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Parametric simulation of MED–TVC units in operation

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ABSTRACT

The main goal of this paper is to simulate the performance of Multiple Effect Desalination (MED) units in operation to predict the influence of variation of different input parameters on the amount of product, GOR value and thermodynamic condition of the unit during the operation. Therefore, for a given system with specified geometry, including thermocompressor, heat transfer area of the effects and condenser, mass and energy balance equations can be written and solved simultaneously in a set of nonlinear equations for different parts of the system. The results obtained by the simulation code are compared the experimental data obtained from an MED–TVC unit installed in Assaluyeh city in Kavian petrochemical company. There is a good agreement between the calculated results and the experimental data. Results show that the variation in the seawater temperature and the scaling amount around the tubes of the first effect have the most influence on variation in production rate.

Keywords: Simulation; MED–TVC; Desalination

1. Introduction

Many countries in the world suffer from shortage of natural fresh water. Demand for fresh water will increase in the future as a result of the rise in population growth rates and enhanced living standards, together with the expansion of industrial and agricultural activities. In the past, many technologies have been developed for making drinkable water from brackish and seawater. Desalination is now generally considered as an economically viable option to solve the water shortage problems [1–3]. Nearly 50% of desalination plants around the world are of the thermal type (Multi-stage Flash, MSF, or Multi-Effect Distillation, MED). MED is occupying higher desalination market share today because of its relatively lower energy consumption (relative to MSF) and considerable technical improvements [4].

Some research has been carried out to investigate the characteristics and the performance of MED–TVC. Morin [4], Temstet et al. [5] and Michels [6] have described the process characteristics of MED–TVC and compared its features with other desalination processes. Darwish and El-Dessouky [7] conducted a study that compared the specific available energy, performance ratio, and specific heat transfer area for MED, MED–TVC, and MSF. Al-Najem et al. [8] proposed a parametric analysis using the first and second laws of thermodynamics for single- and multi-effect TVC systems. El-Dessouky et al. [9–11] presented an analysis for single- and multi-effect TVC systems, which focused on the parameters that affect the product cost. A report by Ophir and Lokiec [12] described the design principles

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and various considerations concerning advanced MED processes including MED–TVC, which results in the most economical desalination system. Thermal analysis of three different configurations of a multi-effect thermal vapor compression desalting system was presented by Alasfour et al. [13]. A design algorithm of MED–TVC units is presented by Kkamali et al. [14,15] which can provide engineers a cost-effective tool for thermodynamic design, development, and optimization of the thermal desalination plants. They developed a mathematical model which would predict the influence of all parameters on heat transfer coefficients, temperature and pressure, total capacity, and performance ratio of the system under design conditions.

Although lots of work have aimed to do sensitivity analysis and parametric study of multiple effect desalination units, this work has important differences with those works in the approach of solving the problem. They considered heat transfer area, condenser area, and other geometrical variables as unknown parameters and obtained them by means of written equations. In other words, they have written a code to design a multi-effect desalination plant as a function of determined process specifications. In contrast, we wrote our code by simulation approach. The geometrical features of the unit are assumed as known parameters in this works so as to simulate the behavior of a constructed MED plant. The main objective of this article is to develop a simulation code for a MED–TVC unit that is able to predict the influence of changing the operating conditions on thermodynamic characteristics and the performance of the unit in terms of production rate and GOR value during the full load and partial load operation. Performance of the unit for a constructed desalination unit can be evaluated in term of production rate, for the most important output process parameter for a specified unit is production rate. The GOR value is also very important in considering the performance of the system, as it shows the energy efficiency of the system. The aim of the simulation is to monitor the changes in the behavior of the unit with any changes in the system's inputs, i.e. flow and thermodynamic properties of entering stream or any change in the process such as formation of fouling on the tube surface.

2. Mathematical modeling

2.1. Modeling of desalination process based on simulation approach

The multi-effect seawater desalination process with thermal vapor compression using horizontal tube evaporators is illustrated schematically in Fig. 1. The system consists of a number of evaporators, a condenser, and a thermocompressor. In each effect, heat is transferred from the condensing water vapor inside the tubes to the evaporating brine around the tube bundle.

Modeling a unit by simulation approach, the specifications of the unit related to its geometry is assumed to be known and constant. Then, the effect of any change or fluctuation in the inlet streams or any other change which may occur by time is investigated on the performance of the unit in terms of production rate. The production rate is selected as representative of the unit's performance, as it is the most important parameter to be modified for a constructed unit with specified geometry.

This process is repeated successively in each of the effects at progressively lower pressure and temperature, driven by the water vapor from the preceding effect. In the last effect, at the lowest pressure and temperature, the water vapor condenses in condenser. The condensate distillate is collected from each effect in the water box of the next effect and flashes there because of the lower pressure and temperature.

The following assumptions are considered in this mathematical modeling:

(1) The operation is in a steady state.
(2) Physical properties of steam flow are functions of temperature.
(3) Physical properties of liquid flow are functions of temperature and salinity.
(4) Steam is condensed completely in each effect and leaves at saturated temperature.
(5) Product is free of salt.
(6) Flow and temperature of feed are equal for all effects.
(7) All effects have the same heat transfer area.

The mathematical model of the system includes the following set of unknown variables:

- Vapor flow rate due to brine evaporation in effects 1 through n, which are defined by $V_1(n)$ and the discharge vapor flow from the thermocompressor which is defined by V (0). This gives $n + 1$ unknowns.
- Brine flow rate in effects 1 through n, which are defined by B (n). This gives n unknowns.
- Temperature of the steam enters the effects and condenser, which are defined by T_V (n) and the temperature of the discharge vapor stream from the thermocompressor which is defined by T_V (0). This gives $n + 1$ unknowns.
- The brine temperature of each effect. They are defined by T_B (n). This gives n unknowns.
- The feed temperature of each effect. They are defined by T_F (n). This gives n unknowns.

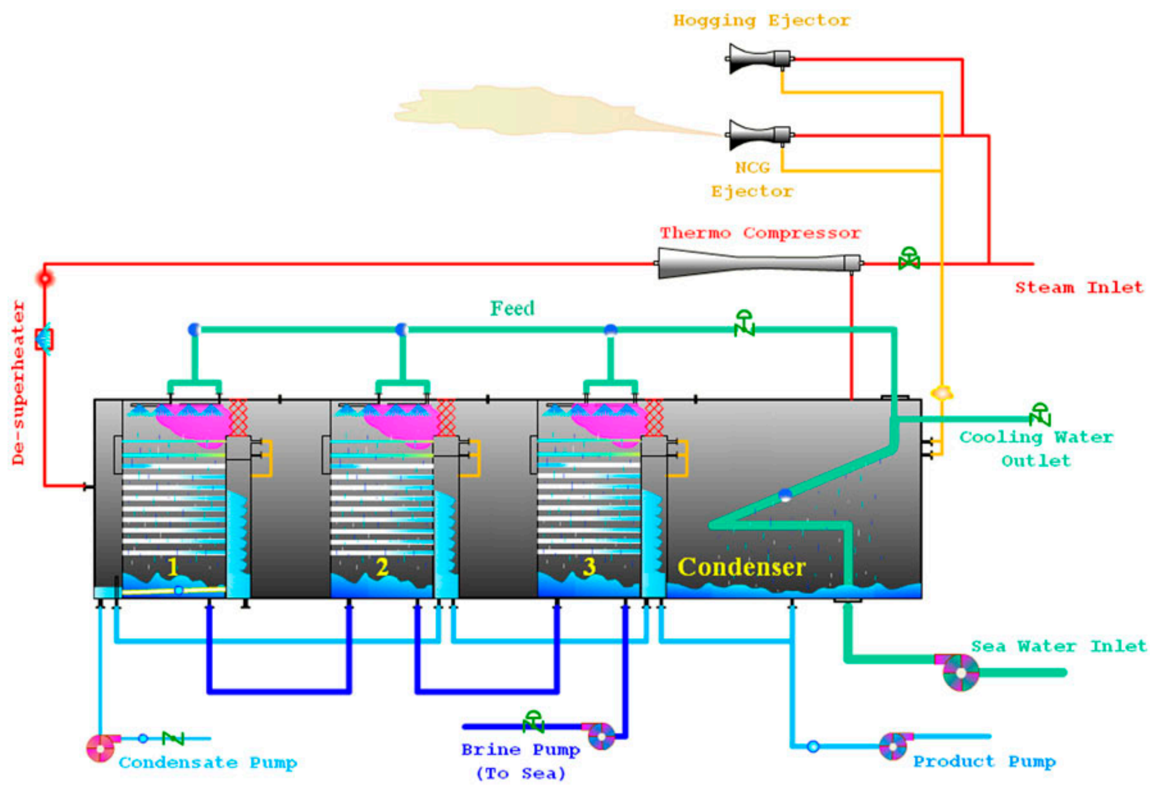


Fig. 1. A schematic of the MED–TVC system.

- Distillate flow rates due to vapor condensation in tubes which are defined by $C\ (n)$. This gives $n+1$ unknowns.
- Temperature of vapor condensates in tubes which are defined by $T_C\ (n)$. This gives $n+1$ unknowns.
- Distillate flow rates due to brine flashing which are defined for effect $2-n$ by $V_B\ (n)$. This gives $n-1$ unknowns.
- Distillate flow rates due to product flashing which are defined for effect $2-n$ by $V_P\ (n)$. This gives $n-1$ unknowns.
- The flow rate of the condenser cooling water, which is defined by M_{CW}.
- The vapor entrained from an effect as suction to thermocompressor which is defined by V_2 (Nthc).

Known variables of the system are as follows:

- Temperature and flow rate of the incoming seawater, which are, respectively, defined by T_{sea} and M_{sea}.
- Feed flow rate and feed salinity, which are, respectively, defined by F and X_F.
- Motive steam flow rate and temperature, which are, respectively, defined by M_{st} and T_{st}.
- Length, material, thickness, and number of tubes in each effect and condenser.

Fig. 2 shows the inlet and outlet streams of an effect of a multiple effect desalination system. $V_1\ (n)$ refers to outlet vapor stream of effect n to effect $n+1$. $V_2\ (n)$ refers to the sucked vapor from effect n to thermocompressor. The value for $V_2\ (n)$ is zero for all effects unless the effect with the number Nthc on which thermocompressor is located. $F\ (n)$ is the feed of the effect n the portion of which evaporates by being heated on the tube bundles of the effect. $B\ (n)$ is the concentrated liquid resulting from the evaporation of feed stream. $C(n)$ is the distillate flow resulting from condensation of the produced vapor in the last effect. In fact, $C(n)$ is the product of the effect n.

Mass balance:

$$V_1(n-1) - C_1(n-1) = 0 \tag{1}$$

$$F(n) - V_1(n) - V_2(n) - B(n) = 0 \tag{2}$$

These two equations are the mass balance conservations inside and outside of the tubes of each effect.

Energy balance:

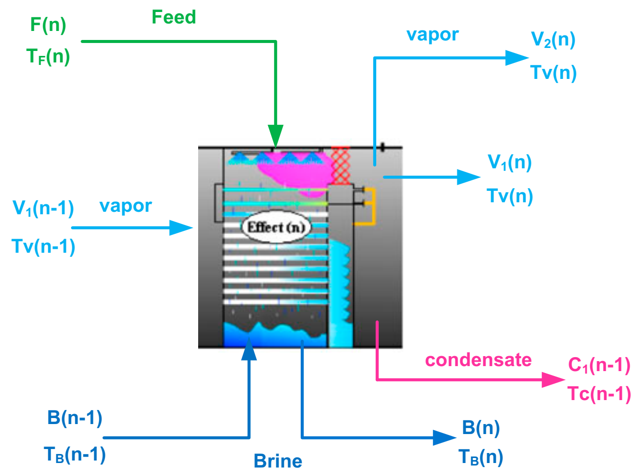


Fig. 2. Effect as a control volume.

The energy balance equation can be written in each effect as following:

$$[V_P(n-1)+V_B(n-1)+V_1(n-1)]L(n-1) + +F(n)Cp_F(n)(T_F(n)-T_B(n))-V_1(n)L(n) - V_2(n)L(n) = 0 \tag{3}$$

As, there is no flashing vapor in effect 1, this equation for the effect 1 one converts to the following equation:

$$V(0)*L(0)+F(1)Cp_F(1)(T_F(1)-T_B(1))-V_1(1)L(1) - V_2(1)L(1)=0 \tag{3.a}$$

Also, the heat transfer rate should be corresponding with that of latent heat of the condensing steam in each effect.

$$V_1(n-1)L(n-1)-U_e(n)A_e(n)(T_V(n-1)-T_B(n))=0 \tag{4}$$

This equation converts to the following equation, when is applied for the first effect:

$$V(0)L(0)-U_e(1)A_e(1)(T_V(0)-T_B(1))=0 \tag{5}$$

In addition, the overall heat transfer coefficient which is necessary to be calculated is calculated based on the following equation:

$$\frac{1}{U_e}=\frac{1}{h_o}+\frac{1}{h_i}\frac{A_O}{A_i}+\frac{Ln\frac{D_O}{D_i}}{2\pi kL}+R''_{f,O} \tag{6}$$

where U_e is the overall heat transfer coefficient based on external surface area of the tubes which is calculated considering heat transfer coefficient inside the tubes, heat transfer coefficient outside the tubes, conduction of tube wall, and fouling factor outside the tubes.

In addition, the pressure drop of the flowing vapor inside the unit may cause temperature loss in the unit and reduce temperature difference, and thus increases required surface area. The following two equations show the relation between temperature of the streams and the temperature loss of the flowing vapor inside and outside of tubes.

$$T_V(n-1)-T_C(n-1)-\Delta T\text{loss}_{\text{Demister}}-\Delta T\text{loss}_{\text{inside}}=0 \tag{7}$$

$$T_V(n)-T_B(n-1)+\text{BPE}(n)+\Delta T\text{loss}_{\text{outside}}=0 \tag{8}$$

where $\Delta T\text{loss}_{\text{Demister}}$, $\Delta T\text{loss}_{\text{inside}}$, and $\Delta T\text{loss}_{\text{outside}}$ are the temperature drops due to vapor pressure drop across flowing across demister, inside the tubes, and outside the tubes.

The following next two equations are energy conservation equations to obtain the amount of flashing vapor of brine stream or distillate stream.

$$V_B(n)L(n)-B(n-1)Cp_B(n-1)(T_B(n-1)-T_B(n))=0 \tag{9}$$

$$V_P(n)L(n)-\left[\sum_{k=1}^{n-1}V_1(k)+\sum_{k=1}^{n-2}V_B(k)\right]Cp_F(n)(T_V(n-1) - T_V(n))=0 \tag{10}$$

It is necessary to mention that the Eqs. (8) and (9) are not applicable for the first effect, and thus these equations can not be written for the first effect.

As shown in Fig. 3, the vapor produced in the last effect is divided to two parts which are not equal. A part of vapor generated in the last effect is passed through condenser and is condensed by inlet seawater. In this process, seawater which is used as feed water of effects is preheated. Another portion of the seawater passing through the condenser is rejected which is called M_{rej} in the Fig. 3. This flow is the same as M_{CW} which is considered as an unknown variable in last section. The mass and energy balance for condenser can be written as follows:

The following three equations are mass and energy balance equations for the outside and inside of condenser tubes and also the mass balance equation for distribution of the feed to the effects, respectively.

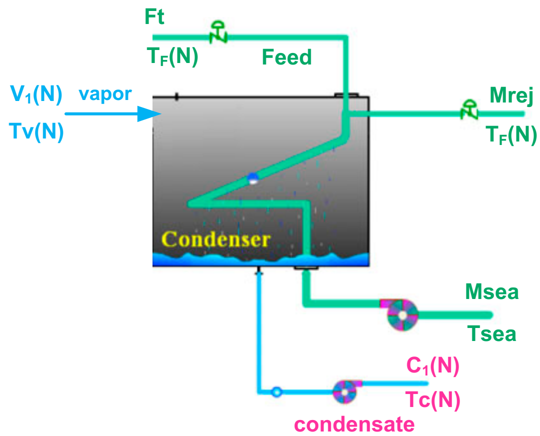


Fig. 3. Condenser as a control volume.

Mass balance:

$$V_1(n_t) - C_1(n_t) = 0 \tag{11}$$

$$M_{Sea} - F_t - M_{CW} = 0 \tag{12}$$

$$F_t = \sum_{n=1}^{N} F(n) \tag{13}$$

The following equation is energy balance equation between the hot fluid which is condensing steam in the condenser and cold medium which is moving seawater inside the tubes.

Energy balance:

$$[V_B(n_t) + V_1(n_t)]L(N) - M_{Sea}Cp_F(n_t)(T_F(n_t) - T_{sea}) = 0 \tag{14}$$

The following equation is the equation which relates temperature difference of the streams with the required surface area of the condenser.

$$[V_B(n_t) + V_1(n_t)]L(n_t) - U_{cond}A_{cond}\left[\frac{(T_F - T_{sea})}{\ln\frac{T_V(n_t)-T_{sea}}{T_V(n_t)-T_F}}\right] = 0 \tag{15}$$

As there is an equilibrium between condensing vapor and distillate in condenser, the temperature of the two streams should be the same as each other.

$$T_V(n_t) - T_C(n_t) = 0 \tag{16}$$

Mass and momentum balance for thermocompressor (Fig. 4) is as follows:

$$V_2(\mathrm{Nthc}) + M_{st} - V(0) = 0 \tag{17}$$

$$V(0) = \frac{P_V(0)}{\sqrt{RT_V(0)}} A_{thc}\gamma^{0.5} Ma\left(1 + \left(\frac{\gamma - 1}{2}\right) Ma^2\right)^{(\gamma+1)/(2-2\gamma)} \tag{18}$$

where $P_V(1)$ is the motive steam pressure, A_{thc} is the cross section area, and R is the universal gas constant. The above equation can be used to calculate the mass flow rate of the discharge flow of thermocompressor which can be found in most of books concerning dynamic gas [16]. In addition, the entrainment ratio of the thermocompressor, which is the ratio of vapor sucked from the MED unit to that of inlet fresh motive steam to the thermocompressor, is calculated according to following Eq. (18):

$$\frac{1}{\mathrm{ER}} = 0.296 * \frac{P_{dis}^{1.19}}{P_{suc}^{1.04}} * \left(\frac{P_{st}}{P_{suc}}\right)^{0.015} * \left(\frac{\mathrm{PCF}}{\mathrm{TCF}}\right) \tag{19}$$

where PCF and TCF are pressure correction factor for motive steam pressure and temperature correction factor for saturation temperature of suction flow which can be obtained using following equations, respectively:

$$\mathrm{PCF} = 3 * 10^{-7}P_{st}^2 - 0.0009P_{st} + 1.6101 \tag{20}$$

$$\mathrm{TCF} = 3 * 10^{-8}T_{suc}^2 - 0.0006T_{suc} + 1.0047 \tag{21}$$

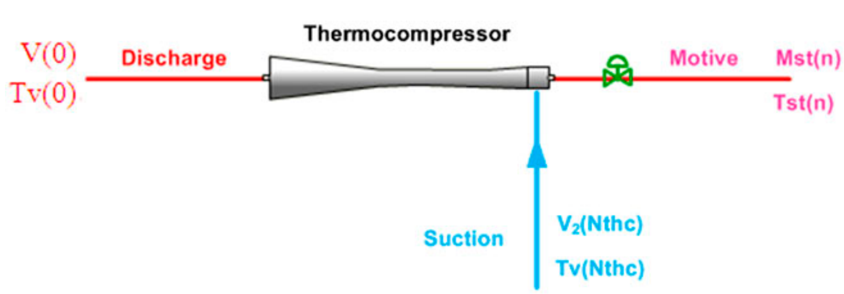
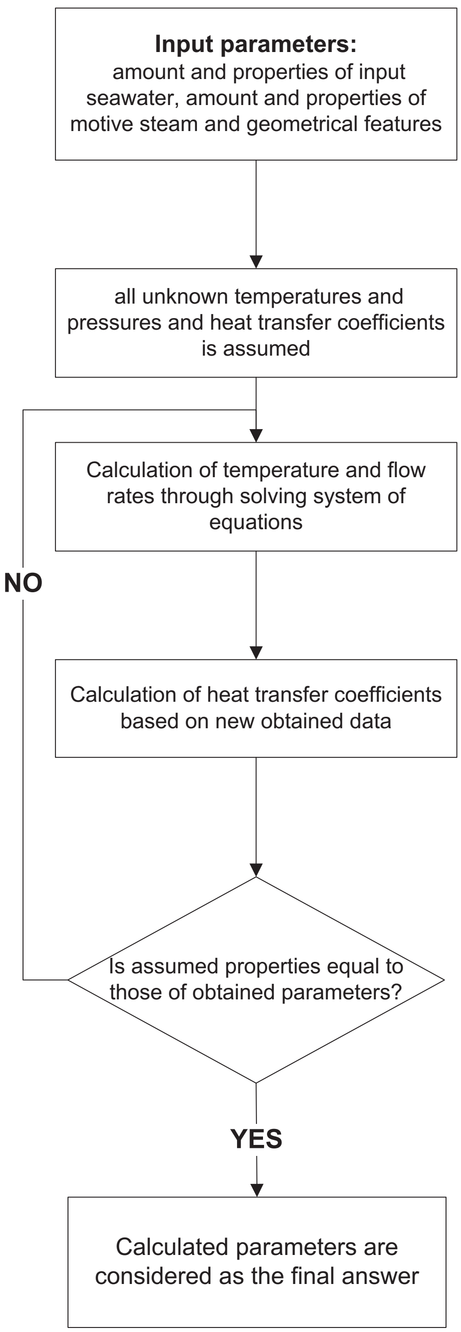


Fig. 4. Thermocompressor as a control volume.

As there are $(9N + 4)$ equations and $(9N + 4)$ unknowns, the set of equations has a unique answer.

2.2. Computational algorithm

The scheme of the stages for solving the set of above equations is shown below:



As it can be seen in the above diagram, firstly known parameters are inserted in the MATLAB simulation code. After that, the set of equations, which was described in the last section, is solved with numerical simulation by MATLAB, using Newton-Raphson Method. The solution of the nonlinear system of equations by Newton-Raphson method is briefly described in the next section.

2.3. Description of the solution of a set of nonlinear equation system using Newton–Raphson method

Assume a system of n non-linear equations in n unknowns given by:

$$
\begin{aligned}
f_1(x_1, x_2, \ldots, x_n) &= 0 \\
f_2(x_1, x_2, \ldots, x_n) &= 0 \\
&\;\,. \\
&\;\,. \\
f_n(x_1, x_2, \ldots, x_n) &= 0
\end{aligned}
$$

This set of equations can be written in the following form:

$$\mathbf{f}(\mathbf{x}) = 0$$

where $\mathbf{x}$ is a vector of the independent variables and the vector $\mathbf{f}$ is consisted of the functions $f_i(x)$:

$$x = \begin{bmatrix} x_1 \\ x_2 \\ . \\ . \\ x_n \end{bmatrix}, \mathbf{f}(\mathbf{x}) = \begin{bmatrix} f_1(x_1, x_2, \ldots, x_n) \\ f_2(x_1, x_2, \ldots, x_n) \\ . \\ . \\ f_n(x_1, x_2, \ldots, x_n) \end{bmatrix} = \begin{bmatrix} f_1(x) \\ f_2(x) \\ . \\ . \\ f_n(x) \end{bmatrix}$$

For solution of the system of nonlinear set of equations above, the Jacobian matrix should be formed as following:

$$\mathbf{J} = \frac{\partial(f_1, f_2, \ldots, f_n)}{\partial(x_1, x_2, \ldots, x_n)} = \left[\frac{\partial f_i}{\partial x_j}\right]_{n*n}$$

Firstly, all the unknown parameters, i.e. x_i are guessed and then the next guess for unknown parameters are obtained as following:

$$x_{k+1} = x_k - J^{-1}.f(x_k)$$

The final and definite answer is obtained when the following condition is reached:

$$\max|f(x_k)| < \varepsilon$$

Then, the value of the vector {**x**} is considered to be final and definite value of unknown parameters.

2.4. Initial guess of unknown parameters

As it is said before, for precise solution of the problem, the unknown parameters as well as heat transfer coefficients should be guessed firstly. Whatever the initial guesses of unknown parameters are more accurate, the final result will be reached more easily and accurately. In this work, it is tried to guess the unknown parameters as much close as possible to the final obtained results. To have better estimation of unknown parameters, a good sense about allowable range of these parameters should be there. Reviewing corresponding references in the literature [13,14,17–20] and also considering physical behavior of desalination units, a good knowledge of the range of unknown parameters is obtained, a part of which is described below:

- The flow rate of the vapor produced in every effect should be around the total capacity divided by number of effects assuming that equal amount of vapor is produced in each effect. Therefore, a good estimation of produced flow of vapor streams may be the total capacity divided by the number of effects. The flow rate of a vapor stream can not be never negative and more than total product flow rate of the unit.
- The flow rate of the brine stream in every effect could be equal to the feed flow rate of the inlet feed to the effect minus the assumed flow rate for the vapor stream which is obtained in last section. It can be never more than the flow rate of the feed of each effect and also can have never a negative value.
- All the temperatures inside the MED unit are assumed based on the temperature of the inlet vapor to the MED unit and the temperature of the inlet seawater. Generally, the temperature of the inlet vapor to the desalination unit does not exceed 70˚C, for it sharply intensifies the risk of fouling in MED units. Furthermore, the temperature of the condenser should be higher than the temperature of the seawater as the seawater is the cooling medium in the condenser. The value of temperature of the effects and the brine and vapor streams is assumed between the value of the temperature of the inlet steam to the unit and temperature of the condenser.
- The heat transfer coefficients in the MED units are measured to be between 1,500 and 4,500 w/m^2 ˚C, according to experiments and the mentioned references in the literature. Therefore, the values of heat transfer coefficients are guessed to be at this range.
- Temperature of the feed streams is guessed based on the temperature of the inlet seawater. Generally, the temperature of the feed water is a few degrees higher than the temperature of the inlet seawater.
- The flow rate of distillate streams are equivalent to the flow rate of the produced vapor in the last effect unless for the effect on which the thermocompressor is located, the next effect of which the flow rate of the distillate stream is equal to the flow rate of the produced vapor in last effect minus the flow rate of the sucked vapor to the thermocompressor. As a rule of thumb, the flow rate of the sucked vapor can be guessed by putting the ER equal to 1. This could be an appropriate assumption for the entrainment ratio of the thermocompressor.
- The flow rate of flashed streams as a result of transmission of the brine streams and product streams is guessed to be 5% of the flow rate of the total stream. This is because generally the temperature difference around 2.5–7.5˚C between the effects results in flashing of 5% of the moving streams.

3. Results and discussion

3.1. Experimental verification

Operating data of an actual MED–TVC unit installed in Assaluyeh city in Kavian petrochemical company (Fig. 5) has been adopted for experimental verification. Process data and geometrical specifications of the unit are presented in Table 1. The GOR value in

Fig. 5. MED–TVC system in Kavian petrochemical company, Iran (http://www.fanniroo.com).

Table 1
Process data of Kavian desalination plant

Parameter	Value	Unit
Motive steam pressure	1,680	kPa
Motive steam temperature	230	˚C
Seawater flow rate	805.3	ton/h
Seawater temperature	29	˚C
Feed salinity	45.6	g/L
Feed flow rate (each effect)	139.3	ton/h
No. of effects	4	–
Motive steam flow rate	24.2	ton/h
Effects area	2,424	m^2
GOR	7.1	–
Condenser area	808	m^2

this table is the ratio of net product flow rate to motive steam flow rate. The control panel of the desalination unit is shown in Fig. 6.

A schematic of input and output data panel of the simulation code is shown in Fig. 7. The results obtained from the simulation code and actual operating conditions are compared in Table 2. As it can be seen, the simulation code can well predict the performance of the actual desalination unit.

3.2. Sensitivity analysis

The influence of the variation in operating conditions for the actual desalination plant (Fig. 5) on the plant performance, i.e. production rate and GOR value is investigated by the developed simulation code. The GOR value is defined as the ratio of the net produced fresh water rate (total product rate minus steam flow rate) to that of fresh high-pressure steam flow rate.

The effect of seawater temperature, while the amount of cooling water is attempted to be kept constant, on the produced fresh water mass flow rate and also GOR value is shown in Fig. 8. As can be seen in the figure, for the seawater flow with the temperature below 30˚C, a 25% rise in the seawater temperature results in 11% increase in the amount of produced fresh water mass flow rate and GOR value. Increasing the seawater temperature causes to increase the feed water temperature which results in higher evaporation and production rate. The maximum discharge pressure (MDP) is the maximum capability of a thermocompressor to increase the sucked vapor. However, increasing the seawater temperature above 30˚C results in the reduction of fresh water production rate. This is because of the fact that the rise in seawater temperature leads into the highly elevation of condenser temperature, and thus first effect temperature. The increase in the pressure of the first effects is equivalent to the need for higher MDP of the thermocompressor. Higher required MDP affects the performance of thermocompressor in a negative manner and thus it leads to the reduction in fresh water production rate due to the drastic decrease in suction flow rate. As it can be seen, the trend of the change in the GOR value is the same as produced fresh water flow rate. As the amount of input steam flow rate to the unit is constant, the GOR value is directly related to the produced fresh water flow rate, and therefore the trend of variation in the GOR value is the same as the produced water flow rate, as it is depicted in the Fig. 8.

Fig. 9 illustrates the influence of variation in seawater flow rate in constant input feed flow rate to the effects. As it is shown in this figure, 10% increase in seawater flow rate causes about 2.4% decrease in total production rate and the GOR value falls from its initial value of 7.13 to the final value of 6.77. This is the result of reduction in feed and effects temperature due to increasing seawater flow rate. Again, it is obvious that in constant flow rate of input steam into the first effect, the trend of the change in the GOR value is the same as produced fresh water flow rate.

Fig. 10 shows the variation of total produced fresh water by increasing the motive steam flow rate. As it can easily be seen, by change of the steam flow rate from 22.5 to 26.27 ton/h, i.e. 16.5% increase in steam flow rate, in constant inlet seawater flow rate causes 12.5% rise in total product flow rate from 182 to 204.8 ton/h where it reaches its maximum value. This is because by addition of steam flow rate, higher energy is available in the first effect for the falling seawater to be evaporated, and thus higher amount of steam flow is produced in the first effect and moves toward the second effect. This continuous increase in the evaporation rate in the sequential effects occurs and thus results in the increase in the elevation of amount of condensate streams, and thus rising of produced fresh water flow rate. However, not only did not the GOR value increase, but also it decreases from its initial value of 7.09 to 6.79. This is because by delivering more motive steam to the system, evaporation rate of each effect increases, so feed water and effects temperature goes up as well. But, as the heat transfer surface area is constant, increasing the motive steam flow rate causes a rise in the temperature difference between the effects and consequently the discharge pressure and temperature goes up. In this case, thermocompressor cannot work in fully stable condition. Therefore, the suction load and the total production rate decreases sharply as a result of partially unstable function of the thermocompressor due to its high compression ratio. By further increase in the amount of motive steam flow rate from 26.27 to 27 ton/h, not only did the GOR value sharply begin to fall sharply from 6.79 to

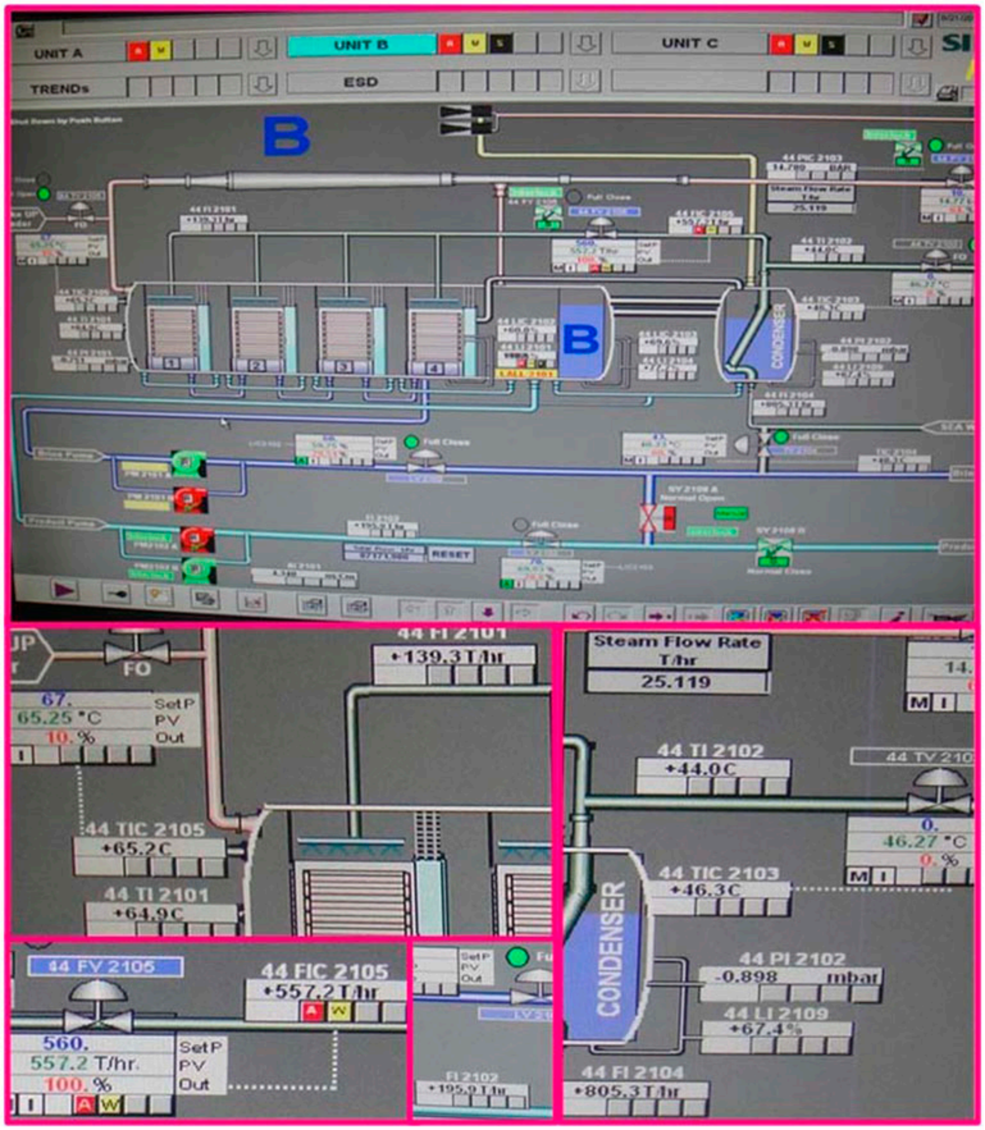


Fig. 6. A schematic of unit control panel.

6.42, but also the amount of produced water flow rate decreases from 204.8 to 200.5 ton/h, for very higher compression ratio of the thermocompressor relative to that it is designed for results in highly unstable function of the thermocompressor in the fluctuated manner. Hence, sometimes the thermocompressor isn't able to suck the low-pressure secondary flow from the condenser until the temperature of the first effect decreases as a result of very lower flow of steam to the first effect. Again, it begin to entrain the secondary flow from condenser and so the temperature of the first effect begin to rise as the result of higher steam flow rate to the first effect, and thus higher heat load of the effects. After the temperature of the first effects becomes too high, again the thermocompressor begins to return the entrained flow to the condenser. As a result of this defect in the performance of the thermocompressor and fluctuated flow of the input steam to the first effect, the production rate and thus GOR value decrease by addition of the amount of motive steam flow rate. This is an interesting result of simulation of a constructed MED–TVC plant, while this behavior cannot be predicted when design approach is utilized to simulate the behavior of an MED–TVC system. In all similar works, product flow rate continuously elevates as the motive steam flow rate increases. While simulating the behavior of a specified MED–TVC unit, product flow rate may decrease by elevation of motive steam flow rate, for the available surface area and thermocompressor's geometry is not suitable to yield a higher product flow rate.

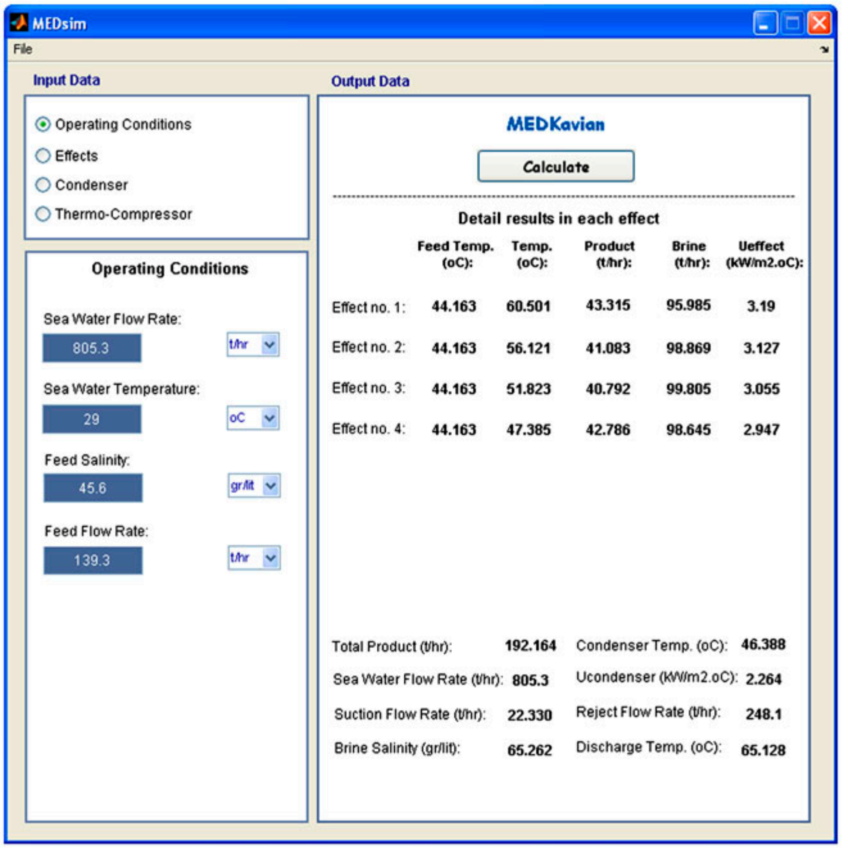


Fig. 7. Input and output data panel of simulation code.

Table 2
Comparison of simulation results and actual plant data

Parameter	Actual plant data	Calculated	Unit	Error %
Total product	195.9	193.1	ton/h	1.91
Feed water temp.	44	44.16	℃	0.37
Condenser temp.	46.27	46.39	℃	0.26
Discharge temp.	65.25	65.13	℃	0.19
Entrained steam	26.2	25.6	ton/h	2.3
M_{CW}	590	586	ton/h	0.6
First effect temp.	60	60.5	℃	0.83
Second effect temp.	56	56.1	℃	0.22
Third effect temp.	52	51.8	℃	0.34
Last effect temp.	Not available	47.4	℃	–

The influence of variation in feed water flow rate on the total net production rate and GOR value is investigated in Fig. 11. As it can be seen in this figure, by increasing the amount of the total input feed flow rate from 447.7 to 674.4 ton/h, the produced fresh water flow rate and the GOR value decrease to 4.9%. This is because of the fact that the increase in the feed flow rate leads to achieve a higher heat transfer coefficient and evaporation rate. But in the other points of view, the higher feed water flow provides higher pressure drop among the tubes and effects and results in waste of energy in the system. Therefore, due to this dissipation, evaporation rate will decrease. As shown in Fig. 11, the influence of increasing the pressure drop dominates the rise in heat transfer coefficient a little bit; so 20% increase in feed flow rate lead to 2.4% decrement in total production rate. There is another reason for the decrease

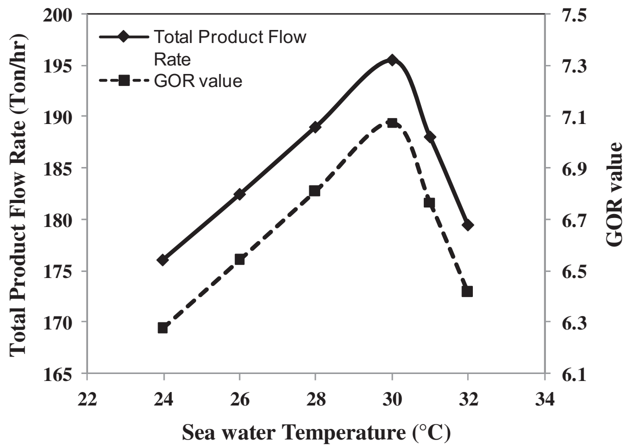


Fig. 8. Variation of total product flow rate and GOR value versus the seawater temperature.

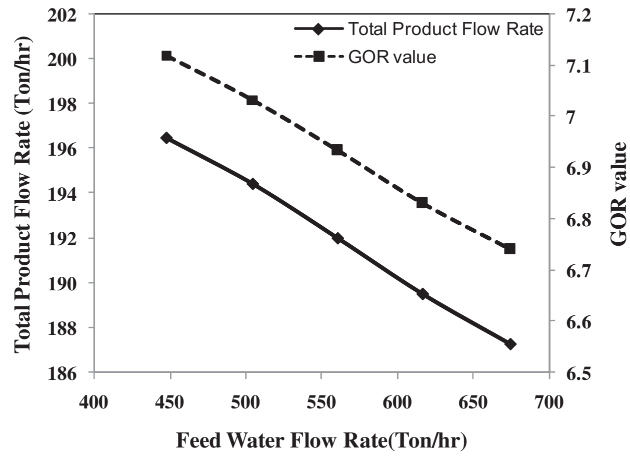


Fig. 11. Variation of total product flow rate and GOR value versus feed flow rate.

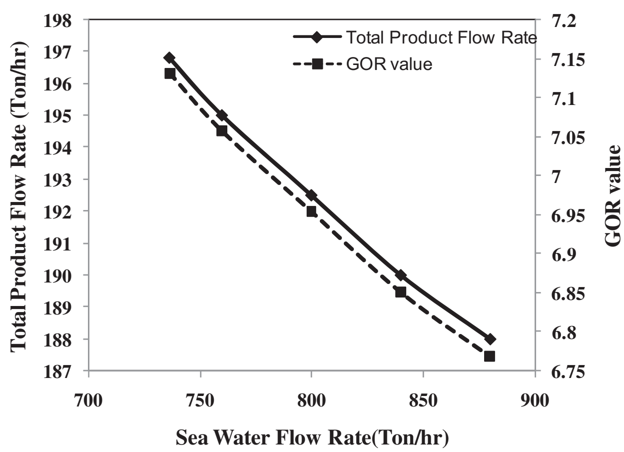


Fig. 9. Variation of total product flow rate and GOR value versus the seawater flow rate.

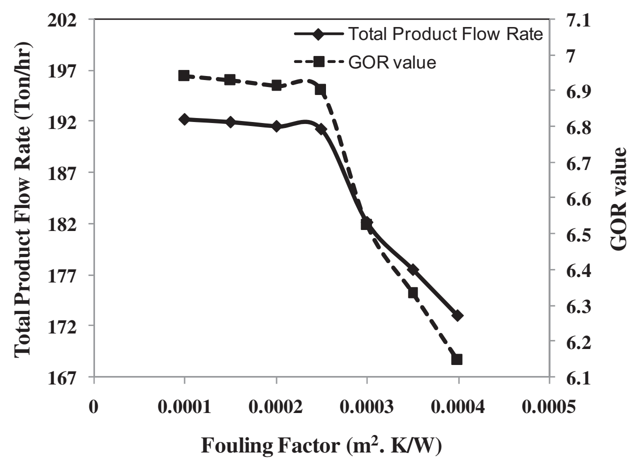


Fig. 12. Variation of total product flow rate and GOR value as a function of scale formation in the first effect.

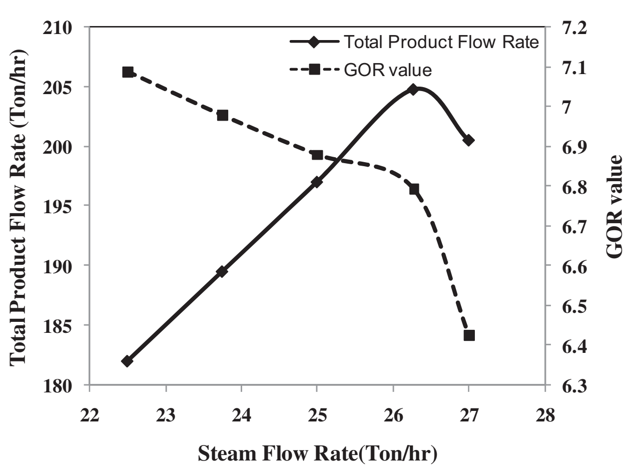


Fig. 10. Variation of total product flow rate and GOR value versus steam flow rate.

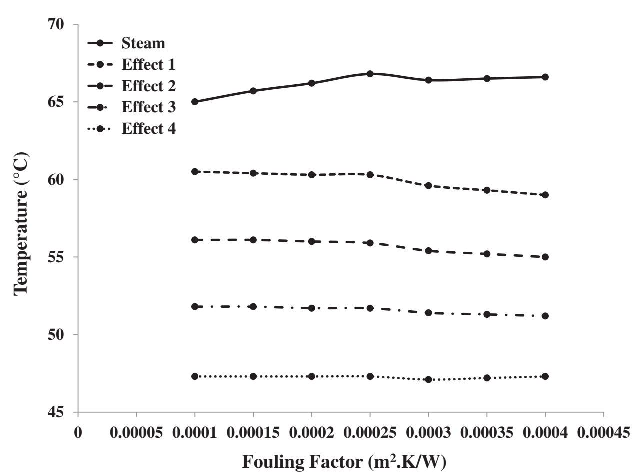
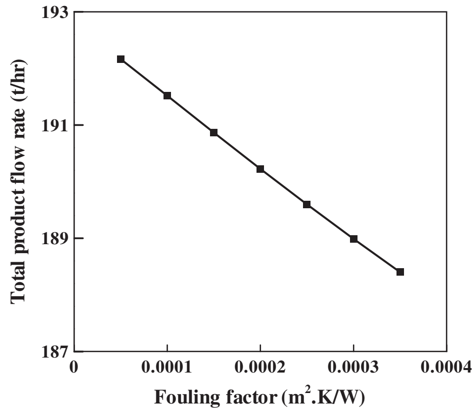


Fig. 13. Variation of effects temperature as a function of scale formation.



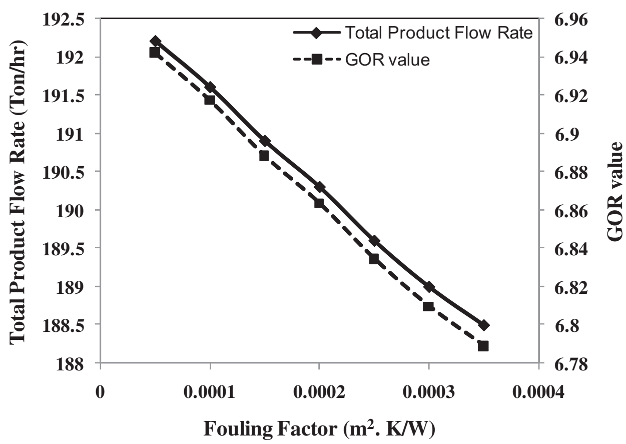
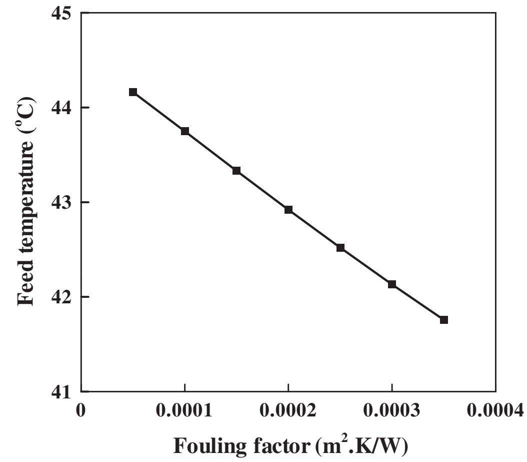


Fig. 14. Variation of total product flow rate as a function of scale formation in condenser.



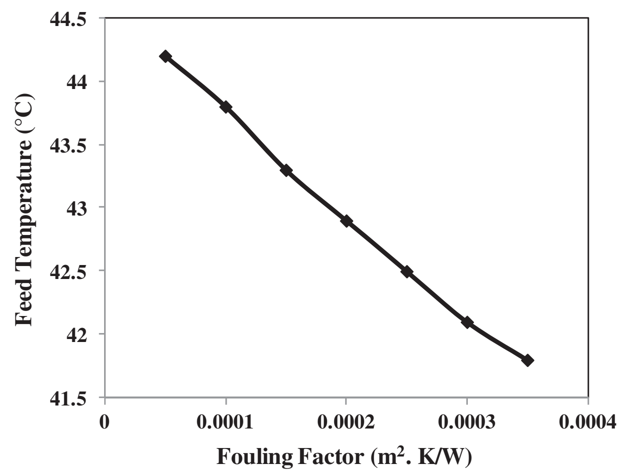


Fig. 15. Variation of feed temperature as a function of scale formation in condenser.

of product flow by increasing feed water flow rate. It is clear that when feed flow rate is increased, the sensible heat which is needed to heat the feed to the saturation temperature is increased and therefore a smaller portion of the total heat load is allocated for evaporation of feed flow and thus product decreases. This interpretation can better be realized by noticing to the energy balance equations is written in the mathematical and modeling section. It is obvious that the reduction in the total net production flow rate is equivalent to the reduction in the GOR value.

As it is clear, after a period of time, the production rate goes down because of scale formation around the tubes as it decreases the effective area of the system. The influence of the scale formation around the tubes of the first effect which is expressed in the form of fouling factor is shown in Fig. 12. The variation of the fouling factor in the range of 0.0001–0.0004 that is equal to the change of 3.6 μm in the scale thickness causes about 10% decrease in total production rate and the GOR value. If the fouling factor increases in such a way that the first effect pressure exceeds the MDP, the inlet vapor to the first effect decreases due to the drastic reduction in suction flow rate. The reduction in the total input steam flow rate causes lower production rate of steam in the unit, and thus intensive reduction in the amount of the total product, as is shown in Fig. 12. As a result of reduction in the total product, the GOR value decreases in the same manner.

According to Fig. 13, as mentioned before, the increase in scale thickness around the tubes due to change in thermocompressor operating condition and suction load changes the total vapor delivered to the effects and affects the temperature difference between them.

The influence of scaling inside the condenser tubes on total net production rate and the GOR value is investigated in Fig. 14. Scale inside the condenser tubes decreases the available heat transfer surface area of condenser. It causes the feed temperature to not to reach its designed temperature and also an increase in condenser temperature due

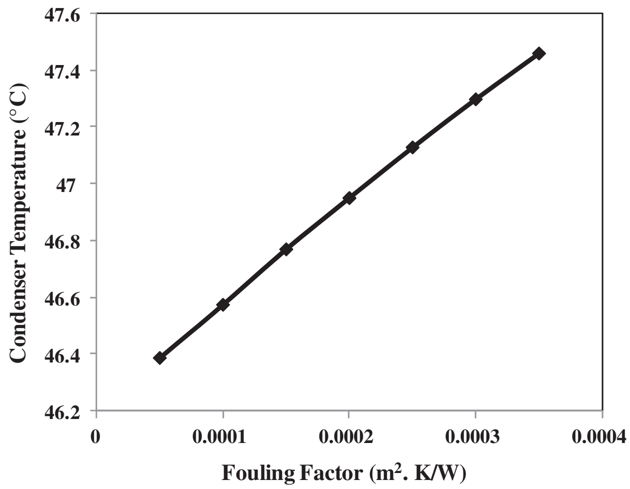


Fig. 16. Variation of condenser temperature as a function of scale formation in condenser.

to insufficient heat transfer surface area of the condenser which results in higher temperature difference across the condenser(Figs. 15 and 16). Therefore, the produced fresh water mass flow rate and GOR are reduced. Fig. 14 represents that changing the fouling factor of the condenser from 0.00005 to 0.00035, which is equal to 4.0 μm increase in the scale thickness, causes about 2% decrease in total production rate and the GOR value.

4. Conclusions

A multi-effect thermal vapor compression seawater desalination unit was simulated in operation. Comparison of the simulation code results and actual operating conditions showed that the code predicts the behavior of the system with good accuracy.

Finally, the following points can be remarked as conclusion;

- The produced fresh water mass flow rate and the GOR value increases 11% with about 25% rise in the seawater temperature.
- 10% increase in seawater flow rate causes about 2.4% reduction in total product rate and the GOR value.
- Delivering 16.5% more steam to the first effect in constant inlet seawater flow rate causes 12.5% rise in total product flow rate, but 4.2% reduction in the GOR value.
- 20% increase in feed flow rate leads to 2.4% decrease in total product flow rate and the GOR value.
- 3.6 μm increase in the scale thickness of the first effect causes about 10% decrease in total production rate and the GOR value.
- 4.0 μm increase in the scale thickness of condenser causes about 2% decrease in total production rate and the GOR value.
- In the case that condenser temperature is too high due to the high temperature of inlet seawater or high flow rate of motive steam, thermocompressor cannot work in a stable condition. In this situation, the suction load decreases, therefore the fresh water production rate decreases as well.

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Nomenclature

A	—	heat transfer area (m^2)
B	—	brine flow rate (kg/s)
BPE	—	boiling point elevation (K)
C	—	condensate flow rate (kg/s)
Cp	—	heat capacity (J/kg K)
F	—	feed flow rate (kg/s)
L	—	latent heat (kJ/kg)
M	—	water flow rate (kg/s)
Ma	—	Mach number
T	—	temperature (K)
U	—	overall heat transfer coefficient (W/m^2 K)
V	—	vapor flow rate (kg/s)
X	—	salinity (g/L)

Subscripts

B	—	brine
C	—	condensate
cond	—	condenser
CW	—	cooling water
Dis	—	discharge steam from thermocompressor
e	—	effect
F	—	feed
Nthc	—	the number of the effect where suction of thermocompressor placed on
n	—	effect number
n_t	—	last effect
P	—	product
Suc	—	suction flow to thermocompressor
st	—	steam
t	—	total
v	—	vapor

Greek

γ	—	specific heat ratio

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