MECHANICAL VAPOR COMPRESSION DISTILLATION

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Contents

Summary

This chapter attempts to provide a brief overview of the mechanical vapor compression distillation process description, analysis, effects of salinity and temperature on energy consumption, system heat balance, heat transfer surface requirement and effects of evaporator temperature on heat transfer area and type of energy. It also presents a comparison of single-effect and multieffect vapor compression distillation systems, forced circulation vapor compression, and comparison of lobe-type and centrifugal compressors.

1. Process Description

Mechanical vapor compression distillation of saline water is a process in which mechanical energy is used as the main driving force. It is basically a heat pump process that pumps heat from a low-temperature reservoir to a higher temperature reservoir, just as in household refrigerators where the heat is pumped from the air inside the refrigerator box to the ambient air outside of the box. The main elements consist of an

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evaporator, a condenser, a compressor, and an expansion valve where refrigerants such as Freon or ammonia are used as working fluid in a closed cycle. The coils inside the refrigerator box form the evaporator and the coils outside the box form the condenser. Here the working fluid inside the coils in the evaporator, evaporates to absorb heat from inside the box, compressed to raise its saturation pressure and temperature and then enters the condenser coils outside the box to release its latent heat of condensation to the ambient air. The high-pressure liquid refrigerant then enters the expansion valve where it flashes down to the evaporator pressure to complete the cycle.

In a vapor compression distillation of saline water, the system is not a closed cycle and the basic system consists of the evaporator-condenser and the compressor. The working fluid is water vapor. Water vapor generated from the water layer on the evaporation side of the heat transfer surface is compressed to raise its saturation temperature and pressure and is then condensed on the condensation side of the heat transfer surface, forming condensate and releasing the latent heat of condensation to be transferred across the heat transfer surface into the water layer on the evaporation side to cause further evaporation of water. The only energy required to pump the heat content in the vapor from the evaporation side to the condensation side of the heat transfer surface is the mechanical energy required to drive the compressor.

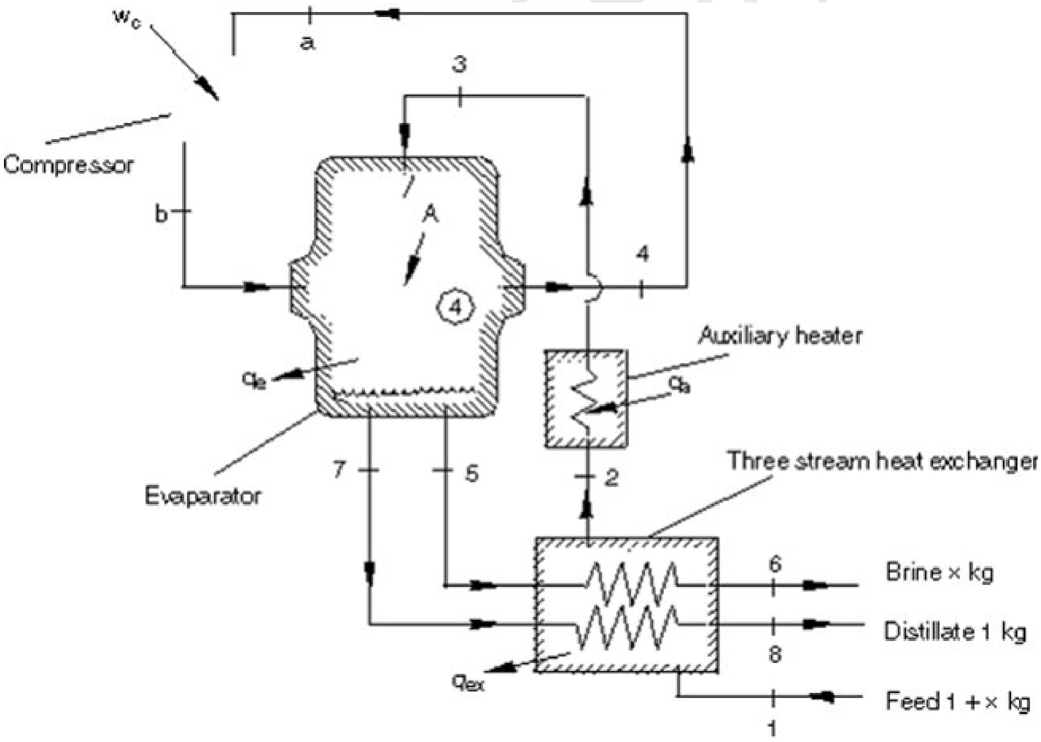

Figure 1. Flow diagram of single-effect vapor compression distillation system.

Figure 1 shows a general simplified flow diagram of a basic mechanical vapor compression distillation system. As stated earlier, the system consists of an evaporator-condenser, shown as heat transfer *A*, a mechanically driven vapor compressor, a three-stream heat exchanger to pre-heat the feed stream by cooling the distillate and brine streams, and an auxiliary heater to supply additional heat to the feed when needed. For simplicity, pumps and auxiliary equipment are not shown here.

In this system the feed enters the three-stream heat exchanger at state point 1, where it is heated to state point 2, and then enters the auxiliary heater to be heated to state point 3, where it enters the evaporation side of the evaporator. Here the feed is brought into contact with the heat transfer surface *A* where part of it becomes vapor at state point 4 and the balance is taken out as brine at state point 5, to be cooled in the three-stream exchanger to state point 6.

The vapor generated at state point 4 enters the compressor at state point a where it is compressed to state point b and then enters the condensation side of the evaporator where it condenses on the other side of the heat transfer surface *A* at state point 7 and is then cooled to state point 8 in the three-stream exchanger.

2. Process Analysis

Figure 2 shows the temperature entropy diagram corresponding to the flow diagram shown in Figure 1. Here T_5 and T_7 represent, respectively, the absolute temperature in the evaporation and condensation sides of the evaporator. Due to the presence of dissolved salts in the brine, the vapor pressure in the evaporator p_e is lower than the vapor pressure p_{se} of pure water corresponding to T_5. Because the vapor generated is in equilibrium with the brine at T_4, it is slightly superheated by an amount α the boiling point elevation. Friction losses in the piping from state point 4 to state point a cause a pressure drop p_e - p_i with a very slight decrease of temperature due to isenthalpic flow of the vapor from state point 4 to a.

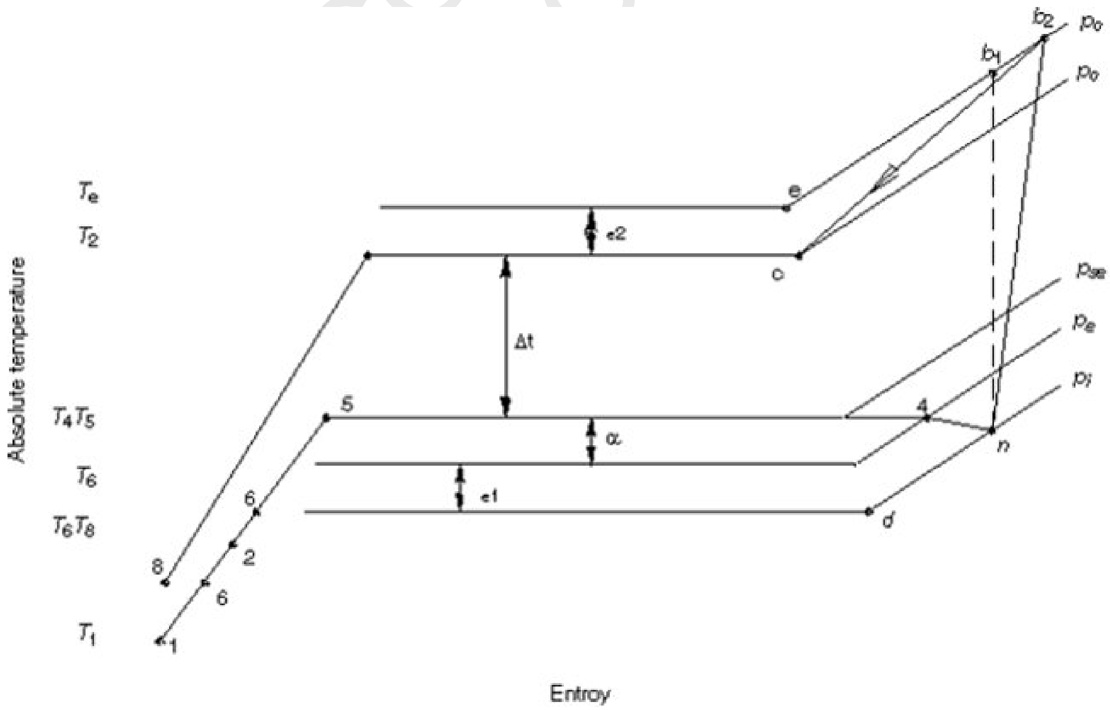

Figure 2. Temperature-Entropy diagram for the single-effect, multieffect vapor compression distillation system.

In the compressor, the vapor is compressed adiabatically from state point a at the inlet pressure p_i to state point b at the outlet pressure p_0 and then enters the condensation side of the evaporator where it condenses on heat transfer surface A at pressure p_c (slightly lower than p_0, due to friction losses in the outlet piping). The condensate is taken out at state point 7 to be cooled to state point 8 in the three-stream heat exchanger.

The total pressure rise, $p_0 - p_i$, supplied by the compressor consists of the following segments.

(a) $p_e - p_i$ pressure rise to overcome ε_1, the friction losses in inlet piping.
(b) $p_{se} - p_e$ pressure rise to overcome the boiling point elevation.
(c) $p_c - p_{se}$ pressure rise to overcome Δt.
(d) $p_0 - p_c$ pressure rise to overcome ε_2, the friction losses in outlet piping.

The energy imparted to the vapor during compression from a to b is given by the difference of enthalpies between state points b and a. For a well-insulated compressor, the specific energy input to the vapor w_c by the compressor is

$$w_c = h_b - h_a = \int T\mathrm{d}s + \int v\mathrm{d}p \tag{1}$$

where $T\mathrm{d}s$ and $v\mathrm{d}p$ are integrated from state point a to state point b. For an ideal compressor, the pressure rise across the compressor would be represented by the isentropic line ab_1. The isentropic work ($\mathrm{d}s = 0$) is then given by

$$w_{ci} = h_{b1} - h_a = \int v\mathrm{d}p \tag{2}$$

where $v\mathrm{d}p$ is integrated from state point a to state point b_1. For most applications using vapor compression distillation in the temperature range of 100°C or lower, water vapor can be assumed to behave as an ideal gas. Using this assumption, the integral $\int v\mathrm{d}p$ can be evaluated using the relation pv^κ = constant. This assumption results in

$$w_{ci} = \kappa R T_a \left[(p_o / p_i)^{\frac{(\kappa-1)}{\kappa}} - 1 \right] \Big/ (\kappa - 1) \tag{3}$$

The Clapeyron equation relates the absolute pressure and absolute temperature along the saturation line by the following equation:

$$\mathrm{d}p / p = \lambda_f \mathrm{d}T / RT^2 \tag{4}$$

Assuming λ_f to be constant in this range and integrating the equation from state point d to state point e (Figure 2) results in

$$\ln(p_o / p_i) = \lambda_f (T_e - T_3) / RT_e T_3 \tag{5}$$

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as $T_e = T_4 + \Delta t + \varepsilon_2$ and $T_3 = T_4 - \alpha - \varepsilon_1$ and as Δt, ε_1, ε_2, and α are small in comparison to T_4, one can make the following approximation:

$$T_e T_3 = (T_4 + \Delta t + \varepsilon_2)(T_4 - \alpha - \varepsilon_1) \cong T_4^2$$

and as $T_a \cong T_4$, Eq. (5) can be written as

$$\ln(p_o / p_i) \cong \lambda_f (\varepsilon_2 + \Delta t + \alpha + \varepsilon_1) / RT_4^2 \tag{6}$$

Substituting Eq. (6) into Eq. (3) and observing that $\kappa R/(\kappa\text{-}1) = c_p$ for ideal gas we obtain

$$w_{ci} = c_p T_4 [\exp\{\lambda_f (\Delta t + \alpha + \varepsilon_1 + \varepsilon_2) / c_p T_4^2\} - 1] \tag{7}$$

Setting $B = \lambda_f (\Delta t + \alpha + \varepsilon_1 + \varepsilon_2)/c_p T_4^2$ and expanding exp B into infinite series gives

$$w_{ci} = c_p T_4 \left[B + \frac{B^2}{2} + \frac{B^3}{6} + \ldots \right]$$

For very small values of $B \le 0.04$ one can neglect all higher order terms with accuracy exceeding 98 per cent and Eq. (7) reduces to the simplified equation:

$$w_{ci} = \lambda_f (\Delta t + \alpha + \varepsilon_1 + \varepsilon_2) / T_4 \tag{8}$$

Assuming further that friction losses are negligible, Eq. (8) is then simplified to

$$w_{ci} = \lambda_f (\Delta t + \alpha) / T_4 \tag{9}$$

In actual systems the compressor is not ideal and is usually driven by an electric motor. The electric energy input to the electric motor per unit mass of vapor is given by

$$w_m = w_{ci} / \eta_c \eta_m \tag{10}$$

where η_c is the isentropic compressor efficiency and η_m is the electric motor efficiency.

Equation (10) shows that the electric energy input to the motor driving the compressor depends on the compressor and motor efficiencies. In large systems using centrifugal compressors, the compressor efficiency can be as high as 85 per cent and motor efficiency can be as high as 95 per cent resulting in a combined efficiency of about 80 per cent. However, in very small systems (10-100 kg h^{-1}), the electric motor is usually larger than needed to drive the compressor for operation at steady state. In such situations the motor and compressor efficiencies are much lower resulting in a combined efficiency that can range from 20 to 50 per cent.

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3. Effects of Salinity and Temperature on Energy Consumption

Examination of Eq. (9) in Section 2 shows that the work required to compress the vapor consists of two components: $\lambda_f\ \Delta t/T4$ and $\lambda_f\ \alpha/T_4$. The properties of water and saline waters show that as evaporator temperature increases, the latent heat of evaporation decreases, resulting in a lower value of $\lambda_f\ /T_4$; however, the value of the boiling point elevation α increases and tends to balance the decrease of the ratio $\lambda_f\ /T_4$.

Table 1 shows the values of w_{ci} calculated as a function of t_4 for Δt = 0, 1, 2, and 4°C assuming a seawater brine salinity of 70 000 p.p.m. The values of α were taken from Fabuss (1980) and the values of λ_f were taken from steam tables (Keyes et al. 1969). It can be seen here that at Δt= 0 (infinite heat transfer surface), w_{ci} increases with increasing t_4. This increase continues until Δt is equal to α, after which the effect of Δt overcomes the effect of α, resulting in the decrease of w_{ci} as t_4 increases.

t_4 (°C)	**40.00**	**60.00**	**80.00**	**100.00**
α (°C)	0.780	0.913	1.047	1.181
λ_f/T_4	7.682	7.076	6.535	6.046
w_{ci}, Δt = 0	5.99	6.46	6.84	7.14
w_{ci}, Δt = 1	13.68	13.54	13.38	13.19
w_{ci}, Δt = 2	21.35	20.61	19.91	19.23
w_{ci}, Δt = 4	36.72	34.76	32.98	31.32

Table 1. Isentropic compression work as a function of evaporator temperature with Δt as a parameter and brine at 70 000 p.p.m.

For example, at t_4 = 100°C and Δt = 4°C, the value of w_{ci} = 31.3 kJ kg^{-1} while at t_4 = 50°C and Δt = 4°C the value of w_{ci} = 35.8 kJ kg^{-1} an increase of 4.5 kJ kg^{-1} or about 12.3 per cent. This suggests that, as long as $\Delta t > \alpha$, vapor compression systems should be operated at as high temperature as possible to reduce energy consumption. However, this advantage must be balanced against the practical disadvantages of larger heat losses from the system with the corresponding need to add auxiliary heat and increasing corrosion and scaling potentials in the evaporator.

At inland locations where brine disposal may be difficult and expensive, the volume of the brine must be reduced to a minimum requiring the brine to be discharged at as high a salinity as possible. As an example, in a 189 m^3 day^{-1} (50 000 gal day^{-1}) vapor compression system (Bechtel National Inc. 1986) used to recover water from agricultural drainage water (about 10 000 p.p.m.) in the San Joaquin Valley, California, the brine was discharged at about 200 000 p.p.m. Data from this plant operating at about 50°C with Δt = 4°C showed that the system can operate without auxiliary heat. The electrical energy consumption by the compressor motor was about 91.3 kJ kg^{-1} (25.4 kWh m^{-3}) resulting in a compressor motor combined efficiency of about 59 per cent.

Table 2 shows the calculated values of w_{ci} for the above application as a function of t_4

for Δt = 0, 1, 2, and 4°C. They show the effect of increased brine salinity on energy consumption.

In Table 2 the brine was assumed to have seawater composition with brine discharged at 200 000 p.p.m. It should be noted here that as long as $\Delta t < \alpha$, the value of w_{ci} increased with increasing operating temperature. It is interesting to note that, when Δt = 4°C, the energy consumption decreased with increasing temperature. Operating the plant at 100°C and Δt = 4°C results in energy savings of 4.1 per cent in comparison to operating the plant at 60°C and Δt = 4°C.

t_4 (°C)	**40**	**60**	**80**	**100**
α (°C)	2.948	3.412	3.877	4.334
w_{ci}, $\Delta t = 0$	22.64	24.14	25.34	26.20
w_{ci}, $\Delta t = 1$	30.33	31.22	31.87	32.25
w_{ci}, $\Delta t = 2$	38.01	38.30	38.41	38.30
w_{ci}, $\Delta t = 4$	53.76	52.45	51.48	50.39

Table 2. Isentropic compression work as a function of evaporator temperature with Δt as a parameter and brine at 200 000 p.p.m.

Comparison of the values of w_{ci} from Tables 1 and 2 for t_4 = 100°C and Δt = 2°C shows that the increase in the brine salinity from 70 000 to 200 000 p.p.m. has doubled the energy required by the compressor due to the large increase in the value of α. This large increase can be reduced significantly by the use of multiple-effect vapor compression distillation instead of single-effect vapor compression distillation as will be shown later in Section 8.

4. System Heat Balance

The heat balance in the system shown in Figure 1 per unit of distillate is given by

$$q_a + w_c = (h_8 - h_1) + (h_6 - h_1)x + q_e + q_{ex} \tag{11}$$

where q_a is the auxiliary heat added to the system.

Rearranging terms and assuming that $t_8 \cong t_6$ and that the specific heat of distillate and brine are equal, Eq. (11) becomes

$$q_a + w_c \cong c_w (1 + x)(t_8 - t_1) + q_e + q_{ex} \tag{12}$$

It can be seen here that if w_c is greater than or equal to the right hand side of Eq. (12), no auxiliary heat is needed and the auxiliary heater can be eliminated from the system. However, if w_c is less than the right hand side of Eq. (12), then auxiliary heat is needed to keep the system operating at steady-state. Obviously, the operating temperature in the evaporator has an important influence on heat losses q_e and q_{ex}; the higher the operating temperature t_5 the larger the heat losses.

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Assuming that the evaporator and heat exchanger are well insulated, i.e. q_e and q_{ex} are negligible, Eq. (12) is further simplified to

$$q_a + w_c = c_w(1+x)(t_8 - t_1) \quad (13)$$

Therefore, in order to eliminate the need for an auxiliary heater, the heat exchanger must be sized so that $w_c \geq c_w\ (1 + x)(t_8 - t_1)$. This is important because the majority of mechanical vapor compression systems are powered by electrical energy and auxiliary heat is usually supplied by electric immersion heaters. The elimination of the auxiliary heater will result in lower capital and operating costs. In reality, heat losses are never negligible. They are a function of the temperature difference between the evaporator and ambient temperatures; the higher this temperature difference, the larger the heat losses. The choice between installing a larger three-stream exchanger or adding an auxiliary electric immersion heater is usually a trade-off between the initial capital cost and electric energy cost. Obviously, if waste heat is available at an appropriate temperature, the electric heater can be replaced by the waste heat and the three-steam heat exchanger can be made smaller to reduce the capital cost. However, another heat exchanger must be added to the system to transfer the waste heat to the feed stream.

As an example, assume a well-insulated vapor compression distillation system operating at the following conditions: seawater feed, t_5 = 100°C, t_1 = 25°C, recovery 50 per cent, and w_c = 20 kWh m^{-3}. Estimate the highest allowable value of outlet brine and condensate temperatures to eliminate the auxiliary heater.

At these conditions c_w = 1.16 Wh kg^{-1} °C^{-1} and x = 1. Substituting these values in Eq. (13) gives t_8 = 33.6°C. This is the highest allowable value for outlet condensate and brine temperatures to eliminate the need for auxiliary heat at these ideal conditions.

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