FUNDAMENTALS OF MULTIPLE EFFECT EVAPORATION

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Contents

Summary

In this section the basics of designing single- and multieffect boiling (MEB) desalination systems are developed. The flow sheet of the conventional multieffect system as well as new developments, are presented. The principles of single- and multieffects VC desalting systems, both mechanical and thermal, are also presented. This section can serve also as an introduction to the MSF desalting system.

Part I: Multieffect Distillation

1. Introduction

Distillation is the oldest but still the most reliable method used to desalt seawater. In this method, water vapor generated from the seawater due to heat addition (thermal energy) is directed to a condenser where it condenses by giving up its latent heat to the incoming cold seawater. Distillation has been largely improved in the last few decades for (a) better usage of the consumed thermal energy, (b) decreasing the cost of the product

water, (c) overcoming design problems associated with the old methods, and (d) improving reliability.

2. Single-effect Submerged Tube System

A single-effect submerged tube system, which is shown in Figure 1, was the first and simplest method used to desalt seawater in large quantities. In this method, the evaporator has one or two horizontal (or vertical) tube bundles representing the heating surface and contained in the lower part of a shell. The tubes in the shell are covered completely (or submerged) with the seawater to be desalted. There is enough space in the shell over the submerged tubes to give a reasonably low released vapor velocity in order to separate the carried-over water droplets and to house demisters. The demisters are used to separate the fine droplets of seawater carried with the vapor.
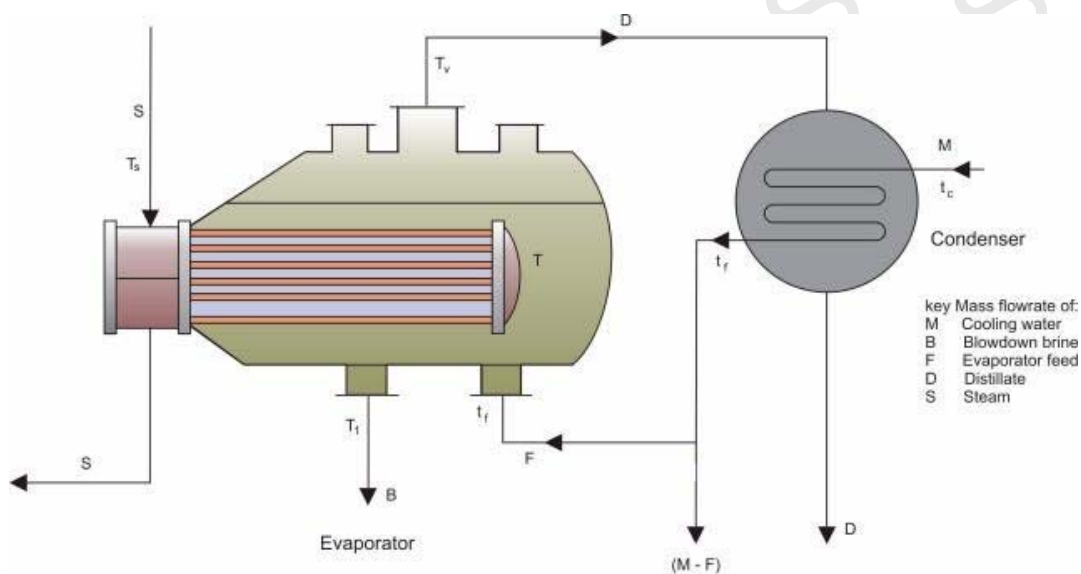

Figure 1. Single-effect and condenser arrangement.

In the evaporator, the seawater fed in is heated to its boiling temperature and is partly evaporated by boiling. Heating is performed by passing steam inside the tube bundle. The heating steam loses its latent heat and condenses before it flows out of the tube bundle. The partial evaporation of the seawater occurs by a nucleate pool boiling process. In this process, the salty solution has a limited movement around the heating surface under the influence of convection current and continuous rise of the vapor bubbles. The limited movement of water gives a low heat transfer coefficient on the boiling side and causes scale accumulation in the hot regions (tubes). The scales formed decrease the overall heat transfer coefficients substantially and these scales should be removed periodically. By using this type of arrangement, 1 kg of steam can evaporate, at most, 0.8 kg of vapor from 2 kg of the seawater supplied. The remaining 1.2 kg of brine has a higher salt concentration than the feed and should be removed (blowdown). The released 0.8 kg of vapor is condensed in the condenser by the use of cooling seawater. Part of this seawater which is partially pre-heated in the condenser, is used as a feed to the evaporator. The remaining cooling seawater is discharged back to the sea.

The temperature distribution of this arrangement is shown in Figure 2.

The disadvantages of scale formation and the large amount of the energy consumed to produce 1 kg of product water practically rule out the application of the submerged tube single-effect desalination method for large capacity production. However, it is very useful to make an analysis of this system since it represents the basis for the developed multieffect systems.
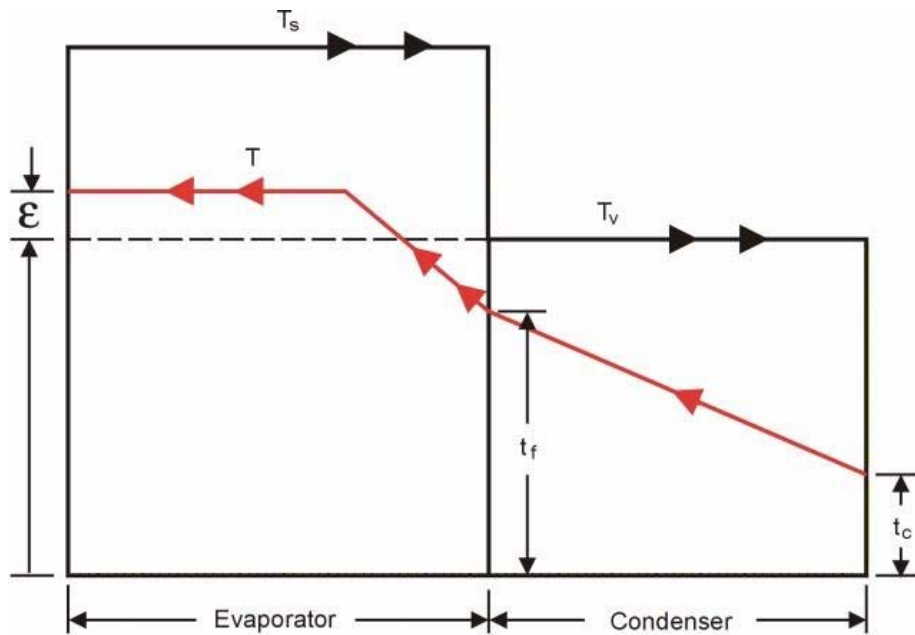

Figure 2. Temperature profile in single-effect and condenser arrangement.

2.1. Performance Ratio of the Single-effect Distillation System

Consider the single-effect distillation (SED) system shown in Figure 1. Assume that S is the mass flow rate of the heating steam to the evaporator in kg s^{-1} and that the steam loses its latent heat L_o due to its condensation inside the tubes at temperature T_s. This amount of heat (SL_o) is used to heat the incoming seawater feed at mass flow rate F from its feed temperature t_f to its boiling temperature T_1 and evaporate part of it at a rate of D, i.e.

$$SL_o = FC_b\,(T_1 - t_f) + DL_1 \tag{1}$$

where C_b is the brine specific heat in kJ kg^{-1} K^{-1}, and L_1 is the latent heat required to evaporate 1 kg of water at saturation temperature T_{v1}. It is noticed here that the saturation temperature of the released vapor T_{v1} is less than the boiling temperature T_1 by the boiling point elevation ε, i.e.

$$T_{v1} = T_1 - \varepsilon \tag{2}$$

The boiling point elevation ε is a function of the brine salinity, s (in p.p.m.), and the

temperature T (in K).

It can be expressed by (in K) (Khan 1986)

$$\varepsilon = s(b + cs)$$

where

$$b = (6.71 + 6.34 \times 10^{-2} T + 9.74 \times 10^{-5} T^2) \times 10^{-3}$$

and

$$c = (22.238 + 9.59 \times 10^{-3} T + 9.59 \times 10^{-3} T + 9.42 \times 10^{-5} T^2) \times 10^{-5}$$

The mass flow rate of the brine blowdown B is equal to the difference between the feed and vapor flow rates, i.e.

$$B = F - D \tag{3}$$

The generated vapor enters the condenser where it condenses and becomes the product water. This vapor loses its latent heat to the cooling seawater entering the condenser at mass flow rate M and temperature t_c and leaves at temperature t_f. Part of the leaving cooling seawater represents the feed water to the evaporator F and the remaining amount $(M - F)$ is rejected back to the sea. An energy balance to the condenser gives

$$MC_c(t_f - t_c) = DL_1 \tag{4}$$

An energy balance for the whole system gives

$$SL_o = BC_b(T_1 - t_c) + DC_d(T_v - t_c) + (M - F)C_c(t_f - t_c) \tag{5}$$

where C_b, C_d, and C_c are the specific heats for the brine, distillate and cooling seawater, respectively. For simplicity consider an average amount C for all the specific heats, i.e.

$$C_d \cong C_b \cong C_c \cong C$$

In addition, consider an average value of the latent heat L, i.e.

$$L \cong L_1 \cong L_o$$

By these simplifications, Eq. (5) reduces to

$$SL = BC(T_1 - t_c) + DC(T_v - t_c) + MC(t_f - t_c) - (B + D)C(t_f - t_c)$$
$$= BC(T_1 - t_f) + DC(T_v - t_f) + DL$$

and

$$\frac{SL}{D} = L + \left(1 + \frac{B}{D}\right) C(T_v - t_f) + \frac{B}{D} \, C\varepsilon \tag{6}$$

The value of *SL/D* in Eq. (6) represents the thermal energy required to desalt 1 kg of water. The performance ratio (PR) is defined by

$$PR = \frac{D}{\text{energy consumed} \, / \, 2330} \tag{7}$$

where 2330 kJ kg^{-1} is the latent heat of steam condensed at an average temperature. By neglecting the energy consumed in pumping the feed, condensate, etc., with respect to the thermal energy, the performance ratio PR can be represented by

$$PR = \frac{2330}{L + C(T_v - t_f) \left(1 + \frac{B}{D}\right) + \frac{BC}{D} \, \varepsilon} \tag{8}$$

Equation (8) shows that the PR is close to or less than the amount of product water obtained for each kilogram of heating steam, *D/S*, if the latent heat of that steam is equal to or more than 2330 kJ kg^{-1}. In addition, Eq. (8) shows that the performance ratio can be improved by the following.

(a) Decreasing the temperature difference $(T_v - t_f)$; this requires an increase in the heat transfer surface area of the condenser. In practice, the value of $(T_v - t_f)$ cannot be less than 5°C. However, increasing t_f has very little effect on the PR since the consumed thermal energy for each kilogram of the product water given by

$$L + C(T_v - t_f)\left(1 + \frac{B}{D}\right) + \frac{B}{D} \, C\varepsilon$$

would change from 2309 to 2351 kJ kg^{-1} if the term $(T_v - t_f)$ increases from 5°C to 10°C. This means that the PR would increase by less than 1.8 per cent by decreasing $(T_v - t_f)$ by 5°C.

(b) The value of *B/D* is determined by the salt concentration of the brine in the evaporator to avoid fast scale formation. The maximum salt concentration is usually set at $C_{sb} = 70\ 000$ p.p.m. (70 g kg^{-1}). So if the feed salt concentration is $C_{sf} = 35\ 000$ p.p.m., then *B/D* is approximately 1 and *F/D* is 2. This can be checked by the mass balance for the salt, i.e.

$$FC_{sf} = BC_{sb} + DC_{sd} \tag{9}$$

(c) When considering the maximum value of the performance ratio when steam condenses at atmospheric pressure and $B/D = 1$, $\varepsilon = 1.2$°C, $C = 4.0$ kJ kg^{-1} K^{-1},

$L = 2257$ kJ kg^{-1}, and $(T_v - t_f) = 5°C$ then

$$PR = \frac{2330}{2257 + 4.0 \times 5 \times 2 + 1 \times 4.0 \times 1.2} = 1.0$$

In practice PR varies from 0.8 to 0.9.

(d) From a theoretical point of view only, the *PR* can be increased by increasing the solution temperature to decrease the latent heat *L* (the value of *L* decreases with the increase in the temperature until it reaches zero at the critical temperature 374.14°C). If boiling occurs at 370°C at which ε reaches 30°C and $C = 12.5$ kJ kg^{-1}°C^{-1} and $L = 441.6$ kJ kg^{-1} then,

$$PR = \frac{2330}{441.6 + 2 \times 12.5 \times 5 + 12.5 \times 30} = 2.5$$

by taking $T_v - t_f = 5°C$.

This example shows that, even theoretically, the PR cannot have a value higher than 2.5. It is necessary to indicate here that there is an acceptable upper limit for the boiling temperature which depends on the method of water treatment before it enters the evaporator. This maximum boiling temperature is 90°C for polyphosphate additives treatment and 120°C for acid treatment. Consequently, the PR is always less than unity for single-effect distillation systems.

2.2. Heat Transfer Area of a Single-effect desalting (SED) System

To determine the heat transfer surface area of the evaporator and the condenser of the SED system, the following equation will be used for each one of them,

$$Q = UA(\Delta t_e) \tag{10}$$

where *Q* is the heat exchanger (evaporator or condenser) thermal load in kW, *U* is the overall heat transfer coefficient kW m^{-2}, *A* is the heat transfer surface area in m^2 and (Δt_e) is the effective temperature difference across the heat transfer surface (usually taken as the logarithmic mean temperature difference).

2.2.1. Evaporator Heat Transfer Area: A_e

The evaporator thermal load is *SL*. The effective temperature across the heat transfer surface is equal to $(T_s - T_1)$ after neglecting the feedwater subcooling. The overall heat transfer coefficient U_e is not a simple function, since it depends on the nucleate pool boiling heat transfer coefficient on the outer surface of the tubes, the condensing heat transfer coefficient inside the tubes, the thermal conductivity of the tube material, the thickness of the tube, and the thermal resistance of the scale formed on the heat transfer surface. The evaluation of U_e will be discussed later. By using Eq. (10), the heat transfer surfaces area A_e can be given as

$$SL = U_e \cdot A_e (T_S - T_1)$$

and the specific heat transfer area (area/unit mass flow rate of the product) A_e/D (m^2 kg^{-1} s^{-1}) is

$$\frac{A_e}{D} = \frac{SL}{D} \frac{1}{U_e(T_s - T_v - \varepsilon)}$$
$$\cong \frac{1}{PR} \frac{L}{U_e(T_s - T_v - \varepsilon)} \qquad (11)$$

2.2.2. Condenser Heat Transfer Surface Area: A_c

$$DL = MC(t_f - t_c) = U_c A_c \, (\text{LMTD})$$

where U_c is the overall heat transfer coefficient and LMTD is the logarithmic temperature difference and is equal to

$$\text{LMTD} = \frac{(T_v - t_c) - (T_v - t_f)}{\ln \dfrac{T_v - t_c}{T_v - t_f}} = \frac{t_f - t_c}{\ln \dfrac{T_v - t_c}{T_v - t_f}}$$

and, consequently,

$$\frac{A_c}{D} = \frac{L}{U_c} \frac{1}{(t_f - t_c)} \ln \frac{T_v - t_c}{T_v - t_f} \qquad (12)$$

It is noticed here that the increase in the condenser heat transfer area increases the effectiveness of the condenser k, defined by

$$k = \frac{t_f - t_c}{T_v - t_c} \qquad (13)$$

The condenser effectiveness k represents the increase in the cooling water temperature $(t_f - t_c)$ in the condenser to the maximum possible increase $(T_v - t_c)$ when the condenser surface area tends to be infinity. The value of A_c/D can be expressed in terms of the condenser effectiveness k as follows:

$$\frac{A_c}{D} = \frac{L}{U_c k(T_v - t_c)} \ln \frac{T_v - t_c}{(T_v - t_c) - (t_f - t_c)} = \frac{L}{U_c k(T_v - t_c)} \ln \frac{1}{1 - k} \qquad (14)$$

2.2.3. The Optimum Total Area of the Condenser and Evaporator: A_t

The total heat transfer surface area of both evaporator and condenser $A_t = (A_c + A_e)$ can be expressed by the use of Eqs (11) and (12), i.e.

$$\frac{A_t}{D} = \frac{L}{(\text{PR})U_e(T_s - T_v - \varepsilon)} + \frac{L}{U_c(t_f - t_c)} \ln \frac{T_v - t_c}{T_v - t_f}$$

(15)

Some of the values in Eq. (15) are fixed and cannot be changed in the design. For example, t_c is fixed by the available cooling water. The steam temperature T_s is usually chosen a few degrees above the maximum permissible temperature of the brine, called the top brine temperature (TBT). Other values can be changed (e.g. T_v). The increase in the evaporator heat transfer area decreases the temperature difference across the heat transfer surface (T_s - ε - T_v) and consequently increases T_v. This increase in T_v increases the LMTD in the condenser and, consequently, decreases the required condenser heat transfer surface area. This means that the increase in T_v increases the evaporator surface area and decreases the condenser surface area and vice versa. This necessitates the choice of an optimum value of T_v that gives minimum total condenser and evaporator heat transfer areas. This can be obtained by differentiating the function A_t with respect to T_v and equating the differentiation with zero to obtain the optimum T_v. This can be done by assuming that the other parameters such as T_s, t_c, t_f, and U have constant values. It can be shown that the optimum value of T_v when PR $\cong$ 1 and $U_c = U_e = U$ is

$$(T_v)_{\text{opt}} = \frac{(T_s - \varepsilon)^2 - t_f t_c}{2(T_s - \varepsilon) - (t_f + t_c)}$$

(15a)

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Bibliography and Suggestions for further study

Akili D. Khawaji, Ibrahim K. Kutubkhanah, Jong-Mihn Wie, (2008), *Advances in seawater desalination technologies*, Desalination **221**, Elsevier, pp. 47-69.

Ali Al-Odwani, Essam E.F. El-Sayed, Mohamed Al Tabtabaei, Mohamed Safar, (2006), *Corrosion resistance and performance of copper-nickel and titanium alloys in MSF distillation plants*, Desalination **201**, Elsevier, pp. 46-57.

C. Sommariva, V.S.N. Syambabu, (2001), *Increase in water production in UAE*, Desalination **138**, Elsevier, pp. 173-179.

Cane D et al. (1970) Conceptual Design Study of 200 MGD, VTE/MSF Desalination Plant and Prototype Module, PB 233-299 NTIS. US Department of Commerce.

Darwish M A (1980) Short Course Notes on Desalination. Jeddah: King Abdul Aziz University Press.

Darwish M A (1983a) A hybrid system for single purpose desalting plants, more water production with less energy (The First Saudi Engineering Conference, King Abdulaziz University, Jeddah, 1983).

Darwish M A (1983b) Better arrangement for multi effect boiling process (The First Saudi Engineering Conference, King Abdulaziz University, Jeddah, 1983).

Duckler A E et al. (1969) OSIN, R&D Rep No. 487.

Egorov A P et al. (1973) The results of the starting tests of a 10-effects once-through desalting plant with forced circulation (Fourth International Symposium on Fresh Water From the Sea) Vol. 1, pp. 245-252.

El Ramley N A et al. (1977) Desalting Plants Inventory Reports, No. 6. US Department of Interior.

Haves R I (1970) Development of vertical tube multi effect process (Third Symposium on Fresh Water from the Sea, Dubrovnik) Vol. 1, pp. 275-289.

Hisham El-Dessouky, S. Bingulac, (1995), *A Stage-by-Stage Algorithm for Solving the Steady State Model of Multi-Stage Flash Desalination Plants*, IDA 141, Volume **IV**, pp. 251-27.

Joachim Gebel, Süleyman Yüce, (2008), *A new approach to meet the growing demand of professional training for the operating and management staff of desalination plants,* Desalination **220**, Elsevier, pp. 150-164.

Khan A H (1986) Desalination Processes and Multistage Flash Distillation Practice. Amsterdam, Elsevier.

Lagana V and Tiraboschi (1982) Snamprogetti Seawater Desalination Process and its Control Devices. Automatic Control in Desalination and the Oil Industry (ed. El Hares and Dali). Pergamon Press.

Lucas M and Tabourier B (1985) Mechanical vapor compression process applied to seawater desalination. Desalination 52, 123-133.

M.A. Darwish , Iain McGregor, (2005), *Five days' Intensive Course on - Thermal Desalination Processes – Fundamentals and Practice*, MEDRC & Water Research Center Sultan Qaboos University, Oman

M.A. Darwish, Ammar Alsairafi, (2004), *Technical comparison between TVC/MED and MSF*, Desalination **170**, Elsevier, pp. 223-239.

M.A. Darwish, Hassan K. Abdulrahim, (2008), *Feed water arrangements in a multi-effect desalting system*, Desalination **228**, Elsevier, pp. 30-54.

M.A. Darwish, N. Al-Najem, N. Lior, (2006), *Towards Sustainable Energy in Seawater Desalting in the Gulf Area*, Tenth International Water Technology Conference, Alexandria, Egypt, pp. 655-684.

M.A. Darwish, S. Alotaibi, S. Alfahad, (2008), *On the reduction energy and its cost in Kuwait*, Desalination **220**, Elsevier, pp. 483-495.

Michaels T (1992) Recent achievement of low temperature multiple effect desalination in western areas of Abu Dhabi, UAE (Desal.'92 Proc. Arabian Gulf Regional water desalination symposium, Al-Ain, UAE).

Mohamed A. Dawoud, (2005), *The role of desalination in augmentation of water supply in GCC countries*, Desalination **186**, Elsevier, pp. 187-198.

Mohamed Al-bahou, Zamzam Al-Rakaf, Hassan Zaki, Hisham Ettouney, (2007), *Desalination experience in Kuwait*, Desalination **204**, Elsevier, pp. 403-415.

Nabil M. Abdel-Jabbar, Hazim Mohameed Qiblawey, Farouq S. Mjalli, Hisham Ettouney, (2007), *Simulation of large capacity MSF brine circulation plants,* Desalination **204**, Elsevier, pp. 501-514.

Nafiz Kahraman, Yunus A. Cengel, (2005), *Exergy análisis of a MSF distillation plant*, Energy Conversion and Management **46**, Elsevier, pp. 2625-2636.

Novikov E P et al. (1969) Nuclear industrial desalination plant with fast neutron reactor at shevchenko. Desalination 6, 349-369.

Power R (1994) Steam Jet Ejectors for the Process Industries. New York: McGraw-Hill.

R.K. Kamali, A. Abbassi, S.A. Sadough Vanini, (2009), *A simulation model and parametric study of MED-TVC process*, Desalination **235**, Elsevier, pp. 340-351.

Roberton Borsani, Silvio Rebagliati (2005), *Fundamentals and costing of MSF desalination plants and comparison with other technologies*, Desalination **182**, Elsevier, pp. 29-37.

Satone H et al. (1977) Experience on VTE distillation plants in Japan (Proceedings of the International Congress on Desalination and Water Reuse).

Takada M and Drake J C (1983) Application of improved high performance evaporator. Desalination 45, 3-12.

Y.M. El-Sayed, (2001), *Designing desalination systems for higher productivity*, Desalination **134**, Elsevier, pp. 129-158