Chapter 6

Multi-Stage Flash Desalination

Objectives

The objective of this chapter is to analyze and evaluate the performance of the multistage flash desalination system. This is made through discussion of the following:

- Process developments
- Standard features of the brine circulation MSF process, which is the most common process
- Modeling and analysis of the single stage flashing system, the once through multistage system, and the brine circulation multistage system.
- Features and performance of novel configurations, which include MSF with brine mixing and MSF with thermal vapor compression.

6.1 Developments in MSF

The MSF process is an innovative concept, where vapor formation takes place within the liquid bulk instead of the surface of hot tubes. The hot brine is allowed to flow freely and flash in a series of chambers; this feature keeps the hot and concentrated brine from the inside or outside surfaces of heating tubes. This is a major advantage over the original and simple concept of thermal evaporation, where submerged tubes of heating steam are used to perform fresh water evaporation. The performance of such configurations was far from satisfactory, where salt scale is formed progressively on the outside surface of the tubes. The formed scale has a low thermal conductivity and acts as an insulating layer between the heating steam and the boiling seawater. Consequently, the evaporation rate is drastically reduced and cleaning becomes essential to restore the process efficiency. Earlier designs were plagued with such problems, where operation lasted for less than two weeks and shutdown and cleaning lasted for more than four weeks.

The brine circulation MSF process is the industry standard. The process elements are illustrated in Fig. 1, where the flashing stages are divided among the heat recovery and heat rejection sections. The system is driven by heating steam, which increases the temperature of the brine recycle or feed seawater to the desired value in the brine heater. The hot brine flashes in the consecutive stages, where the brine recycle or the feed seawater flowing inside the condenser tubes recovers the latent heat of the formed vapor. In the heat rejection section of brine circulation system, the excess heat added to the system by the heating steam is rejected to the environment by the cooling seawater stream. In the MSF process the tubes are arrange in a long or cross tube configuration. The cross tube configuration is the original system design and its units have production capacities in the range of 27,276 – 32,731 m^3/d. In this configuration, the tubes are aligned along the width of the flashing chambers and are connected via external water boxes. The long tube arrangement is geared towards larger production volume with current unit capacities up to 57,734 m^3/d. In this system, a single bundle of tubes span the whole length of a limited number of flashing stages. This eliminates the water boxes found in the cross flow system and allows for the increase of the flow rate per chamber width, which reduces the required chamber width.

The brine circulation process has many attractive features, which makes it distinguishable among other desalination configurations. Since establishment in the late fifties, a huge field experience has been accumulating in process technology, design procedure, construction practice, and operation. This has resulted in development of simple and reliable operational procedures. In addition, the development addressed and solved various operational problems, which include scale formation, foaming, fouling, and corrosion. Gained experience in operation and design of the MSF plants has lead to use inexpensive construction material capable of standing harsh conditions at high salinity. The MSF system does not include moving parts, other than conventional pumps. Construction of the MSF plants is simple and contains a small number of connection tubes, which limits leakage problems and simplifies maintenance works. In the light of the above, we strongly believe that the MSF system will remain the main desalination process, especially in the Middle East. This is due to the following facts:

- The conservative nature of the desalination owner.
- The product is a strategic life-supporting element.
- Extensive experience in construction and operation.
- Process reliability.
- Limited experience, small database, and unknown risks with new technologies.

Since inception, several developments have been achieved in system design and operation. These developments include the following:

- Increasing the unit capacity from 454.6 m^3/d to a current conventional capacity of 27,276 – 32,731 m^3/d. Recently, larger units with a capacity of 57,734 m^3/d are commissioned. Each capacity doubling is associated with 24% reduction in unit product cost.
- Decreasing the specific power consumption from 25-70 kW/m^3 in 1955 to 4-10 kW/m^3.
- Plant operation is drastically improved with introduction of more efficient antiscalent and corrosion control chemicals and use of construction materials capable of withstanding the harsh operating conditions. Earlier, operation was plagued by excessive scaling and damaging corrosion resulting in limited production time followed by prolonged cleaning procedures, Temperley (1995). Currently, conventional MSF units operate continuously for more than 2

years, without the need for complete shut down. This is achieved in part by adoption of on-line acid or sponge ball cleaning.

- Other process developments include use of smaller specific equipment size, reduction of the stand-by units, and minimizing and simplifying control units. For example, more efficient interstage devices are developed to withstand erosion effects and increase the brine flow rate per chamber width.
- Development of more accurate and advanced models capable of various tasks, which includes process design, rating, evaluation of process economics, optimization, process dynamics, and system control, Helal et al. (1986), Darwish (1991), El-Dessouky et al. (1995), Darwish and El-Dessouky (1996), El-Dessouky and Bingulac (1996), El-Dessouky et al. (1998).

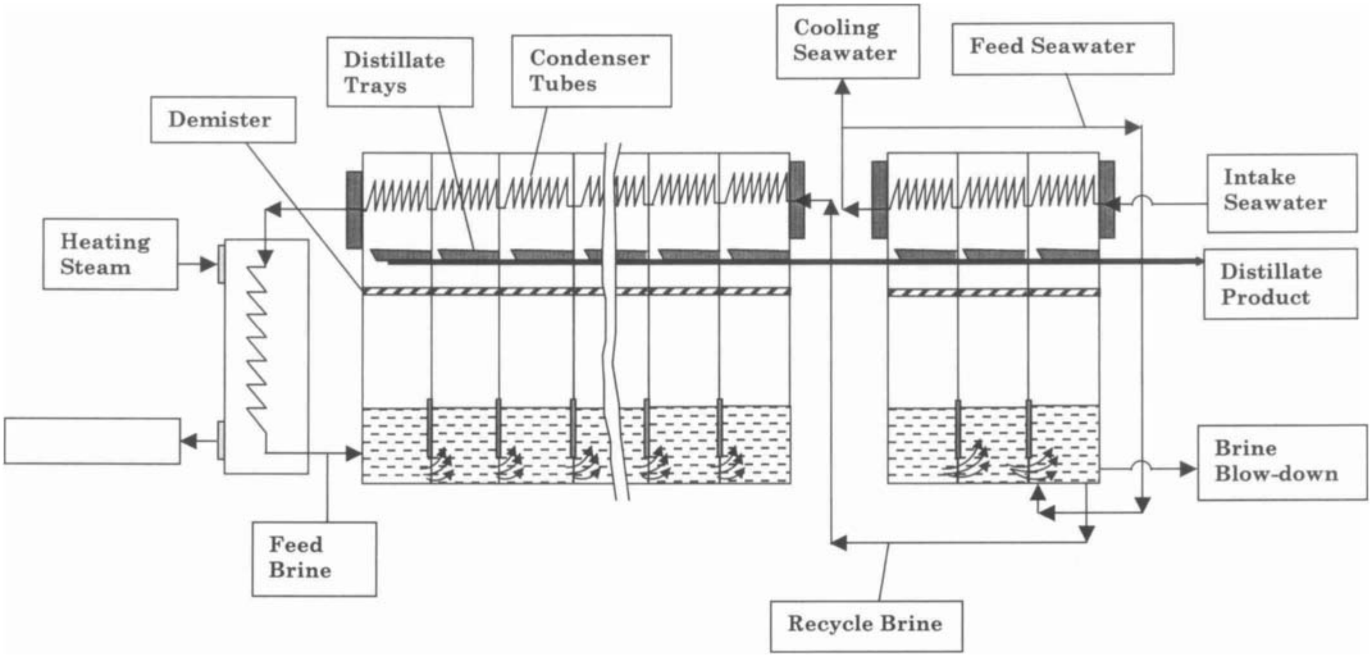

Fig. 1. Brine circulation multi-stage flash desalination process

6.2 *MSF Flashing Stage*

For conventional MSF systems with capacities of 27,000 up to 32,000 m^3/d, the flashing stage has dimensions of 18x4x3 m in width, height, and length. A schematic of the MSF flashing stage is shown in Fig. 2 and it includes the following:

- A large brine pool with similar width and length of the flashing stage and a depth of 0.2-0.5 m.
- A brine transfer device between the stages is designed to seal the vapor space between the stages and to enhance turbulence and mixing of the inlet brine stream. This device promotes flashing; controls formation of vapor bubbles, their growth, and subsequent release.
- The demister is formed of wire mesh layers and the supporting system. The demister function is to remove the entrained brine droplets from the flashed off vapor. This is essential to prevent increase in the salinity of product water or scale formation on the outer surface of the condenser tubes.
- The tube bundle of the condenser/preheater tubes, where the flashed off vapor condenses on the outer surface of the tubes. The released latent heat of condensation results in heating of the brine recycle stream flowing inside the tubes. This energy recovery is essential to maintain high system performance.
- Distillate tray, where the condensed distillate product is collected and cascade through the stages. The distillate product is withdrawn from the tray of the last stage.
- Water boxes at both ends of the tube bundle to transfer the brine recycle stream between adjacent stages.
- Connections for venting system, which removes non-condensable gases (O_2, N_2, and CO_2), which are dissolved in the feed seawater, even after deaeration. Also, CO_2 can be generated during decomposition of the bicarbonate compounds in the high temperature stages. Another important source for the non-condensable gases is air in-leakage from the ambient surroundings into the flashing stages operating at temperatures below 100 °C, which correspond to vacuum conditions.
- Instrumentation, which includes thermocouples, level sensor, and conductivity meter, are placed in the last and first flashing stages. The measured data from these stages are adopted by the control system of the process. Accordingly and subject to disturbances in the system parameters, i.e., feed seawater temperature, increase in fouling thermal resistance, available steam, etc., adjustments are made in the controllers to restore the desired operating conditions. The magnitude of these adjustments depends on the measurements made in the last and first stages.

The MSF process operates over a temperature range of 110-30 °C. This implies the majority of the flashing stages operate at a temperature below 100 °C or vacuum conditions. Therefore, all flashing stages are designed to withstand full vacuum. However, the bottom of the flashing stages is exposed to the hydrostatic pressure of the brine pool. Therefore, the system is designed to withstand a maximum pressure of 2 bar.

The walls, ceilings, and partitions of the flashing stages are constructed of carbon steel with stainless steel or epoxy cladding. Stainless steel cladding is used in locations where higher erosion or corrosion conditions can be found. All stages are reinforced with a stainless steel structure and heavily insulated to minimize heat losses.

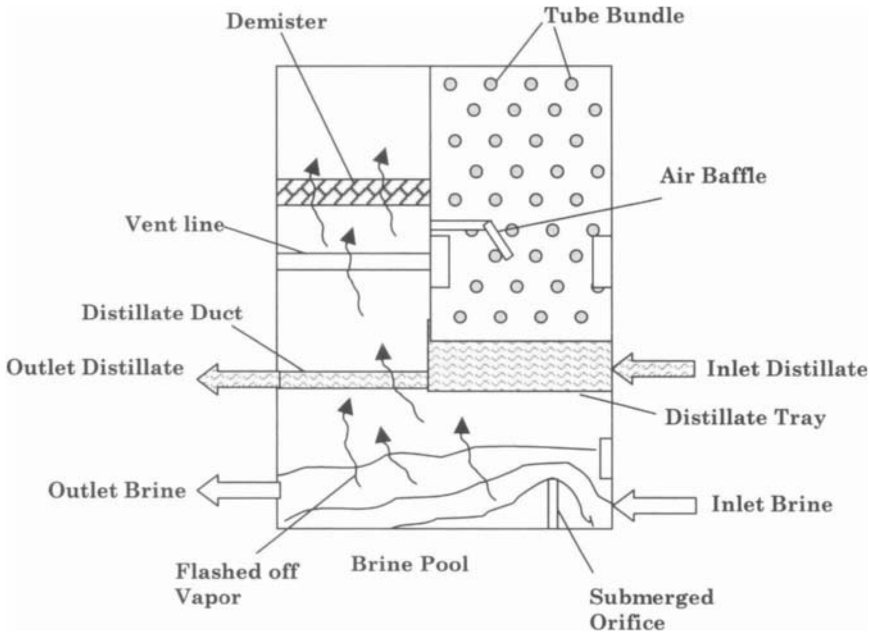

Fig. 2. MSF flashing stage

6.2.1 Condenser/Preheater Tubes

The condenser/preheater tubes are used to recover and reject heat in the MSF process. In the stages forming the heat recovery section, heat is recovered from the condensation of the flashed off vapor to heat the brine recycle stream flowing on the tube side. This heat recovery is essential in obtaining a high thermal performance ratio. In the heat rejection stages, the feed and cooling seawater are heated by absorbing the latent heat of the condensing flashed off vapor. Accordingly, the feed seawater is heated to a temperature equal to the temperature of the brine in the last flashing stage. This is necessary to prevent thermal shock upon mixing of the heated feed stream in the brine pool of the last stage. A thermal shock would result in decomposition of the calcium bicarbonate and calcium carbonate precipitation, which known as the soft scale. The decomposition process is also associated with release of carbon dioxide gas, which would promote corrosion reaction and increases the load on vacuum ejectors. The function of the cooling seawater stream in the heat rejection section is to remove the excess heat added to the system in the brine heater.

The heat transfer area of the condenser tube is a major design feature that controls the temperature of the brine recycle entering the brine heat. This parameter has a strong effect on the system performance ratio. If the heat transfer area is smaller than the thermal load of the condensing vapor, the stage pressure will increase due to accumulation of the non-condensed vapors. This pressure increase will reduce the amount of flashed of vapor. Eventually, the system will reach a new steady state with lower flashing rates and smaller flow rate of the distillate product. Also, the temperature of the brine recycle entering the brine heater will become smaller, which will result in increase of the required amount of heating steam and reduce of the system performance ratio. Although, initial system design provides sufficient heat transfer within various stages; however, poor operation and increase in fouling resistance will reduce the heat transfer coefficient and will create conditions, where the overall heat transfer rate is lower than the design thermal load. However, tube blockage may result in a similar result.

6.2.2 Tube Materials

Table 1 shows properties of the characteristics of preheater/condenser tubes used in the MSF process. As is shown, material selection depends on the stage temperature. In this regard, Cu/Ni 70/30 is used in stages with temperatures higher than 80 ^{o}C. On the other hand, in stages with lower temperatures a number of materials can be used, which includes Cu/Ni 90/10, aluminum brass, high steel alloys, and titanium. The highest thermal conductivity among these materials is the Cu/Ni 90/10 with a value of 44×10^{-3} kW/m ^{o}C. On the other hand, titanium tube provides the highest erosion resistance and the lowest wall thickness. Aluminum bronze provides a cheaper material, however, its copper content dissolves in the seawater and has an adverse impact on the receiving water bodies. The same problem is also found in other types of copper based tubes. In this regard, titanium, although more expensive than the copper alloys, it does not dissolve in the seawater.

6.2.3 Tube Configuration

The first tube configuration in the MSF system is the long tube arrangement, Fig. 3. In the configuration, the tubes are aligned in the same direction as the brine flow. Depending, on the available tube length, a single tube can span more than one stage. The limit on the tube length is imposed manufacturing companies and associated technical difficulties in transportation and handling. On-site tube welding may prove to be useful in constructing the required tube length. In practice, available long tube configurations are limited a maximum length of 28 m, which encompass on average 8-10 flashing stages. Features of the long tube configuration include the following:

- Fouling, blocking, and scaling of a single tube have a strong impact on the system performance, since; loss of a single tube implies reduction of the heat transfer in 8-10 stages for the lost tube.
- In maintenance and cleaning, tube removal is not a simple task. Also, conventional mechanical cleaning, blasting, would require specially designed equipment.
- Expansion consideration for long tubes requires special consideration in stage design.
- Vapor leakage between stages is a serious problem that needs special consideration in design, installation, and during maintenance and cleaning procedures.
- The main advantage for the long tube configuration is the reduction of the tube pressure drop by a factor of 25-30%. This reduces the associated pumping power.
- The long tube configuration can be thought as the optimum choice for plants with capacities higher than 50,000 m^3/d.
- Long tube configuration eliminates the water boxes on both sides of the flashing chamber, which is found in the cross tube configuration.

The second configuration is the cross tube arrangement, where the tubes are arranged in perpendicular direction to the brine flow, Fig. 4. This is a common configuration and is found in most of the MSF plants. Huge field experience is accumulated over the years for the cross tube configuration and it includes design, installation, operation, maintenance, and replacement. Also, less technical experience is required for construction, maintenance, and removal than the long tube system. Tube expansion in this system does not represent a problem in design and construction. The main disadvantage of this system is the need for installing water boxes on both ends of the tubes to transfer the brine recycle or feed seawater between the stages, which will increases the process capital, pressure drop, and pumping power.

Table 1
Properties of tube materials used in the brine heater and the condenser tubes

Material	Temperature	Thermal Conductivity kW/m °C	Brine Velocity m/s	Wall Thickness mm
Cu/Ni 70/30 (66% Cu, 30% Ni, 2% Fe, and 2% Mn)	Above 80 °C	29×10^{-3}	2-4	1.2
Aluminum Brass (76% Cu, 22% Zn, and 12% Al)	Below 80 °C	32×10^{-3}	1.5-2.5	1.2
Titanium	Below 80 °C	16.5×10^{-3}	3-20	0.5
Cu/Ni 90/10	Below 80 °C	44×10^{-3}	2-4	1.2
High Steel Alloy	Below 80 °C	19.9×10^{-3}	3-10	0.7

Table 2
Summary of fouling resistance in m^2 °C/W on the tube side

	Design value	Test data	Actual operation
Heat recovery section	1.5×10^{-4}	1.04×10^{-4}– 2.18×10^{-4}	0.68×10^{-4}– 2.23×10^{-4}
Heat rejection section	1.77×10^{-4}	2.42×10^{-4}	2.51×10^{-4}
Brine Heater	3.01×10^{-4}	1.49×10^{-4}	1.52×10^{-4}

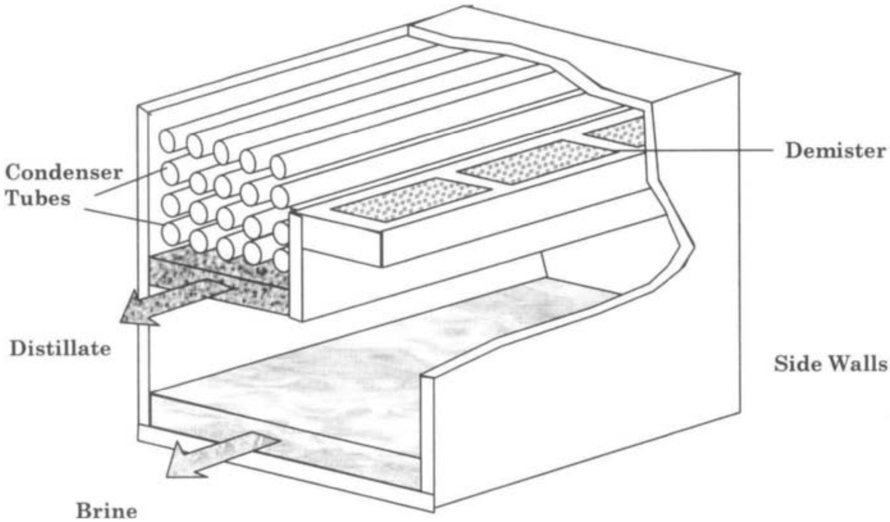

Fig. 3. Long Tube Configuration

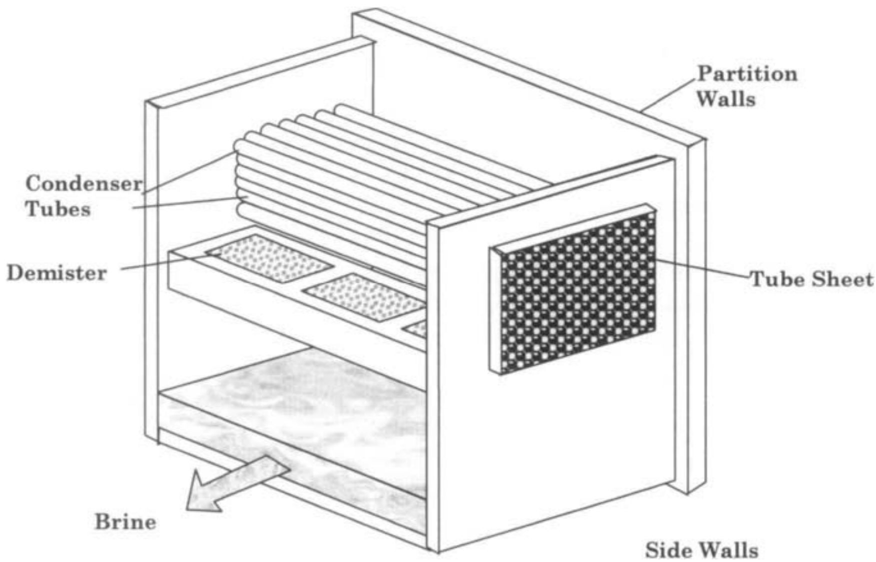

Fig. 4. Cross Tube Configuration

6.2.4 Features of MSF Brine Circulation Plants

The general features of the MSF plants include the following:

- Stable and reliable operation by insuring adequate heat transfer area, suitable materials, and proper corrosion allowance.
- The MSF desalination units operate with dual-purpose power generation plants. Design of the co-generation plants allows for flexible operation during peak loads for power or water. In Kuwait and the Gulf area peak loads for electricity and water occurs during the summer time due to the high ambient temperature, which is associated with massive use of indoor air-conditioning units and increase in domestic and industrial water consumption. The opposite is true for the winter season with mild temperatures and limited use of indoor heating units.
- The majority of the MSF plants are brine circulation, which are more superior to the once through design. Brine circulation result in higher conversion ratio, uses smaller amount of chemical additives, and gives good control on the temperature of the feed seawater.
- Cross tube design has simpler manufacturing and installation properties than long tube arrangement.
- All auxiliaries are motors driven have better operating characteristics than turbine drive units; even for the large brine circulation pumps.
- Additive treatment is superior to acid treatment, where acidic solutions may enhance corrosion rates of tubing, shells, and various metallic parts.
- Proper system design should allow for Load variations between 70-110% of the rated capacity.

6.3 MSF Process Synthesis

As discussed in Chapter 1 and the previous section, the MSF process accounts for more than 60% of the global desalination industry. In addition, it is the major source of fresh water in the Gulf countries. This section focuses on process fundamentals and developing better understanding for various elements forming the MSF process. The layout for the MSF process shown in Fig. 1 is quiet complex and understanding the functions and relations of various elements in the process is essential for successful system operation, analysis, optimization, and control. In addition, comprehensive analysis of the system flow sheet aids the development and design of novel and more efficient desalination processes. The following is a brief description of the plant flow diagram shown in Fig 1. The system includes three major sections: the brine heater, the heat recovery section, and the heat rejection section. The number of stages in the heat recovery section is larger than the heat rejection section. The brine heater drives the flashing process through heating the recycle brine stream to the top brine temperature. Flashing occurs in each stage, where a small amount of product water is generated and accumulated across the stages in the two sections. Vapor formation results because of the reduction of the brine saturation temperature; therefore, the stage temperature decreases from the hot to cold side of the plant. This allows for brine flow across the stages without the aid pumping power. The flashed off vapors condenses on the tubes of the preheater/condenser units. The released latent heat by the condensing vapor is used to preheat the brine recycle stream. On the cold side of the plant, the feed and the cooling seawater are introduced into the condenser/preheater tubes of the last stage in the heat rejection section. As this stream leaves the heat rejection section, the cooling seawater is rejected back to the sea and the feed seawater is mixed in the brine pool of the last stage in the heat rejection section. Also, two streams are extracted from the brine pool in this stage, which include the brine blow down and the brine recycle. The rejection of brine is necessary to control the salt concentration in the plant. As is shown, the brine reject is withdrawn prior to mixing of the feed seawater and the recycled brine is withdrawn from a location beyond the mixing point. The brine blow down is rejected to the sea and the brine recycle is introduced to the last stage in the heat recovery section. Additional units in the desalination plant include pretreatment of the feed and cooling seawater streams. Treatment of the intake seawater is limited to simple screening and filtration. On the other hand treatment of the feed seawater is more extensive and it includes dearation and addition of antiscalent and foaming inhibitors. Other basic units in the system include pumping units for the feed seawater and brine recycle. Also, gas-venting systems operate on flashing stages for removal of non-condensable gases.

From the above brief description, many questions arise regarding the specific arrangement of flashing stages and various streams. These questions include the following:

- Use of a large number of flashing stages.
- Upper limit on the top brine temperature.
- Need for two flashing sections (recovery and rejection).
- Minimum number of stages in the heat rejection section.
- The use of brine recycle.
- Function of the cooling seawater stream.

As mentioned before, complexity of the process makes it difficult for many people in the field to find the proper answers for the above questions. Therefore, finding the answers is pursued through simplifying the complicated MSF diagram to a number of simpler configurations. The simplest of these configurations is the single stage flashing system, Fig. 5. Results and analysis of this simple system are then used to modify the process diagram to a more detailed configuration, which solves the problems encountered in the simple system. As will be shown later, this process involved analysis of four simpler systems before reaching the conventional MSF system.

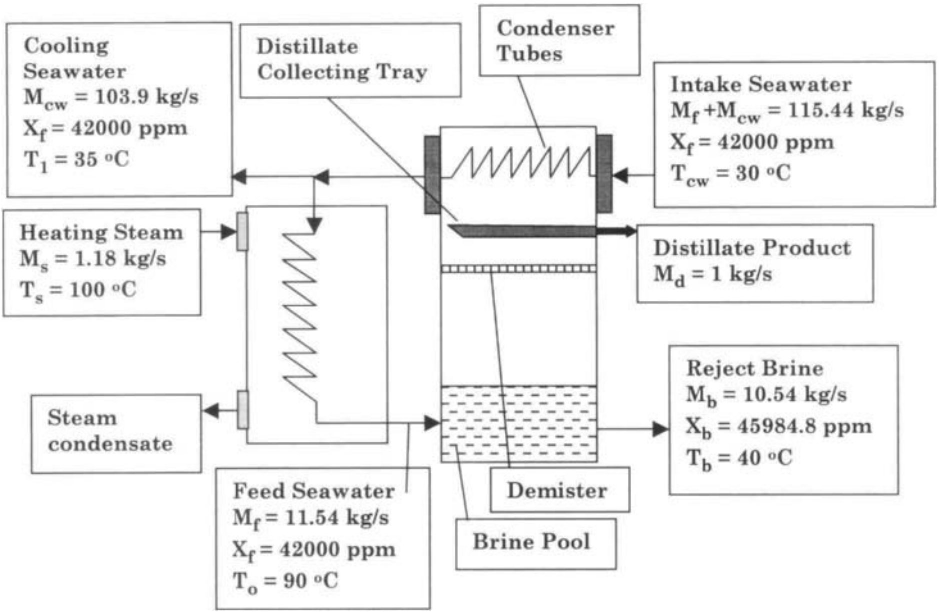

Fig. 5. Single stage flash desalination

The above task is achieved through mathematical modeling and analysis of various configurations. Performance of this type of analysis requires the use of analytical and non-iterative mathematical models rather than models based on numerical analysis. Examples for numerical models can be found in the studies performed by Omar, 1983, El-Dessouky and Assassa, 1985, Helal, et al. (1986), Darwish (1991), Montagna (1991), Hussain et al. (1993), Rosso et al. (1996), and El-Dessouky et al. (1995). This choice is necessary since analytical models generate closed form equations, which can be used to assess effect of various system parameters on the system performance. Of course use of analytical models should be made with caution to avoid simplifying assumptions that result in inaccurate predictions, which are not consistent with process characteristics.

Certainly, the following analysis will provide useful insights to the desalination community into the details of the MSF process. Results and analysis are also of great value to a number of the undergraduate and graduate engineering courses including plant design, process synthesis, modeling and simulation, energy conservation, flow sheet analysis, and of course water desalination. It is interesting to mention that the procedure outlined here can be used to analyze other complicated systems, i.e., multi-effect evaporation (MEE), distillation, etc.

6.3.1 Elements of Mathematical Analysis

To simplify the analysis procedures the following assumptions are used in development of various models:

- Distillate product is salt free. This assumption is valid since the boiling temperature of water is much lower than that of the salt.
- Specific heat at constant pressure, C_p, for all liquid streams, brine, distillate, and seawater is constant and equal to 4.18 kJ/kg °C.
- The overall heat transfer coefficient in the brine heater and preheaters is constant and equal to 2 kW/m^2 °C.
- Subcooling of condensate or superheating of heating steam has negligible effect on the system energy balance. This is because of the large difference of the vapor latent heat in comparison with the sensible heat value caused by liquid subcooling or vapor superheating of few degrees.
- The power requirements for pumps and auxiliaries are not included in the system analysis.
- The heat losses to the surroundings are negligible because the flashing stages and the brine heater are usually well insulated and operate at relatively low temperatures.

Performance analysis of various configurations is determined in terms of the following parameters:

- The thermal performance ratio, which is the ratio of distillate flow rate to the heating steam, $PR = M_d/M_s$.
- The specific heat transfer area is the ratio of the total heat transfer area to the distillate flow rate, $sA = A/M_d$.
- The specific feed flow rate is ratio of the feed to distillate flow rates, $sM_f = M_f/M_d$.
- The specific cooling water flow rate is the ratio of cooling water to distillate flow rates, $sM_{cw} = M_{cw}/M_d$

These variables are the most important factors controlling the cost of fresh water production.

The following data set is used to evaluate the performance of various configurations:

- Top brine temperature, $T_o = 90$ °C.
- Temperature of reject brine, $T_b = 40$ °C.
- Temperature of motive steam, $T_s = T_o+10$ °C.
- Temperature of intake seawater, $T_{cw} = 30$ °C.
- Thermodynamic loss, $\Delta T_{loss} = 2$ °C.
- The condenser terminal temperature difference, $TTD_c = 3$ °C.

- The salinity of intake seawater, $X_f = 42{,}000$ ppm.
- The maximum attainable concentration of the rejected brine, $X_b = 70{,}000$ ppm. This value is imposed by scale formation limits of $CaSO_4$.

6.3.2 Single Stage Flashing

The basic elements of the SSF system include the brine heater and the flashing chamber, which contains the condenser/preheater tubes, the demister, the brine pool, and the collecting distillate tray. A schematic diagram for the unit is shown in Fig. 5. As is shown, saturated steam at a flow rate of M_s and a temperature of T_s drives the single unit. The heating steam condenses outside the tubes of the brine heater, where it releases its latent heat, λ_s. This energy increases the feed seawater temperature from T_1 to the top brine temperature, T_o. The type of chemical additive that used to control scale formation dictates the upper limit on T_o. For acid and modern chemical additives, the limit on T_o is 120 °C and for polyphosphate the limit is 90 °C. The hot brine enters the flashing chamber, which operates at a pressure lower than the saturation pressure corresponding to the temperature of the brine flowing into the stage, T_o. In other words, the temperature difference of T_o-T_1 gives the degree of superheating of the brine as it flows to the flashing stage. During the flashing process, a part of the sensible heat of the brine is changed to latent heat by evaporation of a small portion of the brine, M_d. Distillate formation also results in the increase of the brine salinity from X_f to X_b. The formed vapor flows through the demister pad and then releases its latent heat, λ_v, as it condenses on the seawater condenser/preheater tubes. The condensed vapor is collected on the distillate tray. The latent heat of condensation is transferred to the intake seawater, $M_{cw}+M_f$, and increases its temperature from T_{cw} to T_1. The cooling seawater, M_{cw}, is rejected and the feed seawater, M_f, is introduced into the brine heater. Recovery of the latent heat by the feed seawater improves the overall efficiency of the desalination process. This reduces the amount of heating steam required in the brine heater, since the feed seawater temperature is increased in the brine heater from T_1 to T_o instead of T_{cw} to T_o. From a thermodynamic point of view, the function of the feed seawater preheater is the recovery of the energy added to the system by the heating steam in the brine heater. Also, it controls the saturation pressure inside the flashing chamber. On the other hand, the function of the demister is the removal of any brine droplets entrained with the flashed off vapor. This is necessary to avoid product contamination and lowering of its quality. In addition, removal of entrained brine protects the preheater tubes from fouling.

The mathematical model for the single flash unit is simple and it includes total mass and salt balances, rate equations for the heat transfer units, as well as energy balances for the brine heater and the condenser. The total mass and salt balances are

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

$$X_f M_f = X_b M_b \tag{2}$$

Eq. 2 assumes that the salt concentration, X_d, in the formed vapor is zero. The brine heater and condenser energy balances are given respectively by

$$M_s \lambda_s = M_f C_p (T_o - T_1) \tag{3}$$

$$M_d \lambda_v = (M_{cw}+M_f) C_p (T_1 - T_{cw}) = M_f Cp (T_o - T_b) \tag{4}$$

The heat transfer rate equations for the brine heater is

$$M_s \lambda_s = U_h A_h (LMTD)_h \tag{5}$$

where

$$(LMTD)_h = (T_o - T_1)/\ln((T_s - T_1)/(T_s - T_o)) \tag{6}$$

The heat transfer rate equation for the condenser is

$$M_d \lambda_v = U_c A_c (LMTD)_c \tag{7}$$

where

$$(LMTD)_c = (T_1 - T_{cw})/\ln((T_v - T_{cw})/(T_v - T_1)) \tag{8}$$

In the above system of equations, A is the heat transfer area, Cp is the specific heat at constant pressure, M is the mass flow rate, T is the temperature, X is the salinity of seawater and brine, λ is the latent heat of evaporation. The subscripts 1, b, c, cw, d, h, s, and v refer to stage number, brine, condenser, intake seawater, distillate, brine heater, steam, and vapor, respectively.

The unit thermal performance ratio, defined as the mass ratio of fresh water produced per unit mass of heating steam, is obtained by dividing Eqs. 4 and 3, where

$$PR = \frac{M_d}{M_s} = \frac{M_f C_p (T_o - T_b) \lambda_s}{M_f C_p (T_o - T_1) \lambda_v}$$

which simplifies to

$$PR = \frac{M_d}{M_s} = \frac{(T_o - T_b) \lambda_s}{(T_o - T_1) \lambda_v} \tag{9}$$

The stage temperature drop (ΔT_{st}) is equal to the difference (T_o – T_b) and is known as the flashing range. On the other hand, the term (T_o – T_1), as is shown in Fig. 6, is equal to the sum of the stage temperature drop (ΔT_{st}) the stage terminal temperature difference, TTD_c, and the thermodynamic losses, ΔT_{loss}, or,

$(T_o - T_b) = \Delta T_{st}$, and

$(T_o - T_1) = \Delta T_{st} + \Delta T_{loss} + TTD_c$

The thermodynamic losses (ΔT_{loss}) are the temperature difference of the brine leaving the stage, T_b, and the condensation temperature of the vapor, T_v. In a single stage flashing unit, these losses are caused by the boiling point elevation, the non-equilibrium allowance, and the temperature drop corresponding to the pressure drop in the demister pad and during condensation. The terminal temperature difference of the condenser, TTD_c, is equal to temperature difference of the condensing vapor, T_v, and the seawater leaving the condenser, T_1. The value of TTD_c plays a very important role in the design of the MSF system and its value ranges between 3-5 ^oC. In the brine heater, the temperature difference of the condensing steam and the effluent brine gives the brine heater terminal temperature difference, TTD_h.

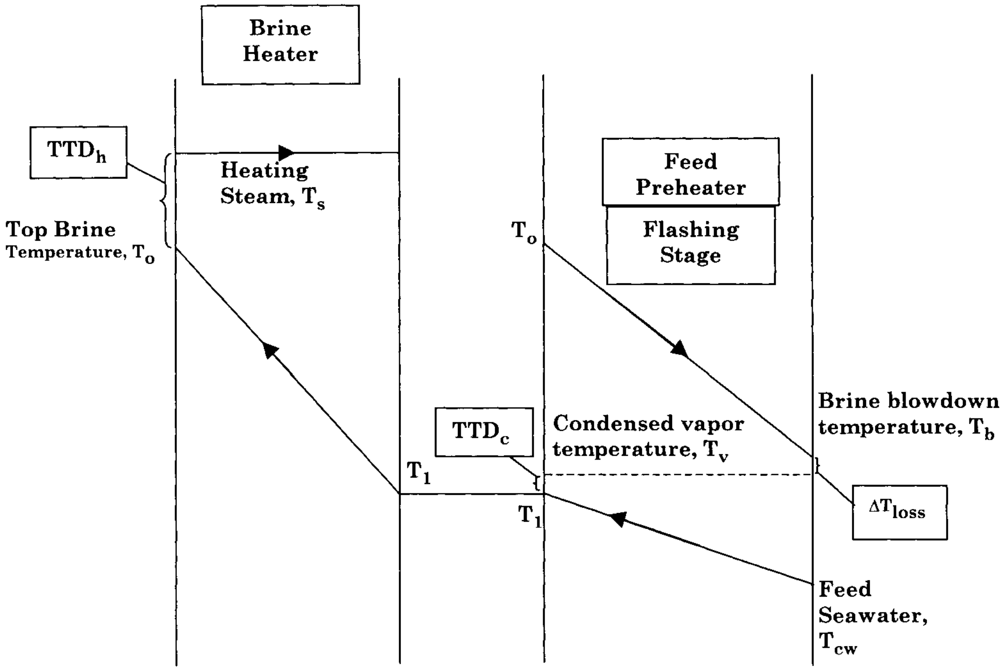

Fig. 6. Temperature profiles of heating steam, seawater, flashing brine, and condensate

The previous relations are substituted in the single stage model equations, Eqs. 3, 4, 6, 8, and 9. This results in the following equations, which are used to determine the single effect performance,

$$M_s\, \lambda_s = M_f\, C_p\, (\Delta T_{st} + \Delta T_{loss} + TTD_c) \tag{10}$$

$$M_d\, \lambda_v = M_f\, C_p\, \Delta T_{st} = (M_f + M_{cw})\, C_p\, (T_o - \Delta T_{st} - \Delta T_{loss} - TTD_1 - T_{cw}) \tag{11}$$

$$(LMTD)_h = (\Delta T_{st} + \Delta T_{loss} + TTD_c)/\ln((TTD_h + \Delta T_{st} + \Delta T_{loss} + TTD_c)/(\,TTD_h)) \tag{12}$$

$$(LMTD)_c = (\Delta T_{st})/\ln((\Delta T_{st} + TTD_c)/(TTD_c)) \tag{13}$$

$$PR = (\lambda_s)(\Delta T_{st})/((\Delta T_{st} + \Delta T_{loss} + TTD_c)\,(\lambda_v)) \tag{14}$$

The above data are used to calculate the stage temperature drop,

$\Delta T_{st} = T_o - T_b = 90 - 40 = 50\ ^oC$,

The terminal temperature difference for the brine heater,

$TTD_h = T_s - T_o = 100 - 90 = 10\ ^oC$,

The temperature of formed vapor,

$T_v = T_b - \Delta T_{loss} = 40 - 2 = 38\ ^oC$,

The latent heat of condensing vapor,

$\lambda_v = 2412.5$ kJ/kg (at 38 °C),

The latent heat of steam,

$\lambda_s = 2256$ kJ/kg (at 100 °C).

The performance ratio of the single stage flashing unit can be obtained from Eqs. 14, where

$$\begin{aligned} PR &= (\lambda_s)(\Delta T_{st})/((\Delta T_{st}+\Delta T_{loss}+TTD_c)\ (\lambda_v)) \\ &= ((2256)\ (50))\ /((50+2+3)(2412.5)) \\ &= 0.85 \text{ kg distillate/kg steam} \end{aligned}$$

The specific flow rates of the feed and cooling seawater are obtained from Eq. 11, where

$$M_d\ \lambda_v = M_f\ C_p\ \Delta T_{st}$$

$$\begin{aligned} M_f/M_d &= \lambda_v/(C_p\ (\Delta T_{st})) \\ &= 2412.5/((4.18)(50)) \\ &= 11.54 \text{ kg feed seawater/kg distillate} \end{aligned}$$

As for the specific flow rate of the cooling water it is given by

$$M_d\ \lambda_v = (M_f+M_{cw})\ C_p\ (T_o-\Delta T_{st}-\Delta T_{loss}-TTD_c-T_{cw})$$

which simplifies to

$$\begin{aligned} M_{cw}/M_d &= \lambda_v/(C_p\ (T_o-\Delta T_{st}-\Delta T_{loss}-TTD_c-T_{cw})) - (M_f/M_d) \\ &= (2412.5)/((4.18)(90-50-2-3-30)) - 11.54 \\ &= 103.9 \text{ kg cooling seawater/kg distillate} \end{aligned}$$

This result together with specification of the amount of distillate water can be used to calculate the brine flow rate and its salinity. Assuming that the distillate flow rate is equal to 1 kg/s gives

$M_b = 11.54 - 1 = 10.54$ kg brine/kg distillate

and

$$\begin{aligned} X_b &= X_f\, M_f/M_b \\ &= (42000)\,(11.54)/(10.54) \\ &= 45984.8 \text{ ppm} \end{aligned}$$

The specific heat transfer areas for the brine heater and the condenser are calculated from Eqs. 5, 7, 12, 13. Eqs. 12 and 13 are used to calculated the LMTD values in each unit, where,

$$\begin{aligned} (\mathrm{LMTD})_c &= \Delta T_{st} / \ln((\Delta T_{st} + \mathrm{TTD}_c)/(\mathrm{TTD}_c)) \\ &= 50/\ln((50+3)/3) \\ &= 17.4\ {}^{o}\mathrm{C} \end{aligned}$$

and

$$\begin{aligned} (\mathrm{LMTD})_h &= \frac{\Delta T + \Delta T_{loss} + \mathrm{TTD}_c}{\ln((\mathrm{TTD}_h + \Delta T + \Delta T_{loss} + \mathrm{TTD}_c)/(\mathrm{TTD}_h))} \\ &= (50+2+3)/\ln((10+50+3+2)/(10)) \\ &= 29.4\ {}^{o}\mathrm{C} \end{aligned}$$

The specific heat transfer area for the condenser and the brine heater are then calculated from Eqs. 5 and 7, where,

$$\begin{aligned} A_h/M_d &= ((M_s)(\lambda_s))/((M_d)(U_h)(\mathrm{LMTD})_h) \\ &= 2256/((0.85)(2)(29.4)) \\ &= 45.1 \text{ m}^2/(\text{kg/s}) \end{aligned}$$

and

$$\begin{aligned} A_c/M_d &= \lambda_v/((U_c)(\mathrm{LMTD})_c) \\ &= 2412.5/((2)(17.4)) \\ &= 69.3 \text{ m}^2/(\text{kg/s}) \end{aligned}$$

The above model and results shows that the main drawbacks of the single stage flash unit are:

- The performance ratio of the single stage flashing unit is always less than one. This result implies that the amount of distillate water produced is less than the amount of heating steam. Thus, the single stage flash unit can not be used on industrial scale
- The flow rate of feed seawater is much larger than the amount of distillate generated. This ratio is above ten. Therefore, a high rate of chemical additives and treatment for the feed/unit product is high.
- The specific flow rate of cooling water is very high. This would increase first cost of intake seawater pumping unit and power consumption.

On the other hand, the single stage flash unit is characterized by:

- The salinity of reject brine is much smaller than allowable design value set by $CaSO_4$ solubility limits.
- The heat transfer area for the brine heater and condenser are small, because of the large temperature driving force.
- The specific heat transfer area for the brine heater is inversely proportional to the performance ratio.
- The thermodynamic losses affect the area of the brine heater, however, it has no effect on the condenser area.

It is important to emphasis that most of the heat added to the system is rejected with the cooling seawater. In other words, the flashing stage does not consume most of the energy provided by the heating steam, but simply it degrades its quality.

6.3.3 Once Through MSF

The objective of the once through MSF system is to overcome the main drawback of the single stage flash unit that is to improve the system performance ratio. This is achieved by dividing the flashing range over a larger number of stages and as a result reducing the stage temperature drop.

A schematic diagram for the system is shown in Fig. 7. As is shown, the system includes a number of stages, n, and the brine heater. The stage elements are identical to those of the single stage unit. In the once through system, the temperature of intake seawater is increased as it flows through the preheater tubes of each stage from T_{cw} to t_1. The intake seawater flows from stage n to stage 1, i.e., from the low temperature to the high temperature side of the plant. The seawater leaving the last condenser enters the brine heater, where its temperature is increased from t_1 to T_o. The heated brine flashes off as it flows through the successive stages, where its temperature decreases from T_o to T_n. Simultaneously, the flashing vapor condenses around the condenser tubes in each stage, where it heats the seawater flowing through the tubes. The collected distillate in the distillate-collecting tray flows across the stages, where it leaves the plant from stage n. The flashing process reduces the brine temperature and increases its salinity from X_f to X_b. The brine leaving the last stage is rejected to the sea. This designates the process as the once through system, since no recycle of any portion of the unevaporated brine is made into the system. It is worth mentioning that the MSF-OT system does not contain a cooling water stream. This is because the reject brine stream, which has a low temperature and a large flow rate, contains the energy that must be removed from the system.

The performance of the once through system is developed in a similar manner to that of the singe stage unit. For this system, the energy balance of the brine heater is identical to Eq. 10. The amount of distillate formed in stage i is determined approximately from the energy balance on unit. This is

$$D_i \, \overline{\lambda}_v = M_f \, Cp \, (t_{i+1} - t_i) = B_{i-1} \, Cp \, (T_{i-1} - T_i)$$

where $\overline{\lambda}_v$ is the average latent heat of vapor condensation, evaluated at $T_{av} = (T_o+T_n)/2$. It should be noted that in the previous equation the brine flow rate in the first stage is equal to feed seawater flow rate, M_f. Summation of the above equation for all stages gives the total amount of distillated

$$M_d = \sum_{i=1}^{n} D_i = M_f \, C_p \, (T_o - T_n)/\overline{\lambda}_v$$

The flashing range term $(T_o - T_n)$ in the above equation is replaced by the stage temperature drop, which gives

$$M_d = \sum_{i=1}^{n} D_i = M_f \, C_p \, (n\Delta T_{st})/\overline{\lambda}_v \qquad (15)$$

Division of Eqs. 15 and 10 gives the performance ratio for the once through system, where,

$$PR = \frac{M_d}{M_s} = \frac{M_f C_p \left(n\Delta T_{st}\right) \lambda_s}{M_f C_p \left(\Delta T_{st} + \Delta T_{loss} + TTD_c\right) \overline{\lambda}_v}$$

Simplifying the above equation gives

$$PR = \frac{M_d}{M_s} = \frac{\left(n\Delta T_{st}\right) \lambda_s}{\left(\Delta T_{st} + \Delta T_{loss} + TTD_c\right) \overline{\lambda}_v} \qquad (16)$$

Eq. 7 is modified to calculate the condenser heat transfer area in each flashing stage, where

$$M_f\, C_p\, \Delta T_{st} = U_c\, A_c\, (LMTD)_c \tag{17}$$

The remainder of the model equations is similar to those of the single stage unit, where, X_b and M_b are obtained from Eqs. 1 and 2, A_h is calculated by Eq. 5, and the LMTD values are determined from Eqs. 12 and 13.

For the same set of system specifications, given for the single stage unit, the performance characteristics of the once through system are calculated for a 23-stage plant. The temperature drop per stage is first determined, where

$\Delta T_{st} = (90\text{-}40)/23 = 2.174\ ^oC$

The latent heat of condensation is calculated at the average stage temperature, T = $(T_o + T_n)/2$, which equal to 65 ^oC. The latent heat value at this temperature is 2346.5 kJ/kg. The performance ratio for the system is calculated from Eq. 16,

$$\begin{aligned} PR &= \frac{n\Delta T_{st}\ \lambda_s}{\left(\Delta T_{st} + \Delta T_{loss} + TTD_c\right)\overline{\lambda}_v} \\ &= (23)(2.174)(2256)/((2.174+2+3)(2346.5)) \\ &= 6.7 \text{ kg distillate water/kg steam} \end{aligned}$$

The specific feed flow rate is obtained from Eq. 17, where

$$\begin{aligned} M_f/M_d &= \overline{\lambda}_v /(C_p\ (T_o - T_n) \\ &= 2346.5/((4.18)(90\text{-}40)) \\ &= 11.22 \text{ kg intake seawater/kg distillate} \end{aligned}$$

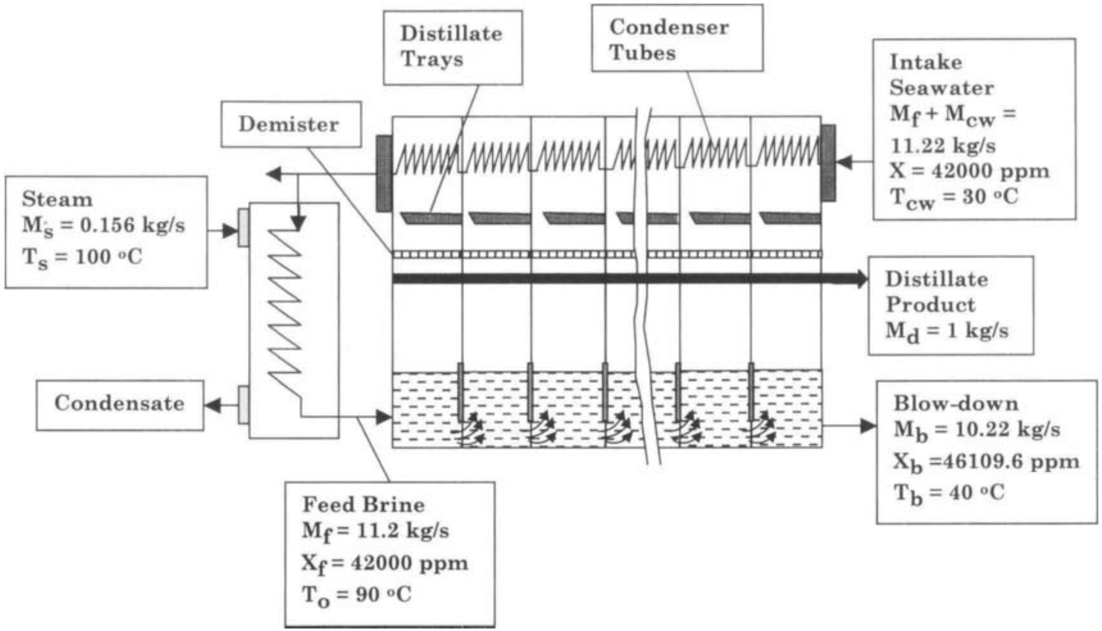

Fig. 7. Once-through multi-stage flash desalination process

This value together with specification of the amount of distillate water is used to calculate the rejected brine flow rate and its salinity. Assuming that the distillate flow rate is equal to 1 kg/s gives

$M_b = 11.22 - 1 = 10.22$ kg/s

and

$$\begin{aligned}X_b &= X_f\, M_f/M_b \\ &= (42000)\,(11.22)/(10.22) \\ &= 46106.6 \text{ ppm}\end{aligned}$$

The LMTD values are calculated from Eqs. 12 and 13, where,

$$\begin{aligned}(LMTD)_c &= \Delta T_{st}/\ln((\Delta T_{st}+TTD_c)/(TTD_c)) \\ &= 2.174/\ln((2.174+3)/3) \\ &= 3.98\ {}^oC\end{aligned}$$

and

$$\begin{aligned}(LMTD)_h &= \frac{\Delta T_{st} + \Delta T_{loss} + TTD_c}{\ln((TTD_h + \Delta T_{st} + \Delta T_{loss} + TTD_c)/(TTD_h))} \\ &= (2.174+2+3)/\ln((10+2.174+2+3)/(10)) \\ &= 13.27\ {}^oC\end{aligned}$$

The specific heat transfer area for the condenser and the brine heater are then calculated from Eqs. 5 and 17, where,

$$\begin{aligned}A_h/M_d &= M_s\, \lambda_s/((M_d)(U_h)(LMTD)_h) \\ &= 2256/((6.7)(2)(13.27)) \\ &= 12.7 \text{ m}^2/(\text{kg/s})\end{aligned}$$

and

$$\begin{aligned}A_c/M_d &= M_f\, C_p\, \Delta T_{st}/((U_c)(LMTD)_c) \\ &= (11.22)(4.18)(2.174)/((2)(3.98)) \\ &= 12.8 \text{ m}^2/(\text{kg/s})\end{aligned}$$

The above heat transfer areas are used to calculate the total specific heat transfer area, where

$$\begin{aligned}sA &= A_h + n\, A_c \\ &= 12.7 + (23)\,(12.8)\end{aligned}$$

$= 307.1\ m^2/(kg/s)$

The above results show that the MSF-OT system is distinguished from the single stage unit by the drastic increase in the thermal performance ratio, which increases from a value below one for the single stage flash unit to a value above six for the once through system. Also, the MSF-OT system does not use cooling water for removal of excess heat added in the brine heater. Irrespective of this improvement, the MSF-OT still has a high flow rate ratio for the intake seawater to the distillate product. Also, the total specific heat transfer area for the condenser increases from 69.3 $m^2/(kg/s)$ for the single stage unit to 294.4 $m^2/(kg/s)$ in the once through MSF. This is caused by the decrease in the temperature driving force from 17.4 °C in the single unit to 3.98 °C in the once through system.

Other features of the MSF-OT system include the following:

- Operation at low salinity for the feed and flashing brine streams, with values of 42000 and 46106.6 ppm, respectively. This reduces fouling and scaling problems in the condenser tubes and the brine heater and the boiling elevation, which reduces the value of the thermodynamic losses.
- The specific heat transfer area for the brine heater is inversely proportional to the performance ratio. This is because the heat added to the system in the brine heater is divided over a larger amount of distillate water. Therefore, the increase in the performance ratio for the once through system reduces the specific heat transfer area from 45.1 $m^2/(kg/s)$ found in the single unit to 12.7 $m^2/(kg/s)$ for the once through system.

6.3.4 Brine Mixing MSF

The purpose of brine recirculation is to decrease the flow rate of the feed seawater. As a result, this lowers the chemical additive consumption rate and the size of the pretreatment facilities for the feed stream. Also, since the recycled brine contains higher energy than the feed seawater, the process thermal efficiency will improve. The simplest brine circulation system is made through mixing part of the blow-down brine with the feed stream. This simple configuration is shown in Fig. 8. In this system, a portion of the blow-down brine, M_r-M_f, is mixed with the intake seawater stream, M_f. The resulting mixture, M_r, has a higher salinity and temperature than the intake seawater. The remaining elements of the system are similar to those of the once through MSF.

The brine recycle system allows for achieving the maximum limit on the salinity of the blow-down brine, which is equal to 70000 ppm. Assuming that the distillate is salt free, the salinity of the feed seawater is 42000, and the distillate flow rate is equal to 1 kg/s, gives

$70000\ (M_b) = 42000\ (M_b + 1)$

This results in M_b = 1.5 kg/s and M_f = 2.5 kg/s. Thus, the brine recirculation reduces the specific feed flow rate from a high value above 11 in the once through system to a lower value of 2.5 in the new system.

The thermal performance ratio of the system is identical to Eq. 16. The system variables, which include T_n, t_1, T_r, and M_r, are determined by performing the following:

- Overall energy balance,

$$M_s\ \lambda_s = M_b\ C_p\ (T_n - T_{cw}) + M_d\ Cp\ (T_d - T_{cw}) \tag{18}$$

- Energy balance on the brine heater,

$$M_s\ \lambda_s = M_r\ C_p\ (T_o - t_1) \tag{19}$$

- Overall balance on the flashing brine,

$$M_r\ C_p\ (T_o - T_n) = M_d\ \bar{\lambda}_v \tag{20}$$

- Energy balance on the mixing point,

$$(M_r - M_f)\ C_p\ (T_n - T_{cw}) = M_r\ C_p\ (T_r - T_{cw}) \tag{21}$$

Eqs. 18 and 19 are combined to express M_r in terms of M_d, where,

$$M_r\ C_p\ (T_o - t_1) = (X_f/(X_b\text{-}X_f))\ M_d\ C_p\ (T_n - T_{cw}) + M_d\ Cp\ (T_d - T_{cw})$$

Neglecting the effect of the thermodynamic losses in the last stage, i.e., setting $T_n = T_d$, simplifies the above equation to

$$M_r\ C_p\ (T_o - t_1) = (1+X_f/(X_b\text{-}X_f))\ M_d\ C_p\ (T_n - T_{cw})$$

Or

$$M_r\ C_p\ (T_o - t_1) = (X_b/(X_b\text{-}X_f))\ M_d\ C_p\ (T_n - T_{cw}) \tag{22}$$

Eliminating M_r from Eqs. 20 and 22 gives

$(M_d \bar{\lambda}_v)(T_o - t_1)/(T_o - T_n) = (X_b/(X_b\text{-}X_f))\ M_d\ C_p\ (T_n - T_{cw})$

which reduces to

$(\bar{\lambda}_v)(T_o - t_1)/(T_o - T_n) = C_p\ (X_b/(X_b\text{-}X_f))\ (T_n - T_{cw})$

Recalling that the ratio of $(T_o - t_1)/(T_o - T_n)$ is approximately equal to 1/PR, changes the above equation to the following form

$$PR = \bar{\lambda}_v /(C_p\ (X_b/(X_b\text{-}X_f))\ (T_n - T_{cw})) \tag{23}$$

The same procedure is applied to Eq. 21, where M_r and M_f are expressed in terms of M_d. The resulting relation is

$$(\bar{\lambda}_v)(T_n - T_f)/(T_o - T_n) = C_p\ (X_b/(X_b\text{-}X_f))\ (T_n - T_{cw}) \tag{24}$$

Division of Eqs. 23 and 24, show that

$(T_n - T_f) = (T_o - t_1)$

or

$$(t_1 - T_f) = (T_o - T_n) \tag{25}$$

This relation implies that the total temperature drop of the flashing brine is equal to the total temperature increase of the brine recycle flowing inside the preheater/condenser tubes.

Iterative solution of the above system of equations is dictated by the dependence of $\bar{\lambda}_v$ on the value of T_n. However, a simple solution procedure can provide quick estimate for the system variables. Starting with Eq. 23 and by assuming that the system performance ratio PR is equal to 8, and the average latent heat of condensing vapor is assumed equal to 2346.5 kJ/kg (at 65 °C), gives

$(\bar{\lambda}_v) = (PR)\ C_p\ (X_b/(X_b\text{-}X_f))\ (T_n - T_{cw})$
$(2346.5) = (8)\ (4.18)\ (70000/(70000\text{-}42000))\ (T_n - 30)$

Solution of the above equation gives

$T_n = 58$ °C.

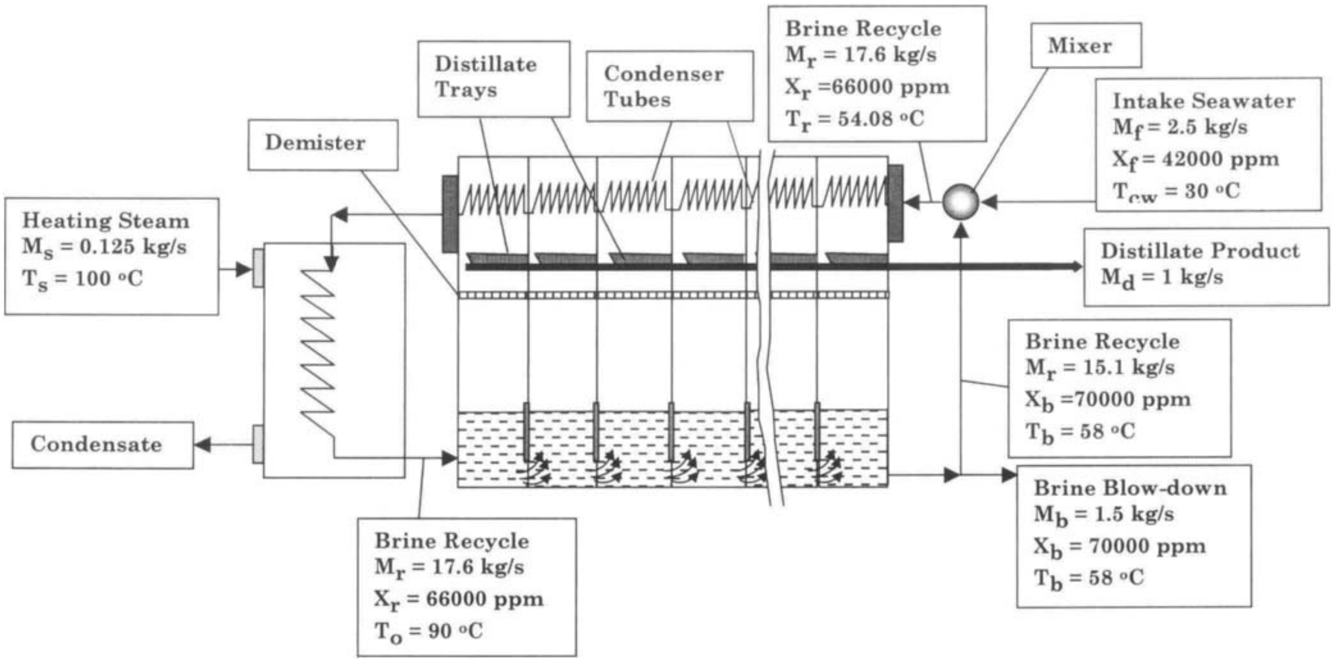

Fig. 8. Simple mixer brine recycle MSF desalination process

This value results in a stage temperature drop of (90-58)/23 or 1.39 °C. The flow rate of recycled brine, M_r, is obtained from Eq. 21, where

$$M_r\, C_p\, (n\Delta T_{st}) = M_d\, \bar{\lambda}_v$$

$$M_r\, (4.18)\, (23)\, (1.39) = (1)\, (2346.5)$$

This gives

$$M_r = 17.6 \text{ kg/s}$$

The flow rate of the recycle brine can be used to calculate the salinity of the feed seawater, X_r. A salt balance at the mixer is given by

$$M_r\, X_r = X_f\, M_f + X_n\, (M_r - M_f)$$

$$X_r\, (17.6) = (42000)\, (2.5) + (70000)\, (17.6 - 2.5)$$

Therefore, X_r is equal to 66018 ppm. The feed seawater temperature, T_r, is calculated from the mixer energy balance, Eq. 22, where

$$(M_r - M_f)\, C_p\, (T_n - T_{cw}) = M_r\, C_p\, (T_r - T_{cw})$$

$$(17.6 - 2.5)\, (4.18)\, (58 - 30) = (17.6)\, (4.18)\, (T_r - 30)$$

which gives

$$T_r = 54.08\ ^{o}\text{C}$$

The temperature of feed seawater leaving the first stage is determined from Eq. 25, where,

$$(t_1 - T_f) = (T_o - T_n)$$

$$(t_1 - 54.08) = (90 - 58.07)$$

This gives

$$t_1 = 86.01\ ^{o}\text{C}$$

The LMTD values are calculated from Eqs. 12 and 13, where,

$$(LMTD)_c = \Delta T_{st}/\ln((\Delta T_{st}+TTD_c)/(TTD_c))$$
$$= 1.39/\ln((1.39+3)/(3))$$
$$= 3.65\ ^oC$$

and

$$(LMTD)_h = \frac{\Delta T_{st} + \Delta T_{loss} + TTD_c}{\ln((TTD_h + \Delta T_{st} + \Delta T_{loss} + TTD_c)/(TTD_h))}$$
$$= (1.39+2+3)/\ln((10+1.39+2+3)/(10))$$
$$= 12.93\ ^oC$$

The specific heat transfer area for the condenser and the brine heater are then calculated from Eqs. 5 and 17, where,

$$A_h/M_d = \lambda_s/((PR)(U_h)(LMTD)_h)$$
$$= 2256/((8)(2)(12.93))$$
$$= 10.9\ m^2/(kg/s)$$

and

$$A_c/M_d = M_r\ C_p\ \Delta T_{st}/((U_c)(LMTD)_c)$$
$$= (17.58)(4.18)(1.39)/((2)(3.65))$$
$$= 13.97\ m^2/(kg/s)$$

This value gives a total specific heat transfer area of

$$sA = A_h + n\ A_c$$
$$= 10.9 + (23)\ (13.97)$$
$$= 332.3\ m^2/(kg/s)$$

The main result of the above analysis is the high temperature of the rejected brine flowing from the last flashing stage, which is larger than the practical limit of 40 ^oC. This reduces the flashing range of the system and in turn results in the following:

- Increase of the recycle brine flow rate per unit of distillate product. This is necessary to account for the reduction in the temperature flashing range, $T_o - T_n$, across the stages.
- Increase of the specific pumping power for the recycle brine.
- Increase of the salt concentration in the first stage. The feed seawater salinity is quite high, 66018 ppm. This value would result in severe operational problems, because of enhanced formation rates for scale and fouling in the preheater/condenser tubes in the flashing stage close to the brine heater. Also,

the high salinity increases the thermodynamic losses, which is caused by the boiling point elevation.

- The temperature drop per stage is low, 1.39 °C, in comparison with 2.174 °C, found in the once through MSF unit. This reduces the temperature driving force for heat transfer and results in the increase of the increase of the heat transfer area of preheater/condenser tubes.

6.3.5 MSF with Brine Recirculation and a Heat Rejection Section

The main aim of adding heat rejection section is the removal of the excess heat added to the system in the brine heater. The heat rejection section is used to control the temperature of the recycled brine. This is achieved through recovery of a controlled amount of energy from the flashing brine into the brine recycle and rejection of the remaining energy into the cooling seawater stream, Fig. 9. A major advantage of the heat rejection section is the reduced pretreatment applied to the large stream of intake seawater, which requires inexpensive screening and filtration. Therefore, extensive pretreatment, which includes deareation, anti-foam, and anti-scalent additions, is applied only to the feed stream, which is less than the feed flow rate in the once through process.

In this analysis, the heat rejection section is assumed to contain a single stage. Figure 9 does not show the elements forming the brine heater and the heat recovery section, which are similar to those of the once through system. As for the heat rejection section, which includes a single stage, it receives the brine leaving the heat recovery section, where it flashes and generates a small amount of vapor. The vapor condenses around the preheater/condenser tubes, where its latent heat is absorbed by the intake seawater, M_f+M_{cw}. As a result, the intake seawater temperature is increased from T_{cw} to T_n. The cooling seawater stream, M_{cw}, is rejected, while the feed seawater stream is mixed in the brine pool within the stage. A portion of the brine is recycled and enters the tubes of the condenser tubes of the last stage in the heat recovery section. The remaining brine is returned back to the sea.

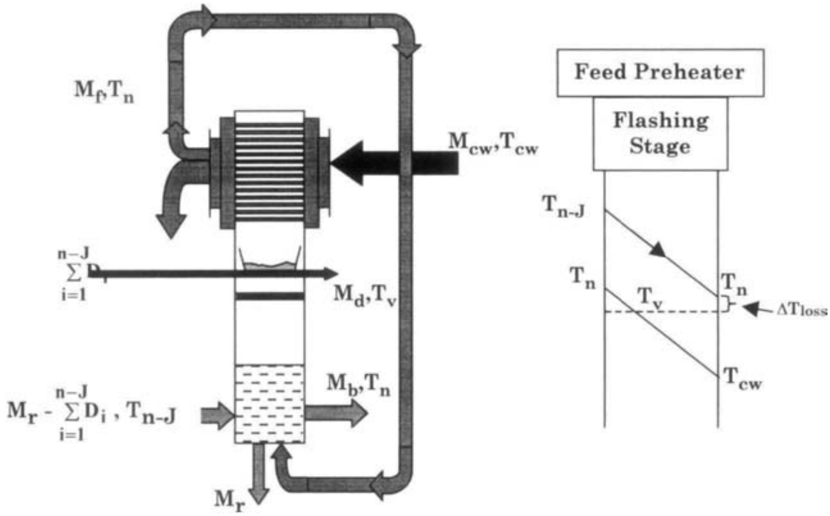

Fig. 9. Single stage heat rejection section and temperature profiles in the condenser

First, analysis is made for an MSF system containing a single stage heat rejection. The stage temperature profile, shown in Fig. 9, is used to determine applicability of this configuration. As is shown, the temperature of the intake seawater is increased from T_{cw} to T_n, where T_n is equal to flashing brine temperature in the stage. This temperature increase is necessary to avoid thermal shock upon mixing of the feed seawater and the flashing brine inside the stage. Since, the condensing vapor temperature, T_v, is less than the flashing brine temperature, T_n, by the thermodynamic losses. Therefore, the hot stream temperature profile, T_v, intersects the cold stream profile, T_{cw}-T_n, and as a result the required heat transfer area will be infinite. This simple analysis indicates that a single stage heat rejection section can not be implemented industrially.

The layout of the two-stage heat rejection and brine recycle MSF system is similar to conventional MSF, shown in Fig. 1, except for the number of stages in the heat rejection section. To determine feasibility of the two-stage system its heat transfer characteristics are compared against those of conventional MSF. Comparison of the two systems is based on performance of the condenser/preheater units in each stage and the value of the terminal temperature difference for these units. Common practice for similar heat exchange units puts a minimum value of 2 °C on the terminal temperature difference of heat exchange units. Operation at values lower than this limit do not yield the desired design values, which may include heating, cooling, condensation, or evaporation.

Results for the analysis are shown in Fig. 10. As is shown for the two-stage system, the values of TTD_{21} and TTD_{22} are 0.273 and 3 °C. On the other hand, the values of TTD_{21}, TTD_{22}, and TTD_{23} for conventional MSF are 2.35, 3.5, and 4.67 °C, respectively. In the light of the above, design and operation of the MSF system with a two-stage heat rejection is feasible, nevertheless, the low terminal temperature difference will increase dramatically the required heat transfer area. Addition of the third heat rejection unit in conventional MSF solves the limit imposed on the terminal temperature difference for the two stage system and consequently the required heat transfer area.

6.3.6 Conventional MSF

The model and analysis performed for the MSF system is made for a three-stage heat rejection section and twenty stages heat recovery section. Most of the model equations are similar to those previously given for other configurations. The overall material and salt balance equations are identical to Eqs. 1 and 2 given in the model of the single stage flash unit. The equations for the amounts of heating steam and recycle brine are identical to Eqs. 19 and 20, respectively. In addition, the heat transfer areas for the brine heater and the condensers in the heat recovery section are given by Eqs. 5 and 17. As for the LMTD values for both units it is calculated from Eqs. 12 and 13.

The salt concentration in the recycle stream, X_r, is obtained by performing salt balance on the heat rejection section. This balance is

$$X_r\,M_r + M_b\,X_b = X_f\,M_f + (M_r - M_d)\,X_{n-j}$$

The above balance is arranged to

$$X_r = (X_f\,M_f + (M_r - M_d)\,X_{n-j} - M_b\,X_b)/M_r$$

Assuming that $X_{n-j} = X_b$, simplifies the above equation to

$$X_r = (X_f\,M_f + (M_r - M_d)\,X_b - M_b\,X_b)/M_r$$

Since $M_f = M_b + M_d$, then,

$$X_r = ((X_f - X_b)\,M_f + M_r\,X_b)/M_r \quad (26)$$

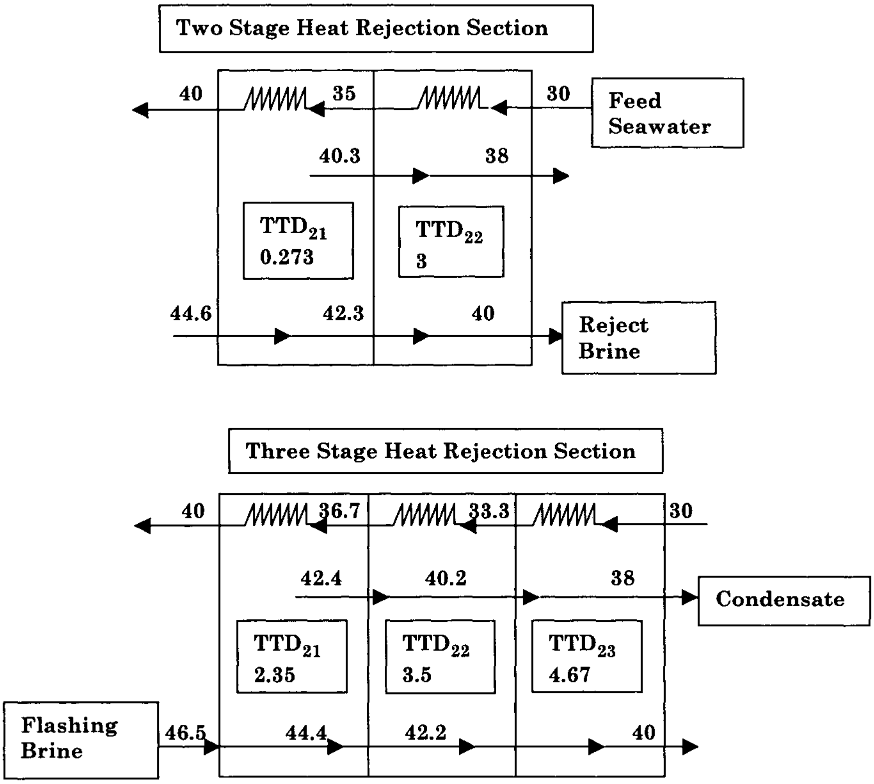

Fig. 10. Variation in stage terminal temperature difference and temperatures of flashing brine, condensate, and feed seawater.

The cooling water flow rate, M_{cw}, is obtained from an overall energy balance around the desalination plant, Fig. 1. The intake seawater temperature, T_{cw}, is used as the reference temperature in the energy balance. This gives

$$M_s \lambda_s = M_{cw} C_p (T_n - T_{cw}) + M_b C_p(T_n - T_{cw}) + M_d C_p (T_n - T_{cw})$$

The above equation is arranged to obtain an expression for M_{cw}

$$M_{cw} = (M_s \lambda_s - M_f C_p(T_n - T_{cw}))/ (C_p (T_n - T_{cw})) \qquad (27)$$

The heat transfer area for the condensers in the heat rejection section, A_j, are obtained from the following equation

$$A_j = (M_f+M_{cw})\ Cp\ (t_i - t_{i+1})/(U\ (LMTD)_j)$$
$$= (M_f+M_{cw})\ Cp\ (\Delta t_j)/(U\ (LMTD)_j) \qquad (28)$$

$$(LMTD)_j = (t_i - t_{i+1})_j/\ln((T_{v_i} - t_{i+1})/(T_{v_i} - t_i))$$
$$= \Delta t_j/\ln((\Delta t_j + TTD_j)/(TTD_j)) \qquad (29)$$

The value of TTD_j is set equal to 3 ^oC and Δt_j is obtained from

$$\Delta t_j = (T_n - T_{cw})/j$$

The total heat transfer area for all condensers in the heat recovery and rejection sections as well as the brine heater is then obtained from

$$sA = A_h + (n\text{-}j)\ A_c + (j)\ A_j \qquad (30)$$

As discussed before, a product flow rate of 1 kg/s and salinity of 42000 ppm and 70000 ppm for the feed and blow down, respectively, gives a feed flow rate of 2.5 kg/s and brine blow down flow rate of 1.5 kg/s. The temperature drop per stage is obtained from

$$\Delta T_{st} = (T_o - T_n)/n = (90\text{-}40)/23 = 2.174\ ^oC$$

The recycle flow, M_r, is then calculated from Eq. 20, where

$$M_d = M_r\ C_p\ (n\square T_{st})/\bar{\lambda}_v$$
$$1.0 = M_r\ (4.18)\ (23)(2.174)/2346.5$$
$$M_r = 11.22\ \text{kg/s}$$

The salinity of the recycle brine is calculated from Eq. 26,

$$X_r = ((X_f - X_b)\ M_f + M_r\ X_b)/M_r$$
$$X_r = ((42000 - 70000)\ (2.5) + (11.22)\ (70000))/(11.22)$$

which reduces to

$$X_r = 63765\ \text{ppm}$$

Equation 19 is used to determine the flow rate of the heating steam. In this equation, the steam latent heat, λ_s, is equal to 2256 kJ/kg (at 100 °C), thus,

$$M_s = M_r\, C_p\, (\Delta T + \Delta T_{loss} + TTD_c)/\lambda_s$$
$$= (11.22)\,(4.18)\,(2.174+2+3)/2256$$
$$= 0.149 \text{ kg/s}$$

Since the total flow rate of the distillate product, M_d, is equal to 1 kg/s, the system thermal performance ratio can be readily obtained

$$PR = M_d/M_s = 1/0.149 = 6.7$$

The cooling water flow rate is then calculated from Eq. 27, where

$$M_{cw} = (M_s\, \lambda_s - M_f\, C_p(T_n - T_{cw}))/(C_p\, (T_n - T_{cw}))$$
$$= ((0.149)(2256) - (2.5)(4.18)(40\text{-}30))/((4.18)(40\text{-}30))$$
$$= 5.55 \text{ kg/s}$$

The heat transfer areas are obtained from Eqs. 5, 17, and 28. The heat transfer area for the brine heater is calculated from Eq. 5,

$$Ms\, \lambda s = U\, Ah\, (LMTD)h$$
$$(0.149)\,(2256) = (2)\,(Ah)(13.26)$$

This gives, $A_h = 12.69$ m^2. The $(LMTD)_h$ is calculated from the Eq. 12, where

$$(LMTD)_h = \frac{\Delta T_{st} + \Delta T_{loss} + TTD_c}{\ln((TTD_h + \Delta T_{st} + \Delta T_{loss} + TTD_c)/(TTD_h))}$$
$$= (2.174+2+3)/\ln((10+2.174+2+3)/(10))$$
$$= 13.26\ ^{o}C$$

The heat transfer area for the preheater/condenser in each stage in the heat recovery section is calculated from Eq. 17

$$A_c = (M_r)\, Cp\, (\Delta T_{st})/(U\, (LMTD)_c)$$
$$= (11.22)\,(4.18)\,(2.174)/((2)(3.315))$$
$$= 12.78 \text{ m}^2$$

The value of $(LMTD)_c$ is calculated using Eq. 13, thus,

$$(LMTD)_c = \Delta T_{st}/\ln((\Delta T_{st} + TTD_c)/(TTD_c))$$
$$= (2.174)/\ln((2.174+3)/(3))$$

$= 3.98\ ^oC$

The same procedure is applied to the stages in the heat rejection section, where the condenser area in rejection stages is given by

$$A_j = (M_f+M_{cw})\ Cp\ (\Delta t_j)/(U\ (LMTD)_j)$$
$$= (2.5+5.55)\ (4.18)\ (3.33)/((2)(4.46))$$
$$= 12.57\ m^2$$

where

$$(LMTD)_j = \Delta t_j/\ln((\Delta t_j + TTD_j)/(TTD_j))$$
$$= (3.33)/\ln((3.33+3)/(3)$$
$$= 4.46\ ^oC$$

The total specific heat transfer area is obtained from Eq. 30, where,

$$sA = A_h + (n\text{-}j)\ A_c + (j)\ A_j$$
$$= 12.69 + (20)\ (12.78) + (3)\ (12.57) = 306.2\ m^2$$

Examining the above results for the MSF system show the following characteristics:

- Reduction in the feed flow rate in comparison with the MSF-OT.
- Low total specific heat transfer area.
- High Performance ratio.
- Low salinity for the recycle brine in comparison with MSF-R.
- Low temperature for the reject brine in comparison with MSF-R.

6.3.7 Effects of Operating Variables

Sensitivity of conventional MSF, the once through (MSF-OT), and the single stage flashing (SSF) are analyzed as a function of the top brine temperature, the number of flashing stages, and the thermodynamic losses. Analysis includes effects of system parameters on the thermal performance ratio, the total specific heat transfer area, the salinity of recycle and blow down brine, and the specific flow rates of feed, cooling water, recycle brine, and blow down brine. Analysis is performed over a temperature range of 90-110 ^oC for the top brine temperature, a total number of stages of 20 to 29, and a thermodynamic loss range of 0.5-2 ^oC.

Figure 11 shows variations in the thermal performance ratio for the three systems as a function of the top brine temperature for different values of the thermodynamic losses. As is shown, the performance ratio for the SSF system is less than 1 and is insensitive to variations in the top brine temperature as well as the thermodynamic losses. This behavior is cleared by inspection of Eq. 11, which is dominated by the flashing range. This parameter is much larger than the thermodynamic losses as well as the terminal temperature difference. As for the MSF and MSF-OT, their performance ratio is identical and is much larger than that for the single stage flashing system. This is because the flashing range is divided over n stages. As is shown in Fig. 11, the thermal performance ratio for the MSF and MSF-OT systems increases by the decrease of the thermodynamic losses and the increase of the top brine temperature. As given in Eq. 16, which applies to either system, the thermal performance ratio is inversely proportional to the thermodynamic losses. In addition, increase of the top brine temperature increases the flashing range and results in the increase of the amount of distillate product per unit mass of recirculated brine.

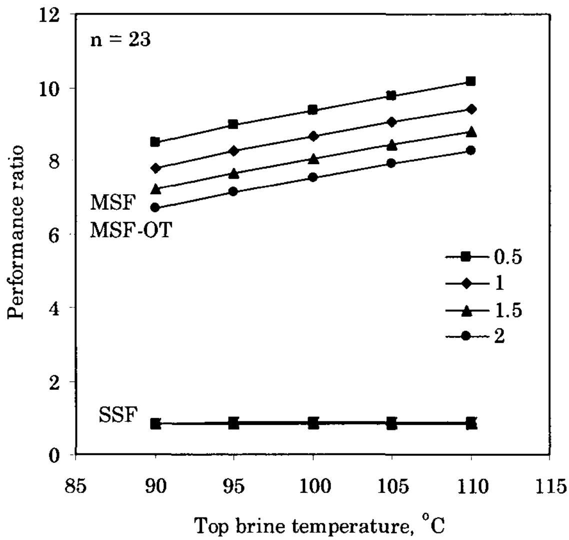

Fig. 11. Variation in system performance ratio as a function of top brine temperature and the thermodynamic losses.

Variations in the specific heat transfer area for the three systems are shown in Fig. 12. As is shown, the specific heat transfer area decreases with the increase of the top brine temperature and the decrease of the thermodynamic losses. Increase of the top brine temperature increases the temperature driving force, which enhances the rate of heat transfer, and as a result reduces the heat transfer area. The same effect is found upon lowering of the thermodynamic losses, which implies increase of the temperature of the condensing vapor. Thus, the temperature driving force increases and results in lowering of the heat transfer area.

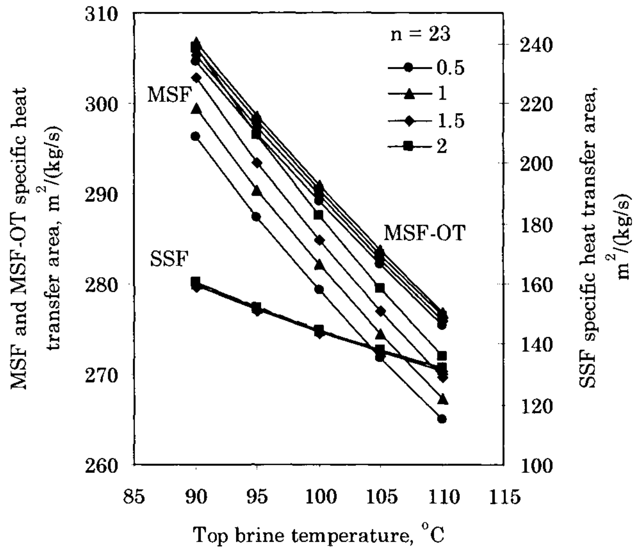

Fig. 12. Variation in specific heat transfer area for MSF, MSF-OT, and SSF as function of the top brine temperature and thermodynamic losses

Figure 13 shows variations of specific system parameters for the MSF configuration, which include the specific flow rate and salinity of the brine recycle as well as the specific cooling water flow rate. Variations are given as a function of the top brine temperature and at different values of the thermodynamic losses. As is shown, the flow rate and the salinity of the brine recycle are independent of the thermodynamic losses. Examining the balance equations for either parameter, Eqs. 20 and 26, respectively, show no dependence on the thermodynamic losses. However, both parameters are inversely proportional to the flashing range or the top brine temperature. The decrease of the specific flow rate of the cooling water with the decrease of the thermodynamic losses and the increase of the top brine temperature is associated with simultaneous increase of the system thermal performance ratio. At higher performance ratios, lower amounts of the heating steam are used, thus, the rate of heat removal in the heat rejection section is reduced, i.e., the specific flow rate of the cooling water.

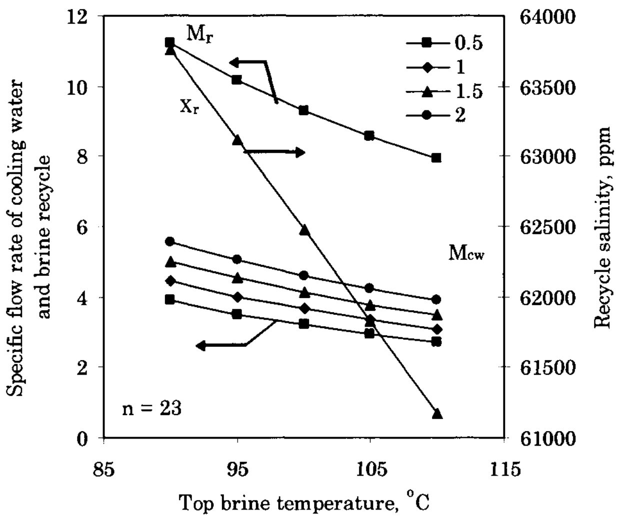

Fig. 13. Variation in specific cooling water flow rate, specific recycle flow rate, and salinity of brine blow down for MSF as a function of the top brine temperature and thermodynamic losses.

Dependence of the system parameters for the MSF and MSF-OT on the number of stage is shown in Figs. 14-16. Identical variations in the system thermal performance ratio and specific total heat transfer area are obtained for the MSF and MSF-OT as a function of the number of stage, Figs. 14 and 15. In either system, the thermal performance ratio is proportional with the number of stages. Increase of the number of stages reduces the stage temperature drop and as a result increases the system thermal performance. Increase in the total specific heat transfer area in both systems with the increase of the number of stages is caused by the reduction in the stage temperature drop and consequently the temperature driving force for heat transfer.

Variations in the MSF parameters as a function of the number of stage, which include the specific flow rates of brine recycle and cooling seawater as well as the salinity of the brine recycle, are shown in Fig. 16. As is shown the salinity and specific flow rate of brine recycle are independent of the number of stages. This is because the specific flow rate of the brine recycle depends only the total flashing range, which is independent of the number of stages. The decrease of specific flow rate of the cooling water with the increase of the number of stages is caused by the increase in the system thermal performance ratio. Increase of the performance ratio reduces the heat load in the brine heater and consequently the amount of heat removal by the cooling seawater.

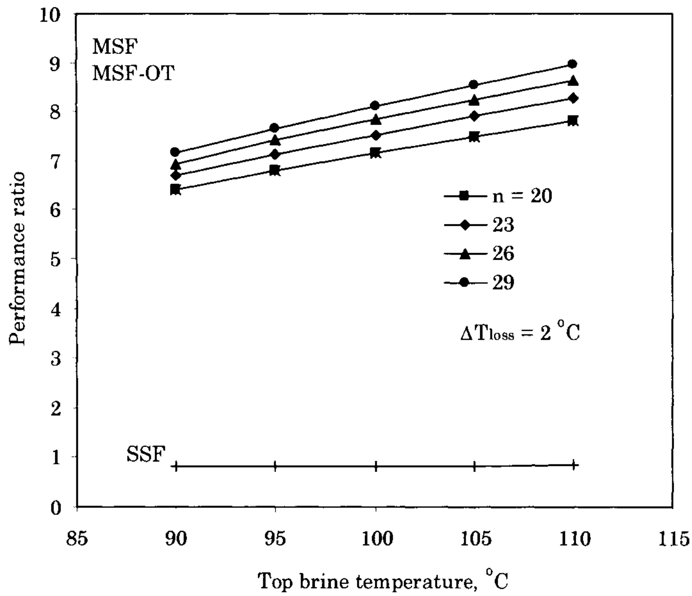

Fig. 14. Variation in performance ratio as a function of the top brine temperature and the number of stages.

Other system parameters, which include the specific flow rates of feed and brine blow down as well as the salinity of blow down brine, are shown in Figs. 17-19. As is shown these parameters are independent of the thermodynamic losses as well as the number of stages. In addition, these parameters are also independent of the top brine temperature for the MSF system. This is because the thermal performance of the MSF system is dependent on the properties of the brine recycle stream rather than the feed seawater stream. For the MSF system, the salinity and specific flow rate of the reject brine are kept constant at 70,000 ppm and 1.5 kg/s, respectively. In addition, the specific feed flow rate in the MSF system is also kept constant at 2.5 kg/s. The decrease in the specific feed flow rate for the MSF-OT and SSF system upon the increase in the top brine temperature is caused by the increase the flashing range and consequently the specific amount of product per unit mass of feed seawater. This decrease is also associated with simultaneous decrease in the specific flow rate of brine blow down for the MSF-OT and SSF, which is less than the specific feed flow rate by 1 kg/s or the distillate flow rate. Decrease in the specific flow rate of the brine blow down causes the increase of the stream salinity, Fig. 19.

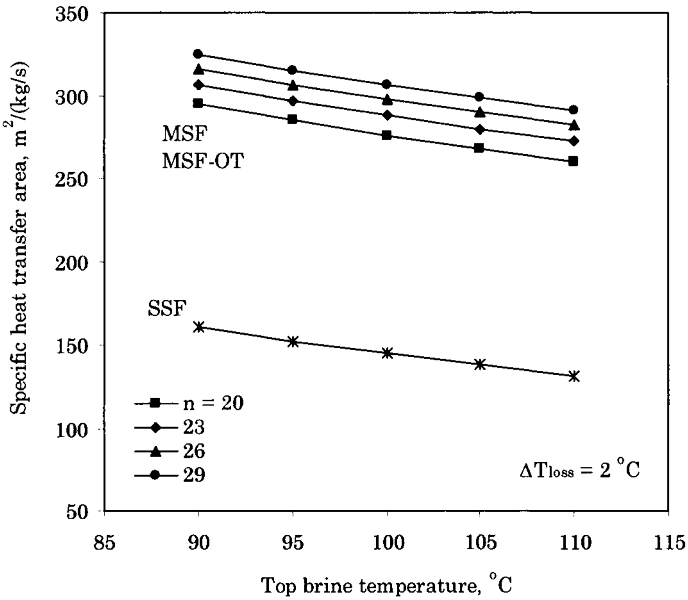

Fig. 15. Variation in specific heat transfer area for MSF, MSF-OT, and SSF as function of the top brine temperature and the number of stages

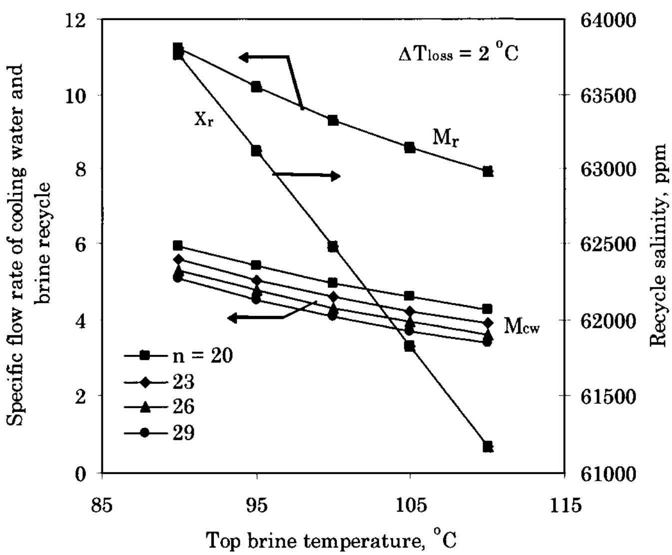

Fig. 16. Variation in specific cooling water flow rate, specific recycle flow rate, and salinity of brine blow down for MSF as a function of number of stages and the top brine temperature.

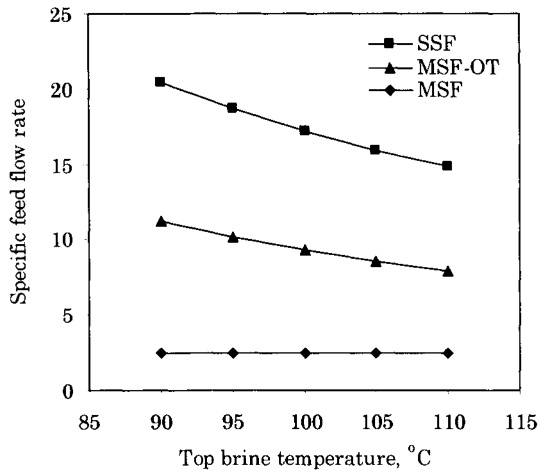

Fig. 17. Variation in specific feed flow rate for MSF, MSF-OT, and SSF as a function of the top brine temperature.

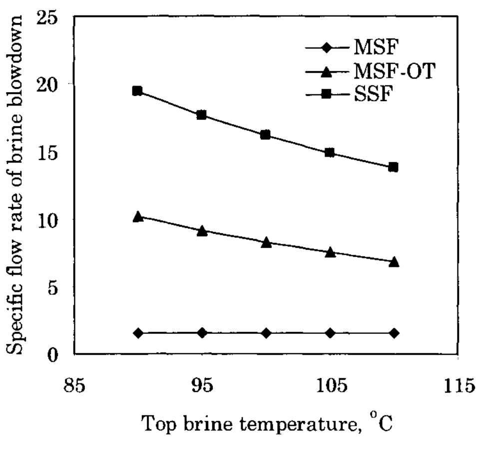

Fig. 18. Variation in specific flow rate of brine blow down for for MSF, MSF-OT, SSF as a function of the top brine temperature.

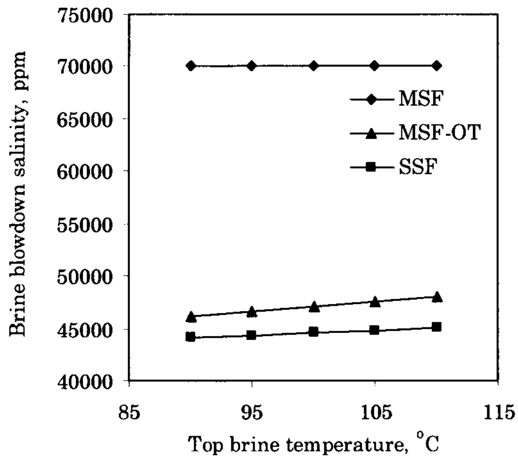

Fig. 19. Variation in salinity of brine blow down for MSF, MSF-OT, SSF as a function of the top brine temperature.

6.3.8 Summary

The following summary is made in the light of results and conclusions made for each flashing system. Figure 20 shows a schematic for the proposed configurations and their drawbacks and merits.

- A single effect flashing unit can not be used for water desalination. Specific power consumption for such system is very high. This is because the amount of steam used is larger than the amount of distillate water. In addition, the pumping power for intake seawater is very high, considering its large amount in comparison with the generated amount of distillate water. These two problems are addressed in the once through and the brine recirculation type desalination systems.
- The once through MSF system solves the performance ratio problem found in the single unit configuration, where the thermal performance ratio is increased from below one to values above 6. However, the problem of the large seawater intake is not solved. The apparent solution of this problem is to recycle part of the blow-down brine and to mix the recycle stream with the intake seawater.
- The simple mixer brine-recycle MSF system improves the thermal performance ratio, where it increases to a higher value of 8. However, several problems are found in the simple mixer configuration, i.e., high brine recirculation rates, and high salinity of the feed seawater.
- In the heat rejection stages, the intake seawater must be heated to the same temperature as that of the brine of the last stage. This is essential to prevent

thermal shock upon mixing of the two streams in the last stage. The thermal shock causes decomposition of the bicarbonate salts and formation of carbonate precipitates and carbon dioxide gas. The formed gas reduces the heat transfer efficiency around the condenser tubes and has harmful effect the steam jet ejector.

- Use of the single-stage heat rejection section is not possible, because of the intersection temperature profiles of the hot and cold streams in the preheater/condenser tubes.
- Use of the two-stage heat rejection section is not practical because of the low terminal temperature difference found in the first flashing stage. This value is well below limiting values that allow for stable and steady operation.
- The MSF system, which includes three stages in the heat rejection gives the desired system performance ratio, salinity for the recycled and blow-down brine, and practical values for the specific cooling water flow rate and heat transfer area.
- Performance of the MSF and MSF-OT systems improve at larger number of stages and higher top brine temperatures. This improvement is reflected in the increase of thermal performance ratio and the specific heat transfer area, and the reduction of the specific flow rate of cooling water. Increase of the thermal performance ratio and reduction of the specific flow rate of cooling water reduces the specific power consumption as well as the capital cost of the cooling water pumping unit. However, a higher capital cost is incurred at larger number of stages. In addition, the specific heat transfer area increases at higher number of stages. This increase has a direct effect on the capital and maintenance cost.

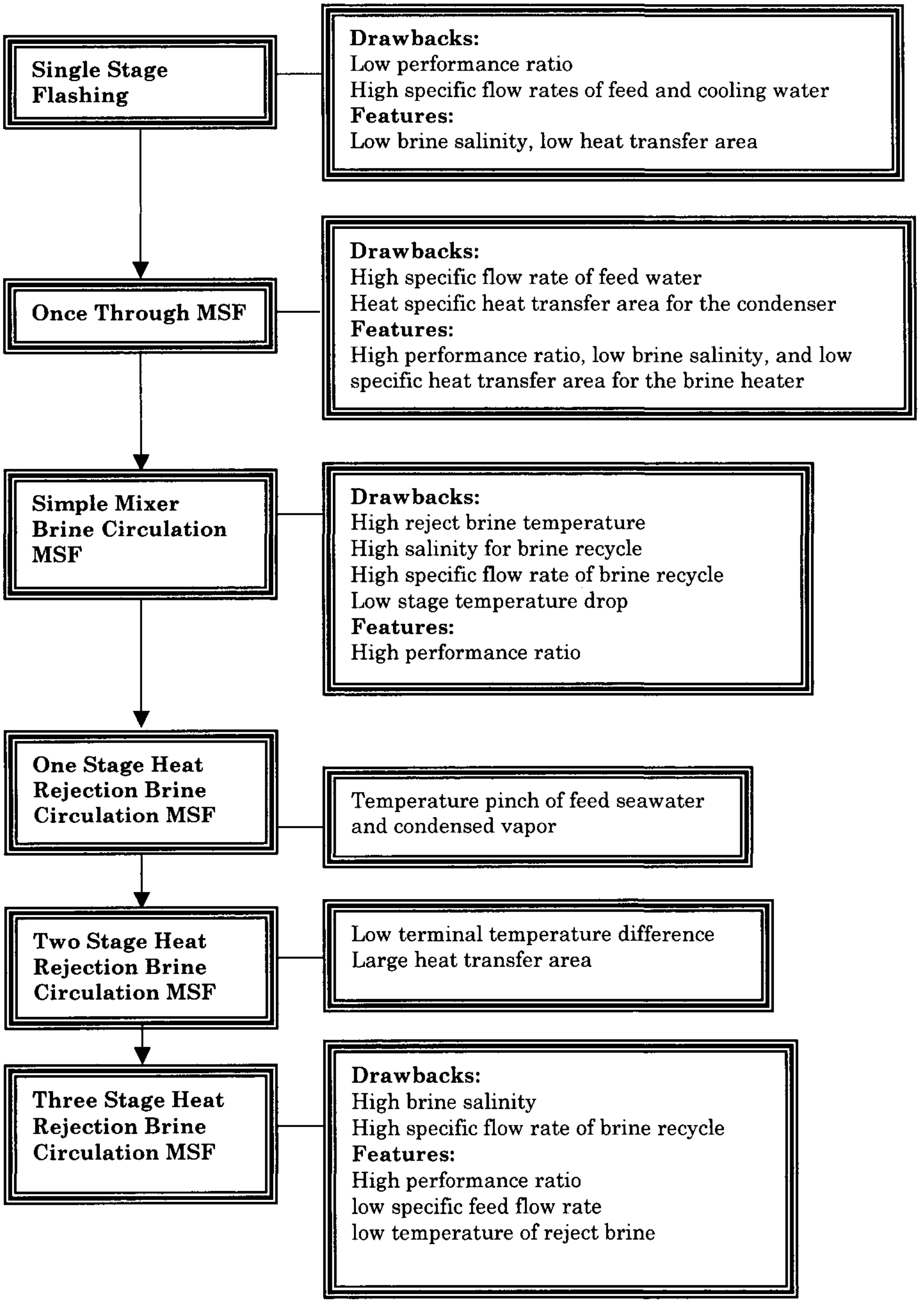

Fig. 20. Performance summary of various MSF configurations.

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Problems

Problem 1

Calculate the performance ratio, specific heat transfer area, specific flow rate of cooling water, conversion ratio, and salinity of brine blowdown for a single stage flash desalination unit operating at the following conditions:
- Feed salinity = 45000 ppm
- Feed temperature = 25 ^{o}C
- Heating steam temperature = 90 ^{o}C
- Production capacity = 1 kg/s
- Brine blowdown temperature = 35 ^{o}C
- Top brine temperature = 80 ^{o}C
- Terminal temperature difference in the condenser = 3 ^{o}C
- Thermodynamic losses = 2 ^{o}C.

Problem 2

An MSF-OT system operates at the following conditions
- Feed salinity = 45000 ppm
- Heating steam temperature = 100 ^{o}C
- Feed temperature = 25 ^{o}C
- Production capacity = 1 kg/s
- Brine blowdown temperature = 35 ^{o}C
- Top brine temperature = 90 ^{o}C
- Terminal temperature difference in the condenser = 3 ^{o}C
- Thermodynamic losses = 2 ^{o}C.

Calculate the system performance parameters if the number of stages is equal to 30.

Problem 3

Compare the performance of MSF-OT and MSF-M systems at the following conditions:
- Feed salinity = 45000 ppm
- Heating steam temperature = 112 ^{o}C
- Feed temperature = 28 ^{o}C
- Production capacity = 1 kg/s
- Brine blowdown temperature = 38 ^{o}C
- Top brine temperature = 105 ^{o}C
- Terminal temperature difference in the condenser = 3 ^{o}C

- Thermodynamic losses = 2 ^{o}C.
- Number of stages = 40.

Problem 4

A brine circulation MSF system has the following operating data
- Feed salinity = 57000 ppm
- Brine blowdown salinity = 70000 ppm
- Heating steam temperature = 116 ^{o}C
- Production capacity = 1 kg/s
- Brine blowdown temperature = 40 ^{o}C
- Feed temperature = 30 ^{o}C
- Top brine temperature = 106 ^{o}C
- Terminal temperature difference in the condenser = 3 ^{o}C
- Number of stages = 24 (with 3 stages in the heat rejection section).

Compare the system performance if the thermodynamic losses are equal to 1.5 ^{o}C.

Problem 5

A brine circulation MSF system operates at the following conditions:
- Feed salinity = 34000 ppm
- Feed temperature = 25 ^{o}C
- Heating steam temperature = 100 ^{o}C
- Production capacity = 1 kg/s
- Brine blowdown temperature = 35 ^{o}C
- Top brine temperature = 90 ^{o}C
- Terminal temperature difference in the condenser = 3 ^{o}C
- Thermodynamic losses = 1.5 ^{o}C

Calculate the system performance parameters for the following conditions:

a) Number of stages 40 with 3 stages in the heat rejection section.
b) Number of stages 19 with 3 stages in the heat rejection section.

6.4. *Once Through MSF*

The flow diagram of the MSF-OT is shown in Fig. 21. As is shown process is primarily formed of the flashing stages and the brine heater. In this system, the flashing stages forming the heat rejection section are eliminated. This simplifies the process layout and system operation. Irrespective, the MSF-OT is found on a limited industrial scale because of the following drawbacks:

- Absence of control system on the temperature of the feed seawater. This limits the use of the MSF-OT process in regions with large seasonal temperature changes. For example, the seawater temperature in the Gulf area drops to low values of 15 oC during the winter. Operation at these conditions would result in drastic reduction in the system performance ratio or the production capacity during wintertime. A thermal performance ratio of 3 is reported for the MSF-OT units during winter operation.
- Reduction in the thermal performance ratio at low intake seawater temperatures can be met by increasing the temperature range or reduction of the last stage temperature. At low temperatures the specific volume of the flashing vapor increases, which would require increasing the flash chamber size. This is necessary to keep the vapor velocity to values below 4 m/s, which limits entrainment of brine droplets.

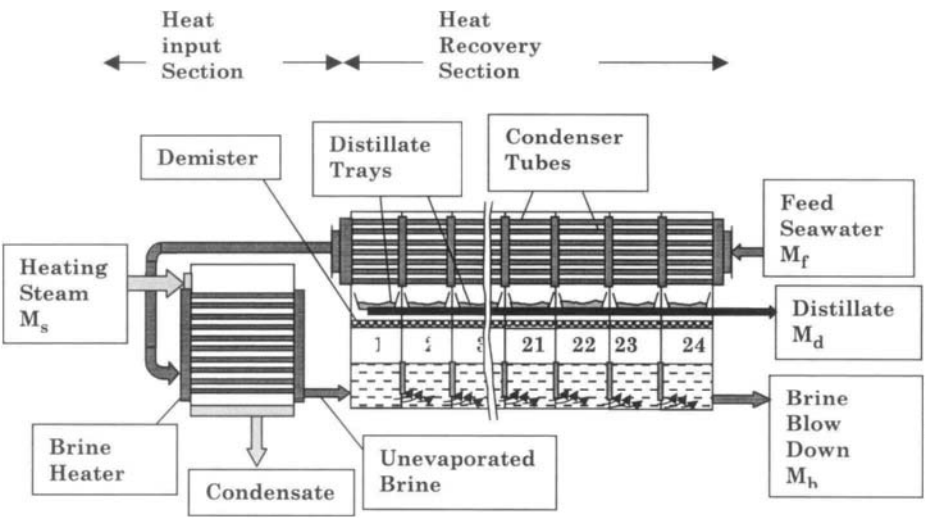

Fig. 21. Multistage flash desalination once through process.

- The flow rate ratio of the feed to product in the MSF-OT process is approximately 10:1. On the other hand, this ratio in the brine circulation MSF is 2.5:1. As a result, the MSF-OT system would consume a larger amount of chemicals for scale, foaming, and corrosion control.

Irrespective of the drawbacks of the MSF-OT process its use can be prove useful especially in equatorial regions, where the seawater temperature remains nearly constant throughout the year.

6.4.1 Process Description

Schematic of the brine circulation MSF process is shown in Fig. 21 and process variables in the brine heater and the flashing stages are shown in Fig. 22. Details of the MSF process are described below:

- The feed seawater (M_f) is deaerated and chemically before being last flashing stage in the heat rejection section, where it flows from stage (n) to stage.
- The seawater temperature increases due to absorption of the latent heat of the condensing fresh water vapor.

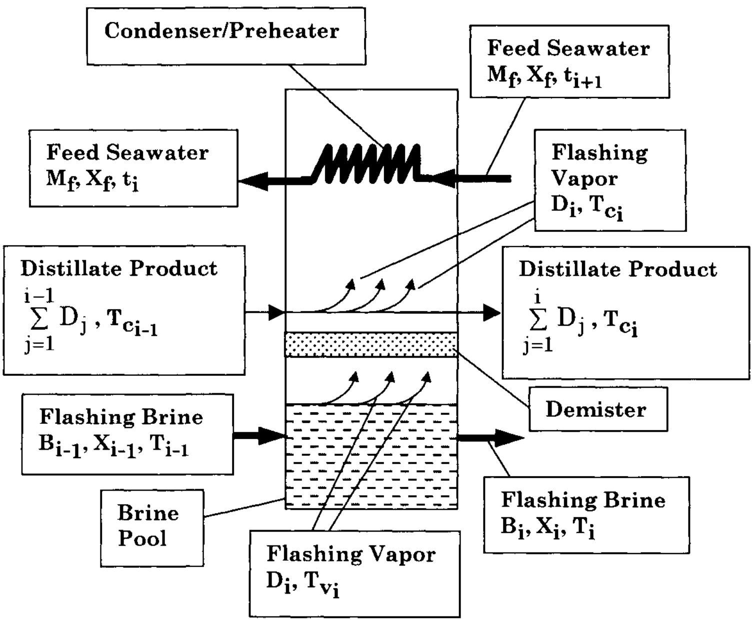

Fig. 22a. Schematics of model variables in flashing stage

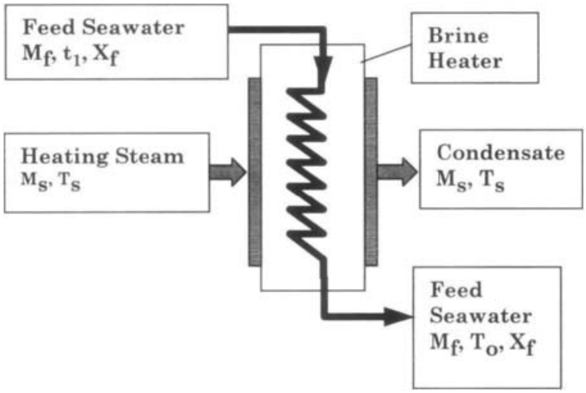

Fig. 22b. Schematics of model variables in brine heater

- The feed seawater (M_f) enters the brine heater tubes, where the heating steam (M_s) is condensed on the outside surface of the tubes. The feed seawater (M_f) absorbs the latent heat of condensing steam and its temperature increases to its maximum design value known as the top brine temperature (T_o). This value depends on the nature of chemicals used to control the scale formation.
- The feed seawater (M_f) enters the flashing stages, where a small amount of fresh water vapor is formed by brine flashing in each stage. The flashing process takes place due to decrease in the stage saturation temperature and causes the reduction in the stage pressure.
- In each stage, the flashed off vapors condenses on the outside surface of the condenser tubes, where the feed seawater (M_f) flows inside the tubes from the cold to the hot side of the plant. This heat recovery improves the process efficiency because of the increase in the feed seawater temperature.
- The condensed fresh water vapor outside the condenser tubes accumulates across the stages and forms the distillate product stream (M_d). This stream cascades in the same direction of the flashing brine from stage to stage and is withdrawn from the last stage.
- The flashing process and vapor formation is limited by increase in the specific vapor volume at lower temperatures and difficulties encountered for operation at low pressures. Common practice limits the temperature of the last stage to range of 30 to 40 °C, for winter and summer operation, respectively. Further reduction in these temperatures results in drastic increase of the stage volume and its dimensions.
- In MSF, most of flashing stages operating at temperatures below 100 °C have vacuum pressure. This increases the possibilities of in-leakage of the outside air. Also, trace amounts of dissolved gases in the flashing brine, which are not

removed in the deaerator or formed by decomposition of $CaHCO_3$. At such conditions, air and other gases are non-condensable and its presence in the system may result in severe reduction in the heat transfer rates within the chamber, increase of the tendency for corrosion, and reduction of the flashing rates. This condition necessitates proper venting of the flashing stages to enhance the flashing process and to improve the system efficiency.

- Treatment of the feed seawater (M_f) is limited simple screening and filtration. On the other hand, treatment of the feed seawater stream is more extensive and it includes dearation and addition of chemicals to control scaling, foaming, and corrosion.

6.4.2 Mathematical Model

The MSF-OT simplified model is a very useful tool, which can be used to obtain quick design data, evaluate system performance, and develop a good initial guess for more detailed mathematical models. The simplified model does not need iterative solution and requires minimal computational effort. Development of the simplified model is based on the following assumptions:

- Constant and equal specific heat for all liquid streams, Cp.
- Equal temperature drop per stage for the flashing brine.
- Equal temperature drop per stage for the feed seawater.
- The latent heat of vaporization in each stage is assumed equal to the average value for the process.
- Effects of the non-condensable gases have negligible effect on the heat transfer process.
- Effects of the boiling point rise and non-equilibrium losses on the stage energy balance are negligible, however, their effects are included in the design of the condenser heat transfer area.

The simplified model includes the following elements:

- Overall material balance.
- Stages and condensers temperature profiles.
- Stage material and salt balance.
- Condensers and brine heater heat transfer area.
- Stage dimensions.
- Performance parameters.

The following sections include the model equations for each of the above items:

Overall Material Balance

The MSF-OT flow diagram shows one input stream, the feed seawater, M_f, and two output streams, the distillate product, M_d, and the rejected brine, M_b.

Therefore, the excess energy added to the system in the brine heater is removed in brine blowdown and distillate product streams.

The overall material balance equations is given by

$$M_f = M_d + M_b \tag{31}$$

Where M is the mass flow rate and the subscript b, d, and defines the brine, distillate, and feed. The overall salt balance is given by

$$X_f M_f = X_b M_b \tag{32}$$

Where X is the salt concentration. Equation (32) assumes that the distillate is salt free.

Stages and Condensers Temperature Profiles

The temperature distribution in the MSF-OT system is defined in terms of four temperatures; these are the temperatures of the steam, T_s, the brine leaving the preheater (top brine temperature), T_o, the brine leaving the last stage, T_n, and the feed seawater, T_f. A linear profile for the temperature is assumed for the flashing brine and the seawater flowing inside the condenser tubes. The temperature drop per stage, ΔT, is obtained from the relation

$$\Delta T = (T_o - T_n)/n \tag{33}$$

where n is the number of stages. Therefore, the temperature in the first stage is given by

$$T_1 = T_o - \Delta T$$

As for the second stage temperature it is equal to

$$T_2 = T_1 - \Delta T$$

Substituting for T_1 in the above equation gives

$$T_2 = T_o - \Delta T - \Delta T = T_o - 2\,\Delta T$$

The same procedure is repeated for subsequent stages and a general expression is developed for the temperature of stage i

$$T_i = T_o - i\,\Delta T \tag{34}$$

The temperature of the feed seawater, M_f, which flows inside the condenser tubes, increases by Δt in the condenser of each stage. This temperature increase, Δt, is equal to the decrease in the brine temperature in each stage, ΔT. This result is arrived at by performing an energy balance on stage i, which gives

$$D_i\, Cp\, T_{v_i} + B_i\, Cp\, T_i - D_{i+1}\, Cp\, T_{v_{i+1}} - B_{i+1}\, Cp\, T_{i+1} = M_f\, C_p\, (t_i - t_{i+1})$$

Assuming the temperature difference, $T_i - T_{v_i}$, is small and has a negligible effect on the stage energy balance. Thus, the above equation reduces to

$$(D_i + B_i)\, Cp\, T_i - (D_{i+1} + B_{i+1})\, Cp\, T_{i+1} = M_f\, C_p\, (t_i - t_{i+1})$$

Recalling that the sum $(D_i + B_i)$ in each stage is equal to M_f, would simplify the above equation to

$$M_f\, Cp\, T_i - M_f\, Cp\, T_{i+1} = M_f\, C_p\, (t_i - t_{i+1})$$

Elimination of the like terms on both sides of the equation gives the pursued relation, thus,

$$T_i - T_{i+1} = t_i - t_{i+1}$$

or

$$\Delta T_i = \Delta t_i$$

The seawater temperature, which leaves the condenser of the first stage, is then defined by

$$t_1 = T_f + n\, \Delta t$$

The seawater temperature leaving the condenser of the second stage, T_2, is less than T_1 by Δt, where

$$t_2 = t_1 - \Delta t$$

Substituting for T_1 in the above equation gives

$$t_2 = T_f + (n - 1)\, \Delta t$$

Similar to equation (34), a general equation is obtained for the condenser temperature in stage i

$$t_i = T_f + (n - (i - 1))\,\Delta t \tag{35}$$

Stage Material and Salt Balance

The amount of flashing vapor formed in each stage obtained by conservation of energy within the stage, where the latent consumed by the flashing vapor is set equal to the decrease in the brine sensible heat. This is

$$D_1 = y\,M_f$$

where D_1 is the amount of flashing vapor formed in the first stage, M_f is the feed seawater flow rate, and y is the specific ratio of sensible heat and latent heat and is equal to

$$y = Cp\,\Delta T/\lambda_{av} \tag{36}$$

Where Cp is the specific heat capacity and λ_{av} is the average latent heat calculated at the average temperature

$$T_{av} = (T_o + T_n)/2 \tag{37}$$

The amount of distillate formed in the second stage is equal to

$$D_2 = y\,(M_f - D_1)$$

Substituting the value of D_1 in the above equation gives

$$D_2 = y\,(M_f - y\,M_f)$$

This simplifies to

$$D_2 = M_f\,y\,(1 - y)$$

The balance equations for D_2 and D_3 will reveal the general form for the formula of D_i. The D_3 balance is

$$D_3 = y\,(M_f - D_1 - D_2)$$

Substituting for the values of D_1 and D_2 in the above equation gives

$D_3 = y\ (M_f - M_f\ y - M_f\ y\ (1 - y))$

Taking (M_r) as a common factor in the above equation gives

$D_3 = M_f\ y\ (1 - y - y + y^2)$

This simplifies to

$D_3 = M_f\ y\ (1 - y)^2$

Accordingly, the resulting general formula for D_i is

$$D_i = M_f\ y\ (1 - y)^{(i-1)} \qquad (38)$$

The total distillate flow rate is obtained by summing the values of D_i for all stages. The summation is performed in steps in order to obtain a closed form equation. Therefore, the summation of D_1 and D_2 gives

$$\begin{aligned} D_1 + D_2 &= M_f\ (y + y\ (1 - y)) \\ &= M_f\ (2\ y - y^2) \\ &= M_f\ (1 - (1 - y)^2) \end{aligned}$$

Addition of D_3 to the above gives

$D_1 + D_2 + D_3 = M_f\ ((2\ y - y^2) + y(1 - y)^2)$

This simplifies to

$$\begin{aligned} D_1 + D_2 + D_3 &= M_f\ (2\ y - y^2 + y - 2\ y^2 + y^3) \\ &= M_f\ (3\ y - 3\ y^2 + y^3) \\ &= M_f\ (3\ y - 3\ y^2 + y^3) \\ &= M_f\ (1 - (1 - y)^3) \end{aligned}$$

Comparison of the summations of D_1+D_2 and D_1+D_2+D_3 gives the general form for the total summation of the distillate formed in all stages, M_d, which is given by

$$M_d = M_f\ (1 - (1 - y)^n) \qquad (39)$$

Equation (39) is used to calculate the feed flow rate, since the distillate flow is always specified in a design problem.

The flow rate of the brine stream leaving stage (i) is given by

$$B_i = M_f - \sum_{k=1}^{i} D_k \tag{40}$$

The salt concentration in the brine stream leaving stage i is given by

$$X_i = M_f X_f / B_i \tag{41}$$

The flow rate of the heating steam, Ms, is obtained the energy balance equation for the brine heater, where

$$M_s \lambda_s = M_f Cp (T_o - t_1)$$

The above equation is arranged to calculate M_s

$$M_s = M_f Cp (T_o - t_1)/ \lambda_s \tag{42}$$

Brine Heater and Condensers Heat Transfer Area

The brine heater area is given by

$$A_b = M_s \lambda_s/(U_b (LMTD)_b) \tag{43}$$

Where LMTD is given by

$$(LMTD)_b=((T_s-T_o)-(T_s-t_1))/\ln((T_s-T_o)/(T_s-t_1)) \tag{44}$$

and U_b is given by

$$U_b=1.7194+3.2063x10^{-3}\, T_s+1.5971x10^{-5}\, (T_s)^2 - 1.9918x10^{-7}\, (T_s)^3 \tag{45}$$

The heat transfer area for the condenser in each stage is assumed equal. Therefore, the calculated heat transfer area for the first stage is used to obtain the total heat transfer area in the plant. The condenser heat transfer area in the first stage is obtained from

$$A_c = M_f Cp (t_1 - t_2)/(U_c (LMTD)_c) \tag{46}$$

where

$$U_c = 1.7194 + 3.2063\text{x}10^{-3}\, T_{v_1} + 1.5971\text{x}10^{-5} T_{v_1}{}^2 - 1.9918\text{x}10^{-7} T_{v_1}{}^3 \quad (47)$$

$$T_{v_1} = T_1 - BPE_1 - NEA_1 - \Delta T_{d_1} \quad (49)$$

and

$$(LMTD)_c = ((T_{v_1} - t_1) - (T_{v_1} - t_2))/\ln((T_{v_1} - t_1)/(T_{v_1} - t_2)) \quad (50)$$

In the above equations (BPE) is the boiling point elevation, (NEA) is the non-equilibrium allowance, (T_v) is the condensing vapor temperature, (ΔT_d) is the temperature drop in the demister, and (U_c) is the condenser overall heat transfer coefficient. Expressions for correlations used calculate the BPE, NEA, and ΔT_d, are given in the appendix.

The total heat transfer in the plant is obtained by summing the heat transfer area for all condensers and the brine heater

$$A = A_b + n\, A_c \quad (51)$$

Stage Dimensions

Calculations of the stage dimensions include the gate height, the height of the brine pool, the stage width, and the stage length. The length of all stages is set equal to the length of the last stage and the width of all stages is set equal to the width of the first stage. The height of the brine pool must be higher than the gate height, this is necessary to prevent bypass of the vapors between stages (vapor blow through). The gate height (GH) is obtained in terms of the stage pressure drop (ΔP), the brine density (ρ_b), the weir friction coefficient (C_d), the stage width (W), and the feed flow rate (M_f). For stage i the gate height is

$$GH_i = (M_f - \sum_{j=1}^{i-1} D_j)\,(2\,\rho_{b_i}\,\Delta P_i)^{(-0.5)}/(C_d\, W) \quad (52)$$

The brine pool height is set higher than the gate height by 0.2 m.

$$H_i = 0.2 + GH_i \quad (53)$$

Where

$\Delta P_i = P_i - P_{i+1}$

$$W = M_f / V_b \tag{54}$$

where P_i and P_{i+1} are the pressures in stages i and i+1, and V_b is the brine mass velocity per chamber width. The length of the last stage is determined as a function of the vapor flow rate, D_n, the vapor density, ρ_{v_n}, the vapor allowable velocity, V_{v_n}, and the stage width, W. This is

$$L = D_n/(\rho_{v_n} V_{v_n} W) \tag{55}$$

The cross section area for each stage, A_s, is then calculated

$$A_s = L\ W \tag{56}$$

Performance Parameters

The system performance parameters are defined by the thermal performance ratio, PR and the specific heat transfer area, sA. The performance ratio is the defined as the amount of distillate product produced per unit mass of the heating steam. This is

$$PR = M_d/M_s \tag{57}$$

The specific heat transfer area is defined by

$$sA = (A_b + n\ A_c)/M_d \tag{58}$$

Solution Method

Solution of the MSF-OT simplified model is non-iterative. The solution proceeds as follows:

- The average stage temperature, T_{av}, is calculated from Eq. (37) and the corresponding latent heat value is obtained from the correlation given in the appendix.
- The temperature drop per stage, ΔT, is calculated from Eq. (33).
- The ratio of the stage sensible and latent heat, y, is calculated from Eq. (36).
- The feed flow rate is calculated from Eq. (39).
- The flow rate of the brine blow down, M_b, as well as its salinity, X_b, are obtained from Eqs. (31) and (32).

- The brine temperatures leaving stages 1 and 2, T_1 and T_2, are calculated from Eq. (34).
- The seawater temperatures leaving the condensers of the first and second stages, t_1 and t_2, are determined from Eq. (35). Also, the seawater temperature leaving the condenser of the last stage, t_n, is calculated from Eq. (35).
- The heat transfer area for the brine preheater (A_b), the first stage condenser (A_c) are calculated from Eqs. (43) and (46), respectively.
- The stage length, width, and cross section areas are determined from Eqs. (54-56). Also, the height of the brine pool, H, and the gate height for stage i are obtained from Eqs. (52) and (53).
- The system performance parameters, PR and sA, are calculated from Eqs. (57) and (58).

The above solution procedure requires specification of the following variables:

- Total distillate flow rate, M_d.
- Total number of stages.
- Feed seawater temperature, T_f.
- Top brine temperature, T_o.
- Steam temperature, T_s.
- Temperature of brine blowdown, T_n.
- Intake seawater salt concentration, X_f.
- Heat capacity of liquid streams, Cp.
- Weir friction coefficient, C_d.
- Vapor velocity in the last stage, V_{vn}.
- Brine mass flow rate per stage width, V_b.

6.4.3 Case Study

An MSF system with 24 stages is used to produce 7.2 MGD of product water. The following specifications are made to obtain the system design parameters and performance characteristics:

- Feed seawater temperature, $T_f = 25\ ^oC$.
- Steam temperature, $T_s = 116\ ^oC$.
- Top brine temperature, $T_o = 106\ ^oC$.
- Brine temperature in the last stage, $T_n = 40\ ^oC$.
- Heat capacity of liquid streams, $C_p = 4.18$ kJ/kg ^oC.
- Salinity of feed seawater, $X_f = 42000$ ppm.

- Vapor velocity in the last stage, $V_{v_{24}} = 6$ m/s
- Brine mass flow rate per stage width, $V_b = 180$ kg/ms.
- Weir friction coefficient, $C_d = 0.5$.

Before proceeding, the product volume flow rate is converted to mass rate in SI units; this is necessary for solution of the energy balance equations. The distillate flow rate in kg/s is

$$M_d = \frac{\left(7.2\left(\frac{MG}{d}\right)\right)\left(10^6\left(\frac{G}{MG}\right)\right)\left(\frac{1}{24x3600}\left(\frac{d}{s}\right)\right)}{\left(\frac{1}{219.96}\left(\frac{m^3}{G}\right)\right)10^3\left(\frac{kg}{m^3}\right)}$$

$M_d = 378.8$ kg/s

Calculation of the y ratio from Eq. (36) is preceded by evaluation of T_{av} and λ_{av}. The T_{av} value is

$$\begin{aligned} T_{av} &= (T_o + T_n)/2 \\ &= (106+40)/2 \\ &= 73\ ^oC \end{aligned}$$

At this temperature λ_{av} is equal to 2330.1 kJ/kg from the correlation given in the appendix. The value of y is calculated

$$\begin{aligned} y &= Cp\ \Delta T/\lambda_{av} \\ &= (4.18)\ (2.75)/2330.1 \\ &= 4.933x10^{-3} \end{aligned}$$

The feed flow rate is obtained from equation (39)

$$\begin{aligned} M_f &= M_d/(1 - (1 - y)^n) \\ &= 378.8/(1 - (1 - 4.933x10^{-3})^{24}) \\ &= 3384.8 \text{ kg/s} \end{aligned}$$

Arranging equations (31) and (32) gives the following expression, which is used to calculate the salinity of the brine blow down

$$\begin{aligned} X_b &= M_f X_f/(M_f - M_d) \\ &= (3384.8)(42000)/(3384.8 - 378.8) \end{aligned}$$

$= 47292.6$ ppm

The flow rate of the blow-down brine is then obtained from the overall balance given in equation (31)

$$\begin{aligned} Mb &= Mf - Md \\ &= 3384.8 - 378.8 \\ &= 3006 \text{ kg/s} \end{aligned}$$

The temperature drop in each effect is obtained from Eq. (33), where

$$\begin{aligned} \Delta T &= (T_o - T_n)/n \\ &= (106 - 40)/24 \\ &= 2.75\ ^oC \end{aligned}$$

This value is used to calculate the temperatures of the first and second stages, T_1 and T_2, where

$$\begin{aligned} T_1 &= T_o - \Delta T \\ &= 106 - 2.75 \\ &= 103.25\ ^oC \end{aligned}$$

As for the second stage temperature it is equal to

$$\begin{aligned} T_2 &= T_1 - \Delta T \\ &= 103.25 - 2.75 \\ &= 100.5\ ^oC \end{aligned}$$

Also the temperatures of the seawater leaving the condensers in the first and second stages are calculated. This is

$$\begin{aligned} t_1 &= T_f + n\ \Delta t \\ &= 25 + (24)\ (2.75) \\ &= 91\ ^oC \end{aligned}$$

and

$$\begin{aligned} t_2 &= t_1 - \Delta t \\ &= 91 - 2.75 \\ &= 88.25\ ^oC \end{aligned}$$

The steam flow rate is calculated from Eq. (42)

$$M_s = M_f\, Cp\, (T_o - t_1)/\lambda_s$$
$$= (3384.8)\,(4.18)\,(106 - 91)/(2222.33)$$
$$= 95.49 \text{ kg/s}$$

The heat transfer area for the brine heater, A_b, is calculated from Eq. (43). This requires calculations of the logarithmic mean temperature difference $(LMTD)_b$ and the overall heat transfer coefficient. The value of $(LMTD)_b$ is obtained from Eq. (44), which gives

$$(LMTD)_b = (t_1 - T_o)/\ln((T_s - T_o)/(T_s - t_1))$$
$$= (91 - 106)/\ln((116 - 106)/(116 - 91))$$
$$= 16.37\ {}^oC$$

The overall heat transfer coefficient is obtained from Eq. (45) where

$$U_b = 1.7194 + 3.2063\text{x}10^{-3}\, T_s + 1.5971\text{x}10^{-5}\, (T_s)^2 - 1.9918\text{x}10^{-7}(T_s)^3$$
$$= 1.7194 + 3.2063\text{x}10^{-3}\,(116) + 1.5971\text{x}10^{-5}\,(116)^2 - 1.9918\text{x}10^{-7}\,(116)^3$$
$$= 2 \text{ kW/m}^2\ {}^oC$$

The brine heater area, A_b, is then calculated from Eq. (43)

$$A_b = M_s\, \lambda_s/(U_b\, (LMTD)_b)$$
$$= (95.49)(2222.33)/((2)(16.37))$$
$$= 6481.68 \text{ m}^2$$

The condenser area, A_c, is determined for the first stage. This requires calculations of the vapor condensation temperature, T_{v_1}, the logarithmic mean temperature difference $(LMTD)_c$, and the overall heat transfer coefficient, U_c. The vapor temperature is given by Eq. (49), where

$$T_{v_1} = T_1 - BPE_1 - NEA_1 - \Delta T_{d_1}$$

The boiling point elevation is calculated from the correlation given in the appendix. This requires calculations of X_1, which is obtained in the following sequence

$$D_1 = y\, M_f$$
$$= (4.933\text{x}10^{-3})(3384.8)$$
$$= 16.697 \text{ kg/s}$$

$$B_1 = M_f - D_1$$

$= 3384.8 - 16.697$
$= 3368.1$ kg/s

$X_1 = M_f X_f/B_1$
$= (3384.8)(42000)/3368.1$
$= 42208$ ppm

The values of B and C in the correlation for the boiling point elevation are

$$B = (6.71 + 6.34x10^{-2}(T_1) + 9.74x10^{-5}(T_1)^2)10^{-3}$$
$$= (6.71 + 6.34x10^{-2}(103.25) + 9.74x10^{-5}(103.25)^2)10^{-3}$$
$$= 0.0143$$

$$C = (22.238 + 9.59x10^{-3}(T_1) + 9.42x10^{-5}(T_1)^2)10^{-8}$$
$$= (22.238 + 9.59x10^{-3}(103.25) + 9.42x10^{-5}(103.25)^2)10^{-8}$$
$$= 2.423x10^{-7}$$

Substituting the values of B and C in the BPE correlation gives

$$BPE_1 = X_1\ (B + (X_1)(C))10^{-3}$$
$$= 42208\left(0.0143 + (42208)\left(2.423x10^{-7}\right)\right)10^{-3}$$
$$= 1.035\ ^oC$$

The non-equilibrium allowance, NEA_1, in the first stage is calculated from the correlation given in the appendix for the MSF system. This involves calculations of the gate height, GH_1, the height of the brine pool, H_1, the stage width, W, the stage pressure drop, P_1-P_2, and the brine density. The width of the first stage is given by

$W = M_f / V_b$
$= 3384.8/180$
$= 18.8$ m

The stage length is calculated for the last stage, where

$D_{24} = 14.9$ kg/s

$\rho_{v_n} = 0.0512$ kg/m^3,

$$L = D_n/(\rho_{v_n} V_{v_n} W)$$
$$= 14.9/((0.0512)(6)(18.8))$$
$$= 2.58 \text{ m}$$

The brine density in the first stage is 1002.413 kg/m^3, which is obtained from the correlation given the appendix at a salinity of 42208 ppm and a temperature of 103.25 °C. The pressures of the first and second stage are obtained from the saturation pressure correlations, where, at T_1 = 103.25 °C, P_1 = 113.72 kPa, and at T_2 = 100.5 °C, P_2 = 103.23 kPa. The resulting gate height in the first stage, GH_1, is calculated from Eq. (52), where

$$GH_1 = M_f (2\, \rho_{b_1}\, \Delta P_1)^{(-0.5)}/(C_d\, W)$$
$$=(3384.8)((2)(1002.41)(113.72-103.23)\text{x}10^3)^{(-0.5)}/((0.5)(18.8)$$
$$= 0.078 \text{ m}$$

It should be noted that the pressure drop in the above equation is in Pa and not kPa. The corresponding brine pool height is obtained by simply adding 0.2 m to the value of GH_1, or

$$H_1 = 0.278 \text{ m}$$

The non-equilibrium allowance is then calculated using the correlation given in the appendix

$$NEA_1 = (0.9784)^{T_o}\, (15.7378)^{H_1}\, (1.3777)^{V_b \text{x} 10^{-6}}$$
$$= (0.9784)^{(106)}\, (15.7378)^{(0.278)}\, (1.3777)^{(180\text{x}10^{-6})}$$
$$= 0.213\ ^{o}\text{C}$$

The temperature drop in the demister is assumed negligible in comparison with the values of BPE_1 and NEA_1. Therefore, the vapor temperature in the first is

$$T_{v_1} = T_1 - BPE_1 - NEA_1 - \Delta T_{d_1}$$
$$= 103.25 - 1.035 - 0.213 - 0$$
$$= 102.002\ ^{o}\text{C}$$

The vapor temperature, T_{v_1}, is used to calculate U_c and $(LMTD)_c$, where

$$(LMTD)_c = (t_2\text{-}t_1)/\ln((T_v-t_1)/(T_v-t_2))$$
$$= (2.75)/\ln((102.002-91)/(102.002-88.25))$$
$$= 12.32\ ^{o}\text{C}$$

$U_c = 1.7194 + 3.2063\text{x}10^{-3}\, T_{v_1} + 1.5971\text{x}10^{-5}\, (T_{v_1})^2 - 1.9918\text{x}10^{-7}(T_{v_1})^3$
$= 1.7194 + 3.2063\text{x}10^{-3}\, (101.207) + 1.5971\text{x}10^{-5}\, (101.207)^2$
$- 1.9918\text{x}10^{-7}\, (101.207)^3$
$= 2$ kW/m^2 ^{o}C

$A_c = M_f\, Cp\, (t_2\text{-}t_1)/(U_c\, (LMTD)_c)$
$= ((3384.8)\, (4.18)\, (97.75\text{-}95))/((2)\, (12.32))$
$= 1579$ m^2

The total heat transfer area is obtained from

$A = A_b + n\, A_c$
$= 6481.68 + (24)\, (1579)$
$= 44377.7$ m^2

Performance Parameters

Performance Ratio

$PR = M_d/M_s$
$= 378.8/95.49$
$= 3.96$

The specific heat transfer area

$sA = A/M_d$
$= (44377.7)/378.8$
$= 117.2$ m^2/(kg/s)

Solution Summary

Flow Rates

$M_d = 378.8$ kg/s

$M_b = 3006$ kg/s

$M_f = 3384.8$ kg/s

$M_s = 52.52$ kg/s

Heat Transfer Areas

$A_b = 6481.68 \text{ m}^2$

$A_c = 1579 \text{ m}^2$

$A = 44377.7 \text{ m}^2$

Stage Dimensions

$W = 18.8$ m

$L = 2.56$ m

$GH_1 = 0.078$ m

$H_1 = 0.278$ m

Performance Parameters

$PR = 3.96$

$sA = 117.2 \text{ m}^2/(\text{kg/s})$

Stage Profiles

The MSF-OT simplified model is used to calculate the temperature and concentration profiles for the 24 stages. The calculations also include the brine and distillate flow rate, the gate height and the brine level in each stage. In the following table, the flow rates are in kg/s, the temperature in °C, the salinity in ppm, and the height in m.

6.4.3 Case Study

Stage	D	$\sum D$	B	X	T	T_f	GH	H
1	16.70	16.70	3368.1	42208.2	103.25	91	0.078	0.278
2	16.61	33.31	3351.5	42417.5	100.5	88.25	0.081	0.281
3	16.53	49.84	3335.0	42627.7	97.75	85.5	0.084	0.284
4	16.45	66.30	3318.5	42839.1	95	82.75	0.087	0.287
5	16.37	82.67	3302.1	43051.4	92.25	80	0.09	0.29
6	16.29	98.96	3285.8	43264.9	89.5	77.25	0.094	0.294
7	16.21	115.16	3269.6	43479.3	86.75	74.5	0.097	0.297
8	16.13	131.29	3253.5	43694.9	84	71.75	0.101	0.301
9	16.05	147.34	3237.5	43911.5	81.25	69	0.105	0.305
10	15.97	163.31	3221.5	44129.2	78.5	66.25	0.11	0.31
11	15.89	179.21	3205.6	44348.0	75.75	63.5	0.114	0.314
12	15.81	195.02	3189.8	44567.8	73	60.75	0.119	0.319
13	15.74	210.75	3174.0	44788.8	70.25	58	0.124	0.324
14	15.66	226.41	3158.4	45010.8	67.5	55.25	0.13	0.33
15	15.58	241.99	3142.8	45233.9	64.75	52.5	0.136	0.336
16	15.50	257.50	3127.3	45458.2	62	49.75	0.143	0.343
17	15.43	272.92	3111.9	45683.5	59.25	47	0.15	0.35
18	15.35	288.27	3096.5	45910.0	56.5	44.25	0.157	0.357
19	15.28	303.55	3081.3	46137.6	53.75	41.5	0.165	0.365
20	15.20	318.75	3066.1	46366.3	51	38.75	0.175	0.375
21	15.12	333.87	3050.9	46596.2	48.25	36	0.185	0.385
22	15.05	348.92	3035.9	46827.2	45.5	33.25	0.197	0.397
23	14.98	363.90	3020.9	47059.3	42.75	30.5	0.211	0.411
24	14.90	378.80	3006.0	47292.6	40	27.75	0.211	0.411

Problems

Problem 1

An MSF-OT circulation plant has the following design data:

Plant capacity:	Unknown
Seawater temperature:	32 °C
Seawater salinity:	49400 ppm
Top brine temperature:	100 °C
Performance ratio:	8
Number of stages:	22
Overall heat transfer coefficient in brine heater:	2 kW/m^2 °C
Overall heat transfer coefficient in heat rejection section:	1.9 kW/m^2 °C
Overall heat transfer coefficient	

in heat recovery section:	2.4 kW/m^2 ^{o}C

Calculate the following:

- The heating steam flow rate.
- The plant capacity
- The specific heat transfer area

Problem 2

An MSF-OT plant has the following design data:

Plant capacity:	Unknown
Top brine temperature:	Unknown
Brine flow rate per chamber width:	Unknown
Number of stages:	20
Boiling temperature in last stage:	40 ^{o}C
Heat transfer area in the brine heater:	1000 m^2
Overall heat transfer coefficient in all sections:	2.527 kW/m^2 ^{o}C
Mass flow rate of heating steam:	16.782 kg/s
Heating steam temperature:	120 ^{o}C
Number of tubes in the brine heater:	1000 tube
Specific flow rate of feed water:	8.422
Diameter of tubes used in brine heater:	31.8 mm

Calculate the following:

- The plant performance ratio
- The specific heat transfer area
- The dimensions of stage 7.

Problem 3

An MSF-OT plant has the following design data:

Seawater temperature:	34 ^{o}C
Seawater salinity:	42000 ppm
Top brine temperature:	100 ^{o}C
Temperature in the last stage:	40 ^{o}C
Temperature of heating steam:	110 ^{o}C
Specific flow rate of brine circulation:	8.478
Heat transfer area in the brine heater:	80 m^2
Overall heat transfer coefficient	

in brine heater: 1.5 kW/m^2 ^{o}C

Calculate the following

- The plant performance ratio
- The terminal temperature difference of the first stage.
- The specific heat transfer area.
- The dimensions of stage 7.

Problem 4

An MSF brine circulation plant has the following design data:

Distillate flow rate:	5000 m^3/d
Seawater temperature:	30 ^{o}C
Seawater salinity:	44000 ppm
Top brine temperature:	112 ^{o}C
Temperature in the last stage:	40 ^{o}C
Temperature of heating steam:	116.7 ^{o}C
Terminal temperature difference:	3 ^{o}C
Number of stages:	20

The overall heat transfer coefficient in the brine heater or the flashing stages is given by the relation

$$U = 6.5 - 0.03(115\text{-}T)$$

With U in kW/m^2 ^{o}C and T in ^{o}C.

Calculate the following

- Thermodynamic losses in the first stage
- Plant performance ratio
- Brine heater surface area
- Flow rate of feed water.

Also, calculate the following parameters for stage number 5

- Boiling point elevation.
- Gate height
- Liquid level
- Demister temperature loss
- Stage height
- Stage length
- Preheater surface area.

Problem 5

An MSF-OT plant has the following design data:

Distillate flow rate:	22750 m^3/d
Seawater temperature:	28 ^oC
Seawater salinity:	45000 ppm
Top brine temperature:	90 ^oC
Temperature in the last stage:	40 ^oC
Temperature of heating steam:	100 ^oC
Terminal temperature difference:	3 ^oC
Number of heat rejection stages:	3
Number of heat recovery stages:	25
Width of stage 10:	16 m
Length of stage 10:	3.5 m

Calculate the following

- The temperature profile of flashing brine
- The temperature profile of seawater flowing in the preheaters
- The flow rates of heating steam, feed water, and brine blowdown.
- The pressure in stages 9,10, and 11.
- The distillate product in stage 10.
- The vapor velocity in stage 10.

6.5. Brine Circulation MSF

The MSF process with brine circulation is one of the major processes of the desalination industry. As mentioned before, the process is developed in the late fifties and since then it has gone several modifications to optimize its performance. At present, this process is considered the most suitable for large scale production capacity, where the conventional capacity of a single unit amounts to 25,000 m^3/d. Regardless, further developments remains to be necessary to improve performance and reduce product cost. Features and performance of the brine circulation process is discussed in this section, while possible process developments are proposed in the next sections.

6.5.1 Process Description

Schematic of the brine circulation MSF process is shown in Fig. 23 and process variables in the brine heater and the flashing stages are shown in Fig. 24. Details of the MSF process are described below:

- The intake seawater stream (M_f+M_{cw}) is introduced into the condenser tubes of the heat reject section, where its temperature is increased to a higher temperature by absorption of the latent heat of the condensing fresh water vapor.
- The warm stream of intake seawater is divided into two parts: the first is the cooling seawater (M_{cw}), which is rejected back to the sea and the second is the feed seawater (M_f), which is deaerated, chemically treated and then mixed in the brine pool of the last flashing stage in the heat rejection section.
- The brine recycle stream (M_r) is extracted from the brine pool of the last stage in the heat rejection section and is introduced into the condenser tubes of the last stage in the heat recovery section. As the stream flows in the condenser tubes across the stages it absorbs the latent heat of condensation from the flashing vapor in each stage.
- The brine recycle stream (M_r) enters the brine heater tubes, where the heating steam (M_s) is condensed on the outside surface of the tubes. The brine stream absorbs the latent heat of condensing steam and its temperature increases to its maximum design value known as the top brine temperature (T_o). Its value depends on the nature of chemicals used to control the scale formation.
- The hot brine enters the flashing stages in the heat recovery section and then in the heat rejection section, where a small amount of fresh water vapor is formed by brine flashing in each stage. The flashing process takes place due to decrease in the stage saturation temperature and causes the reduction in the stage pressure.

- In each stage of the heat recovery section, the flashed off vapors condenses on the outside surface of the condenser tubes, where the brine recycle stream (M_r) flows inside the tube from the cold to the hot side of the plant. This heat recovery improves the process efficiency because of the increase in the feed seawater temperature.
- The condensed fresh water vapor outside the condenser tubes accumulates across the stages and forms the distillate product stream (M_d). This stream cascades in the same direction of the flashing brine from stage to stage and is withdrawn from the last stage in the heat rejection section.
- The flashing process and vapor formation is limited by increase in the specific vapor volume at lower temperatures and difficulties encountered for operation at low pressures. Common practice limits the temperature of the last stage to range of 30 to 40 °C, for winter and summer operation, respectively. Further reduction in these temperatures results in drastic increase of the stage volume and its dimensions.
- In MSF, most of flashing stages operating at temperatures below 100 °C have vacuum pressure. This increases the possibilities of in-leakage of the outside air. Also, trace amounts of dissolved gases in the flashing brine, which are not removed in the deaerator or formed by decomposition of $CaHCO_3$. At such conditions, air and other gases are non-condensable and its presence in the system may result in severe reduction in the heat transfer rates within the chamber, increase of the tendency for corrosion, and reduction of the flashing rates. This condition necessitates proper venting of the flashing stages to enhance the flashing process and to improve the system efficiency.
- Treatment of the intake seawater (M_f+M_{cw}) is limited simple screening and filtration. On the other hand, treatment of the feed seawater stream is more extensive and it includes dearation and addition of chemicals to control scaling, foaming, and corrosion.

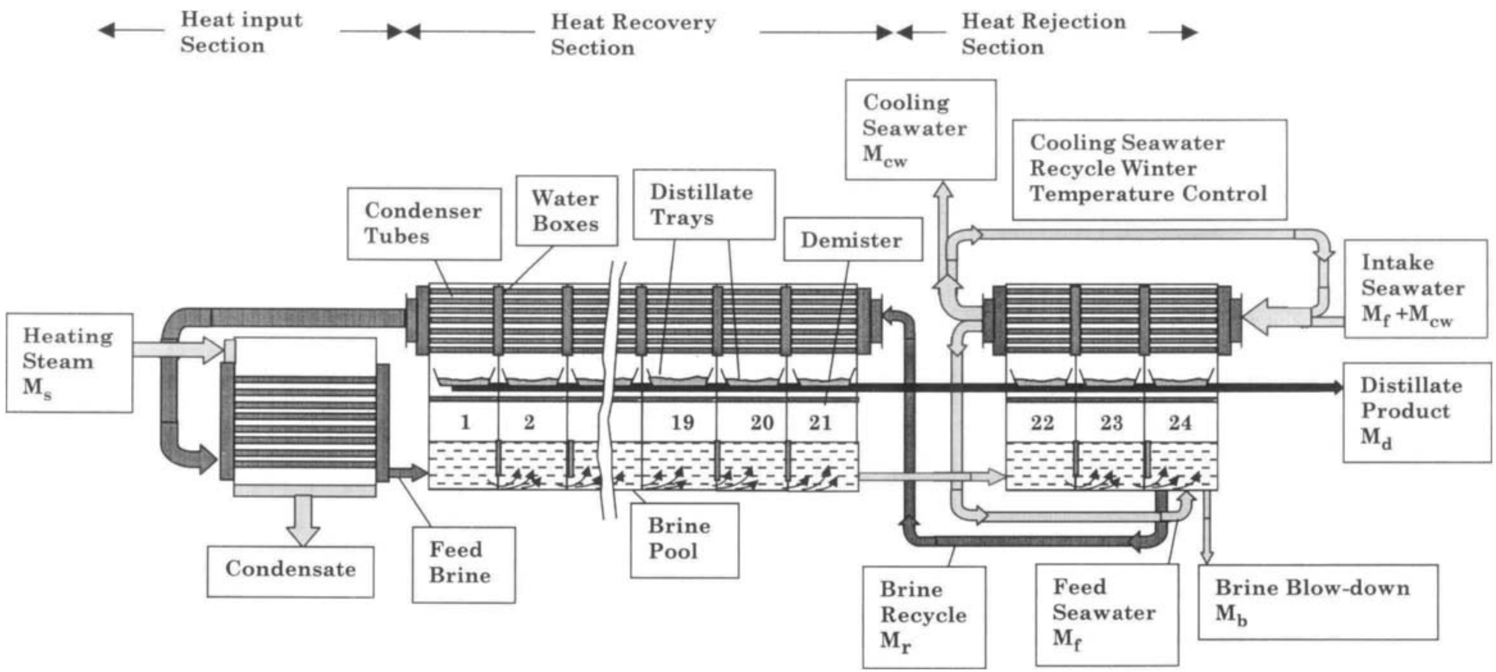

Fig. 23. Multistage flash desalination with brine circulation (MSF)

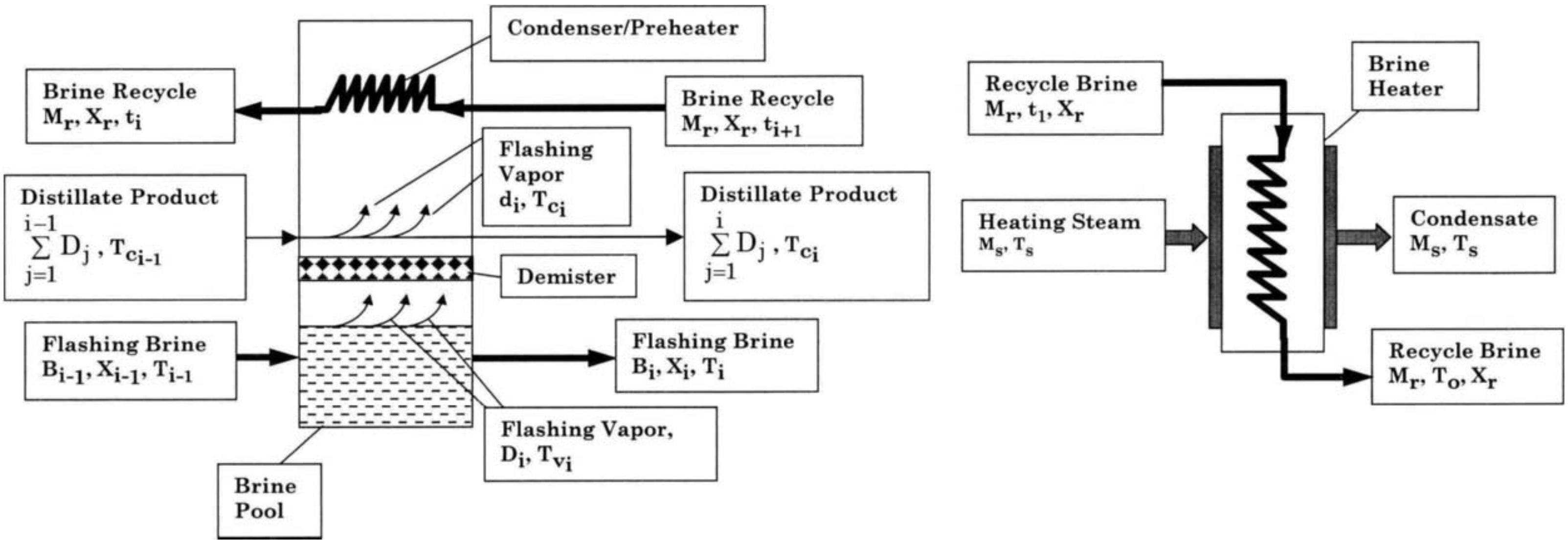

Fig. 24. Schematics of model variables in brine heater and flashing stage.

6.5.2 Mathematical Model

The MSF simplified model is a very useful tool for obtaining quick design data, evaluating system performance, and developing a good initial guess for more detailed mathematical models. This model does not need iterative solution and requires minimal computational effort. Model assumptions include the following:

- Constant and equal specific heat for all liquid streams.
- Equal temperature drop per stage for the flashing brine.
- Equal temperature drop per stage for the feed seawater.
- The latent heat of vaporization in each stage is assumed equal to the average value for the process.
- The non-condensable gases have negligible effect on the heat transfer process.
- Effects of the boiling point rise and non-equilibrium losses on the stage energy balance are negligible; however, their effects are included in the design of the condenser heat transfer area.
- The temperature of the feed seawater leaving the rejection section is equal to the brine temperature in the last stage.

Overall Material Balance

The overall material balance equations is given by

$$M_f = M_d + M_b \qquad (59)$$

where M is the mass flow rate and the subscript b, d, and defines the brine, distillate, and feed. The overall salt balance is given by

$$X_f \, M_f = X_b \, M_b \qquad (60)$$

where X is the salt concentration. Equation (60) assumes that the distillate is salt free. Equations (59) and (60) can be rearranged to obtain the expression for the total feed flow rate in terms of the distillate flow rate; this is

$$M_f = X_b/(X_b - X_f)\, M_d \qquad (61)$$

Equation (61) is used to calculate M_f, since the values of X_b, X_f, and M_d are known.

Stages and Condensers Temperature Profiles

The temperature distribution in the MSF system is defined in terms of four temperatures; these are the temperatures of the steam, T_s, the brine leaving the preheater (top brine temperature), T_o, the brine leaving the last stage, T_n, and the intake seawater, T_{cw}. A linear profile for the temperature is assumed in the stages and the condensers. First, the temperature drop per stage, ΔT, is obtained from the relation

$$\Delta T = (T_o - T_n)/n \tag{62}$$

where n is the number of recovery and rejection stages. Therefore, the temperature in the first stage is given by

$$T_1 = T_o - \Delta T$$

As for the second stage temperature it is equal to

$$T_2 = T_1 - \Delta T$$

Substituting for T_1 in the above equation gives

$$T_2 = T_o - \Delta T - \Delta T = T_o - 2\,\Delta T$$

The same procedure is repeated for subsequent stages and a general expression is developed for the temperature of stage i

$$T_i = T_o - i\,\Delta T \tag{63}$$

The recycle seawater, which flows into the condensers of the heat recovery section, has a temperature equal to T_n. This temperature is assumed to increase by ΔT_r in the condenser of each unit. This temperature increase, ΔT_r, is equal to the decrease in the brine temperature in each stage, ΔT. This result is arrived at by performing an energy balance on stage i, which gives

$$D_i\, Cp\, T_{v_i} + B_i\, Cp\, T_i - D_{i+1}\, Cp\, T_{v_{i+1}} - B_{i+1}\, Cp\, T_{i+1} = M_r\, C_p\, (T_{r_i} - T_{r_{i+1}})$$

Assuming the temperature difference, $T_i - T_{v_i}$, is small and has a negligible effect on the stage energy balance. Thus, the above equation reduces to

$$(D_i + B_i)\, Cp\, T_i - (D_{i+1} + B_{i+1})\, Cp\, T_{i+1} = M_r\, C_p\, (T_{r_i} - T_{r_{i+1}})$$

Recalling that the sum $(D_i + B_i)$ in each stage is equal to M_r, would simplify the above equation to

$$M_r\, Cp\, T_i - M_r\, Cp\, T_{i+1} = M_r\, C_p\, (T_{r_i} - T_{r_{i+1}})$$

Elimination of the like terms on both sides of the equation gives the pursued relation, thus,

$$T_i - T_{i+1} = T_{r_i} - T_{r_{i+1}}$$

or

$$\Delta T_i = \Delta T_{r_i}$$

The seawater temperature, which leaves the condenser of the first stage, is then defined by

$$T_{r_1} = T_n + (n - j)\, \Delta T$$

The seawater temperature leaving the condenser of the second stage, T_{r_2}, is less than T_{r_1} by ΔT, where

$$T_{r_2} = T_{r_1} - \Delta T$$

Substituting for T_{r_1} in the above equation gives

$$T_{r_2} = T_n + (n - j)\, \Delta T - \Delta T$$

Similar to Eq. (63), a general equation is obtained for the condenser temperature in stage i

$$T_{r_i} = T_n + (n - j)\, \Delta T - (i - 1)\, \Delta T \tag{64}$$

As for the temperature drop of the seawater in the condensers of the heat rejection section, it is obtained from the stage energy balance. This is

$$D_i\, Cp\, T_{v_i} + B_i\, Cp\, T_i - D_{i+1}\, Cp\, T_{v_{i+1}} - B_{i+1}\, Cp\, T_{i+1} = (M_f + M_{cw})\, C_p\, (T_{f_j} - T_{f_{j+1}})$$

Assuming the small temperature difference, $T_i - T_{v_i}$, has a negligible effect on the stage energy balance. Thus, the above equation reduces to

$$(D_i + B_i)\, Cp\, T_i - (D_{i+1} + B_{i+1})\, Cp\, T_{i+1} = (M_f + M_{cw})\, C_p\, (T_{f_i} - T_{f_{i+1}})$$

Recalling that the sum (D_i + B_i) in each stage is equal to M_r, would simplify the above equation to

$$M_r\ Cp\ T_i - M_r\ Cp\ T_{i+1} = (M_f + M_{cw})\ C_p\ (T_{j_i} - T_{j_{i+1}})$$

Elimination of the like terms on both sides of the equation gives the pursued relation, thus,

$$(T_{j_i} - T_{j_{i+1}}) = (T_i - T_{i+1})\ (M_r/(M_f + M_{cw}))$$

or

$$(\Delta T_{j_i}) = \Delta T_i\ (M_r/(M_f + M_{cw}))$$

Since the temperature profile is assumed linear, the above relation can also be obtained from the following simple relation

$$(\Delta T_{j_i}) = (T_n - T_{cw})/j$$

The seawater temperature, which leaves the condenser of the last stage, is then defined by

$$T_{j_n} = T_{cw} + (\Delta T_{j_i})$$

This gives the general relation for the seawater temperature in the rejection section

$$T_{j_i} = T_{cw} + (n-i+1)(\Delta T_{j_i}) \tag{65}$$

Stage Material and Salt Balance

The amount of flashing vapor formed in each stage obtained by conservation of energy within the stage, where the latent consumed by the flashing vapor is set equal to the decrease in the brine sensible heat. This is

$$D_1 = y\ M_r$$

where D_1 is the amount of flashing vapor formed in the first stage, M_r is the recycle brine flow rate, and y is the specific ratio of sensible heat and latent heat and is equal to

$$y = Cp\ \Delta T/\lambda_{av} \tag{66}$$

where Cp is the specific heat capacity and λ_{av} is the average latent heat calculated at the average temperature

$$T_{av} = (T_o + T_n)/2 \tag{67}$$

The amount of distillate formed in the second stage is equal to

$$D_2 = y\ (M_r - D_1)$$

Substituting the value of D_1 in the above equation gives

$$D_2 = y\ (M_r - y\ M_r)$$

Which simplifies to

$$D_2 = M_r\ y\ (1 - y)$$

The balance equations for D_2 and D_3 will reveal the general form for the formula of D_i. The D_3 balance is

$$D_3 = y\ (M_r - D_1 - D_2)$$

Substituting for the values of D_1 and D_2 in the above equation gives

$$D_3 = y\ (M_r - M_r\ y - M_r\ y\ (1 - y))$$

Taking M_r as a common factor in the above equation gives

$$D_3 = M_r\ y\ (1 - y - y + y^2)$$

This simplifies to

$$D_3 = M_r\ y\ (1 - y)^2$$

Accordingly, the resulting general formula for D_i is

$$D_i = M_r\ y\ (1 - y)^{(i-1)} \tag{68}$$

The total distillate flow rate is obtained by summing the values of D_i for all stages. The summation is performed in steps in order to obtain a closed form equation. Therefore, the summation of D_1 and D_2 gives

$$\begin{aligned} D_1 + D_2 &= M_r\,(y + y\,(1 - y)) \\ &= M_r\,(2\,y - y^2) \\ &= M_r\,(1 - (1 - y)^2) \end{aligned}$$

Addition of D_3 to the above gives

$$D_1 + D_2 + D_3 = M_r\,((2\,y - y^2) + y(1 - y)^2)$$

This simplifies to

$$\begin{aligned} D_1 + D_2 + D_3 &= M_r\,(2\,y - y^2 + y - 2\,y^2 + y^3) \\ &= M_r\,(3\,y - 3\,y^2 + y^3) \\ &= M_r\,(3\,y - 3\,y^2 + y^3) \\ &= M_r\,(1 - (1 - y)^3) \end{aligned}$$

Comparison of the summations of D_1+D_2 and D_1+D_2+D_3 gives the general form for the total summation of the distillate formed in all stages, M_d, which is given by

$$M_d = M_r\,(1 - (1 - y)^n) \tag{69}$$

Equation 69 is used to calculate the brine recycle flow rate, since the distillate flow is always specified in design problems.

The salt concentration in the recycle stream, X_r, is obtained by performing salt balance on the loop shown in Fig. 25. This balance is

$$X_r\,M_r + M_b\,X_b = X_f\,M_f + (M_r - M_d)\,X_n$$

The above balance is arranged to

$$X_r = (X_f\,M_f + (M_r - M_d)\,X_n - M_b\,X_b)/M_r$$

Assuming that $X_n = X_b$, simplifies the above equation to

$$X_r = (X_f\,M_f + (M_r - M_d)\,X_b - M_b\,X_b)/M_r$$

Since $M_f = M_b + M_d$, then,

$$X_r = ((X_f - X_b)\, M_f + M_r\, X_b)/M_r \tag{70}$$

The flow rate of the brine stream leaving stage i is given by

$$B_i = M_r - \sum_{k=1}^{i} D_k \tag{71}$$

The salt concentration in the brine stream leaving stage i is given by

$$X_i = \left(M_r - \sum_{k=1}^{i} D_k \right) \Big/ B_i \tag{72}$$

The last item in this section is to determine the cooling water flow rate, M_{cw}. This is required to obtain the specific cooling water flow rate, sM_{cw}, which affects the process economics. This flow rate is obtained from an overall energy balance around the desalination plant, Fig. 26. The intake seawater temperature, T_{cw}, is used as the reference temperature in the energy balance. This gives

$$M_s\, \lambda_s = M_{cw}\, C_p\, (T_n - T_{cw}) + M_b\, C_p (T_n - T_{cw}) + M_d\, C_p\, (T_n - T_{cw})$$

The above equation is arranged to obtain an expression for M_{cw}

$$M_{cw} = (M_s\, \lambda_s - M_f\, C_p (T_n - T_{cw}))/\, (C_p\, (T_n - T_{cw})) \tag{73}$$

Brine Heater and Condensers Heat Transfer Area

The motive steam provides the brine heater with the necessary energy to increase the feed seawater temperature from T_{f_1} to the top brine temperature, T_o. This requires calculation of the motive steam flow, which is obtained from the brine heater energy balance

$$M_s\, \lambda_s = M_r\, Cp\, (T_o - T_{f_1})$$

The above equation is arranged to calculate M_s

$$M_s = M_r\, Cp\, (T_o - T_{f_1})/\, \lambda_s \tag{74}$$

The brine heater area is given by

$$A_b = M_s \lambda_s/(U_b\ (LMTD)_b) \tag{75}$$

where LMTD is given by

$$(LMTD)_b=((T_s-T_o)-(T_s-T_{f_1}))/\ln((T_s-T_o)/(T_s-T_{f_1})) \tag{76}$$

and U_b is given by

$$U_b=1.7194+3.2063x10^{-3}\ T_s+1.5971x10^{-5}\ T_s^2-1.9918x10^{-7}T_s^3 \tag{77}$$

The heat transfer area for the condenser in each stage in the heat recovery section is assumed equal. The same assumption is made for the condenser heat transfer area in the heat rejection section. Therefore, the calculated heat transfer area for the first stage is used to obtain the total heat transfer area in the heat recovery section. The condenser heat transfer area in the first stage is obtained from

$$A_r = M_r\ Cp\ (T_{r_1} - T_{r_2})/(U_r\ (LMTD)_r) \tag{78}$$

where

$$U_r= 1.7194+3.2063x10^{-3}\ T_{v_1}+1.5971x10^{-5}T_{v_1}^{\ 2}-1.9918x10^{-7}T_{v_1}^{\ 3} \tag{79}$$

$$T_{v_1} = T_1 - BPE_1 - NEA_1 - \Delta T_{d_1} \tag{80}$$

and

$$(LMTD)_r=((T_{v_1}-T_{r_1})-(T_{v_1}-T_{r_2}))/\ln((T_{v_1}-T_{r_1})/(T_{v_1}-T_{r_2})) \tag{81}$$

In the above equations BPE is the boiling point elevation in the first stage, NEA, is the non-equilibrium allowance, T_v, is condensing vapor temperature, ΔT_d, is the temperature drop in the demister, and U_r is the condenser overall heat transfer coefficient. Expressions for correlations used calculate the BPE, NEA, and ΔT_d, are given in the appendix.

The same procedure is applied to the stages in the heat rejection section, where the condenser area in rejection stages is given by

$$A_j = (M_f + M_r)\ Cp\ (T_{j_n} - T_{cw})/(U_j\ (LMTD)_j) \tag{82}$$

where

$$U_j = 1.7194+3.2063x10^{-3}T_{v_n}+1.5971x10^{-5}T_{v_n}^{2}- 1.9918x10^{-7}T_{v_n}^{3} \quad (83)$$

$$T_{v_n} = T_n - BPE_n - NEA_n - \Delta T_{d_n} \quad (84)$$

and

$$(LMTD)_j=((T_{v_n}-T_{cw})-(T_{v_n}-T_{j_n}))/\ln((T_{v_n}-T_{cw})/(T_{v_n}-T_{j_n})) \quad (85)$$

The total heat transfer area for all condensers in the heat recovery and rejection sections is then obtained from

$$A_c = (n-j)\, A_r + (j)\, A_j \quad (86)$$

Stage Dimensions

Calculations of the stage dimensions include the gate height, the height of the brine pool, the stage width, and the stage length. The length of all stages is set equal to the length of the last stage and the width of all stages is set equal to the width of the first stage. The height of the brine pool must be higher than the gate height, this is necessary to prevent bypass of the vapors between stages (vapor blow through). The gate height, GH, is obtained in terms of the stage pressure drop, ΔP, the brine density, ρ_b, the weir friction coefficient, C_d, the stage width, W, and the brine recycle flow rate, M_r. For stage i the gate height is

$$GH_i = (M_r - \sum_{j=1}^{i-1} D_j)\, (2\, \rho_{b_i}\, \Delta P_i)^{(-0.5)}/(C_d\, W) \quad (87)$$

The brine pool height is set higher than the gate height by 0.2 m.

$$H_i = 0.2 + GH_i \quad (88)$$

where

$$\Delta P_i = P_i - P_{i+1}$$

$$W = M_r / V_b \quad (89)$$

where P_i and P_{i+1} are the pressures in stages i and i+1, and V_b is the brine mass velocity per chamber width. The length of the last stage is determined as a function of the vapor flow rate, D_n, the vapor density, ρ_{v_n}, the vapor allowable velocity, V_{v_n}, and the stage width, W. This is

$$L = D_n/(\rho_{v_n} V_{v_n} W) \tag{90}$$

The cross section area for each stage, A_s, is then calculated

$$A_s = L\ W \tag{91}$$

Performance Parameters

The system performance parameters are defined by the thermal performance ratio, PR, the specific heat transfer area, sA, and the specific cooling water flow rate, sM_{cw}. The performance ratio is the ratio of product to steam flow rates. This is

$$PR = M_d/M_s \tag{92}$$

The specific heat transfer area is defined by

$$sA = (A_b + A_c)/M_d \tag{93}$$

The specific cooling water flow rate is given by

$$sM_{cw} = M_{cw}/M_d \tag{94}$$

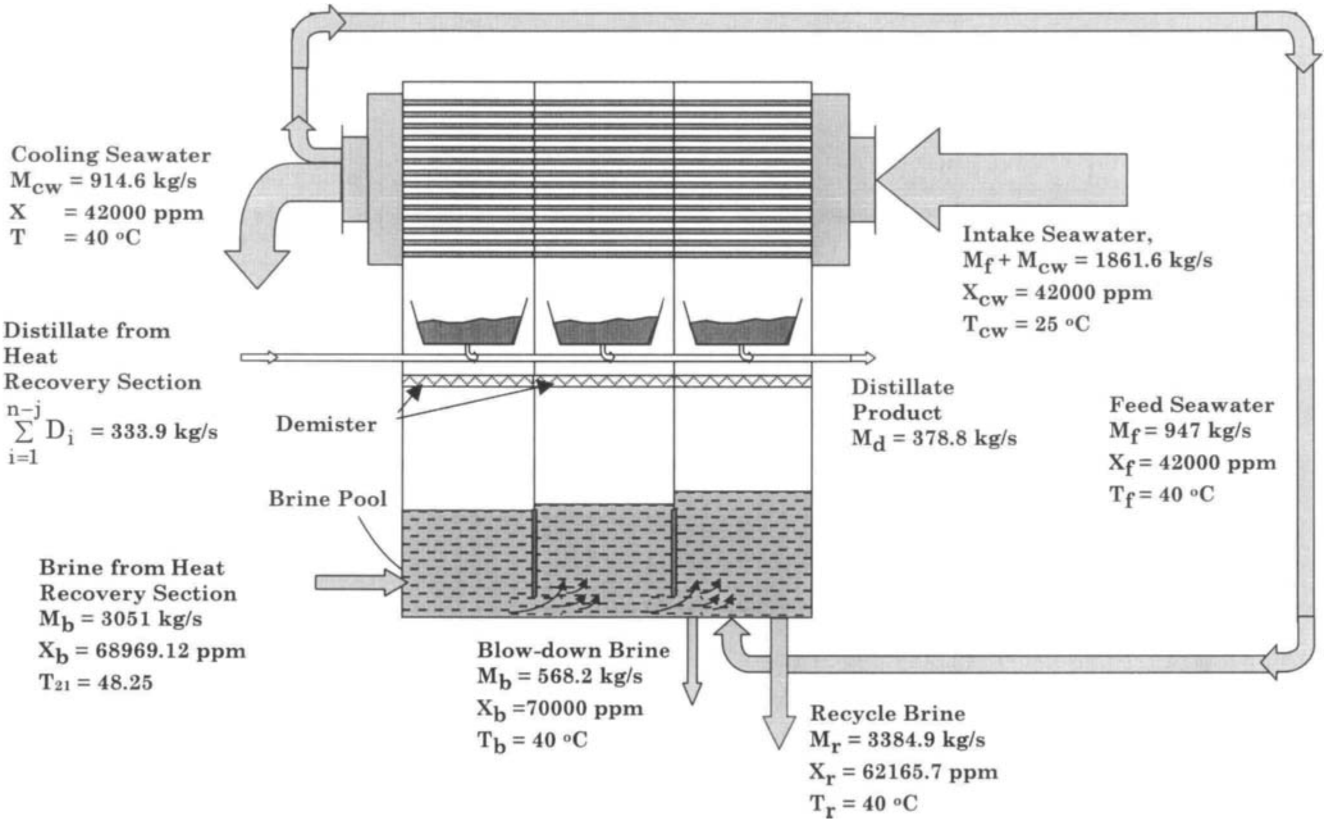

Fig. 25. Material and salt balance on the heat rejection section.

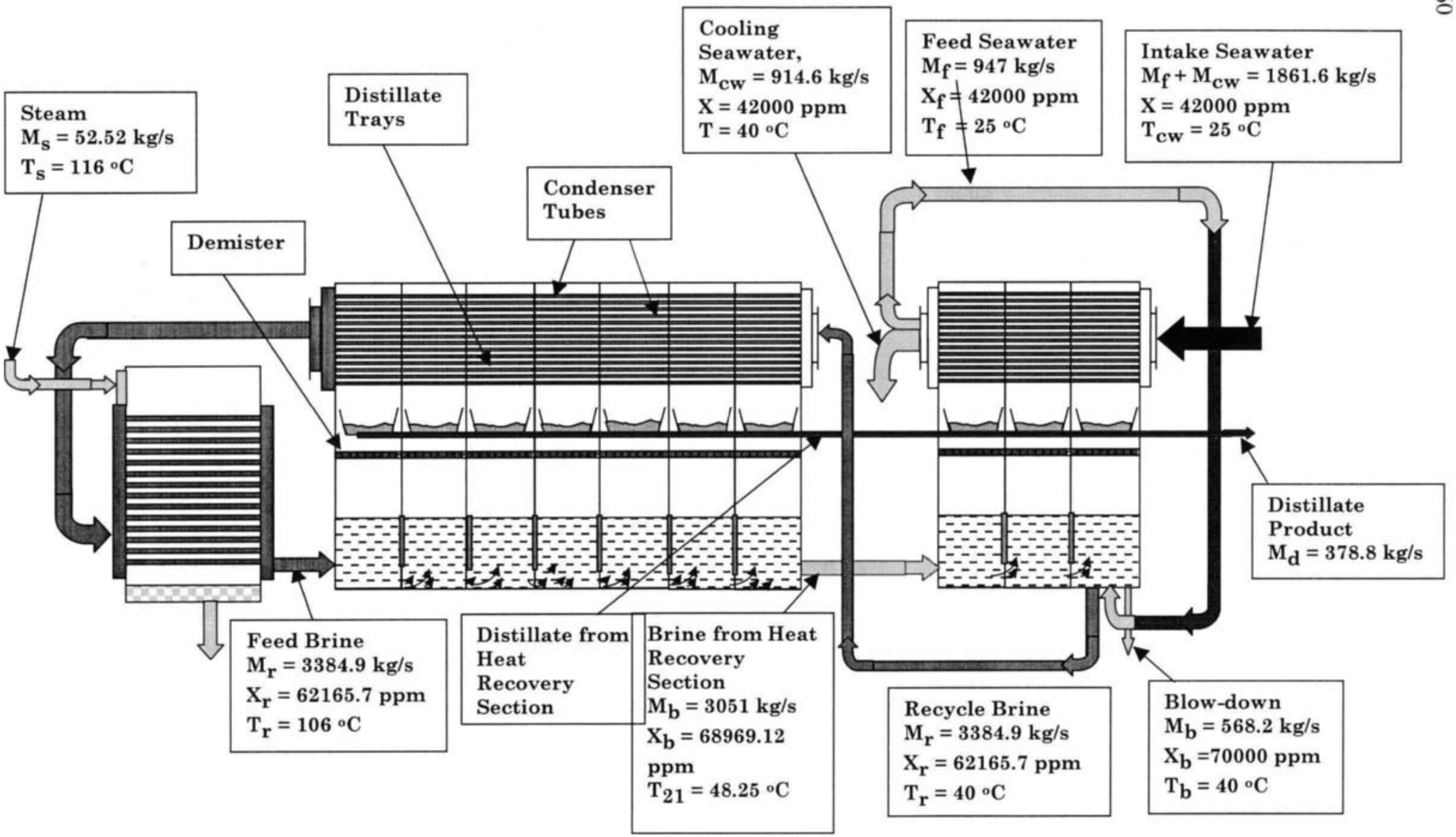

Fig. 26. Stream flow rate, salinity, and temperature in MSF

Solution Method

Solution of the MSF simplified model is non-iterative and can be limited to calculations of the system performance parameters, which include the performance ratio, the specific cooling water flow rate, and the specific heat transfer area. This reduces calculations of the temperature, flow rate, and concentration profiles to those of the first stage and the recycle brine. The solution proceeds as follows:

- The flow rates of the feed seawater, M_f, and blow down brine, M_b, are obtained from Eqs. 60 and 58, respectively.
- The temperature drop per stage, ΔT, is calculated from Eq. 62.
- The temperatures of stages 1 and 2, T_1 and T_2, are calculated from Eq. 63.
- The seawater temperatures leaving the condensers of the first and second stages, T_{r_1} and T_{r_2}, are determined from Eq. 64. Also, the seawater temperature leaving the condenser of the last stage, T_{j_n}, is calculated from Eq. 65.
- The average stage temperature, T_{av}, is calculated from Eq. 67 and the corresponding latent heat value is obtained from the correlation given in the appendix.
- The ratio of the stage sensible and latent heat, y, is calculated from equation 66.
- The flow rates of recycle brine, M_r, cooling seawater, M_{cw}, and steam, M_s, are calculated from Eqs. 69, 73, and 74, respectively.
- The heat transfer area for the brine preheater (A_b) the first stage condenser in the recovery section (A_r) and the last condenser in the rejection section (A_j) are calculated from Eqs. 75, 78, and 82, respectively.
- The stage length, width, and cross section areas are determined from Eqs. 89-91. Also, the height of the brine pool, H, and the gate height for stage i are obtained from Eqs. 87-88.
- The system performance parameters, PR, sA, and sM_{cw}, are calculated from Eqs. 92-94.

The above solution process requires specification of the following variables:

- Total distillate flow rate, M_d.
- Total number of stages (recovery and rejection).
- Number of heat rejection stages, j.
- Intake seawater temperature, T_{cw}.
- Top brine temperature, T_o.
- Steam temperature, T_s.
- Brine temperature of stage n.
- Brine salt concentration, X_b.

- Intake seawater salt concentration, X_f.
- Heat capacity of liquid streams, Cp.
- Weir friction coefficient, C_d.
- Vapor velocity in the last stage, V_{vn}.
- Brine mass flow rate per stage width, V_b.

Case Study

An MSF system with 24 stages is used to produce 7.2 MGD of product water. The system contains 21 stages in the heat recovery section and 3 stages in the heat rejection section. The following specifications are made to obtain the system design parameters and performance characteristics. The specifications include the following:

- Intake seawater temperature, $T_{cw} = 25\ ^oC$.
- Steam temperature, $T_s = 116\ ^oC$.
- Top brine temperature, $T_o = 106\ ^oC$.
- Brine temperature in the last stage, $T_n = 40\ ^oC$.
- Heat capacity of liquid streams, $C_p = 4.18$ kJ/kg ^oC.
- Salinity of intake seawater, $X_f = 42000$ ppm.
- Salinity of the brine blow-down, $X_b = 70000$ ppm.
- Vapor velocity in the last stage, $V_v = 6$ m/s
- Brine mass flow rate per stage width, $V_b = 180$ kg/ms.
- Weir friction coefficient, $C_d = 0.5$.

Before proceeding, the product volume flow rate is converted to mass rate in SI units; this is necessary for solution of the energy balance equations. The kg/s value of M_d is

$$M_d = \frac{\left(7.2\left(\frac{MG}{D}\right)\right)\left(10^6\left(\frac{G}{MG}\right)\right)\left(\frac{1}{24x3600}\left(\frac{D}{s}\right)\right)}{\left(\frac{1}{219.96}\left(\frac{m^3}{G}\right)\right)10^3\left(\frac{kg}{m^3}\right)}$$

$= 378.8$ kg/s

The seawater feed flow rate is obtained from Eq. 60, where

$M_f = X_b/(X_b - X_f)\ M_d$

$= 70000/(70000 - 42000)\ (378.8)$

$= 947$ kg/s

The flow rate of the blow-down brine is then obtained from the overall balance given in Eq. 58,

$$Mb = Mf - Md$$
$$= 947 - 378.8$$
$$= 568.2 \text{ kg/s}$$

The temperature drop in each effect is obtained from Eq. 62, where

$$\Delta T = (T_o - T_n)/n$$
$$= (106 - 40)/24$$
$$= 2.75 \ ^oC$$

This value is used to calculate the temperatures of the first and second stages, T_1 and T_2, where

$$T_1 = T_o - \Delta T$$
$$= 106 - 2.75$$
$$= 103.25 \ ^oC$$

As for the second stage temperature it is equal to

$$T_2 = T_1 - \Delta T$$
$$= 103.25 - 2.75$$
$$= 100.5 \ ^oC$$

Also the temperatures of the seawater leaving the condensers in the first and second stages are calculated. This is

$$T_{r_1} = T_n + (n - j)\ \Delta T$$
$$= 40 + (24 - 3)\ (2.75)$$
$$= 97.75 \ ^oC$$

and

$$T_{r_2} = T_{r_1} - \Delta T$$
$$= 97.75 - 2.75$$
$$= 95 \ ^oC$$

Calculation of the y ratio from Eq. (64) is preceded by evaluation of T_{av} and λ_{av}. The T_{av} value is

$$T_{av} = (T_o + T_n)/2$$
$$= (106+40)/2$$
$$= 73\ ^oC$$

At this temperature λ_{av} is equal to 2330.1 kJ/kg from the correlation given in the appendix. The value of y is calculated

$$y = Cp\ \Delta T/\lambda_{av}$$
$$= (4.18)\ (2.75)/2330.1$$
$$= 4.933x10^{-3}$$

The brine recycle flow rate is obtained from Eq. 59

$$M_r = M_d/(1 - (1 - y)^n)$$
$$= 378.8 = M_r\ (1 - (1 - 4.933x10^{-3})^{24})$$
$$= 3384.8\ kg/s$$

The steam flow rate is calculated from Eq. 74

$$M_s = M_r\ Cp\ (T_o - T_{r_1})/\ \lambda_s$$
$$= (3384.8)\ (4.18)\ (106\text{-}97.75)/(2222.33)$$
$$= 52.52\ kg/s$$

The heat transfer area for the brine heater, A_b, is calculated from Eq. 73. This requires calculations of the logarithmic mean temperature difference $(LMTD)_b$ and the overall heat transfer coefficient. The value of $(LMTD)_b$ is obtained from Eq. 76, which gives

$$(LMTD)_b = ((T_s - T_o) - (T_s - T_{r_1}))/\ln((T_s - T_o)/(T_s - T_{r_1}))$$
$$= ((116 - 106) - (116 - 97.75))/$$
$$\ln((116 - 106)/(116 - 97.75))$$
$$= 13.71\ ^oC$$

The overall heat transfer coefficient is obtained from Eq. 77. This gives

$$U_b = 1.7194+3.2063x10^{-3}\ T_s+1.5971x10^{-5}\ (Ts)^2 - 1.9918x10^{-7}(Ts)^3$$
$$= 1.7194 + 3.2063x10^{-3}\ (116) + 1.5971x10^{-5}\ (116)^2 - 1.9918x10^{-7}\ (116)^3$$
$$= 2\ kW/m^2\ ^oC$$

The brine heater area, A_b, is then calculated from Eq. 75

$$A_b = M_s \lambda_s/(U_b\ (LMTD)_b)$$
$$= (52.52)(2222.33)/((2)(13.71))$$
$$= 4256.6\ m^2$$

The condenser area in the heat recovery section, A_r, is determined for the first stage. Determination of this value requires calculations of the vapor condensation temperature, T_{v_1}, the logarithmic mean temperature difference $(LMTD)_r$, and the overall heat transfer coefficient, U_r. The vapor temperature is given by Eq. 80, where

$$T_{v_1} = T_1 - BPE_1 - NEA_1 - \Delta T_{d_1}$$

The boiling point elevation is calculated from the correlation given in the appendix. Solution of this equation requires calculation of the recycle brine concentration, X_r, which is defined by Eq. 70. This gives

$$X_r = (X_f\ M_f + (M_r - M_d)\ X_n - M_b\ X_b)/M_r$$
$$= ((42000)(947)+(3384.8 - 378.8)(70000) - (568)(70000))/(3384.8)$$
$$= 62170\ ppm$$

The values of B and C in the correlation for the boiling point elevation are

$$B = (6.71 + 6.34x10^{-2}(T_1) + 9.74x10^{-5}(T_1)^2)10^{-3}$$
$$= (6.71 + 6.34x10^{-2}(103.25) + 9.74x10^{-5}(103.25)^2)10^{-3}$$
$$= 0.0143$$

$$C = (22.238 + 9.59x10^{-3}(T_1) + 9.42x10^{-5}(T_1)^2)10^{-8}$$
$$= (22.238 + 9.59x10^{-3}(103.25) + 9.42x10^{-5}(103.25)^2)10^{-8}$$
$$= 2.423x10^{-7}$$

Substituting the values of B and C in the BPE correlation gives

$$BPE_1 = X_r\ (B + (X_r)(C))10^{-3}$$
$$= 62165\left(0.0143 + (62165)\left(2.423x10^{-7}\right)\right)10^{-3}$$
$$= 1.83\ ^oC$$

The non-equilibrium allowance, NEA_1, in the first stage is calculated from the correlation given in the appendix for the MSF system. This involves calculations of the gate height, GH_1, the height of the brine pool, H_1, the stage width, W, the stage pressure drop, P_1-P_2, and the brine density. The width of the first stage is given by

$$W = M_r / V_b$$
$$= 3384.8/180$$
$$= 18.8 \text{ m}$$

The stage length is calculated for the last stage, where

D_{24} = 14.904 kg/s and

ρ_{v_n} = 0.0512 kg/m3,

$$L = D_n/(\rho_{v_n} V_{v_n} W) = 14.904/((0.0512)(6)(18.8))$$
$$= 2.58 \text{ m}$$

The brine density in the first stage is 1002.413 kg/m^3, which is obtained from the correlation given the appendix at a salinity of 62473.9 ppm and a temperature of 103.25 °C. The pressures of the first and second stage are obtained from the saturation pressure correlations, where, at T_1 = 103.25 °C, P_1 = 113.72 kPa, and at T_2 = 100.5 °C, P_2 = 103.23 kPa. The resulting gate height in the first stage, GH_1, is calculated from Eq. 82. This result in

$$GH_1 = M_r\,(2\,\rho_{b_1}\,\Delta P_1)^{(-0.5)}/(C_d\,W)$$
$$= (3384.8)((2)(1002.41)(113.72-103.23)\text{x}10^3)^{(-0.5)}/((0.5)(18.8)$$
$$= 0.078 \text{ m}$$

It should be noted that the pressure drop in the above equation is in Pa and not kPa. The corresponding brine pool height is obtained by simply adding 0.2 m to the value of GH_1, or

$$H_1 = 0.278 \text{ m}$$

The non-equilibrium allowance is then calculated using the correlation given in the appendix

$$NEA_1 = (0.9784)^{T_o}\,(15.7378)^{H_1}\,(1.3777)^{V_b \text{x} 10^{-6}}$$

$= (0.9784)^{(106)}\ (15.7378)^{(0.278)}\ (1.3777)^{(180x10^{-6})}$
$= 0.213\ ^oC$

The temperature drop in the demister is assumed negligible in comparison with the values of BPE_1 and NEA_1. Therefore, the vapor temperature in the first is

$$T_{v_1} = T_1 - BPE_1 - NEA_1 - \Delta T_{d_1}$$
$$= 103.25 - 1.83 - 0.213 - 0$$
$$= 101.207\ ^oC$$

The vapor temperature, T_{v_1}, is used to calculate U_r and $(LMTD)_r$, where

$$(LMTD)_r = ((T_v-T_{r_1})-(T_v-T_{r_2}))/\ln((T_v-T_{r_1})/(T_v-T_{r_2}))$$
$$= ((101.207-97.75)-(101.207-95))/$$
$$\ln((101.207-97.75)/(101.207-95))$$
$$= 4.69\ ^oC$$

$$U_r = 1.7194+3.2063x10^{-3}\ T_{v_1}+1.5971x10^{-5}\ (T_{v_1})^2 - 1.9918x10^{-7}(T_{v_1})^3$$
$$= 1.7194 + 3.2063x10^{-3}\ (101.207) + 1.5971x10^{-5}\ (101.207)^2$$
$$- 1.9918x10^{-7}\ (101.207)^3$$
$$= 2\ kW/m^2\ ^oC$$

$$A_r = M_r\ Cp\ (T_{r_1} - T_{r_2})/(U_r\ (LMTD)_r)$$
$$= ((3384.8)\ (4.18)\ (97.75\text{-}95))/((2)\ (4.68))$$
$$= 4156.9\ m^2$$

The condenser area in the heat rejection section, A_j, is determined for the last stage. Determination of this value requires calculations of the vapor condensation temperature, T_{v_n}, the logarithmic mean temperature difference $(LMTD)_j$, and the overall heat transfer coefficient, U_j. The vapor temperature is given by Eq. 84, where

$$T_{v_n} = T_n - BPE_n - NEA_n - \Delta T_{d_n}$$

The values of B and C in the correlation for the boiling point elevation are

$$B = \left(6.71 + 6.34x10^{-2}(T_n) + 9.74x10^{-5}(T_n)^2\right)10^{-3}$$
$$= \left(6.71 + 6.34x10^{-2}(40) + 9.74x10^{-5}(40)^2\right)10^{-3}$$
$$= 0.0094018$$

$$C = \left(22.238 + 9.59\text{x}10^{-3}(T_n) + 9.42\text{x}10^{-5}(T_n)^2\right)10^{-8}$$
$$= \left(22.238 + 9.59\text{x}10^{-3}(40) + 9.42\text{x}10^{-5}(40)^2\right)10^{-8}$$
$$= 2.277\text{x}10^{-7}$$

Substituting the values of B and C in the BPE correlation gives

$$BPE_n = X_n\ (B + (X_n)(C))10^{-3}$$
$$= 70000\left(0.0094018 + (70000)\left(2.277\text{x}10^{-7}\right)\right)10^{-3}$$
$$= 1.77\ {}^{o}C$$

The gate height and the height of the brine pool in the last stage are assumed equal to those in the previous stage. The brine density in stage n-1 is 1042.4 kg/m^3 , which is calculated at a salinity of 69654.7 ppm and a temperature of 35 ^{o}C. The pressures of stages n-1 and n are $P_{n\text{-}1}$ = 8.35 kPa and P_n = 7.23 kPa, which are calculated at $T_{n\text{-}1}$ = 35 ^{o}C and T_n = 40 ^{o}C. The resulting gate height in the stage n-1, $GH_{n\text{-}1}$, is calculated from Eq. 87. This result is

$$GH_{n-1} = B_{n\text{-}2}\ (2\ \rho_{b_{n\text{-}1}}\ \Delta P_{n-1})^{(-0.5)}/(C_d\ W)$$
$$= (3036)((2)(1043)(8.35\text{-}7.23)\text{x}10^{3})^{(-0.5)}/((0.5)(18.8)$$
$$= 0.21\ \text{m}$$

The corresponding brine pool height is obtained by simply adding 0.2 m to the value of GH, or

$$H_{n-1} = 0.41\ \text{m}$$

The non-equilibrium allowance is then calculated using the correlation given in the appendix

$$NEA_n = (0.9784)^{T_{n\text{-}1}}\ (15.7378)^{H_{n\text{-}1}}\ (1.3777)^{V_b\text{x}10^{-6}}$$
$$= (0.9784)^{(42.75)}\ (15.7378)^{(0.41)}\ (1.3777)^{(180\text{x}10^{-6})}$$
$$= 1.217\ {}^{o}C$$

The temperature drop in the demister is assumed negligible in comparison with the values of BPE_n and NEA_n. Therefore, the vapor temperature in the last stage is

$$T_{v_n} = T_n - BPE_n - NEA_n - \Delta T_{d_n}$$

$= 40 - 1.77 - 1.217 - 0$
$= 37\ ^{o}C$

The vapor temperature, T_{v_n}, is used to calculate U_j and $(LMTD)_j$, where

$$(LMTD)_j = ((T_{vn}-T_{jn})-(T_{vn}-T_{cw}))/\ln((T_{vn}-T_{jn})/(T_{vn}-T_{cw}))$$
$$= ((37-30) - (37\text{-}25))/\ln((37-30)/(37-25))$$
$$= 9.28\ ^{o}C$$

$$U_j = 1.7194+3.2063\text{x}10^{-3}\ T_{v_n}+1.5971\text{x}10^{-5}\ (T_{v_n})^2 -1.9918\text{x}10^{-7}(T_{v_n})^3$$
$$= 1.7194 + 3.2063\text{x}10^{-3}\ (37) + 1.5971\text{x}10^{-5}\ (37)^2 - 1.9918\text{x}10^{-7}\ (37)^3$$
$$= 1.85\ kW/m^2\ {}^{o}C$$

$$A_j = (M_f+M_{cw})\ Cp\ (T_{j_n} - Tcw)/(U_j\ (LMTD)_j)$$
$$= ((1861.7)\ (4.18)\ (30-25))/((1.85)\ (9.28))$$
$$= 2266.4\ m^2$$

The total condenser area is then calculated

$$A_c = n\text{-}j\ A_r + j\ A_j$$
$$= (21)\ (4156.9) + (3)\ (2266.4)$$
$$= 94094.1\ m^2$$

The cooling water flow rate is calculate from Eq. 73, where

$$M_{cw} = (M_s\ \lambda_s - M_f\ C_p(T_n - T_{cw}))/\ (C_p\ (T_n - T_{cw}))$$
$$= ((52.52)(2222.33) - (947)(4.18)(40\text{-}25))/((4.18)(40\text{-}25))$$
$$= 914.64\ kg/s$$

Performance Parameters

Performance Ratio

$$PR = M_d/M_s$$
$$= 378.8/52.52$$
$$= 7.21$$

The specific heat transfer area

$$sA = (A_b + A_c\ )/M_d$$
$$= (4256.6 +94094.1)/378.8$$
$$= 259.6\ m^2/(kg/s)$$

The specific cooling water flow rate

$$sM_{cw} = M_{cw}/M_d$$
$$= 914.6/378.8$$
$$= 2.41$$

Solution Summary

Flow Rates

$M_d = 378.8$ kg/s

$M_f = 947$ kg/s

$M_b = 568.2$ kg/s

$M_r = 3384.8$ kg/s

$M_s = 52.52$ kg/s

$M_{cw} = 914.64$ kg/s

Heat Transfer Areas

$A_b = 4256.6$ m^2

$A_r = 4156.9$ m^2

$A_j = 2266.4$ m^2

$A_c = 94094.1$ m^2

Stage Dimensions

$W = 18.8$ m

$L = 2.56$ m

$GH_1 = 0.078$ m

$H_1 = 0.278$ m

Performance Parameters

PR = 7.21

sA = 259.6 $m^2/(kg/s)$

sM_{cw} = 2.41

Stage Profiles

The MSF simplified model is used to calculate the temperature and concentration profiles for the 24 stages. The calculations also include the brine and distillate flow rate, the gate height and the brine level in each stage. In the following table, the flow rates are in kg/s, the temperature in °C, the salinity in ppm, and the height in m.

Stage	D	$\sum D$	B	X	T	T_f	GH	H
1	16.699	16.699	3368.284	62473.96	103.25	97.75	0.078	0.278
2	16.617	33.316	3351.667	62783.69	100.5	95	0.081	0.281
3	16.535	49.851	3335.133	63094.95	97.75	92.25	0.084	0.284
4	16.453	66.304	3318.68	63407.75	95	89.5	0.087	0.287
5	16.372	82.676	3302.308	63722.11	92.25	86.75	0.09	0.29
6	16.291	98.967	3286.017	64038.03	89.5	84	0.094	0.294
7	16.211	115.178	3269.806	64355.51	86.75	81.25	0.097	0.297
8	16.131	131.309	3253.675	64674.56	84	78.5	0.101	0.301
9	16.051	147.36	3237.624	64995.2	81.25	75.75	0.105	0.305
10	15.972	163.332	3221.652	65317.43	78.5	73	0.11	0.31
11	15.893	179.225	3205.759	65641.25	75.75	70.25	0.114	0.314
12	15.815	195.04	3189.944	65966.68	73	67.5	0.119	0.319
13	15.737	210.777	3174.208	66293.73	70.25	64.75	0.124	0.324
14	15.659	226.436	3158.548	66622.39	67.5	62	0.13	0.33
15	15.582	242.018	3142.966	66952.68	64.75	59.25	0.136	0.336
16	15.505	257.523	3127.461	67284.62	62	56.5	0.143	0.343
17	15.429	272.952	3112.033	67618.19	59.25	53.75	0.15	0.35
18	15.352	288.304	3096.68	67953.42	56.5	51	0.157	0.357
19	15.277	303.581	3081.404	68290.31	53.75	48.25	0.165	0.365
20	15.201	318.782	3066.202	68628.88	51	45.5	0.175	0.375
21	15.126	333.908	3051.076	68969.12	48.25	42.75	0.185	0.385
22	15.052	348.96	3036.024	69311.05	45.5	40	0.197	0.397
23	14.977	363.937	3021.047	69654.67	42.75	35	0.211	0.411
24	14.904	378.841	3006.143	70000	40	30	0.211	0.411

Approximate Form for the Performance Ratio

The approximate form for the performance ratio is based on the assumption that the temperature profile is linear throughout the stages and the condensers, Fig. 27. As is shown the two right-angle triangles, abc and $\bar{a}\bar{b}T_c$, are conforming. Also, the two reciprocal triangles, aT_oT_1 and $\bar{a}\bar{b}T_c$, are identical, so that

$$T_o - T_1 = T_n - T_{cw}$$

The following identity is then written for the two conforming triangles, abc and $\bar{a}\bar{b}T_c$,

$$\frac{ab}{\bar{a}\bar{b}} = \frac{bT_c}{\bar{b}T_c}$$

This relation is expressed as a function of temperatures and number of stages,

$$\frac{T_o - T_{cw}}{T_n - T_{cw}} = \frac{n + j}{j}$$

The numerator on the left-hand side in the above equation is arranged into the following form

$$\frac{T_o - T_n + T_n - T_{cw}}{T_n - T_{cw}} = \frac{n}{j} + 1$$

Simplifying the above equation gives

$$\frac{T_o - T_n}{T_n - T_{cw}} + 1 = \frac{n}{j} + 1$$

The dominator in the above equation is replaced with the relation obtained for the two identical-reciprocal triangles, aT_ot_1 and $\bar{a}\bar{b}T_c$,

$$\frac{T_o - T_n}{T_n - T_1} = \frac{n}{j} \qquad (95)$$

The temperature differences in equation (95) are then expressed in terms of the process thermal loads. The total amount of energy gain by the recycle seawater in heat recovery section is defined in terms of the temperature difference, T_o-T_n, which is approximated by

$$D\ \lambda_{av} = M_r\ Cp\ (T_o - T_n) \tag{96}$$

where D is total amount of distillate product (λ_{av}) is the average latent heat of condensing vapor. The steam thermal load is then defined in terms of the temperature increase of the recycle seawater in the brine heater, T_o-T_1, where

$$M_s\ \lambda_s = M_r\ Cp\ (T_o - T_1) \tag{97}$$

Dividing equations (96) and (97) gives

$$\frac{D}{M_s} = \frac{T_o - T_n}{T_o - T_1}$$

The previous equation assumes that the ratio of the latent heat of steam and distillate product is approximately one. The previous equation gives the process thermal performance ratio, PR = D/M_s. Also, the left hand side of the previous equation is defined in Eq. (95) as n/j. Making these simplifications gives the desired approximate relation for the system thermal performance ratio, where

$$PR = n/j \tag{98}$$

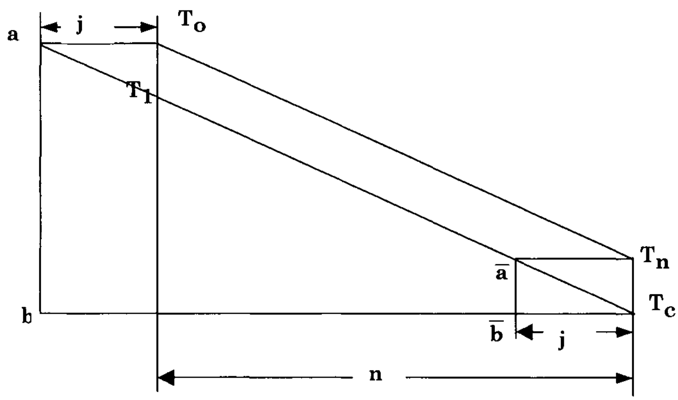

Fig. 27. Schematic of triangles for approximate PR calculations

6.5.3 System Performance

Relations among various system variables of the MSF system include the following:

- Temperature profiles of the flashing brine and seawater flowing into the condenser tubes.
- Thermal performance ratio, defined as the flow rate ratio of the product fresh water to the heating steam.
- Specific heat transfer area, defined as the ratio of the total heat transfer area of the condenser tubes and the brine heater to flow rate of product fresh water.
- Specific flow rate of cooling water, which is defined as the flow rate ratio of cooling seawater to the product fresh water.

The above system variables have a strong effect on the product unit cost. The thermal performance ratio is a measure of the process efficiency, where higher ratios imply larger production rate per unit flow rate of heating steam. As discussed before, a thermal performance ratio of 8 is common among various MSF designs. However, system operation at clean conditions normally gives thermal performance ratios close to 10. As operation proceeds, where fouling and scaling starts to reduce the overall heat transfer coefficient, the thermal performance ratio drops gradually and it is normal to allow the system to operate at thermal performance ratios below 7, where full shutdown takes place and acid cleaning and in some events mechanical cleaning might be necessary to restore the unit characteristics to the original design conditions, Al-Falah and Al-Shuaib, 2000.

As for the specific heat transfer area it is gives a simultaneous measure for the capital cost contribution on the product unit cost as well as efficiency of system operation. Design values vary between 200 to 300 $m^2/(kg/s)$. For clean systems, this parameter has low values close 200 m2/(kg/s) and as fouling and scaling takes place the parameter value reaches the maximum range. Fouling and scaling reduces the overall heat transfer coefficient as well the seawater temperature entering the brine heater. Compensation for this effect and to maintain constant production rate requires increase of the heating steam flow rate. Eventually, the system reaches limiting operating conditions, where fouling effects can no longer be met with the increase in the heating steam flow rate. At these conditions the specific heat transfer area starts to increase and reaches values close to the maximum limit.

The specific flow rate of cooling seawater is also a simultaneous measure for the contribution of the system capital that includes cost of the pumping units and the intake system as well as the operation efficiency on the product unit cost. The cooling seawater is used to remove excess heat added to the system by the heating steam in the brine heater. Proper system design aims at minimizing the amount of cooling seawater. Therefore, the size of the pumping units is reduced as well as the intake system. As discussed above, larger amounts of heating steam are used as fouling and scaling builds inside the condenser and the brine heater tubes. Considering the process thermodynamics and energy conservation implies the increase in the flow rate of the cooling seawater at such conditions. This is necessary to remove the excess heat added to the system.

Figure 28 shows a plot of the temperature distribution for both the brine flowing insider the condenser tubes and in the flashing chambers. Examination of the figure indicates that the temperature distribution for the brine flow inside the tubes of the preheaters deviates more from the straight line when compared to that for the flashing brine. Also, the rate of increase in temperature of the brine flowing in the preheaters at the heat rejection section is higher than the rate of increase at the heat recovery section. For any stage the vertical distance between these two nearly straight line curves is the summation of the stage flash down, non-equilibrium allowance, boiling point elevation, temperature decrease corresponding to the pressure drop in the demister, and the terminal temperature difference.

The effects of number of stages on the plant performance ratio, specific heat transfer area, specific cooling water flow rate and specific flow rate of brine circulation at different values of top brine temperature are shown in Figs. 29-32, respectively. Figure 29 shows the increase in the performance ratio upon the increase in the number of flashing stages and the top brine temperature. Increase of the number stages decreases the temperature drop per stage, which is inversely proportional to the performance ratio. Also, operation at higher top brine temperatures increases the flashing range, which results in the increase of the amount of product water per unit mass of heating steam.

Fig. 30 shows increase in the specific heat transfer area at larger number of stages and lower top brine temperatures. Since all calculations are made at constant total production rate, increasing the number of stages increases the total heat transfer area, which results in the increase of the specific heat transfer area. Similarly, operation at lower top brine temperature reduces the values of the heat transfer coefficient, which in turn necessitates use of larger heat transfer area.

Figure 31 illustrates that the top brine temperature has very little influence on the specific cooling water flow rate. On the other hand, use of larger number of stages reduces the specific flow rate of the cooling seawater. This is explained in terms of the results obtained in Fig. 29, where the performance ratio increases at larger number of flashing stages, which implies use of smaller amount of heating steam at constant production rate. Reduction in the amount of the heating steam is associated with decrease in the amount of energy added to the system as well as decrease in the amount of energy removed by the cooling seawater.

Figure 32 shows the independence of specific flow rate of circulated brine on the number of stages. This value decreases as the brine temperature is increased, which reflects increase in the system efficiency. Figures 33-36 demonstrate the effects of top brine temperature on the same variables, namely, the plant performance ratio, specific heat transfer area, specific cooling water flow and specific recirculated brine flow rate at different values of number of stages.

Figures 29-36 also show the data of six different plants in operation for a long time in Arabian Gulf countries. Plants from the C_3, C_4 and C_5 groups are in Saudi Arabia, Al-Mudaiheem et al. (1993). Umm Al-Nar and Taweelah B plants are in the United Arab Emirates, Hornburg and Watson (1993) and Hornburg et al. (1993). In the Doha-West plants are located in Kuwait, Darwish (1991). It is worthwhile mentioning that design and operation data for these plants are unfortunately scarce. The comparison of the data of operating plants with the developed model predictions shows adequate quantitative agreement, and no qualitative divergence has been observed.

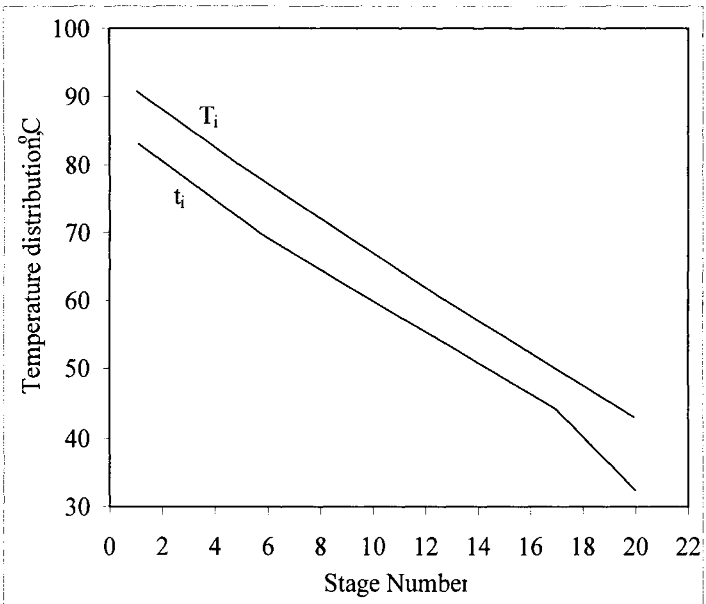

Fig. 28. Temperature distribution for brines flowing in the preheaters and through the flashing chambers

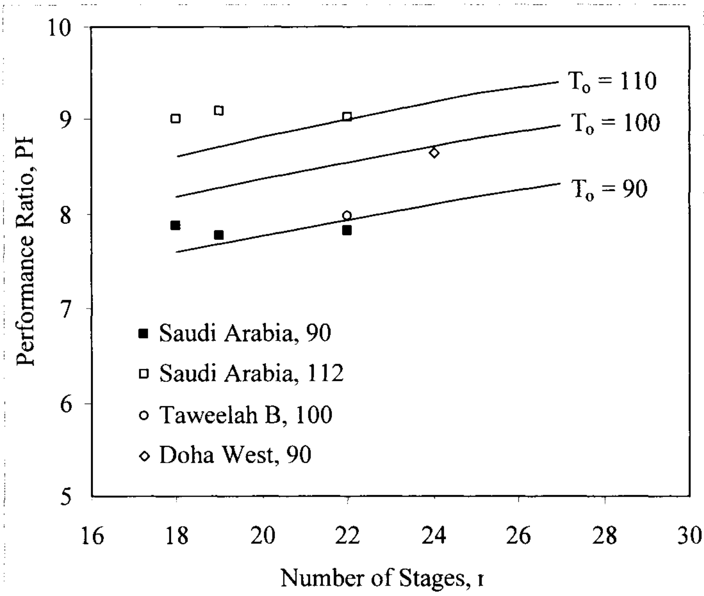

Fig. 29. Effect of number of stages on the plant performance ratio

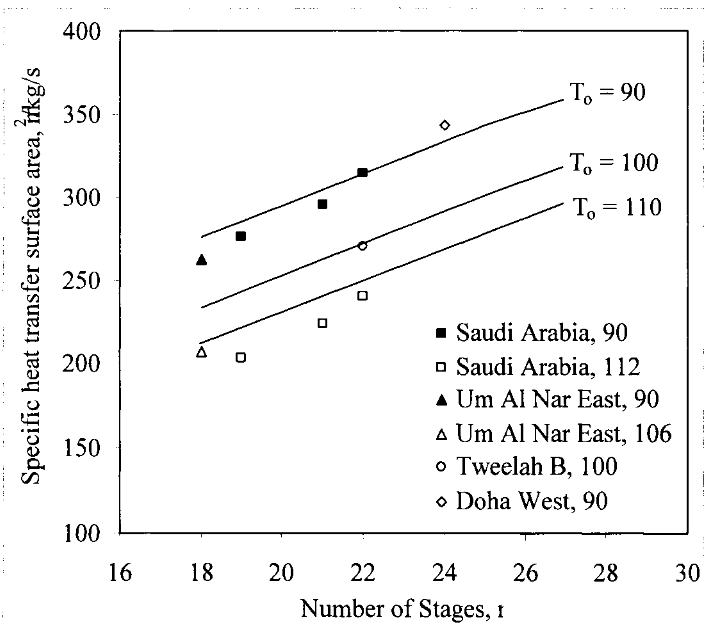

Fig. 30. Effect of number of stages on the specific heat transfer area

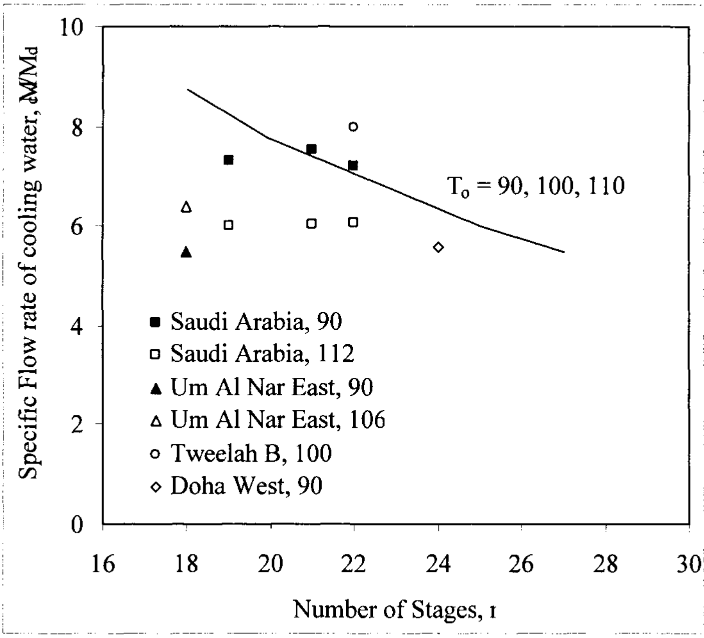

Fig. 31. Effect of number of stages on specific cooling water flow rate

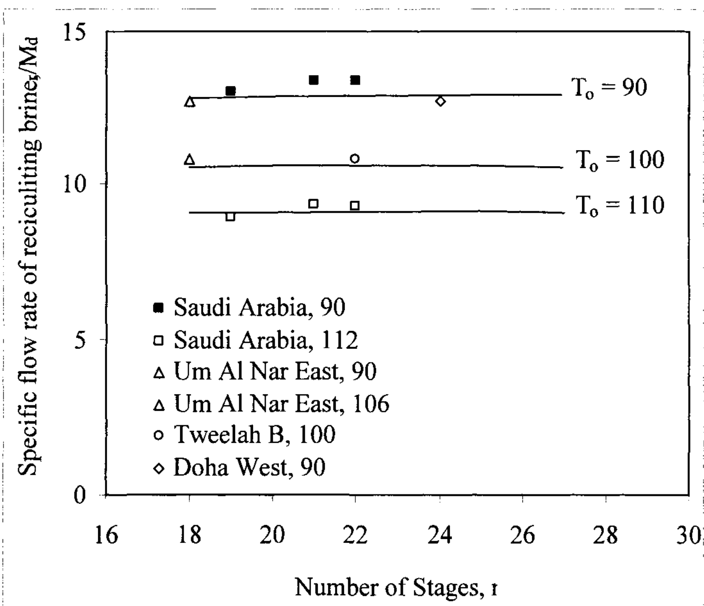

Fig. 32. Effect of number of stages on the specific of recirculating brine flow rate

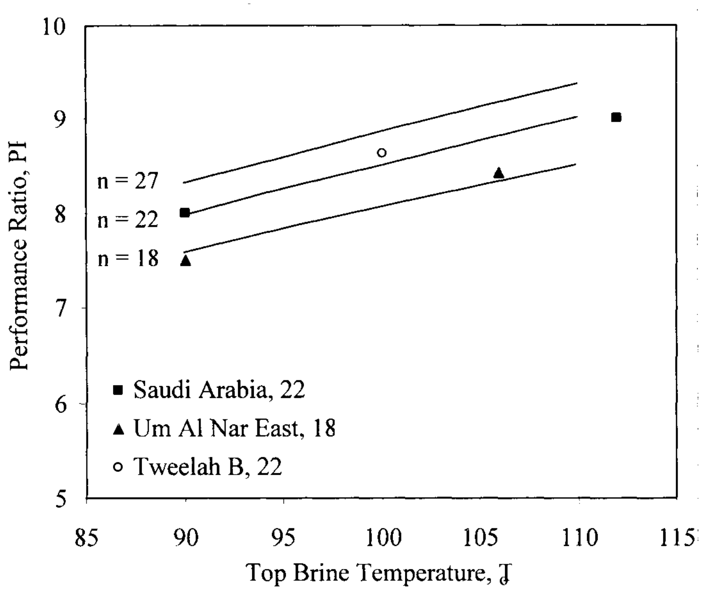

Fig. 33. Effect of top brine temperature on the plant performance ratio

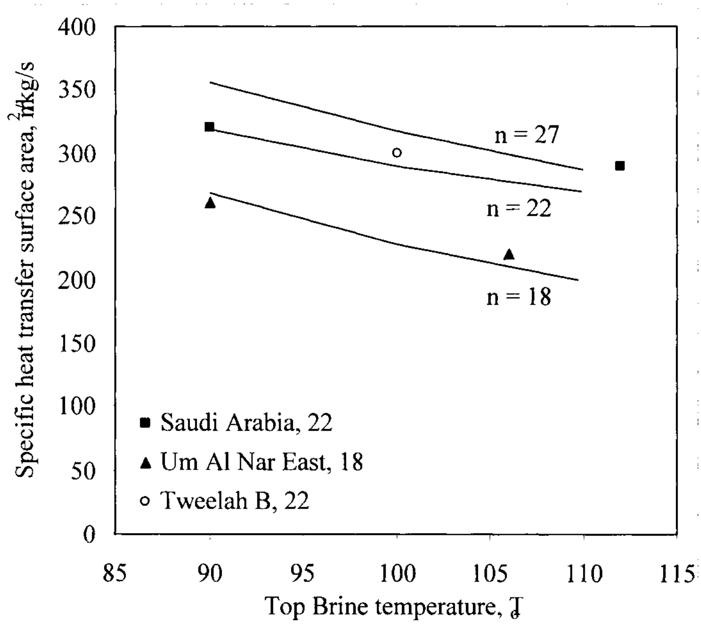

Fig. 34. Effect of number of stages on the specific heat transfer area

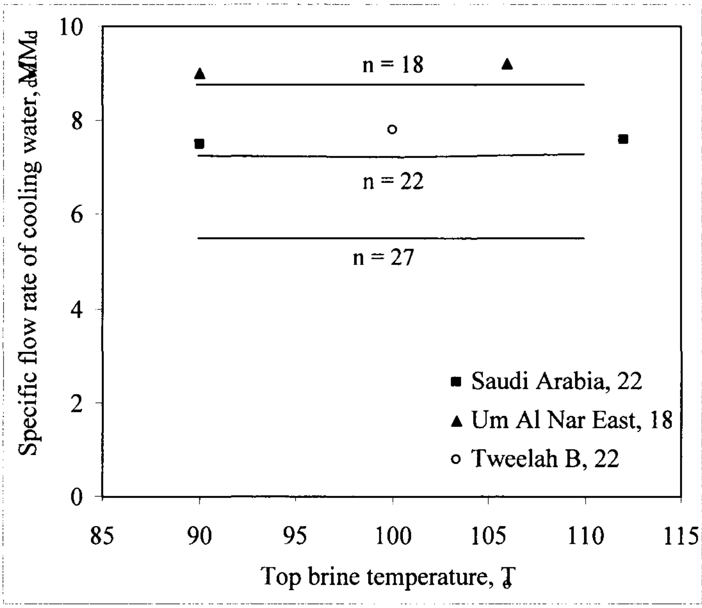

Fig. 35. Effect of number of stages on the specific flow rate of cooling water

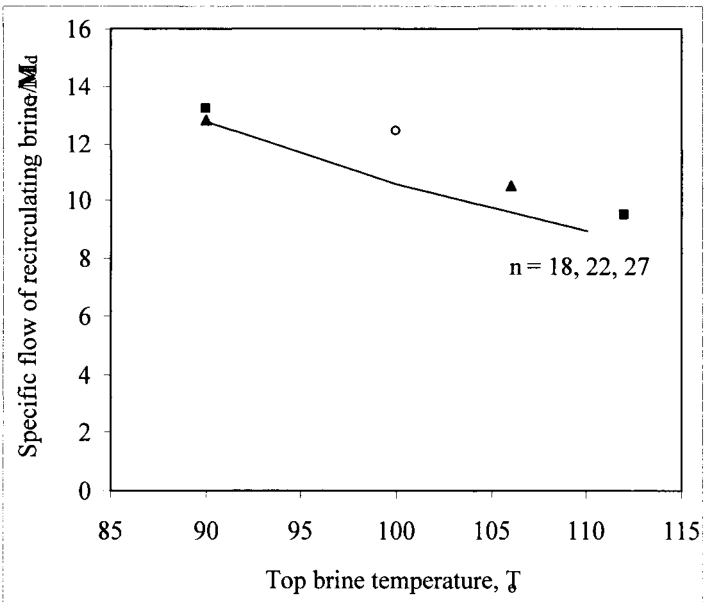

Fig. 36. Effect of top brine temperature on specific recirculating brine flow rate

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Problems

Problem 1

An MSF brine circulation plant has the following design data:

Plant capacity:	Unknown
Seawater temperature:	32 °C
Seawater salinity:	49400 ppm
Top brine temperature:	100 °C
Performance ratio:	8
Number of heat recovery stages:	19
Number of heat rejection stages:	3
Maximum brine concentration:	68600 ppm
Specific flow rate of cooling water:	1
Ratio of demister cross Sectional area to the chamber area:	0.75
Maximum vapor velocity in demister:	4 m/s
Maximum allowable flow rate Of brine per chamber width:	600×10^3 kg/m hr
Overall heat transfer coefficient in brine heater:	2 kW/m^2 °C
Overall heat transfer coefficient in heat rejection section:	1.9 kW/m^2 °C
Overall heat transfer coefficient in heat recovery section:	2.4 kW/m^2 °C

Calculate the following:

- The specific heat transfer area
- The vapor velocity in flashing chamber number 6.
- The liquid depth in flashing chamber number 10.

Problem 2

An MSF brine circulation plant has the following design data:

Plant capacity:	Unknown
Top brine temperature:	Unknown
Brine flow rate per chamber width:	Unknown
Number of stages:	20
Number of heat rejection stages:	3
Boiling temperature in last stage:	40 °C
Heat transfer area in the brine heater:	1000 m^2
Overall heat transfer coefficient in all sections:	2.527 kW/m^2 °C
Mass flow rate of heating steam:	16.782 kg/s
Heating steam temperature:	120 °C
Number of tubes in the brine heater:	1000 tube
Specific flow rate of brine circulation:	8.422
Diameter of tubes used in brine heater:	31.8 mm

Calculate the following:

- The plant performance ratio
- The specific heat transfer area
- The specific flow rate of cooling water
- The dimensions of chamber 7.

Problem 3

An MSF brine circulation plant has the following design data:

Seawater temperature:	34 °C
Seawater salinity:	42000 ppm
Top brine temperature:	100 °C
Temperature in the last stage:	40 °C
Temperature of heating steam:	110 °C
Specific flow rate of brine circulation:	8.478
Heat transfer area in the brine heater:	80 m^2
Number of heat rejection stages:	2

Overall heat transfer coefficient in brine heater:	1.5 $kW/m^2\ ^oC$

Calculate the following

- The plant performance ratio
- The specific flow rate of cooling water
- The terminal temperature difference of the first stage.
- The specific heat transfer area.
- The dimensions of stage 7.

Problem 4

An MSF brine circulation plant has the following design data:

Distillate flow rate:	5000 m^3/d
Seawater temperature:	30 ^oC
Seawater salinity:	44000 ppm
Top brine temperature:	112 ^oC
Temperature in the last stage:	40 ^oC
Temperature of heating steam:	116.7 ^oC
Terminal temperature difference:	3 ^oC
Number of heat rejection stages:	2
Number of heat recovery stages:	18

The overall heat transfer coefficient in the brine heater or the flashing stages is given by the relation

$$U = 6.5 - 0.03(115\text{-}T)$$

With U in $kW/m^2\ ^oC$ and T in ^oC.

Calculate the following

- Thermodynamic losses in the first stage
- Plant performance ratio
- Cooling water flow rate
- Brine heater surface area
- Flow rate of make up water.

Also, calculate the following parameters for stage number 5

- Boiling point elevation.
- Gate height
- Liquid level

- Demister temperature loss
- Stage height
- Stage length
- Preheater surface area.

Problem 5

An MSF brine circulation plant has the following design data:

Distillate flow rate:	22750 m^3/d
Seawater temperature:	28 ^oC
Seawater salinity:	45000 ppm
Top brine temperature:	90 ^oC
Temperature in the last stage:	40 ^oC
Temperature of heating steam:	100 ^oC
Terminal temperature difference:	3 ^oC
Number of heat rejection stages:	3
Number of heat recovery stages:	25
Width of stage 10:	16 m
Length of stage 10:	3.5 m

Calculate the following

- The temperature profile of flashing brine
- The temperature profile of seawater flowing in the preheaters
- The flow rates of heating steam, makeup water, brine circulation, cooling water, and brine blowdown.
- The pressure in stages 9, 10, and 11.
- The distillate product in stage 10.
- The vapor velocity in stage 10.

6.6. MSF with Thermal Vapor Compression

Attempts to improve the overall performance of the MSF system include design of hybrid configurations with the MVC or RO systems. The study by Genthner and El-Allawy (1983) focused on analysis of hybrid parallel configuration of MSF and MVC. As a result, several of the MVC devices are eliminated since their functions are provided from the hybrid MSF system. These functions include feed preheating, venting of non-condensable gases, chemical dosing, and pumps for distillate and brine blowdown. Results show lower chemical and specific fuel consumption with values of 50% and 65-75%, respectively. As for the RO-MSF hybrid system, it has similar advantages since the RO feed is extracted from the cooling seawater stream. This combination eliminates part of the pumping power requirements and treatment of the feed water for the RO system, Darwish et al. (1989) and Glueckstern (1995). Reported economics of the hybrid system are better than those for either system in a stand-alone mode. Regardless, actual use of the RO-MSF or the MVC-MSF systems are not found on industrial scale.

Development of the MSF brine circulation with thermal vapor compression (MSF-TVC) is motivated by the drastic increase in the thermal performance ratio of MEE upon combination with vapor compression. The field study by Temstet and Laborie (1995) and the performance studies by El-Dessouky et al. (1998) show that the thermal performance ratio of the stand-alone MEE system is approximately equal to the number of effects raised to the power 0.9. This gives thermal performance ratios of 5 and 9.4 for systems with 6 and 12 effects, respectively. The field study by de Gunzbourg and Larger (1998) and the performance evaluation of El-Dessouky and Ettouney (1997) show that combining the forward feed MEE system with lithium bromide vapor compression heat pumps increases the thermal performance ratio to a value of 21 for a 14-effect system. Similar findings are found in the studies by El-Dessouky et al. (2000) for the parallel feed MEE combined with thermal and mechanical vapor compression.

This section outlines features, model and analysis of the multistage flashing desalination combined with thermal vapor compression (MSF-TVC). The analysis considers several modes for vapor compression, which differs in the stage from which vapor is extracted. System evaluation focuses on calculations of the parameters that affect the unit product cost, which includes the thermal performance ratio, the specific heat transfer area, and the specific flow rate of cooling water. This report is the first, to the authors' knowledge, to propose and evaluate the performance of a combined MSF and thermal vapor compression. The next sections include brief description of the brine circulation MSF, the vapor compression modes, system model, results, and conclusions.

6.6.1 Process Description

Schematic diagrams for the proposed MSF-TVC processes are shown in Figs. 37 and 38. The systems constitute the brine heater, the flashing stages, and the steam jet ejector. The flashing stages are divided among the heat recovery and rejection sections. Each flashing stage includes brine pool, submerged orifice, vapor space, demister, distillate collection tray, and condenser tubes. The system also includes a cascade of venting units, which removes and prevents accumulation of non-condensable gases within the vapor space. The process is described in the following steps:

- The brine recycle stream (M_r) enters the brine heater tubes, where the compressed vapor (M_s) is condensed on the outside surface of the tubes. The brine stream absorbs the latent heat of condensing steam and its temperature increases to its maximum design value known as the top brine temperature (T_o). Its value depends on the nature of chemicals used to control the scale formation.
- The hot brine enters the flashing stages in the heat recovery section and then in the heat rejection section, where a small amount of fresh water vapor is formed by brine flashing in each stage. The flashing process takes place due to decrease in the stage saturation temperature and causes the reduction in the stage pressure.
- In each stage of the heat recovery section, the flashed off vapors condenses on the outside surface of the condenser tubes, where the brine recycle stream (M_r) flows inside the tube from the cold to the hot side of the plant. This heat recovery improves the process efficiency because of the increase in the feed seawater temperature.
- The condensed fresh water vapor outside the condenser tubes accumulates across the stages and forms the distillate product stream (M_d). This stream cascades in the same direction of the flashing brine from stage to stage and is withdrawn from the last stage in the heat rejection section.
- The intake seawater stream (M_f+M_{cw}) is introduced into the condenser tubes of the heat reject section, where its temperature is increased to a higher temperature by absorption of the latent heat of the condensing fresh water vapor.
- The warm stream of intake seawater is divided into two parts: the first is the cooling seawater (M_{cw}), which is rejected back to the sea and the second is the feed seawater (M_f), which is deaerated, chemically treated and then mixed in the brine pool of the last flashing stage in the heat rejection section.
- The brine recycle stream (M_r) is extracted from the brine pool of the last stage in the heat rejection section and is introduced into the condenser tubes of the last stage in the heat recovery section. As the stream flows in the condenser

tubes across the stages it absorbs the latent heat of condensation from the flashing vapor in each stage.

- The remaining brine in the last stage of the heat rejection section, known as the brine blowdown (M_b), is rejected to the sea.
- The steam jet ejector entrains a specified portion of the vapor formed in the flashing stages in the heat rejection (Fig. 37) or the heat recovery (Fig. 38). The motive steam compresses the entrained vapor to the desired temperature and pressure. The compressed vapor is then used to heat the brine recycle stream in the brine heater.
- Treatment of the intake seawater is limited simple screening. On the other hand, treatment of the feed seawater stream is more extensive and it includes dearation and addition of chemicals to control scaling, foaming, and corrosion.
- Another steam jet ejector is used to remove the non-condensable gases, which are released during flashing from each stage. Presence and accumulation of the gases result in reduction of the overall heat transfer coefficient for the condenser tubes and in turn reduces the process efficiency and the system productivity.
- The steam jet ejector is formed of a converging-diverging nozzle, a converging-diverging diffuser, a throat, a mixing zone, and a suction or entrainment chamber, Fig. 39. The motive steam, M_m, expands in the nozzle from state 1 to state 3, where its static pressure energy is converted to kinetic energy. The motive steam velocity becomes larger than the speed of sound (supersonic) as it leaves the nozzle. The suction chamber keeps the nozzle properly positioned with respect to the diffuser and directs the entrained vapor. The entrained vapor M_{ev} enters the suction chamber at pressure P_3 where it mixes violently and rapidly with the motive steam at point 4. The two streams mix together as they pass through the converging section of the venturi diffuser (from point 4 to 5). The mixture enters the throat section of the diffuser, completely mixed, at the sonic velocity of the mixture. The combined mixed streams is self compressed through the diverging section of the venture diffuser, where the cross sectional area increases and the velocity decreases, converting the kinetic energy of the mixture to static pressure energy. The mixture leaves the ejector at a pressure P_s that is intermediate to the motive (P_m) and suction (P_v) pressures. The steam jet ejector is designed to operate at the critical condition, where the supersonic shock wave is located at the nozzle exit. This condition occurs as the pressure compression ratio (P_m/P_v) is greater than 1.8. During operation, the shock wave may move downstream upon reduction in the condenser pressure. This has a negligible effect on the ejector performance, however, if conditions preventing formation of the shock wave are prevailed, the ejector performance deteriorates and no entrainment or compression takes place. In part stable operation is associated with absence of violent fluctuations in the suction pressure. If the ejector is designed to operate with a full stable range, it will have a constant mass flow rate of the

entrained vapor for different discharge pressures when the upstream conditions remain constant.

Two configurations are considered in this study that differs in the location of vapor entrainment, Figs. 37 and 38. In the first configuration (Fig. 37) vapor of equal thermal load and not flow rate is entrained from the stages forming the heat rejection section. This is necessary because of the equality of the heat transfer area and the thermal load of the flashed off vapor in these stages. Therefore, an equal rise occurs in the temperature of the feed seawater stream flowing on the inside of the condenser tubes of the heat rejection section. This scheme has some similarity to vapor compression in the MEE system, where vapor entrained by the steam jet ejector is at a low temperature of 40 °C, El-Dessouky and Ettouney (1997). However, the difference between MEE and MSF is the requirement to compress the vapor to a higher temperature of 90-110 °C in the MSF, while in MEE the compressed vapor temperature can be as low as 60 °C. In MSF constraints imposed by the flashing range and the temperature drop per stage dictates vapor compression to high temperatures. As is shown, mode (a) requires the use of two steam jet ejectors in series. This is because vapor compression from a temperature of 40 °C to 110 °C gives a pressure compression ratio of 19.4 between the compressed and the entrained vapor. Design and operation of steam jet ejectors puts a maximum limit of 5 on the compression ratio, Power (1994). As a result, operating the first ejector at the maximum limit gives an outlet pressure of 36.92 kPa for the compressed vapor corresponding to a saturation temperature of 73.95 °C. The remaining compression range between 73.95 and 110 °C is 3.88, which is achieved in the second ejector.

In the second configuration (Fig. 38) vapor of equal thermal load is equally entrained from 2 to 4 stages in the heat recovery section. Again the equal heat transfer area and the flashing thermal load in the heat recovery section dictate the above equality constraint. However, the higher vapor temperature in the heat recovery section would require one steam jet ejector only. For example, vapor compression from 70 °C to 110 °C gives a compression ratio of 4.6, which is lower than the maximum limit. As is shown in Figs. 37 and 38 separate ejectors are used to entrain and compress vapor from each stage. Use of a single ejector is not possible, because of differences in the pressure of entrained vapor.

In summary, differences in system design of the conventional and thermal vapor compression MSF include the following:

- As will be discussed later the MSF-TVC system has a higher thermal performance ratio than conventional MSF. This implies reduction in the amount of input energy to the system per unit mass of product distillate water. As a result, lower specific heat transfer area for the condenser tubing and smaller pump capacity for the cooling seawater are required for the MSF-TVC system.

- Elimination of the control loop and associated devices on the temperature and pressure of the low pressure steam used in conventional MSF. This loop reduces the steam temperature and pressure from 155 °C and 5.5 bar to the saturation conditions, which may vary from 100-120 °C. In the proposed MSF-TVC system the motive steam is directly extracted from the medium pressure steam line at 15 bar.
- Increase in the steam pressure for the MSF-TVC would result in reducing the size of the steam piping system. This is because of reduction in the specific volume of the vapor at higher pressures. In conventional MSF use of lower pressure steam, which has higher specific volume, would require use of larger pipe diameter.

6.6.2 Mathematical Model and Solution Procedure

Mathematical model and solution procedure for MSF with thermal vapor compression are similar to the model of the stand-alone brine circulation system. The model elements include the following:
- Energy, mass, and salt balances for each flashing stage.
- Heat transfer equations for condenser/preheater tubes in heat recovery and heat rejection section.
- Ejector equations.

It should be noted that all the model equations and correlations are given in the previous section for the brine circulation system. Also, the model equations for the ejector are given in the single and multiple effect systems.

6.6.3 System Performance

Performance of the MSF-TVC modes (a) and (b) are shown in Figs. 39-42, which includes the temperature profiles, the performance ratio, the specific heat transfer area, the specific flow rate of cooling water, and the piping size.

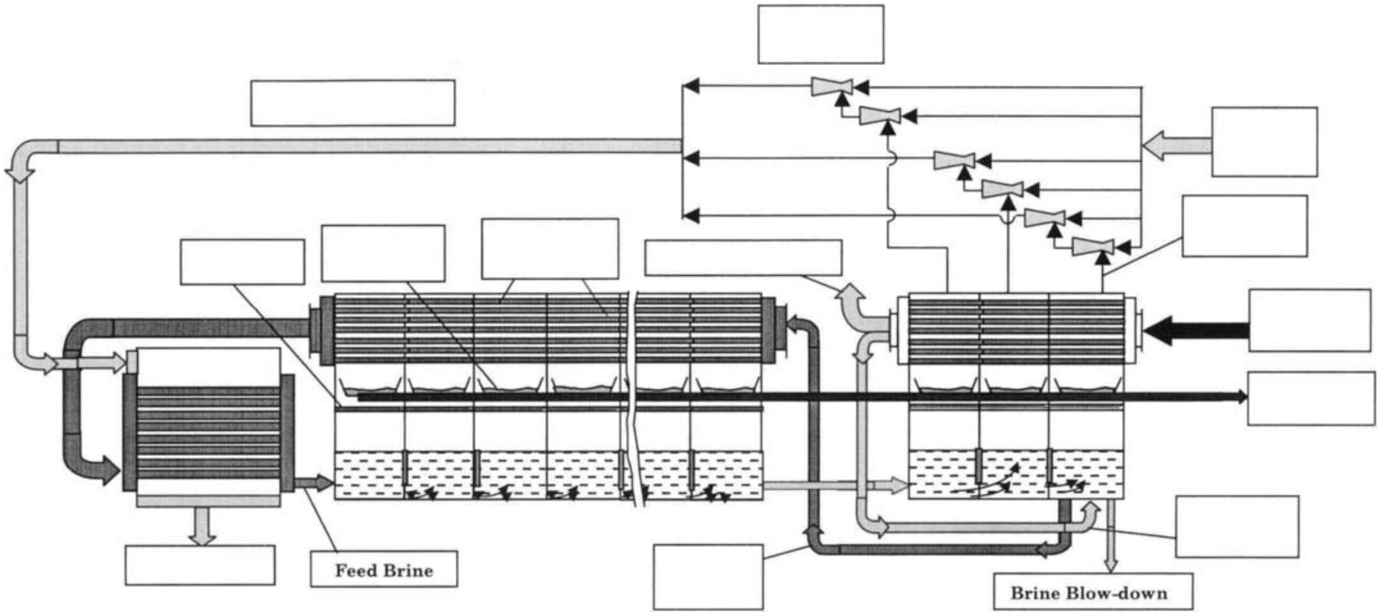

Fig. 37. MSF with thermal vapor compression. Mode (a) Vapor entrainment from the heat rejection stages.

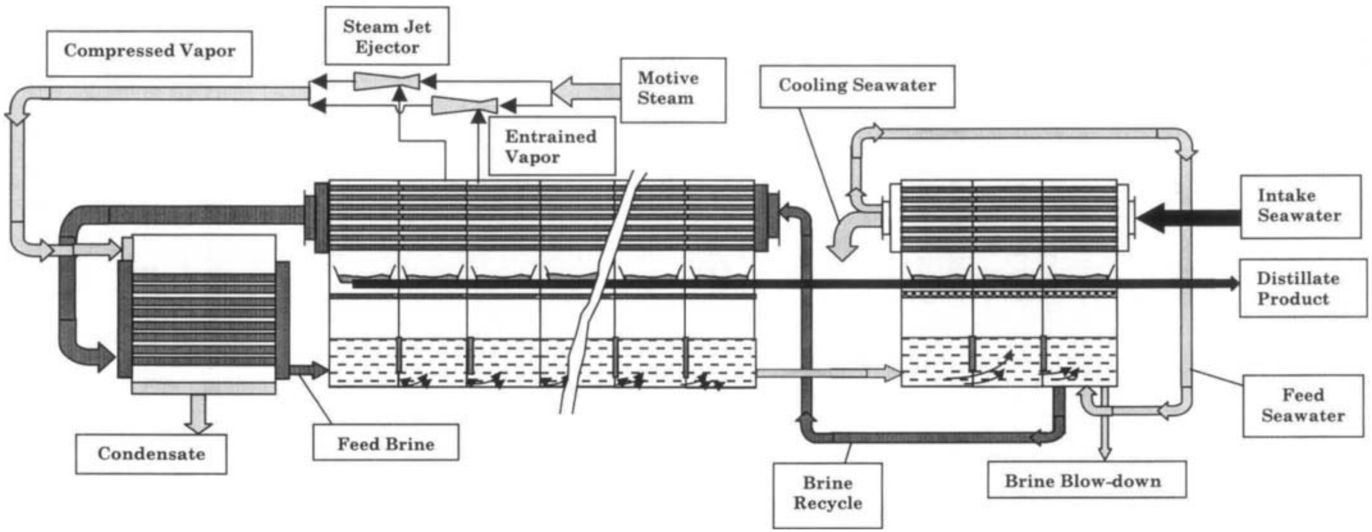

Fig. 38. MSF with thermal vapor compression. Mode (b) Vapor entrainment from the heat recovery stages.

The temperature profiles shown in Fig. 39 are obtained for the flashing brine, the brine recycle, and the feed seawater. In both configurations the temperature profiles for the flashing brine and flashed off vapor are identical to that of conventional MSF. This is because the flashing range and the temperature drop per stage are kept constant for all systems. Similarly, the temperature profiles of the feed seawater and brine recycle stream in configuration (a) is the same as conventional MSF. This is because the temperature differences across the heat recovery or heat rejection sections are the same as those for the MSF system. Therefore, vapor entrainment in the heat rejection section would only result in reduction of the thermal load, which implies reduction of the flow rate of cooling water. In configuration (b) the temperature profile of brine recycle in the heat recovery section differs from those for the MSF system. The difference is found in stages 1-6, where vapor is entrained. As a result, the recycle brine temperature does not increase to the same value as it leaves the heat recovery section as in configuration (a) and conventional MSF.

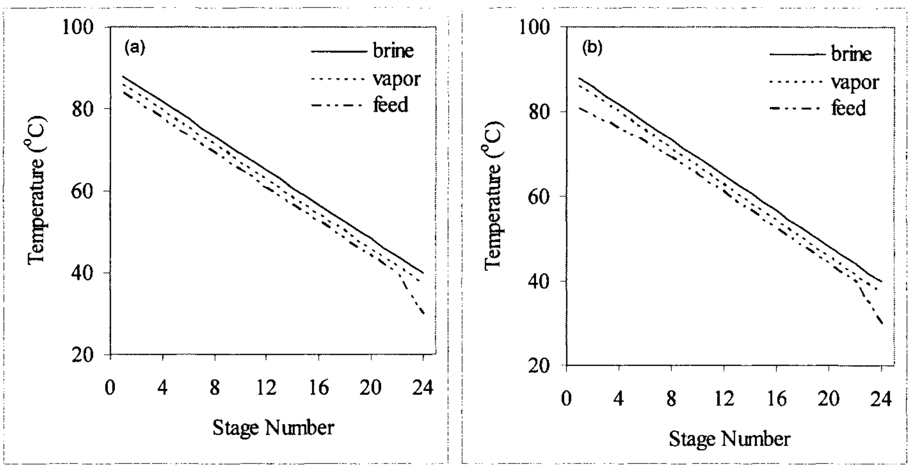

Fig. 39. Temperature profiles in MSF-TVC. (a) Vapor entrainment from the heat rejection section. (b) Vapor entrainment from the heat recovery section.

Fig. 40 shows variations in the thermal performance ratio for modes (a) and (b) as well as the conventional MSF system. For the three systems, the thermal performance ratio increases upon the increase in the top brine temperature. This is because of the flashing range, which results in the increase of the temperature drop per stage and consequently the increase in the amount of flashed off vapor per stage. The highest performance ratio is obtained for the vapor compression mode (b) with values varying between 9 and 10.5 as the top brine temperature increases from 90 to 110 °C. Lower performance ratio is obtained for the vapor compression mode (a) and the lowest values are obtained for the conventional MSF system. The higher performance ratio for mode (b) is caused by reduction in the motive steam requirements for vapor compression at higher temperatures. As a result, the average percentage increase in the thermal performance ratio for mode (a) over mode (b) and the conventional MSF is 6 and 14%, respectively.

Results for the specific heat transfer area for configuration (a), (b) and the MSF system are shown in Fig. 41. The specific heat transfer area for the three configuration decreases upon the increase in the top brine temperature. This is because of the increase in the driving force for heat transfer. It should be noted that the heat transfer area for mode (a) and the conventional MSF system are similar. This is because of the same temperature profiles for both system in the heat recovery and rejection sections. Therefore, the temperature drop per stage in each system and consequently, the driving force for heat transfer are identical. As for mode (b) it has a smaller specific heat transfer area than the other two systems because of the increase in the temperature driving force between the flashing vapor and the brine recycle stream, Fig. 40. The average percentage reduction in the specific heat transfer area for mode (B) is 8% lower than mode (a) and conventional MSF.

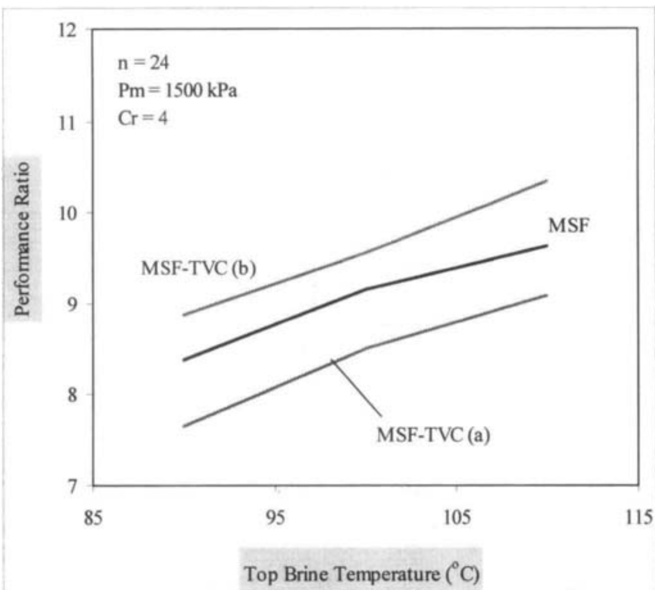

Fig. 40. Variation in the performance ratio as a function of the compression mode and the top brine temperature

Variations in the specific flow rate of cooling water are shown in Fig. 42 for the three configurations. As is shown, the specific flow rate of cooling water decreases with the increase in the top brine temperature. This is caused by reduction in the amount of heating or motive steam per unit mass of product water, or the increase in the system thermal performance ratio. Therefore, the thermal load of the system decreases at higher top brine temperatures and consequently the specific flow rate of the cooling water decreases. The lowest specific flow rate for the cooling water is obtained for mode (a), followed by mode (b), and then the MSF system. The average percentage reduction in the specific flow rate of cooling water in mode (a) over mode (b) and the conventional MSF are 44 and 56%, respectively. The large reduction in the specific flow rate of cooling water for mode is caused by vapor entrainment from the heat rejection section and simultaneous increase in the system thermal performance ratio. Both factor reduces the thermal load of the heat rejection section and consequently the specific flow rate of cooling water.

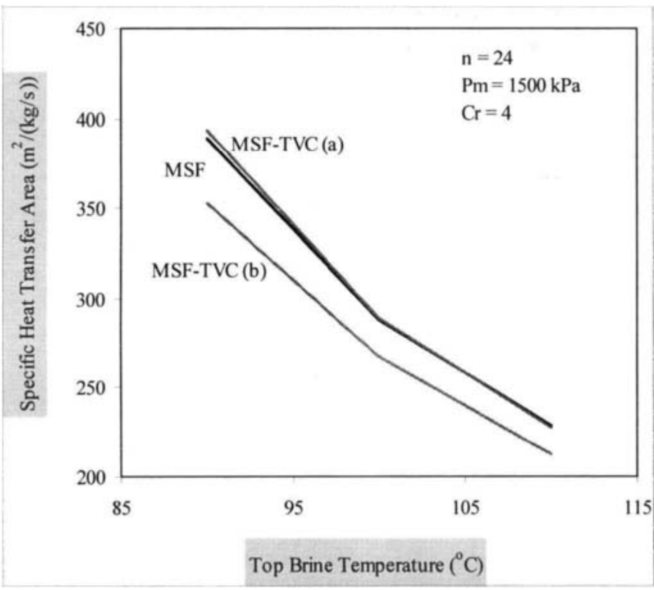

Fig. 41. Variation in the specific heat transfer area as a function of the compression mode and the top brine temperature

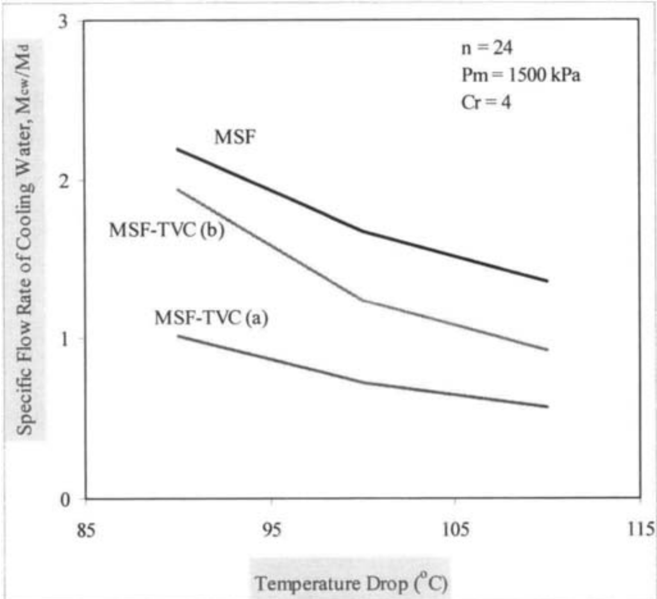

Fig. 42. Variation in the specific flow rate of cooling water as a function of the compression mode and the top brine temperature

6.6.4 Summary

A novel MSF configuration, which is based on thermal vapor compression, is proposed and evaluated as a function of the vapor compression mode and the operating conditions. Analysis of the results shows the following:

- Thermal vapor compression enhances the performance of the MSF system as a result of increase in the performance ratio and reduction in the specific flow rate of cooling water and the specific heat transfer area.
- Vapor compression from stages operating at higher temperatures in the heat recovery section give higher performance ratios than for vapor compression from the heat rejection section.
- The specific heat transfer area for the vapor compression mode from the heat recovery section gives lower specific heat transfer area from the heat rejection section.
- The pipe diameter for the motive steam in MSF-TVC is lower than the diameter for the heating steam by an average of 60%.

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6.7. *MSF with Brine Mixing*

This section presents a novel MSF process with design and operation advantages over the brine circulation MSF or the once through process MSF-OT. The proposed system is based on the essentials, features, and fundamentals of the MSF process. Therefore, implementation of the proposed system requires a limited number of modifications in MSF plants already in operation.

6.7.1 *Process Description*

The layout for the brine mixing process is shown in Fig. 43. Also, the temperature profiles of the unevaporated and flashing brine for the MSF, MSF-M, and MSF-OT are shown in Fig. 44. As illustrated, the configuration is similar, in some extent, to the heat recovery section in conventional MSF as well as the once through MSF. The system contains three main sections; these are the brine heater (or the heat input section), the flashing stages (the heat recovery section), and the brine mixing tank. As is shown the unevaporated brine recycle stream flows in a counter current direction to the flashing brine and distillate product in the flashing stages. The unevaporated brine flows from the cold side of the plant to the hot side, while, the distillate product and flashing brine flow in the opposite direction. In the brine heater, saturated steam at a flow rate of, M_s, is used to increase the temperature of the unevaporated brine to the top brine temperature, T_o. The temperature of the heating steam, T_s, is higher than the top brine temperature by a specified few degrees. The heated brine enters the first flashing chamber, where a small portion of distillate is formed. The flashing process converts the excess sensible heat of the unevaporated brine into latent heat required for vapor formation. The flashing process continues throughout the stages, where small amounts of distillate product, D_i, are formed in each stage. As a result of water evaporation, the brine salinity increases across the stages. The maximum permissible value of salt concentration is limited to 70,000 ppm in order to prevent formation of calcium sulfate scaling.

The formed vapor passes through wire mesh mist eliminator, known as the demister. The demister removes brine droplets entrained by the flashing vapor. Brine droplets are formed because of foaming, bursting of vapor bubbles at the brine surface, and brine splashing at the stage orifice. The separation of entrained brine droplets is necessary to prevent contamination of the distillate product and reduction of its quality; moreover, the separation prevents scale formation on the outside surface of the condenser tubes. The vapor condenses on the outside surface of the preheater/condenser tubes, where the unevaporated brine flows. The vapor releases its latent heat to the brine stream and as a result the temperature of the brine stream increases across the stages. The distillate product is collected in the distillate trays and accumulates across the stages.

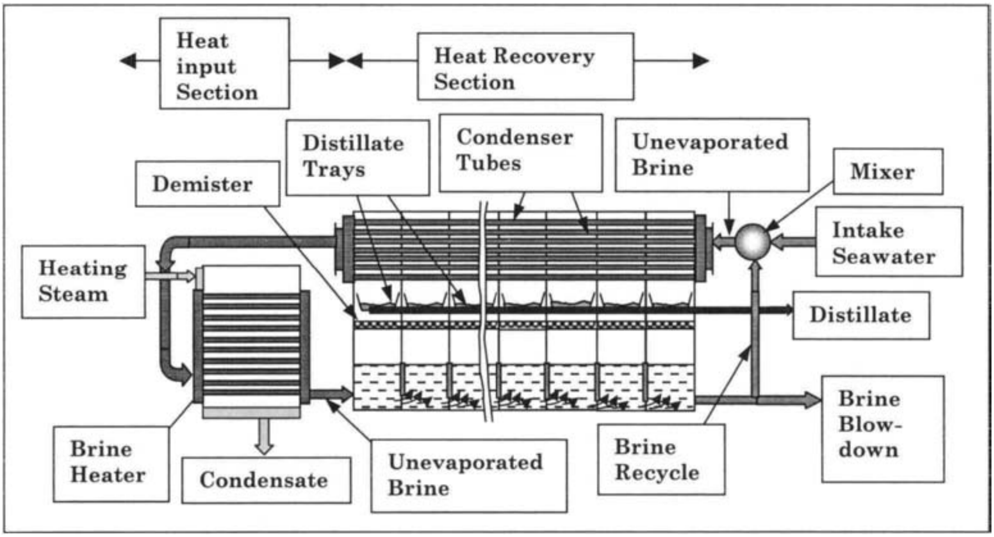

Fig. 43. Schematic of MSF with brine mixing

In each stage, the temperature of the formed vapor, T_{v_i}, is lower than the flashing brine temperature, T_i, by the boiling point elevation. The value of the boiling point elevation is affected by the salinity of the flashing brine and the boiling temperature. Further reduction in the vapor saturation temperature is caused by pressure drop in the demister and during condensation.

The brine leaving the last stage is divided into two parts; the first is rejected back to the sea and the second is recycled. The ratio between these parts can be controlled and depends mainly on the intake seawater temperature. The aim of brine rejection to the sea is to control salt concentration inside the plant. On the other hand, the purpose of brine recirculation is to decrease the flow rate of the feed seawater and recover a part of the heat added to the system in the brine heater. This lowers the chemical additive consumption rate and the size of the pretreatment facilities for the feed stream. Also, since the recycled brine contains higher energy than the feed seawater, the process thermal efficiency will improve. The portion of the brine stream leaving the last stage, M_r-M_f is mixed with the intake seawater stream, M_f. The resulting mixture, M_r, has a higher salinity and temperature than the intake seawater. The mixing process is expected to cause a thermal shock because of differences in the temperatures of the intake seawater and the brine recycle. This would result in dissociation of the bicarbonate compounds and formation of carbon dioxide gas. Therefore, the mixing unit must be properly vented to avoid accumulation of non-condensable gases in the brine recycle stream.

The flashing process and vapor formation is limited by increase in the specific vapor volume at lower temperatures and difficulties encountered for operation at low pressures. Common practice limits the temperature of the last stage to values of 32 and 40 °C, for winter and summer operation, respectively. Further reduction in these temperatures results in drastic increase of the stage volume and its dimensions. In addition, since most of the stages operate at temperatures below 100 °C, the pressure within the stages is at vacuum conditions. This may result in leak of the outside air. At such conditions, air is non-condensable and its presence in the system may result in severe reduction in the heat transfer rates within the chamber, increase of the chamber pressure, and reduction of the flashing rates. Accumulation of non-condensable gases is also generated from trace amounts of dissolved gases, which are not removed in the dearation pretreatment. This condition necessitates proper venting of the flashing stages to enhance the flashing process and to improve the system efficiency.

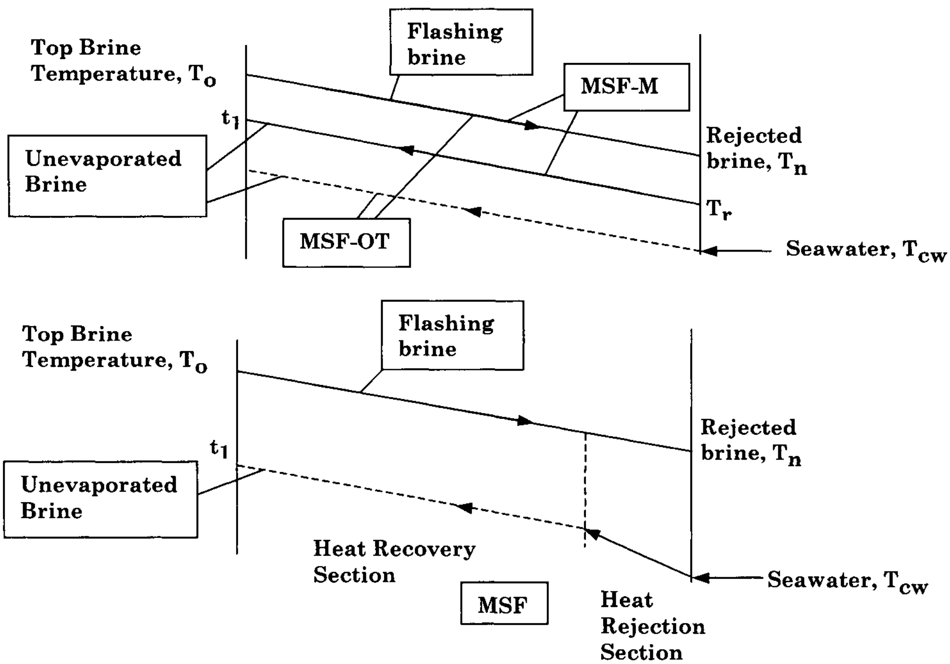

Fig. 44. Temperature profiles for the MSF-M, the MSF-OT, and MSF.

In summary, the main differences between the proposed system and conventional MSF with brine circulation are:

- Removal of the heat rejection section.
- Absence of the cooling water loop used in conventional MSF to control the flashing temperature in the last stage and to remove excess energy added to the system in the brine heater.
- Elimination of the cooling water recycle loop, which is used to adjust the flashing temperature of the seawater of the last flashing stage in the heat rejection section, when the seawater temperature becomes very at winter operation conditions.
- The mixing of the brine recycle and the feed seawater take place in an external mixing tank rather than inside the flashing stages.
- The salinity of the rejected brine can be less than the limiting value of 70,000; this depends on the temperature of feed seawater.
- The flow rate of feed seawater is not constant and is regulated subject to the temperature and salinity of the seawater.

Differences for the proposed system and the once through configuration are:

- Part of the brine flowing in the last stage is recirculated to the system.
- Brine recirculation reduces the flow rate of the feed seawater; consequently lower amounts of chemical additives are used and smaller size pretreatment plant is required, which include screening, filtration, and dearation.
- The use of the mixing tank for the feed stream and the brine recycle stream gives a better control on the temperature of the brine feed to the condenser tubes of the last flashing stage.
- The salinity of the recycle brine is higher than the feed seawater.
- Dearation of the feed seawater takes place outside the stage; this reduces the corrosion rate inside the stages.
- The system is less sensitive to variations in feed seawater temperature; because it can be controlled by the brine circulation rate.

6.7.2 Mathematical Model

The MSF-M mathematical model constitutes the same balance equations and correlations used in the previous MSF systems. The model includes stage energy and material balances as well as the heat transfer equations for condenser/preheater. The model also includes the material and energy equations for the brine mixer, Fig. 45. The salt balance is given by

$$X_f M_f + (M_r - M_f) X_n = M_r X_r$$

This simplifies to

$$X_r = X_n - M_f/M_r\,(X_n - X_f) \qquad (99)$$

The energy balance is given by

$$(M_r - M_f)\,C_p\,(T_n - T_f) = M_r\,C_p\,(T_r - T_f)$$

This relation is rearranged into

$$T_r = T_n - M_f/M_r\,(T_n - T_f) \qquad (100)$$

Solution procedure and system parameters used in performance evaluation are identical to those of the previous MSF systems. The solution procedure is based on Newton's method and it evaluates through iterative sequence the flow rates, temperature, and salinity of various streams.

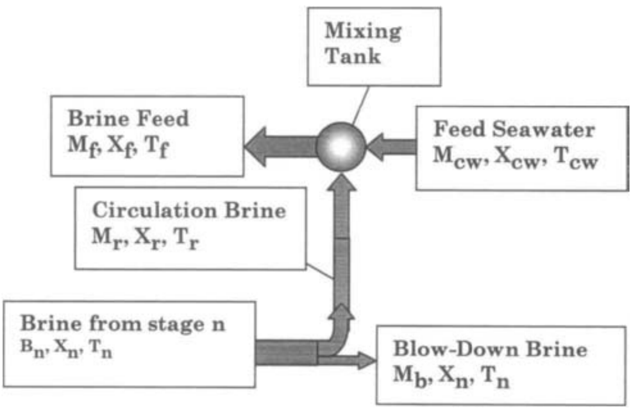

Fig. 45. Temperature profiles for the MSF-M, the MSF-OT, and MSF

6.7.3 System Performance

The performance of the proposed MSF-M system is analyzed as a function of the top brine temperature, the temperature of brine recycle, and the number of stages. Results shown in Figs. 46 and 47 include variations in the thermal performance ratio and the total specific heat transfer area as a function of the recycle temperature and the number of stages, respectively. The results shown in Fig. 46 are made for 24 flashing stages and seawater temperature of 32 °C. As is shown in Fig. 46 the thermal performance ratio increases at higher top brine temperatures and lower recycle temperatures. At these conditions, the flashing range increases resulting in higher temperature drop per stage. This increases the amount of flashing vapor per stage and consequently the total amount of product distillate increases. At the same conditions, the specific heat transfer decreases at higher top brine temperature and lower recycle temperatures. As discussed before, this decrease is caused by the increase in the amount of distillate product. Also, increase of the top brine temperature enhances the heat transfer rate across the condenser tubes. This is caused by the viscosity decrease and thermal conductivity increase for the seawater water and the distillate streams at higher temperatures. Effect of the number of stages on the system performance is shown in Fig. 47, at a recycle temperature of 37 °C and seawater temperature of 32 °C. Calculations are made for systems with 20, 24, 28, and 32 stages. As is shown the performance ratio and the specific heat transfer area increase at larger number of stages. Increase of the number of stages reduces the stage temperature drop. Therefore, the driving force for heat transfer is reduced. This increases the heat transfer area of the condensers in each stage and consequently the amount of distillate condensate per stage.

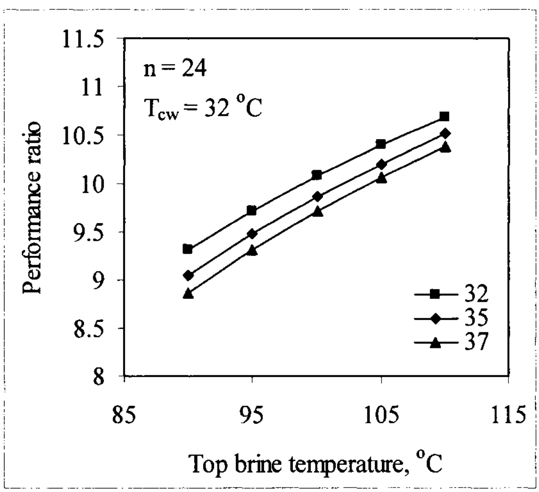

Fig. 46. Effect of the top brine temperature and brine recycle temperature on the performance ratio

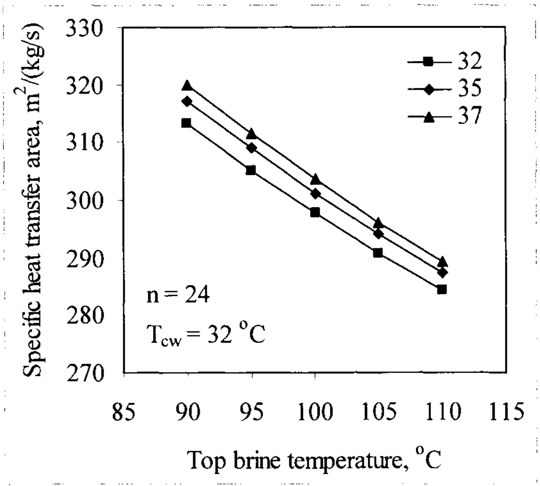

Fig. 47. Effect of the top brine temperature and brine recycle temperature on the specific heat transfer area.

6.7.4 Modification of Existing MSF Plants

Conversion of the MSF with brine recirculation to the MSF-M configuration is simple and primarily involves elimination of the brine circulation and the cooling seawater streams. The conversion includes the following:

- Removal of the cooling seawater loop as well as the temperature control loop on the feed seawater temperature.
- Modify the brine circulation loop to recycle the brine to the storage tank instead of the last stage.
- Addition of the accumulation tank for the recycled brine stream.
- Connect the preheater tubes of the first stage in the heat rejection section and the last stage in heat recovery section.
- Replace the intake seawater pump with a smaller capacity pump; this is necessary, since in the cooling seawater stream in the MSF-M system is eliminated.
- It should be stressed that conversion of the MSF system into MSF-M is feasible since the temperature of the brine blow down and the recycle flow rates, during the whole year, are similar in both systems. Therefore, the two ratios $M_r/\Delta P$ and M_d/ρ_v will have similar value and the design features of the MSF, which includes stage dimensions, brine flow area across the stage, and the non-condensable gases vacuum system, will be suitable for the MSF-M system.

6.7.5 Summary

A novel system is presented for the multi-stage flash desalination. The system is based on the conventional MSF configuration, where it includes two of its basic elements; the brine heater and the heat recovery section. The cooling water stream, typical of the MSF system, is also eliminated. The new system adopts a direct brine recycle stream from the brine stream leaving the last flashing stage. This stream is mixed with the intake seawater stream in an insulated and vented tank. Accordingly, the temperature of the feed stream is adjusted to meet summer and winter operating conditions. In the light of above, the following conclusions are made:

- Control of the intake seawater temperature is an essential feature in all thermal desalination process. This is found in the mixing tank of the MSF-M system and in the heat rejection section of the conventional MSF.
- This temperature control feature is essential in winter operation. Lack of this control, as in the MSF-OT system, reduces the brine temperature in the last stage and results in large increase the pressure drop across the stages.
- Reduction of the brine temperature in the last stage during winter operation for the MSF-OT system, necessitates increase of the stage dimensions to accommodate the large increase in the vapor specific volume. Also, larger flow area for the brine stream is necessary to meet the high pressure drop across the stages. Additionally, increase in the capacity of the non-condensable gases vacuum system, wall thickness, and leakage rate of the outside air are dictated by lack of control of the intake seawater temperature.
- Operation of the MSF-M system with no brine recycle reduces the system to the MSF-OT configuration. This condition can be adopted during the summer period, where the seawater temperature is high enough to ensure high brine temperatures in the last stage.
- In actual operation, brine circulation in the MSF-M system would be favored, even during summer operation. This condition reduces the amount of intake seawater and the amount of chemicals used to control scaling, corrosion, and foaming.
- The MSF-M has a similar consumption rate of the antiscalent to the MSF system. Since MSF-OT system has no brine recycle stream, its antiscalent consumption rate is 3-4 times higher than the other two systems.

Problems

Problem 1

An MSF-M system has the following design data:

Plant capacity:	Unknown
Seawater temperature:	32 °C
Brine recycle temperature:	36 °C
Seawater salinity:	49400 ppm
Top brine temperature:	100 °C
Performance ratio:	8
Number of flashing stages	22
Maximum brine concentration:	68600 ppm
Ratio of demister cross Sectional area to the chamber area:	0.75
Maximum vapor velocity in demister:	4 m/s
Maximum allowable flow rate of brine per chamber width:	600×10^3 kg/m hr
Overall heat transfer coefficient in brine heater:	2 kW/m^2 °C
Overall heat transfer coefficient in flashing stages:	2.4 kW/m^2 °C

Calculate the following:

- The specific heat transfer area
- The thermal performance ratio
- The brine recycle flow rate

Problem 2

An MSF-M plant has the following design data:

Plant capacity:	Unknown
Top brine temperature:	Unknown
Brine flow rate per chamber width:	Unknown
Number of stages:	20
Brine recycle temperature:	36 °C
Boiling temperature in last stage:	40 °C
Heat transfer area in the brine heater:	1000 m^2
Overall heat transfer coefficient	2.527 kW/m^2 °C
Mass flow rate of heating steam:	16.782 kg/s

Heating steam temperature: 120 ^oC
Number of tubes in the brine heater: 1000 tube
Specific flow rate of brine circulation: 8.422
Diameter of tubes used in brine heater: 31.8 mm

Calculate the following:

- The plant performance ratio
- The specific heat transfer area

Problem 3

An MSF-M plant has the following design data:

Seawater temperature: 34 ^oC
Brine recycle temperature: 36 ^oC
Seawater salinity: 42000 ppm
Top brine temperature: 100 ^oC
Temperature in the last stage: 40 ^oC
Temperature of heating steam: 110 ^oC
Specific flow rate of brine circulation: 8.478
Heat transfer area in the brine heater: 80 m^2
Overall heat transfer coefficient
in brine heater: 1.5 $kW/m^2\,^oC$

Calculate the following

- The plant performance ratio
- The terminal temperature difference of the first stage.
- The specific heat transfer area.

Problem 4

An MSF-M plant has the following design data:

Distillate flow rate: 5000 m^3/d
Brine recycle temperature: 34 ^oC
Seawater temperature: 30 ^oC
Seawater salinity: 44000 ppm
Top brine temperature: 112 ^oC
Temperature in the last stage: 40 ^oC
Temperature of heating steam: 116.7 ^oC
Terminal temperature difference: 3 ^oC
Number of flashing stages: 18

The overall heat transfer coefficient in the brine heater or the flashing stages is given by the relation

$$U = 6.5 - 0.03(115\text{-}T)$$

With U in kW/m^2 ^{o}C and T in ^{o}C.

Calculate the following

- Thermodynamic losses in the first stage
- Plant performance ratio
- Brine heater surface area
- Flow rate of make up water.

Problem 5

An MSF-M plant has the following design data:

Distillate flow rate:	22750 m^3/d
Brine recycle temperature:	32 ^{o}C
Seawater temperature:	28 ^{o}C
Seawater salinity:	45000 ppm
Top brine temperature:	90 ^{o}C
Temperature in the last stage:	40 ^{o}C
Temperature of heating steam:	100 ^{o}C
Terminal temperature difference:	3 ^{o}C
Number of stages:	40
Width of stage 10:	16 m
Length of stage 10:	3.5 m

Calculate the following

- The temperature profile of flashing brine
- The temperature profile of seawater flowing in the preheaters
- The flow rates of heating steam, makeup water, brine circulation, cooling water, and brine blowdown.
- The pressure in stages 9, 10, and 11.
- The distillate product in stage 10.
- The vapor velocity in stage 10.