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Effect of seawater-coolant feed arrangement in a waste heat driven multi-stage vacuum membrane distillation system

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Abstract

This paper proposes two types of seawater-coolant feed arrangements of a heat recovery unit (HRU) for improving the performance of a multi-stage vacuum membrane distillation (VMD) system: backward feed (BF) and parallel feed (PF). Theoretical studies were performed to assess the effect of the BF and PF feed arrangements on the system performance. In addition, to comprehensively understand the thermochemical phenomena in both the BF and PF arrangements, spatial variations in the temperature, permeate pressure, permeate flux, and salinity were investigated using a rigorous simulation model that considered the heat and mass transfer across the hollow fibers coupled with the transport behavior on the feed side. To determine the superior HRU configuration between BF and PF, the water production, recovery ratio, and specific energy consumption of the multi-stage VMD system were investigated. It was found that the total water production in the PF arrangement was approximately $2.94 \text{ m}^3/\text{d}$, which was approximately 6% higher than in the BF arrangement; however, the BF arrangement was more efficient for the production of freshwater than the PF arrangement when a smaller number of module stages was employed. Furthermore, the optimum number of HRUs in the BF arrangement was determined based on this theoretical study.

Keyword: vacuum membrane distillation; multi-stage concept; coolant feed arrangement; heat and mass transfer.

1. Introduction

Membrane distillation (MD) is an emerging desalination technology based on thermally driven separation using a hydrophobic microporous membrane [1–5]. Because the hydrophobic membrane surface completely repels water droplets, only water vapor is transported across the membrane pores. The driving force of the MD process is the difference in water vapor pressure resulting from the temperature gradient between the liquid-vapor interfaces of the feed and permeate sides. The advantages of the MD process include a 99.99% rejection rate for non-volatile species in seawater, a lower operating temperature compared to mature evaporation-based desalination processes (e.g., multi-stage flash and multi-effect distillation), possible atmospheric pressure drive, and low sensitivity of system performance to NaCl concentration [1–3,6–9]. As such, MD has attracted attention as a next-generation seawater desalination process. There are three typical MD configurations, classified based on the condensation method of water vapor permeating through the membrane: direct contact MD (DCMD), air gap MD (AGMD), and vacuum MD (VMD). Of these configurations, the VMD process can achieve the highest system efficiency owing to its negligible conductive heat loss and high water production. The VMD process generates a water vapor pressure difference between the feed and permeate sides by applying a vacuum pressure rather than by feeding a cold permeate to the permeate side. Consequently, conductive heat loss and temperature polarization effects are minimized in the VMD process [4]. Furthermore, the application of a vacuum on the permeate side removes the air from the pores of the hydrophobic membrane, resulting in negligible molecular diffusion and a uniform vacuum level on the permeate side [4]. Therefore, Knudsen diffusion dominates the mass transfer through the membrane pores [2,6,7,10–12].

To achieve high energy efficiency in a multi-stage VMD (MVMD) system, the main challenge is determining the appropriate process design, including the heat recovery scheme. Heat recovery is a common approach for enhancing the thermal efficiency in thermally driven processes [13–17]. Previous research on MD process design has focused on a multi-stage system including a heat recovery scheme to enhance water production and reduce specific energy consumption [2,12,18,19]. Kim et al. [2] investigated the performance of a solar-assisted MVMD system and evaluated the water production and specific thermal energy consumption (STEC). To improve system performance, a heat recovery unit (HRU) was integrated into the MVMD system. In addition, Memsys successfully commercialized a vacuum multi-effect membrane distillation (V-MEMD) system featuring internal heat recovery at each stage of the flat-sheet VMD module, and tested it under real-world conditions [13]. The results showed a great

potential to experimentally increase the gain output ratio (GOR) using the V-MEMD system. Chung et al. [19] also applied a HRU to improve the system performance of a MVMD system.

These previous studies investigated the enhancement of system performance using a heat recovery scheme, and also demonstrated the feasibility of increasing the thermal efficiency. This was possible because the recovered thermal energy was recycled to heat the incoming seawater or successive feed brine (next module stage). Furthermore, by designing an appropriate arrangement of HRUs for the MVMD system, the system performance could be further enhanced [20,21]. However, experimental and simulation studies concerning the effect of the HRU arrangement on MVMD system performance are lacking.

In this study, therefore, two different seawater-coolant feed arrangements are suggested to improve the performance of a MVMD system with HRUs: backward feed (BF) and parallel feed (PF). A theoretical performance evaluation was conducted for both arrangements using a rigorous simulation model which takes into account the heat and mass transfer through the membrane coupled with the transport behavior in the bulk feed phase. To achieve a better understanding of the MVMD system performance with BF and PF arrangements, spatial variations in the major operating variables (e.g., temperature, pressure, feed salinity, and permeate flux) of the MVMD system were comprehensively investigated. The system performance with both HRU arrangements was analyzed in terms of water production, recovery rate, and specific energy consumption. In addition, the effect of the number of HRUs on the system performance was explored for the BF arrangement.

2. Theoretical approach

2.1. MVMD desalination system

Figure 1 shows a schematic diagram of a waste heat driven MVMD system with two different HRU arrangements (BF in Fig. 1a and PF in Fig. 1b). The MVMD desalination system with different HRU arrangements consists of VMD modules, HRUs, a circulation pump, and a vacuum pump. The HRUs can be configured to connect to a series of modules in the MVMD system.

As shown in Fig. 1(a), in the BF arrangement, incoming seawater is continuously heated by recovering the latent heat of condensation of the permeate vapor using a series of HRUs in the MVMD system. Waste heat is then applied to attain the desired feed temperature from the seawater discharged via the HRUs. The heated incoming seawater flows into the first stage module as the inlet feed solution for the MVMD system, and then continuously flows into

successive stage modules. Meanwhile, with an increasing number of module stages having a countercurrent-flow configuration between the seawater-coolant and seawater-feed streams, the brine temperature decreases as a result of the latent heat of vaporization induced by the permeate flux through the hollow fiber. On the other hand, the seawater-coolant temperature increases as the coolant flows from the final HRU to the first HRU through recovery of the latent heat of condensation of the permeate vapor, resulting in an increase in permeate pressure. The brine that is cooled and concentrated at each module stage is discharged from the last module of the MVMD system, and the distillate is collected from each HRU stage.

Fig. 1. Schematic diagram of the two different HRU arrangements in the MVMD system: (a) BF and (b) PF.

In a MVMD desalination system with a PF-type seawater-coolant feed arrangement, latent heat recovery from permeate-vapor condensation occurs only at the first HRU stage, which is in communication with the front VMD modules having relatively high permeate fluxes, as shown in Fig. 1(b). The seawater-coolant supplied to each HRU stage (except the first HRU stage) is used only for condensation of the permeate vapor and exits as is. As a result, the permeate pressure in successive modules is highly dependent on the temperature of the incoming seawater-coolant. The influent seawater heated by the first HRU is reheated using waste heat and then flows into the MVMD system.

To compare the system performance with different seawater-coolant feed arrangements, the following fixed operating conditions were applied at the inlet conditions: (i) an inlet feed temperature of 80 °C, (ii) an inlet feed velocity of 1.0 m/s, and (iii) an inlet feed salinity of 45 g/kg. However, the vacuum pressure at the permeate side of the VMD module is determined by the saturation temperature (i.e., the saturation pressure) of the HRU in the MVMD system proposed in this study. The same hydrophobic hollow-fiber membrane module was used in all module stages. Hot feed brine flows into the shell side of the module, and vacuum pressure is applied to the lumen side of the hollow fiber, thereby transferring heat and mass through the membrane.

2.2. Vacuum membrane distillation (VMD) model

Figure 2 shows a schematic of the VMD module (Fig. 2a) and a VMD process diagram and scanning electron microscopy (SEM) image of the hydrophobic hollow-fiber membrane (Fig. 2b). A commercially available polypropylene hollow-fiber membrane (MD020CP2N, Microdyn-Nadir GmbH) was used; detailed module specifications are summarized in Table 1.

Fig. 2. Schematic representation of (a) the shell-and-tube type VMD module, and (b) heat and mass transfer through the polypropylene hollow-fiber membrane in the VMD process.

Table 1 Detailed specifications of the hollow-fiber module used in this study.

MD020CP2N (polypropylene)	
Length of fiber, L_f (m)	0.5
Inner diameter of shell, d_s (m)	0.03
Number of fibers, N_f (-)	100
Inner diameter of fiber, d_i (mm)	1.8
Outer diameter of fiber, d_o (mm)	2.6
Thickness of fiber, δ_m (mm)	0.4
Membrane porosity, ε (%)	70
Mean pore size, d_p (μm)	0.2
Liquid entry pressure of water (kPa)	140

As shown in the schematic diagram of the VMD process (Fig. 2), water vapor is generated by the difference in the partial pressure of water vapor between the feed and permeate (vacuum) sides resulting from heat and mass transfer across the boundary layers near the membrane surface and through the membrane coupled with the mass transfer of water vapor through the microporous membrane.

Thermal energy in the form of a heat flux (Q) is transferred across the boundary of the feed side (Q_f) and through the membrane (Q_m). The convective heat transfer across the feed boundary layer is expressed as follows:

$$Q_f = h_f A_f (T_f - T_{f,m}) \quad (1)$$

where $A_f = \pi N_f d_o$ is the surface area of thermal energy transport across the membrane; N_f is the number of hollow fibers; d_o is the outer diameter of the hollow fiber; and T_f and $T_{f,m}$ are the bulk feed temperature and liquid-vapor interface temperature on the feed side, respectively.

As shown in Fig. 2, the seawater feed inlet is oriented perpendicular to the hollow-fiber array, and the feed flows along the shell side of the module. As a result, cross and parallel flows exist concurrently on the shell side. The convective heat transfer coefficient (h_f) at the shell side is determined by Groehn's correlation [22], which considers a yaw angle (α) as varying from 0° for cross flow to 90° for parallel flow, as follows:

$$Nu_f = \frac{h_f d_h}{k_f} = 0.206 (Re \cos \alpha)^{0.63} Pr^{0.36} \quad (2)$$

where k_f is the thermal conductivity of the bulk feed; and $d_h = d_o (1 - \phi) / \phi$ is the hydraulic diameter of the shell, which is a function of the module packing density, $\phi = N_f (d_o / d_s)^2$, where d_s is the inner diameter of the shell [23].

Owing to the negligible conductive heat transfer across the membrane in the VMD process [10,24], the heat transfer across the membrane, Q_m , can be described only by the latent heat of vaporization. The heat transfer through the membrane is thus expressed as follows:

$$Q_m = A_m J \Delta H \quad (3)$$

where $A_m = \pi N_f d_{lm}$ and $d_{lm} = (d_o - d_i) / \ln(d_o / d_i)$; ΔH is the enthalpy of evaporation at the average temperature of the membrane, T_m .

The overall heat transfer across the membrane at steady state is described as follows:

$$Q_f = Q_m \quad (4)$$

The water vapor permeate flux (J) is a function of the mass transfer coefficient (i.e., the membrane distillation coefficient, C_{mem}), with the partial pressure gradient (i.e., the transmembrane pressure, ΔP) acting as a driving force, as follows:

$$J = C_{mem} (P_{f,m} - P_r) \quad (5)$$

where $P_{f,m}$ is the water vapor pressure at the liquid-vapor interface on the feed side of the membrane, and P_r is the pressure at the permeate side (vacuum side), i.e., the pressure on one side of the HRU. The water vapor pressure of the mixture varies depending on the molar concentration (x_f) of dissolved species on the feed side. Thus, the water vapor pressure is calculated using Raoult's law as follows:

$$P_{f,m} = (1 - x_f) P_{w,m} \quad (6)$$

where $P_{w,m}$ is the vapor pressure of pure water, which is determined using the Antoine equation [11].

In the VMD process, the Knudsen diffusion mechanism dominates the mass transfer through membrane pores [25–27]. The membrane distillation coefficient for the VMD process is thus calculated as follows:

$$C = \frac{8}{3} \frac{\varepsilon d_p}{\tau \delta} \left(\frac{M}{2\pi RT} \right)^{1/2} \quad (7)$$

where ε , d_p , τ , and δ are structural parameters of the hydrophobic microporous membrane, and correspond to the porosity, mean pore size, pore tortuosity, and membrane thickness, respectively; M is the molecular weight of water; and R is the ideal gas constant.

To improve the prediction accuracy of the model, the transport behavior (e.g., the mass, species, energy, and momentum balances) coupled to the heat and mass transfer is also taken into account in the bulk feed phase. Several assumptions are made in describing the transport behavior in the flow channel: (i) the fibers are distributed uniformly in the shell side; (ii) there is constant pressure in the permeate side (vacuum side); and (iii) negligible heat loss to the environment.

Shell-side spatial variations in the velocity (v_f), concentration (x_f), temperature (T_f), and pressure (P_f) can be estimated using mass, species, energy, and momentum balance equations as follows [1]:

$$\frac{1}{X_f} \frac{dv_f}{dz} - \frac{v_f}{X_f^2} \left(\frac{M_s}{\rho_s} - \frac{M_w}{\rho_w} \right) \frac{dx_f}{dz} = - \frac{2N_f r_o J}{M_f (r_s^2 - N_f r_o^2)} \quad (8)$$

$$\frac{x_f}{X_f} \frac{dv_f}{dz} + \frac{v_f M_w}{X_f^2 \rho_w} \frac{dx_f}{dz} = 0 \quad (9)$$

$$\frac{d\rho_f v_f c_{p,f} T_f}{dz} = - \frac{Q_f}{\pi (r_s^2 - N_f r_o^2)} \quad (10)$$

$$\frac{dP_f}{dz} = - \frac{8\mu_f v_f}{r_h^2} \quad (11)$$

where X is the molar volume; ρ is the density; r_o and r_s are the radius of outer fiber and inner shell, respectively; μ is

the dynamic viscosity; and c_p is the specific heat capacity. The subscripts s , w , and f indicate the salt, water, and feed, respectively.

For the simulations, the following boundary conditions are applied to the velocity, concentration, pressure, and temperature at the feed side:

$$v_f(0) = v_{f,in}, \quad x_f(0) = x_{f,in}, \quad P_f(L_f) = P_0, \quad T_f(0) = T_{f,in} \quad (12)$$

where L_f is the fiber length, P_0 is the atmospheric pressure, and the subscript in indicates the module inlet.

The mean transmembrane flux (J_m) and distillate flow rate ($\dot{m}_p$) of the VMD module are estimated as follows:

$$J_m = \frac{1}{L_f} \int_0^{L_f} J dz \quad (13)$$

$$\dot{m}_p = J_m N_f \pi d_o L_f \quad (14)$$

where J is the local permeate flux given in Eq. (5).

The set of coupled ordinary differential equations in Eqs. (8)–(11) describing the mass, species, energy, and momentum balances of the shell-side feed flow were discretized based on the finite volume method. They were then solved by coupling the heat and mass transfer given in Eqs. (1), (3), and (4) and Eqs. (5)–(7), respectively, with the boundary conditions given in Eq. (12). The numerical analysis was conducted using Broyden's method [1], which is a quasi-Newtonian method for solving a system of nonlinear equations. The thermochemical properties of water and seawater were adopted from Ref. [28].

2.3. Heat recovery unit (HRU) model

A shell-and-tube type heat exchanger was employed as a heat recovery unit (HRU) to recover the latent heat of permeate water vapor from the MVMD system to be applied to the seawater as a feed solution. The water vapor produced at the lumen side of the hollow fiber in the VMD module flowed into the shell side of the HRU, in which water vapor was assumed to be completely condensed. Based on this assumption, the energy balance of j th HRU communicating with the VMD modules is given by [29]:

$$\sum_{i=1+k}^{M \times j} \dot{m}_{p,i} [h_v(T_{p,i}) - h_w(T_{r,j})] + \dot{m}_s [h_f(T_{s,in,j}) - h_f(T_{s,out,j})] = 0 \quad (15)$$

where $k = M(j-1)$ and $M = N_s / N_r$, in which N_s and N_r are the total number of module stages and HRUs, respectively; i and j are the number of module stages and HRUs, respectively; h_v , h_w , and h_f are the enthalpies of the

permeate vapor, distillate, and seawater coolant, respectively; T_p , T_r and T_s are the temperatures of the permeate vapor, HRU, and seawater coolant, respectively; and $\dot{m}_s$ is the mass flow rate of seawater coolant.

In Eq. (15), the outlet temperature ($T_{s,out}$) of the seawater from the j th HRU is determined using the log-mean temperature difference (LMTD) method [29,30]:

$$T_{s,out,j} = T_{r,j} + (T_{s,in,j} - T_{r,j}) \exp \left[\frac{-U_{r,j} A_{r,j}}{\dot{m}_s c_{p,j} (T_{r,j})} \right] \quad (16)$$

where $A_{r,j}$ is the outer surface area of the tubes in the HRU; and U_r is the overall heat transfer coefficient, which is calculated using thermal circuit theory. The overall heat transfer coefficient consists of the convective and conductive heat transfer resistances as follows [29,30]:

$$U_r = \left(\frac{r_o}{r_i} \frac{1}{h_{i,j}} + \frac{r_o}{k} \ln \frac{r_o}{r_i} + \frac{1}{h_{o,j}} \right)^{-1} \quad (17)$$

where r_o and r_i are the outer and inner radius of the tube, respectively; h is the convective heat transfer coefficient; and k is the thermal conductivity of the tube.

2.4. Evaluation of specific energy consumption

A theoretical evaluation of the specific thermal and electrical energy consumption was carried out for the MVMD desalination system with HRU in the BF- and PF-type seawater-coolant feed arrangements. For simplicity, the pressure drop and heat loss due to plumbing fixtures (e.g., pipes, fittings, valves, etc.) were not taken into account in the simulation.

In this study, the thermal energy delivered by the waste heat to achieve the desired feed temperature from the seawater discharged through the HRUs was not taken into account in the estimation of specific thermal energy consumption (STEC, which is the ratio of the thermal energy consumption to the water production in kWh/m³), as the waste heat input was considered free. Thus, the specific thermal energy consumption at the i th VMD stage (STEC _{i}) can be estimated as follows:

$$STEC_i = \frac{Q_{f,i}}{\dot{m}_{p,i} / \rho_w (T_{r,j})} \quad (18)$$

with

$$Q_{ft,i} = \dot{m}_{f,in,i} h_f(T_{f,in,i}) - \dot{m}_{f,out,i} h_f(T_{f,out,i}) \quad (19)$$

where Q_{ft} is the thermal energy used in each VMD stage, and $\dot{m}_f$ is the mass flow rate of feed seawater.

The specific electrical energy consumption (SEEC) at the i th VMD stage linked to the HRU ($SEEC_i$) is determined by the following equation [2]:

$$SEEC_i = \frac{Q_{fe,i} + Q_{re,i}}{\dot{m}_{p,i} / \rho_w(T_{r,j})} \quad (20)$$

with

$$Q_{fe,i} = \frac{[P_{f,i}(0) - P_{f,i}(L_f)] V_{f,i}}{\eta} \quad (21)$$

$$Q_{re,i} = \frac{\Delta P_{r,j} V_s}{\eta} \zeta_i \quad (22)$$

where Q_{fe} and Q_{re} are the electrical energy consumption at each VMD stage and the HRU, respectively; V_f and V_s are the volume flow rate of the feed seawater in each VMD stage and the seawater coolant in the HRU, respectively; and η is the pump efficiency (i.e., 0.65). In Eq. (22), ΔP_r is the pressure drop through the tube side of the shell-and-tube HRU, which is calculated by considering the tube friction and internal loss due to return bends [2] as follows:

$$\Delta P_r = \left[\frac{4 f_t L_t N_p}{2 r_i} + 4 N_p \right] \frac{G_t^2}{2 \rho} \quad (23)$$

where N_p is the number of tube passes, L_t is the tube length, G_t is the mass flux velocity through the tube, and $f_t = (0.79 \ln Re - 1.64)^{-2}$. In Eq. (22), ζ_i is the ratio of the latent heat of condensation of permeate vapor transported from each VMD module to the heat recovered by the j th HRU linked to the VMD modules, expressed as follows:

$$\zeta_i = \frac{\dot{m}_{p,i} [h_v(T_{p,i}) - h_w(T_{r,j})]}{\dot{m}_s [h_f(T_{s,in,j}) - h_f(T_{s,out,j})]} \quad (24)$$

In addition, the overall STEC (OSTEC) and overall SEEC (OSEEC) of the MVMD desalination system with HRUs in BF and PF seawater-coolant feed arrangements can be determined as follows:

$$OSTEC = \frac{\sum_1^{N_s} Q_{ft,i}}{\sum_1^{N_s} [\dot{m}_{p,i} / \rho_w(T_{r,j})]}, \quad OSEEC = \frac{\sum_1^{N_s} (Q_{fe,i} + Q_{re,i})}{\sum_1^{N_s} [\dot{m}_{p,i} / \rho_w(T_{r,j})]} \quad (25)$$

3. Results and discussion

3.1. Performance comparison of BF and PF arrangements

To evaluate the performance of a MVMD desalination system with different seawater-coolant feed arrangements (i.e., BF and PF), modules with the specifications listed in Table 1 were used in a 24-stage VMD desalination system. This system consisted of a module at each stage and eight HRUs, such that three VMD modules in series were in communication with one HRU. The feed temperature and velocity entering the first stage were maintained at 80 °C and 1.0 m/s, respectively, and the raw seawater salinity was 4.5 wt.%.

Fig. 3. Axial variations in the bulk feed temperature (T_f) and liquid-vapor interface temperature at the feed side ($T_{f,m}$) for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

Figure 3 shows the profiles for the bulk feed temperature (T_f) and liquid-vapor interface temperature on the feed side ($T_{f,m}$) along the module stages of the MVMD system (i.e., along a normalized axial length) with BF and PF arrangements. The dimensionless axial length was defined as $[L_f(i-1) + z] / L_f$, where L_f is the hollow-fiber length at each MVMD stage, i is the number of module stages in the range of $0 - N_s$, and z is the local axial coordinate in the range of $0 - L_f$. For a feed inlet temperature of 80 °C at the first stage, the outlet temperature of the brine from the last stage was approximately 44 °C with the BF arrangement and approximately 41 °C with the PF arrangement. This qualitative trend was also observed for $T_{f,m}$, and was attributed to latent heat loss due to vaporization along the

module stages. A significant decrease in both T_f and $T_{f,m}$ along the stages was observed in the PF arrangement owing to the latent heat loss due to vaporization resulting from a higher vapor permeation through the hollow fiber compared to the BF arrangement. Meanwhile, the rate of decrease of T_f and $T_{f,m}$ in the PF arrangement was similar to that in the BF arrangement after the ninth module stage. This indicates that more vapor permeation occurred in the front stages of the PF arrangement than the BF arrangement, signifying that the driving forces of the BF and PF arrangements were more similar in the rear stages of the MVMD system.

Fig. 4. Comparison of HRU temperature (T_r) and seawater-coolant outlet temperature from the HRU (T_s) for different seawater-coolant feed arrangement (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

Profiles of the HRU vapor temperature (T_r) and seawater-coolant outlet temperature from the HRU (T_s) at the eight HRUs are shown in Fig. 4. For the BF arrangement, the values of T_r and T_s obtained from Eqs. (15) and (16), respectively, increased significantly from 34 °C to 62 °C and from 31 °C to 58 °C, respectively, along the direction of the seawater-coolant flow from the eighth to the first HRU. Meanwhile, the values of T_r and T_s for the PF arrangement increased from 34 °C to 42 °C and from 31 °C to 36 °C, respectively. As described above, for the MVMD system with a countercurrent-flow configuration between the seawater supplied to the VMD module and the seawater-coolant supplied to the HRU, the significant increase in T_r and T_s along the coolant-flow direction in the BF arrangement was achieved by sequentially recovering the latent heat of condensation of the water vapor

discharged from the VMD module connected to the HRU. In contrast, a slight increase in T_r and T_s was observed in the PF arrangement, as the seawater coolant was discharged after removing the latent heat of permeate vapor in all HRUs except the first.

Fig. 5. Axial variations in the water vapor pressure at the feed side of the membrane ($P_{f,m}$) and the HRU pressure (i.e., permeate-side pressure, P_r) for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

Variations in the water vapor pressure at the feed side of membrane ($P_{f,m}$) and the HRU pressure (i.e., permeate-side pressure, P_r) along the stages for both the BF and PF arrangements are shown in Fig. 5. The results show that the transmembrane pressure ($P_{f,m} - P_r$) decreased from 19.04 kPa at the inlet to 3.39 kPa at the outlet with the BF arrangement, whereas it decreased from 29.49 kPa to 2.06 kPa with the PF arrangement. The permeate-side pressure with the PF arrangement increased slightly from 5.42 kPa at the eighth HRU to 8.02 kPa at the first HRU, while it increased significantly with the BF arrangement, from 5.16 kPa at the eighth HRU to 21.52 kPa at the first HRU. As expected, this significant increase with the BF arrangement was achieved by continuously recovering the latent heat of condensation of the permeate vapor as the seawater-coolant was fed to the eighth HRU and discharged from the first HRU.

Fig. 6. Permeate flux and feed salinity profiles along the module stages for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80\text{ }^{\circ}\text{C}$, $v_{f,in} = 1\text{ m/s}$, $s_{in} = 45\text{ g/kg}$).

Variations in the local permeate flux and brine salinity along the module stages with the different coolant feed arrangements are shown in Fig. 6. Because the thermal energy was consumed primarily by the evaporation of water vapor, i.e., the driving force of the MD process decreased, the permeate flux decreased asymptotically along the module length. In addition, the local permeate flux decreased exponentially along the axial distance because the water vapor pressure at the feed side decreases exponentially with decreasing feed temperature according to the Antoine equation [11], which describes the relationship between vapor pressure and temperature for pure water. It was observed that the permeate fluxes along the module stages had a serrated appearance for both arrangements; this was more pronounced in the front module stages of the MVMD system and in the BF arrangement. In other words, the permeate flux decreased significantly along the fiber length of the module stages in communication with a HRU, followed by a sudden increase and significant decrease in the module stages connected to the subsequent HRU with a lower vapor temperature than the previous HRU. These phenomena were observed to occur consecutively along the module stages, and the magnitude gradually decreased. As previously mentioned, this can be attributed to the exponential decrease in vapor pressure, which resulted in a substantial decline in the permeate flux as the temperature decreased. In addition, the permeate flux in the PF arrangement was greater than in the BF arrangement until the ninth module stage, after which this was reversed as a result of thermal energy consumption from the higher vapor permeation in the front module stages of the MVMD system.

The brine salinities at the feed side increased along the dimensionless axial length for both BF and PF arrangements as a result of vapor permeation across the hydrophobic membrane during the MD process. An asymptotic increase in both salinities was also observed due to the decrease in driving force along the module stages. The outlet brine salinity was higher in the PF arrangement than the BF arrangement, which can be attributed to the higher water production of the former than the latter.

Figure 7 shows profiles of the cumulative water production (W_p) and water recovery (WR) along the dimensionless axial distance. The water recovery is defined as the ratio of distillate production (kg/h) to feed flow rate (kg/h). The W_p and WR in the PF arrangement were approximately 145.3 kg/h and 25.3%, respectively, over the MVMD system, which is approximately 6% higher than in the BF arrangement. Based on performance predictions for the BF and PF arrangements, it was demonstrated that different HRU arrangements in the MVMD system may lead to different system performance. Thus, the STECs for the proposed BF and PF arrangements of the MVMD system were also compared.

Fig. 7. Comparison of water production (W_p) and water recovery (WR) profiles along the module stages for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

Fig. 8. Comparison of STEC profiles along the module stages for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

The STEC profiles along the dimensionless axial distance are shown in Fig. 8 for the BF and PF arrangements. The results show that the STECs for both the BF and PF arrangements increased asymptotically along the module stages. The decrease in $T_{f,m}$ and T_r along the feed flow direction (Figs. 3 and 4) and hence the decrease in transmembrane pressure ($P_{f,m} - P_r$) (Fig. 5), i.e., the driving force of the VMD process, resulted in an asymptotic decrease in the vapor permeation (Fig. 6). Therefore, the STEC increased asymptotically according to the definition of STEC given in Eq. (18), i.e., the ratio of thermal energy consumption to water production. In addition, the STEC values were lower over the entire MVMD system with the BF arrangement than the PF arrangement. However, the difference in STEC between the BF and PF arrangements decreased gradually with increasing module stage. Therefore, it can be concluded that MVMD systems with the BF arrangement and a smaller number of stages can produce freshwater more efficiently than systems with the PF arrangement.

Overall, these results demonstrate that the multi-stage system could increase the total water production; however, use of an inappropriate number of module stages in the MVMD system can lead to reduced system performance in terms of the thermal energy efficiency.

3.2. Effect of the number of HRUs on performance of the MVMD system with BF arrangement

As demonstrated in Section 3.1, when a smaller number of module stages were used in the MVMD system, the BF arrangement could produce freshwater more efficiently than the PF arrangement in terms of energy consumption. Thus, in this section, the effect of the number of HRUs (N_r) on the system performance was investigated in terms of the specific energy consumption and water production to determine the optimum system design. As detailed in Section 2.3, the temperature and pressure at each HRU were determined by the thermohydrodynamic conditions (e.g., the temperature and mass flow rate) of the seawater-coolant supplied to the HRU and the water vapor discharged from the VMD modules in communication with the HRU, according to Eq. (15). Thus, a smaller number of installed HRUs could lead to a lower vacuum level in each VMD module than with a larger number of HRUs. This indicates that the driving force in each VMD module decreases with a decreasing number of HRUs. However, the larger the number of HRUs, the higher the capital and operating costs incurred, such as for purchasing HRUs, maintenance costs, and electrical operating costs to compensate for the higher pressure drop. For instance, the lower water production of later HRUs may lead to an inefficient increase in OSTEC.

Fig. 9. Effect of number of HRUs (N_r) on HWP and DWP with the BF arrangement (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

Figure 9 shows the influence of N_r on the hourly water production (HWP) and daily water production (DWP) with

the BF configuration. HWP and DWP both increased asymptotically by approximately 36%, from 1.02 m³/h to 1.39 m³/h and from 2.45 m³/d to 3.33 m³/d, respectively. However, the increase in HWP and DWP was negligible for four or more HRUs compared to lower N_r .

Fig. 10. Effect of N_r on OSTEC and OSEEC in the BF arrangement (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

Figure 10 shows the effect of N_r on the OSTEC and OSEEC for the BF arrangement. The OSTEC increased asymptotically from 224.39 kWh/m³ to 226.83 kWh/m³ with an increase in N_r from 1 to 10. This was because with increasing module stages, the permeate flux and cumulative water production asymptotically decreased and increased, respectively, as shown in Figs. 6 and 7, while the bulk feed temperature also decreased asymptotically, as shown in Fig. 3. Meanwhile, the OSEEC increased linearly from 0.65 kWh/m³ to 1.47 kWh/m³ to compensate for the pressure loss at high N_r .

In summary, despite the asymptotic increase in STEC within a small range with increasing N_r , the W_p was greatly enhanced, and the OSEEC also increased linearly. However, it was also observed that water production increased negligibly at N_r greater than 4, which indicates that an inappropriate N_r (> 4) may lead to lower energy efficiency and higher capital costs. As a result, based on a comprehensive analysis of the simulation results, four HRUs were found to be optimal in the BF arrangement.

5. Conclusion

In this study, two types of seawater-coolant feed arrangements of a heat recovery unit (HRU) were proposed to enhance the performance of a multi-stage vacuum membrane distillation system: backward feed (BF) and parallel feed (PF). Theoretical studies were conducted using a rigorous simulation model to estimate the effect of BF and PF arrangements on the system performance. Under given operating conditions, it was observed that the MVMD system with the BF arrangement was more efficient for the production of freshwater than the PF arrangement at a smaller number of module stages in terms of the specific thermal energy consumption. Furthermore, it was also found that an inappropriate number of module stages in the MVMD system had a negative effect on the system performance. Based on comprehensive data analysis of the results obtained from various BF arrangement scenarios, it was demonstrated that four HRU stages with the BF arrangement were the optimal number in the 24-stage VMD desalination system in terms of energy efficiency and cost.

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Nomenclature

A	area [m^2]
c_p	heat capacity [$\text{J/kg}\cdot\text{K}$]
C_{mem}	membrane distillation coefficient [$\text{kg/m}^2\cdot\text{s}\cdot\text{Pa}$]
d	diameter [m]
f	friction factor [-]
G_t	mass flux velocity in HRU [$\text{kg/m}^2\cdot\text{s}$]
h	convective heat transfer coefficient [$\text{W/m}^2\cdot\text{K}$]
h_f	enthalpy of seawater-coolant [J/kg]
h_v	enthalpy of permeate vapor [J/kg]

h_w	enthalpy of distillate [J/kg]
ΔH	latent heat of vaporization [J/kg]
J	permeate flux [$\text{kg}/\text{m}^2 \cdot \text{h}$]
k	thermal conductivity [$\text{W}/\text{m} \cdot \text{K}$]
L	length [m]
L_t	tube length in HRU [m]
$\dot{m}_f$	mass flow rate of feed seawater [kg/s]
$\dot{m}_p$	mass flow rate of distillate [kg/s]
$\dot{m}_s$	mass flow rate of seawater-coolant [kg/s]
M	molecular weight [kg/kmol]
N_f	number of hollow fibers [-]
N_p	number of tube passes [-]
N_r	number of HRUs [-]
N_s	number of module stages [-]
P	pressure [Pa]
ΔP_r	pressure drop through the tube side in HRU [Pa]
Q	heat transfer [W]
Q_{ft}	thermal energy consumption through each VMD stage [W]
Q_{fe}	electrical energy consumption through each VMD module [W]
Q_{re}	electrical energy consumption through each HRU [W]
R	gas constant [$\text{J}/\text{kmol} \cdot \text{K}$]
r_o	outer radius of fiber or tube in HRU [m]
r_i	inner radius of fiber or tube in HRU [m]
SEEC	specific electrical energy consumption [kWh/m^3]
STEC	specific thermal energy consumption [kWh/m^3]
T_f	temperature of feed seawater [$^{\circ}\text{C}$]
T_p	temperature of permeate vapor [$^{\circ}\text{C}$]
T_r	temperature of HRU [$^{\circ}\text{C}$]

T_s	temperature of seawater-coolant [$^{\circ}\text{C}$]
U_r	overall heat transfer coefficient in HRU [$\text{W}/\text{m}^2\cdot\text{K}$]
v	velocity [m/s]
V	volume [m^3], volume flow rate [l/min]
x_f	molar fraction in liquid phase [-]
X	molar volume [m^3/mol]
z	axial coordinate [m]

Dimensionless numbers

Nu	Nusselt number, hd_h/k [-]
Pr	Prandtl number, $\mu c_p/k$ [-]
Re	Reynolds number, $\rho v d_h/\mu$ [-]

Greek letters

α	yaw angle [$^{\circ}$]
δ	membrane thickness [m]
ε	membrane porosity [-]

ζ_i ratio of the latent heat of condensation of the permeate vapor transported from each VMD module to the heat recovered by the j th HRU linked to the VMD modules [-]

η	pump efficiency [-]
μ	dynamic viscosity [$\text{kg}/\text{m}\cdot\text{s}$]
ρ	liquid density [kg/m^3]
τ	pore tortuosity [-]
ϕ	packing density [-]

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Table captions

Table 1 Detailed specifications of the hollow-fiber module used in this study.

Figure captions

Fig. 1. Schematic diagram of the two different HRU arrangements in the MVMD system: (a) BF and (b) PF.

Fig. 2. Schematic representation of (a) the shell-and-tube type VMD module, and (b) heat and mass transfer through the polypropylene hollow-fiber membrane in the VMD process.

Fig. 3. Axial variations in the bulk feed temperature (T_f) and liquid-vapor interface temperature at the feed side ($T_{f,m}$) for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80^\circ\text{C}$, $v_{f,in} = 1\text{ m/s}$, $s_{in} = 45\text{ g/kg}$).

Fig. 4. Comparison of HRU temperature (T_r) and seawater-coolant outlet temperature from the HRU (T_s) for different seawater-coolant feed arrangement (feed inlet conditions: $T_{f,in} = 80^\circ\text{C}$, $v_{f,in} = 1\text{ m/s}$, $s_{in} = 45\text{ g/kg}$).

Fig. 5. Axial variations in the water vapor pressure at the feed side of the membrane ($P_{f,m}$) and the HRU pressure (i.e., permeate-side pressure, P_r) for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80^\circ\text{C}$, $v_{f,in} = 1\text{ m/s}$, $s_{in} = 45\text{ g/kg}$).

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Table 1 Detailed specifications of the hollow-fiber module used in this study.

MD020CP2N (polypropylene)	
Length of fiber, L_f (m)	0.5
Inner diameter of shell, d_s (m)	0.03
Number of fibers, N_f (-)	100
Inner diameter of fiber, d_i (mm)	1.8
Outer diameter of fiber, d_o (mm)	2.6
Thickness of fiber, δ_m (mm)	0.4
Membrane porosity, ε (%)	70
Mean pore size, d_p (μm)	0.2
Liquid entry pressure of water (kPa)	140

[illegible]

Fig. 2. Schematic representation of (a) the shell-and-tube type VMD module, and (b) heat and mass transfer through the polypropylene hollow-fiber membrane in the VMD process.

Fig. 3. Axial variations in the bulk feed temperature (T_f) and liquid-vapor interface temperature at the feed side ($T_{f,m}$) for the different seawater-coolant feed arrangements (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

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Fig. 9. Effect of number of HRUs (N_r) on HWP and DWP with the BF arrangement (feed inlet conditions: $T_{fin} = 80 \text{ }^\circ\text{C}$, $v_{fin} = 1 \text{ m/s}$, $s_{in} = 45 \text{ g/kg}$).

Fig. 10. Effect of N_r on OSTEC and OSEEC in the BF arrangement (feed inlet conditions: $T_{f,in} = 80$ °C, $v_{f,in} = 1$ m/s, $s_{in} = 45$ g/kg).

Highlights

- ▶ An effect of BF and PF arrangements on the MVMD system was theoretically evaluated.
- ▶ The MVMD system with the BF arrangement was more efficient for freshwater production than the PF arrangement at a smaller number of module stages.
- ▶ An inappropriate number of module stages had a negative effect on the MVMD system performance.
- ▶ Four HRU stages with the BF arrangement showed an improved efficiency of the MVMD system.