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MAESTRÍA Y DOCTORADO EN CIENCIAS E INGENIERÍA

**“ESTUDIO DE SISTEMAS TÉRMICOS DE DESALINIZACIÓN Y  
ENFRIAMIENTO AL ALTO VACÍO CON EYECTO-CONDENSACIÓN  
BAROMÉTRICA ACTIVADOS CON ENERGÍAS RENOVABLES”**

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EL SIGUIENTE PROYECTO FUE DESARROLLADO  
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LA INVESTIGACIÓN SE LLEVÓ A CABO DENTRO  
DEL MARCO DEL PROGRAMA DE MAESTRÍAS Y  
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*“El tiempo es un regalo que no podemos recuperar; úsalo para servir,  
amar y disfrutar lo que haces” —Anónimo*

### **Dedico esta tesis:**

A Dios, creador de la vida, la sabiduría, la materia y la energía;  
principio y fin de todo lo que existe. Nada se crea ni se destruye  
sin Su voluntad, nada se sostiene sin Su propósito. El verdadero  
conocimiento nace del temor reverente a Él y en conocerle a Él y a  
Jesucristo, y dedico este trabajo doctoral porque todo es de Él, por  
Él y para Él (*Romanos 11:36*)

Dedico también esta tesis a la memoria de Elvia Contreras  
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en la antesala de su titulación. Su vida fue un ejemplo de fe,  
resiliencia, superación y pasión por aprender.

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# RESUMEN

La escasez hídrica y la demanda energética en climatización representan desafíos críticos para las regiones costeras áridas. Esta tesis doctoral demuestra la hipótesis de que un rediseño fundamental de la tecnología de Evaporación de Múltiple Efecto (MED) puede superar las limitaciones de los sistemas convencionales. Para ello, se desarrolla y valida un sistema innovador de poligeneración, que produce simultáneamente agua dulce y enfriamiento, logrando una eficiencia y capacidad superiores a las de las tecnologías existentes. La solución propuesta se basa en una configuración MED que opera a alto vacío (28.3–0.8 kPa) utilizando un eyector-condensador barométrico como componente central. Este diseño, estudiado teóricamente en el software Aspen Plus, permite expandir drásticamente el rango operativo del sistema, alcanzando una temperatura mínima de salmuera de 4 °C, muy por debajo del límite convencional (35–40 °C). Los resultados, validados rigurosamente y publicados en la revista *Desalination*, confirman la superioridad del sistema MED al alto vacío (MED-AV). En cumplimiento del objetivo general, en su modalidad de solo desalinización (modo invierno), el sistema incrementa la Relación de Producción (GOR) en un 34.1% (GOR=13.5) y duplica la Relación de Recuperación (RR=0.83), como consecuencia de una reducción del 25.4% en la energía térmica de activación. Como logro central de la poligeneración, al integrar un subsistema de enfriamiento que utiliza el agua producto como refrigerante, el sistema demuestra su capacidad para entregar 96 m<sup>3</sup>/d de agua potable y 700 TR (2,460 kW) de enfriamiento. El estudio de factibilidad técnica revela un Coeficiente de Desempeño Global (COP<sub>G</sub>) entre 9.3 y 10.3. En línea con el objetivo de comparación, el sistema supera consistentemente a las tecnologías de referencia, con reducciones del 25.4% en el consumo de energía térmica y del 11.19% en el consumo eléctrico. Como productos de divulgación y protección de la investigación, se derivan tres artículos científicos —uno publicado en *Desalination* como primer autor, otro publicado como autor de correspondencia en la revista *Membranes* y un tercero sometido en *Desalination* como primer autor— y dos solicitudes de patente. En conclusión, esta tesis establece un nuevo paradigma para la desalinización térmica, probando que es posible lograr una poligeneración de alta eficiencia y capacidad mediante una integración ingenieril inteligente y simplificada, sentando las bases para una nueva generación de tecnologías sostenibles para comunidades costeras.

# ABSTRACT

Water scarcity and the energy demand for space cooling represent critical challenges for arid coastal regions. This doctoral thesis demonstrates the hypothesis that a fundamental redesign of Multi-Effect Distillation (MED) technology can overcome the limitations of conventional systems. To this end, it develops and validates an innovative polygeneration system that simultaneously produces fresh water and cooling, achieving superior efficiency and capacity compared to existing technologies. The proposed solution is based on an MED configuration operating at high vacuum (28.3–0.8 kPa) using a barometric ejector-condenser as its core component. This design, modeled and theoretically studied in Aspen Plus, allows for a drastic expansion of the system's operational range, achieving a minimum brine temperature of 4 °C, far below the conventional limit of 35–40 °C. The results, rigorously validated and published in the journal *Desalination*, confirm the superiority of the High-Vacuum MED (MED-AV) system. Fulfilling the general objective, in its desalination-only mode, the system increases the Gained Output Ratio (GOR) by 34.1% (GOR=13.5) and doubles the Recovery Ratio (RR=0.83), as a consequence of a 25.4% reduction in the required thermal activation energy. As a central achievement of the polygeneration approach, upon integrating a cooling subsystem that uses the product water as a refrigerant, the system demonstrates its capacity to deliver 96 m<sup>3</sup>/day of potable water and 700 TR (2,460 kW) of cooling. The technical feasibility study reveals a Global Performance Coefficient (COP<sub>G</sub>) between 9.3 and 10.3. Aligned with the comparison objective, the system consistently outperforms benchmark technologies, achieving reductions of 25.4% in thermal energy consumption and 11.19% in electrical energy consumption. As products for knowledge dissemination and intellectual property protection, this research has yielded three scientific articles—one published in *Desalination* as first author, another as corresponding author in *Membranes*, and a third submitted to *Desalination* as first author—along with two patent applications. In conclusion, this thesis establishes a new paradigm for thermal desalination, proving that high-efficiency, high-capacity polygeneration is achievable through intelligent and simplified engineering integration, laying the groundwork for a new generation of sustainable technologies for coastal communities.

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# CAPÍTULO 1

*“La imaginación es más importante que el conocimiento”*  
—Albert Einstein

## INTRODUCCIÓN

---

Este capítulo introduce la investigación al establecer la descripción de las problemáticas de la escasez de agua y la creciente demanda de acondicionamiento térmico de espacios, justificando la necesidad de soluciones. A continuación, se revisan los antecedentes más relevantes para, posteriormente, formular la hipótesis, los objetivos y el alcance del proyecto.

### 1.1. Problemática y justificación

La seguridad hídrica y energética son dos de los desafíos globales más apremiantes del siglo XXI, exacerbados por una triada de factores críticos: el cambio climático, la gestión ineficiente de los recursos y el crecimiento poblacional acelerado que ejerce una presión insostenible sobre los suministros naturales.

La naturaleza de la crisis hídrica global yace en una profunda paradoja: la disparidad entre la abundancia de agua en el planeta y su escasa disponibilidad para uso humano. Aunque aproximadamente el 75% de la superficie terrestre está cubierta por agua, solo el 2.5% corresponde a agua dulce (**Chen et al., 2022**). A su vez, de esta reducida fracción, la mayor parte se encuentra inmovilizada en glaciares y acuíferos profundos, como se aprecia en la Figura 1.1. En consecuencia, menos del 1% del total del agua dulce del planeta se encuentra en fuentes superficiales inmediatamente accesibles, constituyendo el recurso real del que depende la humanidad (**Musie & Gonfa, 2023**). Esta porción crítica de agua dulce, vital para la salud, la seguridad alimentaria y la estabilidad socioeconómica, es la que enfrenta una demanda insostenible y sin precedentes.





**Figura 1.1.** Proporción y disponibilidad de agua en el planeta. Fuente: Centro Virtual de Información del Agua (Agua.org.mx), 2017.

Dicha demanda insostenible, es impulsada por una dinámica demográfica irrefutable: la población mundial ha crecido de 2,500 millones en 1950 a más de 8,200 millones en la actualidad (UN, 2024). Este fenómeno es particularmente severo en regiones áridas y semiáridas, como el noroeste de México, donde la disponibilidad del recurso ha disminuido drásticamente —hecho evidenciado por los altos niveles de sequía que persisten en la región, como se visualiza en la Figura 1.2. (Monitor de Sequía de México, CONAGUA, 2025)— en contraste con una demanda creciente. Esta crítica situación posiciona a la desalinización de agua de mar no solo como una alternativa razonable, sino como una estrategia indispensable (CONAGUA, 2023).

La viabilidad de una desalinización sostenible en regiones se sustenta en su alto potencial de recursos renovables. Las áreas con mayor estrés hídrico, como el noroeste de México, coinciden con las zonas de mayor irradiación solar del país, como se aprecia en la Figura 1.3. (Mapa de Irradiancia Global en México). Esta coincidencia geográfica hace de la desalinización activada por energía solar una solución técnicamente coherente, capaz de garantizar el suministro de agua potable a largo plazo y de romper la dependencia de los combustibles fósiles.

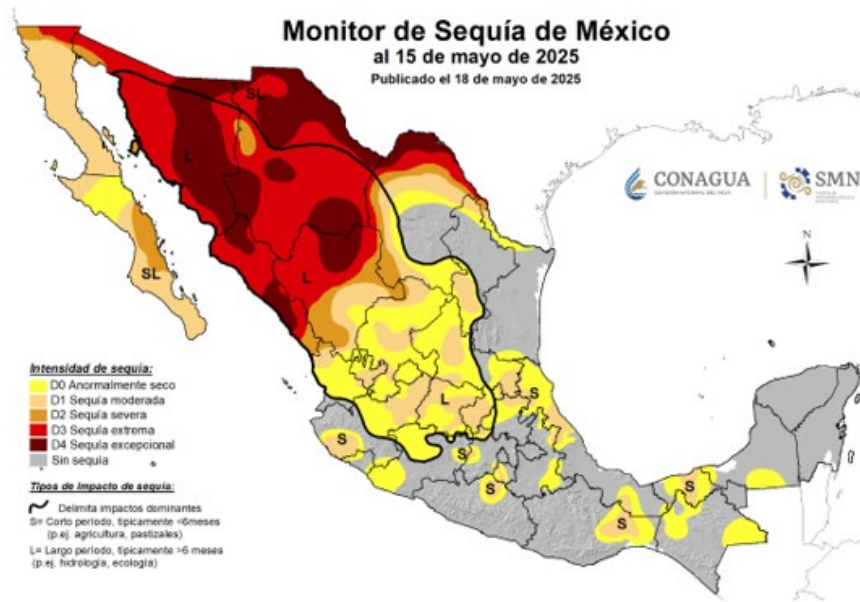


Figura 1.2. Monitor de Sequía de México. Fuente: CONAGUA, 2025.

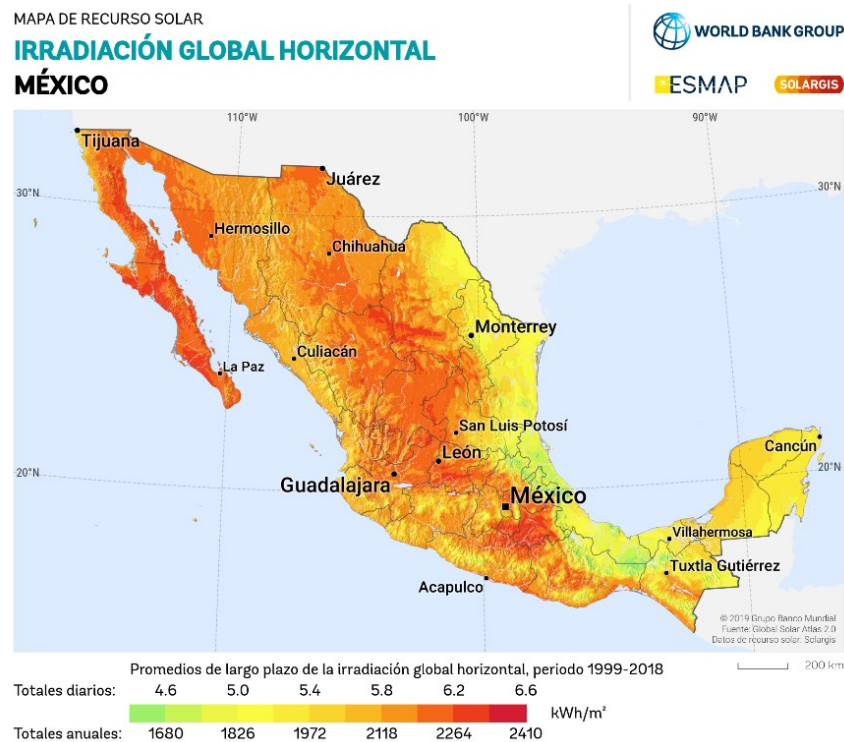


Figura 1.3. Mapa de Irradiancia Global en México. Fuente: The World Bank, Solar resource data: Solargis, 2017

De manera paralela, la creciente demanda de refrigeración emerge como un desafío de igual magnitud a nivel global. Impulsada por el aumento de las temperaturas y la urbanización, la climatización de espacios representa aproximadamente el 8.5% del consumo eléctrico mundial, una cifra que la Agencia Internacional de Energía (IEA, por sus siglas en inglés) anticipa se triplique para el año 2050. Esta demanda energética no solo ejerce una presión extrema sobre las redes eléctricas, sino que se encuentra intrínsecamente ligada a la crisis hídrica. Cuando se satisface con sistemas convencionales, se genera el círculo vicioso de la demanda de enfriamiento planteado por **Velázquez-Limón, (2002)**: un mayor uso de aire acondicionado incrementa la quema de combustibles fósiles y las emisiones de gases de efecto invernadero (GEI), lo que agrava el calentamiento global, y a su vez, intensifica la demanda de refrigeración, cerrando así el ciclo.

Tradicionalmente, los problemas de la demanda hídrica y la necesidad de acondicionamiento térmico se han abordado de forma independiente. No obstante, su intrínseca relación, intensificada por el crecimiento poblacional, ha generado dos círculos viciosos de gran impacto que se retroalimentan, tal y como lo conceptualizó **López-Zavala, (2019)**. El primer ciclo se inicia con la escasez de agua potable, la cual impulsa el uso de tecnologías de desalinización de alto consumo energético. Cuando esta energía proviene de fuentes fósiles, se emiten Gases de Efecto Invernadero (GEI) que aceleran el cambio climático; este, a su vez, altera los patrones de precipitación y reduce la recarga de acuíferos, profundizando así la escasez inicial de agua. Paralelamente, las alteraciones producidas por el cambio climático activan un segundo ciclo. El aumento de las temperaturas globales incrementa la demanda de enfriamiento, la cual, al ser satisfecha con tecnologías que consumen electricidad de origen fósil, genera más emisiones de GEI, agravando el calentamiento global, y en consecuencia intensificando aún más la demanda de enfriamiento.

Frente a esta problemática dual, cuyos detonantes principales son el crecimiento poblacional y la mala gestión de los recursos, la convergencia de estos ciclos representa un desafío socioambiental complejo y actualmente abierto. En este escenario, cualquier solución aislada a un problema corre el riesgo de agravar el otro, haciendo insostenibles los enfoques tradicionales. Precisamente esta interacción negativa, alimentada por la presión demográfica, revela la oportunidad de desarrollar sistemas de poligeneración que produzcan agua potable y enfriamiento de manera simultánea y eficiente.

En este sentido, la desalinización térmica, específicamente la tecnología de Evaporación/Destilación de Múltiple Efecto (MED), se posiciona como una solución robusta para regiones costeras por su tolerancia a aguas de alta salinidad. No obstante, su implementación, en acoplamientos con unidades de enfriamiento para abordar ambos círculos viciosos, se ha visto limitada por los altos consumos energéticos, la complejidad de las configuraciones y la consecuente alta inversión de capital (CAPEX) que requieren.

Es precisamente en la superación de estas limitaciones donde se inserta la presente investigación doctoral. Por ello, en este trabajo se propone el estudio de sistemas térmicos de desalinización y enfriamiento al alto vacío con eyecto-condensación barométrica activados con energías renovables, para la producción simultánea de agua y frío. La novedosa configuración de estos sistemas busca, mediante la reducción de componentes, modificación de sus condiciones de operación al alto vacío y la integración de un subsistema de enfriamiento y eyecto-condensación, superar las ineficiencias y el alto CAPEX de los sistemas térmicos de desalinización y enfriamiento convencionales. El objetivo final es proveer una solución técnico-económica viable y sostenible para comunidades costeras con estrés hídrico. Cabe destacar que las tecnologías de desalinización térmicas se consideran las más adecuadas para este fin, debido a su gran capacidad de acoplamiento con los sistemas de captación solar térmica y su versatilidad para la poligeneración.

## 1.2. Antecedentes

### 1.2.1. Evolución de la Desalinización

*“La historia del agua es equivalente a la historia del mundo y la historia de la calidad del agua es equivalente a la historia de la vida.”*

—*Andreas N. Angelakis*

Rastrear el origen exacto del concepto de desalinización resulta una tarea compleja, ya que es probable que las primeras indagaciones surgieran de manera recurrente sin dejar un registro formal. No obstante, como documenta (**Birkett, 1984**), textos con más de dos milenios de antigüedad preservan reflexiones que pueden interpretarse como los fundamentos de la destilación y, por extensión, de la desalinización. Un caso emblemático se encuentra en el Libro de Job de la Biblia (citado en **Birkett, 1984**), donde se describe un proceso hidrológico que evoca claramente

el principio de la destilación: “*pues Él reduce a pequeñas gotas las aguas, que derraman la lluvia según su vapor, el cual destilan las nubes y lo deja caer abundantemente sobre el hombre*”. Dicha conceptualización fue precisada siglos después por Aristóteles (348-322 a.C.), quien, mediante observación, estableció el principio físico clave: “*El agua salada al evaporarse forma agua dulce, y el vapor no se condensa formando de nuevo agua de mar. Esto se sabe por experimentación*” (citado en **Kalogirou, 2005**). Estas observaciones fundacionales, que equiparan el ciclo natural del agua con un acto de destilación, sentaron la base conceptual sobre la cual se erigiría, siglos después la ingeniería de la desalinización.

Si bien el desarrollo de plantas de desalinización a gran escala no fue técnicamente viable sino hasta finales de la Segunda Guerra Mundial (**Needham, et al., 1980**), existen numerosos ejemplos de experimentación sobre tecnologías de desalinización a lo largo de la antigüedad. Un hito temprano y revelador de esta transición hacia la aplicación ingenieril lo constituye el método descrito por Alejandro de Afrodisias alrededor del año 200 d.C. (citado en **Kalogirou, 2005**). Según estos relatos, los marineros ante la necesidad de producir agua dulce en alta mar, hervían agua de mar y utilizaban esponjas para condensar y absorber el vapor. Este vapor podía luego exprimirse de la esponja y usarse para beber. Debido a estos avances, los viajes marítimos de larga distancia se hicieron posibles.



**Figura 1.4.** Marineros producen agua dulce mediante destilación de agua de mar (Kalogirou, 2005).

La Revolución Industrial sentó las bases tecnológicas que catalizaron la era moderna de la desalinización. Un hito fundamental en este periodo fue el descubrimiento del fenómeno de ósmosis por el físico francés Jean-Antoine Nollet, principio que, siglos después, se convertiría en la base de las tecnologías de membranas (**Angelakis et al., 2021**). Paralelamente, durante el siglo XIX se materializaron los conceptos de los procesos térmicos que dominarían la industria: en la década de 1830, Norbert Rillieux concibió y patentó un sistema de evaporación de múltiple efecto (MED), el cual se aplicó en el ingenio azucarero, mientras que hacia finales del siglo, Waterhouse esbozó en una patente el concepto de destilación flash de múltiple etapa (MSF), aunque este último no se materializaría en una planta operativa hasta medio siglo después (**Birkett, 1984**).

El impulso definitivo para la desalinización a gran escala llegó tras la Segunda Guerra Mundial, con la creación de la Oficina de Agua Salada en los Estados Unidos en 1955, una institución dedicada a la investigación y desarrollo que consolidó tecnológicamente este campo (**Angelakis et al., 2021**). La materialización de la desalinización a escala industrial comenzó a principios del siglo XX. En 1928 se construyó la primera planta de destilación terrestre del mundo en Curazao, Antillas Neerlandesas, la cual operaba bajo principios térmicos (**Lattemann et al., 2010**). Este desarrollo pionero fue seguido, durante la década de 1950, por el establecimiento de las primeras plantas en la Península Arábiga —específicamente en Arabia Saudita, Kuwait, Bahrén y Qatar—, donde se implementaron tecnologías de pretratamiento como filtros de arena a gran escala, marcando el inicio de la desalinización como un componente vital de la infraestructura hídrica nacional (**Angelakis et al., 2021**).

Impulsada por el crecimiento demográfico y la creciente escasez hídrica, la desalinización experimentó una rápida evolución y expansión global a partir de la segunda mitad del siglo XX. En la actualidad, muchos países, especialmente en Oriente Medio, dependen críticamente de esta tecnología, mientras que un número creciente de naciones está diversificando sus alternativas de suministro de agua mediante la incorporación de plantas desalinizadoras como una estrategia robusta para garantizar la seguridad hídrica frente a la variabilidad climática, lo que ha resultado en un incremento acelerado de la capacidad instalada a nivel mundial.

Esta tendencia concreta se refleja en cifras concretas. A partir de 2010, la capacidad global de desalinización ha registrado un incremento porcentual promedio del 6.8% anual, lo que equivale a

una incorporación de aproximadamente 4.6 millones de m<sup>3</sup>/d cada año (**Ríos-Arriola et al., 2022**). Este crecimiento se mantuvo de forma sostenida en la última década, con una capacidad instalada reportada de 95.37 millones de m<sup>3</sup>/d en 2019 (**Jones et al., 2019**), lo que representó un crecimiento del 3.1% interanual. Dicha tendencia continuó al año siguiente, alcanzándose los 97.5 millones de m<sup>3</sup>/d en 2020, con un incremento adicional del 1.9% (**Eke et al., 2020**). Estas estadísticas consolidan a la desalinización como una de las industrias de infraestructura hídrica de más crecimiento a nivel global.

### **1.2.2. Tecnologías actuales de desalinización y sus desafíos**

En la actualidad, las tecnologías de desalinización se clasifican en dos categorías principales: procesos térmicos y procesos de membranas. Entre los primeros destacan la Destilación Flash Multietapa (MSF), la Destilación/Evaporación de Múltiple Efecto (MED) y la Compresión Mecánica de Vapor (MVC); mientras que en la segunda categoría se encuentran la Ósmosis Inversa (RO), la Electrodialisis (ED) y la Destilación por Membrana (MD). Tres tecnologías concentran el 93% del mercado global: la Ósmosis Inversa lidera con el 70% de la capacidad instalada, seguida de la MSF (17%) y la MED (6%) (**Ríos-Arriola et al., 2022**).

La elección de la tecnología está determinada por el contexto geográfico y la calidad del agua de alimentación. Las tecnologías térmicas, particularmente MSF y MED, son predominantes en regiones como el Golfo Pérsico, donde se enfrentan a agua con alta salinidad y recurrente proliferación de algas. Su robustez ante estas condiciones les confiere una clara ventaja, ya que estos contaminantes afecta severamente el rendimiento y la vida útil de las membranas de Ósmosis Inversa, encareciendo los costos de pretratamiento (**Goh et al., 2021**). Adicionalmente, los procesos térmicos generalmente requieren un pretratamiento menos complejo, un menor consumo de energía eléctrica y un mantenimiento menos exigente en comparación con las tecnologías de membranas (**Ahmed et al., 2020; Goh et al., 2021; Khoshgoftar et al., 2013**), lo que los hace técnicamente viables para su acoplamiento con fuentes de energía térmica, como la solar.

Pese a las mejoras sustanciales en eficiencia, capacidad y reducción de costos que han consolidado la viabilidad de la MED, (**Morin, 1993; El-Dessouky et al., 1998; Al-Shammiri and Safar, 1999; Darwish et al., 2006; Sharaf et al., 2011; Druetta et al., 2013; Palenzuela et al.,**

**2013; Palenzuela et al., 2014; Al-Mutaz & Wazeer, 2014**), su optimización se ha visto limitado por los desafíos que representan las pérdidas de energía por efluente en el condensador y a los límites operativos que tiene en la temperatura máxima y mínima de la salmuera, (TBT, por sus siglas en inglés) y (BBT, por sus siglas en inglés), respectivamente (**Ahmed et al., 2020**). Por un lado, la Temperatura Máxima de la Salmuera (TBT) está restringida a 70 °C para evitar la corrosión acelerada por el agua de mar (**Goh et al., 2021**), mientras que la Temperatura Mínima de la Salmuera (BBT) está limitada por la temperatura del agua de mar de enfriamiento ( $\approx 25\text{-}30$  °C), requiriendo un gradiente térmico mínimo de ( $\approx 5$  °C) para la condensación del vapor del último efecto de la MED (**Aly, 1995; Shahzad et al., 2015**). Adicionalmente, los análisis exergeticos identifican al condensador como el componente de mayor pérdida de energía útil, donde se disipa una gran cantidad de calor latente al mar (**Gude, 2018**).

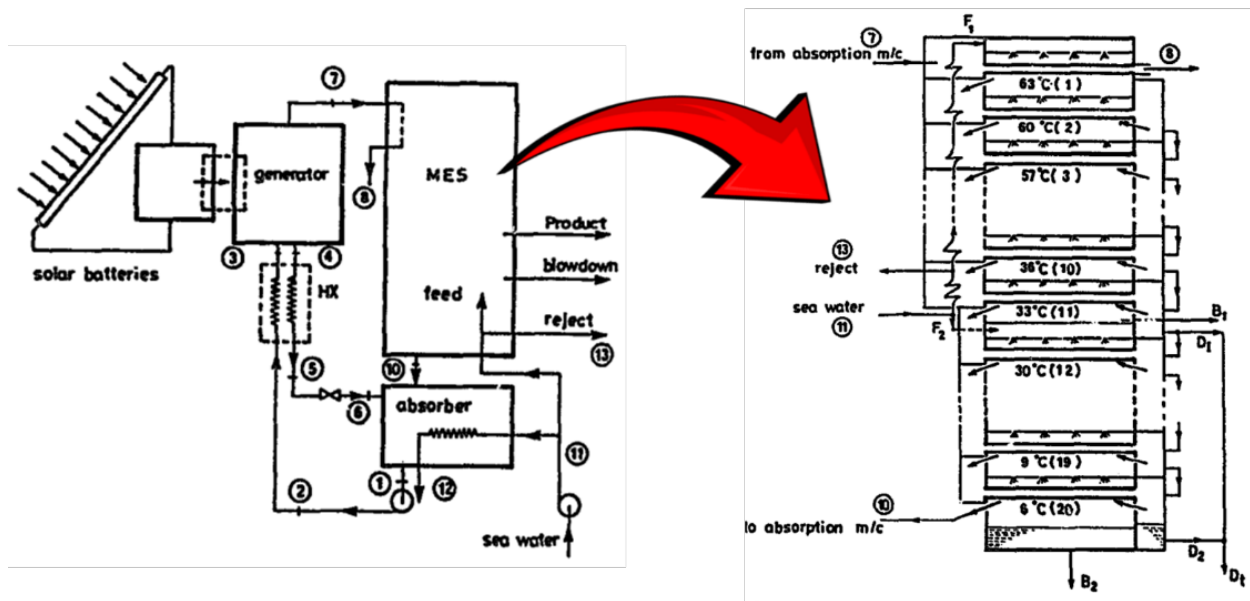
Para superar estas limitaciones, una estrategia recurrente ha sido la hibridación de sistemas MED con tecnologías de enfriamiento por absorción o adsorción. Además, la convergencia de las demandas de agua desalada y acondicionamiento térmico de espacios ha representado una oportunidad para desarrollar soluciones integradas que optimicen el uso de recursos energéticos y reduzcan la inversión inicial (CAPEX). En este contexto, ha aumentado el interés por integrar sistemas de desalinización con tecnologías de enfriamiento, lo que permite la producción simultánea de ambos servicios. Sin embargo, estas integraciones enfrentan retos importantes, como la complejidad de diseño de los sistemas y los costos asociados a su implementación (**López-Zavala et al., 2019; López-Zavala et al., 2023**). Las configuraciones de acoplamiento más exploradas integran procesos de desalinización MED, MSF y humidificación-deshumidificación (HDH) combinados con unidades de enfriamiento por absorción, adsorción o eyección (**Abdulrahim & Darwish, 2015; Caballero-Talamantes et al., 2024**).

Otra línea de investigación se ha centrado en expandir el rango operativo de los sistemas MED reduciendo la Temperatura mínima de la salmuera (BBT, por sus siglas en inglés) (**E. Ali et al., 2020; Farsi et al., 2019; Ghenai et al., 2021**). **Aly, 1995**, fue uno de los primeros en investigar configuraciones innovadoras y explorar nuevos rangos operativos de presión y temperatura. En su estudio, se empleó una TBT de 63 °C y una BBT de 6 °C, siendo esta última significativamente más baja que en los sistemas convencionales (Figura 1.5.). Esto fue posible mediante la integración

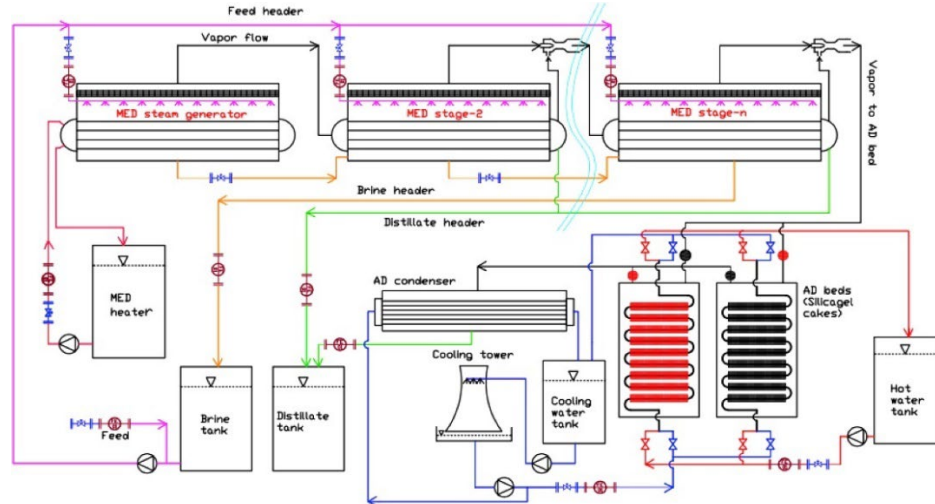


de un sistema de enfriamiento por absorción LiBr-H<sub>2</sub>O de 220 kW, lo que permitió implementar una cascada de 20 efectos y mejorar los indicadores de desalinización, logrando un GOR de 14.8 y un PR de 14.2. Si bien esta configuración aprovecha sinergias térmicas avanzadas como el aprovechamiento del calor retirado del absorbedor y condensador para la activación de la MED, su enfoque no está orientado a la cogeneración de frío, y la configuración resultante conlleva una mayor complejidad del sistema.

Alternativamente, *Shahzad et al., 2014* y *Shahzad et al., 2015*, analizaron el acoplamiento de una MED con un sistema de adsorción (AD) de gel de sílice para manejar el vapor a baja temperatura del último efecto, como se aprecia en la Figura 1.6. La MED operó en los rangos de temperatura (TBT-BBT) de 50–3 °C y 70–5 °C, reportando un incremento de 2 a 3 veces en la producción de agua. La viabilidad a escala industrial de este enfoque, sin embargo, debe considerar los desafíos asociados a la controlabilidad del sistema y la regeneración del material adsorbente.

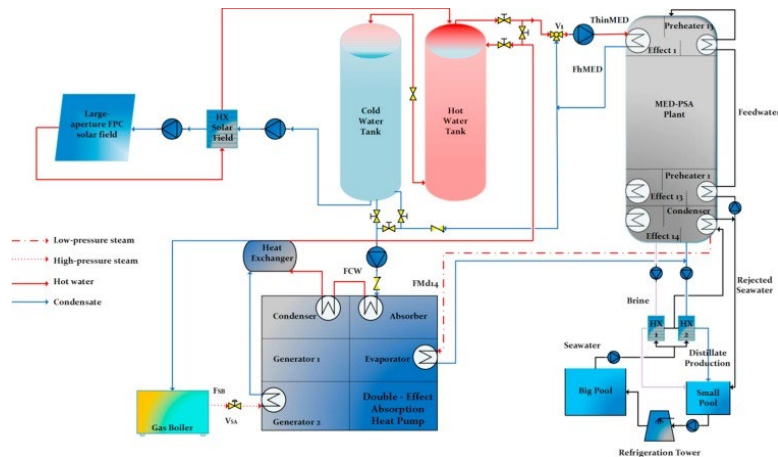


**Figura 1.5.** Diagrama del sistema de enfriamiento por absorción de LiBr/H<sub>2</sub>O y desalinización MED que opera con una BBT de 6 °C (Aly, 1995).



**Figura 1.6.** Diagrama esquemático del sistema MEDAD que acopla un sistema adsorción (AD) y un sistema de desalinización MED que opera con una BBT de 5 °C (Shahzad et al., 2014).

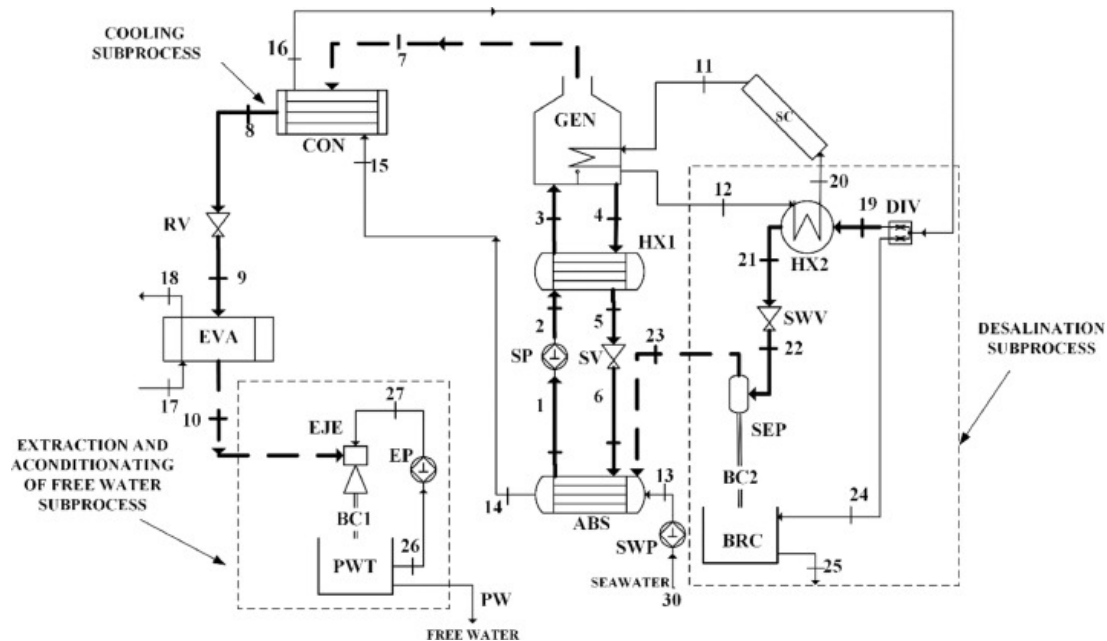
En la búsqueda de una mayor eficiencia, investigaciones recientes han demostrado el potencial de acoplar sistemas MED con sistemas de enfriamiento por absorción **Chorak et al., 2018** y **Alarcón-Padilla et al., 2007** (Figura 1.7.), validaron experimentalmente esta integración, utilizando el efecto frigorífico del evaporador de una máquina de absorción para condensar el vapor del último efecto del sistema MED. Esta estrategia permitió reducir las pérdidas energéticas y reportaron mejorar notables, alcanzando una Relación de Rendimiento (PR, por sus siglas en inglés) de 18.53 y una capacidad de desalinización de 2.42 m<sup>3</sup>/h. Un desafío que emerge de esta y otras configuraciones similares es la complejidad del sistema, la cual incrementa el número de componentes y puede impactar la viabilidad tecno-económica.



**Figura 1.7.** Diagrama de la planta de desalinización DEAHF-MED en la Planta Solar de Almería (Chorak et al., 2018)

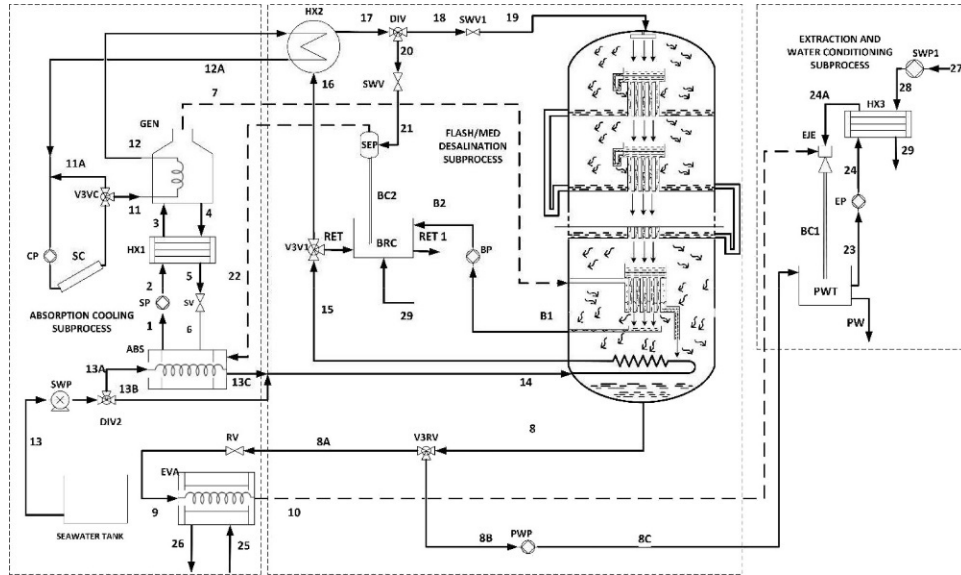
Explorando fuentes de energía sostenibles, **López-Zavala et al., 2019a**, propusieron un innovador sistema de enfriamiento y desalinización, basado en el ciclo de absorción de simple efecto LiBr/H<sub>2</sub>O, con triple nivel de presión activado por energía solar térmica, logrando una producción de 838 L/d con 23 kW de enfriamiento, como se muestra en la Figura 1.8. Su innovadora configuración de presión mejoró la eficiencia en un 22% respecto a los ciclos convencionales, destacando su potencialidad, aunque también evidencia el reto de alcanzar niveles de competitividad en el mercado con configuraciones complejas.

En una contribución posterior, **López-Zavala et al., 2019b**, integró un sistema de Destilación de Múltiple Efecto (MED) de 14 efectos con un sistema abierto de absorción de simple efecto de 23 kW (Figura 1.9.). Se reportó un incremento extraordinario de 87.4 veces en la capacidad de enfriamiento (2,012 kW) y un COP de 6.15. Si bien estos resultados representan un avance conceptual en el desempeño térmico, la configuración del sistema híbrido resultante es más compleja, incorporando un mayor número de componentes.

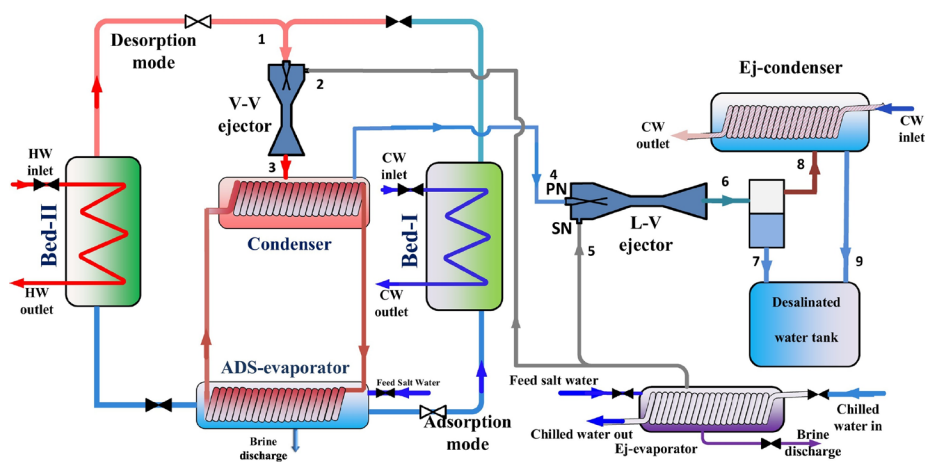


**Figura 1.8.** Diagrama del sistema de enfriamiento y desalinización por absorción de LiBr/H<sub>2</sub>O activado con energía solar térmica (López-Zavala et al., 2019a).

Una estrategia prometedora para mejorar la competitividad es la adaptación de tecnologías existentes para la cogeneración con modificaciones mínimas. En este contexto, **Askalany et al., 2020**, propusieron una unidad de enfriamiento y desalinización por adsorción, modificando el evaporador e incorporando dos eyectores para alcanzar una mejora en el COP del 192% ( $COP=2.2$ ) y un rendimiento de  $79.4 \text{ m}^3$  por tonelada de gel sílice. Este trabajo muestra el potencial de rediseños innovadores para añadir funcionalidad (Figura 1.10.).

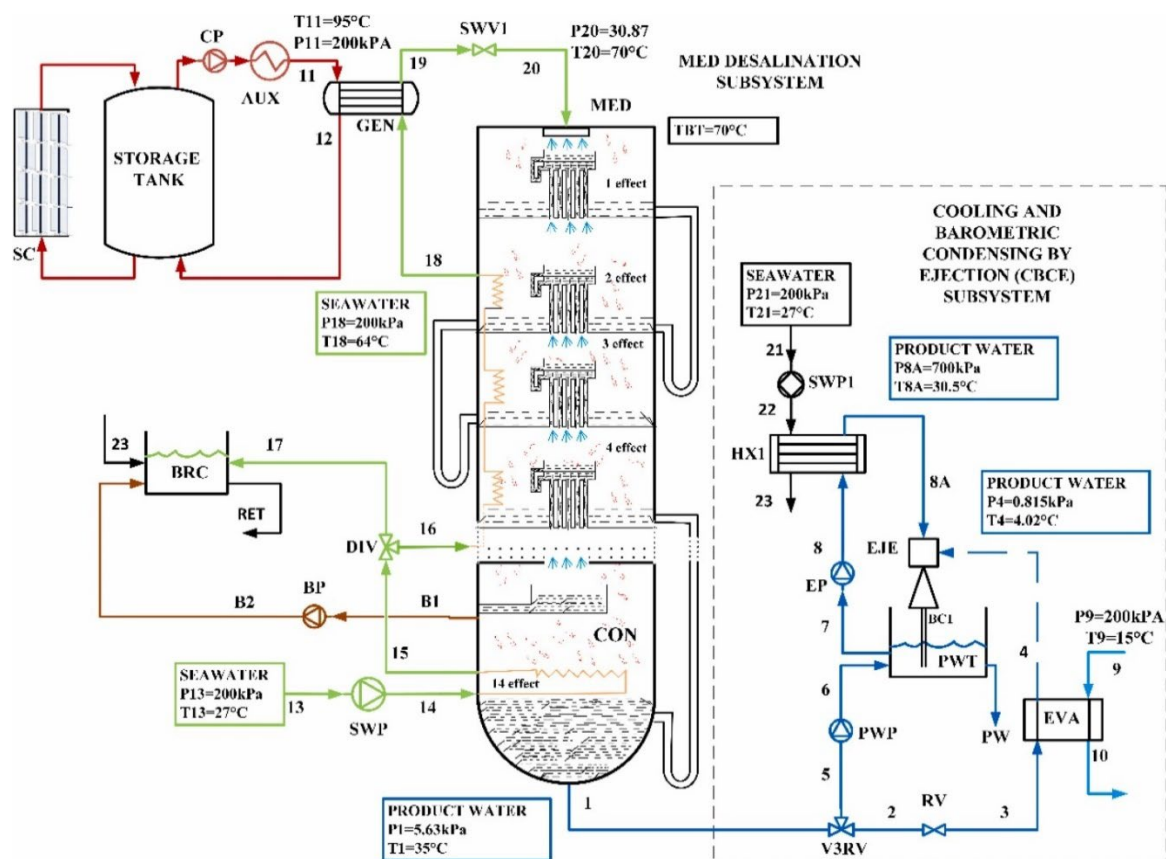


**Figura 1.9.** Diagrama del sistema de enfriamiento por absorción de LiBr/H<sub>2</sub>O y desalinización FLASH/MED activado con energía solar térmica (López-Zavala et al., 2019b).



**Figura 1.10.** Diagrama del sistema AD2EJ-HR que consta de la hibridación entre el ciclo de desalinización por adsorción y dos eyectores bifásicos (Askalany et al., 2020).

Recientemente, un avance significativo hacia la integración directa fue reportado por **López-Zavala et al., 2023**, quienes utilizaron el agua producto de un sistema MED convencional para generar enfriamiento mediante la expansión súbita y un eyector-condensador barométrico, alcanzando 2,043 kW de enfriamiento (Figura 1.11.). Una característica de este diseño es que mantiene los parámetros operativos convencionales de una MED (TBT-BBT de 70–35 °C), lo que se traduce en una capacidad de desalinización y PR comparables a los sistemas base. Además, reporta que el COP es 6.53 veces superior al de un sistema de enfriamiento por absorción de simple efecto. Este desarrollo se encuentra en trámite de patente ante el Instituto Mexicano de la Propiedad Industrial (IMPI), con número de expediente MX/a/2020/008009.



**Figura 1.11.** Diagrama del sistema de desalinización y enfriamiento MED con eyecto-condensación barométrica (López-Zavala et al., 2023).

### 1.2.3. Eyecto-condensadores

Los eyecto-condensadores, o eyectores de condensación (EC), son dispositivos que operan con un flujo bifásico (líquido-vapor). Su principio de funcionamiento se basa en la gran diferencia de velocidad entre la corriente motriz (fluido primario) y la corriente de succión (fluido secundario), produce una transferencia de cantidad de movimiento, masa y calor. Esta interacción provoca la condensación de la fase vapor, resultando una corriente en fase líquida a la salida del eyector. El proceso de cambio de fase es impulsado tanto por la mezcla turbulenta, como por la transferencia de calor entre fases (**Colarossi et al., 2012**).

En las últimas décadas, los esfuerzos de investigación se han centrado en desarrollar trabajos teóricos y experimentales para comprender las interacciones entre los fluidos primario y secundario dentro de las diversas secciones del eyector. Ha predominado el desarrollo de múltiples modelos matemáticos que evalúan el efecto de los aspectos geométricos en el rendimiento del eyector, como lo son la posición de la boquilla motriz, la relación de área en la cámara de mezcla, longitud de la cámara de succión, entre otros (**Keenan et al., 1950; Eames et al., 1995; Rogdakis and Alexis, 2000; Ouzzane and Aidoun, 2003; Ruangtrakoon et al., 2011; Liu and Groll, 2013; Pounds et al., 2013; Dong et al., 2020**). Sin embargo, la mayoría de estos trabajos se ha centrado en eyectores que operan con ambas corrientes en fase de vapor, existiendo una brecha significativa en el modelado y comprensión de los eyectores de condensación propiamente dichos, donde la interacción líquido-vapor es fundamental.

Entre los trabajos especializados en eyectores de condensación, destacan la investigación realizada en el MIT (*Massachusetts Institute of Technology*) por parte de **Levy, (1963)** y el estudio teórico realizado por **Wang et al., (2022)**. En estos trabajos se analiza la interacción líquido-vapor que ocurre en el interior de los eyectores bifásicos de condensación mediante de modelos unidimensionales (1D), así como el comportamiento y el rendimiento del dispositivo a través de estudios paramétricos.

Con base a la problemática y áreas de oportunidad antes expuestas, se muestra que se ha intentado desarrollar tecnologías que satisfagan de manera simultánea las necesidades de agua y frío. Las estrategias predominantes se basan en la hibridación de sistemas MED con sistemas de enfriamiento por absorción/adsorción, lo que conlleva una complejidad que impacta la viabilidad

tecnico-económica. Por otro lado, las propuestas que priorizan la simplicidad en la configuración o el desempeño en desalinización han logrado avances notables en frentes específicos, pero no han logrado conjugar de forma óptima una cogeneración de alta capacidad y eficiencia con una configuración simplificada. Esta brecha motiva la presente investigación, la cual propone un nuevo sistema de producción simultánea de agua y frío que, mediante la modificación de un sistema MED convencional para operar al alto vacío (28.3–0.8 kPa, 68–4 °C) y la integración de un subsistema de enfriamiento y eyectocondensación, utiliza el agua producto directamente como refrigerante. Lo anterior representa un avance hacia una nueva generación de tecnologías térmicas de desalinización y enfriamiento con mayor capacidad y eficiencia.

### **1.3. Hipótesis**

El desarrollo de un sistema térmico de desalinización y enfriamiento simultáneo, basado en evaporación de múltiple efecto con eyecto-condensación barométrica que opere en condiciones de alto vacío y baja temperatura de activación con energía solar térmica, resultará en una tecnología de poligeneración que supere a los sistemas convencionales mediante un incremento en su eficiencia, su capacidad de producción y una reducción significativa de la inversión de capital inicial.

### **1.4. Objetivos**

#### **1.4.1. General**

Desarrollar un sistema térmico para la producción de agua y frío basado en evaporación de múltiple efecto con eyecto-condensación barométrica, que opere al alto vacío, con baja temperatura de activación (TBT), con energía solar térmica, para alcanzar una mayor eficiencia y capacidad en la producción simultánea de agua y enfriamiento en comparación con las tecnologías convencionales.

#### **1.4.2. Específicos**

- Realizar una revisión del estado del arte de las tecnologías de desalinización térmica, los eyectores bifásicos y los sistemas de desalinización y enfriamiento simultáneo.
- Generar propuestas de sistemas de desalinización y enfriamiento simultáneo que operen

al alto vacío con eyecto-condensación y se activen con energías renovables.

- Proponer y estudiar teóricamente el proceso de eyecto-condensación, integrado en los sistemas propuestas, mediante los softwares Aspen Plus y EES.
- Validar el simulador con resultados experimentales reportados en la literatura.
- Realizar un estudio de factibilidad técnica de las propuestas tecnológicas de desalinización y enfriamiento en simulador Aspen Plus, para determinar las condiciones óptimas de operación de los sistemas propuestos y los límites operativos.
- Comparar los sistemas propuestos contra otros equipos presentes en el estado de la técnica.
- Divulgar los resultados de la investigación mediante la publicación de artículos en revistas científicas indexadas.
- Redactar y escribir el documento del trabajo doctoral.

## 1.5. Descripción general de la tesis

La presente investigación se desarrolló bajo la modalidad de tesis por compendio — disertación de artículos científicos—, establecida en el programa de Maestrías y Doctorado en Ciencias e Ingeniería (MyDCI) del Instituto de Ingeniería de la UABC, para la obtención del grado de Doctorado en Ingeniería. Esta modalidad consiste en la integración de un conjunto de publicaciones científicas —en este caso, tres artículos científicos y dos solicitudes de patentes— que constituyen el núcleo de la contribución doctoral.

Para demostrar el cumplimiento de los objetivos planteados, la presente tesis se estructura de la siguiente manera: el **Capítulo 1** ha establecido el marco introductorio, que comprende la problemática, justificación, antecedentes, hipótesis y objetivos. El **Capítulo 2** presenta los fundamentos teóricos que sustentan el desarrollo tecnológico de esta investigación. El **Capítulo 3** presenta las contribuciones centrales de esta investigación, representadas por los artículos científicos y patentes que constituyen el cuerpo y desarrollo tecnológico de este trabajo. El **Capítulo 4** presenta las conclusiones generales derivadas de la investigación, integrando los hallazgos particulares de cada producto y proponiendo líneas de trabajo futuro. Los documentos completos de las publicaciones científicas en las cuales se participó como coautor se incluyen en los **Anexos**.



# CAPÍTULO 2

*“Lo que sabemos es una gota,  
lo que no sabemos es un océano”  
—Isaac Newton*

## FUNDAMENTOS TEÓRICOS

---

Este capítulo establece los fundamentos teóricos que rigen la base de la presente investigación doctoral. Se presenta la composición detallada del agua de mar como materia prima en el proceso de desalinización, analizando su composición fisicoquímica, su clasificación por contenido de sólidos disueltos totales y los límites termodinámicos que gobiernan la solubilidad de sales y los fenómenos de incrustación. Asimismo, se fundamentan los principios de operación de las principales tecnologías de desalinización, destacando la tecnología de evaporación de múltiple efecto MED.

### 2.1. Naturaleza y composición del agua de mar

La efectividad de cualquier proceso de desalinización está determinada por la naturaleza fisicoquímica del agua de mar, constituyendo este fluido la materia prima fundamental. Comprender su composición y comportamiento es fundamental para el diseño, la operación y la optimización de tecnologías de separación, especialmente en procesos térmicos, en los cuales los equilibrios químicos y los mecanismos de transporte de masa y calor definen el rendimiento global.

#### 2.1.1. Composición fisicoquímica

El agua de mar es una solución compleja y estable de sales, gases y materia orgánica, cuya composición iónica a nivel global se mantiene en proporciones notablemente constantes, un principio conocido como de “Proporción Relativa Constante de los Componentes Mayores” o Principio de Marcet (Millero et al., 2008). La composición del agua de mar se basa en seis iones predominantes —cloruro ( $Cl^-$ ), sodio ( $Na^+$ ), sulfato ( $SO_4^{2-}$ ), magnesio ( $Mg^{++}$ ), calcio ( $Ca^{++}$ )

y potasio ( $K^+$ )— que en conjunto constituyen aproximadamente el 99% de los sólidos disueltos. De manera representativa, el cloruro y el sodio constituyen aproximadamente el 55% y 30%, respectivamente, de la masa total de sólidos disueltos, representando en conjunto el 85% de la salinidad. Los iones sulfato y magnesio contribuyen con cerca del 8% y 4%, respectivamente.

Esta composición iónica dominante es un factor determinante en fenómenos críticos para la desalinización, como la presión osmótica, la corrosión y la tendencia a la formación de incrustaciones (*scaling*). Además de esta matriz iónica, el agua de mar contiene una fracción de materia en suspensión —que incluyen arena, arcilla, microorganismos y virus— con tamaños característicos que abarcan desde  $5 \times 10^{-2}$  hasta  $0.15 \mu m$ , como se muestra en la Tabla 2.1. Si bien la salinidad absoluta puede variar entre regiones, la constancia de las proporciones relativas permite el desarrollo de tecnologías de desalinización robustas y estandarizadas.

**Tabla 2.1.** Composición típica del agua de mar con una salinidad de 35,000 ppm.

| Ion          | Símbolo Químico | Porcentaje másico (%) | Concentración promedio (ppm) |
|--------------|-----------------|-----------------------|------------------------------|
| Cloruro      | $Cl^-$          | 55.0                  | 19,345                       |
| Sodio        | $Na^+$          | 30.6                  | 10,752                       |
| Sulfato      | $SO_4^{2-}$     | 7.6                   | 2,701                        |
| Magnesio     | $Mg^{++}$       | 3.7                   | 1,295                        |
| Calcio       | $Ca^{++}$       | 1.2                   | 416                          |
| Potasio      | $K^+$           | 1.1                   | 390                          |
| Bicarbonato  | $HCO_3^-$       | 0.4                   | 145                          |
| Bromuro      | $Br^-$          | 0.2                   | 66                           |
| Borato       | $BO_3^{3-}$     | 0.08                  | 27                           |
| Estroncio    | $Sr^{++}$       | 0.04                  | 13                           |
| Fluoruro     | $F^-$           | 0.003                 | 1                            |
| <b>Total</b> |                 | <b>99.9</b>           | <b>≈ 35,000</b>              |

### 2.1.2. Clasificación por contenido de sólidos disueltos totales (SDT)

La salinidad, cuantificada como la concentración de Sólidos Disueltos Totales (SDT), es el parámetro para clasificar los recursos hídricos y definir la estrategia de desalinización requerida. Las aguas se categorizan típicamente como dulces (hasta ~1,500 ppm), salobres (~3,000-10,000 ppm) y agua de mar (>10,000 ppm), como se presenta en la Tabla 2.2 (Cipollina, 2009).

Sin embargo, lejos de ser un valor constante, la salinidad del agua de mar presenta una significativa variabilidad geográfica, determinada por un balance entre la evaporación, la precipitación, el aporte de aguas continentales y la circulación oceánica. Como se observa en la Tabla 2.3 (López Zavala, 2019), los océanos abiertos presentan una salinidad media alrededor de 35,000 ppm, mientras que mares cerrados o semicerrados en regiones áridas exhiben valores muy superiores. Un caso paradigmático es el Golfo Árabe, donde la salinidad cerca de las costas de Kuwait, Arabia Saudita y Emiratos Árabes Unidos puede alcanzar ~50,000 ppm debido a las altas tasas de evaporación y descargas de salmuera (El-Dessouky et al., 2002). En el extremo opuesto, el Mar Báltico puede presentar salinidades cercanas a 10,000 ppm debido al significativo aporte de agua dulce de ríos y deshielo.

**Tabla 2.2.** Clasificación del agua en función al nivel de SDT. Elaborada de: (Cipollina, 2009).

| Tipo de agua             | SDT           | Nota                         |
|--------------------------|---------------|------------------------------|
| Agua dulce               | ~1,500        | Composición química variable |
| Agua salobre             | 1,500–5,000   | Composición química variable |
| Agua muy salobre         | 5,000–15,000  | Composición química variable |
| Agua salina              | 15,000–30,000 | Composición química variable |
| Agua de mar              | 30,000–50,000 | Composición química fija     |
| Estándar del agua de mar | 35,000        | Composición química fija     |

**Tabla 2.3.** Valor medio de las SDT disueltas en varios mares y océanos. Elaborada de: (López Zavala, 2019).

| Océano o Mar      | SDT (ppm) | Océano o Mar                  | SDT (ppm) |
|-------------------|-----------|-------------------------------|-----------|
| Mar Báltico       | 7,000     | Atlántico Norte 2             | 36,200    |
| Mar Caspian       | 13,500    | Océano Índico                 | 33,800    |
| Mar Negro         | 20,000    | Mediterráneo                  | 39,000    |
| Mar Blanco        | 28,000    | Mar Rojo                      | 43,000    |
| Atlántico Norte 1 | 29,000    | Golfo Árabe                   | 50,000    |
| Mar Adriático     | 31,400    | Bahía del tiburón australiano | 70,000    |
| Océano Pacífico   | 33,600    | Golfo de Kara Bogas           | 164,000   |
| Atlántico del Sur | 36,000    | Mar Muerto                    | 220,000   |

Esta variabilidad tiene implicaciones directas en la ingeniería de desalinización. La alta salinidad, como la del Golfo Árabe, impone una mayor presión osmótica y un mayor potencial de incrustación, lo que justifica técnicamente el uso de tecnologías térmicas robustas como MED y MSF, las cuales son menos sensibles a estos factores que las tecnologías de membranas. Comprender el rango de salinidad esperando es, por tanto, el primer paso para el diseño y selección de una tecnología de desalinización eficiente y económicamente viable.

### 2.1.3. Solubilidad de sales y potencial de incrustación ( $\text{CaSO}_4$ )

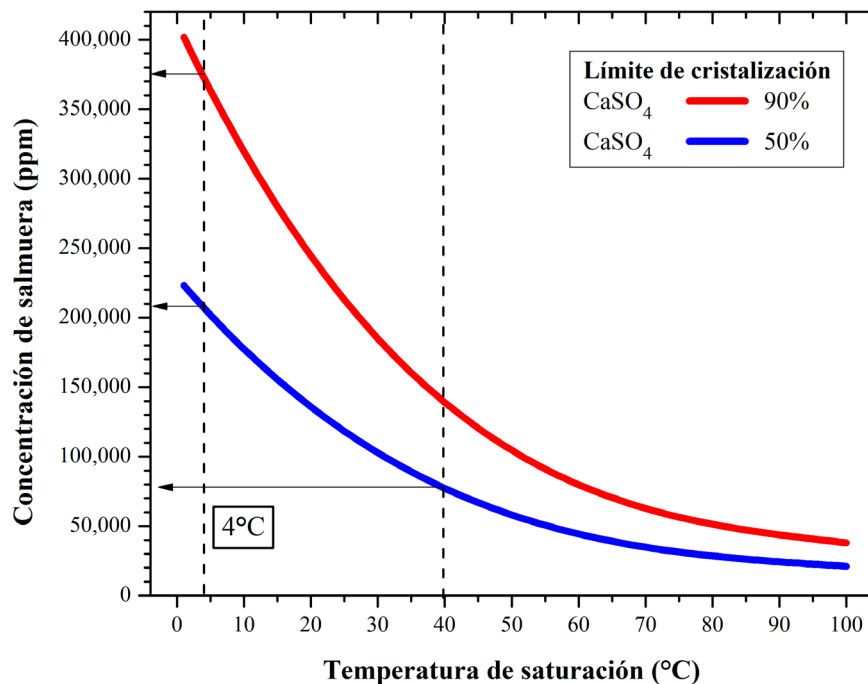
A medida que el agua se concentra en un proceso de desalinización térmica, transformándose en una salmuera con concentraciones superiores a ~50,000 ppm, aumenta significativamente el riesgo de precipitación de sales poco solubles. Este fenómeno, conocido como *scaling* o incrustación, puede reducir la eficiencia de transferencia de calor, obstruir tuberías y dañar equipos. Entre las sales presentes, el sulfato de calcio ( $\text{CaSO}_4$ ) es una de las principales responsables de incrustaciones debido a su solubilidad limitada y su comportamiento inverso: su solubilidad disminuye al aumentar la temperatura, a diferencia de la mayoría de las sales.

Para predecir y evitar este problema, es esencial modelar el punto de saturación. **El-Dessouky et al., (2002)** y posteriormente **Zheng, (2017)**, presentaron una correlación termodinámica (Ecuación 2.1) que define la concentración límite de saturación ( $X_b$ ) del en  $\text{CaSO}_4$  en función de la temperatura de saturación la salmuera ( $T_b$ ):

$$X_b = 0.9(457628.5 - 11304.11T_b + 107.5781T_b^2 - 0.36074T_b^3) \quad (2.1)$$

Esta ecuación permite establecer la frontera operativa segura para un sistema, determinando la concentración máxima permitida antes de que comience la precipitación. La relación se representa gráficamente en la Figura 2.1, la cual muestra cómo la concentración disminuye conforme aumenta la temperatura de saturación, imponiendo un límite termodinámico más restrictivo a elevadas temperaturas de operación.

En los sistemas MED convencionales, que operan con una Temperatura Mínima de Salmuera (BBT) de alrededor de 35–40 °C, este límite de saturación se traduce en una concentración máxima práctica de salmuera de ~70,000 ppm para evitar problemas de corrosión e incrustaciones (**Bamufleh et al., 2017**). Para una operación aún más conservadora y segura, se recomienda no exceder el 50% del valor de saturación (**Azimibavil & Jafarian, 2021**).



**Figura 2.1.** Relación de la salinidad y temperatura de saturación para la solubilidad  $\text{CaSO}_4$ .

Aquí reside una de las ventajas fundamentales de la tecnología de evaporación de múltiple efecto al alto vacío (MED-AV) propuesta en esta tesis. Al operar a una BBT significativamente menor (4 °C), el límite de saturación del  $\text{CaSO}_4$  se desplaza drásticamente hacia concentraciones más altas. Esto permite que el sistema MED-AV opere de manera segura con una concentración de salmuera de hasta 210,000 ppm, un valor muy superior al límite convencional. Esta capacidad no solo se traduce en un incremento en índice de la recuperación de agua (RR), sino que también representa un avance importante hacia el aprovechamiento integral de la salmuera y el concepto de cero descargas líquidas (ZLD).

## 2.2. Fundamentos de la desalinización

La desalinización se define como el proceso de separación de sales minerales del agua de mar o salobre, con el objetivo de obtener un flujo de agua con una calidad apta para el consumo humano, agrícola o industrial. Este proceso se ha erigido como una solución tecnológica para abordar la escasez hídrica en regiones áridas y semiáridas, así como en zonas costeras con estrés poblacional, donde las fuentes convencionales de agua dulce son insuficientes (**Bundschuh et al., 2021; Goh et al., 2016**).

El desafío técnico fundamental de este proceso radica en la alta concentración de sólidos disueltos totales (SDT) que deben ser removidos. Mientras que el agua de mar presenta una salinidad media en el rango de 35,000 ppm, el agua salobre contiene típicamente entre 5,000 y 10,000 ppm de SDT. Ambos valores contrastan drásticamente con el límite máximo de 1,500 ppm establecido por la Organización Mundial de la Salud (OMS) para el agua potable.

### 2.2.1. Definición y clasificación general de procesos

Los métodos de desalinización convencionales se clasifican en dos categorías principales: procesos térmicos y procesos por membranas (**El-agouz et al., 2022; Shalaby et al., 2023**). Los primeros se basan en un cambio de fase, donde el agua salada es calentada para producir vapor que, al condensarse, genera agua dulce destilada y un flujo de salmuera de alta concentración (**Shalaby et al., 2024**). Por otro lado, los procesos por membranas operan mediante la aplicación de una presión externa —superior a la presión osmótica del agua de alimentación— para forzar el paso selectivo de moléculas de agua a través de una barrera semipermeable, reteniendo los iones de sal en una corriente concentrada de salmuera. En contraste con los métodos térmicos

convencionales, esta categoría de tecnologías se caracteriza por un menor consumo energético, ya que evita el cambio de fase; una mayor compactibilidad de sus unidades, y una alta modularidad que facilita el escalado de las plantas desalinizadoras (Alenezi & Alabaiadly, 2025).

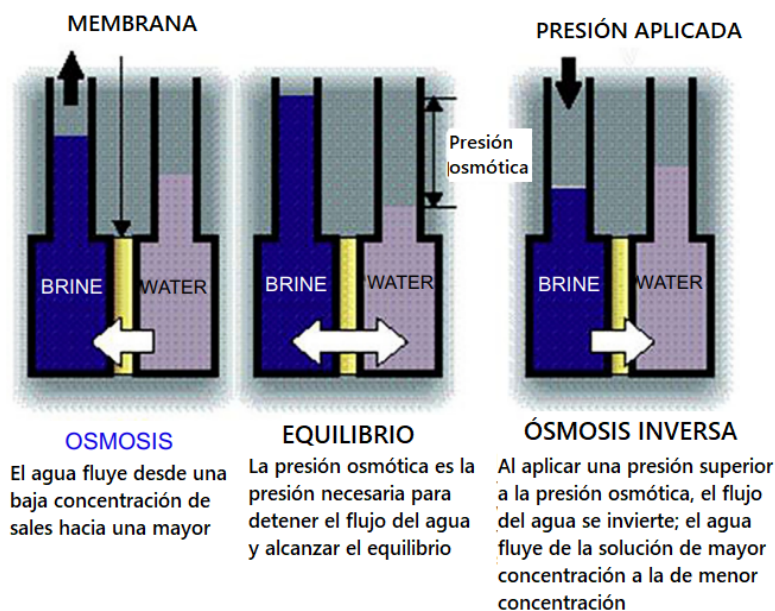
### 2.2.2. Procesos por Membranas

La desalinización por membranas emergió como una tecnología comercial a principios de la década de 1970, considerablemente más tarde que los procesos térmicos. No obstante, su crecimiento ha sido exponencial, llegando a dominar el escenario actual, el mercado global de la desalinización tanto en capacidad instalada total como en número de unidades (Jalihal & Venkatesan, 2019). Los procesos de membrana se clasifican según la fuerza impulsora que gobierna la separación: a) Procesos impulsados por presión y b) procesos impulsados por potencial eléctrico. En la primera clasificación, se aplica una presión externa para favorecer el transporte selectivo de solvente (agua) a través de la membrana. La tecnología representativa de este grupo es la Ósmosis Inversa (OI). En el caso de la segunda clasificación, un campo eléctrico facilita el transporte selectivo de solutos (iones) a través de membranas con carga. La tecnología principal en esta categoría es la Electrodialisis (ED).

#### 2.2.2.1. Ósmosis Inversa (OI)

La Ósmosis Inversa (OI) es la tecnología más predominante en el mercado global de la desalinización (Shalaby et al., 2024). Su principio de operación como se muestra en la Figura 2.2., se basa en la aplicación de una presión superior a la presión osmótica del agua de alimentación, forzando el paso selectivo de moléculas de agua a través de una membrana semipermeable, mientras retiene la mayoría de los iones disueltos. Aunque la Ósmosis Directa (OD) es otro proceso existente (Aboelmaaref et al., 2023), su aplicación comercial es más limitada y especializada en comparación con la OI.

La OI ofrece ventajas decisivas, entre las que destacan un consumo de energía significativamente menor en comparación con los métodos térmicos, ya que no requiere un cambio de fase del agua. Los sistemas modernos incorporan dispositivos de recuperación de energía que mejoran aún más su eficiencia energética. Sin embargo, la tecnología presenta desafíos inherentes, como la necesidad de un pretratamiento exhaustivo del agua de alimentación para minimizar el ensuciamiento (*fouling*) y la incrustación (*scaling*) de las membranas (Alhawsawi et al., 2023).



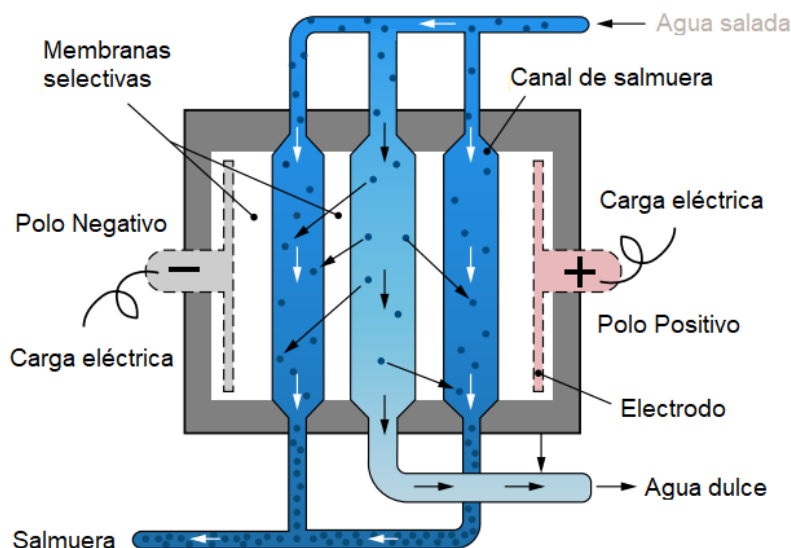
**Figura 2.2.** Concepto de equilibrio osmótico y ósmosis inversa. Elaborado de: (Jalihal & Venkatesan, 2019).

### 2.2.2.2. Electrodialisis (ED)

La Electrodialisis (ED) es un proceso electroquímico de desalinización que utiliza una diferencia de potencial eléctrico como fuerza impulsora para separar iones a través de membranas con carga selectiva, transportándolos desde una solución de alimentación hacia una de concentrado (Jalihal & Venkatesan, 2019). Aunque el costo del agua producto mediante ED es relativamente alto, resulta competitivo al ser inferior al de los procesos de destilación tradicionales de capacidad equivalente.

Esta tecnología, utilizada desde hace décadas, posee un amplio historial de aplicaciones que incluye: la producción de agua potable y de proceso a partir de agua salobre y de mar, el tratamiento de efluentes industriales, la recuperación de subproductos valiosos e, incluso la producción de sal (El-agouz et al., 2022). Como se ilustra en la Figura 2.3., una unidad de ED típica opera con una alta tasa de recuperación, capaz de eliminar entre el 75% y el 98% de los Sólidos Disueltos Totales (SDT). Su variante la Electrodialisis Inversa (EDR), incrementa esta eficiencia —alcanzando hasta un 94% de recuperación— gracias a la circulación interna del agua de alimentación y a la inversión periódica de la polaridad eléctrica, lo que ayuda a mitigar el ensuciamiento de las membranas.



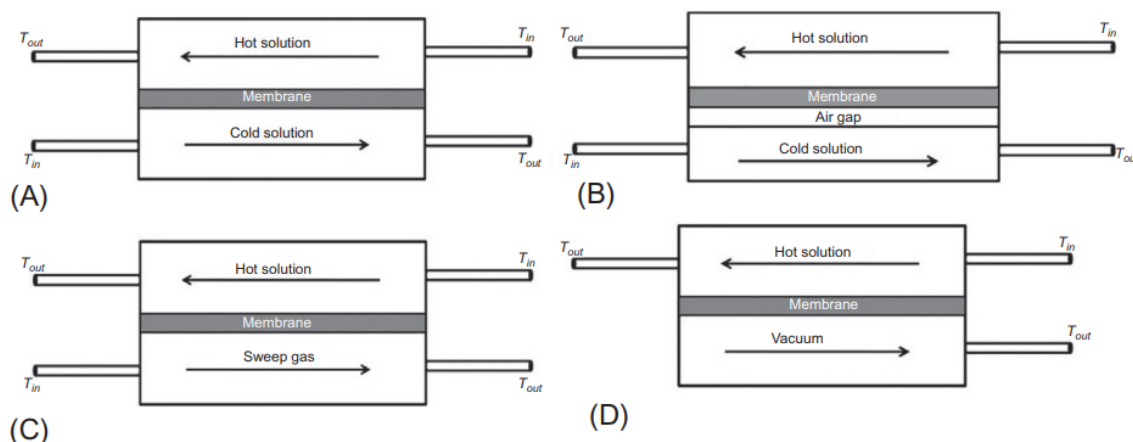


**Figura 2.3.** Diagrama esquemático de la ED. Elaborado de: (Al-Karaghoulí & Kazmerski, 2013).

Entre sus ventajas operativas destacan su capacidad para eliminar una amplia gama de contaminantes iónicos, su baja sensibilidad a variaciones de pH y dureza en el agua de alimentación, su adaptabilidad a distintos parámetros operativos, y sus bajos requerimientos de mano de obra y costos de mantenimiento. No obstante, una limitación es que el costo de tratamiento está directamente correlacionado con la concentración de SDT. Por ello, la ED/EDR, resulta más económica para el tratamiento de agua salobres con SDT inferiores a 4,000 mg/L, perdiendo competitividad frente a tecnologías como la Ósmosis Inversa (OI) a salinidades más elevadas (Younos & Tulou, 2005).

### 2.2.2.3. Destilación de membranas (MD)

La Destilación por Membrana (MD) se define como un proceso de separación híbrido que combina principios de los procesos térmicos y de membranas (Shalaby et al., 2024). Su característica fundamental reside en su fuerza impulsora: un gradiente de presión de vapor a lo largo de la membrana. Este principio la distingue de otras tecnologías de membrana, que dependen de una diferencia de presión absoluta aplicada (como la Ósmosis Inversa), un gradiente de concentración (como la Ósmosis Directa) o un potencial eléctrico (como la Electrodialisis). Su principio de operación permite teóricamente alcanzar recuperaciones mucho más elevada y manejar salmuera de alta concentración (Jalihal & Venkatesan, 2019).



**Figura 2.4.** Esquema simplificado de distintas configuraciones de destilación por membranas. (A) Destilación por Membrana de Contacto Directo, (B) Destilación por Membrana con Espacio de Aire, (C) Destilación por Membrana con Barrido de Gas, (D) Destilación por Membrana al Vacío. (Jalihal & Venkatesan, 2019)

La Destilación por membra se implementa en distintas configuraciones, definidas por la estrategia para generar y recolectar el vapor en el lado del permeado. La Figura 2.4., muestra las principales configuraciones: a) Destilación por Membrana de Contacto Directo (DCMD), donde el permeado se condensa en contacto directo con una corriente líquida fría; b) Destilación por Membrana con Espacio de Aire (AGMD), en la que un espacio de aire estancado separa la membrana del condensador; c) Destilación por Membrana con Barrido de Gas (SGMD), donde un gas inerte arrastra el vapor para su posterior condensación externa y; d) Destilación por Membrana al Vacío (VMD), en esta configuración se aplica vacío en el lado del permeado para aumentar el gradiente de presión de vapor.

### 2.2.3. Procesos Térmicos: principios básicos y ventajas comparativas

La desalinización térmica consiste en la generación de vapor de agua a partir de una solución salina (agua de mar) mediante la aplicación de energía térmica. Este objetivo se logra mediante dos principios fundamentales: la reducción de la presión ambiental o el suministro continuo de calor, dando lugar a los procesos conocidos como “*flashing*” (evaporación instantánea) y destilación, respectivamente (Jalihal & Venkatesan, 2019). Las tecnologías térmicas más utilizadas a nivel industrial son la Destilación Multietapa Flash (MSF), la Destilación/Evaporación

de Múltiple efecto (MED) y la Comprensión de Vapor térmica (TVC) y mecánica (MVC) (Shalaby et al., 2023).

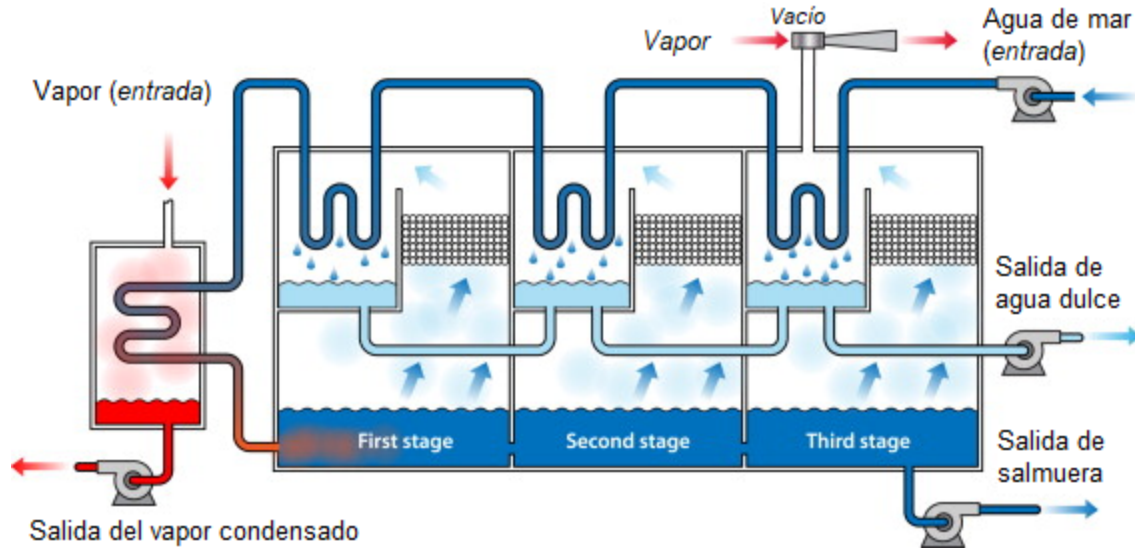
### 2.2.3.1. Flash Multietapa (MSF)

La destilación por Evaporación Instantánea Multietapa (MSF) es un proceso térmico que demanda un alto consumo de energía, tanto térmica como eléctrica. La energía térmica se suministra principalmente como vapor de baja presión (1-3 bares) para calentar la salmuera de alimentación, y como vapor a media presión para accionar los eyectores que mantienen el vacío en las diferentes secciones de la unidad. Por su parte, la energía eléctrica se utiliza para el accionamiento de las bombas de recirculación, producto destilado, extracción de salmuera y dosificación de productos químicos (Al-Karaghoulí & Kazmerski, 2013).

Estas plantas, con capacidades típicas entre 10,000 y 35,000 m<sup>3</sup>/d, constan de una serie de etapas (entre 4 y 40) operando a temperaturas y presiones decrecientes. El principio de operación se basa en la evaporación instantánea (“*flashing*”): el agua de mar precalentada ingresa a la primera etapa, donde se le suministra calor externo —proveniente de fuentes como calderas de combustibles fósiles, calor residual de centrales eléctricas, reactores nucleares o energías renovables— para elevar su temperatura a un rango de 90 °C a 110 °C. Esta salmuera caliente se introduce entonces a una cámara a baja presión, provocando la vaporización instantánea de una parte de la corriente de alimentación.

El vapor generado en cada etapa se condensa en los intercambiadores de calor —que a su vez precalientan el agua de alimentación— y se recoge como agua dulce o agua producto. La salmuera restante, ahora a una temperatura ligeramente inferior, fluye a la siguiente etapa, que se mantiene a una presión aún más baja, repitiéndose el proceso “*flashing*” sucesivamente hasta la descarga final (Jalihal & Venkatesan, 2019).

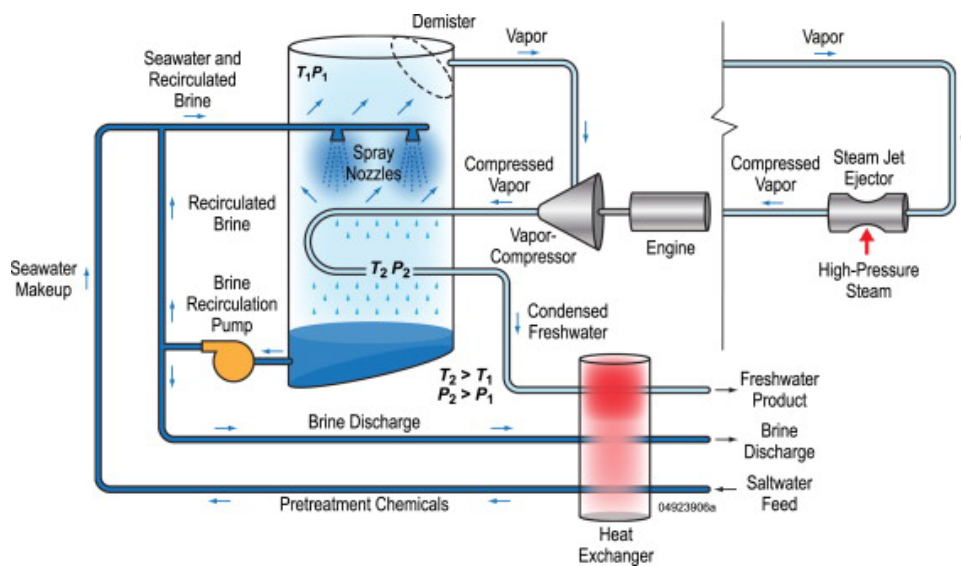
A pesar de su amplia implementación, que la posiciona como el segundo proceso de desalinización más utilizado a nivel global después de la Ósmosis Inversa (OI), la tecnología MSF presenta desafíos operativos significativos. La evaporación instantánea promueve la formación de incrustaciones y depósitos en componentes y tuberías, lo que exige limpieza y mantenimiento periódicos. La Figura 2.5., ilustra un diagrama esquemático de una unidad MSF típica.



**Figura 2.5.** Diagrama esquemático de la unidad MSF. Elaborado de: (Al-Karaghoulí & Kazmerski, 2013).

### 2.2.3.2. Compresión mecánica de vapor (VC)

Las plantas de Compresión de Vapor (VC) son sistemas de desalinización térmica que utilizan el calor de condensación del vapor, previamente comprimido, como fuente de energía principal para evaporar el agua salada. Esta tecnología se implementa principalmente mediante dos configuraciones: Compresión Mecánica de Vapor (MVC) y Compresión Térmica de Vapor (TVC).



**Figura 2.6.** Diagrama esquemático de las unidades VC (MVC y TVC), (Jalihal & Venkatesan, 2019).

El proceso inicia cuando el agua de alimentación, precalentada en un intercambiador de calor, ingresa al evaporador. Allí, el vapor generado es comprimido. En la MVC, un compresor mecánico accionado eléctricamente eleva la presión y temperatura del vapor. Por el contrario, en la TVC, se utiliza un eyector impulsado por un chorro de vapor a alta presión (o “vapor motriz”) para comprimir el vapor. En ambos casos, la compresión incrementa la entalpía del vapor, permitiendo que este sirva como medio de calentamiento al condensarse en los tubos del evaporador, cerrando así el ciclo térmico de manera altamente eficiente.

Posteriormente, la salmuera concentrada es extraída del evaporador mediante una bomba. Este flujo se divide: una porción se recircula para mezclarse con la alimentación nueva y mejorar la eficiencia térmica, mientras que el resto se descarga como corriente de rechazo.

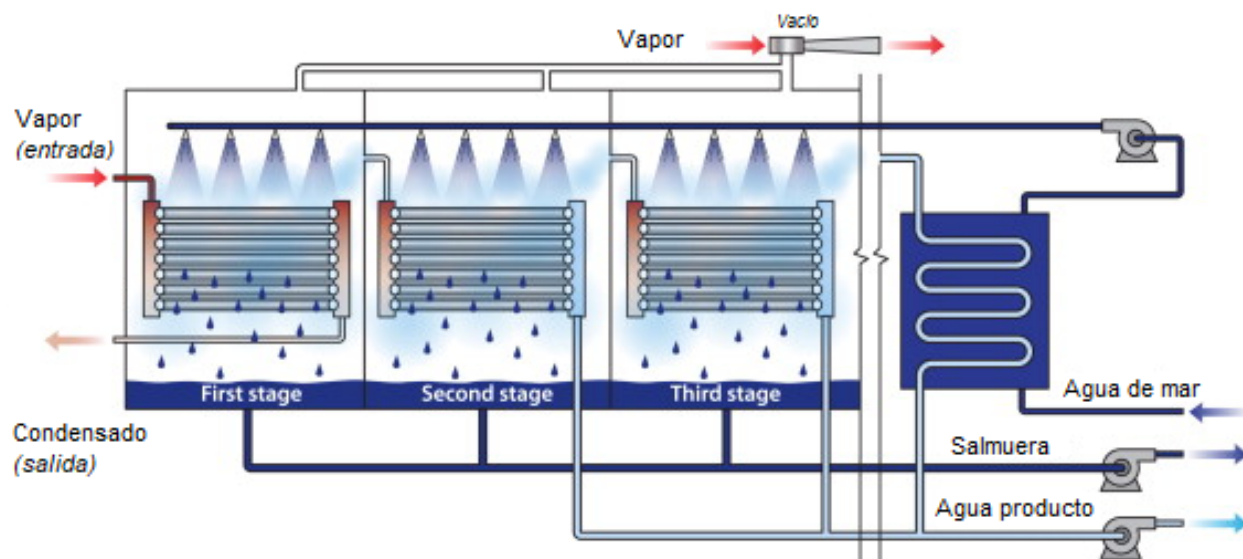
Estas tecnologías se destinan generalmente a aplicaciones de pequeña y mediana escala, con capacidades típicas que oscilan entre 100 y 3,000 m<sup>3</sup>/d para MVC, y entre 10,000 y 30,000 m<sup>3</sup>/d para TVC (Al-Karaghoulí & Kazmerski, 2013). La Figura 2.6., presenta un diagrama esquemático de ambos sistemas (MVC y TVC).

## **2.2.4 La tecnología de Evaporación de Múltiple Efecto (MED)**

### **2.2.4.1. Principio de funcionamiento**

El proceso de Destilación o Evaporación de Múltiple Efecto (MED) opera mediante una serie de evaporadores o “efectos” (generalmente entre 4 y 16) dispuestos en cascada y mantenidos a presiones decrecientes. El principio de funcionamiento se basa en la reutilización del calor latente de condensación del vapor en cada etapa sucesiva. El calor externo —proveniente de calderas, calor residual de centrales eléctricas, energía solar u otras fuentes— se suministra únicamente en el primer efecto, elevando la temperatura de la salmuera hasta aproximadamente 70 °C. El vapor generado en esa primera etapa fluye hacia el siguiente efecto, que se encuentra a una presión menor, y actúa como fuente de calor para evaporar una nueva porción de agua de mar. Este proceso se repite en serie a lo largo de todos los efectos, lo que permite una alta eficiencia térmica global (Alenezi & Alabaiadly, 2025). La Figura 2.7., muestra un diagrama de una unidad MED.

Las unidades MED, con capacidades que suelen oscilar entre 600 y 30,000 m<sup>3</sup>/d, se diseñan principalmente en dos configuraciones:



**Figura 2.7.** Diagrama esquemático de la unidad MED. Elaborado de: (Al-Karaghoulí & Kazmerski, 2013).

a) Tubos Verticales (VTE), donde el agua de mar hierve formando una película delgada en el interior de los tubos, mientras el vapor se condensa en el exterior de los mismos y; b) Tubos Horizontales (HTE), en esta configuración el agua de mar se pulveriza sobre el exterior de los tubos, y el vapor fluye por dentro de ellos, donde se condensa para producir el agua dulce.

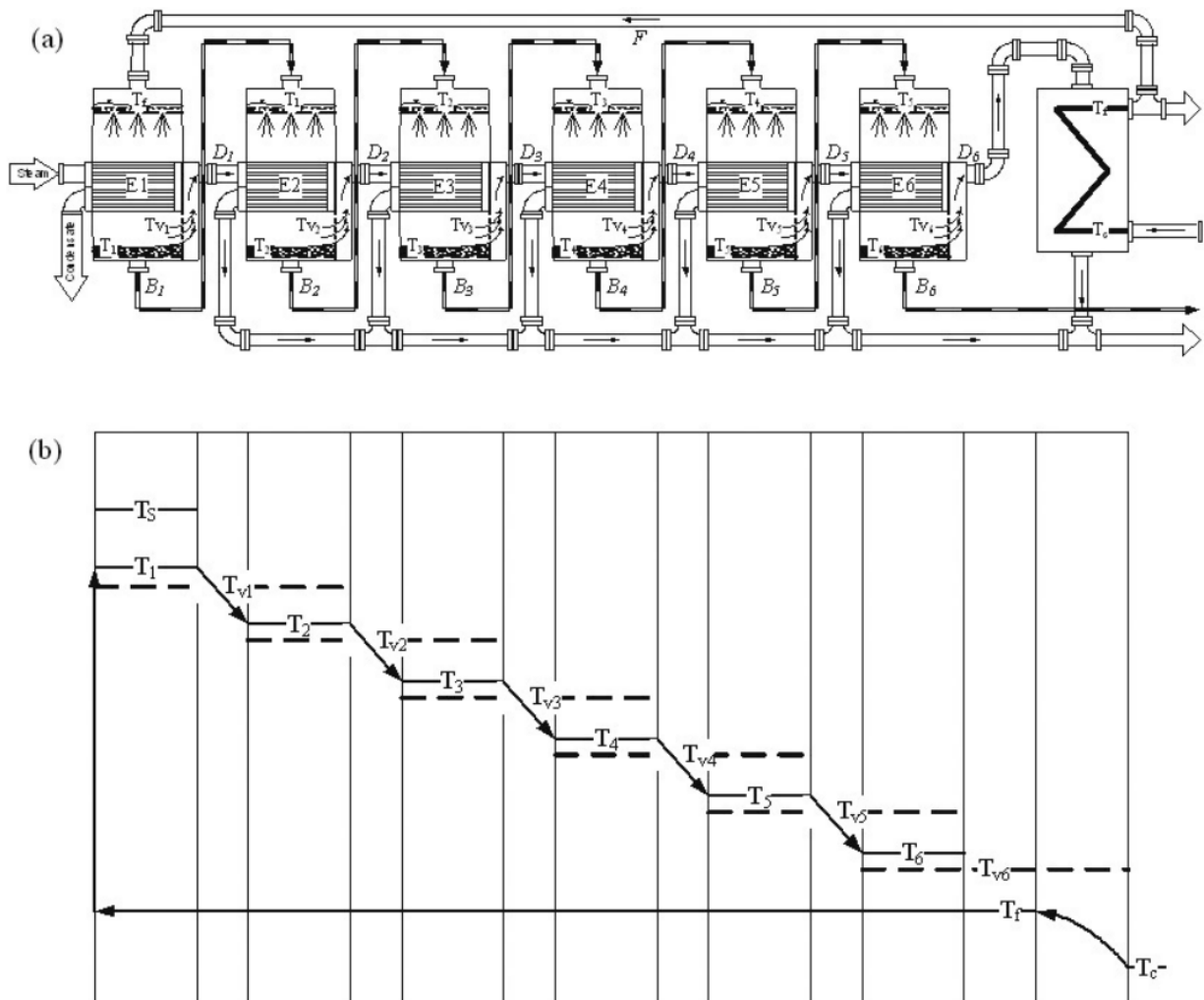
Si bien las primeras plantas de desalinización utilizaban MED, fue desplazada por las tecnologías de Evaporación Instantánea Multietapa (MSF) durante décadas debido a los menos costos de inversión inicial (CAPEX) de esta última y a su menor tendencia a las incrustaciones (“*scaling*”). No obstante, en los últimos años, el interés por el proceso MED se ha renovado debido a su mayor eficiencia energética, lo que le está permitiendo recuperar una creciente cuota de en el mercado.

#### 2.2.4.2. Configuraciones de flujo de alimentación en sistemas MED

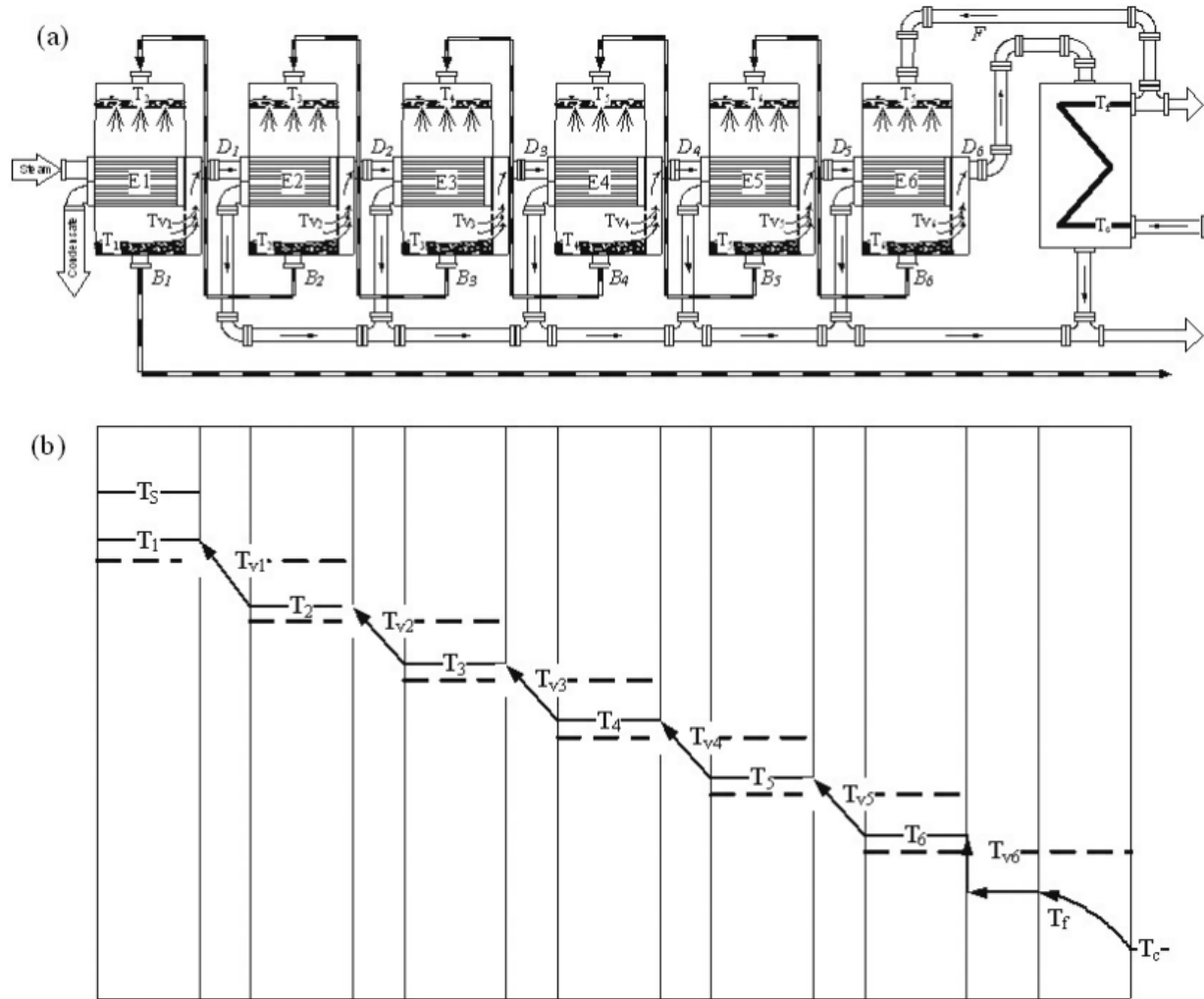
El diseño de una planta de Destilación/Evaporación de Múltiple Efecto (MED) está determinado por el esquema del agua de alimentación a través de los efectos. Existen cuatro configuraciones principales: a) Alimentación Directa (MED-FF, del inglés *Forward Feed*), b) Alimentación Inversa (MED-BF, del inglés *Backward Feed*), c) Alimentación en Paralelo (MED-PF, del inglés *Parallel Feed*) y d) Alimentación Directa con Precalentadores (MED-FFH).

En la configuración **MED-FF**, el agua de alimentación, precalentada en el condensador final, se introduce en el primer efecto —el de mayor temperatura— y fluye sucesivamente hacia los efectos subsiguientes. Esta configuración presenta un flujo en **co-corriente**, dado que el vapor de calentamiento y la salmuera circulan en el mismo sentido, como se ilustra en la Figura 2.8.

La configuración **MED-BF**, presenta un esquema de alimentación de **contracorriente**, donde el flujo de alimentación es opuesto al del vapor. El agua de mar ingresa por el último efecto —el de menor temperatura— y avanza hacia los efectos de mayor temperatura, mientras que la salmuera concentrada se descarga desde el primer efecto. Este arreglo se presenta en la Figura 2.9.



**Figura 2.8.** (a) Configuración MED-FF; (b) perfil de temperatura a través de los efectos (Darwish & Abdulrahim, 2008).

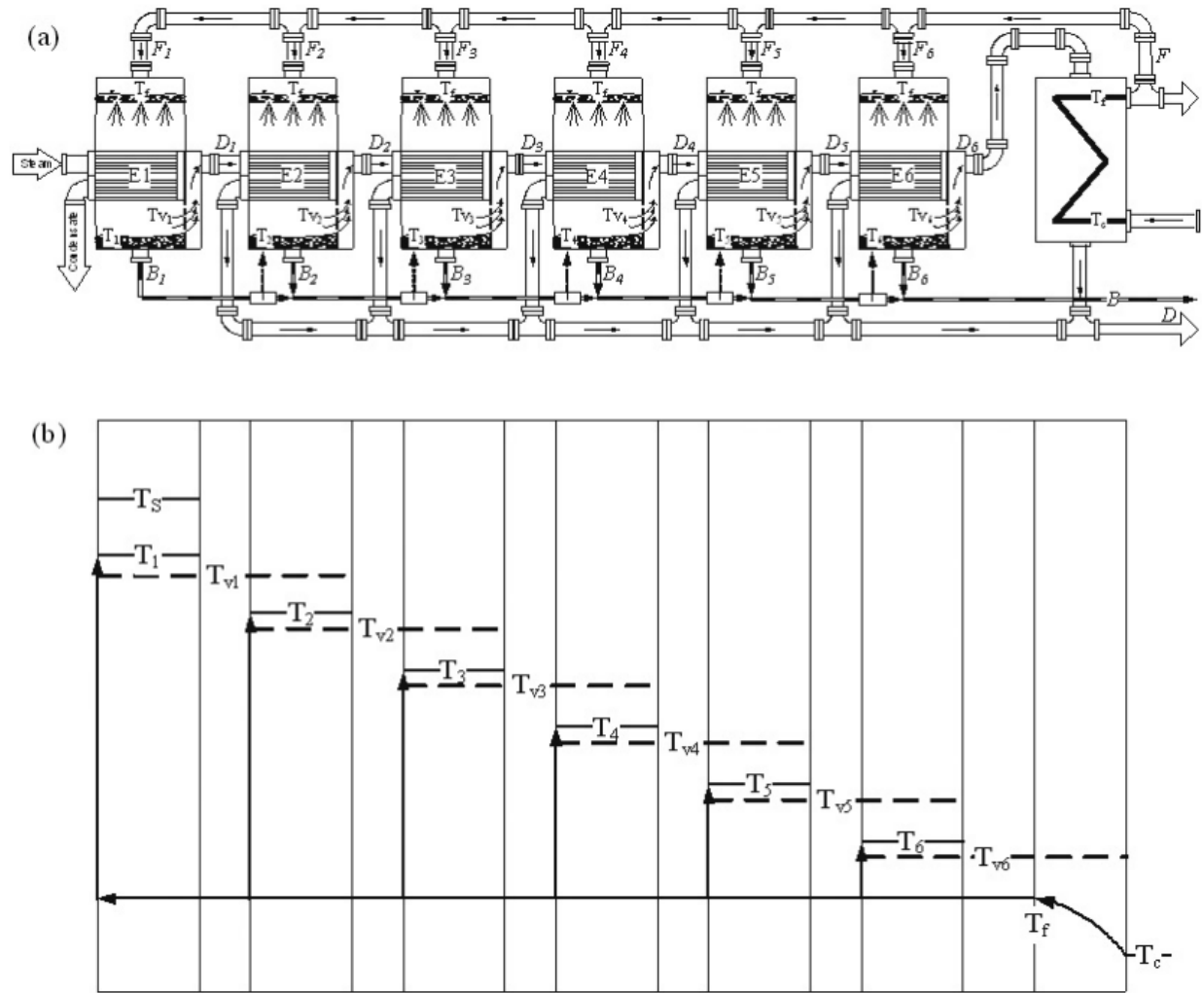


**Figura 2.9.** (a) Configuración MED-BF; (b) perfil de temperatura a través de los efectos (Darwish & Abdulrahim, 2008).

En la configuración **MED-PF**, el caudal de alimentación que sale del condensador se divide en *streams* independientes que se distribuyen de manera simultánea y equitativa a cada uno de los efectos. A este tipo de alimentación también se le denomina **flujo cruzado**, como se muestra en la Figura 2.10.

Finalmente, la configuración **MED-FFH** es una variante de la alimentación directa que incorpora una serie de precalentadores. Parte del agua de enfriamiento que sale del condensador final, una vez pretratada, se convierte en alimentación y circula a través de estos precalentadores, incrementando su temperatura de manera escalonada antes de ingresar al primer efecto, como se representa en la Figura 2.11.





**Figura 2.10.** (a) Configuración MED-PF; (b) perfil de temperatura a través de los efectos (Darwish & Abdulrahim, 2008).

La selección de una configuración específica impacta críticamente en el rendimiento global del sistema, influyendo directamente en parámetros clave como la disposición de los evaporadores, las áreas de transferencia de calor requeridas, la distribución de la generación de vapor, el consumo de energía de bombeo y la Relación de Producción (GOR).

Para cerrar esta sección, y en línea con los hallazgos del presente estudio, se concluye que la configuración MED-PF y la MED-FFH, son las más eficientes y confiables, según los resultados reportados por Sharaf et al., (2011). La configuración MED-BF no es favorable debido al gradiente de salinidad elevado que se genera en el primer efecto, el cual opera a la temperatura más alta (TBT).



### 2.2.4.3. Parámetros clave de desempeño: GOR, RR, SEC, GOR<sub>G</sub> y SEC<sub>G</sub>

Para evaluar el efecto de la desalinización en un sistema de desalinización MED, se utilizan dos indicadores, la Relación de Ganancia —*Gain Output Ratio*— (**GOR**) y la Tasa de Recuperación —*Recovery Ratio*— (**RR**). El GOR, representa una relación entre el calor latente del agua producida y la energía térmica adicionada al sistema, donde  $\dot{m}_{PW}$  representa el agua producto total del sistema en kg/s, que se multiplica por el calor latente a una presión específica en kJ/kg. Mientras que,  $Q_{GEN}$  representa el calor de activación adicionado al sistema en kW. El indicador RR, es una relación entre la cantidad de agua producto y el agua de mar suministrada a la MED ( $\dot{m}_{SW}$ ). Asimismo, para evaluar el requerimiento energético del sistema respecto a la producción de agua desalinizada, se utiliza el indicador Consumo Específico de Energía —*Specific Energy Consumption*— (**SEC**). Las ecuaciones para los indicadores descritos son:

$$GOR = \frac{((\dot{m}_{AP} * 2,326 \text{ kJ / kg}))}{Q_{GEN}} \quad (2.2)$$

$$RR = \frac{\dot{m}_{AP}}{\dot{m}_{AM}} \quad (2.3)$$

$$SEC = \frac{\text{Energía eléctrica (kWh)}}{\text{Agua producto total (m}^3\text{)}} \quad (2.4)$$

Sin embargo, debido a las características del sistema MED al alto vacío (MED-AV) que utiliza un subsistema de eyecto-condensación barométrico, el cual requiere un significativo consumo de energía eléctrica, a diferencia de un sistema MED convencional. Se propone el uso de los indicadores Gain Output Ratio Global (**GOR<sub>G</sub>**) y Specific Energy Consumption Global (**SEC<sub>G</sub>**):

$$GOR_G = \frac{((\dot{m}_{AP} * 2,326 \text{ kJ / kg}))}{(Q_{GEN} + W_{BOMBAS\_TOTAL})} \quad (2.5)$$

$$SEC_G = \frac{\text{Energía térmica} + \text{Energía Eléctrica (kWh)}}{\text{Agua producto total } [\dot{m}_{AP}] \text{ (m}^3\text{)}} \quad (2.6)$$

# CAPÍTULO 3

*“Si he logrado ver más lejos,  
ha sido porque he subido a  
hombros de gigantes”  
—Isaac Newton*

## PUBLICACIONES CIENTÍFICAS DEL PROYECTO DOCTORAL

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Este capítulo presenta el núcleo de la contribución de esta investigación doctoral, al integrar las publicaciones científicas y solicitudes de patente que dan cuerpo y evidencia concreta al desarrollo y los hallazgos del proyecto. A continuación, se presenta una síntesis de cada trabajo, destacando su objetivo, metodología y contribución particular al cumplimiento del objetivo general. Tras cada resumen, se incluye el documento completo de la publicación o solicitud de patente correspondiente, dando evidencia completa del trabajo realizado.

### **3.1. Artículo A: High-vacuum multiple effect desalination system with barometric ejector condensation.**

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En este trabajo se presenta un estudio operativo para demostrar la factibilidad técnica de un novedoso sistema de desalinización de evaporación de múltiple efecto (MED) al alto vacío con eyectocondensación barométrica, diseñado para ser activado con energía solar térmica. El sistema propuesto opera con una cascada de presión que desciende a niveles de alto vacío (28.3 - 0.8 kPa), alcanzando en el último efecto una temperatura de saturación (BBT) de 4 °C. La innovación central reside en la implementación de un eyector que succiona, arrastra y comprime el vapor frío del último efecto de la MED para condensarlo, incrementando la capacidad y eficiencia global del sistema. Al llevar a cabo el estudio se encontró que:

- El sistema MED-AV obtuvo un GOR (Gained Output Ratio) de 13.5, lo que representa un incremento del 34.1% respecto a un sistema MED convencional de igual capacidad, atribuible a una reducción del 25.4% en la energía de activación.
- La configuración propuesta del sistema MED-AV reduce el número de componentes, al eliminar la necesidad de precalentadores en los últimos cinco efectos. Esto es posible porque se elimina el gradiente de temperatura entre el agua de mar de alimentación (25 °C) y la temperatura del vapor generado en dichos efectos.
- El sistema propuesto mostró un  $GOR_G$  (Gained Output Ratio Global) de 11.2 y un RR (Recovery Ratio) de 0.83, lo que equivale a una mejora del 11.7% y del 120% respectivamente, en comparación con un sistema MED convencional de la misma capacidad y número de efectos.

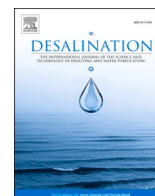
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## **Artículo A**

### **High-vacuum multiple effect desalination system with barometric ejector condensation**

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## High vacuum multiple effect desalination system with barometric ejector condensation

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### HIGHLIGHTS

- A new MED system configuration that operates at high-vacuum;
- A barometric ejector-condensation process is proposed to condense the vapor at 4 °C;
- New operating conditions, which increase the efficiency of the system, are set;
- The maximum sea-water pre-heating level is reached using less pre-heaters;
- A variable-flow pump is used in order to ensure the proper functioning of ejector.

### ARTICLE INFO

#### Keywords:

Barometric condensation  
Desalination  
Ejector  
High vacuum  
MED

### ABSTRACT

In order to search for new desalination technologies to meet the growing water needs, this paper presents a parametric study of a novel, high-vacuum multiple-effect desalination system with barometric ejector-condensation activated with solar thermal energy, to demonstrate its technical feasibility. Aspen Plus simulation software was used for the parametric study, which was validated with the results reported in the literature. The high-vacuum multiple-effect distillation system has 14 effects, Top Brine Temperature of 68 °C and produces 96,033.6 L/d of water. The increase in the range of the pressure cascade (28.6–0.8 kPa), as well as of the temperature (68–4 °C), increases the capacity and efficiency of the desalination system. This is possible due to the implementation of an ejector that pumps, drags, and compresses vapor at 4 °C from the last effect of the system in order to condense it. In comparison to a conventional multiple-effect distillation, the Gain Output Ratio increases in 34.1 % because 25.4 % less energy from the generator is consumed. Furthermore, the system has a Recovery Ratio of 0.83, which corresponds approximately to a 120 % increase. Hence, the proposed desalination system significantly increases the potentiality of multiple-effect distillation systems.

### 1. Introduction

Currently, several communities throughout the world face severe survival challenges due to hydric stress, as the shortage of drinking water becomes greater and more generalized [2,24,34,45,46,58]. Nevertheless, in spite of this severe shortage, seawater is abundant, making up to 97 % of the planet's water [31]. This is the reason why the international scientific community is working on the development of

new technologies, or in the enhancement of existing ones, in order to make desalination a technically and economically feasible option [50].

Desalination emerges, and has been recognized, as one of the most promising alternatives for safe and reliable drinking water supply [14,27]. There are several processes to desalinate seawater; among them, Reverse Osmosis (RO), Multistage Flash distillation, and Multiple-Effect distillation/evaporation, with 70 %, 17 %, and 6 % worldwide installed capacity, respectively, adding up to 93 % of the total world installed capacity [51]. Thermal desalination technologies, such as MSF

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**Nomenclature***Symbols and abbreviations*

|                   |                                                          |
|-------------------|----------------------------------------------------------|
| P                 | Pressure (kPa)                                           |
| T                 | Temperature (°C)                                         |
| X                 | Salt concentration (ppm)                                 |
| $\dot{m}$         | Mass flow (kg/s)                                         |
| VF                | Vapor fraction (–)                                       |
| Q                 | Heat transferred (kW)                                    |
| W                 | Net Work (kW)                                            |
| Ex                | Exergy flow rate (kW)                                    |
| Ex <sub>D</sub>   | Exergy destruction (kW)                                  |
| PR                | Performance Ratio (–)                                    |
| GOR               | Gain Output Ratio (–)                                    |
| RR                | Recovery Ratio (–)                                       |
| COP               | Coefficient of Performance (–)                           |
| SEC               | Specific energy consumption (kWh/m <sup>3</sup> )        |
| GOR <sub>G</sub>  | Gain Output Ratio global (–)                             |
| SEC <sub>G</sub>  | Specific energy consumption global (kWh/m <sup>3</sup> ) |
| Eff               | Efficiency (–)                                           |
| H                 | Height (m)                                               |
| CaSO <sub>4</sub> | Calcium sulfate                                          |
| $\eta_{ex}$       | Exergy efficiency                                        |
| TBT               | Top Brine Temperature (°C)                               |
| BBT               | Bottom Brine Temperature (°C)                            |
| MED               | Multiple effect distillation                             |
| MSF               | Multi-stage flash distillation                           |
| MED-HV            | High vacuum multiple effect distillation                 |
| PWP               | Total pump power                                         |
| $\omega$          | Ejector entrainment ratio                                |

|                   |                                                                 |
|-------------------|-----------------------------------------------------------------|
| RO                | Reverse osmosis                                                 |
| EJC               | Ejector-condenser                                               |
| EV                | Expansion Valve                                                 |
| PH                | Preheaters                                                      |
| HX                | Heat exchanger (evaporator)                                     |
| GEN               | Generator                                                       |
| BC                | Barometric column                                               |
| P <sub>SW</sub>   | Seawater pump                                                   |
| P <sub>PW</sub>   | Product water pump                                              |
| P <sub>CW</sub>   | Cooling seawater pump                                           |
| S <sub>B</sub>    | Brine Separator                                                 |
| S <sub>PW</sub>   | Product water separator                                         |
| x                 | Quality (–)                                                     |
| LS                | Lower section                                                   |
| US                | Upper section                                                   |
| IAPWS             | International Association for the Properties of Water and Steam |
| mno               | Motive nozzle output                                            |
| mcco              | Mixing chamber output                                           |
| U-T <sub>B</sub>  | U Tube Brine                                                    |
| U-T <sub>PW</sub> | U Tube Product Water                                            |

*Subscripts*

|    |                   |
|----|-------------------|
| MF | Motive fluid      |
| SF | Secondary fluid   |
| DC | Ejector discharge |
| c  | Calculated        |
| r  | Required          |
| f  | Final             |

or MED, are predominant in regions with a water type similar to that of the Persian Gulf (high concentration of total dissolved solids and proliferation of algae), where drinking water shortage overlaps with abundant seawater and high levels of solar radiation [48]. MED and MSF systems are robust for operation with water with high levels of salinity and a recurrent proliferation of algae. These factors are very detrimental for technologies such as RO, since they affect the performance of their membranes and forces them to have expensive pre-treatment systems [28,51]. Furthermore, thermal desalination processes, like the MED systems, require less pre-treatment of the feed water and maintenance, and consume less primary electricity than processes that operate with membranes [1,28,33]. However, one of the main disadvantages of MED processes is that they have a higher energy consumption compared to RO processes. The RO process requires approximately 25 % of the total energy consumed by a MED process [30]. Nevertheless, the replacement and pre-treatment procedures of the RO membrane are expensive, which increases RO's total cost [13].

In the last decades, the efforts of the scientific community have focused on developing theoretical and experimental work to improve MED technology and achieve greater techno-economic viability. To a certain degree, the operation conditions have improved, increasing the efficiency and capacity of the systems, as well as reducing the initial investment, and operational costs. [6,7,17,19,21,40,47,48,57]. Another way in which MED has been tried to improve is through the proposal of hybridization, and through the coupling with other cooling and electrical energy generation technologies. This significantly reduces operating costs and allows the production of multiple services, or polygeneration [42].

Trying to improve energetic efficiency of the MED process, the theoretical works of [3,15] propose the coupling of an MED desalination system to an absorption cooling unit. In this hybridization, the cooling power of the absorption machine's evaporator is used to condense the

saturated vapor produced by the last effect of the MED system. In this manner, [15] were able to reduce effluent energy loss in the condenser, and to achieve a better PR value and desalination capacity within the system. However, even though the coupling presents an improvement in the efficiency of the desalination system, a large number of components is required, which increases initial investment as well as maintenance cost, reducing, therefore, the techno-economic feasibility of these kind of systems.

The improvement of MED technology has been limited by the challenges posed by the effluent energy losses in the condenser, as well as the operational limits presented by the Top Brine Temperature (TBT) and the Bottom Brine Temperature (BBT) of the system [1]. The TBT limit depends directly on the corrosion problems that inherent to operating with seawater (35,000 to 50,000 ppm) at temperatures >70 °C [28]. The BBT is restricted by the temperature of the seawater (25–30 °C) used to condense the vapor of the last effect of the MED, requiring a minimum temperature gradient ( $\approx 5$  °C) between the vapor and the seawater [8,56]. Furthermore, different exergetic studies found in the literature [25,36,39,41,43,60], show that in conventional MED systems the greatest exergy loss occurs within the condenser. This is due to the fact that, in the vapor condensation process of the last effect of the MED, too much latent heat is transferred to the seawater, and a large amount of such water is rejected to the sea [29].

In an attempt to increase the technical and economic feasibility of these MED technologies, Aly [8], was one of the pioneers in the study of new configurations and the operative exploration of new pressure and temperature ranges, using a TBT of 63 °C and a BBT of 6 °C –considerably lower than the one used in conventional systems. This was achieved due to hybridization with a 220 kW LiBr-H<sub>2</sub>O absorption cooling system. This allowed to set a 20-effects cascade and to achieve an increase in desalination indicators (GOR of 14.8 and a PR of 14.2). The coupling of technologies consisted in using, for activation, the



withdrawn heat of the absorber and the condenser of the absorption system. Furthermore, the cooling of the evaporator was used to condense the low temperature vapor ( $6^{\circ}\text{C}$ ) of the last effect of the MED. However, this technology coupling does not consider the production of cold; moreover, it poses a greater system complexity and a large number of components, which implies a higher initial investment as well as high operation and maintenance costs.

There have been recent studies that explore new pressure/temperature operation ranges for the MED [4,23,26]. Among these, it's worth noting the works of [55,56], who studied the temperature ranges of  $50$  to  $3^{\circ}\text{C}$  and  $70$  to  $5^{\circ}\text{C}$ . These authors report a water production 2 to 3 times larger than that of a conventional MED system. However, in order to achieve this, it was necessary to couple the MED system with an adsorption (AD) system that uses silica gel, using the latter to adsorb the low temperature vapor from the last effect of the MED. The main disadvantage of this proposal is its high initial investment due to the large number of components of the process. Similarly, López-Zavala et al. [38], presented the study of a novel MED desalination system that uses produce water to generate cooling through sudden expansion and condensation of cold vapor due to the pressure rise caused by a novel barometric ejector-condenser. The authors report a desalination capacity of  $74,736\text{ L/d}$ , a production of  $2043\text{ kW}$  ( $581\text{ tons}$ ) of cooling, and COP, PR, and RR values of  $4.7$ ,  $11.96$  and  $0.53$ , respectively, with a COP  $6.53$  times greater than that of a single-effect absorption cooling system. However, in spite of achieving a considerable increase in cooling and efficiency, the system presents a desalination capability similar to the capability formerly reported in the literature (PR of  $11.96$ ). This is because the system uses the same number of effects and operation limits that conventional MED desalination systems use ( $70\text{--}35^{\circ}\text{C}$ ).

Based on the last-mentioned study, condensation ejectors are presented as an alternative to condense the water vapor coming from the last effect of the MED and also offers the possibility to break the operational limits set by the conventional BBT and increase its techno-economic viability. Condensation ejectors (CE) are a special type of ejectors that involve a biphasic flow (liquid-vapor) in which the great speed difference between the motive current (primary) and the pump current (secondary), as well as the intimate interaction between them, produces a significant movement, mass, and heat transfer, which provokes the condensation of the vapor phase, resulting in a liquid phase current at the end of the ejector. The phase change process is boosted both by the turbulent mix, and by the heat transfer between phases [16]. However, the review of the state of the art of ejectors shows that in the majority of the studies where mathematical models were proposed to understand the interactions between the primary and secondary fluids inside the ejector and to achieve the optimization of the ejectors, only ejectors without condensation -which predominantly use primary and secondary fluids in vapor phase- were considered [18,20,32,37,44,49,52,53].

Nevertheless, it is possible to find some studies on ejector-condensers; among them, the research conducted by the Massachusetts Institute of Technology (MIT) Levy [35] and the theoretical work by Wang et al., [59]. These works analyze the liquid-vapor interaction that occurs within a biphasic condenser-ejector through one-dimensional (1D) models, as well as the behavior of the performance of an ejector using a parametric study.

Based on the issues and areas of opportunity presented above, and having as goals the decrease of effluent energy losses in the condensers and breaking the operative limits set by the conventional BBT -using seawater as a cooling medium-, this work presents a parametric study to prove the technical feasibility of a novel high-vacuum multiple effect evaporation system with barometric ejector-condensation. Due to the configuration and operating conditions proposed, a high internal energetic integration is achieved. The process uses a high-vacuum pressure cascade, which reaches the pressure that corresponds to a saturation temperature (BBT) of  $4^{\circ}\text{C}$  in the last effect of the system. An ejector pumps, drives, and compresses in order to condense the low temperature

vapor of the last effect of the MED. This has a direct impact on the water desalination capacity, since the effects that operate at a temperature below the seawater temperature, having no pre-heaters, use all the vapor generated in the posterior MED effects. Furthermore, the final condenser used in conventional systems is replaced by the ejector-condenser. This reduces the exergy losses that occur when the latent heat of the condensation is transferred to the water that returns to the sea. Compared to the implementation of an absorption or adsorption cooling system in order to obtain high-vacuum presented in previous studies, the proposed technology consists of less components and a greater desalination capacity and efficiency.

## 2. System description

Fig. 1 shows the diagram of the high-vacuum desalination system, which is composed of three subsystems: 1) the MED desalination subsystem; 2) the produce water extraction, condensation, and adaptation subsystem; and, 3) the solar thermal energy collection, storage, and supply subsystem. It is worth noting that the operational study of the last subsystem is beyond the scope of this work.

Fig. 1 shows that sea water pump ( $P_{SW}$ ) pumps seawater flow (1) and drives it to the 14-effects MED desalination subsystem. Seawater flow (2) is then carried to the pre-heaters (PH) in order to receive the total amount of the latent heat of the vapor produced by the flashing of the brine and produce water within the effects. The vapor comes from the brine ( $S_B$ ) and produce water ( $S_{PW}$ ) separators. The vapor from  $S_B$  is a compound of the flashed vapor and the vapor generated by heating within the effect. This occurs from effect 8 to the second effect of the MED, while in the first effect seawater (5) is heated with a percentage of the vapor produced by boiling in the generator (GEN). This way, the consumption of external heating services within the generator is minimal. It is important to note that, due to the temperature profiles, the pre-heating of seawater starts at effect 8 (PH1), and finishes at the first effect (PH8). At the exit of pre-heater PH8, seawater current (6) goes through an expansion valve (EV1), in order to suddenly decrease its pressure until it reaches the pressure that corresponds to the MED's TBT. Under these conditions, sub-cooled seawater (7) enters the GEN and receives thermal energy from the heat carrier fluid (13) that comes from the solar collector field. In the GEN, current (7) receives energy in the form of sensible heat and latent heat, and it exits in two phases as current (15). Then, it enters the liquid-vapor separator ( $S_{B1}$ ), splitting into the brine (21) and water vapor (16) currents. The brine current (21) is carried to the evaporator of the MED's second effect (HX2), and it enters through a U-shaped pipe ( $U - T_{B1}$ ). This allows the formation of a hydraulic seal in order to keep the pressure difference between effects 1 and 2. It is important to mention that brine current (21) is flashed as it goes from pressure P1 to pressure P2 ( $P1 > P2$ ). The current is now in two phases and has a lower temperature (current 22), and, under these conditions, enters HX2 in order to be heated by the liquid-vapor current (17).

On the other hand, water vapor (16) is carried to pre-heater PH8 and it yields a percentage of its latent heat in order to pre-heat seawater current (5). This causes current (17) to leave PH8 in two phases. Then, current (17) enters the HX2 and exchanges heat with the brine current (22), totally condensing and exiting as produce water in current (18). The brine current (23) that exits the HX2 is in two phases. This occurs because when using current (17), vapor is produced in the brine current (22). Meanwhile, the produce water (current 18) from the first effect, once it goes through the U-shaped tube ( $U - T_{PW1}$ ) and is flashed, joins the produce water from the following effects. The two-phased brine current (23) is carried to the brine liquid-vapor separator ( $S_{B2}$ ), where it is split into the vapor (26) and brine (24) currents. Vapor current (26) joins the current of vapor generated by the flashing of produce water (27) that comes from the produce water liquid-vapor separator ( $S_{PW1}$ ), becoming current (28). This current enters pre-heater PH7 and yields a percentage of its latent heat (condensation of the vapor generated by flashing), finishing in two phases (current 29). Then, as well as in the

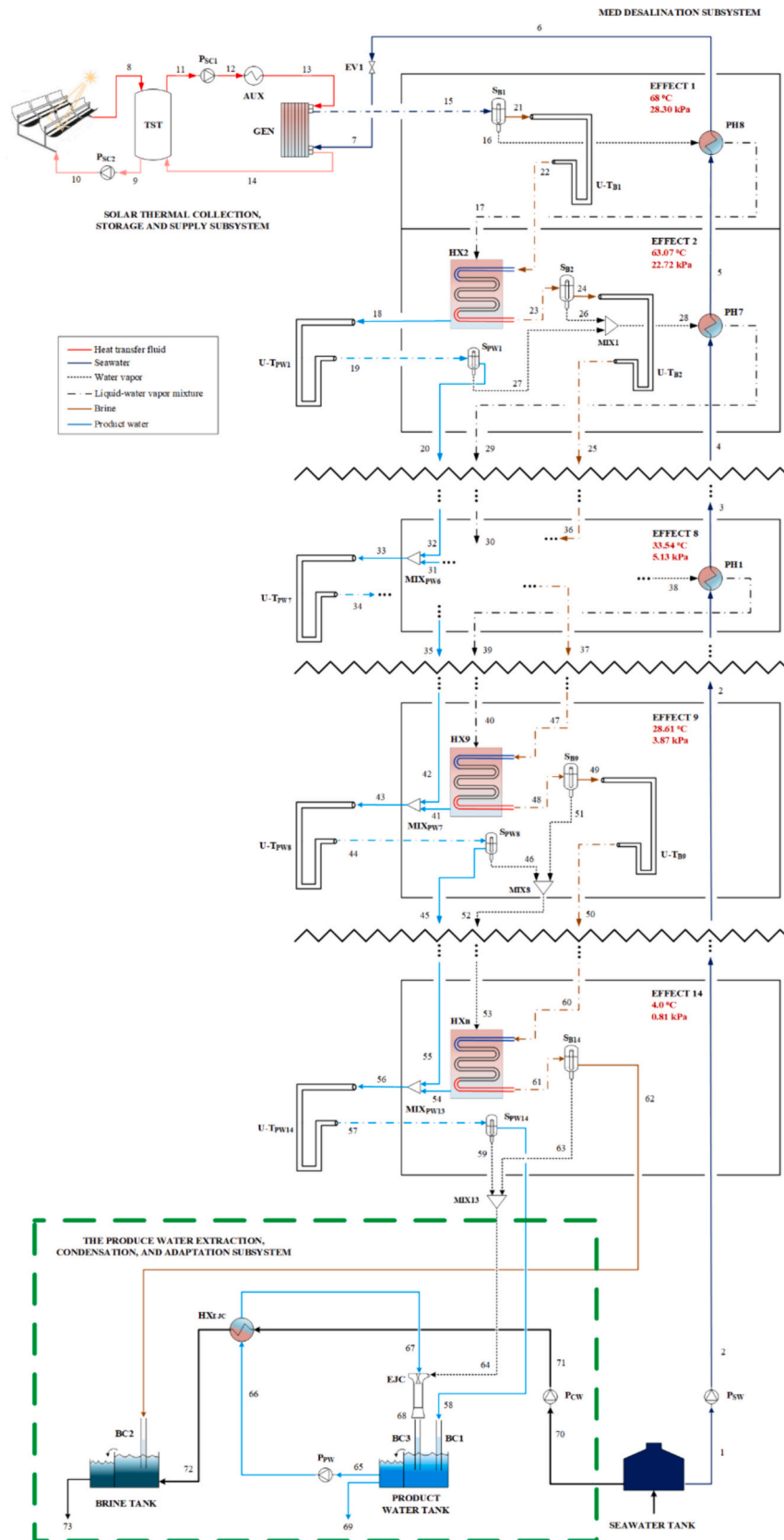


Fig. 1. Diagram of the high-vacuum multiple effect desalination through evaporation system.

second effect, the liquid-vapor current (29) enters the third-effect's evaporator (HX3). Meanwhile, the brine current (24) is carried through the U-shaped tube ( $U - T_{B2}$ ) and then goes to evaporator HX3, where it acquires the pressure of the third effect (P3). What happened in effect 2 also happens in the following effects, up to effect 8. From effect 9 to effect 13 something similar occurs, only that in that section there are no pre-heaters, so all the vapor generated by heating and flashing is used in the following effect.

The produce water from each effect joins the water from the subsequent effect by means of U-shaped tubes. Therefore, a hydraulic seal is formed and the pressure differential between effects is held. This process repeats itself up until the last effect (Effect 14). The total amount of product water obtained from the system's effects (current 58) is carried through a barometric column (BC1) to the produce water extraction and adaptation subsystem. Meanwhile, the brine at the exit of the MED's last effect (current 62) is carried to the brine container through the barometric column (BC2), where it will be diluted by mixing itself with the cooling seawater current (72) before returning either to the sea or to another exploitation process. Within the product water extraction and conditioning subsystem, the water vapor from the MED's last effect (current 64), which has low pressure and temperature, is pumped, carried, and compressed in order to cause its condensation within the ejector-condenser (EJC), which uses the product water current (67) as motive fluid. Once condensed, but still at vacuum pressure, the vapor mixed with the motive fluid, exits as current (68) and enters into the barometric column (BC3), from which it exits at the bottom as product water at atmospheric pressure (69).

The ejector-condenser (EJC) is a special type of ejector that involves vapor condensation. Therefore, it presents thermal-energy and momentum transfer through the interaction between the vapor and liquid currents. This was studied using ASPEN Plus software as a simulation tool. Fig. 2 shows that current (67), when going through the nozzle of the ejector-condenser, expands until it reaches the pressure of point (67a), which is lower than the pressure of the vapor of the MED's last effect (64). The pressure and kinetic energy differential between these two fluids causes the secondary fluid (vapor current 64) to be first suctioned into the discharge nozzle, and then to the ejector's mixing zone. Within the mixing chamber, currents (64) and (67) mix and a portion of the remaining kinetic energy from the driver fluid becomes a rise in pressure (point 67b). Afterwards, the two-phase current (67b) enters the diffuser where, due to the pressure increase (caused by the velocity reduction when the cross-sectional area is increased), the vapor finishes its condensation, exiting as a liquid current (68). The ejector's discharge pressure lies between the pressures of currents (64) and (67).

Inside the ejector (mixing and diffuser zone), the motive fluid cur-

rent (67) receives the condensation heat from vapor current (64), resulting in an increase of the fluid's temperature at the ejector's exit (68). This current (68), condensed and with a greater temperature, is directed towards a barometric column (BC3) in order to create a hydraulic seal and to maintain a vacuum within the system. Afterwards, the fluids comes out of the bottom of the BC3 at atmospheric pressure, and is placed in the water produce container, from which currents (65) and (69) emerge. Pump  $P_{PW}$  pulls current (65) in order to once again, activate the ejector. Product water current (66) enters the heat exchanger  $HX_{EJC}$ , where the seawater current (71) removes the condensation heat that it received from vapor current (64). Hence, produce water current (67) keeps the same temperature at the ejector's entrance. Meanwhile, seawater current (72), after cooling current (66), is taken to the brine container in order to dilute brine current (62), which exits the MED's last effect. With the ejector-condenser the MED is taken to a vacuum level greater than that of ordinary systems because this equipment, having a pressure increase effect, condenses low temperature vapor.

The main characteristics that differentiates the MED-HV system from a conventional one are the following: the proposed system uses a pressure cascade that decreases the pressure to high-vacuum levels (28.3–0.8 kPa). The way in which the last-effect's vapor is condensed changes: instead of condensing vapor current (64) with the seawater input, it is done through an ejector-condensation process. The pressure of the last effect (0.8 kPa) corresponds to a saturation temperature of 4 °C.

### 3. Methodology

For the parametric study to prove the technical feasibility of the proposed system, Aspen Plus was used; the study was validated with the results found in the literature. Using a parametric study, the design conditions, which are a compromise between the capacity and efficiency of the system, were set.

#### 3.1. System simulation

ASPEN PLUS v10 is a software simulation tool used for the design, parametric study, and optimization of different processes within the chemical industry. ASPEN PLUS comes pre-loaded with a large amount of previously-validated devices and components, which facilitate its operation and use. Furthermore, it is capable of computing the thermodynamic properties of pure and mixed fluids. Fig. 3 depicts the flowchart for the methodology used for the simulation with Aspen Plus.

The main components used in the system, with their schematic

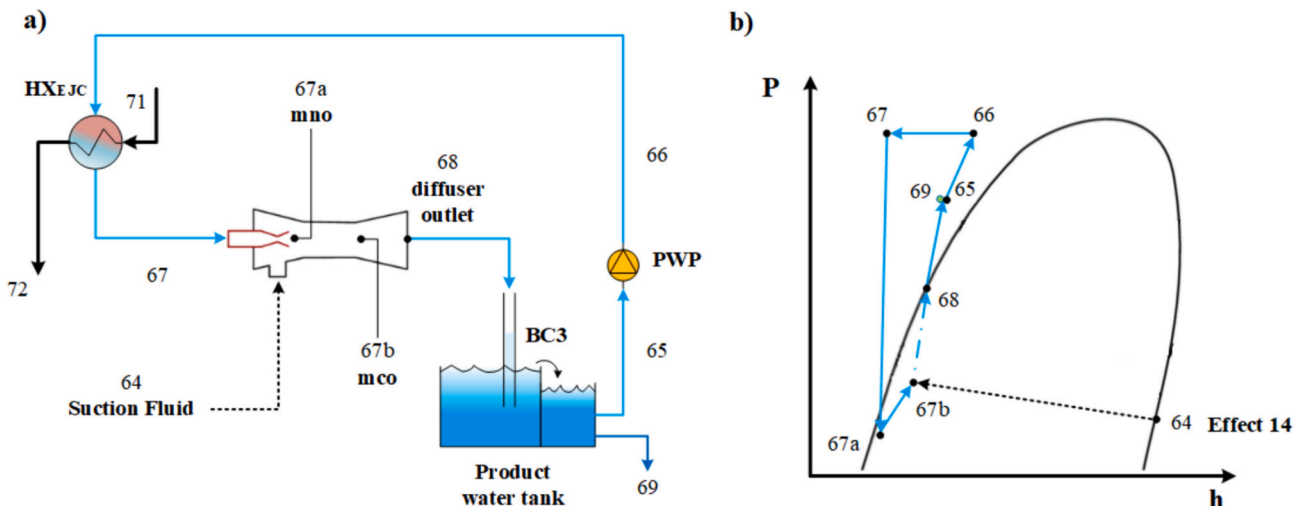


Fig. 2. P-h diagram of the MED-HV system's barometric ejector-condenser.

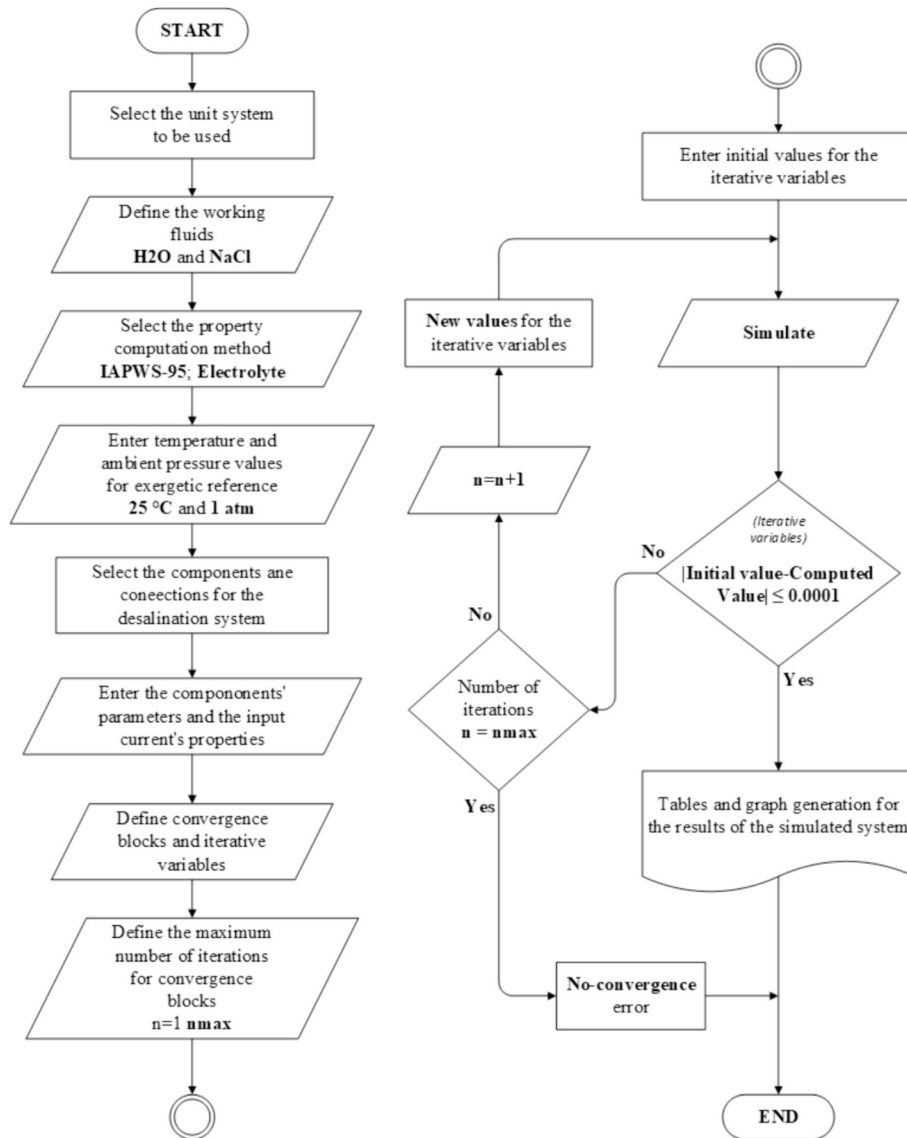


Fig. 3. Flowchart for the process' solution algorithm in Aspen Plus.

representation, the currents involved and the type of working fluid, as well as the energy balance equations governing each component, are presented in Table 1. In order to clarify the simulation process, Table 2 shows the simulation blocks established in Aspen Plus, as well as their connections, to build the different components of the system –Effect 1, Effects 2 to 8, Effects 9 to 13, Effect 14, and Ejector-condenser. It also shows the total simulation model, in order to represent and simulate the MED-HV Forward Flow desalination system.

To simulate in Aspen Plus blocks that operate with seawater and produce water as working fluids, Electrolyte and IAPWS-95 libraries, respectively, were used. In the specific case of the simulation of Effect 1, a 4-currents heat exchanger (HeatX block) was used to represent the generator. However, it is necessary to select the Electrolyte properties method for the cold side (where the seawater flows), while for the hot side (where the heat carrier fluid flows) IAPWS-95 should be used. The rest of the components used for Effect 1, except for PH1, use the Electrolyte properties method.

### 3.2. Parameters for the proposed system

In order to evaluate the operational behavior of the MED-HV system, the flow of the seawater input current, as well as the heat added to the

generator was varied. The simulation used a temperature and pressure cascade in the effects of 68–4 °C and 28.3–0.8 kPa, respectively, as well as concentration and seawater temperatures of 35,000 ppm and 25 °C, respectively. For all the heat exchangers, the  $\Delta T_{min}$  approach for the hot and cold currents is of 2.5 °C. The rest of the parameters are shown in Table 3.

When using Aspen Plus for simulations, if there are recirculation currents within the system, it is critical to define and initialize the iterative variables for the convergence blocks. These variables define the iterative computation process and, therefore, the final solution for each simulation of the parametric study. Specifically, for the high-vacuum MED desalination process, the iterative variables are the following:  $T_6 = 65.5$  °C,  $T_{67} = 28.5$  °C y  $VF_{68} = 0$ .

### 3.3. Procedure for the definition of the design conditions of the ejector-condenser

The methodology for defining the design conditions of the ejector-condenser in Aspen Plus is presented in the flow diagram in Fig. 4. This ensures that the design conditions of the ejector-condenser correspond to conditions where total condensation of the secondary steam is achieved, is within the thermodynamic feasibility zone, and meets the

**Table 1**  
Main components of the MED-HV system.

| Component                      | Schematic view | Stream |                        | Energy balance                                              |
|--------------------------------|----------------|--------|------------------------|-------------------------------------------------------------|
|                                |                | No.    | Fluid                  |                                                             |
| Generator (GEN)                |                | 13     | Heat carrier fluid in  | $m_{13}h_{13} + m_7h_7 = m_{14}h_{14} + m_{15}h_{15}$       |
|                                |                | 15     | Heat carrier fluid out |                                                             |
|                                |                | 7      | Feed                   |                                                             |
|                                |                | 14     | Seawater in            |                                                             |
|                                |                | 15     | Two-phase seawater out |                                                             |
| Evaporator 2 (HX2)             |                | 17     | Water condensate in    | $m_{17}h_{17} + m_{22}h_{22} = m_{18}h_{18} + m_{23}h_{23}$ |
|                                |                | 22     | Water condensate out   |                                                             |
|                                |                | 23     | Two-phase Brine in     |                                                             |
|                                |                | 18     | Two-phase Brine out    |                                                             |
|                                |                |        | Two-phase water in     |                                                             |
| Evaporator n (HXn)             |                | 53     | Water condensate in    | $m_{53}h_{53} + m_{60}h_{60} = m_{54}h_{54} + m_{61}h_{61}$ |
|                                |                | 60     | Water condensate out   |                                                             |
|                                |                | 61     | Two-phase Brine in     |                                                             |
|                                |                | 54     | Two-phase Brine out    |                                                             |
|                                |                |        | Two-phase water in     |                                                             |
| Preheater 8 (PH8)              |                | 5      | Feed seawater in       | $m_5h_5 + m_{16}h_{16} = m_6h_6 + m_{17}h_{17}$             |
|                                |                | 6      | Feed seawater out      |                                                             |
|                                |                | 16     | Water vapor in         |                                                             |
|                                |                | 17     | Two-phase water out    |                                                             |
|                                |                |        | Feed seawater in       |                                                             |
| Preheater 7 (PH7)              |                | 4      | Feed seawater in       | $m_4h_4 + m_{28}h_{28} = m_5h_5 + m_{29}h_{29}$             |
|                                |                | 5      | Feed seawater out      |                                                             |
|                                |                | 28     | Water vapor in         |                                                             |
|                                |                | 29     | Two-phase water out    |                                                             |
|                                |                |        | Feed seawater in       |                                                             |
| Brine Separator (BS1)          |                | 15     | Feed seawater in       | $m_{15}h_{15} = m_{16}h_{16} + m_{21}h_{21}$                |
|                                |                | 16     | Feed seawater out      |                                                             |
|                                |                | 21     | Water vapor in         |                                                             |
|                                |                |        | Water vapor out        |                                                             |
| Product Water Separator (SPW1) |                | 19     | Two-phase water in     | $m_{19}h_{19} = m_{20}h_{20} + m_{27}h_{27}$                |
|                                |                | 20     | Liquid water out       |                                                             |
|                                |                | 27     | Water vapor out        |                                                             |
|                                |                |        | Water vapor in         |                                                             |
| Ejector (EJC)                  |                | 64     | Liquid water in        | $m_{64}h_{64} + m_{67}h_{67} = m_{68}h_{68}$                |
|                                |                | 67     | Liquid water in        |                                                             |
|                                |                | 68     | Water condensate out   |                                                             |
|                                |                |        | Water vapor in         |                                                             |
| Seawater Pump (SWP)            |                | 1      | Feed seawater in       | $m_1h_1 = m_2h_2$                                           |
|                                |                | 2      | Feed seawater out      |                                                             |
| Product water Pump (PWP)       |                | 65     | Feed seawater in       | $m_{65}h_{65} = m_{66}h_{66}$                               |
|                                |                | 66     | Feed seawater out      |                                                             |

**Table 1 (continued)**

| Component                | Schematic view | Stream |                   | Energy balance                |
|--------------------------|----------------|--------|-------------------|-------------------------------|
|                          |                | No.    | Fluid             |                               |
| Cooling water Pump (CWP) |                | 70     | Feed seawater in  | $m_{70}h_{70} = m_{71}h_{71}$ |
|                          |                | 71     | Feed seawater out |                               |

required energy efficiency with the minimum possible pump power. The design conditions of the ejector-condenser of the MED-HV system will be presented in the results section.

### 3.4. Basic assumptions

In order to simulate the operative behavior of the high-vacuum MED system, the following assumptions were considered:

- The high-vacuum MED desalination system operates at a steady state;
- An exergetic reference temperature and ambient pressure of 25 °C and 1 atm, respectively;
- The system uses a Top Brine Temperature (TBT) of 68 °C, a Bottom Brine Temperature (BBT) of 4 °C, and a  $\Delta T$  between effects of 4.93 °C;
- A minimal approach temperature between currents ( $\Delta T_{min}$ ) of 2.5 °C is used for all heat exchangers;
- The maximum possible pre-heating temperature for the seawater input is  $T_6 = 65.5$  °C;
- The mass flow and the temperature for the heat carrier fluid, which comes from the solar collectors, are  $\dot{m}_{13} = 2.5$  kg/s and 90 °C, respectively;
- The efficiency of the hydraulic pumps ( $P_{SW}$ ,  $P_{PW}$  y  $P_{CW}$ ) if of 85 %;
- The pressure drop within the devices and pipelines is negligible.

### 3.5. Thermodynamic evaluation criteria

The thermodynamic equations that govern the processes of the proposed system, based on the first law of thermodynamics, are contained in the following expression:

$$\Delta E = Q_{neto} - W_{neto} \quad (1)$$

Using the energy conservation principle in each of the system's components, we get the equations shown in Table 6.

Two indicators are used for the evaluation of the desalination effect: the Gain Output Ratio (GOR), and the Recovery Ratio (RR). The GOR represents a relation between the latent heat of the product water and the thermal energy added to the system.  $\dot{m}_{PW}$  is the total product water of the system in kg/s, which is multiplied by the latent heat at a specific pressure in kJ/kg.  $Q_{GEN}$  is the activation heat added to the system in kW. RR is a relation between the amount of product water and the seawater supplied to the MED ( $\dot{m}_{SW}$ ). Furthermore, the Specific Energy Consumption (SEC) indicator is used to evaluate the energetic requirement of the system with respect to the desalinated water production. The equations for these indicators are the following:

$$GOR = \frac{(\dot{m}_{PW} \cdot 2,326 \text{ kJ/kg})}{Q_{GEN}} \quad (2)$$

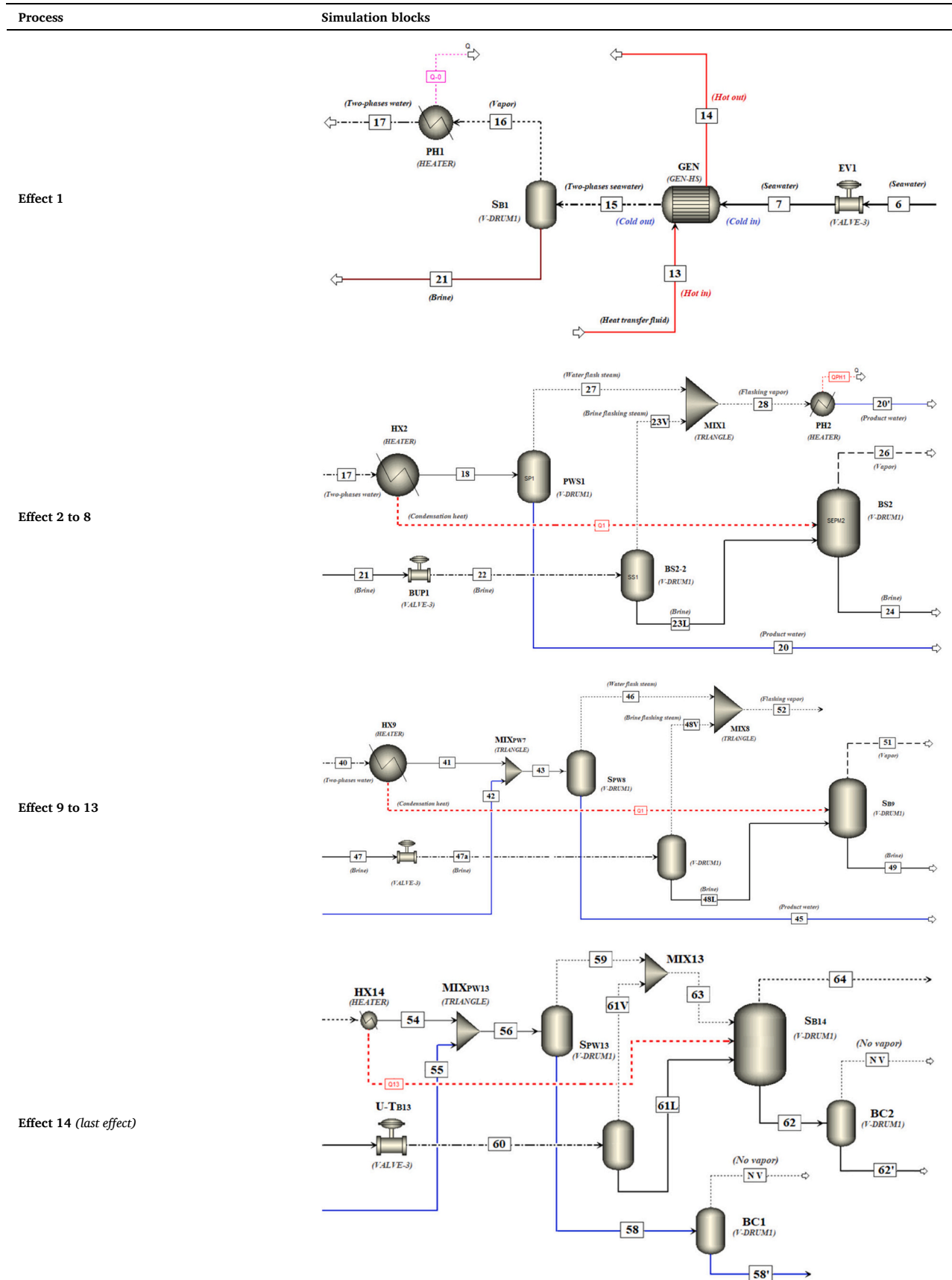
$$RR = \frac{\dot{m}_{PW}}{\dot{m}_{SW}} \quad (3)$$

$$SEC = \frac{\text{Energy input (kWh)}}{\text{Total water produced (m}^3\text{)}} \quad (4)$$

However, due to the characteristics of the MED-HV system (the use of



**Table 2**  
Simulation blocks in Aspen Plus of the MED-HV system.



(continued on next page)

Table 2 (continued)

| Process               | Simulation blocks |
|-----------------------|-------------------|
| Ejector-condenser     |                   |
| MED-HV (forward flow) |                   |

Table 3

Operating conditions of the MED high-vacuum system (MED-HV).

| Parameters                                            |        | Units |
|-------------------------------------------------------|--------|-------|
| Number of effects (HX)                                | 14     | –     |
| Number of preheaters (PH)                             | 8      | –     |
| Seawater temperature at system inlet ( $T_{SW, in}$ ) | 25.0   | °C    |
| Salt concentration of feed seawater ( $X_{SW}$ )      | 35,000 | ppm   |
| Seawater temperature at first effect inlet (TBT)      | 68.0   | °C    |
| System temperature Cascade Range                      | 68–4   | °C    |

a barometric ejector-condenser subsystem that requires a significant electrical energy consumption, unlike a conventional MED), the following indicators are proposed: Gain Output Ratio Global ( $GOR_G$ ), and Specific Energy Consumption Global ( $SEC_G$ ):

$$GOR_G = \frac{(\dot{m}_{PW} * 2,326 \text{ kJ/kg})}{(Q_{GEN} + W_{Pumps\_Total})} \quad (5)$$

$$SEC_G = \frac{\text{Thermal energy input} + \text{Electric energy input (kWh)}}{\text{Total water produced (m}^3\text{)}} \quad (6)$$

In the product water extraction and conditioning subsystem, the low temperature (4 °C) vapor condensation occurs due to an increase in pressure caused by an ejector. This is a fundamental –yet little known– part of the proposed technology. Therefore, in order to evaluate the thermodynamic feasibility of the process, an analysis using the second law of thermodynamics was conducted on the barometric ejector-condenser. The study is based on the following expression:

$$\sum m_{in} ex_{in} + W_{in} = \sum m_{out} ex_{out} + W_{out} + Ex_D \quad (7)$$

#### 4. Validation of the simulation

The validation of the high-vacuum desalination system was done in two stages: 1) MED-desalination sub-process; and, 2) barometric ejector-condenser sub-process for the extraction and conditioning of product water. Theoretical and experimental data were used to validate the MED

process reported by Palenzuela et al. [48], which correspond to the behavior of a production plant MED misalignment, which has the same number of effects as the proposed system. Furthermore a comparative study was conducted with the theoretical results reported by Aly [8], since It presents vacuum conditions similar to those of the proposed system.

The MED system presented by Palenzuela et al. [48], has 14 effects and 13 preheaters, a TBT of 68 °C and a BBT of 35 °C. It is powered with sea water at 25 °C and a concentration of 35,000 ppm, in addition, it uses a massive flow of activation vapor of 295.9 kg/h. The comparison of the obtained results are shown in Table 4. Likewise, the brine that comes out of each effect and the steam generated by ebullition and flashing, are presented in Fig. 5. The results obtained from the simulation have a deviation from the experimental results of <10 %.

In order to compare the simulation results obtained with Aspen Plus, it was necessary to adapt to the same characteristics of the system presented by Aly [8]. Which presents the study of a TVC/MED system, which consists of an absorption machine coupled with a MED. The MED has 20 effects, uses an operating temperature range of 63–6 °C and a heat transfer fluid, coming from solar collectors at 125 °C. It is fed with seawater at 30 °C and a concentration of 57,000 ppm. The comparison of the simulation carried out in Aspen Plus, with respect to the theoretical data reported by Aly [8], are shown in Table 5. It is observed that the percentage of error in all parameters available is <2 %. Due to what was previously mentioned, it is considered that the Aspen Plus software is suitable for simulating MED desalination processes.

For the second validation stage, the results of two experimental studies were used presented by Bilir Sag et al., [11], and Bilir Sag et al., [12], where an exergetic evaluation is carried out for the ejector-compression cooling cycles. In these cycles, a two-phase liquid-vapor ejector is used to increase the pressure of the aspirated vapor. As the two-phase ejector of the cooling cycle, uses liquid as motive fluid and steam as secondary fluid just like the ejector condenser of the MED-HV, the validation of the latter was carried out taking its operation to the same operating conditions of the ejector-compression cooling cycle proposed by Bilir Sag et al., [11]. Table 6 presents the comparison of the results obtained when analyzing them under the first law of

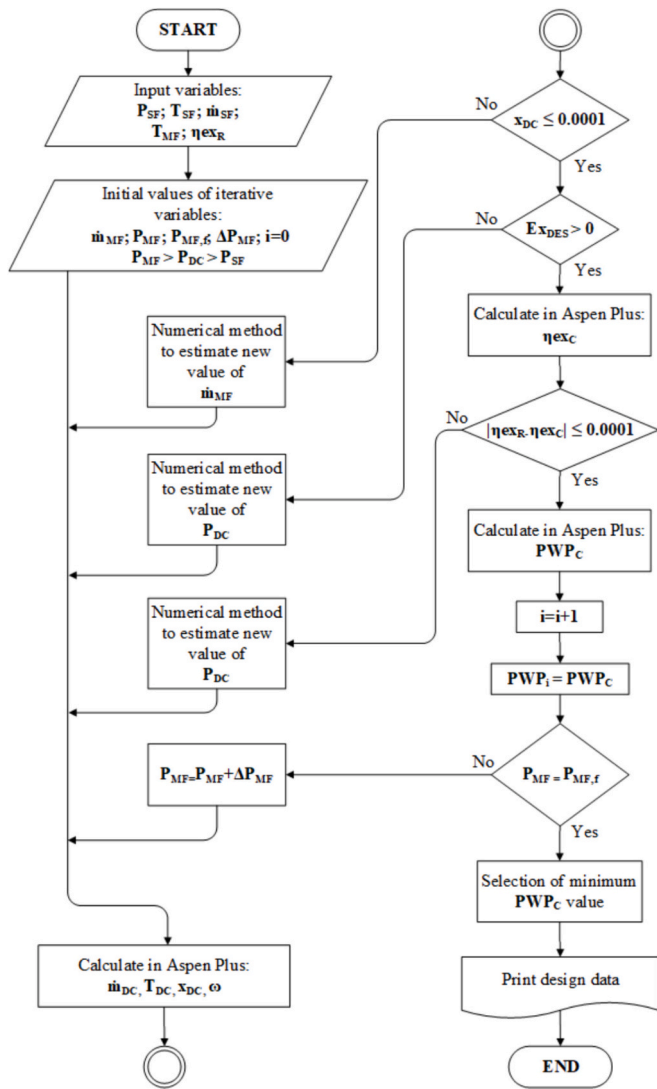


Fig. 4. Flowchart with the Aspen Plus procedure for defining the design conditions of the ejector condenser.

thermodynamics. It is shown that, when using the same pressure, temperature and flow conditions of the mass at the inlet of the primary and secondary fluids of the ejector, the pressure and temperature at the discharge presented errors no  $>3\%$ . The network of the compressor, the capacity of cooling and the COP show errors lower than  $5\%$ , while the largest error ( $4.80\%$ ) occurs on the condenser inlet temperature.

On the other hand, in Table 7, where the exergy destruction is presented, there is an average error of  $3.3\%$  compared to the work of Bilir Sag et al., [11], and  $5.3\%$  for [12]. It is important to mention that the equipment of greatest interest is the two-phase ejector; Due to the aforementioned, it stands out that, in both studies, the error in the ejector was  $<0.3\%$ . From the above, it is determined that Aspen Plus is a tool that allows you to study, based on the first and second law of thermodynamics, real ejection processes that use liquid as a motive fluid

to suck vapor.

## 5. Analysis and discussion of results

### 5.1. Parametric study

The parametric study of the technological proposal is presented, where the internal operation behavior of the system and the variation of desalination capacity and efficiency indicators are shown. The operation of the MED high vacuum desalination system is subject to flow variations, temperature and concentration of the feed seawater, as well as the intermittencies of the solar resource used to activate the generator. Therefore, the evaluation of the technical feasibility of the technological proposal was carried out by studying the effect of the variation of the seawater flow and the amount of heat added to the system, in the system's main efficiency indicators and its desalination capacity.

The behavior of the brine concentration leaving the last effect of the MED and the flow of steam generated by ebullition in the first effect (stream 16), by varying the flow of seawater, is shown in Fig. 6.

Additionally, the amount of steam used to achieve maximum sea water preheating in the first effect is shown ( $65.5^\circ\text{C}$ ). As presented in Fig. 6, by increasing the flow of sea water, it causes a decrease in the steam generated by boiling. Due to the seawater entering to the first

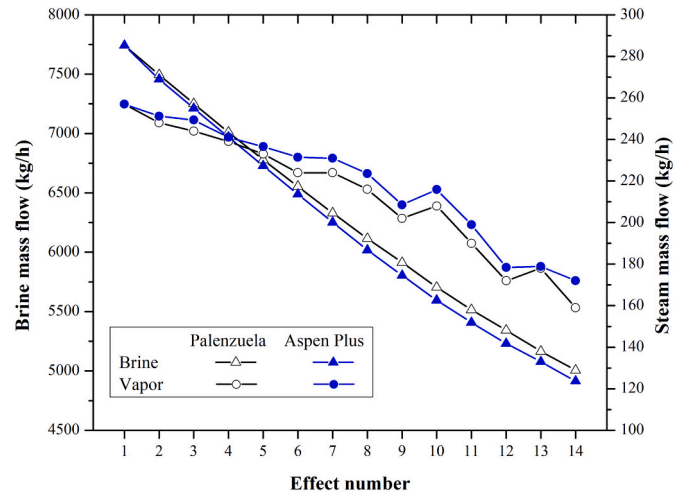


Fig. 5. Comparison between the water vapor production variation and brine for each of the 14 effects of the MED and the experimental results.

Table 5

Validation of the simulation results in Aspen Plus with respect to the theoretical data of [8].

| Parameters                        | Units | S.E. Aly | Simulation Aspen Plus | Error (%) |
|-----------------------------------|-------|----------|-----------------------|-----------|
| Product water (US: Upper Section) | kg/s  | 27.55    | 27.55                 | 0.00      |
| Product water (LS: Low Section)   | kg/s  | 39.76    | 39.77                 | 0.03      |
| Product water capacity            | kg/s  | 67.31    | 67.32                 | 0.01      |
| GOR                               | –     | 14.80    | 14.99                 | 1.27      |

Table 4

Validation of simulation results in Aspen Plus in respect with the experimental and theoretical data reported by [48].

| Parameters                      | Units | Palenzuela (expt.) | Palenzuela (model) | Simulation Aspen Plus | Error (%) expt. | Error (%) model |
|---------------------------------|-------|--------------------|--------------------|-----------------------|-----------------|-----------------|
| Product water capacity          | kg/h  | 2984               | 3003               | 3114                  | 4.2             | 3.5             |
| Seawater entering the condenser | kg/h  | 14,558             | 15,848             | 14,949                | 2.6             | 6.0             |
| GOR                             | –     | 9.7                | 10.2               | 10.5                  | 7.8             | 3.1             |
| RR %                            | –     | 37.5               | 37.6               | 38.9                  | 3.6             | 3.4             |



**Table 6**

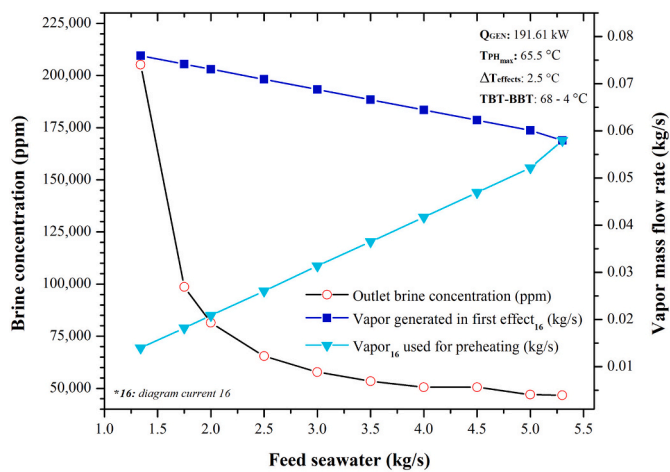
Comparison between the experimental results of Bilir Sag et al. [11,12] and the results obtained from the simulation in Aspen Plus.

| Parameters                 |       | Expt. 2015 | Aspen Plus | Expt. 2017 | Aspen Plus | Error (%) (2015) | Error (%) (2017) |
|----------------------------|-------|------------|------------|------------|------------|------------------|------------------|
| Ejector outlet temperature | (°C)  | 7.68       | 7.49       | 13.12      | 12.88      | 2.54             | 1.86             |
| Ejector discharge pressure | (kPa) | 380.43     | 381.0      | 456.0      | 456.0      | 0.15             | 0.00             |
| Compressor Net work        | (kW)  | 1.35       | 1.30       | 2.11       | 2.07       | 3.76             | 1.79             |
| Cooling capacity           | (kW)  | 4.30       | 4.29       | 4.47       | 4.45       | 0.34             | 0.47             |
| COP                        | (-)   | 3.18       | 3.29       | 2.12       | 2.15       | 3.36             | 1.27             |
| Condenser inlet temp.      | (°C)  | 57.38      | 57.38      | 77.36      | 81.26      | 0.00             | 4.80             |
| Condenser outlet temp.     | (°C)  | 37.78      | 37.78      | 57.64      | 59.02      | 0.00             | 2.34             |

**Table 7**

Comparison of destroyed exergy between the results of Bilir Sag et al. [11,12] and the results obtained with the simulation in Aspen Plus.

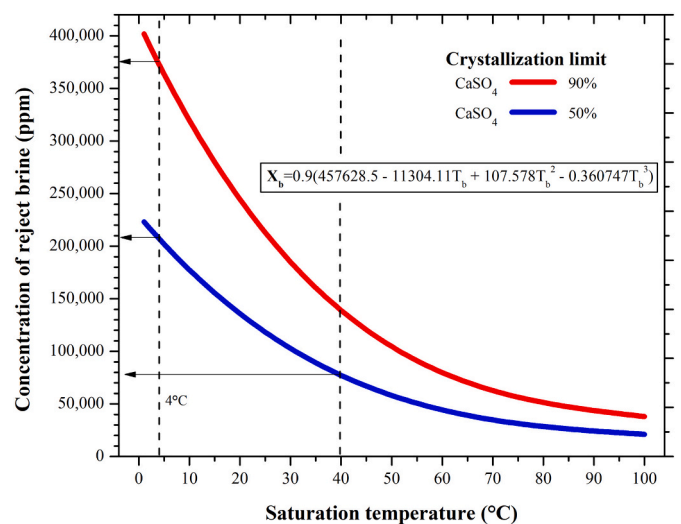
| Component       | Exergy destruction rate (W) |                       |           |            |                       |           |
|-----------------|-----------------------------|-----------------------|-----------|------------|-----------------------|-----------|
|                 | Expt. 2015                  | Simulation Aspen Plus | Error (%) | Expt. 2017 | Simulation Aspen Plus | Error (%) |
| Condenser       | 96.7                        | 92.04                 | 4.81      | 71.88      | 76.64                 | 6.62      |
| Compressor      | 678                         | 629.56                | 7.14      | 974.3      | 905                   | 7.11      |
| Evaporator      | 79.5                        | 81.11                 | 2.02      | 87.03      | 93.72                 | 7.69      |
| Expansion valve | 0.5                         | 0.52                  | 4.27      | 0.47       | 0.49                  | 5.23      |
| Ejector         | 41.1                        | 41.11                 | 0.01      | 188.2      | 188.72                | 0.27      |
| Separator       | 29.1                        | 29.53                 | 1.47      | 35.76      | 37.44                 | 4.71      |
| Total exergy:   | 924.9                       | 873.86                | 5.52      | 1357.64    | 1379.95               | 1.64      |
| Average error:  |                             |                       | 3.29      |            |                       | 5.27      |

**Fig. 6.** Effect of feed seawater variation on brine concentration and steam flow generated for preheating in the first effect.

effect subcooled and the heat addition being constant. The above makes the increase in seawater flow consume more energy in the form of sensible heat, decreasing the amount of energy available for phase change.

With the previous analysis, the operational limits of the MED-HV system are defined. The upper limit on the feed flow is found when the steam generated in the first effect is fully used in the PH8 preheater, leaving no steam for the subsequent effects of the MED system. The above is given with a flow feed of 5.3 kg/s and a constant activation heat additional to the MED of 191.6 kW, where maximum seawater preheating is achieved, but the cascade of multiple evaporation is eliminated.

On the other hand, the lower limit on the feed flow is given by the concentration of the brine in the last effect, given since this increases as the feed flow is decreased. According to [10] the brine concentration in conventional MED systems should not exceed 70 % or what is equivalent to 70,000 ppm, with the purpose of avoiding corrosion and scaling problems. However, as shown in Fig. 7, the authors [22,61], present the equation of a polynomial that allows to calculate the maximum acceptable salinity of the brine, depending on the saturation

**Fig. 7.** Calcium sulfate solubility and saturation temperature of brine, using the equation reported by [22].

temperature. Whereas, the reject brine is restricted to 90 % of the solution saturation as Calcium sulfate ( $\text{CaSO}_4$ ), because it is the least soluble component.

To ensure any precipitation problems, [9], recommend a 50 % saturation in the least soluble component of the brine, which is also graphed in Fig. 7. The previously mentioned allows the proposed MED-HV system to operate in a maximum rejection brine concentration level of 210,000 ppm, because it's BBT is 4 °C, which is very different from the conventional concentration limit of brine that varies between 70,000 and 85,000 ppm, since its BBT usually ranges from 35 to 40 °C. Allowing the proposed technology (MED-HV) to operate with a greater water recovery (RR). Having lower flow and higher brine concentration, not only favors the desalination process, but also the process of using brine and to the concept of zero liquid discharge (ZLD).

In order to show the operational behavior of the MED-HV system, Fig. 8 shows the behavior of the preheaters and evaporators (effects) of the system. The MED-HV system, to reach the maximum seawater preheating temperature, only requires 8 preheaters as opposed to the 13

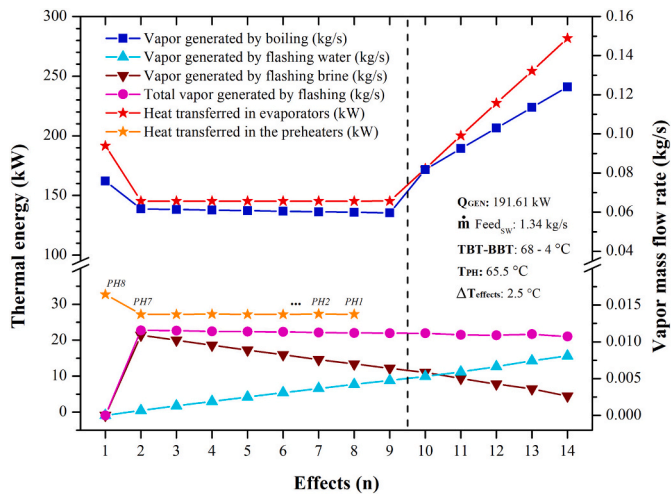


Fig. 8. Performance of evaporators and preheaters and steam generation.

preheaters used in the conventional MED system presented by [48]. The aforementioned is because only preheaters in the effects that have a temperature gradient, with respect to the water of feeding, are used.

Respecting the minimum  $\Delta$  approximation of 2.5 °C between the effects currents and preheaters, we can reach the maximum preheating temperature (65.5 °C). The aforementioned, using in the preheaters (PH1 to PH7), 100 % of steam generated by flashing on the effects (Effect 2–9), complementing the preheating in PH8 by using a percentage of the steam generated by ebullition in the first effect. In the first effect of the MED-HV system, no steam is generated by flashing, just steam ebullition.

It is highlighted that the MED-HV system, compared to conventional MED systems, presents a greater  $\Delta T$  and  $\Delta P$  between effects. Which produces a significant increase in steam generation by flashing. The average heat transferred in preheaters PH1 to PH7 is 27 kW, while in PH8 it is 32 kW.

In Fig. 8, we can observe that with the exception of the first evaporator (HX 1), the steam production generated by boiling and the heat transferred, has an almost constant behavior in the first 9 effects. However, from effect 10 to 14, the generation of steam and heat transferred is significantly increased. This is because both the steam generated by flashing and the steam generated by ebullition, are used in the following effect to heat the brine and produce more steam, which ultimately becomes a larger amount of water product. The aforementioned is the reason why the proposed MED-HV system has greater capacity and efficiency compared to a conventional MED.

On the other hand, Fig. 9 shows the variation of the accumulated product water, the mass flow and brine concentration, effect after effect. The brine flow is decreasing, while its concentration increases. The aforementioned is due to the generation of steam by ebullition and flashing, which subsequently condenses and becomes product water. The product water passes from an effect to another, until exiting in the last effect of the system.

In a conventional MED system, electrical energy consumption is considered negligible with respect to the thermal energy required. However, the MED-HV system proposes two additional hydraulic pumps aside to the feed pump ( $P_{SW}$ ), which are the pump for the traction fluid in the ejector condenser ( $P_{PW}$ ) and the pump for water cooling in the HX<sub>EJC</sub> ( $P_{CW}$ ), which significantly increase the electrical energy consumption of the system.

Therefore, Fig. 10 shows: a) the behavior of the product water and the electric potency required by each of the hydraulic pumps of the system, when varying the flow of seawater and maintaining a fixed activation heat addition. Additionally, it presents: b) the effect of the variation of the seawater flow and activation heat in the total power of

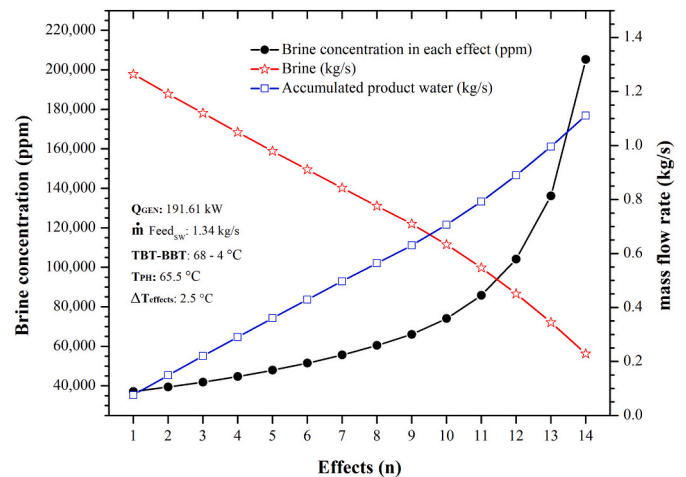


Fig. 9. General operating behavior effect to effect of the main variables in the MED-HV system.

pumps ( $PP_{TOTAL}$ ) and water production product. In both cases, it is shown that as the seawater flow increases, the MED-HV increases its desalination capacity. However, the individual electrical power requirement and total pumps also increase. In part (a) of Fig. 10, it is shown that the pump  $P_{PW}$ , is the one that has the greatest contribution to the electrical energy required. Due to this variable flow pump ( $P_{PW}$ ) in the ejector condenser, providing the pressure and mass flow conditions necessary for the motive fluid (primary fluid) to aspirate, entrain and fully condense a greater amount of vapor from the last effect of the MED (secondary fluid). Since, by increasing the flow of seawater, there is a greater vapor flow in the last effect of the MED. On the other hand, subsection (b) Fig. 10 shows that for the proposed MED-HV system, in addition to the increase in the product water and total electrical power, it also increases the operating range per flow of power, as the heat addition in the generator increases ( $Q_{GEN}$ ).

A study was carried out by the second law of thermodynamics in the barometric ejector condenser, to determine the thermodynamic feasibility zone in which the total condensation of the steam that comes out of the last effect of the MED is achieved. The study was carried out maintaining constant pressure conditions, temperature and mass flow of the steam that enters as secondary fluid (stream 64). In subsection (a) of Fig. 11 the results of the destroyed exergy, under different pressure conditions in the motive fluid (stream 67) and different ejector discharge pressures (stream 68) are shown, always using the minimum flow of the primary fluid that ensures condensation. It is observed that as the pressure in the ejector discharge (diffuser outlet) decreases, the destruction of exergy in the ejector increases. Likewise, when the pressure in the ejector traction fluid increases, the destruction of exergy increases. The aforementioned is due to the fact that in both cases more flow of the motive fluid is used to carry out the total condensation of the steam and there are more energy losses due to friction. Also, using increasingly higher-pressure values in the motive fluid allows for a greater range of operation within the feasible zone where total condensation of the steam in the ejector is achieved. Under the criteria of thermodynamics, the zone with negative exergy destruction refers to a not feasible process not feasible. While in the area where the exergy destroyed is equal to zero, it is equivalent to having a process ideal (reversible process). Therefore, the ejecto-condensation process is feasible in the area of destruction of positive exergy. The exergetic study shown in part (a) of Fig. 11, presents the area in which it is possible to operate the proposed MED-HV system, with the barometric ejector condenser.

On the other hand, within this zone of thermodynamic feasibility, the conditions or operation are sought to find the best compromise between irreversibilities and energy requirements of the system pumps. This is

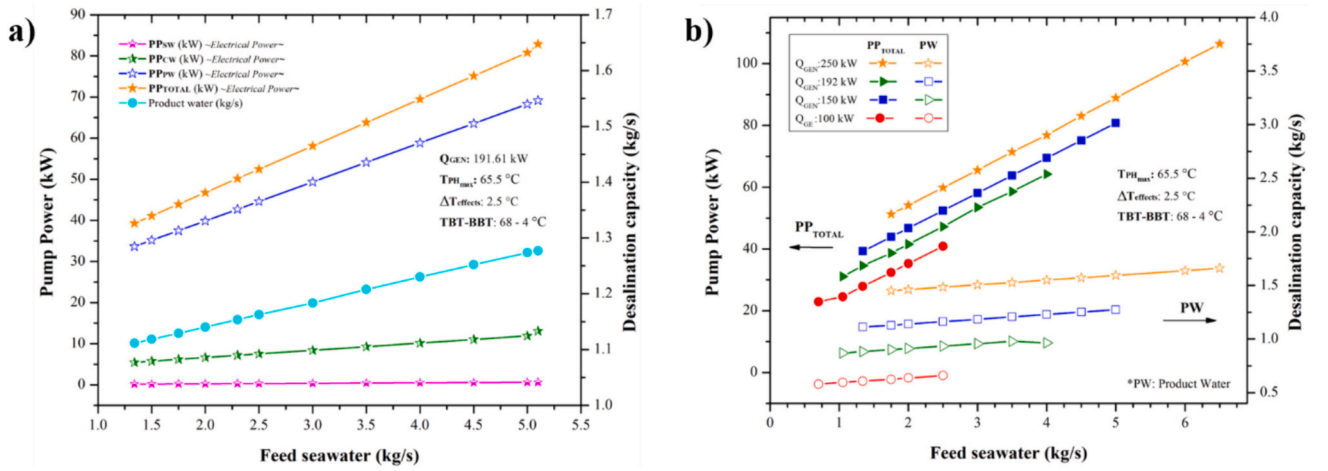


Fig. 10. Variation of product water and electrical power of pumps in the system.

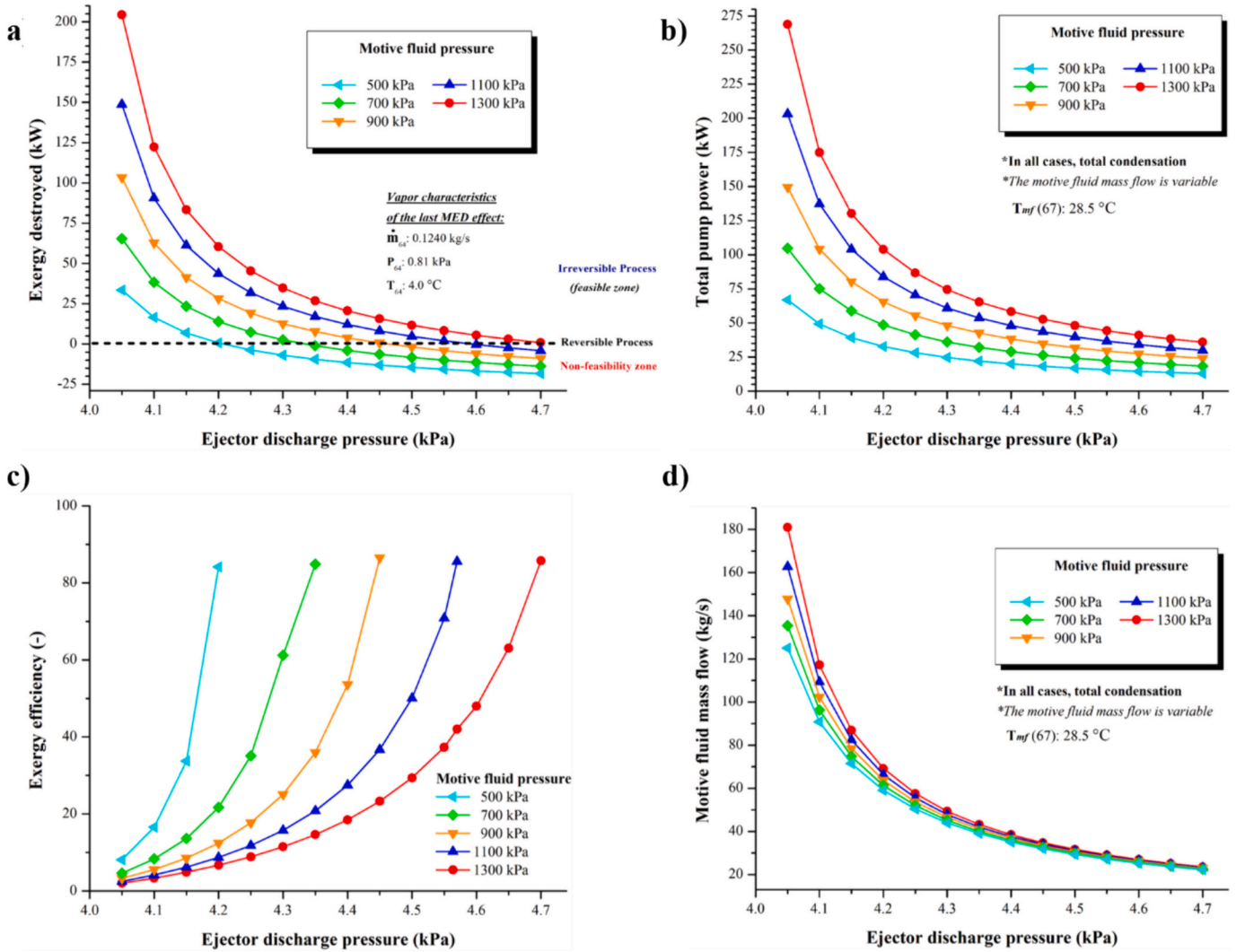


Fig. 11. Variation of exergy destroyed in the ejector, total pump power, mass flow rate of the motive fluid and exergy efficiency under different conditions of pressure at the inlet of primary fluid and the discharge of the ejector.

why, in part (b), (c) and (d) of Fig. 11, the electrical power required by the system pumps, exergy efficiency and the required mass flow rate of the motive fluid, respectively, as the discharge pressure in the ejector is

varied, are presented. It can be observed that the pump potency decreases when the discharge pressure increases, because a lower mass flow of the motive fluid is required to condense the vapor that comes from the last



effect of the MED. However, the electrical potency increases with increasing pressure in the motive fluid (motive fluid), since, together with the increase in pressure, it also increases the mass flow necessary in the primary fluid to achieve total condensation of the vapor (secondary fluid). The selection of the design condition will depend on the exergy efficiency of the ejector used, since said efficiency has a direct relationship with the exergy destruction and pump power, shown in part (a) and (b) of Fig. 11.

Based on the study of the ejector-condenser by the second law of thermodynamics, it is observed that the ejector-condenser can operate under different conditions of pressure and mass flow of the primary fluid. However, the design conditions of the ejector-condenser were conservatively established for an exergy efficiency of 34 % and are shown in Table 8. The exergy efficiency was established based on the work of Šarevski and Šarevski [54], where efficiencies in the range of 0.30–0.47 are indicated for two-phase (liquid-vapor) ejectors using water (R718). On the other hand, the design conditions of the ejector-condenser are similar to those reported in the literature for two-phase ejectors using water as working fluid and operating at high vacuum pressure conditions [5].

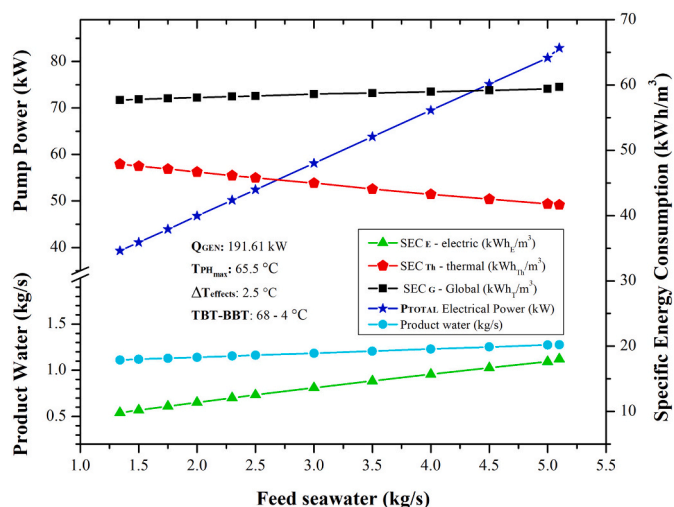
In order to show the best relationship between the production of desalinated water and consumption of electrical energy of the MED-HV system, Fig. 12 shows the behavior of the Specific indicator Energy Consumption Global (SEC<sub>G</sub>). In addition, the specific consumption of thermal energy is presented (SEC<sub>Th</sub>) and also of electrical energy (SEC<sub>E</sub>). It is observed that by increasing the flow of seawater and keeping the addition of heat in the GEN (191.61 kW) fixed, the SEC<sub>Th</sub> indicator, is reduced. This is because it generates a greater amount of water product with the same amount of thermal energy. However, despite that the amount of product water is greater by increasing the flow of seawater, the electrical power required by the system's hydraulic pumps also increases. The aforementioned is due mainly because, the greater the flow of seawater, the greater the amount of steam to condense in the ejector, requiring that the flow of the primary fluid, which passes through pump P PW, be greater, increasing thus his network. Since the electrical energy consumption of the pumps has a greater effect on the production of desalinated water, the SEC<sub>E</sub> indicator increases.

On the other hand, Fig. 13 shows the behavior of the GOR efficiency indicators (conventional) and GOR<sub>G</sub> (global), along with the production of desalinated water. The behavior of product water by varying the feedwater flow is discussed in Fig. 10. In relation to the GOR indicator, which relates the amount of product water and the thermal energy added to the GEN, it shows that the greater the flow of seawater, the GOR increases, since a greater amount is obtained of product water, maintaining the same heat of activation (Q<sub>GEN</sub>).

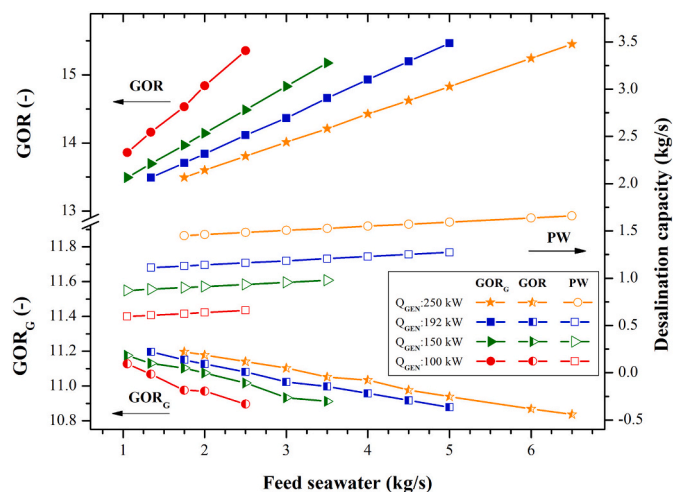
Unlike the GOR indicator, the GOR<sub>G</sub> shows an opposite behavior, since, at a higher flow of feed water, the indicator decreases. This variation is attributed to the fact that the GOR<sub>G</sub>, in addition to the thermal energy added to the system, considers the electrical energy consumed by the pumps. As previously mentioned in the discussion of Fig. 10, it is shown that, the greater the feed water flow, the greater the water product, but it also increases the consumption of electrical energy. This last factor has a greater effect regarding the generation of product water.

**Table 8**  
Ejector-condenser design conditions.

| Parameters                | Units  | Motive Fluid | Suctioned fluid | Ejector outlet |
|---------------------------|--------|--------------|-----------------|----------------|
| Pressure                  | (kPa)  | 500          | 0.81            | 4.15           |
| Temperature               | (°C)   | 28.50        | 4.00            | 29.71          |
| Mass flow rate            | (kg/s) | 71.51        | 0.12            | 71.63          |
| Ejector entrainment ratio | (–)    | –            | –               | 0.0017         |



**Fig. 12.** Variation of the specific thermal, electrical, and overall energy consumption of the MED-HV system.



**Fig. 13.** Effect of seawater and activation heat variation on product water, GOR and on the GOR<sub>G</sub> indicator.

Finally, the amount of heat added to the generator and the mass flow of seawater were varied to evaluate its impact on the main efficiency indicators of the system. The results obtained are presented in Fig. 14, which shows that, by increasing the flow of seawater and reducing the heat added to the generator, the SEC<sub>G</sub> indicator tends to increase. This is because, despite the increase in water production, the effect of electrical energy consumption in the pumps is more significant. The above forces us to select the design point where we have the minimum SEC<sub>G</sub>.

On the other hand, it is shown that as the seawater flow increases, the indicator of GOR<sub>G</sub> efficiency decreases. Due to the aforementioned, it is possible to define that the best operating condition of the proposed system MED-HV, considering the GOR<sub>G</sub>, SEC<sub>G</sub> indicators and respecting the limit of concentration of >210,000 ppm in the rejection brine of the last effect, is achieved by using a activation heat of 192 kW and a feed seawater flow of 1.34 kg/s.

In order to show more clearly the operational functioning of the proposed system, Table 9 presents the thermodynamic properties of each effect in the MED desalination process at high vacuum (MED-HV). Pressure and temperature parameters are presented in cascade form descending throughout the series of effects. The concentration of the brine increases with each effect due to steam generation. In addition, the increase in temperature experienced by the sea water is shown, given the

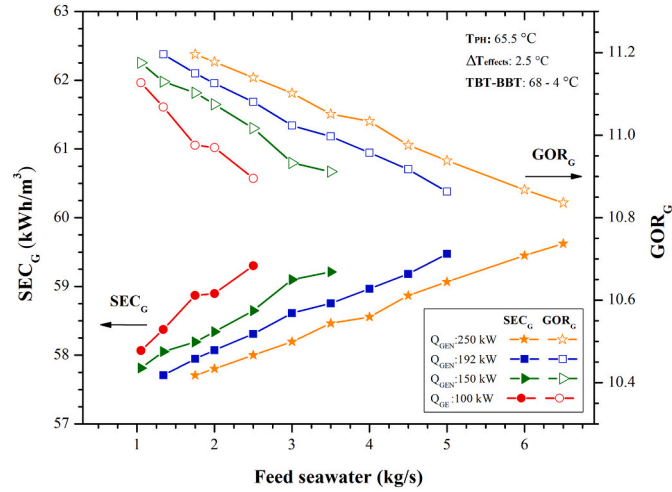


Fig. 14. Effect of seawater and activation heat variation on  $SEC_G$  and  $GOR_G$  indicators.

Table 9  
Thermodynamic and operational characteristics of the MED-HV at its design point.

| Effect | Temperature | Pressure | Accumulated product water | Brine concentration | $Q_{effects}$ | Sea water temperature |
|--------|-------------|----------|---------------------------|---------------------|---------------|-----------------------|
|        | (°C)        | (kPa)    | (kg/s)                    | (ppm)               | (kW)          | (°C)                  |
| 1      | 68.00       | 28.30    | –                         | 37,103.14           | 191.61        | 65.50                 |
| 2      | 63.08       | 22.73    | 0.0753                    | 39,363.26           | 145.29        | 59.58                 |
| 3      | 58.15       | 18.11    | 0.1472                    | 41,880.19           | 145.29        | 54.59                 |
| 4      | 53.23       | 14.32    | 0.2182                    | 44,698.63           | 145.29        | 49.65                 |
| 5      | 48.31       | 11.23    | 0.2882                    | 47,877.22           | 145.28        | 44.72                 |
| 6      | 43.38       | 8.73     | 0.3573                    | 51,490.58           | 145.28        | 39.80                 |
| 7      | 38.46       | 6.72     | 0.4255                    | 55,632.29           | 145.28        | 34.86                 |
| 8      | 33.54       | 5.13     | 0.4927                    | 60,430.12           | 145.28        | 29.93                 |
| 9      | 28.61       | 3.87     | 0.5591                    | 66,056.06           | 145.32        | 25.00                 |
| 10     | 23.69       | 2.91     | 0.6247                    | 74,032.96           | 172.59        | 25.00                 |
| 11     | 18.77       | 2.14     | 0.7006                    | 85,754.87           | 200.20        | 25.00                 |
| 12     | 13.85       | 1.56     | 0.7865                    | 104,089.06          | 227.45        | 25.00                 |
| 13     | 8.92        | 1.12     | 0.8820                    | 136,140.68          | 254.37        | 25.00                 |
| 14     | 4.00        | 0.80     | 0.9874                    | 205,220.60          | 281.77        | 25.00                 |
| Total  |             |          | 1.1115                    |                     | 2490.31       |                       |

Table 10  
Energy evaluation of the proposed MED-HV system under design conditions.

| Components                      | Energy Balance                                                            | Gains (kW)   | Losses (kW)              | Internal Energy Integration (kW) |
|---------------------------------|---------------------------------------------------------------------------|--------------|--------------------------|----------------------------------|
| Generator                       | $Q_{GEN} = m_{13}h_{13} - m_7h_7$                                         | 191.612      |                          |                                  |
| Dissipator (HX <sub>ELC</sub> ) | $Q_{HX_{ELC}} = m_{66}h_{66} - m_{71}h_{71}$                              |              | 230.81                   |                                  |
| Effects (HX)                    | $\sum Q_{HXn} = m_n h_n - m_n h_n$                                        |              |                          | 2298.70                          |
| Preheaters                      | $\sum Q_{PHn} = m_n h_n - m_n h_n$                                        |              |                          | 223.18                           |
| $P_{SW}$                        | $W_{SWP} = m_1 v_1 (P_2 - P_1) / Eff_{SWP}$                               | 0.16         |                          |                                  |
| $P_{PW}$                        | $W_{PWP} = m_{65} v_{65} (P_{66} - P_{65}) / Eff_{PWP}$                   | 33.68        |                          |                                  |
| $P_{CW}$                        | $W_{CWP} = m_{70} v_{70} (P_{71} - P_{70}) / Eff_{CWP}$                   | 5.46         |                          |                                  |
| Flow Streams                    |                                                                           |              |                          |                                  |
| $m_2 h_2$                       | Seawater inlet                                                            | –20,822.31   |                          |                                  |
| $m_{62} h_{62}$                 | Brine outlet                                                              |              | –3202.96                 |                                  |
| $m_{69} h_{69}$                 | Product water output                                                      |              | –17,618.58               |                                  |
| Total                           |                                                                           | –20,591.40   | –20,590.73               | 2521.88                          |
| Product Water                   |                                                                           | 96,033.6 L/d | Daily production         |                                  |
| RR                              | $RR = m_{69} / m_1$                                                       | 0.83         | Recovery Ratio           |                                  |
| GOR                             | $GOR = \frac{m_{69} * 2,326 \text{ kJ/kg}}{Q_{GEN}}$                      | 13.49        | Gain Output Ratio        |                                  |
| BC1                             | $H_1 = (P_{atm} - P_{dis}) / (\rho g)$                                    | 9.998 m      | Barometric Column 1      |                                  |
| BC2                             | $H_2 = (P_{atm} - P_{dis}) / (\rho g)$                                    | 9.966 m      | Barometric Column 2      |                                  |
| BC3                             | $H_2 = (P_{atm} - P_{dis}) / (\rho g)$                                    | 9.655 m      | Barometric Column 3      |                                  |
| GOR <sub>G</sub>                | $GOR_G = \frac{m_{69} * 2,326 \text{ kJ/kg}}{Q_{GEN} + W_{PUMPS\_TOTAL}}$ | 11.20        | Gain Output Ratio Global |                                  |

internal energy integration in the preheaters.

Finally, it is highlighted that the total product water production is 1.11 kg/s, while the Internal energy integration corresponding to the evaporators in the MED effects is 2490.31 kW. The heat shown in effect 1 is not considered as internal energy integration since this represents the heat required in the generator to activate the system.

The indicators of the proposed MED-HV technology, in addition to the amount of product water and the heat transfer by component type, are shown in Table 10.

## 5.2. General comparison

In order to show the main differences and advantages of the system proposed in this work, a general comparison between the MED-HV system and the experimental results of a conventional MED desalination system reported by [48] is presented. The comparison is carried out under the same product water capacity and using the same number of effects, among others parameters shown in Table 11.

Table 12 shows the results of the comparison between the previously mentioned systems. Among the outstanding advantages that the MED-HV system presents with respect to the conventional system, is that the system produces the same amount of product water (0.83 kg/s) using

**Table 11**

Variables and parameters of both systems to carry out the comparative study.

| Parameters                          | Units | MED Palenzuela | MED-HV |
|-------------------------------------|-------|----------------|--------|
| Seawater temperature fed            | °C    | 25             | 25     |
| Salt concentration in feed seawater | ppm   | 35,000         | 35,000 |
| Top Brine Temperature (TBT)         | °C    | 68             | 68     |
| Desalination capacity               | kg/s  | 0.83           | 0.83   |
| Number of effects                   | –     | 14             | 14     |
| Number of preheaters                | –     | 13             | 8      |
| Feed seawater                       | kg/s  | 2.22           | 1.34   |

**Table 12**

Results of the comparative table of the MED-HV and conventional MED systems.

| Parameters         | Units              | MED Palenzuela | MED-HV     | (Δ%)    |
|--------------------|--------------------|----------------|------------|---------|
| $Q_{GEN}$          | kW                 | 191.61         | 142.89     | 25.43   |
| $W_{PSW}$          | kW                 | 0.80           | 0.12       | 84.65   |
| $W_{PCW}$          | kW                 | –              | 4.07       | –       |
| $W_{PPW}$          | kW                 | –              | 25.12      | –       |
| $W_{PUMPS\_TOTAL}$ | kW                 | 0.80           | 29.31      | 3564.19 |
| GOR                | –                  | 10.06          | 13.49      | 34.11   |
| RR                 | –                  | 0.38           | 0.83       | 120.03  |
| $X_{B14}$          | ppm                | 69,326.31      | 205,584.06 | 195.10  |
| $GOR_G$            | –                  | 10.02          | 11.20      | 11.75   |
| $SEC_G$            | kWh/m <sup>3</sup> | 64.49          | 57.71      | 10.51   |

25.4 % less heat in the generator. The aforementioned brings with it a GOR of 13.49, which represents an increase of 34.1 %, compared to the GOR reported in the conventional MED. Likewise, the global indicators that consider the thermal and electrical energy supplied in the proposed system, present an increase in  $GOR_G$  and  $SEC_G$  of 11.75 % and 10.51 % respectively. The aforementioned, despite that the MED-HV system presents a significant increase in the electrical power requirement total for hydraulic pumps, with respect to conventional MED systems.

It should be noted that due to the significant decrease in feed seawater flow, maintaining the same desalination capacity, the MED-HV system presents a value in the indicator RR of 0.83, represents a significant increase of 120 % with respect to the RR reported in the conventional system. The aforementioned, coupled with the fact that the MED-HV system operates in high vacuum conditions, reaching a BBT of 4 °C, being able to operate at concentration levels much higher than the limit of concentration of traditional MED systems of 70,000 ppm as explained above in the results section.

## 6. Conclusions

The proposed multiple effect high vacuum evaporation (MED-HV) technology, which has an integrated ejector condensation system allows operation in a higher pressure and temperature range than that of a conventional MED system, 28.3 at 0.8 kPa and 68 at 4 °C, respectively.

Based on the parametric study, the MED-HV system resulted in a GOR of 13.5, which corresponds to an increase of 34.1 % compared to a conventional MED system. This important improvement is attributed to the fact that the system requires 25.4 % less activation energy to achieve the same capacity of product water.

The MED-HV system reduces the number of components, since there are no preheaters in the last five effects. This is due to the fact that there is no temperature gradient between sea water (25 °C) and the temperature of the steam generated in each effect.

The steam generated in the last five effects of the MED-HV is greater compared to a MED conventional that has the same number of effects (14 effects). This is because all the steam Flashing joins the steam generated by ebullition and is used to heat the brine and generate greater amount of steam in the following effect resulting in a capacity of 96,033.6 L/d of water, which is enough to satisfy the needs of a 960-person community.

The MED-HV system presented a  $GOR_G$  de 11.2, and a RR of 0.83,

which represents an increase in 11.7 % and 120 % respectively, when compared with a conventional MED system of the same capacity and same number of effects. The increase in the RR is due to the high vacuum of the MED allows to have a low temperature brine in the last effect (4 °C) and operate at a concentration higher (205,220 ppm), without having the risk of crystallization.

Under the approach of the first and second law of thermodynamics, it was demonstrated that the two-phase condensation ejector (barometric ejector-condenser), an essential component of the MED-HV system, can condense a vapor at low temperature (4 °C) and pressure (0.8 kPa) and can operate in various pressure and mass flow conditions of the primary fluid (feasibility zone). The selection of the design condition will depend directly from the exergy efficiency of the ejector and as a consequence on the electrical power of the pumps.

Based on the results obtained in this study, a new paradigm is established that allows the development and design of new high vacuum desalination systems, with a longer operating range wide (pressure and temperature cascade). With the increase in capacity and efficiency, along with the Reduction of components in the MED-HV system will allow more efficient attention to the increasing water needs in coastal communities.

## CRedit authorship contribution statement

**C.A. Casares-De la Torre:** Writing – review & editing, Writing – original draft, Visualization, Validation, Supervision, Software, Resources, Project administration, Methodology, Investigation, Funding acquisition, Formal analysis, Data curation, Conceptualization. **N. Velázquez-Limón:** Writing – review & editing, Conceptualization. **R. López-Zavala:** Validation, Formal analysis. **J. Ríos-Arriola:** Writing – original draft, Supervision, Investigation. **S. Islas-Pereda:** Visualization, Methodology, Data curation. **G.E. Dévora-Isiordia:** Writing – review & editing, Resources, Data curation. **J.A. Aguilar-Jiménez:** Writing – review & editing, Project administration, Formal analysis.

## Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

## Data availability

Data will be made available on request.

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### **3.2. Artículo B: High-vacuum multi-effect desalination system with barometric ejector-condensation for simultaneous water and cooling production.**

*Revista: En proceso de revisión en Desalination.*

Este trabajo evalúa la factibilidad técnica de un novedoso sistema de desalinización para la producción simultánea de agua y frío, basado en la modificación de un sistema MED convencional. La modificación consiste en operar al alto vacío, con una cascada presión y temperatura (28.3–0.8 kPa y 68–4 °C, respectivamente), e integrar un subsistema de enfriamiento y eyecto-condensación. La innovación clave radica en el aprovechamiento de las condiciones de operación para utilizar el agua producto directamente como refrigerante, mediante una válvula de tres vías que regula el flujo de agua producto hacia un evaporador y ajusta la capacidad de enfriamiento a la demanda. El análisis del sistema demostró que:

- La configuración permite una producción simultánea de 96.03 m<sup>3</sup>/d de agua producto y 700 TR (2,460 kW) de enfriamiento, capacidad suficiente para satisfacer las necesidades de agua y climatización de espacios de comunidades, hoteles y comercios en zonas costeras.
- El análisis de la capacidad de enfriamiento, en un rango de 0 a 700 TR, revela que, a pesar de un incremento en el consumo energético del eyector a mayores cargas, el sistema mantiene un Coeficiente de Desempeño Global (COP<sub>G</sub>) entre 9.3 y 10.3, superando consistentemente a las tecnologías de enfriamiento convencionales.
- La comparación con sistemas existentes confirma la superioridad del diseño propuesto: Reduce el consumo térmico en un 13.51% y mejora el GOR y GOR<sub>G</sub> en un 16.29% y 8.21%, respectivamente, frente a una MED con enfriamiento integrado; y logra una reducción del 25.43% en consumo térmico, del 17% en consumo eléctrico y una mejora del 25.83% en el COP<sub>G</sub>, frente a un sistema híbrido (MED + enfriador de compresión centrífuga).

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## **Artículo B**

**High-vacuum multi-effect desalination system with  
barometric ejector-condensation for simultaneous  
water and cooling production**

**Revista:** Sometido a revisión en Desalination



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**High-vacuum multi-effect desalination system with barometric ejector-condensation for simultaneous water and cooling production**

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Declarations of interest: none

## Abstract

This paper evaluates the technical feasibility of a desalination system for the simultaneous production of water and cooling, developed from a novel high-vacuum multi-effect distillation (MED-HV) system (28.3–0.8 kPa). The innovation lies in leveraging the operating conditions to use the product water directly as a refrigerant. The system integrates a barometric ejector-condenser and a three-way valve that regulates the product water flow to an evaporator, allowing the cooling capacity to be adjusted according to demand. The study was conducted in Aspen Plus and validated with experimental results reported in the literature. The MED-HV system achieves a water production of 96.03 m<sup>3</sup>/d and a cooling capacity of 700 TR (2,460 kW). The effect of cooling capacity variation on the ejector-condenser performance and the system's global indicators was analyzed. Compared to commercial desalination and cooling technologies operating in a hybrid manner, the proposed system achieves a Global Coefficient of Performance (COP<sub>G</sub>) of 9.22—20.73% higher—and accomplishes a 17.15% reduction in global specific energy consumption, for both desalination (SEC<sub>G\_D</sub>: 127.15 kWh/m<sup>3</sup>) and cooling (SEC<sub>G\_C</sub>: 0.73 kWh/TR). These results confirm the potential of the proposed system to efficiently and simultaneously meet water and space conditioning needs.

Keywords: Cooling, Desalination, MED, High vacuum, Ejector.

## Highlights

Key features of the proposed technology are:

- A novel MED-HV system with EJC for simultaneous water and cooling production.
- The product water is utilized directly as a refrigerant for space conditioning.
- The ejector-condenser (EJC) compresses cold vapor (4 °C) for complete condensation.
- Ejector-condenser performance is analyzed under varying vapor flow rates.
- The proposed system outperforms commercial hybrid configurations in efficiency.

**Nomenclature***Symbols and abbreviations****Symbols***

|             |                                                   |                          |                       |
|-------------|---------------------------------------------------|--------------------------|-----------------------|
| $\dot{E}_x$ | Exergy destruction (kW)                           | RO                       | Reverse osmosis       |
| $\dot{e}_x$ | Exergy flow rate (kW)                             | EXPV                     | Expansion valve       |
| $\dot{m}$   | Mass flow (kg/s)                                  | COND                     | Condenser             |
| T           | Temperature (°C)                                  | COMP                     | Compressor            |
| P           | Pressure (kPa)                                    | SEP                      | Separator             |
| x           | Quality (-)                                       | GEN                      | Generator             |
| VF          | Vapor fraction (-)                                | HV                       | High vacuum           |
| X           | Salt concentration (ppm)                          | HX                       | Heat exchanger        |
| $\eta$      | Efficiency (-)                                    | PH                       | Preheaters            |
| $\eta_{ex}$ | Exergy efficiency (-)                             | mno                      | Motive nozzle output  |
| Q           | Heat transferred (kW)                             | mco                      | Mixing chamber output |
| W           | Net Work (kW)                                     | TR                       | Tons of refrigeration |
| COP         | Coefficient of Performance (-)                    | CAPEX                    | Capital Expenditure   |
| GOR         | Gain Output Ratio global (-)                      | <b><i>Subscripts</i></b> |                       |
| RR          | Recovery Ratio (-)                                | PF                       | Primary fluid         |
| SEC         | Specific energy consumption (kWh/m <sup>3</sup> ) | SF                       | Secondary fluid       |
| TTP         | Total Pumping Power (kW)                          | DC                       | Ejector discharge     |
| $\omega$    | Ejector entrainment ratio (-)                     | G                        | Global                |
| BBT         | Bottom Brine Temperature (°C)                     | D                        | Desalination          |
| TBT         | Top Brine Temperature (°C)                        | C                        | Cooling               |
| H           | Height (m)                                        | E                        | Electric              |
| $\Delta$    | Delta                                             | T                        | Thermal               |

***Abbreviation***

|     |                                 |     |               |
|-----|---------------------------------|-----|---------------|
| BC  | Barometric column               | PW  | Product water |
| EJC | Ejector-condenser               | in  | Inlet         |
| EVA | Evaporator                      | out | Outlet        |
| HDH | Humidification–Dehumidification | c   | Calculated    |
| MED | Multiple effect distillation    | f   | Final         |
| MSF | Multi-stage flash distillation  | r   | Required      |
|     |                                 | t   | Total         |

## 1. Introduction

The growing scarcity of freshwater, particularly acute in arid and semi-arid regions, has intensified the pursuit of efficient desalination solutions [1–3]. However, the high energy consumption associated with conventional desalination technologies threatens their large-scale sustainability [4,5]. Concurrently, the demand for efficient cooling systems for space conditioning continues to rise, posing significant challenges in terms of energy use and environmental impact [6,7]. According to the International Energy Agency (IEA), buildings account for approximately 30% of global energy consumption and 55% of electricity use, with space conditioning alone responsible for 8.5% [8].

Traditionally, freshwater supply has been addressed through various desalination technologies, most notably reverse osmosis (RO), multi-stage flash distillation (MSF), and multi-effect distillation/evaporation (MED). These account for 70%, 17%, and 6% of the global installed capacity, respectively, collectively representing 93% of the worldwide total [9]. On the other hand, space conditioning has primarily relied on mechanical vapor compression or absorption cooling systems [10,11]. Nevertheless, these conventional technologies exhibit limitations in efficiency and sustainability, especially when aiming to simultaneously address the coupled needs for water and cooling.

The convergence of these two demands —desalinated water and thermal conditioning of spaces— represents an opportunity to develop integrated solutions that optimize energy resource use and reduce capital expenditure (CAPEX). In this context, interest has grown in coupling desalination systems with cooling technologies, enabling the simultaneous production of both utilities. However, these integrated systems face significant challenges, including design complexity and the costs associated with their implementation [12,13]. The most widely explored configurations integrate MED, MSF, and humidification-dehumidification (HDH) desalination processes with absorption, adsorption, or ejection-based cooling units [14–16].

In the pursuit of higher efficiency, recent research has demonstrated the potential of coupling MED systems with absorption chillers. **Chorak et al.** and **Alarcón-Padilla et al.** [17,18] experimentally validated this integration by utilizing the refrigeration effect of an absorption chiller's evaporator to condense the vapor from the last effect of the MED system. This strategy reduced energy losses and yielded notable performance improvements, achieving a Performance

Ratio (PR) of 18.53 and a desalination capacity of 2.42 m<sup>3</sup>/h. A key challenge emerging from this and similar configurations is the increased system complexity, which raises the number of components and can impact techno-economic viability.

Exploring sustainable energy sources, **López-Zavala et al.** [13] proposed an innovative cooling and desalination system based on a single-effect LiBr/H<sub>2</sub>O absorption cycle with a triple-pressure level, activated by thermal solar energy. Their system achieved a production of 838 L/d alongside 23 kW of cooling capacity. Their innovative pressure configuration enhanced efficiency by 22% compared to conventional cycles, highlighting its potential; however, it also underscores the challenge of reaching market-competitive levels with complex system architectures.

In a subsequent contribution, **López-Zavala et al.** [19] integrated a 14 effect Multiple-Effect Distillation (MED) system with a 23 kW single-effect open absorption system. The hybrid system reportedly achieved an extraordinary 87.4 fold increase in cooling capacity (2,012 kW) and a Coefficient of Performance (COP) of 6.15. While these results represent a conceptual advance in thermal performance, the resulting hybrid configuration is more complex, incorporating a greater number of components.

Another research thrust has focused on expanding the operational range of MED systems by reducing the Bottom Brine Temperature (BBT) [20–22]. **Aly** [23] demonstrated this concept by hybridizing an MED system with a LiBr-H<sub>2</sub>O absorption cycle, utilizing a Top Brine Temperature (TBT) of 63 °C and achieving a BBT of 6 °C. This enabled a 20-effect cascade that reached a Gained Output Ratio (GOR) of 14.8. Although this configuration leverages advanced thermal synergies, such as recovering heat from the absorber and condenser to activate the MED unit, its focus is not on the cogeneration of cooling, and the resulting system entails greater complexity.

Alternatively, **Shahzad et al.** [24,25] analyzed the coupling of an MED system with a silica-gel adsorption (AD) chiller to handle the low-temperature vapor from the last effect. The MED unit operated within temperature (TBT-BBT) ranges of 50–3 °C and 70–5 °C, reporting a 2 to 3 fold increase in water production. The industrial-scale feasibility of this approach, however, must consider the challenges associated with system controllability and the regeneration of the adsorbent material.



A promising strategy to enhance competitiveness involves the adaptation of existing technologies for cogeneration with minimal modifications. In this context, **A. Askalany et al.** [26] proposed an adsorption-based cooling and desalination unit by modifying the evaporator and incorporating two ejectors. This redesign achieved a 192% improvement in the Coefficient of Performance (COP=2.2) and a specific water yield of 79.4 m<sup>3</sup> per ton of silica gel. This work demonstrates the potential of innovative redesigns to add functionality.

A significant advance towards direct integration was reported by **López-Zavala et al.** [12], who utilized the product water from a conventional MED system to generate cooling through a process of flash evaporation and a barometric ejector-condenser, achieving a cooling capacity of 2,043 kW. A key feature of this design is that it maintains the conventional operational parameters of an MED system (TBT–BBT of 70–35 °C), resulting in desalination capacity and Performance Ratio (PR) figures comparable to baseline systems. Furthermore, the system reportedly achieves a Coefficient of Performance (COP) 6.53 times higher than of a standard single-effect absorption chiller. This development is currently under patent review by the Mexican Institute of Industrial Property (IMPI), with file number MX/a/2020/008009.

Building upon this principle, **C.A. Casares-De la Torre et al.** [27] introduced a fundamental modification: operating the MED system under high-vacuum conditions (BBT of 4 °C). This was enabled by incorporating an ejector-condenser, which extracts, entrains, and compresses the cold vapor (4 °C) from the MED's last effect. In contrast to the system presented by **López-Zavala et al.** [12]—which utilizes an MED operating under conventional conditions—this system does not employ product water expansion through a valve. Instead, the MED's operating conditions are modified to function at high vacuum. This innovation dramatically expands the temperature and pressure cascade, resulting in a 34.1% improvement in the Gained Output Ratio (GOR) and a 120% increase in the Recovery Ratio (RR) compared to a conventional system. This remarkable increase in RR is a direct consequence of operating at a BBT of 4 °C, which enables reaching a maximum brine concentration of 210,000 ppm—behavior supported by the thermodynamic fundamentals reported in the literature [28–30]. This work establishes a new performance paradigm for desalination, although its scope is focused solely on water production. It is noted that this study is supported by a patent application filed with the IMPI under file number MX/a/2025/001564. The present article is a continuation of this work.

Based on the aforementioned challenges and opportunities, previous efforts have sought to develop technologies that simultaneously address the needs for water and cooling. The predominant strategies are based on hybridizing MED systems with absorption/adsorption chillers, which introduces complexity that impacts techno-economic viability. Conversely, proposals that prioritize configuration simplicity [12] or desalination performance [27] have achieved notable advances on specific fronts, but have not successfully combined high-capacity, high-efficiency cogeneration with a simplified configuration.

This gap motivates the present research. This work introduces a novel system for the simultaneous production of water and cooling, which is based on a modified conventional MED system. The modification involves altering its operating conditions (high vacuum) and incorporating a cooling and ejector-condensation subsystem. The system operates with an expanded pressure and temperature cascade (28.3–0.8 kPa and 68–4 °C, respectively). As a consequence of operating under high vacuum, the product water can be used directly as a refrigerant for cooling production by incorporating an evaporator and a three-way valve to control the cooling capacity. This approach paves the way for a new generation of thermal desalination and cooling technologies with enhanced capacity and efficiency.

## 2. System Description

Figure 1 presents the schematic diagram of the high-vacuum desalination system for the simultaneous production of water and cooling. The system consists of two subsystems: 1) a high-vacuum multi-effect distillation (MED-HV) unit, and 2) a cooling production and ejector-condensation subsystem.

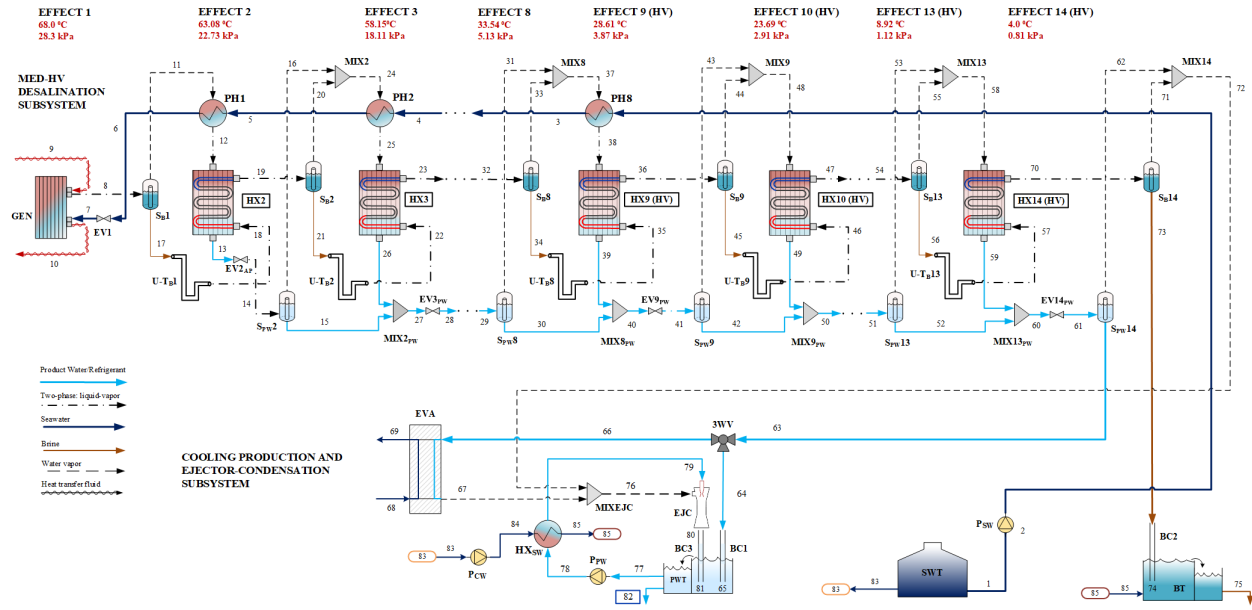
The MED-HV subsystem is based on the 14-effect configuration reported by **Casares-De la Torre et al.** [27], and its detailed description can be found in that reference. The principal innovation of this work lies in the redesign of the cooling production and ejector-condensation subsystem to enable the cogeneration of water and cooling. To achieve this, a heat exchanger functioning as an evaporator (EVA) was integrated, where the product water is utilized as a refrigerant to generate cooling.

The cooling production and ejector-condensation subsystem (Subsystem 2) integrates the key innovative components —the evaporator (EVA), the three-way valve (3WV), and the ejector-condenser (EJC)— along with auxiliary equipment. The latter includes the vapor mixer (MIXEJC), pumps for seawater ( $P_{CW}$ ) and product water ( $P_{PW}$ ), the heat exchanger ( $HX_{SW}$ ), tanks for product water (PWT) and brine (BT), and three barometric columns: two for product water extraction (BC1 and BC3) and one for brine (BC2).

The three-way valve (3WV) is the key component for regulating the system's cooling capacity. As shown in Figure 1, this valve splits the total product water stream (Stream 63) into two: one that is directed to BC1 for extraction (64) and another that feeds the evaporator (EVA) as refrigerant (66). The cooling production is proportional to the mass flow rate directed toward the evaporator (Stream 66). To operate the system exclusively in desalination mode, the entire flow is diverted through Stream (64). Stream (64) descends through the barometric column (BC1) into the product water tank (PWT), where the vacuum is released, and the fluid reaches atmospheric conditions for final extraction (Stream 65).

On the other hand, the product water stream as a saturated liquid (66) enters the evaporator (EVA), where it acts as a refrigerant. Upon receiving heat from the fluid to be cooled (Stream 68), the water (66) completely evaporates (becoming vapor Stream 67), generating the desired cooling effect. Stream (68) exits the evaporator (EVA) as Stream (69) after having transferred its heat to Stream (66). The water vapor (67) is combined in the mixer (MIXEJC) with the vapor from Effect 14 (Stream 72), forming the suction vapor stream (76). Finally, this low-pressure, low-temperature vapor stream (76) is drawn in and compressed within the ejector-condenser (EJC), where it undergoes complete condensation.

Inside the ejector-condenser (EJC), the motive stream —the primary fluid (79)— entrains and compresses the suction vapor stream —the secondary fluid (76)— inducing its condensation. The resulting mixture exits the EJC as a liquid stream (80). As detailed in Figure 2, the expansion of the motive stream (79) through the convergent-divergent nozzle generates a pressure drop (point 79a) that, being lower than the pressure of the secondary vapor (76), is responsible for drawing in the latter. This condition defines the lowest pressure point in the entire system.



**Figure 1.** Schematic diagram of the MED-HV desalination system for the simultaneous production of water and cooling.

The interaction within the mixing chamber (between streams 79a and 76) and the subsequent deceleration in the diffuser compress the two-phase mixture, achieving complete vapor condensation by converting its kinetic energy into a pressure increase. The latent heat of condensation released is absorbed by the motive fluid (79), resulting in a liquid stream at an intermediate pressure with a corresponding temperature rise (Stream 80). The liquid stream (80) is discharged to atmospheric pressure through the barometric column BC3 into the product water tank (PWT). From the PWT, a portion of the water (Stream 77) is recirculated by the product water pump ( $P_{PW}$ ) to be used as the ejector-condenser's motive fluid, while the surplus (Stream 82) represents the final product. It is important to note that the barometric columns (BC1, BC2, and BC3) serve a dual function: the extraction of fluids and the maintenance of the system's vacuum via a hydraulic seal.

To guarantee the thermal stability of the ejector-condenser (EJC), the recirculated water stream (78) is cooled in the heat exchanger (HX<sub>sw</sub>) before returning as the motive fluid (79). In the HX<sub>sw</sub> heat exchanger, the condensation heat is removed by a stream of seawater at 25 °C (84), driven by the seawater pump ( $P_{CW}$ ). This stream, now heated (85), is directed to the brine tank (BT) to dilute the concentrated brine (Stream 73) from the last effect of the MED-HV unit, facilitating its handling prior to final discharge (75).

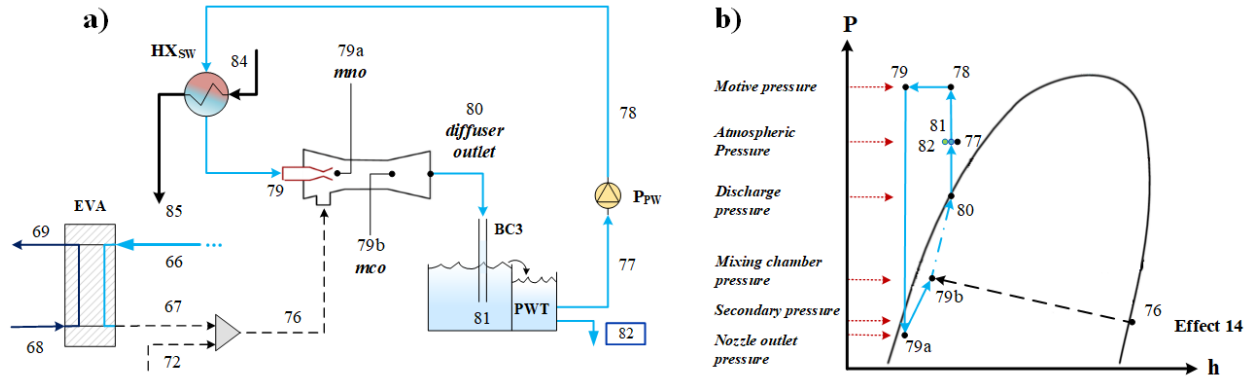


Figure 2. P-h diagram of the barometric ejector-condenser in the MED-HV system with cooling production.

The proposed system is distinguished by its simplified and efficient integration for the cogeneration of water and cooling. Its main features include: (1) the incorporation of a barometric ejector-condenser to condense low-temperature vapor ( $4\text{ }^{\circ}\text{C}$ ) through a pressure increase; (2) high-vacuum operating conditions that enable a reduction in the number of preheaters (from 13 to 8), thereby decreasing the total number of components compared to configurations coupling conventional technologies; (3) the direct use of product water as a refrigerant in an evaporator, without prior expansion; (4) flexible operation ensured by a three-way valve for cooling capacity control and a variable frequency drive on the motive fluid pump, which allows the ejector-condenser to operate effectively under off-design conditions. Collectively, these features result in a system with a simultaneous production of water and cooling that achieves higher capacity and efficiency than traditional hybrid systems.

### 3. Methodology

The technical feasibility of the MED-HV system for the simultaneous production of water and cooling was evaluated through modeling and simulation using Aspen Plus V10 software. The model was validated by comparing its results with operational data reported in the literature for baseline configurations [23,31,32]. Subsequently, a parametric analysis was conducted to establish the optimal design conditions, which were selected by maximizing water production and cooling capacity while maintaining a balance with the overall system efficiency.

### ***3.1. System Simulation***

The system simulation was implemented in ASPEN PLUS V10, leveraging its robust thermodynamic and equipment model libraries to accurately represent the involved processes.

Given that the simulation methodology for the MED-HV subsystem was established by **Casares-De la Torre et al.** [27], this article focuses on the methodology developed for the ejector-condenser model within the cooling production and ejector-condensation subsystem.

#### ***3.1.1. Procedure for Defining the Ejector-Condenser Design Conditions***

Figure 3 shows the iterative procedure followed in Aspen Plus to define the ejector-condenser design conditions. The objective of this methodology was to guarantee the complete condensation of the secondary vapor (76) within a thermodynamically viable range, while maximizing exergy efficiency and minimizing the energy consumption of the pumping system.

Table 1 presents the key components of the cooling production and ejector-condensation subsystem, detailing their scheme, associated streams, working fluid, and the governing energy balance equations for their operation. To ensure model reproducibility, Table 2 specifies the correspondence between the physical components and the Aspen Plus simulation blocks used, along with their connection configuration. The modeling of the MED-HV subsystem followed the implementation reported by **Casares-De la Torre et al.** [27].

The selection of thermodynamic models was based on the working fluids. The Electrolyte method was employed for the seawater streams, while the IAPWS-95 model was used for the product water. Consequently, the HX<sub>SW</sub> heat exchanger (four-stream HeatX block) was configured with the Electrolyte method on the seawater side and IAPWS-95 on the product water side. All other blocks in the cooling production and ejector-condensation subsystem exclusively used the IAPWS-95 model (properties for water).

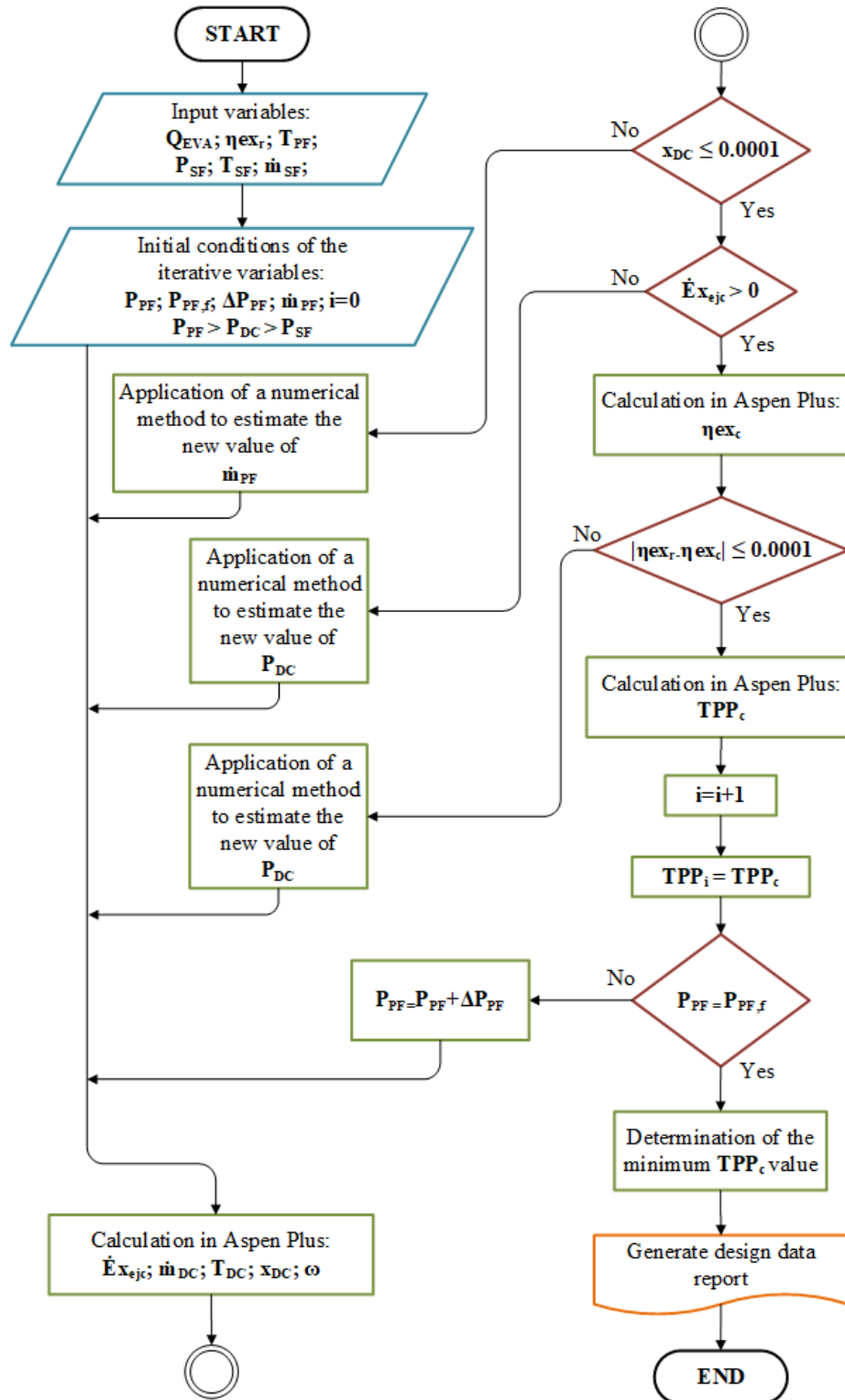
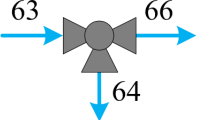
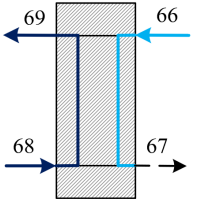
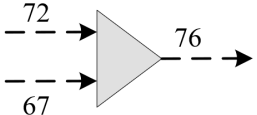
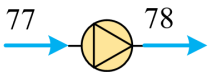
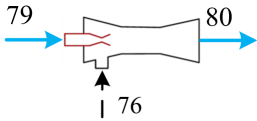
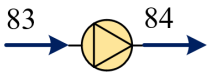
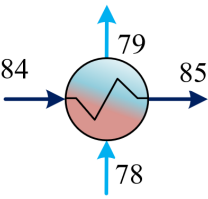


Figure 3. Iterative procedure for defining the ejector-condenser design conditions.

Table 1. Main components of the cooling production and ejector-condensation subsystem.

| Component                                      | Schematic diagram                                                                   | Stream |               | Energy balance                                                                           |
|------------------------------------------------|-------------------------------------------------------------------------------------|--------|---------------|------------------------------------------------------------------------------------------|
|                                                |                                                                                     | No.    | Fluid type    |                                                                                          |
| Three-way valve (3WV)                          |    | 63     | Product water | $\dot{m}_{63}h_{63}$<br>$= \dot{m}_{64}h_{64}$<br>$+ \dot{m}_{66}h_{66}$                 |
|                                                |                                                                                     | 64     | Product water |                                                                                          |
|                                                |                                                                                     | 66     | Product water |                                                                                          |
| Evaporator (EVA)                               |    | 66     | Product water | $\dot{m}_{66}h_{66} + \dot{m}_{68}h_{68}$<br>$= \dot{m}_{67}h_{67} + \dot{m}_{69}h_{69}$ |
|                                                |                                                                                     | 67     | Water vapor   |                                                                                          |
|                                                |                                                                                     | 68     | Cooling water |                                                                                          |
|                                                |                                                                                     | 69     | Cooling water |                                                                                          |
| Mixer (MIXEJC)                                 |    | 67     | Water vapor   | $\dot{m}_{67}h_{67} + \dot{m}_{72}h_{72}$<br>$= \dot{m}_{76}h_{76}$                      |
|                                                |                                                                                     | 72     | Water vapor   |                                                                                          |
|                                                |                                                                                     | 76     | Water vapor   |                                                                                          |
| Product water pump (P <sub>PW</sub> )          |   | 77     | Product water | $\dot{m}_{77}h_{77} + W_{P\_PW}$<br>$= \dot{m}_{78}h_{78}$                               |
|                                                |                                                                                     | 78     | Product water |                                                                                          |
| Ejector-condenser (EJC)                        |  | 76     | Product water | $\dot{m}_{76}h_{76} + \dot{m}_{79}h_{79}$<br>$= \dot{m}_{80}h_{80}$                      |
|                                                |                                                                                     | 79     | Water vapor   |                                                                                          |
|                                                |                                                                                     | 80     | Product water |                                                                                          |
| Seawater pump (P <sub>CW</sub> )               |  | 83     | Seawater      | $\dot{m}_{83}h_{83} + W_{P\_CW}$<br>$= \dot{m}_{84}h_{84}$                               |
|                                                |                                                                                     | 84     | Seawater      |                                                                                          |
| Heat dissipation exchanger (HX <sub>SW</sub> ) |  | 78     | Product water | $\dot{m}_{78}h_{78} + \dot{m}_{84}h_{84}$<br>$= \dot{m}_{79}h_{79} + \dot{m}_{85}h_{85}$ |
|                                                |                                                                                     | 79     | Product water |                                                                                          |
|                                                |                                                                                     | 84     | Seawater      |                                                                                          |
|                                                |                                                                                     | 85     | Seawater      |                                                                                          |

### 3.1.2. System Parameters Under Study

The analysis considers a baseline MED-HV configuration operating with a descending pressure cascade (28.3 to 0.8 kPa) and temperature cascade (68 to 4 °C) across the effects. The



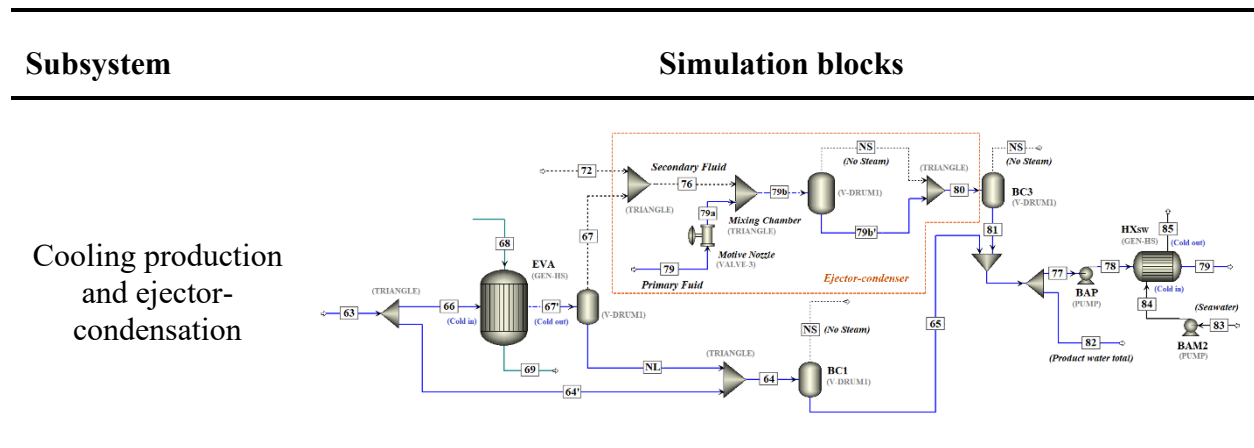
feed consists of seawater at 25 °C with a salinity of 35,000 ppm. A minimum temperature approach ( $\Delta T_{min}$ ) of 2.5 °C was established for all heat exchangers between the hot and cold streams.

The operational behavior of the cooling production and ejector-condensation subsystem was evaluated by varying the percentage of product water directed to the evaporator via the three-way valve (3WV). The remaining simulation parameters are summarized in Table 3.

To guarantee model stability and convergence, particularly due to the recycle streams, the following key iterative variables were initialized:

- $T_6 = 65.5$  °C; maximum preheating temperature.
- $T_{79} = 28.5$  °C; motive fluid temperature at the ejector-condenser inlet.
- $FV_{80} = 0$ ; vapor fraction at the ejector-condenser discharge.
- $T_{69} = 7.0$  °C; cooling temperature at the evaporator outlet.

**Table 2.** Aspen Plus simulation blocks for the cooling production and ejector-condensation subsystem.



**Table 3.** Operating conditions for the high-vacuum multi-effect distillation (MED-HV) system producing water and cooling.

| Parameters                                  |        | Units |
|---------------------------------------------|--------|-------|
| Number of effects                           | 14     | -     |
| Number of preheaters                        | 8      | -     |
| System temperature range (TBT – BBT)        | 68 - 4 | °C    |
| Cooling fluid inlet temperature (Stream 68) | 12.0   | °C    |

|                                              |      |    |
|----------------------------------------------|------|----|
| Cooling fluid outlet temperature (Stream 69) | 7.0  | °C |
| Pump efficiency                              | 0.85 | -  |

---

### 3.2. Basic Assumptions

The system simulation was based on the following fundamental assumptions:

- Steady-state operation.
- Reference environmental conditions for the exergy analysis: 25 °C and 1 atm.
- Pressure in the evaporator (EVA) fixed at the saturation pressure corresponding to 4 °C.
- Inlet and outlet temperatures of the cooled fluid in the evaporator: 12 °C and 7 °C, respectively.
- Top Brine Temperature (TBT) of 68 °C, Bottom Brine Temperature (BBT) of 4 °C, and an inter-effect temperature difference ( $\Delta T$ ) of 4.93 °C.
- Minimum temperature approach ( $\Delta T_{min}$ ) of 2.5 °C in all heat exchangers.
- Maximum seawater preheating temperature:  $T_6 = 65.5$  °C.
- Mass flow rate and temperature of the heat transfer fluid:  $\dot{m}_{13} = 2.5$  kg/s and  $T_9 = 90$  °C.
- Isentropic efficiency of the hydraulic pumps ( $P_{SW}$ ,  $P_{CW}$  and  $P_{PW}$ ): 85%.
- Stream (78) is cooled to  $T_{79} = 28.5$  °C to guarantee the operational stability of the ejector-condenser.
- The mass flow rate of the ejector-condenser's primary fluid varies between 1 kg/s and 700 kg/s.
- Pressure drops in equipment and piping are negligible.

### 3.3. Thermodynamic Evaluation Criteria

The system analysis is based on the first law of thermodynamics. Equation (1) presents the general energy balance, while the individual balances for each component are detailed in Table 1 and 7.

$$\Delta E = Q_{net} - W_{net} \quad (1)$$

#### 3.3.1. Conventional Performance Metrics

The following conventional metrics were employed to evaluate performance:

**Gained Output Ratio (GOR):** Ratio of the latent heat of the produced water to the thermal energy supplied (Equation 2).

$$GOR = \frac{((\dot{m}_{PW} * 2,326 \text{ kJ} / \text{kg}))}{(Q_{GEN})} \quad (2)$$

**Recovery Ratio (RR):** Ratio of the product water mass flow rate to the seawater feed mass flow rate (Equation 3).

$$COP = \frac{Q_{EVA}}{(Q_{GEN} + W_{TOTAL PUMPING})} \quad (3)$$

**Coefficient of Performance (COP):** Ratio of the cooling capacity to the total energy supplied (Equation 4).

$$RR = \frac{\dot{m}_{pw}}{\dot{m}_{sw}} \quad (4)$$

**Specific Energy Consumption (SEC):** Electrical energy consumption ( $SEC_E$ , Equation 5) and thermal energy consumption ( $SEC_T$ , Equation 6) per unit of water produced.

$$SEC_E = \frac{\text{Electric energy input (kWh)}}{\text{Total product water } [\dot{m}_{pw}] (m^3)} \quad (5)$$

$$SEC_T = \frac{\text{Thermal energy input (kWh)}}{\text{Total product water } [\dot{m}_{pw}] (m^3)} \quad (6)$$

### 3.3.2. Proposed Global Performance Metrics

Unlike a conventional MED system, the proposed design integrates a cooling production and ejector-condensation subsystem, which consumes a significant amount of electrical energy. To comprehensively evaluate this cogeneration, this study introduces a set of global metrics. First, the Global Gained Output Ratio ( $GOR_G$ ) and the Global Coefficient of Performance, ( $COP_G$ ), defined in Equations (7) and (8), respectively.

$$GOR_G = \frac{((\dot{m}_{pw} * 2,326 \text{ kJ} / \text{kg}))}{(Q_{GEN} + W_{TOTAL PUMPING})} \quad (7)$$

$$COP_G = \frac{((Q_{EVA} + \dot{m}_{AP} * 2,326 \text{ kJ} / \text{kg}))}{(Q_{GEN} + W_{TOTAL PUMPING})} \quad (8)$$

Additionally, to distinctly evaluate the specific energy consumption, two indicators are introduced: the Global Specific Energy Consumption for desalination ( $SEC_{G\_D}$ ) and the Global Specific Energy Consumption for cooling ( $SEC_{G\_C}$ ), defined in Equations (9) and (10).

$$SEC_{G\_D} = \frac{\text{Thermal energy} + \text{Electric energy (kWh)}}{\text{Total product water } [\dot{m}_{pw}] (m^3)} \quad (9)$$

$$SEC_{G\_C} = \frac{\text{Thermal energy} + \text{Electric energy (kWh)}}{\text{Cooling capacity } [Q_{EVA}] (TR)} \quad (10)$$

### 3.3.3. Exergy Analysis

Given the relevance of the ejector-condenser (EJC) for condensing vapor at 4 °C —a distinctive and scarcely documented mechanism— an exergy analysis based on the second law of thermodynamics was performed. The exergy balance for the EJC is expressed in Equation (11). From this, its exergy efficiency (Equation 12) is defined, which quantifies the fraction of useful exergy recovered.

$$\sum \dot{m}_{in} \dot{ex}_{in} + W_{in} = \sum \dot{m}_{out} \dot{ex}_{out} + W_{out} + \dot{Ex}_{EJC} \quad (11)$$

$$\eta_{ex} = \frac{\dot{ex}_{out}}{(\dot{ex}_{PF} + \dot{ex}_{SF})} \quad (12)$$

## 4. Validation

The model validation was conducted in two independent stages: (i) the high-vacuum multi-effect distillation (MED-HV) subsystem, and (ii) the cooling production and ejector-condensation subsystem. For the MED-HV subsystem, the methodology and reference data reported in (Casares-De la Torre et al., 2024) were followed, which was previously validated against data from **Palenzuela et al.** [31] and **Aly** [23], obtaining maximum errors of 6% and 2%, respectively.

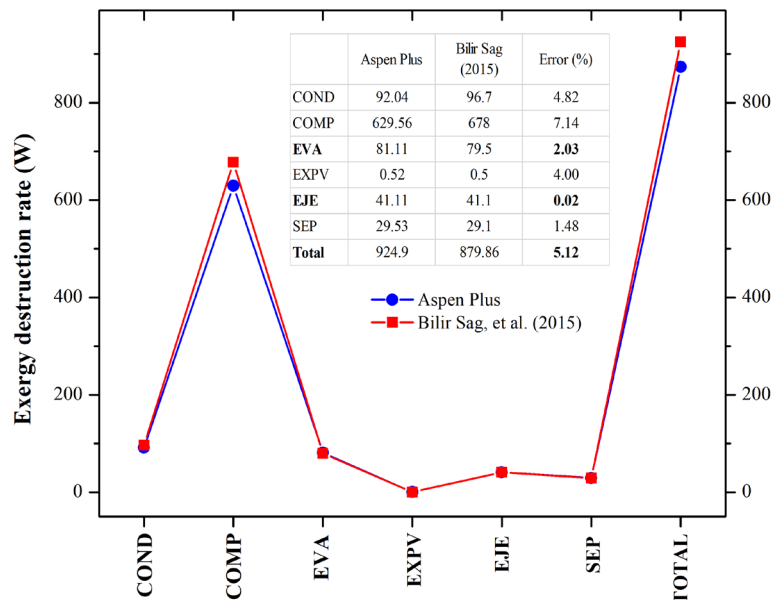
For the cooling production and ejector-condensation subsystem, the ejector-condenser model was validated against the experimental data reported by **Bilir Sag et al.** [32]. Their study on a liquid-vapor two-phase ejector, which uses a liquid motive fluid to compress the secondary vapor,

presents a configuration analogous to that of the proposed ejector-condenser. Consequently, the validation was performed by replicating the operational conditions reported in [32].

The validation results, compared with the experimental data from **Bilir Sag et al.** [32], confirm the model's accuracy. As summarized in Table 4, the first-law parameters —such as discharge pressure and temperature— exhibit discrepancies of less than 3%, while the compressor work, cooling capacity, and COP vary by less than 5%. The largest deviation (4.8%) was recorded in the condenser inlet temperature.

The second-law based analysis, illustrated in Figure 4, reinforces this agreement. The total system exergy destruction rate replicates the experimental data with an average error of 5.1%. The model's precision for key components is particularly noteworthy: the two-phase ejector (EJE), the central element of this research, exhibits a minimal error ( $<0.3\%$ ), and the evaporator an error of 2.03%.

Collectively, these results validate the capability of the Aspen Plus model to accurately simulate the processes of suction, entrainment, and mixing of cold vapor by a liquid primary fluid, satisfying both the first and second laws of thermodynamics.



**Figure 4.** Exergy destruction rate by system component, compared with the results from [32].

**Table 4.** Model validation: comparison of parameters with the experimental data from Bilir Sag et al. [32].

| Parameters                      |       | Aspen<br>Plus | Bilir Sag<br>et. al [32] | Error<br>(%) |
|---------------------------------|-------|---------------|--------------------------|--------------|
| Cooling Capacity                | (kW)  | 4.29          | 4.30                     | 0.34         |
| COP                             | (-)   | 3.29          | 3.18                     | 3.36         |
| Compressor work                 | (kW)  | 1.30          | 1.35                     | 3.76         |
| Condenser (Inlet temperature)   | (°C)  | 57.38         | 57.38                    | 0.00         |
| Condenser (Outlet temperature)  | (°C)  | 37.78         | 37.78                    | 0.00         |
| Ejector (Discharge temperature) | (°C)  | 7.49          | 7.68                     | 2.54         |
| Ejector (Discharge pressure)    | (kPa) | 381.0         | 380.43                   | 0.15         |

## 5. Results and Discussion

This study analyzes the operational behavior and efficiency indicators of the cooling production and ejector-condensation subsystem during the simultaneous production of water and cooling. The analysis is based on the previously established design condition for the complete MED-HV system [27], whose performance was already presented in that reference.

The key control variable is the percentage of product water directed toward the evaporator (EVA) by the three-way valve (3WV), which directly regulates the cooling capacity and determines the mass flow rate of vapor entering the ejector-condenser (EJC) as the secondary fluid. The analysis evaluates the impact of this variable on: 1) the cooling capacity, 2) the internal behavior of the ejector-condenser, and 3) the global indicators of the system's performance.

Figure 5 delineates the thermodynamic feasibility zones for the ejector-condenser (EJC) under two operational modes: **a)** summer mode—simultaneous production of water and cooling—and **b)** winter mode—water production only—. The analysis, based on the second law of thermodynamics, maintained constant conditions for the secondary vapor (Stream 76) with flow

rates of 0.99 kg/s and 0.12 kg/s for each mode, respectively, and evaluated the vapor fraction at the outlet (Stream 80) to verify complete condensation.

The results reveal a fundamental relationship: the exergy destruction is minimized by reducing the ejector-condenser discharge pressure but increases with higher motive fluid pressures (Stream 79). This behavior is attributed to the fact that a higher motive fluid pressure demands a greater primary mass flow rate to achieve condensation, intensifying the frictional losses. Nevertheless, operating with a high motive fluid pressure expands the range of discharge pressures within which complete condensation is achieved in the ejector-condenser.

In summer mode [Figure 5a], under the condition of maximum cooling capacity (100% of the product water used as refrigerant in the evaporator), a critical trade-off point is identified: a discharge pressure of 4.2 kPa represents the minimum pressure level to achieve complete condensation. Operating below this value (e.g., at 4 kPa) increases the exergy destruction, as it would require a prohibitively high primary fluid flow ( $>700$  kg/s) that exceeds the design limit.

Conversely, in winter mode [Figure 5b], the lower vapor load (0.12 kg/s) allows for complete condensation to be achieved under all analyzed conditions. It is shown that a higher secondary vapor flow rate reduces the feasible zone and increases the exergy destruction, a crucial finding that quantifies the impact of the thermal load on the EJC's efficiency. For both modes, it is important to highlight that the ejector-condenser discharge pressure—or back-pressure—must be equal to or less than the critical pressure of the device to ensure its correct operation.

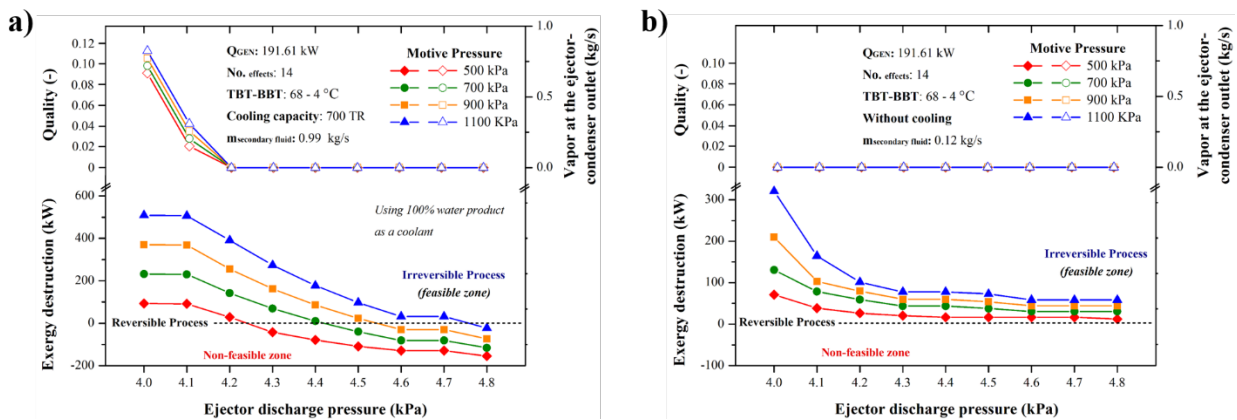


Figure 5. Thermodynamic feasibility zones for the ejector-condenser (EJC).

The feasibility of the ejector-condensation process, from a second-law perspective, is conditional upon positive values of exergy destruction, as delineated in Figure 5. However, within this feasible zone, there is more than one possible operating condition; therefore, the selection of the optimal operating point requires a balance between thermodynamic efficiency and the associated energy consumption. Figure 6 presents this trade-off analysis for summer mode, showing (a) the exergy efficiency and outlet temperature of the EJC, and (b) the total electric pumping power and the motive fluid flow rate, all as a function of the discharge pressure.

The results indicate that as the discharge pressure increases, the total pumping power decreases, since a lower motive fluid mass flow rate is required to achieve vapor condensation. However, this advantage is counteracted by an increase in the outlet temperature of the ejector-condenser. On the other hand, operating with a higher motive fluid pressure—although it increases the consumption of its specific pump—expands the operational range of the system, offering greater design flexibility (ejector sizing). The exergy efficiency of the EJC, therefore, cannot be optimized in isolation but must be evaluated within this context of energy trade-offs.

The final selection of the EJC design conditions must, therefore, balance the exergy efficiency with the pumping power. For this study, a conservative operating point was adopted, setting an exergy efficiency of 43%. This value, indicated in Table 5, aligns with those reported by Šarevski and Šarevski [33] for liquid-vapor ejectors using water (R718) as the working fluid, which range between 0.3 and 0.47. From a second-law perspective, although Figure 6 shows that the EJC can operate over a wide range, the choice of a value within the validated experimental range prioritizes the robustness and practical feasibility of the design.

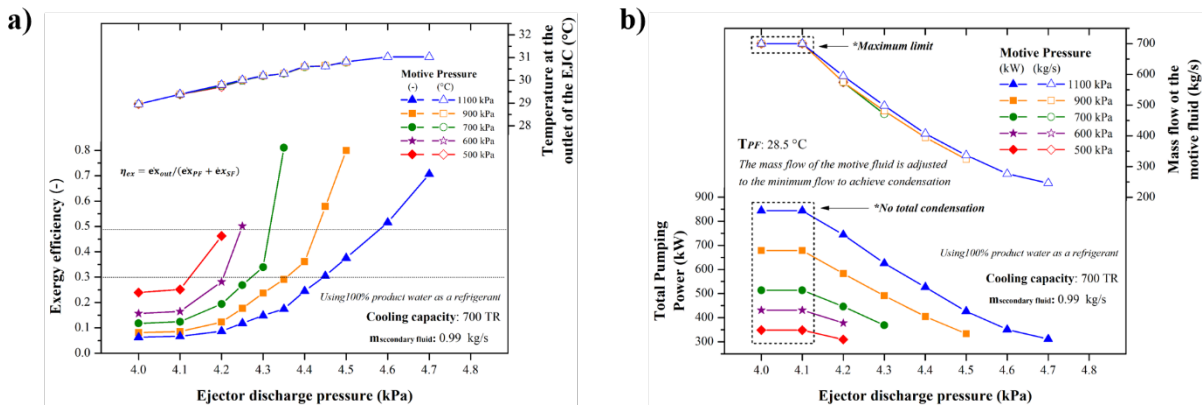


Figure 6. Ejector-condenser operating parameters as a function of discharge pressure.



The proposed design conditions, presented in Table 5, align with those reported for two-phase ejectors operating at high vacuum with water [34] and correspond to the maximum capacity scenario, where 100% of the product water is used as refrigerant (700 TR).

Given that the cooling demand is inherently variable, the system's response was evaluated under five operational scenarios, defined by the fraction of product water directed to the evaporator: 0% (winter mode, desalination only), 25%, 50%, 75%, and 100% (maximum capacity). Figure 7 presents the setup of these scenarios and their corresponding cooling capacity.

For each of these scenarios, a unique design condition was determined within the thermodynamic feasibility zone (Figure 5). The selection criterion was to guarantee total vapor condensation with the pre-established exergy efficiency of  $\eta_{ex} = 43\%$ , while simultaneously minimizing the electric pumping consumption. The resulting ejector-condenser design conditions for each cooling capacity are summarized in Table 6.

The operation of the MED-HV system with a fixed-geometry ejector is controlled by a variable frequency drive (VFD) on the motive fluid pump ( $P_{PW}$ ), a robust strategy for handling moderate load variations. However, when a wider operational range and higher energy utilization are required, it is advisable to explore advanced strategies based on variable-geometry ejectors. Among the solutions reported in the literature, the following stand out: (a) the integration of a VFD with adjustable vortex nozzles, which offer high-efficiency control without intrusive moving parts, and (b) ejectors with a movable spindle, which provide greater operational flexibility at the cost of increased mechanical complexity and maintenance requirements.

**Table 5.** Operational design conditions for the cooling production and ejector-condensation subsystem.

| Parameters        | Units  | Motive fluid       | Secondary fluid | Ejector outlet |
|-------------------|--------|--------------------|-----------------|----------------|
| Pressure          | (kPa)  | 800                | 0.81            | 4.35           |
| Temperature       | (°C)   | 28.50              | 4.00            | 29.81          |
| Mass flow         | (kg/s) | 388.21             | 1.11            | 389.32         |
| <b>Ejector</b>    |        | <b>Motive pump</b> |                 |                |
| Exergy efficiency | (-)    | 0.43               | Efficiency (-)  | 0.85           |

|                   |     |        |            |        |
|-------------------|-----|--------|------------|--------|
| Entrainment Ratio | (-) | 0.0028 | Power (kW) | 352.13 |
|-------------------|-----|--------|------------|--------|

*\*Desalination capacity: 96.03 m<sup>3</sup>/d; Cooling capacity: 700 TR; Heat transferred HX<sub>EJC</sub>: 2,994 kW*

This cooling subsystem introduces a fundamental change in the energy balance of the MED-HV system. Unlike conventional systems [27], where electrical consumption is secondary, the MED-HV configuration—with the  $P_{PW}$  and  $P_{CW}$  pumps—experiences a significant increase in its electrical demand. The present analysis quantifies how this consumption is affected by the increase in cooling capacity, which is proportional to the flow of product water directed to the evaporator.

The increase in cooling capacity entails significant energy requirements. As shown in Figure 8, increasing the fraction of product water in the evaporator requires a significant rise in the mass flow rates of the ejector's motive fluid and the seawater in the  $HX_{SW}$  heat exchanger to dissipate the heat generated during ejector-condensation. This phenomenon translates directly into higher electric power consumption in the  $P_{PW}$  and  $P_{CW}$  pumps, raising the system's total energy demand (Total Pumping Power, TPP).

The analysis of the ejector-condenser under this variable load reveals the cause of this increased consumption. To process the higher flow of refrigerant vapor while maintaining the motive fluid pressure, the system must increase both the ejector's discharge pressure and the primary fluid mass flow rate. This adjustment leads to a 2.89-fold increase in the Total Pumping Power (TPP) when shifting from the 25% to the 100% capacity scenario, an energy cost that outweighs the fourfold gain in cooling capacity.

The impact of these energy requirements is directly reflected in the global efficiency indicators (Figure 9). The Global Specific Energy Consumption for desalination ( $SEC_{G\_D}$ ) increases, primarily attributed to the higher pumping work (TPP) required to ensure the EJC operates in a thermodynamically feasible zone and achieves complete vapor condensation. On the other hand, the Global Gained Output Ratio ( $GOR_G$ ) and the Global Coefficient of Performance ( $COP_G$ ) decrease. The decline in  $GOR_G$  is explained by the fact that the total electrical consumption increases while water production remains constant. The decrease in  $COP_G$  indicates that, although the cooling capacity grows, the increase in the pumps' electrical consumption is proportionally greater.

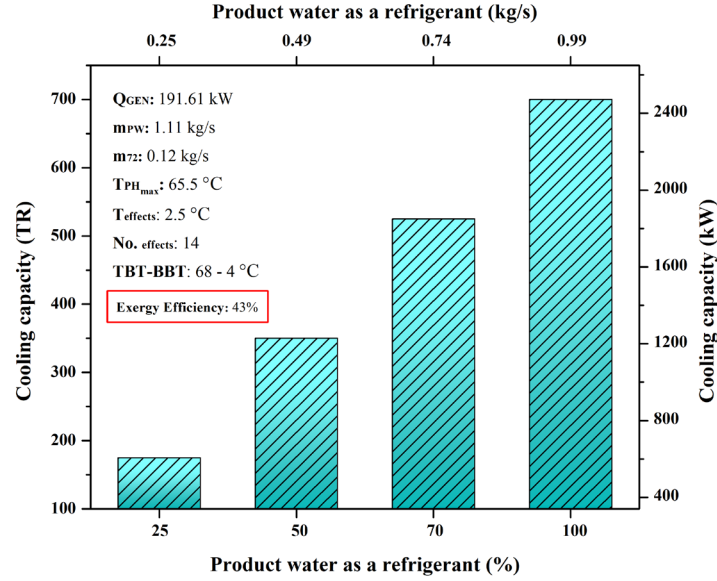


Figure 7. Cooling capacity of the MED-HV system.

These results demonstrate a fundamental trade-off between capacity and efficiency: when increasing the cooling capacity from 25% to 100%, the global performance progressively decreases, evidenced by the reduction in  $COP_G$  and  $GOR_G$  and the increase in  $SEC_{G\_D}$ . Nevertheless, it is important to highlight that, across its entire operational range, the system reports  $GOP_G$  values above 9.2. This performance level surpasses that typically reported in the literature for conventional and hybrid cooling systems. Therefore, the selection of the cooling capacity will depend on the specific cooling demand. The performance indicators of the proposed MED-HV technology and the component energy balances are shown in Table 7.

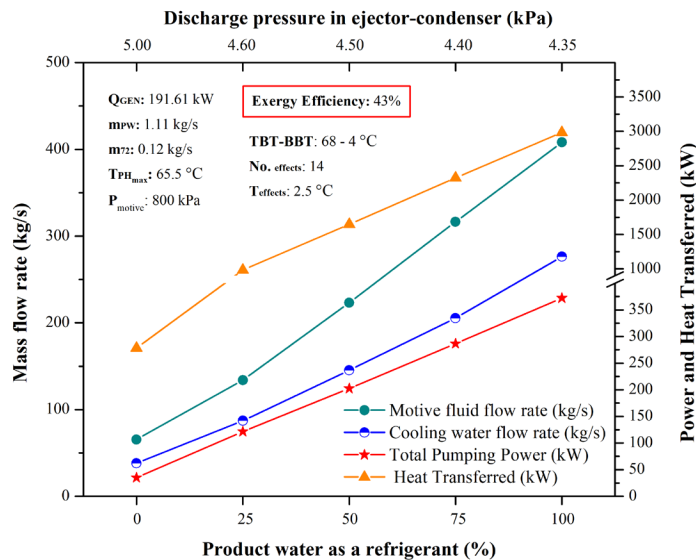


Figure 8. Motive fluid mass Flow rate and electric power required by the pumps.

**Table 6.** Design conditions for different cooling capacities.

| Parameters<br>( $\eta_{ex} = 43\%$ )  | Units | Product water as refrigerant (%) |        |        |        |        |
|---------------------------------------|-------|----------------------------------|--------|--------|--------|--------|
|                                       |       | 0                                | 25     | 50     | 75     | 100    |
| Cooling capacity                      | TR    | -                                | 175    | 350    | 525    | 700    |
| Motive fluid pressure                 | kPa   | 400                              | 800    | 800    | 800    | 800    |
| Discharge pressure                    | kPa   | 5.00                             | 4.60   | 4.50   | 4.40   | 4.35   |
| Secondary fluid mass flow rate        | kg/s  | 0.12                             | 0.38   | 0.62   | 0.86   | 1.11   |
| Primary (motive) fluid mass flow rate | kg/s  | 85.56                            | 184.04 | 273.30 | 336.48 | 388.21 |
| Entrainment ratio                     | (-)   | 0.0014                           | 0.0020 | 0.0023 | 0.0025 | 0.0028 |

The analysis of the cooling capacity (0 to 700 TR) allowed us to observe the behavior of the cooling production and ejector-condensation subsystem. It was found that as the amount of cold vapor to be condensed increases (entrainment ratio), the energy consumption rises due to the increased flow rate and pressure of the motive fluid. Consequently, the efficiency indicators decrease; however, the system maintains a COPG between 9.3–10.3, consistently outperforming conventional technologies.

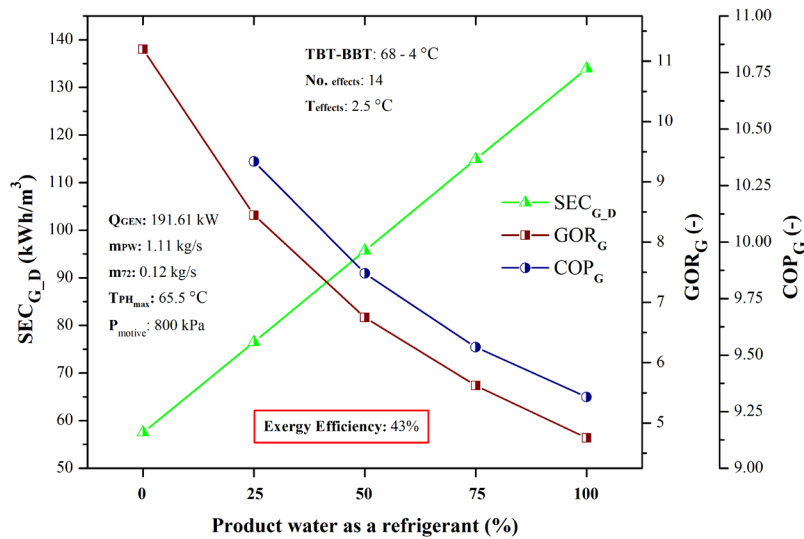
**Figure 9.** Global efficiency indicators when varying the cooling capacity.

Table 7. Energy evaluation of the proposed MED-HV system under design conditions.

| COMPONENTS                     | ENERGY BALANCE                                                                                             | ENERGY GAINS (kW) | ENERGY LOSSES (kW) | INTERNAL ENERGY INTEGRATION (kW)  |
|--------------------------------|------------------------------------------------------------------------------------------------------------|-------------------|--------------------|-----------------------------------|
| GENERATOR (GEN)                | $Q_{GEN} = \sum_{in} \dot{m}_n h_n - \sum_{out} \dot{m}_n h_n$                                             | 191.61            |                    |                                   |
| DISSIPATOR (HX <sub>SW</sub> ) | $Q_{HX_{SW}} = \sum_{in} \dot{m}_n h_n - \sum_{out} \dot{m}_n h_n$                                         |                   | 2,993.60           |                                   |
| EVAPORATOR (EVA)               | $Q_{EVA} = \sum_{in} \dot{m}_n h_n - \sum_{out} \dot{m}_n h_n$                                             | 2450.75           |                    |                                   |
| EFFECTS (HX)                   | $Q_{HX_n} = \sum_{in} \dot{m}_n h_n - \sum_{out} \dot{m}_n h_n$                                            |                   |                    | 2,298.70                          |
| PREHEATERS (PH)                | $Q_{PH_n} = \sum_{in} \dot{m}_n h_n - \sum_{out} \dot{m}_n h_n$                                            |                   |                    | 223.18                            |
| P <sub>SW</sub>                | $W_{P_{SW}} = \dot{m}_1 v_1 (P_2 - P_1) / \eta_{P_{SW}}$                                                   | 0.16              |                    |                                   |
| P <sub>PW</sub>                | $W_{P_{PW}} = \dot{m}_{77} v_{77} (P_{78} - P_{77}) / \eta_{P_{PW}}$                                       | 320.48            |                    |                                   |
| P <sub>CW</sub>                | $W_{P_{CW}} = \dot{m}_{83} v_{83} (P_{84} - P_{83}) / \eta_{P_{CW}}$                                       | 31.49             |                    |                                   |
| FLOW ENTHALPY                  |                                                                                                            |                   |                    |                                   |
| $\dot{m}_2 h_2$                | Seawater inlet                                                                                             | -20,822.31        |                    |                                   |
| $\dot{m}_{73} h_{73}$          | Seawater outlet                                                                                            |                   | -3,202.96          |                                   |
| $\dot{m}_{82} h_{82}$          | Product water outlet                                                                                       |                   | -17,618.58         |                                   |
| TOTAL                          |                                                                                                            | -17,827.81        | -17,827.94         | 2,521.88                          |
| PRODUCT WATER                  |                                                                                                            | 96,033.6 L/d      |                    | Daily production                  |
| RR                             | $RR = \dot{m}_{82} / \dot{m}_1$                                                                            | 0.83              |                    | Recovery Ratio                    |
| GOR                            | $GOR = \frac{(\dot{m}_{82} * 2,326 \text{ kJ} / \text{kg})}{Q_{GEN}}$                                      | 13.49             |                    | Gained Output Ratio               |
| BC1                            | $H_1 = (P_{atm} - P_{discharge}) / (\rho g)$                                                               | 9.998 m           |                    | Barometric Column 1               |
| BC2                            | $H_2 = (P_{atm} - P_{discharge}) / (\rho g)$                                                               | 9.966 m           |                    | Barometric Column 2               |
| BC3                            | $H_2 = (P_{atm} - P_{discharge}) / (\rho g)$                                                               | 9.655 m           |                    | Barometric Column 3               |
| COP <sub>G</sub>               | $COP_G = \frac{((Q_{EVA} + \dot{m}_{82} * 2,326 \text{ kJ} / \text{kg}))}{(Q_{GEN} + W_{TOTAL\_PUMPING})}$ | 9.28              |                    | Global Coefficient of Performance |

### 5.1. General Comparison: Case Study.

To highlight the advantages of the developed system, two comparative exercises were conducted.

#### 5.1.1. First Comparative Exercise

The first exercise compares the proposed MED-HV system with a conventional MED system that also produces cooling, reported by **López-Zavala et al.**, [12]. The fundamental difference lies in the operating conditions: while the proposed system operates at a high vacuum, the conventional system [12] operates at traditional pressure levels, which requires the entire product water stream to be expanded before entering the evaporator.

The comparison was performed under equivalent operating conditions (Table 8), including: product water capacity, number of effects, as well as the temperature and concentration of the feed seawater. For the proposed system, an exergy efficiency of  $\eta_{ex} = 43\%$  was established for the ejector-condenser.

The results presented in Table 9 demonstrate significant advantages of the MED-HV system over the conventional MED system with cooling [12]. The same amount of product water (75.2 m<sup>3</sup>/d) is produced using 13.51% less thermal energy in the generator. Consequently, the GOR and COP reach values of 13.49 and 5.01, respectively, representing an increase of 16.29% and 1.54% compared to the indicators of the system [12]. Likewise, the global indicators that integrate thermal and electrical energy consumption show notable improvements: the GOR<sub>G</sub> and COP<sub>G</sub> increase by 8.21% and 4.93%, respectively, while the SEC<sub>G</sub> decreases by 7.69%. The cooling capacity differs by 5.7%, a direct consequence of the innovative configuration of the MED-HV system, where the vapor from the last effect is directly conveyed to the ejector-condenser for subsequent condensation.

*Table 8. Variables and parameters of the systems evaluated for the first comparison.*

| Parameters                     | Units             | MED [12]<br>(cooling) | MED-HV<br>(cooling) |
|--------------------------------|-------------------|-----------------------|---------------------|
| Desalination capacity          | m <sup>3</sup> /d | 75.2                  | 75.2                |
| Product water as refrigerant   | %                 | 100                   | 100                 |
| Number of effects              | -                 | 14                    | 14                  |
| Number of preheaters           | -                 | 13                    | 8                   |
| Top Brine Temperature (TBT)    | °C                | 70                    | 68                  |
| Bottom Brine Temperature (TBT) | °C                | 35                    | 4                   |
| Feed water temperature         | °C                | 25                    | 25                  |
| Feed water concentration       | ppm               | 35,000                | 35,000              |
| Feed water mass flow rate      | kg/s              | 1.63                  | 1.05                |

*Table 9. Results from the comparative table of the first comparison exercise.*

| Parameters                 | Units | MED [12]<br>(cooling) | MED-HV<br>(cooling) | (Δ%)   |
|----------------------------|-------|-----------------------|---------------------|--------|
| Q <sub>GEN</sub>           | kW    | 173.43                | 150                 | -13.51 |
| W <sub>SW</sub>            | kW    | 0.92                  | 0.13                | -85.87 |
| W <sub>CW</sub>            | kW    | 16.21                 | 31.89               | 96.73  |
| W <sub>PW</sub>            | kW    | 244.15                | 221.67              | -9.21  |
| W <sub>TOTAL_PUMPING</sub> | kW    | 261.28                | 253.69              | -2.90  |
| Q <sub>EVA</sub>           | kW    | 2,042.87              | 1,926.36            | -5.70  |
| COP                        | -     | 4.70                  | 4.77                | 1.54   |
| GOR                        | -     | 11.60                 | 13.49               | 16.29  |
| RR                         | -     | 0.53                  | 0.83                | 56.60  |

|              |           |           |            |        |
|--------------|-----------|-----------|------------|--------|
| $X_{B14}$    | ppm       | 74,575.00 | 204,314.09 | 173.97 |
| $GOR_G$      | -         | 4.63      | 5.01       | 8.21   |
| $COP_G$      | -         | 9.33      | 9.79       | 4.93   |
| $SEC_{G\_D}$ | $kWh/m^3$ | 139.60    | 128.87     | -7.69  |

### 5.1.2. Second Comparative Exercise

The second exercise evaluates the proposed integrated MED-HV system against a conventional hybrid configuration: a conventional MED desalination system (Palenzuela et al., 2014) and a commercial centrifugal chiller (Carrier AquaEdge 19DV 550 TR) [35] operating independently. The latter reports a COP of 6.1 at full load under AHRI conditions. The comparison, whose parameters are detailed in Tables 10 and 11, is based on the same water production ( $71.7 \text{ m}^3/\text{d}$ ) and the same cooling capacity (521 TR).

**Table 10.** Variables and parameters of the desalination systems for the second comparison.

| Parameters                     | Units                 | MED [31]<br>Palenzuela | MED-HV<br>(cooling) |
|--------------------------------|-----------------------|------------------------|---------------------|
| Desalination capacity          | $\text{m}^3/\text{d}$ | 71.7                   | 71.7                |
| Number of effects              | -                     | 14                     | 14                  |
| Number of preheaters           | -                     | 13                     | 8                   |
| Top Brine Temperature (TBT)    | $^{\circ}\text{C}$    | 68                     | 68                  |
| Bottom Brine Temperature (BBT) | $^{\circ}\text{C}$    | 35                     | 4                   |
| Feed water temperature         | $^{\circ}\text{C}$    | 25                     | 25                  |
| Feed water concentration       | ppm                   | 35,000                 | 35,000              |
| Feed water mass flow rate      | $\text{kg/s}$         | 2.22                   | 1.00                |



*Table 11. Variables and parameters of the cooling systems for the second comparison.*

| Parameters       | Units | Carrier       | MED-HV    |
|------------------|-------|---------------|-----------|
|                  |       | AquaEdge [35] | (cooling) |
| Cooling capacity | TR    | 521           | 521       |
| Cooling capacity | kW    | 1,834.95      | 1,834.95  |
| COP              | -     | 6.91          | -         |
| Refrigerant type | -     | R-1233zd      | Agua      |
| Electric power   | kW    | 265.54        | 221.06    |

The results, summarized in Table 12 for a continuous 24-hour operation, demonstrate the advantages of the proposed MED-HV system. Firstly, to achieve the same production of water and cooling, the MED-HV system reduces thermal energy consumption by 25.43% and electrical energy consumption by 11.19% compared to the hybrid scenario.

Consequently, it achieves a GOR of 13.49, equivalent to a 34.1% increase compared to the conventional MED system (Palenzuela et al., 2014). The  $COP_G$  indicator reaches a value of 9.22, corresponding to a 20.73% increase compared to the hybrid scenario, evidencing the efficiency of simultaneous production. Furthermore, the specific indicators of electrical and thermal consumption per cubic meter of water ( $SEC_E$  and  $SEC_T$ ) are favorably reduced:  $SEC_E$  (79.26 kWh/m<sup>3</sup>) by 11.19% and  $SEC_T$  (47.88 kWh/m<sup>3</sup>) by 25.43%.

The global indicators  $SEC_{G\_D}$  (127.15 kWh/m<sup>3</sup>) and  $SEC_{G\_C}$  (0.73 kWh/TR) confirm a 17.15% reduction in total integrated energy consumption compared to the commercial hybrid configuration.

*Table 12. Results from the comparative table of the second comparison exercise.*

| Parameters                | Units     | MED + Carrier<br>AquaEdge | MED-AV<br>(cooling) | ( $\Delta\%$ ) |
|---------------------------|-----------|---------------------------|---------------------|----------------|
| $Q_{GEN}$                 | kW        | 191.61                    | 142.89              | -25.43         |
| $W_{P\_SW}$               | kW        | 0.80                      | 0.12                | -84.65         |
| $W_{P\_CW}$               | kW        | -                         | 29.99               | -              |
| $W_{P\_PW}$               | kW        | -                         | 206.42              | -              |
| $W_{TOTAL\_PUMPING}$      | kW        | 0.80                      | 236.53              | -              |
| $W_{ELECTRICAL\_COOLING}$ | kW        | 265.54                    | -                   | -              |
| $W_{ELECTRICAL\_TOTAL}$   | kW        | 266.34                    | 236.53              | -11.19         |
| GOR                       | -         | 10.06                     | 13.49               | 34.10          |
| RR                        | -         | 0.38                      | 0.83                | 118.14         |
| $X_{B14}$                 | ppm       | 69,326.31                 | 204,584.06          | 195.10         |
| $COP_G$                   | -         | 8.22                      | 9.22                | 20.73          |
| $SEC_E$                   | $kWh/m^3$ | 89.25                     | 79.26               | -11.19         |
| $SEC_T$                   | $kWh/m^3$ | 64.21                     | 47.88               | -25.43         |
| $SEC_{G\_D}$              | $kWh/m^3$ | 153.46                    | 127.15              | -17.15         |
| $SEC_{G\_C}$              | $kWh/TR$  | 0.88                      | 0.73                | -17.15         |

## 6. Conclusions

This article demonstrates the technical feasibility of an innovative multi-effect distillation (MED) system that, for the simultaneous production of water and cooling, incorporates an ejector-condenser and an evaporator, enabling it to operate at a high vacuum (0.8 kPa) and utilize the product water as a refrigerant.

The proposed configuration and high-vacuum operating conditions of the system allowed for the use of product water as a refrigerant, resulting in a technology that produces 96.03 m<sup>3</sup>/d of water and a remarkable 700 TR (2,460 kW) of cooling. This output can satisfy the water and space conditioning needs of communities, hotels, commercial establishments, and other entities located in coastal areas.

The proposed methodological procedure for the thermodynamic feasibility study of the new cogeneration system for water and cooling allowed for the definition of an operational zone and the optimal design conditions for the ejector-condenser. This was based on achieving complete condensation of the cold vapor, an exergy efficiency of 43% within the feasible range, and the minimum energy consumption of the pumps (Total Pumping Power, TPP).

The analysis of the cooling capacity (0 to 700 TR) allowed us to observe the behavior of the cooling production and ejector-condensation subsystem. It was found that as the amount of cold vapor to be condensed increases (entrainment ratio), the energy consumption rises due to the increased flow rate and pressure of the motive fluid. Consequently, the efficiency indicators decrease; however, the system maintains a COP<sub>G</sub> between 9.3–10.3, consistently outperforming conventional technologies.

Varying the cooling capacity with the ejector-condenser presents the challenge of operating away from the design point. A proposed solution is to incorporate a variable frequency drive on the motive fluid pump to adjust the flow and pressure, combined with other control options such as an adjustable vortex nozzle or a movable spindle.

The comparison with existing systems confirms the superiority of the proposed design: Compared to an MED system with integrated cooling [12], the proposed system reduces thermal consumption by 13.51% and improves the GOR and GOR<sub>G</sub> by 16.29% and 8.21%, respectively. When compared to a hybrid system (MED + centrifugal chiller) [31,35], it achieves a 25.43% reduction in thermal consumption, a 11.19% reduction in electrical consumption, and a 25.83% improvement in COP<sub>G</sub> (9.22) alongside a 20.73% reduction in global specific energy consumption (SEC<sub>G\_D</sub> and SEC<sub>G\_C</sub>).

In conclusion, the proposed MED-HV system is established not only as a technically viable alternative but as an energetically superior solution for the simultaneous production of water and cooling. These results pave the way for a new generation of thermal desalination and cooling technologies with enhanced capacity and efficiency.

### **Declaration of competing interest**

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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### **3.3. Artículo C: Evaluation of concentration polarization due to the effect of feed water temperature change on reverse osmosis membranes.**

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Este estudio experimental aborda un desafío operativo crítico en la tecnología de desalinización que predomina en cuestiones de capacidad instalada a nivel global: la Ósmosis Inversa (RO). La investigación se centra en el fenómeno de la polarización de la concentración, un problema que reduce el área efectiva de transferencia de masa en las membranas y aumenta el consumo energético del proceso. El objetivo principal fue cuantificar experimentalmente la relación entre la temperatura del agua de alimentación y la polarización de la concentración ( $\beta$ ) para diferentes salinidades, con el fin de identificar estrategias operativas que reduzcan el impacto en el consumo energético específico ( $\text{kWh/m}^3$ ). Los principales hallazgos del estudio son los siguientes:

- Se determinó que el aumento de temperatura del agua de alimentación (de 23 °C a 35 °C) reduce la polarización de la concentración en un 3.10% para 5,000 mg/L y en un 3.64% para 10,000 mg/L, manteniendo todos los valores de  $\beta$  por debajo del límite crítico de 1.2 que previene las incrustaciones.
- Este beneficio se atribuye a un incremento del 31.68% en la difusividad y del 22.71% en el coeficiente de transferencia de masa, lo que mejora la eficiencia del proceso.
- La optimización energética fue significativa: el consumo específico de energía se redujo en un 12.5% ( $0.59 \text{ kWh/m}^3$ ) para 5,000 mg/L y en un 14.54% ( $0.72 \text{ kWh/m}^3$ ) para 10,000 mg/L.
- Se identificó que la temperatura óptima de operación para aguas de 10,000 mg/L es de 25.5 °C, equilibrando la máxima eficiencia energética con la calidad del permeado.

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## **Artículo C**

### **Evaluation of concentration polarization due to the effect of feed water temperature change on reverse osmosis membranes**

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## Article

# Evaluation of Concentration Polarization Due to the Effect of Feed Water Temperature Change on Reverse Osmosis Membranes

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**Abstract:** Water is a necessary resource for life development. Its excessive consumption has a negative impact, generating scarcity problems worldwide. Desalination is an alternative to solve these problems; its objective is to reduce the concentration of total dissolved solids to levels suitable for consumption. The most widely used desalination technology is reverse osmosis, which works by means of semipermeable membranes; however, lack of knowledge or wrong operation cause phenomena such as concentration polarization, which reduces the effective area for mass transfer in the membrane, increasing the energy consumption of the process. The objective of the present study is to evaluate the concentration polarization ( $\beta$ ) of the concentration in reverse osmosis membranes by varying the temperature in the feed water (23, 25.5, 28, and 35 °C) for different concentrations (5000 and 10,000 mg L<sup>-1</sup>) in order to reduce its impact on energy consumption (kWh m<sup>-3</sup>). The results show that as the temperature increases, the specific energy consumption decreases for both concentrations. In the 5000 mg L<sup>-1</sup> tests, the specific energy consumption decreased by 0.590 kWh m<sup>-3</sup>, representing 12.5%. For 10,000 mg L<sup>-1</sup> tests, the specific energy consumption shows a reduction of 0.72 kWh m<sup>-3</sup>, which represents a percentage decrease of 14.54%.

**Keywords:** desalination; concentration polarization; reverse osmosis; water scarcity

## 1. Introduction

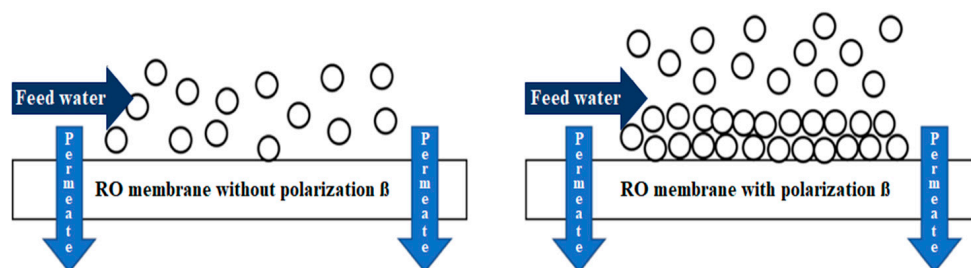
The accelerated population growth and industrialization present during the last decades around the world have significantly increased the demand for resources; therefore, it is becoming more difficult to meet the basic needs of human beings, limiting the availability of drinking water, food, and electrical energy [1]. Although water covers most of the Earth, only about 2.5% of it is available as fresh water. However, the percentage of fresh water available for consumption is much lower because nearly three-quarters of this resource is frozen in glaciers and ice caps, making its use in the short term practically impossible [2,3].

It is estimated that by 2030 there will be a 40% water deficit worldwide [4]. For example, Mexico currently has 94% of its rivers and lakes polluted, dozens of cities with insufficient drinking water, and 50% of its water wasted; this country has been going through a critical water shortage situation for several years, and it has only been increasing [5,6]. About 77%

of the national population, located in the arid and semi-arid states of the northern region of the country, are the most affected in terms of lack and total scarcity of water.

Desalination technologies appear to be an attractive solution to satisfy part of the global water shortage. Desalination consists of taking water with a high concentration of total dissolved solids (TDS) and passing it through a treatment process until it reaches the appropriate concentration ranges for its respective use [7]. In order to obtain fresh water through desalination, two types of systems can be used, which are categorized as: thermal and membrane systems [8]. The latter system separates the feed water into two streams, one of water with low concentration called *-permeate-* and another with the concentrate of the rejected salts called *-rejection-* [9]. Within this category of membrane desalination systems, the most used technology is reverse osmosis (RO).

However, certain factors require special attention during the RO process. One of these factors is the so-called *-concentration polarization factor-*, which causes a series of phenomena that affect RO membranes directly and, consequently, the quality of the product water and the energy consumption of the process [10]. The concentration polarization occurs when, as feed water flows through the RO system, salts are retained by the semipermeable membrane and form a concentrated layer on its surface, as shown in Figure 1. The concentration at this point gradually increases because as the water flow continues to pass through the membranes, salts continue to accumulate, causing an even higher concentration than the inlet concentration *-scaling-* [11]. This phenomenon limits mass transfer in the process, causing an increase in; operating pressure, energy consumption, and scaling formation. It also decreases the quality and flux of the permeate [12].



**Figure 1.** Concentration polarization in reverse osmosis membranes.

McGovern and Lienhard [13] conducted a study with the aim of providing an explanation for the flux asymptote that arises due to concentration polarization when employing ultra-permeable reverse osmosis membranes used to desalinate seawater. The authors report a series of analytical formulas that clarify this limit for both single-stage and batch processes. These asymptotes illustrate that, although improvements in flux are not linear as permeability increases, the limit imposed on the mean flux by transport is approximately four times the current mean flux for seawater desalination and approximately twenty times the current mean flux for brackish water desalination.

Some correlations, new models, and/or the improvement of existing ones have been proposed in order to evaluate the behavior of concentration polarization in reverse osmosis membranes. Rathore et al. [14] developed a transport model that focuses on evaluating the formation of the concentration polarization layer and its behavior in reverse osmosis membranes. Their results show that as filtration increases, the amount of rejected solutes gradually increases; this is reflected in an increased thickness of the concentration polarization layer and a higher rate of mass transfer by counter diffusion in the boundary layer, the latter causing an accumulation of backward transported solutes resulting in higher accumulated concentrations along the boundary layer. Al-Mutaz and Alsubaie [15] combined the solution diffusion transport model and film theory to obtain an explicit expression for water flux through reverse osmosis membranes, and the authors mention that the formula will be helpful for the evaluation of the concentration polarization using parameters such as; permeability coefficients of water and solute, operating conditions and mass transfer coefficient. Their model was compared with data reported in the literature and showed

a good agreement. Gu et al. [16] presented new dimensionless correlations for the concentration polarization and the friction factor obtained by analysis of computational fluid dynamics simulation data. Their results obtained with the new correlations are compared with those of the existing correlations in the literature; the authors report that there is good consistency in the predicted concentration polarization, with mean discrepancies of less than 6%.

Baghdadi et al. [17] theoretically investigated by means of FilmTec's software the effect of temperature on the thickness of the concentration polarization layer in two types of membranes, with ex situ macromolecules and in situ macromolecules. The authors report that the permeate flux does not vary significantly as the temperature increases. However, the concentration polarization layer thickness decreases as the temperature increases and to a greater extent when the macromolecules are embedded in the membrane (in situ), while this decrease was less significant when the macromolecules act as a separate layer on the membrane surface (ex situ).

Li et al. [18] studied the performance of the miniature form of a scaled-up nanoporous membrane centrifuge for reverse-osmosis desalination. They studied the critical angular velocity for reverse osmosis desalination, concentration polarization, and energy efficiency against different design parameters; pore size, porous CNT centrifuge length, and freshwater flux rate. The authors report that the ion concentration density is almost zero at the membrane wall, implying that concentration polarization may not exist for their proposed nanoporous membrane centrifuge.

Zhang et al. [19] discovered a previously unknown mechanism that breaks the permeability–selectivity trade-off in using a rotating nanoporous graphene membrane with pores of two to four nanometers in diameter. Their results show that the rotating membrane exhibits almost 100% salt rejection even when the pore size is larger than that of hydrated ions. The authors report that no concentration polarization is observed from the radial concentration distribution and radial density distributions of water molecules and ions obtained from the molecular dynamics simulations.

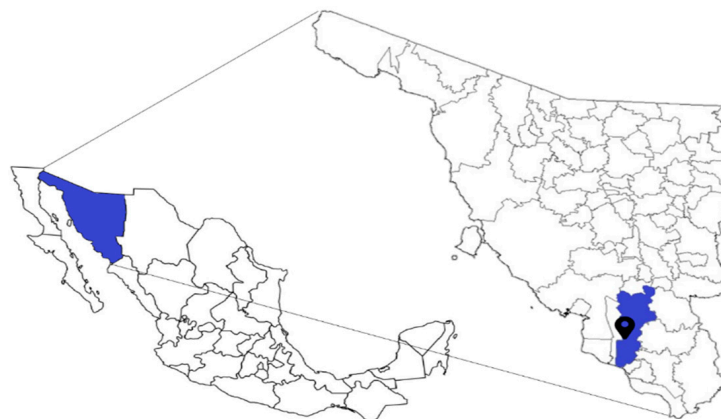
Some recent work has focused on reducing concentration polarization with different types and configurations of spacers in the feed channel of reverse osmosis membranes. Ruiz-García and Pestana [20] simulated the effect of different spacer geometries in the feed channel for reverse osmosis membranes used to desalinate brackish water with different coefficients of permeability. Their results show that the geometry of the spacers has a greater impact on membranes with a higher coefficient of permeability. Wei et al. [21] simulated concentration polarization behavior in spirally wound reverse osmosis membranes. The authors report that for a concentration of  $10,000 \text{ mg L}^{-1}$  of salt and a pressure of 10.91 bar in the feed water, the amount of permeate went from  $27.6 \text{ L/m}^2\text{h}$  to  $24.1 \text{ L/m}^2\text{h}$ , as a result of concentration polarization in the absence of a spacer. On the contrary, the presence of the spacer decreased the concentration polarization, which increased the flow at the outlet up to  $26.5 \text{ L/m}^2\text{h}$ . Sitaraman and Battiato [22] reported that with a non-uniform spacer configuration, substantial savings in pressure drop ( $\sim 40\%$ ), minor variations ( $\sim 2\%$ ) in concentration polarization are obtained, product water quality ( $\sim 1\%$ ), and water recovery ( $\sim 7\%$ ) compared to a uniform spacer configuration.

To summarize, it is clear that the study of polarization concentration reduction is a topic that is currently being addressed in different ways; therefore, models and simulations have been developed to understand its effects on the performance of RO systems and how to minimize them. Although it has been reported that some recently proposed novel membranes do not face this problem, it is still of interest to develop techniques to minimize it in RO systems that currently operate with conventional membranes, which are approximately 17,000 plants, equivalent to 81% of the total existing plants [23]. From a comprehensive review of the state of the art, the authors discovered that there is currently a lack of information on procedures that work to minimize concentration polarization due to increases in feedwater temperature to an RO process experimentally. For this reason, the present research study focuses on the evaluation of concentration polarization in RO

membranes by varying the feed water temperature under different TDS concentrations in order to increase product water quality and reduce the energy consumption of the process.

## 2. Materials and Methods

Experiments on the effect of varying feed water temperature (23, 25.5, 28, and 35 °C) and inlet water concentration (5000 and 10,000 mg L<sup>-1</sup>) on the concentration polarization in the RO membranes were conducted at the Marine and Brackish Water Desalination Laboratory using Renewable Energies located at the Instituto Tecnológico de Sonora campus Náinari in Ciudad Obregón, Sonora with coordinates of 27°29′11.11″ N and 109°56′26.99″ W (Figure 2).



**Figure 2.** Location of the study area.

The delimitations of the present experimental study are established according to the capacity of the RO water desalination plant, which is 10 L min<sup>-1</sup> and has a 260 L storage tank. Working with a recovery rate of 50%, the experimental tests have a maximum duration of 20 min. Another conditioning factor is that RO membranes have a maximum temperature resistance between 40 and 45 °C since these temperatures promote the proliferation of biofouling. Therefore, it is proposed to work with temperatures of 23, 25.5, 28, and 35 °C. Two electrical heating elements of 1100 W each were used to heat the water. Infrared equipment was used to measure the temperature during the whole experiment. The residence time throughout the reverse osmosis process is 20–24 min. In order to control the temperature of the water inside the feed tank that will enter the reverse osmosis process, a stirrer was used for 2 min to homogenize the total mixture of the 260 L of water in the feed tank. If the temperature is within the different required ranges of 23, 25.5, 28, and 35 °C, the test is started according to the established design of experiments. A tolerance of ±0.5 °C was considered for the tests and for the temperature setting considered. For the preparation of each sample, the 260 L feed tank was filled with water from the city network of approximately 270 mg L<sup>-1</sup>, and the temperature was measured using a Fluke Model 62 Mini infrared thermometer; in cases where it was necessary to lower the water temperature, cooling systems were used to bring it to the desired temperature.

Subsequently, the electrical conductivity was measured with YSI Model 556 multiparameter equipment and by means of Equation (1).

$$TDS = EC * 644 \quad (1)$$

where *TDS* is the concentration of the feed water (mg L<sup>-1</sup>) and *EC* is the electrical conductivity (mS cm<sup>-1</sup>).

Once the feed water was characterized, the corresponding amount of synthetic sea salt (mg) of the Reef Crystals Reef Salt type manufactured by Instant Ocean, MN, USA, was



added, which was calculated using Equation (2). Subsequently, the solution was mixed by stirring for approximately 5 min.

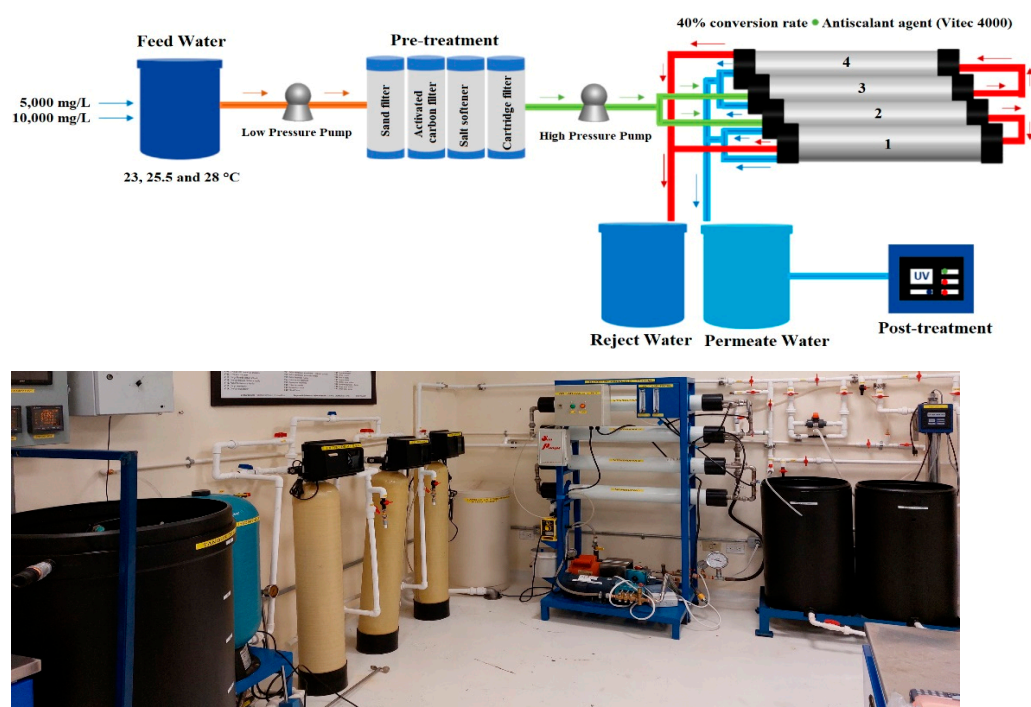
$$\text{Salt} = C_2 V_1 - C_1 V_1 \quad (2)$$

where  $C_1$  is the concentration of the city network water ( $\text{mg L}^{-1}$ ),  $V_1$  is the volume of the feed tank in (L), and  $C_2$  is the desired concentration ( $5000 \text{ mg L}^{-1}$  or  $10,000 \text{ mg L}^{-1}$ ).

In order to protect the RO membranes and ensure the reliability of the experiment, chemical washes were performed with 50 g of Avista Technologies RoClean P112 cleaner, San Marcos CA, USA, diluted in 260 L of city network water, representing a 19.2% concentration. This was followed by a wash with Avista brand Vitec 4000 membrane antifouling agent, San Marcos CA, USA. One liter of solution was prepared where 300 mL corresponded to the chemical while the remaining 700 mL were distilled water, representing a 30% concentration. The liter of antiscalant prepared was supplied to a tank of 3.78 L in total capacity so that the Milton Roy, Pasadena, CA, USA dosing pump injected the solution to the feed water in a combination of 40% stroke and 40% pulsation, in such a way that it promoted the dragging of salts and biofouling out of the system through the reject water.

### 2.1. Description of the Reverse Osmosis System

A  $10 \text{ L min}^{-1}$  RO desalination plant was used, consisting of three stages: physical and chemical pretreatment, treatment, and post-treatment. The design is shown in Figure 3. The feed water circulated through the three pretreatment filters using an ESPA model 2025SL low-pressure pump. The first filter to pass through was the sand filter, which retained particles larger than  $50 \mu\text{m}$ , followed by the activated carbon filter, which removed the chlorine and organic matter present in the water and removed the odor and color of the water. Both the sand filter and the activated carbon filter are Pentair Structural Poly Glass  $8'' \times 44''$  filters with a capacity of  $2.12 \text{ m}^3$  each. Finally, the water was passed to the Aquor PP Sedimeter Filter  $20''$   $5 \mu\text{m}$  SFCPP-205 cartridge filter.



**Figure 3.** Reverse osmosis desalination system. Above schematic; below real [24].

A Cat Pumps Model 231 high-pressure pump was used to convey the water to the treatment stage of the reverse osmosis plant. Four Hydranautics SWC4 Max  $8'' \times 40''$  spiral wound composite polyamide membranes placed in a frame were used for the experiment.

These seawater membranes have an average salt rejection rate of 99.8%, withstand a maximum pressure of 8273.7 kPa, and a maximum temperature of 45 °C, the active area of each membrane is 40.8 m<sup>2</sup>, and a permeate flux of 27.1 m<sup>3</sup> d<sup>-1</sup>. The membranes are classified as 1, 2, 3, and 4 from the lower to the upper membrane. Membranes 2–3 are in parallel with each other; however, 1–2 and 3–4 are arranged in series, as shown in Figure 3.

During the process, Vitec 4000 antiscalant was injected into the system before passing through the high-pressure pump to reach the RO membranes. The permeate and reject volumetric flow rates (L min<sup>-1</sup>) were measured by rotameters. The volumetric flow rate ratio allowed the calculation of the recovery percentage by means of Equation (3). In addition, the permeate concentration and salt rejection were calculated using Equations (4) and (5). In the last stage of the process, the permeate water that came out of the rotameter was directed through the pipes to the Wedeco UV Technologies Inc. ultraviolet light lamp NLR1845WS, New York, NY, USA, and then stored in the permeate tank (100 L).

The percentage of recovery (%R) is determined with:

$$\%R = \frac{Q_p}{Q_f} * 100 \quad (3)$$

where  $Q_p$  is the permeate flow rate (m<sup>3</sup> s<sup>-1</sup>) and  $Q_f$  is the feed flow rate (m<sup>3</sup> s<sup>-1</sup>)

The permeate concentration ( $C_p$ ) and salt rejection ( $R_s$ ) were determined with:

$$C_p = (1 - R_s) * \left( \frac{C_f + C_R}{2} \right) \quad (4)$$

$$R_s = \frac{(C_f - C_p)}{C_f} \quad (5)$$

where  $C_p$  is the permeate concentration (mg L<sup>-1</sup>),  $R_s$  is the salt rejection (%),  $C_f$  is the feed concentration (mg L<sup>-1</sup>) and  $C_R$  is the rejection concentration (mg L<sup>-1</sup>).

## 2.2. Experiment Description

The experiment consists of 6 combinations with 3 replicates each in order to analyze the effect of concentration polarization ( $\beta$ ) at different temperatures and concentrations in the feed water. Therefore, a total of 18 experimental tests were carried out. The 18 experimental tests were carried out at a conversion percentage of 50% so that for every 10 L min<sup>-1</sup> of feed water, 5 L min<sup>-1</sup> of permeate water and 5 L min<sup>-1</sup> of reject water were obtained.

## 2.3. Experiment Description

For the analysis of the concentration polarization ( $\beta$ ), data were measured on the pressures of the low- and high-pressure pumps (kPa), volumetric flow rates (L min<sup>-1</sup>), temperatures (°C), and densities (kg m<sup>-3</sup>) of the feed, permeate and reject water, as well as their concentrations (mg L<sup>-1</sup>) at different stages of the RO desalination process. From the measured data and using Equations (6)–(11), inherent variables were calculated, such as the mass transfer coefficient and diffusivity, which were used for the determination of the concentration polarization and its analysis.

The mass transfer coefficient ( $k_{mt}$ ) is determined with:

$$k_{mt} = \frac{Sh * D_f}{d_h} = \frac{0.023Re^{0.8}Sc^{1/3}D_f}{d_h} \quad (6)$$

where  $k_{mt}$  is the mass transfer coefficient (m s<sup>-1</sup>),  $Sh$  is the sherwood number (-),  $D_f$  is the feed diffusion,  $d_h$  is the hydraulic diameter,  $Re$  is the Reynolds number (-) and  $Sc$  is the Schmith number (-).

The diffusivity of the feed water ( $D_f$ ) is determined with:

$$D_f = \frac{(1.173 \times 10^{-15}) * T_p * (2.26 * 58.44)^{-0.5}}{0.02^{0.6} * \mu_p} \quad (7)$$

where  $T_p$  is the temperature of the permeate water ( $^{\circ}\text{C}$ ) and  $\mu_p$  is the viscosity of the permeate water (Pa s).

The permeate flux rate ( $J_p$ ) is determined with:

$$J_p = \frac{Q_p}{A_m} \quad (8)$$

where  $J_p$  is the permeate flux ( $\text{m s}^{-1}$ ),  $Q_p$  is the permeate flow rate ( $\text{m}^3 \text{s}^{-1}$ ) and  $A_m$  is the membrane area ( $\text{m}^2$ ).

The Sherwood [25,26] and Reynolds number were determined with:

$$Sh = \frac{k_{mt} d_h}{D_f} = 0.023 Re^{0.8} Sc^{1/3} \quad (9)$$

$$Re = \frac{\rho v d_h}{\mu_p} \quad (10)$$

where  $\rho$  is the permeate density ( $\text{kg m}^{-3}$ ),  $v$  is the water velocity ( $\text{m s}^{-1}$ )

The Schmidth number and concentration polarization are determined with:

$$Sc = \frac{\mu_f}{\rho D_f} \quad (11)$$

$$\beta = \exp\left(\frac{J_p}{k_{mt}}\right) \quad (12)$$

where  $\mu_p$  is the feed water viscosity (Pa s) and  $\beta$  is the concentration polarization (-)

The temperature correction factor ( $TCF_p$ ) for permeate ( $TCF_p$ ) and solute ( $TCF_s$ ) are determined by the following equations [27]:

$$TCF_p = \exp[0.0343 (T - 25)] \quad < 25^{\circ}\text{C} \quad (13)$$

$$TCF_p = \exp[0.0307 (T - 25)] \quad > 25^{\circ}\text{C} \quad (14)$$

$$TCF_s = 1 + 0.05 (T - 25) \quad < 25^{\circ}\text{C} \quad (15)$$

$$TCF_s = 1 + 0.08 (T - 25) \quad > 25^{\circ}\text{C} \quad (16)$$

Temperature affects both the osmotic pressure and the permeability of water across the membrane [28].

#### 2.4. Initial Investment and Permeate Water Costs

Using DesalData 2021 data [29], the initial investment costs in brackish water reverse osmosis (BWRO) plants and seawater reverse osmosis (SWRO) plants for Mexico have been estimated. The relationship between the investment cost (\$/USD) and the installed capacity ( $\text{m}^3 \text{d}^{-1}$ ) of desalination plants using brackish and seawater as input water to the process will be established. This will be of great help to decision makers since it will be possible to estimate the initial investment cost or size of the current or future plant to be installed.

### 3. Results

#### 3.1. Feed Water Characterization

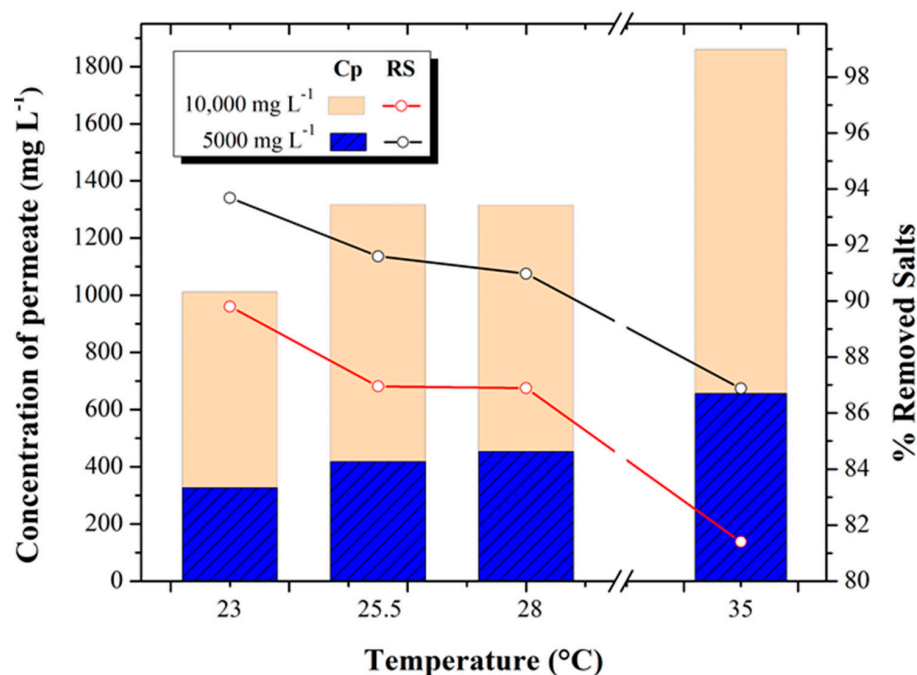
The physicochemical parameters of the feedwater from the 5000 and 10,000 mg L<sup>-1</sup> tests prepared in the laboratory are shown in Table 1.

**Table 1.** Physicochemical characterization of feed water.

| Parameters                                       | 5000 mg L <sup>-1</sup> | 10,000 mg L <sup>-1</sup> |
|--------------------------------------------------|-------------------------|---------------------------|
| Volume (L)                                       | 260.00                  | 260.00                    |
| TDS (mg L <sup>-1</sup> )                        | 5013.00                 | 10,003.00                 |
| pH                                               | 7.80                    | 7.90                      |
| Electrical conductivity (mS cm <sup>-1</sup> )   | 7.78                    | 15.53                     |
| Hardness CaCO <sub>3</sub> (mg L <sup>-1</sup> ) | 718.20                  | 1436.40                   |
| Chlorine (mg L <sup>-1</sup> )                   | 0.00                    | 0.00                      |
| Turbidity (NTU)                                  | 0.00                    | 0.01                      |

#### 3.2. Permeate Concentration and Salt Removal

Figure 4 shows the relationship between the behavior of permeate salinity and percent salt removal of the RO process with feed water at different temperatures (23, 25.5, 28, and 35 °C) and concentrations (5000 and 10,000 mg L<sup>-1</sup> TDS).



**Figure 4.** Permeate concentration (mg L<sup>-1</sup>) and percent salt removal at different temperatures and concentrations.

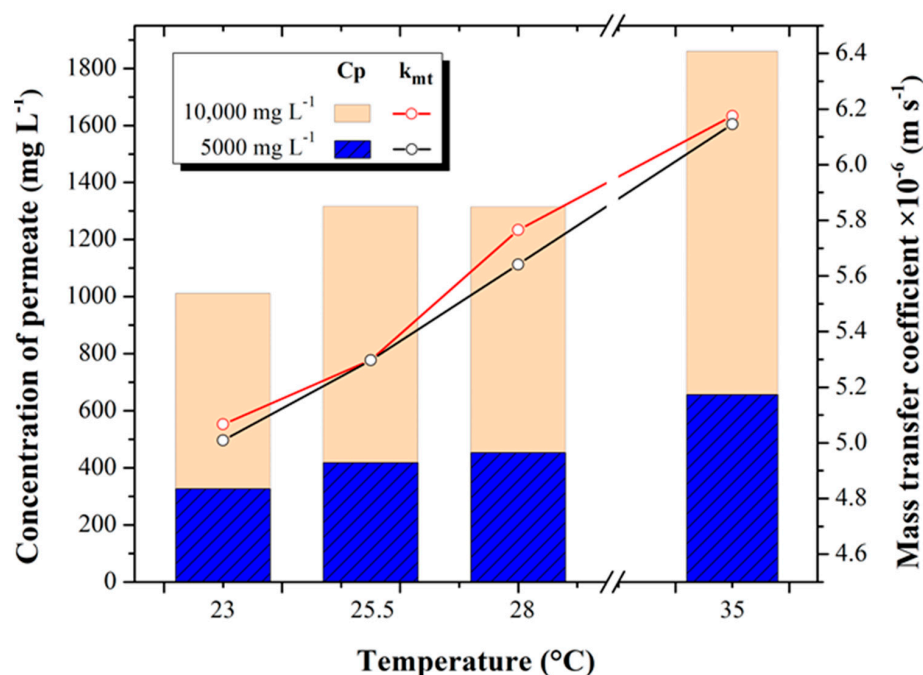
In the 5000 mg L<sup>-1</sup> test, an increase in the permeate salt concentration was obtained from  $326.15 \pm 36.21$  mg L<sup>-1</sup> to  $656.03 \pm 41.79$  mg L<sup>-1</sup>, which represents an increase of 50% for the temperature increase from 23 to 35 °C tests, respectively. On the other hand, in the 10,000 mg L<sup>-1</sup> test, the permeate concentration reached values of  $1011.08 \pm 294.43$  mg L<sup>-1</sup>,  $1316.33 \pm 295.43$  mg L<sup>-1</sup>,  $1314.18 \pm 250.03$  mg L<sup>-1</sup>, and  $1859.13 \pm 81.27$  mg L<sup>-1</sup> in the 23, 25.5, 28 and 35 °C tests, respectively, equivalent to an average increase of 44%. With respect to the percentage of salt removal for the 5000 mg L<sup>-1</sup> tests, it decreased from 93.68% to 86.88%, while in the 10,000 mg L<sup>-1</sup> tests, it decreased from 89.8 to 81.40% for the temperature increase from 23 to 35 °C tests, respectively.



Tests performed at different concentrations (5000 and 10,000 mg L<sup>-1</sup>) have similar trends. In both cases, the permeate concentration increases while the percentage of salt removal decreases as the temperature increases. This occurs because the components that make up the dense surface of the membrane *-polyamide-* expand when exposed to temperature increases so that the salts in the feed water dissolve in greater proportion and pass through the membrane more easily, causing a higher concentration of salts in the permeate stream *-salt passage-*. These behaviors coincide with those reported by Elsayed [30], which show that, at a higher temperature, there is higher salinity in the permeate and lower salt removal. That is why a temperature correction factor is applied. It is usually accepted that the pressure and permeate flow rate increase by about 0.5 bar and 3% for each 1 °C increase in temperature, respectively.

### 3.3. Permeate Concentration and Mass Transfer Coefficient

Figure 5 shows the behavior of the salinity obtained in the permeate with respect to the mass transfer coefficient calculated with the data measured in the experimental desalination tests for the 5000 and 10,000 mg L<sup>-1</sup> tests at different temperature increments 23, 25.5, 28, and 35 °C.



**Figure 5.** Permeate concentration (mg L<sup>-1</sup>) and mass transfer coefficient (m s<sup>-1</sup>) at different concentrations and temperatures.

Permeate salinity concentration increased from  $326.15 \pm 36.21$  to  $656.03 \pm 41.79$  mg L<sup>-1</sup> in the 5000 mg L<sup>-1</sup> tests, while minimum concentrations of  $1011.08 \pm 296.24$  and maximum concentrations of  $1859.33 \pm 52.30$  mg L<sup>-1</sup> were recorded in the 10,000 mg L<sup>-1</sup> tests, for the temperature increase from 23 to 35 °C tests, respectively for both concentrations.

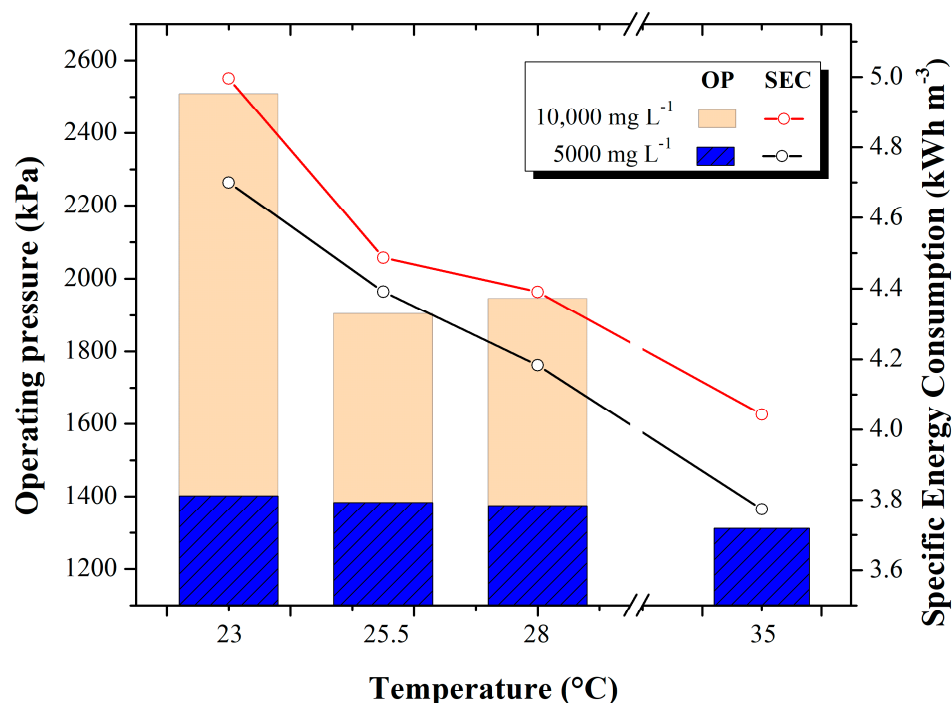
The minimum and maximum results obtained for the mass transfer coefficient are  $5.016 \times 10^{-6} \pm 4.07 \times 10^{-8}$  and  $6.14 \times 10^{-6} \pm 5.54 \times 10^{-8}$  m s<sup>-1</sup> for the 23 and 35 °C tests at 5000 mg L<sup>-1</sup>, respectively. For the 10,000 mg L<sup>-1</sup> tests, the mass transfer coefficient increased from  $5.056 \times 10^{-6} \pm 6.93 \times 10^{-8}$  m s<sup>-1</sup> (at 23 °C) to  $6.174 \times 10^{-6} \pm 1.16 \times 10^{-7}$  m s<sup>-1</sup> (at 35 °C), increasing by 11.87 and 13.8%, respectively.

The behaviors observed above are supported by the results of Joseph [31], who, with brackish and seawater, evaluated the behavior of different variables when working with different temperatures in a RO process; he reports that increasing the concentration and temperature of the feed water also increases both the concentration of the permeate and its diffusivity, the latter directly affecting the mass transfer coefficient.

The results show that for brackish water, it is possible to increase the feed water temperature and favor mass transfer for the three temperatures studied. However, in the ranges from 25.5 °C to 35 °C at a concentration of 10,000 mg L<sup>-1</sup>, the final concentration of the permeate is observed to be above 1000 mg L<sup>-1</sup>, which means that the permeate water is not suitable for consumption, according to Mexican regulation NOM-127-SSA1-2021 [32]. Therefore, a very important finding is that for concentrations higher than 10,000 mg L<sup>-1</sup>, it is not advisable to increase the temperature since the increase in the mass transfer coefficient will increase the passage of salts, causing the permeate to exceed the recommended limits. In the case of the experiment presented by Kim [33], when evaluating the behavior of seawater by increasing the temperature in an RO process, he observed that the permeate water quality decreased significantly.

### 3.4. Operating Pressure and Specific Energy Consumption

The relationship between the operating pressure used for the desalination tests and the specific energy consumption is shown in Figure 6.



**Figure 6.** Behavior of operating pressure (kPa) and specific energy consumption (kWh m<sup>-3</sup>) at different temperatures and concentrations.

In the 5000 mg L<sup>-1</sup> tests, an operating pressure of 1379 kPa was maintained. Energy consumption decreased as feedwater temperature increased. Values in specific energy consumption range from  $4.7 \pm 0.46$  kWh m<sup>-3</sup> (at 23 °C) to  $3.773 \pm 0.109$  kWh m<sup>-3</sup> (at 35 °C). This increase in temperature represents a decrease in specific energy consumption of 0.590 kWh m<sup>-3</sup>. On the other hand, at a concentration of 10,000 mg L<sup>-1</sup>, the changes in operating pressure are greater. That is, at 23 °C feedwater temperatures, the required pressure was 2503 kPa on average, while at 35 °C temperatures, lower pressure was required, i.e., 1944.3 kPa on average. Specific energy consumption values range from  $4.99 \pm 0.22$  kWh m<sup>-3</sup> (at 23 °C) to  $4.042 \pm 2.9 \times 10^{-15}$  kWh m<sup>-3</sup> (at 35 °C). This temperature increase represents a decrease in specific energy consumption of 0.72 kWh m<sup>-3</sup>.

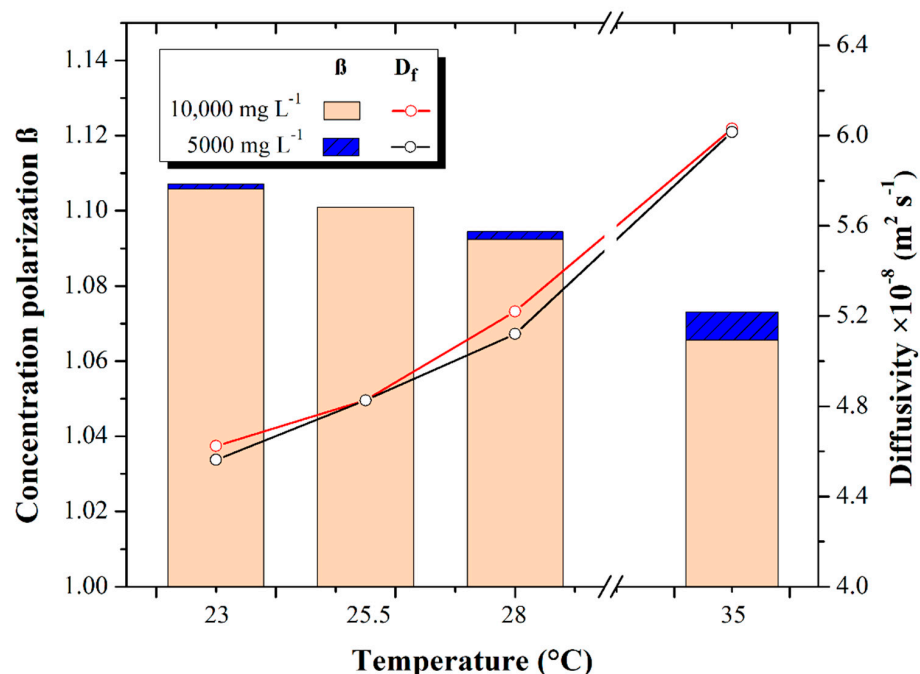
It is important to highlight that a large-scale RO plant at ambient temperature consumes on average from 3 kWh m<sup>-3</sup> to 3.8 kWh m<sup>-3</sup>; in this sense, for this study, a decrease of 0.590 kWh m<sup>-3</sup> and 0.72 kWh m<sup>-3</sup> was achieved at 5000 and 10,000 mg L<sup>-1</sup>, respectively, which enhances the effect of increasing temperature in decreasing specific energy consumption, certainly this finding can be applied on a large scale in the industrial sector, [34].

Although the specific power consumption and the operating pressure decrease with increasing temperature, this is something that the decision maker will have to face since, as discussed in Section 3.3, increasing temperature increases the concentration of water permeated, which is controversial since one of the purposes of the desalination process is to obtain quality water. In this way, a balance point must be found regarding what temperature should be fed to the process that reflects a decrease in energy consumption without jeopardizing the quality of the permeated water. In addition, it is necessary to know this relationship, as there will be an unnecessary expense of the time and energy necessary to heat the feed water.

As the temperature of the feed water increases, it is easier for the water to pass through the RO membrane because the salts are mostly dissolved, and those that are retained by the membrane are carried away with greater fluidity through the rejection flow, avoiding clogging of the membrane and a decrease in the pressure required by the pump to make reverse osmosis. Koutsou [35] conducted research evaluating the behavior of brackish and seawater at different temperatures in a RO process and reports that the physicochemical parameters of the water are directly influenced by the variation of temperature and salinity of the water; however, he did not study the relationship of pressure and energy consumption that is presented in this work. He mentions that when working with brackish and saline waters, optimal results are obtained in terms of the benefit of specific energy consumption. On the other hand, when working with higher concentrations, the same tendency as the 10,000 mg L<sup>-1</sup> tests is presented, where at the beginning, as the temperature increases, the specific energy consumption decreases.

### 3.5. Diffusivity and Concentration Polarization

The behaviors of the concentration polarization factor ( $\beta$ ) and feedwater diffusivity at different temperatures (23, 25.5, 28, and 35 °C) and concentrations (5000 and 10,000 mg L<sup>-1</sup> TDS) are shown in Figure 7.



**Figure 7.** Feedwater diffusivity (m<sup>2</sup> s<sup>-1</sup>) and concentration polarization  $\beta$  at different concentrations and temperatures.

The concentration polarization ( $\beta$ ) presented a value of  $1.106 \pm 0.00096$  as a maximum for the 23 °C test and  $1.094 \pm 0.00097$  as a minimum for the 28 °C test, both at 5000 mg L<sup>-1</sup>. For the 10,000 mg L<sup>-1</sup> tests, the concentration polarization  $\beta$  was found to have a maximum of  $1.106 \pm 0.0015$  when operating at 23 °C and  $1.092 \pm 0.0019$  working with 28 °C feed

water temperature. Such values are within the range established by Kucera [10] for the minimization of scaling formation in the membrane, ranging from 1 to 1.2 for the concentration polarization ( $\beta$ ). From the above, it is stated that there is a correlation between the increase in feedwater temperature and the decrease in the concentration polarization ( $\beta$ ).

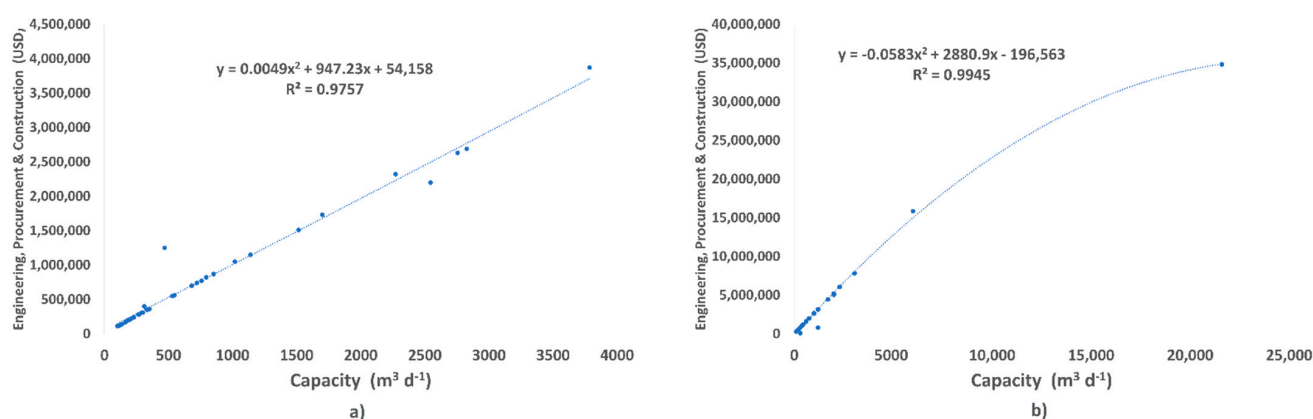
The concentration polarization  $\beta$  decreases because the physicochemical properties of water and salts are directly affected by the temperature increase. The viscosity and density of the water decrease while the salt dissolves better in the solution, so the dissolved salts pass through the membrane, and the retained salts flow more easily through the rejection flow. Rodriguez [24] investigated the behavior of the concentration polarization ( $\beta$ ) with 5000 mg L<sup>-1</sup> feed water at different temperature increments and reported a decrease in the concentration polarization ( $\beta$ ).

The feed water diffusivity obtained was  $4.57 \times 10^{-8} \pm 5.27 \times 10^{-10} \text{ m}^2 \text{ s}^{-1}$  as a minimum in the 23 °C test at 5000 mg L<sup>-1</sup> and  $6.015 \times 10^{-8} \pm 6.03 \times 10^{-10} \text{ m}^2 \text{ s}^{-1}$  as maximum for the 35 °C tests at 5000 mg L<sup>-1</sup>, increasing the feed water diffusivity by 24.18%. Similarly, the water diffusivity for the 10,000 mgL<sup>-1</sup> tests resulted in  $4.623 \times 10^{-8} \pm 5.607 \times 10^{-10} \text{ m}^2 \text{ s}^{-1}$  at its minimum at 23 °C and  $6.030 \times 10^{-8} \pm 2.923 \times 10^{-10} \text{ m}^2 \text{ s}^{-1}$  at its maximum at 28 °C, increasing the diffusivity 23.3%.

The above indicates that the increases in temperature and diffusivity of the feed water are proportional. On the other hand, the concentration polarization decreases. This coincides with the results of Mukesh-Sharma [36], who report that temperature has significant repercussions in the water desalination process and mentions that increasing the temperature results in a reduction in the concentration polarization and in the viscosity of the feed water, which favors better mass transfer and diffusivity through the membrane.

### 3.6. Investment and Desalinated Water Cost

Figure 8 shows the quadratic relationship of the cost of 37 reverse osmosis brackish water plants and 39 reverse osmosis seawater plants installed versus the installed capacity m<sup>3</sup> d<sup>-1</sup> of desalination plants installed in Mexico. The resulting equations describe an approximation to be had on the initial investment cost or the installed or to-be-bid-built plant capacity in a short time.



**Figure 8.** Correlation between initial investment and BWRO (a) SWRO (b) capacity. Desaldata [29].

Also, Table 2 shows that feedwater temperature has a direct impact on the costs of brackish water and seawater desalination plants. For salinity levels of 5000 and 10,000 mg L<sup>-1</sup>, the temperature is inversely proportional to the different desalination plant costs; therefore, as higher-temperature water is fed, the total plant cost decreases.

Table 2 shows the total cost (TC) in \$USD m<sup>-3</sup> of desalinated water for different temperatures in the brackish feedwater at 23, 25.5 and 35 °C at 5000 mg L<sup>-1</sup> and 5000 mg L<sup>-1</sup> with their different costs and percentages. Specific energy consumption (A), maintenance (B), labor (C), management (D), and chemicals (E).

Table 2. Permeate water cost.

| T<br>(°C)                               | A (62%)             |                          | B<br>(27%)          | C<br>(5.2%)         | D<br>(3.2%) | E<br>(2.6%) | TC    |
|-----------------------------------------|---------------------|--------------------------|---------------------|---------------------|-------------|-------------|-------|
|                                         | kWh m <sup>-3</sup> | USD<br>kWh <sup>-1</sup> | USD m <sup>-3</sup> | USD m <sup>-3</sup> |             |             |       |
| Feed Water (5000 mg L <sup>-1</sup> )   |                     |                          |                     |                     |             |             |       |
| 23                                      | 4.7                 | 1.04                     | 0.244               | 0.110               | 0.021       | 0.013       | 0.399 |
| 25.5                                    | 4.45                | 1.04                     | 0.231               | 0.104               | 0.020       | 0.012       | 0.378 |
| 28                                      | 4.18                | 1.04                     | 0.217               | 0.098               | 0.019       | 0.012       | 0.355 |
| 35                                      | 4.11                | 1.04                     | 0.214               | 0.096               | 0.019       | 0.011       | 0.349 |
| Feed Water (10,000 mg L <sup>-1</sup> ) |                     |                          |                     |                     |             |             |       |
| 23                                      | 4.95                | 1.04                     | 0.257               | 0.116               | 0.022       | 0.014       | 0.420 |
| 25.5                                    | 4.52                | 1.04                     | 0.235               | 0.106               | 0.020       | 0.013       | 0.384 |
| 28                                      | 4.38                | 1.04                     | 0.228               | 0.102               | 0.020       | 0.012       | 0.372 |
| 35                                      | 4.23                | 1.04                     | 0.220               | 0.099               | 0.019       | 0.012       | 0.359 |

#### 4. Conclusions

The present investigation evaluated the concentration polarization ( $\beta$ ) and inherent variables in the RO process with feed water at temperatures of 23, 25.5, 28, and 35 °C and at concentrations of 5000 and 10,000 mg L<sup>-1</sup>. Although the effect of temperature on concentration polarization has been studied, it is limited to theoretical studies that consider fixed concentrations for feed water. Therefore, the results of this study allow us to experimentally analyze the effect of feed water temperature for different concentrations (5000 and 10,000 ppm) and to determine whether the decrease in concentration polarization increases or decreases with increasing feed water concentration.

The increase in temperature from 23 °C to 35 °C in the feed water generates a positive impact on the concentration polarization in the RO membranes. For the 5000 mg L<sup>-1</sup> tests, a decrease of 3.10% was achieved, while for the 10,000 mg L<sup>-1</sup> tests, it decreased by 3.64%. All concentration polarization ( $\beta$ ) values obtained resulted below the established limit of 1.2 to prevent the formation of salt scaling in the membrane layer. Furthermore, the results obtained for both concentrations indicate that with increasing temperature from 23 to 35 °C in the feed water, both its diffusivity and mass transfer coefficient increase by 31.86 and 22.71%, respectively. These increase in diffusivity and mass transfer coefficient at concentrations of 5000 and 10,000 mg L<sup>-1</sup> by means of temperature increase (23 to 35 °C) produces an increase in permeate salinity concentration and a decrease in percent salt removal. The temperature of 35 °C causes a greater decrease in the SEC and concentration polarization due to the better diffusion of the solutes in the permeate water, decreasing the presence of fouling in the membrane film.

The specific energy consumption and the operating pressure in the two reverse osmosis desalination tests showed a decrease as the temperature of the feed water increased from 23 to 35 °C. In order not to affect the concentration and quality of the permeated water due to the effect of the increase in temperature in the feed water, it was found that the ideal temperature to work with water of 10,000 mg L<sup>-1</sup> -and probably higher- due to the trend shown is of 25.5 °C. This is due to the fact that the higher the temperature in the feed water, the diffusivity and mass transfer increase.

The research demonstrates the effect of the variation in feed water temperature in a RO system, benefiting the process by reducing energy consumption and the scaling of salts deposited on the membrane surface, causing a reduction of chemicals applied to the process and, therefore, a decrease in the overall cost of the process. Reducing the concentration polarization will allow advances in the scientific community by developing innovative methodologies to understand, study and perform laboratory tests to reduce them. On the other hand, the industrial sector will apply this scientific knowledge for the reduction of the concentration polarization  $\beta$ . All of the above potentiates the implementation of RO desalination technology as an attractive alternative to reduce current water shortage

problems, providing water in quantity and quality to populations located in urban centers and/or remote communities near coastlines.

**Author Contributions:** Conceptualization, G.E.D.-I. and D.P.M.-M.; methodology, G.E.D.-I., C.A.C.-D.I.T., J.R.-A. and D.P.M.-M.; software, C.A.C.-D.I.T. and J.R.-A.; validation, J.A.A.-J., N.V.-L. and R.M.-P.; formal analysis, G.E.D.-I., D.P.M.-M., J.R.-A., C.A.C.-D.I.T. and J.A.A.-J.; investigation, D.P.M.-M., R.M.-P. and G.E.D.-I.; resources, N.V.-L., J.A.A.-J. and R.M.-P.; data curation, D.P.M.-M. and R.M.-P.; writing—original draft preparation, G.E.D.-I. and D.P.M.-M.; writing—review and editing, G.E.D.-I., C.A.C.-D.I.T. and J.R.-A.; visualization, J.A.A.-J., N.V.-L., G.E.D.-I., J.R.-A. and C.A.C.-D.I.T.; supervision, G.E.D.-I., C.A.C.-D.I.T., J.R.-A., N.V.-L. and J.A.A.-J.; project administration, G.E.D.-I., N.V.-L. and J.A.A.-J.; funding acquisition, G.E.D.-I. and N.V.-L. All authors have read and agreed to the published version of the manuscript.

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## Nomenclature

### Symbols

|          |                                              |               |
|----------|----------------------------------------------|---------------|
| $A_m$    | Effective membrane area                      | $m^2$         |
| $C_1$    | Salt concentration of the city network water | $mg\ L^{-1}$  |
| $C_2$    | Desired salt concentration                   | $mg\ L^{-1}$  |
| $D$      | Solute diffusivity                           | $m^2\ s^{-1}$ |
| $d_h$    | Hydraulic diameter                           | $m$           |
| $EC$     | Electrical conductivity                      | $mS\ cm^{-1}$ |
| $J_p$    | Volumetric permeate flux                     | $m\ s^{-1}$   |
| $k_{mt}$ | Mass transfer coefficient                    | $m\ s^{-1}$   |
| $OP$     | Operating pressure                           | $kPa$         |
| $Q$      | Volumetric flow rate                         | $L\ min^{-1}$ |
| $Re$     | Reynolds number                              | -             |
| $Sh$     | Sherwood number                              | -             |
| $Sc$     | Schmidt number                               | -             |
| $SEC$    | Specific energy consumption                  | $kWh\ m^{-3}$ |
| $T$      | Temperature                                  | $^{\circ}C$   |
| $TDS$    | Total dissolved solids                       | $mg\ L^{-1}$  |
| $v$      | Water velocity in membrane module            | $m\ s^{-1}$   |
| $V_1$    | Volume of the feed tank                      | $L$           |
| $\beta$  | Polarization factor                          | -             |
| $\mu$    | Fluid viscosity                              | $Pa\ s$       |
| $\rho$   | Density of water                             | $kg\ m^{-3}$  |

### Subscripts and abbreviations

|      |                 |
|------|-----------------|
| $ED$ | Electrodialysis |
| $RO$ | Reverse osmosis |
| $p$  | permeate        |
| $f$  | feed            |
| $R$  | rejection       |
| $m$  | membrane        |



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### **3.4. Solicitud de patente A: Sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío.**

En la presente invención se refiere a un novedoso sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío, idealmente activado con energía solar térmica. La reivindicación central consiste en una configuración que establece una cascada de presión al alto vacío, la cual alcanza en el último efecto una temperatura de saturación (BBT) de 4 °C, rebasando el límite de la BBT de los procesos MED convencionales (generalmente 35-40 °C). Esta condición de operación permite utilizar el agua producto directamente como refrigerante.

La innovación se materializa mediante la implementación de un eyector-condensador barométrico que succiona, arrastra y comprime el vapor frío del último efecto de la MED para condensarlo. Esta configuración elimina la necesidad de una válvula de expansión y de precalentadores en los efectos finales, lo que se traduce en una reducción de las pérdidas exergéticas y un incremento en los indicadores de eficiencia global del sistema y proceso.

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**Solicitud de Patente A**

**SISTEMA Y PROCESO DE DESALINIZACIÓN  
POR EVAPORACIÓN DE MÚLTIPLE EFECTO AL  
ALTO VACÍO PARA LA PRODUCCIÓN  
SIMULTÁNEA DE AGUA Y FRÍO**

**SOLICITUD DE PATENTE**

MX/a/2025/001564



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**Certificado de acuse de  
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Expediente de Patente **MX/a/2025/001564**

**Asunto:** Se informa resultado del examen de forma.

Ciudad de México, a 13 de febrero de 2025.

No. Folio: **13854**

**Narciso FLORES ROMERO**  
**Apoderado de**  
**UNIVERSIDAD AUTÓNOMA DE BAJA CALIFORNIA**  
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MX/2025/13854

REF: Se da acuse de recibo a su Solicitud

En términos de lo dispuesto por el artículo 106 de la Ley Federal de Protección a la Propiedad Industrial y artículos 18, 19 y 26 del Acuerdo por el que se establecen lineamientos en materia de servicios electrónicos del Instituto Mexicano de la Propiedad Industrial; **se tiene por satisfecho el examen de forma** al haber cumplido con lo dispuesto por la Ley Federal de Protección a la Propiedad Industrial y el Reglamento de la Ley de la Propiedad Industrial, aplicable a la Ley Federal de Protección a la Propiedad Industrial en términos de lo dispuesto por el artículo Cuarto Transitorio del DECRETO por el que se expide la Ley Federal de Protección a la Propiedad Industrial y se abroga la Ley de la Propiedad Industrial publicado en el Diario Oficial de la Federación del día 01 de julio de 2020.

Por lo anterior, de conformidad con los artículos 107 y 108 de la Ley Federal de Protección a la Propiedad Industrial; artículos 39 y 40 del Reglamento de la Ley de la Propiedad Industrial y artículo 9 del Acuerdo por el que se establecen los plazos de respuesta a diversos trámites ante el Instituto Mexicano de la Propiedad Industrial, una vez **aprobado** el examen de forma, **la publicación en Gaceta de la presente solicitud de patente en trámite**, tendrá lugar después del vencimiento del plazo de 18 meses, contado a



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## **SISTEMA Y PROCESO DE DESALINIZACIÓN POR EVAPORACIÓN DE MÚLTIPLE EFECTO AL ALTO VACÍO PARA LA PRODUCCIÓN SIMULTÁNEA DE AGUA Y FRÍO**

5

### **CAMPO TÉCNICO**

La presente invención se relaciona con los procesos de desalinización y climatización de espacios y se refiere a un proceso de alta eficiencia y capacidad para la producción de agua desalinizada y enfriamiento/calefacción de espacios.

### **ANTECEDENTES**

- 10 La tecnología de desalinización por evaporación/destilación de múltiple efecto (MED), ha sido una de las principales tecnologías empleadas en el mundo para realizar el proceso de desalinización del agua de mar y/o agua salobre. Durante las últimas décadas, los esfuerzos de la comunidad científica internacional se han enfocado en la realización de trabajos teóricos y experimentales para la optimización de la tecnología MED. En cierta medida, se ha logrado
- 15 mejorar las condiciones de operación incrementando la eficiencia y capacidad del sistema, así como la reducción de los altos costos de inversión inicial, costos de operación y costos de mantenimiento. (Morin, 1993; El-Dessouky et al., 1998; Al-Shammiri and Safar, 1999; Darwish et al., 2006; Sharaf et al., 2011; Druetta et al., 2013; Palenzuela et al., 2013; Palenzuela et al., 2014; Al-Mutaz & Wazeer, 2014). En busca de seguir mejorando la
- 20 tecnología MED, se propusieron hibridaciones o acoplamientos con otras tecnologías de generación eléctrica y enfriamiento. Lo anterior, reduce de una forma muy importante el costo de operación y permite la producción de múltiples servicios o poligeneración (Nassrullah, Anis, Hashaikeh, & Hilal, 2020).

- Tratando de lograr una mejora en la eficiencia energética del proceso MED, (Alarcón-
- 25 **Padilla and García-Rodríguez, 2007; Chorak et al., 2018)** realizaron trabajos teóricos, donde se presenta el acoplamiento de un sistema de desalinización MED con una unidad de enfriamiento por absorción. En la hibridación, se utiliza el efecto frigorífico del evaporador de la máquina de absorción, para condensar el vapor saturado producido en el último efecto

del sistema MED. Con lo anterior, **(Chorak et al., 2018)**, lograron reducir las pérdidas de energía por efluente en el condensador, y obtener una mejora en la Relación de Rendimiento (PR, por sus siglas en inglés) y capacidad de desalinización del sistema. Sin embargo, aunque el acoplamiento entre tecnologías presenta una mejora en la eficiencia del sistema de desalinización, se requiere de un alto número de componentes, lo cual incrementa la inversión inicial y el costo de mantenimiento, reduciendo la viabilidad tecno económica de este tipo de sistemas.

El mejoramiento de la tecnología MED se ha visto limitado por los retos que representan las pérdidas de energía por efluente en el condensador, aunado a los límites operativos que tiene la Temperatura máxima de salmuera (TBT, por sus siglas en inglés) y la Temperatura mínima de salmuera (BBT, por sus siglas en inglés) del sistema **(Ahmed et al., 2020)**. La limitante en la TBT, depende directamente de los problemas de corrosión que se presentan al operar con agua de mar (35,000 a 50,000 ppm) a temperaturas mayores a 70 °C **(Goh et al., 2021)**; mientras que la BBT, está restringida por la temperatura del agua de mar (25-30 °C), la cual se utiliza para condensar el vapor del último efecto de la MED, requiriéndose un gradiente de temperatura mínimo ( $\approx 5$  °C) entre el vapor y el agua de mar **(Aly, 1995; Shahzad, et al., 2015)**. Asimismo, diferentes estudios exergéticos encontrados en la literatura **(García-Rodríguez and Gómez-Camacho, 2001; Nafey et al., 2006; Mistry and Lienhard, 2013; Warsinger et al., 2015; Lienhard et al., 2017; Ng et al., 2017)**, han mostrado que, en los sistemas MED convencionales, la mayor pérdida de exergía tiene lugar en el condensador. Esto es debido a que en el proceso de condensación del vapor del último efecto de la MED, se transfiere mucho calor latente al agua de mar y una gran cantidad de dicha agua es regresada al mar **(Gude, 2018)**.

Buscando incrementar la viabilidad técnica y económica de este tipo de tecnologías MED, **Aly, (1995)**, fue uno de los pioneros en el estudio de nuevas configuraciones y la exploración operativa de nuevos rangos de presión y temperatura; utilizando una TBT de 63 °C y una BBT de 6 °C, lo cual es una BBT considerablemente menor de lo utilizado en los sistemas convencionales. Lo anterior, se logró gracias a la hibridación con un sistema de enfriamiento por absorción LiBr-H<sub>2</sub>O de 220 kW. La hibridación, permitió establecer una cascada de 20 efectos y obtener un incremento en los indicadores de desalinización (GOR de 14.8 y un PR de 14.2). El acoplamiento de tecnologías consistió en utilizar para la activación de la MED,

- el calor retirado del absorbedor y condensador del sistema de absorción. También se utiliza el enfriamiento del evaporador, para condensar el vapor de baja temperatura (6 °C) del último efecto de la MED. Sin embargo, este acoplamiento de tecnologías no considera la producción de frío. Además, presenta una mayor complejidad del sistema y un alto número de componentes; lo que implica una alta inversión inicial y un alto costo de operación y mantenimiento. Recientemente se han realizado estudios relacionados con la exploración de nuevos rangos de presión/temperatura de operación de la MED (**Farsi and Dincer, 2019; Ali et al., 2020; Ghenai et al., 2021**). Destacando los trabajos de (**Shahzad et al., 2014; Shahzad et al., 2015**), quienes estudiaron los rangos de temperatura de 50 a 3 °C y de 70 a 5 °C, reportando una producción de agua de 2 a 3 veces más que un sistema MED convencional. Sin embargo, para lograr esto, fue necesario el acoplamiento de la maquina MED con un sistema de adsorción (AD) que utiliza gel de sílice, adsorbiendo con este último el vapor a baja temperatura proveniente del último efecto de la MED. La principal desventaja de esta propuesta es su alta inversión inicial, debido al gran número de componentes del proceso.
- Recientemente **López-Zavala et al., (2019), (MX/a/2017/016797)** propusieron un innovador sistema de enfriamiento y desalinización, basado en el principio de absorción de simple efecto LiBr/H<sub>2</sub>O, con tres niveles de presión y activado por energía solar térmica. Este sistema presentó una capacidad de enfriamiento de 23 kW, junto con una notable producción de agua de 838 L/día. La propuesta incluyó una configuración que logró mantener una presión en el absorbedor superior a la del evaporador, lo que resultó en una mejora significativa del 22% en la eficiencia del sistema en comparación con los ciclos convencionales de enfriamiento por absorción de simple efecto. A pesar de este avance, el incremento en la eficiencia aún no es suficiente para posicionar a estas tecnologías en un nivel competitivo en el mercado actual.
- En un estudio adicional, **López-Zavala et al., (2019b) (MX/a/2018/006950) (MX/a/2018/015007)** llevaron a cabo la integración de un sistema de Destilación de Múltiple Efecto (MED) de 14 efectos junto con un sistema abierto de enfriamiento por absorción de simple efecto de 23 kW. Se registró una notable capacidad de enfriamiento de 2,009 kW con un Coeficiente de Rendimiento (COP) de 5.87, y una producción de agua de 73,569 L/día. Respecto a la eficiencia de desalación, se informó un Coeficiente de Recuperación (RR) de 0.52 y una Relación de Rendimiento (PR) de 6.72. No obstante, a pesar de las mejoras

significativas en la eficiencia del proceso de enfriamiento, esta tecnología aún requiere múltiples componentes, lo que resulta en una inversión inicial elevada.

De igual manera, **López-Zavala et al., (2023)**, presentaron el estudio de un novedoso sistema de desalinización MED, que utiliza el agua producto para generar enfriamiento mediante la expansión súbita y condensación del vapor frío, debido al aumento de presión provocado por un eyector-condensador barométrico. Con lo anterior, los autores reportan una capacidad de desalinización de 74,736 L/d, una producción de 2,043 kW (581 toneladas) de enfriamiento y unos indicadores de COP, PR y RR de 4.7, 11.96 y 0.53, respectivamente, siendo el valor del COP 6.53 veces mayor que el de un sistema de refrigeración por absorción de simple efecto. Sin embargo, a pesar de lograr un aumento significativo en la capacidad de enfriamiento y eficiencia, el sistema reporta una capacidad de desalinización similar a lo antes reportado en la literatura (PR de 11.96). Lo anterior, es debido a que utiliza el número de efectos y los límites de operación de los sistemas convencionales de desalinización MED (70-35 °C). Cabe mencionar que el estudio reportado anterior se encuentra en estado de solicitud de patente con número de expediente **(MX/a/2020/008009)**.

Con lo anterior, se muestra que se ha intentado desarrollar tecnologías que satisfagan de manera conjunta y simultánea las necesidades de agua y frío, principalmente mediante la combinación de tecnologías de enfriamiento y desalinización. Sin embargo, estos esfuerzos han resultado en procesos y equipos con una alta inversión inicial y costos operativos elevados, sin lograr un incremento significativo en su capacidad y eficiencia. Para superar estas limitaciones, se propone un **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío**, el cual está constituido por un **subproceso A (desalinización por evaporación de múltiple efecto al alto vacío)** y un **subproceso B (producción de frío y condensación del agua producto)**.

Este **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío**, a diferencia de la patente MX/a/2020/008009 (que utiliza una MED que opera en condiciones convencionales), no realiza la expansión del agua producto, en su lugar se modifican las condiciones de operación de la MED, trabajando al alto vacío. Este cambio permite una mejora en la eficiencia del proceso: el valor de la Relación de Rendimiento (PR) de la invención propuesta es de 14.22,



en comparación con el valor de 11.60 reportado en la patente mencionada, lo que representa una mejora significativa en la eficiencia de la desalinización. Asimismo, la eficiencia en la producción de enfriamiento también se ve incrementada, observándose un Coeficiente de Rendimiento Global ( $COP_G$ , por sus siglas en inglés) de 9.75 para la invención, frente a 9.32 en la patente MX/a/2020/008009. Como resultado, el agua producto sale en condiciones óptimas para ser utilizada directamente como fluido refrigerante en el evaporador y obtener la producción de frío. La invención propuesta tiene como características principales, la utilización de un eyector-condensador barométrico, que permite operar el **subproceso de desalinización por evaporación de múltiple efecto al alto vacío** con una BBT inferior a la convencional (35-40 °C), sin requerir el uso de una válvula de expansión, lo que disminuye las pérdidas exérgicas del sistema.

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### BREVE DESCRIPCIÓN DE FIGURAS

La figura 1, es un diagrama esquemático del **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío**.

20 La figura 2, es un diagrama esquemático del **subproceso A (desalinización por evaporación de múltiple efecto al alto vacío)**,

La figura 3, es un diagrama esquemático del **subproceso B (producción de frío y condensación del agua producto)**.

### DESCRIPCIÓN

A continuación, se presenta la descripción física y operativa del presente **sistema y proceso de desalinización por evaporación de múltiple efecto alto vacío para la producción simultánea de agua y frío** el cual está constituido por dos subsistemas y subprocesos: a) el

5 **subsistema A y subproceso A (desalinización por evaporación de múltiple efecto al alto vacío)**, y b) el **subsistema B y subproceso B (producción de frío y condensación del agua producto)**.

El **subsistema A**, se compone de cuatro etapas: 1) **Etapla del Efecto 0**, 2) **Etapla del Efecto 1 hasta el Efecto n**, 3) **Etapla del Efecto 1AV hasta el Efecto nAV**, y 4) **Etapla del Efecto fAV**. La primera etapa (**Etapla del Efecto 0**), está compuesto por un intercambiador de calor que actúa como generador (**GEN**) para la activación del sistema MED, un separador líquido-vapor (**SEP0**) para salmuera y/o agua salobre, y una válvula de expansión (**VE1**) para agua

10 salada. La segunda etapa (**Etapla del Efecto 1 hasta el Efecto n**), cuenta con n evaporadores (**HX1, HX2, ... HXn**), en donde n puede ir de 1 a cualquier número, n separadores líquido-vapor para salmuera y/o agua salobre (**SEP1, SEP2, ... SEPn**) en donde, n es igual al número

15 de intercambiadores de calor (**HX**) que actúan como evaporadores, dado que, a cada evaporador (**HX**) le corresponde un separador líquido-vapor para salmuera (**SEP**), n separadores líquido-vapor para agua producto (**SEP1AP, SEP2AP, ... SEPnAP**), en donde n es igual al número de evaporadores (**HX**) menos 1, n precalentadores de agua de mar o agua

20 salobre (**PC0, PC1, ... PCn**), n tuberías tipo “U” para mantener un sello hidráulico y mantener la diferencia de presión entre los efectos (**TU0, TU1, ... TUn**), en donde n es igual al número de evaporadores (**HX**) menos 1, n válvulas de expansión para el agua producto (**VE1AP, VE2AP, ... Ven-1AP**), en donde n es igual al número de evaporadores (**HX**) menos

25 1, n mezcladores para mezclar los vapores producidos por el flasheo de la salmuera y el agua producto (**MIX1, MIX2, ... MIXn**), en donde n es igual al número de evaporadores (**HX**) menos 1, n mezcladores para mezclar las corrientes de agua producto (**MIX2AP, MIX3AP, ... MIXnAP**), en donde n es igual al número de evaporadores (**HX**) menos 2. Por otra parte, la **Etapla del Efecto 1AV hasta el Efecto nAV**, cuenta con n evaporadores (**HX1AV, HX2AV, ... HXnAV**), en donde n puede ir de 1 a cualquier número, n separadores líquido-

30 vapor para salmuera y/o agua salobre (**SEP1AV, SEP2AV, ... SEPnAV**) en donde, n es

igual al número de intercambiadores de calor (**HX**) que actúan como evaporadores, dado que, a cada evaporador (**HX**) le corresponde un separador líquido-vapor para salmuera (**SEP**), n separadores líquido-vapor para agua producto (**SEP1AV<sub>AP</sub>**, **SEP2AV<sub>AP</sub>**, ... **SEPnAV<sub>AP</sub>**), en donde n es igual al número de evaporadores (**HX**) menos 1, n tuberías tipo “U” para mantener un sello hidráulico y mantener la diferencia de presión entre los efectos (**TU1AV**, **TU2AV** ... **TUnAV**), en donde n es igual al número de evaporadores (**HX**) menos 1, n válvulas de expansión para el agua producto, en donde n es igual al número de evaporadores menos 1, n mezcladores para mezclar los vapores producidos por el flasheo de la salmuera y el agua producto (**MIX1AV**, **MIX2AV**, ... **MIXnAV**), en donde n es igual al número de evaporadores (**HX**) menos 1, n mezcladores para mezclar las corrientes de agua producto (**MIX1AV<sub>AP</sub>**, **MIX2AV<sub>AP</sub>**, ... **MIXnAV<sub>AP</sub>**). Finalmente, la **Etapas del Efecto fAV**, se compone de un intercambiador de calor que actúa como evaporador (**HXfAV**), un separador líquido-vapor (**SEPfAV**), un mezclador para agua producto (**MIXnAV<sub>AP</sub>**), una válvula de expansión (**VENAV<sub>AP</sub>**), un separador líquido-vapor para agua producto (**SEPf<sub>AP</sub>**), y un mezclador (**MIXfAV**). Además, el subsistema A, se cuenta con un tanque para agua de mar (**TAM**) y una bomba para agua de mar o agua salobre (**BAM1**). A su vez, **subsistema B (producción de frío y condensación del agua producto)**, está compuesto por un eyector-condensador (**EYC**), un evaporador para la producción de frío (**EVA**), una válvula de tres vías para agua producto y/o refrigerante (**V3V1**), una válvula de tres vías para salmuera (**V3V2**), un mezclador para corrientes de vapor (**MIXVE**), una bomba para agua de mar o agua salobre (**BAM2**), una bomba para agua producto (**BAP**), un intercambiador de calor para agua de mar o agua salobre (**HXsw**), un intercambiador de calor para salmuera (**HXBR**), un separador líquido-vapor para salmuera (**SEP15**), ocho válvulas de paso (**VP1**, **VP2** ... **VP8**), un tanque para agua producto (**TAP**), un tanque para salmuera (**TS**) y tres columnas barométricas, una para extracción de salmuera (**CB2**), y dos para la extracción de agua producto (**CB1** y **CB3**).

En el **subproceso A**, se emplea agua de mar o agua salobre, preferiblemente agua de mar con una salinidad típica de 35,000 ppm. En el marco de esta invención, el término de “agua salada” abarca tanto el agua de mar o agua salobre, según corresponda. El **subproceso A**, también se compone de cuatro etapas: 1) **Etapas del Efecto 0**, 2) **Etapas del Efecto 1 hasta el Efecto n**, 3) **Etapas del Efecto 1AV hasta el Efecto nAV**, y 4) **Etapas del Efecto fAV**. En

el **subproceso A**, de acuerdo con las figuras 1 y 2, la bomba para agua salada (**BAM1**) succiona a la corriente de agua salada (**AM1**) y la dirige como la corriente (**AM2**) hacia la **Etapas del Efecto 0**, pasando previamente por los precalentadores de la MED. La corriente de agua salada (**AM2**), es llevada hacia los precalentadores (**PC0, PC1, ... PCn**), desde el precalentador (**PCn**) hasta el precalentador para agua salada (**PC0**), para recibir todo el calor latente del vapor producido por flasheo de las corrientes de salmuera y de agua producto en los efectos. Dichas corrientes de vapor provienen de los separadores de salmuera (**SEP0, SEP1, ... SEPn**) y de los separadores de agua producto (**SEP1<sub>AP</sub>, SEP2<sub>AP</sub>, ... SEPn<sub>AP</sub>**). El vapor de los separadores (**SEP1, SEP2, ... SEPn**), es un compuesto del vapor flasheado y el vapor generado por calentamiento en el efecto. Sin embargo, es preferible utilizar únicamente el vapor producido por flasheo para precalentamiento y utilizar el vapor producido por ebullición para el calentamiento de la corriente de salmuera del efecto posterior. Lo anterior sucede, desde el efecto con el último precalentamiento (**Efecto n**) hasta el **Efecto 1** de la MED. Mientras que, en el precalentador (**PC0**) del **Efecto 0**, la corriente de agua de mar (**AM5**), (que previamente a la salida del **PC1** fue la corriente **AM4** y posterior a la salida del **PCn** fue la corriente **AM3**), se precalienta con el porcentaje necesario del vapor producido por calentamiento (ebullición) en el generador (**GEN**), para alcanzar la máxima temperatura de precalentamiento posible. Con lo anterior, se consume el mínimo de servicios de calentamiento externo en el generador. Es importante resaltar que, el número de precalentadores (**PC**) estará dado por el perfil de temperaturas en los Efectos del sistema MED. La corriente de agua salada (**AM6**) que sale del precalentador (**PC0**), pasa por una válvula de expansión (**VE1**) con la finalidad de bajar súbitamente su presión hasta la presión de saturación correspondiente al **Efecto 0 (TBT)**. En estas condiciones de presión y temperatura, la corriente de agua salada (**AM7**), ingresa al generador (**GEN**) y es calentada por un fluido caloportador (**FC1**) proveniente de un sistema de captación y almacenamiento de energía solar que puede estar conformado por colectores solar tipo Placa Plana, Cilindro Parabólico, Concentrador Parabólico Compuesto, Fresnel, entre otros o cualquier otra fuente de calor. La corriente de fluido caloportador (**FC2**) sale del generador (**GEN**) después de haber calentado a la corriente de agua salada (**AM7**). La corriente de agua salada (**AM8**) sale en dos fases del generador (**GEN**) e ingresa al separador líquido-vapor (**SEP0**), dividiéndose en las corrientes de salmuera (**S0**) y de vapor de agua (**V0**). La corriente de salmuera (**S0**) es

dirigida al evaporador del segundo efecto de la MED (**HX1**), ingresando a través de una tubería en forma de U (**TU0**). Lo anterior permite tener un sello hidráulico para mantener la diferencia de presión entre el **Efecto 0** y **Efecto 1**. Es importante mencionar, que la corriente de salmuera (**S0**) al pasar por la tubería (**TU0**), pasa de tener la presión del **Efecto 0** (**P0**) a la presión del **Efecto 1** (**P1**), con lo que la corriente de salmuera (**S0**) se flashea y a la salida queda como la corriente en dos fases (**S0a**) y con una menor presión y temperatura de saturación. En la **Etapas del Efecto 1 hasta el Efecto n**, la corriente (**S0a**) ingresa al intercambiador del **Efecto 1** (**HX1**), donde recibe todo el calor latente de la corriente líquido-vapor (**V0a**). Por otro lado, la corriente de vapor de agua (**V0**) se dirige al precalentador (**PC0**) y cede un porcentaje de su calor latente para precalentar a la corriente de agua salada (**AM5**), quedando en dos fases (**V0a**). La corriente (**V0a**) ingresa al intercambiador del **Efecto 1** (**HX1**) y calienta a la corriente de salmuera en dos fases (**S0a**) que sale como la corriente de salmuera (**S0b**). Al ceder toda su energía latente, la corriente (**V0a**) se condensa totalmente y sale como agua producto en la corriente (**AP1**). La corriente de agua producto (**AP1**) pasa a través de la válvula de expansión (**VE1AP**) y sale como la corriente (**AP1a**) quedando en dos fases con la misma presión que el separador líquido-vapor (**SEP1AP**); en el separador (**SEP1AP**) se separa la fase de vapor (**Vp1**) y la fase líquida (**Ap1b**), el vapor (**Vp1**) es dirigido al mezclador (**MIX1**) para mezclarse con la corriente de vapor (**V1**), mientras que la corriente (**Ap1b**) se dirige al mezclador (**MIX2AP**) para mezclarse con la corriente (**AP2**) y dirigirse como la corriente (**AP2a**) hacia la válvula de expansión (**VE2AP**), donde cae su presión hasta la presión del efecto posterior, saliendo como la corriente (**AP2b**). Por otra parte, la corriente de salmuera (**S0b**), ingresa al separador líquido-vapor (**SEP1**), en donde se divide en las corrientes de vapor (**V1**) y de salmuera (**S1**). La corriente de vapor (**V1**), se une y se mezcla con la corriente de vapor generado por flasheo del agua producto (**Vp1**), proveniente del separador líquido-vapor (**SEP1AP**), convirtiéndose en la corriente de vapor (**V1a**). La corriente (**V1a**), ingresa al precalentador del **Efecto 1** (**PC1**) y cede un porcentaje de su calor latente (preferentemente el porcentaje correspondiente a la condensación del vapor por flasheo), quedando como la corriente en dos fases (**V1b**); posteriormente, la corriente (**V1b**) ingresa al intercambiador de calor del **Efecto 2** (**HX2**). Por otra parte, la corriente de salmuera (**S1**), hace lo propio, al dirigirse a través del sello hidráulico en forma de tubería U (**TU1**) saliendo como la corriente (**S1a**), y luego pasar hacia el intercambiador

de calor (**HX2**), donde termina con la presión del **Efecto 2**, y posteriormente salir como la corriente de salmuera en dos fases (**S1b**). Lo ocurrido en el **Efecto 1**, sucede en los posteriores efectos, teniéndose una caída de presión y temperatura en efecto cascada, hasta la temperatura del **Efecto n**. Cabe mencionar que en el **Efecto n**, el separador líquido-vapor (**SEPn**), recibe la corriente (**Sn-1b**) proveniente del intercambiador de calor del efecto previo, y divide dicha corriente en las corrientes de salmuera (**Sn**) y de vapor (**Vn**). La corriente de vapor (**Vn**) se dirige al mezclador de vapor (**MIXn**) donde se mezcla con la corriente de vapor (**Vpn**), proveniente del separador líquido-vapor (**SEPnAP**), para salir como la corriente mezclada de vapor (**Vna**) y así dirigirse al precalentador (**PCn**). Además, el separador (**SEPnAP**), recibe la corriente de agua producto (**APnb**) y la divide en dos corrientes: la corriente de vapor (**Vpn**) previamente mencionada, y la corriente de agua (**APnc**), la cual es dirigida al mezclador (**MIXnAP**) para mezclarse con la corriente de agua producto (**AP1AV**) que sale del **Efecto 1AV**. En la **Etapas del Efecto 1AV hasta el Efecto nAV**, sucede algo muy similar, solo que en esa sección no se tiene precalentadores, por lo que, todo el vapor generado por calentamiento (ebullición) y flasheo, se utiliza en el efecto posterior. En el **Efecto 1AV**, el intercambiador de calor (**HX1AV**) recibe la corriente de vapor (**Vnb**) proveniente del precalentador (**PCn**), la cual cede su energía de condensación a la corriente (**Sna**) que proviene de la tubería en forma de “U” (**TUn**). La corriente de vapor (**Vnb**) se condensa y sale del intercambiador (**HX1AV**) como la corriente de agua (**AP1AV**), la cual después de mezclarse con la corriente (**APnc**) en el mezclador (**MIXnAP**), sale como la corriente (**AP1AVa**) que pasa a través de la válvula de expansión (**VENAP**) para disminuir su presión hasta la presión del siguiente efecto (**Efecto 2AV**) y finalmente ingresar al separador líquido-vapor (**SEP1AVAP**) como la corriente (**AP1AVb**). Por otra parte, la corriente de salmuera bifásica (**Sna**) después de ser calentada, sale como la corriente (**Snb**) y se dirige al separador líquido-vapor (**SEP1AV**), del cual salen las corrientes de vapor (**V1AV**) y de salmuera (**S1AV**). La corriente de salmuera (**S1AV**) pasa a través de la tubería en forma de “U” (**TU1AV**), donde disminuye su presión hasta la presión del siguiente efecto (**Efecto 2AV**), y sale como la corriente (**S1AVa**) que se dirige al intercambiador de calor (**HX2AV**). Mientras que la corriente de vapor (**V1AV**) ingresa al mezclador (**MIX1AV**) para mezclarse con la corriente de vapor (**Vp1AV**) proveniente del separador (**SEP1AVAP**), de donde sale como la corriente de vapor (**V1AVa**) que se dirige al intercambiador de calor (**HX2AV**). El

intercambiador de calor (**HX2AV**) del **Efecto 2AV**, cuenta con dos corrientes de salida: la corriente de salmuera (**S1AVb**), que se dirige hacia el separador líquido-vapor del siguiente efecto, y la corriente de agua (**AP2AV**). La corriente de agua (**AP2AV**) se dirige al mezclador (**MIX1AV<sub>AP</sub>**) para mezclarse con la corriente de agua (**AP1AVc**) que proviene del separador líquido-vapor (**SEP1AV<sub>AP</sub>**), para finalmente salir como la corriente (**AP2AVa**). El proceso e interconexiones ocurridas en el **Efecto 2AV**, suceden en los posteriores efectos, hasta el **Efecto nAV**, teniéndose una caída de presión y temperatura en efecto cascada, hasta la temperatura del **Efecto nAV**. Se destaca que en el **Efecto nAV**, el separador líquido-vapor para salmuera (**SEPnAV**) recibe la corriente (**Sn-1AVb**) y la divide en dos corrientes: la corriente de salmuera (**SnAV**) y la corriente de vapor (**VnAV**). La corriente de salmuera (**SnAV**) pasa a través de la tubería en forma de “U” (**TUnAV**), donde disminuye su presión hasta la presión del **Efecto fAV**, y posteriormente sale como la corriente (**SnAVa**), que ingresa al intercambiador de calor del último efecto (**HXfAV**) para ser calentada y salir como la corriente de salmuera (**SnAVb**). A su vez, la corriente de vapor (**VnAV**), se dirige al mezclador (**MIXnAV**), donde se mezcla con la corriente de vapor (**VpnAV**) proveniente del separador líquido-vapor (**SEPnAV<sub>AP</sub>**), para salir como la corriente de vapor (**VnAVa**) e ingresar al intercambiador de calor (**HXfAV**) del **Efecto fAV**. Del separador líquido-vapor (**SEPnAV<sub>AP</sub>**), que previamente recibió la corriente (**APnAVb**), salen las corrientes de vapor (**VpnAV**) y la corriente de agua (**APnAVc**), la cual se dirige al mezclador (**MIXnAV<sub>AP</sub>**) donde se mezcla con la corriente condensada (**APf**) que sale del intercambiador de calor (**HXfAV**), para formar la corriente (**APfa**). La corriente de agua (**APfa**) pasa a través de la válvula de expansión (**VEnAV<sub>AP</sub>**) para disminuir su presión hasta la presión del **Efecto fAV**, y salir como la corriente (**APfb**), que posteriormente ingresa al separador líquido-vapor (**SEPf<sub>AP</sub>**) del cual salen las corrientes de vapor (**VpfAV**) y de agua líquida (**ApT**). Finalmente, en la etapa que comprende al **Efecto fAV**, se lleva a cabo el mismo proceso que en los efectos anteriores, con la diferencia de que sus corrientes de vapor (**VfAVa**), de salmuera (**SfAV**) y de agua producto (**ApT**) se dirigen al **subproceso B**. En el **Efecto fAV**, el separador líquido-vapor (**SEPfAV**) recibe la corriente (**SnAVb**) proveniente del intercambiador de calor (**HXfAV**), y la divide en las corrientes de salmuera (**SfAV**) y de vapor (**VfAV**). La corriente de salmuera (**SfAV**) se dirige a la válvula de tres vías (**V3V2**) del **subproceso B**, mientras que la corriente de vapor (**VfAV**) ingresa al mezclador

(**MIXfAV**) para mezclarse con la corriente de vapor (**VpfAV**), y salir como la corriente de vapor (**VfAVa**), que también se dirige al **subproceso B**. Por otra parte, el agua producto de cada efecto, se une con la del efecto posterior a través de válvulas de expansión para agua producto (**VE<sub>AP</sub>**) y separadores para agua producto (**SEP<sub>AP</sub>**). Este proceso se repite hasta el

5 último efecto (**Efecto fAV**), donde se obtiene el total del agua producto de los efectos del sistema en la corriente (**ApT**) y se dirige al **subproceso B**, para su posterior extracción y/o utilización como refrigerante en el evaporador (**EVA**). Asimismo, las corrientes de vapor (**VfAVa**) y de salmuera (**SfAV**) del último efecto, se dirigen al **subproceso B**, para su correspondiente utilización.

- 10 **En el subproceso B**, de acuerdo con las figuras 1 y 3, la válvula para el control de capacidad de enfriamiento de tres vías (**V3V1**), recibe la corriente del agua producto total de los efectos (**ApT**) y la divide en las corrientes (**ApTa**) y (**R1**) para regular la capacidad de enfriamiento. La corriente (**ApTa**) es llevada por medio de una columna barométrica (**CB3**) al tanque para agua producto (**TAP**), eliminando las condiciones de presión de vacío y quedando a
- 15 condiciones atmosféricas para su extracción, mientras que la corriente de agua producto (**R1**) es llevada al evaporador (**EVA**), en donde recibe calor del espacio o fluido a enfriar (**AF1**) y la corriente de agua producto (**R2**) termina de evaporarse, produciendo el efecto frigorífico o de enfriamiento. La corriente (**AF1**) después de transferir calor con la corriente (**R1**), sale del evaporador (**EVA**) como la corriente (**AF2**). El agua producto (**R2**) en fase de vapor, se
- 20 mezcla con la corriente de vapor (**VfAVa**) proveniente del **Efecto fAV** y con la corriente de vapor (**R3**) en el mezclador (**MIXVE**), convirtiéndose en la corriente de vapor (**FS**). La corriente de vapor (**R3**) solo se genera y se mezcla en los casos donde se utiliza la salmuera fría para el enfriamiento del eyector-condensador (**EYC**). La corriente de vapor de agua (**FS**) a baja presión y temperatura, es succionada, arrastrada y comprimida para provocar su
- 25 condensación en el eyector-condensador (**EYC**), el cual es un tipo especial de eyector que involucra la condensación de vapor. En este elemento (eyector-condensador), la interacción entre la corriente motriz (**corriente FP4**) y la corriente de succión (**corriente FS**) genera una transferencia significativa de cantidad de movimiento, masa y calor, entre ambas corrientes. Este proceso favorece la condensación del vapor, transformando la corriente succionada en
- 30 una corriente líquida al salir del eyector-condensador (**EYC**). El fluido motriz utilizado en el **EYC**, es la corriente de agua producto (**FP4**), que opera a una presión superior tanto a la de



la corriente de vapor de baja presión (**FS**) como a la presión intermedia de operación del sistema, la cual se presenta en la **corriente FP5**. La presión de la corriente (**FS**) puede ajustarse para garantizar un nivel adecuado de enfriamiento en el evaporador (**EVA**), siendo una condición preferente una presión cercana a 0.8 kPa, ideal para aplicaciones en sistemas de aire acondicionado. En el interior del eyector-condensador (**EYC**), las corrientes (**FS**) y (**FP4**) se mezclan y salen como la corriente líquida (**FP5**) a una presión intermedia que se encuentra por debajo de la presión del fluido motriz (**FP4**), pero superior a la del vapor succionado (**FS**), asegurando así la completa condensación del vapor. La condensación total del vapor se presenta a la salida del eyector-condensador (**EYC**), donde la corriente (**FP5**) experimenta una reducción en su velocidad debido al incremento en el área transversal del eyector-condensador (**EYC**). Este fenómeno provoca un aumento adicional de presión, completando la transformación del vapor en líquido (condensación). La corriente mezclada y condensada a la salida del eyector-condensador (**EYC**), sale como la corriente (**FP5**) y entra a la columna barométrica (**CB1**), de donde sale por el fondo como agua producto a presión atmosférica (**FP6**) hacia el tanque para agua producto (**TAP**). La columna barométrica (**CB1**) se utiliza para lograr un sello hidráulico y mantener el vacío en el sistema. Posteriormente, el fluido sale por el fondo de la (**CB1**) a presión atmosférica y se deposita en el tanque para agua producto (**TAP**), de donde salen las corrientes (**FP1**) y (**ApTb**). La bomba para agua producto (**BAP**) succiona a la corriente (**FP1**) para activar nuevamente al eyector-condensador. La corriente de agua producto (**FP2**) que sale de la bomba (**BAP**), ingresa al intercambiador de calor (**HXsw**), en donde la corriente de agua salada (**AM11**), le retira el calor de condensación que recibió del vapor (**FS**). Por lo que, la corriente de agua producto (**FP4**) mantiene la misma condición de temperatura a la entrada del eyector-condensador (**EYC**). Por otra parte, la corriente de agua salada (**AM11**), previamente sale del tanque para agua de mar (**TAM**) como la corriente (**AM9**), la cual pasa por la válvula de paso (**VP1**) y es succionada por la bomba para agua de mar (**BAM2**) como la corriente (**AM10**) y es llevada hacia el intercambiador de calor (**HXsw**) como la corriente (**AM11**). Dicha corriente de agua salada (**AM11**) después de enfriar a la corriente (**FP2a**), sale como la corriente de agua salada (**AM12**) y es dirigida al contenedor de salmuera para diluir la corriente de salmuera (**Sne**) que proviene de la válvula de tres vías (**V3V2**) y que a la salida de la columna barométrica (**CB2**) se convierte en la corriente (**Snf**).

El **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío**, presenta diferentes modos de operación; 1) producción de agua desalinizada y 2) producción de frío y agua desalinizada. En el caso donde únicamente se produce agua desalinizada, es necesario que la válvula tres vías (**V3V1**) mantenga cerrado el paso hacia la corriente (**R1**), evitando con esto el paso hacia el evaporador (**EVA**) y dirija todo el flujo de la corriente (**ApT**) hacia la salida de la corriente (**ApTa**). Debido a que en este caso el evaporador (**EVA**) no se encuentra en operación, la corriente de vapor (**R2**) no es generada, por lo cual el eyector-condensador (**EYC**) sólo tiene que recibir y condensar la corriente de vapor (**VfAVa**) y en el caso que aplique, la corriente de vapor (**R3**). En el caso donde se produce frío y agua desalinizada de manera simultánea, la válvula de tres vías (**V3V1**) para el control de la capacidad de enfriamiento, recibe la corriente del agua producto total de los efectos (**ApT**) y la divide en las corrientes (**ApTa**) y (**R1**). La corriente (**ApTa**) se dirige a la columna barométrica (**CB3**) para ser extraída directamente como agua desalinizada, mientras que la corriente de agua producto (**R1**) es utilizada como refrigerante para la producción de frío. La capacidad de enfriamiento es regulada por la cantidad de flujo de refrigerante (**R1**) que ingresa al evaporador (**EVA**). Lo anterior, también determina la cantidad de flujo de vapor (**R2**) que se dirige al mezclador (**MIXVE**) y posteriormente al eyector-condensador (**EYC**).

Cuando se quiera enfriar el eyector-condensador (**EYC**) de manera convencional utilizando únicamente agua de mar para disipar el calor generado, las válvulas de paso (**VP1**), (**VP3**), (**VP6**) y (**VP8**) estarán abiertas, mientras que las válvulas de paso (**VP2**), (**VP4**), (**VP5**) y (**VP7**) permanecerán cerradas. Por su parte, la válvula de tres vías (**V3V2**) mantendrá cerrado el paso de la corriente (**Snb**) y redireccionará todo el flujo hacia la salida de la corriente (**Sna**). La corriente motriz de agua producto (**FP1**) sale del tanque para agua producto (**TAP**) y es succionada por la bomba (**BAP**) para pasar por la válvula de paso (**VP3**) y llegar al intercambiador de calor (**HXsw**) y posteriormente salir como la corriente (**FP2c**) y pasar por las válvulas de paso (**VP6**) y (**VP8**) y llegar a la entrada del eyector-condensador (**EYC**) como la corriente motriz (**FP4**). En el caso que se quiera enfriar el eyector-condensador (**EYC**) utilizando de manera híbrida agua salada y salmuera, la válvula de tres vías (**V3V2**) permitirá de manera parcial o total (dependiendo el nivel de enfriamiento requerido en el eyector-condensador) el paso de la corriente de salmuera (**Snb**). Las válvulas de paso (**VP1**),

(VP3), (VP5) y (VP7) estarán abiertas, mientras que las válvulas de paso (VP2), (VP4), (VP6) y (VP8) permanecerán cerradas, para que el fluido motriz (FP2) que sale de la bomba (BPA) pase por la válvula de paso (VP3) y disipe la ganancia de calor del proceso al intercambiar energía en forma de calor con la corriente de agua salada (AM11) en el  
5 intercambiador (HXsw). Posteriormente la corriente (FP2c) pasa a través de la válvula de paso (VP5) y sale como la corriente (FP3a), la cual se dirige hacia el intercambiador de calor (HXBR) para enfriarse a una menor temperatura al intercambiar calor con la corriente de salmuera saturada y baja temperatura (Snb), y salir como la corriente (FP3b) que ingresa por la válvula de paso (VP7) antes de dirigirse al eyector-condensador (EYC). Posteriormente la  
10 corriente de fluido motriz (FP4) ingresa con menor temperatura al eyector-condensador (EYC). Finalmente, cuando se quiere enfriar el eyector-condensador (EYC) únicamente con la corriente de salmuera saturada y a baja temperatura (SfAV) que sale del último efecto del proceso MED, es necesario que la válvula de tres vías (V3V2) cierre el paso a la corriente (Sna) y dirija todo el flujo de salmuera hacia la salida de la corriente (Snb), además las  
15 válvulas de paso (VP1), (VP3), (VP6) y (VP8) permanecerán cerradas, mientras que las válvulas de paso (VP2), (VP4), (VP5) y (VP7) deben estar abiertas para permitir que el fluido motriz (FP1) que sale del tanque para agua producto (TAP), pase a través de la válvula de paso (VP2) y salga como la corriente (FP2b) hacia la válvula de paso (VP4), y posteriormente intercambie energía en forma de calor con la corriente de salmuera (Snb) en  
20 el intercambiador para salmuera (HXBR). En los últimos dos casos, cuando se utiliza la corriente de salmuera saturada (Snb) de manera parcial o total para el enfriamiento del eyector-condensador (EYC), la corriente de salmuera sale del intercambiador de calor (HXBR) como la corriente en dos fases (Snc), la cual es llevada al separador líquido-vapor (SEP15) de donde sale dividida en las corrientes de salmuera líquida (Snd) y de vapor de  
25 agua (R3). La corriente de salmuera (Snd) se une a través de una tubería con la corriente de salmuera (Sna) para convertirse en la corriente de salmuera (Sne) y dirigirse hacia la columna barométrica (CB2), la cual produce un sello hidráulico y mantiene el vacío en el sistema. Posteriormente la corriente de salmuera (Sne) es extraída del fondo de la columna barométrica (CB2), como la corriente (Snf) a presión atmosférica y ser depositada en el  
30 tanque para salmuera (TS) de donde sale como la corriente de salmuera (RETM). Por su parte la corriente de vapor (R3) es dirigida hacia el mezclador (MIXVE) para mezclarse con

las corrientes de vapor (**VfAVa**), (**R2**) que se encuentran a las mismas condiciones de presión y temperatura y convertirse en la corriente de vapor (**FS**) que es succionada por el eyector-condensador (**EYC**) como el fluido secundario. El eyector-condensador (**EYC**) permite operar la unidad de desalinización por evaporación de múltiple efecto (MED) a un nivel de vacío significativamente superior al convencional. Mientras que los valores típicos presión de vacío en el último efecto de sistemas convencionales se encuentran en el rango de 5-7 kPa, la invención propuesta alcanza presiones de vacío en el último efecto que oscilan entre 0.7-1 kPa, lo cual mejora sustancialmente la eficiencia del proceso.

El **subproceso B** anteriormente descrito, puede ser integrado a cualquier proceso de evaporación al vacío como el de múltiple efecto, múltiple etapa flash, de destilación por membranas o cualquier otro subproceso de desalación que opere en presiones de vacío, dado que en todos ellos se tiene un agua producto que puede ser utilizada como refrigerante.

## REIVINDICACIONES

Habiendo descrito suficientemente mi invención, considero como una novedad y por lo tanto reclamo como de mi exclusiva propiedad, lo contenido en las siguientes cláusulas:

1. Un **sistema de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío**; caracterizado porque comprende un **subsistema A (desalinización por evaporación de múltiple efecto al alto vacío)** y un **subsistema B (producción de frío y condensación del agua producto)**, en donde:

a) El **subsistema A de desalinización por evaporación de múltiple efecto al alto vacío**, está constituido por al menos un efecto, donde el **Efecto 0** está compuesto de un intercambiador de calor que actúa como generador de vapor (**GEN**), interconectado con una válvula de expansión (**VE1**) para agua de mar y un separador líquido-vapor para salmuera (**SEP0**), que a su vez tiene dos salidas; una para la corriente de vapor (**V0**) conectada con el precalentador para agua de mar (**PC0**) y otra para la corriente de salmuera (**S0**) que se conecta con una tubería tipo U (**TU0**), las cuales a su vez se interconectan posteriormente con el intercambiador de calor que funciona como el evaporador del **Efecto 1 (HX1)**; interconectando en el **Efecto 1**, el **HX1** con el separador (**SEP1**) y con la válvula de expansión para agua producto (**VE1AP**) que a su vez se conecta con el separador para agua producto (**SEP1AP**), cuyas dos salidas se conectan al mezclador de vapor (**MIX1**), que posteriormente se interconecta con el precalentador para agua de mar (**PC1**) y con el mezclador de agua producto (**MIX2AP**), de modo que la interconexión entre los elementos del **Efecto 1** se repite hasta el **Efecto n**; mientras que en la **Etapa del Efecto 1 hasta el Efecto n**, presentan los mismos componentes e interconexión que los efectos anteriores a excepción de los precalentadores para agua de mar;

b) El **subsistema B**, está constituido por un tanque para agua de mar (**TAM**) que se conecta a una válvula de paso (**VP1**) y posteriormente a la bomba para agua de mar (**BAM2**), que se conecta con el intercambiador de calor para agua de

mar (**HXsw**), que a su vez se conecta con el tanque para salmuera (**TS**); dicho tanque para salmuera (**TS**) contiene una columna barométrica (**CB2**) que se interconecta con una válvula de 3 vías para salmuera (**V3V2**), asimismo la válvula **V3V2** se conecta con el intercambiador de calor para salmuera (**HXBR**) por medio de la corriente de salmuera (**Snb**); por otra parte, el **subsistema B**, incluye una válvula de tres vías para agua producto (**V3V1**) que tiene dos vías de salida, una para la corriente (**ApTa**) que se conecta a través de una tubería con la columna barométrica (**CB3**) y otra para la corriente (**R1**) que se conecta con el intercambiador de calor que funciona como evaporador (**EVA**), el cual se conecta con el mezclador de vapor (**MIXVE**) mediante la corriente (**R2**); en el mezclador de vapor (**MIXVE**) se conectan las corrientes de vapor (**R2**), (**R3**) y (**VfAVa**), que provienen del **EVA**, **SEP15** y el **MIXfVA**, respectivamente, teniendo como salida la corriente de vapor (**FS**), que a su vez se conecta con el eyector-condensador (**EYC**) que se encuentra interconectado con la columna barométrica (**CB1**) contenida en el tanque para agua producto que tiene dos salidas, una para la corriente de agua producto (**ApTb**) y otra para la corriente de agua producto (**FP1**), que a su vez se conecta con la bomba (**BAP**) y asimismo esta se interconecta con las válvulas de paso (**VP2**) y (**VP3**) para posteriormente interconectarse con el intercambiador (**HXsw**) y/o el intercambiador (**HXBR**), este último conectado a un separador para salmuera (**SEP15**) que tiene dos salidas, una para la corriente de salmuera (**Snd**) que se une por una tubería a la corriente (**Sna**) y se convierte en la corriente de salmuera (**Sne**), y otra para la corriente de vapor (**R3**) que se conecta al mezclador de vapor (**MIXVE**);

2. Un **proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío**; que está compuesto de un **subproceso A (desalinización por evaporación de múltiple efecto al alto vacío)** y un **subproceso B (producción de frío y condensación del agua producto)**, caracterizado porque:

- a) El **subproceso A de desalinización por evaporación de múltiple efecto al**

**alto vacío**, que utiliza agua de mar o agua salobre y consta de las etapas:

- i. Etapa del **Efecto 0**, que recibe la corriente de agua salada de alimentación (**AM5**) que previamente fue precalentada en los precalentadores para agua de mar (**PCn** hasta el **PC1**) y que proviene del **subproceso B**; la corriente de agua salada de alimentación (**AM5**) ingresa al precalentador para agua de mar (**PC0**) donde termina de ser precalentada por la corriente de vapor (**V0**) y sale como la corriente de agua salada (**AM6**), mientras que la corriente de vapor (**V0**) sale en dos fases como la corriente (**V0a**) y se dirige al **Efecto 1**; la corriente de agua salada (**AM6**) se dirige hacia la válvula de expansión (**VE1**) donde disminuye su presión hasta la presión del **Efecto 0**, que corresponde a la temperatura de saturación del **Efecto 0** y sale como la corriente de agua salada (**AM7**); la etapa del **Efecto 0**, tiene un intercambiador de calor que actúa como generador de vapor (**GEN**) que recibe a la corriente del fluido caloportador (**FC1**) y a la corriente de agua salada (**AM7**); la corriente del fluido caloportador (**FC1**), la cual puede ser cualquier tipo de fluido que tenga la temperatura suficiente para generar vapor en el generador (**GEN**) y además puede provenir de cualquier tipo de proceso o tecnología, preferentemente de tecnologías termosolares como: Concentradores Parabólicos Compuestos, Concentradores solares tipo Cilindro Parabólico, Fresnel, entre otros; en el generador (**GEN**), la corriente de agua salada (**AM7**) es calentada por el fluido caloportador (**FC1**), provocando que la corriente de agua salada (**AM7**) se convierta en dos fases (líquido-vapor) y salga del generador (**GEN**) como la corriente de agua salada (**AM8**) para dirigirse hacia el separador líquido-vapor para salmuera (**SEP0**), del cual salen dos corrientes, la corriente de vapor (**V0**) y la corriente de salmuera (**S0**); la corriente de vapor (**V0**) se dirige hacia el precalentador para agua de mar (**PC0**) donde cede su energía en forma de calor latente para terminar de precalentar a la corriente de agua de mar (**AM5**) y salir en dos fases como la corriente (**V0a**), y así posteriormente ingresar al intercambiador de calor que actúa como evaporador del **Efecto 1** (**HX1**); por otra parte la corriente de salmuera (**S0**) se dirige hacia la

tubería U (**TU0**), donde se logra un sello hidráulico que mantiene la diferencia de presión entre el **Efecto 0** y **Efecto 1**, por lo cual la corriente de salmuera se flashea y sale como la corriente (**S0a**) para ingresar posteriormente al intercambiador de calor (**HX1**);

- 5           ii. Etapa del **Efecto 1** hasta el **Efecto n**, donde el **Efecto 1** tiene un intercambiador de calor que actúa como evaporador del **Efecto 1** (**HX1**) que recibe dos corrientes; la corriente en dos fases líquido-vapor (**V0a**) que proviene del precalentador del **Efecto 0** (**PC0**) y la corriente de salmuera en dos fases (**S0a**) que proviene de la tubería U (**TU0**); en el intercambiador
- 10 de calor (**HX1**) la corriente (**V0a**) cede su energía en forma de calor latente hacia la corriente de salmuera (**S0a**), saliendo como la corriente condensada de agua producto (**AP1**) que se dirige a la válvula de expansión para agua producto (**VE1<sub>AP</sub>**) donde reduce su presión hasta la presión del siguiente efecto (**Efecto 1**) y salir como la corriente de agua en dos fases (**AP1a**) para
- 15 posteriormente ingresar al separador líquido-vapor para agua (**SEP1<sub>AP</sub>**) del cual salen las corrientes de vapor (**Vp1**) y la corriente de agua producto (**AP1b**), mientras que la corriente de salmuera (**S0a**) sale del intercambiador de calor (**HX1**) como la corriente (**S0b**) con una mayor cantidad de vapor, dirigiéndose hacia el separador líquido-vapor para salmuera (**SEP1**), del
- 20 cual salen dos corrientes, la corriente de vapor (**V1**) y la corriente de salmuera (**S1**); la corriente de vapor (**V1**) se dirige hacia el mezclador de vapor (**MIX1**) donde se mezcla con la corriente de vapor (**Vp1**) que proviene del separador líquido-vapor para agua (**SEP1<sub>AP</sub>**) y salen juntas como la corriente de vapor (**V1a**) que ingresa al precalentador del siguiente efecto (**PC1**), en tanto que la corriente de salmuera (**S1**) se dirige a la tubería
- 25 U (**TU1**) para posteriormente bajar su presión hasta la presión del siguiente efecto (**Efecto 2**); en cambio la corriente de agua producto (**AP1b**) se dirige al mezclador de agua (**MIX2<sub>AP</sub>**) para mezclarse con la corriente de agua producto (**AP2**) que sale del intercambiador de calor que actúa como evaporador del **Efecto 2** (**HX2**); en el mezclador para agua producto
- 30 (**MIX2<sub>AP</sub>**) se mezclan las corrientes de agua producto (**AP1b**) y (**AP2**) y



salen mezcladas como la corriente de agua producto (**AP2a**), dirigiéndose hacia la válvula de expansión (**VE2AP**) donde se reduce su presión hasta la presión del siguiente efecto, saliendo como la corriente de agua producto (**AP2b**) para mezclarse con la corriente de agua producto del siguiente; lo ocurrido en el **Efecto 1**, sucede en los posteriores efectos hasta el **Efecto n**, presentando una caída de presión y temperatura en efecto cascada desde el **Efecto 1** hasta el **Efecto n**;

- iii. Etapa del **Efecto 1AV** hasta el **Efecto nAV**, donde el proceso que se lleva a cabo es semejante que el de la etapa anterior (**Etapa del Efecto 1 hasta el Efecto n**), no obstante, se destaca como diferencia que en esta sección se reduce el número de componentes dado que no se utilizan precalentadores para agua de mar al no haber requerimiento de los mismos por la propia naturaleza del **subproceso A**; en el **Efecto 1AV** a diferencia de los efectos de la etapa anterior, no se cuenta con un precalentador para agua de mar, por lo que todo el vapor que se genera tanto por flasheo y ebullición ingresa al mezclador de vapor (**MIX1AV**) a través de las corrientes de vapor (**Vp1AV**) y (**V1AV**), provenientes del separador líquido-vapor para agua (**SEP1AVAP**) y del separador líquido-vapor para salmuera (**SEP1AV**), para mezclarse y posteriormente ingresar al intercambiador de calor (**HX2AV**) que actúa como evaporador del **Efecto 2AV** y calentar a la corriente de salmuera (**S1AVa**); lo ocurrido en el **Efecto 1AV**, sucede en los posteriores efectos hasta el **Efecto nAV**, presentando una caída de presión y temperatura en efecto cascada desde el **Efecto 1AV** hasta el **Efecto nAV** y utilizando todo el vapor generado en cada efecto para calentar la salmuera del efecto posterior y generar mayor cantidad de vapor;
- iv. Etapa del **Efecto fAV**, donde se presenta el mismo proceso a los efectos de la etapa previa (**Etapa del Efecto 1AV hasta el Efecto nAV**), sin embargo, al ser el último efecto de los múltiples efectos del **subproceso A**, la corriente del vapor total (**VfAVa**) generado en el **Efecto fVA**, la corriente de salmuera (**SfAV**) y la corriente de agua producto total (**ApT**) que acumula los condensados de agua producto de los múltiples efectos, se dirigen al

**subproceso B**; la corriente de vapor total (**VfAVa**) generado en el **Efecto fAV** tanto por flasheo como por ebullición, se dirige al **subproceso B** para ingresar al mezclador de vapor (**MIXVE**), mientras que la corriente de salmuera (**SfAV**) y la corriente de agua producto (**ApT**) que también se dirigen al **subproceso B**, ingresan a la válvula de tres vías para salmuera (**V3V2**) y a la válvula de tres vías para agua (**V3V1**), respectivamente;

b) El **subproceso B de producción de frío y condensación del agua producto**, consta de las etapas:

- i. Etapa de **producción de frío**, en donde la corriente de agua producto (**ApT**) proveniente del **Efecto n** del **subproceso A** ingresa a una válvula de tres vías para agua (**V3V1**), la cual se encarga de separar dicha corriente en las corrientes de agua producto (**ApTa**) y de refrigerante (**R1**); la corriente de refrigerante (**R1**) ingresa a un intercambiador de calor que actúa como evaporador (**EVA**) para la producción de frío; en el evaporador (**EVA**) la corriente de refrigerante (**R1**) recibe el calor de la corriente (**AF1**) para salir como la corriente de vapor (**R2**) y dirigirse al mezclador de vapor (**MIXVE**) de la siguiente etapa; por otra parte, la corriente de agua producto (**ApTa**) se dirige hacia la columna barométrica (**CB3**), la cual se utiliza para lograr un sello hidráulico y mantener el vacío en el sistema, saliendo por el fondo de la columna barométrica (**CB3**) como la corriente (**FP6**) a presión atmosférica y posteriormente depositarse en el tanque para agua producto (**TAP**); la función de la válvula de tres vías para agua (**V3V1**) es la de controlar la capacidad de enfriamiento regulando la cantidad de flujo de la corriente refrigerante (**R1**) que ingresa al evaporador (**EVA**);
- ii. Etapa de **eyecto-condensación barométrica**, que consta de un tanque para agua producto (**TAP**) el cual tiene dos salidas, la corriente de agua producto (**ApTb**) y la corriente de agua producto (**FP1**); en la corriente de agua (**ApTb**) se extrae el total de agua desalinizada, mientras que la corriente de agua producto (**FP1**) es succionada por la bomba para agua producto (**BAP**) donde incrementa su presión y sale como la corriente de agua (**FP2**); el nivel

de presión de la corriente de agua (**FP2**) que sale de la bomba (**BAP**) debe ser una presión superior a la presión de la corriente (**FP5**) de descarga del eyector-condensador (**EYC**) y a la corriente de vapor de baja presión (**FS**) que se utiliza como fluido secundario en el eyector-condensador (**EYC**);

5 por otra parte, el nivel de presión de la corriente de vapor de baja presión (**FS**) puede ser cualquier presión correspondiente a la temperatura de saturación capaz de producir frío en el evaporador (**EVA**), siendo una modalidad preferente 0.8 kPa para aire acondicionado; la **etapa de eyecto-**

10 **condensación barométrica** tiene la capacidad de operar en tres diferentes modalidades: 1) cuando el eyector-condensador barométrico (**EYC**) es enfriado por agua de mar, 2) cuando el eyector-condensador barométrico (**EYC**) es enfriado con agua de mar y salmuera y 3) cuando el eyector-

15 condensador barométrico (**EYC**) es enfriado únicamente con salmuera; cuando el eyector-condensador barométrico (**EYC**) es enfriado solo con agua de mar (en la primera modalidad mencionada), las válvulas de paso (**VP2**), (**VP4**), (**VP5**) y (**VP7**) deben permanecer cerradas, mientras que las

20 válvulas de paso (**VP1**), (**VP3**), (**VP6**) y (**VP8**) deben estar abiertas para permitir que la corriente de agua producto (**FP2**) que actúa como fluido motriz en el eyector-condensador (**EYC**) pase a través de la válvula (**VP3**) y se dirija al intercambiador de calor que actúa como disipador de calor (**HXsw**); en el intercambiador de calor (**HXsw**) la corriente de agua de agua

25 de mar (**AM11**) que proviene de la bomba (**BAM2**) para agua de mar, recibe el calor de la corriente de agua producto (**FP2a**) proveniente de la válvula de paso (**VP3**) y posteriormente sale del intercambiador de calor (**HXsw**) como la corriente de agua de mar (**AM12**) que se dirige al tanque para salmuera (**TS**); por otra parte, la corriente de agua producto (**FP2c**) sale del

30 intercambiador de calor (**HXsw**) y se dirige hacia la válvula de paso (**VP6**) donde sale como la corriente de agua producto (**FP3**) pasando también por la válvula de paso (**VP8**) y salir como la corriente de agua producto (**FP4**), la cual ingresa al eyector-condensador (**EYC**) como fluido primario; en cuanto a la válvula de tres vías (**V3V2**) para salmuera, debe cerrar

totalmente el paso de la corriente de salmuera (**Snb**) para que todo el flujo de salmuera de la corriente (**SfAV**) sea dirigido por la corriente (**Sna**); en la segunda modalidad donde el eyector-condensador barométrico (**EYC**) es enfriado con agua de mar y posteriormente con salmuera, las válvulas de paso (**VP2**), (**VP4**), (**VP6**) y (**VP8**) deben permanecer cerradas, mientras que las válvulas de paso (**VP1**), (**VP3**), (**VP5**) y (**VP7**) deben estar abiertas para permitir que la corriente de agua producto (**FP2**) pase a través de la válvula de paso (**VP3**) y dirigirse al intercambiador de calor (**HXsw**) donde es enfriada por la corriente de agua de mar (**AM11**); posteriormente la corriente de agua producto (**FP2**) sale del intercambiador de calor (**HXsw**) como la corriente (**FP2c**) y se dirige hacia el intercambiador de calor (**HXBR**) pasando a través de la válvula de paso (**VP5**), para ser enfriado por la corriente de salmuera (**Snb**) y salir como la corriente de agua producto (**FP3b**) que se dirige a la válvula de paso (**VP7**) y sale como la corriente de agua producto (**FP4**), la cual ingresa al eyector-condensador (**EYC**) como fluido primario; por otra parte, la válvula de tres vías (**V3V2**) para salmuera, permite que la corriente de salmuera (**SfAV**) pase total o parcialmente por la vía de la corriente de salmuera (**Snb**); en la tercera modalidad donde el eyector-condensador (**EYC**) es enfriado con salmuera, las válvulas de paso (**VP3**), (**VP6**) y (**VP8**) deben permanecer cerradas, mientras que las válvulas de paso (**VP1**), (**VP2**), (**VP4**), (**VP5**) y (**VP7**) deben estar abiertas, para permitir que la corriente de agua producto (**FP2**) pase a través de las válvulas de paso (**VP2**) y (**VP4**) de donde sale como la corriente de agua producto (**FP2c**); la corriente de agua producto (**FP2c**) fluye a través de la válvula de paso (**VP5**) y se dirige al intercambiador de calor (**HXBR**) donde es enfriada por la corriente de salmuera (**Snb**); en el intercambiador de calor (**HXBR**) la corriente de agua producto (**FP3a**) se enfría al transferir calor con la corriente salmuera (**Sna**) y sale como la corriente de agua producto (**FP3b**) que se pasa por la válvula de paso (**VP7**) e ingresa al eyector-condensador (**EYC**) como la corriente de agua producto (**FP4**) que actúa como fluido primario; por otra parte, la corriente de salmuera (**Sna**) que

- 5                    enfría a la corriente de agua producto (**FP3a**) al recibir su calor en forma de calor latente en el intercambiador de calor (**HXBR**), sale como la corriente (**Snc**) en dos fases (líquido-vapor), dirigiéndose al separador líquido-vapor para salmuera (**SEP15**) del cual salen dos corrientes, la corriente de vapor (**R3**) y la corriente de salmuera (**Snd**); en esta modalidad donde eyector-condensador (**EYC**) es enfriado con salmuera, la corriente de vapor (**R3**) se dirige al mezclador de vapor (**MIXVE**) para mezclarse con las corrientes de vapor (**VfAVa**) y (**R2**), y convertirse en la corriente de vapor (**FS**) e ingresar al eyector-condensador (**EYC**) como el fluido secundario, mientras
- 10                    que la corriente de salmuera (**Snd**) sale del separador líquido-vapor para salmuera (**SEP15**) y se convierte en la corriente de salmuera (**Sne**) que ingresa a la columna barométrica para salmuera (**CB2**), la cual se utiliza para lograr un sello hidráulico y mantener el vacío en el sistema, saliendo por el fondo de la columna barométrica (**CB2**) como la corriente de
- 15                    salmuera (**Snf**) a presión atmosférica y posteriormente depositarse en el tanque para salmuera (**TS**) de donde sale la corriente de salmuera (**RETM**); en cada una de las tres diferentes modalidades en la que se puede operar la etapa de **eyecto-condensación barométrica**, la corriente de vapor (**FS**) que actúa como fluido secundario en el eyector-condensador (**EYC**), es
- 20                    succionada, arrastrada y comprimida por la corriente (**FP4**) en el eyector-condensador (**EYC**); el eyector-condensador (**EYC**) utiliza como fluido motriz la corriente de agua producto (**FP4**) proveniente de la bomba (**BAP**), provocando que el fluido motriz (**FP4**) de alta presión, se mezcle con la corriente de vapor (**FS**) de baja presión y salgan como la corriente líquida
- 25                    (**FP5**) a cualquier presión intermedia (menor a la presión de la **corriente FP4** y mayor a la presión de la **corriente FS**) que asegure la condensación de la corriente de vapor (**FS**).
- 30                    3. El proceso de acuerdo con la **reivindicación 2** caracterizado además por que en la **etapa de producción de frío**, que pertenece al **subproceso B**, la cantidad de flujo de la corriente refrigerante (**R1**) es nula o igual a cero, con lo cual el proceso se destina

únicamente a brindar el servicio de desalinización, sin brindar el servicio de producción de frío o calefacción para climatización de espacios.

4. El proceso de acuerdo con la **reivindicación 2**, caracterizado por qué el proceso de evaporación al vacío en el **subproceso A**, puede ser de múltiple efecto, múltiple etapa, de membranas o cualquier otro subproceso de desalación que opere en presiones de vacío.

## RESUMEN

En la presente invención se presenta un novedoso **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío para la producción simultánea de agua y frío** activado con energía solar térmica, con la finalidad de contribuir en el desarrollo de alternativas tecnológicas que se presenten como una opción viable para contrarrestar las problemáticas que representan la escasez de agua para uso humano y la climatización de espacios en comunidades costeras. El proceso opera con una cascada de presión al alto vacío, la cual alcanza la presión correspondiente a una temperatura de saturación (BBT) de 4 °C en el último efecto del sistema, rebasando el límite de la BBT de los procesos MED convencionales. Además, como consecuencia de operar en condiciones de alto vacío, el agua producto es utilizada de manera directa como fluido refrigerante para la producción simultánea de frío. Lo anterior, se logra con la implementación de un eyector-condensador que succiona, arrastra y comprime para condensar, el vapor frío del último efecto de la MED. Esto tiene un impacto directo en la capacidad y en el incremento de los indicadores de eficiencia del proceso. Este proceso está constituido por un **subproceso A de desalinización por evaporación de múltiple efecto al alto vacío** y el **subproceso B de producción de frío y condensación del agua producto**. Las características antes descritas, hacen de esta invención una opción recomendable para los sectores residencial, agrícola, ganadero, turístico, comercial e industrial que se encuentran en zonas costeras, con clima cálido-húmedo donde se tenga problemas con la disponibilidad de agua potable y altas temperaturas.



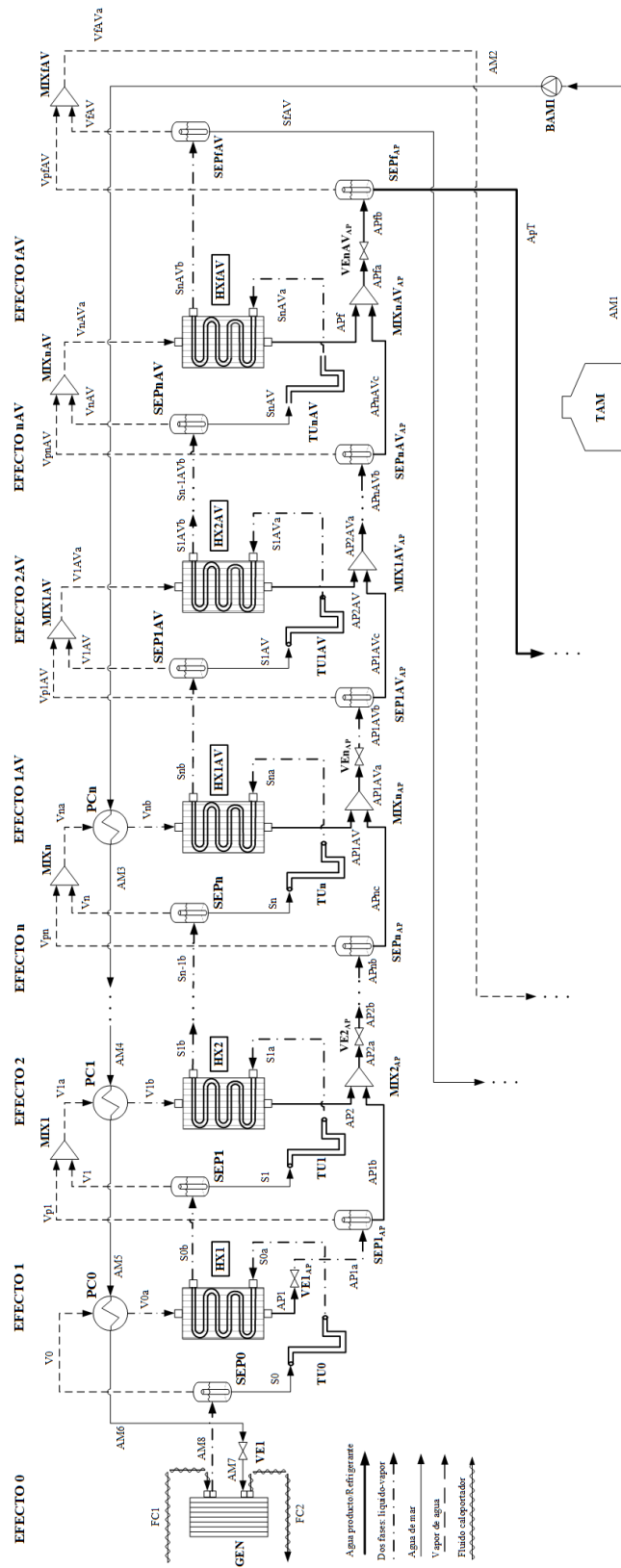


FIGURA 2



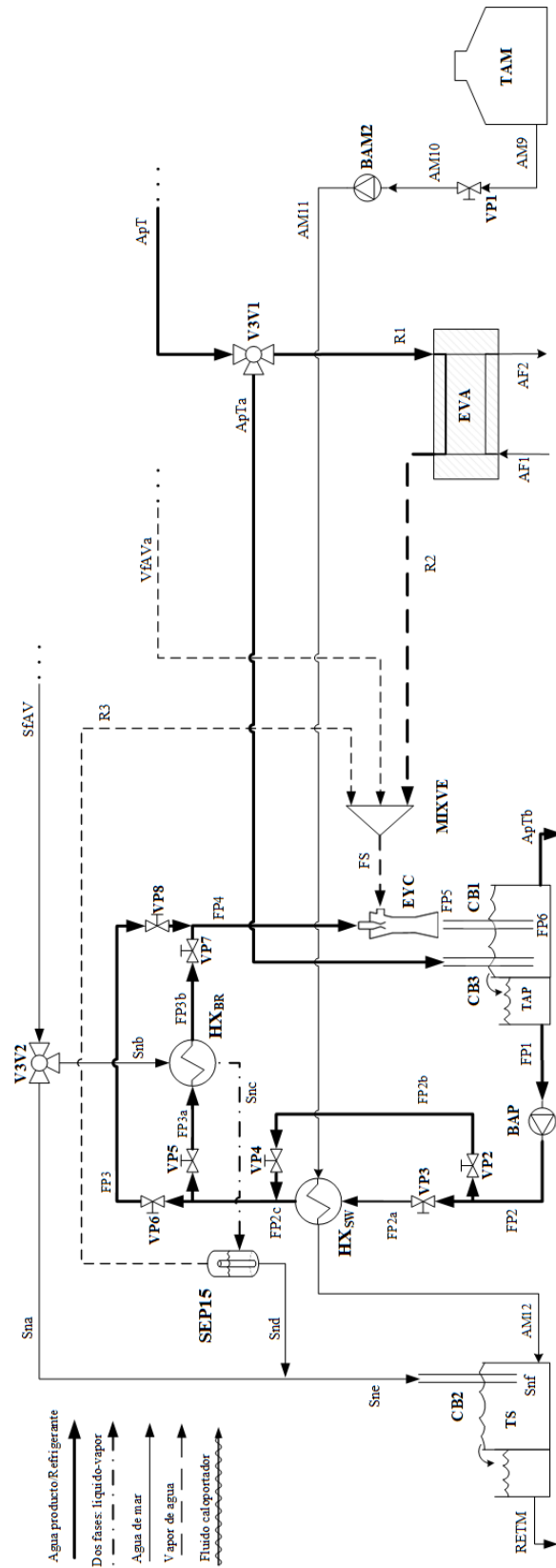


FIGURA 3

### **3.5. Solicitud de patente B: Sistema y proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios.**

Esta invención se refiere a un nuevo sistema y proceso de desalinización que redefine los límites operativos de la tecnología MED al permitir la activación con bajas temperaturas. La reivindicación principal radica en operar con una Temperatura Mínima de la Salmuera (BBT) muy por debajo de los valores convencionales (35-45 °C), alcanzando, en un modo de realización, los 4 °C. Esta drástica reducción de la BBT permite, a su vez, disminuir la Temperatura Máxima de la Salmuera (TBT) hasta valores tan bajos como la temperatura de alimentación del agua salada (25 °C).

Este avance tecnológico se logra mediante una configuración que opera a alto vacío (por ejemplo 0.8 kPa en el último efecto) e integra un eyector-condensador barométrico para condensar el vapor a bajas temperaturas. Como consecuencia directa, el agua producto se utiliza directamente como refrigerante en un evaporador. La ventaja fundamental de esta invención es su acoplamiento con fuentes de calor de activación de baja temperatura, como colectores solares térmicos, para conformar sistemas de poligeneración altamente eficientes y sostenibles para una amplia gama de sectores.

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## **Solicitud de Patente B**

# **SISTEMA Y PROCESO CON BAJA TEMPERATURA DE ACTIVACIÓN PARA LA PRODUCCIÓN SIMULTÁNEA DE AGUA Y ACONDICIONAMIENTO TÉRMICO DE ESPACIOS**

## **SOLICITUD DE PATENTE**

MX/a/2025/007161



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**Certificado de acuse de**  
**recibo registro:**  
**MX/2025/042107**

Expediente de Patente **MX/a/2025/007161**

**Asunto:** Se informa resultado del examen de forma.

Ciudad de México, a 23 de junio de 2025.

No. Folio: **60489**

**Dora Luz FLORES GUTIERREZ**  
**Apoderado de**  
**UNIVERSIDAD AUTÓNOMA DE BAJA CALIFORNIA**



REF: Se da acuse de recibo a su Solicitud

MX/2025/60489

En términos de lo dispuesto por el artículo 106 de la Ley Federal de Protección a la Propiedad Industrial y artículos 18, 19 y 26 del Acuerdo por el que se establecen lineamientos en materia de servicios electrónicos del Instituto Mexicano de la Propiedad Industrial; **se tiene por satisfecho el examen de forma** al haber cumplido con lo dispuesto por la Ley Federal de Protección a la Propiedad Industrial y el Reglamento de la Ley de la Propiedad Industrial, aplicable a la Ley Federal de Protección a la Propiedad Industrial en términos de lo dispuesto por el artículo Cuarto Transitorio del DECRETO por el que se expide la Ley Federal de Protección a la Propiedad Industrial y se abroga la Ley de la Propiedad Industrial.

Por lo anterior, de conformidad con los artículos 107 y 108 de la Ley Federal de Protección a la Propiedad Industrial; artículos 39 y 40 del Reglamento de la Ley de la Propiedad Industrial y artículo 9 del Acuerdo por el que se establecen los plazos de respuesta a diversos trámites ante el Instituto Mexicano de la Propiedad Industrial, una vez **aprobado** el examen de forma, **la publicación en Gaceta de la presente solicitud de patente en trámite**, tendrá lugar después del vencimiento del plazo de 18 meses, contado a



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Cadena Original

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TRIBUTARIA|1796|FOR|MX/2025/60489|MX/a/2025/007161|Favorable|187  
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## **SISTEMA Y PROCESO CON BAJA TEMPERATURA DE ACTIVACIÓN PARA LA PRODUCCIÓN SIMULTÁNEA DE AGUA Y ACONDICIONAMIENTO TÉRMICO DE ESPACIOS**

### **CAMPO TÉCNICO**

- 5 La presente invención se enmarca en el campo de los sistemas de desalinización y climatización de espacios y se refiere a un sistema y proceso con baja temperatura de activación que incorpora un eyector-condensador barométrico para la producción simultánea de agua desalada y el acondicionamiento térmico de espacios mediante enfriamiento y calefacción optimizando el uso de los recursos energéticos.

10

### **ANTECEDENTES**

- La desalinización de agua es una solución crucial para enfrentar la creciente escasez de recursos hídricos en diversas regiones del mundo, especialmente en aquellas con climas áridos o semiáridos. Sin embargo, los procesos tradicionales de desalinización suelen ser altamente demandantes en términos de energía, lo que limita su implementación sostenible.
- 15 Asimismo, la necesidad de sistemas eficientes de acondicionamiento térmico de espacios también plantea desafíos importantes en términos de consumo energético. La combinación de ambas necesidades representa una oportunidad para desarrollar soluciones integradas que optimicen el uso de recursos energéticos mientras abordan de manera simultánea la producción de agua desalada y el acondicionamiento térmico de espacios.
- 20 A nivel mundial, se ha incrementado el interés por integrar sistemas de desalinización con tecnologías de enfriamiento, con el objetivo de optimizar el consumo energético y ofrecer ambos servicios de manera simultánea. No obstante, estas integraciones suelen enfrentar desafíos significativos, como los altos costos iniciales asociados a la adquisición de ambos sistemas y la implementación de una red compleja de equipos para lograr su conexión. En
- 25 este contexto, la presente invención propone una solución innovadora al producir agua y enfriamiento de manera simultánea mediante un sistema y proceso de desalinización al alto vacío con baja temperatura de activación que incorpora un eyector-condensador barométrico, sin requerir la combinación con otras tecnologías existentes. Dicha invención, presenta una configuración innovadora y diferente a los sistemas de evaporación de múltiple efecto (MED)

convencionales, reduciendo la cantidad de componentes y modificando las condiciones de operación, logrando así una disminución significativa en la inversión inicial y los costos operativos, además que incrementa la viabilidad técnica y económica del proceso.

5 Dentro de las principales tecnologías térmicas de desalinización destacan la múltiple etapa flash (MSF) y la evaporación de múltiple efecto (MED). El desarrollo de la tecnología de evaporación de múltiple efecto (MED, por sus siglas en inglés) ha enfrentado limitaciones debido a los desafíos asociados con las pérdidas de energía en el efluente del condensador. Además, su rendimiento se ha visto restringido por las condiciones operativas que definen la temperatura máxima de salmuera (TBT, por sus siglas en inglés) y la temperatura mínima de salmuera (BBT, por sus siglas en inglés), según lo reportado por **(Ahmed et al., 2020)**. La TBT de un sistema MED, está condicionada por los problemas de corrosión asociados con la operación a temperaturas superiores a 70 °C al trabajar con agua de mar, que normalmente tiene una salinidad de 35,000 a 50,000 ppm **(Goh et al., 2021)**. Por otra parte, la BBT se encuentra limitada por la temperatura del agua de mar de alimentación utilizada para  
10 condensar el vapor en el último efecto del sistema MED, generalmente en un rango de 25-30 °C. Este proceso requiere un gradiente térmico mínimo de aproximadamente 5 °C entre el vapor y el agua de mar para asegurar un intercambio térmico eficiente **(Aly, 1995; Shahzad, et al., 2015)**.

Con el objetivo de mejorar la viabilidad técnica y económica de las tecnologías MED, **Aly, 20 (1995)**, fue uno de los primeros en investigar configuraciones innovadoras y explorar nuevos rangos operativos de presión y temperatura. En su estudio, se empleó una TBT de 63 °C y una BBT de 6 °C, siendo esta última significativamente más baja que en los sistemas convencionales. Esto fue posible mediante la integración de un sistema de enfriamiento por absorción LiBr-H<sub>2</sub>O de 220 kW, lo que permitió implementar una cascada de 20 efectos y  
25 mejorar los indicadores de desalinización, logrando un GOR de 14.8 y un PR de 14.2. El sistema híbrido utilizó el calor extraído del absorbedor y condensador del sistema de absorción para activar la MED, mientras que el enfriamiento del evaporador permitió condensar el vapor de baja temperatura (6 °C) generado en el último efecto de la MED. Sin embargo, esta integración no consideró la producción de frío como un servicio adicional y

presentó una alta complejidad debido al elevado número de componentes, lo que resultó en costos iniciales, operativos y de mantenimiento elevados.

En los últimos años, se han llevado a cabo investigaciones enfocadas en explorar nuevos rangos de presión y temperatura para la operación de sistemas MED (**Farsi and Dincer, 2019; Ali et al., 2020; Ghenai et al., 2021**). Entre estos estudios destacan los realizados por **Shahzad et al., (2014, 2015)**, quienes analizaron sistemas MED operando con una TBT de 50-70 °C y BBT de 3-5 °C. Lograron reportar una producción de agua entre 2 y 3 veces mayor que la de un sistema MED convencional. Para alcanzar estos resultados, fue necesario acoplar el sistema MED con una unidad de adsorción (AD) basada en gel de sílice, que permite adsorber el vapor de baja temperatura proveniente del último efecto de la MED. No obstante, esta solución presenta como principal desventaja un elevado costo inicial debido al número considerable de componentes que conforman el proceso.

En un estudio reciente, **López-Zavala et al., (2019) (MX/a/2017/016797)** presentaron un sistema innovador que combina enfriamiento y desalinización mediante un ciclo de absorción de simple efecto LiBr/H<sub>2</sub>O con tres niveles de presión y un proceso de desalinización por evaporación flash. Este sistema integró un subproceso para la extracción y acondicionamiento del agua producto, que utiliza un eyector para condensar el vapor generado en el evaporador, caracterizado por operar a baja presión y temperatura. Esta integración permitió incrementar tanto la capacidad de enfriamiento como la producción de agua, representando un avance significativo en la tecnología. Sin embargo, aunque el diseño del sistema representa un avance significativo, la mejora en la eficiencia energética aún no alcanza los niveles necesarios para hacer estas tecnologías plenamente competitivas en el mercado actual. En cuanto a las temperaturas de operación del subproceso de desalinización, que se basa en una etapa de flasheo, presenta una temperatura de saturación de 58 °C, correspondiente a la TBT. Posteriormente, el agua de mar pasa por una válvula de expansión que reduce su presión hasta alcanzar una temperatura de saturación de 9.26 °C, correspondiente a la BBT, resultando en una corriente en dos fases. Aunque el sistema mantiene una TBT convencional, logra alcanzar una BBT por debajo de los niveles convencionales gracias a la absorción del vapor de baja temperatura en el absorbedor del



ciclo de absorción. No obstante, esta integración incrementa la complejidad del sistema, el número de componentes y, consecuentemente, el costo inicial.

En un trabajo complementario, **López-Zavala et al. (2019b) (MX/a/2018/006950; MX/a/2018/015007)** presentaron la integración de un sistema de Destilación de Múltiple Efecto (MED) de 14 efectos y una etapa de desalinización por evaporación flash, acoplado a un sistema abierto de enfriamiento por absorción de simple efecto con tres niveles de presión. Además, el diseño del sistema incluyó un subsistema para la extracción y adecuación del agua producto, el cual emplea un eyector para condensar el vapor generado en el evaporador, el cual presenta baja presión y temperatura. Este diseño permitió un notable incremento en la producción de agua producto y, especialmente, en la capacidad de enfriamiento, alcanzando un Coeficiente de Rendimiento (COP) de 5.87. Sin embargo, a pesar de las mejoras logradas en el rendimiento del sistema de enfriamiento, la tecnología enfrenta limitaciones significativas debido a su alta complejidad y al elevado costo inicial, derivados del gran número de componentes requeridos. Es importante mencionar que, aunque la integración propuesta representa un avance tecnológico, las condiciones de operación del proceso de desalinización MED no presentaron innovaciones sustanciales, ya que se mantuvieron en los rangos convencionales, con una TBT de 70 °C y una BBT de 35 °C.

Asimismo, **López-Zavala et al. (2023)** analizaron un innovador sistema de desalinización MED que aprovecha el agua producto para generar enfriamiento mediante la expansión súbita y posterior condensación de vapor frío, inducida por el incremento de presión en un eyector-condensador barométrico. Sin embargo, a pesar de las mejoras significativas en la capacidad de enfriamiento y eficiencia global, la capacidad de desalinización se mantuvo en valores comparables a lo reportado previamente en la literatura (PR de 11.96). Esta limitación se debe a la operación dentro de los rangos tradicionales de temperatura de los sistemas MED convencionales (70-35 °C). Es importante señalar que dicho estudio está respaldado por una solicitud de patente bajo el expediente **MX/a/2020/008009**. A diferencia de este sistema (que utiliza una MED que opera en condiciones convencionales), la presente invención no realiza la expansión del agua producto, en su lugar se modifican las condiciones de operación de la MED, trabajando al alto vacío.

Por último, **C.A. Casares-De la Torre et al. (2024)** analizaron un novedoso sistema y proceso de desalinización por evaporación de múltiple efecto (MED) al alto vacío con eyector-condensación barométrica, operando con una TBT de 68 °C y una BBT de 4 °C, activado mediante energía solar térmica. Si bien la TBT se mantiene dentro del rango convencional, la BBT supera el límite de los procesos MED tradicionales. Esto se debe a la incorporación de un eyector-condensador, el cual succiona, arrastra y comprime el vapor frío (4 °C) del último efecto de la MED, condensándolo mediante el efecto de presión. La notable reducción de la BBT incrementa el rango de la cascada de presión y temperatura (28.6-0.8 kPa y 68-4 °C, respectivamente), lo que impacta directamente en la capacidad y en los indicadores de eficiencia del proceso. En particular, el sistema presenta un GOR de 13.49, y un RR de 0.83, lo que representa un incremento del 34.1% y 120% en comparación con un sistema MED convencional. Aunque el sistema propuesto demuestra mejoras significativas en eficiencia y capacidad, las condiciones de operación en términos de TBT no presentan innovaciones, ya que se mantienen dentro del rango convencional. Cabe destacar que este estudio está respaldado por una solicitud de patente (**MX/a/2025/001564**).

Con el objetivo de mejorar las tecnologías térmicas de desalinización, también se han explorado hibridaciones o integraciones con sistemas de generación eléctrica y enfriamiento. Dichas hibridaciones han demostrado ser efectivas para reducir significativamente los costos operativos, al tiempo que posibilitan la producción simultánea de diversos servicios, un concepto conocido como poligeneración (**Nassrullah et al., 2020**). Sin embargo, estas integraciones suelen incrementar los costos por inversión inicial y complejidad en el sistema.

Con base a lo expuesto, la presente invención introduce condiciones operativas que modifican los límites convencionales de temperatura superior e inferior (TBT y BBT, respectivamente). Esto es posible gracias a la incorporación de un eyector-condensador barométrico, que permite al sistema operar bajo condiciones de presión de alto vacío. Este diseño innovador facilita reducir la BBT desde los valores convencionales de 35-45 °C hasta valores cercanos al punto de congelamiento de la salmuera. Un valor normal y no limitativo es una BBT de 4 °C. Al disminuir la BBT, se amplía el rango operativo del sistema, logrando una cascada de presión y temperatura más eficiente. Como resultado, también se modifica y reduce la TBT desde los valores convencionales de 65-95 °C hasta valores cercanos al punto

de congelamiento de la salmuera. Un valor normal y no limitativo es una TBT de 25 °C, que pudiera coincidir con la temperatura de alimentación del agua salada. Lo anterior, modifica de manera novedosa la configuración del sistema, reduce el número de equipos y componentes y brinda nuevas condiciones de operación, reduciendo de manera significativa la inversión inicial y costos de operación, lo que representa un avance considerable en la eficiencia y viabilidad de los sistemas térmicos de desalinización.

Con base a lo anterior, se propone un **sistema y proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, el cual está constituido por un **subproceso A (desalinización por evaporación de múltiple efecto al alto vacío y baja TBT)**, y un **subproceso B (alimentación de agua salada, climatización y adecuación del agua producto mediante agua salada y salmuera fría)**.

### BREVE DESCRIPCIÓN DE FIGURAS

La figura 1, es un diagrama esquemático del **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios.**

- 5 La figura 2, es un diagrama esquemático del **subproceso A (desalinización por evaporación de múltiple efecto al alto vacío y baja TBT).**

La figura 3, es un diagrama esquemático del **subproceso B (alimentación de agua salada, climatización y adecuación del agua producto mediante agua salada y salmuera fría).**

- 10 La figura 4, es una variante a la figura 1 y es un diagrama esquemático del **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, cuando en el **subproceso A** no está presente la etapa intermedia que abarca desde el **Efecto 1** hasta el **Efecto n**, por lo que solamente consta de la etapa inicial (**Efecto 0**) y de la etapa final (**Efecto f**).

- 15 La figura 5, es una variante a la figura 4 y es un diagrama esquemático del **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, cuando en la etapa final del **subproceso A**, que corresponde al **Efecto f**, **f** es igual a cero, y por lo tanto el **subsistema y subproceso A** solo opera con un único efecto de evaporación (**Efecto 0**) y además se incluye un elemento estrangulador, por ejemplo una válvula de expansión (**VR**) en el **subproceso B**.
- 20

- La figura 6, es una variante a la figura 5 y es un diagrama esquemático del **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, cuando en el **subproceso B**, se sustituye el elemento estrangulador (**VR**) para introducir un expansor generador (**EXP**) para generación de energía eléctrica mediante la diferencia de presión.
- 25

La figura 7, es una variante a la figura 5 y es un diagrama esquemático que presenta el **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, operando sin el intercambiador de calor que actúa como evaporador (EVA) y por lo tanto solo produce el servicio de desalinización.

La figura 8, es una variante a la figura 1 y es un diagrama esquemático del **sistema y proceso de desalinización por evaporación de múltiple efecto al alto vacío con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, cuando el **subproceso A** opera con una TBT que sea menor a los valores convencionales pero mayor a la temperatura del agua salada de alimentación, por ejemplo, un rango normal y no limitativo es 35-65 °C.

### DESCRIPCIÓN

A continuación, se detalla la descripción física y operativa del presente **sistema y proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, el cual comprende de dos subsistemas y subprocesos: a) el **subsistema A y subproceso A (desalinización por evaporación de múltiple efecto al alto vacío y baja TBT)**, y b) el **subsistema B y subproceso B (alimentación de agua salada, climatización y adecuación del agua producto mediante agua salada y salmuera fría)**.

El **subsistema A**, se compone de tres etapas: la etapa inicial, correspondiente al **Efecto 0**, la etapa intermedia, que abarca del **Efecto 1** hasta el **Efecto n** y la etapa final, correspondiente al **Efecto f**. La etapa del efecto inicial (**Efecto 0**), está compuesta por un intercambiador de calor que actúa como generador (**GEN**) para la activación del sistema de evaporación de múltiple efecto (MED, por sus siglas en inglés) al alto vacío y baja temperatura superior de la salmuera (TBT, por sus siglas en inglés), una válvula de expansión (**VE1**) para agua salada y un separador líquido-vapor (**SEP0**) para salmuera y/o agua salobre. La etapa que abarca los efectos del **Efecto 1** hasta el **Efecto n**, cuenta con n intercambiadores de calor que actúan como evaporadores (**HX1, HX2, ... HXn**), donde n puede variar desde cero hasta cualquier número, n separadores líquido-vapor para salmuera y/o agua salobre (**SEP0, SEP1, ...**

- SEP<sub>n</sub>**), en donde  $n$  coincide con el número de intercambiadores de calor que actúan como evaporadores (**HX**) menos uno, ya que cada evaporador (**HX**) cuenta con un separador líquido-vapor para salmuera (**SEP**),  $n$  separadores líquido-vapor para agua producto (**SEP<sub>1AP</sub>**, **SEP<sub>2AP</sub>**, ... **SEP<sub>APn</sub>**), donde  $n$  equivale al número de evaporadores (**HX**) menos uno,  $n$  tuberías en forma de “U” (**TU<sub>0</sub>**, **TU<sub>1</sub>**, ... **TU<sub>n</sub>**), diseñadas para mantener el sello hidráulico necesario y preservar la diferencia de presión entre efectos consecutivos, donde  $n$  corresponde al número de evaporadores (**HX**) menos uno,  $n$  válvulas de expansión para el agua producto (**VE<sub>AP1</sub>**, **VE<sub>AP2</sub>**, ... **VE<sub>APn</sub>**), donde  $n$  es igual al número de evaporadores (**HX**) menos uno,  $n$  mezcladores para combinar los vapores generados durante el flasheo de la salmuera y el agua producto (**MIX<sub>1</sub>**, **MIX<sub>2</sub>**, ... **MIX<sub>n</sub>**), en donde  $n$  coincide con el número de evaporadores (**HX**) menos uno, y  $n$  mezcladores para mezclar las corrientes de agua producto (**MIX<sub>2AP</sub>**, **MIX<sub>3AP</sub>**, ... **MIX<sub>nAP</sub>**), donde  $n$  se define como el número de evaporadores (**HX**) menos dos. Finalmente, la etapa que abarca al último efecto (**Efecto f**), se compone de un intercambiador de calor que actúa como evaporador (**HX<sub>f</sub>**), un separador líquido-vapor para salmuera y/o agua salobre (**SEP<sub>f</sub>**), una tubería en forma de “U” (**TU<sub>f</sub>**), un mezclador para agua producto (**MIX<sub>fAP</sub>**), una válvula de expansión para el agua producto (**VE<sub>fAP</sub>**), un separador líquido-vapor para agua producto (**SEP<sub>f</sub>**), y un mezclador (**MIX<sub>f</sub>**) para combinar los vapores producidos durante el flasheo de la salmuera y el agua producto. En esta etapa final correspondiente al **Efecto f**,  $f$  puede ser igual a cero o a uno.
- Por otra parte, el **subsistema B**, está compuesto por un tanque para agua salada (**TAM**), cuatro válvulas de paso (**VP<sub>1</sub>**, **VP<sub>2</sub>**, **VP<sub>3</sub>** y **VP<sub>4</sub>**), dos bombas para agua salada (**BAM<sub>1</sub>**) y (**BAM<sub>2</sub>**), un divisor para agua salada (**DIVAM**) y un mezclador para agua salada (**MIXAM**). Además, este **subsistema B**, incluye un intercambiador de calor que actúa como evaporador (**EVA**), interconectado con una válvula de tres vías (**V3V<sub>1</sub>**) para el agua producto y un mezclador para vapor de agua (**MIXVE**), el cual se conecta a un eyector-condensador (**EYC**) que utiliza una corriente líquida como fluido primario y vapor de agua como fluido secundario. Este **subsistema B**, también cuenta con una bomba para agua producto (**BAP**), un intercambiador de calor (**HX<sub>sw</sub>**) que actúa como disipador de calor, cuatro válvulas de expansión (**VP<sub>5</sub>**, **VP<sub>6</sub>**, **VP<sub>8</sub>** y **VP<sub>9</sub>**), un tanque para agua producto (**TAP**) y dos columnas barométricas (**CB<sub>1</sub>**) y (**CB<sub>3</sub>**) diseñadas para la extracción del agua producto. Además, se

compone por diversos elementos, entre los que se incluyen: una válvula de tres vías para salmuera (**V3V2**), un intercambiador de calor (**HXBR**) que actúa como disipador de calor y dos válvulas de paso (**VP10**) y (**VP11**). Además, cuenta con un separador líquido-vapor para salmuera (**SEPBR**) y un tanque para el almacenamiento de salmuera (**TS**), el cual está  
5 interconectado a una columna barométrica (**CB2**) para la extracción de la salmuera.

En el **subproceso A (desalinización por evaporación de múltiple efecto al alto vacío y baja TBT)**, se utiliza agua de mar o agua salobre, siendo preferible el uso de agua de mar con una salinidad típica de 35,000 ppm. En el marco de esta invención, el término de “agua salada” abarca tanto el agua de mar o agua salobre, según corresponda. En el **subproceso A**,  
10 de acuerdo con las figuras 1 y 2, la corriente de agua salada (**1**) proveniente del mezclador para agua salada (**MIXAM**) del **subproceso B**, se dirige directamente hacia la etapa inicial (**Efecto 0**), ya que el proceso no permite utilizar precalentadores para el agua salada cuando la TBT sea igual o menor a la temperatura de alimentación del agua salada, dado que no se tiene gradiente de temperatura entre el vapor generado en los efectos y el agua salada. El  
15 **subproceso A**, modifica la temperatura máxima de salmuera (TBT), reduciéndola desde sus valores convencionales (65-95 °C) a temperaturas inferiores. Dichas temperaturas pueden corresponder a una TBT igual a la temperatura de alimentación del agua salada (por ejemplo, un valor normal y no limitativo es 25 °C) o hasta valores cercanos al punto de congelamiento de la salmuera, donde la TBT pudiera coincidir con la BBT. Un valor normal y no limitativo  
20 es una BBT de 4 °C. La corriente de agua salada (**1**), ingresa al sistema a través de una válvula de expansión (**VE1**), reduciendo súbitamente su presión hasta la presión de saturación correspondiente al **Efecto 0**. En estas condiciones de presión y temperatura, la corriente de agua salada (**2**), ingresa al generador (**GEN**), donde es calentada por un fluido caloportador (**4**), el cual puede ser cualquier fluido con la capacidad térmica suficiente para generar vapor  
25 en el generador (**GEN**) y además puede provenir de cualquier tipo de proceso o tecnología, preferiblemente proveniente de fuentes termosolares como: Concentradores Parabólicos Compuestos, Concentradores solares tipo Cilindro Parabólico, Fresnel, entre otros. La corriente de fluido caloportador (**5**) sale del generador (**GEN**) después de haber calentado a la corriente de agua salada (**2**). La corriente de agua salada (**3**), sale del generador (**GEN**)  
30 como una corriente bifásica e ingresa al separador líquido-vapor (**SEP0**), dividiéndose en las

corrientes de salmuera **(11)** y de vapor de agua **(6)**. La corriente de salmuera **(11)** es dirigida al evaporador del segundo efecto de la MED **(HX1)** a través de una tubería en forma de “U” **(TU1)**, que asegura un sello hidráulico para mantener la diferencia de presión entre el **Efecto 0** y **Efecto 1**. Durante este proceso, la corriente de salmuera **(11)** se flashea al pasar por la tubería **(TU1)**, reduciendo su presión y temperatura de saturación, correspondiente a la presión del **Efecto 0**, hasta a la presión del **Efecto 1**, lo que da lugar a la corriente de salmuera en dos fases **(12)**, la cual sale de la tubería en forma de “U” **(TU1)**. La corriente **(12)** ingresa al intercambiador de calor del **Efecto a (HX1)**, donde recibe todo el calor latente de la corriente de vapor **(6)**. A su vez, la corriente de vapor de agua **(6)** ingresa al intercambiador del **Efecto 1 (HX1)** y calienta a la corriente de salmuera en dos fases **(12)** que sale como la corriente de salmuera **(13)**. Al ceder toda su energía latente, la corriente **(6)** se condensa totalmente y sale como agua producto en la corriente **(7)**. La corriente de agua producto **(7)** pasa a través de la válvula de expansión **(VE1AP)**, quedando en dos fases (**corriente 8**) con la misma presión que el separador líquido-vapor **(SEP1AP)**. En el separador **(SEP1AP)**, se separa la fase de vapor **(10)** y la fase líquida **(9)**. La corriente de vapor **(10)** es dirigida al mezclador **(MIX1)** para mezclarse con la corriente de vapor **(14)**, mientras que la corriente en fase líquida **(9)** se dirige al mezclador **(MIX2AP)** para mezclarse con la corriente de agua producto **(19)** y dirigirse como la corriente **(20)** hacia la válvula de expansión **(VE2AP)**, donde cae su presión hasta la presión del efecto posterior y sale como la corriente **(21)**. Simultáneamente, la corriente de salmuera **(13)**, ingresa al separador líquido-vapor **(SEP1)**, en donde se divide en las corrientes de vapor **(14)** y de salmuera **(15)**. La corriente de vapor **(14)**, se mezcla con la corriente de vapor generado por flasheo del agua producto **(10)**, proveniente del separador líquido-vapor **(SEP1AP)**, formando la corriente de vapor **(18)**, que es dirigida al intercambiador de calor del **Efecto 2 (HX2)**. La corriente de salmuera **(15)**, hace lo propio, al dirigirse a través de la tubería en forma de “U” **(TU2)**, en donde reduce su presión, saliendo como la corriente de salmuera **(16)** y luego pasa hacia el intercambiador de calor **(HX2)**, donde termina con la presión del **Efecto 2** y sale como la corriente de salmuera en dos fases **(17)**. Lo ocurrido en el **Efecto 1**, se repite en los efectos subsiguientes hasta el **Efecto n**, presentando una disminución progresiva de presión y temperatura en efecto cascada. En el **Efecto n**, sucede el mismo proceso, la corriente de salmuera en dos fases **(25)**, ingresa al separador para salmuera **(SEPn)** de donde salen las corrientes de vapor **(26)** y de



salmuera (27). La corriente de vapor (26), se mezcla en el mezclador **MIXn** con la corriente de vapor generado por flasheo del agua producto (23), proveniente del separador líquido-vapor (**SEPn<sub>AP</sub>**), formando la corriente de vapor (30), que es dirigida al intercambiador de calor del **Efecto f (HXf)**. La corriente de salmuera (27), hace lo propio, al dirigirse a través

5 de la tubería en forma de “U” (**TUf**), en donde reduce su presión, saliendo como la corriente de salmuera (28) y luego pasa hacia el intercambiador de calor (**HXf**), donde termina con la presión del **Efecto f** y sale como la corriente de salmuera en dos fases (29). La corriente de salmuera en dos fases (29), se dirige al separador para salmuera (**SEPf**) de donde salen las corrientes de vapor (41) y de salmuera (43). Por otra parte, el separador líquido-vapor

10 (**SEPn<sub>AP</sub>**) recibe la corriente de agua (22) y la divide en las corrientes de vapor (23) y de salmuera (24). La corriente en fase líquida (24) se dirige al mezclador (**MIXf<sub>AP</sub>**) para mezclarse con la corriente de agua producto (31) y dirigirse como la corriente (32) hacia la válvula de expansión (**VEf<sub>AP</sub>**), donde cae su presión hasta la presión del efecto posterior y sale como la corriente (33). La corriente de agua (33) ingresa al separador líquido-vapor

15 (**SEPf<sub>AP</sub>**) y se divide en las corrientes (34) y (35).

En el último efecto (**Efecto f**), se lleva a cabo el mismo proceso que en los efectos anteriores, con la diferencia de que sus corrientes de vapor (42), de salmuera (43) y de agua producto (35) se dirigen al **subproceso B**. La corriente de vapor (42) se forma de la mezcla de las corrientes de vapor (34) y (41) en el mezclador (**MIXf**). Por otra parte, el agua producto de

20 cada efecto, se combina con la del efecto posterior a través de válvulas de expansión para agua producto y separadores para agua producto, acumulándose en la corriente final de agua producto (35), que se dirige al **subproceso B**, para su posterior extracción y/o utilización como refrigerante en el evaporador (**EVA**).

**El subproceso B (alimentación de agua salada, climatización y adecuación del agua**

25 **producto mediante agua salada y salmuera fría)**, se compone de tres etapas: la etapa de **alimentación de agua salada**, la etapa de **producción de frío** y la etapa de **eyecto-condensación barométrica**. De acuerdo con las figuras 1 y 3, la etapa de **alimentación de agua salada**, presentan dos modalidades diferentes para realizar la alimentación del agua salada al proceso: 1) cuando el agua salada se precalienta en el intercambiador de calor para

30 agua salada (**HXsw**) y 2) cuando el agua salada se dirige directamente al efecto inicial (**Efecto**

**0)** del **subproceso A**. En la primera modalidad de alimentación del agua salada (cuando el agua salada se precalienta en el intercambiador de calor para agua salada (**HX<sub>sw</sub>**)), la corriente de agua salada (**1a**) que sale del tanque para agua salada (**TAM**), pasa a través de la válvula de paso (**VP1**) y es succionada por la bomba para agua salada (**BAM1**) para ser dirigida como la corriente (**2a**) hacia el intercambiador de calor para agua salada (**HX<sub>sw</sub>**). Una vez que la corriente de agua salada (**2a**), que actúa como la corriente fría en el intercambiador de calor (**HX<sub>sw</sub>**), recibe el calor sensible de la corriente de agua producto (**55**) proveniente de la válvula de paso (**VP5**), sale como la corriente de agua salada (**3a**) que se dirige a un divisor de flujo (**DIVAM**), del cual salen las corrientes de agua salada (**4a**) y (**5a**). La corriente (**4a**), que contiene solo el flujo másico necesario para activar el sistema MED del **subproceso A**, pasa a través de la válvula de paso (**VP2**) para ingresar al mezclador (**MIXAM**) de agua salada, y convertirse en la corriente de agua salada (**1**), que posteriormente se dirige al efecto inicial (**Efecto 0**) del **subproceso A**, mientras que la corriente de agua salada (**5a**) que conlleva el flujo másico necesario para enfriar a la corriente (**55**) de agua producto, se redirige al tanque de salmuera (**TS**) para ser utilizada en la dilución de la corriente de salmuera (**50**). En esta modalidad las válvulas de paso (**VP3**), (**VP4**), (**VP6**), (**VP7**), (**VP10**) y (**VP11**) deben permanecer cerradas, mientras que las válvulas de paso (**VP1**), (**VP2**), (**VP5**), (**VP8**) y (**VP9**), deben estar abiertas. En la segunda modalidad, cuando el agua salada se dirige directamente al **subproceso A**, las válvulas de paso (**VP1**), (**VP2**), (**VP5**), (**VP8**) y (**VP9**), deben de permanecer cerradas, mientras que las válvulas de paso (**VP3**), (**VP4**), (**VP6**), (**VP7**), (**VP10**) y (**VP11**) deben estar abiertas, para permitir que la corriente de agua salada (**1b**) que proviene del tanque para agua salada (**TAM**), pase a través de la válvula de paso (**VP3**) y sea succionada por la bomba para agua salada (**BAM2**) y posteriormente salga como la corriente (**2b**) para pasar por la válvula de paso (**VP4**) antes de ingresar al mezclador (**MIXAM**) para agua salada, de donde sale como la corriente de agua salada (**1**) que se dirige al efecto inicial (**Efecto 0**) del **subproceso A**.

En el **subproceso B**, también se lleva a cabo la etapa de **producción de frío**, en la cual la corriente de agua producto (**35**) proveniente del **Efecto f** del **subproceso A**, es dirigida hacia una válvula de tres vías (**V3V1**), que separa dicha corriente en dos: la corrientes de agua producto (**36**) y la corriente de agua producto (**37**). La corriente de agua producto (**37**) es

conducida a un intercambiador de calor que opera como evaporador (EVA) para la generación de frío. Dentro del evaporador (EVA), la corriente de agua producto (37) recibe el calor de la corriente (39), transformándose en la corriente de vapor (38), la cual es posteriormente dirigida hacia el mezclador de vapor (MIXVE). Mientras que la corriente (39) sale del evaporador (EVA) a menor temperatura como la corriente (40). En el mezclador (MIXVE), la corriente de vapor (38), se mezcla con la corriente de vapor (42) proveniente del Efecto f y con la corriente de vapor (47) proveniente del separador (SEPBR), convirtiéndose en la corriente de vapor (52). La corriente de vapor (47) solo se genera y se mezcla con las corrientes (38) y (42), en los casos donde se utiliza la salmuera fría para el enfriamiento del eyector-condensador (EYC). Por otra parte, la corriente de agua producto (36) se dirige a la columna barométrica (CB1), que tiene como función principal crear un sello hidráulico y mantener el vacío dentro del sistema, saliendo por la base de la columna barométrica (CB1) en condiciones de presión atmosférica y posteriormente depositarse en el tanque para agua producto (TAP). La válvula de tres vías (V3V1) desempeña un papel clave en el control de la capacidad de enfriamiento, regulando el flujo másico de la corriente refrigerante (37) que ingresa al evaporador (EVA). En caso de que el flujo másico de la corriente de agua producto (37) sea nulo, el **proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, queda limitado únicamente a la desalinización, sin proveer el servicio de generación de frío o calefacción para el acondicionamiento térmico de espacios.

En la etapa de **eyector-condensación barométrica**, que forma parte del **subproceso B**, se destaca como elemento clave el eyector-condensador (EYC), el cual es un tipo especial de eyector que involucra la condensación de un vapor, en donde la gran diferencia de velocidad del fluido motriz (**corriente 61**) con respecto a la corriente de succión (**corriente 52**) y la íntima interacción entre las mismas, produce una significativa transferencia de cantidad de movimiento, masa y calor, lo cual provoca la condensación de la fase vapor, resultando una corriente en fase líquida a la salida del eyector-condensador. En esta etapa, la corriente de vapor de agua (52), caracterizada por su baja presión y temperatura, es succionada, arrastrada y comprimida para provocar su condensación en el eyector-condensador (EYC). Este dispositivo emplea como fluido motriz la corriente de agua producto (61), que debe tener una

presión superior tanto a la de la corriente de vapor de baja presión **(52)** como a la presión intermedia de operación. La presión del vapor de baja presión **(52)** puede ser ajustada para garantizar un adecuado nivel de enfriamiento en el evaporador **(EVA)**, siendo una condición preferente operar a aproximadamente 0.8 kPa para sistemas de aire acondicionado. Al

5 ingresar al eyector-condensador **(EYC)**, las corrientes **(52)** y **(61)** se mezclan y salen en forma líquida a cualquier presión intermedia. Esta presión se encuentra por debajo de la presión del fluido motriz (**corriente de agua producto 61**), pero es mayor a la del fluido secundario (**corriente de vapor de baja presión 52**), asegurando así la condensación de la corriente de vapor de baja presión **(52)**. La condensación total del vapor (**corriente de vapor**

10 **de baja presión 52**) se presenta a la salida del eyector-condensador **(EYC)**, donde la velocidad de la corriente mezclada de descarga **(62)** disminuye al incrementarse el área transversal, y a su vez este cambio provoca un aumento abrupto de presión, completando la condensación del vapor de baja presión **(52)**. Posteriormente, la corriente mezclada y condensada **(62)** que sale del eyector-condensador **(EYC)**, entra en la columna barométrica

15 **(CB3)**, cuya función es crear un sello hidráulico y mantener el vacío en el sistema. Desde el fondo de la columna barométrica **(CB3)**, el fluido emerge a presión atmosférica como agua producto **(63)** y se almacena en el tanque para agua producto **(TAP)**. Desde el tanque **(TAP)**, salen las corrientes **(53)** y **(64)**, que se dirigen a otras etapas del sistema o procesos relacionados.

20 El enfriamiento del fluido motriz (**corriente de agua producto 61**) en la etapa de **eyecto-condensación barométrica**, puede llevarse a cabo mediante diferentes configuraciones, dependiendo de los requerimientos del sistema. Las posibles configuraciones son: 1) **enfriado únicamente mediante agua salada**, 2) **enfriado mediante agua salada y salmuera fría**, y 3) **enfriado únicamente mediante salmuera fría**. En la primera

25 configuración, donde el eyector-condensador **(EYC)** es **enfriado únicamente mediante agua salada** para disipar el calor de condensación, las válvulas de paso **(VP1)**, **(VP2)**, **(VP5)**, **(VP8)** y **(VP9)** permanecen abiertas, mientras que las válvulas de paso **(VP3)**, **(VP4)**, **(VP6)**, **(VP7)**, **(VP10)** y **(VP11)** permanecerán cerradas. En esta configuración, la válvula de tres vías **(V3V2)**, bloquea el flujo de la corriente **(44)** y redirige la totalidad de la corriente hacia

30 la salida **(48)**. El agua producto **(53)** es extraída del tanque para agua producto **(TAP)**

- mediante la bomba (**BAP**), de la cual sale como la corriente (**54**) pasando por la válvula de paso (**VP5**) hacia el intercambiador de calor (**HX<sub>sw</sub>**) como la corriente (**55**). Posteriormente, esta corriente que sale del intercambiador de calor (**HX<sub>sw</sub>**) como la corriente (**57**), atraviesa la válvula (**VP8**) e ingresa a la válvula de paso (**VP9**) como la corriente (**58**), y finalmente
- 5 ingresa al eyector-condensador (**EYC**) como corriente motriz (**corriente de agua producto 61**). En la configuración híbrida, donde el eyector-condensador (**EYC**) es **enfriado mediante agua salada y salmuera fría**, la válvula de tres vías (**V3V2**), permite pasar de forma parcial o total el flujo de la corriente de salmuera (**44**), según el nivel de enfriamiento requerido en el eyector-condensador (**EYC**). En este caso, las válvulas de paso (**VP1**), (**VP2**), (**VP5**),
- 10 (**VP10**) y (**VP11**) permanecerán abiertas, mientras que las válvulas de paso (**VP3**), (**VP4**), (**VP6**), (**VP7**), (**VP8**) y (**VP9**) permanecerán cerradas. La corriente de agua producto (**54**), impulsada por la bomba (**BAP**), pasa a través de la válvula de paso (**VP5**), intercambia calor con la corriente de agua salada (**2a**) en el intercambiador de calor (**HX<sub>sw</sub>**), pasa a través de la válvula de paso (**VP10**) y continúa como la corriente (**59**) hacia el intercambiador de calor
- 15 (**HX<sub>BR</sub>**), donde se enfría aún más al transferir calor a la corriente de salmuera (**44**) la cual se encuentra en su punto de saturación. Por último, la corriente de fluido motriz (**corriente de agua producto 61**) ingresa al eyector-condensador (**EYC**) a una temperatura menor que en la primera configuración, en la que el eyector-condensador (**EYC**) es **enfriado únicamente mediante agua salada**.
- 20 Finalmente, cuando se opta por enfriar al eyector-condensador (**EYC**) **únicamente mediante salmuera fría**, se utiliza la corriente de salmuera (**43**), que proviene del último efecto de la MED (**Efecto f**) y se encuentra en estado de saturación y baja temperatura. La válvula de tres vías (**V3V2**) cierra el paso hacia la corriente (**48**) y dirige todo el flujo hacia la corriente (**44**). En esta configuración, las válvulas de paso (**VP1**), (**VP2**), (**VP5**), (**VP8**) y (**VP9**)
- 25 permanecerán cerradas, mientras que las válvulas de paso (**VP3**), (**VP4**), (**VP6**), (**VP7**), (**VP10**) y (**VP11**) permanecen abiertas. Así, la corriente de agua producto (**53**) que proviene del tanque para agua producto (**TAP**), es succionada por la bomba (**BAP**) y sale como la corriente (**54**), para pasar por las válvulas de paso (**VP6**), (**VP7**) y (**VP10**) saliendo como las corrientes (**56**), (**57**) y (**59**), respectivamente. La corriente (**59**) es enfriada en el
- 30 intercambiador de calor (**HX<sub>BR</sub>**) mediante el intercambio térmico con la corriente de salmuera

(44), saliendo a menor temperatura como la corriente (60). En las dos últimas configuraciones de enfriamiento del eyector-condensador (EYC) mencionadas, se utiliza la corriente de salmuera (44) de manera total o parcial como fluido de enfriamiento, saliendo del intercambiador de calor (HXBR) como una corriente de salmuera en dos fases (45), que se dirige al separador líquido-vapor (SEPBR). Este separador divide la corriente de salmuera (45) en las corrientes de salmuera líquida (46) y la corriente de vapor de agua (47). La corriente de salmuera (46) se une a través de una tubería con la corriente de salmuera (48) para formar la corriente de salmuera (49), que se dirige hacia la columna barométrica (CB2), donde se asegura el sello hidráulico y el mantenimiento del vacío en el sistema.

Posteriormente, la corriente líquida es extraída del fondo de la columna barométrica (CB2) a presión atmosférica como la corriente (50) y se deposita en el tanque para salmuera (TS) de donde sale como la corriente de salmuera (51). Por otro lado, la corriente de vapor de agua (47) es conducida al mezclador (MIXVE), donde se combina con los vapores (38) y (42), generando la corriente de vapor (52), que es succionada por el eyector-condensador (EYC) como fluido secundario (**corriente de vapor de baja presión 52**).

El **subproceso B** previamente mencionado puede incorporarse a cualquier proceso de evaporación al vacío como el de múltiple efecto, múltiple etapa flash, de destilación por membranas, entre otros métodos de desalación que operen bajo condiciones de vacío. Esto se debe a que, en todos estos casos, el agua producto generada puede emplearse como fluido refrigerante.

El **sistema y proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, presenta dos modos de operación; 1) producción exclusiva de agua desalinizada y 2) producción simultánea de frío y agua desalinizada. En el primer caso, cuando únicamente se genera agua desalinizada, la válvula de tres vías (V3V1) se configura para cerrar el flujo hacia la corriente (37), evitando así el ingreso al evaporador (EVA) y dirigiendo la totalidad de la corriente (35) hacia la columna barométrica (CB1) como la corriente (36). En esta modalidad, el evaporador (EVA) permanece inactivo, lo que implica que no se genera la corriente de vapor (38). Por lo tanto, el eyector-condensador (EYC) únicamente recibe y condensa la corriente de vapor (42) y, en el caso que aplique, la corriente de vapor (47). En el segundo caso, cuando el sistema opera

para producir simultáneamente frío y agua desalinizada, la válvula de tres vías (**V3V1**) divide la corriente total de agua producto proveniente de los efectos (**35**) en dos flujos: las corrientes (**36**) y (**37**). La corriente (**36**) es dirigida directamente a la columna barométrica (**CB1**), desde donde se extrae como agua desalinizada. Mientras tanto, la corriente (**37**) se utiliza como refrigerante en el evaporador (**EVA**) para la generación de frío. La capacidad de enfriamiento del sistema se ajusta regulando el flujo de la corriente de agua producto (**37**) que entra al evaporador (**EVA**). Este ajuste también determina el volumen de vapor (**38**) generado, el cual se dirige al mezclador (**MIXVE**) y posteriormente al eyector-condensador (**EYC**).

La implementación del eyector-condensador (**EYC**), permite condensar el vapor de baja temperatura (**corriente 52**) mediante un efecto de incremento de presión, ya que logra alcanzar un nivel de vacío superior al convencional en el proceso MED, lo que conlleva a que la BBT descienda de sus valores convencionales (35-40 °C) hasta valores que pueden oscilar entre los 2 y 6 °C. Además, la disminución de la BBT permite modificar y reducir la TBT, pasando de los valores convencionales (65-95 °C) hasta temperaturas más bajas (25 °C, por ejemplo), incluso operando cerca de los valores de la BBT (2-6 °C), lo que representa un avance significativo en la eficiencia y viabilidad de los sistemas MED.

Por último, es importante destacar que el **sistema y proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, presenta variantes en su configuración. Tal como se describió en el **subproceso A**, la etapa intermedia, que comprende desde el **Efecto 1** hasta el **Efecto n**, puede ajustarse de manera que n varíe entre cero y hasta cualquier número como lo muestra la figura 1, sin embargo, el **subproceso A** y **subsistema A**, cuentan con la variante de solo estar conformado por la etapa inicial (**Efecto 0**) y la etapa final (**Efecto f**), según se muestra en la figura 4. En esta configuración, donde el número de efectos totales del **subsistema A** y **subproceso A** se reduce a dos (**Efecto 0** y **Efecto f**), el separador líquido-vapor para salmuera (**SEP0**) se conecta directamente con la tubería en forma de “U” (**TUf**) y con el intercambiador de calor (**HXf**) que pertenecen al **Efecto f**, a través de las corrientes de salmuera (**11**) y vapor de agua (**6**), respectivamente. El resto del proceso que se lleva a cabo en la etapa final (**Efecto f**) se desarrolla de manera idéntica a lo descrito previamente.

- Otra variante, según la figura 5, se presenta cuando el **subsistema A** y **subproceso A** solo cuenten con la etapa inicial que corresponde al **Efecto 0**. En esta configuración donde el **subsistema A** y **subproceso A** operan con un único efecto de evaporación (**Efecto 0**), la corriente de agua salada (**3**) proveniente del generador (**GEN**), ingresa al separador líquido-vapor para salmuera (**SEP0**) y se divide en las corrientes de vapor de agua (**42**) y de salmuera (**43**). La corriente de salmuera (**43**) se dirige a la válvula de tres vías (**V3V2**) del **subsistema B** y **subproceso B**, mientras que la corriente de vapor de agua (**42**) se redirige al mezclador (**MIXVE**) que también pertenece al **subsistema B** y **subproceso B**. Se destaca que en el **subsistema B** y **subproceso B** se introduce un nuevo componente: un elemento estrangulador, que puede ser una válvula de expansión (**VR**), correspondiente a la etapa de **producción de frío**. En este caso, la corriente de agua producto (**64**) que sale del tanque para agua producto (**TAP**), pasa a través de la válvula de expansión (**VR**) antes de ingresar al evaporador (**EVA**). El resto del proceso en el **subproceso B**, se desarrolla como se describió previamente.
- 15 Otra variante, según la figura 6, se presenta cuando el **subsistema A** y **subproceso A** se desarrolla exactamente igual que en la variante presenta en la figura 5, sin embargo, en el **subsistema B** y **subproceso B** se sustituye el elemento estrangulador (**VR**) para introducir un expansor generador (**EXP**) para la generación de energía eléctrica, aprovechando la notable diferencia de presión entre las corrientes (**64**) y la corriente (**37**).
- 20 Una variante adicional, mostrada en la figura 7, es similar a la presentada en la figura 5, con la diferencia de que se prescinde del evaporador (**EVA**) y la válvula de expansión (**VR**), correspondientes al **subproceso B**. Esto implica que el **subproceso B** del sistema y proceso **con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, no incluye la etapa de **producción de frío**, proporcionando únicamente el servicio de desalinización.
- 25

Finalmente, la última variante, mostrada en la figura 8, se presenta cuando el **subsistema A** y **subproceso A** operan con una TBT que sea menor a los valores convencionales pero mayor a la temperatura del agua salada de alimentación, por ejemplo, un rango normal y no limitativo es 35-65 °C. Lo anterior, implica que el primer efecto del sistema y proceso se



- activa a una temperatura mayor a la del agua de alimentación, por lo cual, en este escenario, es conveniente la utilización de precalentadores (PC1, PC0... PCn). Lo anterior, para lograr la mayor integración energética interna posible y reducir la adición de calor de calor de forma sensible o bien, reducir la cantidad de calor suministrada en el GEN. En esta variante, la
- 5 cantidad de precalentadores necesarios, está directamente relacionado con la TBT establecida y el gradiente de temperatura entre el vapor generado en los efectos y el agua salada de alimentación. A mayor TBT, mayor será el número de precalentadores a utilizar.

### EJEMPLO PROFÉTICO DE OPERACIÓN

- Como ejemplo ilustrativo y no limitativo del **sistema y proceso con con baja temperatura**
- 10 **de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, se presenta un estudio paramétrico realizado mediante simulación computacional utilizando el software Aspen Plus™ versión 10, con el objetivo de evaluar la factibilidad técnica.

- Aspen Plus™ es una herramienta ampliamente utilizada y validada en la industria para el
- 15 diseño y análisis de sistemas termoquímicos, ya que permite calcular con precisión propiedades termodinámicas de mezclas y fluidos puros, así como representar con fidelidad diversos equipos de proceso como generadores, eyectores e intercambiadores de calor que operan como evaporadores, y condensadores.

- La simulación se llevó a cabo considerando una cascada descendente de temperatura y
- 20 presión a lo largo de los efectos, con un rango de de 25 a 4 °C y 3.15 a 0.8 kPa, respectivamente, La corriente de alimentación corresponde a agua salada con una concentración salina de 35,000 ppm y una temperatura de ingreso de 25 °C. En todos los intercambiadores de calor del sistema y proceso se consideró un valor mínimo de aproximación térmica ( $\Delta T_{mín}$ ) de 2.5 °C entre las corrientes calientes y frías.

- 25 Durante el estudio paramétrico, se evaluaron distintas combinaciones de flujo másico de alimentación y cantidad de calor suministrado al generador, observándose el comportamiento del sistema en términos de producción de agua, recuperación de energía térmica y destrucción de exergía. Particularmente, se analizó el desempeño del eyector mediante un enfoque de

segunda ley de la termodinámica. Los resultados obtenidos, presetados en la Tabla 4, muestran valores positivos de destrucción de exergía, lo cual indica que el sistema opera dentro de los principios termodinámicos y bajo un régimen energético eficiente y consistente. En la Tabla 1, se presentan las principales condiciones de operación del ejemplo aplicativo

5 de la invención. En la Tabla 2, se presentan las principales características termodinámicas y operativas de la invención, incluyendo agua producto por efecto, concentración de salmuera y temperatura y presión de cada efecto. En la Tabla 3, se presenta la evaluación energética del ejemplo aplicativo.

Tabla 1. Condiciones de operación del proceso de la invención, para un ejemplo aplicativo.

| Corrientes | Temperatura<br>(°C) | Presión<br>(kPa) | Flujo másico<br>(kg/s) | Concentración<br>(ppm) | Entalpía<br>(kJ/kg) | Fluido        |
|------------|---------------------|------------------|------------------------|------------------------|---------------------|---------------|
| <b>1b</b>  | 25.00               | 101.32           | 5.00                   | 35,000.00              | -15,539.03          | Agua salada   |
| <b>2b</b>  | 25.00               | 200.00           | 5.00                   | 35,000.00              | -15,538.91          | Agua salada   |
| <b>2</b>   | 25.00               | 3.15             | 5.00                   | 35,000.00              | -15,538.95          | Agua salada   |
| <b>3</b>   | 25.00               | 3.15             | 5.00                   | 35,000.00              | -15,533.95          | Agua salada   |
| <b>4</b>   | 45.00               | 200.00           | 0.39                   | 0.00                   | -15792.24           | Agua          |
| <b>5</b>   | 30.00               | 200.00           | 0.39                   | 0.00                   | -15,854.94          | Agua          |
| <b>6</b>   | 25.00               | 3.15             | 0.01                   | 0.00                   | -13,433.02          | Vapor de agua |
| <b>11</b>  | 25.00               | 3.15             | 4.99                   | 35,074.30              | -15,538.41          | Agua salada   |
| <b>12</b>  | 22.66               | 2.73             | 4.99                   | 35,074.30              | -15,538.41          | Agua salada   |
| <b>35</b>  | 4.00                | 0.81             | 0.78                   | 0.00                   | -15,965.03          | Agua          |
| <b>38</b>  | 4.00                | 0.81             | 0.78                   | 0.00                   | -13,473.03          | Vapor de agua |
| <b>39</b>  | 12.00               | 200.00           | 91.99                  | 0.00                   | -15,929.99          | Agua          |
| <b>40</b>  | 7.00                | 200.00           | 91.99                  | 0.00                   | -15,951.23          | Agua          |
| <b>42</b>  | 4.00                | 0.81             | 4.03                   | 43,413.20              | -15,544.37          | Agua salada   |
| <b>43</b>  | 4.00                | 0.81             | 0.18                   | 0.00                   | -13,472.09          | Vapor de agua |
| <b>51</b>  | 4.00                | 0.81             | 4.03                   | 43,413.20              | -15,544.37          | Agua salada   |
| <b>52</b>  | 4.00                | 0.81             | 0.96                   | 0.00                   | -13,472.90          | Vapor de agua |

|           |       |        |        |      |            |      |
|-----------|-------|--------|--------|------|------------|------|
| <b>53</b> | 29.59 | 101.32 | 558.40 | 0.00 | -15,856.52 | Agua |
| <b>54</b> | 29.59 | 500.00 | 558.40 | 0.00 | -15,856.28 | Agua |
| <b>57</b> | 28.50 | 500.00 | 558.40 | 0.00 | -15,860.74 | Agua |
| <b>61</b> | 28.50 | 500.00 | 558.40 | 0.00 | -15,860.74 | Agua |
| <b>62</b> | 29.59 | 4.15   | 559.37 | 0.00 | -15,856.80 | Agua |
| <b>63</b> | 29.59 | 101.32 | 559.37 | 0.00 | -15,856.52 | Agua |
| <b>64</b> | 29.59 | 101.32 | 0.9696 | 0.00 | -15,856.52 | Agua |

Tabla 2. Características termodinámicas y operativas de la invención.

| <b>Efecto</b> | <b>Temperatura<br/>(°C)</b> | <b>Presión<br/>(kPa)</b> | <b>Agua producto<br/>acumulada<br/>(kg/s)</b> | <b>Concentración<br/>de salmuera<br/>(ppm)</b> | <b>Q<sub>efectos</sub><br/>(kW)</b> | <b>Temperatura<br/>Agua salada<br/>(°C)</b> |
|---------------|-----------------------------|--------------------------|-----------------------------------------------|------------------------------------------------|-------------------------------------|---------------------------------------------|
| <b>1</b>      | 25.00                       | 3.15                     | -                                             | 35,074.30                                      | 25.00                               | 25.00                                       |
| <b>2</b>      | 22.66                       | 2.73                     | 0.0753                                        | 35,286.90                                      | 25.91                               | 25.00                                       |
| <b>3</b>      | 20.33                       | 2.37                     | 0.1472                                        | 35,641.20                                      | 73.71                               | 25.00                                       |
| <b>4</b>      | 18.00                       | 2.05                     | 0.2182                                        | 36,144.20                                      | 121.39                              | 25.00                                       |
| <b>5</b>      | 15.66                       | 1.76                     | 0.2882                                        | 36,806.80                                      | 168.96                              | 25.00                                       |
| <b>6</b>      | 13.33                       | 1.52                     | 0.3573                                        | 37,644.40                                      | 216.42                              | 25.00                                       |
| <b>7</b>      | 11.00                       | 1.30                     | 0.4255                                        | 38,679.80                                      | 263.79                              | 25.00                                       |
| <b>8</b>      | 8.66                        | 1.11                     | 0.4927                                        | 39,943.20                                      | 311.55                              | 25.00                                       |
| <b>9</b>      | 6.33                        | 0.94                     | 0.5591                                        | 41,473.40                                      | 359.79                              | 25.00                                       |
| <b>10</b>     | 4.0                         | 0.81                     | 0.7773                                        | 43,322.90                                      | 408.29                              | 25.00                                       |
| <b>EYC</b>    | 4.0                         | 0.81                     | 0.1833                                        | ---                                            | ---                                 | ---                                         |
| <b>Total</b>  | <b>0.9696</b>               |                          |                                               | <b>1,974.81</b>                                |                                     |                                             |

Tabla 3. Evaluación energética de la invención.

| COMPONENTES                           | BALANCE DE ENERGÍA                                                                     | ENTRADA ENERGÍA (kW) | PÉRDIDA ENERGÍA (kW)                         | INTEGRACIÓN ENERGÉTICA INTERNA (kW) |
|---------------------------------------|----------------------------------------------------------------------------------------|----------------------|----------------------------------------------|-------------------------------------|
| <b>GENERADOR</b>                      | $Q_{GEN} = m_4 h_4 - m_2 h_2$                                                          | 25.00                |                                              |                                     |
| <b>DISIPADOR (HX<sub>EJC</sub>)</b>   | $Q_{HXS_W} = m_{55} h_{55} - m_{2a} h_{2a}$                                            |                      | 2,598.63                                     |                                     |
| <b>EFFECTOS (HX)</b>                  | $\sum Q_{HXn} = m_n h_n - m_n h_n$                                                     |                      |                                              | 1,974.81                            |
| <b>BOMBA<sub>BAM2</sub></b>           | $W_{BAM2} = m_{1b} v_{1b} (P_{2b} - P_{1b}) / \eta_{BAM2}$                             | 0.61                 |                                              |                                     |
| <b>BOMBA<sub>BAP</sub></b>            | $W_{BAP} = m_{53} v_{53} (P_{54} - P_{53}) / \eta_{BAP}$                               | 260.71               |                                              |                                     |
| <b>BOMBA<sub>BAM1</sub></b>           | $W_{BAM1} = m_{1a} v_{1a} (P_{2a} - P_{1a}) / \eta_{BAM1}$                             | 36.70                |                                              |                                     |
| <b>CORRIENTES DE ENTRADA Y SALIDA</b> |                                                                                        |                      |                                              |                                     |
| $m_1 h_1$                             | Entrada del agua de mar                                                                | -15,538.90           |                                              |                                     |
| $m_{2a} h_{2a}$                       | Agua salada de enfriamiento                                                            | -15,538.90           |                                              |                                     |
| $m_{43} h_{43}$                       | Salida de la salmuera                                                                  |                      | -15,545.20                                   |                                     |
| $m_{35} h_{35}$                       | Salida del agua producto                                                               |                      | -15,534.52                                   |                                     |
| <b>TOTAL</b>                          |                                                                                        | <b>-31,077.80</b>    | <b>-31,079.72</b>                            | <b>1,974.81</b>                     |
| <b>AGUA PRODUCTO</b>                  |                                                                                        | 83,017.3 L/día       | <b>Producción diaria</b>                     |                                     |
| <b>RR</b>                             | $RR = m_{64} / m_1$                                                                    | 0.20                 | <b>Tasa de recuperación</b>                  |                                     |
| <b>GOR</b>                            | $GOR = \frac{(m_{64} * 2,326 \text{ kJ/kg})}{Q_{GEN}}$                                 | 89.37                | <b>Relación de ganancia de salida</b>        |                                     |
| <b>CB1</b>                            | $H_1 = (P_{atm} - P_{descarga}) / (\rho g)$                                            | 9.998 m              | <b>Columna Barométrica 1</b>                 |                                     |
| <b>CB2</b>                            | $H_2 = (P_{atm} - P_{descarga}) / (\rho g)$                                            | 9.966 m              | <b>Columna Barométrica 2</b>                 |                                     |
| <b>CB3</b>                            | $H_3 = (P_{atm} - P_{descarga}) / (\rho g)$                                            | 9.655 m              | <b>Columna Barométrica 3</b>                 |                                     |
| <b>GOR<sub>G</sub></b>                | $GOR_G = \frac{(m_{64} * 2,326 \text{ kJ/kg})}{Q_{GEN} + W_{BOMBAS\_TOTAL}}$           | 6.92                 | <b>Relación de ganancia de salida global</b> |                                     |
| <b>COP<sub>G</sub></b>                | $COP_G = \frac{Q_{EVA} + (m_{64} * 2,326 \text{ kJ/kg})}{Q_{GEN} + W_{BOMBAS\_TOTAL}}$ | 12.91                | <b>Coefficiente de operación global</b>      |                                     |

Tabla 4. Condiciones de operación en las que es posible la implementación del eyector.

| Fluido Motriz |            |              | Descarga EYC |            | Bombas    |           |          | Eficiencias          |                      | Eyector           | Posibilidad Termodinámica |
|---------------|------------|--------------|--------------|------------|-----------|-----------|----------|----------------------|----------------------|-------------------|---------------------------|
| Pres. (kPa)   | Temp. (°C) | Flujo (kg/s) | Pres. (kPa)  | Temp. (°C) | BAM1 (kW) | BAM2 (kW) | BAP (kW) | COP <sub>G</sub> (-) | GOR <sub>G</sub> (-) | Exergía destruida |                           |
| 300           | 28.5       | 534.7        | 4.1          | 29.6       | 0.61      | 35.3      | 125.5    | 12.1                 | 22.6                 | -55.5             | <i>Imposible</i>          |
|               | 28.5       | 444.9        | 4.2          | 29.8       | 0.61      | 31.7      | 104.4    | 13.9                 | 26.0                 | -82.6             | <i>Imposible</i>          |
|               | 28.5       | 268.6        | 4.4          | 30.6       | 0.61      | 22.9      | 63.1     | 20.2                 | 37.7                 | -137.5            | <i>Imposible</i>          |
|               | 28.5       | 194.4        | 4.6          | 31.4       | 0.61      | 18.1      | 45.7     | 25.2                 | 47.1                 | -162.2            | <i>Imposible</i>          |
|               | 28.5       | 159.4        | 4.8          | 32.0       | 0.61      | 15.5      | 37.4     | 28.7                 | 53.6                 | -174.8            | <i>Imposible</i>          |
|               | 28.5       | 130.7        | 5.0          | 32.8       | 0.61      | 13.2      | 30.7     | 32.4                 | 60.5                 | -186.0            | <i>Imposible</i>          |
| 500           | 28.5       | 558.4        | 4.1          | 29.6       | 0.61      | 37.0      | 263.0    | 6.9                  | 12.9                 | 62.1              | <i>Posible</i>            |
|               | 28.5       | 461.2        | 4.2          | 29.8       | 0.61      | 33.0      | 217.2    | 8.2                  | 15.3                 | 13.5              | <i>Posible</i>            |
|               | 28.5       | 274.5        | 4.4          | 30.6       | 0.61      | 23.4      | 129.3    | 12.6                 | 23.6                 | -81.5             | <i>Imposible</i>          |
|               | 28.5       | 197.5        | 4.6          | 31.4       | 0.61      | 18.4      | 93.0     | 16.4                 | 30.7                 | -122.3            | <i>Imposible</i>          |
|               | 28.5       | 161.9        | 4.8          | 32.0       | 0.61      | 15.7      | 76.3     | 19.2                 | 35.8                 | -142.1            | <i>Imposible</i>          |
|               | 28.5       | 132.8        | 5.0          | 32.7       | 0.61      | 13.4      | 62.6     | 22.2                 | 41.4                 | -159.2            | <i>Imposible</i>          |
| 700           | 28.5       | 584.3        | 4.1          | 29.6       | 0.61      | 38.9      | 413.2    | 4.7                  | 8.8                  | 190.5             | <i>Posible</i>            |
|               | 28.5       | 478.7        | 4.2          | 29.8       | 0.61      | 34.4      | 338.6    | 5.7                  | 10.6                 | 116.9             | <i>Posible</i>            |
|               | 28.5       | 280.6        | 4.4          | 30.6       | 0.61      | 24.0      | 198.5    | 9.1                  | 17.0                 | -23.0             | <i>Imposible</i>          |
|               | 28.5       | 200.6        | 4.6          | 31.4       | 0.61      | 18.7      | 142.0    | 12.1                 | 22.6                 | -81.1             | <i>Imposible</i>          |
|               | 28.5       | 164.5        | 4.8          | 32.0       | 0.61      | 15.9      | 116.4    | 14.3                 | 26.6                 | -108.3            | <i>Imposible</i>          |
|               | 28.5       | 134.9        | 5.0          | 32.7       | 0.61      | 13.6      | 95.5     | 16.7                 | 31.2                 | -131.5            | <i>Imposible</i>          |
| 900           | 28.5       | 612.7        | 4.1          | 29.6       | 0.61      | 41.0      | 578.0    | 3.5                  | 6.5                  | 331.4             | <i>Posible</i>            |
|               | 28.5       | 497.6        | 4.2          | 29.8       | 0.61      | 35.9      | 469.5    | 4.2                  | 7.9                  | 228.4             | <i>Posible</i>            |
|               | 28.5       | 287.0        | 4.4          | 30.6       | 0.61      | 24.6      | 270.9    | 7.0                  | 13.1                 | 38.2              | <i>Posible</i>            |
|               | 28.5       | 203.9        | 4.6          | 31.4       | 0.61      | 19.0      | 192.5    | 9.5                  | 17.7                 | -38.6             | <i>Imposible</i>          |
|               | 28.5       | 167.2        | 4.8          | 32.0       | 0.61      | 16.2      | 157.8    | 11.3                 | 21.1                 | -73.5             | <i>Imposible</i>          |
|               | 28.5       | 137.1        | 5.0          | 32.7       | 0.61      | 13.8      | 129.5    | 13.3                 | 24.9                 | -102.9            | <i>Imposible</i>          |
| 1100          | 28.5       | 643.9        | 4.1          | 29.6       | 0.61      | 43.4      | 759.7    | 2.7                  | 5.1                  | 486.7             | <i>Posible</i>            |
|               | 28.5       | 518.0        | 4.2          | 29.8       | 0.61      | 37.5      | 611.2    | 3.3                  | 6.2                  | 349.1             | <i>Posible</i>            |
|               | 28.5       | 293.7        | 4.4          | 30.6       | 0.61      | 25.2      | 346.6    | 5.7                  | 10.6                 | 102.2             | <i>Posible</i>            |
|               | 28.5       | 207.2        | 4.6          | 31.4       | 0.61      | 19.4      | 244.6    | 7.8                  | 14.5                 | 5.3               | <i>Posible</i>            |
|               | 28.5       | 169.9        | 4.8          | 32.0       | 0.61      | 16.5      | 200.6    | 9.3                  | 17.3                 | -37.5             | <i>Imposible</i>          |
|               | 28.5       | 139.3        | 5.0          | 32.7       | 0.61      | 14.0      | 164.5    | 11.0                 | 20.6                 | -73.3             | <i>Imposible</i>          |
| 1300          | 28.5       | 678.6        | 4.1          | 29.6       | 0.61      | 45.9      | 960.9    | 2.2                  | 4.1                  | 658.7             | <i>Posible</i>            |
|               | 28.5       | 540.2        | 4.2          | 29.8       | 0.61      | 39.3      | 765.0    | 2.7                  | 5.1                  | 480.1             | <i>Posible</i>            |
|               | 28.5       | 300.7        | 4.4          | 30.6       | 0.61      | 25.8      | 426.0    | 4.7                  | 8.8                  | 169.3             | <i>Posible</i>            |
|               | 28.5       | 210.7        | 4.6          | 31.4       | 0.61      | 19.7      | 298.5    | 6.6                  | 12.2                 | 50.6              | <i>Posible</i>            |
|               | 28.5       | 172.8        | 4.8          | 32.0       | 0.61      | 16.8      | 244.8    | 7.8                  | 14.7                 | -0.2              | <i>Imposible</i>          |
|               | 28.5       | 141.7        | 5.0          | 32.7       | 0.61      | 14.3      | 200.8    | 9.4                  | 17.5                 | -42.8             | <i>Imposible</i>          |
| 1500          | 28.5       | 717.1        | 4.1          | 29.6       | 0.61      | 48.8      | 1184.9   | 1.8                  | 3.3                  | 850.1             | <i>Posible</i>            |
|               | 28.5       | 564.4        | 4.2          | 29.8       | 0.61      | 41.2      | 932.6    | 2.3                  | 4.2                  | 622.8             | <i>Posible</i>            |
|               | 28.5       | 311.2        | 4.4          | 30.6       | 0.61      | 26.5      | 514.3    | 4.0                  | 7.4                  | 244.3             | <i>Posible</i>            |
|               | 28.5       | 214.3        | 4.6          | 31.4       | 0.61      | 20.1      | 354.3    | 5.6                  | 10.5                 | 97.5              | <i>Posible</i>            |
|               | 28.5       | 165.6        | 4.8          | 32.1       | 0.61      | 16.3      | 273.8    | 7.1                  | 13.3                 | 22.5              | <i>Posible</i>            |
|               | 28.5       | 135.8        | 5.0          | 32.9       | 0.61      | 13.8      | 224.6    | 8.5                  | 15.9                 | -24.4             | <i>Imposible</i>          |

## REIVINDICACIONES

Habiendo descrito suficientemente mi invención, considero como una novedad y por lo tanto reclamo como de mi exclusiva propiedad, lo contenido en la siguientes cláusulas:

1. Un **sistema con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**; caracterizado porque comprende un subsistema A (desalinización por evaporación de múltiple efecto al alto vacío y baja TBT) y un subsistema B (alimentación de agua salada, climatización y adecuación del agua producto mediante agua salada y salmuera fría), en donde:

- a) El subsistema A de **desalinización por evaporación de múltiple efecto al alto vacío y baja TBT**, está conformado por tres etapas: la etapa inicial, correspondiente al **Efecto 0**, la etapa intermedia, que abarca del **Efecto 1** hasta el **Efecto n** y la etapa final, correspondiente al **Efecto f**, donde la etapa inicial (**Efecto 0**), incluye un intercambiador de calor que funciona como generador de vapor (**GEN**), interconectado a una válvula de expansión (**VE1**) para agua salada y un separador líquido-vapor para salmuera (**SEP0**), que a su vez cuenta con dos salidas: una para la corriente de vapor (**6**) y otra para la corriente de salmuera (**11**) que se conecta con una tubería tipo U (**TU1**), las cuales a su vez se interconectan con el intercambiador de calor (**HX1**) que funciona como el evaporador del siguiente efecto (**Efecto 1**); las corrientes de vapor generado en cada uno de los efectos (como es el caso de la **corriente 6**), se dirigen al intercambiador de calor que actúa como evaporador en el efecto posterior, debido a que este sistema no requiere precalentadores de calores para el agua salada de alimentación (**corriente 1**); la etapa que abarca los efectos del **Efecto 1** hasta el **Efecto n**, cuenta con n intercambiadores de calor que actúan como evaporadores (**HX1, HX2, ... HXn**), donde n puede variar desde cero hasta cualquier número, n separadores líquido-vapor para salmuera y/o agua salobre (**SEP0, SEP1, ... SEPn**), en donde n coincide con el número de intercambiadores de calor que actúan como evaporadores (**HX**) menos uno, ya que cada evaporador (**HX**) cuenta con un separador líquido-vapor para salmuera (**SEP**), n separadores líquido-vapor para agua producto (**SEP1AP, SEP2AP, ...**

**SEP<sub>APn</sub>**), donde  $n$  equivale al número de evaporadores (**HX**) menos uno,  $n$  tuberías en forma de “U” (**TU<sub>0</sub>**, **TU<sub>1</sub>**, ... **TU<sub>n</sub>**), diseñadas para mantener el sello hidráulico necesario y preservar la diferencia de presión entre efectos consecutivos, donde  $n$  corresponde al número de evaporadores (**HX**) menos uno,  $n$  válvulas de expansión para el agua producto (**VE<sub>AP1</sub>**, **VE<sub>AP2</sub>**, ... **VE<sub>APn</sub>**), donde  $n$  es igual al número de evaporadores (**HX**) menos uno,  $n$  mezcladores para combinar los vapores generados durante el flasheo de la salmuera y el agua producto (**MIX<sub>1</sub>**, **MIX<sub>2</sub>**, ... **MIX<sub>n</sub>**), en donde  $n$  coincide con el número de evaporadores (**HX**) menos uno, y  $n$  mezcladores para mezclar las corrientes de agua producto (**MIX<sub>2AP</sub>**, **MIX<sub>3AP</sub>**, ... **MIX<sub>nAP</sub>**), donde  $n$  se define como el número de evaporadores (**HX**) menos dos; el **Efecto 1** de etapa intermedia, cuenta con un intercambiador de calor que actúa como evaporador (**HX<sub>1</sub>**) el cual se interconecta con el separador líquido-vapor para salmuera (**SEP<sub>1</sub>**) y con la válvula de expansión para agua producto (**VE<sub>1AP</sub>**), que a su vez se conecta con el separador para agua producto (**SEP<sub>1AP</sub>**) que cuenta con dos salidas: una para la corriente de vapor (**10**) que se conecta mediante una tubería al mezclador de vapor (**MIX<sub>1</sub>**), y otra para la corriente líquida (**9**), que se conecta al mezclador de agua producto (**MIX<sub>2AP</sub>**), de modo que la interconexión entre los componentes del **Efecto 1** se replica en los efectos subsecuentes, hasta el penúltimo efecto (**Efecto n**); mientras que la etapa final que abarca al último efecto (**Efecto f**), se compone de un intercambiador de calor que actúa como evaporador (**HX<sub>f</sub>**), un separador líquido-vapor para salmuera y/o agua salobre (**SEP<sub>f</sub>**), una tubería en forma de “U” (**TU<sub>f</sub>**), un mezclador para agua producto (**MIX<sub>fAP</sub>**), una válvula de expansión para el agua producto (**VE<sub>fAP</sub>**), un separador líquido-vapor para agua producto (**SEP<sub>f</sub>**), y un mezclador (**MIX<sub>f</sub>**) para combinar los vapores producidos durante el flasheo de la salmuera y el agua producto, presentando los los mismos componentes que los efectos anteriores, con excepción de la tubería tipo “U”, además la etapa final, presenta la diferencia de que interconecta directamente sus corrientes de vapor (**42**), de salmuera (**43**) y de agua producto (**35**) con los componentes del **subsistema B** (**alimentación de agua salada, climatización y adecuación del agua**

**producto mediante agua salada y salmuera fría);**

- b) El **subsistema B**, está compuesto por un tanque para agua salada (**TAM**), cuatro válvulas de paso (**VP1, VP2, VP3 y VP4**), dos bombas para agua salada (**BAM1**) y (**BAM2**), un divisor para agua salada (**DIVAM**) y un mezclador para agua salada (**MIXAM**), además, este **subsistema B**, incluye un intercambiador de calor que actúa como evaporador (**EVA**), interconectado con una válvula de tres vías (**V3V1**) para el agua producto y un mezclador para vapor de agua (**MIXVE**), el cual se conecta a un eyector-condensador (**EYC**) que utiliza una corriente líquida como fluido primario y vapor de agua como fluido secundario; el **subsistema B**, también cuenta con una bomba para agua producto (**BAP**), un intercambiador de calor (**HXsw**) que actúa como disipador de calor, cuatro válvulas de expansión (**VP5, VP6, VP8 y VP9**), un tanque para agua producto (**TAP**) y dos columnas barométricas (**CB1**) y (**CB3**) diseñadas para la extracción del agua producto, además, se compone por diversos elementos, entre los que se incluyen: una válvula de tres vías para salmuera (**V3V2**), un intercambiador de calor (**HXBR**) que actúa como disipador de calor, dos válvulas de paso (**VP10**) y (**VP11**), un separador líquido-vapor para salmuera (**SEPBR**) y un tanque para el almacenamiento de salmuera (**TS**), el cual está interconectado a una columna barométrica (**CB2**) para la extracción de la salmuera.

2. Un **proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**; que está compuesto de un **supbroceso A (desalinización por evaporación de múltiple efecto al alto vacío y baja TBT)**, y un **subproceso B (alimentación de agua salada, climatización y adecuación del agua producto mediante agua salada y salmuera fría)**, caracterizado porque:

- a) El **subproceso A desalinización por evaporación de múltiple efecto al alto vacío y baja TBT**, que utiliza agua salada, consta de las etapas:
- i. Etapa inicial, correspondiente al **Efecto 0**, que recibe la corriente de agua salada de alimentación (**1**) directamente desde el **subproceso B**, ya que el proceso pudiera no requerir precalentadores para el agua salada cuando la



TBT sea igual o menor a la temperatura de alimentación del agua salada, dado que no se tiene gradiente de temperatura; el subproceso A, modifica la temperatura máxima de salmuera (TBT), reduciéndola desde sus valores convencionales (65-95 °C) a temperaturas inferiores; dichas temperaturas

5 pueden corresponder a una TBT igual a la temperatura de alimentación del agua salada (por ejemplo, un valor normal y no limitativo es 25 °C) o hasta valores cercanos al punto de congelamiento de la salmuera, donde la TBT pudiera coincidir con la BBT; un valor normal y no limitativo es una BBT de 4 °C; la corriente de agua salada de alimentación **(1)** se dirige hacia la válvula

10 de expansión **(VE1)**, donde su presión se reduce hasta la presión del **Efecto 0**, que corresponde a la temperatura de saturación del **Efecto 0** y sale como la corriente de agua salada **(2)**; la etapa del **Efecto 0**, está equipada con un un intercambiador de calor que opera como generador de vapor **(GEN)**, que recibe la corriente del fluido caloportador **(4)** y a la corriente de agua salada

15 **(2)**; la corriente del fluido caloportador **(4)**, la cual puede ser cualquier fluido con la capacidad térmica suficiente para generar vapor en el generador **(GEN)** y además puede provenir de cualquier tipo de proceso o tecnología, preferiblemente proveniente de fuentes termosolares como: Concentradores Parabólicos Compuestos, Concentradores solares tipo Cilindro Parabólico,

20 Fresnel, entre otros; en el generador **(GEN)**, la corriente de agua salada **(2)** es calentada por el fluido caloportador **(4)**, lo que provoca su transformación en una corriente bifásica (líquida-vapor) y salga del generador **(GEN)** como la corriente de agua salada **(3)** para dirigirse hacia el separador líquido-vapor para salmuera **(SEP0)**; la corriente de fluido caloportador **(5)** sale del

25 generador **(GEN)** después de haber calentado a la corriente de agua salada **(2)**; en el separador **SEP0**, la corriente de salmuera **(3)** se divide en dos corrientes: la corriente de vapor **(6)**, que posteriormente transfiere su energía de condensación en el intercambiador de calor **(HX1)** que actúa como evaporador del **Efecto 1**, y la corriente de salmuera **(11)**, que se dirige hacia

30 una tubería “U” **(TU1)**, donde se logra un sello hidráulico que mantiene la diferencia de presión entre el **Efecto 0** y **Efecto 1**, por lo cual la corriente de

salmuera se flashea y sale como la corriente **(12)** para ingresar posteriormente al intercambiador de calor **(HX1)** y recibir el calor transferido por parte de la corriente de vapor **(6)** y después de ser calentada salir como la corriente de salmuera **(13)**;

- 5            ii. Etapa intermedia que comprende del **Efecto 1** hasta el **Efecto n**, donde el **Efecto 1** tiene un intercambiador de calor que actúa como evaporador del **Efecto 1 (HX1)** y recibe dos corrientes, la corriente de vapor **(6)** proveniente del separador líquido-vapor **(SEP0)** para salmuera del efecto anterior y la
- 10            corriente de salmuera en dos fases **(12)** que proviene de la tubería en forma de “U” **(TU1)** del efecto anterior; en el intercambiador de calor **(HX1)** la corriente **(6)** transfiere su energía en forma de calor latente hacia la corriente de salmuera **(12)**, saliendo como la corriente condensada de agua producto **(7)** que se dirige hacia la válvula de expansión para agua producto **(VE1<sub>AP</sub>)** donde se reduce su presión hasta igualarla con la presión del siguiente efecto,
- 15            saliendo como la corriente de agua en dos fases **(8)** que ingresa al separador líquido-vapor para agua **(SEP1<sub>AP</sub>)** del cual salen las corrientes, una de vapor **(10)** y otra de agua producto **(9)**, en tanto que la corriente de salmuera **(12)** sale del intercambiador de calor **(HX1)** como la corriente **(13)** con una mayor proporción de vapor y se dirige hacia el separador líquido-vapor para salmuera
- 20            **(SEP1)**, del cual salen la corriente de vapor **(14)** y la corriente de salmuera **(15)**; la corriente de vapor **(14)** se dirige hacia el mezclador de vapor **(MIX1)** donde se mezcla con la corriente de vapor **(10)** que proviene del separador líquido-vapor para agua **(SEP1<sub>AP</sub>)**, dando lugar a la corriente de vapor **(18)** que ingresa al intercambiador de calor **(HX2)** que actúa como evaporador del
- 25            **Efecto 2**, mientras que la corriente de salmuera **(15)** se dirige a la tubería en forma de “U” **(TU2)** para disminuir su presión hasta igualarla con la presión del siguiente efecto (**Efecto 2**), en cambio la corriente de agua producto **(9)** se dirige hacia el mezclador de agua **(MIX2<sub>AP</sub>)** donde se mezcla con la corriente de agua producto **(19)** proveniente del intercambiador de calor **(HX2)**, que
- 30            actúa como evaporador del **Efecto 2**; en el mezclador para agua producto **(MIX2<sub>AP</sub>)** las corrientes de agua producto **(9)** y **(19)** se unen formando la

corriente de agua producto **(20)**, dirigiéndose hacia la válvula de expansión **(VE2<sub>AP</sub>)** para reducir su presión hasta la presión correspondiente al siguiente efecto, saliendo como la corriente de agua producto **(21)** que se integra con la corriente de agua producto del siguiente; lo ocurrido en el **Efecto 1**, sucede en los posteriores efectos hasta el **Efecto n**, presentando una reducción progresiva de presión y temperatura en cada efecto, formando una cascada desde el **Efecto 1** hasta el **Efecto n**;

- iii. Etapa del **Efecto f**, en la cual se replica el proceso descrito en los efectos de la etapa previa, desde el **Efecto 1** hasta el **Efecto n**, no obstante, al ser el último efecto dentro del **subproceso A**, las corrientes resultantes adquieren particular importancia, la corriente total de vapor **(42)** generada en el **Efecto f**, la corriente de salmuera **(43)** y la corriente de agua producto total **(35)** que acumula los condensados de agua producto de los múltiples efectos, se dirigen al **subproceso B**; la corriente de vapor total **(42)**, generada en el **Efecto f** tanto por flasheo como por ebullición, es enviada al **subproceso B** para ingresar al mezclador de vapor **(MIXVE)**, mientras que la corriente de salmuera **(43)** y la corriente de agua producto **(35)** también se dirigen al **subproceso B**, ingresando respectivamente a la válvula de tres vías para salmuera **(V3V2)** y a la válvula de tres vías para agua **(V3V1)** del **subproceso B**;

- b) El **subproceso B** de **alimentación de agua salada, climatización y adecuación del agua producto mediante agua salada y salmuera fría**, consta de las etapas:

- i. Etapa de **alimentación de alimentación de agua salada**, en donde se presentan dos modalidades diferentes para realizar la alimentación de agua salada al proceso: 1) cuando el agua salada se precalienta en el intercambiador de calor **(HX<sub>sw</sub>)** y 2) cuando el agua salada se dirige directamente al efecto inicial **(Efecto 0)** del **subproceso A**; en la primera modalidad, cuando el agua salada se precalienta en el intercambiador de calor **(HX<sub>sw</sub>)**, las válvulas de paso **(VP3)**, **(VP4)**, **(VP6)**, **(VP7)**, **(VP10)** y **(VP11)**, deben permanecer cerradas, mientras que las válvulas de paso **(VP1)**, **(VP2)**, **(VP5)**, **(VP8)** y **(VP9)**, deben estar abiertas para permitir que la corriente de agua salada **(1a)**

que proviene del tanque para agua salada (**TAM**), pase a través de la válvula de paso (**VP1**) y se dirija hacia la bomba para agua salada (**BAM1**) y posteriormente al intercambiador de calor que actúa como disipador de calor (**HXsw**); en el intercambiador de calor (**HXsw**) la corriente de agua de agua salada (**2a**) que proviene de la bomba (**BAM1**) para agua salada, recibe el calor de la corriente de agua producto (**55**) proveniente de la válvula de paso (**VP5**), y posteriormente sale del intercambiador de calor (**HXsw**) como la corriente de agua salada (**3a**), que se dirige a un divisor de flujo (**DIVAM**), del cual salen las corrientes de agua salada (**4a**) y (**5a**); la corriente (**4a**), que contiene solo el flujo másico necesario para activar el sistema MED del **subproceso A**, pasa a través de la válvula de paso (**VP2**) para ingresar al mezclador (**MIXAM**) de agua salada, y convertirse en la corriente de agua salada (**1**), que posteriormente se dirige al efecto inicial (**Efecto 0**) del **subproceso A**, mientras que la corriente de agua salada (**5a**) que conlleva el flujo másico necesario para enfriar a la corriente (**55**) de agua producto, se redirige al tanque de salmuera (**TS**) para ser utilizada en la dilución de la corriente de salmuera (**50**); en la segunda modalidad, cuando el agua salada se dirige directamente al efecto inicial (**Efecto 0**) del **subproceso A**, las válvulas de paso (**VP1**), (**VP2**), (**VP5**), (**VP8**) y (**VP9**) deben de permanecer cerradas, mientras que las válvulas de paso (**VP3**), (**VP4**), (**VP6**), (**VP7**), (**VP10**) y (**VP11**) deben estar abiertas, para permitir que la corriente de agua salada (**1b**) que proviene del tanque para agua salada (**TAM**), pase a través de la válvula de paso (**VP3**) y sea succionada por la bomba para agua salada (**BAM2**) y posteriormente salga como la corriente (**2b**) para pasar por la válvula de paso (**VP4**) antes de ingresar al mezclador (**MIXAM**) para agua salada, de donde sale como la corriente de agua salada (**1**) que se dirige al efecto inicial (**Efecto 0**) del **subproceso A**;

ii. Etapa de **producción de frío**, en la cual la corriente de agua producto (**35**) proveniente del **Efecto f** del **subproceso A**, se dirige hacia una válvula de tres vías para agua (**V3V1**), la cual separa dicha corriente en las corrientes de agua producto (**36**) y (**37**), la corriente de agua producto (**37**) se dirige a un

intercambiador de calor que funciona como evaporador (EVA) destinado a la generación de frío; dentro del evaporador (EVA) la corriente de agua producto (37) absorbe el calor de la corriente (39), transformándose en la corriente de vapor (38), que posteriormente es conducida hacia el mezclador de vapor (MIXVE), mientras que la corriente (39) sale del evaporador (EVA) a menor temperatura como la corriente (40); en el mezclador (MIXVE), la corriente de vapor (38), se mezcla con la corriente de vapor (42) proveniente del Efecto f y con la corriente de vapor (47) proveniente del separador (SEPBR), convirtiéndose en la corriente de vapor (52); la corriente de vapor (47) solo se genera y se mezcla con las corrientes (38) y (42), en los casos donde se utiliza la salmuera fría para el enfriamiento del eyector-condensador (EYC); por otra parte, la corriente de agua producto (36) se dirige hacia la columna barométrica (CB1), que cumple con la función de crear un sello hidráulico y mantener el vacío dentro del sistema, saliendo por la base de la columna barométrica (CB1) en condiciones de presión atmosférica y posteriormente depositarse en el tanque para agua producto (TAP); la función de la válvula de tres vías para agua (V3V1) es la de controlar la capacidad de enfriamiento regulando la cantidad de flujo másico de la corriente refrigerante (37) que ingresa al evaporador (EVA); en el caso de que el flujo másico de la corriente de agua producto (37) sea nulo, el **proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, queda limitado únicamente a la desalinización, sin proporcionar el servicio de generación de frío o calefacción para el acondicionamiento térmico de espacios;

iii. Etapa de **eyecto-condensación barométrica**, compuesta por un tanque para agua producto (TAP) con dos salidas, una correspondiente a la corriente de agua producto (64) que extrae el total del agua desalinizada del **proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios**, y otra para la corriente de agua producto (53) que es succionada por la bomba para agua producto (BAP), donde aumenta su presión y sale como la corriente de agua (54); la presión de

la corriente de agua **(53)** debe ser superior a la presión de la corriente **(62)** que se descarga en el eyector-condensador **(EYC)** y a la de la corriente de vapor de baja presión **(52)** utilizada como fluido secundario en el eyector-condensador **(EYC)**; por otro lado, la presión de la corriente de vapor de baja presión **(52)** puede ser cualquier presión correspondiente a una temperatura de saturación adecuada para la producción de frío en el evaporador **(EVA)**, siendo una presión preferida de 0.8 kPa para aplicaciones de aire acondicionado; al ingresar al eyector-condensador **(EYC)**, las corrientes **(52)** y **(61)** se mezclan y salen en forma líquida a cualquier presión intermedia, dicha presión se encuentra por debajo de la presión del fluido motriz **(corriente de agua producto 61)**, pero es mayor a la del fluido secundario **(corriente de vapor de baja presión 52)**, asegurando así la condensación de la corriente de vapor de baja presión **(52)**; la condensación total del vapor **(corriente de vapor de baja presión 52)** se presenta a la salida del eyector-condensador **(EYC)**, donde la velocidad de la corriente mezclada de descarga **(62)** disminuye al incrementarse el área transversal, y a su vez este cambio provoca un aumento abrupto de presión, completando la condensación del vapor de baja presión **(52)**; el enfriamiento del fluido motriz **(corriente de agua producto 61)** en la etapa de **eyecto-condensación barométrica**, puede llevarse a cabo mediante las diferentes configuraciones, dependiendo de los requerimientos del sistema: 1) **enfriado únicamente mediante agua salada**, 2) **enfriado mediante agua salada y salmuera fría**, y 3) **enfriado únicamente mediante salmuera fría**; en la primera modalidad, donde el eyector-condensador **(EYC)** es **enfriado únicamente mediante agua salada**, la válvula de tres vías **(V3V2)**, bloquea el flujo de la corriente **(44)** y redirige la totalidad de la corriente hacia la salida **(48)**, además, las válvulas de paso **(VP3)**, **(VP4)**, **(VP6)**, **(VP7)**, **(VP10)** y **(VP11)** permanecen cerradas, mientras que las válvulas **(VP1)**, **(VP2)**, **(VP5)**, **(VP8)** y **(VP9)** permanecen abiertas, permitiendo que la corriente de agua producto **(54)** pase por la válvula de paso **(VP5)** hacia el intercambiador de calor **(HXsw)** como la corriente **(55)**, donde la corriente de agua salada **(2a)**, proveniente de la bomba

(BAM1), extrae el calor de la corriente de agua producto (55) y sale como la corriente (3a) hacia el tanque para salmuera (TS), mientras que la corriente de agua producto (57) atraviesa mediante la válvula de paso (VP8) e ingresa a la válvula de paso (VP9) como la corriente (58) y finalmente ingresa al eyector-condensador (EYC) como la corriente motriz (**corriente de agua producto 61**); en la segunda modalidad, donde el eyector-condensador (EYC) es **enfriado mediante agua salada y salmuera fría**, la válvula de tres vías (V3V2), permite pasar de forma parcial o total el flujo de la corriente de salmuera (44), según el nivel de enfriamiento requerido en el eyector-condensador (EYC), además, las válvulas (VP3), (VP4), (VP6), (VP7), (VP8) y (VP9) permanecerán cerradas, y las válvulas de paso (VP1), (VP2), (VP5), (VP10) y (VP11) permanecen abiertas, permitiendo que la corriente (55) ingrese al intercambiador de calor (HX<sub>sw</sub>) para ser enfriada por la corriente de agua salada (2a), salga como la corriente (57), pase através de la válvula de paso (VP10) y continúe como la corriente (59) hacia el intercambiador de calor (HX<sub>BR</sub>), donde es enfriada por la corriente de salmuera (44) la cual se encuentra en su punto de saturación, y sale como la corriente (60) hacia la válvula de paso (VP11), ingresando al eyector-condensador (EYC) como fluido primario (**corriente de agua producto 61**); en la tercera modalidad, donde el eyector-condensador (EYC) es **enfriado únicamente mediante salmuera fría**, se utiliza la corriente de salmuera (43), que proviene del último efecto de la MED (**Efecto f**) y se encuentra en estado de saturación y baja temperatura; la válvula de tres vías (V3V2) cierra el paso hacia la corriente (48) y dirige todo el flujo hacia la corriente (44), además, las válvulas (VP1), (VP2), (VP5), (VP8) y (VP9) permanecen cerradas, mientras las válvulas (VP3), (VP4), (VP6), (VP7), (VP10) y (VP11) están abiertas, permitiendo que la corriente (53) sea succionada por la bomba (BAP) y salga como la corriente (54), para pasar por las válvulas (VP6), (VP7) y (VP10), saliendo como las corrientes (56), (57) y (59), respectivamente; la corriente (59) ingresa al intercambiador de calor (HX<sub>BR</sub>) para enfriarse mediante el intercambio térmico con la corriente de salmuera (44), saliendo a

- menor temperatura como la corriente **(60)** hacia la válvula **(VP11)**, y posteriormente ingresar al eyector-condensador **(EYC)** como fluido primario (**corriente de agua producto 61**); en las dos últimas configuraciones de enfriamiento del eyector-condensador **(EYC)** mencionadas, se utiliza la
- 5 corriente de salmuera **(44)** de manera total o parcial como fluido de enfriamiento, saliendo del intercambiador de calor **(HXBR)** como una corriente de salmuera en dos fases **(45)**, que se dirige al separador líquido-vapor **(SEPBR)**; este separador divide la corriente de salmuera **(45)** en las
- 10 corrientes de salmuera líquida **(46)** y la corriente de vapor de agua **(47)**, la corriente de salmuera **(46)** se une a través de una tubería con la corriente de salmuera **(48)** para formar la corriente de salmuera **(49)**, que se dirige hacia la columna barométrica **(CB2)**, donde se asegura el sello hidráulico y el mantenimiento del vacío en el sistema; posteriormente, la corriente líquida es
- 15 extraída del fondo de la columna barométrica **(CB2)** a presión atmosférica como la corriente **(50)** y se deposita en el tanque para salmuera **(TS)** de donde sale como la corriente de salmuera **(51)**; por otro lado, la corriente de vapor de agua **(47)** es conducida al mezclador **(MIXVE)**, donde se combina con los vapores **(38)** y **(42)**, generando la corriente de vapor **(52)**, que es succionada por el eyector-condensador **(EYC)** como fluido secundario (**corriente de**
- 20 **vapor de baja presión 52**).
3. El proceso de acuerdo con la **reivindicación 2** caracterizado además porque en la **etapa intermedia** que pertenece al **subproceso A** y que comprende desde el **Efecto 1** hasta el **Efecto n**, n es igual a cero y por lo tanto el **subproceso A**, se limita a dos etapas: la etapa
- 25 inicial, correspondiente al **Efecto 0**, y la etapa final, correspondiente al **Efecto f**; en esta configuración, donde el número de efectos totales del **subproceso A** se reduce a dos (**Efecto 0** y **Efecto f**), el separador líquido-vapor para salmuera **(SEP0)** se conecta directamente con la tubería en forma de “U” **(TUf)** y con el intercambiador de calor **(HXf)** que pertenecen al **Efecto f**, a través de las corrientes de salmuera **(11)** y vapor de
- 30 agua **(6)**, respectivamente; el resto del proceso que se lleva a cabo en la etapa final (**Efecto f**) se desarrolla de manera idéntica a lo descrito previamente.



4. El proceso de acuerdo con la **reivindicación 2** caracterizado además porque la etapa intermedia que pertenece al **subproceso A** y que comprenden desde el **Efecto 1** hasta el **Efecto n**, n es igual a cero, y además, en la etapa final (**Efecto f**), f también es igual a  
5      cero; en esta configuración, el **subproceso A** opera con un único efecto de evaporación (**Efecto 0**), la corriente de agua salada (**3**) proveniente del generador (**GEN**), ingresa al separador líquido-vapor para salmuera (**SEP0**) y se divide en las corrientes de vapor de agua (**42**) y de salmuera (**43**); la corriente de salmuera (**43**) se dirige a la válvula de tres vías (**V3V2**) del **subproceso B**, mientras que la corriente de vapor de agua (**42**) se redirige  
10     al mezclador (**MIXVE**) que pertenece al **subproceso B**; se destaca que en el **subproceso B** se introduce un nuevo componente: una válvula de expansión (**VR**), correspondiente a la etapa de **producción de frío**; en este caso, la corriente de agua producto (**64**) que sale del tanque para agua producto (**TAP**), pasa a través de la válvula de expansión (**VR**) antes de ingresar al evaporador (**EVA**); el resto del proceso en el **subproceso B**, se desarrolla  
15     como se describió previamente.
5. El proceso de acuerdo con la **reivindicación 2** caracterizado además porque el **subproceso A** opera con un único efecto de evaporación (**Efecto 0**), y en donde la corriente de agua salada (**3**) proveniente del generador (**GEN**), ingresa al separador  
20     líquido-vapor para salmuera (**SEP0**) y se divide en las corrientes de vapor de agua (**42**) y de salmuera (**43**); la corriente de salmuera (**43**) se dirige a la válvula de tres vías (**V3V2**) del **subproceso B**, mientras que la corriente de vapor de agua (**42**) se redirige al mezclador (**MIXVE**) que pertenece al **subproceso B**; se destaca que en el **subproceso B** se introduce un nuevo componente: un expansor generador (**EXP**) para la generación de  
25     energía eléctrica, aprovechando la notable diferencia de presión entre las corrientes (**64**) y la corriente (**37**).
6. El proceso de acuerdo con la **reivindicación 2** caracterizado además porque la etapa intermedia, que comprenden desde el **Efecto 1** hasta el **Efecto n**, n es igual a cero, y  
30     además, en la etapa final (**Efecto f**), f también es igual a cero; además con la diferencia de que se prescinde del evaporador (**EVA**) y la válvula de expansión (**VR**),

correspondientes al **subproceso B**; lo anterior implica que el **subproceso B**, no incluye la etapa de **producción de frío**, proporcionando únicamente el servicio de desalinización.

- 5        7. El proceso de acuerdo con la **reivindicación 2** caracterizado además porque el **subsistema A** y **subproceso A** operan con una TBT que sea menor a los valores convencionales pero mayor a la temperatura del agua salada de alimentación, por ejemplo, un rango normal y no limitativo es 35-65 °C; lo anterior, para lograr la mayor integración energética interna posible y reducir la adición de calor de calor de forma sensible o bien, reducir la cantidad de calor suministrada en el GEN; en esta variante, la
- 10       cantidad de precalentadores necesarios, está directamente relacionado con la TBT establecida y el gradiente de temperatura entre el vapor generado en los efectos y el agua salada de alimentació; a mayor TBT, mayor será el número de precalentadores a utilizar.
- 15       8. El proceso de acuerdo con la **reivindicación 2** caracterizado además porque el proceso de evaporación al vacío en el **subproceso A**, puede ser de múltiple efecto, múltiple etapa flash, de membranas o cualquier otro subproceso de desalación que opere en presiones de vacío.

## RESUMEN

La presente invención propone un novedoso **sistema y proceso con baja temperatura de activación para la producción simultánea de agua y acondicionamiento térmico de espacios** que puede ser activado con energía solar térmica. Este desarrollo busca ofrecer una alternativa tecnológica eficiente que permita enfrentar los desafíos asociados con la escasez de agua potable y la necesidad de climatización de espacios. El proceso opera con una temperatura mínima de la salmuera (BBT) inferior a los valores convencionales (35-45 °C). Un valor normal y no limitativo es una BBT de 4 °C. Esto amplía el rango operativo del sistema, posibilitando la reducción de la temperatura máxima de la salmuera (TBT) desde los valores tradicionales de 65-95 °C a temperaturas inferiores. Dichas temperaturas pueden corresponder a una TBT igual a la temperatura de alimentación del agua salada (por ejemplo, un valor normal y no limitativo es 25 °C) o hasta valores cercanos al punto de congelamiento de la salmuera, donde la TBT pudiera coincidir con la BBT. Un valor normal y no limitativo es una BBT de 4 °C. Este avance representa un salto significativo en términos de eficiencia y viabilidad para los sistemas MED. Adicionalmente, gracias al funcionamiento en condiciones de alto vacío (por ejemplo 0.8 kPa en el último efecto), el agua desalinizada puede utilizarse como fluido refrigerante, permitiendo la producción simultánea de frío. Este efecto se consigue mediante un eyector-condensador que succiona, arrastra y comprime para condensar, el vapor frío del último efecto de la MED, optimizando así los indicadores de rendimiento y capacidad del proceso. La invención está compuesta por tres subprocesos: un **subproceso A de desalinización por evaporación de múltiple efecto al alto vacío y baja TBT**, y un **subproceso B de alimentación de agua salada, climatización y adecuación del agua producto mediante agua salada y salmuera fría**. Estas características convierten al sistema en una opción versátil y eficaz para sectores como el residencial, agrícola, ganadero, turístico, comercial e industrial, especialmente en regiones costeras de clima cálido-húmedo donde la disponibilidad de agua potable es limitada y las temperaturas ambientales son elevadas

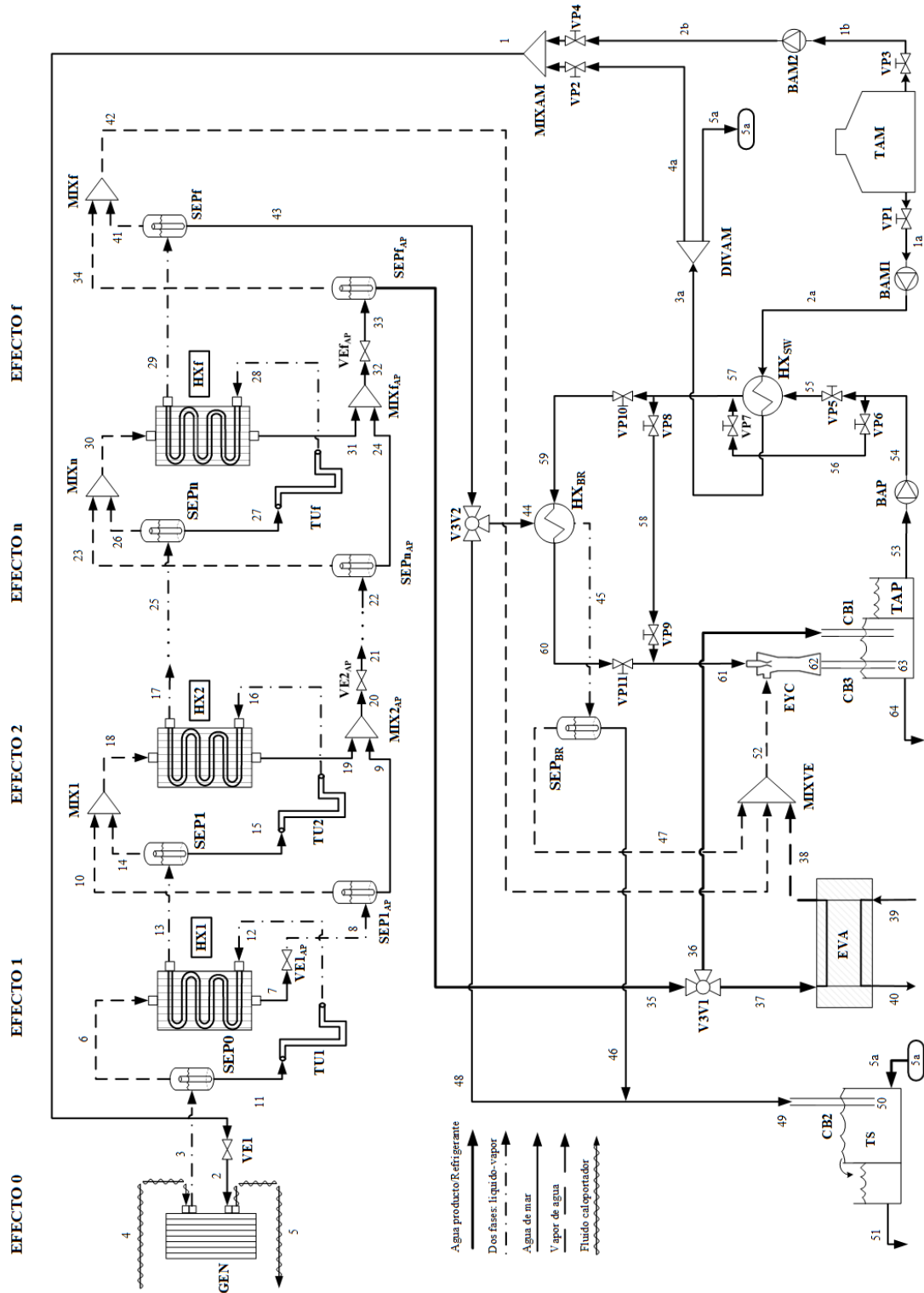


FIGURA 1

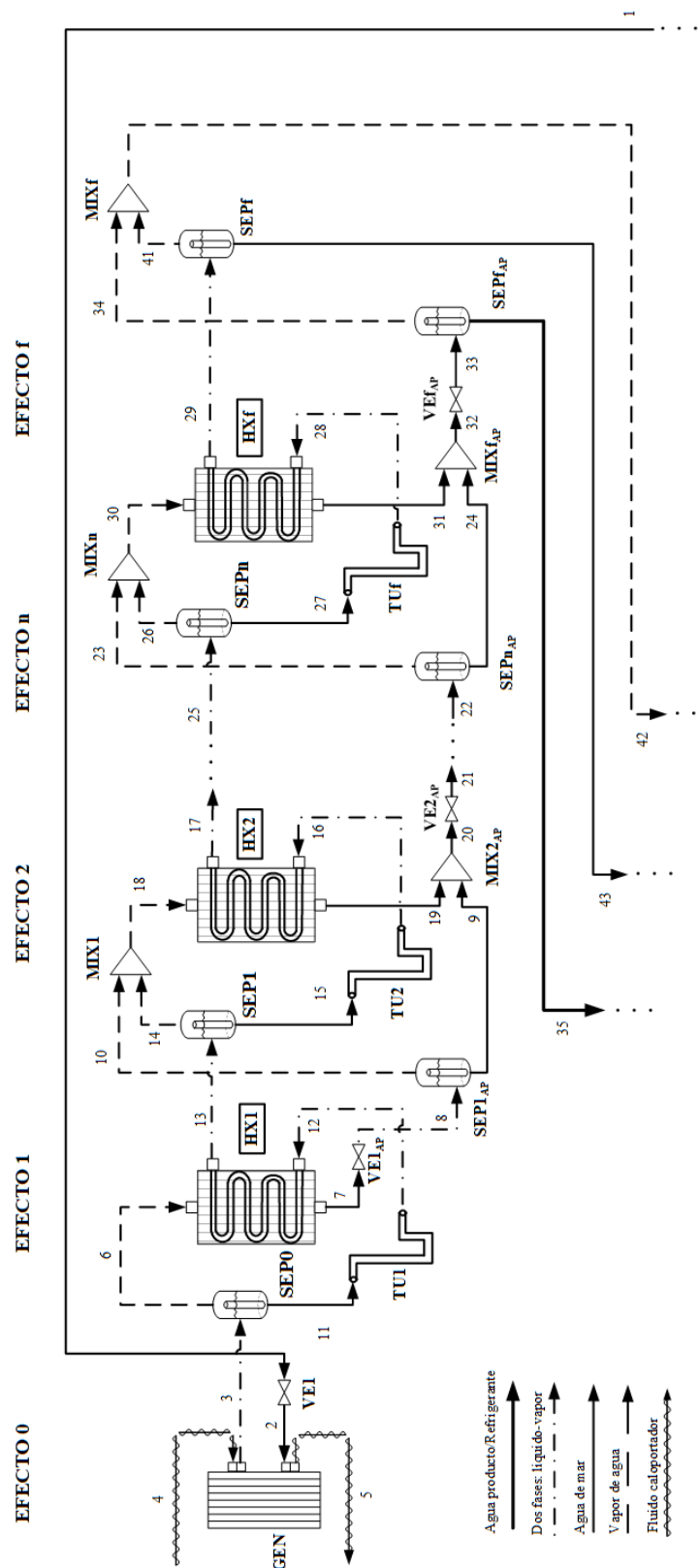


FIGURA 2



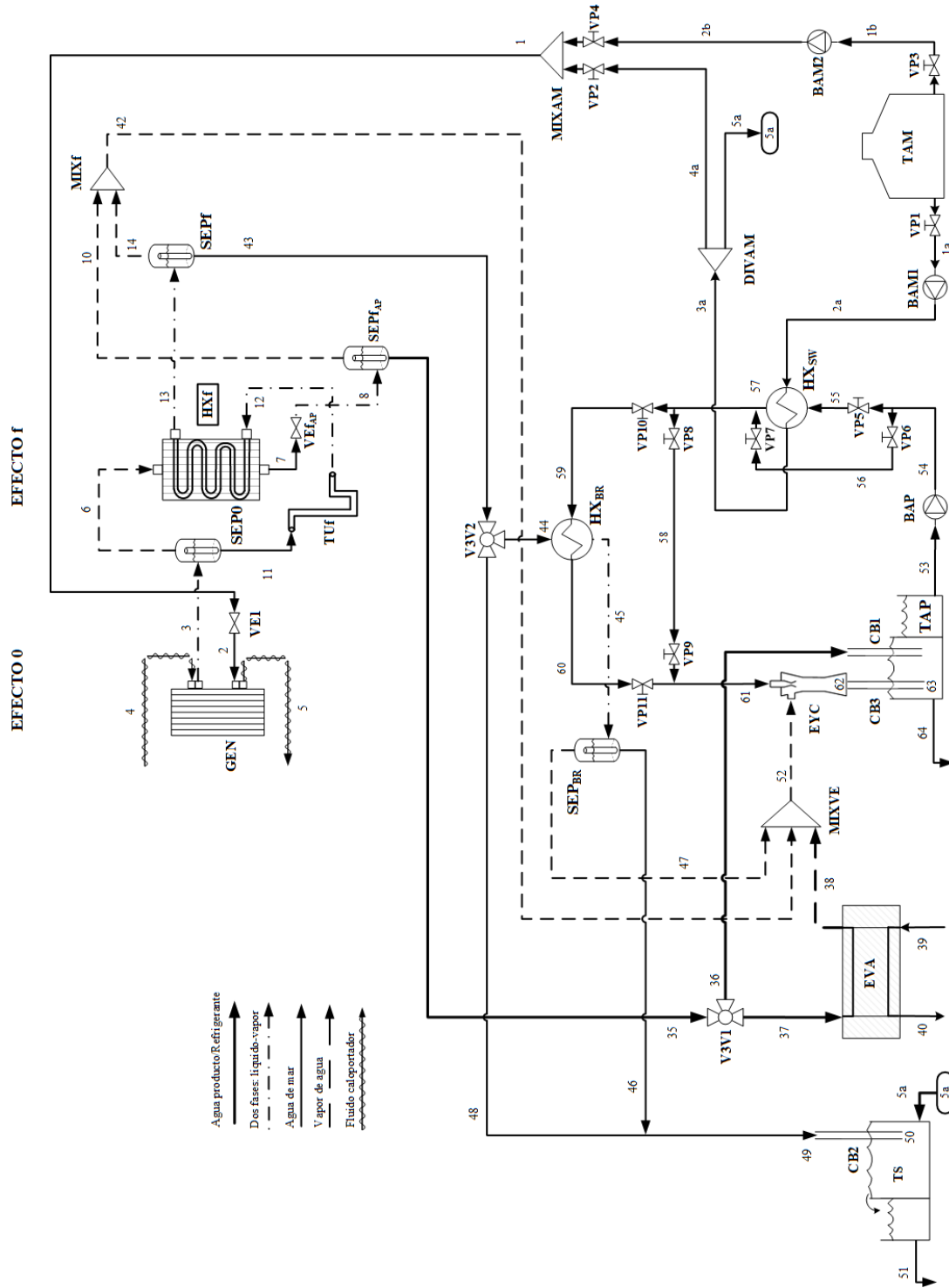
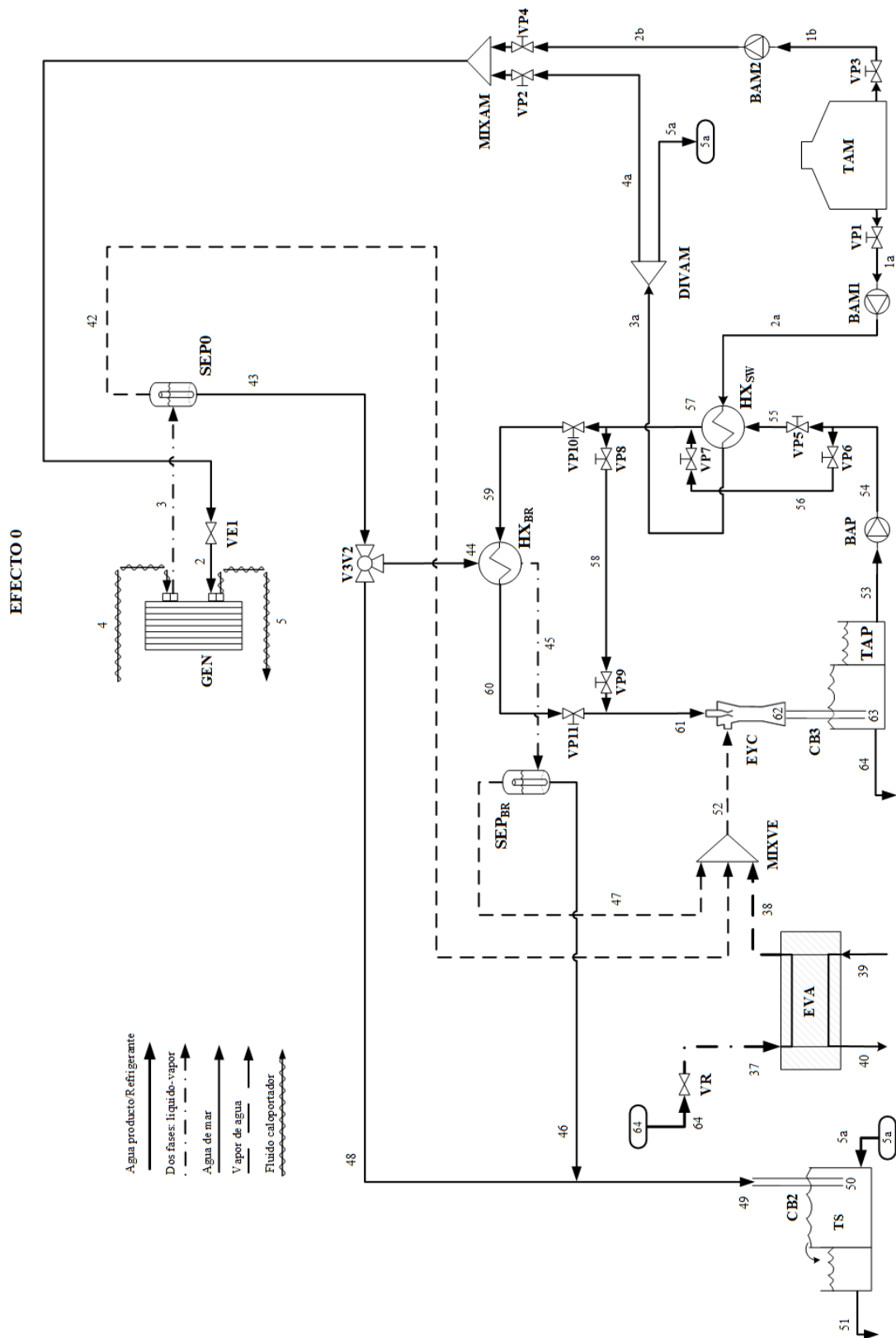
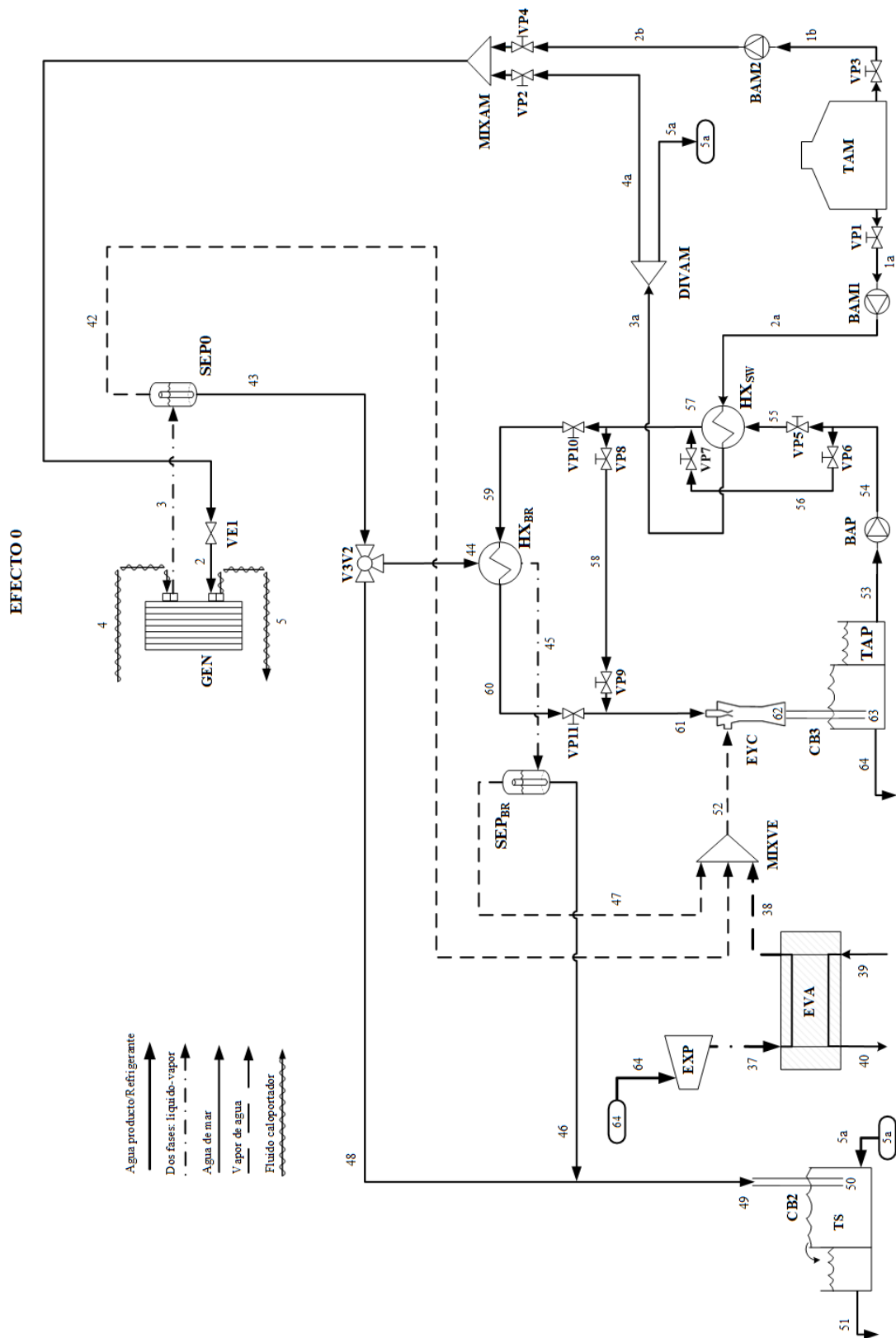


FIGURA 4

**FIGURA 5**



**FIGURA 6**

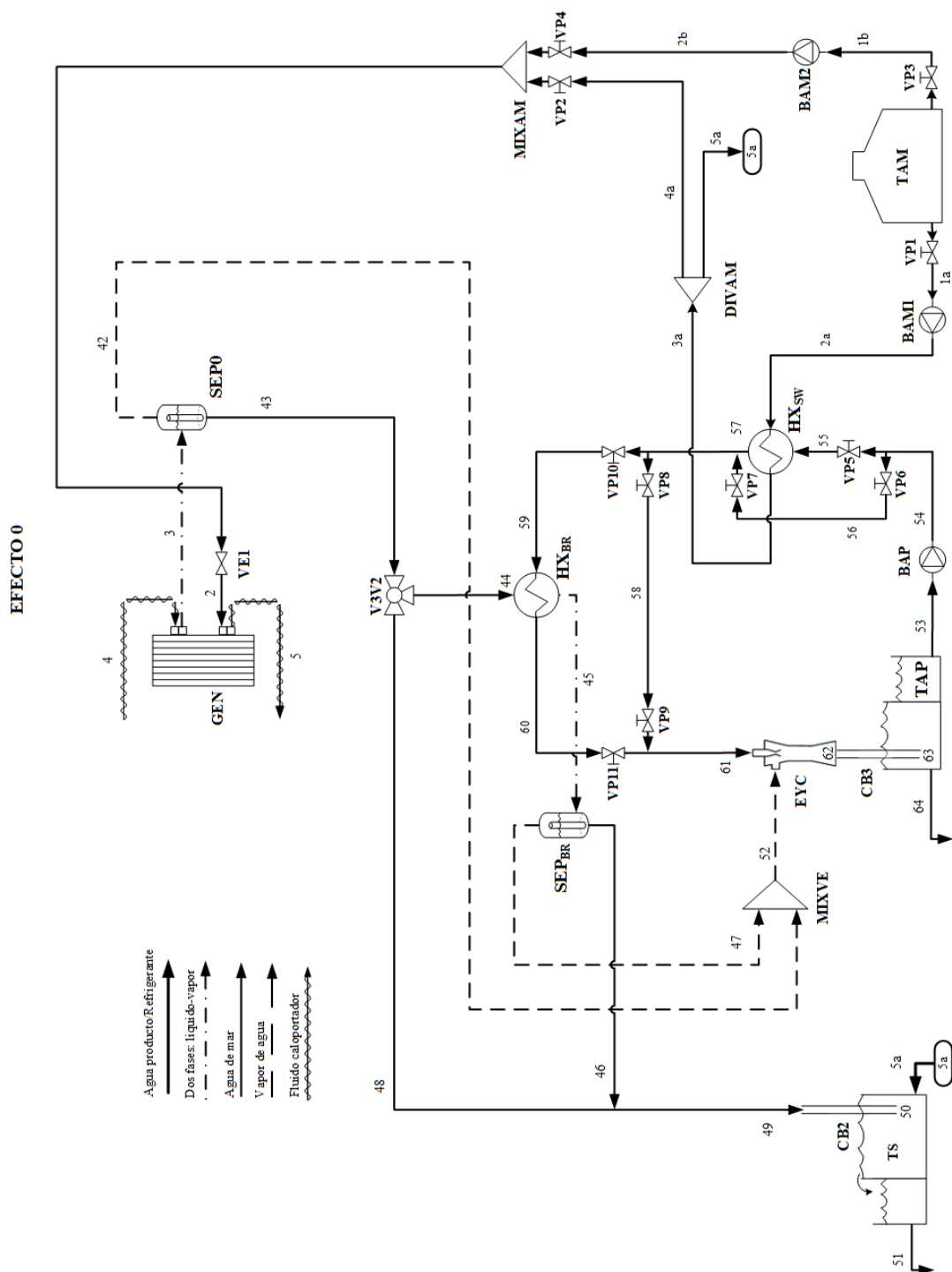


FIGURA 7

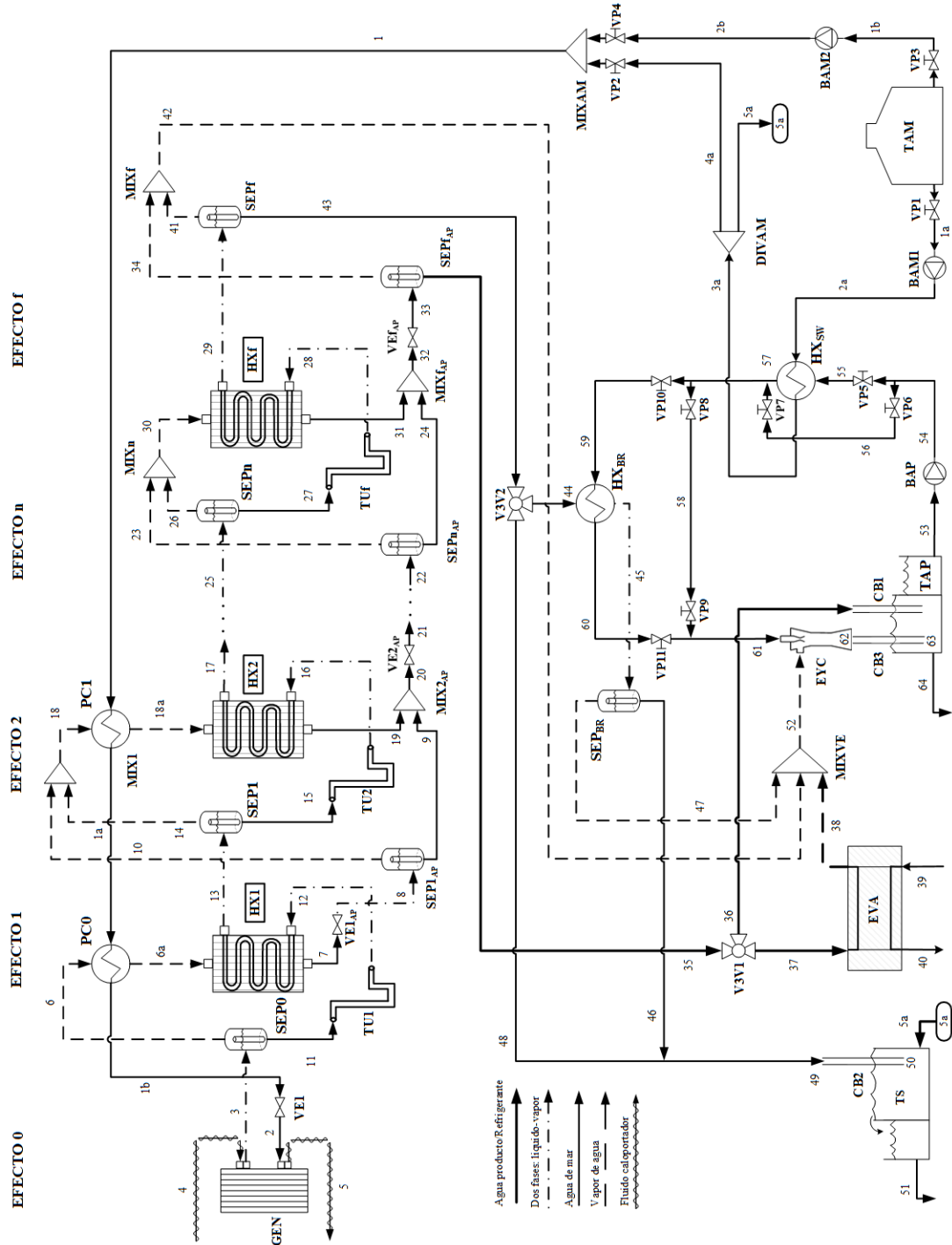


FIGURA 8

### 3.6. Artículos en los que se participó como coautor.

- F.J. Caballero-Talamantes, N. Velázquez-Limón, J.A. Aguilar-Jiménez, C.A. Casares-De la Torre, R. López-Zavala, J. Ríos-Arriola, S. Islas-Pereda, **A Novel High Vacuum MSF/MED Hybrid Desalination System for Simultaneous Production of Water, Cooling and Electrical Power, Using Two Barometric Ejector Condensers**, Processes. 12 (2024). doi:10.3390/pr12122927.
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- Ríos-Arriola, J., Velázquez-Limón, N., Aguilar-Jiménez, J. A., Islas, S., López-Sánchez, J. D., Caballero-Talamantes, F. J., Corona-Sánchez, J. A., & Cásares-De la Torre, C. A., **Comparison between Air-Exposed and Underground Thermal Energy Storage for Solar Cooling Applications**, Processes, 11 (2023). doi:10.3390/pr11082406.
- J.A. Aguilar-Jiménez, N. Velázquez-Limón, R. López-Zavala, S. Islas-Pereda, J. Ríos-Arriola, C.A. Cásares, **Use of sensible and latent condensation heat of an organic Rankine cycle in a multiple-effect desalination system: A comparative study of novel energy integrations**, Int. J. Energy Res. 46 (2022) 18363–18378. doi:10.1002/er.8450.
- J.A. Aguilar-Jiménez, N. Velázquez-Limón, S. Islas-Pereda, J. Ríos-Arriola, J.A. Suástegui-Macias, C.A. Cásares-De la Torre, J.A. Gómez-Hernández, **Technical and economic assessment of hybrid PVT solar systems compared to independent solar thermal and photovoltaic collectors in high-temperature zones: A case study in Mexicali, Mexico**, Renew. Energies. 2 (2024). doi:10.1177/27533735241278104.

# CAPÍTULO 4

*“El conocimiento habla,  
pero la sabiduría escucha”  
—Jimi Hendrix*

## CONCLUSIONES Y TRABAJOS FUTUROS

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El presente capítulo presenta las conclusiones generales de cada propuesta tecnológica y de los estudios realizados en esta Tesis. Finalmente se presentan las líneas de trabajo futuro dirigidas a transitar de la validación conceptual hacia la implementación práctica y optimización de esta tecnología, con el objetivo último de posicionarla como una solución competitiva y sostenible hacia las problemáticas del agua y acondicionamiento térmico de espacios.

### 4.1. Conclusiones

El primer avance consistió en redefinir los límites operativos de la tecnología MED, posibilitado por la implementación de un eyector-condensador barométrico. La configuración de alto vacío (MED-AV), con un rango de presión y temperatura 28.3 a 0.8 kPa y 68 a 4 °C, respectivamente, demostró ser técnicamente viable y energéticamente superior a un sistema convencional.

Con base al estudio operativo, el sistema MED-AV resultó con un GOR de 13.5, lo cual corresponde a un incremento del 34.1% en comparación con un sistema MED convencional. Esta importante mejora se atribuye a que el sistema requiere 25.4% menos de energía de activación para lograr la misma capacidad de agua producto.

El sistema MED-AV reduce el número de componentes, dado que no se tienen precalentadores en los últimos cinco efectos. Lo anterior, se debe a que no existe gradiente de temperatura entre el agua de mar (25 °C) y la temperatura del vapor generado en cada efecto.

El vapor generado en los últimos cinco efectos de la MED-AV, es mayor respecto a una MED convencional que tiene el mismo número de efectos (14 efectos). Esto es debido a que, todo el vapor de flasheo se une al vapor generado por ebullición y se utiliza para calentar la salmuera y generar una mayor cantidad de vapor en el siguiente efecto. Dando como resultado, una capacidad de 96,033.6 L/d.

El sistema MED-AV presentó un  $GOR_G$  de 11.2, y un RR de 0.83 que representa un incremento del 11.7% y del 120% respectivamente, al compararse con un sistema MED convencional de la misma capacidad y mismo número de efectos. El incremento del RR se debe a que el alto vacío de la MED permite tener una salmuera de baja temperatura en el último efecto (4 °C) y operar a una concentración más elevada (205,220 ppm), sin tener riesgo de cristalización.

Con base a un estudio paramétrico se demostró la viabilidad técnica de un sistema innovador de desalinización de múltiple efecto (MED) que, para la producción simultánea de agua y frío, incorpora un eyector-condensador y un evaporador que le permiten operar al alto vacío (0.8 kPa) y utilizar al agua producto como refrigerante.

La configuración propuesta y condiciones de operación de alto vacío del sistema permitieron utilizar el agua producto como refrigerante y obtener una tecnología que produce 96.03 m<sup>3</sup>/d y la notable cantidad de 700 TR (2,460 kW) de enfriamiento, lo cual puede satisfacer la necesidad de agua y climatización de espacios de comunidades, hoteles, comercios, entre otros que están en las zonas costeras.

Bajo el enfoque de la segunda ley de la termodinámica, el eyector bifásico de condensación, componente esencial del sistema MED-AV, puede operar en diversas condiciones de presión y flujo másico del fluido primario (zona de factibilidad).

El procedimiento metodológico propuesto para el estudio de factibilidad termodinámica del nuevo sistema que cogenere agua y enfriamiento, permitió definir una zona de operación y las mejores condiciones de diseño del eyector-condensador. Lo anterior fue considerado la condensación total del vapor frío, una eficiencia exergética del 43% en el rango factible y con el mínimo de consumo de energía de las bombas (TPP).

El análisis de la capacidad de enfriamiento (0 a 700 TR) nos permitió observar el comportamiento del subsistema de producción de frío y eyectocondensación, encontrando que conforme aumenta la cantidad de vapor frío que debe condensar (relación de arrastre), se incrementa el consumo energético debido al aumento del flujo y presión del fluido motriz). Lo anterior hace que los indicadores de eficiencia disminuyen, no obstante, el sistema mantiene un  $COP_G$  entre 9.3–10.3, superando consistentemente a las tecnologías convencionales.

La variación de capacidad de enfriamiento con el eyector-condensador presenta el desafío de operar fuera del punto de diseño. Una solución propuesta es incorporar un variador de frecuencia en la bomba de fluido motriz para ajustar flujo y presión, en combinación con otras opciones de control, como, por ejemplo, boquilla de vórtice ajustable y aguja móvil.

La comparación con sistemas existentes confirma la superioridad del diseño propuesto: Frente a una MED con enfriamiento integrado, el sistema reduce el consumo térmico en un 13.51% y mejora el GOR y  $GOR_G$  en un 16.29% y 8.21%, respectivamente. Comparando con un sistema híbrido (MED + enfriador de compresión centrífuga), se logra una reducción del 25.43% en el consumo térmico, 11.19% en consumo eléctrico, y una mejora el  $COP_G$  (9.22) del 20.73%, junto a una disminución del 17.15% en el consumo específico global de energía ( $SEC_{G\_D}$  y  $SEC_{G\_E}$ ).

Con base a los estudios de las propuestas previas, se estudió un sistema MED al alto vacío (MED-AV) y con baja temperatura de activación (MED-AV-LTBT), el cuál presentó un  $GOR_G$  de 6.92 y un  $COP_G$  de 12.91, y una capacidad de 83.01 m<sup>3</sup>/d.

## 4.2. Trabajos futuros

Con relación a las propuestas tecnológicas planteadas en esta tesis, se propone lo siguiente:

- Se recomienda el estudio y optimización experimental del proceso de eyecto-condensación al alto vacío.
- Construir y operar una planta piloto para validar empíricamente los resultados de simulación, caracterizar el comportamiento del sistema y afinar las estrategias de control propuestas, incluyendo el uso de variadores de frecuencia con boquillas de vórtice ajustable y eyectores con aguja móvil.

- Estudiar a profundidad la tecnología de desalinización híbrida MSF/MED-LTBT eléctrico con eyecto-condensación al alto vacío y con bajo TBT para diferentes aplicaciones y escenarios de operación.
- Estudiar y proponer sistema de cogeneración, trigeneración y poligeneración que brinden los servicios de desalinización, enfriamiento, calefacción y energía eléctrica.
- Implementar y estudiar los hallazgos destacados de este trabajo doctoral: eyecto-condensación barométrica, operación al alto vacío, baja temperatura de activación (TBT) y utilización del agua producto como refrigerante, en tecnologías térmicas como múltiple etapa flash, de membranas o cualquier otro subproceso de desalinización que opere en presiones de vacío.



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## **ANEXO A**

**A novel high vacuum MSF/MED hybrid desalination system for simultaneous production of water, cooling and electrical power, using two barometric ejector condensers**

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## Article

# A Novel High Vacuum MSF/MED Hybrid Desalination System for Simultaneous Production of Water, Cooling and Electrical Power, Using Two Barometric Ejector Condensers

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**Abstract:** This work presents a novel trigeneration system for the simultaneous production of desalinated water, electrical energy, and cooling, addressing the challenges of water scarcity and climate change through an integrated and efficient approach. The proposed system combines an 8-stage Multi Stage Flash Distillation (MSF) process with a 6-effect Multiple Effect Distillation (MED) process, complemented by an expander-generator to optimize steam utilization. Cooling production is achieved through a dual ejectocondensation mechanism, which enhances energy recovery and expands operational flexibility. The system's performance was analyzed using Aspen Plus simulations, demonstrating technical feasibility across a broad operating range: 28.3 to 0.8 kPa and 68 to 4 °C. In cogeneration mode, the system achieves a Performance Ratio (PR) of 12.06 and a Recovery Ratio (RR) of 54%, producing 67,219.2 L/day of desalinated water and reducing electrical consumption by 12.03%. In trigeneration mode, it achieves a PR of 17.81 and an RR of 80%, with a cooling capacity of 1225 kW, generating 99,273.6 L/day of desalinated water while reducing electrical consumption by 3.69%. These results underscore the system's capability to significantly enhance the efficiency and capacity of thermal desalination technologies, offering a sustainable and high-performing solution for coastal communities worldwide.

**Keywords:** barometric condensation; cooling; desalination; ejector; expander; high vacuum; MED; MSF; power



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## 1. Introduction

Access to clean water, electricity, and air conditioning is essential for ensuring an adequate quality of life and promoting sustainable development, particularly in marginalized communities. However, these resources are increasingly under threat from global challenges, including the projected population growth to 9.3 billion by 2050 [1,2], drinking water scarcity, and climate change. These challenges drive a growing demand for electrical energy, which is required to power air conditioning systems as well as water purification and desalination technologies.

The outlook for water resource availability is increasingly unfavorable. Although 97% of the Earth's surface is covered by water [3], inadequate management and uncontrolled exploitation are expected to result in a 40% global water deficit by 2030 [4]. Simultaneously, global energy demand and CO<sub>2</sub> emissions have risen significantly [5], primarily due to the burning of fossil fuels for electricity generation, which powers conventional desalination and air conditioning systems. This exacerbates climate change, with projections indicating

a global temperature increase of up to 1.5 °C by 2100 [6]. Furthermore, the demand for air conditioning and refrigeration systems is predicted to rise sharply, from 4.4% in 2010 to 35% by 2050, and potentially reaching 61% by 2100 [7].

The development of systems capable of simultaneously providing multiple services—such as desalinated water, electricity, and air conditioning—offers a comprehensive solution to these challenges. By operating with high energy efficiency, these systems can significantly reduce energy consumption, lower operating costs, and promote a more environmentally sustainable approach [8,9].

Desalination is now considered a key solution to global water scarcity. The two dominant technologies in the market are Reverse Osmosis (RO) and thermal desalination. RO systems account for approximately 70% of desalination plants worldwide, while Multiple Stage Flash (MSF) and Multiple Effect Distillation (MED) systems represent 21% and 7%, respectively [10]. RO systems report an electrical energy consumption of 3.5 to 5.5 kWh/m<sup>3</sup> of desalinated water produced. In contrast, MSF and MED technologies consume 1.5 to 3.5 kWh/m<sup>3</sup> of electrical energy, in addition to 80 kWh/m<sup>3</sup> of thermal energy [11]. MSF and MED systems are more prevalent in regions where water contains high concentrations of dissolved solids, as they require less pretreatment of the feedwater and lower maintenance. Although RO systems consume up to 25% less energy than MSF and MED systems, the higher cost of pretreatment increases operational expenses, making RO less viable in these regions [12].

Continuous advancements in MSF and MED technologies have shown that lowering the top brine temperature (TBT) mitigates corrosion issues and enhances energy efficiency [13]. These systems are commonly referred to as LT-MED. Aly [14] introduced a hybrid system that extended the temperature range from a TBT of 63 °C to a bottom brine temperature (BBT) of 6 °C, achieving a record-low BBT, cost reductions, and high performance, with a Performance Ratio (PR) of 14.2 and a Gain Output Ratio (GOR) of 14.8. Subsequent studies by Shahzad et al. [15], and Saren et al. [16] investigated the hybridization of MED systems with adsorption desalination (AD) technologies, further reducing the BBT to 5 °C. This improvement increased production capacity while simultaneously decreasing the specific energy consumption (SEC).

Research on the hybridization of MED desalination systems has also focused on their integration with conventional or organic Rankine cycles (ORCs), gas turbines, combined cycles, and the utilization of industrial waste heat. These approaches aim to simultaneously produce potable water and electricity, enhancing both the efficiency and overall performance of the systems [17,18]. ORCs are particularly notable for their ability to operate with low-capacity, low-temperature thermal energy sources by utilizing low-boiling organic fluids. Baccioli et al. [19] proposed integrating waste heat recovery from industrial processes into ORC systems to power MED desalination. Two configurations were analyzed, showing improved performance based on the second law of thermodynamics, with the first configuration proving more efficient by utilizing waste heat from the ORC to activate the MED system. Furthermore, Aguilar-Jiménez et al. [20] demonstrated the technical feasibility of integrating the ORC evaporator with the first effect of the MED system, achieving simultaneous operation of both subsystems. This integration resulted in increased water production capacity and PR for the MED/ORC system, with PR improvements of 8.8%, 13%, 17.65%, and 22% for electrical generation capacities of 20, 30, 40, and 50 kW<sub>power</sub>, respectively.

The simultaneous production of desalinated water and cooling has been extensively studied. Hybridization proposals combining absorption heat pump cycles or cooling systems with MED desalination technologies are among the most commonly discussed in the literature [1,14,21,22].

Open absorption cycles integrated with desalination systems were investigated by López-Zavala et al. [23], who introduced a LiBr-H<sub>2</sub>O absorption cooling system coupled with a single-effect flash desalination system powered by solar thermal energy. Under design conditions, this system achieved a production of 838 L/day of desalinated water and a cooling capacity of 23 kW. Furthermore, they proposed a system integrating two sub-

processes: a 14-effect FLASH/MED desalination unit and a LiBr-H<sub>2</sub>O absorption cooling system. This integrated energy and mass configuration resulted in a cooling capacity of 2012 kW and a daily fresh water production of 73,569 L/day, with a coefficient of performance (COP) of 6.15 and a PR of 6.63 [24].

The development and conceptualization of trigeneration systems for the simultaneous production of water, cooling/heating, and electric power have been widely reported in the literature, integrating various technologies to achieve this. For example, Maraver et al. [25] presented a trigeneration system combining an organic Rankine cycle (ORC) powered by biomass combustion, a MED desalination system, and an absorption chiller cycle (ACC) for the production of electricity, potable water, and cooling. The analysis focused on the optimal distribution of heat generated by the ORC, with 40% allocated to the MED and ACC systems. The economic evaluation indicated a payback period of 4 to 20 years, depending on biomass resource costs. The study provides an optimistic outlook for the development of trigeneration systems based on renewable energy sources.

In the research by Ghaebi and Abbaspour [26], a novel trigeneration system is proposed, integrating a MED desalination system with thermal vapor compression (TVC), an open-cycle gas turbine as the thermal source, a heat recovery steam generator (HRSG) operating at three pressure levels, and an absorption chiller cooling system (ACC) to produce desalinated water, cooling, and electric power. Heat from the combustion chamber is distributed through the HRSG, adjusting to the three pressure levels. The results show a thermal efficiency of 84.6% and an exergy efficiency of 57.57%.

To provide a clear comparison of existing systems, Table 1 summarizes key findings, performance metrics, and economic metrics for recent state-of-the-art single, hybrid and trigeneration desalination technologies.

**Table 1.** Comparative Overview of State-of-the-Art Hybrid and Trigeneration Desalination Systems.

| Reference                | Key Findings                                                                                                                                                          | Desalination Technologies | Performance Metrics                                                            | Economic Metrics                                                                   |
|--------------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------------|--------------------------------------------------------------------------------|------------------------------------------------------------------------------------|
| Mehrpooya et al. [27]    | CSP-powered system integrates steam power, multi-effect desalination, and absorption refrigeration, emphasizing exergy efficiency and economic viability.             | MED                       | $\eta_{\psi} = 66.05\%$ .<br>$\eta_{Th} = 80.70\%$ .                           | Return Period = 5.738 years.<br>Profit = \$6.828 million/year.                     |
| Lawal et al. [28]        | Heat pump-powered HDH system achieves simultaneous water production and cooling, with feasibility assessed through parametric and economic analysis.                  | HDH                       | PR = 4.07.<br>RR = 4.86%.<br>COP = 4.85.<br>SEEC = 160.16 kWh/m <sup>3</sup> . | Water Cost: \$10.68–\$20.39/m <sup>3</sup> .                                       |
| Abdelhay et al. [29]     | Solar-driven polygeneration integrates MED desalination and absorption refrigeration with a steam Rankine cycle, focusing on thermodynamic and economic optimization. | MED                       | $\eta_{\psi} = 23.95\%$ .<br>$\eta_{\psi, Losses} = 82.42\%$ in PTC.           | Water Cost: \$1.247/m <sup>3</sup> .<br>Cooling Cost = \$0.003/kWh.                |
| Ali et al. [30]          | Combines adsorption desalination (AD) with humidification-dehumidification (HDH) and ejectors for energy reuse, validated through energy and cost modeling.           | HDH<br>AD                 | SDWP = 83.1 m <sup>3</sup> /ton.<br>PR = 2.7.                                  | Water Cost = \$1.49/m <sup>3</sup> (solar),<br>\$0.54/m <sup>3</sup> (waste heat). |
| López-Zavala et al. [31] | Novel MED system uses an ejector for vapor condensation and cooling, validated through simulation for coastal applications.                                           | MED                       | PR = 11.96.<br>COP = 4.7.<br>COP <sub>global</sub> = 9.325.<br>RR = 53%        | -                                                                                  |
| La Torre et al. [12]     | High-vacuum MED system with barometric ejector improves efficiency and production via pressure and temperature cascading.                                             | MED                       | PR = 34.1% higher than conventional MED.<br>RR = 83%                           | -                                                                                  |

Finally, this work presents a novel trigeneration system for the simultaneous production of water, cooling, and electric power, integrating a hybrid MSF/MED desalination and cooling subsystem. This system proposes the energetic and mass integration of two thermal desalination technologies: Multi Stage Flash (MSF) with 8 stages and Multi Effect Distillation (MED) with 6 effects. The technological contribution of this hybrid MSF/MED system lies in the extension of the operating range, with a top brine temperature (TBT) of 68 °C and a bottom brine temperature (BBT) of 4 °C in the sixth effect. The elimination of preheaters in the MED section allows all generated steam to be used for brine evaporation in the subsequent effects. Additionally, an electrical expansion-generation subsystem is integrated, which extracts a fraction of the mass flow from the first flash stage to generate electricity and meet the system's electrical demand. The expanded steam is reinserted into the first effect of the MED section, increasing capacity and efficiency in both desalination and cooling. Furthermore, a condensation process using two barometric ejectocondensers is employed instead of a conventional condenser, facilitating the extraction and adaptation of cold steam to produce desalinated water. The 4 °C product water stream from the final MED effect is then used to generate a cooling effect, further enhancing system efficiency.

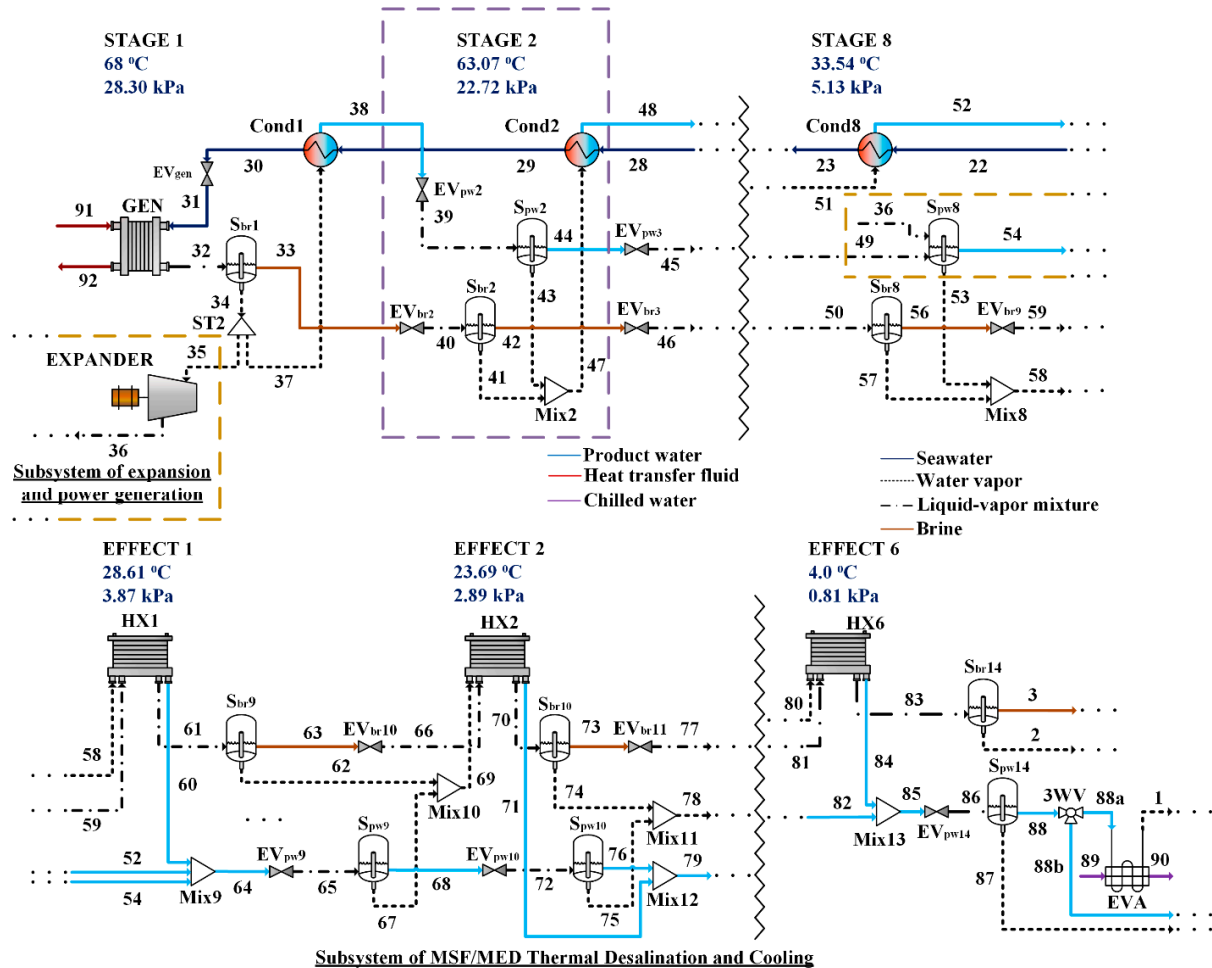
The proposed trigeneration system introduces an innovative technology that simultaneously produces desalinated water, cooling, and electrical power. Key innovations include the elimination of seawater preheaters in the MED section, improving overall system efficiency, and the integration of components that optimize energy use. This configuration provides a sustainable approach to meeting water, cooling, and electricity demands simultaneously.

## 2. System Description

Figure 1 shows the proposed trigeneration system, which consists of: (1) the hybrid MSF/MED thermal desalination and cooling subsystem, (2) the expansion and power generation subsystem, (3) the product water extraction and adaptation subsystem, and (4) the motive fluid cooling and brine dilution subsystem.

The operation of the hybrid MSF/MED thermal desalination and cooling subsystem begins when the seawater stream (18b) is drawn from the Seawater Tank and pressurized to the transport pressure (3 to 5 bar) by the hydraulic pump ( $P_{SW2}$ ), exiting as stream (22). This is introduced to the condenser ( $Cond_8$ ), where it absorbs the latent heat of condensation of the water vapor stream (51), preheating in the process. Then, the seawater stream (23) passes through the condensers of the subsequent stages, gradually increasing its temperature by acquiring the latent heat of condensation. This process continues until it reaches the maximum preheating temperature (65.5 °C) in the condenser ( $Cond_1$ ), exiting as the seawater stream (30). Subsequently, it is introduced to the expansion valve ( $EV_{gen}$ ), where a sudden pressure drop is performed until it reaches the saturation pressure of the TBT of the MSF/MED hybrid desalination subsystem. The stream (31) then enters the generator, where it receives both sensible and latent heat from the heat transfer fluid, which can come either from the solar collector field (stream 91) or from another thermal source. At the generator (GEN) outlet, the seawater stream (32) is in a two-phase (liquid-vapor) state and is introduced into a liquid-vapor separation chamber ( $S_{Br1}$ ), where it is divided into two streams: the concentrated brine (33) and the vapor (34). The concentrated brine stream (33) is directed to the second flash distillation stage, where it is introduced into the expansion valve ( $EV_{Br2}$ ). This valve suddenly reduces the pressure to the operating condition of stage 2, causing the flashing effect and steam generation, which leads to the concentrated brine stream (40) being in two phases. In turn, the mass flow of the water vapor stream (34) is divided by the splitter ( $ST_2$ ) which directs a fraction of the mass flow to the expander (stream 35) and another to the condenser ( $Cond_1$ ) (stream 37). The water vapor stream (37) is completely condensed in the condenser ( $Cond_1$ ) and then directed to the second flash distillation stage, where it is introduced into an expansion valve ( $EV_{PW2}$ ) to suddenly reduce its pressure to the operating condition of stage 2, which causes the flashing effect and vapor generation, resulting in a two-phase product water stream (39). The flashing vapors generated by the sudden pressure drop in the brine and product water

of stage 2, are directed to their respective liquid-vapor separators ( $S_{PW2}$ ) and ( $S_{Br2}$ ), and then mixed in the mixer ( $Mix_2$ ) and subsequently condensed in the condenser ( $Cond_2$ ). The liquid streams of brine and product water are directed to the expansion valves of the next stage ( $EV_{Br3}$ ) and ( $EV_{PW3}$ ) to repeat this process successively until the eighth flash distillation stage.



**Figure 1.** Subsystem Diagram: MSF/MED Thermal Desalination and Cooling, Expansion and Power Generation.

At the end of the eighth flash distillation stage, multiple distillation effects (MED) are initiated. MED effects involve the use of heat exchangers for the evaporation or release of water from brine streams, using the water vapor generated in a previous effect. In addition, two liquid-vapor separators are used in each effect; one for the product water and one for the concentrated brine, as well as two expansion valves. It is important to note that, in the multiple distillation effects, due to their respective operating temperatures, seawater preheating equipment cannot be used, which is characteristic of high vacuum systems. The water vapor that activates the MED section is obtained from the sudden expansion of the brine and product water from stage eight of the MSF section, where the two-phase product water stream (49) enters the liquid-vapor separator ( $S_{PW8}$ ) and splits into two: a vapor stream (53) and a product water stream (54).

The two-phase concentrated brine stream (50) enters the liquid-vapor separator ( $S_{Br8}$ ), where it is divided into a vapor stream (57) and a brine stream (56). The streams (53) and (57) are mixed in the mixer ( $Mix_8$ ) and exit as stream (58), which will activate the first MED effect by releasing its latent heat and condensing in the heat exchanger ( $HX_1$ ), leaving as stream (60).



The concentrated brine stream (56) enters the expansion valve ( $EV_{Br9}$ ), where this valve suddenly reduces the pressure to the operating conditions of effect 1, in addition to causing a flashing effect and the generation of steam, this pressure reduction also leads to a decrease in the temperature of the stream (59). The two-phase stream (59) is directed to the heat exchanger ( $HX_1$ ) where it receives the latent heat of condensation from the steam streams (58). This heat exchange is possible since the aforementioned vapor stream (stream 58) are at a higher temperature than the two-phase brine stream (59), maintaining a minimum approach temperature of ( $\Delta T_{min} = 2.5\text{ }^{\circ}\text{C}$ ) to carry out the heat transfer.

After exiting the condenser ( $Cond_8$ ), the product water stream (52) is directed to the mixer ( $Mix_9$ ) together with the product water streams (54) and (60), exiting as the product water stream (64). The product water stream (64) enters the expansion valve ( $EV_{PW9}$ ), where the pressure is suddenly reduced to effect 1 operating conditions. Simultaneously, the two-phase concentrated brine stream (61) is directed into the liquid-vapor separator ( $S_{Br9}$ ) and splits into two: a vapor stream (62) and a concentrated brine stream (63), where the latter enters the expansion valve ( $EV_{Br10}$ ), where the pressure is suddenly reduced to the operating conditions of effect 2. This process is repeated sequentially until the sixth multiple distillation effect is reached. The expansion and power generation subsystem begins by receiving a fraction of the vapor mass flow (stream 35) from the brine vapor-liquid separator ( $S_{Br1}$ ). The vapor stream (35) is then introduced into the volumetric expander, where it is drawn in and admitted into an expansion chamber. During this expansion process, the specific volume of the vapor increases ( $v_{36} > v_{35}$ ), resulting in a simultaneous decrease of its pressure and temperature. The study of the expansion subsystem details two different modes of operation, characterized by working with different discharge pressures. In mode 1, the specific volume of the stream (35) is expanded to reach the operating pressure of MED effect 1, while in mode 2 it is expanded to reach the operating pressure of MED effect 6. Finally, the subsystem concludes with the insertion of the two-phase stream (36) in the liquid-product water vapor separator corresponding to the effect in which the outlet pressure of the volumetric expander coincides. By reinserting the two-phase stream (36), the vapor mass flow of that stream is combined with the vapor mass flow fraction of the two-phase stream (65) within the vapor-liquid separator ( $S_{PW9}$ ). This results in an increase in the mass flow in the vapor stream (67) and, consequently, in a higher heat transfer within the exchanger ( $HX_2$ ), which causes a higher generation of water as vapor from the two-phase sea brine stream (66). The described phenomenon is repeated successively in the subsequent MED effects, which increases the product water generation and cooling capacity of the proposed trigeneration system.

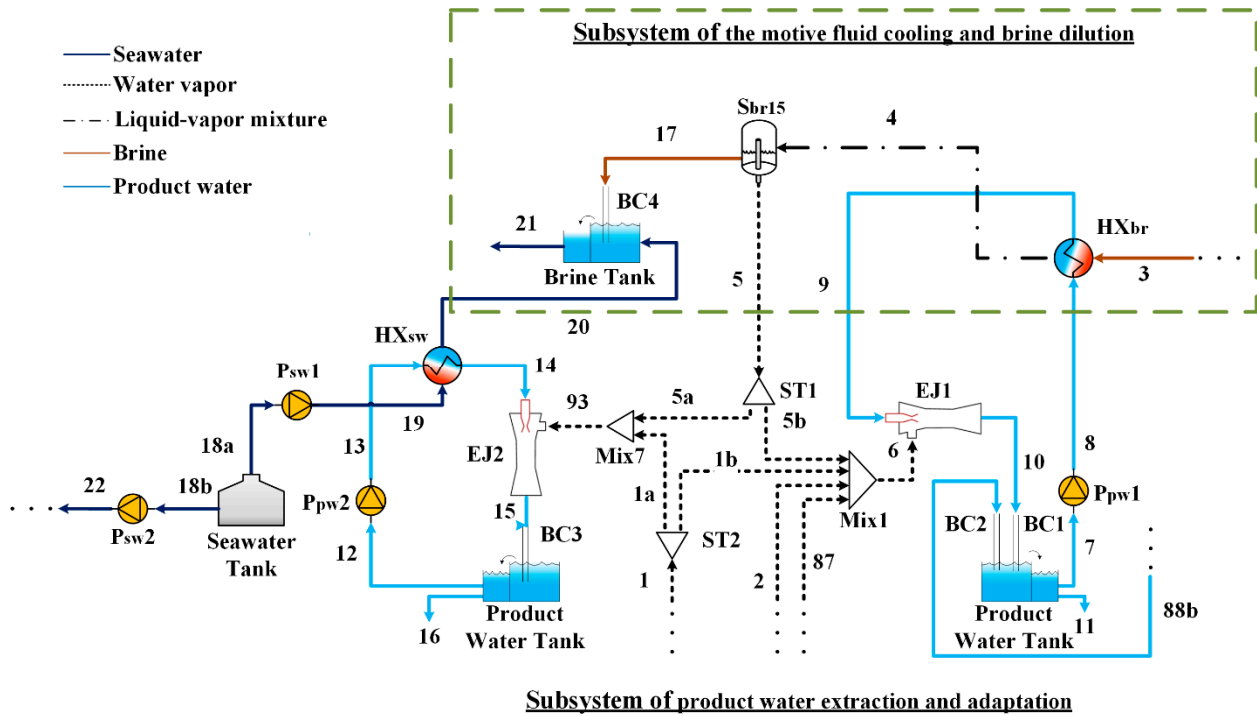
It is worth mentioning that the kinetic energy of the fluid (stream 35) is extracted by the expansion process and converted into mechanical work. This mechanical work is transferred to the rotary motion by the rotor to the shaft, then the shaft work is channeled to an electric generator for final conversion into usable electrical energy. The electrical energy produced by the expander is used for the self-consumption of the proposed system.

After leaving the sixth effect, the product water stream (84) is mixed in the mixer ( $Mix_{13}$ ) with the accumulated product water stream (82) from previous distillation stages and effects, resulting in the final product water stream (85). The latter passes to a 3-way valve (3WV), whose main function is to control the direction and/or division of the product water stream (85). It is important to note that the product water stream (88a) is used as the cooling fluid because it is in a saturated state, with a temperature of  $4\text{ }^{\circ}\text{C}$  and a pressure of 0.81 kPa. The supply of the cooling fluid to the (stream 88a) is conditioned by the 3-way valve (3WV), allowing its partial or total use, depending on the cooling needs required by the user. On the other hand, the product water stream (88b) is directed to a barometric column (BC2) where its pressure is increased and discharged into the product water tank. The vapor streams (1) and (2) at low temperature and vacuum pressure are conducted to the ejector condenser (EJ1) for condensation by pressure increase.

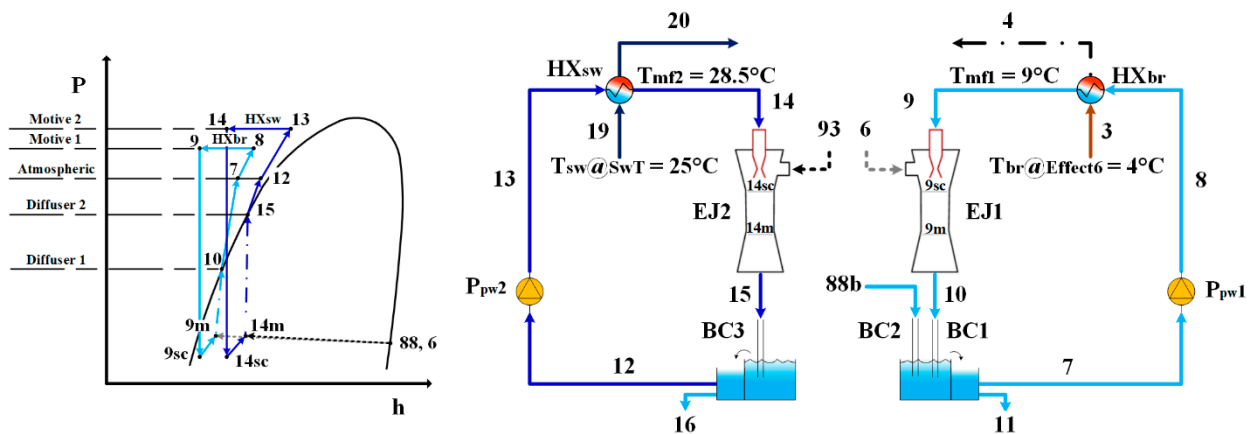
Figures 2 and 3 show the product water extraction and adaptation subsystem, which includes two ejectocondensers; one that cools the motive fluid with brine and the other



with seawater. The principle of operation of the ejectocondenser, like a conventional ejector, begins with the motive fluid entering the ejector (streams 9 and 14), where it passes through the convergent-divergent motive nozzle, increasing its velocity and reducing its pressure (streams 9sc and 14sc). This process draws and entrains the secondary fluid (6 and 93) into the ejector suction chamber. Subsequently, both streams enter the mixing chamber, where they combine, resulting in the transfer of momentum, heat, and mass, exiting as streams (9m and 14m).



**Figure 2.** Subsystem diagrams: the product water extraction and adaptation, and the motive fluid cooling and brine dilution.



**Figure 3.** P-h diagram of the Product Water Extraction and Adaptation Subsystem.

Inside the mixing chamber of the ejector, a shock wave is produced due to the deceleration of the mixed stream, resulting in an increase in pressure and temperature of the mixture, all as a result of the transport phenomena mentioned above. Subsequently, the mixture of motive and suctioned fluid (also called primary and secondary) enters the diffuser, where, by increasing the cross-sectional area of the ejector, its velocity decreases, causing a pressure increase and thus achieving total condensation of the water vapor.

Finally, the condensed mixture (10 and 15) is directed towards the barometric columns (BC1 and BC3) because it is under vacuum pressure, before being deposited in the product water storage tank at atmospheric pressure.

The ejectocondensation process causes a temperature increase in the mixed streams at the outlet of the ejector due to the transfer of condensation heat from the secondary fluid to the primary fluid. This increase in temperature causes energy accumulation in the product water storage tank, from which the motive fluid is extracted. To counteract the temperature increase, the heat exchangers ( $HX_{Brine}$ ) and ( $HX_{SW}$ ) are used in their respective condensation circuits. The heat exchanger ( $HX_{Brine}$ ) uses the concentrated brine stream (3) as a cold stream (4 °C), which makes it possible to reduce the temperature of the motive fluid stream (9) to 9 °C. On the other hand, the heat exchanger ( $HX_{SW}$ ) uses the seawater stream (19) as the cold stream (25 °C), allowing a temperature of the motive fluid stream (14) of 28.5 °C. The two-phase brine stream (4) is split into two in the vapor-liquid separator ( $S_{Br15}$ ). The concentrated brine stream (17) is directed to the barometric column (BC4), to increase its pressure and be subsequently deposited in the brine dissolution tank, where it joins with the seawater stream (20). The steam stream (5) enters the flow splitter (ST1) to control the steam mass flow to be condensed in both ejectocondensers. The ejectocondenser (EJ1) condenses the vapor stream (6), which consists of the sum of the vapors coming from the evaporator (1), the sixth effect (2) and the vapor stream (5b). On the other hand, the ejectocondenser (EJ2) is in charge of condensing the remaining vapor (88) that was not introduced in the first ejectocondenser.

#### *Main Characteristics of the System*

- Integration of Advanced Thermal Desalination Technologies

The system combines an 8 stage Multi Stage Flash (MSF) process and a 6 effect Multiple Effect Distillation (MED) process, operating under high vacuum conditions. With MSF pressures ranging from 28.3 to 5.1 kPa and MED pressures from 3.8 to 0.8 kPa, the system enhances desalination efficiency and expands operational capabilities compared to conventional designs.

- Dual Barometric Ejectocondensation Process

Two two-phase ejectors operating at different temperatures extract and condense low-temperature steam from the final MED effects, improving energy recovery and efficiency. The concentrated brine stream acts as a heat dissipating fluid in the first ejectocondensation circuit, further enhancing water production.

- Elimination of Preheaters in the MED Section

The system eliminates seawater preheaters, as the absence of a temperature gradient allows direct use of generated steam for brine evaporation in subsequent effects. This simplifies the system, reduces heat losses, and increases water production capacity.

- Expansion-Electrical Generation Subsystem

An expander-generator subsystem uses stage 1 flash steam to produce electrical power, with the expanded steam reintroduced into the MED effects. This dual-use of steam optimizes energy integration and supports the system's trigeneration capabilities.

- Efficient Cooling Mechanism

The final product water, in a saturated state at 4 °C, is used as a cooling fluid for the evaporator, minimizing the need for external cooling resources and increasing system efficiency.

- Control of Steam Generation and Brine Concentration

The system incorporates control measures to limit steam generation and prevent calcium sulfate ( $CaSO_4$ ) precipitation, ensuring operational stability and maintaining long-term system reliability.

### 3. Methodology

The parametric study of the trigeneration system proposed in this work focused on analyzing the integration of components such as heat exchangers, vapor-liquid separators, expansion valves, and an expander. An extensive literature review was conducted to identify the most suitable simulation software for evaluating the system's operating behavior and design conditions, thereby determining the production limits for water, cooling, and power.

#### 3.1. Software Simulation Process

The technical feasibility of the proposed system was evaluated using Aspen Plus simulation software, which enables the development, operational analysis, and economic assessment of chemical and petrochemical processes. The software includes extensive databases of thermochemical properties for substances and compounds, facilitating the calculation of the thermodynamic states of the study streams. Its modular architecture allows seamless interaction between specialized blocks, such as heat exchangers and separators, making it ideal for representing and simulating the thermodynamic processes and phenomena relevant to the operational study of the subsystems in this work. Figure 4 illustrates the general methodology used for the simulation process in Aspen Plus.

The simulation process for the proposed system begins with the calculation of thermodynamic properties using the selected methods: IAPWS-95 for water and Electrolyte for streams and modules interacting with seawater. These calculations are applied to the process streams and key modules (HEAT X, PUMP, FLASH2, HEATER, FSPLIT, MIXER, VALVE, COMPR) to ensure precise and reliable results. Subsequently, the system configuration is defined, and the simulation is carried out to model the interactions and behavior of the trigeneration system under specific operating conditions.

To ensure numerical stability and robust outputs, convergence blocks with iterative variables are employed throughout the simulation process. The outcomes of these variables are recorded and thoroughly analyzed to assess system performance and validate its operation against the predefined design criteria.

Table 2 presents the names of the system components, the corresponding Aspen Plus blocks, the symbology, and the general equations applied in the system's construction.

**Table 2.** Components, blocks in aspen plus and general equation of the proposed system.

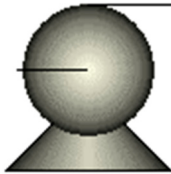
| Component                                                                                                                                                                        | Block in Aspen Plus | Symbology                                                                           | General Equation                                                                                                                                                                                                                                                                                                                                                                                                                             |
|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------|-------------------------------------------------------------------------------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| <ul style="list-style-type: none"> <li>Generator (Gen)</li> <li>Evaporator (EVA)</li> <li>Exchanger 1 (HX<sub>Br</sub>)</li> <li>Exchanger 2 (HX<sub>SW</sub>)</li> </ul>        | HEAT X              |  | <ul style="list-style-type: none"> <li><math>\dot{m}_{31}h_{31} + \dot{m}_{88}h_{88} = \dot{m}_{32}h_{32} + \dot{m}_{89}h_{89}</math></li> <li><math>\dot{m}_{85a}h_{85a} + \dot{m}_{86}h_{86} = \dot{m}_1h_1 + \dot{m}_{87}h_{87}</math></li> <li><math>\dot{m}_3h_3 + \dot{m}_8h_8 = \dot{m}_4h_4 + \dot{m}_9h_9</math></li> <li><math>\dot{m}_{13}h_{13} + \dot{m}_{19}h_{19} = \dot{m}_{14}h_{14} + \dot{m}_{20}h_{20}</math></li> </ul> |
| <ul style="list-style-type: none"> <li>Pump 1 (P<sub>PW1</sub>)</li> <li>Pump 2 (P<sub>PW2</sub>)</li> <li>Pump 3 (P<sub>SW1</sub>)</li> <li>Pump 4 (P<sub>SW2</sub>)</li> </ul> | PUMP                |  | <ul style="list-style-type: none"> <li><math>\dot{W}_1 = \dot{m}_7v_7(P_8 - P_7)/\text{Eff}</math></li> <li><math>\dot{W}_2 = \dot{m}_{12}v_{12}(P_{13} - P_{12})/\text{Eff}</math></li> <li><math>\dot{W}_3 = \dot{m}_{18a}v_{18a}(P_{19} - P_{18a})/\text{Eff}</math></li> <li><math>\dot{W}_4 = \dot{m}_{18b}v_{18b}(P_{22} - P_{18b})/\text{Eff}</math></li> </ul>                                                                       |
| <ul style="list-style-type: none"> <li>Stage n</li> </ul>                                                                                                                        | FLASH2              |  | <ul style="list-style-type: none"> <li><math>\dot{m}_{in}h_{in} + \dot{Q}_{in} = \dot{m}_{vap}h_{vap} + \dot{m}_{liq}h_{liq}</math></li> </ul>                                                                                                                                                                                                                                                                                               |

Table 2. Cont.

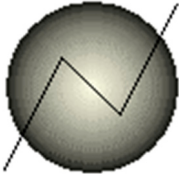
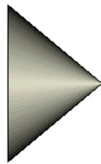
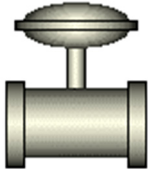
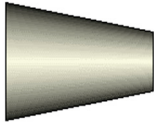
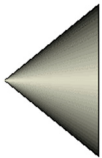
| Component                                                                                                                                                  | Block in Aspen Plus | Symbology                                                                           | General Equation                                                                                                                                                                                                                                                                                                                                                 |
|------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------|-------------------------------------------------------------------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| <ul style="list-style-type: none"> <li>Effect n</li> </ul>                                                                                                 | FLASH2              |    | <ul style="list-style-type: none"> <li><math>\dot{m}_{in}h_{in} + \dot{Q}_{in} = \dot{m}_{vap}h_{vap} + \dot{m}_{liq}h_{liq}</math></li> </ul>                                                                                                                                                                                                                   |
| <ul style="list-style-type: none"> <li>Condenser 1 (PH<sub>1</sub>)</li> <li>Condenser 2 (PH<sub>2</sub>)</li> <li>Condenser 8 (PH<sub>8</sub>)</li> </ul> | HEATER              |    | <ul style="list-style-type: none"> <li><math>\dot{m}_{29}h_{29} + \dot{m}_{37}h_{37} = \dot{m}_{30}h_{30} + \dot{m}_{38}h_{38}</math></li> <li><math>\dot{m}_{28}h_{28} + \dot{m}_{47}h_{47} = \dot{m}_{29}h_{29} + \dot{m}_{48}h_{48}</math></li> <li><math>\dot{m}_{22}h_{22} + \dot{m}_{51}h_{51} = \dot{m}_{23}h_{23} + \dot{m}_{52}h_{52}</math></li> </ul> |
| <ul style="list-style-type: none"> <li>Mixer n</li> </ul>                                                                                                  | MIXER               |    | <ul style="list-style-type: none"> <li><math>\sum \dot{m}_{in}h_{in} = \dot{m}_{out}h_{out}</math></li> </ul>                                                                                                                                                                                                                                                    |
| <ul style="list-style-type: none"> <li>Expansion Valve n</li> </ul>                                                                                        | VALVE               |   | <ul style="list-style-type: none"> <li><math>\dot{m}_{in} = \dot{m}_{out}</math></li> <li><math>h_{in} = h_{out}</math></li> </ul>                                                                                                                                                                                                                               |
| <ul style="list-style-type: none"> <li>Expander</li> </ul>                                                                                                 | COMPR               |  | <ul style="list-style-type: none"> <li><math>\dot{W}_{exp} = \dot{m}_{in}v_{out}(P_{in} - P_{out})/Eff</math></li> </ul>                                                                                                                                                                                                                                         |
| <ul style="list-style-type: none"> <li>Splitter 1 (ST<sub>1</sub>)</li> <li>Splitter 2 (ST<sub>2</sub>)</li> </ul>                                         | FSPLIT              |  | <ul style="list-style-type: none"> <li><math>\dot{m}_5h_5 = \dot{m}_{5a}h_{5a} + \dot{m}_{5b}h_{5b}</math></li> <li><math>\dot{m}_1h_1 = \dot{m}_{1a}h_{1a} + \dot{m}_{1b}h_{1b}</math></li> </ul>                                                                                                                                                               |

Table 3 details the interconnection configuration of the blocks used to represent the subsystems of the proposed system in this work, as well as their organization within the main flow diagram in Aspen Plus. The hybrid MSF/MED thermal desalination and cooling subsystems, along with the Expander-generator, are represented in the processes STAGE 1 and ACTIVATION SOURCE, STAGES 2 to 8, EFFECTS 1 to 5, and EFFECTS 6. Meanwhile, the subsystems for extraction and product water adaptation, as well as cooling of the motive fluid and brine dilution, are represented in the SUBSYSTEM C-ECB (Cooling-Ejector Condenser Barometric) process.

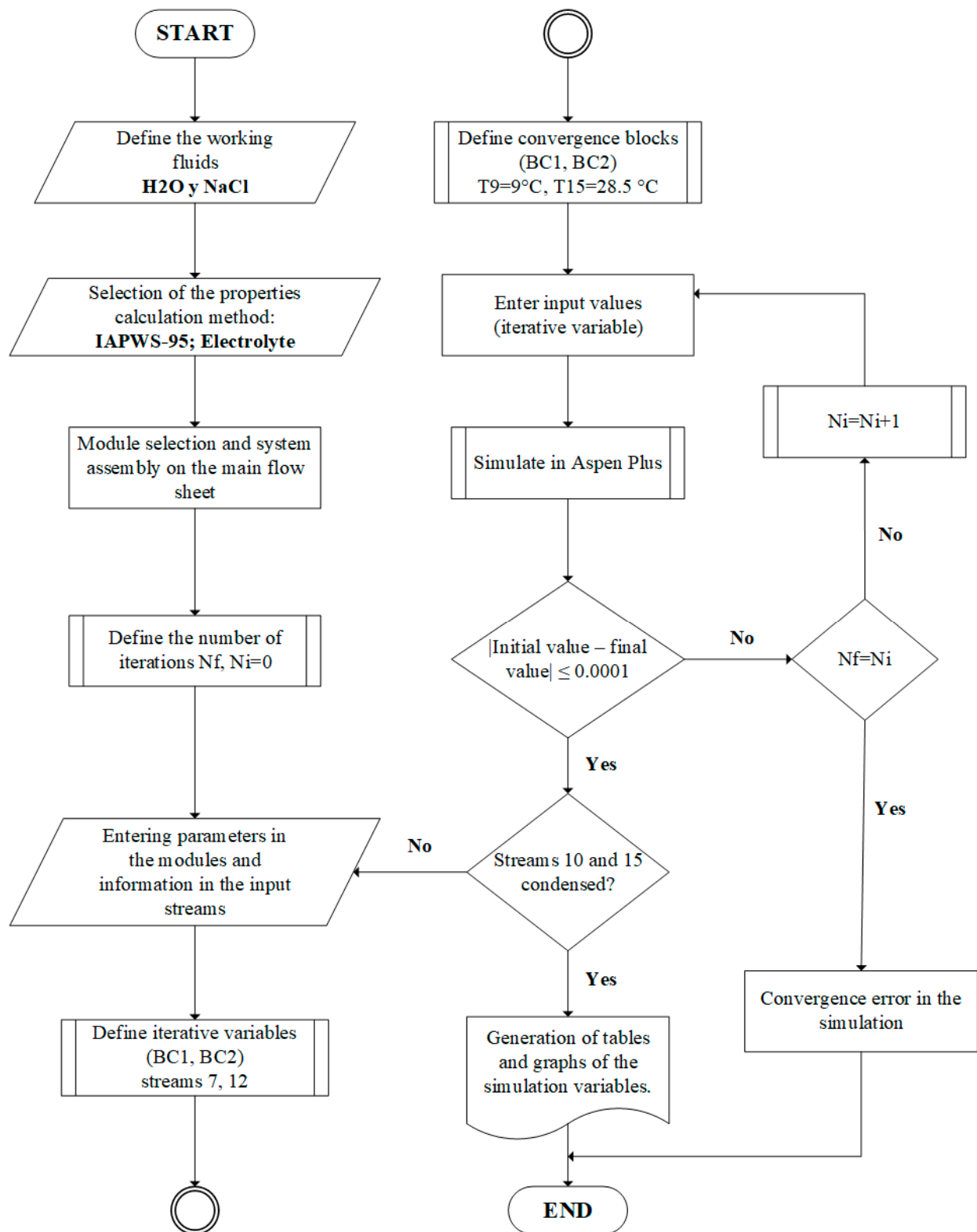


Figure 4. General methodology diagram and the simulation process in Aspen Plus.

Process

Blocks in Aspen Plus

STAGE 1 AND ACTIVATION SOURCE

Process flow diagram for Stage 1 and Activation Source. The diagram illustrates the integration of various unit operations and streams:

- Activation Source:** P<sub>gen</sub> (Generator) provides heat to the system.
- Heat Recovery:** PH<sub>1</sub> and PH<sub>tot</sub> (Heat Exchangers) facilitate heat transfer between seawater and other streams.
- Separation:** Sep<sub>gen</sub>, SEP<sub>2</sub>, SEP<sub>3</sub>, and SEP<sub>4</sub> (Separators) separate different phases (Vapor, Product water, Brine, Two-phases water).
- Heat Exchangers:** Cond<sub>1</sub> (Condenser) and ST<sub>1</sub> (Steam Turbine) are part of the heat recovery system.
- Mixing:** Mix<sub>1</sub> (Mixer) combines different streams.
- Streams:** The diagram shows the flow of Vapor, Hot water, Seawater, Product water, and Brine, along with heat streams and activation sources like P<sub>gen</sub> and P<sub>sw2</sub>.

STAGE 2-8

Process flow diagram for Stage 2-8. The diagram illustrates the integration of various unit operations and streams:

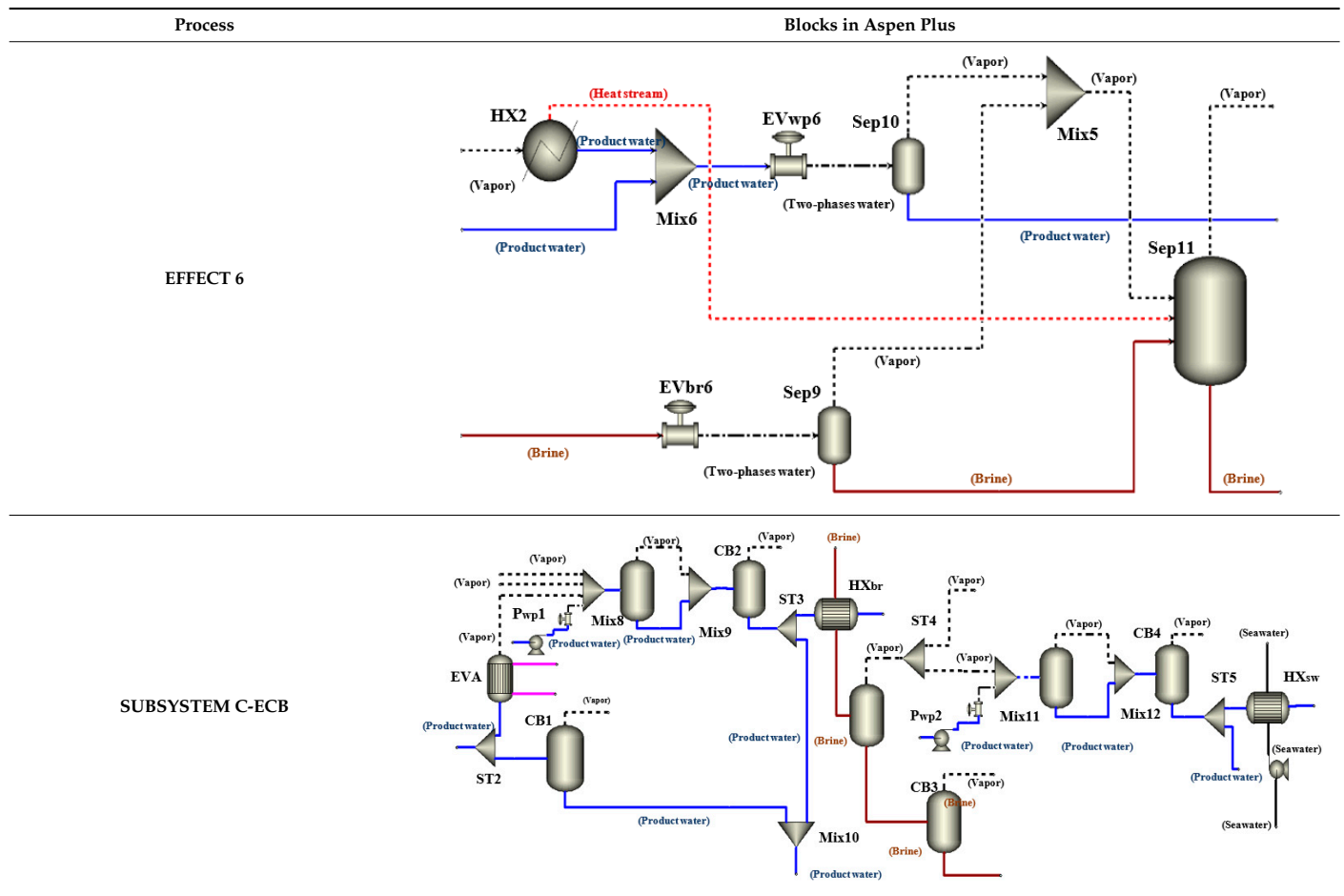
- Heat Recovery:** Cond<sub>2</sub> (Condenser) and HX<sub>1</sub> (Heat Exchanger) facilitate heat transfer between product water and other streams.
- Separation:** SEP<sub>5</sub>, SEP<sub>6</sub>, SEP<sub>7</sub>, and SEP<sub>8</sub> (Separators) separate different phases (Vapor, Product water, Brine, Two-phases water).
- Mixing:** Mix<sub>2</sub>, Mix<sub>3</sub>, and Mix<sub>4</sub> (Mixers) combine different streams.
- Expansion:** EV<sub>wp2</sub>, EV<sub>wp3</sub>, EV<sub>br2</sub>, EV<sub>br3</sub>, and EV<sub>br4</sub> (Expansion Valves) control the flow of different phases.
- Streams:** The diagram shows the flow of Product water, Vapor, Two-phases water, and Brine, along with heat streams and activation sources.

EFFECT 1-5

Process flow diagram for Effect 1-5. The diagram illustrates the integration of various unit operations and streams:

- Heat Recovery:** HX<sub>1</sub> (Heat Exchanger) facilitates heat transfer between product water and other streams.
- Separation:** SEP<sub>7</sub> and SEP<sub>8</sub> (Separators) separate different phases (Vapor, Product water, Brine, Two-phases water).
- Mixing:** Mix<sub>4</sub> (Mixer) combines different streams.
- Expansion:** EV<sub>br4</sub> (Expansion Valve) controls the flow of different phases.
- Streams:** The diagram shows the flow of Product water, Vapor, Two-phases water, and Brine, along with heat streams and activation sources.

Table 3. Cont.



### 3.2. System Parameters

In the operational study of the proposed system presented in this paper, the parameters outlined in Table 4 were defined, varying the seawater feed flow and the activation heat supplied to the generator. The seawater feed flow enters the system at 25 °C with a concentration of 35,000 ppm of NaCl, limiting the Top Brine Temperature (TBT) to 68 °C to mitigate corrosion issues, as reported by Palenzuela et al. [32].

Table 4. Operating parameters of the MSF/MED thermal desalination subsystem at high vacuum.

| Parameters                          | Units | Value  |
|-------------------------------------|-------|--------|
| No. of Flash stages                 | -     | 8      |
| No. of MED effects                  | -     | 6      |
| No. of preheaters                   | -     | 8      |
| Top brine temperature (TBT)         | °C    | 68     |
| Bottom brine temperature (BBT)      | °C    | 4      |
| $\Delta T$ stages/effects           | °C    | 4.92   |
| Feed seawater temperature           | °C    | 25     |
| Salt concentration of feed seawater | ppm   | 35,000 |

### 3.3. Basic Assumptions

Simulation in Aspen Plus, especially when dealing with recirculation streams, requires the initialization and definition of iterative variables, as described in the methodology diagram. These variables iteratively solve the simulation process to achieve convergence.

The operational simulation study of the proposed system considers the following basic assumptions, applicable to both trigeneration and cogeneration modes:

- The technical feasibility and the behavior of the proposed MSF/MED system to the high vacuum in steady state is analyzed.
- The hybrid MSF/MED to high vacuum thermal desalination subsystem uses a TBT of 68 °C and a BBT of 4 °C.
- The heat exchange between streams considers a minimum approach temperature ( $\Delta T_{\min}$ ) of 2.5 °C. for stages and effects.
- A maximum seawater preheating temperature of  $T_{30} = 65.5$  °C is established.
- An isentropic efficiency of 85% of the hydraulic pumping equipment ( $P_{SW1}$ ,  $P_{SW2}$ ,  $P_{WP1}$ ,  $P_{WP2}$ ) is considered.
- An isentropic efficiency of 85% is considered in the expander (Expander).
- Expander discharge pressure is 5.129 kPa.
- All the condensed product water is used as refrigerant fluid, in trigeneration operation.
- The pressure drop in the equipment and piping is negligible.
- An exergy efficiency of 85% is considered in the ejectocondenser (EJ1).
- An exergy efficiency of 28% is considered for the ejectocondenser (EJ2).
- The properties of the equilibrium state: temperature of 25 °C and absolute pressure of 1 atm.

### 3.4. System Energy Performance Indicators

The selected performance indicators evaluate in single, cogenerative and trigeneration production the system's products. The equations are defined based on the energy balance by the first law of thermodynamics.

The following indicators were used in the desalination performance evaluation: Recovery Ratio (RR), Performance Ratio (PR) and global Performance Ratio ( $PR_{\text{global}}$ ). The RR indicator (Equation (1)) represents the fraction of seawater that is converted to desalinated water. To evaluate the thermal supply of the system, the PR indicator (Equation (2)) was used, which describes the ratio between the latent heat of the product water and the thermal energy supplied. In addition, another indicator, the  $PR_{\text{global}}$  (Equation (3)), is proposed, which considers the thermal and electrical energy requirement supplied to the system (La Torre et al. [12]).

$$RR = \frac{(\dot{m}_{11} + \dot{m}_{16})}{\dot{m}_{22}} \times 100 \quad (1)$$

$$PR = \frac{(\dot{m}_{11} + \dot{m}_{16}) \times (2326 \frac{\text{kJ}}{\text{kg}})}{\dot{Q}_{\text{GEN}}} \quad (2)$$

$$PR_{\text{global}} = \frac{(\dot{m}_{11} + \dot{m}_{16}) \times (2326 \frac{\text{kJ}}{\text{kg}})}{\dot{Q}_{\text{gen}} + \sum \dot{W}_{\text{Tot. power}} - \dot{W}_{\text{exp}}} \quad (3)$$

The cooling production performance of the system was analyzed using the Coefficient of performance (COP) indicator (Equation (4)), which relates the cooling capacity of the system to the energy supply (thermal and/or electrical) required. The  $COP_{\text{global}}$  equation (Equation (5)) proposed by López-Zavala et al. [31] was used to evaluate the performance of simultaneous desalination and cooling production.

$$COP = \frac{\dot{Q}_{\text{EVA}}}{\dot{Q}_{\text{GEN}} + \sum \dot{W}_{\text{eléc,tot}} - \dot{W}_{\text{exp}}} \quad (4)$$

$$COP_{\text{Global}} = \frac{\dot{Q}_{\text{EVA}} + (\dot{m}_{11} + \dot{m}_{16}) \times (2326 \frac{\text{kJ}}{\text{kg}})}{\dot{Q}_{\text{GEN}} + \sum \dot{W}_{\text{Tot. power}} - \dot{W}_{\text{exp}}} \quad (5)$$



The following indicators were used to study energy consumption: Electrical Coverage (EC), Specific Electrical Energy Consumption (SEEC), Specific Thermal Energy Consumption (STEC) and Specific Energy Consumption (SEC). Electrical Coverage describes the fraction of the system's electrical demand that is satisfied by the electrical generation of the expander (Equation (6)).

$$EC = \frac{\dot{W}_{\text{exp.}}}{\sum \dot{W}_{\text{elec,tot}}} \quad (6)$$

SEEC and STEC are defined as the ratio between the amount of energy consumed (electrical or thermal) and the water capacity produced (Equation (7)). The total specific energy consumption (SEC) is defined as the sum of the specific consumption of electrical and thermal energy (Equation (8)).

$$\text{SEEC or STEC} = \frac{\text{Total electrical and thermal energy consumption (kW)}}{\text{Water product capacity (m}^3/\text{h)}} \quad (7)$$

$$\text{SEC} = \text{SEEC} + \text{STEC} \quad (8)$$

The operating performance of the ejectocondenser was evaluated with respect to the second law of thermodynamics. The exergy content of the currents entering the ejector was quantified using (Equation (9)). On the other hand, for the thermodynamic feasibility study, (Equation (10)) was used, which determines the exergetic destruction, likewise, for the calculation of the exergetic efficiency of the ejectocondenser, (Equation (11)) was used.

$$\dot{E}x = \dot{m}[h - h_0 - T_0(s - s_0)] \quad (9)$$

$$\sum \dot{Q}(1 - \frac{T_0}{T}) + \sum \dot{E}x_{\text{in}} - \sum \dot{E}x_{\text{out}} - \sum \dot{W} - \dot{E}x_{\text{Destroy}} = 0 \quad (10)$$

$$\eta_{\psi} = \frac{\text{Exergy}_{\text{out}}}{\text{Exergy}_{\text{in}}} \quad (11)$$

#### 4. Validation of Simulation

The validation of the proposed system was performed in two parts: (1) the dual barometric ejectocondensation subprocess of the product water extraction and adaptation subsystem, and (2) thermal desalination subprocess concerning the MSF/MED subsystem to high vacuum. For the first part, the experimental work reported by Sağ et al. [33] which corresponds to the study of exergy destruction of the components of an ejecto-compression cooling system was used.

The proposed system, like the work reported by Sağ et al. [33], uses a liquid-vapor two-phase ejector in an ejectocompression cooling system. However, unlike the system proposed in this work, the author considers partial condensation at the ejector outlet as a design condition. Therefore, a comparison was made between the operating results of the experimental system and those obtained by simulation in Aspen Plus, with the objective of analyzing the operating performance and studying the partial condensation process in the two-phase ejector simulated in Aspen Plus.

As shown in Table 5, the average error of the results is less than 1.5%. When evaluating the operating performance of the system using the first law of thermodynamics, it is observed that the cooling capacity, compressor work and COP do not present errors greater than 0.5%. In addition, the operating performance of the two-phase ejector shows a maximum error of 1.829%, which demonstrates the feasibility of performing the ejector study in Aspen Plus.

**Table 5.** Comparison of operational results of the experimental study reported by (Sağ et al. [33]) and this work.

| Parameters                    | Units | Sağ et al. | This Work | Error (%) |
|-------------------------------|-------|------------|-----------|-----------|
| Condenser outlet temperature  | °C    | 57.64      | 59.02     | 2.394     |
| Evaporator outlet temperature | °C    | 11.63      | 11.03     | 5.159     |
| Ejector outlet temperature    | °C    | 13.12      | 12.88     | 1.829     |
| Ejector outlet pressure       | kPa   | 456.00     | 456.00    | 0.000     |
| Entrainment ratio             | -     | 0.65       | 0.647     | 0.767     |
| COP                           | -     | 2.12       | 2.115     | 0.236     |
| Compressor duty               | kW    | 2.11       | 2.10      | 0.332     |
| Cooling capacity              | kW    | 4.47       | 4.45      | 0.470     |
| Average duty                  |       |            |           | 1.398     |

The experimental study conducted by Sağ et al. [33] on the exergy destruction of each component of the ejectocompression cooling system was used to compare the experimental results with those generated by Aspen Plus.

Table 6 shows the exergy destruction of each component of the ejectocompression cooling system. The results show a total average error of 4.44%, highlighting the condenser as the component with the highest percentage of error, reaching 9.4%. In contrast, the ejector, which is the component of greatest interest, presents an error of 0.27%.

**Table 6.** Comparison of exergy destruction between experimental results from (Sağ et al. [33]) and simulation in Aspen Plus.

| Component               | Units | Sağ et al. | This Work | Error (%) |
|-------------------------|-------|------------|-----------|-----------|
| Condenser               | W     | 71.88      | 78.64     | 9.4       |
| Compressor              | W     | 974.3      | 917.5     | 5.82      |
| Evaporator              | W     | 87.03      | 84.78     | 2.58      |
| Expansion Valve         | W     | 0.46       | 0.5       | 8.76      |
| <b>Ejector</b>          | W     | 188.2      | 188.71    | 0.27      |
| Separator               | W     | 35.76      | 35.61     | 0.42      |
| Exergy destroyed total: | W     | 1357.63    | 1305.74   | 3.82      |
| Average error:          |       |            |           | 4.44      |

In the second part of the validation, the work presented by López-Zavala et al. [31] was used. The desalination and cooling system with barometric ejectocondenser that they describe shares similarities in the operating principles with the proposed hybrid MSF/MED to HV system. Detailed information on the key parameters and considerations for the simulation of this system is presented in Table 7.

**Table 7.** Main considerations and value ranges of the system parameters proposed by (López-Zavala et al. [31]) and the simulation in Aspen Plus.

| Parameters                          | Units | López-Zavala et al. | This Work |
|-------------------------------------|-------|---------------------|-----------|
| No. Effects                         | -     | 14                  | 14        |
| TBT                                 | °C    | 70                  | 70        |
| BBT                                 | °C    | 37                  | 37        |
| ΔP effects                          | kPa   | 3.13                | 3.13      |
| ΔT effects                          | °C    | 2.5                 | 2.5       |
| Operating temperatura range         | °C    | 70–35               | 70–35     |
| Salt concentration of feed Seawater | ppm   | 35,000              | 35,000    |
| Seawater feed water temperature     | °C    | 27                  | 27        |
| Hot source Flow rate                | kg/s  | 1.611               | 1.611     |
| Hot source temperature              | °C    | 95                  | 95        |

Table 8 shows the comparison of both simulations with respect to the desalination and cooling performance indicators, as well as the internal energy integration and heat transfer in the barometric condensation ejection subsystem. A maximum error of 0.638% is observed in the desalination and cooling performance indicators, a maximum error of 0.156% in the internal energy integration (preheating) and a maximum error of 2.8% in the barometric ejectocondensation subsystem section.

**Table 8.** Simulation validation between the theoretical results of López-Zavala et al. [31] and the simulation in Aspen Plus.

| Parameters                               | Units | López-Zavala et al. | This Work | Error (%) |
|------------------------------------------|-------|---------------------|-----------|-----------|
| Seawater temperature at condenser outlet | °C    | 32                  | 31.96     | 0.125     |
| Seawater temperature at the PH@12 outlet | °C    | 64                  | 64.1      | 0.156     |
| Product water capacity                   | kg/s  | 0.865               | 0.859     | 0.694     |
| PR                                       | -     | 11.6                | 11.53     | 0.603     |
| RR                                       | -     | 0.53                | 0.5274    | 0.491     |
| COP                                      | -     | 4.7                 | 4.67      | 0.638     |
| COP <sub>global</sub>                    | -     | 9.325               | 9.291     | 0.365     |
| Cooling capacity                         | kW    | 2042.87             | 2028.59   | 0.699     |
| Heat exchanged (condenser)               | kW    | 152.82              | 151.76    | 0.694     |
| Entrainment ratio                        | -     | 0.0025              | 0.00243   | 2.800     |
| Heat exchanged (heatsink)                | kW    | 2297.54             | 2310.49   | 0.564     |
| Average error:                           |       |                     |           | 0.47      |

## 5. Analysis and Discussion of Results

The parametric study presented in this work on the proposed trigeneration system is divided into two parts: the analysis of the system's behavior in cogeneration mode and its behavior in trigeneration mode.

### 5.1. Operational Study of the System in Cogeneration Mode

To operate in cogeneration mode, cooling production is eliminated in the MSF/MED hybrid thermal desalination subsystem under high vacuum conditions. This is achieved through the control of the 3-way valve (3WV), which diverts the condensed product water stream (88), preventing its entry into the evaporator (EVA) and thereby stopping the generation of the cooling effect in the system. The bypassed condensed product water stream (88b) is directed to the barometric column (BC2). The function of this column is to maintain the hydraulic vacuum seal of the system and increase the pressure to atmospheric levels, preparing the water for storage in the product water tank.

### 5.2. Operational Performance Results of the MSF/MED Thermal Desalination Subsystem Under High Vacuum in Cogeneration/Trigeneration Modes

The amount of heat transferred in the stages of the MSF section and the effects of the MED section, along with the heat gained by the seawater to reach the maximum preheating temperature of 65.5 °C, is presented in Figure 5, in relation to the variation in activation heat.

In the MSF section, heat transfer occurs only in the first stage due to the presence of the generator, which produces the steam necessary to reach the preheating temperature. The excess steam generated when operating above  $Q_{\text{gen}} = 50$  kW is used for the expander to produce electrical energy. As activation heat increases, the heat transferred for seawater preheating decreases, since the required heat is supplied by the mixture of flashed vapors from the product water and brine within the MSF. More steam generated in the generator reduces the brine flow to the second stage, decreasing the amount of flashed steam and, consequently, the heat transfer for preheating. This results in an increased steam flow to the condenser (Cond<sub>1</sub>) to achieve the maximum preheat temperature.

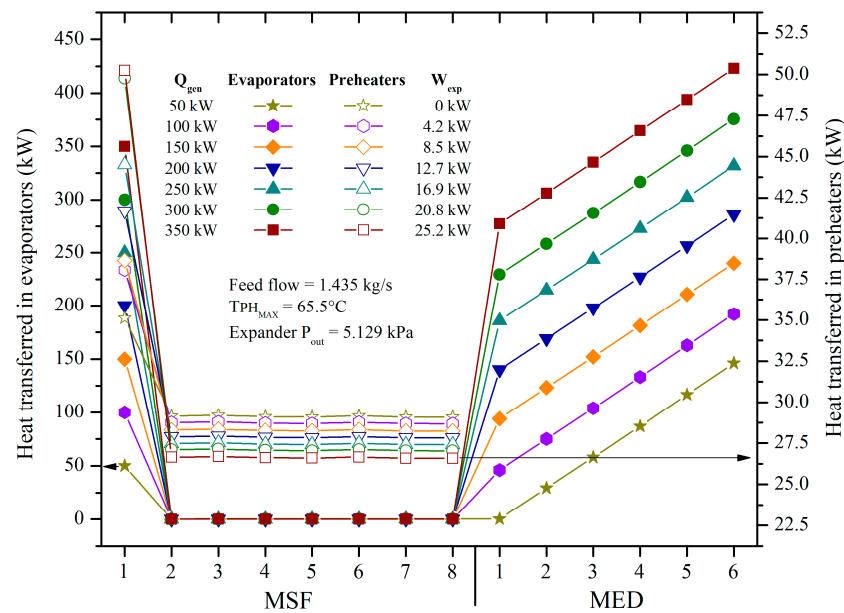


Figure 5. Preheat/heat transferred in stages/effects within the MSF/MED HV thermal desalination system.

After the MSF section, the flashed vapors are no longer used to preheat the seawater due to the lack of a minimum temperature ( $\Delta T_{\min} = 2.5^\circ\text{C}$ ) difference. Instead, they are used to boil off the brine in the first effect of the MED section, gradually increasing heat transfer. Additionally, the reinjection of steam expanded by the expander into the eighth stage of the MSF increases the mass of available steam, enhancing heat transfer in the MED section and creating a cascading effect throughout the other effects in the system.

The vapors generated by boiling and flashing in each stage or effect of the thermal desalination subsystem are shown in Figure 6. In the first stage of the MSF section, vapor is primarily produced through boiling to supply the condenser ( $\text{Cond}_1$ ) and reach the maximum preheating temperature ( $65.5^\circ\text{C}$ ). Subsequently, the desalination capacity decreases due to the flashing of water vapor caused by the pressure differential between stages.

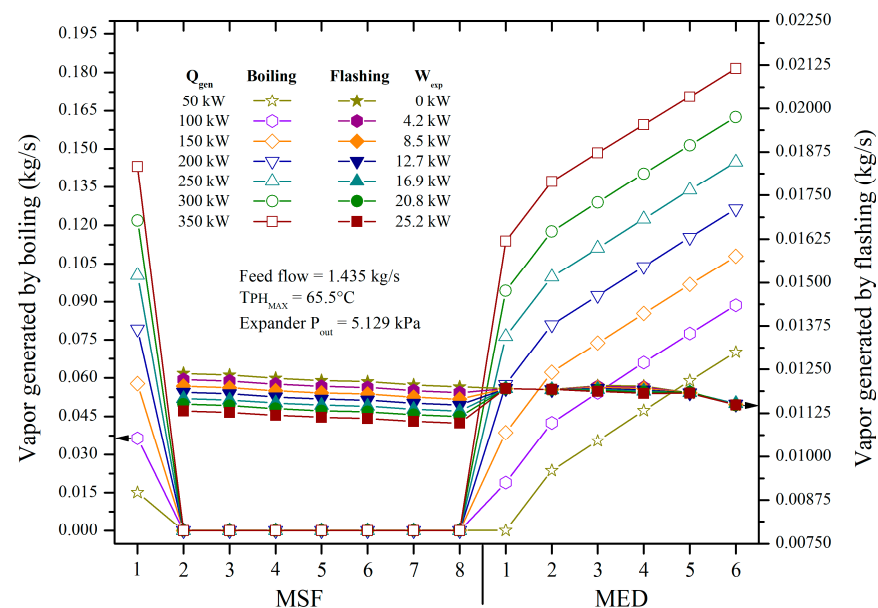
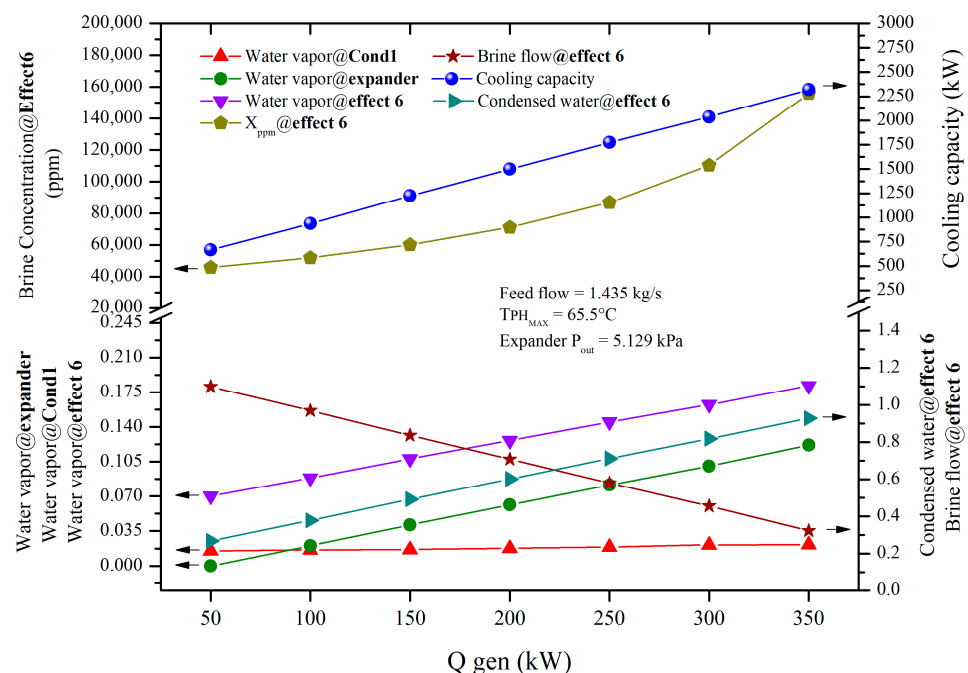


Figure 6. Boiling/flash steam generated in the stages/effects within the MSF/MED at high vacuum thermal desalination system.

By increasing the activation heat, more steam is generated in the generator. The excess steam produced, which is not required to feed the Condenser (Cond<sub>1</sub>) in stage 1, is used as a working fluid for the expander and is reintroduced in the eighth stage. The MED section is activated by utilizing the expanded steam from the expander, along with the vapors generated by the flashing of product water and the brine flow from the eighth stage. This creates a cascading effect, where the vapors generated by flashing are no longer used for preheating, but instead for boiling, allowing more water to be extracted from the brine stream as it passes through the effects. The increase in heat added to the generator leads to an increase in product water production capacity and, consequently, to a higher cooling capacity of the system in trigeneration operation.

Figure 7 shows the production capacity of condensed water, water vapor directed to the expander and to Cond<sub>1</sub>, brine flow, and water vapor produced by flashing and boiling in effect 6. The internal variables in Figure 7 refer to the vapors generated in the first stage of the desalination subsystem, specifically the vapors from the condenser (Cond<sub>1</sub>) and the expander, as well as the brine flow at the outlet of the thermal desalination subsystem. The vapors (Cond<sub>1</sub> and Expander) tend to increase as more heat is added to the generator. This occurs because the water vapor directed to Cond<sub>1</sub> supplies the remaining heat required to reach the maximum preheating temperature of the seawater, which had previously been preheated by the vapors generated through flashing in stages 2 to 8, which released their condensation heat for this purpose. As a result, a higher release of vapor from the brine leads to a reduced brine flow directed to the MSF section, causing an increase in condenser (Cond<sub>1</sub>). On the other hand, the steam flow to the expander increases because the water vapor originating from the generator, which is not used by condenser (Cond<sub>1</sub>), is directed to the expander.



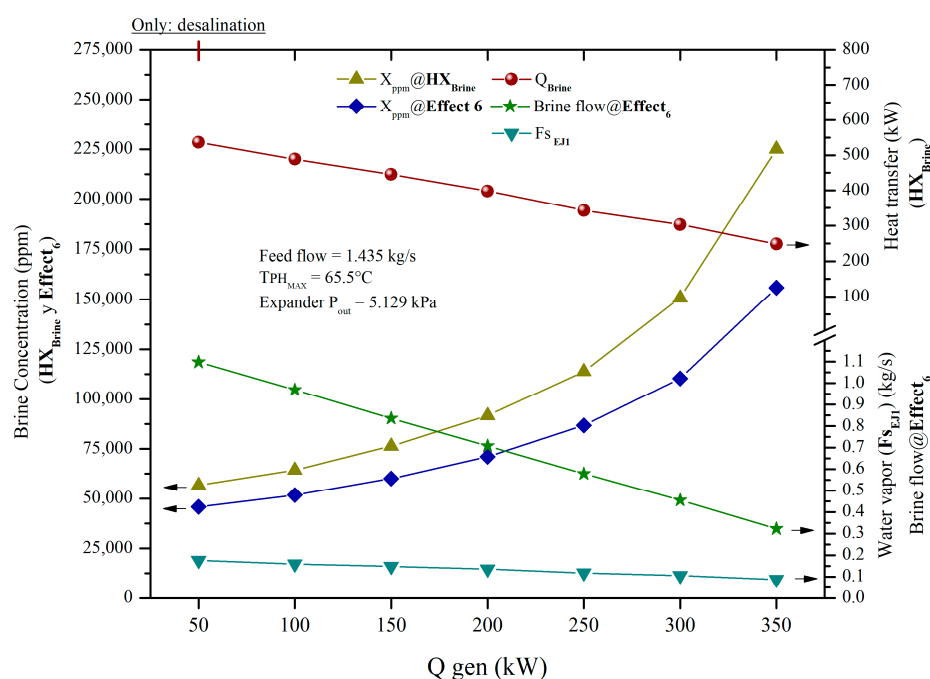
**Figure 7.** Effect of activation heat variation on internal variables of the thermal desalination subsystem.

The water vapor and condensed product water from effect 6 tend to increase, as shown in Figure 7. This occurs because the expanded steam at the expander outlet mixes with the steam from the eighth stage, and this mixture is used to boil the brine flow in the first heat exchanger of the MED section. This process gradually increases the desalination capacity as the steam flow from the expander rises, leading to enhanced heat exchange. Finally, the water vapor extracted from the brine in cogeneration or trigeneration operation modes tends to decrease as heat is added to the generator. This is due to the reduction in heat

dissipated in the brine exchanger ( $HX_{Brine}$ ), as the heat transfer depends on the brine flow (stream 3) and the amount of steam to be condensed in the ejectocondenser (EJ1).

### 5.3. Operational Performance Results of Subsystems: Product Water Extraction, Adequacy, and Cooling of the Motive Fluid in Cogeneration Mode

Figure 8 shows the brine concentrations at the outlet of the thermal desalination subsystem and the heat exchanger ( $HX_{Brine}$ ). The brine concentration ( $X_{ppm@effect6}$ ) in the brine stream (3) at the MED section outlet increases due to the intensified boiling of water vapor in the generator and the higher steam flow to the expander, used in effect 1 ( $HX_1$ ). Similarly, the brine concentration ( $X_{ppm@HXBrine}$ ) also rises. This is attributed to the increased brine concentration at the MED section outlet and the dissipation of condensation heat from the motive fluid by the saturated brine stream (3). This process causes water boiling and a subsequent rise in concentration within the heat exchanger ( $HX_{Brine}$ ), with an average increase of 32.1% compared to the concentration at the outlet of effect 6, reaching a maximum of 225,087 ppm when operating at  $Q_{gen} = 350$  kW. Furthermore, heat transfer in the heat exchanger ( $HX_{Brine}$ ) and secondary flow ( $F_{sEJ1}$ ) decrease due to the reduced brine flow, diminishing the capacity to remove condensation heat. This lowers the condensing capacity of the ejectocondenser (EJ1) from 0.1758 to 0.0869 kg/s, redirecting the remaining cold vapor to the second ejectocondenser (EJ2).

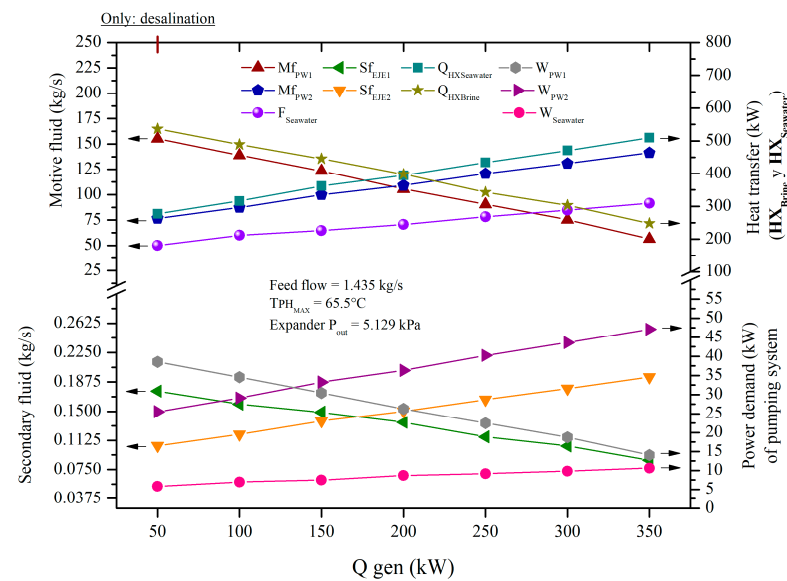


**Figure 8.** Effect of activation heat variation on brine concentration, heat exchanger, brine flow and secondary flow ( $Sf_{EJ1}$ ), in cogeneration mode.

Cold vapors from the brine separator ( $S_{Br15}$ ) and effect 6 are used as secondary flows, while product water from a reservoir tank serves as the motive flow in the ejectocondenser. Figure 9 shows a decrease in the motive flow ( $Mf_{PW1}$ ) from 155 to 56.56 kg/s in ejectocondenser (EJ1), and an increase in the motive flow ( $Mf_{PW2}$ ) from 75 to 125 kg/s in (EJ2), along with an increase in the seawater flow ( $F_{Sea}$ ) from 50 to 91.58 kg/s. This is due to the reduction in secondary flow ( $Sf_{EJ1}$ ), driven by a decrease in brine flow, which reduces the demand for motive flow in (EJ1) and directs more steam to be condensed in (EJ2). The increase in secondary flow ( $Sf_{EJ2}$ ) requires a higher motive flow, increasing the electrical demand of the hydraulic equipment ( $P_{PW2}$ ) by 10.77%, while the reduction in ( $Sf_{EJ1}$ ) decreases the demand on ( $P_{PW1}$ ) by 15.33%. Heat exchange follows the trends of the motive



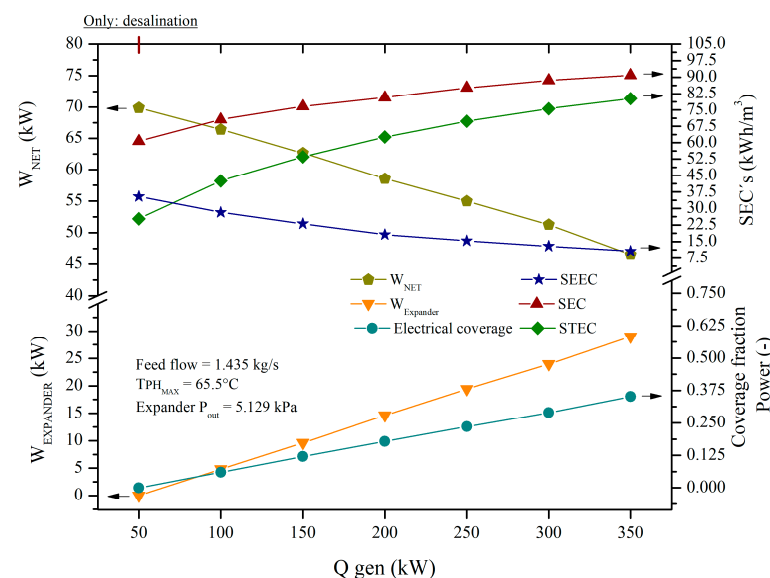
flow: the heat exchanger ( $HX_{Brine}$ ) shows a reduction from 526.61 to 248.12 kW, while the heat exchanger ( $HX_{Sea}$ ) increases from 277.57 to 509.47 kW.



**Figure 9.** Influence on the variation of the activation heat in the motive flows, heat exchanger, secondary flows and electrical demand of the pumping system, in cogeneration mode.

#### 5.4. Overall Operating Performance Results in Cogenerative Mode

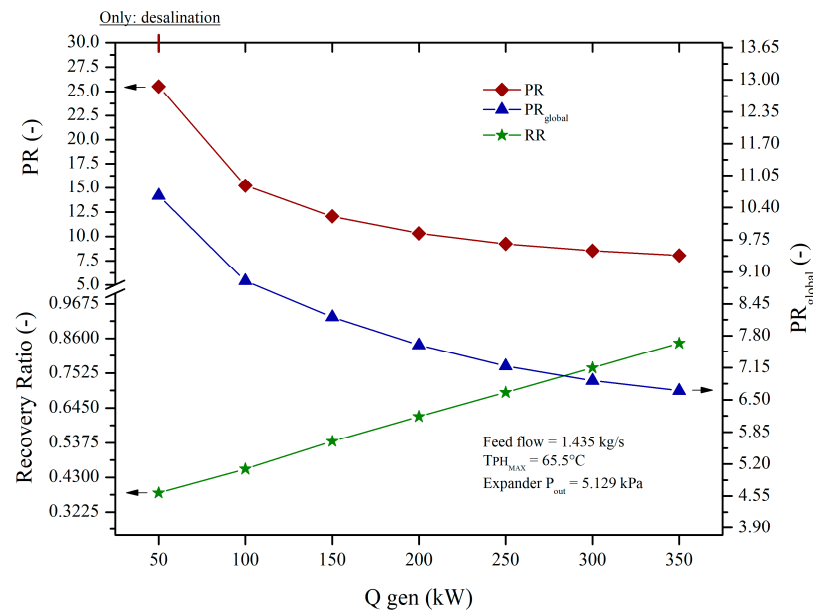
As shown in Figure 10, increasing the activation heat boosts the amount of steam produced in the generator, which is used as the working fluid by the expander-generator. This leads to a rise in the expander's mechanical work, increasing power generation from 4.19 to 25.24 kW<sub>power</sub>, thereby improving the system's electrical self-consumption coverage from 5.94% to 35.16%. The reinjection of expanded steam into the eighth stage of the MSF section in the MSF/MED hybrid thermal desalination subsystem, operating under high vacuum, enhances steam boiling, which improves both desalination capacity and steam condensation in the product water extraction and treatment subsystem, significantly raising the system's electrical power demand.



**Figure 10.** Effect of activation heat variation on electrical power, electrical demand, SECs and electrical coverage in cogeneration mode.

Additionally, as depicted in Figure 10, the SEC indicator increases with the heat supplied to the generator, starting at 60.71 kWh/m<sup>3</sup> and reaching 90.86 kWh/m<sup>3</sup>. This positions the proposed system above conventional MED and MSF desalination technologies in terms of SEC, as noted by Al-Karaghoul and Kazmerski [34].

As shown in Figure 11, the behavior of the PR and PR<sub>global</sub> indicators reveals a decrease in the system's desalination performance. This effect is observed because the system maintains constant product water production through the operation of the high-vacuum hybrid MSF/MED desalination subsystem and the motive fluid cooling and brine dilution subsystem, allowing the concentration process to reach a product water output of 1.212 kg/s. However, the increase in activation heat and the power allocated to the expander-generator's self-consumption do not enhance desalination performance, leading to a decline in efficiency as more heat is added to the system.



**Figure 11.** Effect of activation heat variation on desalination indicators: PR, PR<sub>global</sub> and recovery rate, in cogeneration mode.

According to Figure 7 an increase in activation heat correlates with a rise in water production capacity, escalating from 0.548 to 1.212 kg/s due to heightened seawater boiling (31). This phenomenon results in an increase in product water generation and RR from 38.2% to 84.4%.

As discussed in Figure 11 this increase also contributes to higher electrical demand. However, the power input from the expander has a notable impact on desalination performance. The PR indicator diminishes from 25.51 to 8.05, representing a reduction of 68.4%, while the thermal power output ( $Q_{gen}$ ) increases to 350 kW. The PR<sub>global</sub> decreases from 10.64 to 6.68, indicating a reduction of 37.21%. This decrease is less pronounced than that of the PR, attributed to the electrical self-consumption of the expander.

The proposed system attains a PR of 25.51, which is 1.83 times higher than the value reported by Aguilar-Jiménez et al. [20] for a MED/ORC cogenerative system. Additionally, the recovery rate of 38.2% to 84.4% is comparable to that observed in cooling desalination systems, which exhibit the highest recovery rates among phase-change desalination technologies [35,36].

##### 5.5. Operational Study of the System in Trigenerative Mode

To operate the proposed system in trigeneration mode, the condensed product water stream (84) exits the sixth effect in a saturated state at 4 °C and 0.81 kPa. This stream is directed to a three-way valve (3WV) that regulates its flow based on cooling demand.

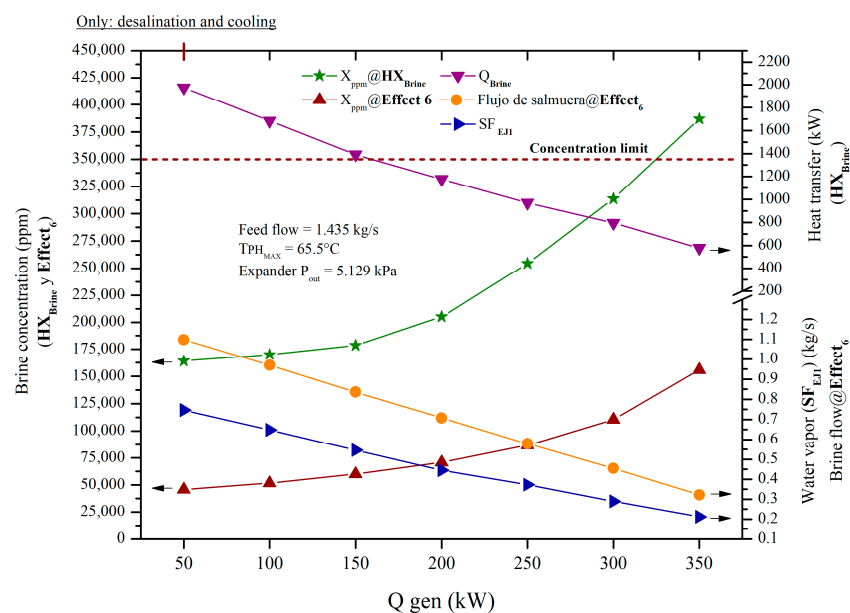


When cooling is required, the valve redirects stream (88a) to the evaporator (EVA), where its low temperature efficiently absorbs heat to produce the cooling effect. If cooling is not needed, the valve diverts stream (88b) to a barometric column (BC2), where it undergoes a controlled pressure increase before being discharged into a product water tank. Additionally, low-temperature vapor streams (1) and (2) from earlier stages are processed in the ejectocondenser (EJ1), where they mix with a high-pressure driving fluid. This process condenses the vapor, releasing latent heat and maintaining vacuum conditions, while the condensed fluid is directed to BC2. By combining chilled product water and vapor condensation via the ejector, the system achieves efficient and flexible cooling production, optimizing its performance in trigeneration mode.

This trigeneration operation, involving the simultaneous production of product water, cooling, and electrical energy, has a direct impact on all subsystems, particularly in the cooling of the motive fluid and brine dilution. Power generation requires the production of additional steam in the generator, which is used as the working fluid to feed the expander. Subsequently, the expanded steam at the expander outlet is reinserted into the vapor-liquid separator ( $S_{PW9}$ ), increasing product water production and cooling capacity, as a greater mass flow of steam is available for boiling the concentrated brine stream (66) entering the heat exchanger ( $HX_2$ ). This leads to a higher concentration of brine at the outlet of the sixth effect. These factors result in a reduction of the maximum limit of the reinjected steam mass flow percentage and a decrease in the final concentration of the brine entering the brine dilution tank.

#### 5.6. Operational Performance Results of Subsystems: Product Water Extraction, Adequacy, and Cooling of the Motive Fluid in Trigeneration Mode

The brine concentrations, measured in parts per million (ppm), at the outlet of the thermal desalination subsystem and the concentrated brine stream at the outlet of the heat exchanger ( $HX_{Brine}$ ) are presented in Figure 12. The brine concentration ( $X_{ppm@effect6}$ ) represents the concentration in stream (3). The increase in this brine concentration is caused by two factors: the increased boiling of water vapor in the generator and the increased steam flow directed to the expander, which is then used for brine vapor boiling in the heat exchanger of effect 1 ( $HX_1$ ). Similarly, the brine concentration ( $X_{ppm@HXBrine}$ ) also rises, primarily due to the increase in brine concentration at the outlet of the MED section.



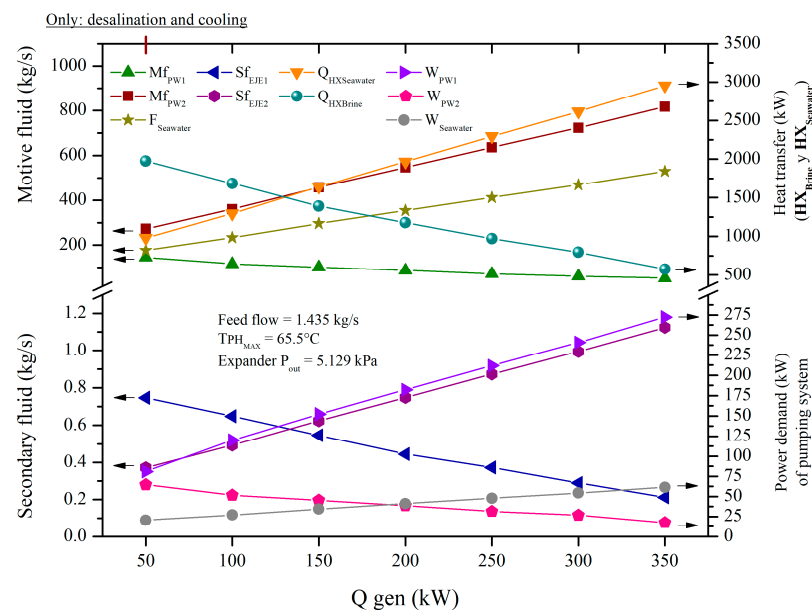
**Figure 12.** Effect of activation heat variation on brine concentration, heat exchanger, brine flow and secondary flow ( $Sf_{EJ1}$ ), in trigeneration.

Additionally, the dissipation of condensation heat from the motive fluid by the saturated brine stream (stream 3) causes this stream to enter the saturation region, releasing water vapor and resulting in a higher concentration. The brine concentration at the outlet of the heat exchanger ( $HX_{Brine}$ ) shows an average increase of 203.01% compared to the concentration at the outlet of effect 6, exceeding the limiting concentration of 350.00 ppm when operating at  $Q_{Q_{gen}} = 350$  kW, with a reported concentration of 386.959 ppm.

Furthermore, heat transfer in the heat exchanger ( $HX_{Brine}$ ) and the secondary flow ( $Sf_{EJ1}$ ) tend to decrease due to the reduction in brine flow, resulting in a diminished capacity to remove condensation heat. This reduces the condensing capacity of the ejectocondenser (EJ1) from 0.1758 to 0.0869 kg/s, directing the remaining cold vapor to the second ejectocondenser (EJ2).

The cold vapors from the brine separator ( $S_{Br15}$ ), evaporator, and effect 6 are used as secondary flows, while the product water, coming from a reservoir tank, is used as the motive flow in the corresponding ejectocondenser. In Figure 9, a decrease in the motive flow ( $Mf_{PW1}$ ) of the ejectocondenser (EJ1) from 145.9 to 52.794 kg/s is observed, alongside an increase in the motive flow ( $Mf_{PW2}$ ) of ejectocondenser (EJ1) from 272.8 to 817.05 kg/s, and an increase in the seawater flow ( $F_{Sea}$ ) from 176.8 to 529.4 kg/s. This occurs because the motive flow is closely linked to the secondary flow; a decrease in the secondary flow indicates a lower amount of motive flow needed to ensure condensation.

The reduction in secondary flow ( $Sf_{EJ1}$ ), as shown in Figure 13, is caused by the decrease in brine flow, which limits the amount of heat transferred in the heat exchanger ( $HX_{Brine}$ ). With an increase in activation heat, the steam flow to be condensed in the ejectocondenser (EJ1) is reduced, transferring the remaining steam to the ejectocondenser (EJ2) for condensation.



**Figure 13.** Influence on the variation of the activation heat in the motor flows, heat exchanger, secondary flows and electrical demand of the pumping system, in trigeneration.

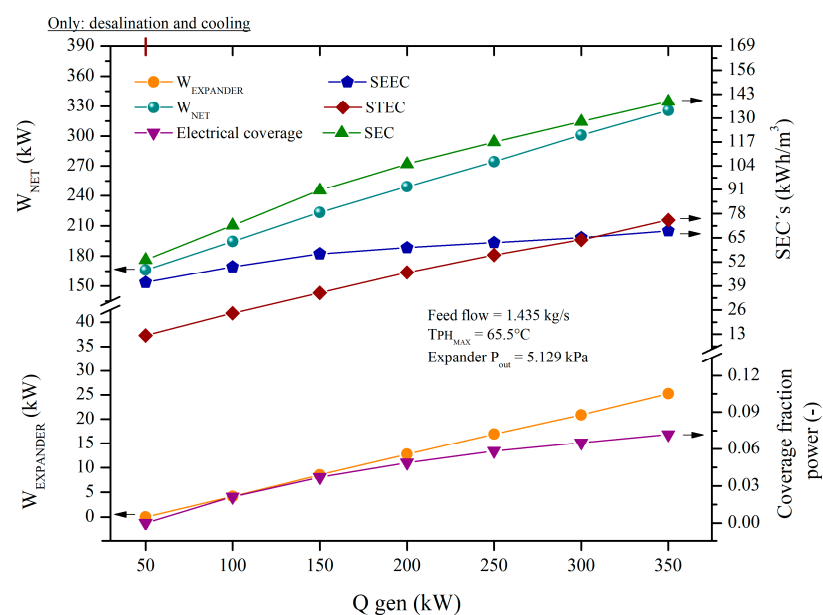
The increase in secondary flow ( $Sf_{EJ2}$ ) requires a higher motive flow to achieve condensation, leading to an average increase of 17.86% in the electrical demand of the hydraulic equipment ( $P_{PW2}$ ). Conversely, the reduction in secondary flow ( $Sf_{EJ1}$ ) results in an average decrease of 24.9% in the electrical demand of the hydraulic equipment ( $P_{PW1}$ ). Additionally, heat exchange follows the trends of both secondary and motive flows. As mentioned, condensing a larger amount of steam requires a higher motive flow, resulting in increased heat transfer in the heat exchanger to remove the condensation heat generated in the ejection-condensation process. The heat transfer in the heat exchanger ( $HX_{Brine}$ ) decreases from

1974.25 kW to 459.35 kW, while the heat transfer in the heat exchanger ( $HX_{Sea}$ ) increases from 979.02 kW to 2954 kW.

### 5.7. Overall Operating Performance Results in Trigeneration Mode

The power demand of the proposed system is primarily driven by the hydraulic pumping equipment in the product water extraction and adaptation subsystem. In trigeneration mode, the electrical demand increases due to the higher flow of motive fluid required for the condensation of the product water stream (88a) evaporated in the ejectocondensers (EJ1 and EJ2).

The expander increases its electrical generation capacity as more heat is supplied, utilizing the additional steam as the working fluid. The reintroduction of the expanded vapor into the vapor-liquid separator ( $S_{PW9}$ ) enhances both water production and cooling, which subsequently raises the system's electrical demand. This behavior is shown in Figure 14, where both electrical demand and electrical coverage tend to increase.



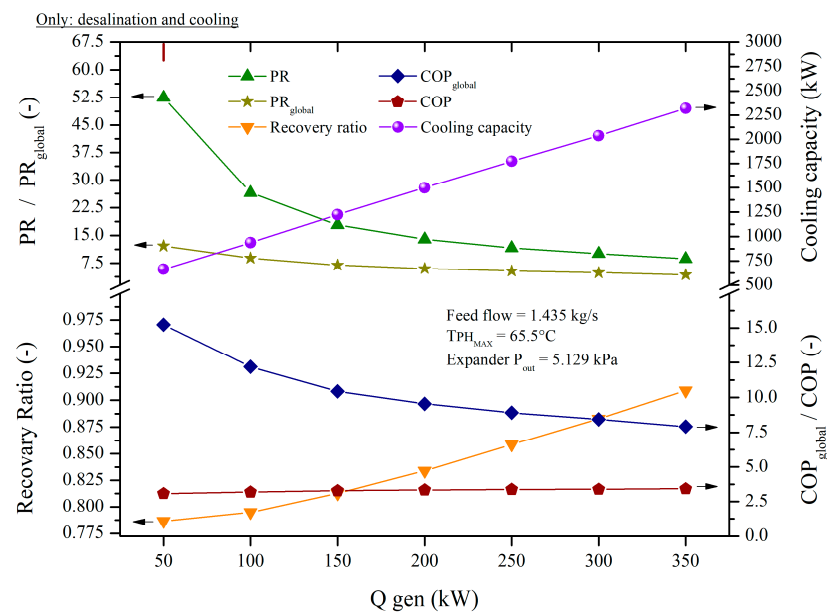
**Figure 14.** Influence of activation heat variation on electrical power, electrical demand, SEC's and electrical coverage in trigeneration.

This is because the electrical generation provided by the expander is less than the increase in the electrical demand of the system's hydraulic equipment, which is caused by the impact of the expanded steam on the MSF/MED thermal desalination subsystem operating at high vacuum, as well as the increase in cold steam mass flow from effect 6 and the evaporator. The system's electrical demand increases to a maximum of 351.14 kW, but the power generated by the expander reduces the system's overall electrical consumption by an average of 5.32%. Additionally, the self-consumption electrical coverage varies between 2.11% and 7.18%.

Although the product water capacity increases from 1128 to 1304 m<sup>3</sup>/day, this increase is not significant. The additional steam production in the generator leads to a rise in both water production and cooling capacity, increasing the power demand, which the expander fails to fully compensate. As a result, SEC indicator increases from 53.16 to 143.95 kWh/m<sup>3</sup>, reflecting higher energy consumption relative to the activation heat.

The system's cooling capacity also increases with the additional heat supplied to the generator, rising from 665.07 to 2320.3 kW, as shown in Figure 15. This is due to the reinsertion of expanded steam into the thermal desalination subsystem, which enhances product water capacity. The liquid product water at 4 °C is used as the cooling fluid. After producing the cooling effect, the product water transitions to the vapor phase and is

divided into two streams ( $ST_2$ ), fractionating the vapor mass flow for the proper operation of the ejectocondensers (EJ1 and EJ2).



**Figure 15.** Effect on the variation of activation heat on the performance indicators PR, PR<sub>global</sub>, RR, COP and COP<sub>global</sub>, cooling capacity, in trigeneration.

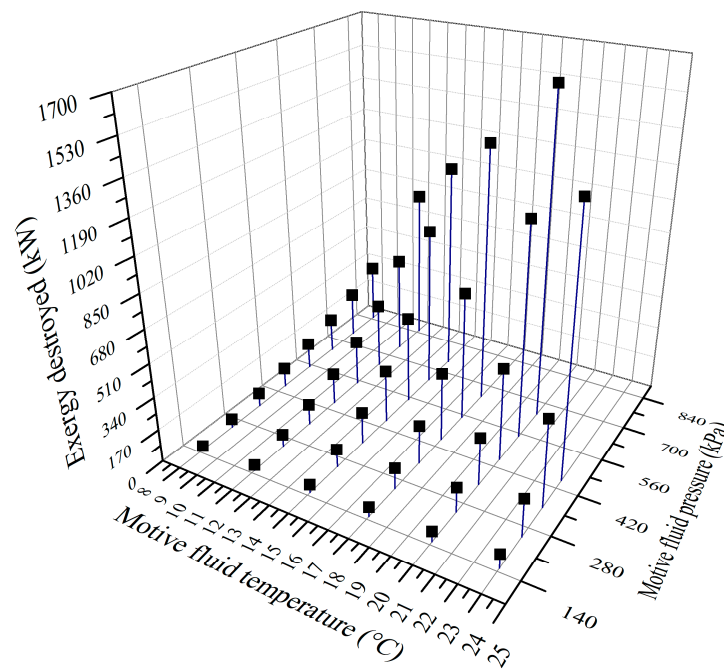
The maximum COP achieved is 3.43, which is 4.70 times higher than the COP of 0.73 reported by Gomri [37] for single-effect absorption systems. Additionally, the COP<sub>global</sub> indicator proposed by López-Zavala et al. [31] was 9.325, which is 1.643 times lower than the COP achieved by the proposed system, which reached 15.231.

The PR decreases from 52.51 to 8.66 as the heat input to the generator increases, while water production rises from 1.128 to 1.304 kg/s. The maximum PR of 52.51 is 3.76 times higher than that reported by Aguilar-Jiménez et al. [20]. The PR<sub>global</sub>, which accounts for both the thermal and electrical energy of the system, drops from 12.15 to 4.32, as the reduction in electrical demand from expander generation does not offset the increased heat input to the generator. The RR reaches a maximum of 86.7%, which is 2.22 times higher than the MED-HDH system reported by Tahir and Al-Ghamdi [38]. This suggests that future studies should investigate the technical feasibility of incorporating a crystallizer in the motive fluid cooling subsystem and brine dilution process.

### 5.8. Operational Performance Results of the Ejectocondensation Process

A study based on the second law of thermodynamics was conducted on the barometric ejectocondensation process, with the objective of determining the operating parameters of the motive pressure for the hydraulic pumping equipment ( $P_{PW1}$  and  $P_{PW2}$ ). The study assumed constant thermodynamic properties for the secondary fluid, with a pressure of 0.804 kPa, temperature of 4 °C, and mass flow of 0.2167 kg/s. For the motive fluid, an initial mass flow rate of 130 kg/s was considered. However, as mentioned, the hydraulic pumping equipment features a flow variator that adjusts to ensure condensation.

The exergy destruction results, shown in Figure 16, indicate that regardless of the temperature condition of the motive fluid, an increase in motive pressure at each temperature condition leads to an increase in exergy destruction. Additionally, it is observed that higher driving temperatures result in greater exergy destruction, suggesting a decrease in exergy efficiency. This occurs because increasing the driving pressure requires a higher motive flow to ensure complete condensation of the steam inside the ejectocondenser. A higher motive flow in the ejectocondenser leads to greater energy losses due to friction.



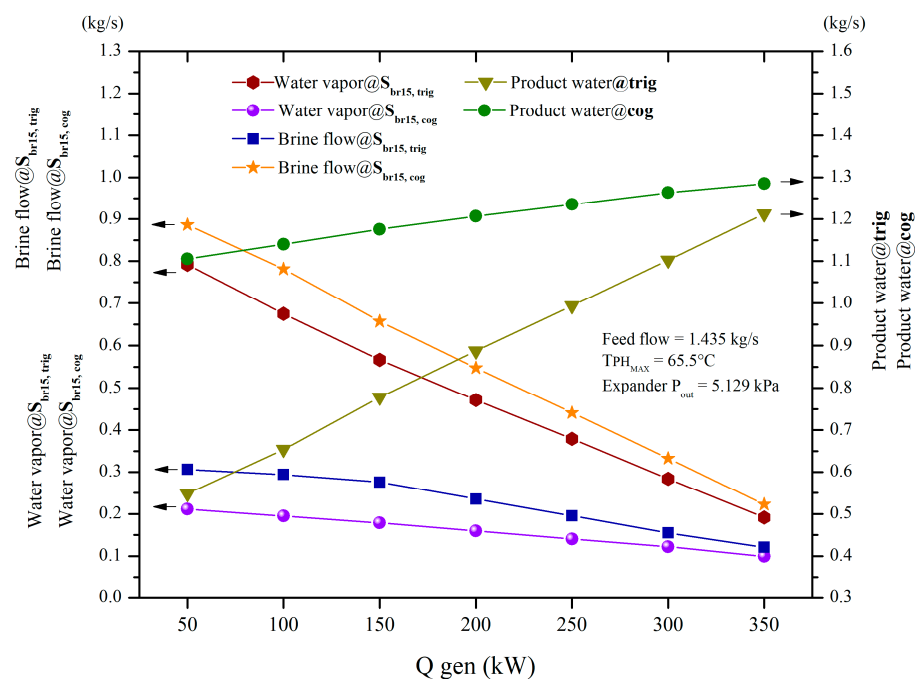
**Figure 16.** Variation of exergy destroyed in the ejectocondenser under different conditions of temperature and driving pressure.

### 5.9. General Comparison

#### 5.9.1. Operating Mode Comparison

The technological system developed in this work exhibits a distinctive operating behavior in terms of water production and electrical demand, depending on the operation mode.

In trigeneration mode, the water production capacity ranges from 1.128 to 1.304 kg/s, while in cogeneration mode it ranges from 0.548 to 1.212 kg/s, reflecting an average increase of 38.18% in trigeneration mode (Figure 17).



**Figure 17.** Influence on activation heat variation in product water, brine flow, steam generated in cogeneration and trigeneration.

In trigeneration mode, the condensed water in the MED section is sent to the evaporator (EVA), where it changes phase to steam. This additional steam is directed to the ejectocondenser (EJ1) for condensation, increasing the motive flow ( $F_{mpW1}$ ) and heat dissipation in the heat exchanger ( $HX_{Brine}$ ). The saturated brine (stream 3) is further heated and separated in the liquid-vapor separator ( $S_{Br15}$ ). Water vapor extraction in trigeneration mode varies from 0.791 to 0.1915 kg/s, while in cogeneration mode it ranges from 0.211 to 0.099 kg/s. This process reduces the final brine flow (stream 17), enhancing the potential for valuable mineral extraction and achieving zero liquid discharge.

### 5.9.2. Literature-Based Operating Comparison

To emphasize the main distinctions and advantages of the proposed MSF/MED-HV system, a comparative analysis is conducted against the MED-HV system reported by La Torre et al. [12] and the MED system described by López-Zavala et al. [31]. The comparison evaluates operational performance and design capacities, considering the specific design conditions established by the respective authors and those defined for the MSF/MED-HV system in this study. Key metrics such as desalination capacity, cooling output, and energy efficiency are presented in Table 9 to highlight the relative performance of each system.

**Table 9.** Results of comparative table of the MSF/MED-HV and MED systems reported in literature.

| Parameters            | Unitz | MSF/MED-HV (1) | MED-HV La Torre (2) | MED López-Zavala (3) | ( $\Delta_{1-2}$ %) | ( $\Delta_{1-3}$ %) |
|-----------------------|-------|----------------|---------------------|----------------------|---------------------|---------------------|
| Desalination capacity | kg/s  | 1.14           | 1.11                | 0.86                 | 2.70                | 32.55               |
| Cooling               | kW    | 1224.32        | -                   | 2042.87              | -                   | 40.06               |
| $Q_{gen}$             | kW    | 150            | 191.61              | 173.43               | -21.71              | -13.50              |
| $W_{pumps\ total}$    | kW    | 231.98         | 39.3                | 261.37               | 490.27              | 11.24               |
| PR                    | -     | 17.81          | 13.49               | 11.60                | 32.02               | 53.53               |
| RR                    | -     | 0.80           | 0.83                | 0.53                 | 3.61                | 50.94               |
| $PR_{global}$         | -     | 6.99           | 11.20               | 4.65                 | 37.58               | 50.32               |
| COP                   | -     | 3.27           | -                   | 4.7                  | -                   | 30.42               |
| $COP_{global}$        | -     | 10.43          | -                   | 9.32                 | -                   | 11.84               |

Among the key advantages of the proposed MSF/MED-HV system is its PR of 17.81, surpassing the systems by La Torre et al. [12] and López-Zavala et al. [31] by 32.02% and 53.53%, respectively. The system by La Torre et al. also operates under high-vacuum (HV) conditions (28.3 to 0.8 kPa; 68 to 4 °C), while López-Zavala et al. operates under conventional conditions (30.87 to 5.63 kPa; 70 to 35.37 °C). Despite similar or less demanding operational ranges, the proposed system achieves superior desalination performance, with a RR of 0.8, exceeding López-Zavala et al. RR of 0.53 by 50.94%.

In terms of energy demand, the MSF/MED-HV system requires 150 kW of generator heat input ( $Q_{gen}$ ), 21.71% lower than the  $Q_{gen}$  reported by La Torre et al. and 13.5% lower than that of López-Zavala et al., highlighting its improved thermal efficiency. However, the total power requirement for hydraulic pumps ( $W_{pumps\ total}$ ) is higher (231.98 kW), primarily due to the dual barometric ejectocondensation process. Despite these demands, the system achieves superior global performance indicators, with a  $PR_{global}$  of 6.99 and a  $COP_{global}$  of 10.43, representing an improvement of 50.32% and 11.84%, respectively, compared to López-Zavala et al. system. These advances highlight the significant thermodynamic and operational efficiency of the proposed design.

After the parametric study, the following system design conditions were determined for both cogeneration and trigeneration modes. Tables 10–13 provide the thermodynamic properties, operating characteristics, and energy evaluation of the system under the selected design conditions.

**Table 10.** Thermodynamic and operational characteristics of the MSF/MED at high vacuum trigenerative system.

| No. Stage/Effects        | Pressure (kPa) | Temperature (°C) | Accumulated Product Water (kg/s) | Brine Concentration (ppm) | $Q_{\text{cond}}$ (kW) | $Q_{\text{effects}}$ (kW) | Seawater Temperature (°C) |
|--------------------------|----------------|------------------|----------------------------------|---------------------------|------------------------|---------------------------|---------------------------|
| 1                        | 28.3           | 68.00            | -                                | 36,467                    | 38.65                  | 150                       | 65.5                      |
| 2                        | 22.73          | 63.08            | 0.0164                           | 36,785.2                  | 28.3                   | -                         | 56.55                     |
| 3                        | 18.1           | 58.15            | 0.0281                           | 37,104.2                  | 28.34                  | -                         | 52.05                     |
| 4                        | 14.32          | 53.23            | 0.0398                           | 37,423.4                  | 28.27                  | -                         | 47.53                     |
| 5                        | 11.23          | 48.31            | 0.0512                           | 37,743.6                  | 28.24                  | -                         | 43.03                     |
| 6                        | 8.72           | 43.38            | 0.0625                           | 38,065.5                  | 28.32                  | -                         | 38.52                     |
| 7                        | 6.72           | 38.46            | 0.0737                           | 38,387.9                  | 28.24                  | -                         | 34.01                     |
| 8                        | 5.13           | 33.54            | 0.0847                           | 38,711.4                  | 28.23                  | -                         | 29.5                      |
| 1                        | 3.87           | 28.61            | 0.1364                           | 40,245                    | -                      | 94.13                     | 25.0                      |
| 2                        | 2.89           | 23.69            | 0.1854                           | 42,298.2                  | -                      | 123.15                    | 25.0                      |
| 3                        | 2.14           | 18.77            | 0.2455                           | 45,014.5                  | -                      | 152.01                    | 25.0                      |
| 4                        | 1.56           | 13.85            | 0.3166                           | 48,617                    | -                      | 181.17                    | 25.0                      |
| 5                        | 1.12           | 8.92             | 0.3986                           | 53,451.1                  | -                      | 210.72                    | 25.0                      |
| 6                        | 0.79           | 4.00             | 0.4913                           | 60,092                    | -                      | 240.09                    | 25.0                      |
| HX <sub>Brine,cog</sub>  | 0.79           | 4.00             | 0.1788                           | 76,443                    | -                      | 445.61                    | -                         |
| HX <sub>Brine,trig</sub> | 0.79           | 4.00             | 0.5498                           | 175,605                   | -                      | 1371.1                    | -                         |
| Total <sub>cog</sub>     |                |                  |                                  |                           | 236.63                 | 1596.8                    |                           |
| Total <sub>trig</sub>    |                |                  |                                  |                           | 236.63                 | 2422.5                    |                           |

**Table 11.** Energy evaluation of the proposed MSF/MED at high vacuum system under design conditions in cogenerative operation.

| Components:                                                      | Energy Balance                                                                                                                                      | Gains (kW)        | Losses (kW)       | Internal Energy Integration (kW) |
|------------------------------------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------|-------------------|-------------------|----------------------------------|
| Generator                                                        | $\dot{m}_{31}h_{31} + \dot{m}_{88}h_{88} = \dot{m}_{32}h_{32} + \dot{m}_{89}h_{89}$                                                                 | 150               |                   |                                  |
| HX <sub>Brine</sub>                                              | $\dot{m}_3h_3 + \dot{m}_8h_8 = \dot{m}_4h_4 + \dot{m}_9h_9$                                                                                         |                   | 445.618           |                                  |
| HX <sub>Sea</sub>                                                | $\dot{m}_{13}h_{13} + \dot{m}_{19}h_{19} = \dot{m}_{14}h_{14} + \dot{m}_{20}h_{20}$                                                                 |                   | 361.69            |                                  |
| Condensers                                                       | $\dot{m}_{in}h_{in} + \dot{m}_{vap}h_{vap} = \dot{Q}_{out} + \dot{m}_{liq}h_{liq}$                                                                  |                   |                   | 236.63                           |
| Effects                                                          | $\dot{m}_{in}h_{in} + \dot{Q}_{in} = \dot{m}_{vap}h_{vap} + \dot{m}_{liq}h_{liq}$                                                                   |                   |                   | 1051.39                          |
| Expander                                                         | $\dot{W}_{exp} = \dot{m}_{in}v_{out}(P_{in} - P_{out})/\text{Eff}$                                                                                  |                   | 8.568             |                                  |
| P <sub>PW1</sub>                                                 | $\dot{W}_1 = \dot{m}_7v_7(P_8 - P_7)/\text{Eff}$                                                                                                    | 32.43             |                   |                                  |
| P <sub>PW2</sub>                                                 | $\dot{W}_2 = \dot{m}_{12}v_{12}(P_{13} - P_{12})/\text{Eff}$                                                                                        | 33.26             |                   |                                  |
| P <sub>SW1</sub>                                                 | $\dot{W}_3 = \dot{m}_{18a}v_{18a}(P_{19} - P_{18a})/\text{Eff}$                                                                                     | 7.505             |                   |                                  |
| P <sub>SW2</sub>                                                 | $\dot{W}_4 = \dot{m}_{18b}v_{18b}(P_{22} - P_{18b})/\text{Eff}$                                                                                     | 0.176             |                   |                                  |
| <b>Streams:</b>                                                  |                                                                                                                                                     |                   |                   |                                  |
| $\dot{m}_{22}h_{22}$                                             | Seawater intake                                                                                                                                     | -22,298.3         |                   |                                  |
| $\dot{m}_{17}h_{17}$                                             | Brine outlet                                                                                                                                        |                   | -10,002.7         |                                  |
| $\dot{m}_{10}h_{10} + \dot{m}_{15}h_{15} + \dot{m}_{82b}h_{82b}$ | Product water outlet                                                                                                                                |                   | -12,401.4         |                                  |
| $\dot{m}_{35}h_{35}$                                             | Expander inlet steam                                                                                                                                |                   | -550.98           |                                  |
| $\dot{m}_{36}h_{36}$                                             | Expander outlet steam                                                                                                                               |                   | -559.55           |                                  |
| <b>Total:</b>                                                    |                                                                                                                                                     | <b>-22,074.92</b> | <b>-22,698.75</b> | <b>1287.99</b>                   |
| Product Water                                                    | $\dot{m}_{\text{prod. water}} = \dot{m}_{15} + \dot{m}_{10} + \dot{m}_{85a}$                                                                        |                   |                   | 0.778 kg/s                       |
| PR                                                               | $PR = (\dot{m}_{\text{prod. water}} \times 2326 \frac{\text{kJ}}{\text{kg}}) / (\dot{Q}_{\text{gen}})$                                              |                   |                   | 12.062                           |
| PR <sub>global</sub>                                             | $PR_{\text{global}} = (\dot{m}_{\text{prod. water}} \times 2326 \frac{\text{kJ}}{\text{kg}}) / (\dot{Q}_{\text{gen}} + \sum \dot{W}_{\text{tot.}})$ |                   |                   | 8.179                            |
| Recovery Ratio                                                   | $RR = \left( \frac{\dot{m}_{\text{prod. water}}}{\dot{m}_{\text{Feed}}} \right) \times 100$                                                         |                   |                   | 54.2%                            |
| Total power demand                                               | $\sum \dot{W}_{\text{tot. cog.}} = \dot{W}_1 + \dot{W}_2 + \dot{W}_3 + \dot{W}_4$                                                                   |                   |                   | 71.20 kW                         |
| E.C. <sub>Cog</sub>                                              | $E.C._{\text{cog.}} = \dot{W}_{\text{exp}} / \sum \dot{W}_{\text{tot. cog.}}$                                                                       |                   |                   | 0.1203                           |
| BC1                                                              | $H_1 = (P_{\text{atm}} - P_{\text{dis}}) / (\rho_{10} \times g)$                                                                                    |                   |                   | 10.179 m                         |
| BC2                                                              | $H_2 = (P_{\text{atm}} - P_{\text{dis}}) / (\rho_{88b} \times g)$                                                                                   |                   |                   | 10.247 m                         |
| BC3                                                              | $H_3 = (P_{\text{atm}} - P_{\text{dis}}) / (\rho_{15} \times g)$                                                                                    |                   |                   | 9.952 m                          |
| BC4                                                              | $H_4 = (P_{\text{atm}} - P_{\text{dis}}) / (\rho_{17} \times g)$                                                                                    |                   |                   | 11.475 m                         |



**Table 12.** Energy evaluation of the proposed MSF/MED at high vacuum system under design conditions in trigenerative operation.

| Components:                               | Energy Balance                                                                                                                             | Gains (kW)         | Losses (kW)       | Internal Energy Integration (kW) |
|-------------------------------------------|--------------------------------------------------------------------------------------------------------------------------------------------|--------------------|-------------------|----------------------------------|
| Generator                                 | $\dot{m}_{31}h_{31} + \dot{m}_{88}h_{88} = \dot{m}_{32}h_{32} + \dot{m}_{89}h_{89}$                                                        | 150                |                   |                                  |
| Evaporator                                | $\dot{m}_{85a}h_{85a} + \dot{m}_{86}h_{86} = \dot{m}_1h_1 + \dot{m}_{87}h_{87}$                                                            | 1224.32            |                   |                                  |
| HX <sub>Brine</sub>                       | $\dot{m}_3h_3 + \dot{m}_8h_8 = \dot{m}_4h_4 + \dot{m}_9h_9$                                                                                |                    | 1371.12           |                                  |
| HX <sub>Sea</sub>                         | $\dot{m}_{13}h_{13} + \dot{m}_{19}h_{19} = \dot{m}_{14}h_{14} + \dot{m}_{20}h_{20}$                                                        |                    | 1640.41           |                                  |
| Condensers                                | $\dot{m}_{in}h_{in} + \dot{m}_{vap}h_{vap} = \dot{Q}_{out} + \dot{m}_{liq}h_{liq}$                                                         |                    |                   | 236.63                           |
| Effects                                   | $\dot{m}_{in}h_{in} + \dot{Q}_{in} = \dot{m}_{vap}h_{vap} + \dot{m}_{liq}h_{liq}$                                                          |                    |                   | 1051.39                          |
| Expander                                  | $\dot{W}_{exp} = \dot{m}_{in}v_{out}(P_{in} - P_{out})/Eff$                                                                                |                    | 8.568             |                                  |
| P <sub>PW1</sub>                          | $\dot{W}_1 = \dot{m}_7v_7(P_8 - P_7)/Eff$                                                                                                  | 45.29              |                   |                                  |
| P <sub>PW2</sub>                          | $\dot{W}_2 = \dot{m}_{12}v_{12}(P_{13} - P_{12})/Eff$                                                                                      | 152.32             |                   |                                  |
| P <sub>SW1</sub>                          | $\dot{W}_3 = \dot{m}_{18a}v_{18a}(P_{19} - P_{18a})/Eff$                                                                                   | 34.24              |                   |                                  |
| P <sub>SW2</sub>                          | $\dot{W}_4 = \dot{m}_{18b}v_{18b}(P_{22} - P_{18b})/Eff$                                                                                   | 0.176              |                   |                                  |
| <b>Streams:</b>                           |                                                                                                                                            |                    |                   |                                  |
| $\dot{m}_{18b}h_{18b}$                    | Seawater intake                                                                                                                            | −22,298.3          |                   |                                  |
| $\dot{m}_{17}h_{17}$                      | Brine outlet                                                                                                                               |                    | −4079.88          |                                  |
| $\dot{m}_{10}h_{10} + \dot{m}_{15}h_{15}$ | Product water outlet                                                                                                                       |                    | −18,258           |                                  |
| $\dot{m}_{35}h_{35}$                      | Expander inlet steam                                                                                                                       |                    | −550.98           |                                  |
| $\dot{m}_{36}h_{36}$                      | Expander outlet steam                                                                                                                      |                    | −559.55           |                                  |
| $\dot{m}_{88a}h_{88a}$                    | Evaporator coolant inlet                                                                                                                   |                    | −7843.97          |                                  |
| $\dot{m}_1h_1$                            | Evaporator refrigerant fluid outlet                                                                                                        | −6619.65           |                   |                                  |
| <b>Total:</b>                             |                                                                                                                                            | <b>−27,311.604</b> | <b>−28,272.28</b> | <b>1288.02</b>                   |
| Product water                             | $\dot{m}_{prod. water} = \dot{m}_{15} + \dot{m}_{10} + \dot{m}_{85a}$                                                                      |                    |                   | 1.149 kg/s                       |
| PR                                        | $PR = (\dot{m}_{prod. water} \times 2326 \frac{kJ}{kg}) / (\dot{Q}_{gen})$                                                                 |                    |                   | 17.81                            |
| PR <sub>global</sub>                      | $PR_{global} = (\dot{m}_{prod. water} \times 2326 \frac{kJ}{kg}) / (\dot{Q}_{gen} + \sum \dot{W}_{tot.} - \dot{W}_{exp})$                  |                    |                   | 6.995                            |
| Recovery ratio                            | $RR = \left( \frac{\dot{m}_{prod. water}}{\dot{m}_{Feed}} \right) \times 100$                                                              |                    |                   | 80.06%                           |
| COP                                       | $COP = \dot{Q}_{eva} / (\dot{Q}_{gen} + \sum \dot{W}_{tot.})$                                                                              |                    |                   | 3.278                            |
| COP <sub>global</sub>                     | $COP_{global} = (\dot{Q}_{eva} + \dot{m}_{prod. water} \times 2326 \frac{kJ}{kg}) / (\dot{Q}_{gen} + \sum \dot{W}_{tot.} - \dot{W}_{exp})$ |                    |                   | 10.435                           |
| Total power demand                        | $\sum \dot{W}_{tot. trig.} = \dot{W}_1 + \dot{W}_2 + \dot{W}_3 + \dot{W}_4$                                                                |                    |                   | 231.983 kW                       |
| E.C. <sub>Trig</sub>                      | $E.C._{trig.} = \dot{W}_{exp} / \sum \dot{W}_{tot. trig.}$                                                                                 |                    |                   | 0.0369                           |
| BC1                                       | $H_1 = (P_{atm} - P_{dis}) / (\rho_{10} \times g)$                                                                                         |                    |                   | 10.185 m                         |
| BC2                                       | $H_2 = (P_{atm} - P_{dis.}) / (\rho_{88b} \times g)$                                                                                       |                    |                   | -                                |
| BC3                                       | $H_3 = (P_{atm} - P_{dis.}) / (\rho_{15} \times g)$                                                                                        |                    |                   | 9.952 m                          |
| BC4                                       | $H_4 = (P_{atm} - P_{dis.}) / (\rho_{17} \times g)$                                                                                        |                    |                   | 13.069 m                         |

**Table 13.** Ejectocondenser design conditions.

| Ejectocondenser | Parameters  | Units | Motive Fluid | Secondary Fluid | Ejectocondenser Output |
|-----------------|-------------|-------|--------------|-----------------|------------------------|
| EJ1             | Temperature | °C    | 9            | 4               | 12.19                  |
|                 | Pressure    | kPa   | 350          | 0.79            | 1.5                    |
|                 | Mass flow   | kg/s  | 102.236      | 0.525           | 102.761                |
| EJ2             | Temperature | °C    | 28.5         | 4               | 29.38                  |
|                 | Pressure    | kPa   | 400          | 0.79            | 4.10                   |
|                 | Mass flow   | kg/s  | 457.192      | 0.623           | 457.815                |

## 6. Conclusions

This study presents a novel trigeneration system that simultaneously produces desalinated water, electrical energy, and cooling, addressing critical challenges related to water scarcity and energy demand in coastal regions. The system integrates hybrid MSF and MED technologies under high vacuum, coupled with ejector-condensation mechanisms and an expander-generator subsystem. These innovations enhance energy recovery, operational flexibility, and resource efficiency.



Simulation results using Aspen Plus demonstrated the system's technical feasibility and superior performance across different operating modes. In trigeneration mode, the system achieved a RR of 80%, a PR of 17.81, and a cooling capacity of 1225 kW while producing 99,273.6 L/day of desalinated water. In cogeneration mode, the system achieved a PR of 12.06 and reduced electrical consumption by 12.03%, emphasizing its adaptability to seasonal and demand-based operational shifts.

Key innovations include the elimination of seawater preheaters in the MED section, which reduces heat losses and increases desalination capacity, and the use of barometric ejector condensers, which optimize steam utilization and cooling production. These features not only improve energy efficiency but also simplify system design, making it more cost-effective and scalable for real-world applications.

The proposed system demonstrates significant potential for mitigating water and energy challenges while promoting sustainability. Future research should focus on validating the system under experimental and field conditions, assessing its long-term reliability, and exploring opportunities for scaling up to meet larger community needs. Additionally, economic analysis and life-cycle assessments would provide deeper insights into the system's commercial viability and environmental impact.

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## Nomenclature

### Symbols and abbreviations:

|                       |                                       |                  |                               |
|-----------------------|---------------------------------------|------------------|-------------------------------|
| T                     | Temperature (°C)                      | EVA              | Evaporator                    |
| P                     | Pressure (kPa)                        | EJ               | Ejector condenser             |
| Q                     | Heat transfer (kW)                    | PH               | Preheater                     |
| m                     | Mass flow (kg/s)                      | MSF              | Multi stage distillation      |
| h                     | Specific enthalpy (kJ/kg)             | MED              | Multiple effect distillation  |
| v                     | Specific volume (m <sup>3</sup> )     | RO               | Reverse osmosis               |
| s                     | Specific entropy (kJ/kg·K)            | TBT              | Top brine temperature (°C)    |
| Ex                    | Exergy flow (kW)                      | BBT              | Bottom brine temperature (°C) |
| W                     | Power (kW)                            | NaCl             | Sodium chloride               |
| Eff                   | Efficiency (%)                        | H <sub>2</sub> O | Water                         |
| RR                    | Recovery Ratio (-)                    | ΔT               | Temperature difference (°C)   |
| PR                    | Performance Ratio (-)                 | ΔP               | Pressure difference (°C)      |
| PR <sub>global</sub>  | Performance Ratio global (-)          | <b>Subindex:</b> |                               |
| COP                   | Coefficient Of Performance (-)        | SW               | Seawater                      |
| COP <sub>global</sub> | Coefficient Of Performance global (-) | SWT              | Seawater tank                 |
| EC                    | Cobertura eléctrica (%)               | PW               | Producto water                |

|      |                                                             |                   |                 |
|------|-------------------------------------------------------------|-------------------|-----------------|
| SEEC | Specific Electrical Energy Consumption(kWh/m <sup>3</sup> ) | i                 | Initial         |
| STEC | Specific Thermal Energy Consumption (kWh/m <sup>3</sup> )   | f                 | Final           |
| SEC  | Specific Energy Consumption (kWh/m <sup>3</sup> )           | Br                | Brine           |
| GEN  | Generator                                                   | mf                | Motive fluid    |
| HX   | Heat Exchanger                                              | sc                | Suction chamber |
| Cond | Condenser                                                   | m                 | Mixer           |
| SEP  | Separator liquid-vapor                                      | Number of streams | 1, 2, 3. . .    |
| EV   | Expansion valve                                             |                   |                 |
| ST   | Split flow                                                  |                   |                 |
| Mix  | Mixer                                                       |                   |                 |
| EXP  | Expander                                                    |                   |                 |
| P    | Hydraulic pump                                              |                   |                 |
| BC   | Barometric column                                           |                   |                 |

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## **ANEXO B**

### **State of the art of desalination in Mexico**

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## Review

# State of the Art of Desalination in Mexico

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**Abstract:** This research paper presents a review of the state of the art of desalination in Mexico, with the aim of clarifying the main challenges and opportunity areas for desalination as the main solution to overcome water stress. First, the current situation and forecasts on the availability of water resources in Mexico are described, followed by the main economic, social, and legislative issues of desalination. Mexico's installed capacity for the different desalination technologies and their evolution in recent years was investigated, followed by a comparison with global trends. The current state of research and development in desalination technologies carried out by Mexican institutions was also studied. The results show that membrane technology plants account for 88.85%, while thermal technology plants account for the remaining 11.15%. Although Mexico presented a 240% increase in its desalination capacity in the last 10 years, it has not been enough to overcome water stress, so it is concluded that in the future, it is necessary to increase its capacity in greater proportion, specifically in the areas with greater scarcity, which can be achieved with the joint participation of academy–industry–government through the creation of autonomous organizations, social programs, and/or public policies that promote it.

**Keywords:** desalination; developing countries; environmental impact



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## 1. Introduction

Water is a crucial element for the development and survival of living beings; its availability declines as the population increases and as a consequence of climate change [1]. In the last decades, the percentage increase in water use worldwide has doubled the percentage increase in population [2]. It is estimated that if the current rate of exploitation continues, in the year 2030, there will be a 40% water deficit worldwide, a deficit predicted to be greater in arid regions [3]. It is worth noting that, according to regional forecasts, the regions with greater solar radiation—*Mexico, for instance*—are the ones that will face severe droughts [4].

Since the sixteenth century, water desalination systems have been developed; nevertheless, those early systems were unable to produce water that could be used for human consumption [5]. Desalination technologies are classified into two categories: thermal (phase change) and membrane desalination [6]. Currently, membrane desalination technologies are the most widely used due to the expansion of Reverse Osmosis (RO) technology. However, thermal technologies are attractive because of their potential for integration with thermal energy sources such as waste heat and/or renewable energy sources, and they also perform better for the desalination of high-salinity water [7].

For the year 2022, the worldwide installed capacity percentages for membrane technologies are the following: Reverse Osmosis (RO), 70%; Nanofiltration (NF), 4%; and

Electrodialysis (ED), 2%. It is worth noting that the high percentage of membrane desalination systems' installed capacity is closely related to the rise of RO technology since its installed capacity has significantly increased in comparison to the rest of the membrane technologies. The worldwide installed capacity percentages for thermal desalination systems are the following: Multi-Stage Flash (MSF) at 17%; and Multiple-Effect Distillation (MED) at 6%. The remaining 1% is distributed among the rest of the technologies; Membrane Distillation (MD), Vapor Compression (VC), Freezing (FR), Humidification-Dehumidification (H-DH), and Solar Still (SS) [8]. Thermal technologies, also known as *phase-change technologies*, were the pioneers in the desalination area, and they dominated the world's installed capacity until the year 2000 when Reverse Osmosis technologies reached the same installed capacity [9].

Since 2010, the percentage increase in capacity and installation of new desalination plants worldwide has been 6.8% per year (4.6 million m<sup>3</sup>/d per year). According to Zotalis et al. [10], in 2012, the worldwide desalination capacity was 79 million m<sup>3</sup>/d. Four years later, Catrini et al. [11] reported that the worldwide installed desalination capacity in the year 2016 was 88.6 million m<sup>3</sup>/d, resulting in a 12.15% increase. In 2018, the total capacity reported was 92.5 million m<sup>3</sup>/d, representing a 4.4% increase. In 2019, Jones et al. [9] reported an installed capacity of 95.37 million m<sup>3</sup>/d, a 3.1% increase. In the year 2020, Eke et al. [12] reported an installed capacity of 97.5 million m<sup>3</sup>/d, representing a 1.9% increase.

The worldwide desalination installed capacity for the year 2022 is 115.62 million m<sup>3</sup>/d, distributed in 20,956 desalination plants, presenting an increase of 18.9% with respect to the year 2020 [8]. This is a considerable increase with respect to those obtained in previous years; even though the increase spans over two years, the yearly percentage increase is approximately 9%.

An analysis of the worldwide installed capacity indicates that the country with the largest installed capacity is Saudi Arabia, with 14.58 million m<sup>3</sup>/d, followed by the United States of America, with 11.90 million m<sup>3</sup>/d, and by the United Arab Emirates, with 9.47 million m<sup>3</sup>/d. Membrane desalination technologies amount to 88.96% of the worldwide desalination capacity, while thermal (phase change) technologies amount to 10.56%. The remaining percentage (≈0.5%) corresponds to hybrid and/or unknown technology plants [8].

The installed capacity percentages of membrane and thermal technologies worldwide are mostly similar at a country level. Nevertheless, in the Persian Gulf region, the percentages are inverted; thermal technologies amount to 85% of the total installed capacity. This is due mainly to the availability of fossil fuels in oil-producing countries, where the residual heat of electrical energy generation processes is used to activate desalination systems, mainly through MED or MSF technologies. It is also worth noting that these two technologies easily adapt to the high salinity and high-temperature conditions of the Persian Gulf's water. Furthermore, MED and MSF systems are more robust for the desalination of water with a proliferation of marine biology, which is highly detrimental to RO technology [13].

Dévora-Isiordia et al. [14] conducted a comparative evaluation of the desalination processes used in Mexico in terms of production costs (USD/m<sup>3</sup>) and energy consumption (kWh/m<sup>3</sup>). The authors report that the most widely used desalination technology in Mexico in 2013 is reverse osmosis, and the results of their evaluation support this fact because this technology consumes 2 to 2.8 kWh/m<sup>3</sup> at the cost of 0.6 USD/m<sup>3</sup>, while its direct competition (MED and MSF) consume between 3.4 to 4 and 5 to 8 kWh/m<sup>3</sup> at the cost of 1.5 and 1.1 USD/m<sup>3</sup>, respectively.

In summary, reverse osmosis technology is the current leader in the desalination area. However, there are factors that potentialize the implementation of thermal technologies, such as thermo-physical conditions of the water to be desalinated or the availability of energetic resources (waste heat, electrical energy, renewable sources) for the activation of desalination systems. Another potential area of improvement that can accelerate the



growth of thermal desalination technologies is their hybridization with cooling [15–18] and/or electrical energy generation systems [19–21].

Despite having a water availability of 3620 m<sup>3</sup> per capita per year, Mexico is a country that faces a severe water shortage, mainly due to its territorial extension, because the concentration of water resources is located in the southern part of the country, while the main industrial activity is carried out in the north (which presents scarcity conditions). The distance and the technical limitations of transporting water between the northern and southern zones make water desalination the main tool to combat water stress in Mexico.

RO technology is currently the most widely used; however, Mexico has some bodies of water with high salinity and a high potential for biofouling, factors that limit the proper performance of RO technology. In addition, there is a great diversity of renewable energy sources, such as solar, geothermal, and biomass, that can directly activate thermal desalination systems. For this reason, it is of interest to know the state of the art of the different desalination technologies used in Mexico and their respective advances in technological development to increase their viability. Additionally, modifications or hybridizations with other systems allow the cogeneration of electrical energy and/or cooling.

This research paper briefly and concisely analyzes the current state of desalination worldwide and focuses on the study of the state of the art of desalination plants in Mexico in order to clarify the main challenges and opportunity areas of desalination as the main solution to overcome water stress. The current conditions in Mexico are analyzed, beginning from the water availability for each administrative hydrological region (AHR), the main socio-economic issues of desalination, and the existing public policies/development programs, as well as those in the process of consolidation. Mexico's installed capacity for the different desalination technologies and their evolution in recent years was investigated, followed by a comparison with global trends. It also reports some of the research and development work applied to desalination technologies carried out by researchers affiliated with Mexican institutions. Finally, the authors present their perspective on the current situation and future challenges of desalination in Mexico as the main way to solve the problem of water stress. The main contributions of this work, which have not been presented before, are the following:

- The evolution of desalination in Mexico is analyzed in terms of its total capacity and percentages by technology.
- The main contributions and findings of R&D of Desalination in Mexico are reported.
- The main limitations for desalination in Mexico to be a sustainable solution to overcome water stress are defined.
- An overview of existing and ongoing regulations is given, as well as program and policy options proposed by the authors.

## 2. Current State of Desalination in Mexico

### 2.1. Water Availability in Mexico

The population growth rate in Mexico has caused its population to quintuplicate in the time lapse between 1950 and 2020 [22]. In the same period, its per capita (pc) amount of available water has decreased by almost 80% [23], mainly due to the decrease in annual precipitation in some regions of the country and to its extraction rate. The average annual precipitation in Mexico is 718.3 mm, which presents a decrease of  $\approx 7\%$  from the year 2000 to date. However, in regions such as the Baja California peninsula, the average annual precipitation is  $\approx 168$  mm, which presents a decrease of 10% with respect to the year 2000 and is 77% lower than the national average. It is worth noting that Mexico rates as the fourth country with the largest water extraction ( $88.84 \times 10^9$  m<sup>3</sup>/year), only below China, the United States of America, and Indonesia [24].

The Water Stress Indicator (WSI) was proposed by Falkenmark and Lindh in 1974 [25] and is currently the most utilized indicator to measure water scarcity [26]. This method defines water scarcity in terms of the total amount of water available for the population of a delimited region. Therefore, it is expressed as the amount of renewable drinking water

available for each person every year [27]. Table 1 shows the limits established for the classification of the level of water scarcity.

**Table 1.** Water scarcity-level classification in accordance with the Water Stress Indicator.

| WSI (m <sup>3</sup> /pc/Year) | Category          |
|-------------------------------|-------------------|
| WSI < 500                     | Absolute scarcity |
| 500 < WSI < 1000              | Chronic scarcity  |
| 1000 < WSI < 1700             | Regular scarcity  |

The national average per capita (pc) water availability in Mexico reported for the year 2019 is 3620 m<sup>3</sup>/pc/year. However, there is a marked disparity in the per capita water availability among the administrative hydrological regions (AHR) in Mexico. In the Northern, Central, and Northeastern regions, the water availability is 1558 m<sup>3</sup>/pc/year, while in the Southern region of the country, it is 10,508 m<sup>3</sup>/pc/year [28]. Therefore, currently, the largest part of the country is under regular scarcity conditions, although there are regions that present chronic and absolute scarcity conditions. Figure 1 shows that AHRs 1 (Baja California and Baja California Sur) and 6 (Chihuahua, Coahuila, Nuevo León, and Tamaulipas) are facing regular scarcity, and the forecast is that, in the year 2030, they will be under chronic scarcity conditions. AHR 13, located in the central zone of Mexico, presents conditions of absolute scarcity and is forecast to remain in this status until 2030, this is due to the fact that AHR 13 has the second largest population (23,721,664 inhabitants) only behind AHR 8 (24,981,524 inhabitants), but its territorial extension is 957.23% smaller.



**Figure 1.** Water availability per AHR in Mexico [28].

It is worth emphasizing that the Northern, Central, and Northeastern zones of Mexico cover 78% of the national territory, host 77% of the Mexican population, and contribute



83% of the Gross Domestic Product (GDP). In these regions, there is a larger population concentration and, therefore, greater activity in the agricultural, industrial, and services sectors, which results in a greater water requirement in the zones with less availability [29]. The economic sectors with the largest water consumption are the following: agricultural (75.7%), public supply (14.7%), industrial (4.9%), and electrical energy generation (4.7%). It should be noted that human consumption through public supply should be the main usage of water and that in some AHRs that have water scarcity, there have been shortages reflected in supply cuts for the housing sector [30]. There have also been socio-hydric problems derived from the population's disapproval of the construction of new industrial plants with high water-consumption processes.

Throughout the different AHRs, 1186 monitoring points have been installed. Their purpose is to assess the quality of underground water in terms of their total dissolved solids (TDS). Their results for the year 2018 show that in the Northern, Central, and Northeastern zones of Mexico, 78.53% of the underground water is sweet water, 13.42% is slightly brackish, 7.92% is brackish water, and 0.11% is saline water. In the Southern zone of the country, 81.58% of the underground water is sweet water, 13.92% is slightly brackish, 4.48% is brackish water, and there is no presence of saline water (or saltwater intrusion).

It is projected that the population of Mexico will increase by 10% in the year 2030, 141.8 million inhabitants, even though the growth rate tends to reduce. It is estimated that 56% of the population growth will occur in the Central and Northern regions of the country, which currently have low water availability. This will worsen their status, moving from regular scarcity ( $WSI \approx 1000$ ) to chronic or absolute scarcity ( $WSI \approx 500$ ), as shown in Table 1.

## 2.2. Desalination Plants in Mexico

Mexico's installed desalination capacity in the year 2022 is  $749,751 \text{ m}^3/\text{d}$ , distributed throughout 351 desalination plants located in 28 of its 32 federative entities—also called states. The federative entity with the largest installed capacity is Mexico City, located in AHR 13, with  $276,453 \text{ m}^3/\text{d}$ , followed by Baja California, located in AHR 1, with  $97,382 \text{ m}^3/\text{d}$ , and Quintana Roo, located in AHR 12, with  $63,523 \text{ m}^3/\text{d}$ . Mexico City's desalination capacity is 183.89% higher than Baja California's, but its WSI is 91.28% lower. Therefore, this can be interpreted as a response to the absolute scarcity faced by AHR 13. Another factor to take into account is that Mexico City (and all of AHR 13) is located far from the coastline; therefore, its main feed waters are brackish, river, and wastewater. Figure 2 shows the installed desalination capacity in Mexico compared to the five countries with the largest installed capacity in the world.

The increase in desalination capacity in Mexico from 2013 to 2022 is 240% (2.4 times) [14]. Until 2013, only 19 of the 32 federative entities in Mexico had desalination plants (59% of the federal entities); in 2020, 28 federative entities had desalination plants (87.5%). Figure 3 shows the number of desalination plants per federative entity. Considering the federative entities that already had desalination plants, those with the largest increase in capacity during the last 10 years are Nuevo Leon, with 1734%, Tamaulipas, 666%, and Jalisco, 595%. It is worth mentioning that Nuevo Leon is one of the entities with the lowest WSI ( $839 \text{ m}^3/\text{hab}/\text{year}$ ), the highest industrial activity, and, therefore, the highest contribution to GDP. Currently, its government continues to seek to increase its desalination capacity despite its distance from the coast.

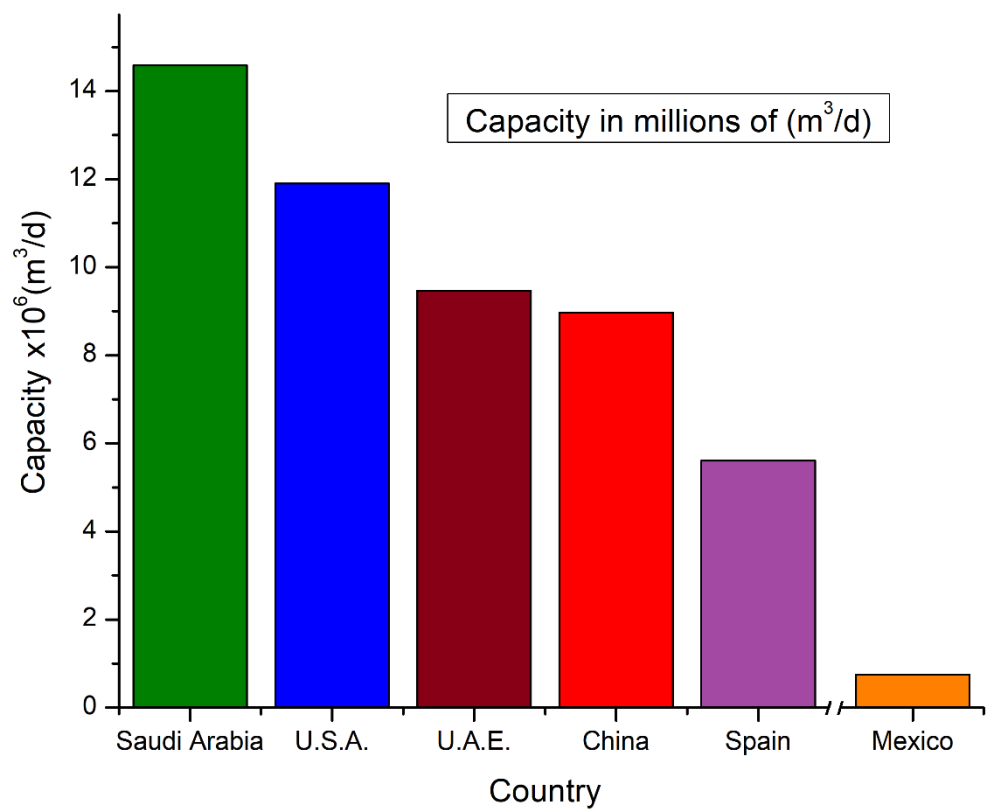


Figure 2. Comparison between the countries with the largest desalination capacity and Mexico [8].

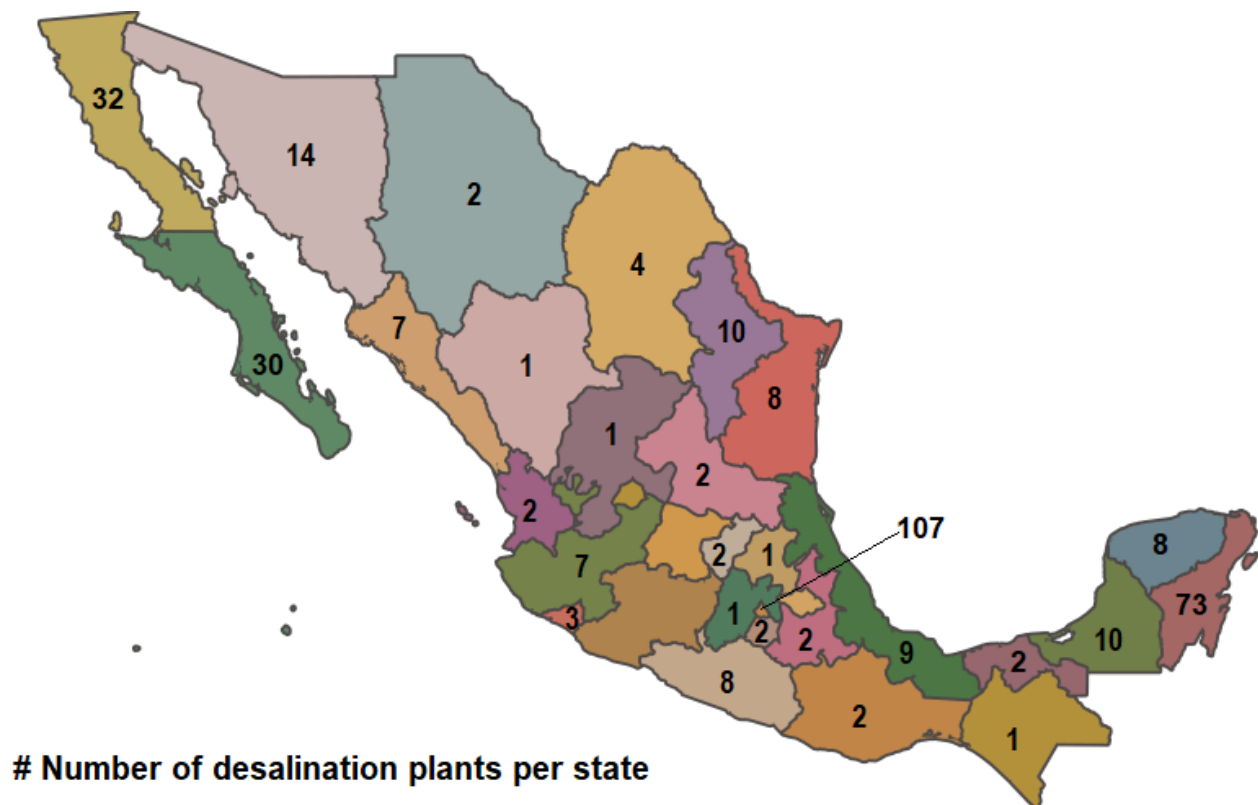
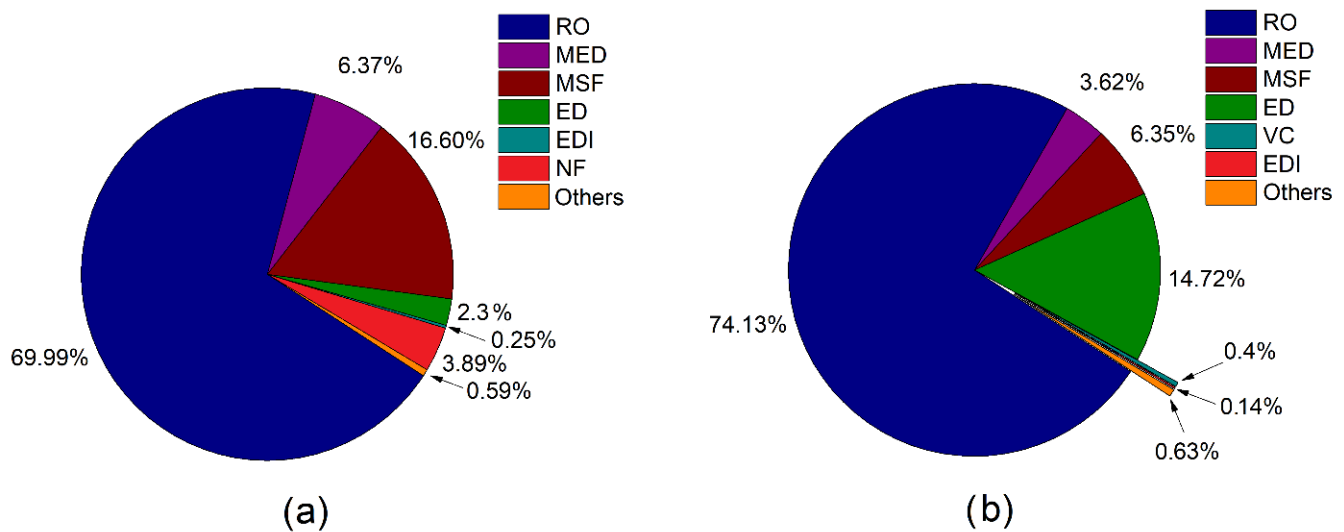


Figure 3. Number of desalination plants per state in Mexico, 2022.

The number of installed desalination plants that use membrane technology in Mexico amount to 85.7% of the total plants; the plants based on thermal (phase change) technology

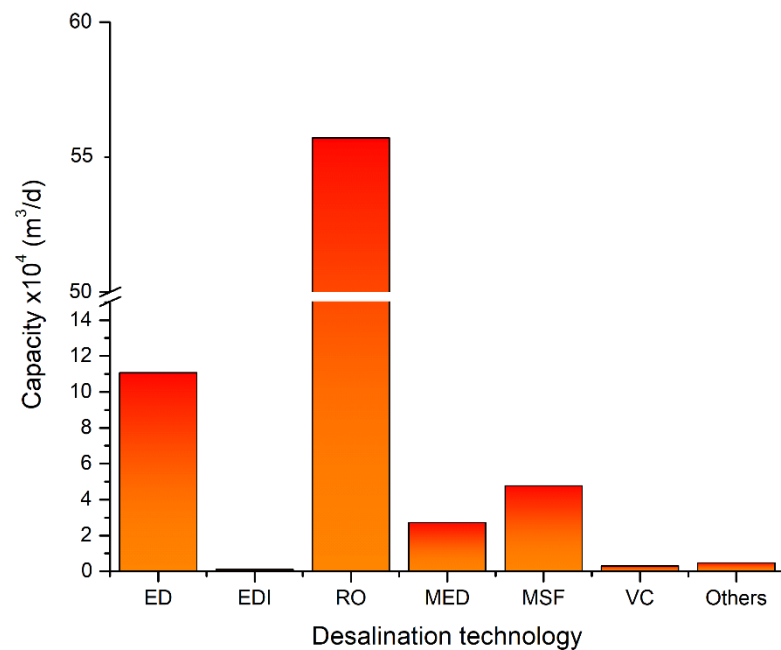
amount to 14.3% [8]. However, in terms of installed capacity, membrane technology plants amount to 88.85%, while thermal technology plants amount to the remaining 11.15%. These percentages are similar to those of worldwide installed capacity, where membrane technology amounts to 90%, while thermal technology amounts to 10%. Figure 4 shows the percentages of installed capacity in Mexico and in the rest of the world. In Figure 4(a), the category -Other- groups; forward osmosis (FO), freezing (FR), humidification–dehumidification (HDH), membrane distillation (MD), solar still (SS), and vapor compression (VC), due to their low percentages of installed capacity worldwide. In Figure 4(b), the category -Other- groups; humidification–dehumidification (HDH) and solar still (SS) due to their low percentages of installed capacity in Mexico.



**Figure 4.** Percentages of installed capacity by technology: (a) worldwide; (b) México [8].

The technologies used in the desalination plants in Mexico are the following: electro-dialysis (ED), electrodeionization (EDI), reverse osmosis (RO), multiple-effect distillation (MED), multiple-stage flash (MSF), and vapor compression (VC). Within the category -others- we can find humidification-dehumidification (HDH) and solar distillation (SS) technologies, which present a lower capacity and efficiency compared to the rest of the desalination technologies. However, they are appropriate for the satisfaction of low water demand ( $1\text{--}100\text{ m}^3/\text{d}$ ) [31] and highlight their low and null electrical energy consumption required for their activation, respectively [32]. Figure 5 shows the installed capacity ( $\text{m}^3/\text{d}$ ) per type of technology in Mexico.

The number of desalination plants installed in the Northern and Southern zones of the country is similar; however, in terms of installed capacity, the Northern zone has a larger capacity. For instance, AHR 1, which corresponds to the Baja California Peninsula, whose population is 4,672,579 inhabitants, has 91 desalination plants and an installed capacity of  $161,470\text{ m}^3/\text{d}$ . On the other hand, AHR 12, which corresponds to the Yucatan Peninsula, whose population is 4,857,556 inhabitants, has 91 desalination plants and a capacity of  $81,480\text{ m}^3/\text{d}$ . This shows that in the Southern zone of the country, there are more low-capacity installed plants, a situation that could be related to the installation of autonomous desalination systems by the private, touristic activities-related sector in the Yucatan Peninsula.



**Figure 5.** Installed capacity per type of desalination technology in Mexico in 2022 [8].

AHR 4, which covers the northern states of the country, has regular scarcity conditions, and it is expected that in 2030 it will be in chronic scarcity conditions, with WSI levels similar to those in AHR 1. However, Nuevo León, the most affected federative entity belonging to AHR 4 is approximately 300 km away from the coast of Tamaulipas, where seawater desalination plants could be installed. Regardless of having the second largest number of aquifers (102), only below RHA 6, AHR 4 has the second lowest per capita water availability in Mexico (Figure 1). To solve this problem, aqueducts are needed for either the transportation of water from zones with availability or for the transportation of desalinated water from the coasts near the urban zones, as proposed by Roggenburg et al. [33].

The difference in per capita water availability between the AHRs in Mexico is an indicator of the need to increase per capita water availability in the Northern, Central, and Northeastern zones of the country. This has induced federative entities governments, as well as members of the private industry, to continue pursuing the construction of desalination plants under the BOT scheme in order to guarantee a continuous supply, especially in the federative entities of Nuevo Leon, Baja California, Baja California Sur, and Mexico City.

#### 2.2.1. Reverse Osmosis

In the 1980s, derived from the Sonntlan project, a reverse osmosis desalination plant was constructed in the fishing community of Barrancas, which belongs to the municipality of Comondú, Baja California Sur. It had a capacity of 20 m<sup>3</sup>/d and was activated using solar PV energy, as well as a backup diesel generator. The project worked for some time, but nowadays, it is abandoned.

Currently, the largest reverse osmosis desalination plant has a capacity of 94,500 m<sup>3</sup>/d, 12.6% of the total desalination capacity in Mexico, and is located in Mexico City, where it desalinates brackish water. It started operations in 2011 and is still in operation. The second largest reverse osmosis desalination plant, with a capacity of 29,184 m<sup>3</sup>/d, is located in the municipality of Cadereyta, in the federative entity of Nuevo León, where it desalinates wastewater to be used in industrial processes. It started operations in 2015 and is still in operation. The third place in installed capacity (21,600 m<sup>3</sup>/d) belongs to the Ensenada desalination plant located in the federative entity of Baja California, which started operations in 2011 and still operating to desalinate seawater to be used for human consumption.

### 2.2.2. Electrodialysis

The electrodialysis desalination plant with the largest capacity is located in Mexico City. It has a capacity of 29,331 m<sup>3</sup>/d, started operation in 2001, and desalinates brackish water for human consumption. The second largest plant is located in Zacatecas, has a capacity of 6480 m<sup>3</sup>/d, started operations in 1995, and is still in operation. The third largest plant is located in Veracruz, has a capacity of 6240 m<sup>3</sup>/d, started operation in 1996, and is still operational.

It is worth noting that Mexico has a high percentage of electrodialysis technology installed capacity (14.72%) compared to world averages (2%). This is due to the fact that Mexico is a country with a high level of beer production, and this industry usually uses electrodialysis to desalinate water for its processes [34].

### 2.2.3. Electrodeionization (EDI)

The only electrodeionization technology plant is located in Mexico City; it has operated since 1999 with a capacity of 1091 m<sup>3</sup>/d, desalinating drinking water to be used as distilled water for industrial processes.

### 2.2.4. Multi-Stage Flash

In the 1970s, a collaborative project between the governments of Mexico and Germany resulted in the construction of a 10 m<sup>3</sup>/d MSF desalination plant in the municipality of La Paz, Baja California Sur. This MSF plant started operations in 1980, and it used a 194 m<sup>2</sup> field of flat plate solar collectors and a 160 m<sup>2</sup> field of 2-axis parabolic trough channel concentrators for activation. In addition, it had 324 m<sup>2</sup> of flat-plate solar collectors to heat a working fluid and store thermal energy to ensure continuous operation of the plant for 28 h.

The Federal Electricity Commission (CFE) operates the MSF desalination plants with the largest capacity. The largest plant has a capacity of 28,388 m<sup>3</sup>/d, is located in the municipality of Rosarito, Baja California, started operations in 1966, desalinates seawater, produces distilled water for electrical energy generation processes, and is still in operation. The plant with the second largest capacity (4800 m<sup>3</sup>/d) is also located in Rosarito, started operations in 1987, and is still in operation. In the third place of MSF desalination capacity, there are two 2400 m<sup>3</sup>/d plants; one is located in the municipality of Tuxpan, Veracruz, and the other in the municipality of Manzanillo, Colima. Both plants started operations in 1985, are still in operation, and desalinate seawater, generating distilled water for electrical energy generation processes.

### 2.2.5. Multiple-Effect Distillation (MED)

The MED desalination plant with the largest capacity (10,900 m<sup>3</sup>/d) is located in Mexico City, started operation in 1998 (is still in operation), and desalinates wastewater, producing distilled water for industrial use. The plant with the second largest capacity (2424 m<sup>3</sup>/d) is located in Topolobampo, Sinaloa, started operations in 1991, is still in operation, administered by the CFE, and desalinates seawater for electrical energy generation processes. The third place in capacity (2400 m<sup>3</sup>/d) is shared by two plants, both located in the municipality of Tuxpan, Veracruz. One of them started operations in 1991, is still in operation, and desalinates seawater for human consumption. The other plant started operations in 1993, is still in operation, and desalinates seawater for industrial processes.

### 2.2.6. Vapor Compression (VC)

There are six vapor compression desalination plants in Mexico. The one with the largest capacity (772 m<sup>3</sup>/d) is located in the municipality of Rosarito, Baja California, has been in operation since 1999, is managed by the CFE, and desalinates seawater in order to produce distilled water for electrical energy generation processes. The second place of installed capacity (768 m<sup>3</sup>/d) is shared by two plants, both controlled by the CFE. These plants desalinate seawater in order to produce distilled water for electrical energy generation processes. One is located in the municipality of Ciudad Victoria, Tamaulipas,

and has been in operation since 2003. The other plant is located in Mazatlán, Sinaloa, and has been in operation since 2005. The plant with the third largest capacity ( $360 \text{ m}^3/\text{d}$ ) is located in Rosarito, Baja California, and has been operated by a private company since 2007. It desalinates seawater for electrical energy generation processes.

### 2.3. Social, Economic, and Legislative Scenario for Desalination in Mexico

#### 2.3.1. Social

Due to its large extension ( $1,964,375 \text{ km}^2$ ) [35], Mexico has a large number of remote rural communities without access to the water distribution network. Even though the national percentage of access to water is 97.4% in urban areas, only 88% of the population in rural areas have access to drinking water [28]. Some of these remote communities are located near 11,122 km of the Mexican coastline; therefore, water desalination is an attractive solution to this problem. However, it is necessary to conduct the corresponding environmental, economic, and technical feasibility studies in order to compare the cost of the process to that of extending the water distribution network from the closest interconnection point.

Overall, rural communities have a low number of inhabitants, which results in a low water demand—or *decentralized demand* ( $<100 \text{ m}^3/\text{d}$ ). The market-leading desalination technologies (RO, MED, MSF) have more feasibility when used on a large scale [36]. It is common for rural populations to be in conditions of social marginalization and without access to the electrical energy transmission grid. Such is the case of Puertecitos, Baja California, and Punta Chueca, in the Seri indigenous community of Sonora, just to list a few examples. For these reasons, it is complicated to make the initial investment required to install a new desalination plant without external investment (government or private). Moreover, the activation and continuous operation of the plant could be affected by the lack of a continuous and reliable supply of electrical energy, as well as qualified personnel for preventive and/or corrective maintenance and for in site decision making regarding the plant's operation.

The National Hydric Policy (2019–2024) [37] sets as its priorities to progressively guarantee the human right to water access, to make more efficient use of water in order to contribute to sustainable development, and to reduce vulnerability during droughts, especially for indigenous populations. In remote coastal communities with decentralized demands, it may be more feasible to implement thermal desalination systems such as humidification-dehumidification or distillation, to mention a few examples. For this reason, it is of national interest to advance in the technological development of emerging or not commercially mature technologies and their implementation in demonstration projects or pilot plants. In addition, the availability of renewable energy sources in almost all territorial extensions can increase the feasibility of implementing decentralized desalination systems activated with renewable energy in remote or indigenous communities. Rural, indigenous, or remote communities usually access drinking water by exploiting underground aquifers. These, when exploited at a rate greater than their natural renewal rate, start to present a decrease in their availability, an increase in the depth in which the phreatic water is found (static level), or changes in the physical-chemical composition of the water. For instance, aquifers close to the sea may present marine intrusion, which increases the salinity of the aquifer to levels inappropriate for direct consumption. The number of aquifers with marine intrusion in Mexico has increased, mainly in the Baja California Peninsula, where there is a convergence of low rainfall rate, high levels of solar radiation, and the presence of high-salinity congenital water and easily soluble evaporitic minerals [28]. In these cases, the implementation of reverse osmosis desalination systems could represent an attractive solution because the energy consumption of this technology is proportional to the salinity of the water, which is generally found in brackish water conditions. In addition, the membrane lifetime, which is usually a limiting factor for the viability of this technology, is longer compared to its lifetime when desalinating seawater directly.



### 2.3.2. Economic

When there is no nearby aquifer, the population carries water in containers in order to have access to it. Worldwide, millions of women and children travel approximately 6 km in order to have access to water, and Mexico is not an exception [38]. Furthermore, there are people that commercialize water carriage in containers from a site with availability to remote communities in which it is scarce. Some documented cases reach costs of 40 USD/m<sup>3</sup>, a price above that reported by IDA (2017) of 15 USD/m<sup>3</sup> of desalinated water. Moreover, this carriage is conducted using trucks with metallic containers in which iron sediments that damage the quality of the water and, consequently, human health have been found [14]. Furthermore, water carriage in vehicles that use internal combustion engines causes a greater environmental impact due to the emissions caused by the consumption of the fuel required to carry large amounts of water, which also increases the economic cost of water with the aim of seeking business profitability.

Being a medium-low-income country, most of the desalination plants installed in Mexico have been built and operated by private sector companies in alliance with state and/or municipal governments in the BOT (building, operating, and transfer) modality. The private company is in charge of the plant's building and operation, obtaining an economic remuneration for the water produced for an agreed-upon number of years ( $\approx 20$  years); after this period, the plant is transferred to the government to be managed through its Public Services State Commissions. This is the case of the first desalination plant installed in the state of Sonora, which has a capacity of 200 L/s.

Another factor to take into account in the economic aspect is the operating cost of the desalination plants; although the reverse osmosis technology leads the market, it shares with the rest of the technologies the limiting factor of the specific energy consumption, to reduce it, it has been proposed; The integration of energy recovery devices, the activation with energy coming from renewable energy sources, the hybridization with systems for cooling and/or electric energy generation and the high-pressure pump with solar thermal energy, etc. In some regions of Mexico, as in AHR 1, the availability of electricity is a problem for the industrial, commercial, and residential sectors. For this reason, the construction of desalination plants must have correct energy planning that does not compromise the availability of electricity for other sectors. Some of the measures to reduce the energy consumption of desalination plants may mean a higher initial investment, but if they significantly reduce the cost of operation, they may be more economically viable.

Desalination continues to be a technologically viable solution, but its economic viability is still a limiting factor; for this reason, it is most widely used in developed countries. On the global scale, the cost of producing desalinated water dropped by 50% from 1985 [39]; nonetheless, studies in California and Mexico identify desalination as having the highest marginal costs among all realistic water-supply alternatives [40]. The final USTDA (2009) report estimates that Puerto Peñasco, Mexico, could build a plant with the capacity to produce 0.5 m<sup>3</sup>/s in the first phase, with the expected expansion of up to 2 m<sup>3</sup>/s by 2020 at the cost of 2.29 USD/m<sup>3</sup> (not including conveyance and storage). This is approximately seven times more expensive than the current cost of water production and delivery (0.339 USD/m<sup>3</sup>) (USTDA, 2009) [41]. For this reason, in countries such as Mexico, the economic impact derived from the investment required to build high-capacity plants is minimized because the design, construction, and operation are carried out jointly with private industry under the BOT modality.

With respect to environmental impact, the greatest challenge of desalination at present is the disposal of brine since it is usually discharged directly into seas and oceans near the coast; in Mexico, there are draft standards that establish guidelines for desalination plant discharge works, but they are in the process of being approved (PROY-NOM-013-CONAGUA-2015) [42]. However, Mexico has water bodies with different hydrodynamic characteristics; information reported in the literature indicates that the environmental impact caused by brine discharge is lower in water bodies with turbulence (waves and currents), such as the Pacific Ocean, but discharging brine in calm seas such as the Sea

of Cortez has a greater environmental impact [43]. Therefore, the discharge work must be carried out in accordance with the regulations (when approved) and considering the characteristics of the water body in question. In second place are the emissions of gases into the atmosphere derived from energy consumption.

To cite a recent example reported by Wilder et al. [44] on the Puerto Peñasco desalination plant project, which could produce  $0.5 \text{ m}^3/\text{s}$  of freshwater (in its first stage), an estimated  $0.73 \text{ m}^3/\text{s}$  of brine concentrate will be produced. Project engineers recommend discharging near the surface to take advantage of prevailing winds and stronger currents to help disperse the brine concentrate [42]. An environmental impact assessment was initiated but not completed [45]. In addition, energy use at the desalination plant would likely come from fossil fuel sources that produce GHG emissions.

### 2.3.3. Legislative

Regarding legislation, the 1992 National Water Law established that the exploitation of national water will be conducted through concession titles emitted by the National Water Commission (CONAGUA, by its Spanish acronym). Therefore, in order to install a desalination plant, a concession for the water extraction is required, whether from the sea—*direct intake*—or from underground aquifers—*wells*—. For the rejected water discharge, as is the case of brine, discharge permission emitted by the same institution is also required.

Other Official Mexican Norms (NOM, by its Spanish acronym) applicable for desalination are the following. NOM-001-SEMARNAT-2021 [46] established the maximum level of pollutants allowed for wastewater discharge into national waters, which applies to the discharge of brine from desalination plants. NOM-003-CONAGUA-1996 [47] established the requirements for the construction of water extraction wells looking for the prevention of aquifer pollutants, which applies in the case of the construction of an extracting well for a desalination plant. NOM-006-ENER-2015 [48] established the energetic efficiency required for pumping systems for water extraction wells in operation, limits, and testing method, which applies to the selection of the pumping equipment used in desalination plants that extract water from underground wells.

In 2015, the Mexican Official Norm project PROY-NOM-013-CONAGUA-2015 [42] was proposed. This project seeks to establish specifications and requirements for the extraction and discharge works that must be met in desalination plants or processes (public or private) that generate brackish or saline rejection waters and discharge them to the coastal, marine, and/or continental environment, in order to protect the environment. The Norm project was presented by the National Consultative Committee for Standardization of the Water Sector, with the collaboration of private sector organizations, institutions, and companies.

Regarding the quality of the product water obtained from desalination plants, NOM-0127-SSA1-2004 [49], originally published in 1994, establishes the maximum permissible levels that produce water should meet for human consumption. This norm determines that water for human consumption should have a maximum of 1000 ppm of TDS.

## 3. Mexico's Research and Development in Desalination

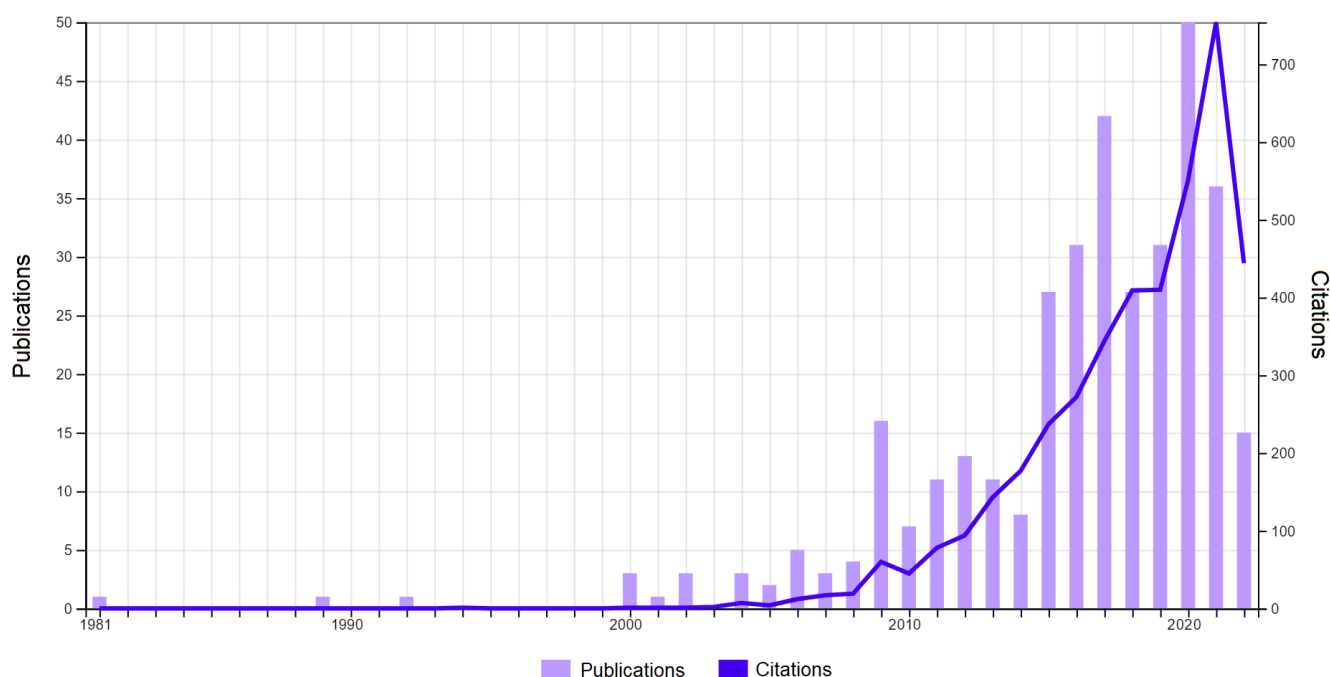
The main barrier that has limited the widespread of desalination technologies is the production cost ( $\text{USD}/\text{m}^3$ ), which is mostly (50–60%) dependent on the energy consumption during the process ( $\text{kWh}/\text{m}^3$ ) [50], for this reason, the efforts of different desalination research groups have been focused on reducing energy consumption, initial investment and increasing the useful life of the components involved in the different desalination technologies. This section summarizes the research and development work focused on optimizing the process of different desalination technologies published by researchers from Mexican institutions.

According to the extended citation index (SCI-EXPANDED) of the Web of Science (WOS) Core Collection database, from 1980 to 2022, 352 papers related to desalination have been published with total or partial participation of researchers from institutions located



in Mexico. These papers have been cited 4084 times and 3865 times without considering self-citations, giving an average of 11.6 citations per article, with an h-index of 32.

Figure 6 shows that research in Mexico started in 1981, although with occasional publications, one in 1989 and another in 1992. It was from the year 2000 when the rate of publications increased, reporting approximately three publications per year, until 2009, when 16 papers were published. From that year, the publication rate increased considerably; from 2015, approximately 30 papers per year began to be published, obtaining a maximum record of 50 publications per year in 2020. As well as the publication rate, the number of citations to Mexican papers has increased considerably, reaching a maximum record of approximately 750 citations in 2021.



**Figure 6.** Statistics of publications related to desalination in Mexico [51].

### 3.1. Membrane Technologies

#### 3.1.1. Reverse Osmosis (RO)

One of the main areas of improvement in the reverse osmosis desalination process is the optimization of the membrane; most of the efforts are focused on increasing the efficiency of the membrane, reducing its fabrication cost, increasing its useful lifetime, and/or reducing the frequency of maintenance. This section shows some of the main research papers published by researchers and/or research groups affiliated with institutions in Mexico.

Armendariz-Ontiveros et al. [52] studied the effectiveness of adhering to a coating formed by iron nanoparticles on the membrane of a reverse osmosis system in order to avoid biofouling. Under laboratory conditions, they compared the performance of the coating using seawater from two sites: Mar de Cortés, Mexico, and Playa del Sol, Chile. According to their results, the coated membranes presented greater biofouling (>74%) with water from the Sea of Cortez due to the differences in temperature and pH, but at the same time, the coating presented a greater biocide effect under these conditions. The authors concluded that temperature, dissolved oxygen, and pH have a more significant effect than salinity on the biofouling of membranes coated with iron nanoparticles.

Murillo-Verduzco et al. [53] manufactured a pulsation valve that generates pressure variations to a water flow in ranges previously defined by simulations. They studied pressure levels between 4826 and 6205 kPa, volumetric flow rates from 0.6 to 1.2 L/min, and frequencies from 10 to 45 Hz, with the purpose of increasing the permeate flow in

reverse osmosis desalination systems. The authors report that they obtained their best results when operating at a pressure of 6205 kPa, a volumetric flow rate of 0.6 L/min, and a frequency of 40 Hz in a non-turbulent flow regime.

Kitano et al. [54] studied the hydraulic compaction process in nitrogen-containing carbon-based reverse osmosis membranes at a laboratory scale using cross-flow membrane desalination systems. Their results show that high water pressure induces an overall reduction of interstitial spaces in the amorphous carbon structure. They also noted that the nitrogen-containing carbon-based membrane performed better in salt rejection and water permeability compared to the carbon-based membrane. The authors attribute the improved performance to the higher mechanical stability of the carbon structure due to the presence of nitrogen sites.

Feria-Díaz et al. [55] conducted a review study on the most recent publications focused on the hybridization of the reverse osmosis process with other desalination technologies in order to reduce the weaknesses and improve the strengths of each technology. For example, they mentioned RO hybridization with MED to initially desalinate seawater with the RO process and later use the reject as the feed stream of a MED system. The authors conclude that activating RO systems with renewable energy and implementing strategies to achieve zero liquid discharge (ZLD) and/or minimum liquid discharge (MLD), as well as the use of valuable products obtained from the brine, are strategies to reduce the environmental impact of reverse osmosis desalination.

Rodríguez-López et al. [56] evaluated the energy consumption of RO desalination systems under different feedwater concentrations (5000–36,000 mg/L), maintaining a constant feed volumetric flow rate of 14.4 m<sup>3</sup>/d. The authors considered three solar PV systems (fixed, single-axis tracking, and dual-axis tracking) for the power supply. With this study, it is possible to verify that the specific energy consumption per unity of produced water increases as the salinity of the feed stream increases; their results show that it varied from 2.33 to 3.83 kWh/m<sup>3</sup>.

Torres-Valenzuela et al. [57] developed and characterized a new composite membrane for reverse osmosis desalination systems. A commercial TFC membrane was modified by impregnating silver nanoparticles on its surface (5, 10, and 15 mg per 100 mL of distilled water) via interfacial polymerization in order to improve resistance to biofouling. Their results show that the salt rejection was 97.6, 97.8, and 96.7%, with a reduction in total cell layer and organic carbon of 72, 48, and 45%, using 5, 10, and 15 mg of AgNP, respectively.

### 3.1.2. Electrodialysis (ED)

In Mexico, the electrodialysis process is most commonly used to separate specific compounds or ions from process water and/or groundwater. Lambert et al. [58] studied the separation of Chromium (III) present in high concentrations in effluents from the tanning process in the textile industry in order to reduce the environmental impact of this industry in Guanajuato, Mexico. Ortega et al. [59] studied a hybrid process of electrodialysis with ion exchange to reduce the concentration of dilute solutions of As (V) ions in water destined for human consumption to values below the limit (10 µg/L) recommended by the United Nations (UN). Arreola-Castro et al. [60] proposed an electrodialysis process to remove iron from an aqueous solution, effluent from the mining industry in Durango, Mexico. They obtained an iron removal of 97.15%.

Enciso et al. [61] studied the hydrodynamic behavior and mass transfer in a filter-press type ED reactor using computational fluid dynamics (CFD) and validated their results with experimental data. Their theoretical data presented a good correlation with respect to those obtained experimentally for hydrodynamics, which made possible the validation of mass transfer and electrode potential distribution. The authors performed a second validation comparing the theoretical mass transfer coefficients with those obtained experimentally, presenting a maximum error of 16%.

Dévora-Isiordia et al. [62] studied the effect of temperature on the final concentration of the product water of a batch electrodialysis reversal (EDR) process, varying the

input voltage and feed water concentration in order to reduce specific energy consumption ( $\text{kWh}/\text{m}^3$ ) and the economic cost of the diluate ( $\text{USD}/\text{m}^3$ ). The variations were concentrations of 2000 and 5000  $\text{mg}/\text{L}$ ; temperature of 25, 30, and 35  $^{\circ}\text{C}$ ; and input voltages of 10 and 20 V. The authors mention that temperature has a positive effect on the EDR process, especially at 10 V, where increasing 10  $^{\circ}\text{C}$  increases the efficiency by 10.83 and 24.69% for the 2000 and 5000  $\text{mg}/\text{L}$  samples, respectively.

### 3.1.3. Nanofiltration (NF)

The studies published by Mexican institutions on nanofiltration desalination technology are mainly focused on the incorporation of thin films of different chemical compounds in order to improve their permeability and salt rejection properties. A case study of pilot plants of the technology is also presented, using a small-scale desalination system activated with renewable energies.

Pérez-Moreno et al. [63] studied the performance of ceramic membranes modified with metal impregnation for nanofiltration desalination on the Mexican Pacific coast. The authors report that the impregnation of metals in the membrane increases approximately 20% the rejection of TDS compared to conventional membranes, and in terms of divalent ions, the rejection was 60–80%, values close to those obtained with polymeric membranes in a reverse osmosis desalination process. The authors conclude that nanofiltration technology can be used for partial seawater desalination or as the main process in the separation of specific ions.

Flores-Prieto et al. [64] proposed a solution to treat brackish groundwater using low-pressure solar PV nanofiltration (LP-PV-NF). The study site is located in the desert of Chihuahua, where the groundwater presented high sulfate content (1863  $\text{mg}/\text{L}$ ) and 2195  $\text{mg}/\text{L}$  TDS. The authors reported that sulfates and TDS were reduced by 98.21 and 75.15%, respectively, under solar radiation equal to or greater than 750  $\text{W}/\text{m}^2$ , with an energy consumption of 1.94  $\text{kWh}/\text{m}^3$ . Their experimental results show that the production of their system was 3.2  $\text{m}^3/\text{d}$  for an average insolation time of 6.3 h/d.

García-Picazo et al. [65] studied the effect of incorporating chemically modified graphene oxide (GO) as a crosslinking agent in thin film nanofiltration membranes for water desalination applications. They studied the effect of GO on membrane properties and performance; they also analyzed the chemical and morphological composition of the GO surface on the membrane by thermogravimetric analysis. Their results show that the membranes present an increase in permeability from 1.12 to 1.93  $\text{L}/\text{m}^2\text{-h-bar}$  and also in salt rejection;  $\text{Na}_2\text{SO}_4$  from 95.9 to 98.9% and NaCl from 46.2 to 61.7% at 2000 ppm, compared to conventional membranes.

Ounifi et al. [66] studied a synthesis of novel polyamide composite nanofiltration membranes (NF-TFC) for water purification. The polyamide was deposited on a synthetic cellulose acetate (CA) supported by the interfacial polymerization method. The membranes were characterized by scanning electron microscopy (SEM) and Fourier transform spectroscopy (FT-IR). The authors studied the effect of varying CHMA concentration (0.2–2 wt %) on the morphology, porosity, water permeability, and rejection properties of the proposed membranes. The membrane water permeability ranged from 36.02 to 17.09  $\text{L}/\text{h-m}^2\text{-bar}$ . The rejection of  $\text{Na}_2\text{SO}_4$  and NaCl salts increased from 9 to 68% and from 38.41 to 89.4%, respectively, by increasing the CHMA concentration from 0.2 to 2 wt %.

### 3.1.4. Electrodeionization (EDI)

Electrodeionization technology has been studied in Mexico by Alvarado et al. [67] with the purpose of removing chromium (VI) from synthetic solutions with pH 5; their results show that an EDI process with an anionic bed is capable of removing 97.55% of chromium (VI) with an energy consumption of 0.91  $\text{kWh}/\text{m}^3$ , while an EDI process with a mixed bed is capable of removing 99.8% of chromium (VI) with 0.167  $\text{kWh}/\text{m}^3$ . Alvarado et al. [68] proposed a continuous ion exchange and electrodeionization process to remove chromium (VI) from effluents from the mining and electroplating industry. The

authors report that the implementation of anionic resin combined with a strongly acidic macro reticular cation exchange resin continuously removes 98.5% of chromium (VI) in the electrodeionization process.

Huang et al. [69] studied the use of activated carbon modified with nitric acid as electrodes in a desalination process of aqueous electrolyte solutions by capacitive electrodeionization. Their experimental results show that the modification they proposed can significantly increase the removal of salts from the solution because the desalination efficiency increases by approximately 15%, and the desalination kinetics improved in the form of a rate constant from 0.09208 to 0.09922. The authors found that the modification greatly increased the oxygen-containing functional groups on the activated carbon surfaces, which resulted in increases in capacitance and reduction in charge resistance, which is reflected in the improvement of the desalination process.

Otero et al. [70] experimentally studied the electrodeionization process to desalinate brackish water using a 4-compartment EDI cell. The authors proposed a mathematical model to analyze the distribution of current density in the EDI cell and its effect on ion removal in order to identify the ineffective regions of the cell. Their results show that the model they proposed is able to describe the effect caused by changes in concentration (0.01 and 0.02 M) and cell voltage (7, 10.5, and 14 V) on current density and desalination flux.

### 3.2. Phase Change Technologies

#### 3.2.1. Multi-Stage Flash

Although MSF technology arrived in Mexico in the 1980s [71] and, nowadays, there are several large-scale desalination plants that use this technology, research for the improvement of some point of this process or cases of study are limited.

Priego et al. [72] proposed different configurations to activate two thermal desalination systems (MSF and MED) by steam extraction in nuclear reactors, comparing them by means of exergy and thermoeconomic analysis. The evaluation proposed by the authors considers the reduction in electrical energy generation due to steam extraction at different positions in the secondary circuit of the nuclear reactor and the implications of the cost of electrical energy in each case. The performance of both desalination technologies was analyzed for GORs of 5, 10, and 15. Their results show the economic competitiveness and feasibility of nuclear desalination using the reactor steam mainstream. A GOR of 15 produces the highest amount of product water for all possible extraction positions in both desalination technologies. The authors conclude that the MSF technology offers the lowest cost for water production.

#### 3.2.2. MED

One of the thermal technologies that in recent years has gained interest in the area of thermal desalination research and development is the Multiple-Effect Distillation technology. López-Zavala et al. [15] studied a modified single-effect absorption cooling system (23 kW) with a novel energy and mass integration to a FLASH/MED desalination process that allows desalinating water and using it as a coolant to increase the cooling capacity. The proposed integration increases the cooling capacity from 23 to 2012 kW and increases the COP from 0.72 to 6.15.

Aguilar-Jiménez et al. [21] proposed a novel energy integration between a MED desalination system and an Organic Rankine Cycle (ORC) for the cogeneration of water and electrical energy. The thermal energy to activate both systems is supplied at the evaporator of the MED system, while the ORC is activated by a fraction of the steam produced in the first effect of the MED. A hybrid MED/ORC system producing 50 kW of electrical energy is 22% more efficient than a MED system without integration and requires only a 6.9% increase in the heat transfer area.

### 3.2.3. Humidificación-Deshumidificación (HDH)

In 1963, the University of Arizona, in collaboration with the University of Sonora, developed a system for the desalination of seawater activated with low-temperature solar energy. Hodges et al. [73] report that the system is composed of a flat plate solar collector field with a plastic cover ( $836 \text{ m}^2$ ), a packed tower humidifier with a height of 14.32 m and a diameter between 1.21 and 1.52 m, and a finned tube heat exchanger used as a dehumidifier through which  $13.62 \text{ m}^3/\text{h}$  of seawater circulates. The seawater to be desalinated is used as a cooling medium in the dehumidifier and then transported to the solar collector field to increase its temperature by between 5 and  $10^\circ\text{C}$ . Finally, it is atomized in the humidifier to evaporate and diffuse into the air stream, entering the dehumidifier to produce between  $9.46$  and  $10.92 \text{ m}^3/\text{d}$  of distilled water.

Martín-Dominguez et al. [74] studied a solar-activated HDH system designed to meet water demand in rural communities. Their system had two solar collectors, one for heating water and the other for heating the air before both entered into the humidifier. They also optimized the dehumidifier by incorporating a heat pipe to keep the condensation temperature close to the ambient temperature. Using a simulator, they concluded that it is recommendable to use the maximum amount of solar radiation available to heat the seawater and air previous to the evaporation process to produce about  $7.5 \text{ kg}/\text{m}^2\text{d}$ .

Tariq et al. [75] proposed a novel humidifier based on the Maisotsenko cycle for being integrated into an HDH desalination process. Their air saturation process included flow infiltration from the dry channel to the wet channel; their results show that for infiltration of 0.6 (60%), the maximum amount of water evaporation is obtained. The authors report that their proposed humidifier increases; by 30% water production, 46% recovery ratio, and 11% GOR compared to an HDH desalination system with direct contact humidifier.

Álvarez et al. [76] numerically studied a direct contact humidification-dehumidification system with open air and water cycle (OAOW). Their system does not use any type of packing material in the humidifier or dehumidifier; instead, it humidifies the air by atomizing hot water over it, and subsequently dehumidifies it by atomizing cold (distilled) water over it, both processes occur under counter-current flow patterns. They studied the influence of geometry by varying the height of the humidifier and dehumidifier. Their results show that water production is benefited by reducing the diameter and velocity of the atomized water droplets and increasing the temperature and flow rate of seawater. Their maximum production obtained is  $242.2 \text{ kg}/\text{h}$  with a height ratio of 2 between the humidifier and the dehumidifier. Their system did not present any type of energy integration; therefore, the maximum GOR obtained was 0.58.

### 3.2.4. Solar Still

The implementation of solar stills in Mexico began during the 1970s in the fishing villages of Puerto Chale and Punta Eugenia ( $1$  and  $1.5 \text{ m}^3/\text{d}$ , respectively), both located in the federative entity of Baja California Sur, funded by the Commission for the Exploitation of Saline Waters (CAAS, by its Spanish acronyms) for research purposes.

Later, in the 1980s, the Centro de Investigaciones Biológicas del Noroeste (CIBNOR) developed and tested—*channel solar stills*—built with concrete and glass cover with an inclination of  $\approx 30^\circ$ . Its first prototype had  $54 \text{ m}^2$  of the collection area and produced approximately  $0.2 \text{ m}^3/\text{d}$ . The same prototype was installed in Puerto Chale, Baja California Sur, with a catchment area of  $300 \text{ m}^2$  consisting of 6 solar channel stills with the aim of producing  $1 \text{ m}^3/\text{d}$ . The best production recorded for this plant was  $1.6 \text{ m}^3/\text{d}$  [73].

The Autonomous University of Baja California Sur (UABCS) also conducted research in the area of solar stills, studying prototypes similar to those built by CIBNOR, but their increased dimensions made them choose to call it a long-section solar still (LSSS). The dimensions of the prototype were 5 m wide, 10 m long, and 2.5 m high; their system produced  $0.175 \text{ m}^3/\text{d}$  in a typical day. An LSSS system was built and put into operation at El Partido Island, Baja California Sur, in 1993 [73].



Aguilar Castro [77] experimentally studied three configurations of a solar still under the climatic conditions of Ensenada, Baja, California, Mexico. The first configuration consisted only of the solar still, the second configuration included a solar collector as a water preheater, and the third configuration included a condenser external to the solar still. The maximum production obtained with each configuration was 3.86, 4.74, and 5 L/m<sup>2</sup>, with thermal efficiencies of 48.8, 60.1, and 66.8%, respectively.

Feria-Díaz et al. [78] studied the performance of a solar still using the model proposed by Dunkle and validated it with an experimental study on the coast of Veracruz, Mexico. They studied different operating parameters and variations in climatic conditions, obtaining an average production of 1.57 L/m<sup>2</sup>d. The authors conclude that solar distillation is a viable solution to meet low potable water demands in remote coastal populations.

#### 4. Conclusions, Outlook, and Future Challenges of Desalination in Mexico

Desalination capacity has increased by 46.35% worldwide from 2013 to 2022, while in Mexico, it has increased by 240% during the same period of time. This increase can be interpreted as a response to the water scarcity faced by Mexico in recent decades and to its coastline length, which allows the availability of space not to be a constraint for the installation and/or expansion of desalination plants. The main limitation of the expansion of desalination plants in Mexico is the initial investment. The main operators of desalination plants belong to the private sector, which generally has capital from external investment.

Desalination technologies began to be applied in Mexico at the end of the last century; however, there is still a lack of regulations for water extraction and brine discharge, which is a vital point to ensure that the desalination plant process does not have a negative environmental impact on nearby ecosystems. There are proposals for official Mexican standards (NOM) that were proposed more than 7 years ago by groups of experts, but they are still in the evaluation stage by the responsible agencies. Despite the fact that desalination capacity has increased significantly and thus brine production, the authorities have not expedited approval of the initiative, which helps to regulate the environmental impact caused by existing plants and future plants to be built. It is important to mention that brine discharge is generally carried out in coastal lines that receive tourist activity, having a negative impact on their flora and fauna due to the lack of environmental impact studies.

It is also important to create or update existing regulations on water exploitation in the sectors with the highest water consumption. In the agricultural, aquaculture, and livestock sectors, verify that the requirements for granting concessions of aquifer exploitation are met and increase the thoroughness of audits to ensure that the concessions granted are being carried out within the established limits. Seek to ensure that subsidies are granted preferentially to crops that are destined for national food security and not to private companies that base their profits on export sales. In this way, water will be used in the national territory, avoiding its virtual transfer [79].

In the industrial sector, promote the installation of new industries with high water consumption in the southeast region to increase its GDP, and not in the northern and central regions of the country, where it has the highest impact due to water stress. Increasing industrial activity in the southeast region implies an increase in the cost and time of exportation; however, it also implies a positive impact on the phenomenon of migration to the North of the country, which is currently a problem. Likewise, identify the existing industries that currently contribute the highest percentage to the water stress present in the north and center regions of the country, and propose their total or partial relocation gradually and/or their expansion towards the southeast zone, where their consumption does not compromise the availability of water for human consumption. In addition, this contributes to homogenizing the WSI in all the AHR of Mexico. Finally, establishing that the high consumer will have high tariffs or penalties when exceeding a permissible limit of consumption and/or failing to pay during a determined period of time, thus aiming to give greater value to water and promote its conservation as a good of national interest.

On the other hand, looking for the direct participation of research groups focused on optimizing water consumption in industrial processes similar to the already established water-intensive companies, with the purpose of making their processes more efficient or proposing alternative processes for obtaining water such as desalination, purification and/or potabilization on site. Furthermore, it is expected that these activities will increase the economic profitability and reduce the environmental impact of these industries, as well as increase their autonomy for water supply, reducing dependence on external factors that could compromise their reliable and continuous supply. Another intrinsic result is the increased training of personnel as a result of collaboration and interaction with groups of academic experts. In order to achieve a triple helix of academy–industry–government, it is necessary to promote the creation of programs or organizations with long-term projection and total autonomy to plan, manage and execute water and energy projects independently of the government in turn.

In the residential sector, state or municipal operating agencies should improve drinking water distribution networks to reduce leakage and spills ( $\approx 30\%$ ) because increasing desalination capacity and, thus, the availability of water in such places increases the waste of the latter.

Solar energy resources are abundant in most of Mexico's land area and other renewable energy sources, such as biomass or geothermal energy, are available in some regions. Different studies at the pilot plant scale have demonstrated the technical feasibility of developing desalination plants activated by renewable energy sources. In order to develop them quickly and efficiently, the combined participation of the academic, governmental, and industrial sectors is required, promoting common projects and/or economic incentives that can increase their socio-economic feasibility.

In Mexico, spatially and temporally isolated efforts have been made to satisfy the water demand of remote communities with medium and low-capacity desalination systems mostly powered by renewable energy due to the unavailability of access to the electrical grid. Successful pilot plant projects, such as the one reported by [80], should be taken as a model to replicate in remote coastal communities because the characteristics are similar. There are such projects, such as the case of the community Punta Chueca, Sonora, which has a PV-RO system with a capacity of  $150 \text{ m}^3/\text{d}$ . The government should consider the construction of desalination plants ideal to satisfy low capacities with low energy consumption. No matter how much the desalination capacity per federal entity is increased, if it is installed only in urban centers, it will have no impact on the quality of life of remote communities.

According to the latest projections on global water scarcity reported in the literature, Mexico is within the most affected zone. Therefore, it is necessary to improve water resource management and increase the installed desalination capacity in order to overcome the predicted water stress. The AHR 6, which includes most of the federative entities located in the north region of the country, currently presents the greatest water scarcity, together with low aquifer availability and remoteness from the coastline, which requires the installation of desalination plants and transport networks, the latter can be activated with renewable energy to seek self-sustainability, as proposed by Roggenburg et al. [33].

Based on the aforementioned, it is demonstrated that the 240% increase in desalination capacity during the last 10 years was not enough to overcome the water stress of recent years in Mexico, so it is necessary to seek greater increases through public policies that promote:

- Academy–Industry–Government joint participation.
- The improvement of the infrastructure of water transport/distribution networks to avoid water waste.
- Spatial homogenization of industrial activity to reduce consumption in water-stressed areas and increase GDP in the southeast regions.
- The implementation of new desalination plants powered by renewable energy in areas with energy deficits, primarily in remote communities.
- The value of water through rate increases or penalties in cases of excessive consumption and/or in the event that water is used for purposes other than those previously established.

- Seeking the sustainability of the virtual transfer of water carried out through the export of agricultural and industrial products.

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## **ANEXO C**

### **Comparison between Air-Exposed and underground thermal energy storage for solar cooling applications**

**Revista: Processes**

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## Article

# Comparison between Air-Exposed and Underground Thermal Energy Storage for Solar Cooling Applications

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**Abstract:** Solar energy is one of the main alternatives for the decarbonization of the electricity sector and the reduction of the existing energy deficit in some regions of the world. However, one of its main limitations lies in its storage, since this energy source is intermittent. This paper evaluates the potential of an underground thermal energy storage tank supplied by solar thermal collectors to provide hot water for the activation of a single-effect absorption cooling system. A simulator was developed in TRNSYS 17 software. Experimentally on-site measured data of soil temperature were used in order to increase the accuracy of the simulation. The results show that the underground tank reduces thermal energy losses by 27.6% during the entire hot period compared with the air-exposed tank. The electrical energy savings due to the reduction in pumping time during the entire hot period was 639 kWh, which represents 23.6% of the electrical energy consumption of the solar collector pump. It can be concluded that using an underground thermal energy storage tank is a feasible option in areas with high levels of solar radiation, especially in areas where ambient temperature drops significantly during night hours and/or when access to electrical energy is limited.

**Keywords:** underground thermal energy storage; UTES; thermal energy storage tank; solar thermal energy storage



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## 1. Introduction

Solar energy is recognized as one of the main alternatives for decarbonizing the electricity sector and reducing the energy deficit that exists in several regions of the world. In places with high levels of solar radiation, the energy collected by a solar capture system (i.e., photovoltaic panels, solar collectors) can be higher than the energy demand during sunny hours; however, the maximum demand can occur during non-sunny hours. Therefore, energy storage technologies have an important role in such applications.

The implementation of an energy storage device allows mitigating/reducing the intermittency of renewable energy sources such as solar energy. The inadequate design of an energy storage device can have negative impacts on the initial investment and/or the operating cost of the whole system. For this reason, the efforts of different research groups are focused on designing energy storage devices that provide an optimal matching between the temporal variation of users demand and the availability of the solar resource.

The selection of an energy storage technology depends directly on the type of solar energy application. In systems for electrical energy generation mainly with photovoltaic technologies, the electrical energy is generally stored in a battery bank. In systems for solar thermal energy collection, thermal energy is generally stored in a volume of fluid inside a container called a “tank”. Thermal energy storage can contribute significantly to



meeting society's desire for more efficient, environmentally benign energy use, particularly in the areas of building heating and cooling and electric power generation [1]. In addition, their high thermal power density has increased their implementation in industrial thermal processes [2].

Despite the fact that thermal storage systems have a low initial investment, long useful life, and simplicity of operation, they still present areas of opportunity to improve their efficiency. Some authors have studied the implementation of two thermal energy storage tanks with the purpose of storing fluid at low temperatures and feeding it into the solar collectors to increase its energy content and store it in another high-temperature tank.

Dannemand et al. [3] used two tanks to store thermal energy provided by PVT collectors to activate a solar-assisted heat pump. The authors report that compared with the single-tank thermal energy storage, the power consumption of the double-tank thermal energy storage system was reduced by 23%, and the heat loss was reduced by 11%.

They have also implemented phase change materials because they offer high thermal storage density with moderate temperature variation. Abdelsalam et al. [4] reported that increasing the number of PCM modules can increase the efficiency of heat storage and delay the time of water temperature drop. However, when the volume fraction of PCM reaches a certain value, it will affect the heat transfer of the heat exchange coil, and the heat storage efficiency becomes no longer sensitive. Therefore, increasing the charge rate requires an increase in the surface area of the heat exchanger coil.

Shen et al. [5] conducted a literature review on three types of solar-driven short-term low-temperature thermal energy storage technologies: thermal energy storage water tanks, thermal energy storage with phase change materials, and thermochemical thermal energy storage. The authors conclude that from an economic point of view, thermal energy storage water tanks are completely feasible in the global market. The application prospect of phase change material thermal energy storage in moderate climates is obviously stronger than that of tropical climates, while the thermochemical heat storage system can be economically feasible around 2030.

Other authors have proposed to take advantage of soil thermal inertia to improve the efficiency of the thermal storage subsystem. For this reason, they are called underground thermal energy storage systems (UTES). These underground storage systems are classified according to their geometry and enclosure material; the main technologies are: aquifer thermal energy storage (ATES), borehole thermal energy storage (BTES), pit thermal energy storage (PTES), and tank thermal energy storage (TTES).

The first attempt to store thermal energy in an underground tank occurred in 1939 in Cambridge, Massachusetts, with the purpose of storing solar thermal energy in the summer months to supply heating to a residential building during the winter months [6]. Since then, different UTES technologies for hot water storage have been investigated and proposed, with the main use being long-term storage between seasons.

Yumrutaş et al. [7] studied a space heating system consisting of a heat pump driven by flat-plate solar collectors and an underground cylindrical thermal energy storage tank. The authors developed an analytical computational model to determine the annual variation of the water temperature in the underground tank, the earth temperature field surrounding the tank, and the annual performance of their heating system. Based on their results, the authors report that their heating system is an alternative to fossil-fuel-fired heating systems. In addition, they mention that the soil surrounding the tank behaves as an additional storage medium, since it absorbs energy during summer and rejects the absorbed energy to the tank in winter.

Banjac [8] presented a sizing methodology for a space heating and cooling system which consists of a heat pump, solar collectors, and an underground thermal storage tank for seasonal (or long-term) thermal energy storage. Numerical calculations show that the size of the underground tank has the greatest impact on water temperature variation, and that the amplitude of temperature oscillation decreases with the increase in the tank

volume. Furthermore, it is shown that the tank burial depth had a minor effect on the change of water temperature in the tank.

Nhut et al. [9] established a mathematical model for the study of a residential heating system activated by solar thermal energy with a buried thermal tank for seasonal storage. Their results revealed that the optimum volume of the underground tank is 5 m<sup>3</sup> and that on days with clear sky, intermittent cloudy sky, and cloudy sky, the solar fraction obtained was 45.8%, 17.26%, and 0%, respectively. The authors recommend that a residential house solar-assisted heating system with a thermal storage tank should be replaced by the solar-assisted heating system with a boiler and a seasonal underground thermal energy storage tank, because it will increase the solar fraction.

Meister and Beausoleil-Morrison [10] experimentally studied a solar thermal system with a 36 m<sup>3</sup> underground thermal energy storage tank for seasonal storage. The authors report that low efficiency of solar collectors has the most detrimental effect on system performance. The results indicate that the system can achieve 100% solar fraction for space heating loads (15 GJ) and 86% for domestic hot water loads (13 GJ). The authors report that the seasonal storage tank achieves an annual energy storage efficiency of 42%.

The main application of the UTES is for seasonal or “long-term” thermal energy storage, in order to have solar thermal energy availability in places where the solar radiation decreases significantly during autumn–winter months. Their main application is to provide heating services in buildings. This research gap presents an opportunity to evaluate the potential of UTES for daily or “short-term” thermal energy storage for solar cooling purposes in zones with hot climates. This paper evaluates the potential of a UTES for short-term thermal energy storage supplied by solar thermal collectors to provide hot water for the activation of a single-effect absorption cooling system, considering as a case of study the off-grid remote community of Puertecitos, located in northwest Mexico where rational consumption of electrical energy is essential and solar radiation levels present high values throughout the year.

## 2. Case of Study

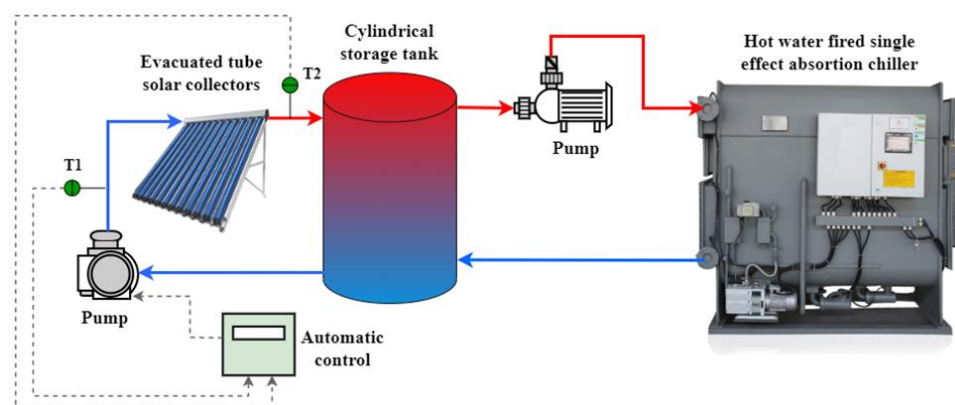
This work considers as the case of study the current conditions of the solar absorption cooling system (35 kW) of Puertecitos school reported by Aguilar-Jiménez et al. [11]. The solar absorption cooling system is activated by a solar thermal energy collection and storage subsystem shown in Figure 1, consisting of 25 evacuated tube heat-pipe solar collectors with 110 m<sup>2</sup> of collection area (arrangement of five collectors in series and five in parallel) and a thermal energy storage tank exposed to the ambient air with a height of 2.5 m, glass fiber insulation thickness of 0.025 m, and a volume of 12 m<sup>3</sup>.



**Figure 1.** Solar thermal energy collection and storage subsystem (Aguilar-Jiménez et al., 2020) [11].

Figure 2 shows schematically the arrangement of the thermal energy storage tank. Thermal energy input comes from solar collectors, increasing the water temperature inside the tank and positioning the water with higher temperature in the upper part of the tank through the physical phenomenon of stratification caused by the density differential. The

water with lower density and higher temperature inside the tank is used for the activation of the solar absorption cooling system. The water with higher density and lower temperature is used for energy capture in the solar collectors.



**Figure 2.** Schematic diagram of the system under study.

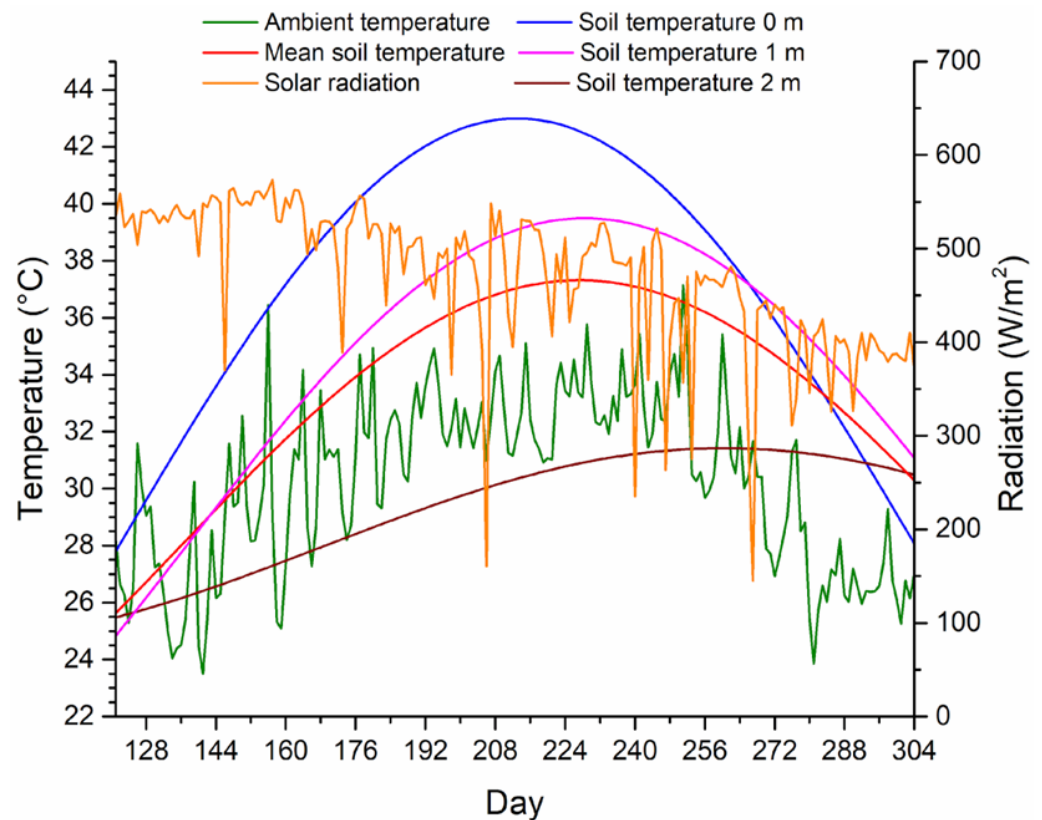
During weekdays, the solar absorption cooling system operates during the occupancy hours of the Puertecitos school, which are from 8:00 h to 15:00 h. During weekends, the absorption cooling system does not operate, but the solar thermal energy collection and storage subsystem may be turned on to increase the water temperature in the thermal energy storage tank. Sunny hours in this community generally range from 6:00 h to 18:00 h. Therefore, during weekdays the solar energy collection and storage subsystem begins collecting energy before the solar absorption cooling system is turned on and can be found working after school hours to increase the water temperature in the thermal energy storage tank.

The pump that circulates water between the thermal energy storage tank and the solar collectors has a rated power of 3.5 HP ( $\approx 2612$  W) and starts up when the water temperature at the outlet of the solar collector rows is higher than the average temperature of the hot water tank.

Figure 3 shows the variation of daily ambient temperature, soil temperature, and solar radiation averages during the warm season (May–October), since these are the two surrounding mediums with which the thermal energy storage tank of the solar absorption cooling system exchanges heat. Although the ambient temperature reaches values of 40 °C during sunny hours in the period from June to September, at night it presents values between 25–30 °C, which increases the temperature difference with respect to the tank temperature ( $\approx 90$  °C) and, subsequently, the heat losses.

The community of Puertecitos is a coastal remote community without access to the electrical grid. The electrical energy supply comes from a solar PV microgrid with electrical battery storage. For this reason, it is important to develop and implement actions and/or strategies to reduce the energy consumption of electrical devices that consume electrical energy from the microgrid. The proposal of burying the thermal energy storage tank of the solar absorption cooling system aims to reduce the heat losses dissipated to the surrounding ambient air, and thus maintaining the desired temperature in the tank ( $\approx 90$  °C) for a longer period. As a result, reducing the operation time of the pump that circulates water between the thermal energy storage tank and solar collectors, decreases its electrical energy consumption.





**Figure 3.** Variation of ambient temperature, soil temperature, and solar radiation during the hot period.

### 3. Methodology

This section presents the methods used to carry out an operational study on the performance of the proposed underground thermal energy storage tank, which consists of three main points: features of TRNSYS simulation, experimental validation of soil temperature at the study site, and calculation of the heat transfer coefficients for the air exposed (base case) and the underground thermal energy storage tank cases.

#### 3.1. TRNSYS Simulation

TRNSYS 17 software [12] was used in order to quantify the potential of the proposed underground thermal energy storage tank for short-term energy storage, because this software is suitable for simulating the annual performance of different energy systems.

Meteorological data from Puertecitos, Baja California, Mexico ( $30.21^\circ$ ,  $114.38^\circ$ ) measured during the year 2020 by a weather station were used in this simulation. The data generated by the weather station was entered into the Meteonorm v7 software [13] in order to obtain an Energy Plus Weather file (EPW) compatible with the TRNSYS “Type 15”.

Output meteorological variables such as ambient temperature and solar radiation processed by “Type 15” are fed as input variables into the mathematical model of the evacuated tube solar collector (Type 71) and the thermal energy storage tank (Type 534). Subsequently, the solar collector (Type 71) output variables (temperature and mass flow rate) are connected to the thermal energy storage tank (Type 534). The hydraulic pump (Type 3) enables the flow of hot water between the thermal energy storage tank and the solar collectors as shown in Figure 4.

The hot water ( $\approx 90^\circ\text{C}$ ) stored in the thermal energy storage tank is pumped to the absorption cooling system (Type 107). Table 1 shows the TRNSYS “Types” used to evaluate both the base case and the proposed underground thermal energy storage tank.



The system simulation was conducted during the hot period of the year at the study site, which starts on 1 May (2880 h) and ends on 31 October (7296 h). Time steps of 1 h were set and “Type 41” was configured to turn off the absorption cooling system during weekends, holidays (15 May and 16 September), and vacations (1 July–25 August). The solar thermal energy collection and storage subsystem pump is turned off during vacations only. The performance analysis of both cases (air-exposed and underground tank) was focused on the operational start-up week of the solar thermal energy collection and storage subsystem (2977–3096 h) and the week with the highest thermal load in the classrooms during the hot period (5833–5952 h). Table 2 shows the parameters specified to “Type 534”.

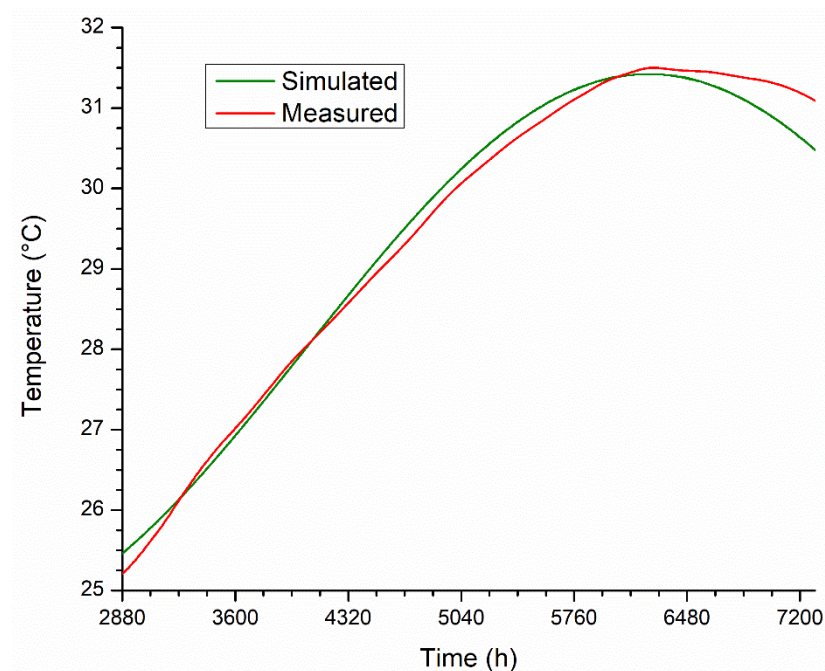
**Table 2.** Specifications of the thermal storage fluid and tank.

| Parameter                          | Specification |
|------------------------------------|---------------|
| Tank volume [m <sup>3</sup> ]      | 12            |
| Tank height [m]                    | 2.5           |
| Fluid specific heat [kJ/kg·K]      | 4.2           |
| Fluid density [kg/m <sup>3</sup> ] | 965           |
| Fluid thermal conductivity [W/m·K] | 0.68          |
| Fluid viscosity [kg/m·h]           | 1.134         |

### 3.2. Soil Temperature Validation

The experimental data were measured with HOBO model TMCx-HD temperature sensors (ONSET BRANDS, Norwalk Connecticut, CT, USA) placed at a depth of 2 m in a vertical well drilled at the Puertecitos school. An analog recorder HOBO model UX120-006M was used for data acquisition.

Figure 5 shows the results of the experimental validation for the soil temperature variation in Puertecitos at 2 m depth. The coefficient of determination  $R^2$  is equal to 0.9901 and the mean absolute error (MAE) value is 0.1572 which indicates that the TRNSYS model to obtain the soil temperature is representative of the real phenomenon. The largest temperature difference between the model and the experimental measurement occurs at the end of the hot period (7293 h) and it is equal to 0.61 °C.



**Figure 5.** Validation of soil temperature modeling at 2 m depth.

The parameters used for “Type 77” to calculate the soil temperature variation based on the Kusuda and Achenbach [14] model are presented in Table 3. The parameters  $T_m$ ,  $A_0$ , and  $t_0$  represent the annual average ambient temperature ( $^{\circ}\text{C}$ ), the amplitude of annual ambient temperature variation ( $^{\circ}\text{C}$ ), and the time at which the lowest annual temperature occurs ( $^{\circ}\text{C}$ ). The parameters  $k$ ,  $\rho$ , and  $C_p$  represent the thermal conductivity, density, and specific heat of the soil, respectively, and were obtained from the ground thermal properties database reported by Dalla Santa et al. [15] considering a wet clayey-silt soil.

**Table 3.** Parameters used for soil temperature calculation.

| $T_m$ ( $^{\circ}\text{C}$ ) | $A_0$ ( $^{\circ}\text{C}$ ) | $t_0$ (h) | $k$ (W/m·K) | $\rho$ (kg/m <sup>3</sup> ) | $C_p$ (J/kg·K) |
|------------------------------|------------------------------|-----------|-------------|-----------------------------|----------------|
| 28                           | 10                           | 360       | 1.45        | 2000                        | 1200           |

### 3.3. Heat Transfer Coefficient

#### 3.3.1. Air-Exposed Tank

The heat transfer phenomenon studied to determine the heat losses through the thermal energy storage tank exposed to ambient air is the natural convection on the top and walls. The equations to calculate the heat transfer natural convective coefficient “ $h$ ” depend mainly on its geometry and orientation.

In the literature, a wide variety of correlations and dimensionless numbers are available that vary according to the geometry and orientation of the tank; as reported by Çengel and Ghajar [16], a vertical cylindrical container can be treated as a vertical plate when the condition stated in Equation (1) is met:

$$D \geq \frac{35 * L}{Gr_L^{1/4}} \quad (1)$$

where “ $D$ ” represents the tank diameter, “ $L$ ” the tank height, and “ $Gr_L$ ” the Grashof number. The Grashof number must be calculated using Equation (2); this dimensionless number represents the effects of natural convection:

$$Gr_L = \frac{g\beta(T_s - T_{\infty})L_c^3}{\nu^2} \quad (2)$$

where “ $g$ ” represents the acceleration of gravity, “ $\beta$ ” the coefficient of volumetric expansion, “ $T_s$ ” the plate surface temperature, “ $T_{\infty}$ ” the temperature of the fluid in contact with the plate surface, “ $L_c$ ” the height of the plate, and “ $\nu$ ” the kinematic viscosity of the fluid.

For this case study, the condition established in Equation (1) is satisfied, and hence the equations corresponding to a vertical plate were used. Subsequently, it is necessary to calculate the Rayleigh number using Equation (3), which describes the relationship between buoyancy and viscosity within the fluid:

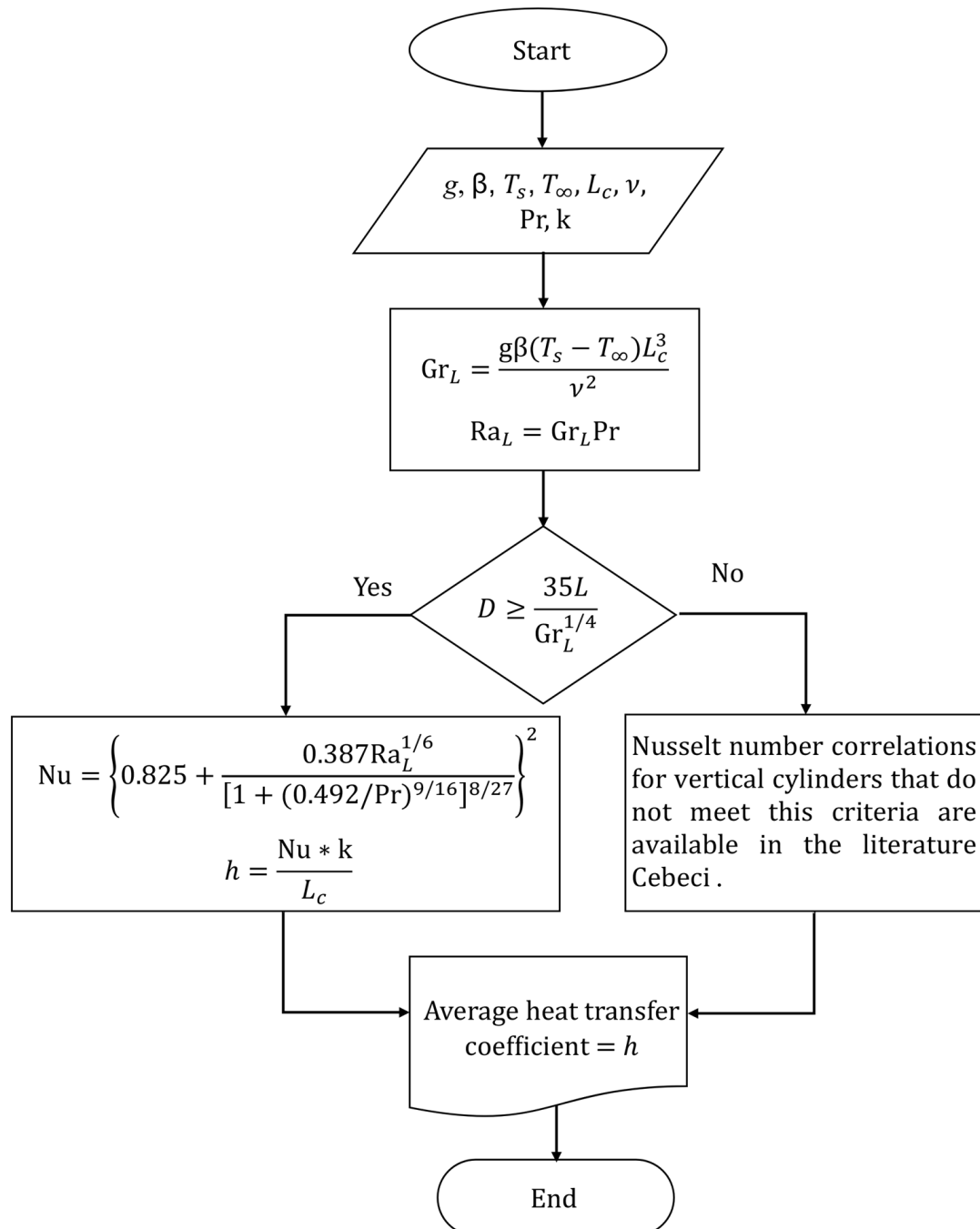
$$Ra_L = Gr_L Pr \quad (3)$$

where “ $Pr$ ” represents the Prandtl number, used to describe the relationship between momentum diffusivity and thermal diffusivity. The last dimensionless number required for the calculation of the convective coefficient is the Nusselt number obtained by Equation (4), which is an empirical correlation in natural convection for vertical plates:

$$Nu = \left\{ 0.825 + \frac{0.387 Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2 \quad (4)$$

Finally, the natural convection heat transfer coefficient “h” was calculated using Equation (5). Figure 6 shows a flow chart summarizing the methodology used for the calculation of the natural convection heat transfer coefficient.

$$h = \frac{Nu * k}{L_c} \quad (5)$$



**Figure 6.** Calculation sequence of natural convection heat transfer coefficient (Cebeci, 1974) [17].

### 3.3.2. Underground Tank

For the calculation of the heat transfer coefficient corresponding to the underground tank case, Equation (6) reported by Big Ladder software (Denver, CO, USA) was used. This software provides software tools, training, support, and consulting services for Energy-

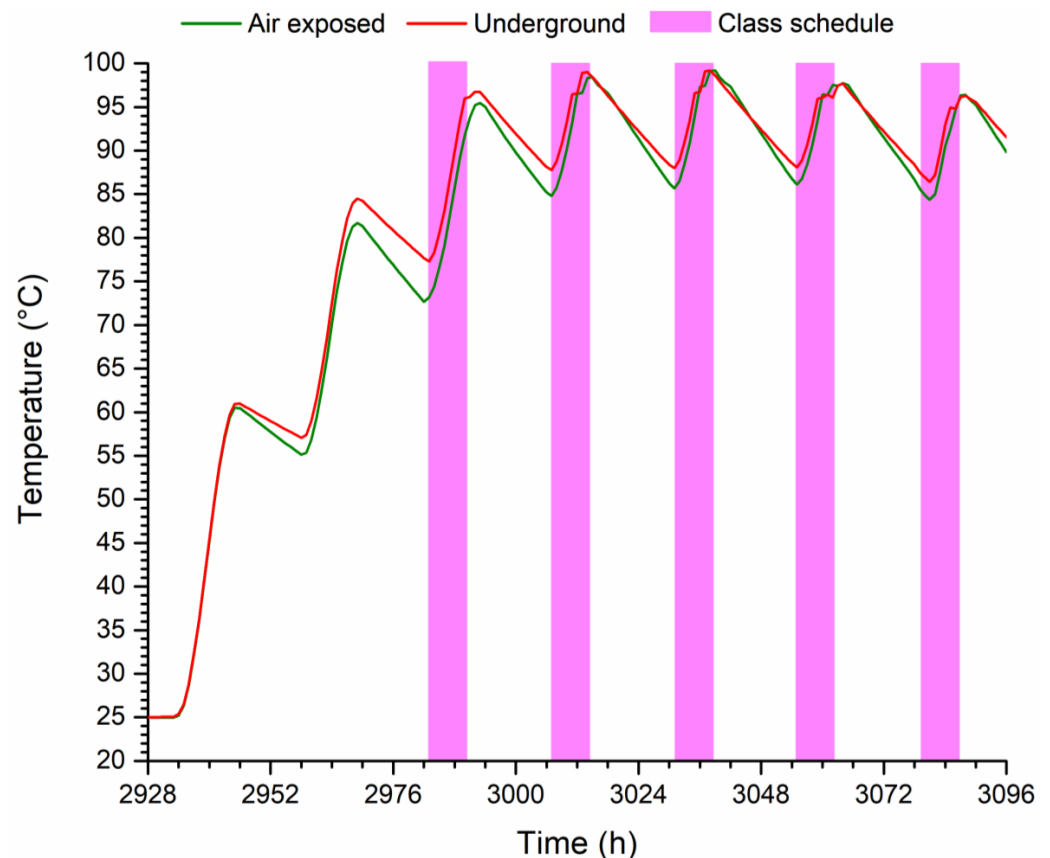
Plus [18] to mechanical, architectural, and design engineering firms focused on the building energy modeling sector.

$$R_{soil} = 0.0607 + 0.3479 \cdot z \quad (6)$$

where “ $R_{soil}$ ” represents the effective thermal resistance of the soil and is the inverse of the heat transfer coefficient, and “ $z$ ” represents the depth into the soil from the surface.

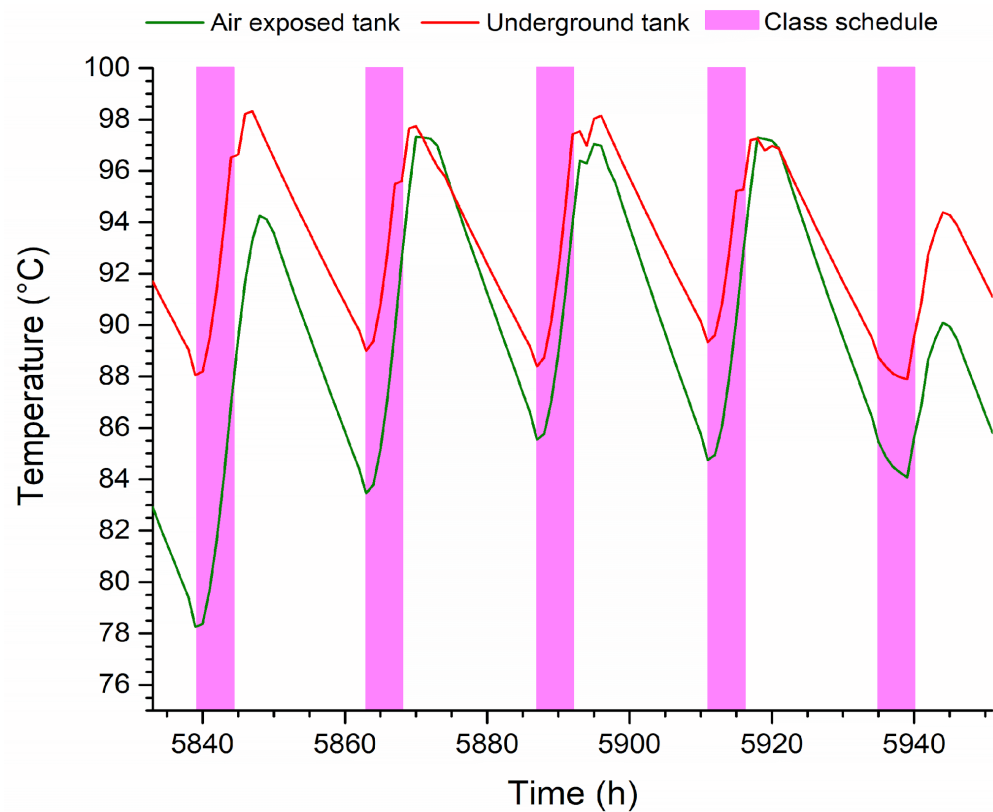
#### 4. Results

Figure 7 shows the tank’s water temperature behavior during the absorption cooling system in the first week of operation (2977–3096 h). The solar thermal energy collection and storage subsystem started operating at the weekend (2929–2976 h) in order to reach the required temperature ( $\approx 90^\circ\text{C}$ ) to activate the absorption cooling system. For Monday, at the beginning of school hours (2982 h) the temperature of the underground tank is  $77.6^\circ\text{C}$  and the air-exposed tank is  $72.6^\circ\text{C}$ ; it is worth mentioning that this is the largest temperature difference ( $\approx 5^\circ\text{C}$ ) between both cases during this week. During the rest of this week, the temperature of the underground tank is on average  $2.1^\circ\text{C}$  higher than the air-exposed tank at the start of the absorption cooling system operation.



**Figure 7.** Variation of the tank water temperature during the start-up week.

Figure 8 shows that the temperature in the underground tank has a more stable dynamic during the highest classroom thermal load week, and the amplitude of the temperature variation is lower compared with the air-exposed tank. At the first hour of Monday (5833 h), the temperature of the buried tank is  $91.6^\circ\text{C}$ , whereas the temperature of the air-exposed tank is  $82.75^\circ\text{C}$ . During school hours, when water is extracted from the tank to activate the absorption cooling system, the temperature difference in the tanks decreases considerably. At the end of the week, the temperature of the underground tank is  $91.1^\circ\text{C}$  and  $85.8^\circ\text{C}$  for the air-exposed tank.



**Figure 8.** Tank water temperature behavior during the highest thermal load week.

Figure 9 shows that the temporal variation of heat losses follows the same trend as the water temperature in the tank, because as the temperature in the tank increases, the temperature difference with respect to the heat sink (air or soil) increases. The underground tank reduces heat losses during the critical week by 27.33% compared with the air-exposed tank. The maximum rate of thermal energy that dissipated to the soil surroundings is 8.1 kWh, while, for the case of the air-exposed tank, this is 11.5 kWh. The increase/decrease in thermal energy dissipated to the soil surroundings has a more linear trend because the soil temperature change during a day is negligible, as opposed to ambient air which has a greater temperature variation.

Figure 10 shows the electrical energy savings generated by the reduction of pumping time, because the pump that circulates water between the tank and the solar collectors turns on when the temperature at the solar collector row outlet is higher than the temperature of the tank. The reduction in thermal energy losses in the underground tank causes its temperature to be higher than the outlet temperature of solar collectors for a longer period of time. Monthly electrical energy savings were 146.21 kWh (22.3%) in May, 193.2 kWh (24.5%) in June, 31.3 kWh (18.7%) in August, 172.32 kWh (24.26%) in September, and 96.6 kWh (25.17%) in October.



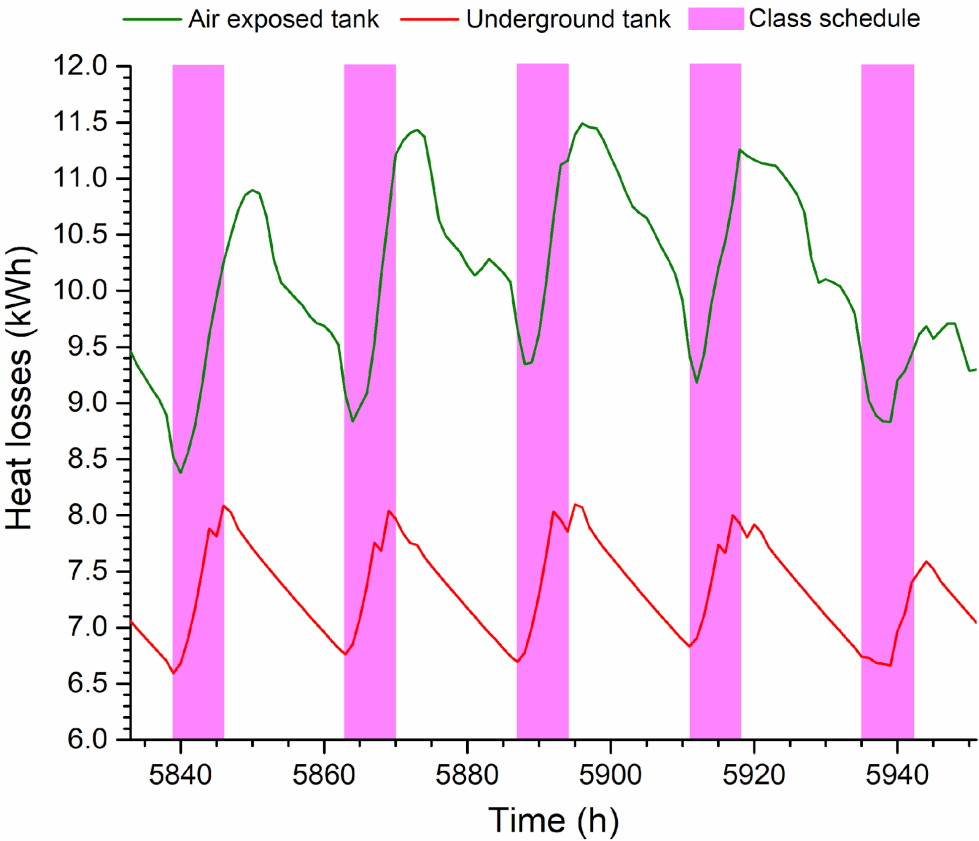


Figure 9. Variation of tank heat losses during the highest classroom thermal load week.

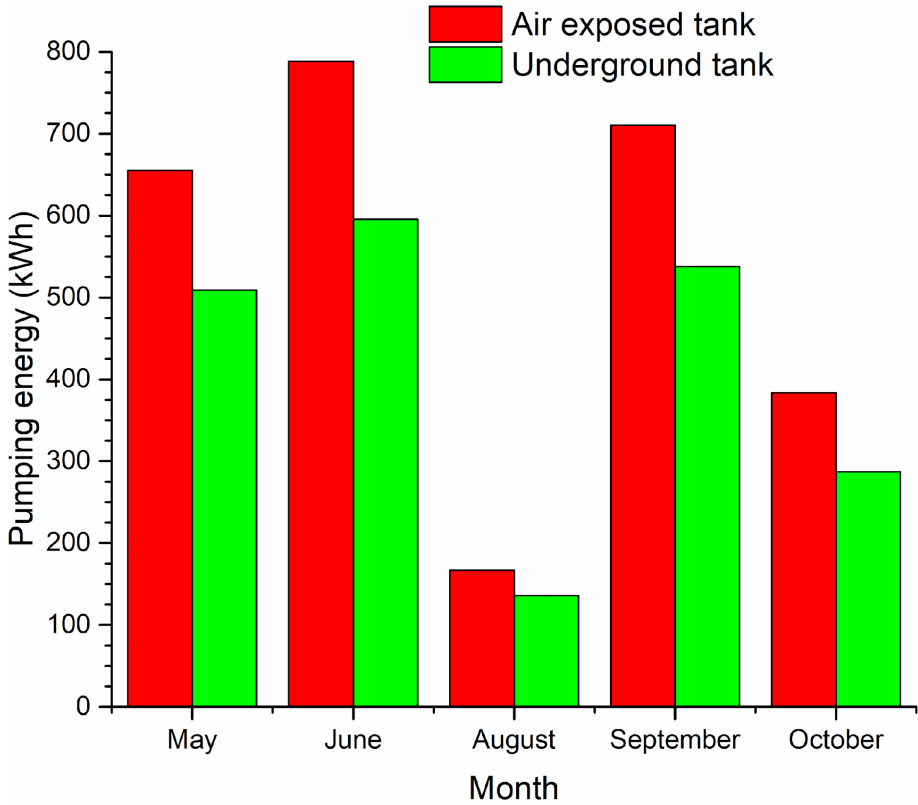


Figure 10. Electrical energy savings due to reduction in pumping time.



## 5. Conclusions

This paper evaluates the potential of a UTES for short-term thermal energy storage supplied by solar thermal collectors to provide hot water to activate a single-effect absorption cooling system, considering as a case of study the off-grid remote community of Puertecitos, located in northwest Mexico where rational consumption of electrical energy is essential and solar radiation levels present high values throughout the year.

The results show that the underground tank reduces thermal energy losses by 27.6% during the entire hot period, during the first stage of the hot period (May–June) by 28.1%, and during the second stage (August–October) by 24.9%, compared with the air-exposed tank. The underground tank presented higher savings during the first stage of the warm period (May–June) due to higher solar radiation values but lower ambient temperature values compared with the second warm period (August–October).

The electrical energy savings due to the reduction in pumping time during the entire hot period was 639 kWh, which represents 23.6% of the electrical energy consumption of the pump that circulates water between the tank and the solar collectors. During the first stage of the hot period (May–June), the electrical energy savings was 339.43 kWh and for the second stage it was 300.2 kWh, representing 23.5% and 23.8%, respectively. Although during the second stage the percentage of savings was higher, the performance of the underground tank was better during the first stage because it saved more electricity by 39.2 kWh and its duration was shorter by 8 days.

Based on the results, it can be concluded that using an underground tank for short-term thermal energy storage is a feasible option in areas with high levels of solar radiation, especially in areas where ambient temperature drops significantly during night hours and/or when access to electrical energy is limited. To reduce the initial investment, the excavation for burying the tank can be carried out simultaneously and using the same machinery required for constructing the foundation of a control room or other structure. In addition, the underground tank may have a longer service life by avoiding corrosion due to contact with saline environments in coastal communities; it also increases the availability of space at the installation site and can prevent vandalism.

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## **ANEXO D**

**Use of sensible and latent condensation heat of an organic rankine cycle in a multiple-effect desalination system: A comparative study of novel energy integrations**

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# Use of sensible and latent condensation heat of an organic Rankine cycle in a multiple-effect desalination system: A comparative study of novel energy integrations

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## Summary

This paper presents a comparative study of novel energy integrations between Multiple-Effect Desalination (MED) and Organic Rankine Cycle (ORC) systems for the simultaneous production of desalinated water and electric power. Two new MED/ORC configurations are proposed, where part of the sensible and/or latent heat of condensation of the ORC is used to potentiate the generation of steam in the last effect of the MED, taking advantage of the temperature levels present in the MED. A simulator of the proposals was developed and validated with experimental results reported in the literature. These configurations are compared energetically with the MED and ORC systems working individually and against another previously proposed configuration, also considering the impact of the integrations on the heat transfer area required by the systems. The results show a decrease in the thermal energy requirements for the activation of the MED system with the new integrations. The energy requirements of the MED are reduced by 28% when integrated into the first effect with the ORC evaporator. The utilization of the sensible heat of condensation of the ORC in configuration 3 did not present a considerable energy decrease compared to configuration 2, due to the limited sensible heat available for its recovery, of approximately 22 kW. The usage of the latent heat of condensation of the ORC in configuration 4 implies that increasing the condensation temperature and pressure of the ORC, decreases its efficiency by 1.8% and increasing the energy required by 221 kW provides the two services simultaneously.

## KEYWORDS

desalination, electricity, energy integration, MED, ORC

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**Abbreviations:** MED, Multiple-Effect Desalination; ORC, Organic Rankine Cycle; MSF, Multi-Stage Flash; RO, Reverse Osmosis; TBT, Top Brine Temperature; PR, Performance Ratio; ppm, Parts per-million; LMTD, Log-Mean Temperature Difference.

## 1 | INTRODUCTION

The simultaneous production of electric power and water desalination is an area that has triggered great interest in the scientific community, due to the importance of these resources and the different environmental, energy, and economic problems, among others, involved in their production and distribution. Water scarcity is a problem that has been increasing over time in most parts of the world, and it is becoming more difficult to have a safe and clean source of water for human consumption. The United Nations incorporates it into its Sustainable Development Goal (SDG) 6.<sup>1</sup> On the other hand, SDG 7 claims to ensure universal access to affordable, reliable, and modern energy services. Globally, great efforts are being made to achieve this goal, but we are still far from meeting the target set for 2030 at the current rate of growth.<sup>2</sup> Therefore, new, more efficient, and cheaper technologies should be proposed, seeking a sustainable development in the generation of electric power and water desalination.

The integration of desalination technologies with power cycles for the production of fresh water and electricity has been crucial in recent years in the scientific community.<sup>3,4</sup> The wide variety of desalination technologies that use thermal and electrical activation energy sources allows their integration with energy systems of different characteristics, to take advantage of the thermal energy effluents and/or directly with the electrical energy. Examples of these widely studied integrations are those made between MED,<sup>5</sup> multi-stage flash (MSF),<sup>6</sup> solar distillation<sup>7</sup> and reverse osmosis (RO)<sup>8</sup> with Rankine cycles, as well as with Brayton cycles<sup>9</sup> and ORC.<sup>10</sup> In the article by Omar et al,<sup>3</sup> the couplings of solar thermal electric power generation systems with desalination systems were reviewed, highlighting five main configurations, among which the most studied in the state of the art is related to the use of waste heat from power cycles in desalination systems. Ortega-Delgado et al<sup>11</sup> studied the effect of using different pressures in the steam activation of a MED with thermal vapor compression, using steam extractions at different stages of a Rankine cycle turbine. They mention that the best configuration is to take advantage of the low-pressure turbine steam by placing the thermocompressor close to the fifth effect of the MED, which allows for a better electrical production, but reduces the efficiency and water production of the MED. The large amounts of thermal energy that must be removed in the condenser of power cycles such as Rankine, allow it to be used in low temperature applications, such as desalination. In this context, Askari and Ameri<sup>5</sup> studied the coupling of a MED system to a solar-powered Rankine cycle, where the MED replaced the power cycle condenser. To achieve the operating temperature

conditions of the MED requires modifications in the condensing temperatures of the Rankine cycle, which ended up reducing its thermal efficiency from 37% to 31%, increasing the cost for the services produced.

In scenarios where a Rankine cycle, due to its operating conditions, makes its application impossible, such as isolated populations with low population and low electricity demands, as well as low availability of activation energy resources, the hybridization of MED systems with ORC can be a technically attractive option. ORC's have the advantage that they can be more efficient than traditional Rankine cycles in applications where the energy source is of low temperature, such as waste heat, solar energy and low enthalpy geothermal energy.<sup>12</sup> Also, by using organic fluids as working fluids, with temperatures and evaporation pressures lower than those of water, they can be integrated with different technologies to take advantage of energy effluents.

Baccioli et al<sup>13</sup> studied the integration between MED and ORC systems when waste heat is used as a source of activation energy. For this purpose, they proposed two integrations; first, part of the ORC condensation heat is used as preheating of the MED, while subsequently all the activation energy of the MED comes from the ORC condensation heat. When only condensation heat is used for preheating, and considering the condensation and first effect temperatures of the MED, this integration is limited to only the utilization of sensible heat by the ORC fluid, which energetically is not as significant as the latent heat that must be removed in the condenser. On the other hand, if the MED is fully activated with the condensing heat of the ORC, it is necessary to raise the condensing temperature and pressure of the ORC in order to use both sensible and latent heat, which causes the efficiency of the ORC to decrease. The same effect of decreasing ORC efficiency by using its condensation heat in MED systems can be seen in several studies. Kerme et al<sup>14</sup> proposed a solar polygeneration system and they found that by increasing the condensing temperature from 75 to 150°C, in order to take advantage of the condensation heat in the MED, the ORC conversion efficiency decreases from 16.5% to 11.4%. The use of gas turbine combustion gases to first generate steam for the activation of MED systems and, subsequently, to generate more electrical energy through ORC systems<sup>15,16</sup> has also been studied. This configuration does not modify the operating conditions of the systems, but it also does not make use of the effluent between the MED and the ORC.

Makkeh et al<sup>17</sup> studied a cogenerative system for electric power production using an ORC and wind turbines, and seawater desalination using a MED and RO system. Using solar thermal energy as the driving energy and under the climatic conditions of Chabahar, Iran, they achieved exergy efficiencies of 26.2% with a desalinated

water cost of 3.08 USD/m<sup>3</sup>. They decreased the energy required by the RO system by 8.9% by combining its feed water with the hot water at the outlet of the MED condenser. Cao et al.<sup>18</sup> analyzed a multi-generative system activated with concentrating solar power and biomass for the simultaneous production of electricity, desalinated water, and cooling. The novelty of this system is that it takes advantage of the cold recovery of liquefied natural gas to condense the ORC, while using biomass as a means of energy activation of the MED system.

Furthermore, Mohammed et al.<sup>19</sup> studied an ORC integrated with a MED for the simultaneous production of electricity, cooling, heating, and desalinated water. Cooling is achieved by an expansion valve that works with the same fluid as the ORC by separating streams, while heating is generated using some of the waste heat from the power cycle. The desalination system is fully activated with the condensation heat from the ORC. It can be seen that increasing the condensing temperature of the ORC from 26 °C to 40 °C increases water desalination by 25% while reducing electricity production by 20%. In a similar study, Tayyeban et al.<sup>20</sup> proposed a system composed of an ORC and a MED system, where the MED replaces the condenser of the ORC. The working fluid after thermally activating the MED is taken to an expansion valve for cold production and then brought to the evaporating pressure of the ORC by means of a compressor. Their results show that electrical power production in the ORC is very sensitive to variations in operating pressure; increasing the ORC pressure impairs electrical power production but benefits water desalination and cooling.

Hekmatshoar et al.<sup>21</sup> proposed a multi-generative system for the electricity production, desalinated water, and hydrogen by means of a MED system, an ORC, and an electrolyzer, all of them activated with geothermal energy. Only the effect of the variation of the mass flow rate of the geothermal fluid on factors such as system efficiency and production costs was studied, without any energy integration or heat recovery between them. The literature shows a variety of integration of MED and ORC systems with other technologies to offer multiservices, such as gas turbines and fuel cells,<sup>22</sup> traditional Rankine cycles,<sup>23</sup> in combination with RO<sup>24</sup> and using geothermal energy,<sup>25</sup> to mention a few, but there is no novel integration that benefits both systems or takes advantage of energy effluents.

The studies described above prove that there is great interest and potential for improvement between MED/ORC technologies, especially in the way in which they can be energetically integrated to favor their conversion efficiency. The research shows that the technical feasibility of these couplings is directly influenced by the operating characteristics of both systems, since most of the results mention that, by improving the operating

conditions of one system, it ends up affecting the other, so a coupling must be achieved to minimize the impact between both technologies or to allow an improvement in general. When energy integration between an ORC and a MED is proposed, the condensation heat of the ORC is usually used as a means of thermal activation of the MED, which causes an increase in the condensation pressure of the ORC and therefore decreases its efficiency. It is necessary to propose new energy integrations that increases the efficiency of both systems in a novel way, without having to choose to affect one technology to benefit the other.

This paper presents a comparative study of novel energy integrations between MED and ORC systems that have not been proposed before, analyzing their coupling to simultaneously produce desalinated water and electric power. Two new MED/ORC integrations are proposed; one where part of the sensible heat is used and the other where the entirety of the sensible heat is used with part of the latent heat of condensation of the ORC, to increase the generation of steam in the last effect of the MED, taking advantage of the temperature levels. These energy integrations are compared with the systems operating individually and against the integration proposed by Aguilar-Jiménez et al.<sup>26</sup> and the benefits and limitations of the new integrations are presented. A comparative study of heat transfer areas is also presented to verify the impact of the proposed integrations on the different configurations. This article aims to demonstrate new energy integration options between water desalination systems and electric power generation, taking advantage of the energy effluents of one system for the benefit of the other. It is worth mentioning that the configurations where the sensible and latent heat of condensation of the ORC are used in the last effect of the MED have not been presented in previous works, and there is no energy comparison between these new integration proposals, so this work allows us to dimension the benefits of the new energy integrations between MED and ORC systems.

Having defined the state of the art, the main contributions and novelties of this article are:

- A new energy integration between MED and ORC systems is proposed, where the sensible heat of condensation of the ORC is used to potentiate the desalination of seawater in the last effect of the MED, which has not been proposed before.
- The effect of taking advantage of sensible and latent heat from the ORC in the last effect of the MED is studied, making modifications to the operating conditions of the ORC, to potentiate seawater desalination, which has not been yet been proposed.
- An energy comparison of 4 different MED and ORC configurations, never before studied as a whole, is carried out.



## 2 | DESCRIPTION OF CONFIGURATIONS

### 2.1 | Configuration 1: Independent MED and ORC

The comparative study considered the MED and ORC systems operating independently, that is, without energy integration between them, as shown in Figure 1. The ORC has the basic configuration consisting of an evaporator, an expansion device, a condenser, and a pump. The organic fluid enters as saturated vapor to the expander and exits as superheated, to later remove its sensible and latent heat in the condenser and be turned into saturated liquid. During this process, electrical energy is generated in the expander. The pump is responsible for pressurizing the liquid again to the evaporation pressure, thus repeating the cycle.

On the other hand, the MED system consists of an arrangement of 14 effects where the activation heat is supplied directly in effect 1, evaporating part of the incoming seawater at a temperature close to TBT. This steam is then used as a medium to generate more steam

in the next effect, which is at a lower pressure and temperature, repeating the process until the last effect. Seawater is used to condense the last steam generated in effect 14, where some of the water is fed into the MED system as feedwater. The latent heat of condensation of the steam generated by flash, due to the decrease in pressure that occurs when the brine flow passes from effect to effect, is used as preheating of the feed water before entering the first effect.

### 2.2 | Configuration 2: MED and ORC with integration in the first effect

The second configuration studied integrates the ORC energetically with the MED desalination system. The activation energy is supplied in the MED evaporator 1, which is used for steam production in the first effect. A part of this steam is condensed by transferring latent heat to the organic fluid in the ORC evaporator, activating the power cycle. The remaining steam is used to activate the following effects and repeat the MED operating process mentioned in configuration 1. This integration can be

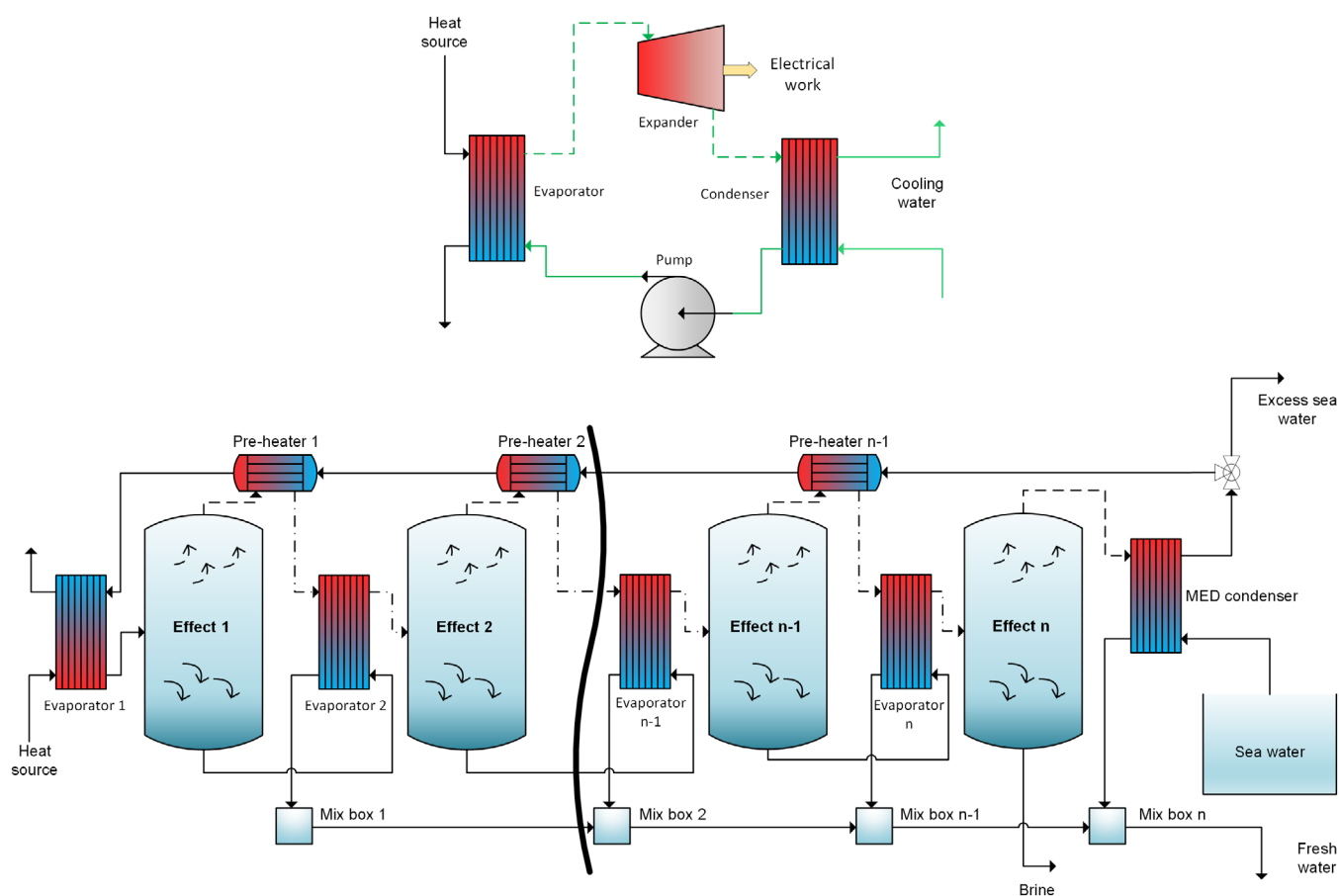
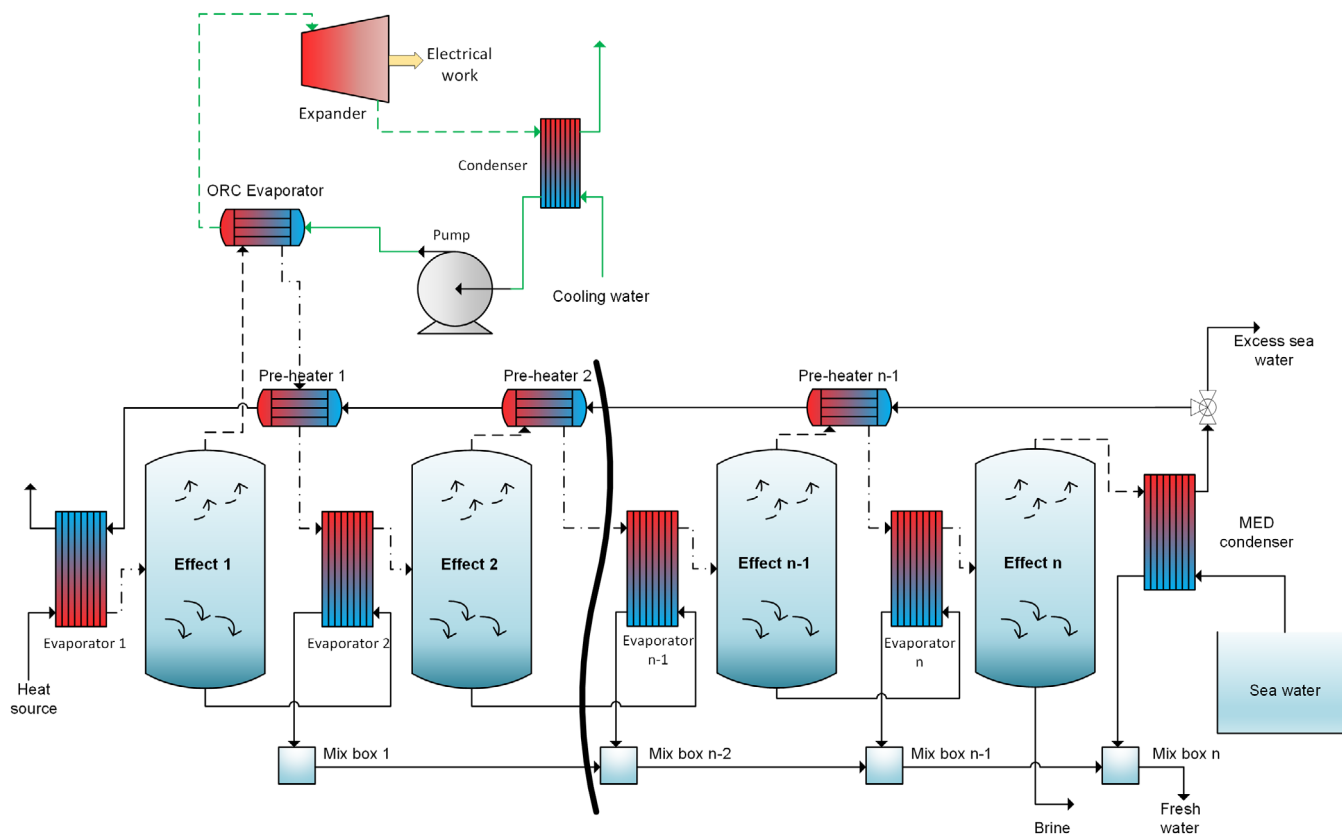


FIGURE 1 Configuration 1: Independent MED and ORC



**FIGURE 2** Configuration 2: MED/ORC integrated in the first effect

seen in Figure 2 and was technically analyzed in the work of Aguilar-Jiménez et al.<sup>26</sup>

To achieve this energy integration, the ORC evaporation temperature must be lower than the TBT of the MED system in order to perform the heat exchange. Since it is a phase change heat exchange, the steam coming from the MED remains at a constant temperature and its operating pressure and temperature conditions are not modified. By performing this integration in both systems, a greater amount of water vapor is generated in the first effect of the MED, which is reflected in a greater amount of fresh water. A more detailed operational description of this configuration can be found at Aguilar-Jiménez et al.<sup>26</sup>

### 2.3 | Configuration 3: MED and ORC with first effect integration and sensible heat utilization

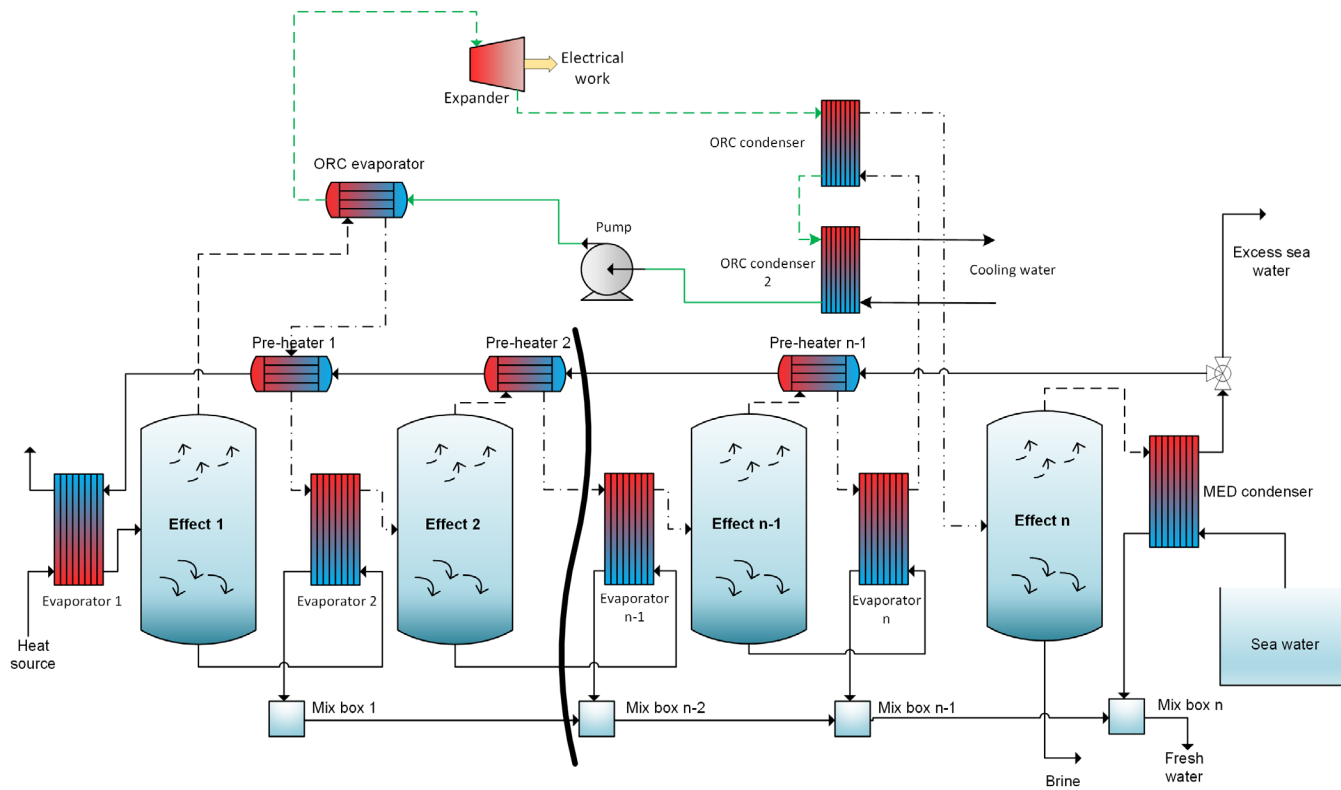
Since low-temperature ORC has a low thermoelectric conversion efficiency compared to other thermodynamic cycles with higher activation temperatures, there is a large amount of waste heat to be removed from the cycle in the condenser. However, this waste energy is of low

quality, making it difficult to use in other processes or services. In configuration 3, also considering the integration presented in configuration 2, it is proposed to use the brine stream as a cooling medium for the ORC before entering the last MED effect, taking advantage of this energy to produce a greater amount of steam and reduce part of the cooling requirements in the ORC condenser. Under this scenario, only part of the sensible heat of the ORC organic fluid at the expander outlet will be used to increase steam production until the temperature approach reaches the established limit and, subsequently, condenses the organic fluid in its entirety with external cooling water. This integration can be seen in Figure 3. The ORC operating conditions, both temperature and pressure, will remain unchanged from those of configuration 2.

### 2.4 | Configuration 4: MED and ORC with first-effect integration and sensible and latent heat utilization

To fully condense the organic fluid at the outlet of the expander, it is necessary to remove firstly sensible heat and then the corresponding part of the latent heat. In





**FIGURE 3** Configuration 3: MED-ORC integration in the first effect and use of sensible heat from the ORC condenser

configuration 3, only sensible heat is considered until the established approach temperatures allow it. However, as mentioned above, the largest amount of usable energy belongs to latent heat. Therefore, in configuration 4, it is proposed to use both sensible and latent heat to increase the production of steam in the MED. The configuration is identical to that shown in Figure 3, differing only in the ORC operating conditions which will be used to achieve a latent heat exchange to the brine. The ORC condensing temperature and pressure must be increased so that the temperature difference with respect to the seawater stream allows adequate heat transfer, that is, the ORC condensing temperature must be higher than the operating temperature of the last MED effect. The latent and sensible heat transfer will occur in the first condenser. However, depending on the electrical production capacity and efficiency of the ORC, perhaps only part of the condensing energy is required, not all of the latent heat, so the second condenser of the ORC is required to remove the remaining heat.

simulation of the proposed MED/ORC configurations, the following considerations were taken into account:

- Steady-state analysis.
- Heat losses and pressure drops in the equipment are not considered.
- Variable organic fluid flow in the ORC to ensure saturated steam conditions in the evaporator outlet.
- ORC expander and pump operate on the principle of isentropic efficiency.
- Variable seawater flow in the MED condenser to ensure steam condensation.
- The feed sea water and the inlet brine to each of the effects decrease its pressure inside the evaporator.
- The flashed steam in the mix-boxes is neglected.

The performance ratio is used to evaluate the energy performance of the MED system, expressed as follows<sup>27</sup>:

$$PR = \frac{\dot{m}_d}{\dot{Q}_{in,MED}} \times \frac{2,326 \text{ kJ}}{1 \text{ kg}} \quad (1)$$

### 3 | METHODOLOGY

The Aspen Plus software was used to perform a detailed analysis of the MED/ORC configurations. For the

where  $\dot{m}_d$  corresponds to the final distilled water (kg/s) and  $\dot{Q}_{in,MED}$  to the heat supplied for the MED subsystem activation (kW).

The thermal efficiency of the ORC is calculated as a function of the electrical power produced,  $\dot{W}_{exp}$ , by reducing the power consumed by the pump,  $\dot{W}_{pump}$ , divided by the heat supplied to the ORC,  $\dot{Q}_{in,ORC}$ , as shown in Equation 2:

$$\eta_t = \frac{\dot{W}_{exp} - \dot{W}_{pump}}{\dot{Q}_{in,ORC}} \quad (2)$$

The energy balance performed in the ORC evaporator, to calculate the heat transferred from the steam coming from the MED system, is calculated according to Equation 3:

$$\dot{Q}_{in,ORC} = \dot{m}_{ORC}(h_{out} - h_{in}) = \dot{m}_{v,1}(h_{v1,in} - h_{v1,out}) \quad (3)$$

The flow diagram of this heat exchanger is shown in Figure 4.

### 3.1 | Simulator validation

The validation of the MED/ORC system simulation was performed separately, first the MED system and second the ORC system. For the MED, theoretical and experimental data were used, while for the ORC, theoretical results were used. The validation of the simulator used can be verified in the work presented by Aguilar-Jiménez et al<sup>26</sup>

### 3.2 | Heat transfer area analysis

To compare the configuration with respect to the size of the equipment, the heat transfer area of the heat

exchangers was considered. To calculate the heat transfer area, Aspen Plus uses the Equation (4):

$$A_i = \frac{\dot{Q}_i}{U_i \times LMTD} \quad (4)$$

where  $A_i$  is the heat transfer area of the heat exchanger equipment,  $\dot{Q}_i$  is the heat transferred,  $U_i$  is the overall heat transfer coefficient, and  $LMTD$  is the log-Mean Temperature Difference. Overall heat transfer coefficients,  $U_i$ , were calculated using the correlations presented in Xu et al<sup>28</sup> for the ORC equipment, while those used for the MED were presented in El-dessouky et al,<sup>29</sup> El-Dessouky and Ettouney<sup>30</sup> and Palenzuela et al<sup>31,32</sup>

The specific area of the ORC was used to analyze the size of the heat exchanger equipment as a function of the electrical power generated, calculated according to Equation (5):

$$sA_{ORC} = \frac{A_{eva} + A_{cond} + A_{hx}}{E_{ORC}} \quad (5)$$

where  $sA_{ORC}$  is the ORC specific area in  $m^2/kWe$ ,  $A_{hx}$ ,  $A_{cond}$  and  $A_{eva}$  are the heat transfer areas of the heat exchangers, condenser and evaporator for the ORC integration, respectively, and  $E_{ORC}$  the electrical power delivered by the ORC. For the specific area of the MED, the Equation (6) is used:

$$sA_{MED} = \frac{\sum A_{eff} + \sum A_{ph} + A_c}{\dot{m}_d} \quad (6)$$

where  $sA_{MED}$  is the MED specific area in  $m^2/kg/min$ ,  $\sum A_{ph}$  and  $\sum A_{eff}$  are the sum of the heat transfer area for the preheating and effects, respectively,  $A_c$  is the

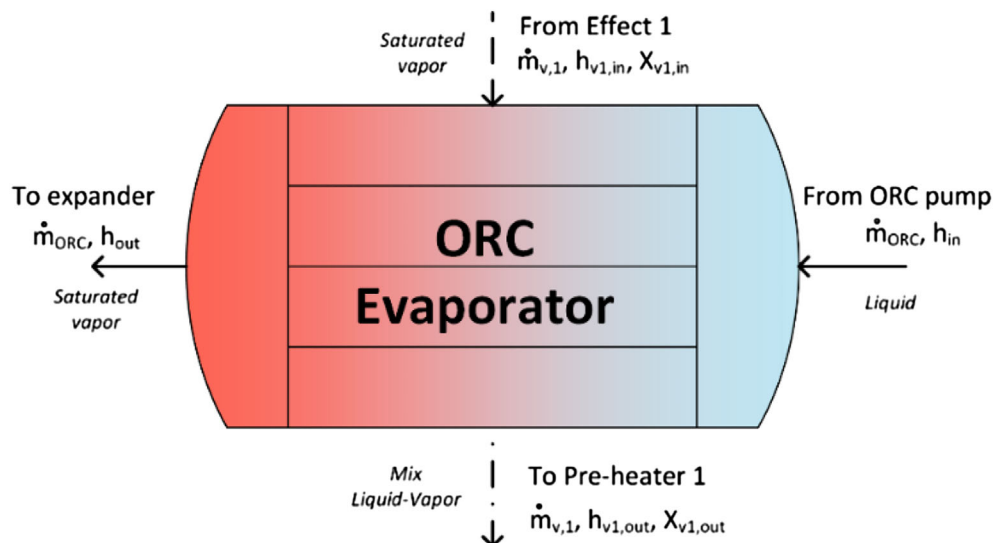


FIGURE 4 Flow diagram of the ORC evaporator

condenser area, and  $\dot{m}_d$  the mass flow of fresh water produced by the MED.

## 4 | RESULTS AND DISCUSSION

### 4.1 | Operating conditions

Due to the different configurations used, it is necessary to make modifications to the operating conditions of the systems to carry out the heat transfers properly. Table 1 presents the operating conditions of the configurations studied. The main source of activation energy is considered at a temperature of 80°C for all configurations and an electrical output of 50 kW is set for the ORC, varying the heat supplied to the MED. Therefore, the TBT in the MED was set at 70°C, causing the ORC evaporation temperature to be lower than this temperature in the proposed integrations in order to perform a heat exchange with the steam produced in the first effect of the MED. However, seeking not to benefit our proposals from the systems operating independently since in configuration 1 the ORC evaporation temperature is not limited to the TBT of the MED, the ORC activation temperature in

configuration 1 was considered the same as the MED because the activation heat is at 80°C, having a 10°C difference. This benefits the ORC and is an advantage that should not be excluded when operating independently.

It is important to mention that the heat exchanged between the steam generated in the first MED effect and the ORC fluid is sufficient to thermally activate the power cycle. In configurations 2 and 3, this heat is the same since the ORC operates under the same temperature and pressure conditions, so they present the same efficiency. However, in configuration 4 and as mentioned above, it is necessary to raise the condensing temperature and pressure conditions of the ORC so that a latent heat exchange can occur. In configuration 3, the ORC fluid at the turbine outlet is at 44.7°C and exchanges sensible heat down to 37.8°C with the vapor–liquid mixture coming from the evaporator of the last MED effect, which is at 35°C, increasing the steam content of the mixture. The remaining sensible and latent heat of the organic fluid are removed with cooling water in the ORC condenser. For configuration 4, by increasing the condensation temperature of the ORC, the organic fluid leaves the turbine at 49.5°C and exchanges sensible heat (in its entirety) and part of the latent heat up to its condensation

TABLE 1 Operating conditions of the MED-ORC configurations

|                                                | Config. 1 | Config. 2 | Config. 3 | Config. 4 |
|------------------------------------------------|-----------|-----------|-----------|-----------|
| <b>ORC</b>                                     |           |           |           |           |
| Electrical power (kW)                          | 50        | 50        | 50        | 50        |
| Heat source temperature (°C)                   | 80        | 70.3      | 70.3      | 70.3      |
| Cooling water temperature (°C)                 | 25        | 25        | 25        | 25        |
| Evaporation temperature (°C)                   | 70        | 67        | 67        | 67        |
| Condensation temperature (°C)                  | 30        | 30        | 30        | 37.7      |
| Evaporation pressure (kPa)                     | 605       | 556       | 556       | 556       |
| Condensation pressure (kPa)                    | 176       | 176       | 176       | 231       |
| Mass flow (kg/s)                               | 3.132     | 3.36      | 3.36      | 4.4       |
| Expander efficiency (–)                        | 0.7       | 0.7       | 0.7       | 0.7       |
| Pump efficiency (–)                            | 0.8       | 0.8       | 0.8       | 0.8       |
| <b>MED</b>                                     |           |           |           |           |
| Heat source temperature (°C)                   | 80        | 80        | 80        | 80        |
| TBT (°C)                                       | 70        | 70        | 70        | 70        |
| Condenser temperature (°C)                     | 35        | 35        | 35        | 35        |
| Sea water inlet temperature effect 1 (°C) (°C) | 64.9      | 64.8      | 64.9      | 64.9      |
| Sea water temperature (°C)                     | 25        | 25        | 25        | 25        |
| Sea water inlet condenser (kg/s)               | 5.221     | 3.446     | 4.076     | 9.28      |
| Feed sea water temperature (°C)                | 32.25     | 32.25     | 32.25     | 32.25     |
| Feed sea water mass flow (kg/s)                | 1.875     | 1.8       | 1.95      | 2.175     |
| Sea water salt concentration (ppm)             | 35 000    | 35 000    | 35 000    | 35 000    |

temperature of 37.8°C with the liquid-steam mixture coming from the MED, which is at 35°C. This achieves the correct temperature conditions for an effective heat exchange between the organic fluid of the ORC and the liquid-steam mixture of the MED.

## 4.2 | Water product

The effect on the total desalinated water production of each configuration, due to the variation of the heat supplied to the MED, is presented in Figure 5. Configuration 4 has the highest total water production because it takes advantage of the benefits of configuration 2, with the waste heat from the ORC, while configuration 1 produces the lowest desalinated water because it does not incorporate any energy integration. It is worth mentioning that, for the purpose of the analysis of the variation of the heat supplied to the MED, an approach temperature of 2°C was considered between the streams of the first condenser of the ORC, in addition to a heat exchange of 200 kW (all of the sensible heat and part of the latent heat of condensation of the ORC) only for configuration 4. Configurations 2 and 3 increase the final water production with respect to configuration 1 from 26% to 53% depending on the heat supplied to the MED, while taking advantage of the sensible and latent heat of the ORC in configuration 4 increases the water production between 37% and 76% with respect to configuration 1 and between 9% and 15% with respect to configurations 2 and 3. There is a linear increasing trend for all the scenarios studied as the heat supplied to the MED increases, this is due to the fact that there is a higher steam generation in the first

effect, causing a cascading increase in the remaining effects. It is interesting to note that the total water production for configurations 2 and 3 is approximately identical. This is a consequence of the small amount of sensible heat that is utilized in configuration 3, approximately 22 kW. It appears to be an attractive amount of energy, but when supplied to the brine entering the last effect, this heat is not cascaded as it is in the other effects and the amount of extra steam generated is not enough to positively impact the efficiency of this configuration, so the increase in final product water is negligible.

## 4.3 | Brine concentration

The level of salt concentration in the brine, depending on the configuration and amount of heat supplied to the MED, is shown in Figure 6. The amount of salt in the final brine is of great importance, since certain operating conditions will depend on it, such as the addition of anti-fouling chemicals, as well as the environmental impact it has when discharged into the marine ecosystem.<sup>33</sup> It is recommended to establish a maximum of 70 000 ppm salt in the final brine to avoid problems, but there are large capacity systems with concentrations above 100 000 ppm, depending on the technology used.<sup>34,35</sup> Configuration 1 produces the lowest salt concentration. The proposed integrations show a noticeable increase, mainly due to the higher separation of water from brine in effect 1 (configurations 2, 3 and 4) and in the last effect (configurations 3 and 4). As the heat supply in the first effect of the MED increases, so does steam production, and the final salt concentration shows the same increasing trend. The

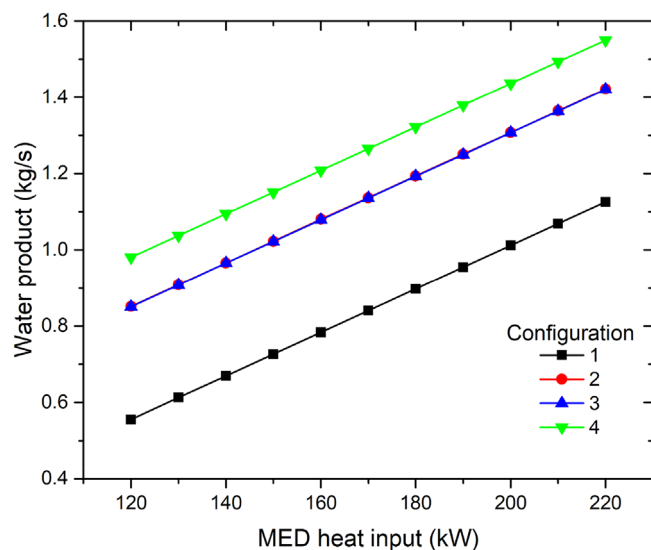


FIGURE 5 Effect on desalinated water production by varying the heat supplied to the MED

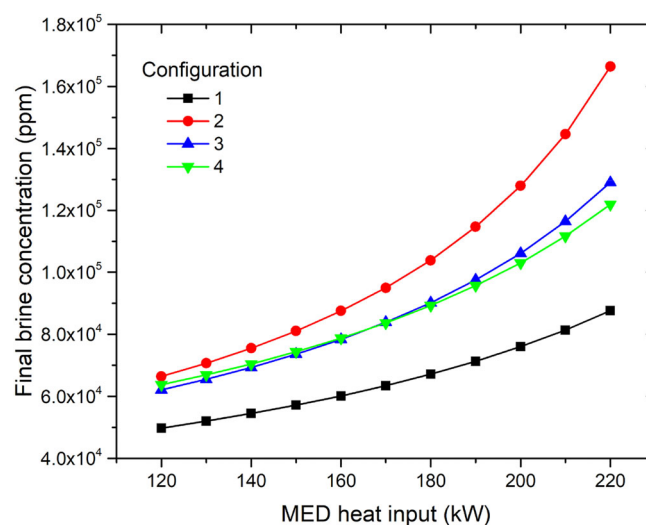


FIGURE 6 Brine concentration variation as a function of heat supplied to the MED

last two configurations have lower concentration levels than configuration 2 because they have a higher mfw. The salinity level is of utmost importance, so this study identifies that, under the operating conditions considered and seeking to maintain levels close to 70 000 ppm, configuration 1 could operate with a power supply of approximately 190 kW at the MED, while configurations 2 and 3 can operate with a maximum supply of 150 kW and 130 kW for configuration 4.

#### 4.4 | Configurations performance ratio

In comparing the water production efficiencies of the different configurations, Figure 7 shows the variation of the PR as a function of the heat supplied to the MED. Configuration 1 presents an increase in the PR as the activation heat increases, but remains with efficiency values below the other proposals. The remaining configurations tend to decrease their PR as the heat supplied increases, due to the way this indicator is calculated. In configurations 2, 3 and 4, there is always an amount of extra steam generated due to energy integration with the ORC. If these integrations are analyzed using Equation 1, as the heat supplied for the MED decreases and the part of energy belonging to the ORC activation is maintained, the PR will increase, although this does not mean a greater amount of water produced. This can be seen by comparing Figures 7 and 9. When 120 kW are supplied to the MED, maximum PR values are obtained, while water production is minimal, and as heat is increased, PR decreases and water production increases. However, in spite of the aforementioned, the MED-ORC integrations

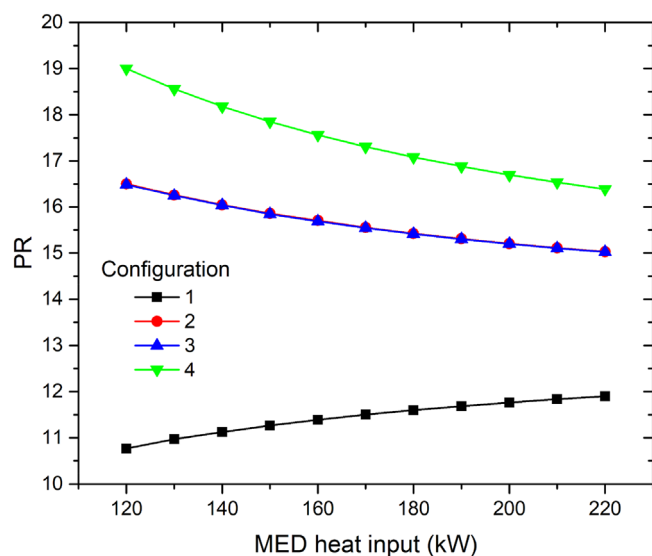


FIGURE 7 Effect of varying the heat supplied to the MED on the PR of different configurations

analyzed show a better use of energy, when taking advantage of the power cycle effluents to increase desalination in the MED.

#### 4.5 | Latent heat utilized in configuration 4

As mentioned above, a heat transfer rate of 200 kW was used in the first ORC condenser of configuration 4 during the conducted studies (see Figure 3). However, there is still a large potential for harnessing condensing energy from the ORC to increase water desalination. The amount of energy that needs to be removed from the ORC is directly related to the generation capacity and its

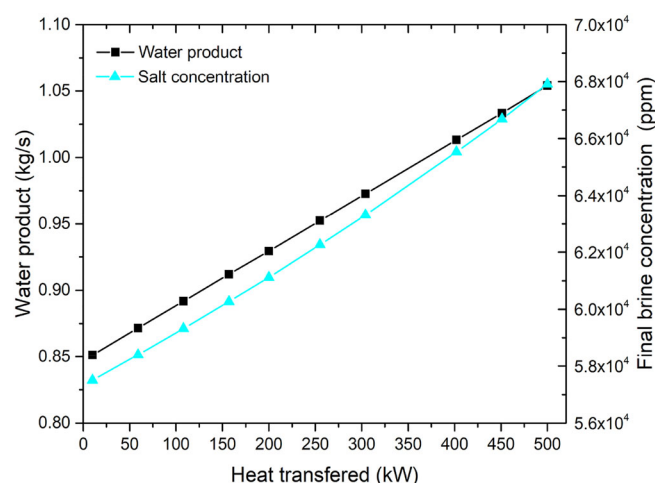


FIGURE 8 Variation of water production and final salt concentration of configuration 4 with respect to heat transferred in the ORC condenser

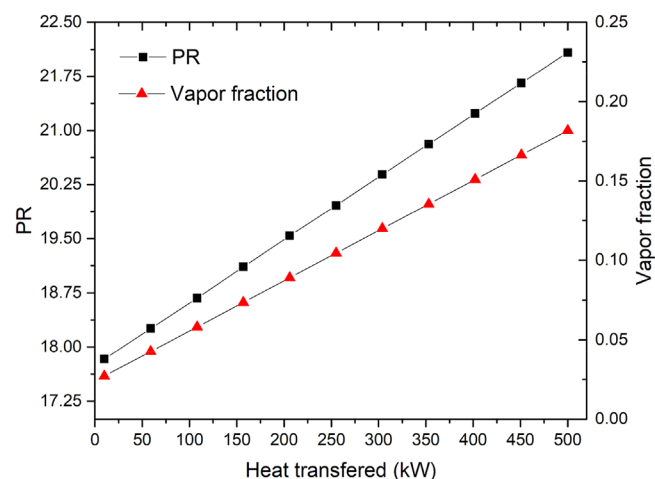


FIGURE 9 Effect of heat transferred in the ORC condenser of configuration 4 on the PR and fraction of steam returning to the MED of the liquid-steam mixture



thermoelectric conversion efficiency; the greater the capacity of the ORC, the greater the heat that can be used to enhance desalination in the MED. Note that the condensation temperature of the ORC was raised to 37.7°C in order to have an acceptable difference from the brine entering the last effect, which is 35°C, and to be able to transfer the latent heat adequately. Figure 8 shows the variation of desalinated water production and final brine concentration as a function of the heat transferred in the ORC condenser. It was analyzed from 0 kW (in which there would be no integration, so the configuration would be identical to 2), up to an exchange of 500 kW, considering a thermal energy supply to the MED of 180 kW. A linear increase in the production of desalinated water is observed with respect to the increase in the heat exchange rate, as well as an increase in the final salt concentration. However, since the integration contributes energy only to the last effect, there is no cascade steam generation, so the increase in final water production represents 23.5% when considering 500 kW of thermal exchange. It is important to mention that this configuration implies that, as an extra amount of thermal energy is added to generate more steam in the last effect, a larger amount of cooling water is required to condense the last steam. This results in a higher energy consumption per water pumping, as can be seen in the msw value in Table 1, where configuration 4 requires a flow of 77.7%, 169.2%, and 127.6% higher than configuration 1, 2, and 3, respectively.

To analyze the performance of the MED as a function of the heat transferred in configuration 4, Figure 9 shows the effect on the PR and the fraction of steam leaving the liquid–steam mixture. The trend is the same for water production, as the heat transfer increases the PR and the amount of steam generated increases in the same manner. This is because the more heat is transferred to the liquid–steam mixture coming from the last evaporator of the MED, the more steam is generated and, therefore, the amount of final product water increases causing an increase in the PR. The decision on the amount of heat to be transferred will depend on the size of the ORC and the economic possibilities for the acquisition of a heat

exchanger suitable for the desired transfer. The PR increases from 17.8 to 22.1 when using up to a maximum of 500 kW of ORC condensing heat (sensible and latent), equivalent to an increase of 23%. By integrating the last effect, the final concentration does not exceed the recommended values, these with maximums of 68 000 ppm. A positive side effect of this integration is that all the heat that is transferred from the ORC, while increasing water desalination, greatly reduces the cooling requirements of the power cycle, which can be translated into energy savings.

#### 4.6 | Overall MED-ORC comparison

A desalinated water flow rate of 0.92 kg/s and 50 kW electricity were established for all configurations in order to compare the indicators and energy requirements. In this comparison, the thermal energy required by the ORC for its activation was considered both individually (configuration 1) and by means of the integration proposed in effect 1 of the MED (configurations 2, 3, and 4). The results of this comparison can be seen in Table 2. In relation to desalinated water production and energy requirements for desalination, configuration 4 presents the highest values, while configuration 1 is less efficient. However, when the thermal energy required by the ORC is incorporated into the study, the latter energy integration becomes the least efficient, requiring a total of 1016.6 kW to provide electric power and desalination services. This is mainly due to the decrease in the efficiency of the ORC as the condensing temperature increases, which forces it to require a large amount of extra thermal energy to generate the 50 kW of electricity required. This generates a greater amount of condensation heat that can be used in desalination, but by supplying it only to the last effect, it does not greatly enhance water production. Increasing the condensing temperature of the ORC from 30 to 37.7°C represents a decrease in efficiency of 1.8% compared to configuration 1, forcing the supply of 221 kW of extra thermal energy. This indicates that the modification of the ORC operating conditions that

**TABLE 2** Simulation results of the different MED/ORC configurations

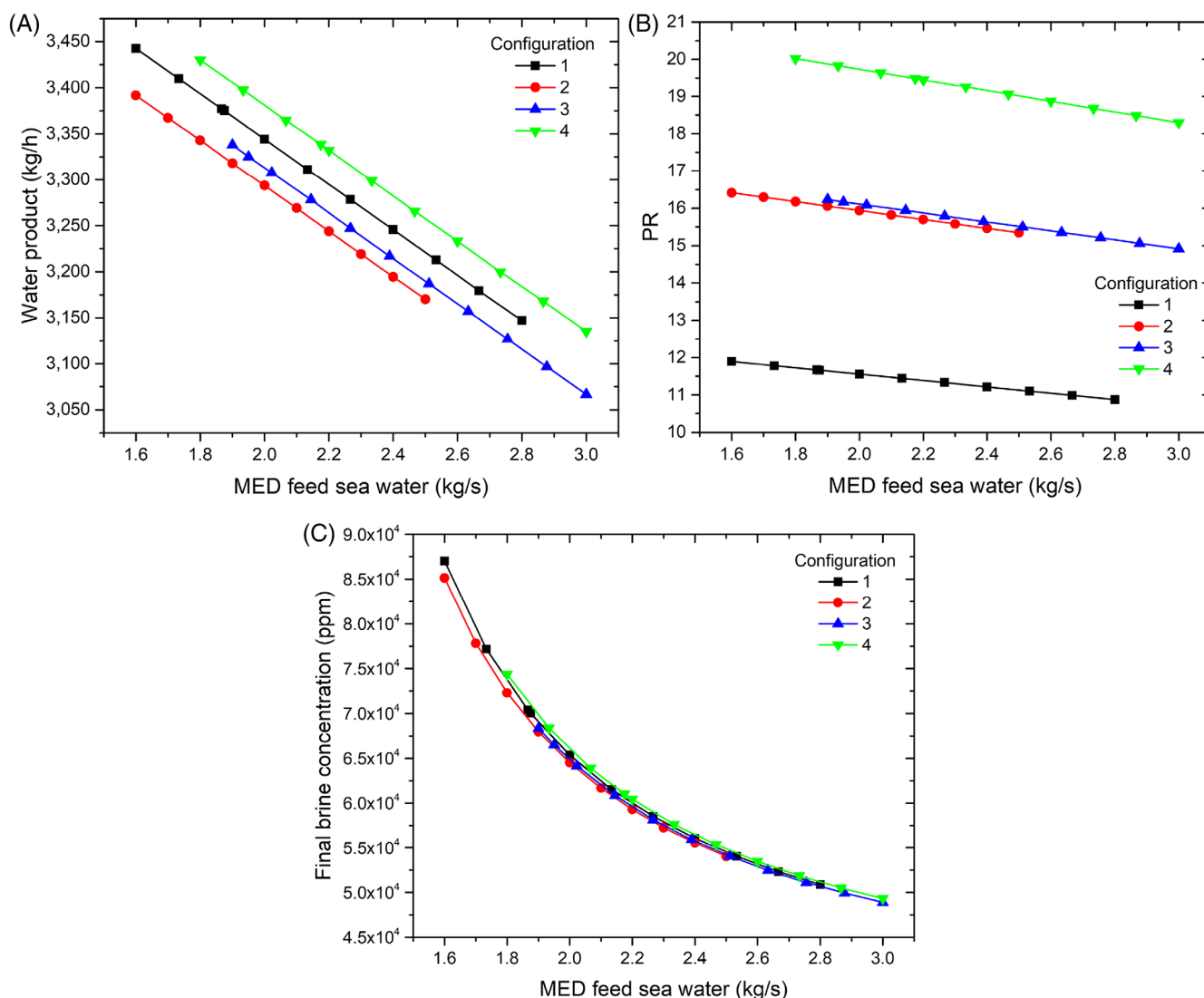
| Parameter                           | Config. 1 | Config. 2 | Config. 3 | Config. 4 |
|-------------------------------------|-----------|-----------|-----------|-----------|
| Electrical energy (kW)              | 50        | 50        | 50        | 50        |
| ORC thermal energy required (kW)    | 684.5     | 726.8     | 727.9     | 905.9     |
| Thermal efficiency (%)              | 7.1       | 6.7       | 6.7       | 5.3       |
| Desalinated water production (kg/s) | 0.92      | 0.92      | 0.92      | 0.92      |
| MED thermal energy required (kW)    | 187       | 133.5     | 132.8     | 110.7     |
| Performance ratio (PR)              | 11.66     | 16.17     | 16.17     | 19.47     |
| Total thermal energy required (kW)  | 871.5     | 860.3     | 860.7     | 1016.6    |

decrease its efficiency negatively impacts the overall energy requirements, ruling out the proposed integration within the comparison. On the other hand, configurations 2 and 3 decrease the energy requirements of the MED by about 28% compared to configuration 1, but increase the amount of energy needed in the ORC by 42.8 kW thermal on average. However, the overall balance is favorable for configurations 2 and 3, with a lower amount of thermal energy needed to provide the two services.

#### 4.7 | Effect of the variation of seawater feed to the MED

The variation of the seawater feed flow to the MED is of utmost importance to achieve acceptable efficiency

levels as a function of the desalinated water and the amount of salts in the final brine. Figure 10 shows the effect of this variation on (a) product water, (b) PR, and (c) final brine concentration. To perform this study, the heat supplied to the MED was kept constant with the values used in the energy comparison in Section 4.6, which are 187, 133.5, 132.8, and 110.7 kW for configurations 1, 2, 3 and 4, respectively. Figure 10A shows a decrease in all configurations of the final product water of the MED as the feed seawater flow rate is increased. This is because, by keeping the heat supply constant, steam generation in the first effect decreases, which causes effect after effect to decrease as well. The same effect causes the PR, as seen in Figure 10B, to tend to decrease as the flow rate increases. However, a positive effect is achieved in the final brine concentration, which decreases as the feed seawater flow rate increases. This



**FIGURE 10** Effect of varying seawater flow rate to the MED on (A) desalinated water production, (B) PR and (C) final brine concentration

is because it decreases steam generation in the MED effects by keeping the supplied heat fixed. Proper selection of the feed seawater flow rate to the MED is of utmost importance in maintaining a balance between product water, salt concentration levels in the final brine, and system efficiency.

#### 4.8 | Heat transfer area comparison

For comparison purposes from the viewpoint of the size of the heat exchange components, since they are the equipment with the greatest presence and impact on the sizing of the proposals, Figure 11 is presented. It can be seen that the configuration with the greatest amount of heat transfer area is configuration 1, where there is no energy integration, thus being 14% of the total area for the ORC. When performing the MED/ORC integration in the first effect, corresponding to configuration 2, the area required by the ORC increases by 48% with respect to configuration 1, but the MED area is benefited by being reduced by 23%, resulting in a total system area of 367 m<sup>2</sup>, a decrease of 13% with respect to configuration 1. This is due to the fact that for the integration of the ORC with the MED, the evaporation temperature was reduced, which caused a decrease in efficiency and therefore a higher heat requirement, increasing the evaporator transfer area of the ORC. As the heat transfer rates in the MED heat exchanger equipment decreased by cause of the increased efficiency due to the integration, the MED heat transfer area decreased. Configuration 3 presents a 2.8% increase in total heat transfer area with respect to

configuration 2 due to the increased condensation requirements of the ORC through integration with the last effect of the MED. Configuration 4 presented a 27% decrease in total area with respect to configuration 1, this being the configuration with the lowest total heat transfer area requirements. This is mainly due to the fact that the integration benefits the efficiency of the MED, causing a 55.4% decrease with respect to the MED transfer area of configuration 1, considering that to generate the same amount of water as the previous configurations require less exchange energy between the equipment.

To encompass the electrical power and water desalination with respect to the heat transfer area requirements, Figure 12 presents the specific areas for the ORC and MED of each configuration, calculated according to Equation (5) and Equation (6). It can be seen that the specific area has an opposite trend between the ORC and the MED, since that of the power cycle tends to increase while that of the MED decreases when performing the energy integrations. This is due to the decrease in the efficiency of the ORC due to the modifications in the operating conditions for its integration with the MED. This ends up benefiting the MED by increasing seawater desalination and decreasing heat transfer needs. Although the percentage of heat transfer area of the ORC increases considerably, the benefit to the MED ends up decreasing the total area required to produce electricity and desalinated water services.

The four configurations studied have advantages and disadvantages when it comes to MED/ORC integration. Attention must be paid to the operating conditions of both systems, especially the ORC evaporation and

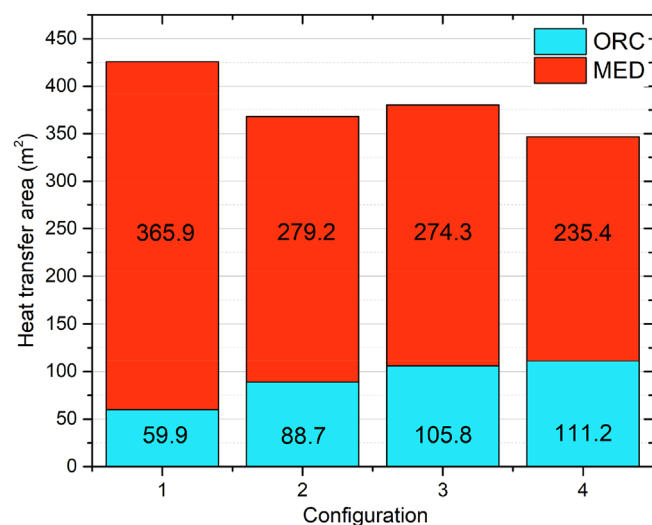


FIGURE 11 Total and individual heat transfer area for the four configurations

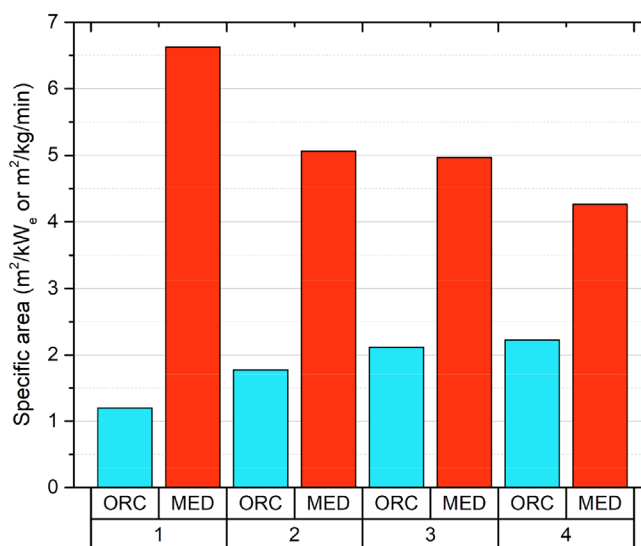


FIGURE 12 Individual specific heat transfer area for the four configurations



condensation temperatures, which will determine their efficiency. It was found that although the proposed novel integrations benefit desalination without harming its operating conditions and salt concentration levels, they end up affecting the ORC operation, impacting the overall energy requirements to produce desalinated water and electric power together. However, in two scenarios, the results of the integration are positive and allow energy savings, in addition to presenting a considerable decrease in the heat transfer area, it is necessary to conduct more detailed studies on the proposed configurations and determine both their technical and economic feasibility.

## 5 | CONCLUSIONS

A comparative energy study of novel MED/ORC integration configurations for simultaneous production of water and electric power was presented. Two new couplings were proposed where a part of the sensible and/or latent heat of condensation of the ORC is used to potentiate the steam production in the last effect of the MED. The MED and ORC systems working separately were also included in the study to have a benchmark for comparison of the benefits of the new proposals. The main results of the analysis are:

- By integrating MED/ORC in the first effect evaporator (configuration 2), the efficiency of the ORC is affected because its evaporation temperature is lowered in order to be activated with the steam from the first effect of the MED. However, the energy requirements of the MED are reduced by 28%, so the thermal activation energy needed to produce the two services is lower than operating the systems separately.
- The utilization of sensible heat (configuration 3) was not as beneficial compared to configuration 2. This is a consequence of the small amount of sensible heat that is utilized, approximately 22 kW. It appears to be an attractive amount of energy, but when supplied to the brine entering the last effect, this heat is not cascaded, as it is in the other effects, and the amount of extra steam generated is not enough to positively impact the efficiency of this configuration, so the increase in final product water is negligible.
- Configuration 4, which takes advantage of the sensible heat and part of the latent heat of condensation of the ORC in conjunction with the integration of configuration 2, allows a decrease in the energy requirements for the MED, increasing its efficiency. However, to take advantage of the latent heat of

the ORC inside the MED, the condensation temperature of the power cycle must be increased in order to perform an adequate heat transfer, which negatively impacts its efficiency and, therefore, the thermal activation energy requirements. Increasing the ORC condensing temperature from 30 to 37.7°C represents a decrease in efficiency of 1.8% compared to configuration 1, forcing the supply of 221 kW of extra thermal energy, overshadowing the benefits in MED.

- The proposed configurations present a considerable decrease in the total heat transfer area compared to the systems operating independently, thus the overall system cost is expected to be reduced.

Since this work focused on aspects for a comparison of energy requirements and a presentation of the new integrations, future work will be related to this specific study of the proposed configurations under different operating conditions, such as ORC evaporation temperature, integration of condensation heat in other MED effects, and study of the technical and economic feasibility of increasing/decreasing the number of MED effects in the configurations.

## NOMENCLATURE

### Symbols

|           |                                   |
|-----------|-----------------------------------|
| P         | Pressure                          |
| T         | Temperature                       |
| $\dot{Q}$ | Heat                              |
| $\dot{m}$ | Mass flow                         |
| h         | Enthalpy                          |
| x         | Steam fraction                    |
| U         | Overall heat transfer coefficient |
| A         | Heat transfer area                |
| s         | Specific                          |
| E         | ORC electrical power              |

### Subscripts

|             |                 |
|-------------|-----------------|
| <i>eva</i>  | Evaporator      |
| <i>cond</i> | Condenser       |
| <i>hx</i>   | Heat exchanger  |
| <i>d</i>    | Distilled water |
| <i>in</i>   | Inlet           |
| <i>out</i>  | Outlet          |
| <i>exp</i>  | Expander        |
| <i>t</i>    | Thermal         |
| <i>eff</i>  | Effect          |
| <i>ph</i>   | Pre-heater      |
| <i>c</i>    | MED condenser   |
| <i>sw</i>   | Sea water       |

$v1$  First effect steam  
 $i$  Effect number

## Greek

$\eta$  Efficiency

## ACKNOWLEDGEMENTS

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## DATA AVAILABILITY STATEMENT

The data that support the findings of this study are available from the corresponding author upon reasonable request.

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## **ANEXO E**

**Technical and economic assessment of hybrid PVT  
solar systems compared to independent solar thermal  
and photovoltaic collectors in high-temperature zones:  
A case study in Mexicali, Mexico**

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# Technical and economic assessment of hybrid PVT solar systems compared to independent solar thermal and photovoltaic collectors in high-temperature zones: A case study in Mexicali, Mexico

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## Abstract

This study analyzes the technical and economic feasibility of hybrid photovoltaic/thermal (PVT) solar energy systems, comparing them with independent flat plate solar thermal collectors (FP) and photovoltaic (PV) modules. Using TRNSYS software, different configurations were simulated under the climatic conditions of Mexicali, Baja California, Mexico. The results indicate that, although independent FP and PV systems exhibit better energy performance in terms of energy production and solar fraction, PVT collectors offer higher specific global energy production (electrical + thermal). However, the economic feasibility of PVT systems is limited due to high investment and maintenance costs, coupled with the low cost of thermal and electrical energy in the study region. Independent FP and PV collectors also face similar economic challenges, with return on investment extending beyond the 20-year lifespan of the project. Additionally, the low thermal energy demand for domestic hot water in the studied region negatively impacts the efficient utilization of these systems. This analysis suggests that in contexts where available installation space is limited, PVT systems can be an efficient solution. However, in scenarios with sufficient space and high energy costs, energy generation with separate equipment remains more viable. Moreover, climatic conditions and DHW demand affect the efficiency of the systems. The discrepancy between solar energy generation and hourly DHW demand necessitates the implementation of TEST, which significantly increases costs and impacts the economic viability of the project.

## Keywords

solar energy, solar PVT, solar fraction, TRNSYS, solar thermal collectors

## 1. Introduction

In the current context, energy consumption continues to increase at an alarming rate. In 2022, global energy consumption reached 442 EJ, accompanied by the emission of 36,930 million tons of CO<sub>2</sub>e (International Energy Agency, 2023). Of the total energy consumed, 30% was attributed to the building sector, equivalent to 133 EJ. Additionally, 10% of the world's energy consumption was allocated to electricity use in buildings, representing 8% of global CO<sub>2</sub>e emissions (International Energy Agency, 2023). This situation is alarming because the continuous increase in energy consumption exacerbates environmental pollution, especially since most of this energy comes from fossil fuels. This creates a vicious cycle where the increase in energy demand leads to higher

levels of pollution, intensifying the negative effects of climate change. Therefore, it is crucial for governments, businesses, and citizens to collaborate in adopting measures that

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promote more efficient and responsible energy use, ensuring a more sustainable and healthier future for all (Aguilar-Jiménez et al., 2018). The implementation of advanced technologies and sustainable practices can significantly reduce the carbon footprint (Nesmachnow and Risso, 2024; Velázquez-Limón et al., 2020).

The hybrid photovoltaic/thermal (PVT) solar collector technology has been proven to be a technically and, in some scenarios, economically viable option for the simultaneous generation of electrical and thermal energy (Kramer et al., 2023). This technology offers several technical advantages compared to other systems, such as better energy utilization of the installation area (Shirinbakhsh and Harvey, 2024), reduced installation time and costs compared to technologies that provide the same services independently (Kazem et al., 2024), and increased photovoltaic efficiency in certain scenarios (Chavarría-Domínguez et al., 2024). These reasons have led to a notable growth in recent years in the multidisciplinary research interest in this technology, revealing a broad and innovative field of application in the energy sector and in reducing pollutant emissions. Due to the great versatility offered by the integration of photovoltaic and thermal solar technology, numerous PVT hybridization options have been proposed, with different geometries, sizes, energy integration methods, applications, and efficiencies, among others (Ahmed et al., 2024). Among the various applications of these technologies are the generation of electrical and thermal energy in the domestic (Ben Seddik et al., 2022), commercial, and industrial sectors (Kramer et al., 2023; Oh et al., 2023), seawater desalination (Anand et al., 2021), cooling (Hamzat et al., 2021), heating (You et al., 2021), hydrogen production (Salari et al., 2022) and food dehydration (Tiwari et al., 2021). However, this technology still faces technological and economic challenges that need to be addressed to accelerate its implementation (Hamid et al., 2023).

The technical aspect of PVT solar collector technology can be considered mature, as there are companies that have been marketing PVT collectors for some time with overall efficiencies (electrical and thermal) of up to 89% (Abora Solar, n.d.). One of the main areas of opportunity for PVT collectors is their economic viability, as their investment costs are usually high (Mostakim and Hasanuzzaman, 2022). Factors such as high costs of electrical and thermal energy, as well as good solar resources at the installation site, can make this technology more economically attractive (Acosta-Pazmiño et al., 2022).

Seddik et al. (Ben Seddik et al., 2023) evaluated the performance of a hybrid PVT system in two climatic zones of Morocco, Errachidia and Tangier, compared to a parallel installation of PV and flat plate (FP) modules. Using simulation models in TRNSYS18, dynamic electrical and thermal yields were estimated, and energy, economic, and environmental analyses were performed. The results show that the PV/FP system outperforms the PVT system in all aspects in both climatic zones. However, the PVT system

remains an important renewable alternative for projects with limited space and the need for thermal and electrical energy, especially in the construction sector in Morocco.

On the other hand, Odeh and Aden (Odeh and Aden, 2025) evaluated the performance of PVT systems in the residential sector of Sydney, focusing on a PVT collector with heat pipe technology. The results reveal that the overall energy efficiency of the proposed PVT panel, on a typical day, is approximately 45%, significantly outperforming traditional PV panels by more than double. Furthermore, the payback period for a typical residential PVT system, providing both hot water and electricity, is substantially shorter than that of installing separate PV and solar hot water systems, highlighting the economic and environmental benefits of the proposed hybrid system.

Pino et al. (Pino et al., 2024) evaluated the energy output and economic viability of PV and PVT systems for a microbrewery. Simulations using TRNSYS and real load profiles from a Spanish microbrewery show that in a hot coastal Mediterranean location, a PVT system can cover up to 26.4% of electricity and 47.6% of heat demand. These figures vary by location, with a central European site covering 27.6% of electricity and 35.7% of heat. PVT systems outperform PV in total energy yield by up to 67% due to the inclusion of useful heat. However, the economic effectiveness of PVT is lower, with a payback period over 12 years, suggesting the need for improved manufacturing techniques to reduce costs.

Aguilar-Jiménez et al. (Aguilar-Jiménez et al., 2020) studied the performance and economic viability of hybrid PVT systems for simultaneous electricity and heat production under different climates and energy tariffs. Using TRNSYS, a simulator was validated to compare energy production across five cities: Mexicali, Soria, Bigene, Fresno, and Madison. Results indicate Mexicali achieves the highest energy output, with an 80% solar thermal fraction achievable using three modules. However, Soria offers the most attractive investment returns and monetary savings. Subsidy scenarios show a payback period of 1.8 years with a 75% subsidy. The study highlights how climate conditions, such as solar radiation and ambient temperature, influence PVT system performance, aiding in decision-making and financial planning for support programs.

Previous studies agree that PVT technology is energetically efficient and can significantly contribute to the electrical and thermal energy demands of a wide range of applications. However, the authors also concur that one of the main challenges of these collectors lies in their economic feasibility. This article presents an operational study of a system for generating electrical and thermal energy using PVT collectors, compared to flat plate solar thermal and photovoltaic collectors operating simultaneously, applied to the domestic sector in the city of Mexicali, Baja California, Mexico, known for its low winter temperatures and high summer temperatures. Additionally, the economic feasibility is analyzed, considering the initial investment costs and the local prices of electrical and thermal energy.



## 2. System description

The system considered in this study is illustrated in the schematic diagram in Figure 1. The solar circuit pump draws water from the bottom of the thermal energy storage tank (TEST) and transports it to the PVT or flat plate (FP) collector, depending on the type of collector being studied. In the collector, solar radiation is converted into thermal energy, which is transferred to the heat transfer fluid from the TEST. The fluid, which in this study is water, returns at a higher temperature to the top of the TEST.

The hot water used for the domestic hot water system (DHW) and to meet the thermal demand is taken from the top of the TEST. Depending on the temperature of the water and the temperature required by the user, it can be further heated by an auxiliary heater. If the heat transfer fluid at the outlet of the TEST is at a higher temperature than required by the demand, its mass flow will be reduced and mixed with tap water to lower its temperature, using a mixing valve. The DHW flow is used by the user, so the same amount of water extracted to meet the demand is replenished to the TEST with tap water, whose temperature depends on the locality and climatic conditions. The electrical energy generated by the PVT and PV collectors is delivered entirely to the demand and analyzed separately.

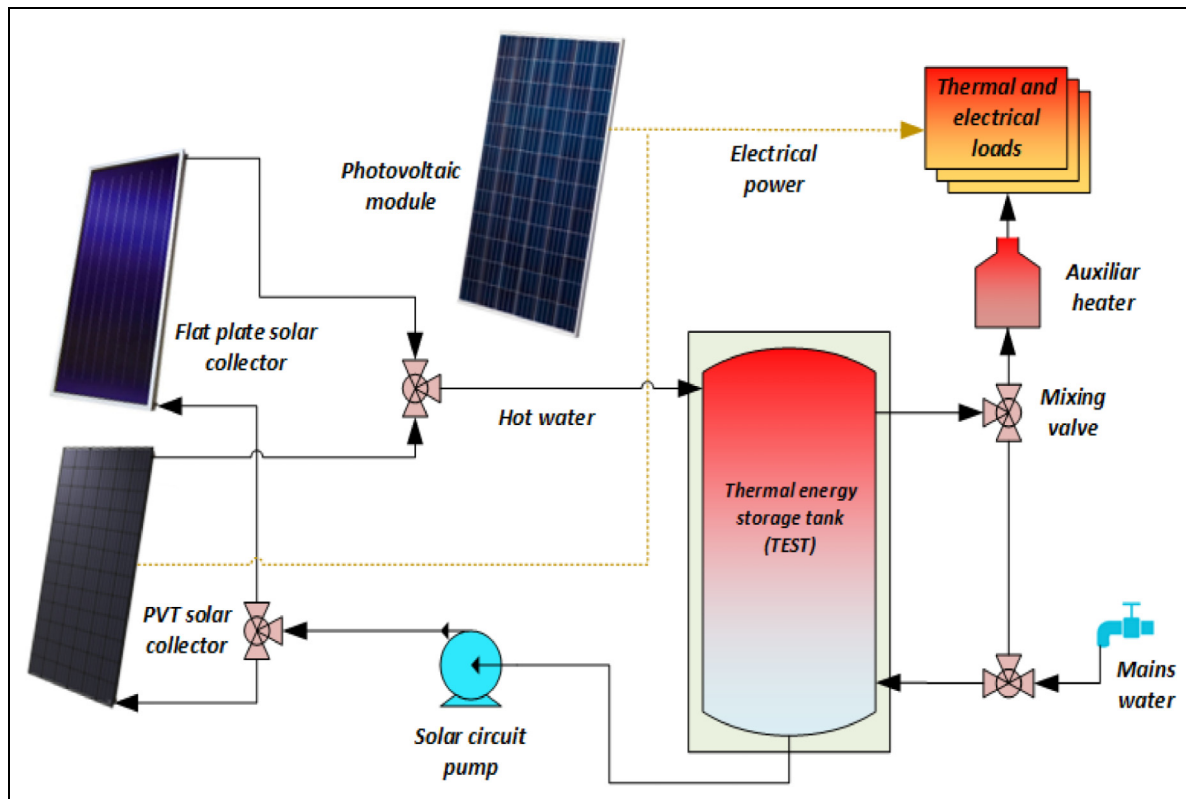
### 2.1. Solar circuit control strategy

To control the operation of the PVT and FP collector systems, a series of equipment and accessories are considered, such as differential temperature controllers, pumps, and mixing valves. For both the PVT solar collector scenario and the FP scenario, the solar circuit pump is used to operate the solar heating circuit. This pump turns on or off based on the temperature differential between the collector's outlet and inlet. If the outlet temperature of the solar collector is higher than the inlet temperature, the differential temperature controller sends a signal to turn on the solar circuit pump, initiating the heating of the fluid in the TEST. Conversely, if the outlet temperature is lower than or equal to the inlet temperature, the controller sends a signal to turn off the solar circuit pump to prevent thermal energy loss.

The controller also has a minimum upper temperature differential (hysteresis) that the outlet temperature must exceed relative to the inlet temperature to trigger the pump to turn on (in our case, 2 °C) to avoid frequent on/off cycling in response to the presence or absence of solar radiation, a common practice in real solar installations.

### 2.2. Equipment characteristics

For the study of the energy system, commercially available PVT, FP, and PV technologies were considered, as detailed



**Figure 1.** Schematic diagram of the solar heating and electric generation system using PVT, FP and PV modules.

in Table 1. The selected PVT collector is the aH72 model from Abora Solar (Abora Solar, n.d.), which has overall efficiencies (thermal and electrical) of up to 87%, making it one of the most efficient commercially available hybrid solar collectors. This collector has a capture area of 1.88 m<sup>2</sup> and, under standard operating conditions, can produce an electrical output of 350 W. The FP collector considered is the Maxol MS2.5 model from Modulo Solar (Módulo Solar, n.d.), with a capture area of 2.32 m<sup>2</sup> and the capability to produce a thermal output of up to 1.8 kW under standard conditions. The PV module used is from JA Solar, with a peak power of 350 W and an area of 1.95 m<sup>2</sup> (JA Solar, n.d.). The complete characteristics of the equipment can be found in the technical data sheets from each referenced company.

### 3. Methodology

#### 3.1. Case study: SDHW in Mexicali, Mexico

Solar heating and photovoltaic generation technologies were studied under the climatic conditions of Mexicali,

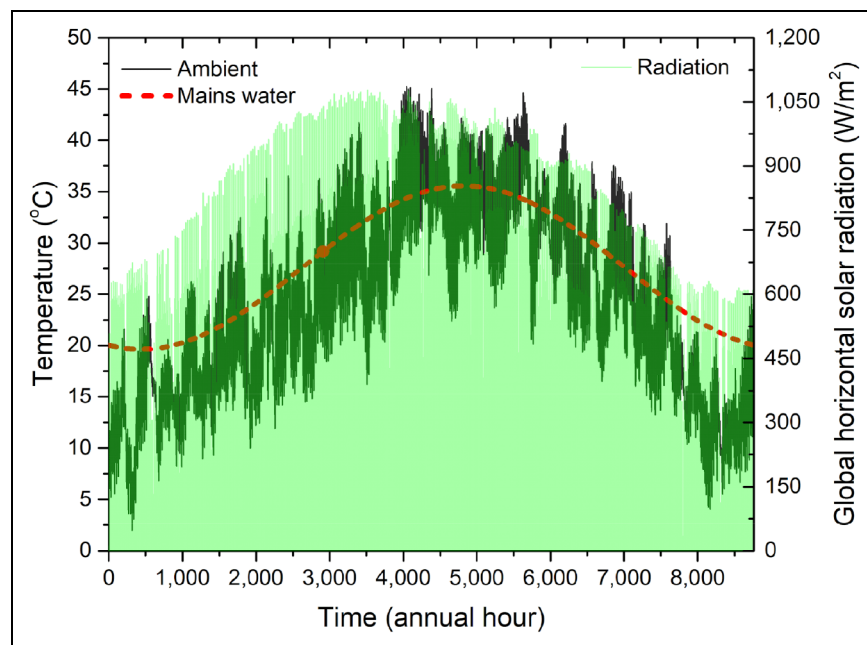
Baja California, Mexico (32°35'35.6"N 115°26'08.7"W), due to the extreme climate conditions present in this city. Mexicali is characterized by its low annual precipitation and has a hot desert climate (BWh), according to the Köppen climate classification. Its winters rarely have ambient temperatures below 0 °C, while summers often experience temperatures above 45 °C, with highs reaching up to 50 °C. This results in lower annual domestic hot water needs compared to other regions with colder climates.

Figure 2 shows the ambient temperature, tap water temperature, and global horizontal solar radiation for Mexicali, information obtained from PVGIS (JRC Photovoltaic Geographical Information System (PVGIS) - European Commission, n.d.). The tap water temperature is an important variable in the study, as its variation will influence the operating conditions of the solar heating system when entering the TEST and mixing with the hot water.

In this study, the Tapping program No. 3 of the EN 15316-3-1 standard (BS EN 15316-3-1:2007 - Heating systems in buildings, n.d.), or the Load profile L of the EN 13203-2 standard, was employed. This particular

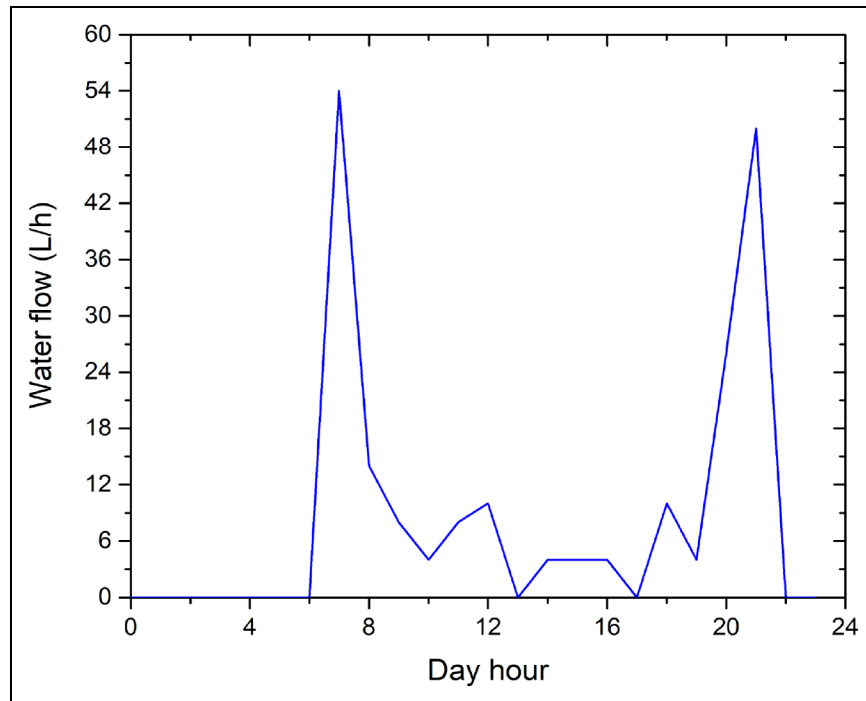
**Table 1.** Characteristics of the collectors and modules used in the study.

|                           | PVT                     | FP                                   | PV                        |
|---------------------------|-------------------------|--------------------------------------|---------------------------|
| Brand and model           | Abora Solar aH72        | Modulo Solar Maxol MS2.5             | JA Solar JAM72D00 –350/PR |
| Electrical power          | 350 W                   | -                                    | 350 W                     |
| Module area               | 1.88 m <sup>2</sup>     | 2.32 m <sup>2</sup>                  | 1.95 m <sup>2</sup>       |
| Intercept efficiency (a0) | 0.7                     | 0.777                                | -                         |
| Efficiency slope (a1)     | 5.98 W/m <sup>2</sup> K | 3.1 W/m <sup>2</sup> K               | -                         |
| Efficiency curvature (a2) | -                       | 0.01 W/m <sup>2</sup> K <sup>2</sup> | -                         |



**Figure 2.** Climatological information for the city of Mexicali, showing ambient temperature, tap water temperature and global horizontal solar radiation, obtained from PVGIS (JRC Photovoltaic Geographical Information System (PVGIS) - European Commission, n.d.).





**Figure 3.** DHW profile based on Tapping program No. 3 of the EN 15316-3-1 standard (BS EN 15316-3-1:2007 - Heating systems in buildings, n.d.).

profile, which corresponds to a daily hot water consumption of approximately 200 L, represents the needs of a single-family household. It was consistently used across all case studies to establish a comparable benchmark for hot water supply requirements. The graphical representation of this profile can be observed in Figure 3.

### 3.2. Energy indicators

To quantify the energy generation of the different technologies and perform a technical and economic comparison, indicators of energy generation and loss, auxiliary energy needs, and solar energy contribution to the demand were used, based on the number and area of PVT collectors, flat plate solar thermal collectors, and photovoltaic modules.

**3.2.1. Solar fraction.** Depending on the scenario, it may happen that not all the solar thermal energy produced by the solar collectors is utilized by the demand or user, as the period when solar energy is generated does not always coincide with the heat demand. This can result in solar thermal energy losses.

The solar fraction, also known as the coverage ratio or solar contribution, is an indicator used in solar thermal energy systems to measure the proportion of the total energy demand that is met by solar energy. Specifically, the solar fraction indicates how much of the energy demand for a specific service (such as domestic hot

water, heating, etc.) is directly covered by solar energy instead of conventional energy sources, such as electricity or fossil fuels. A value of 1 indicates that the entire heat demand for water heating was supplied with solar energy, while values less than 1 denote a contribution from another heat source, usually the auxiliary heating system. The solar fraction is calculated according to Eq. 1 (Gagliano et al., 2019a; Nualboonrueng et al., 2013):

$$f_{sol} = \frac{Q_{DHW} - Q_{aux}}{Q_{DHW}} \quad (1)$$

where  $Q_{DHW}$  is the annual thermal energy required to meet the needs of DHW and  $Q_{aux}$  the annual thermal shortage supplied by auxiliary equipment, after the solar contribution. The  $Q_{DHW}$  is calculated by considering the thermal energy required to raise the temperature of the mains water used for the demand, according to the consumption profile presented in Figure 3, to the setpoint temperature. For the case study in the city of Mexicali,  $Q_{DHW}$  resulted in a value of 1641 kWh annually, considering the mains water temperature presented in Figure 2 and a setpoint temperature of 45°C.

**3.2.2. Auxiliary energy.** In the event that there is insufficient solar energy capture and/or not enough energy stored in the TEST, the hot water supply to the demand must be ensured by an auxiliary heater, providing the required quantity and temperature level. To calculate the

thermal energy required by the mass flow of the heat transfer fluid to reach the user-defined temperature, in this case the set-point temperature for the DHW, Eq. 2 is used:

$$\dot{Q}_{need} = \dot{m}_{in,aux} \cdot C_{p,in,aux} \cdot (T_{set-point} - T_{in,aux}) \quad (2)$$

where  $\dot{Q}_{need}$  is the heat required by the stream entering the auxiliary to reach set-point temperature,  $\dot{m}_{in,aux}$  is the mass flow rate entering the auxiliary heater,  $C_{p,in,aux}$  is the specific heat of the heat transfer fluid, and  $T_{set-point}$  and  $T_{in,aux}$  are the set-point temperature and the temperature of the heat transfer fluid entering the auxiliary heater, respectively.

The annual auxiliary heat is calculated as shown in Eq. 3:

$$Q_{aux} = \sum_{i=0}^{8,760} \dot{Q}_{DHW}(i) - \dot{Q}_{TEST}(i) \quad (3)$$

where  $\dot{Q}_{DHW}(i)$  is the heat required by the demand in a given period, and  $\dot{Q}_{TEST}(i)$  is the heat stored in the TEST during the same period. If the value of  $Q_{aux}$  is positive, it means there is not enough solar energy stored in the TEST, thus requiring the auxiliary heater to supply the necessary energy. If the resulting value is negative, it means there is sufficient thermal energy in the TEST, and no auxiliary heating is needed, which is then accounted for in the annual summation. The result of this equation shows the annual auxiliary heat required to meet the DHW demand.

### 3.3. Economic analysis

To compare the economic viability of the different solar heating and electrical generation systems studied, a detailed economic assessment was conducted. This assessment accounted for the prices of both electrical and thermal energy, as well as the projected savings over the system's expected lifespan of 20 years. The goal was to evaluate the return on investment for the PVT, FP, and PV solar collectors.

The justification for this economic analysis lies in its ability to provide a comprehensive understanding of the financial feasibility of deploying solar energy systems in residential settings. By incorporating key factors such as energy prices, installation costs, and system efficiency over a significant timeframe (20 years), the analysis ensures that the economic performance of each technology is thoroughly evaluated. This approach aligns with established practices in the field, allowing for meaningful comparisons with similar studies and providing valuable insights for stakeholders considering the adoption of solar energy technologies. The focus on long-term economic benefits, despite the high initial investment costs, is essential for assessing the sustainability and practicality of these systems in real-world applications.

The annual monetary savings,  $R_y$ , were calculated using Eq. 4. This calculation considered the reduction in energy consumption due to the production of electrical and

thermal energy by the solar collectors (Gagliano et al., 2019a):

$$R_y = E_{el} \cdot Price_{el} + \frac{Q_{DHW} - Q_{aux}}{\eta_{th,aux}} \cdot Price_{fuel} \quad (4)$$

where  $E_{el}$  is the electrical energy produced by the PVT and PV collectors, and  $Price_{el}$  and  $Price_{fuel}$  is the cost of electrical and thermal energy per kWh, respectively.  $\eta_{th,aux}$  is the conversion efficiency of the auxiliary heater, which in this case was considered as 0.9.

To determine the total monetary savings over the project's lifespan, the discounted sum (RRR) was calculated using Eq. 5 (Gagliano et al., 2019a):

$$R = \sum_{i=1}^n R_y \cdot \frac{q^n - 1}{r \cdot q^n} \quad (5)$$

It is considered a capitalization rate,  $r$ , of 3% and a projection of the useful life of the project,  $n$ , of 20 years, and  $q = 1 + r$ . The above values were considered in order to have the same economic criteria used by other authors in systems related to the proposal. To provide a clear and comprehensive comparison, the costs associated with the equipment used in this study, as well as the average costs of electrical and thermal energy in Mexicali, Mexico, were included. These costs, shown in Table 2, were based on market data and official energy pricing from the national energy provider.

In addition to these costs, the study considered two different capacities for the TEST (200 L and 300 L) across the various solar heating systems. This was done to evaluate the impact of increased hot water storage capacity on economic indicators such as energy savings, solar fraction, and overall system efficiency.

### 3.4. Simulation process

To analyze the operational behavior of the different solar heating and electricity generation technologies, the TRNSYS software (Klein, 2015) was used to simulate the

**Table 2.** Cost of equipment and energy considered in this study.

|                                                                               |           |
|-------------------------------------------------------------------------------|-----------|
| Panels and structure (USD/m <sup>2</sup> ) (Gagliano et al., 2019b)           | \$ 300.00 |
| Inverter and accessories (USD/W) (Exel Solar, 2024)                           | \$ 0.19   |
| Hydraulic circuit (USD/m <sup>2</sup> ) (Herrando and Markides, 2016)         | \$ 112.00 |
| Solar tank (USD) (Herrando and Markides, 2016)                                | \$ 951.00 |
| Installation cost (USD/m <sup>2</sup> ) (Herrando and Markides, 2016)         | \$ 100.00 |
| Price of thermal energy (USD/kWh) (Baja Gas & Oil, 2024)                      | \$ 0.0191 |
| Price of electrical energy (USD/kWh) (Comisión Federal de Electricidad, 2024) | \$ 0.1000 |

energy system under the selected site's transient climatic conditions. TRNSYS is widely used in the development and study of energy systems, as it features an extensive library of models for energy generation equipment and processes in general, allowing for the creation of new models that can be incorporated into the simulation study. Since TRNSYS includes models for processing typical meteorological year (TMY) datasets, the proposed systems can be studied under expected operational conditions at the site where the energy system is intended to be developed, providing accurate and representative results.

For the simulation of the energy systems considered in this study, models of flat plate solar heating equipment (type1), crystalline photovoltaic modules (type103), pumps (type114), TEST (type158), auxiliary heaters (type122), mixing valves (type11 h), and differential controller with hysteresis (type165) were used. The PVT solar module was simulated following the methodology used in previous works (Aguilar-Jiménez et al., 2020; Gagliano et al., 2019a), combining a photovoltaic module model (type103) and a flat plate collector (type1), mutually feeding back to calculate both electrical and thermal production. The validation of this methodology can be verified in those works. The solar FP heating system and the photovoltaic system, operating independently, were developed considering the characteristics of the equipment presented in Table 1 and their technical datasheets. The mathematical models used to develop the different types employed in this simulation can be verified in the TRNSYS mathematical references (Klein et al., 2014). The explanation of the operation of the FP solar collector and the PV module are presented below.

**3.4.1. FP solar collector (type1).** This component models the thermal performance of an FP solar collector using a standard quadratic (or linear) efficiency equation, which is presented in Eq. 6:

$$\eta_{fp} = a_0 - a_1 \frac{(\Delta T)}{I_T} - a_2 \frac{(\Delta T)^2}{I_T} \quad (6)$$

where  $\eta_{fp}$  is the collector solar thermal efficiency,  $a_0$ ,  $a_1$  and  $a_2$  are the collector efficiency coefficients,  $\Delta T$  is the temperature difference of the heat transfer fluid at the inlet of the collector with respect to the ambient temperature, and  $I_T$  is the global radiation incident on the collector. This model also considers the modification of the angle of incidence (IAM) on the efficiency of the FP, both transversely and longitudinally (Klein et al., 2014).

**3.4.2. Photovoltaic module (type103).** For the mathematical representation of the PV module, the type103 uses an empirical equivalent circuit model to predict the current-voltage characteristics of a module. The strength of the current source is dependent on solar radiation and

the current-voltage characteristics of the diode are temperature-dependent. The results for a single module equivalent circuit are extrapolated to predict the performance of a multi-module array (Klein et al., 2014). The model takes into consideration the solar radiation incident on the PV module, the operating temperature and the IAM to calculate the power delivered by the PV module or array. For a detailed description of the electrical, thermal and optical model, we suggest reviewing the TRNSYS mathematical references (Klein et al., 2014).

## 4. Results

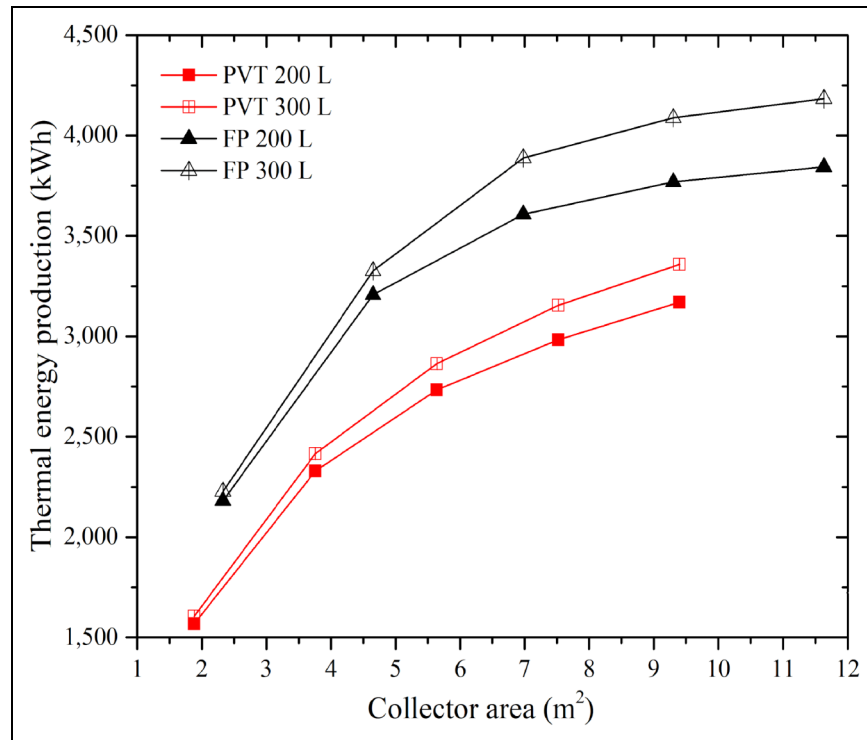
This section presents the comparative energy analysis between PVT and FP collectors, based on the solar capture area and the volume of the TEST. For the analysis, the total solar capture area was considered, as the area of the FP collector is 23% larger than that of the PVT collector. Since the variation in the volume of the TEST affects solar utilization, two storage tanks were considered for comparison: one with 200 L, which is the typical commercially available TEST volume for solar DHW, and the other with 300 L to observe the effect of increased volume on the indicators.

### 4.1. Thermal energy production

In Figure 4, the annual thermal energy production for PVT and FP collectors is presented as a function of the total capture area and TEST volume. When considering one collector for both technologies, it was found that the FP can generate up to 39% more thermal energy per year compared to the PVT. This is primarily due to the larger solar capture area and higher solar thermal efficiency of the FP. When comparing thermal energy production based on the capture area, an FP collector produces 12% more energy per year than the PVT, with 937.6 kWh/m<sup>2</sup>, while the latter produces 834.1 kWh/m<sup>2</sup>.

Increasing from one to two collectors for each technology (PVT and FP) and considering a 200 L TEST, the increases in thermal energy production are 48% and 47%, respectively. As the solar capture area increases and the TEST volume remains constant, the increase in thermal energy production is not linear. Increasing from two to three collectors (a 50% larger capture area), the annual thermal energy increase is 12% and 17% for FP and PVT collectors, respectively, while increasing from four to five collectors results in increases of only 2% and 6%, respectively.

Increasing the TEST volume from 200 L to 300 L benefits the thermal energy production for both types of collectors. This is because having a larger mass of water available for heat storage allows for a greater amount of solar thermal energy to be supplied to the demand, resulting in a lower operating temperature of the solar circuit and improving solar-thermal conversion efficiency. However, the benefit of increasing the TEST volume from 200 L to 300 L is only significant for capture areas greater than 5.5 m<sup>2</sup> for



**Figure 4.** Annual thermal energy production as a function of collector area and TEST volume.

PVT collectors and 7 m<sup>2</sup> for FP collectors, as heat losses occur with a 200 L volume, which can be stored with the 300 L TEST.

To meet the thermal energy demand of the case study, 1641 kWh per year are required, as mentioned earlier. Considering the thermal energy production from Figure 4, it can be seen that the annual heat demand is exceeded by 33% with a single FP collector, and only 4.4% of heat would be missing with a single PVT collector. However, due to the nature of solar thermal energy and hot water consumption patterns, there are times when hot water demands do not coincide with solar heating. In our case study, the highest demands occur in the mornings and evenings, which do not align with the necessary generation of solar thermal energy. Therefore, it is essential to use TEST to store thermal energy when the solar resource is available, so it can be used during times when demand is present but solar contribution is absent. However, this process results in heat losses because storing hot water leads to heat transfer to the environment due to the temperature difference, which represents a loss of solar energy. The amount of energy lost depends on the number of collectors, the volume of the TEST, climatic conditions, and the DHW profile.

#### 4.2. Solar fraction

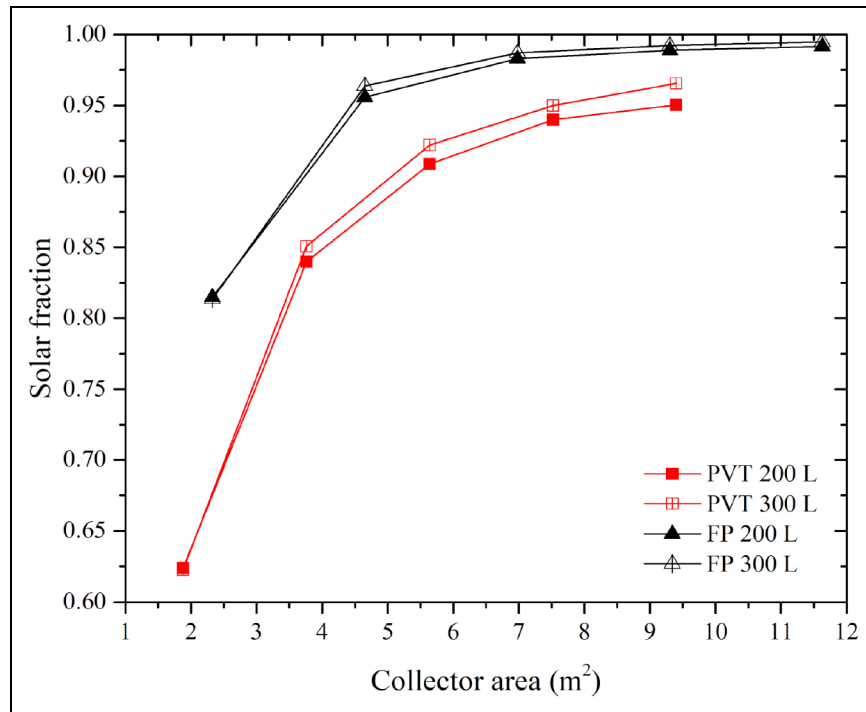
To contextualize the thermal energy production of both technologies and their contribution to the DHW demand, Figure 5 shows the solar fraction. It can be seen that,

despite a single FP collector producing 33% more than the annual energy required for DHW, the resulting solar fraction is 0.81, necessitating the auxiliary system to supply the remaining 19% of the energy. Similarly, a single PVT collector achieves a solar fraction of 0.62, despite producing 95.6% of the heat demand.

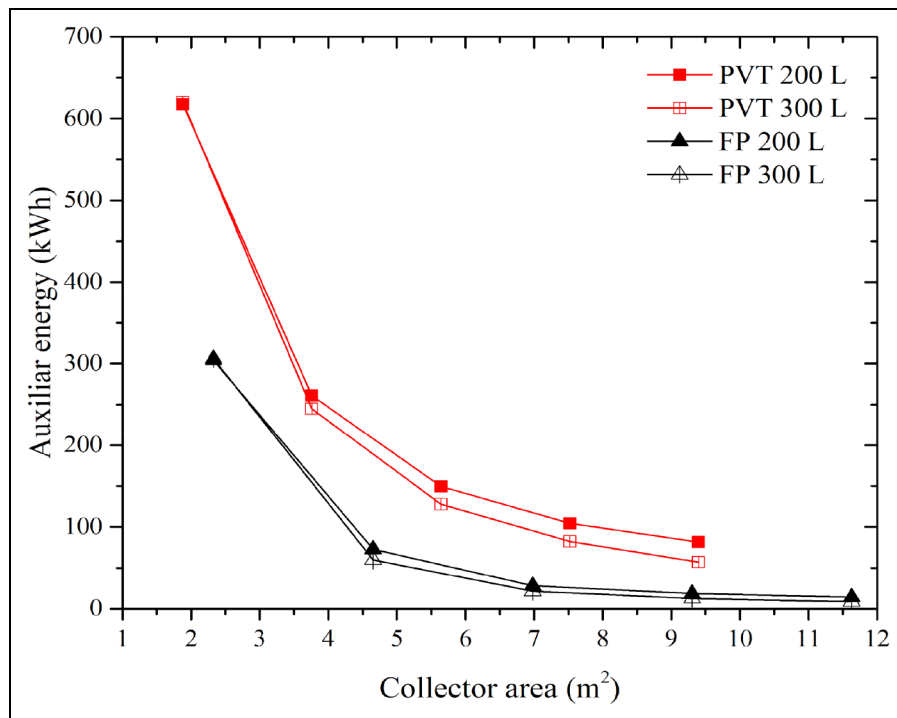
When the solar capture area is doubled, there are increases in the solar fraction to 0.96 and 0.84, equivalent to percentage increases of 17.3% and 34.6% for FP and PVT collectors, respectively. Further increases in the solar capture area for the FP technology are not significant in improving the solar fraction, as nearly the entire heat demand for DHW (96%) is met by two FP collectors. However, to achieve the same solar fraction with PVT collectors, more than double the solar capture area and a 300 L TEST are required. The analysis of the solar fraction helps quantify the actual solar energy contribution from the solar collectors. However, the final number of collectors to be used should be based on a technical and economic analysis.

#### 4.3. Auxiliary energy

In Figure 6, the auxiliary energy requirements to meet the DHW demand are presented as a function of the solar capture area and the TEST volume. Notably, when using a PVT and an FP collector, the PVT requires 103% more auxiliary thermal energy than the FP, regardless of the TEST volume. This difference in auxiliary heat requirement tends to decrease as the number of collectors increases, with



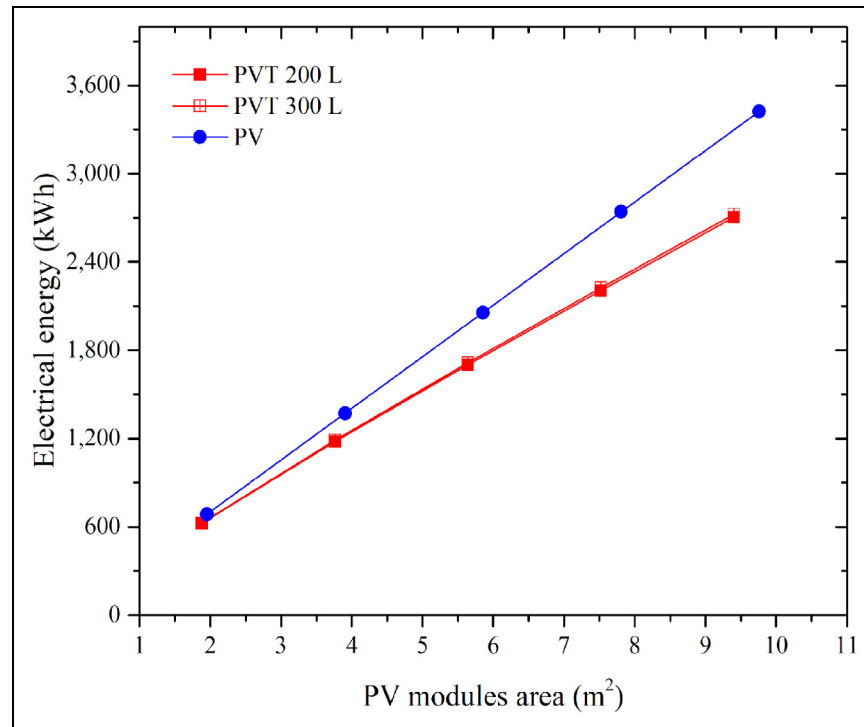
**Figure 5.** Solar fraction as a function of collector area and TEST volume.



**Figure 6.** Auxiliary energy requirements to meet DHW demand, based on solar collector area and TEST volume.

a reduction of 57% and 76% in auxiliary energy when increasing from one to two PVT and FP collectors, respectively, considering a 200 L TEST.

Increasing the TEST volume tends to reduce the auxiliary energy requirement with the use of two or more collectors for both technologies, with a greater reduction for the



**Figure 7.** Comparison of annual PV electrical energy production as a function of the catchment area for PVT collectors and PV modules separately.

PVT. This is because, with a single collector, most of the generated heat is used to meet the demand, leaving no energy for storage. When the number of collectors increases, excess heat is generated, which can be stored in a larger TEST to be used during periods of low or no solar radiation. However, the reduction in auxiliary energy when increasing the TEST volume to 300 L shows maximum values of 25 kWh, equivalent to 1.5% of the total DHW demand (1641 kWh), when using five PVT collectors. The percentage reduction is even smaller for FP collectors, with a maximum reduction of 13 kWh when using two collectors, equivalent to 0.8% of the total DHW demand.

#### 4.4. Electrical energy

A comparison was made of the annual PV electrical energy production by PVT collectors with different TEST volumes and PV modules, based on the photovoltaic capture area. This can be visualized in Figure 7. Photovoltaic conversion efficiency tends to decrease as the operating temperature of the PV cell increases. This is the main factor why PVT technology tends to generate less electrical energy based on the capture area as the number of collectors increases.

When the capture area of PVT collectors increases, the operating temperature of the solar heating circuit also increases, resulting in a higher operating temperature of the cell in this technology compared to PV modules operating independently, leading to lower electrical production. A

PV module produces 9.7% more electrical energy than a PVT collector. As the capture area increases, PV modules produce up to 350 kWh/m<sup>2</sup>, while PVTs produce a maximum of 287 kWh/m<sup>2</sup>, 22% more annual energy per square meter.

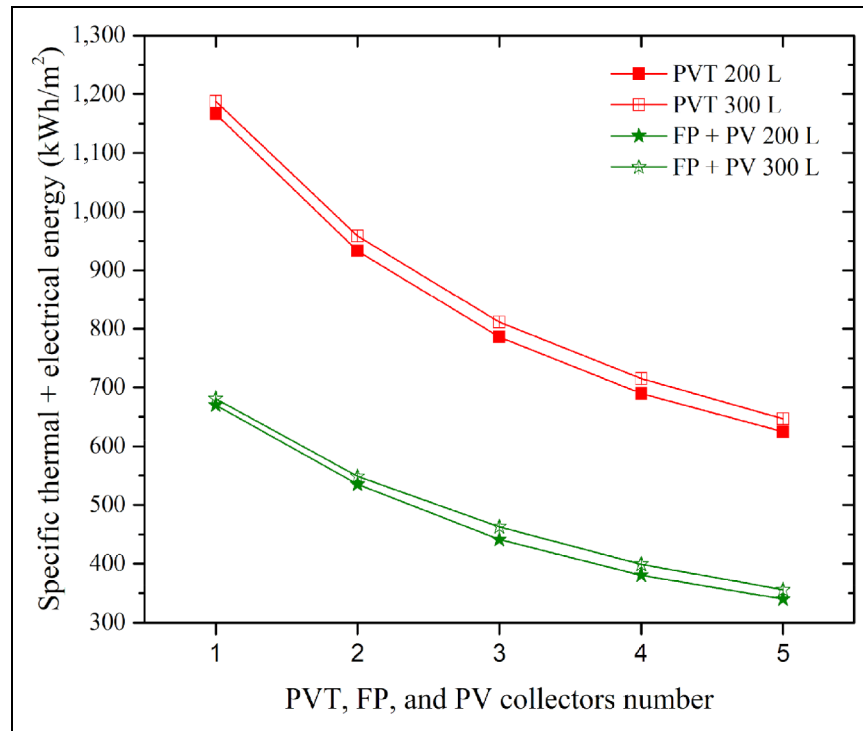
It is noteworthy that the TEST volume does not have a significant impact on the electrical energy production of PVT collectors, with maximum increases of 0.77% when having a 300 L tank compared to a 200 L tank. This is because a larger tank volume slightly reduces the operating temperature of the water heating system, increasing conversion efficiency, but not significantly.

#### 4.5. Specific thermal and electrical energy

In Figure 8, the global specific energy production (thermal and electrical) per square meter of capture area is shown as a function of the number of PVT and FP collectors with PV, with TEST volumes of 200 and 300 L. This comparison allows for analyzing the combined thermal and electrical production per square meter of installed solar capture area, unlike the previous results that discussed thermal and electrical energy separately.

It can be seen that the PVT technology, when using one collector and a 200 L TEST, presents an annual global specific energy production of 1166 kWh/m<sup>2</sup> (approximately 60% thermal energy), 74% more global energy per square meter than a system with one FP collector and one PV





**Figure 8.** Overall annual specific energy production (thermal and electrical) per square meter of collector as a function of the number of PVT and FP collectors with PV, considering TEST of 200 and 300 L.

**Table 3.** Comparison of investment costs and economic recovery over the life of the project in USD.

| Collec-tors | PVT 200 L |         | PVT 300 L |         | FP + PV 200 L |         | FP + PV 300 L |         | FP 200 L |         | FP 300 L |         |
|-------------|-----------|---------|-----------|---------|---------------|---------|---------------|---------|----------|---------|----------|---------|
|             | Cost      | Revenue | Cost      | Revenue | Cost          | Revenue | Cost          | Revenue | Cost     | Revenue | Cost     | Revenue |
| 1           | 1735      | 1251    | 1980      | 1255    | 1935          | 1441    | 2180          | 1440    | 1634     | 422     | 1879     | 421     |
| 2           | 2764      | 2191    | 3009      | 2208    | 2613          | 2533    | 2858          | 2538    | 2011     | 495     | 2256     | 499     |
| 3           | 3793      | 3002    | 4038      | 3027    | 3290          | 3567    | 3535          | 3569    | 2387     | 509     | 2632     | 511     |
| 4           | 4822      | 3767    | 5067      | 3800    | 3968          | 4589    | 4213          | 4591    | 2764     | 512     | 3009     | 514     |
| 5           | 5851      | 4513    | 6096      | 4552    | 4646          | 5610    | 4891          | 5611    | 3141     | 513     | 3386     | 515     |

module. As the number of collectors increases, the global energy production per square meter tends to decrease for all technologies. For PVT and FP technology, this is due to the impact on thermal and electrical energy production caused by the increase in the system's operating temperature, while PV modules are not affected.

When considering a 300 L TEST, an increase in global specific production is observed for both comparisons. For the PVT system, average increases of 3% are seen depending on the number of collectors, while for the FP and PV combination, average increases of 3.7% are observed.

#### 4.6. Investment and revenue costs

To compare the economic feasibility of the different systems studied, Table 3 shows the investment costs and

economic recovery of PVT, FP, and FP + PV technologies, with TEST volumes of 200 and 300 L. The comparison considers the equipment and energy costs presented in Table 2, as well as a system lifespan of 20 years, so the revenue is the total over that period.

Notably, with PVT technology, both with 200 L and 300 L TEST, the revenue never exceeds the investment costs. The low cost of thermal and electrical energy is one of the main factors why there is no return on investment. Additionally, the high cost of the hydraulic and electrical installation of PVT systems further increases the investment cost. Another factor impacting the return on investment is the low thermal energy demand for DHW, which prevents nominal use of the solar installation for much of the year.

From an economic perspective, the high initial costs of PVT and FP technologies, along with the associated

installation and maintenance costs, represent a significant challenge. Although these systems can offer economic benefits in regions with high thermal and electrical energy costs, the case study demonstrates that no significant economic benefit is achieved during the project's lifetime. For the PVT technology, over the 20-year lifespan considered in this study, the returns on initial investment reach up to 79% and 75% in the best-case scenario, with TEST volumes of 200 L and 300 L, respectively. When services are provided separately using FP and PV technologies, returns of up to 121% and 115% of the initial investment are achieved with TEST volumes of 200 L and 300 L, respectively. For the FP technology alone, returns of up to 26% and 22% of the initial investment are achieved with TEST volumes of 200 L and 300 L, respectively.

Similarly, for the FP + PV combination, although there is a return on investment from a system with 3 FP collectors and 3 PV modules, both for 200 L and 300 L TEST, this return occurs after 18 and 20 years of operation, respectively, which is not economically attractive. The same factors affecting PVT systems contribute to this result. For standalone FP collectors, the high initial investment costs combined with the low cost of thermal energy in the case study prevent economic recovery.

## 5. Discussion

The obtained results indicate that when analyzing energy production, solar fraction, and auxiliary energy requirements, the thermal and electrical generation systems using separate collectors and modules perform better compared to hybrid PVT collector systems. However, when comparing these technologies in terms of specific global energy efficiency, PVT collectors prove to be competitive against energy generation with separate equipment. This is mainly due to the utilization of the capture area; using the same surface area for simultaneous thermal and electrical energy generation achieves up to 74% more energy utilization per square meter of installation.

In the domestic sector context, where available space for such installations may be very limited, PVT technology offers an efficient solution for generating both types of energy. However, in scenarios where area availability is not an issue, energy generation using separate equipment remains the most attractive option.

Incorporating the economic analysis significantly changes the perspective on technical feasibility. Currently, the costs associated with PVT technologies and FP collectors, along with installation and maintenance costs, represent significant challenges. The initial investments for such projects are high compared to their economic benefits. Unlike the installation costs of PV systems, solar heating systems tend to be more expensive due to the complexity of the hydraulic installation, increasing the total project cost.

In scenarios where thermal and electrical energy costs in the domestic sector are high, the economic viability of these

projects improves, partially offsetting the high initial investment cost. However, in the case study, with thermal and electrical energy costs of \$0.0191 and \$0.1 USD/kWh, respectively, an economic benefit is not achieved during the project's lifespan.

The ambient temperature conditions in the case study region result in a low amount of thermal energy required to meet the DHW demand. This leads to achieving solar fractions of 0.62 and 0.81 with a single PVT collector and an FP collector, respectively. A higher DHW demand would improve the utilization of the solar energy installation. However, it is important to highlight that a PVT collector generates 95.6% of the energy required for DHW in the case study, while an FP collector exceeds the 1641 kWh needed annually by 33%. The main issue is that the hourly DHW demand does not coincide with solar generation, necessitating the use of a TEST to store the excess heat. Nevertheless, the cost of the TEST is one of the main expenses in such projects, significantly impacting their economic viability.

When comparing the results of this study with those from other research, it is consistent that the greatest economic benefit is achieved when the photovoltaic system operates independently, due to the lower installation costs compared to PVT technology (Gagliano et al., 2019a). Similarly, the findings of this study, as well as those of other authors, indicate that the negative economic outcomes of PVT technologies in the domestic sector are primarily due to low energy costs (Gagliano et al., 2019a), which prevent the recovery of the initial investment within a reasonable timeframe.

## 6. Conclusion

This study evaluated the technical and economic viability of PVT collector systems compared to separate thermal and electrical generation systems. The results indicate that although separate systems offer better performance in terms of energy production, solar fraction, and auxiliary energy requirements, PVT collectors are a competitive option in terms of overall specific energy efficiency, particularly in applications where available space is limited.

The efficiency of PVT systems is highlighted by their ability to maximize the utilization of the capture area, achieving up to 74% more energy utilization per square meter compared to separate systems. This makes them an attractive solution for the domestic sector, where space is often a significant constraint. However, in scenarios where the availability of area is not an issue, separate energy generation remains the most efficient option. Using a single PVT collector achieves a solar fraction of 0.62, while an FP collector reaches 0.81. When two collectors are considered, the solar fractions increase to 0.96 and 0.84, corresponding to percentage increases of 17.3% and 34.6% for FP and PVT collectors, respectively. The solar fraction analysis indicates that a small number of collectors



is sufficient to meet the DHW demand, suggesting that the available installation area would not be a primary factor in choosing this technology under these conditions.

From an economic perspective, the high initial costs of PVT and FP technologies, along with the associated installation and maintenance costs, represent a significant challenge. Although these systems can offer economic benefits in regions and sectors with high thermal and electrical energy costs, as the commercial and industrial sector with the highest possible thermal energy demand during the longest possible time of the year. The case study demonstrates that no significant economic benefit is achieved during the project's lifetime in this scenario in DHW. In the best scenarios, the returns on initial investment during the project's lifespan are 79%, 121%, and 26% for PVT, FP + PV, and FP technologies, respectively, considering a 200 L TEST volume. It was identified that reducing installation expenses is a critical factor for the economic feasibility of PVT technology. Therefore, efforts should focus on reducing material costs and developing accessories that facilitate rapid installation, thereby reducing the time required for specialized labor.

Additionally, climatic conditions and DHW demand affect the efficiency of the systems. The discrepancy between solar energy generation and hourly DHW demand necessitates the implementation of TEST, which significantly increases costs and impacts the economic viability of the project. Opportunities to improve the technical and economic feasibility of these technologies are directed towards their adaptability in extreme climates and low thermal energy demands, as presented in the case study. While PVT systems offer an innovative and efficient solution for the simultaneous generation of thermal and electrical energy, their large-scale adoption is conditioned by space availability and economic considerations. Future research and technological developments that reduce installation costs and improve thermal storage efficiency could enhance the viability of PVT systems, making this technology an even more competitive option in the renewable energy field.

### Declaration of conflicting interests

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## **ANEXO F**

**Sistema de desalinización MSF/MED-LTBT: que opera al alto vacío, con baja TBT y sin la necesidad de energía térmica de activación.**

### **REPORTE TÉCNICO**

## **Sistema de desalinización MSF/MED-LTBT: que opera al alto vacío, con baja TBT y sin energía térmica de activación.**

Elaboración: M.I. Cristian Ascencion Casares De la Torre

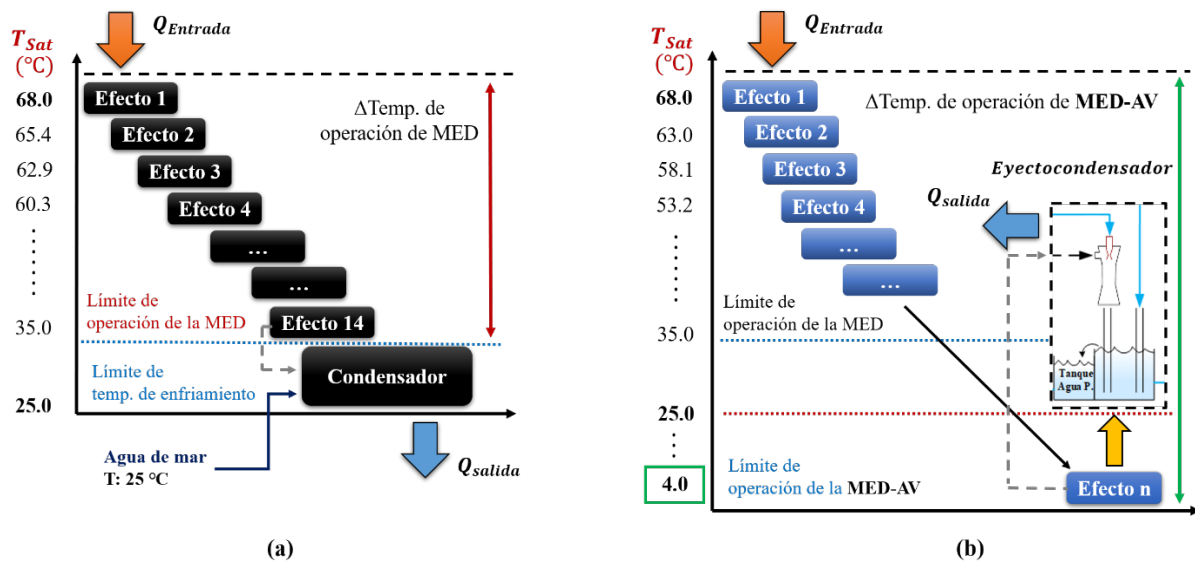
### **1. Introducción**

La tecnología de desalinización mediante destilación de múltiple efecto (MED, por sus siglas en inglés) ha sido ampliamente utilizada para la producción de agua desalinizada en zonas áridas, donde la disponibilidad de recursos hídricos es limitada. Sin embargo, la eficiencia energética y el rendimiento de estos sistemas están estrechamente vinculados a la temperatura de ebullición superior (TBT, Top Brine Temperature) y a la temperatura de ebullición inferior (BBT, Bottom Brine Temperature).

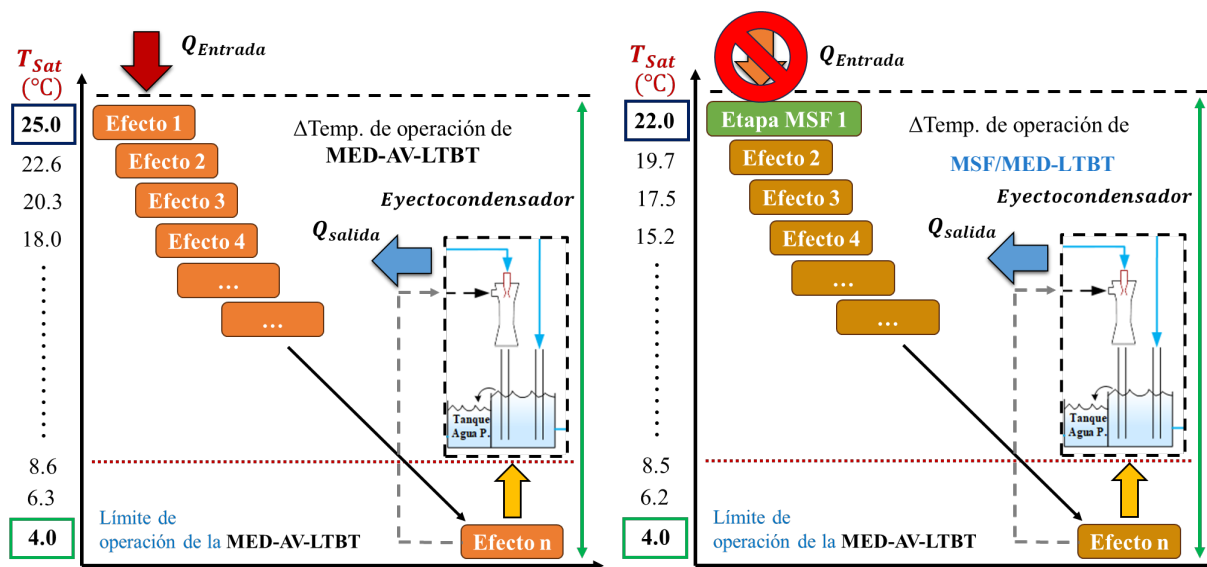
Generalmente, se establece una TBT de alrededor de 70 °C para evitar problemas de incrustación (scaling) debido a la precipitación de sales como el carbonato de calcio y el sulfato de calcio, que son más solubles a temperaturas más bajas. Aumentar la TBT por encima de los 70 °C incrementa el riesgo de formación de estas incrustaciones, lo que puede reducir la eficiencia en la transferencia de calor y aumentar los costos de mantenimiento. En los sistemas convencionales, la BBT suele oscilar entre 35°C y 40°C, ya que es necesario mantener un delta de temperatura respecto al agua de mar de alimentación, cuya temperatura suele estar entre 25 °C y 27 °C. Este delta de temperatura entre la temperatura del agua de alimentación y la BBT, es crucial para garantizar la condensación del vapor del último efecto del sistema MED.

Recientes avances en el campo de la desalinización han sugerido la posibilidad de operar a temperaturas de alimentación más bajas, reduciendo la TBT a valores por debajo de los 70 °C y hasta los 25 °C. Este enfoque innovador, respaldado por la implementación de un subsistema de eyecto-condensación barométrica, tiene el potencial de minimizar los problemas de incrustación y corrosión, y principalmente lograr una mejorar la eficiencia y rendimiento operativo de las plantas de desalinización MED.

El presente reporte tiene como objetivo evaluar el impacto de la disminución de la TBT en un sistema MED, analizando sus efectos sobre la eficiencia térmica ( $GOR_G$ ), el consumo específico de energía ( $SEC_G$ ) y el rendimiento global del proceso.



**Figura 1.** Concepto básico de la operación del sistema MED-AV (a) Rango de operación MED convencional y su límite de BBT, y (b) Rango operativo MED-AV y su límite de BBT



**Figura 2.** (a) Rango de temperatura operación del sistema MED-AV-LTBT con baja TBT y (b) Rango de temperatura operación del sistema MSF/MED-LTBT.

## 2. Descripción del sistema

En la Figura 3 se presenta el diagrama del sistema de desalinización **MSF/MED-LTBT** para la producción simultánea de agua y frío que implementa la eyecto-condensación barométrica y opera a alto vacío y baja temperatura de activación. El sistema **MSF/MED-LTBT** está conformado por tres subsistemas: 1) el subsistema de desalinización FLASH, 2) el subsistema de desalinización por evaporación de múltiple efecto al alto vacío (MED-AV) y 3) el subsistema de eyecto-condensación y producción de frío.

De acuerdo con la figura 3, la bomba (*BAM1*) succiona a la corriente de agua de mar (1) y la dirige como la corriente (2) al subsistema de desalinización FLASH, que consiste en una etapa de flasheo. La corriente de agua de mar (2) pasa por la válvula de expansión (*VE1*), para bajar súbitamente su presión hasta la presión de saturación correspondiente a la Top Brine Temperature (TBT) de la MED. Es importante mencionar que la corriente de agua de mar (2) al pasar por la *VE1* se flashea quedando en una corriente bifásica (líquido-vapor). En estas condiciones; la corriente de agua de mar (3), ingresa a la etapa flash (*EFH1*) donde se divide en las corrientes de salmuera (12) y vapor de agua (7). La corriente de salmuera (12) es dirigida al evaporador del primer efecto de la MED (*HX2*), ingresando a través de una tubería en forma de U (*TU<sub>1</sub>*). Lo anterior permite tener un sello hidráulico para mantener la diferencia de presión entre la etapa 1 y el efecto y 2. Es importante mencionar, que la corriente de salmuera (12) al pasar de la presión *P1* a *P2* ( $P1 > P2$ ) se flashea quedando en dos fases y con una menor temperatura (corriente 13); y en estas condiciones ingresa al *HX2* para ser calentada por la corriente de líquido-vapor (7).

Por otro lado, el vapor de agua (7) ingresa al *HX2* e intercambia calor con la corriente de salmuera (13), condesándose totalmente y saliendo como agua producto en la corriente (8). La corriente de salmuera (13) que sale del *HX2*, se encuentra en dos fases. Esto es debido a que, utilizando la corriente (7), se produce vapor en la corriente de salmuera (13). Por otra parte, el agua producto (corriente 8) del primer efecto, una vez que pasa a través de la válvula de expansión (*VE1<sub>AP1</sub>*) y se flashea, se unirá con el agua producto de los efectos posteriores. La corriente de salmuera en dos fases (14) es dirigida al separador líquido-vapor de salmuera (*SEP2*), en donde se divide en las corrientes de vapor (15) y de salmuera (16). La corriente de vapor (15), se une con la corriente de vapor generado por flasheo del agua producto (11), proveniente del separador líquido-

vapor de agua producto ( $SEP1_{AP}$ ), convirtiéndose en la corriente (19). Esta corriente ingresa al evaporador del tercer efecto (HX3). Por otra parte, la corriente de salmuera (16), hace lo propio, al dirigirse a través del sello hidráulico en forma de U ( $TU_2$ ) y luego pasar hacia el evaporador HX3, donde termina con la presión del tercer efecto (P3). Lo ocurrido en el Efecto 2, sucede en los posteriores efectos, hasta el Efecto 7.

El agua producto de cada efecto, se une con la del efecto posterior a través de tubos en forma de U. Con lo anterior, se forma un sello hidráulico y se logra mantener la diferencia de presión entre efectos. Este proceso se repite hasta el último efecto (EFECTO 8).

El subsistema de eyecto-condensación y producción de frío, integra los componentes clave de la innovación —el evaporador (EVA), la válvula de tres vías (V3V1) y el eyector-condensador (EYC)— junto con equipos auxiliares. Estos últimos incluyen el mezclador de vapor (MIXVE), las bombas para agua de mar (BAM2) y para agua producto (BAP), el intercambiador de calor  $HX_{SW}$ , los tanques para agua producto (TAP), y para salmuera (TS), y tres columnas barométricas: dos para la extracción de agua producto (CB1 y CB3) y una para salmuera (CB2).

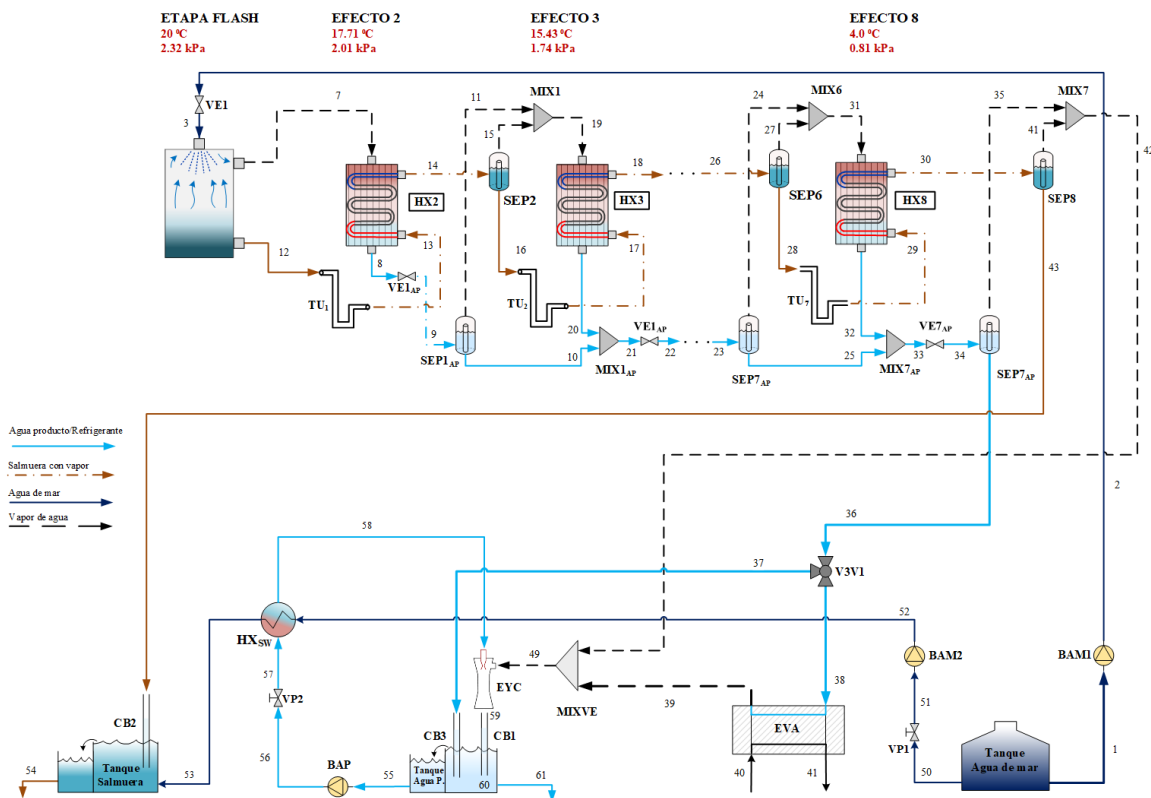


Figura 3. Diagrama del sistema de desalinización al alto vacío MSF/MED-LTBT.

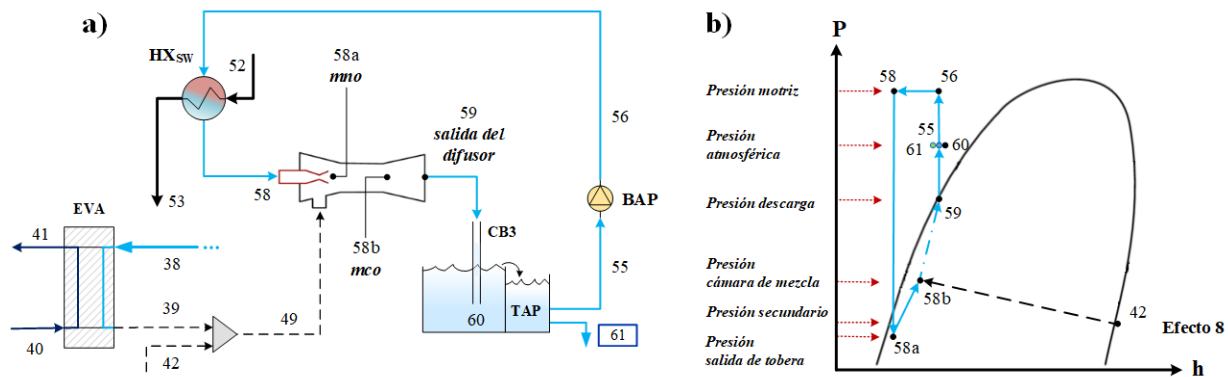
La válvula de tres vías (V3V1) es el componente clave para regular la capacidad de enfriamiento del sistema. Como se muestra en la Figura 3, esta válvula divide la corriente de agua producto total (corriente 36) en dos: una que se dirige hacia la CB3 para extracción (37) y otra que alimenta al evaporador (EVA) como refrigerante (38). La producción de frío es proporcional al caudal másico que fluye hacia el evaporador (corriente 38). Para operar el sistema exclusivamente en modo de desalinización, todo el caudal se desvía a través de la corriente (37). La corriente (37) desciende por la columna barométrica (CB1) hacia el tanque para agua producto (TAP), donde se libera el vacío y el fluido alcanza condiciones atmosféricas para su extracción final (corriente 61).

Por otra parte, la corriente de agua producto como líquido saturado (38) ingresa al evaporador (EVA), donde actúa como refrigerante. Al recibir calor del fluido a enfriar (corriente 40), el agua (38) se evapora completamente (convirtiéndose en la corriente de vapor 39), generando el efecto de enfriamiento deseado. La corriente (38) sale del evaporador (EVA) como la corriente (39), después de haber transferido calor con la corriente (40). El vapor de agua (39) se combina en el mezclador (MIXEV) con el vapor del Efecto 8 (corriente 42), formando la corriente de vapor (49). Finalmente, esta corriente de vapor a baja presión y temperatura (49), es succionada y comprimida en el eyector-condensador (EYC) donde se condensa totalmente.

Dentro del eyector-condensador (EYC), la corriente motriz —fluido primario (58)— arrastra y comprime la corriente de vapor de succión—fluido secundario— (corriente 49) induciendo su condensación. La mezcla resultante sale del EYC como una corriente líquida (59). Es fundamental destacar que la presión de la corriente motriz (58) es superior a la presión de succión (49) y a la presión de descarga (59). Como se detalla en la Figura 4, la expansión de la corriente motriz (58) en la tobera convergente-divergente, genera una disminución de presión (punto 58a) que, al ser inferior a la presión del vapor secundario (49), es la responsable de succionar este último. Esta condición define el punto de presión más bajo en todo el sistema.

La interacción en la cámara de mezcla (entre las corrientes 58a y 49) y la posterior desaceleración en el difusor comprimen la mezcla bifásica, logrando la condensación completa del vapor al convertir su energía cinética en incremento de presión. El calor latente de condensación liberado es absorbido por el fluido motriz (49), resultando en una corriente líquida a presión intermedia con un incremento de temperatura (corriente 59).





**Figura 4.** Diagrama P-h del eyector-condensador barométrico del sistema MSF/MED-LTBT con producción de frío.

La corriente líquida (59) se descarga a presión atmosférica a través de la columna barométrica CB1 hacia el tanque de agua producto (TAP). Del TAP, una parte del agua (corriente 55) es recirculada por la bomba BAP utilizándola como fluido motriz del eyector-condensador, mientras que el excedente (corriente 61) representa el producto final. Cabe destacar que las columnas barométricas, (CB1, CB2 y CB3) cumplen con una doble función: la extracción de fluidos, y el mantenimiento del vacío en el sistema mediante un sello hidráulico.

Para garantizar la estabilidad térmica del eyector-condensador (EYC), la corriente recirculada de agua (corriente 56), se enfría en el intercambiador de calor  $HX_{SW}$  antes de regresar como fluido motriz (58). En el intercambiador  $HX_{SW}$ , el calor de condensación es removido por una corriente de agua de mar a 25 °C (52), impulsada por la bomba BAM2. Esta corriente, ahora calentada (53) se conduce al tanque para salmuera (TS) para diluir la salmuera concentrada (corriente 43) proveniente del último efecto del MSF/MED-LTBT, facilitando su manejo antes de su descarga final (54).

La propuesta se distingue por una integración simplificada y eficiente para la cogeneración de agua y frío. Las características principales incluyen: (1) operación con un nuevo rango de temperatura TBT, la cual está por debajo de la temperatura del agua de mar de alimentación; (2) el sistema opera sin la necesidad de energía térmica para su activación, requiriendo únicamente energía eléctrica para el sistema de bombeo; (3) la incorporación de un eyector-condensador barométrico para condensar por incremento de presión un vapor a baja temperatura (4 °C); (4) el uso directo del agua producto como refrigerante en un evaporador, sin expansión previa; (5) una

operación flexible garantizada por una válvula de tres vías para control de la capacidad de enfriamiento y un variador de frecuencia en la bomba del fluido motriz, que permite que el eyector-condensador opera fuera de sus condiciones de diseño. En conjunto, estas características resultan en un sistema con una producción simultánea de agua y frío con baja temperatura de activación (bajo TBT) y sin necesidad de energía térmica para su activación, de mayor capacidad y eficiencia que los sistemas de acoplamientos tradicionales.

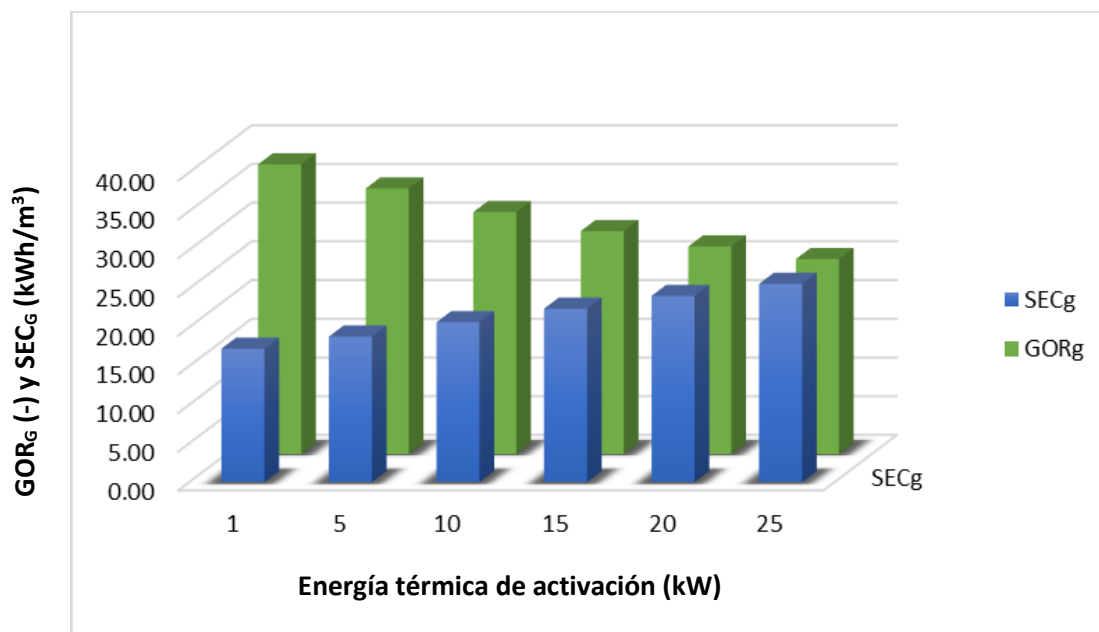
### **3. Metodología**

El estudio operativo para demostrar la factibilidad técnica del sistema de desalinización y enfriamiento de múltiple efecto al alto vacío con baja TBT y eyecto-condensación barométrica (**MSF/MED-LTBT**), se realizó en Aspen Plus. Por medio de un estudio paramétrico, se definieron las condiciones de diseño, que corresponden a un compromiso entre la capacidad y eficiencia del sistema.

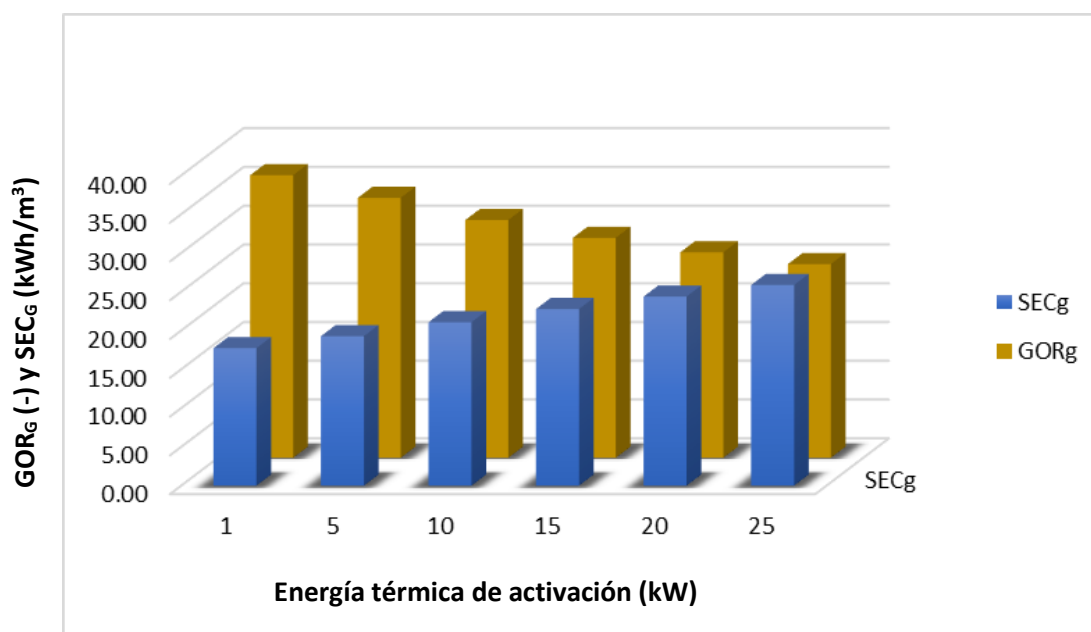
Se presenta el estudio paramétrico del sistema propuesto, donde se muestra el comportamiento operativo interno del sistema y la variación de las variables críticas. La operación del sistema está sujeta a las variaciones de presión, temperatura, calidad y flujo del fluido motriz y fluido secundario, así como a las variaciones de presión de descarga a la salida del difusor.

### **4. Resultados preliminares**

Se presenta el estudio paramétrico de la propuesta tecnológica, donde se muestra el comportamiento operativo interno del sistema y la variación de la capacidad de desalinización e indicadores de eficiencia. La operación del sistema de desalinización MSF/MED-LTBT, está sujeta a las variaciones de flujo, temperatura y concentración del agua de mar de alimentación, así como a las intermitencias del recurso solar utilizado para la activación del generador. Por lo anterior, la evaluación de la viabilidad técnica de la propuesta tecnológica, se realizó estudiando el efecto que tiene la variación del flujo de agua de mar y la cantidad de calor adicionado al sistema, en los principales indicadores de eficiencia del sistema y su capacidad de desalinización.



**Figura 3.** Comportamiento de los indicadores  $GOR_G$  y  $SEC_G$  del sistema MSF/MED-LTBT utilizando una TBT de 25 °C y 10 efectos.



**Figura 4.** Comportamiento de los indicadores  $GOR_G$  y  $SEC_G$  del sistema MSF/MED-LTBT utilizando una TBT de 22 °C y 9 efectos.

**Tabla 1.** Resultados del sistema con TBT de 25 °C y 10 efectos variando el calor de activación.

\*Máximo número de efectos: 10

\* $\Delta T_{\text{efectos}}$ : 2.33

| TBT<br>(°C) | $Q_{GEN}$<br>(kW) | $\dot{m}_{feed}$<br>(kg/s) | $\dot{m}_{ap}$<br>(kg/s) | $W_{B\_total}$<br>(kW) | $Q_{Efectos}$<br>(kW <sub>T</sub> ) | $Q_{Disipador}$<br>(kW <sub>T</sub> ) | $Ex_{Destruida}$<br>(kW) | Entropía $\epsilon$<br>(kJ/kg * K) | $V_1$<br>(kg/s) | $V_{14}$<br>(kg/s) | $X_{Salmuera\_out}$<br>(ppm) | RR<br>(-) | $SEC_G$<br>(kWh/m <sup>3</sup> ) | GOR <sub>Global</sub> |
|-------------|-------------------|----------------------------|--------------------------|------------------------|-------------------------------------|---------------------------------------|--------------------------|------------------------------------|-----------------|--------------------|------------------------------|-----------|----------------------------------|-----------------------|
| 25          | 1                 | 3.5                        | 0.611                    | 38.01                  | -1226.45                            | 282.26                                | 7.90                     | 0.0265                             | 0.001           | 0.122              | 42,405.87                    | 0.17      | 17.73                            | 36.44                 |
|             | 5                 | 3.5                        | 0.627                    | 38.54                  | -1266.72                            | 285.36                                | 8.01                     | 0.0269                             | 0.002           | 0.124              | 42,644.81                    | 0.18      | 19.28                            | 33.52                 |
|             | 10                | 3.5                        | 0.649                    | 39.25                  | -1320.48                            | 289.65                                | 8.17                     | 0.0274                             | 0.004           | 0.127              | 42,968.90                    | 0.19      | 21.08                            | 30.65                 |
|             | 15                | 3.5                        | 0.670                    | 39.94                  | -1372.18                            | 293.79                                | 8.31                     | 0.0278                             | 0.006           | 0.129              | 43,285.39                    | 0.19      | 22.78                            | 28.36                 |
|             | 20                | 3.5                        | 0.691                    | 40.62                  | -1423.85                            | 297.94                                | 8.45                     | 0.0283                             | 0.008           | 0.131              | 43,606.01                    | 0.20      | 24.38                            | 26.50                 |
|             | 25                | 3.5                        | 0.711                    | 41.26                  | -1475.48                            | 301.66                                | 8.59                     | 0.0288                             | 0.010           | 0.133              | 45,627.64                    | 0.20      | 25.87                            | 24.97                 |

**Tabla 2.** Resultados del sistema con TBT de 22 °C y 10 efectos variando el calor de activación.

\*Máximo número de efectos: 10

\* $\Delta T_{\text{efectos}}$ : 2.33

| TBT<br>(°C) | $Q_{GEN}$<br>(kW) | $\dot{m}_{feed}$<br>(kg/s) | $\dot{m}_{ap}$<br>(kg/s) | $W_{B\_total}$<br>(kW) | $Q_{Efectos}$<br>(kW <sub>T</sub> ) | $Q_{Disipador}$<br>(kW <sub>T</sub> ) | $Ex_{Destruida}$<br>(kW) | Entropía $\epsilon$<br>(kJ/kg * K) | $V_1$<br>(kg/s) | $V_{14}$<br>(kg/s) | $X_{Salmuera\_out}$<br>(ppm) | RR<br>(-) | $SEC_G$<br>(kWh/m <sup>3</sup> ) | GOR <sub>Global</sub> |
|-------------|-------------------|----------------------------|--------------------------|------------------------|-------------------------------------|---------------------------------------|--------------------------|------------------------------------|-----------------|--------------------|------------------------------|-----------|----------------------------------|-----------------------|
| 25          | 1                 | 3.5                        | 0.611                    | 38.01                  | -1226.45                            | 282.26                                | 7.90                     | 0.0265                             | 0.001           | 0.122              | 42,405.87                    | 0.17      | 17.73                            | 36.44                 |
|             | 5                 | 3.5                        | 0.627                    | 38.54                  | -1266.72                            | 285.36                                | 8.01                     | 0.0269                             | 0.002           | 0.124              | 42,644.81                    | 0.18      | 19.28                            | 33.52                 |
|             | 10                | 3.5                        | 0.649                    | 39.25                  | -1320.48                            | 289.65                                | 8.17                     | 0.0274                             | 0.004           | 0.127              | 42,968.90                    | 0.19      | 21.08                            | 30.65                 |
|             | 15                | 3.5                        | 0.670                    | 39.94                  | -1372.18                            | 293.79                                | 8.31                     | 0.0278                             | 0.006           | 0.129              | 43,285.39                    | 0.19      | 22.78                            | 28.36                 |
|             | 20                | 3.5                        | 0.691                    | 40.62                  | -1423.85                            | 297.94                                | 8.45                     | 0.0283                             | 0.008           | 0.131              | 43,606.01                    | 0.20      | 24.38                            | 26.50                 |
|             | 25                | 3.5                        | 0.711                    | 41.26                  | -1475.48                            | 301.66                                | 8.59                     | 0.0288                             | 0.010           | 0.133              | 45,627.64                    | 0.20      | 25.87                            | 24.97                 |

**Tabla 3.** Resultados del sistema con TBT de 10, 15, 20, 22, 23, 24 y 25 C sin calor de activación

| TBT<br>(°C) | $Q_{GEN}$<br>(kW) | $\dot{m}_{feed}$<br>(kg/s) | $\dot{m}_{ap}$<br>(kg/s) | $W_{B\_total}$<br>(kW) | $Q_{Efectos}$<br>(kW <sub>T</sub> ) | $Q_{Disipador}$<br>(kW <sub>T</sub> ) | $Ex_{Destruida}$<br>(kW) | Entropía $\epsilon$<br>(kJ/kg * K) | $V_1$<br>(kg/s) | $V_{14}$<br>(kg/s) | $X_{Salmuera\_out}$<br>(ppm) | RR<br>(-) | $SEC_G$<br>(kWh/m <sup>3</sup> ) | GOR <sub>Global</sub> |
|-------------|-------------------|----------------------------|--------------------------|------------------------|-------------------------------------|---------------------------------------|--------------------------|------------------------------------|-----------------|--------------------|------------------------------|-----------|----------------------------------|-----------------------|
| 10          | 0                 | 3.5                        | 0.313                    | 38.04                  | -476.93                             | 310.56                                | 7.88                     | 0.0264                             | 0.087           | 0.122              | 38,432.24                    | 0.09      | 33.80                            | 19.11                 |
| 15          | 0                 | 3.5                        | 0.537                    | 37.90                  | -1040.10                            | 288.29                                | 7.87                     | 0.0264                             | 0.059           | 0.122              | 41,348.33                    | 0.15      | 19.59                            | 32.98                 |
| 20          | 0                 | 3.5                        | 0.601                    | 37.85                  | -1201.59                            | 281.96                                | 7.87                     | 0.0264                             | 0.030           | 0.122              | 42,261.29                    | 0.17      | 17.49                            | 36.95                 |
| 22          | 0                 | 3.5                        | 0.624                    | 37.84                  | -988.29                             | 279.69                                | 7.87                     | 0.0264                             | 0.018           | 0.122              | 42,599.79                    | 0.18      | 16.83                            | 38.38                 |
| 23          | 0                 | 3.5                        | 0.598                    | 37.85                  | -924.96                             | 282.27                                | 7.87                     | 0.0264                             | 0.012           | 0.122              | 42,216.91                    | 0.17      | 17.57                            | 36.76                 |
| 24          | 0                 | 3.5                        | 0.573                    | 37.87                  | -862.08                             | 284.82                                | 7.87                     | 0.0264                             | 0.006           | 0.122              | 41,844.63                    | 0.16      | 18.37                            | 35.16                 |
| 25          | 0                 | 3.5                        | 0.605                    | 37.80                  | -1211.60                            | 280.97                                | 7.86                     | 0.0263                             | 0               | 0.122              | 42,313.91                    | 0.17      | 17.36                            | 37.23                 |

## 5. Conclusiones

El estudio paramétrico realizado en el simulador Aspen Plus permitió evaluar el rendimiento de un sistema de desalinización MED bajo distintas configuraciones de temperatura de ebullición superior (TBT), que se variaron desde los 68 °C hasta los 25°C. Adicionalmente, se investigaron escenarios donde la TBT se redujo aún más, hasta los 10°C, en los que el sistema ya no requiere energía térmica de activación. En esta zona de bajas temperaturas, se observó que el primer efecto del sistema MED se convierte en una etapa de flasheo, donde el vapor generado por este proceso se utiliza para alimentar los múltiples efectos posteriores del sistema.

La mejor condición de diseño identificada en este estudio corresponde a un sistema MSF/MED-LTBT operando al alto vacío con una TBT de 22°C, el cual presenta las siguientes características principales:

- No requiere energía térmica de activación.
- El primer efecto del sistema se convierte en una etapa flash.
- Los múltiples efectos posteriores se activan mediante el vapor generado por el flasheo en la primera etapa flash.
- Utiliza un total de 9 efectos en su configuración.
- Presenta un  $GOR_G$  (relación ganancia de energía) de 38.3 y un  $SEC_G$  (consumo específico de energía global) de 16.8.

Estos resultados demuestran que, mediante el uso de bajas temperaturas y un sistema de eyecto-condensación barométrico, es posible mejorar la eficiencia energética y reducir los costos operativos, lo que abre nuevas oportunidades para optimizar la desalinización a gran escala en entornos con limitados recursos energéticos.