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Optimization of the mechanical vapor compression (MVC) desalination process using mathematical programming

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ABSTRACT

A mathematical model for the optimization of the mechanical vapor compression (MVC) desalination process is presented. The mathematical model involves the real physical constraints for the evaporation process. Nonlinear equations in terms of chemical–physical properties and design equations are used to model the process. A general algebraic modeling system (GAMS) is used to implement the model. The generalized reduced gradient algorithm CONOPT 2.041 is used as an NLP solver. The effects of some relevant process parameters on the system performance are studied. The output results from the proposed model were successfully compared with those of the literature.

Keywords: Mechanical vapor compression; Single-effect evaporation; Modeling and optimization of seawater desalination processes; General algebraic modeling systems (GAMS)

1. Introduction

Desalination of seawater is one of the main alternatives to overcome the problem of fresh water supply. Various types of desalination systems are known and are in use. Typically, such systems include a water pre-treatment system, a desalination unit and a post-treatment system. The desalination of seawater in such systems is achieved through thermal processes or through membrane processes. The thermal processes for seawater desalination include multistage-flash distillation (MSF), multi-effect distillation (MED) and vapor compression (VC). Further, membrane processes include reverse osmosis (RO) and electrodialysis (ED).

VC makes a product of similar quality to the other distillation processes. Its source of driving force is rotating mechanical energy generally from a motor. VC units tend to be small plants in isolated locations while the other processes are usually used for large freshwater productions. On the other hand, certain desalination systems can employ renewable energy sources for powering desalination systems. In fact, a mechanical vapor compression (MVC) desalination system may be powered by a wind turbine. Typically, wind powering of a MVC desalination

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system may be achieved either by direct mechanical coupling of the turbine shaft to the compressor axle of the desalination system, or by generating electrical power that is utilized to drive the electrical compressor drive. However, the mechanical coupling does not provide any means for power regulation or speed control of the compressor drive.

A brief outline of a number of articles focusing on mathematical modeling and analysis of single-effect VC desalination units can be found in the literature [1–3]. Also comparisons of the MVC system with other thermal compression processes can be found [4].

Aly Narmine et al. [1] investigated the thermal performance of the MVC system. Some details of the operational features of the unit were presented, including a comparison of the data of the unit with both theoretical and experimental results. The experimental and theoretical results indicated that the production rate increases by increasing the operating temperature from 70°C to about 98°C, evaporator designed temperature 70°C. Also, increasing the evaporator temperature has a good effect on the heat transfer coefficient.

Helal et al. [3] developed a complete model for the design of a hybrid solar-diesel powered MVC unit to obtain 120 m^3/d. All process equipment was rigorously modeled. Also, a sensitive analysis of the main cost elements of the plant and the environmental impact of the solar assisted unit with respect to CO_2 emissions and atmospheric oxygen consumption are illustrated.

Ettouney [2] investigated a comprehensive mathematical model for the design of the single-effect MVC process. The model equations include fundamental mass and energy equations of all process equipment (compressor, evaporator/condenser and pumps). Also, the proposed model includes a well tested set of correlations for calculations of the physical properties of all streams (distillate, brine and seawater), heat transfer coefficients and thermodynamic losses. With simulations, the system performance was analyzed as a function of the product flow rate, brine boiling temperature and dimensions of evaporator tube.

Al-Juwayhel et al. [4] compared four different types of single-effect evaporator desalination systems. The systems are driven by VC heat pumps including thermal (TVC), mechanical (MVC), absorption (ABVC), and adsorption (ADVC). The study included the development of mathematical models for the mentioned systems. The models included equations for energy and mass conservation. In addition, design equations were used to determine the heat transfer areas in the evaporator and the condenser. The analysis is based on comparison of the performance ratio, specific power consumption, specific heat transfer area, and specific cooling water flow rate. For the four systems the specific surface area for the evaporator and the condenser is decreased upon increase of the boiling temperature. The performance ratio for the TVC system is decreased as the boiling temperature and pressure of motive steam are increased. In the MVC system, the specific power consumption is found to decrease with the increase of the boiling temperature and its difference with the temperature of the compressed vapor. The ABVC and ADVC systems have higher potential than the other two configurations. This is because of the higher performance ratio found in both systems and the generation of hot utility water.

In this paper a mathematical model for the optimization of the MVC desalination process is presented. The mathematical model involves the real physical constraints for the evaporation process. Nonlinear equations in terms of chemical-physical properties and design equations are used to model all plant equipment.

The paper is outlined as follows. The process is briefly described in Section 2. Section 3 introduces the problem formulation. The mathematical model is summarized in Section 4. Section 5 presents applications of the developed model and results analysis. Finally, the conclusions and future works are presented in Section 6.

2. Process description

In the VC process, vapor is recompressed and introduced into the equipment. Two primary methods can be used for compressing vapor: (1) TVC and (2) MVC. Fig. 1 shows the single-effect MVC desalination process where the energy input is entirely mechanical power to drive the compressor. No live steam is required except for preliminary heating to raise the plant to working temperature. The main pieces of equipment used in the MVC desalting process are the evaporator, compressor, pumps and preheaters.

As shown in Fig. 1, the incoming seawater [Mf] is passed through two heat exchangers [HEX1 and HEX2] where it is preheated by the heat transferred from the discharged brine [Mb] and product [Md] streams. The seawater is then recycled and sprayed on the outside of a bundle where it boils and partially evaporates. Then, the produced steam is drawn through the demister to the centrifugal compressor [VC] which increases the pressure and temperature of the steam by compression. This steam is then discharged into the inside of the heat transfer tube bundle where it condenses into distillate [Md]. The compressor provides, through its suction, a pressure lower than the equilibrium of the brine, facilitating the evaporation of the seawater. The energy performance of the system depends on the pressure increase in the mechanical compressor, on the thermodynamic efficiency of polytropic process and on the efficiency of the electric motor.

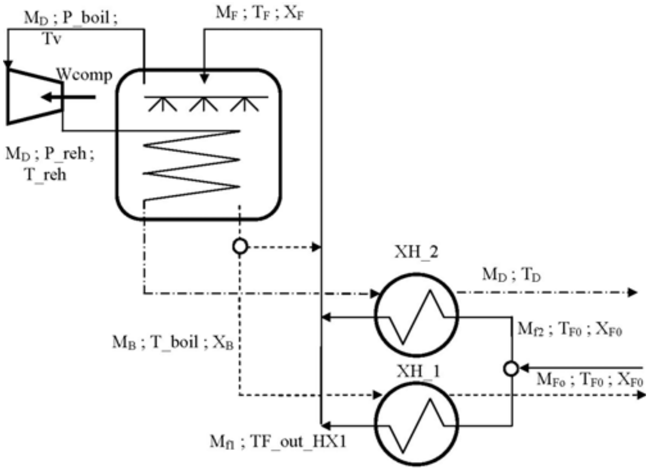

Fig. 1. Single-effect MVC desalination system.

MVC plants are in service with energy consumption around 11 kWh/m^3, and designs have been developed with power consumption as low as 8 kWh/m^3. This is the lowest energy consumption of any distillation so far developed and is competitive with seawater RO with energy recovery.

Capital and energy costs are significant factors on the total water production cost. The main energy consumption of the MVC distillation unit is represented by the electricity which is mainly required to drive the compressor motor, while there is no steam requirement. The operation and maintenance of the compressor motor may be half of the total operating cost. The process includes pumps for the seawater, brine and product. Part of the discharge brine is recycled by using a recirculation pump. MVC unit ratings have so far been limited to about 1500 m^3/d. The greatest disadvantages of the MVC system are the maximum allowable tip velocity of the compressor blades and mechanical compressor, which limit fresh water production.

3. Problem formulation

The optimization problem can be stated as follows: Given the fresh water demand and seawater conditions, the goal is to determine the optimal operating conditions in order to minimize the total annual cost. Other optimization problems can be formulated depending on the objective function to be optimized. For example, it is also possible to maximize the ratio of the fresh water production to the electricity used by the compressor and total heat transfer area of the process. Another optimization problem could be formulated as follows. Given the total heat transfer area (evaporator and pre-heaters), the goal of the problem is to maximize the ratio of fresh water production to the electricity used by the compressor.

4. Mathematical model of the MVC process

In this section, the assumptions and the mathematical model for the MVC system shown in Fig. 1 are presented.

4.1. Assumptions

The purpose of this model is to describe the MVC desalination process mathematically. The resulting mathematical model is based on the following assumptions:

- Product is pure water.
- Heat losses from the evaporator surface are negligible.
- No recycle is considered.
- Equal overall heat transfer coefficients in both heat exchangers.

4.2. Mathematical model

According to Fig. 1, the following mathematical model is proposed.

- Total mass and energy balance:

$$M_f = M_b + M_d \tag{1}$$

$$M_f X_f = M_b X_b + M_d X_d \tag{2}$$

The steam temperature leaving the evaporator is calculated as follows:

$$T_v = T_{boil} - BPE \tag{3}$$

The outside temperature at the compressor is computed as:

$$\frac{T_{sobrec}}{T_v} = Rp^{\frac{\gamma-1}{\gamma}} \tag{4}$$

where

$$RpP_{boil} = P2 \tag{5}$$

The mechanical power consumed by the compressor is given by:

$$W_{comp}\eta MW3600 = M_d \frac{\gamma}{\gamma-1} RT_v \frac{T_{sobrec}}{T_v - 1} \tag{6}$$

The energy balance on the evaporator is as follows:

$$M_d\left(H_{sobrec} - H_{vap\_sat}\right) + M_d \lambda_{TD} = M_f\left(H_{Tboil} - H_{Tf}\right) + M_d \lambda_{Tv} \tag{7}$$

The energy balances on the two pre-heaters are given by:

$$Q_{HX1} = M_b\left(H_{boil} - H_{b\_disch}\right) \tag{8}$$

$$Q_{HX1} = M_{f1}\left(HTF_{out\_HX1} - H_0\right) \tag{9}$$

The mass balance on the seawater splitter is:

$$M_f = M_{f1} + M_{f2} \tag{10}$$

$$Q_{HX2} = M_d\left(H_{Td} - H_{Td\_prod}\right) \tag{11}$$

$$Q_{HX2} = M_{f2}\left(H_{TF\_out\_HX2} - T_{SW}\right) \tag{12}$$

$$M_f H_{TF} = M_{f1} H_{TF\_out\_HX1} + M_{f2} H_{TF\_out\_HX2} \tag{13}$$

The temperature differences on hot/cold sides of pre-heaters are computed as follows:

$$\Delta t1_{HX1} = T_{boil} - TF\_out\_HX1 \tag{14}$$

$$\Delta t2_{HX1} = Tb_{disch} - T_{SW} \tag{15}$$

$$\Delta t1_{HX2} = Td - \mathrm{TF\_out\_HX2} \tag{16}$$

$$\Delta t2_{HX2} = T_{d\_prod} - \mathrm{T_{SW}} \tag{17}$$

The logarithmic mean temperature differences to compute the heat transfer area of pre-heaters (LMTD) are given by:

$$\mathrm{LMTD_{HX1}} = \frac{\Delta t1_{HX1} - \Delta t2_{HX1}}{\log \dfrac{\Delta t1_{HX1}}{\Delta t2_{HX1}}} \tag{18}$$

$$\mathrm{LMTD_{HX2}} = \frac{\Delta t1_{HX2} - \Delta t2_{HX2}}{\log \dfrac{\Delta t1_{HX2}}{\Delta t2_{HX2}}} \tag{19}$$

Then, the heat transfer areas are given by:

$$Q_{HX1} = U_{HX1} A_{HX1} LMTD_{HX1} \tag{20}$$

$$Q_{HX2} = U_{HX2} A_{HX2} LMTD_{HX2} \tag{21}$$

On the other hand, the evaporator area is calculated by:

$$A_{evap} = \frac{M_d \lambda_{TD}}{U_{evap}\left(T_d\ -\ T_{boil}\right)} + \frac{M_d Cp_v\left(T_{sobrec} - T_{boil}\right)}{U\ \ LMTD} \tag{22}$$

Then, the total heat transfer of the process is given by:

$$A_{TOTAL} = A_{HX1} + A_{HX2} + A_{evap} \tag{23}$$

The following logic and inequality constraints on temperatures are included in order to avoid temperature crossovers:

$$T_d \geq T_{boil} + 0.5$$

$$Tb_{disch} \geq T_{SW} + 0.5$$

$$T_{boil} \geq TF\_out\_HX1 + 0.5$$

$$TF\_out\_HX1 \geq T_{sw} + 1$$

$$Td_{product} \geq T_{SW} + 1;$$

$$T_d \geq TF\_out\_HX2 + 0.5$$

$$Td \geq Td_{product} + 0.5$$

On the other hand, the following analytic expression developed by El-Dessouky et al. [5] is used to compute the overall heat transfer:

$$U_{evap} = \left(1970 + 12.057T_{boil}\ 0.085989T_{boil}^2 + 0.25651\,103\ T_{boil}^3\right) \times 10^3$$

where U_{evap} is in kW/m^2 K and T_{boil} in °C.

The saturation temperature is calculated according to:

$$T_v = 42.6776 - \frac{3892.70}{\log \dfrac{P_{boil}}{1000} - 9.48654}$$

where P_{boil} is in kPa and T_v in K.

Finally, the functionality of heat capacity coefficients temperature and salinity for seawater, brine and distillate streams were taken from Helal and Al-Malek [3]. The optimization mathematical model involves 35 variables and 27 constraints. The model was implemented in the General Algebraic Modelling System GAMS [6]. CONOPT was used as the solver for the resulting NLP model [7].

Finally, it is important to notice that global optimal solutions cannot be guaranteed due to the non-convex constraints involved in the mathematical model.

5. Applications of the developed NLP model

In this section, the proposed model validation with the literature was performed and two optimization problems were solved under different objective functions. The parameter data set listed in Table 1 was assumed for all examples.

5.1. Model validation: Example 1

In this example, model outputs are compared with data from the literature to validate the proposed model. Design data previously reported [1–4] are considered for comparison purposes because enough information on the MCV desalination systems is reported by the authors. Many of such designs were obtained by simulations without optimization algorithms. In order to validate the proposed model with those designs, the optimization variables are fixed in the model at the same values as in Helal and Al-Malek [3]. Thus, the proposed model was here used more as a "simulator" than an optimizer and consequently no free-design variables were considered in this example.

In Table 2, the resulting values for the main process variables are reported. The values are compared with those obtained by Helal and Al-Malek [3]. From these results, it can be concluded that the obtained solution agree satisfactorily with the design reported [3]. Also, the proposed model was used as simulator to investigate the effect of the temperature difference between the saturated vapour and boiling brine on the process performance. The electricity consumed by the compressor increases with the temperature approach while the heat transfer area decreases.

5.2. Optimization problem:. Example 2

Once the model validation was conducted, the model was solved for the following optimization problem. Given the seawater conditions, the problem consisted of maximizing the ratio of fresh water production to electricity consumed and total heat transfer area. It should be notice that temperatures, pressures and mass flow-rates of all streams are variables to be optimized. Table 3 shows the optimal solutions.

The obtained results show that keeping constant T_{boil} (boiling temperature) and by increasing T_f (seawater temperature), the specific power consumption of the compressor decreases while the M_b (brine flow-rate) and M_d (distillate production) increase.

5.3. Economic model

Finally, a second optimization model was developed. The mathematical model proposed in Section 5.2 was extended in order to include an economic model. The goal was to minimize the total annual cost (TAC) of the system

Table 1
Problem data

Parameter	Value
X_f [ppm]	45,000
T_f [K]	298
Water production [kg/h]	4,790
Γ	1.34
Motor efficiency	0.78
Compressor efficiency	0.75

Table 2
Model validation (Example 1)

	Helal [2006]	NLP model
W_comp [kW]	76.53	71.02
Specific power consumption [kWh/m^3]	15.39	14.28
Rp	1.22	1.202
Td [K][a]	332.00	331.00[a]
T_{boil} [K][a]	329.00	329.00[a]
T_reheat [K]	345.00	343.76
Mb [kg/h]	6270.03	6332.93
Md [kg/h][a]	4972.00	4972.00[a]
LMTD_HX1 [K]	3.6	3.92
LMTD_HX2 [K]	1.2	1.00

[a]Values fixed in the model.

Table 3
Optimal values corresponding to Example 2

	Tf = 298 [K] *Xf* = 45,000 [ppm]	*Tf* = 301 [K] *Xf* = 45,000 [ppm]	*Tf* = 303 [K] *Xf* = 45,000 [ppm]
W_comp [kW]	36.656	37.216	40.552
Specific power consumption [kWh/m^3]	53.90	52.51	51.75
Rp	2.04	2.06	2.10
Td [K]	330.09	331.01	330.09
Tboil [K][a]	317.000	317.00	317.00
T_reheat [K]	376.882	378.18	377.98
Mb [kg/h]	628.226	637.83	695.02
Md [kg/h]	698.029	708.69	772.23
LMTD_HX1 [K]	3.91	2.63	1.76
LMTD_HX2 [K]	7.91	6.03	5.33

[a]Value fixed in the model.

for given fresh water demands. The TAC includes investment in pumps, pre-heaters, evaporator and compressor. Operating cost includes the electricity consumed by the compressor and pumps.

Two cost models were developed: linear and power models. The following analytic expressions were used to compute the cost of equipments for both model types. Those cost equations were taken from Hayani Mounir et al. [8].

1. Pump investment cost:

- Linear cost equation

$$C_{inv(pump)} = C_{pump}\, q_v + k1$$

where $C_{pump} = 61{,}65/\mathrm{m^3\ h^{-1}})$ and $k1 = 53{,}625$ US$

- Non-linear cost equation

$$C_{inv(pump)} = C_{pump}\, q_v^{\alpha}$$

where $C_{pump} = 4840{,}4$ US$/$(\mathrm{m^3 h^{-1}})$ and $\alpha = 0.111$

2. Exchanger investment cost:

- Linear cost equation

$$C_{inv(rec)} = V_{rec}\, A_{rec} V$$

where $V_{rec} = 189\ \mathrm{m^2}$ and $V = 25{,}071.90$ US$

- Non-linear cost equation

$$C_{inv(ec)} = V_{rec}\, A_{rec}^{\alpha}$$

where $\mathrm{V_{rec}} = 2221.1$ U$S/m^2 and $\alpha = 25071.9$

3. Evaporator investment cost:

- Linear cost equation

$$C_{inv(evap)} = V_{evap} A_{evap} + k2$$

$V_{evap} = 376.2$ US$/m^2 and $k2 = 63{,}584$ US$

- Non-linear cost equation:

$$C_{inv(ec)} = V_{rec}\, A_{rec}^{\alpha}$$

$V_{evap} = 3221.4$ U$S/m^2 and $\alpha = 0.2453$

4. Compressor investment cost:

- Linear cost equation

$$C_{inv(comp)} = V_{comp} W_{elec\text{-}comp} + k3$$

$V_{comp} = 1201.7$ US$/kWh and $k3 = 86{,}599$ US$

- Non-linear cost equation:

$$C_{inv(comp)} = C_{comp}\, W_{elec\text{-}comp}^{\beta}$$

$C_{comp} = 6475$ US$/kWh and $\beta = 0.7354$

5. Operation cost — The operating cost is computed as follows:

$$C_{oper} = V_{comp} W_{elec\text{-}comp} + V_{pump} W_{elec\text{-}pump}$$

where $V_{comp} = V_{pump} = 0.04$ US$/kW/h; $W_{elec\text{-}comp}$ and $W_{elec\text{-}pump}$ refer to the electricity consumed by the compressor and pumps.

As indicated in the following equation, the TAC is computed as the sum of capital investments and operating cost.

$$TAC = C_{inv} + C_{oper}$$

The results obtained show that economic optimal designs depend strongly on equipment costs. As a consequence of this strong dependence, the cost of all equipment should be based on all their variables in order to take into account the trade-offs between variables. In other words, rigorous and detailed cost models should be considered for detailed and realistic designs.

5.4. Computational aspects of the proposed model

Despite the simplicity of the model, non-convex constraints involved by the mathematical model such as logarithms to compute the logarithmic mean temperature differences (LMTD) and bilinear terms lead to local optimal solutions. As regards the initialization procedure, it is possible to identify "basic" variables for initialization "by hand" and then, from these variables, the remaining variables are automatically initialized.

From a sensitive analysis of the effect of initial values on the model convergence it is possible to conclude that the convergence of the proposed model strongly depends on the initial values. The model convergence is always guaranteed when initial values are near to the optimal ones. However, the goal is to develop a flexible mathematical model in order to incorporate it into other optimization models in order to study hybrid systems.

Therefore, a robust and efficient methodology should be developed in order to assure the convergence model and quasi-global optimal solutions. In order to improve the model convergence, it is interesting to involve simplified models to provide initial values to solve the proposed model. In addition, global optimization strategies can be also applied to solve the simplified models in order to provide "feasible" initial values for the rigorous model. Special attention on the scaling on variables and equations as well as appropriate lower and upper bounds on all variables should be taken into account by the solution methodology.

6. Conclusions

A simple optimization mathematical model of the MVC desalination system was presented. The model results agree satisfactorily with those reported by other authors.

In order to get more realistic designs, the presented model should be extended to a more rigorous one. For example, the effect of the non-condensable gases on the process, the velocity of steam inside the evaporator, among others must be considered by the model and they will be addressed in future works. Also, a robust solution methodology will be developed in order to guarantee the model convergence.

7. Symbols

A_HX1	—	Heat transfer area for the pre-heater HX1, m^2
A_HX2	—	Heat transfer area for the pre-heater HX2, m^2
A_evap	—	Heat transfer area for the evaporator, m^2
BPE	—	Boiling point elevation, K
dt1_HX1	—	Temperature difference at the hot side of HX1, K
dt2_HX1	—	Temperature difference at the cold side of HX1, K
dt1_HX2	—	Temperature difference at the hot side of HX2, K
dt2_HX2	—	Temperature difference at the cold side of HX2, K
LTDT_HX1	—	Logarithmic mean temperature difference at the exchanger HX1
LTDT_HX2	—	Logarithmic mean temperature difference at the exchanger HX2
efic_comp	—	Compressor efficiency
HTF_out_HX1		Enthalpy of seawater leaving pre-heater HX1, kcal/kg
H0	—	Enthalpy, kcal/kg
L_TD	—	Latent heat, kcal/kg
Md	—	Distillate product mass flow rate, kg/h
Mb	—	Brine mass flow rate, kg/h
Mf	—	Total seawater mass flow rate, kg/h
Mf1	—	Seawater mass flow rate to pre-heater HX1, kg/h
Mf2	—	Seawater mass flow rate to pre-heater HX1, kg/h
Pboil	—	Pressure of vapor leaving the evaporator, kPa
P2	—	Pressure of vapor leaving the compressor, kPa
Rp	—	Compression ratio
Tv	—	Temperature of vapor leaving the evaporator, kPa
Tboil	—	Boiling temperature, K
Tsobrec	—	Temperature of vapor leaving the compressor, K
Tb_disch	—	Temperature of discharged brine, K
Tseaw	—	Seawater temperature, K
TF_out_HX1	—	Temperature of seawater leaving pre-heater HX1, K
TF_out_HX2	—	Temperature of seawater leaving pre-heater HX2, K
Qexc_HX1	—	Thermal load of the pre-heater HX1, kcal/kg
Qexc_HX2	—	Thermal load of the pre-heater HX2, kcal/kg
W_comp	—	Power consumed by the compressor, kW
Xf	—	Seawater concentration, ppm
Xb	—	Brine concentration, ppm

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