander
$(\cdot)_H$	humidifier
$(\cdot)^{\mathrm{ideal}}$	TTD is zero
$(\cdot)^{\mathrm{max}}$	maximum
$(\cdot)_p$	product water
$(\cdot)_r$	ratio
$(\cdot)^{\mathrm{rev}}$	reversible
$(\cdot)_S$	steam
$(\cdot)_{\mathrm{sat}}$	saturated
$(\cdot)_{sw}$	seawater
$(\cdot)_w$	water
GOR	gained output ratio [-]
HD	humidification-dehumidification
RO	reverse osmosis
SNH	specific net heat input [kJ/kg]
TTD	terminal temperature difference [K]
TVC	thermal vapor compressor
VPR	vapor productivity ratio [kg/kg]

REFERENCES

Cooper, J.R., 2007. Revised Release on the IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam. The International Association for the Properties of Water and Steam , 1–48.

Gill, P.E., Murray, W., Saunders, M.A., 2002. SNOPT: An SQP Algorithm for Large-Scale Constrained Optimization. SIAM Journal on Optimization 12, 979–1006.

Lemmon, E.W., Huber, M.L., McLinden, M.O., 2007. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 8.0.

McGovern, R.K., Narayan, G.P., Lienhard V, J.H., 2011. Analysis of reversible ejectors and definition of an ejector efficiency. Under Review.

Mistry, K.H., 2010. Second Law Analysis and Optimization of Humidification-Dehumidification Desalination Cycles. Master's thesis. Massachusetts Institute of Technology. Cambridge, MA.

Mistry, K.H., Lienhard V, J.H., Zubair, S.M., 2010. Effect of entropy generation on the performance of humidification-dehumidification desalination cycles. International Journal of Thermal Sciences 49, 1837–1847.

Mistry, K.H., McGovern, R.K., Thiel, G.P., Summers, E.K., Zubair, S.M., Lienhard V, J.H., 2011a. Entropy generation analysis of desalination technologies. Entropy 13, 1829–1864.

Mistry, K.H., Mitsos, A., Lienhard V, J.H., 2011b. Optimal operating conditions and configurations for humidification-dehumidification desalination cycles. International Journal of Thermal Sciences 50, 779–789.

Morin, O., 1993. Design and operating comparison of MSF and MED systems. Desalination 93, 69–109.

Narayan, G.P., McGovern, R.K., Zubair, S.M., Lienhard V, J.H., 2011. High-temperature steam-driven varied-pressure humidification-dehumidification system coupled with reverse osmosis for energy-efficient seawater desalination, Under Review.

Narayan, G.P., McGovern, R.K., Zubair, S.M., Lienhard V, J.H., 2011a. Varied pressure humidification dehumidification desalination, in: Proceedings of the ASME/JSME 2011, 8th Thermal Engineering Joint Conference, Honolulu, HI.

Narayan, G.P., Mistry, K.H., Lienhard V, J.H., Zubair, S.M., 2011b. High-efficiency thermal-energy-driven water purification system. US Patent filing no. USSN 13/028,170.

Narayan, G.P., Mistry, K.H., Sharqawy, M.H., Zubair, S.M., Lienhard V, J.H., 2010a. Energy effectiveness of simultaneous heat and mass exchange devices. Frontiers in Heat and Mass Transfer 1, 1–13.

Narayan, G.P., Sharqawy, M.H., Lienhard V, J.H., Zubair, S.M., 2009. Water separation under varied pressure. US Patent number 2011/0079504 A1.

Narayan, G.P., Sharqawy, M.H., Lienhard V, J.H., Zubair, S.M., 2010b. Thermodynamic analysis of humidification dehumidification desalination cycles. Desalination and Water Treatment 16, 339–353.

Narayan, G.P., Sharqawy, M.H., Summers, E.K., Lienhard V, J.H., Zubair, S.M., Antar, M., 2010c. The potential of solar-driven humidification-dehumidification desalination for small-scale decentralized water production. Renewable and Sustainable Energy Reviews 14, 1187–1201.

Numerica Technology, 2009. JACOBIAN Modeling and Optimization Software. `http://www.numericatech.com`. Accessed March 2010.

Sharqawy, M.H., Lienhard V, J.H., Zubair, S.M., 2010. Thermophysical properties of seawater: A review of existing correlations and data. Desalination and Water Treatment 16, 354–380.

Sommariva, C., 2010. Desalination and Advanced Water Treatment: Economics and Financing. Balaban Desalination Publications, Hopkinton, MA 01748.

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