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DESARROLLO DE UN SISTEMA PARA LA DESALACIÓN DE AGUA
DE MAR Y PRODUCCIÓN DE ENERGÍA ELÉCTRICA MEDIANTE
INTEGRACIÓN DE MED Y ORC

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EL SIGUIENTE PROYECTO FUE DESARROLLADO
EN EL CENTRO DE ESTUDIOS DE LAS ENERGÍAS
RENOVABLES.



Este trabajo está dedicado a mi familia *et al*

Así de simple

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RESUMEN

En este trabajo se presenta una novedosa integración energética de un sistema de desalinización térmica por múltiple efecto (MED) y un ciclo Rankine orgánico (ORC) para la producción simultánea de agua potable y electricidad, aprovechando fuentes energéticas de baja temperatura. La energía térmica requerida para el funcionamiento del sistema es suministrada en el evaporador de la MED, mientras que el ORC es activado utilizando parte del calor latente de condensación del vapor de agua producido en el primer efecto de la MED. Con esto, se incrementa la producción de vapor de agua en la primera etapa del sistema de desalinización y, por tanto, la producción de destilado final. Se realizó el simulador y validación de la propuesta utilizando el software Aspen Plus. Se comparó la propuesta con sistemas MED de distintas capacidades y efectos. Se presenta el análisis de la variación de producción de energía eléctrica del ORC y su efecto en el rendimiento y área total de transferencia de calor del sistema MED. El sistema MED/ORC tiene un incremento del 3.95 % en su eficiencia en promedio al aumentar 10 kW la producción eléctrica, con solamente 1.57 % de incremento en el área de transferencia de calor total (MED y ORC). Un sistema MED/ORC con una producción eléctrica de 50 kW es 22 % más eficiente en la desalinización de agua que un sistema MED sin integración, requiriendo solamente 6.9 % más área de transferencia de calor. Los resultados muestran que la integración energética MED/ORC estudiada beneficia a la producción de agua desalinizada final y la eficiencia de la MED, sin incrementar considerablemente el área de transferencia de calor requerida.

Se realizó un estudio comparativo con dos nuevas configuraciones MED/ORC, en donde se aprovecha parte del calor sensible y/o latente de condensación del ORC para potencializar la generación de vapor de agua en el último efecto de la MED, aprovechando los niveles de temperatura presentes. Estas configuraciones son comparadas energéticamente con los sistemas MED y ORC trabajando individualmente y la integración energética en el primer efecto. Los resultados muestran una disminución en los requerimientos de energía térmica de activación del sistema MED con las nuevas integraciones. El aprovechamiento del calor

sensible de la configuración 3 no resultó en una disminución energética considerable en comparación con la configuración 2, debido al poco calor sensible que se aprovecha. El utilizar el calor latente de condensación del ORC implica incrementar su temperatura y presión de condensación, por lo que disminuye su eficiencia y aumenta la energía necesaria para proveer los dos servicios simultáneamente. Sin embargo, se encontró que en dos de las configuraciones estudiadas se disminuyen los requerimientos globales de energía en comparación con los sistemas trabajando de forma individual.

La investigación que se presenta contribuirá en la generación de una nueva tecnología que ayudará a disminuir la problemática de escasez de agua y acceso a energía eléctrica en comunidades aisladas, de una manera más eficiente y empleando recursos renovables para disminuir el impacto en el medio ambiente, propiciando una mejor calidad de vida en las poblaciones marginadas.

ABSTRACT

This work presents a novel energy integration of a multiple effect thermal desalination system (MED) and an organic Rankine cycle (ORC) for the simultaneous production of fresh water and electricity, taking advantage of low temperature energy sources. The thermal energy required for the operation of the system is supplied in the MED evaporator, while the ORC is activated using part of the latent heat of condensation of the water vapor produced in the first effect of the MED. This increases the water vapor production in the first stage of the desalination system and, therefore, the final distillate production. The proposal was simulated and validated using Aspen Plus software. The proposal was compared with MED systems of different capacities and effects. The analysis of the variation of the ORC electrical energy production and its effect on the performance and total heat transfer area of the MED system is presented. The MED/ORC system has a 3.95 % increase in efficiency on average with a 10 kW increase in electrical output, with only a 1.57 % increase in total heat transfer area (MED and ORC). A MED/ORC system with an electrical output of 50 kW is 22 % more efficient in water desalination than a MED system without integration, requiring only 6.9 % more heat transfer area. The results show that the MED/ORC energy integration studied benefits the final desalinated water production and the efficiency of the MED, without considerably increasing the required heat transfer area. A comparative study was carried out with two MED/ORC variant configurations, where part of the sensible and/or latent heat of condensation of the ORC is used to boost the generation of water vapor in the last effect of the MED, taking advantage of the present temperature levels. These configurations are compared energetically with the MED and ORC systems working individually and the energy integration in the first effect. The results show a decrease in the activation thermal energy requirements of the MED system with the new integrations. Utilizing sensible heat from configuration 3 did not result in a considerable energy decrease compared to configuration 2, due to little sensible heat being utilized. Utilizing the latent heat of condensation of the ORC implies increasing its condensing temperature and pressure, thus decreasing its efficiency and

increasing the energy required to provide the two services simultaneously. However, it was found that in two of the configurations studied, the overall energy requirements are reduced compared to the systems working individually. The research presented will contribute to the generation of a new technology that will help reduce the problem of water scarcity and access to electricity in isolated communities, in a more efficient way and using renewable resources to reduce the impact on the environment, promoting a better quality of life in marginalized populations.

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Introducción al proyecto

El conocimiento es valioso en sí, pero es invaluable cuando beneficia a la sociedad.

Anónimo

1.1. Problemática y justificación

En el mundo existen dos escenarios respecto al acceso a la energía eléctrica. Por un lado, están los países desarrollados, principalmente potencias europeas y países norteamericanos, que gozan de un acceso a la energía eléctrica para el 100 % de su población desde hace décadas. Por el otro lado se encuentran los países subdesarrollados con serios problemas para cubrir con la totalidad de sus necesidades eléctricas. Mientras que el promedio mundial de población con acceso a la electricidad está por el rango del 88 %, el promedio de la región sub-Sahara, en África, ronda el 45 % (The World Bank, 2017). Es clara la diferencia de las dos caras de la moneda. De los más de 780 millones de personas en el mundo que no cuentan con electricidad, la gran mayoría se encuentran en zonas rurales, alejadas de los centros de consumo, donde por cuestiones técnicas, geográficas y/o económicas, no es atractivo llevar electricidad a esas zonas. Por lo anterior, es obligación del sector público, privado y académico proponer y dar soluciones a esta problemática, teniendo siempre en mente un desarrollo sustentable.

1.1.1. Situación energética mundial

En el año 2019, se produjeron en el mundo poco más de 27,044 TWh de electricidad por medio de distintas fuentes de energía, como se aprecia en la Fig. 1.1. La electricidad aportada por las plantas de carbón correspondió al 37 % del total anual, 9,914 TWh. Seguido del carbón, el gas natural se situó como la segunda mayor fuente para la generación de energía eléctrica con 6,345 TWh. En conjunto, estos dos combustibles aportaron el 60 % de la electricidad de todo el mundo durante el 2019 (IEA, 2021c). Las energías renovables, en su conjunto, aportaron un total de 7,085 TWh correspondiente al 26 % de la generación eléctrica mundial. Cabe mencionar que el 49 % de ese carbón se quemó solamente en China para la generación eléctrica. La tendencia actual nos dice que el uso de los combustibles fósiles va a la baja, mientras que la aportación de fuentes renovables se está incrementado. En el 2015 se tenía una generación eléctrica del 62 % por medio del carbón y gas natural, mientras que las renovables aportaban el 25.5 % del total mundial (IEA, 2021c). Sin embargo, la diferencia sigue siendo abrumadora y nos sigue mostrando la gran dependencia energética mundial a los combustibles fósiles y contaminantes.

Las energías renovables han estado en constante desarrollo y crecimiento prácticamente en todo el mundo, la capacidad instalada pasó de 1,330 GW a 2,800 GW en el periodo de 2011 al 2020, un incremento de más del 100 %, como se puede apreciar en la Fig. 1.2. La capacidad actual (2021) corresponde al 37 % de la capacidad de generación eléctrica total mundial. A pesar de tener más de un tercio de la potencia eléctrica mundial en fuentes renovables de energía, la aportación a la generación eléctrica es de aproximadamente un cuarto del total, lo anterior debido a las intermitencias por las cuales se ven limitadas algunas tecnologías por la variabilidad del recurso, como la radiación solar, disponibilidad de viento y variación de las lluvias, por mencionar algunas. El crecimiento de las energías renovables es claro, un 2 % promedio anual permite vislumbrar un futuro basado en fuentes limpias de energía, pero, para contrarrestar los impactos ambientales actuales y frenar el calentamiento global, se debe de acelerar su implementación masiva.

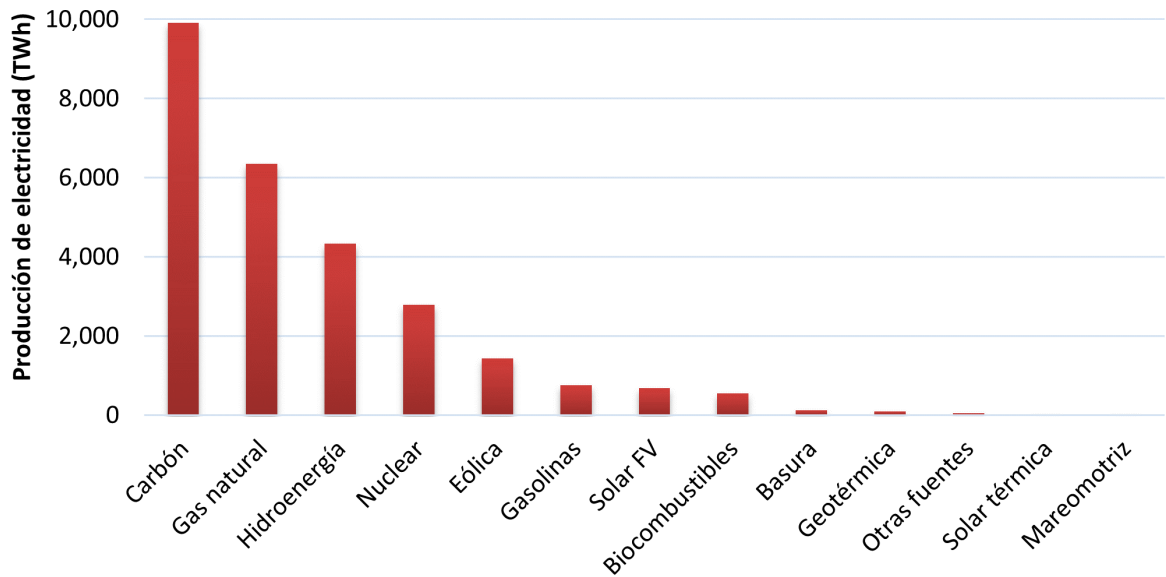


Figura 1.1: Producción de electricidad a partir de distintas fuentes en el mundo durante el 2019 (IEA, 2019).

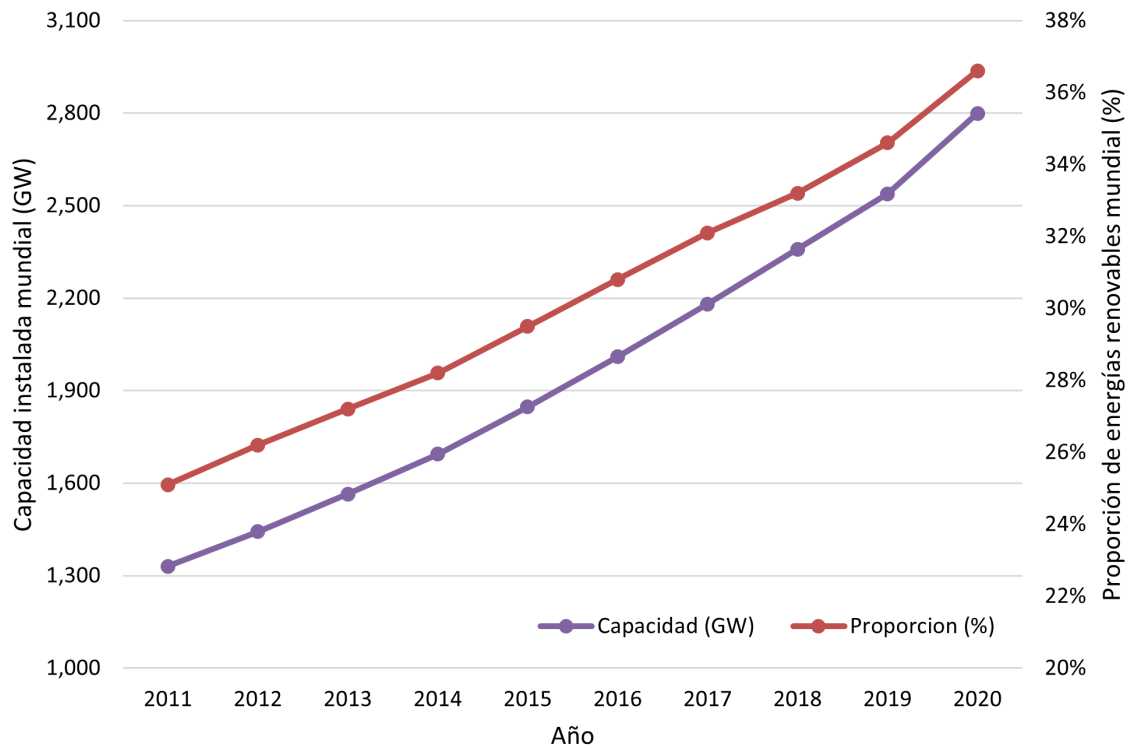


Figura 1.2: Incremento en la capacidad instalada y proporción de energías renovables en el mundo (IRENA, 2021a).

1.1.2. La energía en México

La situación energética en México no difiere en gran medida con el escenario mundial. En el 2020 en México se tenía una capacidad total instalada para generación eléctrica de 87.97 GW, de los cuales 28.35 GW corresponden a fuentes renovables de energía, es decir, el 32 % del total nacional (ver Fig. 1.3). Es de resaltar la aportación que tiene la energía hidráulica en el país, con un 14 % de la potencia total instalada, seguida de la energía eólica con el 9 % y la solar con 7 %. Con una aportación de solamente el 2 % a la matriz energética de México se encuentra la bioenergía y la geotérmica. Hace no mucho tiempo México era un país líder en la generación de energía geotérmica, pero en los últimos años no se han instalado plantas geotermoeléctricas nuevas y las actuales han presentado una notable disminución en su generación. Solo en el 2020 se redujo en un 3 % la capacidad geotérmica instalada en el país, al pasar de 936 MW a 906 MW (IRENA, 2021b). Sin embargo, hay que denotar el constante crecimiento de la potencia renovable en México durante los últimos 10 años. Se paso de una capacidad de energía renovable de 13.48 GW en el 2011 a 28.35 GW en el 2020, un aumento del 110 % con una tasa de incremento promedio del 7.8 % anual. La energía solar y eólica son las que más incrementaron durante ese periodo, al pasar de 39 MW a 5,644 MW y 601 MW a 8,128 MW, respectivamente, equivalente a incrementos anuales promedio de 38.5 y 84.3 %, respectivamente.

Es importante señalar la reducción de la capacidad de fuentes convencionales de energía, al reducirse un 3.5 %, equivalente a 2,177 MW menos en el 2020. Lo anterior es debido a que se han estado cerrando paulatinamente plantas de generación eléctrica basadas en la combustión del carbón y gasolinas.

La generación de electricidad en México contrasta en gran medida con la distribución de la matriz energética. En la Fig. 1.4 se aprecian los distintos generadores en nuestro país y la electricidad que se produjo durante el 2019 y 2020. Aquí se comprueba de forma más clara la gran dependencia energética a los combustibles fósiles, en especial al gas natural y gasolinas. A pesar de que, como se mencionó anteriormente, el porcentaje de energías renovables en la matriz energética nacional en el 2020 fue del 32 %, la aportación en la generación de electricidad fue de solo el 19.7 %. Se produjeron 217 GWh de electricidad por medio del

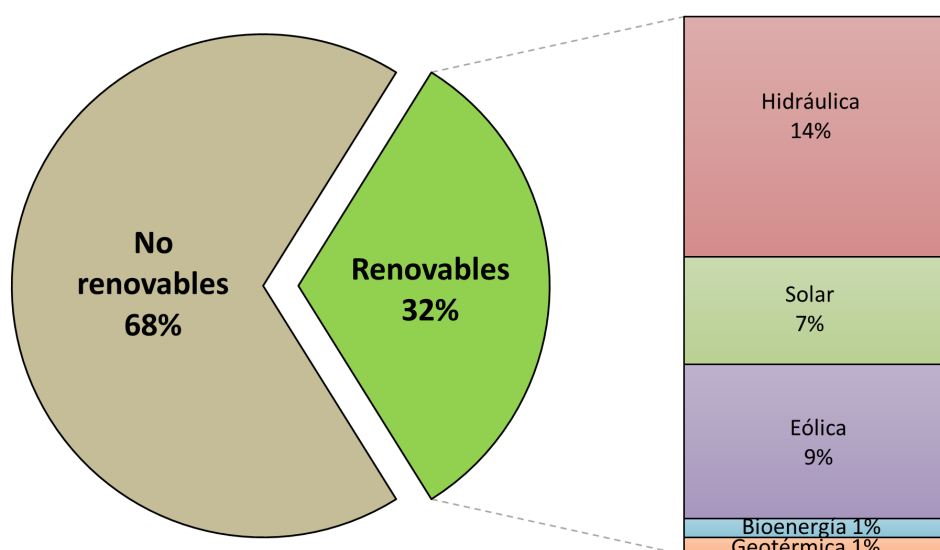


Figura 1.3: Porcentaje de capacidad instalada de fuentes renovables de energía en México hasta el 2020 (IRENA, 2021b).

gas natural en el 2020, que corresponde al 64.5 % de la electricidad total, un incremento del 12.7 % respecto a la producción del mismo generador durante el año anterior. La reducción de 28.1 GWh eléctricos generados con gasolinas y carbón en el 2020 se vieron opacados con el incremento de 24.5 GWh con gas natural. Incluso ese incremento es mayor que la generación total de energía eólica y geotérmica en conjunto.

El depender en más de la mitad de la producción de electricidad total del país en un solo energético provoca dependencia e inestabilidad en el costo de la energía. Además, considerando que la producción de gas natural nacional disminuyó un 26 % desde el 2012 a la actualidad y las importaciones de este se incrementaron un 180 %, siendo nuestro principal suministrador Estados Unidos, nos convierte en un país vulnerable energéticamente. En el 2020 se importó un 93 % más gas natural que el que se produjo en el país (IEA, 2021b). Lo anterior resulta después de apostar mayormente en grandes plantas de generación eléctrica basadas en ciclos combinados, muy populares en la actualidad debido a su alta eficiencia y por qué utilizan gas natural como combustible.

Durante las dos últimas décadas en México se ha optado por incrementar de forma constante la capacidad de generación eléctrica basada en la combustión del gas natural, como

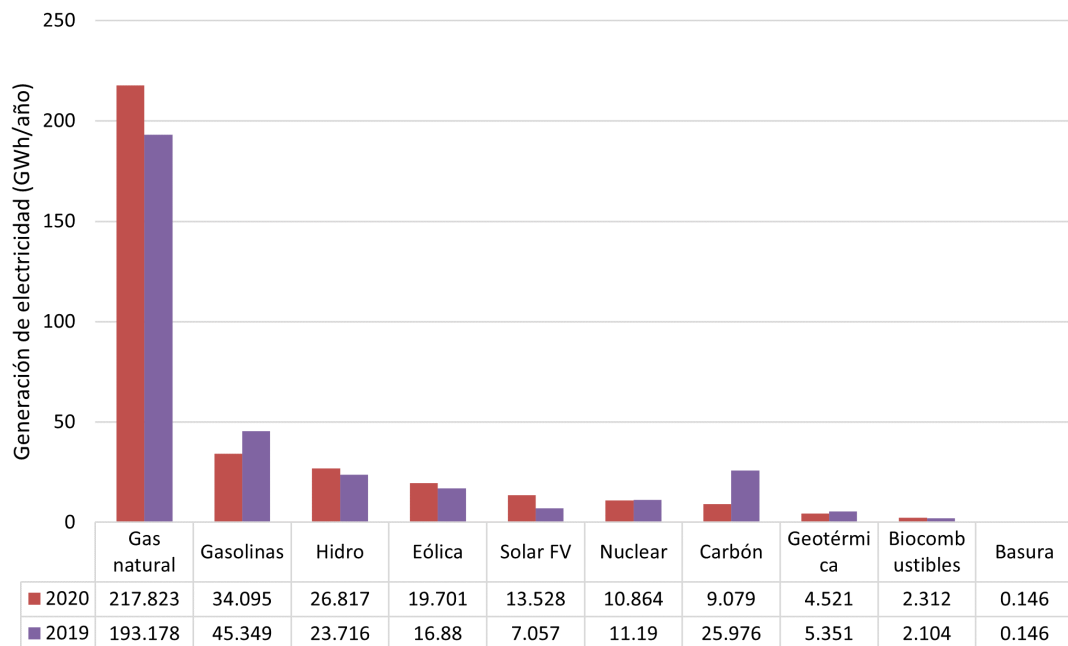


Figura 1.4: Producción de electricidad de distintas fuentes en México durante el 2019 y 2020 (IEA, 2019).

se aprecia en la Fig. 1.5. El costo de este combustible a tenido un promedio durante el 2020 de \$2.05 USD/MBTUs (dólares por millón de British Thermal Units) con mínimos de \$1.6 USD/MBTUs, relativamente barato en comparación con otros combustibles más contaminantes. Sin embargo, la alta inestabilidad en su costo provoca una gran incertidumbre entre los consumidores, ya que se han tenido realces en su costo de hasta \$17 USD/MBTUs durante el 2005, y un poco menores en el 2008. A finales de 2021 el precio del gas natural ronda los \$5.51 USD/MBTUs (Administration, 2021). El consumidor final es el que termina pagando esta inestabilidad energética debido al constante crecimiento de nuestra dependencia al gas natural. Una matriz energética diversificada, combinando energías renovables y otras fuentes limpias, permite estabilidad en los precios, soberanía energética y un desarrollo sustentable.

1.1.3. Problemas energéticos y sus impactos socioambientales

Desde un aspecto socioambiental, son diversas las problemáticas que conllevan la fuerte penetración de combustibles fósiles en el sector energético mundial. En las últimas tres

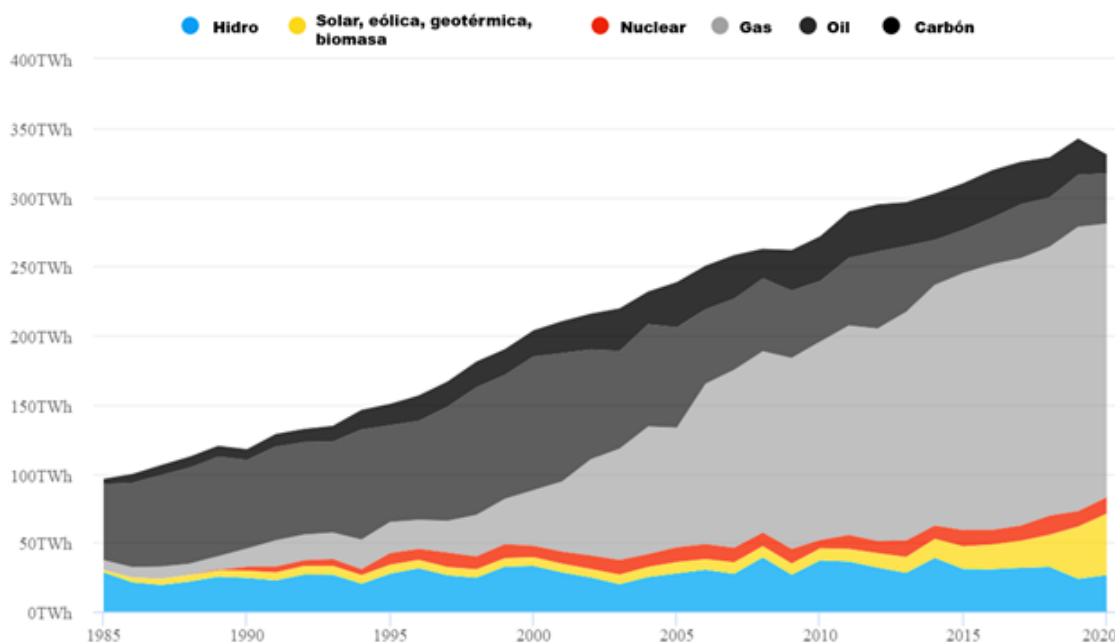


Figura 1.5: Producción de electricidad en México acumulada a partir de distintas fuentes en el periodo de 1985-2020. Tomada de Gabbatiss (2021).

décadas no ha parado de incrementar año con año las emisiones de dióxido de carbono (CO₂) debido a la generación de energía eléctrica (ver Fig. 1.6). La contaminación ambiental provocada por la quema de carbón es la principal, seguida por las gasolinas y por último el gas natural. Dentro de los distintos sectores, las emisiones de CO₂ por la producción de electricidad son las de mayor cantidad, superando al sector transporte e industrial. De las 33,621 Mt CO₂ (millones de toneladas de dióxido de carbono) emitidas por todos los sectores durante el 2019, el sector eléctrico se llevó el 41.8 % del total de emisiones de gases contaminantes. Solamente China generó por sí solo en el 2019 el 29.5 % de las emisiones de CO₂ mundiales (IEA, 2021a). Según la Organización Mundial de la Salud, en el 2016 alrededor de 249,000 muertes prematuras fueron atribuibles a la contaminación del aire exterior, por lo que es considerada el principal riesgo ambiental para la salud pública en el continente americano (de la Salud, 2016). Transitar hacia un esquema energético basado en tecnologías de generación renovables y limpias, permitiría una reducción considerable en las emisiones globales de gases contaminantes, reduciendo problemáticas en la salud, como lo son enfermedades respiratorias, así como mitigar los efectos del calentamiento global.

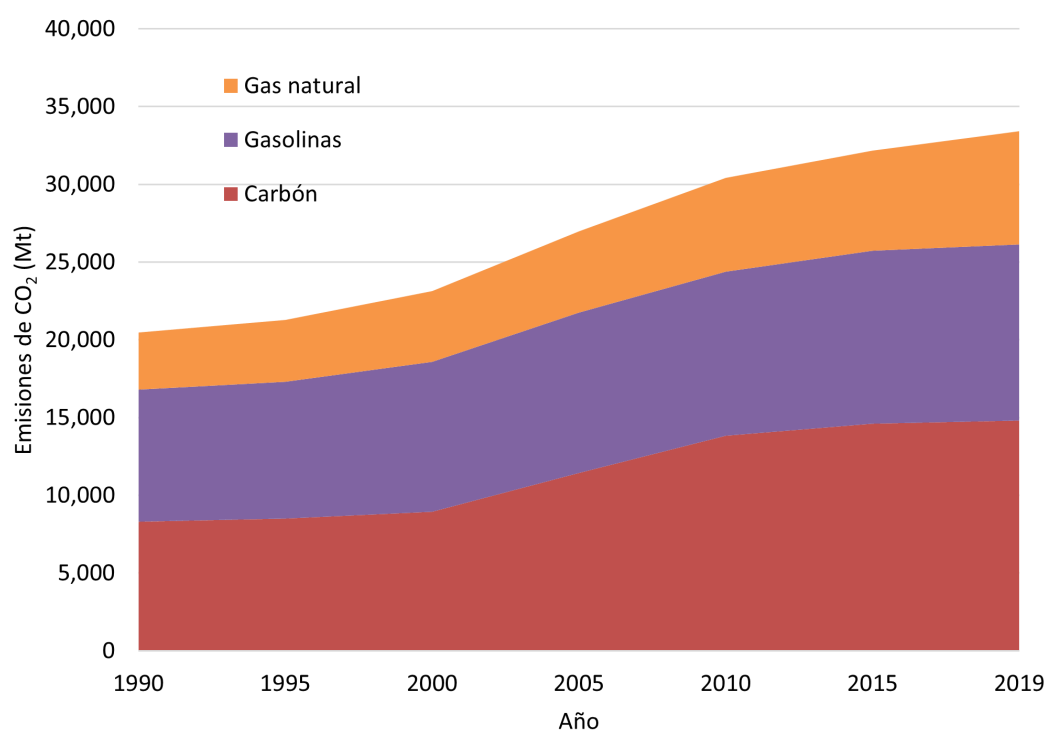


Figura 1.6: Emisiones de CO₂ en el mundo a partir de distintas fuentes energéticas durante 1990-2019 (IEA, 2021a).

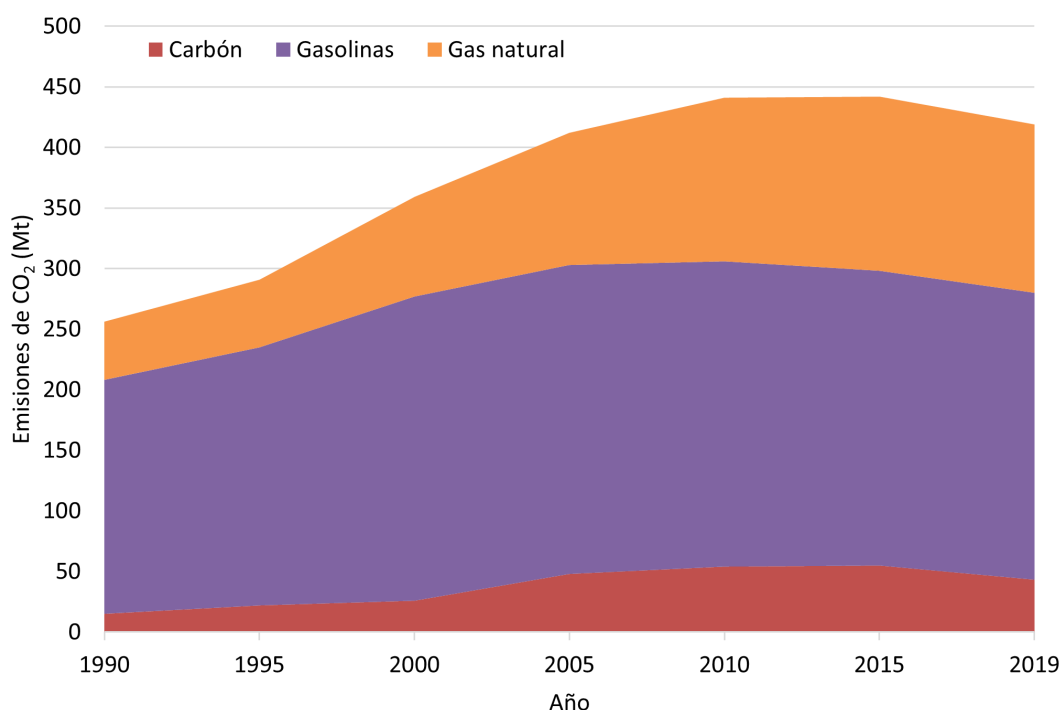


Figura 1.7: Emisiones de CO₂ en México a partir de distintas fuentes durante 1990-2019 (IEA, 2021a).

Por su parte, México aportó el 1.2 % de las emisiones totales de CO₂ mundiales durante el 2019, un porcentaje relativamente bajo en comparación con Estados Unidos y su generación de 4,743 Mt CO₂, correspondientes al 14.1 % del total de emisiones en el 2019. Un dato interesante son las emisiones de CO₂ per cápita ya que, mientras que el promedio mundial es de 4.4 t CO₂/per cápita, México cuenta con emisiones de 3 t CO₂/per cápita, China 7.1 t CO₂/per cápita y Estados Unidos de 13.6 t CO₂/per cápita (IEA, 2021a). La tendencia de generación de emisiones en México tiende a la baja desde el 2015, como se puede apreciar en la Fig. 1.7, en gran medida debido a la sustitución de fuentes de generación de electricidad basadas en gasolinas y carbón, por otras fuentes más limpias o con menos emisiones de CO₂, como lo es el gas natural y las energías renovables.

No por el hecho de que México sea uno de los países con menor aportación a la emisión de gases contaminantes CO₂ al ambiente significa que nos debemos de quedar con los brazos cruzados y permitir la contaminación ambiental. Estudios recientes en México encontraron asociación entre la contaminación atmosférica en ciudades mexicanas y mortalidad por

Covid-19 (Ángel Arturo Cabrera-Cano y cols., 2021) por lo que mantener niveles bajos de contaminación ambiental debe ser tema de interés nacional y salud pública. El sector transporte y la electricidad son los sectores que más contaminan en México con 147 y 137 Mt CO₂ emitidas, respectivamente. Considerando que en el 2019 se generó el 58 % de la electricidad con gas natural en México, sustituyendo en su totalidad esta fuente de energía por energías renovables, se disminuiría el 33 % del CO₂ emitido en México, cumpliendo con los compromisos en cuanto a reducción de gases contaminantes. Sin embargo, el principal generador de gases contaminantes es el sector transporte por la combustión de gasolinas. La transición energética debe de considerar a este sector como primordial, empleando nuevas tecnologías y combustibles limpios como el biodiesel y bioetanol.

1.1.4. Problemática Agua-Energía

Los servicios de energía eléctrica y agua para uso humano son necesarios para contar con una buena calidad de vida de la población, basada en una adecuada salud e higiene, educación, actividades productivas y entretenimiento, por mencionar algunos factores que determinan el nivel de bienestar de las personas. Estos recursos son primordiales para el desarrollo de cualquier sociedad y es obligación de instituciones públicas proveerlos de una manera equitativa y sustentable. Sin embargo, el acceso a estos servicios sigue siendo inalcanzable para algunas poblaciones, principalmente aquellas que se ubican en regiones aisladas, donde se complica técnica y/o económicamente instalar y mantener tecnologías para su producción.

Por otro lado, la cantidad de agua que se tiene disponible en el planeta no varía, solamente el lugar, la forma y la calidad en que se encuentra sí es variable (FEA, 2006). El cambio climático ha provocado que las sequías sean más recurrentes en regiones donde no se presentaban tan cotidianamente y que, con el aumento exponencial de la población, la falta de agua sea una problemática de gran interés para todo el mundo. El 40 % de la población mundial sufre de escasez de agua y alrededor de 1,800 millones de personas utilizan una fuente de agua potable que está fecalmente contaminada (United Nations, 2018). Se prevé que en 2030 el mundo tendrá que enfrentarse a un déficit mundial del 40 % de agua en un escenario climático en que todo sigue igual (2030 WRG, 2013). En México el 92 % de

la población cuenta con acceso al servicio del agua potable. Sin embargo, la distribución geográfica del agua no coincide con la distribución geográfica de la población. Existe un volumen de 3,687 metros cúbicos de agua per cápita por año y se espera que disminuya a 3,279 para el año 2030, pero se presenta una gran diferencia entre las regiones sureste y noreste del país (CONAGUA, 2017), con una disponibilidad del recurso completamente distinta.

Asimismo, cuando se producen estos servicios se hace de una manera insostenible, contaminando el medio ambiente debido al uso de combustibles fósiles para el funcionamiento de los equipos tradicionales. Queda claro que la lejanía de las poblaciones marginadas con las urbanidades que cuentan con grandes centros de producción de electricidad y consumo de agua es el principal problema para el acceso a estos servicios, agregando también los estragos del cambio climático cambiando los patrones de disponibilidad del agua. Es necesario desarrollar tecnologías eficientes y adaptables a este tipo de regiones aisladas, basadas en el aprovechamiento de las energías renovables, buscando combatir dichas problemáticas y propiciar una mejora en la calidad de vida de las personas, teniendo siempre en mente la ruta del desarrollo sustentable.

La investigación que se presenta contribuirá en la generación de una nueva tecnología que ayudará a disminuir la problemática de escasez de agua y acceso a energía eléctrica en comunidades aisladas, de una manera más eficiente y empleando recursos renovables para disminuir el impacto en el medio ambiente, propiciando una mejor calidad de vida en las poblaciones marginadas.

1.2. Antecedentes

La producción simultánea de energía eléctrica y desalinización de agua es un área que ha tomado principal interés entre la comunidad científica mundial, debido a la importancia de contar con estos servicios y las distintas problemáticas ambientales, energéticas y económicas, entre otras, que conlleva su producción y distribución. La escasez de agua es un problema global que ha llamado la atención entre las instituciones públicas y privadas. Las Naciones Unidas incorpora esta problemática en el Objetivo de Desarrollo Sostenible (ODS) 6, mencionando que este recurso, en la suficiente cantidad y calidad, es esencial para todos los aspectos de la vida (Nations, 2018). Por otro lado, el ODS llama a asegurar un acceso universal al servicio de energía seguro, asequible y moderno. Se está haciendo un esfuerzo global para lograrlo, pero se está lejos de cumplir la meta para el 2030 a la velocidad actual (IEA y cols., 2019). Por lo anterior, se deben de proponer nuevas tecnologías más eficientes y económicas, buscando un desarrollo sustentable en la producción de energía eléctrica y desalinización de agua.

La hibridación de tecnologías de desalinización y producción de energía eléctrica es una opción eficientemente atractiva, ya que se pueden aprovechar efluentes de unos equipos para disminuir los requerimientos de energía de otro, o también se pueden realizar novedosas integraciones energéticas en las que se benefician ambos sistemas (Omar y cols., 2020). Existen una gran cantidad de tecnologías para producir dichos servicios, el reto se encuentra en determinar las mejores condiciones de hibridación para obtener la mejor relación costo-beneficio. Estos acoplamientos pueden ser entre sistemas activados tanto térmica como eléctricamente (Delgado-Torres y cols., 2019). Los sistemas de desalinización por ósmosis inversa dominan el mercado mundial (Goh y cols., 2018), debido principalmente al avance de la tecnología de membranas y la simplicidad del proceso de desalinización (Qasim y cols., 2018). Utilizan energía eléctrica para su activación, por lo que pueden ser acoplados a tecnologías eléctricas convencionales, incluyendo la energía fotovoltaica. Dentro de las tecnologías de desalinización activadas térmicamente se encuentran la múltiple etapa flash (MSF), destilación por múltiple efecto (MED) y compresión de vapor (Ullah y Rasul, 2019).

A pesar de que los sistemas MSF predominan en el mercado (IDA, 2006), la tecnología MED ha mejorado y en la actualidad presenta eficiencias mayores que los sistemas MSF (Zheng y Zheng, 2017).

Se han realizado distintos acoplamientos entre sistemas MED y ciclos termodinámicos para la producción de energía eléctrica, en donde destacan las integraciones energéticas entre ciclos Rankine y Rankine orgánicos (ORC) (Askari y Ameri (2018), Omar y cols. (2020), Valenzuela y cols. (2017)). Palenzuela y cols. (2011) estudiaron distintas configuraciones para realizar una eficiente integración entre un ciclo Rankine solar con una MED o un sistema de ósmosis inversa en regiones áridas, como el Medio Oriente y el Norte de África. Sus resultados muestran que la integración de un sistema MED de baja temperatura a un ciclo Rankine, para reemplazar el condensador del vapor de salida de la turbina, es más eficiente termodinámicamente que acoplar el mismo ciclo a una planta de osmosis inversa, lo que también resulta en un área de captación solar menor. En otro estudio similar, Mata-Torres y cols. (2017) analizaron dos formas de integrar un ciclo de potencia con un sistema MED, para la producción simultánea de agua desalinizada y energía eléctrica bajo condiciones climatológicas de Venezuela y Chile. En la primera integración, la MED sustituía en su totalidad el condensador del ciclo de potencia, mientras que en la segunda se extraían una parte del vapor de la turbina para el proceso de desalinización, ambas configuraciones propuestas anteriormente en Palenzuela y cols. (2011). Askari y Ameri (2018) estudiaron el acoplamiento de un ciclo Rankine activado con energía solar en donde aprovechaban el calor de condensación del ciclo de potencia para la desalinización de agua por medio de un sistema MED. Sus resultados muestran que, debido a que en lugar del condensador se integró la MED, las condiciones de operación del ciclo Rankine solar tuvieron que modificarse para contar con una temperatura de condensación de 70 °C y, así, poder activar a la MED. Esto provocó una disminución de la eficiencia térmica del ciclo Rankine de 37 % a 31 %. En cuanto al acoplamiento de ciclos orgánicos con sistemas MED, Baccioli y cols. (2018) estudiaron ambos sistemas con dos distintas integraciones. La primera, una hibridación serie-cascada, utiliza una fuente energética de calor de desecho para activar, en una primera instancia, al ORC y después la MED; también una parte del calor de condensación del ORC es utilizado

como precalentamiento del agua de mar de la MED antes de entrar al primer efecto. En la segunda, se estudió una configuración cascada en donde el calor de desecho es suministrado únicamente al ORC, mientras que el sistema MED aprovecha el calor de condensación para su operación. Para una disponibilidad de energía fija de 20 MW, concluyeron que la configuración en serie incrementa el desempeño termodinámico y económico de la MED, permitiendo una reducción en el costo del agua producida. Por su parte, Zhou y cols. (2020) propusieron un sistema para la producción de frío, desalinización y energía eléctrica, mediante la integración de un ciclo de eyector enfriamiento, MED y ciclo Rankine, que utiliza el gradiente térmico del agua de mar y colectores solares térmicos para proveer los servicios de calentamiento y enfriamiento del sistema. Una parte del vapor que ingresa a la turbina es utilizado como fuente térmica de activación de la MED antes de terminar la etapa de expansión, mientras que el resto del fluido de trabajo a la salida de la turbina es el fluido motriz en el eyector del ciclo de enfriamiento. Lograron eficiencias exergéticas de hasta 29.3 % en el sistema global debido a las integraciones.

En un trabajo de revisión, Omar y cols. (2020) analizaron los distintos estudios de hibridaciones entre sistemas de desalinización y concentración solar de potencia disponibles en la literatura. Concluyeron que este tipo de integraciones ofrecen un complemento beneficioso substancial para generar energía y agua sustentablemente, pero aún es necesario más tiempo para un buen desarrollo y esparcimiento de estas tecnologías híbridas. La ventaja de utilizar sistemas híbridos ORC/MED para proveer los servicios de energía eléctrica y agua desalada es que son tecnologías escalables a capacidades bajas y, siendo activados con energías renovables, pueden ser utilizados en aplicaciones fuera de red para llegar a las comunidades rurales más necesitadas (Aberilla y cols. (2020), Aguilar-Jiménez y cols. (2018), López-Zavala y cols. (2019)). Se han realizado grandes esfuerzos para la producción de energía eléctrica y agua potable en baja capacidad (menor a 100 m³/d) en comunidades rurales de países en desarrollo, en donde se han construido distintos sistemas de baja eficiencia basados en la conversión de la energía solar (Pouyfaucou y García-Rodríguez, 2018). Astolfi (2017) investigaron el uso de tecnologías para la desalinización de agua utilizando energía solar aplicado a una pequeña isla, bajo las distintas dinámicas de

demandas de agua y electricidad, así como también contemplando los pronósticos del clima. Concluyeron que una combinación de tecnologías fotovoltaicas/osmosis inversa y térmica/ORC/MED es la mejor opción para extender la operación durante un mayor tiempo, aprovechando las ventajas técnicas y económicas de ambos sistemas. Maraver y cols. (2012) estudiaron un sistema poligenerativo en donde se utilizaba un ORC de alta temperatura y activaban una MED con el calor residual. Analizaron la influencia de distintos fluidos de trabajo dentro del ORC y concluyeron que utilizar el calor de condensación del ORC en la MED limita severamente la eficiencia de conversión eléctrica. Mouaky y Rachek (2020) estudiaron la aplicación de un sistema poligenerativo solar/biomasa para proveer los servicios de electricidad, agua potable, agua caliente doméstica y calentamiento/enfriamiento de espacios, de una comunidad rural de 40 hogares en Marruecos. Sus resultados muestran que el sistema propuesto puede satisfacer los servicios requeridos, pero solo un 62 % de la demanda de enfriamiento de espacios es cubierta, con un consumo aproximado de 320 toneladas de biomasa con una contribución solar promedio del 20 %.

Los trabajos anteriormente descritos comprueban que existe un gran interés y potencial de mejoramiento entre las tecnologías MED/ORC, especialmente en la forma en que estas pueden ser integradas energéticamente para favorecer su eficiencia de conversión. Las investigaciones demuestran que la factibilidad técnica de estos acoplamientos está directamente influenciada por las características operativas de ambos sistemas, ya que en la gran mayoría de los resultados mencionan que, al mejorar las condiciones de operación de un sistema, este termina por afectar al otro, por lo que se debe de buscar un acoplamiento óptimo entre ambas tecnologías, que permita una mejora en general.

En este trabajo se proponen nuevas integraciones energéticas entre sistemas MED y ORC, para potencializar la desalinización de agua y la eficiencia global del sistema. Estas novedosas integraciones aprovechan el calor latente y sensible de ambos sistemas para beneficiarse energéticamente. Se presenta una descripción de las propuestas de integración, el desarrollo de los simuladores y su validación, para posteriormente realizar un estudio operativo y comparativo.

1.3. Hipótesis

El aprovechamiento del calor latente de condensación en el proceso de desalación por múltiple efecto puede ser aprovechado para la activación de un ciclo Rankine orgánico, incrementando la producción de agua desalada y reduciendo los servicios externos del ciclo de potencia.

1.4. Objetivos

1.4.1. Objetivo general

Desarrollar un sistema para la desalación de agua de mar y producción de energía eléctrica para una comunidad aislada de 20 familias en Puertecitos, Baja California, México, mediante la integración energética interna de una destiladora por múltiple efecto y un ciclo Rankine orgánico, siendo activado con energía solar térmica y convirtiéndolo en una unidad demostrativa que permita la transferencia tecnológica

1.4.2. Objetivos específicos

- Analizar la factibilidad técnica de integración de un ciclo Rankine orgánico a una destiladora por múltiple efecto para el aprovechamiento del calor latente de condensación de las distintas etapas de desalación mediante simulación
- Determinar las condiciones óptimas de operación del sistema utilizando como fuente de activación energía solar térmica u otra energía limpia.
- Comparar el sistema propuesto contra otros equipos presentes en el estado de la técnica.
- Conceptuar la ingeniería básica y de detalle del sistema para la construcción de un prototipo experimental, comprobación del fenómeno físico y validación de resultados teóricos.

- Analizar técnica y económicamente la factibilidad de escalamiento de prototipo experimental para su instalación en microrred de una comunidad aislada

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Descripción del contenido de los artículos

La felicidad no está en la chequera, está en el privilegio de servir a quien lo necesite.

José Mujica

Las propuestas tecnológicas, métodos de evaluación operativa, resultados y conclusiones del proyecto son presentados en los artículos contenidos en los anexos de esta disertación. En las siguientes secciones se presenta un resumen de las contribuciones más importantes de esta investigación.

2.1. ARTÍCULO: Low-temperature multiple-effect desalination/organic Rankine cycle system with a novel integration for fresh water and electrical energy production

Revista: *Desalination*, Vol. 477 (2020) 114269 DOI:<https://doi.org/10.1016/j.desal.2019.114269>
FI: 9.5

En este artículo se presenta un estudio operativo de una novedosa integración entre un ORC y un sistema de desalinización de agua de mar tipo MED. Se propuso suministrar el calor de activación del ORC en el generador de vapor en conjunto con el calor de la MED. De esta

forma se produce una mayor cantidad de vapor en el primer efecto de la MED, mientras que parte del calor latente de condensación del mismo es utilizado para la activación del ORC.

Al llevar a cabo este estudio se encontró que:

- La integración energética propuesta beneficia la producción de agua desalinizada y la eficiencia de la MED sin un incremento significativo en el área total de transferencia de calor.
- El incremento en la producción de agua para una capacidad del ORC de 20, 30, 40 y 50 kWe es de 8.8, 13, 17.6 y 22 %, respectivamente.
- El sistema MED/ORC propuesto presenta en promedio un incremento del PR del 3.9 % cuando se incrementa la producción eléctrica en 10 kW, resultando en solamente un incremento del 1.6 % en el área total de transferencia de calor. Un sistema integrado MED/ORC con una producción eléctrica de 50 kW es 22 % más eficiente en la desalinización de agua que un sistema MED sin la integración del ORC, requiriendo solamente un 6.9 % más de área de transferencia de calor.
- Entre mayor sea la integración MED/ORC, menor será el área específica y mayor el PR de la MED, debido al incremento de la producción de vapor de agua en el primer efecto.

2.2. ARTÍCULO: Comparison of novel energy integrations between MED and ORC systems

Revista: *Primera revisión realizada en International Journal of Energy Research.*

En este artículo se presenta un estudio operativo de una novedosa integración entre un ORC y un sistema de desalinización de agua de mar tipo MED. Se propuso suministrar el calor de activación del ORC en el generador de vapor en conjunto con el calor de la MED. De esta forma se produce una mayor cantidad de vapor en el primer efecto de la MED, mientras que

parte del calor latente de condensación del mismo es utilizado para la activación del ORC.

Al llevar a cabo este estudio se encontró que:

- La integración energética propuesta beneficia la producción de agua desalinizada y la eficiencia de la MED sin un incremento significativo en el área total de transferencia de calor.
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- Entre mayor sea la integración MED/ORC, menor será el área específica y mayor el PR de la MED, debido al incremento de la producción de vapor de agua en el primer efecto.

2.3. ARTÍCULO: Potential for Thermal Water Desalination Using Microgrid and Solar Thermal Field Energy Surpluses in an Isolated Community

Revista: *Communications in Computer and Information Science*, 1152, DOI: https://doi.org/10.1007/978-3-030-38889-8_13

En este artículo se presentó el estudio de un sistema híbrido para la desalinización de agua de mar mediante el aprovechamiento energético de un campo de colectores solares térmicos y una microrred eléctrica basada en energías renovables. Un sistema MED para desalinización

de agua aprovecha la energía solar térmica generada por los colectores y la energía eléctrica de sobra de la microrred (conversión eléctrica-térmica). Se estudió el sistema bajo el caso de estudio de la comunidad aislada de Puertecitos, Baja California, México, donde se cuenta con una microrred para satisfacer las necesidades eléctricas de la población, y un campo de colectores solares de tubos evacuados para activar un enfriador de agua por absorción en la escuela primaria de la comunidad. El periodo de estudio comprendió la temporada de invierno, cuando existe un gran desperdicio de energía eléctrica de la microrred y no se utiliza el enfriador de agua de la escuela, por lo que se tiene una gran disponibilidad energética para desalinización de agua, un servicio muy requerido en la región.

Los principales resultados de este estudio son los siguientes:

- Utilizar los excesos energéticos de la microrred y el campo solar térmico en su totalidad, es posible desalinizar agua de un sistema MED con una producción de 2,500 kg al día.
- El consumo térmico del sistema MED de 25 kW, siendo posible mantener su operación por 6 horas al día, en el periodo de 9:00 a 15:00.
- Al utilizar ambos generadores se obtiene una producción semanal de agua de aproximadamente 17,500 kg; cuando se utiliza solamente el campo de colectores solares térmicos esta disminuye a 8,300 kg a la semana.
- El exceso de energía de la microrred contribuye dos veces más a la desalinización de agua que el campo de colectores solares.
- De acuerdo a la Organización Mundial de la Salud, es posible cubrir las necesidades básicas de higiene, hidratación y nutrición de 25 personas con la producción calculada.

2.4. ARTÍCULO: Optimum operational strategies for a solar absorption cooling system in an isolated school of Mexico

Revista: *International Journal of Refrigeration*, Vol. 112 (2020) 1-13, DOI: <https://doi.org/10.1016/j.ijrefrig.2019.12.010> FI: 3.62

En este trabajo se presentó el estudio de simulación de un sistema de enfriamiento por absorción BrLi-H₂O activado con energía termosolar, instalado en la escuela primaria de la comunidad aislada de Puertecitos, México. El sistema de captación solar consiste en un total de 110 m² de colectores de tubos evacuados con reflector parabólico, un tanque de almacenamiento de agua caliente de 12 m³ y un sistema de enfriamiento por absorción de 10 toneladas de capacidad. El agua fría se distribuye en cuatro aulas de la escuela para contar con las condiciones de confort adecuadas durante el horario escolar. El objetivo principal de este estudio fue determinar las mejores condiciones de operación del sistema para proveer de enfriamiento a las aulas aprovechando a su máximo el recurso solar.

Los resultados de este trabajo se mencionan a continuación:

- Se establecieron las estrategias operativas adecuadas para un funcionamiento estable.
- Si las estrategias no se siguen, no se logran las condiciones de confort durante todo el periodo escolar.
- Con las estrategias propuestas, el sistema logra satisfacer las necesidades de enfriamiento durante los días de clases (lunes a viernes), mientras que los fines de semana se utiliza exclusivamente el campo de colectores solares para volver a condiciones de inicio óptimas.

2.5. ARTÍCULO: Simulation of a Solar-Assisted Air-Conditioning System Applied to a Remote School

Revista: *Applied Sciences*, Vol. 9 (2019) 3398, DOI:10.3390/app9163398 FI: 2.67

En este artículo se estudió la influencia que tienen las condiciones ambientales, la demanda térmica y el consumo de agua de enfriamiento en el sistema de enfriamiento por absorción termosolar de la escuela de Puertecitos, Baja California, México. Al ser una comunidad aislada con condiciones climatológicas extremas y con ausencia de servicios como el del agua potable, es importante determinar los límites operativos del sistema así como las necesidades de agua para enfriamiento.

Los principales resultados de este estudio son:

- Durante una operación a plena carga, el sistema es capaz de satisfacer las necesidades de climatización de las cuatro aulas de la escuela durante los cinco días de clases a la semana, en un periodo de 7:00 am a 3:00 pm.
- Los fines de semana se utilizan exclusivamente para almacenar energía térmica en el tanque y elevar la temperatura del mismo, utilizando el campo de colectores solares.
- Bajo una carga térmica del 75 % de la capacidad del chiller o menor, el sistema puede operar toda la semana sin necesidad de dedicar los fines de semana únicamente para almacenamiento de energía térmica.
- A su capacidad máxima, el sistema consume alrededor de 4 m³ de agua para retirar el calor del equipo por semana normal de operación.

2.6. ARTÍCULO: Techno-economic analysis of hybrid PV/T systems under different climate scenarios and energy tariffs

Revista: *Solar Energy*, Vol. 212 (2020) 191-202, DOI: <https://doi.org/10.1016/j.solener.2020.10.079>, FI: 5.74

En este trabajo se presenta el estudio tecno-económico de un sistema híbrido PV/T para la producción simultánea de energía eléctrica y térmica, bajo distintos escenarios climáticos y tarifas energéticas. Se desarrolló y validó un simulador utilizando el software TRNSYS para realizar el estudio energético comparativo de cinco ciudades con distintas condiciones meteorológicas y precios de la energía: Mexicali, Soria, Bigene, Fresno y Madison. Bajo el mismo perfil de demanda de DHW, se analizó la producción energética durante todo un año de operación, utilizando distintos números de módulos PV/T.

Los principales resultados de este artículo se listan a continuación:

- En ciudades con elevadas temperaturas ambientales y alto recurso solar, los sistemas híbridos PV/T presentan una generación energética, tanto eléctrica como térmica, superior a las ciudades con temperaturas más bajas, siendo más atractivos técnicamente.
- Las tarifas energéticas de la localidad tienen un impacto mayor en la factibilidad del proyecto que contar con un excelente recurso solar.
- Las ciudades de Mexicali y Bigene, sin importar que son los emplazamientos estudiados con mayor insolación anual, no alcanzan a recuperar la inversión inicial en instalaciones de hasta tres módulos PV/T, presentando pérdidas económicas. Por otro lado, Soria y Fresno son las regiones con una mayor factibilidad económica en este tipo de proyectos, en parte por su recurso solar (caso de Fresno), pero en mayor

importancia por el costo de la energía (caso de Soria).

- Los retornos de inversión pueden estar entre los 2 y 4.5 años cuando se consideran subsidios. Estos resultados pueden ser útiles a la hora de formular políticas gubernamentales o programas de apoyo para la instalación de tecnologías renovables, propiciando el ahorro de energía y la reducción de gases contaminantes.

2.7. ARTÍCULO: Study of a Hybrid Solar Absorption-Cooling and Flash-Desalination System

Revista: *Energies*, Vol. 13 (2020) 3943, DOI: 10.3390/en13153943, FI: 3.0

En este trabajo se estudió operativamente un sistema híbrido de enfriamiento por absorción BrLi/H₂O y desalinización flash, activado con energía solar térmica. Bajo el escenario de un complejo hotelero de 20 habitaciones en la ciudad de San Felipe, Baja California, México, se realizó el dimensionamiento del campo de captación solar y almacenamiento de energía térmica. La operación dinámica durante la semana con mayor temperatura ambiental del año es analizada.

Los resultados encontrados se presentan a continuación:

- Se propuso un novedoso sistema híbrido solar de enfriamiento por absorción BrLi/H₂O y desalinización flash de agua de mar. Este sistema acopla un ciclo abierto de enfriamiento por absorción de simple efecto con tres niveles de presión, por lo que permite realizar una integración energética y másica interna con un proceso de desalinización flash de una etapa, produciendo simultáneamente enfriamiento y agua potable.
- El subproceso de enfriamiento por absorción al utilizar tres niveles de presión, presentó un incremento del COP promedio de 13.88 %, respecto a los sistemas convencionales de absorción de BrLi/H₂O. También, al considerar que el sistema brinda enfriamiento

y desalinización de manera simultánea, se tiene un COP promedio de 1.64, siendo 2.27 veces superior al COP de las unidades convencionales de absorción BrLi/H₂O de simple efecto.

- El sistema, durante la semana crítica, presentó una producción de agua desalada de 16.94 m³ con un PR promedio de 0.83; mientras que la producción diaria promedio de agua producto fue de 2,406 kg/día, suficientes para satisfacer los requerimientos hídricos diarios de 4 personas en un hotel costero en México o para cubrir los servicios básicos de 24 personas según la World Health Organization (WHO).
- El dimensionamiento del campo de colectores solares térmicos y TEST resultó en un área total de captación de 620 m² con 30 m³ de almacenamiento en forma de calor sensible. Con lo anterior se logró una fracción solar del 69 % requiriendo 53 MWh extras de energía auxiliar para satisfacer los requerimientos de energía térmica.

2.8. Artículos en los que se participó como coautor

- R. López-Zavala, N. Velázquez-Limón, L. A. González-Uribe, J.A. Aguilar-Jiménez, J. Álvarez-Mancilla, A. Acuña, S. Islas. (2019). A novel LiBr/H₂O absorption cooling and desalination system with three pressure levels. *International Journal of Refrigeration* 99, 469-478. <https://doi.org/10.1016/j.ijrefrig.2019.01.003>.
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**A. Low-temperature multiple-effect
desalination/organic Rankine cycle system
with a novel integration for fresh water and
electrical energy production**

Revista: Desalination

ANEXO A. LOW-TEMPERATURE MULTIPLE-EFFECT DESALINATION/ORGANIC RANKINE CYCLE SYSTEM WITH A NOVEL INTEGRATION FOR FRESH WATER AND ELECTRICAL ENERGY PRODUCTION

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Low-temperature multiple-effect desalination/organic Rankine cycle system with a novel integration for fresh water and electrical energy production



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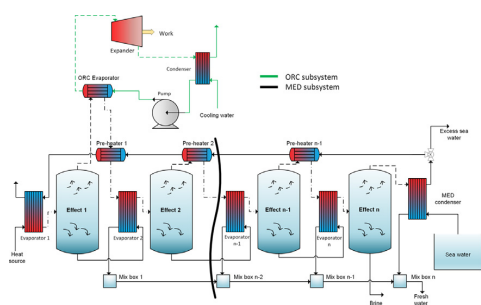
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GRAPHICAL ABSTRACT



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ABSTRACT

This paper presents a novel energetic integration of a Multiple Effect Thermal Desalination System (MED) and an Organic Rankine Cycle (ORC) for simultaneous production of potable water and electrical energy, using low-temperature energetic sources. The thermal energy required for the system's operation is supplied by the MED's evaporator, while the ORC is activated using a fraction of the latent heat of condensation of the water vapor produced in the first effect of the MED. By doing this, the production of water in the first stage of the desalination system increases and, thus, the final production of distillate also increases. A simulation and validation of the proposal was conducted. The MED/ORC system has a 3.95% increase on the average Performance Ratio when the electrical energy production increases in 10 kW, presenting only a 1.57% increase on the total heat transfer area. MED/ORC system with an electrical energy production of 50 kW is 22% more efficient in water desalination than a MED system without integration, while requiring only 6.9% more heat transfer area. The results show that the MED/ORC energetic integration studied benefits both the final production of desalinated water, and the MED's efficiency without considerably increasing the required heat transfer area.

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ANEXO A. LOW-TEMPERATURE MULTIPLE-EFFECT DESALINATION/ORGANIC RANKINE CYCLE SYSTEM WITH A NOVEL INTEGRATION FOR FRESH WATER AND ELECTRICAL ENERGY PRODUCTION

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1. Introduction

Water and electrical energy are first order necessities. The quality of our lives depends on the availability of these resources. In isolated and arid regions, the demand of electrical energy and water usually comes hand in hand [1]; hence, there is particular interest in the integration of the production of both fresh water and electrical energy. The global trend is the integration of technologies [2] in order to increase savings and energy use efficiency, as well as the production of diverse services with a common energy source. However, the challenge resides in the availability of systems to produce efficiently both drinking water, and electrical energy [3].

There are several processes for seawater desalination, as well as for the production of electrical energy. The best choice depends on the scale of the project and the nature of the available energetic activation source. When low-cost fuel or waste energy is available, MED systems can be the best option for the production of drinking water since the heating and electrical energy requirements are lower than in other desalination systems [4,5]. MED technology requires almost half the energy required for pumping, and almost the same amount of thermal energy, within Multiple Stage Flash systems (MSF), assuming that both systems have the same gain ratio [6]. Nevertheless, the current trend of using low-temperature MED (LTMED) systems allows for the usage of low quality energy sources (within the range of 70 °C) such as renewable energies [7], and, thus, to consume an amount of work close to that required by Reverse Osmosis (RO), which is one of the most efficient desalination systems [6].

There is ample research on MED integration to Rankine cycles for the simultaneous production of water and electrical energy for the case of large capacity systems, proving that the hybridization of two or more technologies provides an energetic benefit. Sharaf et al. [8] compared different solar thermal energy-activated MED systems architectures, one of which incorporated a Rankine cycle for electrical energy production; they also stated that a portion of the electrical energy produced could satisfy the pumping requirements, while the remainder could be incorporated into the main grid. Ortega-Delgado et al. [9] analyzed the integration of thermal vapor compression MED systems to a Rankine cycle, and studied its effect on efficiency, production of fresh water, and the specific heat transfer area. In this setup, mid-high pressure water vapor from the power cycle turbine enters a thermo-compressor (ejector), which absorbs part of the vapor produced at vacuum pressure in some of the effects of the MED system. This vapor is then carried into the evaporator that activates the desalination system, increasing the efficiency of the MED. Mata-Torres et al. [10] studied the implications of integrating large MED and Rankine systems for the joint production of fresh water and electrical energy, and concluded that the part-load operation is the key factor for these kind of setups.

On the other hand, in recent years ORC's have received a lot of attention due to their capability to use low-capacity and low-temperature thermal energy sources for the production of electrical energy [11]. Since these systems use an organic fluid with a low boiling point instead of water as working fluid, they allow for a thermal electric conversion with sources those are unavailable for traditional Rankine cycle [12–14]. Their operating conditions allow for a simple energetic integration with other technologies for different services such as refrigeration [15], heating [16], and desalination [17]; thus, its hybridization with LTMED systems are a subject of great interest. ORC's have been thoroughly studied when combined with other desalination technologies, such as RO [18], humidification/dehumidification [19] and membranes [20]. Astolfi et al. [21] studied different options for the production of electrical energy and fresh water to satisfy the needs of a community, using renewable energies instead of fossil fuel. Their results present a number of benefits obtained when different desalination technologies and solar activation sources (such as day-night complementarity when using photovoltaic modules during periods of high insolation, or thermosolar driven ORC systems to extend the operating

hours, which, in turn, activate the RO and the MED systems, respectively) are combined.

There are different ways to conduct an integration of MED and ORC systems. Baccioli et al. [22] analyzed two integration setups: (a) hybrid serial-cascade setup, where the ORC is thermally activated, first, with waste heat, and afterwards with the remnant heat from MED's energetic source, where the heat of the condensation of the work fluid of the ORC is used to pre-heat the seawater input of the MED; and, (b) cascade setup, where the MED system is completely activated with the condensation heat from the ORC. The serial-cascade setup greatly increased the profitability index for almost all the considered cases. Using a different way of integrating ORC with desalination, Al-Weshahi et al. [23] proposed the recovery of the distillate's heat of an MSF system for the activation of the power cycle, comparing different organic fluids. The authors mention that the amount of net power generated is the key issue for design and optimization. It has been shown that the number of MSF stages exploited has a significant impact. Li et al. [24] proposed the integration of a supercritical ORC, an ejector, and a MED desalination system, in which the organic fluid at the exit of the expander of the ORC is used to drive the MED and to condense the vapor produced in the last effect. Ameri and Jordani [25] analyzed a MED and an ORC connected in cascade, driven by the combustion gases of a gas cycle. Shahzad et al. [26] proposes a methodology for traditional power generation systems coupled to thermal desalination systems (MED or MSF) to account for the energy consumed by the desalination system as the primary energy source, and thus be able to analyze the effects of hybridization from an approach that affects the electrical power cycle.

The integration of MED systems to polygenerative cycles is a way for the use of different effluents for the production of other services. You et al. [27] presented a polygenerative cycle to produce cooling, heating, electrical energy and desalination, through the coupling of a gas turbine, a MED and an ORC. In this configuration, the high-temperature combustion gases are used to generate, in the first instance, steam for the activation of the MED, followed by the activation of the ORC and finally uses the remaining thermal energy for domestic water heating. They have an overall efficiency of up to 46.7% under design conditions. Sharan et al. [28] propose a supercritical carbon dioxide Brayton cycle which combines a MED system for the cogeneration of electrical energy and desalination of water. These power cycles have high combustion gas temperatures, so their use as a source of thermal activation of MED systems becomes attractive, as it does not become a parasitic energy load, as is usually the case when coupled to Rankine cycles. In another study by Sharan [29] carried out modifications to the previous MED system and achieved thermal power cycle efficiencies of 49.2% and a distillate cost 17.8% lower than those of reverse osmosis plants. Leiva-Illanes et al. [30] analyzed different coupling schemes of a system for the production of electricity, fresh water, cooling, and process heat, activated with solar thermal energy. When studying twenty-one configurations, their results reveal that the most recommendable are when the desalination plant replaces the power cycle condenser, as well as the refrigeration plant and the process heat module is coupled to the extractions of the turbine. Also, according to the analysis presented by Shahzad et al. [31] performing a comparison of primary energy and conversions of thermal energy to electricity, desalination plants MED and MSF being bottoming cycle of a cogeneration plant can consume only 2.5 kWh_e/m³ of desalinated water, consuming only 2–3% of primary energy [32].

Based on the aforementioned, it is clear that the energetic integration of desalination and electrical energy production systems is a technologically attractive, and particularly interesting, option. The studies tend to analyze different operating conditions and effluent usage in order to diminish the heating and/or cooling requirements of the systems. Typically, the cascade configuration (where latent heat from the ORC condensation is used to thermally activate the desalination process), is the most attractive due to the inefficiency of the power cycle. However, this architecture requires high temperature and ORC

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condensation conditions in order to achieve the heat transfer phenomena and the subsequent activation of the desalination process. Considering that the Top Brine Temperature (TBT) is usually 70 °C, low temperature ORC's encounter significant limitations for the provision of necessary energy to the MED. This paper presents a novel MED/ORC energetic integration in which, unlike the aforementioned integrations, the ORC's evaporator is coupled directly to the first effect of the MED system. The required thermal energy for the activation of both systems is supplied by the MED's evaporator, which allows for a greater production of water vapor and, thus, of final desalinated water. With the proposed integration, the ORC activation is possible using a portion of the latent heat of condensation of the vapor produced in the first effect, simultaneously producing electrical energy and desalinated water, using low temperature thermal energy. The proposed integration differs from traditional Rankine-MED coupling architectures because the desalination system does not use the energy effluents from the power cycle, in which the operating conditions of the ORC or Rankine must be modified in order to achieve the quality of energy needed to activate the MED, decreasing its electrical conversion efficiency [33], even consume between 2 and 3% of the primary energy source [26]. One of the main advantages with the proposed configuration is that the ORC, being activated energetically with an internal MED stream, maintains its operating conditions without affecting its efficiency, causing an increase in the final production of desalinated water. Hence, the system can be driven with renewable energy sources and/or waste heat, can be used in applications in coastal communities, isolated and with needs for desalinated water and electricity, including the domestic (multi-family), commercial and industrial sectors, due to its low temperatures and operating pressures.

Section 2 shows the description of the system, presenting the proposed operation and integration. Section 3 explains the methodology used for the validation and simulation of the proposal, as well as the considerations for this study. Section 4 analyzes the MED/ORD system with the proposed energetic integration, and compares it to traditional MED systems with different number of effects, showing the technical advantages of the ORC coupling.

2. System description

The proposed technology consists of two main subsystems, the seawater desalination subsystem, and the electrical energy generation subsystem, shown in Fig. 1. The seawater desalination subsystem is composed of a 14-effects MED in which seawater is used to condense the vapor generated in the last effect, and its latent heat is used to reduce the heating requirements. Afterwards, a part of that current serves as supply water for the MED subsystem, and is pre-heated using a portion of the latent heat of condensation of the vapor generated in each of the effects. When this seawater current reaches pre-heater 1, it receives a portion of the latent heat of condensation of the vapor generated in the first effect, after having delivered the rest of the heat into the electrical energy generation subsystem. The supply water, now at a temperature close to the TBT of the first effect, is heated in evaporator 1 using the external activation energy source, to a temperature greater than that of the operation temperature of the first effect. The amount of energy supplied to the evaporator 1 is the required for the generation of the necessary water vapor in the first effect, and for a part of the latent heat to be delivered as activation source to the ORC subsystem and to the subsequent effects of the MED. The first effect has an operating pressure lesser than the atmospheric pressure; hence, the supply current, when it enters this effect, suddenly expands and part of it becomes vapor, due to the flash phenomena and to evaporation.

The power cycle evaporator condenses the portion of the vapor needed for the energetic activation of the ORC, which evaporates the organic fluid and carries it to its saturation temperature for further expansion and electrical energy production by means of an expander-generator device. The liquid-vapor current that exits the ORC's

evaporator enters pre-heater 1 in order to deliver another portion of latent heat, the heat of condensation of the portion of the vapor flashed in the first effect, in order to rise the temperature of the supply seawater. The remaining vapor activates the subsequent effects, where it delivers, within the evaporator of each stage, heat of condensation for the brine that comes from the previous effect, but now with lower pressure and saturation temperature. Hence, an adequate heat transfer occurs. This process repeats effect after effect. In the last effect, there is no pre-heating of the supply water, since the approach temperatures between the involved currents are too low to achieve an effective heat transfer. Therefore, this device is dismissed and both the vapor produced by the flash and the evaporation in this last separator enter the final condenser, where seawater is used to condense the final vapor. The total distillate equals the sum of all the condensates of the MED evaporators.

The ORC operates under a basic setup, in which the organic fluid exits the evaporator as saturated vapor. Afterwards, it enters the expander in order to exploit the mechanic energy produced by its pressure reduction. The low pressure-organic fluid that exits the expander condenses until it becomes saturated liquid. Then, the pump raises its pressure until it matches the evaporator's operating pressure in order to proceed with the generation of electrical energy, evaporating it using a portion of the latent heat of condensation of the water vapor that comes from the first effect of the MED, thus, repeating the cycle. To achieve the proposed integration the ORC must maintain its evaporation temperature below the MED's TBT. Hence, it must be insured that the evaporation pressure, $P_{eva, ORC}$, is such that $P_{eva, ORC} < TBT$, considering a temperature differential that allows for an effective heat exchange. This allows the organic fluid to be converted to saturated vapor within the ORC's evaporator, and MED's latent heat of condensation can be used to change phase. Otherwise, if $T_{eva, ORC} > TBT$, the latent heat of the MED will only serve as a source of sensible heat for the organic fluid, which is only a small portion in comparison to its latent heat. This would require the incorporation of an additional evaporator in order to change phase.

The advantage and novelty of the proposed energetic integration of the MED and ORC subsystems relies on the fact that the required heat for the activation of the ORC, since it is supplied jointly with the MED's activation heat from evaporator 1, is used to generate an additional amount of vapor within the first effect, unlike a system that is not integrated to an ORC in a serial-cascade cycle. Afterwards, this vapor becomes produce water and the amount of final distillate increases. The advantages and novelties of our technological proposal can be summarized as follows:

The total amount of energy supplied to the system is the same as that required by both equipments operating individually, but our integration increases the production of desalinated water in the MED.

The proposed integration increases the capacity and efficiency of the MED without greatly increasing the total heat transfer area required.

Unlike traditional ORC/MED hybrids, our proposal integrates the ORC in cascade with the MED, so the operating conditions of the power cycle do not have to be modified, and therefore its efficiency does not decrease as it usually happens with the configurations present in the literature.

3. Methodology

In order to conduct a detailed analysis on the MED/ORC system and to prove its technical feasibility under different operating conditions, Aspen Plus was used. This software allows to conduct simulation studies in sequential-modular fashion, using existing modules coded with the mathematical models of a wide range of devices, among them heat exchangers, separators, pumps, and expanders, just to mention some of the ones involved in this proposal. Having an ample database of thermodynamic and heat transfer properties from a variety of components,

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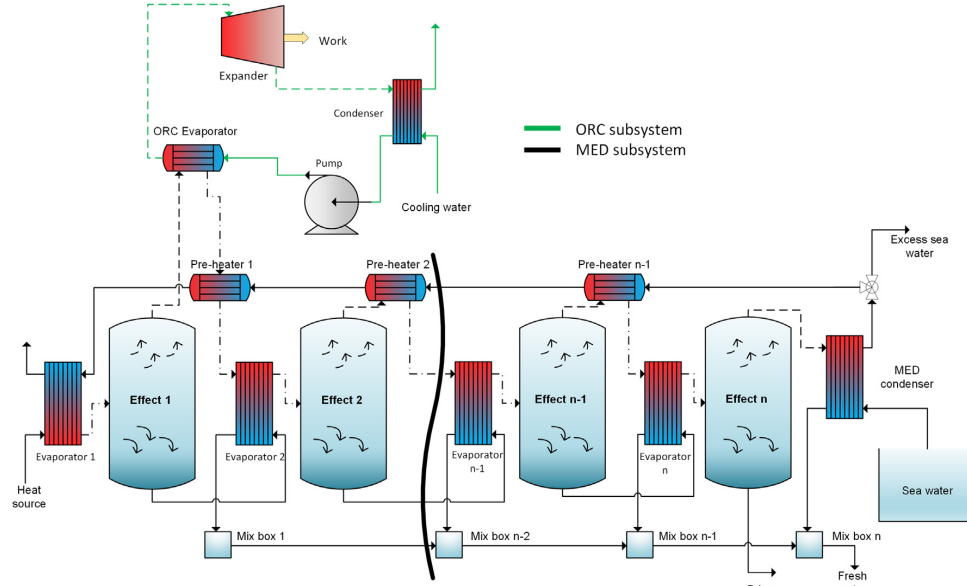


Fig. 1. Schematics of the desalination and electrical energy generation system by means of MED/ORC.

Aspen Plus becomes a very useful tool for the simulation and analysis of energetic systems such as power cycles and desalination systems. Furthermore, the use of modules interconnected through input/output variables allows to modify and to test new system configurations easily. In order to simulate the MED/ORC technological proposal, the following considerations were addressed:

- Steady-state analysis;
- Heat losses and drop pressures were not considered;
- The mass flow provided by the ORC's pump is variable in order to insure saturated vapor conditions at the exit of the evaporator;
- Both the expander and the ORC's pump operate under the isentropic-efficiency principle;
- Neither Boiling Point Elevation (BPE), nor Non-Equilibrium Allowance (NEA) are considered for the MED;
- The seawater flow is variable within the MED's condenser in order to insure water condensation;
- The seawater supplied to the first effect, and the input brine for each of the remaining effects diminish their pressure within the corresponding evaporator;
- Flash vapor within the mix-boxes is dismissed.

For the evaluation of the desalination and the electrical energy generation subsystems' performance under different operating conditions, as well as its comparison with other technologies, the indicators described in the following lines are used. The MED subsystem was evaluated using the Performance Ratio (PR), which is the ratio between the final distillate over the thermal energy supplied to the subsystem, multiplied by the latent water vapor heat at 73 °C [34]. This is expressed in Eq. (1):

$$PR = \frac{\dot{m}_d}{\dot{Q}_{in,MED}} \times \frac{2,326 \text{ kJ}}{1 \text{ kg}} \quad (1)$$

where $\dot{m}_d$ is the final distilled water flow (kg/s) and $\dot{Q}_{in,MED}$ is the heat provided for the activation of the MED subsystem (kW).

The electrical energy generation subsystem was evaluated based on the thermal efficiency of the thermodynamic cycle (2):

$$\eta_t = \frac{\dot{W}_{exp} - \dot{W}_{pump}}{\dot{Q}_{in,ORC}} \quad (2)$$

where η_t is the thermal efficiency of the ORC, $\dot{W}_{exp}$ is the work produced by the expander, $\dot{W}_{pump}$ is the electrical energy consumption for the pump, and $\dot{Q}_{in,ORC}$ is the fraction of the energy used for the activation of the ORC. Both $\dot{Q}_{in,MED}$ and $\dot{Q}_{in,ORC}$ are supplied simultaneously to evaporator 1.

The energy balance within the ORC evaporator's, where the energetic integration with the MED is conducted, is calculated using (3):

$$\dot{Q}_{in,ORC} = \dot{m}_{ORC}(h_{out} - h_{in}) = \dot{m}_{v,1}(h_{v,1,in} - h_{v,1,out}) \quad (3)$$

where $\dot{m}_{ORC}$ is the mass flow of the ORC evaporator's organic fluid, h_{out} and h_{in} its corresponding input and output enthalpies, $\dot{m}_{v,1}$ the mass flow of the vapor produced in the first effect of the MED, and $h_{v,1,in}$ and $h_{v,1,out}$ its input and output enthalpies. Fig. 2 shows the flowchart of this heat exchanger.

To compare between different setups and operating conditions for the MED subsystem, the specific heat transfer area was used [35], which relates the total heat transfer area for all the MED components to the final distillate. This is expressed in (4):

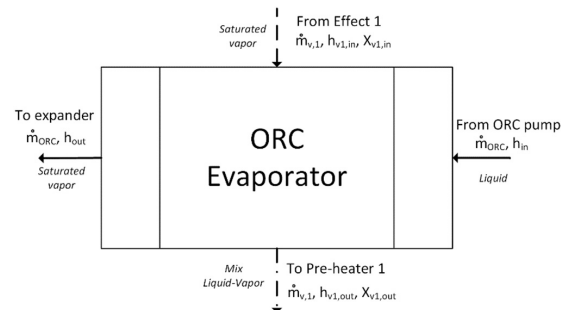


Fig. 2. ORC evaporator flowchart.

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$$sA = \frac{\sum A_{eff} + \sum A_{ph} + A_c}{\dot{m}_d} \quad (4)$$

where sA is the specific heat transfer area, $\sum A_{eff}$ and $\sum A_{ph}$ are the sums of the heat transfer area for the effects and the pre-heaters, respectively, and A_c is the required exchange area within the MED's condenser.

To calculate the devices' heat transfer area Aspen Plus uses the following formula (5):

$$A_i = \frac{\dot{Q}_i}{U_i \times LMTD} \quad (5)$$

where A_i is the device's heat transfer area, $\dot{Q}_i$ the transferred heat, U_i is the global heat transfer coefficient, and $LMTD$ is the Log-Mean Temperature Difference.

To calculate the global heat transfer coefficient for the first effect's evaporator, the correlation proposed by El-Dessouky and Ettouney [36] was used, as seen in Eq. (6):

$$U_h = 1.9695 + 1.2057 \times 10^{-2} T_{v,1} - 8.5989 \times 10^{-5} T_{v,1}^2 + 2.5651 \times 10^{-7} T_{v,1}^3 \quad (6)$$

where U_h is the global heat transfer coefficient for the first effect's evaporator, and $T_{v,1}$ is the operating temperature of the first effect.

The correlation used to calculate the global heat transfer coefficient for the pre-heaters was based on Palenzuela et al. [37], as seen in Eq. (7):

$$U_{ph} = -0.000540399 + 0.836569 \dot{m}_f \quad (7)$$

where U_{ph} is the heat transfer coefficient for the pre-heaters, and $\dot{m}_f$ is the mass flow of the seawater supplied to the MED.

For the evaporators within effects 2 to i , the correlation presented by El-Dessouky et al. [38] was used, as seen in Eq. (8):

$$U_e = 1961.9 + 12.6 T_v - 9.6 \times 10^{-2} T_v^2 + 3.16 \times 10^{-4} T_v^3 \quad (8)$$

where T_v is the vapor temperature at effect i .

For the condenser, Palenzuela et al.'s Eq. (9) was used [39]:

$$U_c = 0.93 - 0.0013 \dot{m}_{cw} - 0.08 \left(\frac{T_{cw,in} + T_{cw,out}}{2} \right) + 0.11 T_v(i) \quad (9)$$

where $\dot{m}_{cw}$ is the mass flow of the cooling water in the condenser, and $T_{cw,in}$ and $T_{cw,out}$ are its input and output temperature, respectively.

The Aspen-Plus validation of the MED/ORC system's simulation was conducted separately for: (1) the desalination process; and, (2) the electrical energy generation process. For (1), the modelled and experimental results provided by Palenzuela et al. [35] were used. For part (2), the results provided by Liu et al. [40] were used. To conduct the validation, the individual simulations were adjusted to the type and operation conditions of the devices used by the authors in their respective studies.

The MED system used for comparison has 14 effects and 13 pre-heaters, an activation vapor mass flow of 295.92 kg/h at 70.8 °C, a TBT of 68 °C, a 35 °C temperature at the last effect, and seawater with a concentration of 35 g/kg at a temperature of 25 °C. Table 1 and Fig. 3 show the results of the validation of the simulation, regarding performance and water vapor production per effect indicators.

To validate the ORC an ideal basic cycle working with R123 cooler, a condensation temperature of 30 °C, and evaporation temperature increments in the range of 50–150 °C, were used, as seen in Fig. 4.

Table 1

Validation of the seawater desalination process.

	Palenzuela et al. model	Palenzuela et al. Experimental	Simulation
Fresh water product (kg/h)	3003	2984	3002
Seawater entering condenser (kg/h)	15,848	14,558	11,670
PR	10.2	9.7	10.1
Recovery ratio (%)	37.6	37.5	37.5

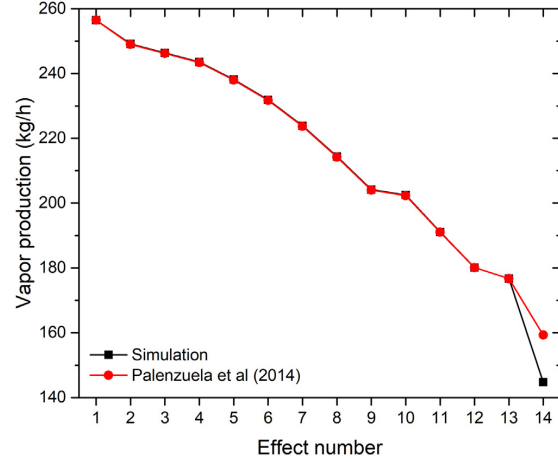


Fig. 3. Comparison between the water vapor production variation for each of the 14 effects of the MED and the experimental results [35].

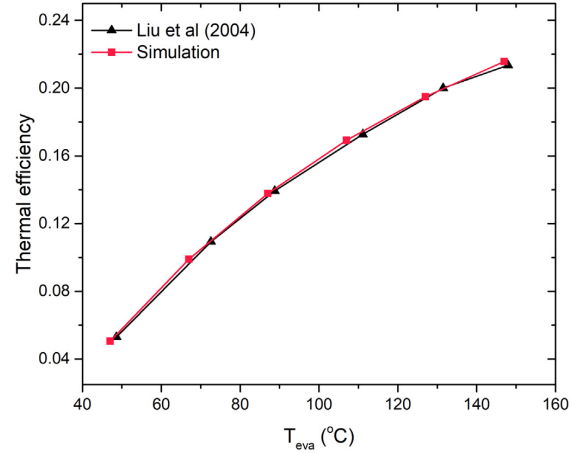


Fig. 4. Comparison between simulation and experimental results presented by Liu et al. [40] of the impact of evaporation temperature on the ORC's cycle thermal efficiency at a condensation temperature of 30 °C.

4. Results

The proposed MED/ORC's performance is influenced by the operating conditions and the amount of thermal energy exchanged between the subsystems. Therefore, the impact of these variables should be studied in order to determine the best operating and design conditions. It is well known that the devices' heat transfer area is one of the main constraints when determining the number of effects of a MED system, and, thus, it restricts the possibility of achieving higher PR levels. Hence, the greater the number of effects, the greater the water

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Table 2

Operative specifications for the MED subsystem.

Effects number	14–18
Pre-heaters number	13–17
Effect for the ORC integration	1
Top brine temperature	70 °C
Condensation temperature	35 °C
Sea water salt concentration	35,000 ppm
Sea water temperature	27 °C
Feed water temperature	32.2 °C
Sea water mass flow in the condenser	10.39 kg/s
Feed water mass flow	2.20 kg/s
Heat source temperature	80 °C
Thermal energy supply	260 kW

production. However, the heat transfer area increases exponentially, so the cost-benefit relationship ends up being the key factor for the definition of the capacity and efficiency of the desalination system. Because of this, the total heat transfer area and the PR are used as benchmarks.

4.1. MED without ORC integration

In order to have a comparison between the benefits of the proposed integration, we start with traditional 14, 15, 16, 17, and 18 effects MED's. The specifications of this subsystem are shown in Table 2, while the results of the simulation can be seen in Fig. 5. For a 14-effects MED, under the described assumptions, a fresh water production of approximately 4800 kg/h can be achieved, requiring a total heat transfer area of 520 m². When the number of effects increases to 18, with the same operating conditions, the production of water increases in 28.7% and the heat transfer area in 75.2%, 6150 kg/h and 918 m², respectively. The temperature differences between the separation stages are the main cause of this. The larger the number of effects, the lower this difference will be, and, therefore, a larger heat transfer area is required to exchange the latent heat of condensation of the vapor generated with the brine from the previous effect. The 14-effects MED presents temperature steps of 2.7 °C, while the 18-effects MED presents 2.05 °C steps.

Fig. 6 shows with more detail the increase of the heat transfer area respective to the number of effects. The figure shows the variation of this parameter for each of the different effect sizes of the MED. For a 14-effects device the heat transfer area averages 35 m² between effects 2–14, while for an 18-effects MED the average is 49.6 m², which corresponds to a 41% increase for each heat exchanger. It is worth noting that all MED's, regardless of their size, require an area of 10 m² for their first effect, significantly lower than the area required for the rest of the effects. This is due to the fact that, in the first-effect evaporator the

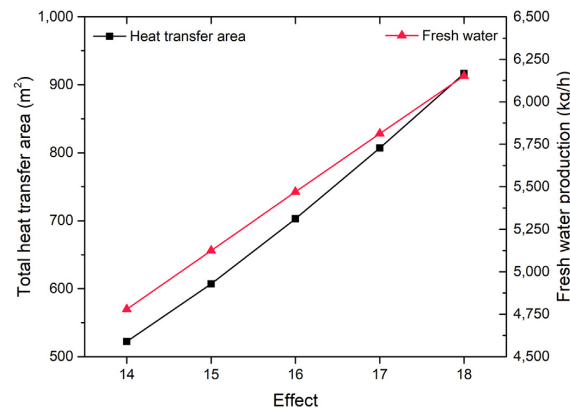


Fig. 5. Required heat transfer area variation, and desalinated seawater in MED simulated systems of 14, 15, 16, 17 and 18 effects.

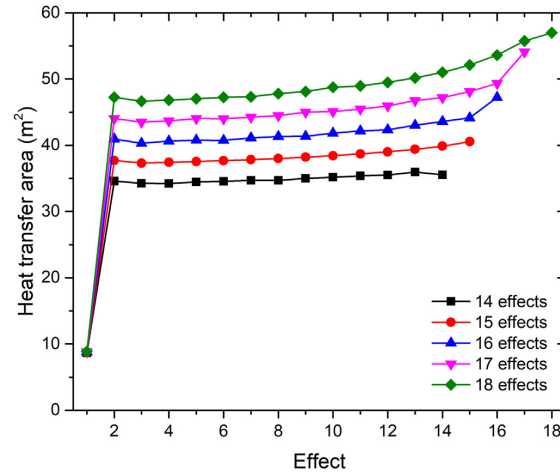


Fig. 6. Variation in heat transfer area for different effect size MED systems.

system is fully thermally activated with the external energy source, which has a temperature of 80 °C, which corresponds to a 10 °C difference with the TBT, which allows for a heat transfer area and an approach temperature lesser than those of the remaining effect.

The results show that an increase in the desalination due to an increment of effects is not linear with respect to the increase in total heat transfer area required. Therefore, in order to define the capacity and size of the MED desalination systems it is necessary to conduct an economic feasibility analysis for the cost of the heat transfer area. This aspect was not considered in this paper, since the goal was to determine the technical feasibility of the MED/ORC integration.

4.2. MED/ORC

To integrate the ORC subsystem to the first effect of the MED subsystem, the thermal energy required for the activation of the power cycle should be considered together with the energy needed for the MED, and it should be supplied to evaporator 1. This way, using a greater amount of energy for the heating of seawater, more water vapor is produced, increasing, thus, the final amount of distillate. However, the heat transfer area's requirement for evaporator 1 increases due to the increase in energy supplied. Therefore, the increase in desalinated water provided by this integration should be analyzed. Table 3 shows the operational characteristics of the ORC subsystem used for the simulation of the proposed system. The TBT in the MED subsystem is of 70 °C; hence, evaporation and condensation temperatures of 67 °C and 30 °C, respectively, where defined for the ORC. Under these operational constraints, and considering an isentropic efficiency of 0.7 for the expander and the pump, the ORC operates with a thermal efficiency of 6.2% when work fluid R245fa is used.

For this study, the integration of the ORC subsystem to a 14-effects

Table 3

Operational characteristics for the ORC subsystem.

Evaporation temperature	67 °C
Evaporation pressure	556 kPa
Condensation temperature	30 °C
Condensation pressure	175 kPa
Expander isentropic efficiency	0.7
Pump isentropic efficiency	0.7
Fluid	R245fa
Method for properties calculation	EOS Peng-Robinson [41,42]
Thermal efficiency	6.2%

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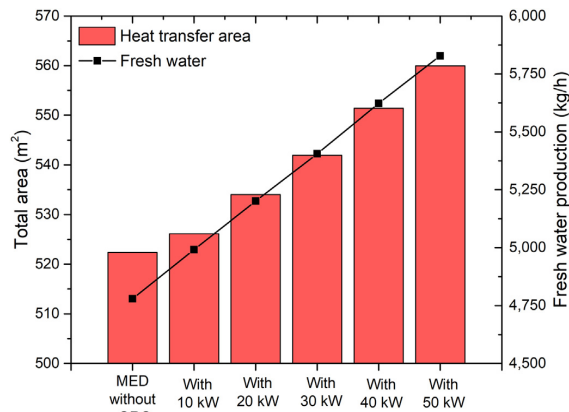


Fig. 7. Total heat transfer area and desalinized water production for a 14-effects MED integrated to a 10, 20, 30, 40, and 50 kW electric generation capacity ORC.

MED was considered. The impact of the electrical energy production on the total amount of desalinized water, keeping the MED's operating conditions constant, as seen in Table 2, were analyzed. The benefit of the incorporation of low-temperature activation ORC technology to established thermal desalination systems was determined. Fig. 7 shows the MED subsystem's shifts in total heat transfer area and in the production of fresh water when it is energetically integrated to an ORC subsystem with electric capacities of 10, 20, 30, 40, and 50 kW. A 14-effects MED system without ORC integration was used as a benchmark. It can be seen that when a 10 kW ORC is integrated, total desalinized water production increases from 4780 to 4990 kg/h, an increase in 4.4%, while the required area increases in 0.43%. This is due to the fact that the area of effects 2–14 remain constant, and the only variable affected is the area of the first effect because of the integration to the ORC. The percentages of produce water increase for 20, 30, 40, and 50 kW capacities are 8.8, 13, 17.65, and 22%, respectively, with a maximum increase of 6.9% in the total heat exchange area for the case of a 50 kW ORC. The increase in desalinized water production only occurs in the first effect since the extra latent condensation heat is transferred to the power cycle's evaporator because of the integration with the ORC subsystem. The same amount of vapor enters the second effect's evaporator, keeping the same operation conditions regardless of the electric production capacity of the ORC.

In order to compare the MED subsystem's capacity with and without ORC integration, Fig. 8 shows the variation in PR and the total heat transfer area for different cases of electrical energy production and MED sizes. Using the baseline 14-effects, no integration setup, a PR of 11.87 is achieved, with a total area of 523 m². As the size of the MED (number of effects) and energetic integration with the ORC increase, the PR and the area tend to rise at a different rate. For the independent MED, the PR increases an average of 6.52% as the device size increases, i.e. going from 14 to 15, or from 17 to 18 effects, while the area increases an average of 15%. An 18-effects MED requires a heat transfer area 75% larger than a 14-effects one. On the other hand, the MED/ORC setup has a 3.95% increase in PR when the electric production raises 10 kW, but the heat transfer area only increases in 1.57%. A MED/ORC system with an electrical energy production of 50 kW is 22% more efficient in water desalination than a MED system without ORC integration and it requires only 6.9% more heat transfer area.

Fig. 9 shows the shift in specific area and PR for the aforementioned setups. The specific area for the MED/ORC integration tends to diminish from 0.105 to 0.095 as the electrical energy production increases from 10 to 50 kW. This means that, the greater the MED/ORC

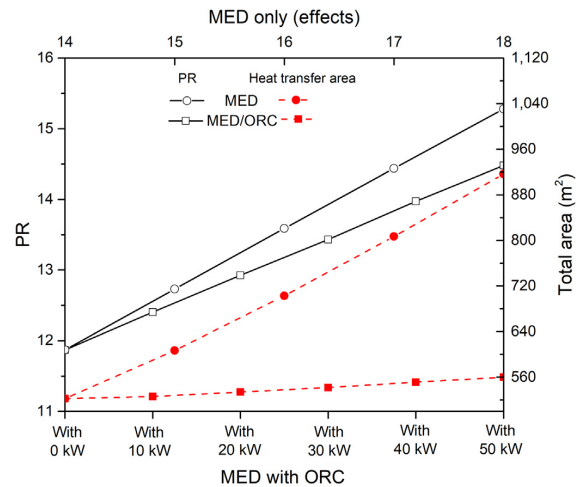


Fig. 8. PR and total heat transfer area for a traditional 14–18 effects MED (upper axis) and the 0–50 kW MED/ORC technological proposal (lower axis).

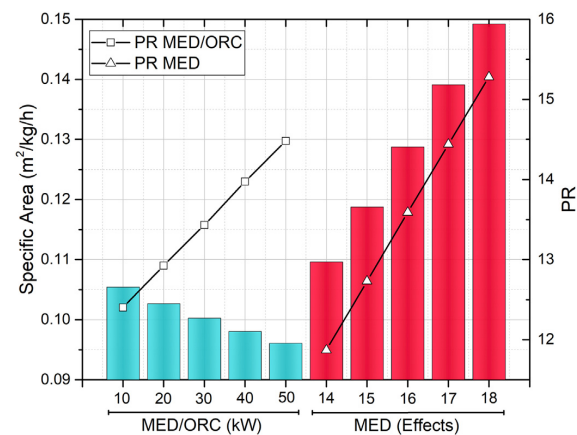


Fig. 9. Shift in specific area and PR for MED and MED/ORC systems with different capacities.

integration, the lesser the total heat transfer area for the production of a given amount of desalinized water will be. Furthermore, the MED subsystem's PR tends to be more efficient with the increase of integration, rising from 11.87 to 14.50. On the other hand, as the number of effects in the MED system increases, the efficiency of the system increases as well, but so does the specific area; hence, heat transfer area requirements will be larger and larger.

Table 4 shows the values of the properties of the different streams involved in the proposed cycle, for the nominal operating conditions of the hybridization of a MED of 14 effects and an ORC of 30 kW electric.

5. Conclusions

This paper analyzed the feasibility of an energetic integration of an electrical energy production system (ORC) to a MED seawater desalination system. The proposal is to supply the necessary thermal energy for the simultaneous activation of the ORC and the MED to the evaporator of the first effect of the desalination subsystem, increasing the heat of the supply seawater flow and causing a greater water vapor production within the first effect. There, a part of its latent

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Table 4

Streams properties in the design condition for a MED of 14 effects and an ORC of 30 kW.

	Temperature (°C)	Pressure (kPa)	Mass flow (kg/s)	Vapor flow (kg/s)	Brine flow (kg/s)
MED					
Sea water	27	101.33	10.22	–	–
Feed water	32.2	200	2.2	–	–
Feed water in effect 1	64.85	200	2.2	–	–
Effect 1	70.0	31.2	2.2	0.2754	1.9246
Effect 2	67.3	27.7	1.9155	0.0901	1.8254
Effect 3	64.6	24.6	1.8167	0.0899	1.7268
Effect 4	61.9	21.8	1.7187	0.0888	1.6299
Effect 5	59.2	19.24	1.6215	0.0886	1.5329
Effect 6	56.5	16.96	1.5258	0.0883	1.4374
Effect 7	53.8	14.91	1.4307	0.0881	1.3426
Effect 8	51.2	13.07	1.3364	0.0879	1.2485
Effect 9	48.5	11.44	1.2428	0.0876	1.1552
Effect 10	45.8	9.98	1.1499	0.0874	1.0625
Effect 11	43.1	8.68	1.0577	0.0872	0.9705
Effect 12	40.4	7.54	0.9661	0.0869	0.8792
Effect 13	37.7	6.52	0.8752	0.0867	0.7885
Effect 14	35.0	5.63	0.7885	0.0900	0.6986
ORC					
Evaporator inlet	66.93	556	2	0	–
Expander inlet	66.93	556	2	1	–
Condenser inlet	44.71	176	2	1	–
Pump inlet	29.77	176	2	0	–

condensation heat is used to thermally activate the OCR in order to produce electrical energy.

The study considered the effect of electrical energy production capacity on both the efficiency and the total heat transfer area required in the MED subsystem, and compared it to a 14, 15, 16, 17, and 18 effect MED without ORC integration. The following results were found:

- The proposed energetic integration benefits the desalinated water production and the MED's efficiency without significantly increasing the required total heat transfer area;
- The increase in produced water for 20, 30, 40, and 50 kW MED/ORC capacities are of 8.8, 13, 17.65 and 22%, respectively;
- The MED/ORC system has an average PR increase of 3.95% when the electrical energy production increases 10 kW, but only a 1.57% increase in the total heat transfer area. A MED/ORC system with an electrical energy production of 50 kW is 22% more efficient in water desalination than a MED system with no ORC integration, requiring only 6.9% more heat transfer area;
- The greater the MED/ORC integration, the lesser the specific area, and the greater the MED's PR, because of the increase of water vapor produced in the first effect;
- If the goal is to transfer the largest possible amount of heat to the ORC through the proposed integration, the ORC evaporation temperature must be lower than the TBT.

Declaration of competing interest

We wish to confirm that there are no known conflicts of interest associated with this publication and there has been no significant financial support for this work that could have influenced its outcome.

We confirm that the manuscript has been read and approved by all named authors and that there are no other persons who satisfied the criteria for authorship but are not listed. We further confirm that the order of authors listed in the manuscript has been approved by all of us.

We confirm that we have given due consideration to the protection of intellectual property associated with this work and that there are no impediments to publication, including the timing of publication, with respect to intellectual property. In so doing we confirm that we have followed the regulations of our institutions concerning intellectual property.

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B. Comparison of novel energy integrations between MED and ORC systems

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Comparison of novel energy integrations between MED and ORC systems

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Abstract

This paper presents a comparative study of novel energy integrations between MED and 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 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. Utilizing the latent heat of condensation of the ORC in configuration 4 implies increasing the condensation temperature and pressure of the ORC, thus decreasing its efficiency and increasing the energy required to provide the two services simultaneously.

Keywords: MED, ORC, desalination, electricity, energy integration.

1 Introduction

The simultaneous production of electric power and water desalination is an area that has become of great interest to the world 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 global problem that has taken on great importance among public and private institutions. The United Nations incorporates it in the Sustainable Development Goal 6, mentioning that this resource, in sufficient quantity and quality, is essential for all aspects of life [1]. On the other hand, the Sustainable Development Goal 7 claim 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 the year 2030 at the current rate of growth [2]. Therefore, new, more efficient and cheaper technologies should be proposed, seeking a sustainable development in the generation of electric power and water desalination.

The hybridization of desalination technologies with power cycles for the simultaneous production of fresh water and electricity has been of great interest in recent years in the scientific community. [3,4]. 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 the electrical energy. Examples of these widely studied integrations are those made between Multiple Effect Desalination systems (MED) [5], Multi-Stage Flash (MSF) [6] and Reverse Osmosis (RO) [7] with Rankine cycles, as well as with Brayton cycles [8] and Organic Rankine cycles [9]. In the article by Omar et al [10] the couplings of solar thermal electric power generation systems with desalination systems were reviewed, highlighting 5 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 [11] studied the effect of using different pressures in the activation steam 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 5th effect of the MED, which allows 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 & Ameri [12] 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, hybridization of MED systems with ORC can be a technically attractive option. ORC 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 [13]. Also, by using organic fluids as working fluids, with temperatures and evaporation pressures lower than those of water, it can be integrated with different technologies to take advantage of energy effluents.

Baccioli et al [14] 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; in the first one, part of the ORC condensation heat is used as preheating of the MED, while in the second one 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 utilize both sensible and latent heat, which causes the efficiency of the ORC to decrease. The same effect of decreasing ORC efficiency by utilizing its condensation heat in MED systems can be seen in several studies. Kerme et al [15] proposed a solar polygeneration system and they found that by increasing the condensing temperature from 75 °C 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%. It has also been studied to use gas turbine combustion gases to first generate steam for the activation of MED systems and, subsequently, to generate more electrical energy through ORC systems.

[16,17]. This configuration does not modify the operating conditions of the systems, but it also does not make use of effluent between the MED and the ORC.

The works 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. 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 whole 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 in [18] 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 show 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.

2 Description of configurations

2.1 Configuration 1: Independent MED and ORC

In order to have an adequate point of comparison, the MED and ORC systems were incorporated into the study operating independently, without energy integration between them, as shown in Fig. 1. The ORC has a basic configuration, where the organic fluid is found as saturated steam at the outlet of the evaporator and is taken to the expander to take advantage of the mechanical energy produced by the decrease in its pressure. The same fluid, in the superheating phase, is condensed to saturated liquid so that the pump raises its pressure to the operating pressure in the evaporator

and, to continue with the production of electrical energy, evaporates it again by means of the external energy source, repeating the cycle.

The seawater desalination system consists of a 14-stage MED desalination plant, where seawater is used to condense the steam generated in the last effect of the MED and take advantage of its latent heat to reduce heating requirements. Then, a fraction of that stream is used as feed water to the system and preheated using a portion of the latent heat of condensation of the steam generated in each of the effects. The feed water, at a temperature close to the top brine temperature (TBT) of the first effect, is heated in evaporator 1 with the external energy source to a temperature higher than the operating temperature of the first effect. In effect 1, the operating pressure is lower than atmospheric pressure, so that the feed stream, upon entering it, expands suddenly and a part of it turns into steam, both by flashing and evaporation phenomena. The part of the steam corresponding to flashing is used as preheating of the feed seawater in preheater 1. The remaining steam is used for the activation of the subsequent effects, where it delivers the heat of condensation to the brine coming from the previous effect, now at a lower pressure and saturation temperature, so that an adequate heat transfer can take place, repeating the process effect after effect. In the last effect there is no preheating of the feed water, since the approach temperatures between the streams involved should be very small to achieve an effective heat transfer, so this equipment is disregarded and both the steam produced by flashing and evaporation in this last separator is entered to the final condenser, where seawater is used to condense the final steam. The total distillate is equivalent to the sum of all the condensates from the MED evaporators.

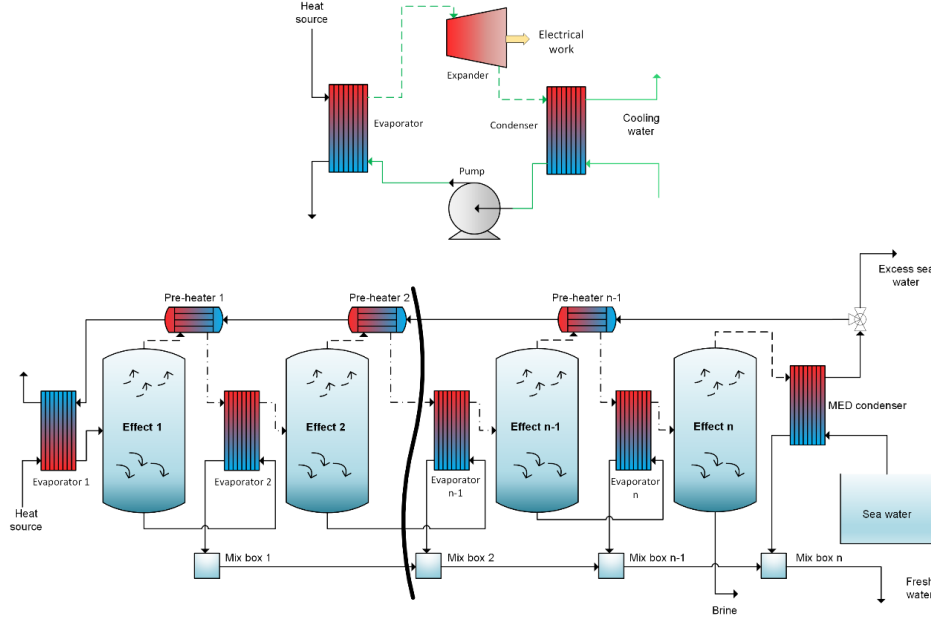


Fig. 1. Configuration 1: Independent MED and ORC.

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 seen in Fig. 2 and was technically analyzed in the work of Aguilar-Jiménez et al [18].

In order to carry out the proposed integration, the ORC must limit its evaporation temperature to one below the TBT of the MED, so it must be ensured that the evaporation pressure of the ORC, $P_{eva,ORC}$, is a value such that $T_{eva,ORC} < TBT$, by contemplating a temperature difference that allows an effective heat exchange. With this, the organic fluid can be brought to saturated vapor inside the ORC evaporator and can take advantage of the latent heat of condensation of the first MED effect to change phase. Otherwise, if $T_{eva,ORC} > TBT$, the usable latent heat of the MED will

serve only as a source of sensible heat for the organic fluid, which is a small fraction compared to its latent heat, having to incorporate an extra evaporator for its phase change.

The advantage of the energy integration of the MED and ORC subsystems is that the heat required for ORC activation, when supplied together with the MED activation heat in evaporator 1, is in turn used to generate an extra amount of steam in the first effect, which is converted into product water and increases the final distillate production.

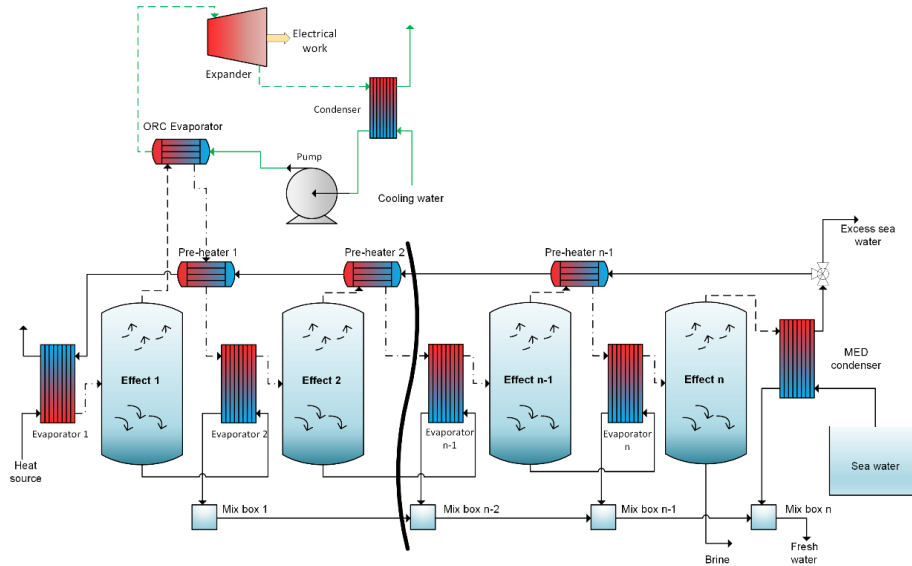


Fig. 2. Configuration 2: MED/ORC integrated in the first effect.

2.3 Configuration 3: MED and ORC with first effect integration and sensible heat utilization

Since the 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 approach temperatures reach the established limit and, subsequently, condense the organic fluid in its entirety with external cooling water. This integration can be seen in Fig. 3. The ORC operating conditions, both temperature and pressure, will remain unchanged from those of configuration 2.

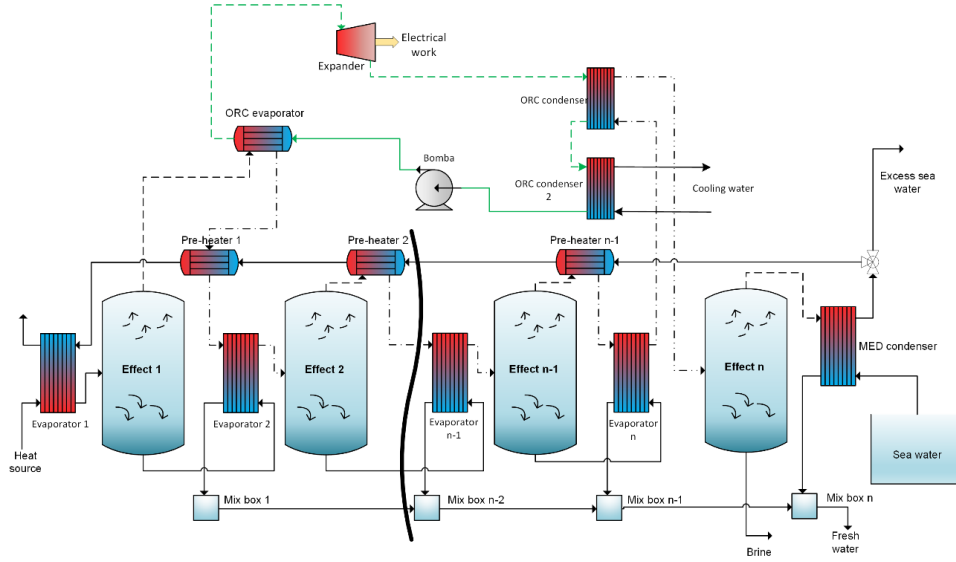


Fig. 3. Configuration 3: MED-ORC integration in the first effect and use of sensible heat from the ORC condenser.

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 first sensible heat and then the corresponding part of latent heat. In 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 the one shown in Fig. 3, only that the ORC operating conditions will be different 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, i.e., the ORC condensing temperature must be higher than the operating temperature of the last MED effect. The latent and sensible heat transfer will take place in the first condenser. However, depending on the electrical production capacity and efficiency of the ORC, it may be that 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.

3 Methodology

The Aspen Plus software was used to perform the detailed analysis of the MED/ORC configurations and to check their technical feasibility under different operating conditions. This software allows simulation studies to be carried out in a sequential-modular approach, using modules programmed with the mathematical models of a large amount of equipment, including heat exchangers, separators, pumps and expanders, to mention a few of those involved in the proposed system. By having an extensive database of thermodynamic and heat transfer properties of a wide variety of fluids and compounds, it becomes a very useful tool for the simulation and analysis of energy systems, such as power and desalination cycles. For the 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 mass flow provided by the ORC pump to ensure saturated steam conditions at 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 to the first effect and the inlet brine to each of the subsequent effects decreases its pressure inside the corresponding evaporator.
- The flashed steam in the mix-boxes is neglected.
- For the evaluation and comparison of the performance of the water desalination subsystem and the electric power generation subsystem, the following indicators are used. The MED subsystem is evaluated with the Performance Ratio (PR), which is the ratio of the final distillate over the thermal energy supplied to the subsystem, all multiplied by the latent heat of vaporization of water at 73 °C, expressed as follows [19]:

$$PR = \frac{\dot{m}_d}{\dot{Q}_{in,MED}} \times \frac{2,326 \text{ kJ}}{1 \text{ kg}} \quad (1)$$

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 power generation subsystem is evaluated based on the thermal efficiency of the thermodynamic cycle:

$$\eta_t = \frac{\dot{W}_{exp} - \dot{W}_{pump}}{\dot{Q}_{in,ORC}} \quad (2)$$

where η_t is the ORC thermal efficiency, $\dot{W}_{exp}$ the electrical work produced by the expander, $\dot{W}_{pump}$ the pump electrical consumption, and $\dot{Q}_{in,ORC}$ the portion of energy used for ORC activation. Both $\dot{Q}_{in,MED}$ and $\dot{Q}_{in,ORC}$ are supplied simultaneously to evaporator 1.

The energy balance in the ORC evaporator, where the energy integration with the MED is performed, is calculated as follows:

$$\dot{Q}_{in,ORC} = \dot{m}_{ORC}(h_{out} - h_{in}) = \dot{m}_{v,1}(h_{v1,in} - h_{v1,out}) \quad (3)$$

where $\dot{m}_{ORC}$ is the ORC refrigerant mass flow rate, h_{out} and h_{in} their evaporator outlet and inlet enthalpies, respectively, $\dot{m}_{v,1}$ the mass flow rate of the steam produced in the first MED effect, and $h_{v1,in}$ and $h_{v1,out}$ their inlet and outlet enthalpies, respectively. The flow diagram of this heat exchanger is shown in Fig. 4.

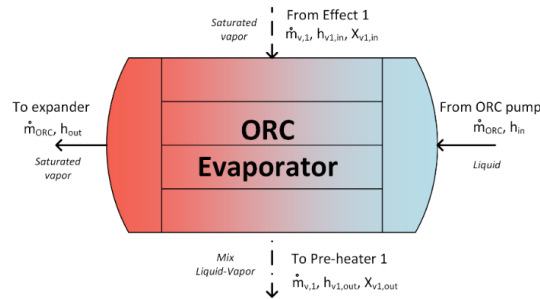


Fig. 4. Flow diagram of the ORC evaporator

3.1 Simulator validation

The validation of the MED/ORC system simulation was performed separately on: (1) the desalination process and (2) the electric power generation process. For part (1), modeled and experimental results presented by Palenzuela *et al* [20] were used, while for part (2) the results presented by Liu *et al* [21] were used. To carry out the validation, the individual simulators were adjusted to the type and operating conditions of the equipment used by the authors in their studies.

The MED system used for comparison is of 14 effects and 13 pre-heaters, with a steam mass flow for its activation of 295.92 kg/h at 70.8 °C, TBT of 68 °C, temperature of 35 °C in the last effect and seawater with a salt concentration of 35 g/kg at 25 °C. Table 1 and Fig. 5 show the results of the validation of the developed simulator in terms of performance indicators and steam production per effect.

Table 1. Validation of the seawater desalination process.

	Palenzuela <i>et al</i> model	Palenzuela <i>et al</i>	Simulation
Fresh water product (kg/h)	3,003	2,984	3,002
Seawater entering condenser	15,848	14,558	11,670
PR	10.2	9.7	10.14
Recovery Ratio (%)	37.6	37.5	37.5

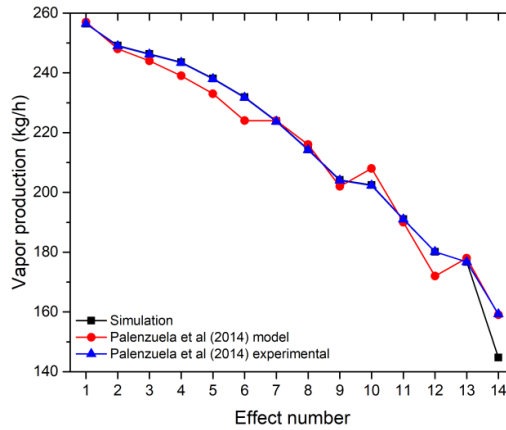


Fig. 5. Comparison of the developed MED simulator with respect to the theoretical and experimental results presented by Palenzuela *et al* [20].

For the validation of the ORC, an ideal basic cycle was used working with refrigerant R123, a condensing temperature of 30 °C and evaporating temperature increments in the range of 50-150 °C, as shown in Fig. 6.

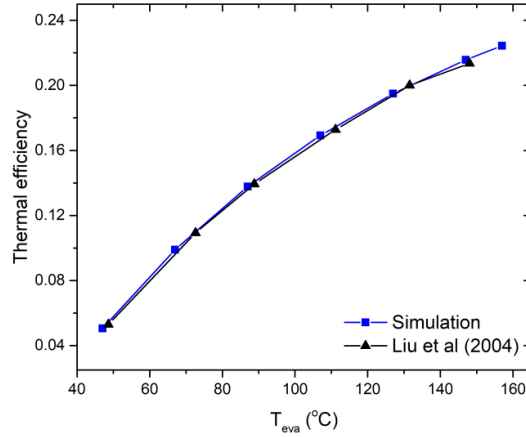


Fig. 6. Comparison of the ORC simulator with the results presented by Liu *et al* [21].

3.2 Heat transfer area analysis

To analyze the configurations from a broader perspective, the analysis of the heat transfer area required by the proposals' heat exchanger equipment was considered for comparison. To calculate the heat transfer area, Aspen Plus uses the Eq. (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 with the correlations presented in Xu *et al* [22] for the ORC equipment, while those used for the MED were the ones presented in El-Dessouky *et al* [23,24] and Palenzuela *et al* [25,26].

For the purpose of analyzing the heat transfer area of the ORC, the specific area will be used, calculated according to Eq. (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_{eva} , A_{cond} and A_{hx} are the heat transfer areas of the evaporator, condenser and heat exchangers for the ORC integration, respectively, and E_{ORC} the electrical power delivered by the ORC. For the specific area of the MED the Eq. (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 $m^2/kg/min$, $\sum A_{eff}$ and $\sum A_{ph}$ are the sum of the heat transfer area for the effects and preheating, respectively, A_c is the required area in the condenser, and $\dot{m}_d$ the mass flow of fresh water produced by the MED.

4 Results

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 2 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.

Table 2. Operating conditions of the MED-ORC configurations.

	Config.	Config. 2	Config. 3	Config. 4
ORC				
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
MED				
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)	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

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 take place. 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 is 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 temperature of 37.8 °C with the liquid-steam mixture coming from the MED, which is at 35 °C. This achieves the 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 of the configurations, due to the variation of the heat supplied to the MED, is presented in Figure 7. Configuration 4 has the highest total water production because it takes advantage of the benefits of configuration 2 together 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. There is a linear increasing trend for all the scenarios studied as the heat supplied

to the MED increases, and 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 are 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.

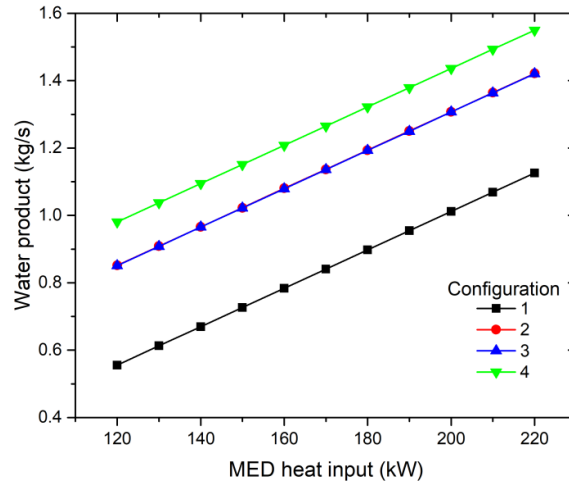


Fig. 7. Effect on desalinated water production by varying the heat supplied to the MED.

4.3 Brine concentration

The level of salt concentration in the brine, depending on the configuration and the amount of heat supplied to the MED, is shown in Figure 8. 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 antifouling chemicals, as well as the environmental impact when discharged into the marine ecosystem [27]. It is recommended to establish a maximum of 70,000 ppm of salt in the final brine to avoid problems, but there are large capacity systems with concentrations above 100,000 ppm, depending on the technology [28,29]. 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, steam production increases, so the final salt concentration shows the same increasing trend. The last two configurations have lower concentration levels than configuration 2 because they have a higher m_{fw} .

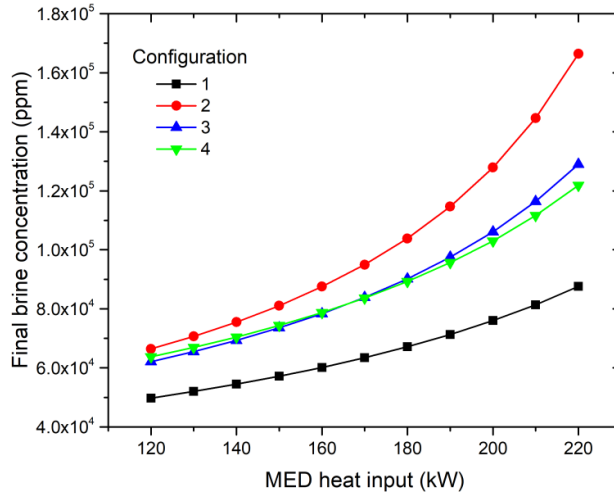


Fig. 8. Brine concentration variation as a function of heat supplied to the MED.

4.4 Configurations Performance Ratio

To compare the water production efficiency of the different configurations, Fig. 9 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 Eq. 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 Fig. 7 and Fig. 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 above, the MED-ORC integrations analyzed show a better use of energy, taking advantage of the power cycle effluents to increase desalination in the MED.

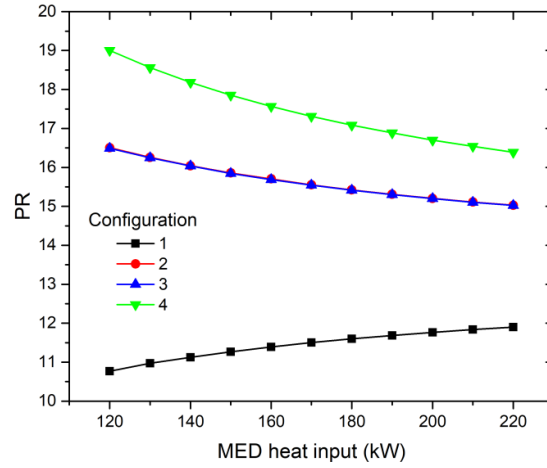


Fig. 9. Effect of varying the heat supplied to the MED on the PR of different configurations.

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 studies (see Fig. 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 thermoelectric conversion efficiency; the greater the capacity of the ORC, the greater the heat that can be used to enhance desalination in the MED. It should be noted that the condensation temperature of the ORC was raised to 37.7 °C in order to have an acceptable difference with the brine entering the last effect, which is 35 °C, and to be able to transfer the latent heat adequately. Fig. 10 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 m_{sw} value in Table 2, where configuration 4 requires a flow 77.7 %, 169.2 % and 127.6 % higher than configuration 1, 2 and 3, respectively.

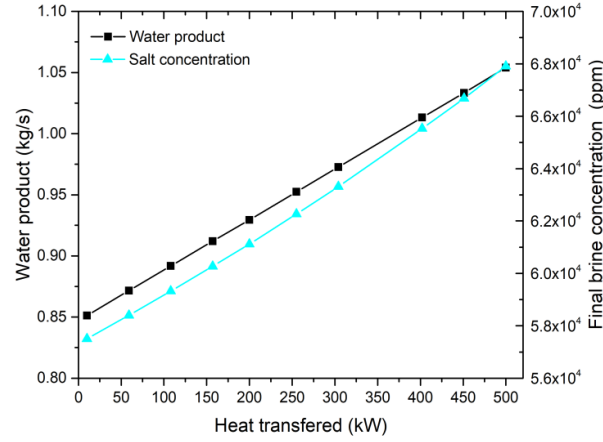


Fig. 10. Variation of water production and final salt concentration of Configuration 4 with respect to heat transferred in the ORC condenser.

To analyze the performance of the MED as a function of the heat transferred in configuration 4, Fig. 11 shows the effect on the PR and the fraction of steam leaving the liquid-steam mixture. The trend is the same as for water production, since as the heat transfer increases the PR and the amount of steam generated increases in the same way. This is because the more heat 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 of 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.

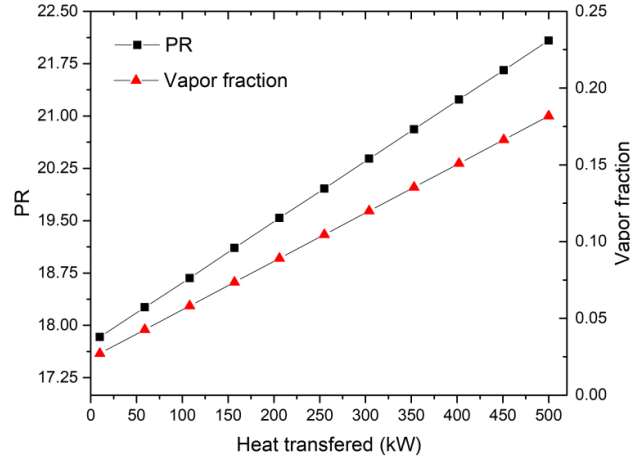


Fig. 11. Effect of heat transferred in the ORC condenser of configuration 4 on the PR and fraction of steam returning to the MED of the liquid-steam mixture.

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 3. In relation to desalinated water production and energy requirements for desalination, configuration 4 presents the highest values, while configuration 1 is the 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 1,016.6 kW to provide electric power and desalination services. This is mainly due to the decrease in 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 °C 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 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.

Table 3. 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

4.7 Heat transfer area comparison

For comparison purposes from the point of view 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 12 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, 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 m², 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 due to 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, since to generate the same amount of water as the previous configurations it requires less exchange energy between the equipment.

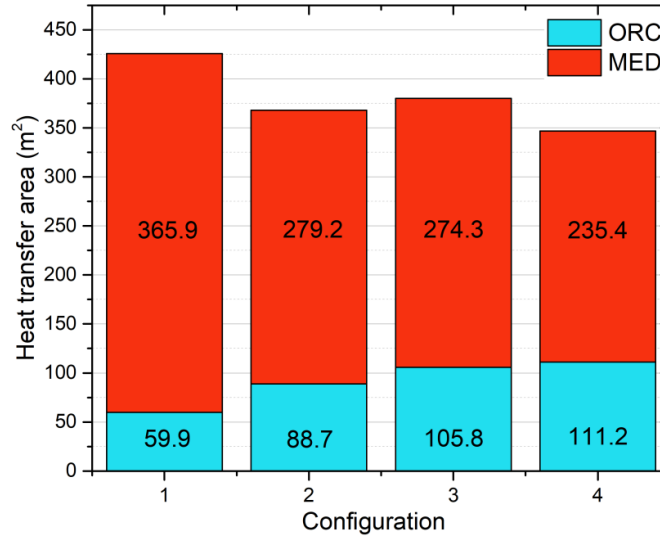


Fig. 12. Total and individual heat transfer area for the four configurations.

To encompass the electrical power and water desalination with respect to the heat transfer area requirements, Fig. 13 presents the specific areas for the ORC and MED of each of the configurations, calculated according to Eq. (5) and Eq. (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 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.

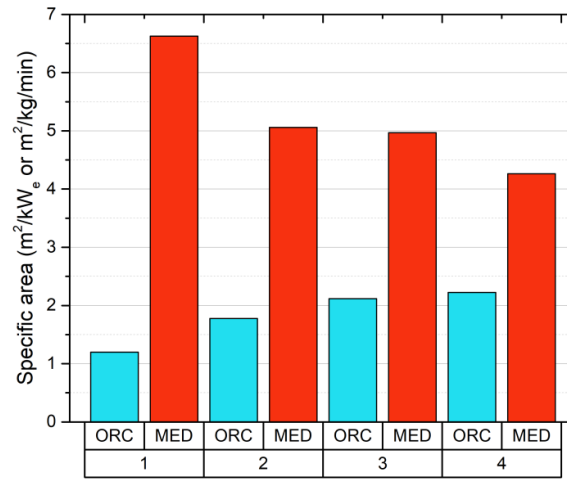


Fig. 13. Individual specific heat transfer area for the four configurations.

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 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, directly impacting the initial investment cost, so it is necessary to conduct more detailed studies on the proposed configurations and determine both their technical and economic feasibility.

5 Conclusions

The 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 activation thermal 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 °C 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.

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Preface

This CCIS volume presents selected articles from the II Ibero-American Congress of Smart Cities (ICSC-CITIES 2019), held in Soria, Spain, during October 7–9, 2019.

Smart Cities are the result of the increasingly urgent need to orient our lives towards sustainability. Therefore, these cities use infrastructure, innovation, and technology to reduce energy consumption and CO₂ emissions, in order to improve the quality of life of their citizens.

Being a strategic issue that brings new challenges, ICSC-CITIES 2019 was a forum for discussion with the main goal of creating synergies among different research groups to favor the development of Smart Cities and contribute to their knowledge and integration in different scenarios, their possible development, and the strategies to address them.

Subject areas defined by the Steering Committee of ICCS-CITIES 2019 included Energy Efficiency and Sustainability; Infrastructures, Energy, and the Environment; Mobility and Internet of Things; and Governance and Citizenship.

ICSC-CITIES 2019 was organized by the CITIES network, and financed as part of the Ibero-American Program for Science and Technology Development (CYTED).

The conference received 98 submissions for peer review. The reviewing process followed a single-blind procedure using a panel of experts and external reviewers (outside the Program Committee). Each submission had an average of three independent reviews and each reviewer was assigned an average of two submissions. The best 21 articles were selected to be part of this CCIS volume.

Sergio Nesmachnow
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Potential for Thermal Water Desalination Using Microgrid and Solar Thermal Field Energy Surpluses in an Isolated Community

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Abstract. In this work, we present the study of seawater desalination potential using the energy surpluses of a microgrid based on renewable energies and a thermosolar absorption cooling system, installed in the isolated community of Puertecitos, Mexico and its primary school, respectively. Given the profile of electricity demand of the community in winter and the non-need for air conditioning, both systems can be used for the desalination of seawater, a resource greatly needed in the region because of the scarcity that is presented. Using the software TRNSYS and Aspen Plus, the simulation of the generating systems was carried out, activating a multiple-effect seawater desalination system during a typical week of February with measured data of electrical consumption. The results show that, with the energy available from both systems, it is possible to desalinate 2,500 kg/day of water with a thermal consumption of 25 kW, during 6 h daily operation. The electrical energy supplied by the microgrid contributes four times more to the desalination of water than the thermal solar field. With this production, it is possible to satisfy the basic requirements of hygiene, hydration and food for 25 people.

Keywords: Microgrid · Solar energy · Desalination · Isolated community

1 Introduction

Water scarcity is a global problem that has taken on great importance among public and private institutions. The United Nations incorporates it in the Sustainable Development Goal 6, mentioning that this resource, in sufficient quantity and quality, is essential for all aspects of life [1]. The decline in the availability of this resource is the result of several

factors, such as the increase in world population, living standards and the pollution of water deposits; projections estimate that they will reach critical levels within the first half of this century [2]. It is for this reason that the efforts of scientists should be directed towards solving this problem, proposing new technologies that are more efficient, use renewable energies and can be applied in regions where there is a shortage of water.

The only methods to increase water supply are through desalination and reuse [3]. Of these two, seawater desalination offers a virtually unlimited and consistent supply of high quality water [4]. Desalination uses large amounts of energy to remove pure water from a salt water source [5]. Kalogirou [6] estimated that the production of 1,000 m³ per day of fresh water requires 10,000 tons of oil per year. This is significant if the price of energy is expensive and populations with water needs cannot afford it.

The integration of desalination processes to solar concentration plants is today the best alternative to solve simultaneously the problems of water scarcity and exhaustion of fossil fuels [2]. Furthermore, if we relate that the regions where water is needed present a good solar resource and, in addition, can be found close to the sea, it becomes an attractive and sustainable combination. Using solar energy, the desalination process can be carried out in two ways, using electrical energy in reverse osmosis equipment, or thermally in multiple-stage flash (MSF), multiple-effect desalination (MED) or humidification/dehumidification systems [7, 8]. Although MSF systems predominate in the market [9], MED technology has improved and now has efficiency levels higher than MSF [10].

Numerous works have been carried out seeking to make desalination processes more efficient by using renewable energies as the source of activation, in which the studies carried out on the Almeria Solar Platform with a 14-effect solar MED system are highlighted [11–14]. López-Zavala *et al.* [15] proposed the coupling of an absorption cooling system to a desalination system through internal energy integration. With that configuration, the cooling system was 19.44% more efficient, while producing 838 L/day of fresh water with the same amount of energy supplied. Mata-Torres *et al.* [16] analyzed energetically the coupling of a MED system with a Rankine cycle for the simultaneous production of water and electrical energy, finding that largest MED plant sizes achieve the lowest unit exergy cost of electricity and water under all the conditions evaluated.

On the other hand, seawater desalination systems have been analyzed and implemented in isolated regions being activated with renewable energies [17–19]. Astolfi *et al.* [20] mention that solar power plants and desalination units can be integrated synergistically as long as their programming is optimized by an advanced energy management system. They also conclude that the combination of two solar plants, such as photovoltaic (PV) and thermal, allows to take advantage of the characteristics of both; the first for its low costs of electricity production during periods of high insolation, while thermal systems extend the hours of operation using thermal energy storage.

Puertecitos, Mexico, is a coastal community isolated from the national electricity grid that has generators based on renewable energy. However, in the region there is a serious problem with access to fresh water, a service that is in high demand among the population due to the low quality of life. For this reason, this paper presents a study of the potential for desalination of seawater in the community by means of a multi-effect desalination system, using the energy capacity during the winter season of the generators

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installed, a period in which they are partially or completely stopped. This is intended to satisfy the basic requirements of clean water to the maximum possible number of residents, improving their quality of life and taking advantage of the energy waste of the renewable energy systems of the community.

2 Renewable Energy Systems Installed in the Community

The community of Puertecitos is located in the state of Baja California, Mexico. ($30^{\circ}21'19.7''$ N, $114^{\circ}38'26.3''$ W) and it's a village isolated from the national power grid. It has a variable number of residents, mainly due to weather conditions, availability of public services and fishing seasons, the latter is the predominant work activity.

Since it does not have electricity service, the Autonomous University of Baja California designed and installed a microgrid based on renewable energy to meet basic requirements, such as conservation of perishable foods, public lighting and air conditioning of spaces [21]. This system has 55.2 kW of photovoltaic modules, a 5 kW wind turbine, a 75 kVA diesel generator and a 522 kWh lead-acid battery bank, as shown in Fig. 1.



Fig. 1. Microgrid of the community of Puertecitos, Mexico.

In the same community of Puertecitos there is a primary school for the children who live there. This consists of four classrooms, where one is dedicated to preschool. However, the school suffers from a high rate of academic desertion due to the factors mentioned above, together with the high environmental temperatures present in the summer, which make it impossible to give classes under suitable comfort conditions. Since they did not have air conditioning systems or the economic resources to acquire them, besides not being able to pay the price of electricity from the microgrid, an absorption cooling system activated by solar thermal energy was installed [22], seeking that the school has the necessary comfort conditions for students to carry out their activities properly, reducing academic desertion or eventually leaving the community for lack of

basic education. The system consists of a field of solar thermal collectors with 110.25 m^2 of caption area, which store thermal energy in a tank of 12 m^3 of capacity for subsequent use in an absorption water cooling machine of 35 kW, as shown in Fig. 2.



Fig. 2. Thermosolar absorption cooling system of 35 kW capacity installed in the primary school of Puertecitos, Mexico.

2.1 Operation in Winter Season

The climate in the community of Puertecitos differs greatly depending on the season. In the winters temperatures can reach close to 0°C , while summers record temperatures in the order of 45°C , with peaks of up to 50°C in the shade. This causes a very different profile in the demand for electricity between these seasons. In the summer, when there is a need for air conditioning due to high temperatures, community demand can be up to five times greater than expected in winter. When dimensioning a system for the production of electrical energy, the problem of determining the appropriate size of the plant occurs; if the community's demand for summer is taken as a reference, the plant will be oversized for the winter period; on the other hand, if it is dimensioned with the winter demand, it will be very limited for the electrical needs of the hot months. The sizing study must contemplate a cost-benefit analysis between demand and production and storage capacity, including different generators and energy storage systems.

Figure 3 shows the predicted electricity demand of the community studied, based mainly on a socioeconomic study carried out on the population and extrapolated to a total of 20 houses, providing enough energy to have an acceptable quality of life [21]. During the first months of the year there is a demand of no more than 15 kW, mainly for public lighting, refrigerators, domestic lighting and entertainment. However, as ambient temperatures rise, forecasted demand tends to rise to a peak of around 60 kW in August, mostly due to the use of air-conditioning equipment to deal with high temperatures. During the months of September to December demand returns to its lowest levels. It was

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decided to install a 60 kW electrical system based on renewable energy and the energy shortfall during the critical months was covered by the diesel generator. In this way, a greater amount of economic resource could be allocated to a larger energy storage system. This resulted in a shortage of electricity for a short period of time per year, remedied with conventional energy, while the remaining time there is an excess of energy production by photovoltaic and wind generators.

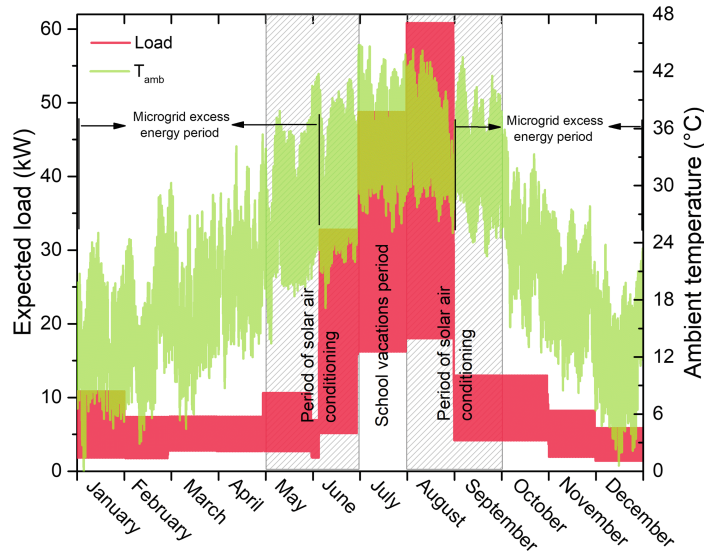


Fig. 3. Annual variation of the electric demand forecast for the community of Puertecitos and ambient temperature, where the periods of excess production of electric energy and use of the thermosolar absorption cooling system of the primary school are indicated.

On the other hand, the community primary school's thermosolar absorption cooling system has something similar to its operation. The need for air conditioning in the classrooms due to the region's high temperatures is present in the May-June and August-September periods, since July is considered a vacation period, as shown in Fig. 3, so the system will operate alone for four months of the year. During the remaining eight months there is no specific work defined, wasting the heating capacity of the solar field or the air conditioning of the absorption machine.

2.2 System for the Use of Energy Surpluses and Seawater Desalination

In the region there is a serious problem for the acquisition of clean water. The easiest way to obtain this resource is in a well 32 km away from the community, which is also overexploited and with high salinity indexes. In a direct questioning with the population of Puertecitos, they defined clean water as the second most desirable service in the community, only after electricity and even more desired than public health or safety services.

Taking into account the above, and seeking to take advantage of existing facilities in the community, such as the microgrid and field of solar thermal collectors, without affecting their main functions (meet the electricity needs and air conditioning of the classrooms of the school, respectively), it is proposed to evaluate the capacity of desalination of seawater in Puertecitos using surplus electricity from the microgrid and field of solar collectors during the periods mentioned above. In this way, the technical feasibility would be analyzed to provide this basic service so required in the region, without altering the current operation of the plants.

Figure 4 presents the schematic diagram of the proposed seawater desalination system in the community. During the months in which the energy storage batteries are at their maximum capacity, the instantaneous electric demand of the community is being satisfied and there is still plenty of energy, this will be used to activate electrical resistances inside the thermal energy storage tank (TEST) of the thermosolar cooling system, converting electrical energy into thermal energy and transferring it to the TEST. The field of solar thermal collectors, during the months when there is no need to cool the classrooms of the school, will be working to increase the thermal energy of the TEST. With the energy supplied by the two systems, a MED system will be activated.

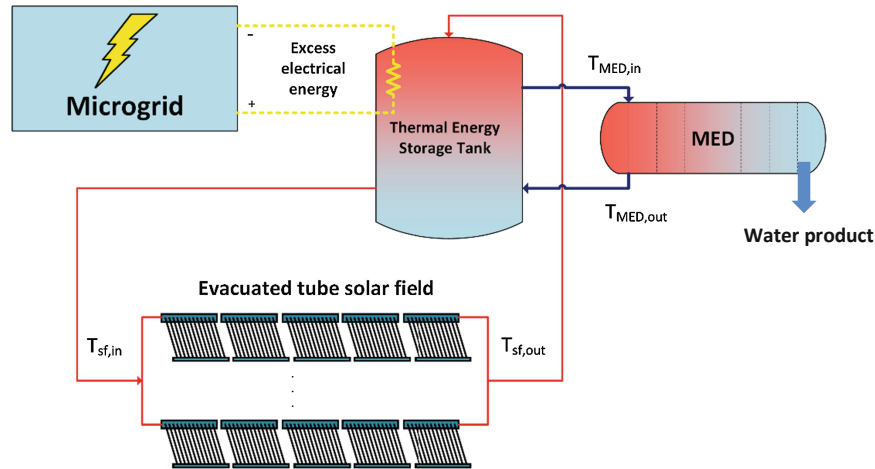


Fig. 4. Schematic diagram of the proposed system for the desalination of seawater using the surplus energy of the microgrid and thermosolar cooling of the community of Puertecitos.

3 Methodology

For the feasibility analysis of the proposal, the TRNSYS software was used to simulate the solar heating and microgrid systems, while Aspen Plus was used for the MED system. TRNSYS has a library of “types” programmed with mathematical models validated from a great amount of equipment, among them renewable technologies such as photovoltaic

modules and solar collectors, which, interconnected with other programmed control modules or other processes, form a system that can be analyzed in a semi dynamic state. The advantage of this software is that it can use meteorological databases typical of the locality of study, so the analysis is carried out under more real operating conditions, subject to environmental variations, such as temperature, radiation and wind speed, among others.

The microgrid, solar collectors' field and storage tank, as well as the dynamics of control and operation of the general system, were simulated in TRNSYS since their performance depends directly on solar radiation conditions and temperatures, in addition to being also used as references for decision making regarding the operation of the global system. The meteorological information used in the simulation study represents a typical year, based on average measurements over the last 10 years. On the other hand, the MED system is simulated in Aspen Plus software due to its advantages in terms of databases of thermodynamic properties and heat transfer of a large number of substances, which facilitate the resolution of mathematical models used in their blocks, representative of physical and chemical processes.

The characteristics of the systems installed in the Puertecitos community, as well as those of the proposed MED system, are presented in Table 1. This information is declared in the TRNSYS blocks to calculate the available heat for the activation of the desalination system. In the case of the microgrid, field of thermal solar collectors and TEST, the information supplied to the simulator is that provided by the manufacturer of the equipment, as well as it is considered that the mathematical models used in TRNSYS are validated. The MED system developed in Aspen Plus was based on the experimental results of a 14-effect plant and validated with experimental results presented by Palenzuela *et al.* [13] and can be found at [15]. It is important to mention that the absorption system is not in operation due to the temperatures present during the simulation period, so only the field of solar collectors was taken as a study system. Also, as previously mentioned, the excess of electrical energy produced by the microgrid was taken into account. The amount of available energy was quantified in-situ and declared as input values in the simulator developed in TRNSYS.

The following considerations were taken into account in the simulation study in order to simplify the mathematical resolution of the proposed system:

- Semi-dynamic state in TRNSYS with simulation periods of 25 min.
- No heat loss from the equipment or piping was considered, only the heat transfer to the environment due to the temperature difference of the TEST and the solar collector field.
- Pressure drops in equipment and pipes were not considered.
- Since there is no direct link between the TRNSYS software and Aspen Plus, they were analyzed separately.
- The MED system was dimensioned based on the maximum usable thermal load of the TEST, looking for an operation for defined periods of time.

Table 1. System characteristics

<i>Solar collector</i>	
Brand/model	Suntask/SHC24
Collector type	Evacuated tube with CPC reflector
Number of tubes	24
Aperture area	4.41 m ²
Optical efficiency (a_0)	0.668
First order efficiency coefficient (a_1)	1.496 W/m ² *K
Second order efficiency coefficient (a_2)	0.005 W/m ² *K ²
Fluid	Water
Mass flow	0.02 kg/s*m ²
Number in series	5
Number of loops	5
<i>Thermal energy storage tank</i>	
Volume	12 m ³
Height	2.5 m
Material	Fiberglass
Insulation thickness	0.025 m
Loss coefficient	1.4 W/m ² *K
Fluid	Water
<i>Microgrid system</i>	
Brand/model	Solartec S72MC6-300
Power	300 W
Number of modules	184
Wind generator	5 kW
Inverter	Kehua 100 kVA
Controller	Kehua SPC348150-M
Batteries	2 V, 1,500 Ah each
Number of batteries	172 connected in serie
Diesel generator	75 kVA
<i>Multiple-effect desalination system</i>	
Number of effects	14
Thermal energy input	25 kW
Top brine temperature	70 °C
Condensation temperature	35 °C
Fresh water production	2,490 kg/h

4 Results

In order to carry out the feasibility study of seawater desalination with the excess energy of the microgrid and the entire solar heater field, a week of February was considered (period of 750–885 h per year) where the environmental conditions of radiation and temperature allow a stable operation of the system, as can be seen in Fig. 5. As it is a winter week, the present temperature allows the air conditioning systems to be switched off, both the school's thermosolar absorption and the domestic electrical systems, which is why an energy surplus is expected from the microgrid and the availability of using the school's collector field for the desalination of seawater. However, the global solar radiation available does not exceed 700 W/m^2 and the hours of use of the resource are less than would be expected in summer, reducing the amount of solar energy that can be acquired during the day.

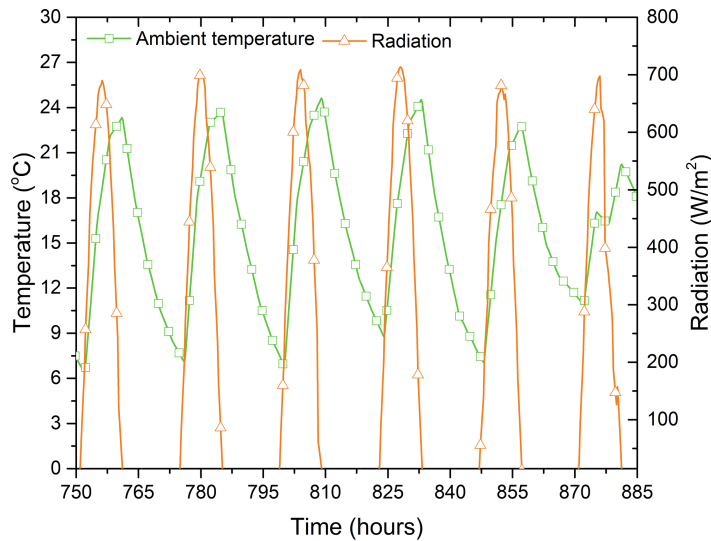


Fig. 5. Variation of ambient temperature and global radiation in the region of study during a week in February.

Figure 6 presents the operating dynamics of the Puertecitos community microgrid, as well as the electrical demand profile and the electrical energy surplus. This information corresponds to data measured in the community during the aforementioned analysis period. It can be seen that, during the whole week studied, there is a surplus of electrical energy due to low-demand. The first day begins with the state of charge (SOC) of the storage system in the order of 0.8, due to the discharge of the same during the night; as the solar resource increases and the production of the photovoltaic field satisfies the instantaneous demand of the community, the excess energy is used to charge the batteries to their maximum. When the storage system reaches a SOC of 1.0, the electrical surplus is used to activate the electrical resistances within the TEST.

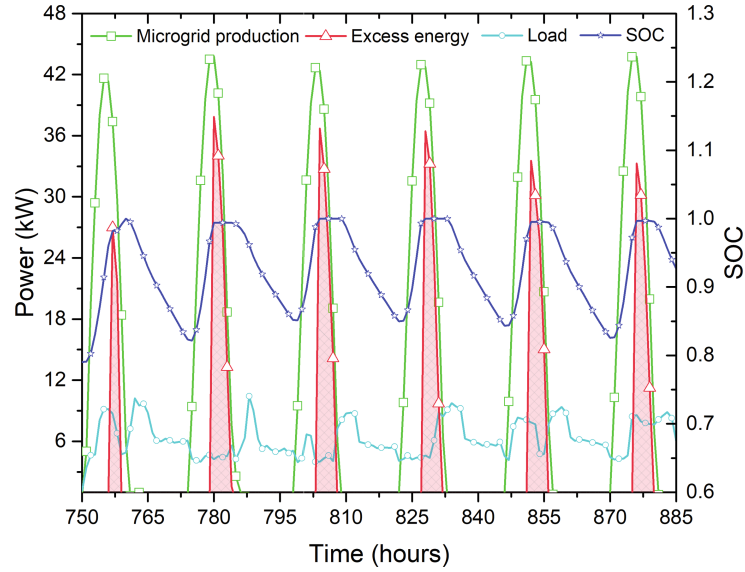


Fig. 6. Dynamics of electrical energy production by the microgrid, excess of energy due to the low demand of the Puertecitos community and SOC of the batteries of the system.

On the first day, due to the community's demand profile, a maximum of 27 kW of transferred energy was obtained for only a short period of time, giving a total of 49 kWh of harnessed energy. However, the rest of the days, after fully charging the battery bank, potencies in the order of 38 kW were reached, equivalent to an average daily energy of 96 kWh, the result of integrating the instantaneous power during the period of time presented, represented by the shaded area under the curve. It should be noted that, being winter season, the storage system in batteries does not fall below 0.8 of its SOC during the night period, allowing the use of excess energy for the desalination of seawater.

On the other hand, the field of thermal solar collectors of the cooling system is available in its entirety to be used as a medium for the activation of the MED system, together with the energy coming from the microgrid. Under the environmental conditions mentioned above, Fig. 7 shows the behavior of the solar thermal collection system during the analysis week. The output temperature of the solar field, $T_{out,sf}$, is directly influenced by the input temperature, $T_{in,sf}$, and the available solar radiation, and indirectly related to the temperature of the TEST. This can reach 95 °C during the solar mid-day, which is the set point defined as the maximum operating limit. The decrease in $T_{in,sf}$ is due to the fact that the solar heating system is not in operation for the night period. The thermal power supplied by the collector field increases with respect to the increased solar radiation, having a maximum of up to 35 kW at mid-day. This energy is transferred in its totality to the TEST to be used by the MED system.

Given the amount of energy available, both from the excesses of the microgrid and the collector field, Fig. 8 shows the production of clean water calculated by the proposed system. The MED will run for a period of 6 h daily, starting at 9:00 and ending at 15:00, as

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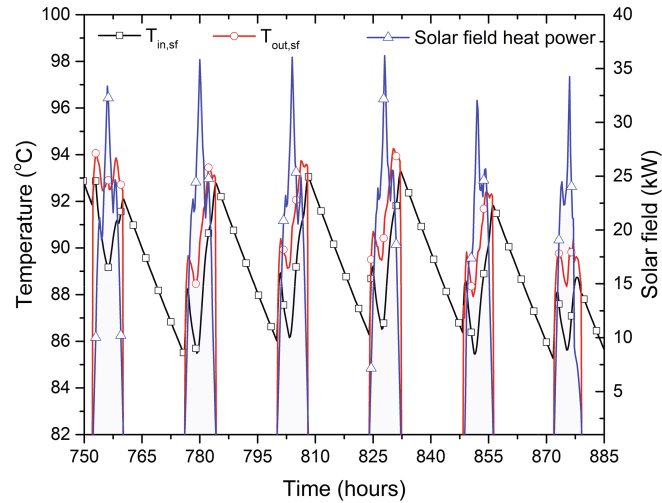


Fig. 7. Behavior of the solar thermal system of the Puertecitos school used in the seawater desalination system.

this allows a programmed operation throughout the week without affecting the operating temperature and TEST conditions, as well as using only the daily energy supplied by the generators. On the other hand, if you have a longer daily operation, the system will not be able to return to the initial conditions of temperature and stored energy of the TEST, affecting the quality of the energy and the subsequent days of operation. Using the energy of the two systems, it is possible to produce 414 kg/h of desalinated water during the 6 h a day, using 25 kW as thermal energy to activate the MED, giving a total of approximately 2,500 kg per day and a weekly accumulated of 17,500 kg.

It also presents the scenario in which only the solar field is available, without considering the electrical surpluses of the microgrid. For this case, during the same period of operation, it is possible to use a constant thermal power of 12 kW without negatively influencing the conditions of the TEST and, for a new capacity and design of the MED, it is capable of producing approximately 1,183 kg daily of desalinated water with a weekly accumulated of 8,305 kg. This results in a 52.5% reduction in water desalination when using only the thermal solar field, or seen in another way, the electrical excesses of the microgrid contribute two times more to the production of desalinated water than the entire solar collector field. Considering the recommendations of the World Health Organization [23], the quantity of 2,500 kg/day of desalinated water can satisfy the basic needs of food, hydration and hygiene of 25 residents of the community of Puertecitos, while using only the energy of the solar field satisfies the water requirements for 5 residents.

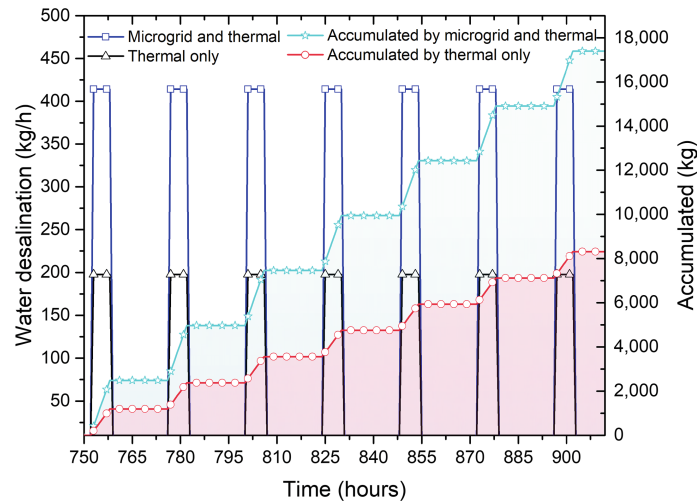


Fig. 8. Production rate of desalinated water and accumulated during the week of analysis.

5 Conclusions

A study was conducted to determine the potential for seawater desalination in the isolated community of Puertecitos, Mexico, using a multi-effect thermal desalination system to provide this scarce service in the region. This population has a microgrid based on renewable energy and, given that the profile of demand differs greatly depending on the seasonal season, we seek to take advantage of excess electricity in the winter period. In addition, a field of thermal solar collectors belonging to an absorption cooling system installed in the primary school of the community is available, which for eight months is without use because no air conditioning is needed in the classrooms.

Considering a week in February as the study period, the system was simulated using the TRNSYS and Aspen Plus software, based on the profile of electricity demand measured in the community, and the following was found:

- Using the energy surplus of the microgrid and the entire solar thermal field, it is possible to desalinate water using a MED system with a production of 2,500 kg/day.
- The thermal consumption of the MED system is 25 kW, being possible to maintain its operation for 6 h/day, during the period from 9:00 to 15:00.
- Using both generators, a weekly production of approximately 17,500 kg of water is obtained, while using only the field of solar collectors this decreases to 8,305 kg weekly.
- The excess energy of the microgrid contributes two times more to the desalination of water than the solar thermal field.
- According to the World Health Organization, it is possible to meet the basic hygiene, hydration and nutritional needs of 25 people with calculated water production.

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**D. Optimum operational strategies for a solar
absorption cooling system in an isolated
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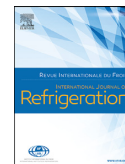
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Optimum operational strategies for a solar absorption cooling system in an isolated school of Mexico



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ABSTRACT

In this work, we present the simulation study of a LiBr-H₂O solar thermal absorption air conditioning system, installed in an isolated primary school in the community of Puertecitos, Mexico. The collection system consists of 110 m² of evacuated tube solar collectors that produce the thermal energy needed to activate the chiller, while four cooling coils of 8.75 kW each cool the classrooms of the school with the chilled water produced. The characteristics of the system's equipment are used and a simulator is developed to determine the optimum operational strategies, given the conditions to which it will be subjected before the start-up of the plant. This is intended to ensure the comfort conditions of the students within the classrooms during the weekly class periods. The results show that the installed system is capable of satisfying the cooling needs of the school with the simulated strategies, but for a stable operation, periods of time must be dedicated for the recovery of energy inside the thermal storage tank. Some experiences are mentioned, both of the start-up and economic, related to an installation in an isolated community. The cost of the balance of plant was the highest with 36% of the total project, followed by the chiller and the solar field with 25 and 18%, respectively. The preliminary experimental results obtained during the implementation of the system are presented, where some of the operational strategies resulting from the simulation study can be confirmed.

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Stratégies de fonctionnement optimales pour un système de refroidissement à absorption solaire dans une école isolée de Mexico

Mots-clés: Conditionnement d'air; Communauté isolée; Énergie renouvelable; Sorption solaire

1. Introduction

The extreme temperatures that occur in regions around the world, which have been aggravated due to climate change, make it a top-level need to have air conditioning (AC) equipment for spaces, both for cooling and heating. 72% increases in globally energy demand are forecast for the year 2100 and a 34% decrease for heating, mainly due to climate change (Isaac and van Vuuren, 2009), so we must opt for efficient and non-polluting

technologies. The use of renewable energies as an external energy source for activating AC systems, especially solar energy, is a suitable option for the cooling of spaces, as typically energy consumption due to AC equipment increases when solar radiation levels are high (Chua et al., 2013) being able to take advantage of the coincidence of the abundance of the resource.

Solar energy can be used in two different ways for the AC of spaces, namely solar photovoltaic (PV)-based electrical cooling systems and collector-based thermal cooling systems (Huang and Zheng, 2018), which in turn can be subdivided into solar PV cooling, solar thermo-electrical cooling, solar thermo-mechanical cooling, and solar thermal cooling (Sarbu and Sebarchievici, 2013). The

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Nomenclature

Symbols

T	Temperature (°C)
a_0	Collector intercept (optical) efficiency (-)
a_1	Collector 1st order loss coefficient ($Wm^{-2}K^{-1}$)
a_2	Collector 2nd order loss coefficient ($W/m^{-2}K^{-2}$)
$\dot{Q}$	Heat rate (kW)
$\dot{m}$	Mass flow ($kg\ s^{-1}$)
G	Incident Solar Radiation ($W\ m^{-2}$)

Subscripts

amb	Ambient
in	Inlet
out	Outlet
cw	Cooling water
w	Water
hw	Hot water
sf	Solar field
chw	Chilled water

Greek

η	Efficiency (%)
PV	Photovoltaic
AC	Air conditioning
CC	Cooling coils
TEST	Thermal energy storage tank
TMY	Typical meteorological year
COP	Coefficient of Performance
RH	Relative Humidity
USD	United States dollar
BoP	Balance of plant

use of PV modules for the activation of AC technologies can be very appropriate since cooling equipment with high technological maturity and a great penetration in the world market would be used. However, these systems need a high amount of electrical energy for their operation which must be satisfied with the PV modules, in addition to needing storage of energy by means of batteries for night and low solar radiation periods (Zeyghami et al., 2015), increasing the system cost. Currently, these types of systems are no more economical than other solar cooling equipment, but the large decrease in prices of PV technologies in recent years and those predicted for the following ones, is making electric AC solar systems competitive with other technologies (Ge et al., 2018; Lazzarin and Noro, 2018). Also, solar photovoltaic refrigeration systems have mechanical components (compressor) that use greenhouse gasses, which are dangerous for the environment.

On the other hand, the AC systems activated with thermal energy are very attractive in applications where access to electric power is null or limited, when there is a waste heat available (Ustaoglu et al., 2017) or in a cogenerative and trigenerative systems making an energy integration for generation of cooling, heat, power (Bellos and Tzivanidis, 2018) and/or desalination (López-Zavala et al., 2019), to mention some. Within these thermally activated technologies, sorption cooling is dominated and contain both absorption and adsorption (Deng et al., 2011). Absorption cooling cycles are the most popular option to take advantage of thermal energy with low temperature, being able to generate cold with temperatures from 75 °C (Xu and Wang, 2018). This opens the possibility of using this cooling technology in many applications, such as the use of waste heat in industries and businesses, as well as being able to be activated with renewable energies such as solar thermal energy, biogas and geothermal. In addition, it has great

technological maturity, economic profitability, and high efficiency (Shirazi et al., 2018). Numerous studies mention aspects of this type of AC systems using solar energy, where they emphasize the impact of the climate where they are installed (Al-Ugla et al., 2016; Bellos and Tzivanidis, 2017), the lack of economic competitiveness with vapor compression technologies (Gabbrielli et al., 2016; Li and Sumathy, 2000) and can be used in isolated applications (Ammari et al., 2018).

Ammari et al. (2018) carried out the simulation of an autonomous absorption cooling system activated by solar energy operating in a residential building located in Batna, Algeria without any auxiliary heating equipment. Using the software TRNSYS and EES they developed the dynamic model of the cooling system and analyzed the behavior under the climatic conditions of the region. They proved that cold energy storage allows continuous operation of the system even when the solar resource is insufficient. A large number of theoretical studies have used simulation tools to analyze solar-activated absorption chillers in different applications, such as agro-industrial processes (Cascetta et al., 2018), education buildings (Khan et al., 2018), indoor air quality control for human health (Yu et al., 2009) and other innovative scenarios (Montagnino, 2017). Regarding experimental studies, Agyenim et al. (2010) developed and tested a thermosolar absorption cooling system at Cardiff University. They mention that this type of projects doesn't compete economically with mechanical compression cooling systems due to the strong initial investment, but they propose that they also be used as domestic hot water generators and heating in winter periods, seeking to be more commercially competitive. On the other hand, Francisco et al. (2002) developed a low-capacity thermosolar absorption cooling prototype to be used in isolated regions with high levels of solar radiation.

It is very clear that the technical aspects of this technology are at a mature stage, where its operational feasibility is more than proven, but they are still systems with a high cost to be considered an attractive option for the air conditioning of spaces. However, there are applications where social welfare is sought and the economic aspect is not a representative factor, as are rural communities isolated from the electricity transmission network, which have a low quality of life and this type of projects impact positively in their daily activities. This work presents the simulation and acquired experiences of an absorption cooling system activated with solar thermal energy that was implemented in an elementary school in the isolated community of Puertecitos, Baja California, Mexico, seeking to generate practical knowledge that implies the development of this type of projects in regions with extreme climates and difficulty in obtaining freshwater and electricity services on a continuous basis. The technical and economic feasibility of this type of systems depends to an adequate operative strategy, for which the system simulation was carried out to determine the strategies to satisfy the cooling needs under the environmental conditions of the locality, as well as subject to the class periods of a typical elementary school in Mexico. Considering the application and context of the project (school in a community isolated from the electricity grid), it is extremely important to perform preliminary operational studies to avoid problems in the operation of the system in general. Therefore, and seeking to have a system as robust as possible, this simulation study was conducted as a first stage before experimentation and completion of the project.

2. Project background and system description

The community of Puertecitos, Baja California, Mexico (30°21'19.7" N, 114°38'26.3" W) is a fishing village with a high rate of social marginalization in which around 20 families of low economic resources live. Given its remoteness to the electric



Fig. 1. Evacuated tube solar collector field.

transmission network, this community did not have electrical service, which was very important to resist the high environmental temperatures of up to 50°C in the summer. A renewable energy-based microgrid was installed with an energy storage system seeking to satisfy the basic needs of this population, such as AC and food conservation, as well as promoting economic activities for the benefit of the community (Aguilar-Jiménez et al., 2018). However, the children who attended the elementary school of the town suffered the high temperatures during the hot summers since they did not have AC systems or the economic resource for the acquisition of them, besides not being able to pay the price of the electric energy coming from the microgrid. This meant that students stopped attending or had to travel long distances daily to take courses in other schools where they did have adequate study conditions. In view of this problem of air conditioning of spaces and the high energy consumption of current technologies, the Autonomous University of Baja California, in collaboration with the Mexican Center for Innovation in Solar Energy, implemented absorption cooling technologies activated with solar thermal energy in order to evaluate their technical and economic viability (Aguilar-Jiménez et al., 2019). This type of systems can be used in different scenarios, including applications in isolated communities, so a first plant was installed in the elementary school of Puertecitos during 2018. It also seeks to demonstrate that solar thermal absorption cooling systems can be installed in regions with extreme environmental and operational conditions, resulting in an improvement in the quality of life of marginalized populations, using the renewable resources present in the region.

The driving force of this system is a field of solar thermal collectors, as shown in Fig. 1, which delivers the thermal energy necessary for the operation of the absorption chiller. The solar collectors are of the evacuated tube type with a parabolic reflector in the lower part and are connected in five parallel arrays with five collectors in series each, giving a total of 110.25 m² of solar collection area. The general characteristics of this collector are presented in Table 1.

It also has a thermal storage tank of 12 m³ to contain the hot fluid coming from the solar field and to supply the necessary en-

ergy to the cooling system in periods of zero or low solar radiation. The absorption cooling machine that was acquired is of simple effect and uses the LiBr-H₂O mixture as working fluid. The cooling capacity is 35 kW and can be activated in a temperature range of 75–95°C, suitable to be coupled to solar collectors. The cooling medium of the machine has an evaporative cooling tower and tanks for the replacement water, very necessary to store as much as possible due to the great shortage of this resource in the region. These components can be seen in Fig. 2, as well as their general characteristics in Tables 1 and 2.

The cooling system is designed to meet the AC requirements of three classrooms of elementary school and one of the kindergarten, where each space to be air-conditioned has a cooling coil of 8.75 kW capacity. The installation of the solar thermal cooling system in the school can be seen in Fig. 3.

The schematic diagram of the components and connections is shown in Fig. 4. The system can be divided into three main circuits: hot water circuit, which has the function of supplying the necessary thermal energy to the absorption machine; cold water circuit, whose function is to remove heat and air-condition the classrooms of the school; and the cooling water circuit, responsible for removing heat from the chiller entered by the two previous circuits. The solar thermal field is responsible for maintaining the fluid temperature of the storage tank in a range of 75–96°C, enough to have an adequate operation of the chiller. Regardless of whether the cooling machine is operating or not (for example, on weekends the chiller will not operate because there are no classes), when there is enough solar resource, the solar field pump will continue to supply fluid for heating until the tank is completely at the set point temperature, in this case 96°C, always looking to have the system in operating conditions. When there is a need for AC in the classrooms, the chiller will turn on and the storage tank pump will supply hot water from the top. Once the temperature of the generator of the absorption machine is in optimal operating conditions, the pump of the cold water circuit will be turned on to take it to the cooling coils of the classrooms, at a temperature between 7–10°C, and to remove the heat of the spaces to be cooled. This flow is taken to the chiller to be cooled again and enter the buffer.

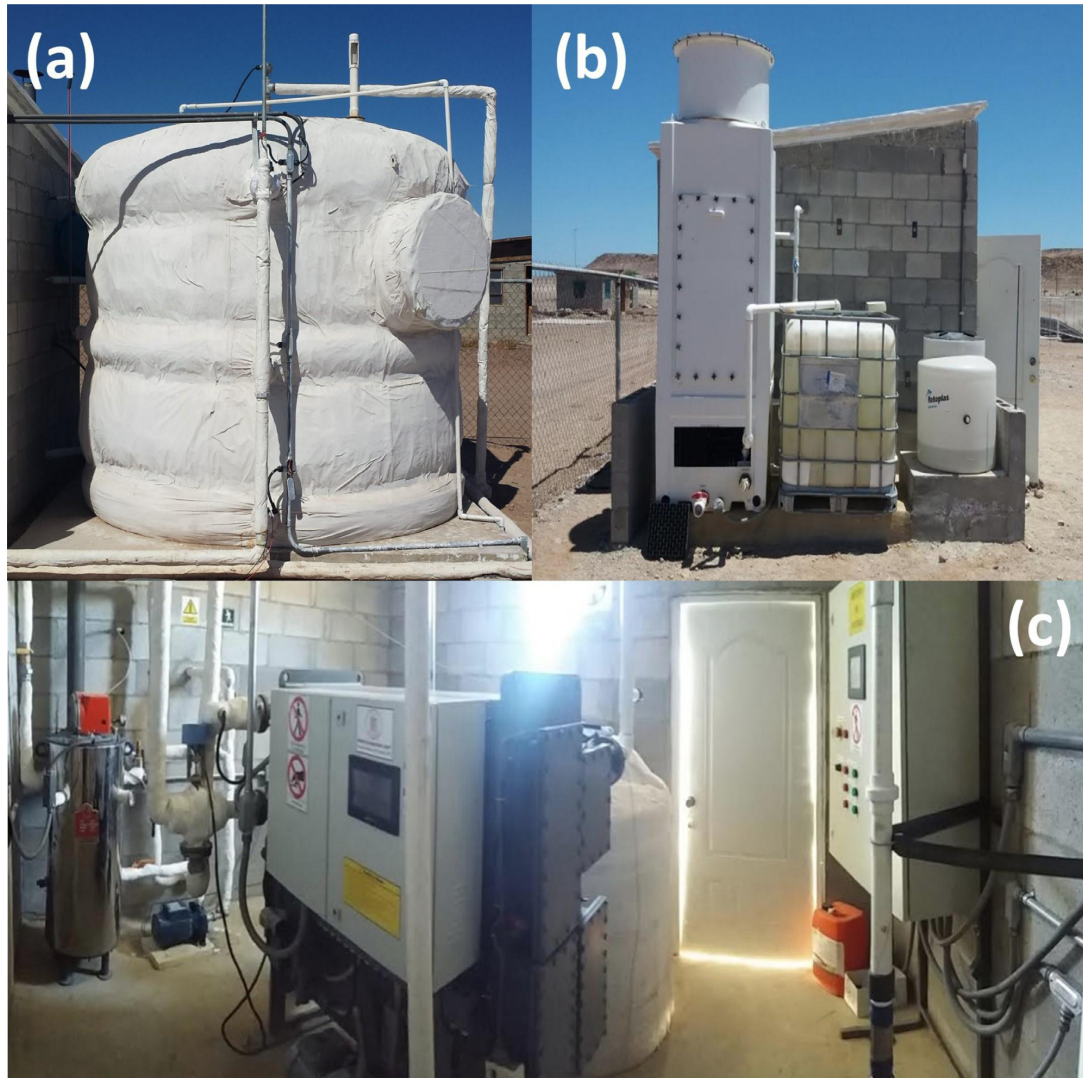


Fig. 2. Components of the thermosolar absorption cooling system of the Puertecitos school, where (a) thermal energy storage tank, (b) cooling tower and (c) auxiliary heater and chiller inside the machine room are shown.

The heat supplied to the system through the hot water and cold water circuits is removed by the cooling water circuit and released to the environment in the evaporative cooling tower.

In this project, the main limitation was the economic budget available for system engineering and equipment purchase. The total cost of the thermosolar absorption cooling system was \$80,000 USD, equivalent to \$2285 USD kW⁻¹ of cooling, which is a high specific cost compared to mechanical compression technologies (SHC, 2016). Fig. 5 shows the percentage of the total cost of equipment and engineering that was needed for this project. It is worth mentioning that the logistics and costs of the different items for the implementation of this type of project (application in an elementary school in an isolated community) is very different from the one that would be carried out in a facility where there are all services, mainly electric power and availability of machinery and materials (Montagnino, 2017). The balance of plant (BoP) was the

most expensive aspect in the project with 36% of the total, equivalent to \$28,560 USD. This includes the cost of materials for installation, labor, transportation of equipment, pumps and commissioning. The absorption cooling machine corresponds to the most expensive equipment in the system with a price of \$19,850 USD, equivalent to \$567 USD kW⁻¹ of cooling. It is worth mentioning that the selected chiller has a high specific price due to the low cooling capacity of this model. However, the scaling factor in this type of machines causes a considerable decrease in costs. The solar collector's field represents a quarter of the total cost of the project, equivalent to \$134 USD/m². The equipment manufactured and purchased in Mexico has a low cost, such as the cooling tower, storage tank, cooling coils and auxiliary heater.

The available electrical energy is limited by the generation of the community microgrid, which in the end must be paid at the price fixed to the school. In total there are four pumps that need to

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Table 1
Solar field and Thermal energy storage characteristics.

Solar collector	
Brand/model	Suntask/SHC24
Collector type	Evacuated tube with CPC reflector
Number of collectors	25 (24 evacuated tubes per collector)
Aperture area	4.41 m ²
Optical efficiency (a_0)	0.668
First order efficiency coefficient (a_1)	1.496 Wm ⁻² K ⁻¹
Second order efficiency coefficient (a_2)	0.005 Wm ⁻² K ⁻²
Fluid	Water
Mass flow	0.02 kg sec ⁻¹ m ⁻²
Number in series	5
Number of loops	5
Thermal energy storage tank	
Volume	12 m ³
Height	2.5 m
Material	Fiberglass
Insulation thickness	0.025 m
Loss coefficient	1.4 Wm ⁻² K ⁻¹
Fluid	Water

Table 2
Absorption chiller characteristics.

Model	Lucy New Energy/RXZ-35
Refrigerant	Water-Lithium Bromide
Cooling capacity	35 kW
COP	0.7
Hot water nominal temperature	90 °C
Hot water nominal flow rate	8.3 m ³ hr ⁻¹
Chilled water nominal inlet temperature	15 °C
Chilled water nominal flow rate	6 m ³ hr ⁻¹
Cooling water nominal temperature	30 °C
Cooling water nominal flow rate	15 m ³ hr ⁻¹
Power consumption	0.3 kW

be activated electrically and also have to add loads of the cooling tower fans and cooling coils, being, in the end, an electrical power of at least 6.71 kW. Being an operative study, these loads were not considered in the analysis.

The availability of drinking water in the community is critical, the nearest point to obtain it is in a well 32 km away, so it becomes a resource much appreciated and cared for among the population. The cooling tower is indispensable for the operation of the chiller and represents a considerable consumption of water, and its supply may be a problem. The amount of water in the circuit of the solar field and storage tank must also be taken into account, if the system is in stagnation due to lack of electrical power or another factor, the water will evaporate causing a decrease in the stored level, having to replace it. To combat the problem of water availability, it is studied the possibility of using seawater in the cooling tower with a circuit independent of the water entering the chiller, avoiding the evaporation of drinking water or being cooled with air with the limitation of electricity consumption.

3. Methodology for the simulation study of the cooling system

A simulator of the absorption cooling system activated with solar collectors was developed using the TRNSYS software (Klein et al., 2009) in order to establish the optimal strategies of operation of the system under the different dynamics that will be presented during a daily operation in the school. This program allows performing quasi-dynamic studies having as a peculiarity that it can work with typical meteorological data bases (TMY), analyzing the behavior of the system under operating conditions closer to reality. The characteristics of the equipment used in the project were supplied to each of the types. Fig. 6 shows the user interface

programmed for the simulation of the solar cooling system. The TMY data correspond to the geographic location of Puertecitos.

Since the purpose of the simulation study is to determine the operative strategies of the system under the changes in the variables mentioned above, the following considerations were taken into account in order to simplify the processing of the mathematical models used in TRNSYS:

- Peak thermal load of 8.75 kW per cooling coil at 12:00 hrs of the day.
- No pressure drop or heat loss in the equipment was considered, only the heat transfer from the storage tank to the environment.
- Control on/off by temperature difference in the solar field ($T_{sf,out} - T_{sf,in} > 0$).
- 15-minute simulation intervals.

3.1. Heat transfer calculation

The thermal load to which the absorption cooling system will be subjected will depend on different factors, such as the ambient temperature, class schedule and the number of students present in the classrooms. For this study, we considered a peak thermal load of 35 kW present at 12:00 hr divided between the four cooling coils (CC₁, CC₂, CC₃ y CC₄), which is the maximum cooling capacity of our equipment. However, outside this time the load will be lower due to the decrease in the ambient temperature during the mornings and sunsets. Therefore, the heat that must be removed from the classrooms is calculated by the following expression:

$$Q_{chw} = \dot{m}_{chw} \cdot C_{pw} \cdot (T_{chw,in} - T_{chw,out}) \quad (1)$$

where Q_{chw} corresponds to the heat removed from the cooling coils and taken to the absorption chiller, $\dot{m}_{chw}$ is the mass flow of the chilled water circuit and C_{pw} the specific heat of the water.

To cool the fluid coming from the classroom cooling coils the system requires an amount of heat from our thermal storage tank, Q_{hw} , previously heated by our field of solar collectors. This is calculated based on the Performance Coefficient of our absorption chiller, COP, as shown below:

$$Q_{hw} = \frac{Q_{chw}}{COP} \quad (2)$$

Similarly, knowing Q_{hw} and the temperature of the hot water to which it is being supplied, $T_{hw,in}$, it is possible to obtain the return temperature to the tank according to the following expression:

$$Q_{hw} = \dot{m}_{hw} \cdot C_{pw} \cdot (T_{hw,in} - T_{hw,out}) \quad (3)$$

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Fig. 3. Puertecitos elementary school and kindergarten, with the absorption cooling system activated with solar thermal energy.

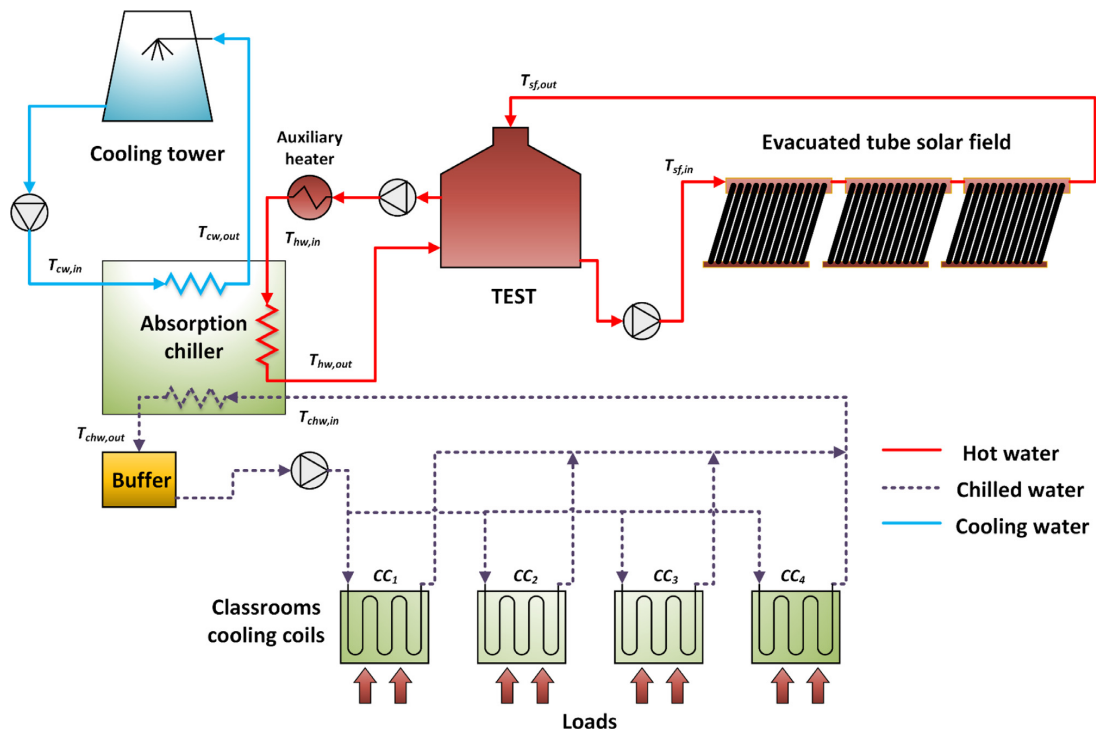


Fig. 4. Schematic diagram of an absorption cooling system activated with solar energy.

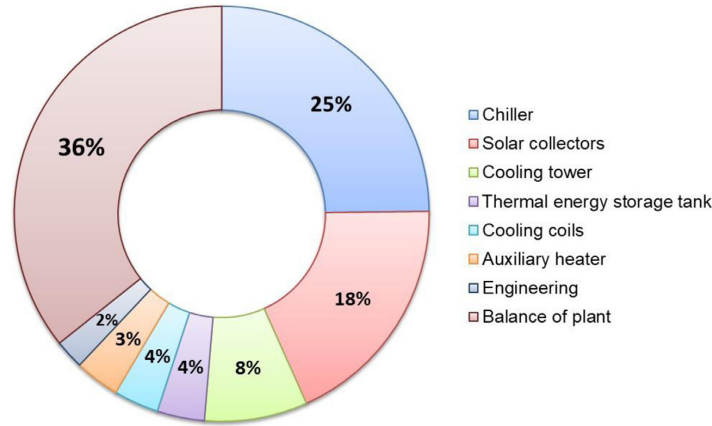


Fig. 5. Percentage of the cost of the different equipment and engineering of the project.

where $\dot{m}_{hw}$ corresponds to the mass flow of hot water from the storage tank to the chiller. The heat entered into the system by the two previous processes is transferred to the environment using the cooling tower, therefore:

$$Q_{cw} = Q_{chw} + Q_{hw}. \quad (4)$$

And in the same way, the outlet temperature of the cooling water circuit can be obtained, $T_{cw, out}$, according to the following equation:

$$Q_{cw} = \dot{m}_{cw} \cdot C_{pw} \cdot (T_{cw, out} - T_{cw, in}). \quad (5)$$

3.2. Hours of operation and environmental conditions

Since it is an elementary school and kindergarten, the system's operating hours differ from those that would be expected for an application in a domestic, commercial or industrial building. In this case, school hours are from 8:00 to 15:00 Monday to Friday, so that a constant operation of the system can be programmed and controlled. The period of holidays of the students corresponds to the month of July and the first week of August, same days that coincide with the dates where the ambient temperature is of the highest of the year. Puertecitos is located in a region with extreme climates; during the summer temperatures of up to 50°C can be reached, while in the winter it is close to 0°C. Fig. 7 shows the annual ambient temperature and relative humidity for the project site. As can be seen, the cooling needs of the classrooms practically begin in May and last until September, so the chiller will work around four months contemplating the school holidays.

4. Results

4.1. System start-up strategy

In the first instance, the conditions of the system will not be adequate to initiate the cooling of the classrooms of the school. This is because the minimum temperature in the storage tank will not be available to generate cooling by means of the absorption chiller. This is why it is necessary that, days prior to the start of classes or the period with AC needs, the temperature of the tank will be raised by means of the solar collector field. As a result of the simulation study, Fig. 8 shows the proposed strategy for starting the system, where four sunny days in August (period of 5450–5540 h of the year) were contemplated. Considering an initial temperature of 25°C, it is necessary to start heating the tank for at

least three sunny days prior to starting the system. During the course of the first day, the temperature rises to about 55°C, which decreases by about 3°C during the night due to energy losses to the environment. Subsequently, during the second day of heating, 80°C is reached, a temperature sufficient to partially operate the cooling machine, but not adequate to maintain it for the rest of the day. By the end of the third previous day, there is already a temperature of 96°C in the entire storage tank, enough to start the week of classes with efficient operation of the system.

4.2. Strategy for a day-night operation

Continuing the previous strategy and having the initial system conditions in place, we proceed with the proposed strategy for a day-night operation during school class periods. The simulation of the thermosolar absorption cooling system was carried out subject to a typical weekly operation expected from August 11 to 18 (period of 5360–5525 h of the year), corresponding to a week with high ambient temperature. The cooling demand expected during the operation of the system and the variations in the temperatures of the different circuits are presented in Fig. 9. There is no need for cooling during the first day, corresponding to a weekend, so temperatures of the flows involved do not perceive any change, only that of heat transfer between the environment because they seek the balance between them. At the beginning of the next day, the operation of the system returns, since classes are resumed at school; to achieve thermal comfort in the classrooms, the system is started one hour before class time, that is, at 7:00 the chiller is turned on. In the first hours, the thermal load is minimal and this tends to increase with respect to the hour's pass until reaching the maximum expected load of 35 kW at solar noon. In this period the maximum capacity of the chiller is used, so the consumption of thermal energy is considerably high, causing a decrease in $T_{hw, in}$ and $T_{hw, out}$. With the decrease of the solar resource, the thermal load also tends to be reduced. However, as mentioned above, the class schedule is from 8:00 to 15:00, Monday through Friday, so the system stops. This operating dynamic can be seen in the temperatures of the different circuits, which tend to equalize. It is worth mentioning that the temperatures of the hot water circuit, coming from the storage tank, decrease with respect to the passage of days, mainly due to the use of thermal energy for the operation of the chiller and the energy losses of our tank during the night periods, which does not recover fully during the day. This causes that, when starting the operation the next day, the initial

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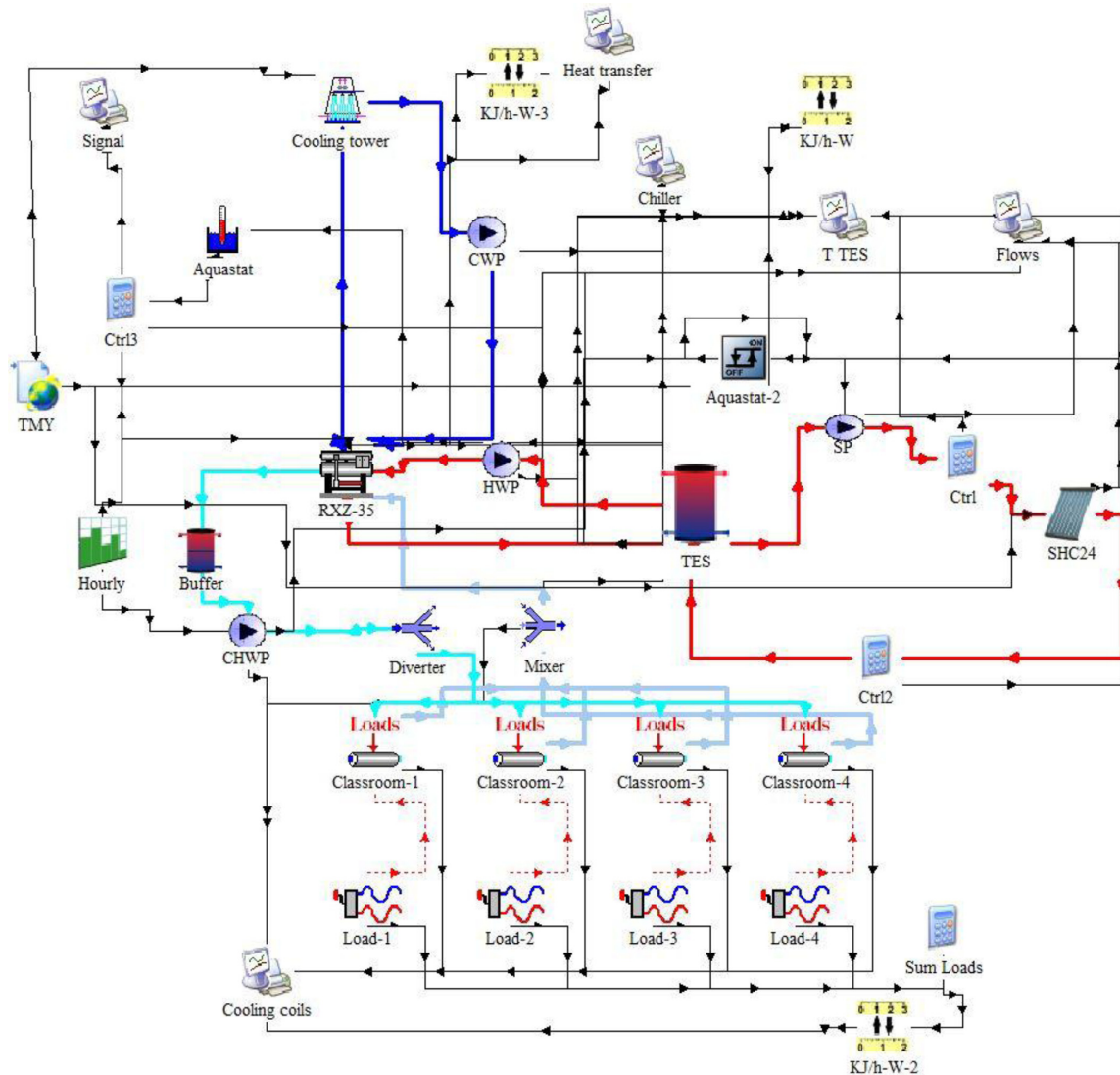


Fig. 6. Simulator user interface in TRNSYS of the solar powered absorption cooling system. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

temperature levels of the previous day are not reached. However, the temperatures are kept at the appropriate levels for the correct operation of the chiller. At the end of the week of classes, the temperature conditions of the system in general are not adequate to continue operating constantly, so it is necessary to raise the temperature of the storage tank without the chiller working.

In order to recover the optimal operating conditions of the system to cover the cooling demand during the weekly class period, the solar resource during the weekend is used by the solar field to heat the storage tank up to the defined set-point. This allows to start the cooling of the classrooms the first day of classes of the week and to maintain a stable operation until the end of the week. As shown in Fig. 9, only one day is necessary to bring the system back to optimal operating conditions. However, the strategy resulting from the simulation study proposes that during the two days

of the weekend constant heating of the tank is maintained regardless of whether a day is sufficient, seeking to recover the energy losses to the environment at night and in case of clouds that prevent normal heating of the tank.

The variation in the inlet and outlet temperature of the solar field during the simulation period mentioned above, as well as the solar radiation incident at the opening of the collector, can be seen in Fig. 10. The outlet temperature of the collector field during this period of the day is oscillating above 100 °C due to the stagnation and evaporation of the fluid, since the temperature of the storage tank is at its upper limit (96 °C) and the pump is stopped because there is no need to continue storing thermal energy. A steam recovery circuit is installed between the thermal tank and the solar collector field to avoid water losses and the problem of stagnation. This circuit is controlled by a pressure valve that redirects

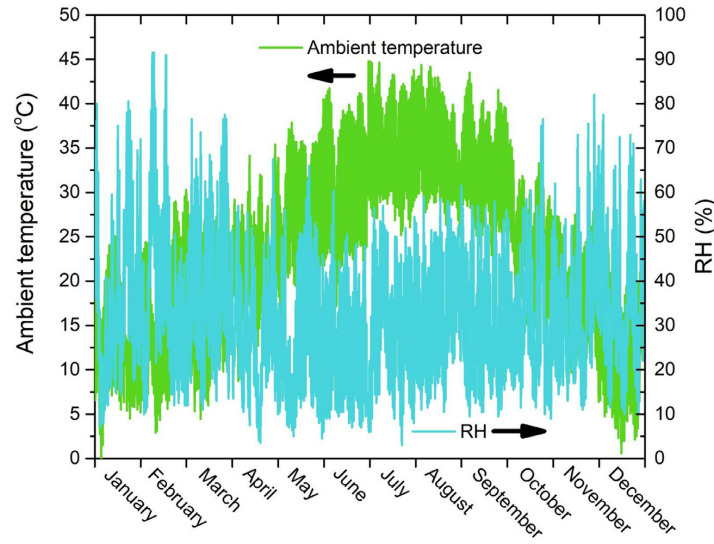


Fig. 7. Ambient temperature and annual relative humidity for the region of Puertecitos, Mexico.

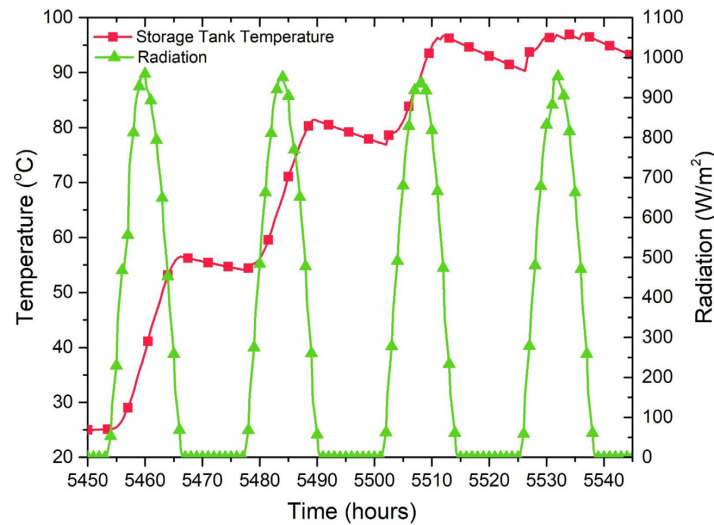


Fig. 8. Temperature increase in storage tank prior to absorption system startup.

the steam to a recovery tank, condenses and returns to the thermal storage tank. Since solar radiation is absent during the evening, $T_{sf, out}$ tends to decrease until it is in equilibrium with the ambient temperature. The following day, when it is the first day of classes of the week, as the solar resource increases with the passage of hours, it also rises $T_{sf, out}$ with respect to $T_{sf, in}$. However, it is no longer stagnant due to the need for air conditioning in the classrooms and the chiller operates normally, taking thermal energy from the storage tank and lowering its temperature, so there is no evaporation of the fluid. The remaining days of the week the same process of heating and cooling of the tank occurs, without any problem by the evaporation of the fluid. Until Saturday, when classes in the school are paused again, there is no need to cool the classrooms, although the temperature of the storage tank is not at its upper limit, so the solar field is responsible for taking it and

keeping it at set point temperature, seeking to have the system ready to operate when needed.

These operating strategies, resulting from the type of project in an elementary school in an isolated community, are beneficial in terms of dimensioning and control of the system, as it can reduce energy requirements, as well as having a typical and constant operation, becoming equipment with a robust and simple control.

4.3. Circuit mass flows during operation

Fig. 11 shows the on/off dynamics of the cooling system circuits. The solar field pump, during a day with no school activities, will operate only when the storage tank is not completely at set point temperature, switching on to recover energy lost to the

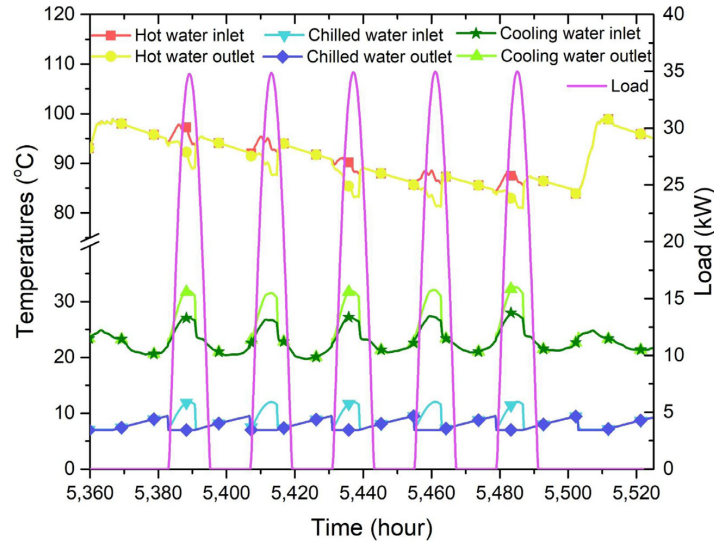


Fig. 9. Temperature variation of the different water circuits and the expected thermal load of the cooling system.

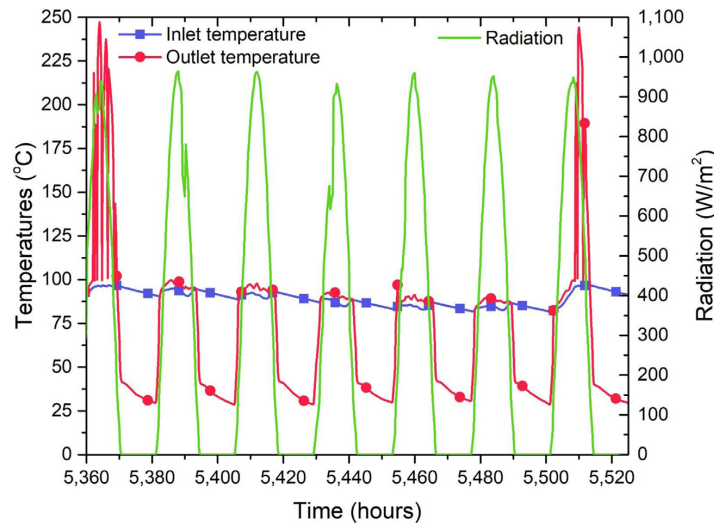


Fig. 10. Collector field inlet and outlet temperature and incident solar radiation in the collector plane during the simulation period.

environment. On a typical class day, this will stay on until the solar resource is not sufficient to raise $T_{sf, in}$ or when no more thermal energy can be stored. In contrast, the pumps of the three chiller feed circuits will be kept on during class hours only. At the end of the weekly class period, due to the lowering of the storage tank temperature mentioned above, the solar field will be operating to increase the stored water temperature by around 14 °C, reaching the set point.

4.4. Heat transfer between flows

The thermal energy that feeds and removes the streams from the circuits in the cooling system will vary depending on the air conditioning needs present in the classrooms of the school. Fig. 12 shows the heat transferred between the different streams during

a typical day of operation. Since classes start at 8:00, the system will automatically switch on one hour before to have the spaces to be air-conditioned in comfortable conditions, as well as to give enough time for the chiller and other equipment to start operating. As the hour's pass, the heat load will increase to reach its expected maximum of 35 kW at 12:00, as seen in the chilled water flow responsible for removing heat from the classrooms, to proceed with decreasing it and turning the system off at 15:00.

4.5. Preliminary experimental results

Experimental tests were carried out during the implementation of the system. Fig. 13 shows the temperatures measured in the different chiller circuits over a period of 450 min. The operating dynamics of the cooling system are very notable, mainly due to the

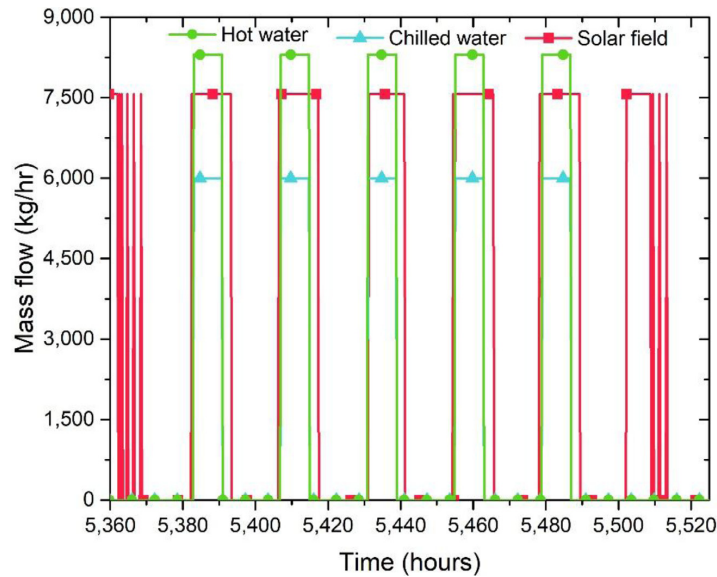


Fig. 11. On/off dynamics of the pumps of the different cooling system circuits.

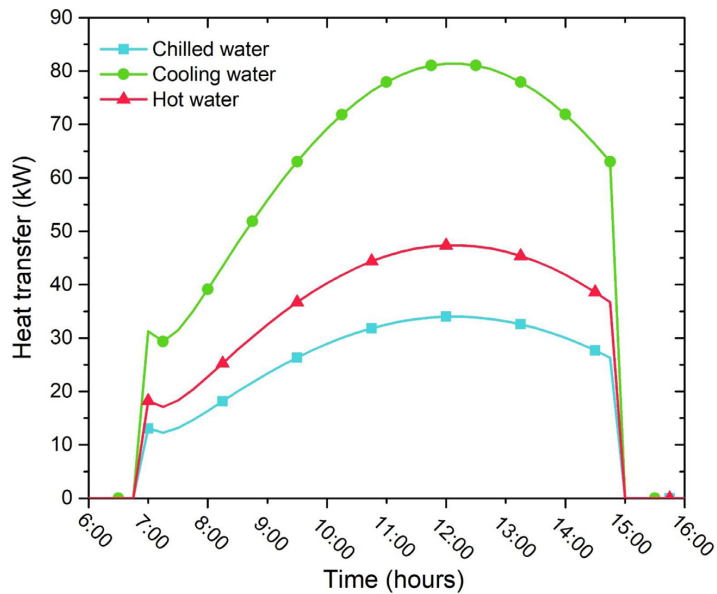


Fig. 12. Heats of the different cooling system flows during a typical day of operation.

programmed hysteresis. As determined in the day-night operating strategy of Section 4.2, in order to bring the operating temperature conditions to the chiller, it is necessary to turn on the system one hour in advance. With this, the temperature of the generator allows the cooling effect to be carried out properly and to initiate the air conditioning of the classrooms in the established schedules. For this test, a chilled water set-point of 10 °C with a hysteresis of ± 4 °C was selected. The variation in the temperature of hot water entering and leaving the chiller is remarkable. This is due to the high energy consumption of the chiller to meet the cooling needs. During operation, temperature differences of up to 16 °C between

the hot water inlet and outlet can be seen, as during the heating period is the time in which most energy is required to bring the chiller generator to operating temperature. This can be seen in the minute 30 of the experimental test. It is only just before the minute 60 when the temperature of the chilled water reaches the set temperature, taking into account the hysteresis, so that the thermal energy supply is almost completely reduced. From this point on, the temperature of the chilled water will increase until it reaches the set-point, so the generator heating process starts again and the cooling effect at the required temperature levels is achieved.

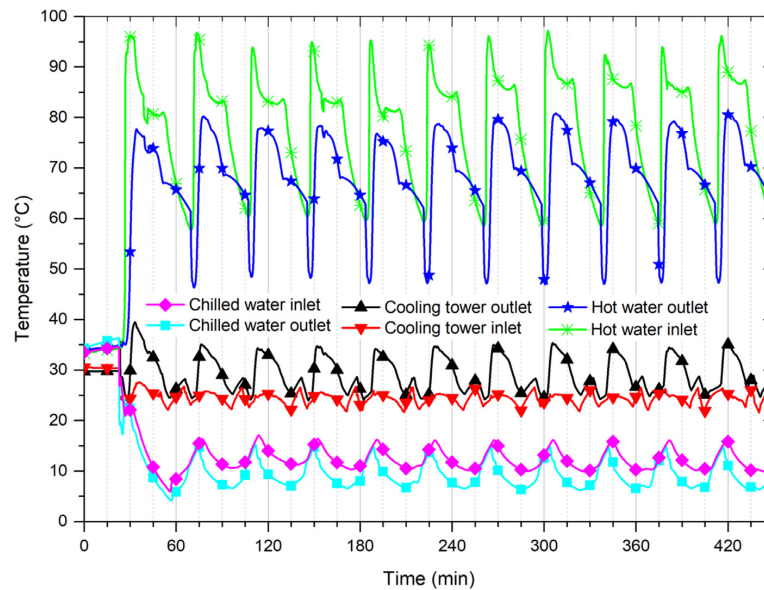


Fig. 13. Experimental temperatures of the different circuits of the cooling system.

5. Conclusion

A 35 kW LiBr-H₂O absorption cooling system, being activated with solar thermal energy from an evacuated tube solar collector field, was implemented and operationally analyzed in the elementary school of Puertecitos, Baja California, Mexico. Since it is an elementary and kindergarten school, the operating schedules of the system differ from those that would be expected for an application in a domestic, commercial or industrial building. In this case, the class schedule is from 8:00 to 13:00 Monday to Friday. Considering the application and context of the project (school in a community isolated from the electricity grid), it is extremely important to perform preliminary operational studies to avoid problems in the operation of the system in general. Therefore, and seeking to have a system as robust as possible, this simulation study was conducted as a first stage before experimentation and completion of the project.

The simulation results show that the proposed operational strategies are adequate for stable functioning of the absorption chiller and if they are not implemented correctly, the cooling needs of the elementary and kindergarten classrooms will not be met. The system is able to remove the thermal load from the classrooms during a typical week of class, while on weekends the solar field is used to bring the system to optimum starting conditions. These operating strategies, resulting from the type of project in an elementary school, are beneficial in terms of sizing and control of the system, as it can reduce the requirements for energy storage, as well as having a typical and constant operation, becoming equipment with a robust control and easy to use. The preliminary experimental results of the plant start-up were presented, where some of the operational strategies proposed from the simulation studies can be validated.

The experiences acquired by this type of project generate valuable knowledge since the installation of absorption cooling systems using solar thermal energy in an isolated elementary school is a novel application and the economic information of the equipment, construction and start-up of the plant will help future research.

The experimental study will be presented in future work, validating the operational strategies resulting from this simulation study.

Declaration of Competing Interest

None.

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**E. Simulation of a Solar-Assisted
Air-Conditioning System Applied to a
Remote School**

Revista: Applied Sciences



Article

Simulation of a Solar-Assisted Air-Conditioning System Applied to a Remote School

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Abstract: In this work, we present an absorption cooling system with 35 kW capacity driven by solar thermal energy, installed in the school of Puertecitos, Mexico, an off-grid community with a high level of social marginalization. The cooling system provides thermal comfort to the school's classrooms through four 8.75-kW cooling coils, while a 110-m² field of evacuated tube solar collectors delivers the thermal energy needed to activate the cooling machine. The characteristics of the equipment installed in the school were used for simulation and operative analysis of the system under the influence of typical factors of an isolated coastal community, such as the influence of climate, thermal load, and water consumption in the cooling tower, among others. The aim of this simulation study was to determine the best operating conditions prior to system start-up, to establish the requirements for external heating and cooling services, and to quantify the freshwater requirements for the proper functioning of the system. The results show that, with the simulated strategies implemented, with a maximum load operation, the system can maintain thermal comfort in the classrooms for five days of classes. This is feasible as long as weekends are dedicated to raising the water temperature in the thermal storage tank. As the total capacity of the system is distributed in the four cooling coils, it is possible to control the cooling demand in order to extend the operation periods. Utilizing 75% or less of the cooling capacity, the system can operate continuously, taking advantage of stored energy. The cooling tower requires about 750 kg of water per day, which becomes critical given the scarcity of this resource in the community.

Keywords: renewable energy; solar cooling; isolated community; absorption chiller; TRNSYS

1. Introduction

Renewable energies are an excellent option to provide electrical energy services, cooling, heating, food dehydration, and desalination, among others, in remote places where traditional fuels are not available or are difficult to acquire [1]. Decentralized low-capacity electrical systems installed in consumption centers, also known as microgrids, enable services to be brought to isolated regions where traditional electrification technologies are difficult to access [2] and, based on a combination with renewable energy sources, prevent the continued emission of large quantities of pollutants into the environment.

On the other hand, solar energy can be used for the direct production of electrical energy, as well as for heating and/or cooling of spaces or products [3]. If the air conditioning of spaces is required,

it can be done using a thermal or electrical source by means of solar collectors or photovoltaic modules, respectively, as well as some source based on fossil fuels. However, thinking of sustainable development of marginalized and isolated communities, traditional fuels must make way for renewable energies, both for reducing pollutants and saving money. Because of this, the cooling and/or air-conditioning systems driven by clean energies attracted attention for their use in isolated communities, giving comfort conditions to the population and increasing their quality of life [4–6]. Absorption cooling technologies are an attractive option for the climatization of spaces because they can be driven by low-temperature thermal energy [7] without the use of large amounts of electrical energy [8], being able to use solar heating technologies as sources of activation [9]. In addition, they have great technological maturity, economic profitability, and high efficiency [10], as well as a great possibility of energy integration with other technologies to offer other services simultaneously, such as desalination [11].

A large number of studies related to solar-assisted absorption cooling technologies were carried out. Aliane et al. [12] conducted an investigation of experimental results and experiences presented in the state of the art of these types of systems, mentioning that it is necessary to have experience in the installation and to keep the design as simple as possible to ensure efficient operation. Bataineh and Taamneh [13] mentioned that low performance and high cost are the main disadvantages of sorption technologies; however, given the synchrony of solar radiation with cooling needs, these systems are still attractive when coupled with solar energy; thus, research continues to seek a solution to their technological, economic, and environmental problems. Lazzarin and Noro [14] mentioned that cooling systems coupled with photovoltaic modules consume half of the investment cost compared to cooling systems driven by thermal energy; thus, a large reduction in the cost of solar collectors must be achieved for them to be competitive.

Soto and Rivera [15] developed an absorption air-conditioning system cooled by air, which works using an ammonia–lithium nitrate mixture. Their experimental results showed that it is possible to cool the absorption system using only air; thus, there is no need for a cooling tower. They achieved a coefficient of performance (COP) in the range of 0.1 to 0.33 with cooling capacities in the order of 0.8 kW to 3.4 kW. However, the effect on electricity consumption due to the motors of the heat sink fans was not mentioned. Chen et al. [16] carried out experimental tests on an air-cooled solar-assisted absorption cooling system applied to residential buildings. They mentioned that, with this type of system, it is possible to save water, as well as reduce maintenance needs and space due to the absence of cooling tower. In addition, they commented that there are still no low-capacity commercial air-cooled absorption chillers, mainly due to the risk of crystallization of the working mix under environmental conditions. Huang et al. [17] presented the results of a 35-kW capacity solar thermal cooling system that began operation in 2018, which was evaluated during the summer cooling and winter heating periods. The average annual COP was in the range of 0.68 and 0.76, while the solar fraction for heating and cooling averaged 56.6 and 62.5%, respectively, generating electrical savings of 10,158 kWh annually. In another experimental study of thermosolar cooling, Rosiek [18] implemented operational strategies to optimize the exergetic efficiency of the system, such as the best absorption chiller–heat source coupling and equipment control. Lubis et al. [19] tested a 239-kW-capacity chiller activated in hybrid manner with solar energy and gas. By characterizing their system under a wide range of operating conditions, they simulated it under a tropical climate scenario of the Asian region, showing good operating potential. Sokhansefat et al. [20] performed a simulation and experimental validation of a five-ton solar absorption cooling system installed in Tehran, Iran. They analyzed different factors that affect the performance of the system and established the optimal sizes of the equipment to achieve a 28% increase in its performance. Li et al. [21] investigated the performance of a 23-kW solar absorption cooling system driven by a parabolic trough collector field for cooling a 102-m² meeting room. They attributed the low performance of the system to the fact that the pipes were too long and the heat loss was too high.

One of the main tools for the analysis of thermosolar absorption systems is the TRNSYS software; a large number of simulation studies used it to determine the best operating conditions, sizing,

and optimizations [22–27], given the advantages of this software when analyzing the systems under conditions closer to reality. In a work related to the use of these systems in schools, Praene et al. [6] presented modeling and simulation studies, as well as preliminary experimental results, of a solar-driven 30-kW LiBr–H₂O system installed at a university in France. They performed a complete analysis of the thermal loads of the classrooms using the TRNSYS software during the periods from 8:00 to 12:00 a.m. and from 1:00 to 5:00 p.m., corresponding to the class schedule. Uçkan and Yousif [28], using the TRNSYS software, analyzed the implementation of a 35-kW LiBr–H₂O cooling system driven by solar thermal energy in the arid region of northern Iraq, seeking to reduce the consumption of fossil fuels and promote the use of renewables. In another similar study, Dakheel et al. [29] simulated the use of renewable energy-saving technologies in the United Arab Emirates, including solar–thermal absorption cooling systems. Reductions of 19.35% in annual cooling energy use, and 7.2% reduction in total annual energy use were achieved. Abrudan et al. [30] studied the implementation of absorption cooling equipment under different climatic conditions. They proposed new correlations between the solar hot water temperature and the cooling water temperature in order to avoid both crystallization and the reduction of the degassing zone below 6%. They concluded that, depending on the present climatic conditions, it is necessary to implement different operative strategies looking for greater efficiency of the system. Analyzing options to improve the efficiency of this type of system, Bellos and Tzivanidis [31,32] studied the incorporation of nanofluids into solar heating technologies. They found that the use of nanoparticles increased the exergetic efficiency of the collector by about 4% and the cooling output by about 0.84% on a daily basis. Mendecka et al. [33] analyzed two thermal storage options for adsorption cooling systems using solar energy. Their results showed that energy storage using phase change materials was slightly more efficient than storage using water. Another option for the activation of absorption systems is hybrid photovoltaic/thermal (PV/T) technology. Alobaid et al. [34], in their review of the state of the art of these systems, identified that up to 50% primary energy can be saved by using hybrid collectors in absorption machines compared to mechanical vapor compression equipment for cooling. The hybridization of mechanical vapor compression technologies was even proposed, working in combination with absorption technologies to improve the system efficiency [35,36].

It should be noted that very few studies mentioned the benefits of this type of solar-assisted cooling system in applications with difficult access to electricity, such as isolated communities. By using an energy resource that can be exploited in an excellent way in most of the world's territory, it makes its implementation attractive in this type of scenario where there is no electricity supply. This type of application presents factors that affect the operation of the system, different from those that would be expected in absorption systems installed in towns or cities with access to continuous fresh water and electricity services. However, as mentioned in the previous works, a detailed analysis of the engineering of these systems must be carried out so that they become both technically and economically attractive. This work presents the simulation study of an absorption air-conditioning system driven by a solar thermal collector field, installed in the school of Puertecitos, an off-grid community with a high level of social marginalization. The system was simulated using the TRNSYS software in order to determine its performance under the typical operating conditions of a school in Mexico, as well as to determine the effect of different distinctive factors of a remote community, such as the scarcity of clean water and electricity. The aim of this simulation study was to determine the best operating conditions prior to system start-up, to establish the requirements for external heating and cooling services, and to quantify the freshwater requirements for the proper functioning of the system. This study allows establishing the operation and maintenance strategies, as well as the operating limits, of the cooling system under a real scenario that was not studied in the literature.

2. System Description

The thermosolar absorption air-conditioning system installed in the primary school of the community of Puertecitos, Baja California, Mexico (30°21'19.7" north (N), 114°38'26.3" west (W)) is

composed of a field of solar thermal collectors, containing evacuated tubes with a parabolic reflector in the lower part with a total aperture area of 110 m²; these solar heating technologies are responsible for maintaining optimal temperature conditions using a water storage tank of 12 m³ capacity, necessary for proper operation of the cooling system. Both the collector field and the thermal energy storage tank (TEST) compose the hot water circuit.

An LiBr–H₂O absorption cooling system of 35 kW capacity is thermally driven by the hot circuit fluid. Due to the absorption process, heat is removed from the water that is used as a cooling medium in the chilled water circuit. For abrupt temperature changes not to affect the operation of the chiller, a 1-m³ buffer tank is used in the chilled water circuit. The chilled water passes through four cooling coils with a capacity of 8.75 kW each, one in each classroom of the school, removing the thermal load when required. The heat supplied to the chiller, both by the hot and chilled water circuit, is discharged through the cooling water circuit and released into the environment with an evaporative cooling tower.

Given the problems of water availability in the region and, being a community next to the sea, the cooling water circuit has two variants: normal water mode and seawater mode. In the first mode of operation, freshwater is used as a direct cooling medium for the chiller; it is fed into the absorption machine to remove the heat and transfer it to the environment with the cooling tower. When seawater is used, the cooling of the system is carried out indirectly. The cooling water circuit is divided into two; the tower is responsible for cooling the seawater and recirculates it in an independent circuit, while, in the other circuit, freshwater is used to remove heat from the chiller and, by means of a titanium coil submerged in the seawater reservoir of the cooling tower, the removed heat is transferred. This is done in order to not introduce the incrusting and corrosive compounds that the seawater carries with it into the internal heat exchangers of the absorption machine, as well as to not depend on replenishment of freshwater due to evaporation by cooling the system. There is also a diesel auxiliary heater in the hot water circuit so that, if the temperature conditions in the TEST are not met, the system can be operated. The previous equipment installed in the Puertecitos school is shown in Figures 1 and 2.



Figure 1. Evacuated tube solar collector field with a bottom parabolic reflector.

The schematic diagram of the components and connections is shown in Figure 3. The solar collector field is responsible for maintaining the temperature of the TEST in the range of 75–96 °C, enough for proper chiller operation. Regardless of whether the cooling machine is operating or not (for example, on weekends, the chiller does not operate because there are no classes), the solar field pump continues to provide fluid for heating until the tank is completely at the set point temperature, in this case 96 °C, always seeking to have the system in operating conditions. When there is a need for classroom air conditioning, the chiller turns on and the TEST pump supplies hot water from the top. Since the temperature of the generator of the absorption machine is at an optimal level for operation, the pump of the chilled water circuit is turned on to bring it to the cooling coils of the classrooms, at a temperature between 7 and 10 °C, removing the heat from the spaces to be air-conditioned. This current is taken to the chiller to be cooled again. It is worth mentioning that the configuration of the

chilled water circuit allows controlling the number of cooling coils used for air conditioning, allowing a saving of thermal energy when air conditioning is not required in all classrooms.

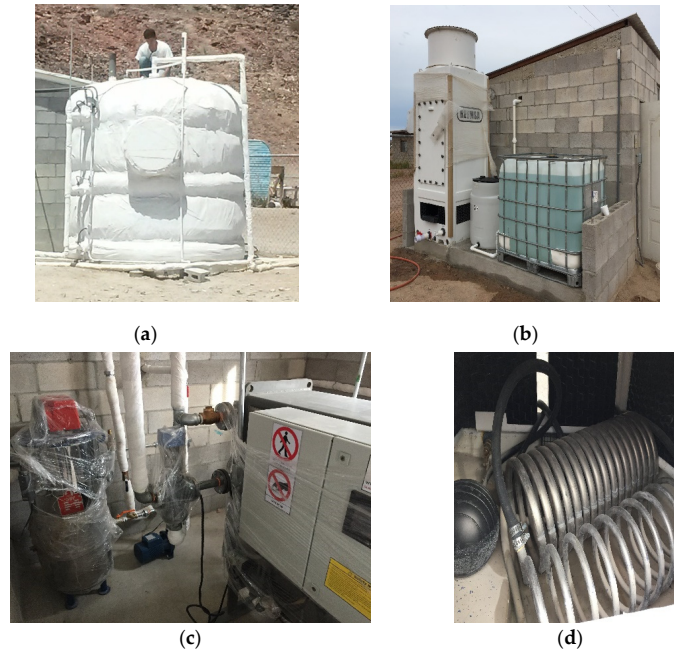


Figure 2. Components of the thermosolar absorption cooling system at Puertecitos school, where we can appreciate the (a) thermal energy storage tank, (b) cooling tower with freshwater and seawater storage tanks, (c) auxiliary heater and chiller inside the machine room, and (d) titanium coil inside the cooling tower.

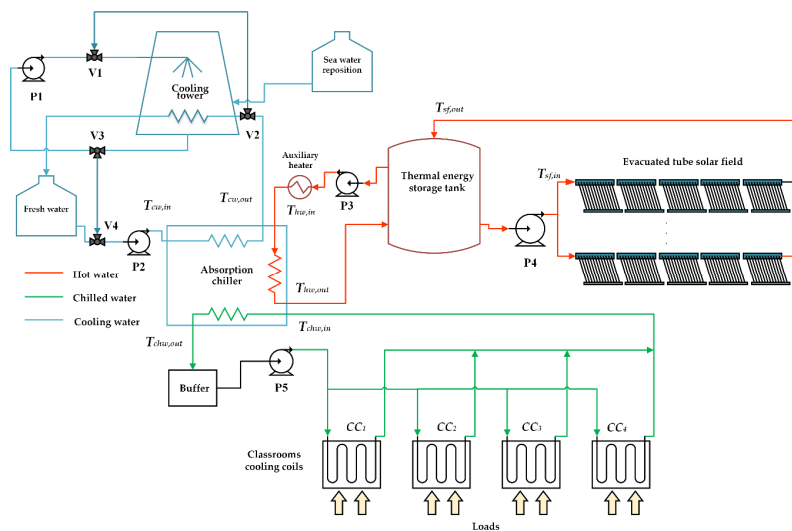


Figure 3. Schematic diagram of the equipment and circuits of the solar thermal cooling system.

Depending on the mode of operation in the cooling tower mentioned above, the cooling water circuit is modified with manual valves for the use of seawater and/or freshwater only. When seawater is required as cooling fluid in the tower, the seawater reposition tank fills the cooling tower tank while the P1 pump recirculates the fluid in a closed circuit between the V1 and V3 valves, cooling the seawater by evaporation. During this process, the P2 pump takes water from the freshwater tank and supplies it to the chiller to remove the heat generated. Then, it transfers it to the seawater by means of a coil heat exchanger submerged in the cooling tower reservoir (Figure 2d) and returns it to the freshwater tank, all in the closed circuit between V2 and V4. This configuration avoids the problem of freshwater evaporation in the cooling tower by using seawater, an easier resource to acquire in the community. On the other hand, when freshwater is used as a direct cooling medium, the P2 pump takes water from the cooling tower reservoir through the V3–V4 circuit, it removes the heat from the chiller, and, through the V2–V1 circuit, it cools the freshwater through evaporation in the tower. The freshwater tank is positioned in a way that when the water level in the tower tank decreases, the water is replaced by gravity and controlled by a float. In the latter configuration, the P1 pump does not work.

3. Methodology

The TRNSYS software was used to carry out the operational simulation of the solar thermal cooling system. This program allows analyzing the system in a semi-dynamic way, having the advantage of working with typical meteorological databases (TMY) in the region, studying the system under more realistic operating conditions. By using modules programmed with validated mathematical models of various equipment, such as pumps, motors, heat exchangers, storage tanks, solar collectors, power cycles, and refrigeration, among many others, TRNSYS becomes an ideal software for the simulation of this type of process. The validated mathematical models used in the simulator modules can be reviewed in Reference [37]. Table 1 shows the characteristics of the cooling system equipment installed in the Puertecitos school, which were used as input parameters in the developed simulator, and Table 2 presents the description of the components within TRNSYS.

Table 1. System characteristics.

Solar Collector	
Brand/model	Suntask/SHC24
Collector type	Evacuated tube with parabolic reflector
Number of tubes	24
Aperture area	4.41 m ²
Optical efficiency (a ₀)	0.668
First order efficiency coefficient (a ₁)	1.496 W/m ² K
Second order efficiency coefficient (a ₂)	0.005 W/m ² K ²
Fluid	Water
Mass flow	0.02 kg/sec m ²
Number in series	5
Number of loops	5
Thermal Energy Storage Tank	
Volume	12 m ³
Height	2.5 m
Material	Fiberglass
Insulation thickness	0.025 m
Loss coefficient	1.4 W/m ² K
Fluid	Water

Table 1. Cont.

Absorption Chiller	
Model	Lucy New Energy/RXZ-35
Refrigerant	Water–lithium bromide
Cooling capacity	35 kW
COP	0.7
Hot water nominal temperature	90 °C
Hot water nominal flow rate	8.3 m ³ /h
Chilled water nominal inlet temperature	15 °C
Chilled water nominal flow rate	6 m ³ /h
Cooling water nominal temperature	30 °C
Cooling water nominal flow rate	15 m ³ /h
Power consumption	0.3 kW

Table 2. TRNSYS type for the components simulated.

Component	TRNSYS Type	Description
Temperature control	2	On/off control of feed pumps of the different circuits.
Pumps	3	Pumps for mass flow feeding of circuits.
Time-dependent forcing function	14	Cooling system on/off time control.
Weather data processor	15	Climatological database.
Cooling tower	51	Evaporative cooling tower with constant volumetric flow.
Solar thermal collector	71	Evacuated tube solar thermal collector.
Absorption chiller	107	LiBr/H ₂ O hot water-driven absorption chiller.
Thermal energy storage tank	534	Stratified thermal storage tank with variable nodes, ports, and insulation.
Buffer tank	534	Thermal storage tank with variable nodes, ports, and insulation.
Mass flow diverter	647	Mass flow diverter with variable outlets.
Mass flow mixer	649	Mass flow mixer with variable inlets.
Mass flow heat exchanger	682	Energy and mass balance heat exchanger.
Heat load	686	Heat load profile dependent on the day, weekday, month, or season.

In order to carry out the analysis of the system with the aforementioned software, the following considerations were taken into account:

- Steady state;
- 15-min simulation intervals;
- Maximum thermal load of 8.5 kW per classroom at 12:00 p.m.;
- Pressure drops and heat losses in the equipment were not considered, only the heat transferred to the environment by the TEST losses was considered;
- On/off control by temperature difference in the solar field ($T_{sf,out} - T_{sf,in} > 0$);
- Only fresh water was used in the cooling water circuit;
- System operation with solar energy only.

The cooling system operates under an established school schedule of 8:00 a.m. to 3:00 p.m. Monday through Friday, which are typical class hours in Mexico. The vacation periods correspond to the months of July and part of August, the same days in which the highest ambient temperatures are present; thus, a constant operation is expected during May, June, mid-August, and September, where the penultimate month is the critical one due to the extreme temperatures present. Figure 4 shows the ambient temperature, relative humidity, and global radiation conditions based on a TMY file of the study region for the simulation period, corresponding to warm and sunny days in August. Being a coastal community, the relative humidity plays a very important role in the proper operation of the cooling water circuit and, therefore, the system in general.

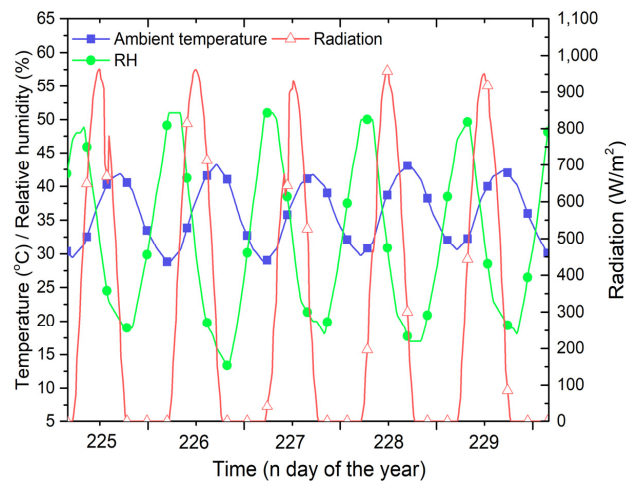


Figure 4. Ambient temperature, relative humidity, and global radiation in the simulation lapse for Puertecitos, México.

4. Results and Discussion

4.1. Operation at Nominal Capacity

Seeking adequate temperature conditions inside the classrooms, the chiller is turned on every day that cooling is needed from 7:00 a.m. to 3:00 p.m., i.e., one hour before the beginning of classes. This results in a constant eight-hour operation from Monday to Friday, allowing robust system programming and control without much user intervention.

Figure 5 shows the variation of the thermal load present in the classrooms during a typical operating day and the temperature of the chilled water coming from the absorption chiller. The thermal load starts at 6:00 a.m.; however, the chiller starts cooling the water at 7:00 a.m. As the hours pass, the load increases until it reaches 35 kW, with the maximum demand established at 12:00 p.m. corresponding to 8.75 kW per cooling coil. At 6:00 p.m., the thermal load is zero, but the cooling system is stopped 3 h beforehand as classes are finished and the facilities are empty, avoiding the consumption of thermal and electrical energy. The chilled water temperature remains constant at 7 °C during chiller operation due to the control of the heat supplied to the system by means of the hot water circuit. If the cooling load increases, the heat is taken out of the TEST to raise the capacity of the chiller and keep the operating conditions as stable as possible. Otherwise, if the load increases but the heat supplied to the system remains constant, it results in an increase in the temperature of the chilled water and, therefore, of the classrooms. In addition, the heat transfer rate to the chiller cooling water must be sufficient to maintain stable operating conditions, so that, as the cooling capacity increases, more heat is removed from the system by the cooling water circuit. It should be noted that the temperatures

of the hot water and cooling water circuit do not remain constant, as they depend directly on the environmental conditions present during the operation of the system; however, they are always within the minimum ranges, as shown in Figure 6.

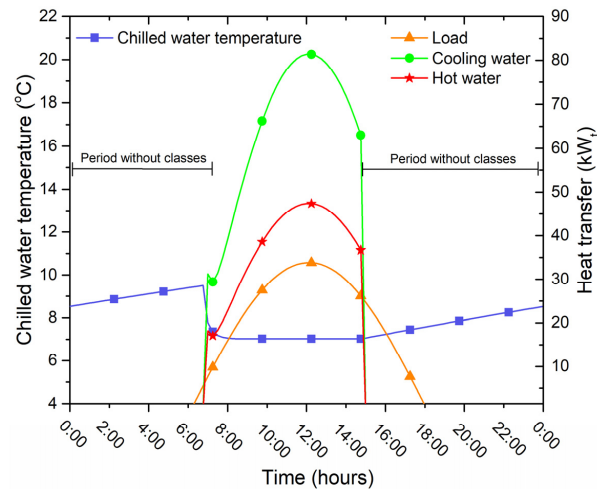


Figure 5. Chilled water temperature and heat transfer of the circuits.

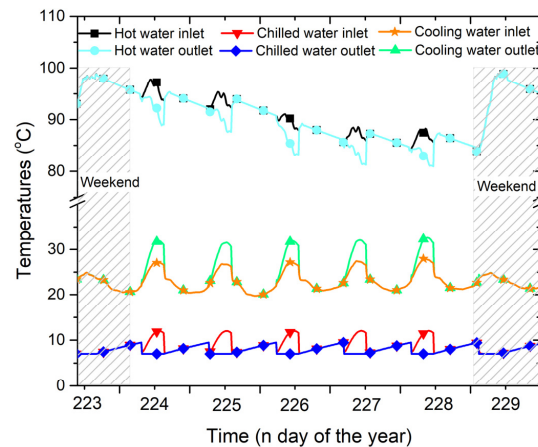


Figure 6. Variation in system circuit temperatures.

A typical week of operation is shown in Figure 6, where the temperature variations of the different circuits of the cooling system at its inlet and outlet can be appreciated. At the beginning of the week, the temperature of the TEST is in optimal conditions to start the system, around 97 °C for all its levels, as the field of solar collectors is dedicated to its increase in temperature on Saturday and Sunday. During the days of classes, the TEST tends to decrease its temperature significantly because the collection and storage of daily solar energy is not enough to return it to initial conditions. Since the operation of the chiller stops at 3:00 p.m., while there is still a solar resource, the TEST recovers a bit of its temperature level by keeping the solar collector pump on; however, this is not enough to counteract the loss of heat during the day. With the passing of operating days, the decrease in temperature becomes greater; thus, when the last day of classes arrives, the system operates in limited conditions

of activation temperature (around 75–80 °C), while being able to satisfy the cooling demand of the classrooms. However, it is necessary for weekends to be exclusively dedicated to heating the TEST and bringing it to its initial conditions so that, at the beginning of the following week, the complete system is in optimal operating conditions and meets the cooling requirements during the five days of classes.

4.2. Partial Load Operation

The school where the thermosolar cooling system is installed has fluctuations in student attendance, mainly due to migration and immigration phenomena of its population, depending on weather or working conditions. For this reason, depending on the number of students in the school, it may not be necessary to air-condition the four classrooms, but only some of them.

Figure 7 shows the behavior of the mean temperature of the TEST with respect to the variation of the maximum thermal load of the classrooms, using the profile presented in Figure 5 at 25, 50, 75, and 100% of the chiller capacity. Working the system with a maximum load of 35 kW ensures operation during the five days of classes as long as weekends are used to heat the TEST, as mentioned in the previous section. However, with a maximum load of 25.5 kW, corresponding to 75% of the capacity of the cooling system, it is possible to maintain constant operation for more than five days without having to spend the entire weekend heating it. The decrease in the average temperature of the TEST is recovered in the period with the availability of the solar resources after typical operation of 7:00 a.m. to 3:00 p.m. On the other hand, when operating at 50% of the maximum capacity or less, the temperature conditions of the system are kept stable, being able to operate normally for the whole week in the aforementioned period.

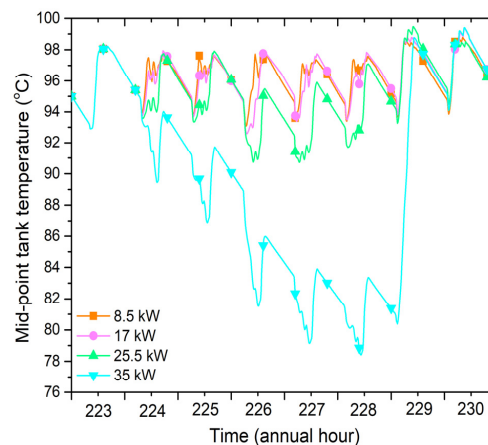


Figure 7. Thermal energy storage tank (TEST) temperature when varying the thermal load.

4.3. Evaporation of Cooling Water

The availability of water and electricity in this region is a very severe problem. There is no water distribution network and the easiest way to acquire it is in a well located 32 km away from the community; thus, it becomes a natural resource much appreciated and cared for. On the other hand, electricity is supplied by a microgrid based on renewable energy, but the school does not have the economic resources to pay for it. However, the heat entering the cooling system needs to be removed; thus, the use of a cooling tower is essential. There are two options: cooling with water or air. The first consumes a considerable amount of water with low electricity consumption, while the second does not consume water, but a large amount of electricity. Water cooling was selected by installing equipment that allows both seawater and freshwater as a working fluid. Thus, if freshwater is not available, seawater can be used given the ease of acquisition.

Figure 8 shows the weekly water consumption in the cooling tower. Depending on the thermal load of the chiller, the water evaporation rate can reach 140 kg/h. During a typical eight-hour operating day, about 750 kg of water would be needed per day, giving a total of 4000 kg ($\sim 4 \text{ m}^3$) per normal week of classes. This is critical considering the scarcity of the resource in the region and the work involved in maintaining tank levels in these conditions, as well as maintenance related to using seawater in the cooling tower due to salt concentration. With these results, it is possible to program the filling of the tanks and the cleaning of the cooling tower tank.

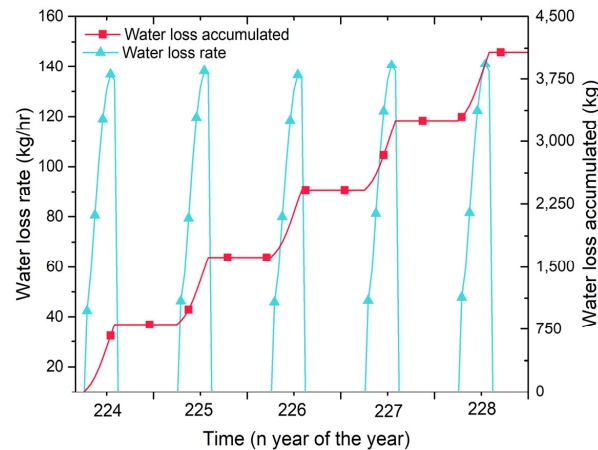


Figure 8. Rate of water loss due to evaporation in the cooling tower and its weekly accumulation.

5. Conclusions

An operational analysis of a 35-kW absorption cooling system driven by solar thermal energy was carried out and installed in a primary school in the isolated community of Puertecitos, Baja California, Mexico. The system was simulated using TRNSYS software in order to determine its performance under the typical operating conditions of a school in Mexico, as well as to determine the effect of different distinctive factors of a remote community, such as the scarcity of freshwater and electricity. The simulation results show that, under full load operation, the cooling system can satisfy the air-conditioning requirements of the four classrooms during the five days of the week, from 7:00 a.m. to 3:00 p.m. The weekends are used exclusively for the storage of thermal energy and elevation of the temperature of the TEST, using the solar collector field. The operation of the system under different thermal load profiles was also studied, and it was concluded that, under a thermal load of 75% of the chiller capacity or less, the system can operate the entire week without the need to dedicate weekends solely to thermal energy storage. With the configuration in the chilled water circuit, the individual operation of each cooling coil is possible; thus, it is possible to control the thermal load of the classrooms to extend the operation of the chiller system.

On the other hand, the consumption of freshwater in the cooling tower is a critical factor due to the scarcity of this resource in the region. At its maximum capacity, the system consumes about 4 m^3 of water to remove heat from the equipment per normal week of operation. Therefore, the system is designed with two independent cooling water circuits, where seawater can be used in the cooling tower as it is easier to acquire in the project community. This study allowed establishing the operation and maintenance strategies, as well as determining the best operating conditions prior to system start-up, establishing the requirements for external heating and cooling services, and quantifying the freshwater requirements for the proper functioning of the system under a real scenario not studied in the literature.

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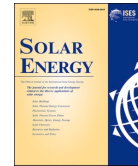
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**F. Techno-economic analysis of hybrid PV/T
systems under different climate scenarios and
energy tariffs**

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Techno-economic analysis of hybrid PV/T systems under different climate scenarios and energy tariffs

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ABSTRACT

This paper presents the techno-economic study of a hybrid PV/T system for the simultaneous production of electric and thermal energy under different climate scenarios and energy tariffs. A simulator was developed and validated using TRNSYS software to perform a comparative energy study of five cities with different weather conditions and energy prices: Mexicali, Soria, Bigene, Fresno and Madison. Our work proposes an extended analysis with sites that have not been studied before, and complements the scope of previous work. Under same requirements of thermal energy for DHW demand, energy production was analyzed during an entire year of operation, using different numbers of PV/T modules. Results show that the city of Mexicali is the place where most energy, both thermal and electrical, is produced. In this same locality a solar thermal fraction of 80% can be reached with only three modules installed. Energy feasibility contrasts drastically with economic benefit, where Soria presents the most attractive investment returns and monetary savings of the scenarios analyzed. Moreover, a comparison of investment returns is presented, considering subsidies in the initial investment, where return of up to 1.8 years result with a 75% subsidy. In addition, a relation of the climatic conditions of the studied cities is presented, such as solar radiation and ambient temperature, in the performance of the PV/T technologies. The conclusions of the study help decision making and planning of financial support programs, since it was considered extreme climates and representative of most of the sites where PV/T systems would be installed.

1. Introduction

Global energy demand is increasing rapidly (International Energy Agency, 2018) and is directly related to population growth, rising living standards and industrialization. At the same time, the levels of environmental pollution are increasing alarmingly because the main energy source is based on fossil fuel use (Aguilar-Jiménez et al., 2018). The use of renewable energy is the most sustainable option, and increasingly economically attractive, to make a transition to an energy matrix with fewer pollutants. In particular, photovoltaic (PV) solar energy is considered the fastest growing technology for electric power production and pollutant reduction, but PV manufacturing is a major energy consumer and pollutant generator (Liu et al., 2019). Therefore, it is of great

importance to increase efficiency and control in the manufacture of this type of systems, seeking a reduction in its carbon footprint throughout its life cycle.

Temperature affects the conversion efficiency of semiconductor materials, especially monocrystalline silicon (Vaishak and Bhale, 2019). At present, the efficiency of commercial (monocrystalline) PV modules can reach approximately 22%, with very few options available, while at the laboratory level maximums of up to 26.7% are recorded (Fraunhofer Institute for Solar Energy Systems, 2019; NREL, 2019), so there is great potential for energy use and improvement in this technology. During the conversion of electrical energy, the largest amount of energy is absorbed and dissipated into the environment (Kalogirou and Tripanagnostopoulos, 2006). An alternative to increase the efficiency of the PV technology is to hybridize with heat exchangers, for the combined

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ANEXO F. TECHNO-ECONOMIC ANALYSIS OF HYBRID PV/T SYSTEMS UNDER DIFFERENT CLIMATE SCENARIOS AND ENERGY TARIFFS

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Nomenclature		
PV/T	Photovoltaic/Thermal	T_{RC} Reference conditions temperature (°C)
DHW	Domestic Hot Water	T_a Ambient temperature (°C)
MPP	Maximum Power Point	T_{in} Inlet temperature (°C)
NOCT	Nominal Operating Cell Temperature	T_{out} Outlet temperature (°C)
TMY	Typical Meteorological Year	T_{ST} Thermal storage tank temperature (°C)
G_T	Solar thermal collector incident energy (W/m ²)	a_0 Optical efficiency (-)
G	Incident solar radiation (W/m ²)	a_1 First order heat loss coefficient (W/K/m ²)
η_{el}	Electrical efficiency (-)	a_2 Second order heat loss coefficient W/K ² /m ²)
$\eta_{PV,RC}$	Photovoltaic cell efficiency at reference conditions (-)	f_{th} Solar thermal fraction (-)
η_{th}	Thermal efficiency (-)	$\dot{Q}_{DHW}$ Domestic hot water thermal energy needs (kWh)
P_{el}	Electrical power (W)	$\dot{Q}_{aux}$ Auxiliary heater energy (kWh)
P_{th}	Thermal power (W)	R_y Yearly economic savings (USD)
A_{PV}	Photovoltaic module area (m ²)	E_{el} Annually electrical energy production (kWh)
A_{SC}	Solar thermal collector gross area (m ²)	$Price_{el}$ Electrical energy price (USD/kWh)
γ	Photovoltaic power loss coefficient by temperature (%/°C)	$Price_{gas}$ Natural gas energy price (USD/kWh)
T_{PV}	Photovoltaic cell temperature (°C)	R Discounted sum (USD)
T_{PVT}	Photovoltaic cell temperature of PV/T module (°C)	r Capitalization rate (%)
		n Useful life of the project (years)

production of electricity and thermal energy (Tripagnagnostopoulos et al., 2002). With this, it is possible to decrease the operating temperature of the PV modules, increasing their electrical efficiency, while taking advantage of the heat removed for another process, such as domestic, commercial and/or industrial. PV/T (photovoltaic/thermal) collectors are a very promising technology, but need further development in terms of overall efficiency and cost reduction (Sultan and Ervina Efsan, 2018). There are different ways of coupling these two technologies. The heat-carrying fluid used can be a liquid or gas, and the collectors can be based on flat plate, concentrating or other designs (Jia et al., 2019; Yazdanifard and Ameri, 2018), although there are a large number of subcategories due to the new technological proposals that have emerged (Chauhan et al., 2018; Joshi and Dhoble, 2018; Kumar et al., 2015; Tyagi et al., 2012; Zhang et al., 2012).

The hybridization of thermal and photovoltaic technologies has the following advantages (Jia et al., 2019; Ramos et al., 2017):

- The area covered by PV/T collectors is less than that required by the separate systems.
- Installation costs can be reduced by using structure for only one system.
- PV/T collectors are more efficient globally than conventional PV systems and thermal collectors by offering power generation and heating services separately.
- They can easily be integrated with other energy technologies (conversion, storage, etc.).

There is a wide variety of applications for PV/T systems, mainly for low temperature. The integration of PV/T systems into buildings has taken great interest today because two services can be provided, electrical and thermal (Mishra and Tiwari, 2020), necessary for adequate comfort in the facilities, using a smaller area than would be required by individual systems. Herrando et al. (2019) analyzed the use of these hybrid technologies on a university campus to produce heating, cooling, and electricity. Gagliano et al. (2019a) compared the performance of PV/T systems against separate equipment used in buildings where installation area is limited. The collector consists of a PV module incorporating a coil type heat exchanger at the backside. Using TRNSYS software, they found that hybridization of these technologies can increase energy production by up to 8.5% compared to conventional PV and thermal modules. Khordehgah et al. (2019) examined the domestic water heating potential of a PV/T installation in the United Kingdom. Jouhara et al. (2016) experimented on a prototype PV/T collector

installed on the roof of a house. The heat exchanger coupled with the PV module is a multi-channel flat heat pipe that allows efficient heat transfer. They found that, with the technology analyzed, electrical efficiency can be increased by up to 15% compared to systems without uncooled PV. They also proved its potential as a water heating system (Jouhara et al., 2017). Among the works that evaluate both energetically and economically the PV/T collectors in the domestic sector, Herrando et al. (2018a) proposed a flat-box PV/T collector for domestic applications, and evaluated its techno-economic performance under the operating conditions and energy costs of the cities of Athens (Greece), London (UK) and Zaragoza (Spain). Their results show that the optimized system is capable of satisfying up to 65% of the annual electricity demand of homes, while providing 60, 30 and 45% of the required thermal energy, respectively. The techno-economic analysis they presented shows that the geographic location and climate have a significant effect on the performance and economic feasibility of their proposal, as well as being particularly sensitive to the utility prices, specific to each region. In another work more focused on the economic aspect, Herrando et al. (2018b) consider possible economic incentives and fluctuations in the cost of energy in the three cities mentioned in the previous study, and their effect on the economic feasibility of the same collector. They recommend that government incentives be promoted and properly applied to make their implementation attractive to households in the short-term. This paper extends the efforts of the previous work by considering five cities that have not been studied in the state of the art, which contemplate a wide range of climatic conditions and energy costs. In addition, a relationship is presented between the climatic conditions of the study site and the performance of the system.

Bombarda et al. (2016) experimentally compared the performance of PV/T systems with roll-bond absorbers on the back, with and without thermal insulation. They found that with this type of heat exchanger, it is possible to increase thermal efficiency by 10% compared to a PV/T collector with tube and plate heat exchanger. Electricity production even increased slightly. Kuo et al. (2019) conducted studies on a bifacial PV/T collector and found economic benefits and reduced carbon footprint compared to conventional PV and PV/T technologies over a typical 20-year lifespan. Recently, with the rise of nanofluids due to benefits in heat exchange systems, a remarkable amount of studies have been conducted incorporating them into PV/T solar collectors (Al-Shamani et al., 2014; Elsheikh et al., 2018; Said et al., 2018; Yazdanifard et al., 2017).

On the other hand, hybrid PV/T systems with concentrator geometry have shown an increase in solar energy capture and potential economic

ANEXO F. TECHNO-ECONOMIC ANALYSIS OF HYBRID PV/T SYSTEMS UNDER DIFFERENT CLIMATE SCENARIOS AND ENERGY TARIFFS

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Table 1

Previous studies of PV/T systems under different scenarios.

Authors	Study locations	Different climates	Economic study	Remarks
Braun et al. (2020)	Moscow, Stuttgart and Dubai	✓	✓	Its application was to be a heat source and heat sink for a heat pump, not for DHW.
Chen et al. (2019)	Guangzhou	×	✓	Studied the behavior of a trigenerative system activated with PV/T collectors under a subtropical climate.
Salem Ahmed et al. (2019)	Egypt	×	×	Analyzed the operation of a PV/T collector under a hot climate in the city of Egypt.
Bigorajski and Chwieduk (2019)	Athens	×	×	Studied energetically under the climatic conditions of Athens in mid-October, considering that these conditions are similar to those of mid-August in Warsaw (Poland).
Hossain et al. (2019)	Kuala Lumpur	×	×	Performed an energy analysis on a prototype PV/T collector under a single climate.
Barbu et al. (2019)	Bucharest and Strasbourg	✓	×	Compared a commercial PV/T module energetically under similar climatic conditions, without incorporating the economic aspect.
Herrando et al. (2018a, 2018b)	Zaragoza, London, Athens	✓	✓	The study focused on the economic aspect, considering government incentives and fossil fuel price fluctuations.
Ramos et al. (2017)	Ten different locations in Europe.	✓	✓	The cost for auxiliary energy and its effect on economic feasibility were not considered.
Ouyang et al. (2017)	Ten different cities in China	✓	×	Analyzed the electrical connection modes between the PV/T collectors, determining the optimal one for each radiation zone.
Aste et al. (2016)	Milan	✓	×	Experimentally evaluated a PV/T collector in Milan, and extrapolated the results through TRNSYS to cities such as Paris and Athens. The study was limited to an energy analysis.
Al-Shamani et al. (2016)	Malaysia	×	×	Compared a PV collector against a PV/T under the Malaysia tropical climate conditions.
	Tunisian	×	×	

Table 1 (continued)

Authors	Study locations	Different climates	Economic study	Remarks
Hazami et al. (2016)				Experimentally evaluated a PV/T collector and extrapolated the energy results under typical climate area of Tunisia by using TRNSYS.
Herrando et al. (2014) and Herrando and Markides (2016)	London, UK.	×	✓	Different configurations were studied for a typical house in London.

benefits (Coventry, 2005). In the review conducted by Sharaf and Orhan (2015) they found that overall efficiencies of up to 85% have been achieved (George et al., 2019), temperatures of the heat-carrying fluid at the outlet of the collector up to 196 °C (Zhao et al., 2011) and costs as low as 2.37 \$/W_e (Li et al., 2011). With the temperatures that the thermal fluid can reach in this hybridization, it is possible to contemplate new scenarios of application of this technology, such as medium and high temperature industrial process heat, activation of thermodynamic cycles such as the organic Rankine cycle, and high temperature desalination. PV/T systems are still emerging technologies with a small market (Herrando and Markides, 2016). However, a large number of studies show that, with an appropriate installation area and operating strategy, PV/T systems can satisfy a significant amount of energy with a reasonable return on investment (Wang et al., 2019). In the report presented by the International Energy Agency, within its Task 60 related to PV/T systems, the most representative PV/T projects in Europe were shown, as well as the current state of companies and total installed capacity, where a great increase in this sector is appreciated during the last years (Ramschak, 2020).

It is important to mention that not always the use of the thermal energy of the PV module benefits its electrical efficiency. Usually, when looking to use the heat obtained from the PV/T module in some application such as domestic hot water (DHW), the PV cell of the module will have a higher surface temperature than it would have without thermal integration, due to the temperature level needed for DHW (Kalogirou, 2001; Kalogirou and Tripanagnostopoulos, 2006), so their electrical conversion efficiency is in some cases impaired.

Factors such as ambient temperature, available solar radiation, application temperature and the local cost of energy are fundamental aspects in the decision making of this type of facilities. However, most of the studies of this type of technologies focus on the energy and/or economic part of similar locations, so it is necessary to focus on those regions where extreme climates can be contrasted, due to their high and low environmental temperatures, as well as their levels of solar radiation. In addition, it is necessary to contemplate the economic aspect, since not always for having an excellent solar resource it is meaning of economic feasibility. Table 1 shows a brief description of some recent work related to PV/T technologies.

It can be seen that most works focus on the energy or economic aspect, and few under a wide range of climates or considering the effect of the cost of auxiliary energy on the technical feasibility.

Therefore, this paper presents the study of a PV/T system installed in five cities with different climatic conditions and energy tariffs, analyzing the effect of these on technical and economic feasibility. The selected cities are Mexicali (Mexico), Soria (Spain), Bigene (Guinea-Bissau), Fresno (USA) and Madison (USA), which present different climatic conditions and energy prices, so a comparison of this type can identify those regions where certain factors determine whether the installation

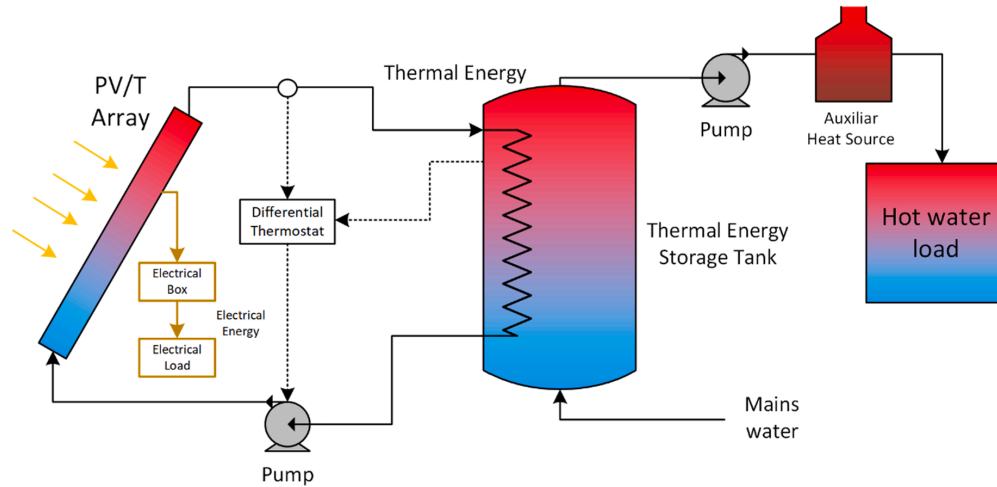


Fig. 1. Schematic diagram of the PV/T system studied.

of PV/T systems is attractive or not. The main contribution is reflected in the wide range of scenarios analyzed, which allow for in-depth comparisons in energy and economic terms. When contemplating cities with extreme climates, such as Madison with very low temperatures and, on the other hand, Mexicali with very high ambient temperatures, as well as intermediate climates (Soria, Bigene and Fresno), the energy results will determine which are the most attractive. However, when considering the installation costs and, much more importantly, the energy cost related to the implementation of PV/T technologies, the contrast with the energy feasibility is interesting. Therefore, our work proposes an extended analysis with different sites, and complements and extends the scope studied in previous works, such as those presented in [Herrando et al. \(2019, 2018a\)](#), to mention some, where the climatic conditions, as well as such disproportionate prices of electric and thermal energy, allow us to analyze a wide range of assumptions that will help decision-making. In addition, a relation of the climatic conditions of the studied sites and their effect in the performance of the PV/T collectors is presented. A comparison of PV and PV/T electrical production is also presented, seeking to analyze the effect of the operating temperature of this application.

[Section 2](#) of this work describes the selected PV/T system, where the characteristics of the main equipment are presented; [Section 3](#) shows the methodology used, a description of the simulation tool and its operation, the mathematical models considered for the analysis of the results and the description of the cities selected for the study; [Section 4](#) presents the technical and economic results and concludes in [Section 5](#).

2. System description

The scheme of the system studied can be seen in [Fig. 1](#). The system consists of an array of PV/T modules connected in series and/or parallel, depending on the needs of the application, working in an independent circuit. The thermal energy captured by the module field is transferred by means of a heat exchanger immersed inside a thermal energy storage tank with a capacity of 200 L. This tank contains water, which is heated with the energy collected for use in a particular application. It also incorporates an auxiliary heater for periods when the stored energy is not sufficient to meet the demand for hot water. The turning on and off of the solar modules field pump is carried out by means of a differential thermostat, which senses the output temperature of the solar circuit and the temperature of the upper part of the storage tank. The flow of water into the thermal storage tank is the same as that extracted from it, to

Table 2
Characteristics of the PV/T module ([DualSun, 2019](#)).

Thermal data	
Aperture area	1.65 m ²
Optical efficiency (a_0)	47.2%
First order heat loss coefficient (a_1)	9.1 W/K/m ²
Second order heat loss coefficient (a_2)	0 W/K ² /m ²
Working fluid	Water
Maximum operating temperature	80 °C
Maximum operating pressure	1.2 Bar
Pressure drop per module	6 kPa a 200 L/h
Thermal storage tank volume	200 L
Heat loss coefficient	0.83 W/m ² K
Electrical data	
Photovoltaic cell type	Monocrystalline
Cells number	60
Nominal power	280 W
Module efficiency	16.87%
MPP voltage	31.64 V
MPP current	8.87 A
Open circuit voltage	39.16 V
Short circuit current	9.46 A
NOCT	45 °C
V_{oc} thermal coefficient	−0.31%/K
I_{sc} thermal coefficient	0.045%/K
Efficiency loss by temperature	−0.41%/K

meet the water demands of the application. This water enters at a temperature that depends on the region and period of the year. The electrical energy produced by the PV/T modules is taken to the junction box and protections, and then supplied to the load and/or electrical network. The auxiliary system is switched on when the upper temperature of the storage tank is less than 45 °C, minimum temperature for DHW. The energy supplied by the auxiliary burner is that necessary to bring the water to the set-point temperature, in this case 45 °C.

The simulated hybrid PV/T module was based on the product of the French company DualSun. The module features presented by the manufacturer can be seen in [Table 2](#).

The maximum operating temperature of the PV/T collector is 80 °C, as mentioned in [Table 2](#). The setpoint of 45 °C set is to control the auxiliary power system, i.e. when the output temperature from the storage tank to the DHW is lower than the setpoint, the auxiliary system will switch on and raise its temperature to this level. The temperature of the storage tank, depending on the operating conditions, can be above 45 °C as it is intended to store the largest amount of thermal energy, in

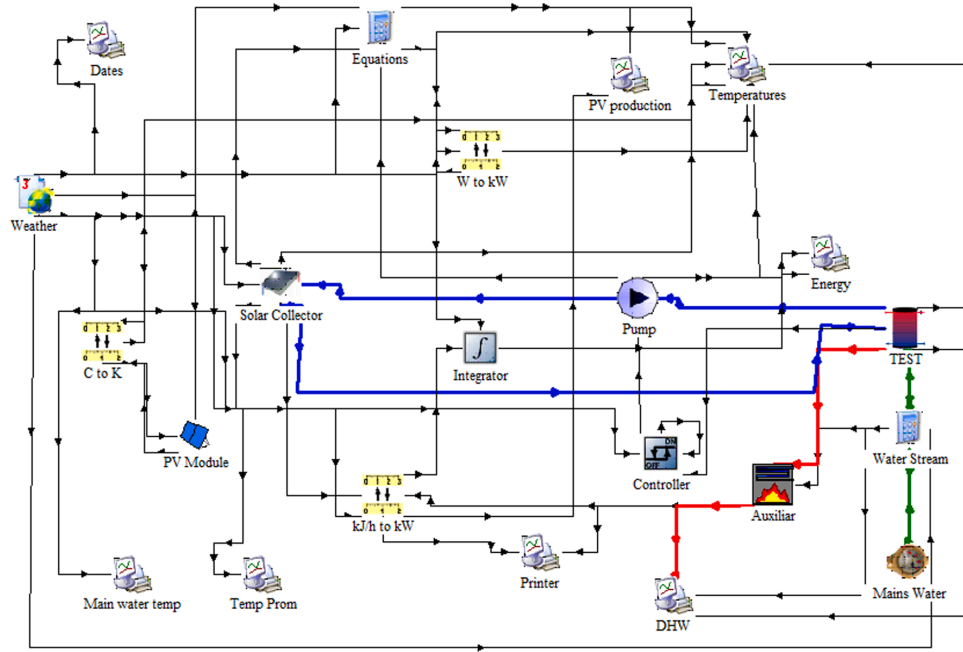


Fig. 2. PV/T system simulator developed in TRNSYS.

Table 3
Types used in the simulation in TRNSYS.

Type	Model
1b	Flat plate solar thermal collector
94a	Photovoltaic module
60d	Thermal storage tank
3d	Pump
2b	Differential temperature controller
700	Boiler
14b	Water consumption profile
24	Integrator
9a	Information provided by the user
15	Weather data processor
57	Unit conversion
65a	Print and data storage

our case in the form of sensible heat. Therefore, our tank will not exceed 80 °C, which is the maximum temperature of the collector. In case the temperature conditions of the tank are at this level, the control inside our simulator forces to turn off the circulation pump and leaves the PV/T collector in stagnant conditions, and turns on the pump again when the temperature level of the tank decreases below this value. It should be mentioned that this control strategy is for simulation purposes only. In practice, the right strategy is to dissipate thermal energy from the storage tank in order to lower its temperature. If, on the other hand, the PV/T collectors are left in a stagnant condition, there is a risk of problems due to the increase in pressure, as well as the degradation of the PV/T collector materials.

3. Methodology

The simulation study was carried out using the TRNSYS® software (Klein, 2015) for the modeling of the system in general. This software has the advantage of having programmed models, called “types” of a large number of equipment, within which can be found photovoltaic

modules, solar thermal collectors, storage tanks and other renewable technologies, to mention some. One of the main features of TRNSYS is that it uses a typical meteorological database (TMY) within its simulation process, where variations in radiation, wind speed, ambient temperature, loads, among many others, are contemplated, allowing the system to be studied under real operating conditions. Fig. 2 shows the simulator developed and Table 3 shows the different types used.

The simulation methodology presented by Gagliano et al. (2019a, 2019b) were used, where, to model a hybrid PV/T module, they used the types of a flat-plate thermal solar collector and the photovoltaic module operating in parallel. Other works have also used this methodology, but with types developed by the authors, to simulate the behaviour of a PV/T collector (Jonas et al., 2019; Lämmle et al., 2017). In TRNSYS it is not a requirement to directly connect the types to exchange information between them, this can be done with previously declared variables. In our case, with the Type Equations the calculations are made and the results are sent as input variables to the different types, such as that of the solar thermal collector.

For the above, the incident energy on the thermal collector G_T is calculated with the Eq. (1):

$$G_T = G(1 - \eta_{el}) \quad (1)$$

where G is the solar radiation incident on the PV module and η_{el} its efficiency. The energy that reaches the thermal solar collector is the solar radiation incident on the PV module minus the energy it converted to electrical energy because of its efficiency. In turn, the efficiency of the PV module is estimated based on the Eq. (2):

$$\eta_{el} = \frac{P_{el}}{A_{PV}G} \quad (2)$$

where P_{el} is the power produced by the PV module and A_{PV} the solar collection area. Gagliano et al. (2019b) proposes to calculate the photovoltaic efficiency according to the temperature of the cell T_{PV} , according to Eq. (3)

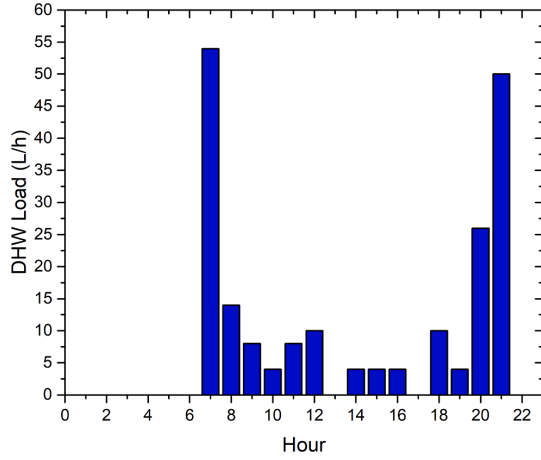


Fig. 3. DHW demand profile used in simulation.

$$\eta_{el} = \eta_{PV,RC} [1 - \gamma(T_{PV} - T_{RC})] \quad (3)$$

where $\eta_{PV,RC}$ is the efficiency of the PV module at the reference conditions (nominal), γ is the power loss of the PV module in %/K due to the T_{PV} , and T_{RC} is the ambient temperature at the reference conditions. For T_{PV} calculation the equation (4) is used, presented by Skoplaki et al. (2008) and Skoplaki and Palyvos (2009a, 2009b):

$$T_{PV} = 30 + 0.0175(G - 300) + 1.14(T_a - 25) \quad (4)$$

Eq. (4) applies only to temperatures of the polycrystalline silicon photovoltaic cell. T_{PV} will be different for a PV module than for a PV/T module due to the removal of heat from the PV surface by the heat-carrying fluid of the thermal integration. To calculate the T_{PVT} of the hybrid module, Eq. (5) proposed by Kalogirou and Tripanagnostopoulos (2006) is used:

$$T_{PVT} = T_{PV} + \left(\frac{T_{in} + T_{out}}{2} - T_a \right) \quad (5)$$

where T_{in} , T_{out} and T_a are the inlet and outlet temperatures of the thermal collector and the ambient temperature, respectively.

Having the efficiency values of the PV module, it is possible to know G_T and calculate the efficiency of the thermal collector η_{th} according to the Eq. (6) and the thermal power P_{th} of the same with the Eq. (7):

$$\eta_{th} = a_0 - a_1 \frac{T_{in} - T_a}{G_T} - a_2 \frac{(T_{in} - T_a)^2}{G_T} \quad (6)$$

$$P_{th} = A_{SC} G_T \eta_{th} \quad (7)$$

Within the software it is possible to use the Eq. (6) according to the input, output or mean temperatures, depending on the availability of the information on the characteristics of the solar collector. It is important to differentiate between P_{th} and $\dot{Q}_{DHW}$; the first is the amount of thermal energy generated by the solar collector field, while the second is the heat demand set by the users to meet DHW requirements, due to the difference in temperature between the setpoint and the mains water temperature. The solar fraction represents the thermal energy contribution of the PV/T system to the DHW heat consumption, and is calculated based on the equation (8):

$$f_{th} = \frac{\dot{Q}_{DHW} - \dot{Q}_{aux}}{\dot{Q}_{DHW}} \quad (8)$$

where $\dot{Q}_{DHW}$ is the thermal energy required to meet the needs of DHW (demand set by the users) and $\dot{Q}_{aux}$ the extra energy supplied by the boiler, after the solar contribution, to meet the demand for DHW.

The hourly consumption profile for DHW, specified in the standard EN 15316-3-1, corresponding to the Tapping program No. 3, or the Load profile L of the standard EN 13203-2, with an approximate consumption of 200 L per day of hot water, was used in this study and can be appreciated in the Fig. 3. This profile is considered for a single family. It was used in all the case studies to have an equivalent point of comparison between the different scenarios, in terms of hot water supply needs.

3.1. Study cities

In order to analyze the effect of hybrid PV/T systems under different climate scenarios, where solar resource and ambient temperature allow a comparison between different regions, as well as different costs of thermal and electrical energy, five cities were selected: Mexicali (Mexico), Soria (Spain), Bigene (Guinea-Bissau), Fresno (USA) and Madison (USA), for the study. The meteorological information for each city corresponds to a typical year (TMY file). Table 4 shows the main characteristics of the cities analysed, as well as the price of electrical and thermal energy. Since the five cities studied have different ambient temperatures and solar radiation, to mention some main features, the thermal energy generation requirements by the PV/T system to meet DHW demand will be different, so the importance of quantifying energy requirements in a wide climatic range. The annual thermal energy required for DHW was calculated based on the thermal energy that

Table 4
Cities selected for the study and main energy characteristics.

	Mexicali	Soria	Bigene	Fresno	Madison
Latitude and longitude	32.6, −115.4	41.7, −2.4	12.4, −15.5	36.7, −119.7	43.0, −89.3
Köppen-Geiger climate classification (Kottek et al., 2006)	BWh	Cfb	Aw	Bsk	Dfa
Annual thermal energy required for DHW (kWh)	1993	3223	1572	2589	3690
Annual irradiation (MWh/m ²)	2.1	1.62	1.94	1.88	1.42
Average annual temperature (°C)	23.37	12.08	27.23	17.9	7.72
Thermal energy price (USD/kWh) ^{**}	\$ 0.0359 (Naturgy, 2019)	\$ 0.0980 (Eurostat, 2019)	\$ 0.0388 ^a	\$ 0.0417 (EIA, 2019)	\$ 0.0429 (EIA, 2019)
Electrical energy price (USD/kWh)	\$ 0.1436 (CFE, 2019)	\$ 0.2774 (Eurostat, 2019)	\$ 0.1713 ^a	\$ 0.1990 (Electric-Choice, 2019)	\$ 0.1428 (Electric-Choice, 2019)

^{*} It was calculated using the simulator, taking into account the climatic conditions of the region and the demand profile of the Fig. 3.

^{**} Natural gas price.

^a An average between Mexicali and Fresno costs was used due to the public absence of the energy price.

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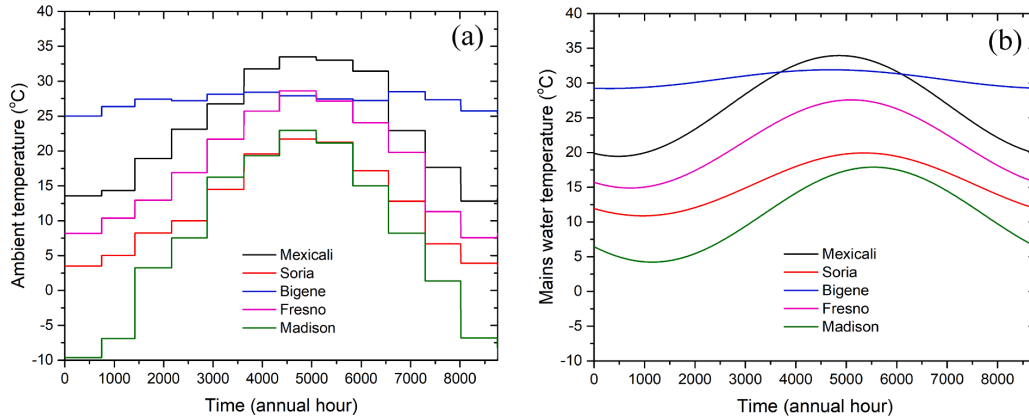


Fig. 4. (a) Average monthly temperature and (b) annual mains water temperature for cities considered in the study.

Table 5

Equipment and PV/T system installation costs (Gagliano et al., 2019a; Herrando and Markides, 2016; Wang et al., 2019).

Panels and structure (USD/m ²)	336
Inverter and accessories (USD/W)	0.27
Hydraulic circuit (USD/m ²)	112
Solar tank (USD)	918
Installation cost (USD)	1120

would be required, using a conventional heater, to bring the mains water temperature to the setpoint temperature. Because of this, and because the mains water temperature at each location is different throughout the year, the thermal energy requirement to cover DHW differs for each city.

To show in a more complete way the climatic scenarios studied, Fig. 4a and b presents the monthly ambient average and annual mains water temperatures of the five selected cities. This information was generated with Meteonom software and used in the TMY format. The ambient and mains water temperature, as well as the availability of the solar resource, determine the thermal energy needs to satisfy DHW. Unlike the average annual temperature in Table 4, it can be seen that Mexicali has higher temperatures than Bigene, since it presents more notable deviations in summer, where it can reach up to 50 °C in the shade (Aguilar-Jiménez et al., 2018). Bigene has a stable ambient temperature throughout the year, that is why it was considered in this study.

On the other hand, cities with very cold climates were also analyzed, as is the case of Madison, where its winter can reach average temperatures of up to −10 °C. With the remaining cities, Soria and Fresno, which present intermediate temperatures, a wide range of climatic scenarios is covered, and considering the energy tariffs so variable in the localities, it is possible to analyze the effects of both factors on the feasibility of these projects.

3.2. Economical analysis

In order to make a comparison of the economic feasibility of this type of installation in the different study cities, an economic analysis was carried out, taking into account the price of energy (electric and thermal) and the projected savings during the useful life of the system. With Eq. (9) the annual monetary savings, R_y , are calculated due to the energy that was no longer consumed due to the production of the PV/T modules (Gagliano et al., 2019a):

$$R_y = E_{el} \cdot Price_{el} + \left(\dot{Q}_{DHW} - \dot{Q}_{aux} \right) \cdot Price_{gas} \quad (9)$$

where E_{el} is the electrical energy produced by the PV/T and $Price_{el}$ and $Price_{gas}$ is the cost of electrical and thermal energy per kWh, respectively.

Their discounted sum, R , is calculated by the Eq. (10) (Gagliano et al., 2019a):

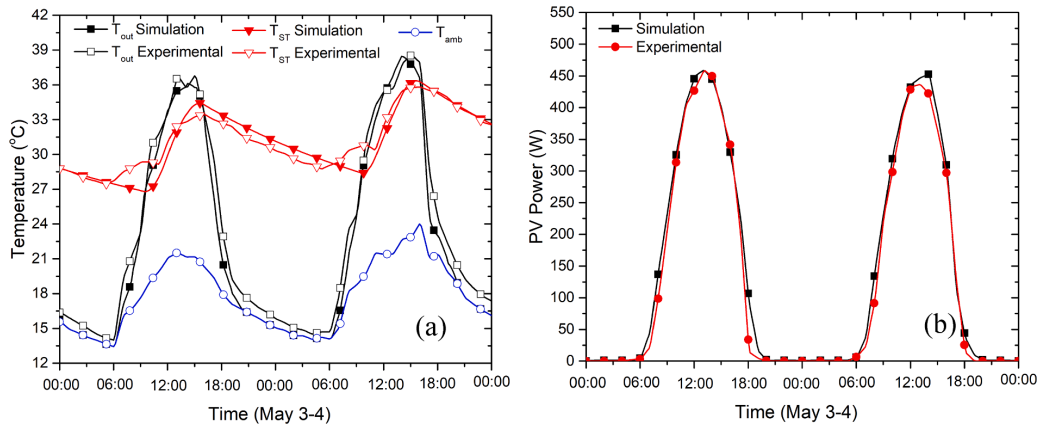


Fig. 5. Comparison of temperatures (a) and electrical power (b) resulting from simulation against experimental.

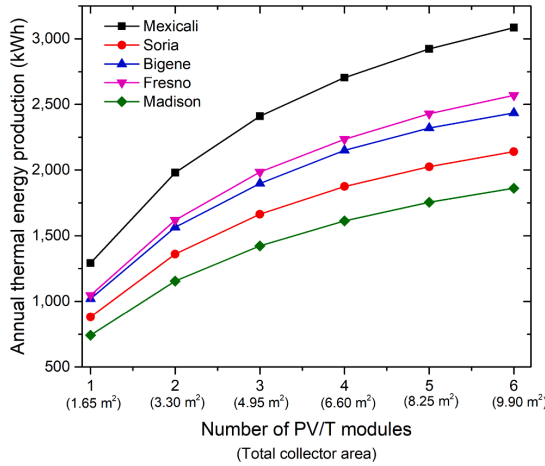


Fig. 6. Annual production of thermal energy with different number of PV/T modules.

$$R = \sum_{i=1}^n R_i \frac{q^i - 1}{r \cdot q^i} \quad (10)$$

It is considered a capitalization rate, r , of 3% and a projection of the useful life of the project, n , of 20 years, and $q = 1 + r$. Table 5 shows the costs considered for equipment, materials and installation of the PV/T system.

3.3. Simulation validation

The experimental information presented by Gagliano et al. (2019b) was used for numerical validation of the developed simulator. The simulation parameters were established as described by the author, using two PV/T modules connected in series. For the simulation, the solar radiation and ambient temperature present during the experimental tests were considered. Fig. 5a shows the comparison of the output temperatures of the PV/T modules, T_{out} , and Fig. 5b the electrical powers during August 3 and 4, both simulated and experimental. T_{out} and T_{ST} (thermal storage tank) have an acceptable relationship to the experimental temperatures, while the power generated has a similar tendency to the experimental one. A standard deviation of 0.75 °C and 13.17 W was obtained in the output temperature and power generation, respectively, of the simulator with the experimental.

4. Results

4.1. Energy analysis

The thermal and electrical energy production of the PV/T modules is directly related to the availability of the solar resource and the ambient temperature of the study site. Fig. 6 shows the production of thermal energy with different numbers of PV/T modules installed for the five locations studied. It can be seen that Mexicali, the city with the highest irradiation compared to the remaining cities, produces a greater amount of thermal energy throughout the year. Madison produces the least amount of thermal energy annually due mainly to its low levels of irradiation and ambient temperature, which causes a greater loss of energy from the PV/T modules. It is noted that by increasing the number of hybrid modules, the increase in thermal energy production is not proportional. In the case of Mexicali, one module produces around 1290 kWh of thermal energy per year, while with two modules it increases to 1980 kWh, equivalent to an increase of 53% and, when increasing from

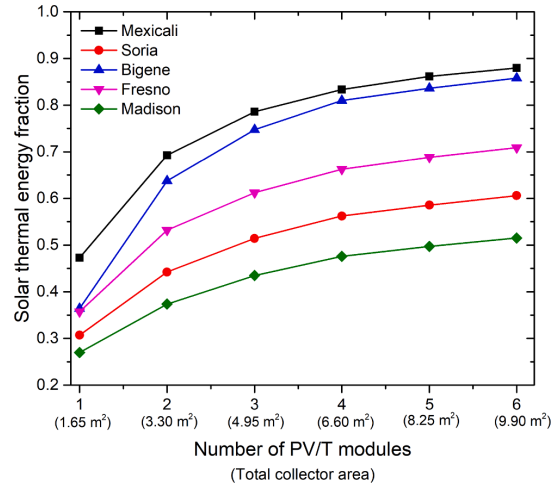


Fig. 7. Solar thermal fraction with different number of PV/T modules.

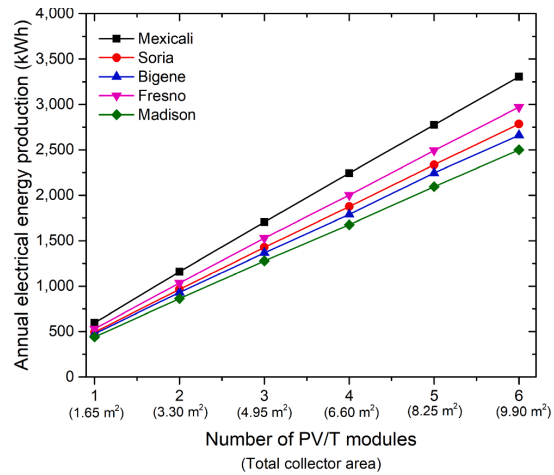


Fig. 8. Annual production of electrical energy with different number of PV/T modules.

two to three modules, it means an increase of 21%. Doubling the number of hybrid modules from three to six results in an increase of only 28%. The same is also the case for the rest of the cities, even with smaller increases in thermal production. This is mainly due to two situations: the first is that, as more PV/T modules are installed, the operating temperature of the system rises and the modules are less efficient; and the second is due to the fact that DHW's needs are being met and thermal energy production is being wasted. It should be mentioned that the hydraulic arrangements of the PV/T collectors were considered in series to achieve the required temperature level for DHW.

The contribution of the thermal energy generation of the PV/T modules to the demand for DHW, calculated on the basis of the solar fraction, is presented in Fig. 7. For Mexicali, a single module represents 47% of the thermal energy required for DHW, while for Madison it is 27%. A notable increase in the solar fraction can be seen when using two modules. However, the upward trend decreases as more than two modules are used. By increasing from four to five and from five to six PV/T modules, the increase in the solar fraction is only 2.5% and 2%,

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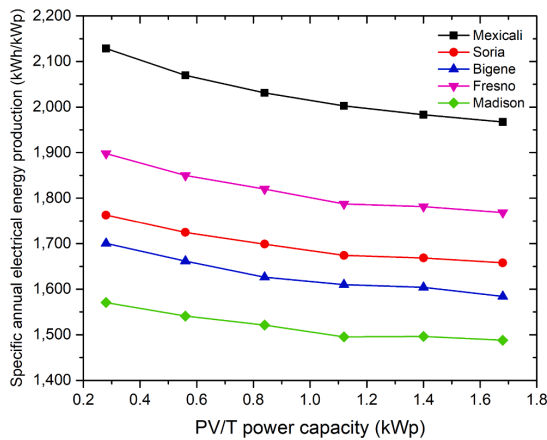


Fig. 9. Specific electricity production of PV/T modules in relation to installed capacity.

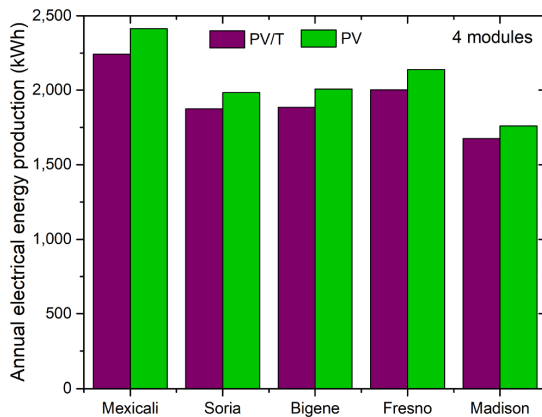


Fig. 10. Comparison of electrical output of PV/T and PV modules.

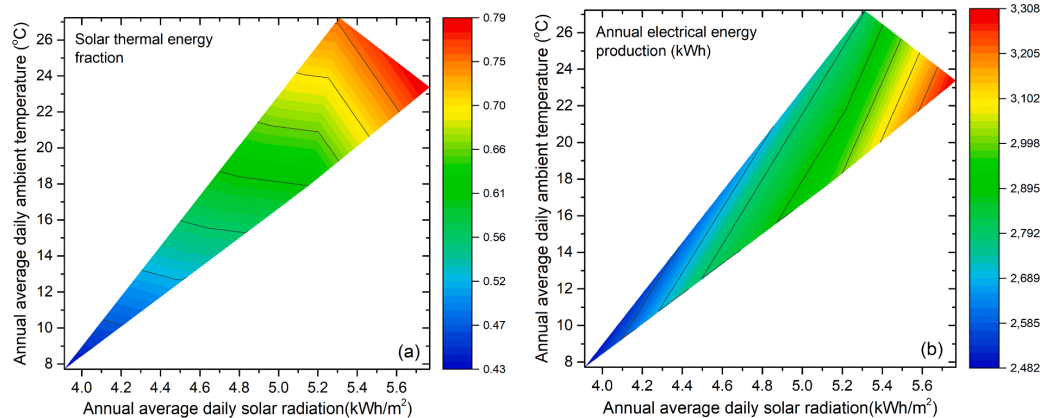


Fig. 11. Effect of the annual average daily solar radiation and ambient temperature on (a) solar thermal energy fraction and (b) annual electrical energy production.

respectively. This is largely due to the fact that at times of greatest demand for DHW there is not enough solar resource to fully satisfy the heating needs, so that, no matter how much we add modules, their energy contribution to demand will not be representative. For a greater contribution of solar energy, the DHW demand patterns must be adjusted to coincide with the periods when there is a greater solar resource.

The annual electricity production with the PV/T modules is shown in Fig. 8. It can be seen that, unlike thermal energy, photovoltaic energy presents an increase in its electricity production proportional to the number of PV/T modules. Mexicali produces a greater amount of energy, while Fresno ranks as the second-best electricity producer, although its annual irradiation level is lower than Bigene. This is mainly due to the environmental temperature of the locality which is colder, making the electrical conversion more efficient. It is worth mentioning that the operating temperature of the thermal system will affect the electrical energy production of these modules due to the fact that it will increase the temperature of the cell, decreasing its efficiency. Therefore, the greater the number of PV/T modules in the system, the higher the operating temperature of the system, and therefore a lower photovoltaic production per PV/T collector.

The effect of PV/T capacity on specific electricity generation is shown in Fig. 9. The decrease is directly related to the increase in PV cell temperature from operating at a higher fluid temperature. Mexicali presents a decrease of 7.6% in the annual electrical energy production per kW installed by increasing from 0.28 to 1.7 kW of PV/T power, while the average among the rest of the cities is around 6.2%. Despite being the city most affected by the operating temperature, it continues to be the one with the highest energy production, compared to the rest of the cities.

An important aspect in the production of electrical energy by means of PV modules is the operating temperature. Although Fig. 9 shows how the local ambient temperature can influence the electrical output of the PV/T modules, it will be different from the one that would produce PV modules working independently of the thermal system.

Fig. 10 shows a comparison in photovoltaic production between four PV/T and PV modules for the different cities in the study. During the comparison, the characteristics of the PV module used were identical to those of the PV module incorporated into the PV/T, which can be seen in Table 2. In all localities the electrical production of the PV/T modules is lower, mainly due to the increase in the temperature of the PV cell due to the use of thermal energy. For Mexicali it represents an 8% decrease in electricity production, while for Madison it is 5%. This can be considered as the electrical cost to take advantage of the thermal energy of the PV/T module.

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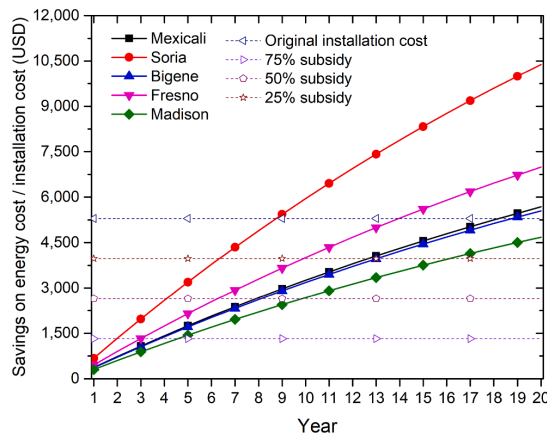


Fig. 12. Economic analysis projected over 20 years of the economic savings of a four-module hybrid PV/T system for the different cities. It also shows the time of return of the investment considering the original installation cost and government subsidies of 25, 50 and 75% of the initial investment cost.

With the values of annual solar radiation and average annual ambient temperature of all the studied locations, a relation was made between the solar fraction and the electric power generation of the PV/T collectors. The results are shown for a capacity of 3 PV/T collectors. It can be seen that there is a relationship between radiation levels and ambient temperature in the amount of solar energy contributed to the DHW demand; a location that presents annual average daily radiation levels above 5.4 kWh/m² can reach a solar fraction of up to 0.79 when its annual average daily temperature is above 22. This is because the higher the ambient temperature present at the location, the lower the thermal energy requirements to meet DHW demand, so the solar fraction increases. On the other hand, contrary to the solar fraction, the annual electricity production is affected by the high ambient temperatures at the system location, mainly due to the operating temperatures and their effect on the efficiency of the PV. However, it can be seen that the availability of solar radiation from the site has a greater impact on electricity generation than the ambient temperature, unlike the solar fraction which is more affected by the ambient temperature.

It is important to mention that the previous results have a scope on the locations studied in our work, as well as other locations with similar climatic values. As shown in Fig. 11, there is still a wide range of values where no information is available because the characteristics of the five sites studied are not within those values of ambient temperature and solar radiation. However, there is a clear trend in the effect of climatic variables on system performance. In future work, it is proposed to analyze more locations that will allow to associate the performance of the PV/T systems with the climatic variables of the study site with a greater scope.

4.2. Economical results

Fig. 12 shows the economic analysis for a four-module PV/T system in the different locations. It can be seen that, unlike the previous graphs where the thermal and electrical production was shown, the city of Soria becomes the main locality in terms of economic benefit from the installation of the PV/T system. The cost of electric and thermal energy determines the economic feasibility of this type of installation. This can be clearly seen in the case of Mexicali which, due to its climatic conditions and solar resource, is the most attractive energy location, followed by Bigene, Fresno, Soria and finally Madison. However, when considering the effect of monetary savings related to the cost of energy for each location, Soria takes the lead due to its high cost of thermal and

Table 6

Economic savings during the useful life of the PV/T system without considering subsidy.

Modules	Mexicali	Soria	Bigene	Fresno	Madison
1	\$ -1,076.4	\$ 627.7	\$ -1,309.1	\$ -705.7	\$ -1,282.8
2	\$ -453.4	\$ 2,396.5	\$ -717.3	\$ 254.5	\$ -953.4
3	\$ -0.9	\$ 3,823.1	\$ -322.4	\$ 1,028.1	\$ -743.1
4	\$ 382.4	\$ 5,083.0	\$ 252.4	\$ 1,693.9	\$ -617.7
5	\$ 735.7	\$ 6,279.9	\$ 370.7	\$ 2,377.3	\$ -490.6
6	\$ 1,071.3	\$ 7,415.2	\$ 634.0	\$ 3,008.6	\$ -401.4

electrical energy, followed by Fresno, Mexicali, Bigene and finally Madison again. Mexicali will be able to produce a greater amount of energy, but its electricity and natural gas tariffs harm the attractiveness of installing these systems. Fig. 12 also presents an analysis of the return on investment with the savings generated by the energy production of the PV/T system. This is represented by the intersection between the original installation cost and savings on energy cost. Considering that the end user will cover 100% of the total cost of the installation, Soria has a return time of 9 years, followed by Fresno with 14 years, Mexicali and Bigene 18.5 years approximately and Madison does not recover the investment during the useful life of the system. In addition, scenarios are added where subsidies of 75, 50 and 25% of the initial investment cost are considered. These incentives can enable end users to recover what they invested in the system from 2 years, in the case of Soria and a subsidy of 75%, while Fresno, Mexicali, Bigene and Madison do so from 3, 3.5, 3.5 and 4.5 years, respectively. These scenarios can serve for the establishment of public policies and support programs for the installation of this type of technologies based on renewable energy, promoting energy savings and the reduction of polluting gases, which is sought to promote according to international treaties against climate change. The results of the return on investment time for the cities studied are attached in Appendix A, depending on the number of PV/T modules and the subsidy on the initial investment.

Table 6 shows the projected economic savings during the 20 years of useful life of the PV/T system, for different system capacity and without considering subsidy in the initial investment. The economic savings were estimated with the investment cost minus the savings over the operational 20 years. It can be seen that the cities of Mexicali and Bigene, regardless of the fact that they are the sites studied with the highest annual insolation, do not recover the initial investment in installations of up to three PV/T modules, presenting economic losses. On the other hand, Soria and Fresno are the regions with the greatest economic feasibility in this type of project, partly because of their solar resource (Fresno case), but more importantly because of the cost of energy (Soria case). These results are consistent with the conclusions reached by other authors in the past. In different countries, economic support is considered for the installation of this type of system based on renewable energies, so that, depending on the national context for each installation, it can be more economically attractive and monetary benefits could be obtained from the use of a module, depending on the stimulus. On the other hand, the market for PV/T technologies is constantly growing, there are more companies offering more efficient products, so it would be expected that in the future the costs of these systems would decrease as is happening with PV technology, benefiting the economic savings. The remarkable global decline in the price of photovoltaic technology benefits PV/T modules. However, the economic part related to the thermal collector needs a remarkable reduction so that the whole PV/T technology will be benefited in costs.

5. Conclusions

The study of a hybrid PV/T system for the simultaneous production of electric and thermal energy, installed in five cities with different climate conditions, solar resources and energy tariffs, was presented in this paper. An analysis was made of installed capacity and its effect on

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energy production, influenced by solar resource availability and DHW demand for all locations. The electrical production of the PV/T system was compared with that of an individual PV system and the effect of operating conditions on power generation was tested. This work facilitates decision-making, since extreme climates were considered representative of most of the sites where PV/T systems would be installed. The results show that, in a city with high environmental temperatures and high solar resources, hybrid PV/T systems generate more energy, both electric and thermal, than cities with lower temperatures, and are more technically attractive. However, the economic study concludes that local energy tariffs have a more significant impact on the feasibility of the project than having an excellent solar resource. An analysis of the climatic conditions of the contemplated cities and their possible relationship with the performance of the PV/T collectors was carried out. It can be seen that the availability of solar radiation from the site has a greater impact on electricity generation than the ambient temperature, unlike the solar fraction which is more affected by the ambient temperature. In addition, different scenarios were carried out in which economic subsidies were considered in the initial investment of the PV/T systems, in which the returns of the investment can be between 2 and 4.5 years. These results are completely in line with the results of previous studies, thus extending the scope that was available with respect to PV/T collectors. Moreover, these results can be useful when formulating government policies or support programs for the installation of renewable technologies, promoting energy savings and the reduction of polluting gases.

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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Appendix A

Tables A1–A5.

Table A1

Time in years for the return on investment in Mexicali, considering different capacity and subsidy.

Modules	0%	25%	50%	75%
1	(-)	(-)	15	6.7
2	(-)	16.5	10	4.5
3	20	13.75	8.5	4
4	18	12.65	8	3.75
5	17.5	12	7.5	3.6
6	16.5	11.5	7.25	3.4

Table A2

Time in years for the return on investment in Soria, considering different capacity and subsidy.

Modules	0%	25%	50%	75%
1	15.5	10.8	6.75	3.25
2	10.7	7.75	4.9	2.3
3	9.25	6.75	4.3	2
4	8.7	6.3	4.1	1.9
5	8.4	6.15	3.9	1.85
6	8.2	5.9	3.8	1.8

Table A3

Time in years for the return on investment in Bigene, considering different capacity and subsidy.

Modules	0%	25%	50%	75%
1	(-)	(-)	18	7.75
2	(-)	18.15	11	5
3	(-)	15	9.25	4.25
4	18.6	13	8	3.75
5	18.4	12.85	7.9	3.7
6	17.8	12.3	7.6	3.6

Table A4

Time in years for the return on investment in Fresno, considering different capacity and subsidy.

Modules	0%	25%	50%	75%
1	(-)	20	12	5.5
2	18.25	13.7	8	3.6
3	15.2	10.7	6.7	3.25
4	14	9.75	6.2	3
5	13	9.3	5.8	2.8
6	12.1	9	5.75	2.75

Table A5

Time in years for the return on investment in Madison, considering different capacity and subsidy.

Modules	0%	25%	50%	75%
1	(-)	(-)	18	7.75
2	(-)	(-)	12.1	5.5
3	(-)	17.3	10.5	4.9
4	(-)	16	9.8	4.5
5	(-)	15.25	9.3	4.3
6	(-)	14.8	9	4.25

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G. Study of a Hybrid Solar Absorption-Cooling and Flash-Desalination System

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Article

Study of a Hybrid Solar Absorption-Cooling and Flash-Desalination System

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Abstract: In this work, the analysis of a hybrid LiBr/H₂O absorption-cooling and flash-desalination system, using solar thermal energy as heat source, is presented. An absorption open-cycle with three pressure levels is used in combination with a single-stage flash-desalination process to use the coolant as product water, resulting in an increase in cooling and desalination efficiency. For the application, a 20-room coastal hotel complex in San Felipe, Baja California, Mexico, is taken as a case study and the sizing of the solar collection and storage system is carried out for the operation of the proposed hybrid system, during the summer operative period. The operational dynamics during the week with the highest ambient temperatures are presented. The dimensioning of the solar collector's area and the energy storage resulted in a collection area of 620 m² with 30 m³, respectively, reaching a solar fraction of 69%. The absorption-cooling subprocess showed an increase of 13.88% in the average coefficient of performance (COP) compared to conventional LiBr/H₂O absorption systems. Also, considering that the system provides cooling and desalination simultaneously, the average COP_G is 1.64, which is 2.27 times higher than the COP of conventional LiBr/H₂O single-effect absorption units. During the critical week, the system presented a desalinated water production of 16.94 m³ with an average performance ratio (PR) of 0.83, while the average daily water production was 2406 kg/day; enough to satisfy the daily water requirements of four people in a coastal hotel in Mexico or to cover the basic services of 24 people according to the World Health Organization.

Keywords: solar energy; absorption cooling; desalination

1. Introduction

Future energy technologies must be designed to use clean and renewable sources of energy, as well as waste energy, in order to achieve sustainable development that ensures the well-being of the world's population. Energy generation systems need to consume less energy and be more efficient in order to reduce environmental pollution and increase their profitability.

The increase in population has brought with it a substantial increase in the demand for energy to meet basic needs, such as air conditioning and access to drinking water. In the United States alone, 10% of total annual energy consumption in 2019 was used for space cooling in buildings, equivalent to 380 billion kWh [1], while in other developed countries 30% of the energy produced is consumed by heating, ventilation, and air conditioning systems (HVAC) [2].

On the other hand, seawater desalination is one of the main methods for meeting the needs for drinking water in regions where it is difficult to acquire it by other means [3,4]. For this purpose, there are different technologies, such as multiple-effect desalination (MED), multi-stage flash (MSF), humidification-dehumidification (HDH), reverse osmosis (RO), membrane distillation (MD), among others, that are widely used and with a high level of technological development. However, at present the energy cost of desalinating water is considerably high [5] and, if fossil fuels are used as energy sources, the environmental pollution generated by water desalination is harmful [6]. Renewable energies are a sustainable option to generate the energy required by this type of system, eliminating the problem of pollution and with competitive costs [7–9].

In recent years, great efforts have been made to integrate renewable energies into desalination processes [10] and space cooling [11], and clearly there is a wide use of solar energy, both thermal and photovoltaic, to produce both services in a sustainable way [12,13]. Within desalination, solar thermal energy has been successfully used in MED, MSF, MD, and HDH systems [14], while photovoltaic is undoubtedly an excellent option for RO [15]. This is mainly because there is a relationship between water scarcity and the availability of solar radiation in many regions [16]. On the other hand, absorption-cooling systems are efficiently coupled with solar heating technologies in regions with difficult access to electricity [17,18]. Currently, the trend is in the hybridization of technologies, seeking greater efficiency and less pollution [19], performing energy and mass integrations between the different streams.

Anand & Murugavelh [20] studied a vapor refrigeration system operating with an HDH system for fresh water production. The results mention that the energy consumption may decrease and the cooling capacity increase when the ambient temperature tends to decrease, while at high ambient temperature the water desalination system is seen to benefit, but the cooling capacity is negatively affected. However, in this study the activation energy came from the combustion of fossil fuels. In turn, Qasem et al. [21] analyzed the integration between an HDH desalination system and an absorption-cooling system. The seawater from the HDH is used as a heat removal medium from the absorber and condenser of the cooling system, acquiring sufficient energy to activate the HDH. The optimal cooling capacity was 62 tons and a water production of 1145 L/h. Its results show a 2.2-times improvement in HDH efficiency and an 8.24-times reduction in the cost of fresh water compared to traditional HDH systems. The HDH desalination systems are attractive in low water demand and cooling applications; however, they are severely limited by climatic conditions, especially high levels of air humidity, so their use in large coastal buildings may not be suitable. Another very interesting energy integration is to use the heat of condensation from a power cycle to activate desalination and absorption thermal systems. Kerme et al. [22] analyzed a system for desalination, power generation, and cooling using solar energy. This system consists of a parabolic trough solar concentrator field that generates the thermal energy to activate an organic Rankine cycle, which, in turn, uses the heat of condensation from the working fluid to activate a MED system and an absorption system. With this energy integration it is possible to reduce the thermal energy requirements of process activation.

In India, Sahoo et al. [23] presented a theoretical study of a polygenerative system that produces electrical energy, water, and cooling. A power cycle of steam activated with solar thermal energy and biomass was proposed, where the steam coming out of the turbine is used to activate the first effect of a LiBr/H₂O double-effect absorption-cooling unit, while the heat of condensation from the absorption system is used for the activation of an HDH desalination process. To activate the system, a parabolic trough collector field with an aperture area of 31,339 m² and a biomass system with a capacity of 17.4 MJ/kg were used. A cooling coefficient of performance (COP) of 1.42 was reported, with a cooling capacity of 2805.5 kW and a freshwater desalination of 900 L/h. However, the coupling of these technologies involves a large number of components, increasing the initial investment. Alelyani et al. [24] proposed novel integrations between absorption-cooling systems and MED desalination systems, where the heat from the condenser and absorption rectifier equipment is used to thermally activate the MED in the first effect. Their results show that with the proposed

integration cost of water production increases by 9% with respect to a MED without integration, but with a reduction of 42% in the cost of cooling production. Mohammadi & McGowan [25] analyzed the integration of an absorption and vapor compression cooling system with a MED system and found that the desalination system produces an appreciable amount of water using the steam discarded by the absorption-cooling cycle.

Different authors have studied the application of hybrid systems that include space cooling and desalination in isolated regions. Calise et al. [26] proposed a polygenerative system based on solar energy and biomass to meet the needs of cooling, seawater desalination, heating, and electricity of an isolated small community. In another study, Calise et al. [27] made a modification to their previous system and analyzed it under typical operating conditions of a building on Pantelleria Island. Recently, a solar thermal activated polygenerative process was proposed to provide electrical power, fresh water, and space cooling to a hotel in Eastern Azerbaijan, Iran [28]. In this system, it is proposed to use a Stirling solar dish activated with solar thermal energy to generate electricity for the building; in addition, to activate the HDH desalination process, the heat rejected by the Stirling engine is used and, for space cooling, a vacuum vessel is used to cool a closed air system. On a typical day, the temperature inside the building was 25 °C; the maximum efficiency of the solar Stirling engine was 36.3% and the electrical power output was 38.8 kW. The maximum activation energy was reported to be 97.12 kW, with a cooling capacity of 229 kW and a Gained Output Ratio (GOR) of 2.35. However, controlling the generation of electrical energy in this system is complex and the initial investment may be high.

In the works mentioned above, a trend can be appreciated in proposing systems that produce different services in a more efficient and sustainable way. The hybridization and energy integration of technologies has reduced energy consumption by taking advantage of the waste heat among them. However, it is necessary to continue improving the current technologies, seeking a better energy and mass integration that allows a decrease in energy consumption, as well as in the system components, making use of renewable energy sources available at the project location. Following the global efforts, this paper presents a novel hybrid LiBr/H₂O absorption-cooling and seawater flash-desalination system that uses solar thermal energy as a heat source. This system couples a single-effect absorption-cooling open-cycle with three pressure levels, thus allowing an internal energy and mass integration with a single-stage flash-desalination process. The activation temperature levels are suitable to enable the system to be coupled to a solar thermal collector field to obtain the energy required for its operation. This system is an improvement to the work presented in [29] and is studied under the dynamics of solar resource availability and cooling demand variation of a 20-room hotel complex as a case study.

The description of the hybrid system, as well as the process of cooling and thermosolar desalination, is presented in Section 2. It also mentions the methodology used to carry out the study, defining the simulation considerations and describing the characteristics of the case study, such as its meteorological conditions that affect the operation of the system and the calculation of the cooling demand. Section 3 shows the results of the energy-operating analysis of the hybrid system during the summer period and during one week with the highest environmental temperatures of the year, where both cooling and water desalination performances can be appreciated. Finally, the most important conclusions are presented in Section 4.

2. Materials and Methods

2.1. System Description

The proposed system consists of the coupling of a LiBr/H₂O single-effect absorption-cooling open-cycle that has three levels of pressure, with a one-stage flash-desalination process. The system is divided into four subprocesses: (1) the cooling subprocess; (2) the desalination subprocess; (3) the product water extraction and conditioning subprocess, and (4) the solar thermal energy collection and storage subprocess, which are shown in Figure 1.

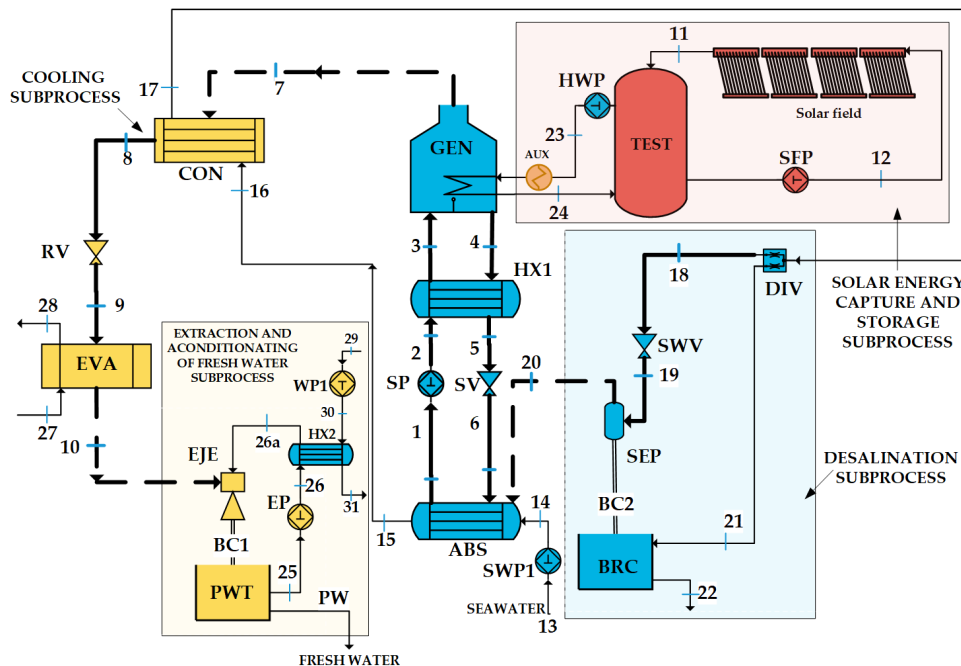


Figure 1. Schematic diagram of hybrid solar absorption-cooling and flash-desalination system.

In the cooling subprocess, the hot water flow (23) coming from the solar thermal energy collection and storage subprocess enters the generator (GEN) to heat the weak solution LiBr/H₂O flow (3), producing the separation of water vapor (7). The vapor (7) is taken to the condenser (CON), to undergo a phase change by exchanging latent heat with the seawater stream (16) and leave as a saturated liquid (stream 8). On the other hand, the strong solution (stream 4) exchanges sensitive heat with the stream (2) of diluted solution, in order to bring both streams closer to the process of desorption and absorption, respectively. The stream (4) after decreasing its temperature, leaves the HX1 as the stream (5) and is taken to a solution expansion valve (SV). When leaving this valve, the stream (6) decreases its pressure and in these conditions enters the absorption equipment (ABS). In the absorber, the stream (6) absorbs water vapor (20) from the desalination subprocess, producing an exothermic reaction. The pump (SWP) suctions in the seawater stream (13) and conducts it as stream 14 to dissipate the heat of the exothermic reaction in the absorber and leaves at a higher temperature (stream 15 = 16) and is conducted to condense the steam (7) produced in the GEN. The stream (17), after receiving the condensation heat, leaves the condenser and enters the desalination subprocess and is taken to a stream divider (DIV), where it is divided into streams (18) and (21). The stream (21) is taken to a brine dilution vessel (BRC); while the stream (18) goes through an expansion process, so it flashes, due to the pressure decrease, and remains in two phases (stream 19). The two-phase stream (19) enters a liquid-vapor separator (SEP); here the vapor is separated (20) and taken to the cooling subprocess to enter the ABS and be absorbed. On the other hand, the liquid brine leaving the SEP is taken to a barometric column (BC2) and, then, exits to the BRC; the function of the barometric column (BC2) is to serve as a hydraulic seal by maintaining the vacuum pressure and allowing the brine to be raised to atmospheric pressure.

In the cooling subprocess, the liquid coolant (8) passes through an expansion valve (RV) producing a slight flash, caused by the decrease of its pressure and temperature (stream 9); in these conditions, it enters the evaporator (EVA) to produce the cooling effect. The cooling water (27) enters to the evaporator equipment and is cooled, causing the refrigerant (10) to change its state at the exit of the

evaporator and leave as saturated steam; in these conditions the stream (10) enters to the expansion and product water conditioning subprocess.

In the subprocess of extraction and adaptation of product water, the current (10) after the cooling effect in the EVA is suctioned by the ejector unit (EJE), which is activated with the driving fluid (26) coming out of the cooler (HX2). The low pressure vapor stream 10 is mixed with the high-pressure liquid stream (26), both mixed fluids are compressed by the ejector diffuser, at a pressure intermediate between the pressures of streams (10) and (26); this pressure has the appropriate value for vapor 10 to condense, while the liquid stream (26) receives this condensation heat. This causes both streams, now in liquid phase, to increase their temperature at the outlet of the ejector diffuser. The liquid stream leaving the ejector is discharged into the barometric column (BC1), which allows the vacuum to be maintained in the system and the water leaving the ejector to be deposited in the product water tank. The stream (25) is suctioned by the pump (EP) and is taken (stream 26) to dissipate the heat of condensation in the cooling equipment (HX2); the stream 26a coming out of HX2, is again able to receive the heat of condensation from the steam 10.

It is important to note that the hybrid system uses three pressure levels where the absorber has a higher pressure than the evaporator, which results in a thermal compressor with less ΔP , improving efficiency and allowing the evaporator to work with adequate pressures for cooling [29]. On the other hand, the system has an internal energy integration in which the heat from the absorber and condenser is used to preheat the seawater before it is taken to the desalination subprocess; this allows the cooling of the CON and ABS without requiring an expensive cooling tower. Finally, to condense the cooling steam, barometric ejecto-condensers are used, which are passive elements that allow maintaining the vacuum in the system, besides compressing the cooling steam until it is condensed, without a high energy cost.

The solar thermal energy collection and storage subprocess convert solar radiation into high quality thermal energy, which is used to drive the hybrid system's generator. During the day, water is taken from the bottom of the thermal energy storage tank (TEST) to be conducted to the solar collector field through the stream (12); the collectors are connected in a series and parallel arrangement to ensure the temperature level and amount of energy required for the operation of the system. As long as there is enough solar radiation to raise the temperature of the stream (11) above that of the stream (12), the solar fluid pump (SFP) will remain on. During operation of the hybrid system, energy is extracted at the rate of the cooling demand by the stream (23) and the HWP, resulting in nonconstant conditions within the TEST. The solar collectors in a certain period of heating will be supplying the amount of energy sufficient for the operation of the hybrid system and, also, for storage in sensible heat form within the TEST for the periods of null (night) or low solar radiation. If the temperature of the stream (23) falls below 78 °C because the stored energy and/or the energy produced by the collector field is not sufficient to meet the demand, the auxiliary heater (AUX) shall be used to reach the minimum operating temperature.

Table 1 shows the characteristics of the equipment used in the study. The capacity of the equipment is analyzed below.

Table 1. System characteristics.

Hybrid Cooling-Absorption and Flash-Desalination System	
Max. cooling capacity	80 kW
Min. cooling capacity	16 kW
Mass flow activation m_{11}	5.62 kg/s
Temp. of seawater (stream 13)	27 °C
Mass flow of seawater (stream 13)	3.39 kg/s
Evaporator temp.	6 °C
Condenser temp.	34.30 °C
Absorber temp. (stream 1)	32 °C
Inlet chiller temp.	12 °C
Flow solution	0.28 kg/s
Solar Collector	
Brand/model	Suntask/SHC24
Collector type	Evacuated tube with CPC reflector
Number of tubes	24
Aperture area	4.41 m ²
Optical efficiency (a0)	0.668
First order efficiency coefficient (a1)	1.496 W/m ² °C
Second order efficiency coefficient (a2)	0.005 W/m ² °C ²
Fluid	Water
Mass flow	0.02 kg/s m ²
Thermal Energy Storage Tank	
Material	Fiberglass
Insulation thickness	0.025 m
Loss coefficient	1.4 W/m ² K
Fluid	Water

2.2. Methods

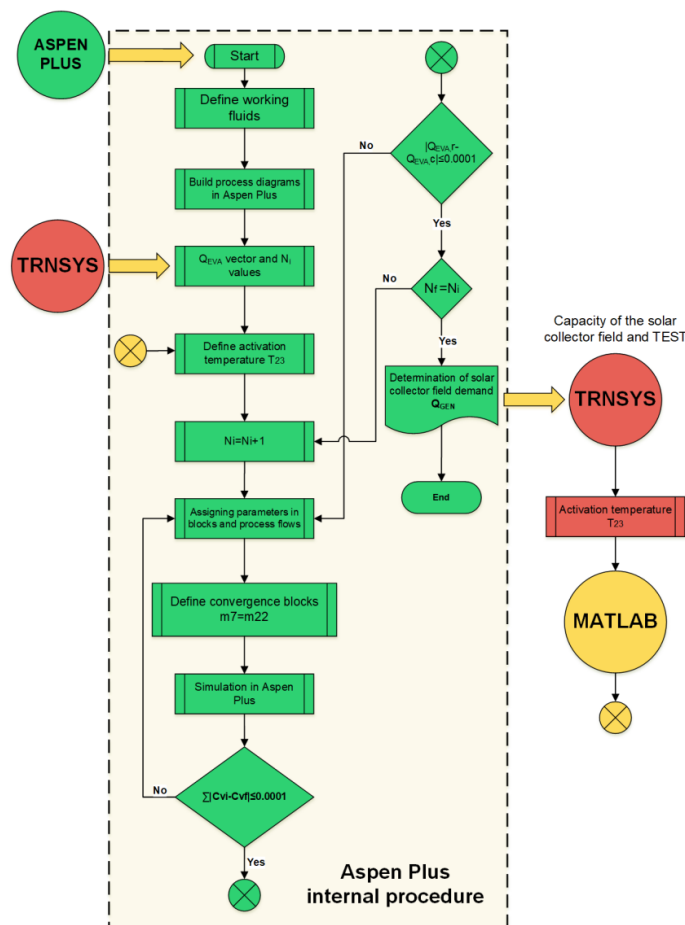
The simulation of the hybrid system activated with solar thermal energy was carried out by means of three software:

1. Aspen Plus was used to evaluate the internal operation of the absorption-cooling and flash-desalination system.
2. Through the TRNSYS software, the thermal load and capacity of the solar thermal collector and TEST system were determined.
3. MATLAB was used to evaluate the coupling of the solar thermal collector, TEST, and thermal load with the proposed hybrid system.

The considerations for making the operational study are presented below:

- The proposed cycle operates in stable conditions. The LiBr/H₂O solution is homogeneous and in balance. The weak solution is at 51%.
- The steam (22) generated in sudden evaporation must be equal to the steam (7) produced in the GEN.
- Operating pressures were set as reported in [29], unless otherwise mentioned.
- The maximum operating cooling capacity was set at 80 kW, and a minimum partial load operation of 20% (16 kW).
- The approach temperatures of the heat exchanger equipment EVA, GEN, ABS, CON and HX1 are those reported by [29] and those of HX2 are those reported by [30].
- The design flows of the external system circuits were considered according to the AHRI standard [31]. Seawater was considered to have a concentration of 35,000 ppm with a temperature of 27 °C, as reported by [32] and the composition based on [33].

- The minimum operating temperature of the hybrid system is 78 °C; a lower temperature will force the use of the auxiliary heater.
- The on/off switching of the solar field pump was controlled by temperature difference ($[T_{out} - T_{in}] > 0$).
- 30-min simulation intervals.



2.3. System Simulator Validation

concluded that Aspen Plus is a good option to study the operability of the system. The validation can be verified in the work of López-Zavala et al. [29].

2.4. Case Study

To operatively study the technological proposal influenced by real operating conditions, a 20-room hotel complex located in the city of San Felipe, Baja California, Mexico ($31^{\circ}01'22.1''$ N $114^{\circ}49'56.5''$ W) was taken as a case study. This city is characterized by its coastal location, with a high level of tourism and high environmental temperatures during the summer, which can reach 45°C in the shade. It presents difficulty in accessing drinking water, its main source being the waters coming from the Colorado River that are transported from Mexicali, which is approximately 200 km away. However, San Felipe has an excellent solar resource for its energy exploitation, so this location is suitable for analyzing the proposal.

The calculation of the cooling demand was made with the Type 56 (TRNBuild) of TRNSYS, an interior temperature of 25°C was established during 24 h of the day for a building formed by concrete blocks ($k = 1.64 \text{ W/m}^{\circ}\text{C}$, $\rho = 2011 \text{ kg/m}^3$, $C_p = 0.91 \text{ kJ/kg}^{\circ}\text{C}$) that were 8 m long, 7.5 m wide, and 2.75 m high. Thermal gains generated by four people performing resting activities (480 W), artificial lighting (75 W), and a television (60 W) were considered. Figure 3 shows the profiles of (a) ambient temperature, (b) global radiation, and (c) cooling demand and daily mean ambient temperature of the hotel complex during the June–September period, considering the months with the highest tourist affluence and high ambient temperatures.

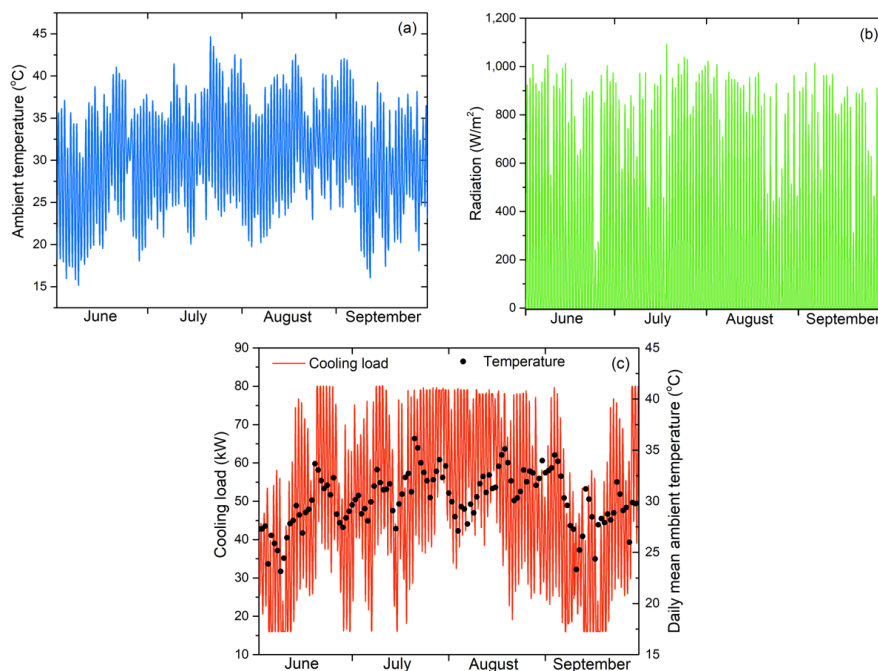


Figure 3. Profile of (a) ambient temperature, (b) global radiation, and (c) cooling demand and daily mean ambient temperature used in the simulation of the solar cooling and desalination system.

2.5. System Evaluation

The absorption system is evaluated with the COP indicator, which represents the ratio of the desired effect to the energy cost of producing it, according to Equation (1). Q_{EVA} is the cooling capacity, Q_{GEN} is the heat supplied to the system, SP is the solution pump power, W_{SWP} and W_{SWP1} represent

the work of the ABS and HX2 cooling pumps, respectively, and W_{EP} is the work of the ejector fluid pump. However, considering that this system produces fresh water and space cooling simultaneously, with the same main source of energy, a COP_G is proposed in accordance with Lopez-Zavala et al.'s proposal [30], calculated as shown in Equation (2), where m_{10} is the product water flow:

$$COP = \frac{Q_{EVA}}{Q_{GEN} + W_{SP} + W_{SWP} + W_{SWP1} + W_{EP}} \quad (1)$$

$$COP_G = \frac{Q_{EVA} + m_{10} \times \left(2326 \frac{kJ}{kg}\right)}{Q_{EVA}} \quad (2)$$

To evaluate desalination, the performance ratio (PR) indicator is used, which relates the amount of water produced in terms of energy with respect to the main activation energy of the desalination process [37], and is calculated with Equation (3):

$$PR = \frac{m_{10} \times \left(2326 \frac{kJ}{kg}\right)}{Q_{GEN}} \quad (3)$$

The thermal efficiency of solar collectors, E_{fsf} , is evaluated based on Equation (4), as well as the heat contributed by each of the collectors, Q_{sf} , with Equation (5):

$$E_{fsf} = a_0 - a_1 \frac{T_{in} - T_a}{G_T} - a_2 \frac{(T_{in} - T_a)^2}{G_T} \quad (4)$$

$$Q_{sf} = A_{SC} G_T E_{fsf} \quad (5)$$

where a_0 , a_1 , and a_2 are the optical efficiency of the solar collector and the first and second order efficiency coefficient, respectively. T_{in} corresponds to the collector inlet temperature, T_a the ambient temperature, G_T the incident solar radiation in the collector aperture area, and A_{SC} the solar collection area.

The solar fraction represents the thermal energy contribution of the solar field and TEST to the hybrid system consumption and is calculated based on the Equation (6):

$$f_{sf} = \frac{\dot{Q}_{GEN} - \dot{Q}_{aux}}{\dot{Q}_{GEN}} \quad (6)$$

where $\dot{Q}_{GEN}$ is the thermal energy required to meet the needs of the hybrid system and $\dot{Q}_{aux}$ is the extra energy supplied by the auxiliary heater after the solar contribution to meet the demand for cooling and desalination.

3. Results

3.1. Analysis in Summer Period

In order to carry out a specific operational study of the system, it is necessary to determine the capacity of the solar collector area and the size of the TEST to satisfy the activation energy requirements throughout the operational period considered in the summer months. For such situation, an operational parametric analysis was performed during June–September using the meteorological conditions of the region and the load profile presented in Figure 3. The results on different variables are presented in Figure 4. Figure 4a analyzes the amount of auxiliary energy required to meet the thermal energy requirements of the cooling and desalination system. It can be seen how a low-capacity solar thermal collector field has no influence on the decrease in auxiliary energy consumption, regardless of the size of the TEST. From the 500 m² of solar collector area the volume of the TEST begins to have a positive effect; this is because the solar energy generated, as shown in Figure 4b, is sufficient to cause

an increase in the temperature of the water stored in the TEST and can be used to activate the hybrid system. From 600 m² onwards, there is a noticeable decrease in the auxiliary energy requirements and a greater effect of the size of the TEST. The aim of this parametric study is to find the appropriate size of the solar collector field, so Figure 4c shows the effect of these values on the cost of auxiliary energy and Figure 4d the solar fraction achieved. By increasing from 460 to 620 m², corresponding to a 25% increase in collector area, a 28% reduction in consumption and cost of auxiliary energy is achieved, while the solar fraction increases from 57% to 69%. However, increasing the capacity of the solar field by 20% and reaching 770 m² represents only a 7% increase in the solar fraction. Therefore, a capacity of 620 m² in the solar collector area and a TEST of 30 m³ was selected to carry out the operational study of the hybrid cooling and desalination system, which achieves a solar fraction of 69%, requiring an extra 53 MWh of auxiliary energy to satisfy the thermal energy requirements.

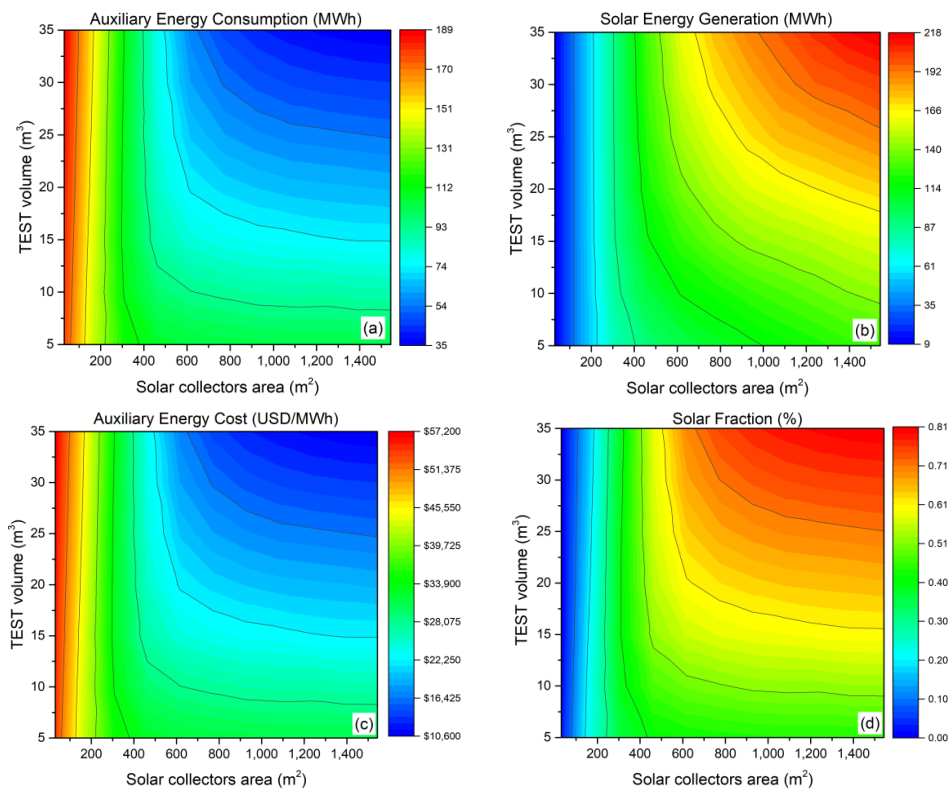


Figure 4. Effect of total solar collector area and total volume of the TEST on (a) auxiliary energy consumption, (b) solar energy generation, (c) auxiliary energy cost, and (d) solar fraction. A fuel cost of \$1.15 USD/L and an equivalence of 1 L = 3.79 kWh were considered, considering a conversion efficiency of 38%.

With the above, Figure 5 shows the energy accumulation generated or consumed by the solar collector field, the cooling and desalination system, and the auxiliary equipment. During the first weeks of June, when the cooling load is not so high, the solar energy and thermal storage capacity is sufficient to meet the energy requirements of the hybrid system. However, as time passes, the cooling load increases and the solar energy is no longer sufficient to cover the heat demand, especially for night periods, so the use of the auxiliary heater is required. During the four summer months analyzed, a total of 183 MWh of thermal energy was required to activate the cooling and desalination system, while the solar field and the auxiliary heater generated a total of 153 and 60 MWh, respectively. Approximately

31 MWh of solar thermal energy is lost due to energy storage capacity and the time lag between the cooling load and solar generation.

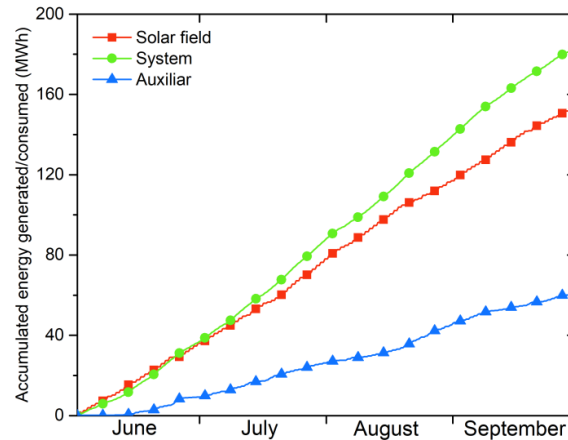


Figure 5. Accumulated energy generated or consumed by the solar field, cooling and desalination system, and auxiliary generator.

3.2. Weekly Operation

In order to further explore the operation of the solar hybrid system, the week of July 19–25 was selected because it has the highest ambient temperatures during the study period, reaching up to 45 °C in the shade. Figure 6 shows the behavior of the different temperatures that affect the system operation, as well as the horizontal global radiation. The average temperature dynamics of the TEST, as well as the temperature of the fluid entering the system's generator, can be seen. The stream entering the generator is taken from the top of the TEST, so it has a higher temperature than the average of the TEST. The minimum operating temperature of the cooling and desalination system is 78 °C, so during the day the solar field is sufficient to maintain the TEST temperature above this value. However, as the solar resource decreases, the tank also decreases its temperature due to the continuous extraction of energy, to the point that there is not enough solar energy to maintain the minimum operating temperature, so the auxiliary heater is used and has a constant temperature of 78 °C until the solar resource is sufficient to meet the heating demand.

The dynamics related to the heat generated by the solar collector field and the auxiliary heater, as well as that consumed by the generator, are shown in Figure 7. It can be seen that at the beginning of the week there is no solar resource stored in the TEST, so the auxiliary heater is responsible for satisfying the system's generator. As it is daytime and the solar radiation is sufficient to activate the generator, the auxiliary heater is stopped. During maximum solar exposure, the energy generated by the solar collectors is sufficient to meet the demand and store energy in the TEST. When solar power generation is completely finished, it can be seen that for a period of time energy continues to be supplied to the system's generator without the use of the auxiliary heater. This is because the TEST has sufficient quantity and quality of energy to keep the cooling and desalination system in operation. When the TEST temperature is below the minimum operating temperature, as shown in Figure 6, the auxiliary heater is turned on.

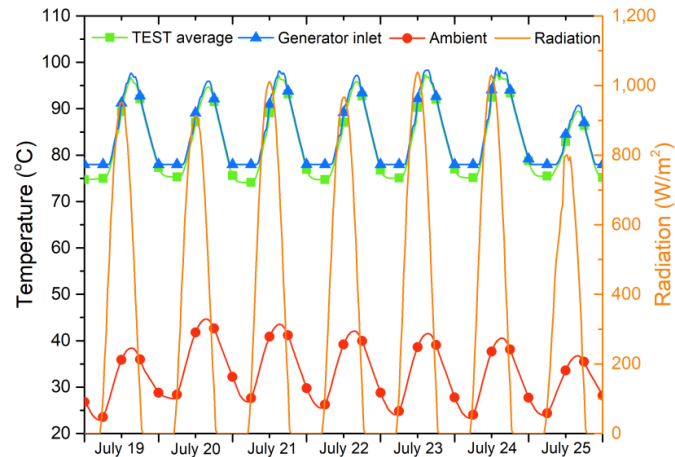


Figure 6. Variation of the average TEST temperature, system generator input temperature, and ambient temperature during an operational week.

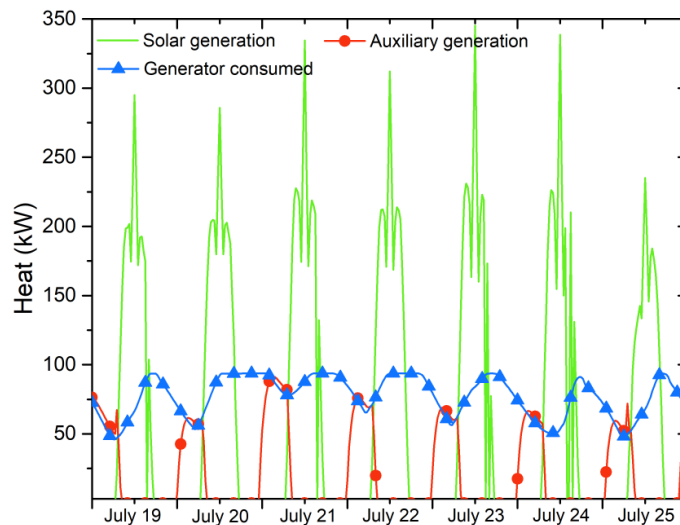


Figure 7. Variation in heat generation of the solar field and the auxiliary heater, and generator power consumption of the cooling and desalination system.

Figure 8 shows the heat transfer behavior of the GEN, EVA, CON, and ABS equipment of the system during the critical week of ambient temperature in summer. It can be seen that the increase in the cooling capacity causes an equal increase in the other heat-exchanger equipment. This is due to the fact that the cooling capacity is controlled by manipulating the addition of heat in the GEN; that is, when Q_{EVA} increases it is necessary to transfer more refrigerant fluid (7) to the evaporator, so the heat in the GEN increases. Furthermore, this vapor, needing to condense to go to the refrigerant expansion valve (VR), causes the increase of heat transfer in the CON. It is important to highlight that the steam flow (7) has to be equal to the steam (20) produced in the desalination subprocess to keep the thermal compressor in stable conditions and avoid possible crystallization problems. If the vapor (7) produced in the GEN is greater than the vapor (20), it will cause crystallization to occur and, if the vapor (7) is less than the vapor (20), then there would be an accumulation of mass in the absorption system, resulting in a change in pressure levels. To maintain the equality between the currents (7) and (20), the current divider (DIV) is used, in order to manipulate the flow of the current 19 and only use the sea

water necessary for the production of the steam (20); consequently, the increase of the steam (7) causes an increase in the capacity of the ABS heat exchanger.

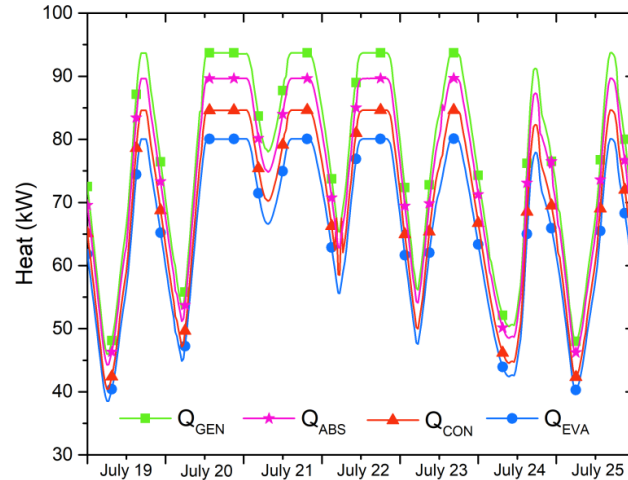


Figure 8. Operational performance of Q_{GEN} , Q_{EVA} , Q_{CON} , and Q_{ABS} heat exchangers during the critical summer week.

Figure 9 shows the heat transfer in EVA and fresh water production during the critical summer week. It can be seen that the increases in cooling capacity and fresh water production are proportional. This is because, as mentioned above, in order to meet the demand for cooling, it is necessary to control the production of steam (7) and bring it into the EVA to provide the cooling effect. This coolant flow, when suctioned by the ejector, is converted into product water, therefore, if the cooling capacity is increased, the product water also increases. The system operating at its maximum cooling capacity can satisfy a demand of 80 kW and a water production of 122 kg/h. During the week analyzed, a total production of 16.94 m³ was reached—while the average daily production of 2406 kg/day—enough to satisfy the daily water requirements of four people in a coastal hotel in Mexico [38] or to cover the basic services of 24 people according to World Health Organization (WHO) [39].

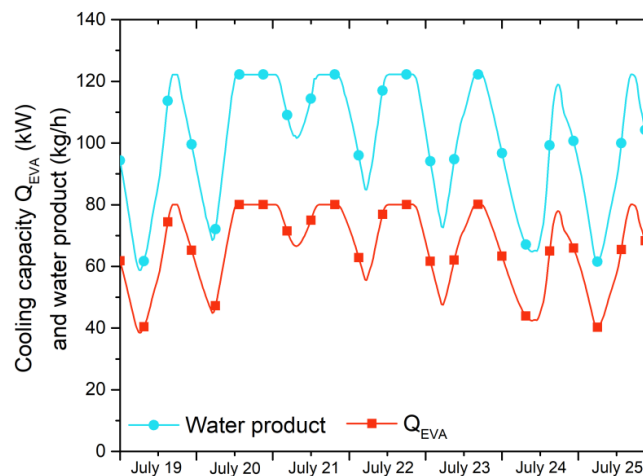


Figure 9. Behavior of cooling capacity and water product during the critical summer week.

Figure 10 shows the behavior of the cooling and desalination indicators, considered to evaluate the performance of the proposed hybrid system. It can be seen that the COP and the PR have a similar behavior, because both depend directly on the cooling demand. It is also observed that both indicators decrease when there is low cooling demand; this is because the decrease in cooling capacity implies less steam generation in the GEN, causing the high pressure in the GEN to decrease and the latent heat of the LiBr/H₂O solution to increase. Therefore, more heat is needed in the GEN to produce cooling steam. On the other hand, with low cooling capacities, the COP shows a greater decrease than the PR due to the energy consumption of the pumps. The average COP during the week of operation was 0.82, which is 13.88% higher than the COP reported in conventional single-effect LiBr/H₂O absorption-cooling systems. This is due to the use of three pressure levels, which allowed the ΔP between the GEN and the ABS to be reduced, allowing the EVA to reach adequate pressure levels for cooling. On the other hand, considering that the system can provide two services simultaneously, it can be seen that the maximum COP_G is 1.66, while the average COP_G is 1.64, being 2.27 times higher than the COP of conventional single-effect LiBr/H₂O absorption units. The system is capable of providing space cooling more efficiently than conventional systems, in addition to producing water with an average PR of 0.83.

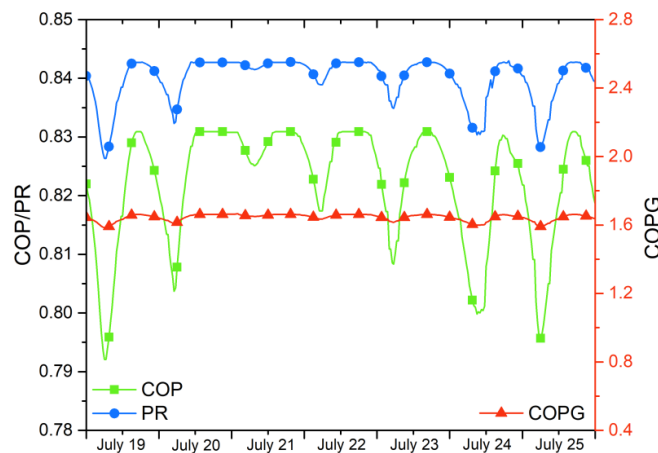


Figure 10. Performance of COP, COP_G, and PR indicators during the critical summer week.

3.3. Technological Considerations

The technological proposal analyzed in this work presents different challenges in terms of current technological development. The commercial maturity of solar heating and absorption systems is sufficient to be used in the current proposal, with simple modifications in the operating conditions. However, the main challenge is focused on the product water extraction and conditioning subprocess. At present, this type of system is not commercially available, so these components must first be developed so that they can work efficiently under the operating conditions mentioned in the study. Once the technological maturity of all the systems involved is achieved, their energy integration will be carried out in order to experimentally corroborate the results. On the other hand, given the nature of the system, an adequate application for the capacity of the equipment must be sought, where sufficient energy resources are available: in this case, high levels of solar radiation. As it is a thermal machine, the capacity–cost ratio can be fundamental. At lower capacity, the specific cost of the system can be increased as with other technologies, such as power cycling and vapor compression cooling systems. However, this will be analyzed in detail in future work, when the main components of the system will be further developed.

4. Conclusions

A novel solar hybrid system of LiBr/H₂O absorption cooling and flash seawater desalination was proposed. This system couples a single-effect absorption-cooling open-cycle with three pressure levels, thus allowing an internal energy and mass integration with a single-stage flash-desalination process, producing simultaneously cooling and drinking water. A solar thermal collector field and a TEST are used to generate the thermal energy required for its activation. Its operation is analyzed under the dynamics of a case study of a 20-room hotel complex in the city of San Felipe, Baja California, Mexico, where the influence of the solar resource, ambient temperature and cooling demand was studied.

The proposed novel system was evaluated during the summer period and in the week of highest cooling requirement. The absorption-cooling subprocess, using three levels of pressure, presented an increase of 13.88% in the average COP, compared to conventional LiBr/H₂O absorption systems. Also, considering that the system provides cooling and desalination simultaneously, it has an average COP_G of 1.64, being 2.27 times higher than the COP of conventional LiBr/H₂O absorption units of single-effect.

The system, during the critical week, presented a production of desalinated water of 16.94 m³ with an average PR of 0.83; while the average daily production of product water was 2406 kg/day, enough to satisfy the daily water requirements of four people in a Mexico hotel or to cover the basic services of 24 people according to the WHO. The dimensioning of the solar thermal collector field and TEST resulted in a total collection area of 620 m² with 30 m³ of storage in the form of sensible heat. This resulted in a solar fraction of 69% requiring an extra 53 MWh of auxiliary energy to meet the thermal energy requirements.

Author Contributions: Conceptualization, J.A.A.-J. and R.L.-Z.; methodology, N.V.-L. and L.H.-C.; software, J.A.A.-J., R.L.-Z., and J.R.-A.; validation, R.L.-Z.; formal analysis, S.O.-B.; investigation, N.V.-L. and L.H.-C.; resources, N.V.-L.; writing—original draft preparation, J.A.A.-J. and R.L.-Z.; writing—review and editing, N.V.-L. and L.H.-C.; supervision, S.O.-B.; project administration, N.V.-L.; funding acquisition, N.V.-L. All authors have read and agreed to the published version of the manuscript.

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Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

GEN	Generator
ABS	Absorber
CON	Condenser
EVA	Evaporator
HX1	Recovery heat exchanger
HX2	Driving fluid cooler
AUX	Auxiliary heat exchanger
TEST	Thermal energy storage tank
SFP	Solar fluid pump
HWP	Hot water pump
SP	Solution pump
SWP	Seawater pump
SWP1	Seawater pump for cooling
EP	Ejector pump
SV	Solution valve
RV	Refrigeration valve

SWV	Seawater expansion valve
DIV	Streams divisor
SEP	Separator equipment
EJE	Ejector
BC1	Ejector barometric column
BC2	Separator barometric column
PWT	Product water tank
BRC	Brine container
COP	Coefficient of operation
PR	Performance ratio
PW	Product water

Nomenclature

T	Temperature [°C]
P	Pressure [kPa]
Q	Heat transfer [kW]
ASC	Solar collection area
GT	Incident solar radiation in the collector opening area
a_0	Optical efficiency of the solar collector
a_1	First order efficiency coefficient
a_2	Second order efficiency coefficient
T_{in}	Collector inlet temperature
T_a	Ambient temperature
f	Solar fraction
E_f	Thermal efficiency of solar collectors

Subscripts

a	Ambient
in	Inlet
1,2,3 ...	Number of streams
sf	Solar field

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H. A novel LiBr/H₂O absorption cooling and desalination system with three pressure levels

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A novel LiBr/H₂O absorption cooling and desalination system with three pressure levels



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ABSTRACT

This paper presents a novel LiBr/H₂O absorption cooling and desalination system with three pressure levels which uses seawater as a cooling medium and is activated by solar thermal energy. This system helps counteract coastal populations' thermal conditioning and water supply problems. In order to show technical feasibility, Aspen Plus was used to simulate the system with the following parameters: a 23 kW cooling capacity; different seawater temperatures; and, a concentration of 35,000 ppm. It was found that the proposed technology has 19.4% more efficiency than a conventional single-effect absorption cooling system, with an FR of 0.08 and an R of 1.84, and produces 838 additional L/day of water, enough to satisfy the requirements of a household for up to 7 people. With the configuration of the proposed technology, which manages three levels of pressure, and using seawater as a cooling medium, the simultaneous production of cooling and desalination is benefited.

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Nouveau système de dessalement et de refroidissement à absorption de LiBr/H₂O à trois niveaux de pression

Mots-clés: Froid; Absorption; Dessalement; Éjecteur; Thermosolaire

1. Introduction

Recently, sustainable provision of water supply and climatization of spaces has become a priority (Gude et al., 2010). Therefore, the scientific community is seeking to develop technologies that provide freshwater and cooling jointly, through renewable energies (Veera Gnanaswar et al., 2011; Negewo, 2012; Sharon and Reddy, 2015; M and Yadav, 2017; Shahzad et al., 2017). Absorption cooling systems have increased their potential due to their flexibility to integrate polygeneration systems activated by renewable energies (Rosiek and Batlles, 2009; Mammoli et al., 2010; Ortiz et al., 2010; Zhai et al., 2011). Moreover hand, multiple effect (MED), and multiple stage (MSF) distillation systems have achieved great com-

patibility with solar thermal energy, and hybridization with absorption and adsorption cycles (Milow and Zarza, 1997; El-Nashar, 2001; Hala Faisal and Al-Fulaij, 2011; Li et al., 2011; Al-Karaghoul and Kazmerski, 2013).

In order to increase the technical and economic viability of these technologies, several studies propose the hybridization of cooling systems with desalination units, as well as the simultaneous production of both cooling and potable water. Aly (1995) presented a theoretical study of the coupling of a 220 kW LiBr/H₂O absorption machine with a 20-stage MSF, which uses the heat of the former's condenser to activate the MSF, while the absorber is cooled with seawater. This proposal reduces the operation costs of the desalination process, achieving, without cooling production, a Gained Output Ratio (GOR) of 14.8 and a Performance Coefficient (FR) of 14.2. Mandani et al. (2000) studied the coupling of a LiBr/H₂O heat pump with a single-effect desalination system, in order to improve the performance of the desalination system; they

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Nomenclature

Symbols

Q	heat transfer (kW)
T	temperature (°C)
P	pressure (kPa)
GOR	gained output ratio (-)
COP	coefficient of performance (-)
FR	performance coefficient (-)
R	production ratio (-)
UPC	unit product cost (\$/m ³)
m	mass flow (kg/s)
X	concentration LiBr (%) or NaCl (ppm)
xc	LiBr/H ₂ O strong solution concentration
SP	solution pump
SWP	seawater pump
EP	ejector activation pump
PW	product water
BC1	product water barometric column
BC2	brine barometric column
BRC	brine container
PWT	product water tank

Subscripts

Number of streams 1,2,3....

GEN	generator
ABS	absorber
CON	condenser
HX1	recovery heat exchanger
HX2	heat exchanger
SC	solar collectors
DIV	stream divider
SV	solution expansion valve
SWV	seawater expansion valve
RV	refrigerant expansion valve
EJE	ejector

reported that the FR improved from 2.4 to 2.8 –from 50 to 70%– in comparison to the single-effect thermal vapor compression system.

Various experimental studies have sought to corroborate the results of the hybridization of cooling and desalination systems. Among the most relevant we find the works of Alarcón-Padilla et al. (2007), Alarcón-Padilla and García-Rodríguez (2007), and Alarcón et al. (2008), who analyzed the coupling of a 14-effects MED with a double effect LiBr/H₂O absorption cooling system. This coupling exploited the heat generated by the condensation and absorption cycles, in order to activate the MED. It showed a reduction of 50% in energy consumption for the MED; furthermore, a 2.2 Coefficient of Performance (COP) at full load is reported for the absorption cycle. However, the activation temperature increased from 70 to 180 °C. Cool water from the absorption machine was not used for the cooling of spaces but, instead, for condensation in the last stage of the MED.

Veera Gnanaswar and Nagamany (2008) and Veera Gnanaswar et al. (2011) theoretically and experimentally studied the coupling of a single-effect vacuum distillation unit with a LiBr/H₂O absorption system of 0.975 tons of cooling. Heat discarded by the condenser was used to activate the desalination process. Product water was obtained at a rate of 4.5 kg/h, which led the authors to conclude that the proposal yielded better results than an MSF with a capacity of approximately 338 kJ/kg of freshwater.

Wang and Lior (2011a, b) theoretically evaluated the integration of a six-effect MED with a LiBr/H₂O single-effect heat pump. The heat of both the condenser and the absorber was used to activate

the MED. A COP of 1.6 was reported for the heat pump, obtaining energy savings of 42% when compared to conventional processes.

Abdulrahim and Darwish (2014) theoretically studied the coupling of a LiBr/H₂O absorption system with an 8-effects MED activated with thermosolar energy. The absorption cooling unit had a capacity of 2148 kW and presented a COP of 0.74, while the MED produced 459 m³ of water per day, with a constant FR of 5.7 when controlling the level of desalination of the last effect.

More recently, Alelyani et al. (2017), applying the methodology of exergy analysis, carried out a study of the coupling of a 14-effect MED with one and two-stages NH₃/H₂O absorption cooling systems. The heat of both the condenser and the rectifier was used to activate the MED. When conducting the coupling, the total destruction of exergy decreased by 55% on average. The Unit Product Cost (UPC) of cooling decreased by 43%, while, for the production of water, with the coupling of the single and double-stage absorption units, the UPC increased by 19 and 3%, respectively.

Several efforts have been made to integrate absorption cooling and thermal desalination technologies through renewable energies. However, there are very few systems that provide both services simultaneously, and those that do have low internal energy integration and require a high initial investment because they use many components, making them only viable for large capacities. Therefore, the challenge is to develop new machines that require a lower initial investment, present lower operating and maintenance costs, along with a prolonged life, have no negative environmental impact, and are capable of working with low capacities in order to cover the residential and commercial market. This article presents a novel LiBr/H₂O absorption cooling and desalination system with three pressure levels. The proposed system takes advantage of the low pressure of the absorption cooling subprocess to potentiate desalination, eliminate components, and increase efficiency as a result of better internal energy integration.

2. Description of the system

The conventional absorption refrigeration cycle was modified, making it interact directly with the flash evaporation desalination process, resulting in a system that is composed of an **absorption cooling subprocess**, a **desalination subprocess**, and an extraction and conditioning of free water subprocess (Fig. 1).

The absorption **cooling subprocess** is composed of an absorber (ABS) that receives the steam (23) generated in the desalination subprocess. Pump SP sends the LiBr/H₂O solution from the ABS to the generator (GEN), first passing through the heat recovery exchanger (HX1). In the GEN, heat from the solar collectors is added to the LiBr/H₂O solution (SC) so that the water refrigerant is released in the form of vapor (7). The solution with a higher concentration of LiBr (4) returns to the ABS, passing first through the HX1 and through the expansion valve of the solution (SV), in order to absorb the steam (23). To remove the heat from the exothermic absorption process, ABS is cooled with seawater from pump SWP. Steam (7) leaving the GEN, passes to the condenser (CON), and when it is cooled with seawater it condenses, moving then through the refrigerant expansion valve (RV). As the refrigerant reduces its pressure, it lowers its temperature and is ready to cause the cooling effect in the evaporator (EVA), where it becomes vapor by means of receiving heat from the water required for cooling. The steam (10) leaving the EVA goes through to the extraction and conditioning of freshwater subprocess.

During the cooling subprocess, seawater (30) is suctioned by pump SWP and used to cool ABS and CON. Subsequently, stream (30) is introduced into the stream divider (DIV), in order to obtain a lower flow in stream (19), which is heated by the heat transfer fluid of the SC in the heat exchanger (HX2). Then, stream (21)

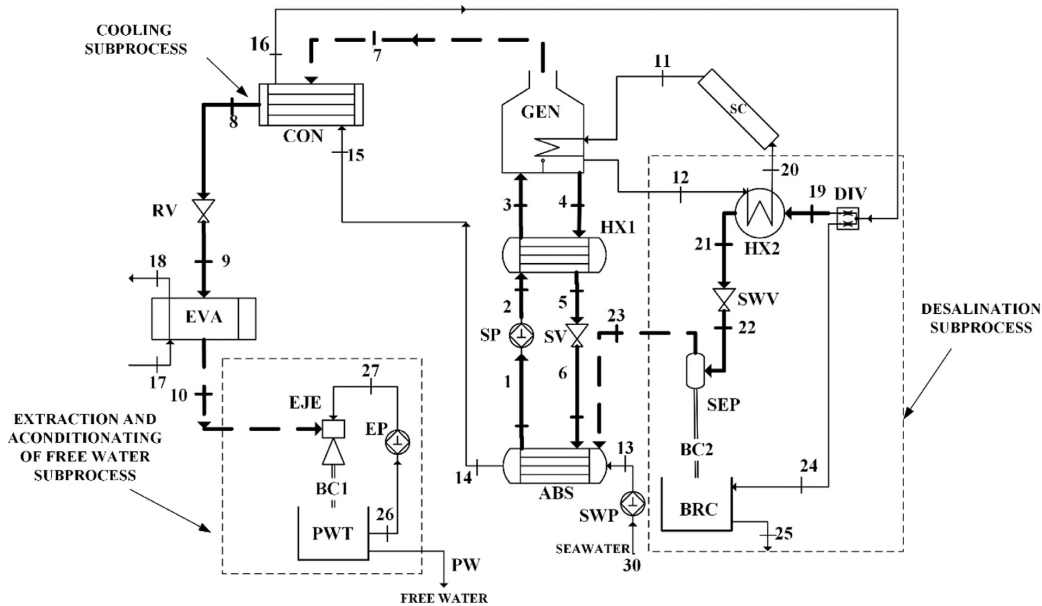


Fig. 1. Diagram of the LiBr/H₂O absorption cooling and desalination system activated with solar thermal energy.

goes to the expansion valve (SWV), where stream (22) is obtained in two liquid/vapor phases. The liquid leaving the SEP, in order to maintain the vacuum pressure, falls to a barometric column (BC2) to make a hydraulic seal and then passes to the brine container (BRC). The concentrated seawater from BC2 is mixed with the seawater stream coming from DIV (24), in order to reduce its concentration before returning to the sea as stream (25), reducing its environmental impact.

During the subprocesses of extraction and adaptation of product water, water vapor (10) is suctioned by the ejector (EJE), which is activated by the liquid water driving fluid (27) supplied by the pump (EP). Both the high-pressure (27) and the low-pressure streams (10) are mixed in the EJE, and leave it in liquid form at an intermediate pressure and room temperature, conditions required by the product water (PW). The water that leaves the EJE is conducted to a barometric column (BC1), and then to the product water tank (PWT), from where pump EP provides the fluid that activates the EJE. The purpose of this ejector within the extraction and adaptation of product water subprocess is to condense a vapor that has low pressure and temperature, in order to release it with proper environmental conditions.

The proposed system presents a series of characteristics that make the most of the existing compatibility between the flash evaporation desalination process and that of absorption cooling:

- Exploitation of the absorber's pressure in the absorption cooling subprocess, in order to potentiate desalination with low energy consumption.
- The provision of greater internal energy integration by taking advantage of absorption heat and condensation from the absorption cooling subprocess, in order to heat the seawater. When seawater is used to cool the absorber and condenser, the cooling tower becomes unnecessary, reducing initial investment, as well as operation and maintenance costs, resulting, also, in a more compact machine.
- Three levels of pressure, which as a result of a higher pressure within the absorber, increase the efficiency of the cooling subsystem without significantly affecting the desalination process.

- Integration of a field of solar collectors that first activates the absorption cooling subprocess, and then heats the seawater to be desalinated. Hence, the heat transfer fluid returns to the collector field at a lower temperature, achieving, thus, a more efficient operation.
- Integration of passive components, such as barometric ejector condensers, to preserve the vacuum in the absorption system and to condition the product water without increasing energy consumption.

3. Methodology

The study was conducted to show the technical feasibility of the system under different operating conditions by means of a general methodology for the generation of the technological proposal, and the simulation procedure in Aspen Plus (see diagram in Fig. 2). Once the technological proposal was generated and the decision to use Aspen Plus software package was taken, it was necessary to figure out which were the most appropriate modules, define the design variables, establish the type of properties to be calculated, determine the critical variables, and explore the system's behavior in order to identify the best operating conditions and system sizing. Special care was given to the variables directly affecting capacity and efficiency.

When running the simulation with Aspen Plus in scenarios where there are recirculation streams in the system, it is critical to define and initialize the iterative variables of the convergence block, which define the iterative calculation procedure and, consequently, the final solution for each simulation of the parametric study. In order to simulate the operating system's behavior the following considerations were assumed:

- The absorption and cooling desalination system operates in a stable state, so the vapor generated in flash evaporation 23 must be equal to the product water stream PW;

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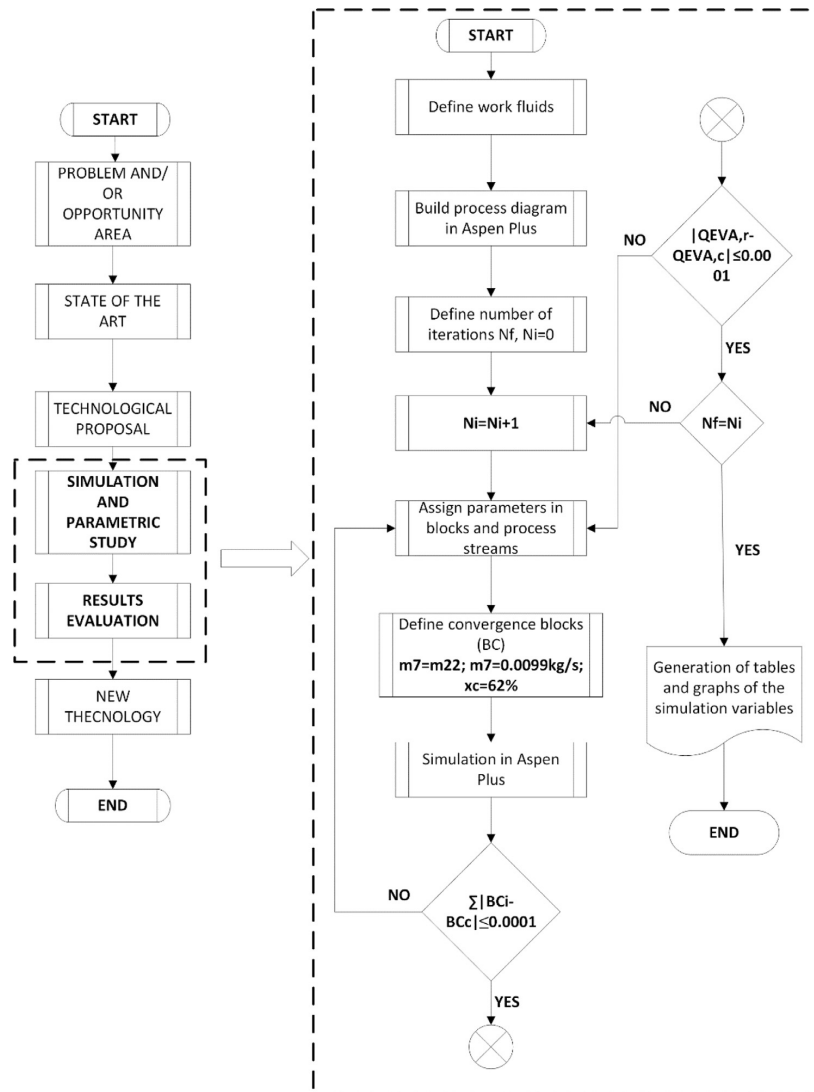


Fig. 2. Diagram of the general methodology and the simulation procedure in Aspen Plus.

- The LiBr/H₂O solution is homogeneous and in equilibrium;
- The high pressure of the system is the saturation pressure corresponding to the coolant temperature at the outlet of the CON, and is equal to the pressure in the GEN;
- The absorber pressure (medium pressure) is the saturation pressure corresponding to the temperature of the diluted solution (1) at the absorber outlet;
- The low pressure in the EVA was established as the saturation pressure of the refrigerant vapor at 1.2 °C, allowing for a temperature gradient with the cool water (18), and capable of rising it up to the 7 °C required in an air conditioning application;
- The design flows of the external circuits of the system adhere to the AHRI standard (AHRI, 2000);
- The process of the expansion of fluids, when passing through the valves of the system, is considered isenthalpic; outside of the specified the system, the process is considered adiabatic;

- The equipment, GEN, EVA, CON, and ABS work in saturation conditions, and the pressure drop in equipment and pipes is negligible.

In order to evaluate and compare the cooling and desalination absorption system under different operating conditions and with other technologies, the following indicators are used: the absorption refrigeration process is evaluated by the Coefficient of Performance (COP).

$$COP = \frac{Q_{EVA}}{Q_{GEN}} \quad (1)$$

Due to the similarity between the operating principle of the MSF and the desalination process of the proposed system, the desalination section is evaluated with the Performance Coefficient (FR) and with the production ratio of pure water (R) (Zheng, 2017a), where FR represents the ratio of fresh water production to

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Table 1

Validation of the absorption subprocess simulation.

Heat exchanger	Heat transfer (Watts)			Outlet external water temperature °C		
	Florides et al. (2003)	Aspen Plus	Error %	Florides et al. (2003)	Aspen Plus	Error %
Generator	1350	1350.5	−0.022	88.00	87.98	−0.023
Condenser	1070	1070.2	−0.018	28.50	28.50	0.000
Evaporator	1000	1000.0	0.000	11.00	11.01	−0.090
Absorber	1280	1280.3	−0.023	31.00	31.09	−0.290
COP	0.74074	0.74046	0.037			

Table 2

Validation of the seawater desalination process in Aspen Plus.

Parameters	(Kahraman and Cengel, 2005)	Aspen Plus	Error %
Seawater discharge [kg/s]	1589	1589	0
Seawater discharge temperature [K]	308	316.94	−2.90
Product water [kg/s]	272	271.24	−0.28
Product water temperature [K]	308	314.50	−2.11
Brine concentration [ppm]	70,093	70,025.50	0.10
Brine flow [kg/s]	536	536.72	−0.14
Brine temperature [K]	308	314.93	−2.25

seawater supplied to the desalination process (Zheng, 2017a), and is calculated as:

$$FR = \frac{m_{23}}{m_{19}} \quad (2)$$

The R factor indicates the ratio of steam production to the energy supplied in the external preheating of seawater (Zheng, 2017a), and is calculated as:

$$R = \frac{(T_{21} - T_{22})}{(T_{21} - T_{19})} \quad (3)$$

4. Validation of the simulation

The simulation of the absorption cooling and desalination system was done with Aspen Plus. This simulator is widely used in chemical processes due to its large database and experimental correlations, which allow for the calculation of thermodynamic properties of pure fluids and mixtures. The validation of the simulation was done separately for: (1) the cooling subprocess; and, (2) the desalination subprocess. For part 1), the experimental results presented by Florides et al. (2003) were used, and for part 2) a comparison was made to experimental results reported by Kahraman and Cengel (2005) and Palenzuela et al. (2014). To perform the validation of the proposed system, the simulation of the absorption subprocess was adjusted to the type and conditions of the machine reported by Florides et al. (2003). As can be seen in Table 1, deviation of the results of the simulation with respect to the experimental results is less than 0.3%, which is why it is considered that the simulation of the LiBr/H₂O absorption processes in Aspen Plus is satisfactory.

In order to validate the desalination process and the calculation of thermodynamic properties of seawater in Aspen Plus, the simulations of an MSF process of 22 stages and a 14-effects MED were performed. The results were compared with the results presented by Kahraman and Cengel (2005) and Palenzuela et al. (2014), respectively. The seawater composition reported by (Zheng, 2017b), and the concentration and temperature provided by Mejía Mercado et al. (2014) were used. The results of the simulation of MSF and MED are shown in Table 2 and in Fig. 3, respectively. It is noticeable that the largest deviation was in the concentration of seawater, being −2.9% for an MSF, while for the MED the results are reasonable. It is concluded that Aspen Plus is one of

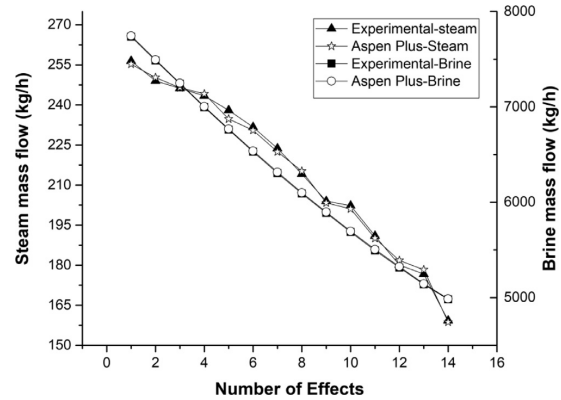


Fig. 3. Comparing MED simulation against experimental data presented by Palenzuela et al. (2014).

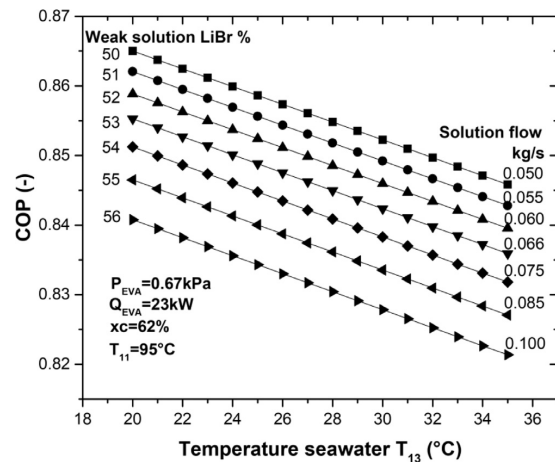


Fig. 4. Coefficient of Performance (COP) of the system under different cooling temperatures, concentration and flow of the working solution.

the best ways to perform the theoretical study of the technological proposal.

5. Results and discussion

The operation of the absorption cooling and desalination system is subject to the intermittences of the solar resource, to seawater temperature variation, and to cooling demand. Due to the aforementioned, the technical feasibility or operational behavior of the system was carried out by studying: (1) the effect of seawater when used as a cooling medium and its impact on COP, R and FR indicators, as well as on the internal pressures of the system

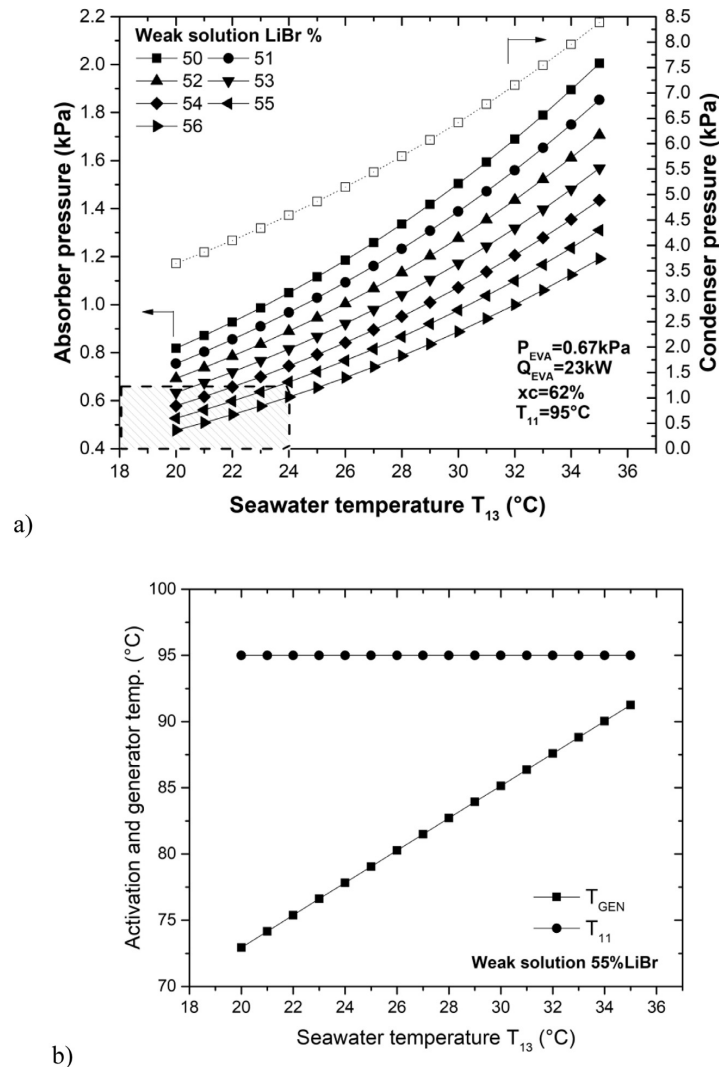


Fig. 5. High and low system pressure under different cooling and concentration temperatures.

when there are changes in the concentration of the LiBr/H₂O solution and in its absorption potential with a third level of pressure; and, (2) the operation of the system under a different activation fluid (thermal solar energy) and cold water temperatures, in order to establish design conditions.

In order to favor the absorption process, three levels of pressure were established: “high”, “medium”, and “low” pressures for the condenser/generator, absorber and evaporator respectively. A temperature difference of 5 °C was considered between the temperature of the incoming seawater and the absorber (13), with respect to the temperature of the diluted solution (1), and between the temperature of incoming seawater and the condenser (15), with respect to the condensed stream (8).

In order to evaluate the behavior of the proposed machine under different seawater conditions (depending on the coastal zone), Fig. 4 shows the behavior of the efficiency and the internal variables (concentration and flow of diluted solution) for 23 kW

cooling capacity, 95 °C activation temperature, 0.67 kPa evaporator pressure (air conditioning application), and a strong solution concentration of 62% of LiBr at the output of the generator to prevent crystallization. Fig. 4 shows an inverse correlation between the weak solution concentration and the COP; as the former decreases, the latter increases and takes values higher than those reported in conventional single-effect absorption systems (COP=0.72) (Kim and Infante Ferreira, 2009; Gomri, 2010; Xu and Wang, 2016). For example, using seawater at 20 °C, and a 50% concentration diluted solution, the proposed machine presents an efficiency 20.14% higher than that of the conventional cycle. The improvement stems from increased pressure in the absorber, which decreases the concentration of the diluted solution at the exit of the absorber, thus, favoring the absorption. Therefore, the solution enters the generator with a greater refrigerant availability and, by keeping a fixed concentration of the strong solution and the capacity of the machine, the weak solution flow decreases (handling a smaller

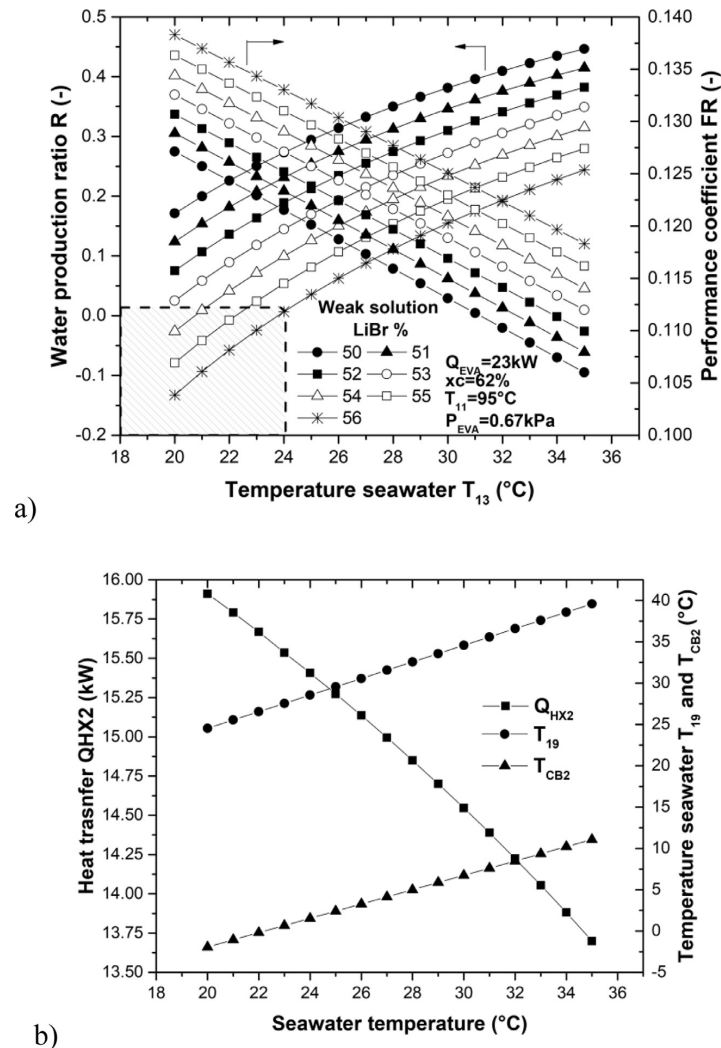


Fig. 6. Indicators of the desalination subsystem under different cooling and concentration temperatures.

amount of working fluid), which increases efficiency. The pressure in the absorber has a direct relation with the cooling medium (seawater temperature), as mentioned before; by defining an approximate temperature of 5 °C in the absorber, the saturation temperature of the solution is automatically set, and by establishing its concentration, the work pressure is directly found. By increasing the pressure in the absorber there is a lower pressure differential between the absorber and the generator, requiring a solution pump of lower capacity, which consumes less energy. The absorber pressures of Fig. 5 are presented in Fig. 5a.

Fig. 5 shows the effect that the seawater temperature has on the pressure of the absorber and the generator/condenser. It can be seen that as the seawater temperature increases, it is necessary to design the machine with greater pressure in the absorber and generator/condenser. If the results in the absorber are analyzed, a greater efficiency is achieved by decreasing the concentration as mentioned earlier; however, the system can operate at a lower pressure, sacrificing a little efficiency. For example, with a seawater temperature of 35 °C and a 50% concentration, there is a

COP of 0.845, and this decreases by an average of 0.15% for every 1.00% increase in the diluted solution concentration. The increase in seawater temperature has a negative effect on the dimensioning of the generator, since, as seen in Fig. 5b, when the pressure is increased, the boiling point (T_{GEN}) of the solution is increased, and this causes a lower temperature gradient between the activation fluid and the LiBr/H₂O solution, causing a higher heat exchange area requirement. Therefore, in order to favor the processes of absorption and heat transfer, it is recommended, when designing the machine, to increase the pressure in the absorber and decrease the pressure in the generator within what is allowed by the seawater. It should be noted that, for any given operating condition of the generator/condenser, the absorber can operate with different levels of pressure and concentration.

It is worth noting that when seawater temperatures are between 20 and 24 °C (shaded area, Fig. 5a) and concentrations of diluted solution of 53–56% are used, the resulting pressure of the absorber is lower than that of the evaporator. This is possible because the proposed system separates the evaporator from the

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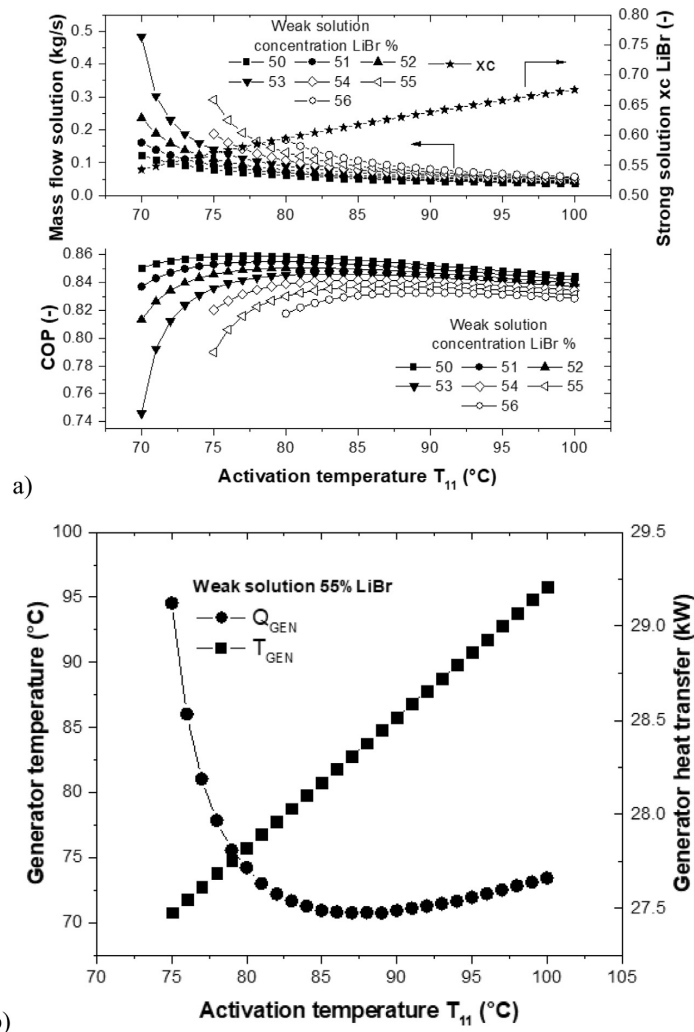


Fig. 7. Influence of the activation temperature on the COP, on the solution flow and concentration of the strong solution. $Q_{EVA} = 23$ kW, $T_{13} = 27$ °C, $P_{EVA} = 0.67$ kPa, $P_{CON} = P_{GEN} = 5.44$ kPa, $P_{ABS} = 1.1$ kPa, $x_1 = 55\%$.

absorber and is able to handle three levels of pressure. This is an advantage over a conventional system, which is unable to handle such pressure levels within the evaporator, since freezing may occur. The study showed that the absorber can operate at pressures lower than those of the evaporator; thus, care must be taken to avoid crystallization of the solution. When the machine operates with a seawater temperature lower than 24 °C, there is the option of using a system configuration whereby part of the cooling water emerging from the absorber is recirculated in such a way that the temperature is adjusted to the desired levels, or to a level where the absorption process is favored.

As previously discussed, if the proposed cooling system is compared against the cooling tower system, in humid climates and with temperatures higher than 40 °C, the former has clear advantages, since the COP increases up to 20.14%, compared to the 0.72 COP reported in the literature for a conventional system. Moreover, the proposed technology generates more seawater desalination.

In Fig. 6a and b, different operating conditions for the desalination subsystem, as well as its interaction with the cooling subsys-

tem, are analyzed for different seawater temperatures, using the R and FR indicators. In Fig. 6a, it can be seen that, as seawater temperature increases (13), the performance coefficient (FR) decreases, which is due to the fact that increasing the pressure in the absorber reduces the pressure drop in the SWV expansion valve, causing less flash and a higher temperature in the seawater (22). By keeping cooling capacity constant and a temperature difference of 15 °C between the fluid leaving the generator (12) and the seawater leaving the exchanger HX2 (21), it becomes necessary to increase the flow of the stream (19) to keep the steam flow entering the absorber (23) constant, preventing an unbalance in the cooling subprocess. Therefore, by increasing the seawater (19) and keeping the product water flow constant (23), FR decreases. The latter indicator has no great influence on the design of the proposed machine because, in order to increase the flow of seawater (19), no change is necessary other than the manipulation of the stream.

Water production ratio (R) tends to increase as seawater temperature increases (13). This is due to the increase in the temperature of stream 19, as can be seen in Fig. 6b. As a consequence, and

ANEXO H. A NOVEL LiBr/H₂O ABSORPTION COOLING AND DESALINATION SYSTEM WITH THREE PRESSURE LEVELS

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Table 3
Properties of LiBr/H₂O absorption cooling and desalination system streams under final design conditions.

Streams	Fluid	Pressure kPa	Temperature °C	Enthalpy kJ/kg	Mass Flow kg/sec	X % (ppm)
1	LiBr/H ₂ O	1.16	32.00	−10,056.7	0.0805	51
2	LiBr/H ₂ O	5.44	32.00	−10,056.7	0.0805	51
3	LiBr/H ₂ O	5.44	60.34	−995.3	0.0805	51
4	LiBr/H ₂ O	5.44	72.79	−9153.3	0.0707	58
5	LiBr/H ₂ O	5.44	36.98	−9223.2	0.0707	58
6	LiBr/H ₂ O	1.16	46.19	−9223.2	0.0707	58
7	H ₂ O	5.44	72.79	−13,343.4	0.0097	0
8	H ₂ O	5.44	34.32	−15,836.9	0.0097	0
9	H ₂ O	0.67	1.19	−15,836.9	0.0097	0
10	H ₂ O	0.67	1.19	−13,477.6	0.0097	0
11	H ₂ O	200	77.00	−15,658.3	1.6110	0
12	H ₂ O	200	73.00	−15,675	1.6110	0
13	Seawater	200	27.00	−15,530.7	2.7000	(35,000)
14	Seawater	200	29.32	−15,521.2	2.7000	(35,000)
15	Seawater	200	29.32	−15,521.2	2.7000	(35,000)
16	Seawater	200	31.50	−15,512.2	2.7000	(35,000)
17	H ₂ O	200	12.00	−15,930.3	1.1110	0
18	H ₂ O	200	7.06	−15,951	1.1110	0
19	Seawater	200	31.50	−15,512.2	0.1203	(35,000)
20	H ₂ O	200	71.07	−15,683	1.6110	0
21	Seawater	200	58.01	−15,403.2	0.1203	(35,000)
22	Seawater	1.16	9.26	−15,403.2	0.1203	(35,000)
23	H ₂ O	1.16	9.26	−13,462.3	0.0097	0
24	Seawater	200	31.50	−15,512.2	2.5790	(35,000)
25	Seawater	101.38	30.61	−15,499	2.6902	(35,100)
BC2	Brine	1.16	9.26	−15,574.1	0.1105	(38,000)

Table 4
Energetic evaluation of LiBr/H₂O absorption cooling and desalination system streams under final design conditions.

COMPONENTS	INPUTS (kW)	OUTPUTS (kW)	ENERGETIC INTEGRATION (kW)
GENERATOR	26.92		
CONDENSER		24.31	
ABSORBER		25.73	
HX1			4.94
EVAPORATOR	23.08		
HX2			13.11
SP	0.000461		
TOTAL	50.00	50.04	18.05
COP	0.86	Coefficient of Performance	
Product Water	838 L/day	Daily Production	
FR	0.08	Performance Coefficient	
R	1.84	Production Ratio	

as seen in Fig. 6b, as seawater temperature increases, and keeping stream 21 temperature's constant, fewer external heating services (QHx2) are required in the desalination subsystem. The result is a machine with smaller heat exchanger HX2 and solar collector field SC. However, coupling the cooling and desalination subprocesses, and increasing the pressure in the absorber, as mentioned before, decreases the expansion process in the SWV valve, while also increasing the temperature of current 22 (see Fig. 6b), which tends to decrease R. With the operative study it was found that the increase in the temperature of stream 19 benefits R, but an increase in the pressure of the absorber caused by the rise in seawater temperature has adverse effects on it. However, the results show that the positive effect predominates and causes an increase in the R value.

For a given application, where seawater temperature is fixed, Fig. 6a shows that the desalination process has a higher R value since the machine is designed to work with lower concentrations of weak solution. This coincides with the cooling subprocess, which obtains a higher COP with lower weak solution concentration levels.

Since the solar power source is variable and intermittent, it becomes especially interesting to analyze the efficiency behavior COP, as well as the internal variables of the proposed technology, by making variations in activation temperature (T11) and maintaining capacity constant. Fig. 7a shows that the COP increases to a max-

imum point and then falls slightly as the activation temperature increases. This is due to the fact that the mass flow and the temperature of the weak solution decrease and increase accordingly, causing the decay of the heat added to the generator, as shown in Fig. 7b. It is also shown in Fig. 7a that, as the concentration of the weak solution decreases, the system's efficiency becomes greater, which coincides with the aforementioned.

Table 3 is the result of the operative analysis for the technical feasibility of this proposal. It shows the summary of the properties of all system currents for a seawater temperature of 27 °C and a cooling capacity of 23 kW. Table 4 shows the indicators of the cooling and desalination system and the amount of product water, as well as the heat transfer in each component. For the latter conditions a COP of 0.86 was obtained, which is 19.44% higher than that of a conventional single-effect absorption system. Water production is 838 L/day, considering a constant design operating condition, enough to meet the needs of a family of up to 7 people, according to the World Health Organization (WHO) (Howard and Bartram, 2003).

6. Conclusions

This paper presents a novel LiBr/H₂O absorption cooling and desalination system with three pressure levels, which uses

ANEXO H. A NOVEL LiBr/H₂O ABSORPTION COOLING AND DESALINATION SYSTEM WITH THREE PRESSURE LEVELS

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seawater as a cooling medium and is activated by solar thermal energy. It was found that the proposed technology has 19.44% more efficiency than a conventional absorption cooling system with a capacity of 23 kW, producing, additionally, 838 L/day of freshwater, with an FR of 0.08 and an R of 1.84.

In the study performed here, a constant design operating condition was considered to generate 838 L/day of water product because the variability of the solar resource can be covered and controlled with an auxiliary heating system. The actual application will present different operational dynamics throughout the year due to the availability of solar resources. Such a study should be specifically addressed in an upcoming work, reporting a full analysis of such variations.

By increasing the pressure in the absorber, the cooling effect was significantly favored, affecting the desalination process only slightly, allowing the production of enough water to meet the daily needs of up to 7 people.

The proposed system works with an activation temperature (77 °C) lower than that of a conventional system (93 °C), managing a small flow (0.0807 kg/s) and a low concentration of the diluted solution (51%), making it possible to use low-cost and efficient thermal solar collectors available in the market.

In order to maintain the equilibrium or stability of the cooling subsystem, it is very important that the water outflow from the evaporator as water product be equal to the incoming water flow from the desalination subsystem into the absorber. Therefore, it becomes necessary to install a control loop that senses the flow of said streams or the electrical conductivity of the diluted solution outgoing from the absorber, and that manipulates the stream divider (DIV).

With the configuration of the proposed technology, managing three levels of pressure and using seawater as a cooling medium, the simultaneous production of cooling and desalination is significantly benefited, providing greater internal energy integration, reducing the number of components, and increasing cooling and desalination efficiency.

At this time, a prototype system is in construction. In future work, a validation of the prediction by the simulation with experimental data will be conducted. In addition, a water quality study as a result of the absorption cooling and desalination system will be performed.

Declarations of interest

None.

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**I. Absorption cooling and desalination system
with a novel internal energetic and mass
integration that increases capacity and
efficiency**

Revista: Desalination



Absorption cooling and desalination system with a novel internal energetic and mass integration that increases capacity and efficiency



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ABSTRACT

This paper presents a novel cooling and desalination system that uses seawater as coolant, and that has a high energetic and internal mass integration, as well as higher capacity and efficiency levels. The proposed system was simulated using Aspen Plus software; its operative behavior was also evaluated in order to prove its technical feasibility. Using a 23 kW simple-effect absorption cooling machine as a basis, and utilizing the produce of desalination as coolant, this technological approach increases the system's capacity by a factor of 87.4, to 2012 kW. It also increases the COP value to 6.15, 8.54 times greater than the 0.72 COP reported for this type of systems. Furthermore, the system produces 73,569 L/day of water, with an RR of 0.56 and a PR of 6.63. The proposed technology presents a setup that allows for a high energetic and internal mass integration, which increases the system's capacity and efficiency.

1. Introduction

In order to satisfy spaces conditioning, and water availability needs [1–6], there is, currently, an ongoing search for the development of renewable energy-activated technologies that, sustainably, both cool spaces, and produce drinking water.

The potential of absorption systems has increased due to their flexibility, which allows them to integrate with renewable energy-activated poly-generative systems [7–10]. On the other hand, Multiple Effect Desalination (MED) and Multi-Stage Desalination (FLASH) systems have a large compatibility with thermal solar energy, as well as with absorption and adsorption cycles hybridization [11–15].

Looking forward to increase the technical and economic feasibility of this type of technologies, Wang and Lior [16,17] theoretically evaluated the integration of a 6-effects MED with a simple-effect LiBr/H₂O heat pump. The heat from both the condenser and the absorber activated the MED. The heat pump presented a COP of 1.6, proving that an energy saving of 42%, when compared to conventional processes, is achievable.

Abdulrahim and Darwish [18] theoretically studied the coupling of a thermo-solar energy activated LiBr/H₂O absorption system and an 8-

effects MED. The absorption-cooling unit had a 2148 kW capacity and a COP of 0.74, while the MED produced 459 m³/day of water, with a constant PR of 5.7 when controlling for the desalination level of the last effect.

In another study, an energetic and internal mass integration was conducted to couple a 6-effects MED with a simple effect LiBr/H₂O absorption thermal compressor powered with 40 kW [19]. In this study, the authors used water vapor from the generator to activate the first effect of the MED; they also incorporated a fraction of the vapor generated by the last effect of the MED into the absorber. After an exergy optimization, the authors reported a water production of 15,000 m³/day with a GOR of 9.44, as well as an exergy efficiency and a total product cost rate of 10.32% and 0.11 \$/s, respectively, an improvement of 6.47% and 4.87% compared to the MED-Thermal Vapor Compression (TVC) system.

Alelyani et al. [20], using the exergy analysis methodology, conducted a study of the coupling of a 14-effects MED with 1-effect and 2-effects NH₃/H₂O absorption cooling systems. The heat from the condenser and the rectifier activated the MED. With this coupling, the total exergy destruction decreased in an average of 55%. The cooling Unit Production Cost (UPC) decreased in 43% and, with the coupling of the

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ANEXO I. ABSORPTION COOLING AND DESALINATION SYSTEM WITH A NOVEL INTERNAL ENERGETIC AND MASS INTEGRATION THAT INCREASES CAPACITY AND EFFICIENCY

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Nomenclature		PWT	product water tank
Symbols		V3V1	3 way valve
Q		Subscripts	
T		Number of streams: 1,2,3...	
P		Abbreviations	
GOR		GEN	generator
COP		ABS	absorber
COP_G		CON	condenser
PR		HX1	recovery heat exchanger
RR		HX2	heat exchanger
UPC		HX3	cooling heat exchanger
m		SC	solar collectors
X		DIV	stream divider
xc		DIV2	seawater divider
SP		SV	solution expansion valve
SWP		SWV	seawater expansion valve
SWP1		SWV1	MED seawater expansion valve
EP		RV	refrigerant expansion valve
PW		V3RV	3 way refrigeration valve
BP		EJE	ejector
PWP		Dish	ejector discharge
BC1		MC	MED condenser
BC2			
BRC			

simple and double state absorption units, the UPC for water production increased 19% and 3%, respectively.

Experimental studies to corroborate the results of the hybridization of cooling and desalination systems have been conducted. Chorak et al. [21] analyzed the coupling of a 14-effect MED to a double-effect LiBr/H₂O absorption cooling system. The maximum values for COP, distilled water, and PR values obtained were 2.08, 2.42 m³/h, and 18.53, respectively. The cold water from the absorption machine was not used for space cooling, but instead to condense the last stage of the MED.

Recently, López-Zavala et al. [22], proposed an open, simple effect LiBr/H₂O absorption cooling and desalination system with three pressure levels and activated with solar-thermal energy. The system had a cooling capacity of 23 kW and a water production rate of 838 L/day. The authors proposed a configuration that allowed to have an absorber operation pressure greater than the one of the evaporator, allowing to increase the system's efficiency in 22% when compared to the conventional simple-effect absorption cooling cycle.

There have been several efforts to integrate absorption and thermal

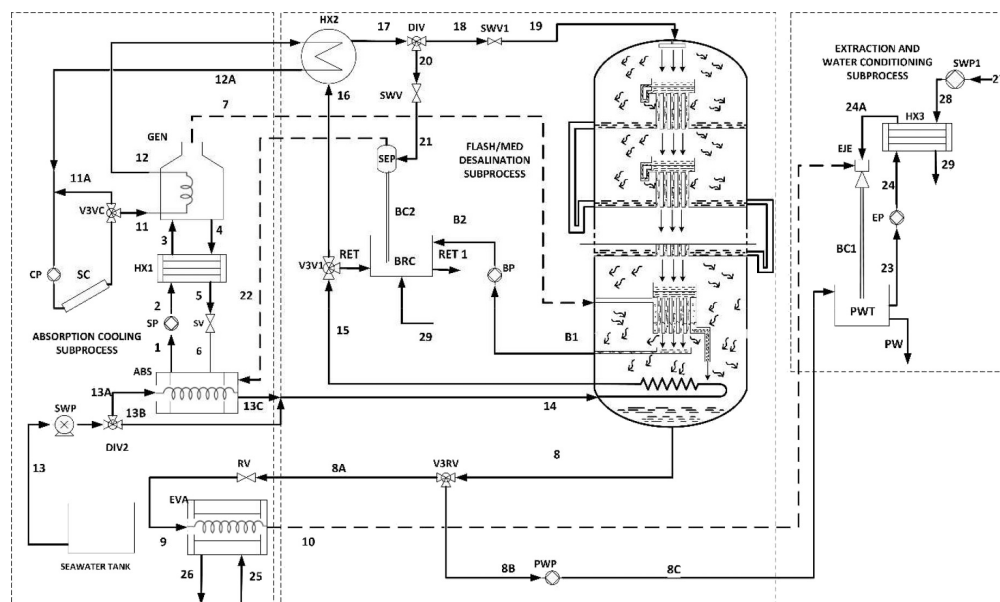


Fig. 1. Diagram of the solar thermal absorption LiBr/H₂O cooling and FLSH/MED desalination system.

ANEXO I. ABSORPTION COOLING AND DESALINATION SYSTEM WITH A NOVEL INTERNAL ENERGETIC AND MASS INTEGRATION THAT INCREASES CAPACITY AND EFFICIENCY

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desalination cooling technologies. This article proposes an absorption cooling and desalination system with a novel energetic and internal mass integration that increases capacity and efficiency. The system presents a configuration, and operation conditions that allow for an optimal coupling between the cooling and the desalination subsystems, resulting in a high energetic and internal mass integration. The simple-effect absorption cooling subsystem has three pressure levels; the pressure of the generator equals the pressure of the last effect of the MED, and the pressure of the absorber is greater than the evaporator pressure. The system uses seawater as coolant, and the vapor produced in the generator to increase the desalination in the last effect of the MED. The proposed operating conditions achieve a triple mass integration: a) the water produced by the MED is used as coolant in the evaporator, significantly increasing the system's cooling capacity; b) the vapor from the generator is incorporated into the MED, achieving an increase in produced water; and, c) the vapor from the FLASH desalination system is carried to the absorber to maintain the system's equilibrium. The technological proposal is now in patent phase, with file number MX/a/2018/015007.

2. Description of the system

Fig. 1 presents a system that takes advantage of the operational characteristics of simple-effect LiBr/H₂O absorption cooling systems, and of FLASH and MED desalination processes. The system is composed of three subprocesses: absorption cooling; FLASH/MED desalination; and, extraction and water conditioning.

The **absorption cooling subprocess** is composed of an absorber (ABS) that receives the water vapor generated in the **FLASH/MED desalination subprocess** (22). Pump (SP) sends the ABS's LiBr/H₂O solution into the generator (GEN), previously going through the heat exchanger-energy recuperator (HX1). Within the GEN, heat from the solar collectors (SC) incorporates into the LiBr/H₂O solution in order to release the cooling water as vapor (7). This solution, which has a higher concentration of LiBr (4), goes through the HX1 and the solution's expansion valve (SV) in order to absorb the water vapor (22) that comes out of the **FLASH/MED desalination subprocess**, and, finally, returns to the ABS. In order to remove the heat generated by the exothermal process, seawater from pump SWP cools the ABS. The water vapor (7) that comes out of the GEN goes to the last effect of the MED desalination process. When it interchanges latent heat with the brine from the last effect, it condenses and mixes with the produced water from the MED. Once they are mixed and in thermal equilibrium, the mix leaves the MED (8) and goes through the three-way cooling valve (V3RV), which divides the flow depending on the required cooling capacity. The PWP sucks current (8B) in order to incorporate it into the produced water deposit (PWP) at an atmospheric pressure. Current (8A) goes through the coolant's expansion valve (RV), where, after diminishing its pressure and temperature, it is ready for the cooling effect in the evaporator (EVA). The water intended to be cooled grants its heat to the coolant, whose pressure and temperature have been lowered in the previous step, and becomes vapor. Then, the vapor (10) that comes out of the EVA goes to the extraction and water conditioning subprocess.

The **FLASH/MED desalination subprocess** uses seawater flow (13), which is carried by the SWP to cool the ABS and to condense the vapor generated in the last effect of the MED. The current divider (DIV2) separates the seawater into currents (13A) and (13B). Current (13B) cools the ABS, maintaining a fixed flow. Current (13B)'s flow is variable, so that, after mixing with current (13C), which comes out of the ABS, it condenses the vapor generated in the last effect of the MED. Afterwards, the seawater (15) that leaves the last effect of the MED goes through the three way valve (V3V1) in order to reduce current (16)'s flow. Within the heat exchanger (HX2), the heat-carrier fluid from the SC heats current (16). Afterwards, this current goes through the three-way valve (DIV) in charge of splitting it (17), using a portion of it for the MED desalination process (18), and the rest for the FLASH

evaporation process (20). Current (20) goes through the seawater expansion valve (SWV), where it comes out in two stages (liquid/vapor) as current (21). The two-stage current (21) goes into the barometric stage separator (SEP), and the outgoing vapor (22) is introduced into the absorption cooling subprocess, where the corresponding absorption is carried out. The liquid that exits the SEP falls through a barometric column (BC2), where it makes a hydraulic seal, which maintains the vacuum pressure. Afterwards, the liquid goes into the brine, or concentrated seawater, deposit (BRC). In order to reduce its concentration, so that it can properly return to the sea (RET1), current (BC2) mixes with seawater that comes out of the V3V1 (RET current), and with the brine that comes out of the MED desalination process (B2). This occurs within the BRC. On the other hand, current (18) goes through an expansion valve (SWV1), where its pressure drops, in order to split current (19) into two stages so it can enter, in that state, into the first effect of the MED. The liquid part of current (19) falls into a tray, pushing the brine in it, so that it goes through a U-shaped pipe connected to the second effect, which has a lower pressure. The column of the U pipe allows for a hydraulic seal that keeps the pressure differential. The brine in the upper tray of the second effect falls through the internal falling film pipes and, since it has a temperature lower than that of the first-effect's vapor, receives its heat and, again, leaves in two stages. The vapor of the first effect of the MED condenses and leaves as produced water from the first effect, mixing with the produced water from the following effects. The produced water from each effect communicate through a U pipe, which maintains the pressure differentials within each effect. This process repeats until the water reaches the final effect. The total produced water that comes through the external U pipes at the last effect is carried to the absorption cooling subprocess as current (8), while the BP extracts brine concentration (B1) in order to incorporate it into the BRC.

In the **extraction and water conditioning subprocess**, the water vapor (10) is suctioned by the ejector (EJE) which is activated by the liquid water driving fluid (24A) supplied by the pump (EP). The high-pressure stream 24A and the low-pressure stream 10 are mixed in the EJE and leave in liquid form at an intermediate pressure. The heat of condensation of the steam 10 is received by the current 24A, causing the mixture of both liquids to increase its temperature. The water that leaves the EJE passes to a barometric column (BC1) and then to the product water tank (PWT), from where the pump EP takes the fluid that activates or makes the EJE work. To remove the heat of condensation from the steam 10, the stream (24) is cooled with seawater (27) in the heat exchanger HX3. Seawater (stream 29) after cooling, is sent to the BRC. The application that this ejector has in the extraction and adaptation of the product water subprocess, is to condense a vapor that is at low pressure and temperature in order to take it to environmental conditions.

The main characteristics of the proposed system are the following:

- A configuration that allows for the increase of the cooling and desalination capacity. To achieve this, the vapor produced in the generator goes into the MED in order to increase desalination. Afterwards, this vapor joins the produced water. The water desalinated by the MED serves as coolant within the absorption cooling subprocess; after producing the cooling effect, it is extracted as produced water.
- A greater internal energetic integration, because the seawater is used to cool the absorber and to condense the vapor of the last effect of the MED. This reduces the external cooling and heating services within the process.
- An adjustment of the operating conditions of the absorption cooling and of the MED desalination subprocesses in order to conduct their coupling, and to maximize their energetic and internal mass integration.
- Removal of the condenser and the cooling tower within the absorption cooling subprocess, as well as a modification of the MED's

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last-effect condenser.

- Implementation of passive elements such as ejector condensers.
- A dilution of currents (RET), (29) and (CB2) in order to reduce the environmental impact caused by the brine.

3. Methodology

Fig. 2 shows a diagram with the general methodology for the development of the proposed technology, as well as the Aspen Plus simulation procedure.

Once the technological proposal was generated, and the decision to use Aspen Plus was taken, it was necessary to choose the adequate modules, establish the type of properties to calculate, determine the critical variables, and explore the operational behavior, in order to find the best operating conditions and to size the system. Special attention was paid to the variables that directly affect capacity and efficiency.

When conducting an Aspen Plus simulation, it is critical, whenever

there is current recirculation, to define and initialize the iterative variables of the convergence block. These variables establish the iterative calculation procedure and, thus, the final solution of each simulation within the parametric study.

The operational behavior of the system was simulated considering the following assumptions:

- The FLASH/MED absorption cooling and desalination system operates at a stable state; thus, the vapor generated by the FLASH evaporation (22) should be equal to vapor current (7) produced by the GEN. This vapor corresponds to the one generated by a conventional simple-effect LiBr/H₂O absorption cooling machine with a 23 kW capacity.
- The pressure within the GEN is the saturation pressure that corresponds to the temperature of the last effect of the MED. The pressure of the first effect of the MED is the pressure that corresponds to a Top Brine Temperature (TBT) of 70 °C.

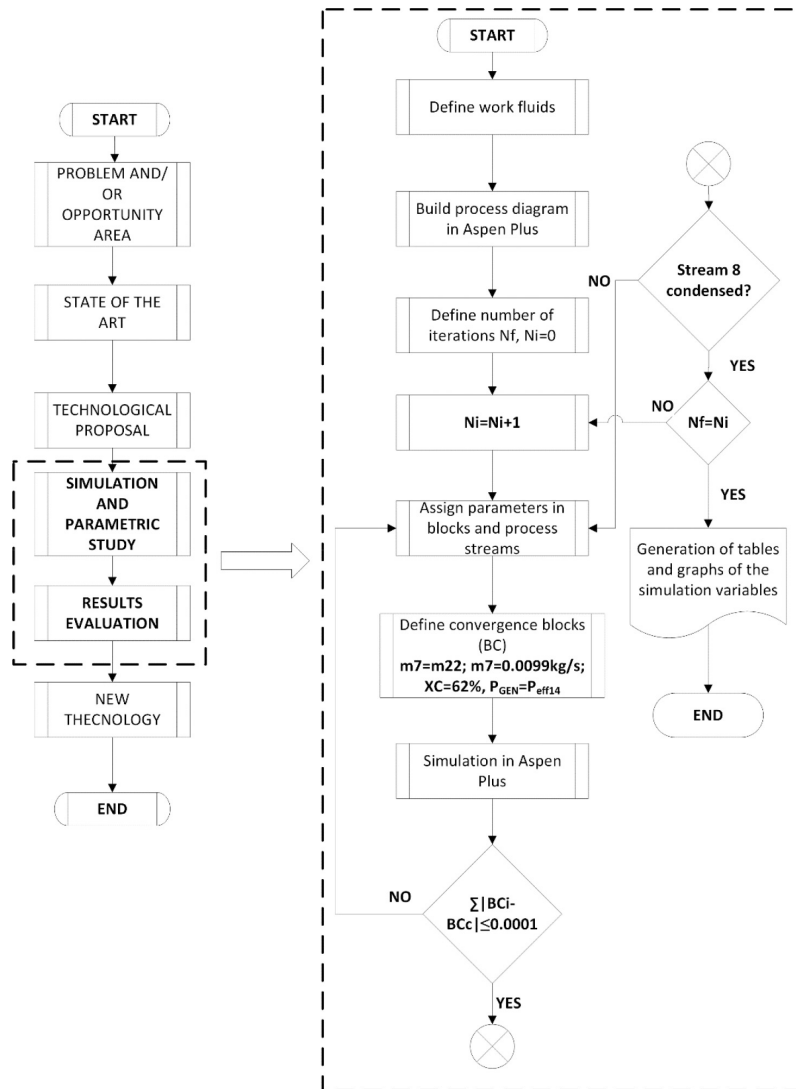


Fig. 2. Diagram of the general methodology and Aspen Plus simulation procedure.

- The absorber pressure is the saturation pressure corresponding to the temperature of the diluted solution (1) at 51% LiBr at the absorber outlet [22].
- The pressure in the EVA is different to that of the absorber, and is set to the saturation pressure of the coolant vapor at 1.2 °C, which allows to have a gradient of temperature within the cold water (26), and to take it down from 12 °C to the 7 °C required for an air conditioning application.
- An approximation temperature of 5 °C was considered between currents (13) and (1) of the absorber, and between currents (14) and (8) of the condenser of the last effect of the MED.
- The external circuits' design mass flows (heat carrying fluid, coolant water, and cold water) of the cooling subprocess are set in accordance to the AHRI standard [23]. The seawater flow (m13A) was set to 1.5 kg/s; the heat carrying fluid (m11), to 1.611 kg/s; and the cold-water flow is variable in order to maintain the evaporator's exit temperature at 7 °C.
- Current (8) is always condensed; hence, seawater flow m13 is variable.
- The GEN, EVA, CON, and ABS devices work at saturation conditions, and the pressure drop within the devices and the pipelines is negligible.
- The expansion process, when the fluids go through the valves of the system, is isenthalpic. Beyond the specification, the system is adiabatic.

The absorption cooling subprocess is gauged using the Coefficient of Operation (COP). This indicator represent the relationship between the cooling capacity and the total heat produced by such cooling, and is expressed as follows:

$$COP = Q_{EVA} / (Q_{GEN} + Q_{HX2} + W_{EP} + W_{SP} + W_{SWP} + W_{SWP1}) \quad (1)$$

The MED desalination subprocess is gauged using two indicators: Performance Ratio (PR) and Recovery Ratio (RR) [24] [25,26]. It is important to highlight that the PR represents a ratio between the amount of water produced and the energetic requirement to produce it; thus, for this indicator, the two energy inputs involved in this subprocess were included. The RR indicator is the ratio of the water produced between the seawater supplied to the MED. Even though vapor flow (7) is not a seawater flow, it is a mass flow that enters the MED and increases desalination. Therefore, it is considered for this indicator.

$$RR = \frac{m_8}{m_{19} + m_7} \quad (2)$$

$$PR = \frac{m_8 \left(2326 \frac{kJ}{kg} \right)}{Q_{MED} + m_7 \left(2326 \frac{kJ}{kg} \right)} \quad (3)$$

The energy contributed by the heat exchanger QHX2 is used in the FLASH and MED subprocesses; hence, the heat used for the MED (Q_{MED}) can be expressed as a QHX2 relationship as follows:

$$Q_{MED} = \left(\frac{m_{19}}{m_{16}} \right) Q_{HX2} \quad (4)$$

The solar collector field must satisfy the GEN and the HX2 requirements. The collector's heat can be expressed as follows:

$$Q_{SC} = Q_{HX2} + Q_{GEN} \quad (5)$$

4. Validation of simulation

Aspen Plus software was used for the simulation of the absorption cooling, and MED desalination system. This simulator is widely used for chemical and heat and mass transfer processes because of its large database and its experimental correlations for the calculation of thermodynamic properties of pure and mixed fluids. Furthermore, it can

model a wide range of devices for different, already validated, industrial applications. The cooling subprocess, and the FLASH and MED desalination subprocesses were separately validated through simulation. For the cooling subprocess, the experimental results obtained by Florides et al. [27] were used. For the FLASH and MED subprocesses, a comparison was made with the experimental results reported by Karaman et al. [28] and Palenzuela et al. [26]. The results of this validation can be found in López-Zavala et al. [22].

5. Results and discussion

The operational behavior of this system depends on the intermitencies of the solar resource, as well as on the flow and temperature of the seawater. The variability of the solar energy has a direct impact, both in the capacity and in the efficiency of the solar collector field, and it manifests directly in the temperature of the heat carrying fluid, causing different operating conditions for the proposed system. Hence, in order to establish the standard design conditions, the technical feasibility of the technological proposal was assessed through the study of the effect that different activation temperatures, as well as different seawater flow conditions, have on the main indicators of the system.

A 95 °C activation temperature for the absorption cooling subsystem, and a seawater flow of 1.5 kg/s, with a temperature of 27 °C [28], for the cooling of the absorber, were used to assess the system's operational behavior. The seawater mass flow entering the FLASH/MED desalination subprocess was varied; and the impact of these variations on the heat transfer of the HX2 and EVA (cooling capacity), the COP, water production, and the PR and RR desalination indicators, was analyzed.

Fig. 3 shows the cooling capacity (Q_{EVA}) and the efficiency of the absorption cooling subprocess (COP) when the water supplied to the desalination subprocess (16) is varied. In this figure, we can see the strong impact that using the water produced from the desalination subsystem as a coolant has on the cooling capacity. It should be noted that the cooling capacity of the proposed machine can be controlled through the manipulation of the produced water flow using the three way valve (V3RV), unlike conventional cooling systems, where the cooling capacity is controlled through the manipulation of the amount of heat added to the generator. Hence, in order for the proposed system to achieve its highest capacity, the absorption cooling subsystem should operate at full capacity.

When the activation fluid's temperature, the vapor flow produced by the GEN, and the concentration of the strong current (4) are constant, the heat transfer within the GEN shows no significant change. Thus, the temperature of the activation fluid (T₁₂) at the exit of the GEN remains

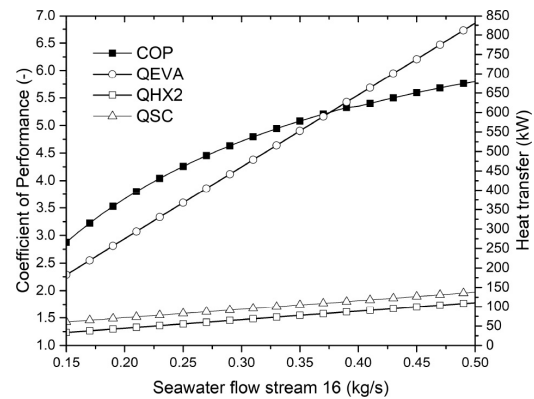


Fig. 3. Behavior of the cooling COP and heat transfer in the evaporator, HX2 and field of solar collectors. T₁₃ = 27 °C, m_{13A} = 1.5 kg/s, T₁₁ = 95 °C, m₁₁ = 1.6 kg/s.

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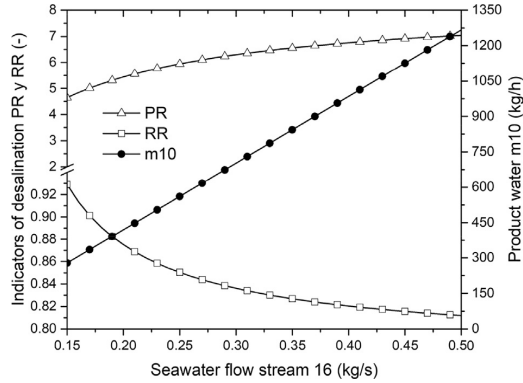


Fig. 4. Effect of seawater flow variation on desalination capacity and, RR and PR indicators; $T_{13} = 27^\circ\text{C}$, $m_{13A} = 1.5\text{ kg/s}$, $T_{11} = 95^\circ\text{C}$, $m_{11} = 1.6\text{ kg/s}$.

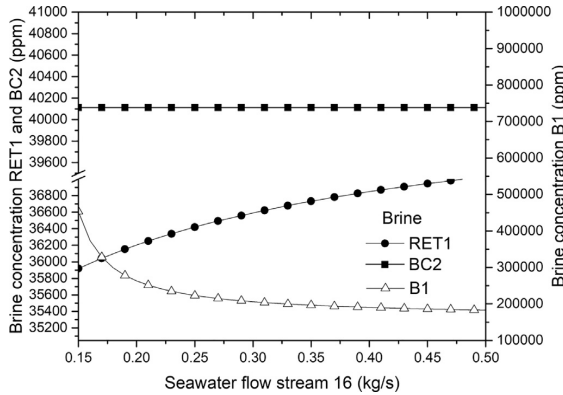


Fig. 5. Effect of variation of seawater flow in the concentration of the brine RET1, BC2 and B1; $T_{13} = 27^\circ\text{C}$, $m_{13A} = 1.5\text{ kg/s}$, $T_{11} = 95^\circ\text{C}$, $m_{11} = 1.6\text{ kg/s}$.

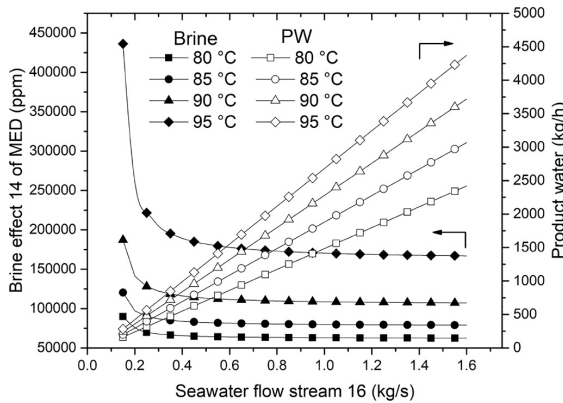
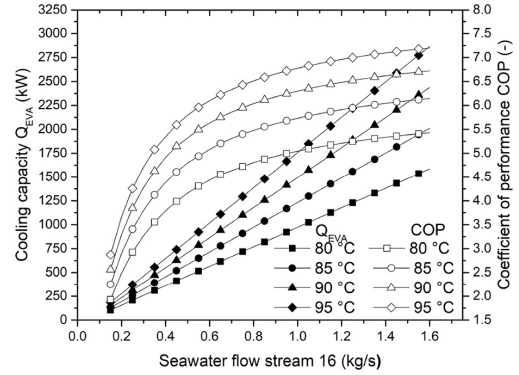
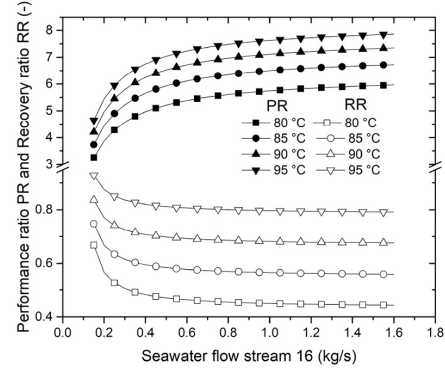


Fig. 6. Effect of variation of temperature T_{11} and flow of seawater 16 in product water and concentration brine B1. $T_{13} = 27^\circ\text{C}$, $m_{13A} = 1.5\text{ kg/s}$, $T_{11} = 95^\circ\text{C}$, $m_{11} = 1.6\text{ kg/s}$.

constant. Given these conditions for T_{12} , as well as an approximation temperature (T_{approx}) between the currents of the HX2 of 5°C , the temperature (T_{17}) of the seawater at the exit of the HX2 does not vary. Therefore, when the flow of current (16) is increased, the heat transference within HX2 also increases (see Fig. 3).



a)



b)

Fig. 7. Effect of temperature variation T_{11} and seawater flow 16 in a) capacity and efficiency of cooling; b) PR and RR desalination indicators. $T_{13} = 27^\circ\text{C}$, $m_{13A} = 1.5\text{ kg/s}$, $m_{11} = 1.611$, $T_{25} = 12^\circ\text{C}$.

In order to replace the coolant removed from the GEN and to keep the concentration of the diluted solution within the ABS (1) stable, it is necessary that the vapor supplied to the ABS (22) equals the vapor produced by the GEN. On the other hand, the pressure drop within the expansion valve of the FLASH section (SWV) is subject to the ABS pressure. This last pressure does not change because it depends on the temperature of the seawater T_{13} (coolant). Hence, the flow of current (20) does not change, meaning that, when seawater flow (16) is increased, the current divider (DIV) increases current (18). Current (18), then, goes through the expansion valve SWV1 and its pressure suddenly drops until it reaches the level that corresponds to the first-effect of the MED. When the flow of current (18) is increased, and when the vapor of current (7) is incorporated into the MED, the generation of produced water (8) increases. Furthermore, when current (8) is used as coolant, the cooling capacity of the system increases significantly.

When the pressure of the generator is set to the same pressure level of the last effect of the MED, the latent heat of current (7) can be used to produce a greater water evaporation, and to achieve a greater energetic and internal mass integration. This, in turn, increases desalination and substantially benefits the cooling capacity. Furthermore, this allows us to remove the condenser from the absorption cooling system.

The contribution of the absorption cooling subsystem in the total cooling capacity depends on the energy and mass integration, since these causes a greater amount of product water. However, when establishing the capacity ratio of the absorption cooling subsystem with

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Table 1
Thermodynamic properties of the currents of the proposed system, for a design condition.

Streams	Fluid	Pressure (kPa)	Temperature (°C)	Enthalpy (kJ/kg)	Mass flow (kg/s)	X %(ppm)
1	LiBr/H ₂ O	1.16	32.00	-10,056.7	0.0564	51
2	LiBr/H ₂ O	5.44	32.00	-10,056.7	0.0564	51
3	LiBr/H ₂ O	5.44	60.55	-9998.0	0.0564	51
4	LiBr/H ₂ O	5.44	80.80	-8705.0	0.0466	62
5	LiBr/H ₂ O	5.44	36.98	-8788.0	0.0466	62
6	LiBr/H ₂ O	1.16	36.98	-8788.0	0.0466	62
7	H ₂ O	5.44	80.8	-13,328.3	0.0098	0
8	H ₂ O	5.44	34.31	-15,836.9	0.8515	0
9	H ₂ O	0.76	3.05	-15,836.9	0.8515	0
10	H ₂ O	0.76	3.05	-13,477.6	0.8515	0
11	H ₂ O	200	85.00	-15,624.7	1.6110	0
12	H ₂ O	200	80.99	-15,641.6	1.6110	0
12A	H ₂ O	200	38.32	-15,820	1.6110	0
13	Seawater	200	27.00	-15,530.7	14.01	(35,000)
13A	Seawater	200	27.00	-15,530.7	1.50	(35,000)
13B	Seawater	200	27.00	-15,530.7	12.59	(35,000)
13C	Seawater	200	31.18	-15,641.6	1.50	(35,000)
14	Seawater	200	27.44	-15,528.9	14.01	(35,000)
15	Seawater	200	32.32	-15,508.8	14.01	(35,000)
16	Seawater	200	32.32	-15,508.8	1.60	(35,000)
17	Seawater	200	75.96	-15,329.2	1.60	(35,000)
18	Seawater	200	75.96	-15,329.2	1.5116	(35,000)
19	Seawater	31.2	70.24	-15,339.2	1.5116	(35,000)
20	Seawater	200	75.96	-15,329.2	0.0883	(35,000)
21	Seawater	1.16	9.27	-15,329.1	0.0883	(35,000)
22	H ₂ O	1.16	9.27	-13,462.3	0.0098	0
23	H ₂ O	101.32	34.291	-15,836.9	127.0938	0
24	H ₂ O	120	34.291	-15,836.9	127.0938	0
24A	H ₂ O	120	30.5	-15,852.7	127.0938	0
25	H ₂ O	200	12.0	-15,930.0	95.75	0
26	H ₂ O	200	7.00	-15,951.0	95.75	0
27	Seawater	101.32	27.00	-15,530.8	114.02	(35,000)
28	Seawater	150	27.00	-15,530.8	114.02	(35,000)
29	Seawater	150	31.29	-15,513.1	114.02	(35,000)
BC1	H ₂ O	5.44 a 101.32	34.291	-15,836.9	127.0938	0
BC2	Brine	1.16 a 101.32	9.27	-15,561.8	0.0785	(39,361)
RET	Seawater	200	32.32	-15,508.9	12.49	(35,000)
RET1	Brine	101.32	31.41	-15,487.3	13.24	(35,233)
B1	Brine	5.44	34.83	-15,076.6	0.6698	(78,979)
B2	Brine	101.32	34.85	-15,076.2	0.6698	(78,979)
PW	H ₂ O	101.32	34.29	-15,836.9	0.8515	0

respect to the capacity of the MED, the initial investment increase must be taken into account. As a result of this study, it was found that, if you wanted to reduce the initial investment, a MED that uses the product water as a coolant can meet cooling and desalination needs without requiring absorption cooling subsystem.

Even though the heat transfer within the **HX2** augments, the increase in the cooling capacity, combined with the fact that the heat within the generator remains constant, causes an increase in the COP values up to 8 times more than those of conventional simple effect absorption systems (COP = 0.72) [29–31]. It should be stressed that, with seawater flows (**16**) below 0.15 kg/s (see Fig. 3) there is no water flow available for the MED. Thus, the system cannot operate under these conditions because there is neither produced water, nor cooling. The current divider (**DIV2**) allows us to determine the adequate flow design for the absorber, as well as the necessary seawater for the condensation of all the vapor of the last effect of the MED. This maximizes the system's desalination and cooling capacities, driving to a maximum the heat transferred within the **HX2**.

Both, PR and RR indicators, as well as the attainment of produced water (**m₁₀**), were assessed under different conditions for current (**16**) (see Fig. 4). An increment in the amount of seawater entering the MED (**m₁₉**) increases the vapor produced in the first effect, which causes a cascade effect for the next effects; as previously explained, the mass integration—carrying the vapor of current (**7**) into the MED—increases the produced water.

Even though, in the calculation of **RR** the water produced is divided between the amount of seawater **19** and vapor (**7**), it can be observed that the mass integration has a positive effect on this indicator when seawater flows **19** are low. This is due to the fact that with low water production, the mass integration mentioned above, represents a large part of the water produced by the MED. However, as the flow of the stream **19** increases, the steam flow **7** present a low value respect to production of water.

When the seawater flow entering the MED increases, **Q_{MED}** increases, while the heat added to the **GEN** remains constant. However, it can be seen that the increase in the produced water that results from the variation of **m₁₆**, as well as the mass integration of vapor from current (**7**), have a larger effect than the increase of **Q_{MED}**. This, in turn, positively affect the PR indicator.

Fig. 5 shows the behavior, under different current (**16**) flow conditions, of the brine concentration within the MED (**B1**), the FLASH unit (**BC2**), and the current that returns to the sea (**RET1**). **B1** decreases as flow **m₁₆** increases, since low seawater input flows cause a high percentage of vapor generation; conversely, as the flow increases, the vapor generation significantly decreases. **BC2** remains constant since the flow and thermodynamic conditions of current (**20**), which enters the **SEP**, do not vary, and the withdrawn vapor flow is constant. On the other hand, **RET1** increases as flow **m₁₆** augments since, as a larger flow is diverted within the **V3 V1** towards current (**16**), the amount of seawater that goes to the **RET** current decreases, reducing the amount

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Table 2

Energy evaluation of the proposed system.

Components	Energy balance	Inlet energy (kW)	Outlet energy (kW)	Energy integration (kW)
Generator	$Q_{GEN} = m_7h_7 + m_4h_4 - m_3h_3$	27.11		
MED condenser	$Q_{MC} = m_{15}h_{15} - m_{14}h_{14}$			282.52
Absorber	$Q_{ABS} = m_{22}h_{22} + m_6h_6 - m_1h_1$			25.77
Evaporator	$Q_{EVA} = m_{10}h_{10} - m_9h_9$	2012.00		
HX2	$Q_{HX2} = m_{12}h_{12} - m_{12A}h_{12A}$	287.44		
QMED	$Q_{MED} = (m_{19}/m_{16})Q_{HX2}$			
QFLASH	$Q_{FLASH} = (m_{20}/m_{16})Q_{HX2}$			
HX1	$Q_{HX1} = m_3h_3 - m_2h_2$			3.87
HX3	$Q_{HX3} = m_{28}h_{28} - m_{29}h_{29}$			2014
SP	$W_{SP} = m_1v_1(P_2 - P_1)/Eff_{SP}$	0.000323		
SWP	$W_{SWP} = m_{13}v_{13}(P_{13} - P_{13E})/Eff_{SWP}$	1.47		
EP	$W_{EP} = m_{23}v_{23}(P_{24} - P_{23})/Eff_{EP}$	2.38		
BP	$W_{BP} = m_{B1}v_{B1}(P_{B2} - P_{B1})/Eff_{BP}$	0.073		
SWP1	$W_{SWP1} = m_{28}v_{28}(P_{29} - P_{28})/Eff_{SWP1}$	5.88		
Flow streams				
$m_{13}h_{13}$	Seawater inlet	-218,966.05		
$m_{27}h_{27}$	Cooling seawater inlet	-1,772,717.63		
$m_{ret1}h_{ret1}$	Brine to return to the sea		-1,975,862.00	
$m_w h_w$	Product water outlet		-13,485.85	
Total		-1,989,347.41	-1,989,348.00	2326.16
BC1	$H_1 = (P_{atm} - P_{at1})/(\rho g)$	9.8 m		Barometric column 1
BC2	$H_2 = (P_{atm} - P_{at2})/(\rho g)$	10.2 m		Barometric column 2
COP	$COP = Q_{EVA}/(Q_{GEN} + Q_{HX2} + W_{EP} + W_{SP} + W_{SWP} + W_{SWP1})$	6.15		Coefficient of operation
Product Water		73,569 L/day		Daily production
RR	$RR = m_6/(m_{19} + m_7)$	0.56		Recovery ratio
PR	$PR = (m_8 * 2,326 \text{ kJ/kg}) / (Q_{MED} + m_7 * 2,326 \text{ kJ/kg})$	6.63		Performance ratio
COP _G	$COP_G = (Q_{EVA} + m_8 * 2,326 \text{ kJ/kg}) / (Q_{GEN} + Q_{HX2} + W_{EP} + W_{SP} + W_{SWP} + W_{SWP1})$	12.21		Global COP

of water available for dilution.

Regarding brine concentration, it should be considered that: 1) mineral salt precipitations must be avoided; and, 2) the environmental impact caused by the return to the sea of a very concentrated fluid should be reduced to a minimum. The concentration level of the brine in the last effect of the MED (B1) is greater than 100,000 ppm. Even though the concentration of B1 is high, according to Bamufleh et al. [29], where the authors state that the brine concentration should not exceed 70‰, there is still an ample operation range that does not present salt precipitation. Fig. 5 shows how B1 and BC2 are diluted with the RET current, and that they exit through RET1, entering the sea with a minimal environmental impact.

Both the activation temperature T_{11} of the absorption cooling subsystem, and the flow of seawater (16) were varied in order to drive B1 to typical MED concentration levels. The impact of this variation on water production, cooling capacity, COP, RR, and PR was also assessed. Fig. 6 shows the concentration of current B1 and the production of water for a variation on T_{11} ranging between 80 and 95 °C with different seawater flows (16). It can be seen that the decrease in the activation temperature causes a reduction in the concentration level of the brine leaving the MED, as well as a reduction of water production. This is due to the fact that, having a 62% concentration level of LiBr and a constant vapor production (m_7) of 0.0098 kg/s permits to have a constant heat transfer within the GEN for every activation temperature. A decrease in the activation temperature implies a lower T_{12} ; and having a 5 °C T_{approx} within the HX2 reduces the temperature of current (17). Hence, the DIV sends a greater flow towards current (20) in order to keep flows (22) y (7) equal. The flow increase in current (20) conveys a reduction of the seawater flow entering the MED (19), which reduces the water production. As previously stated, the lower the seawater flow entering the MED, the greater the concentration of brine B1.

It should be noted that, with a seawater flow (16) of 1.6 kg/s, we reach the thermodynamic limit of the heat exchanger HX2, with a hot water activation flow of 1.611 kg/s. Also, activation temperatures in the range of 80–85 °C produce brine concentration B1 values between 70,000–90,000 ppm, reducing the risk of salt precipitations within the equipment and reducing environmental impact.

Fig. 7 shows the behavior of the different indicators of the proposed system; the operational behavior for different activation and seawater (°C) temperatures can be observed. Fig. 7a shows that greater activation temperatures increase cooling capacity and efficiency. Fig. 7b shows that desalination indicators PR and RR also increase with higher activation temperatures. Considering a system operation with a brine concentration B1 lesser than 90,000 ppm (see Fig. 6) and with maximum capacity and efficiency, the design activation temperature is set at 85 °C (see Fig. 7a).

After the operative analysis conducted to prove the technical feasibility of the proposed system, Table 1 shows a summary of the properties of all currents within the system with the following settings: seawater temperature of 27 °C; activation fluid temperature of 85 °C; and a seawater flow (16) of 1.6 kg/s.

Table 2 presents the equations of the energy balances and the heat transferred in each of the system components (open cycle). The flow enthalpy of the input and output currents to corroborate the global energy balance is also shown, as well as the indicators for the absorption cooling and desalination system, and the amount of water produced. The negative values of the enthalpies of flow of the input and output currents of the open cycle are due to the fact that the Aspen Plus simulator uses a reference temperature of 47,101.0 °C in the calculation of enthalpies, which does not affect the calculations, since all enthalpies have the same state of reference.

Table 2 shows the indicators for the absorption cooling and desalination system, and the amount of water produced, as well as the heat transfer for each of the components. It can be seen that the COP is 8.54 times greater than that of conventional simple-effect absorption systems (COP = 0.72 [29–31]). Furthermore, using a 23 kW thermal compressor, the cooling capacity increased 87.4 times compared to that of a conventional absorption cooling system with the same compressor. The system can produce 73,569 L/day of water, with an RR of 0.56 and a PR of 6.63; enough to satisfy the needs of a 735-person housing complex, according to the World Health Organization (WHO) [32], or the needs of the 126 visitors of a coastal Hotel in Mexico, according to Becken [33]. Brine concentration B1 is 78,979 ppm, an adequate level to avoid salt precipitation within the equipment. The concentration level of the

ANEXO I. ABSORPTION COOLING AND DESALINATION SYSTEM WITH A NOVEL INTERNAL ENERGETIC AND MASS INTEGRATION THAT INCREASES CAPACITY AND EFFICIENCY

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brine that returns to the sea (RET1) is 35,233 ppm, which minimizes its environmental impact.

6. Conclusions

This paper presents an absorption cooling and desalination system with a novel energetic and internal mass integration that significantly increases capacity and efficiency. Based on a 23 kW simple-effect absorption cooling machine, the technological proposal of using desalination produced water as coolant increases the cooling capacity to 2012 KW, 87.4 times greater, with a COP value of 6.15, 8.54 times greater than the value reported for this type of systems. Furthermore, the water production is 73,569 L/day, with an RR of 0.56 and a PR of 6.63.

The disturbances that result from the cooling demand are controlled with the modulation of the coolant flow using a three-way valve, instead of controlling the cooling capacity through the manipulation of the flow and temperature of the activation fluid. In order to produce 73,569 L/day of water, a constant heat supply for the generator was assumed, since the variability of the solar resource can be hedged and controlled for with an auxiliary heating system.

Using the last effect of the desalination subprocess with the pressure conditions of the GEN allows for the incorporation of the vapor produced in the last effect of the MED, the removal of the condenser from the absorption cooling subprocess, and the boosting of desalination and cooling capacity. With this design's cooling capacity, the water production satisfies the hydric needs of a 735-person house complex, or a 126-person coastal hotel in Mexico.

At the time of the writing of this paper, a prototype of this system is being built. In future work the validation of the prediction will be conducted through a simulation with experimental data obtained from the system. Furthermore, a study on the water quality obtained by the system will be conducted.

Declaration of competing interest

None.

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**J. A numerical study of the influence of design
variable interactions on the performance of a
Stirling engine System**

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A numerical study of the influence of design variable interactions on the performance of a Stirling engine System



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HIGHLIGHTS

- A new method to determine the degree of interaction of system variables is proposed.
- 17 critical interactions were found for the optimized design of a Stirling engine.
- The cylinder and cooler variables present the most significant interactions.
- Piston stroke affects the performance of 60% of the variables studied.
- Phase angle $< 90^\circ$ shows system improvement when the heat input is insufficient.

ARTICLE INFO

Keywords:
Stirling engine
Design variable interaction
Second order model
Optimization

ABSTRACT

This paper presents a study to determine the influence of design variable interactions on the performance of a Stirling engine, as well as a new methodology to identify and quantify such influence. An Alpha-type 2 kW Stirling engine was assessed using a second order model where the engine, heater, regenerator and cooler variables were simultaneously analyzed in pairs.

The study includes 240 interactions corresponding to the 16 selected design variables. It was observed that the piston stroke and cylinder diameter variables have more significant interactions with the rest of the parameters due to their strong influence over mass flow. The piston crown length does not show significant interactions; however, it reduces individual thermal losses in the expansion zone. The cycle achieves its maximum net power at angles less than 90° due to pressure, heat losses, and the ratio between mass flow rates and swept volume. Finally, this work presents recommendations for the analyzed interactions.

The global interactions scheme provides a qualitative and quantitative identification of the variables that affect the system, synthesized into a hierarchy of variables that indicates a critical path for their exploration for system design and optimization purposes.

1. Introduction

The Stirling engine was developed in 1816 by Robert Stirling as an alternative to vapor machines. In principle, it works by the expansion and compression of a working fluid (helium, hydrogen, nitrogen, argon and air) [1–3] without phase change. Due to the great flexibility in its activation, the external combustion engine is an excellent option for electric energy generation through non-conventional and environmentally friendly means such as biomass, biofuels, geothermal, and solar energy [4–6]. However, its development, implementation and commercialization have been hindered by high manufacturing cost,

weight, and the difficulty of sealing the work chamber [7,8].

In 1930's, given its energetic flexibility and low emissions, Philips Research Laboratories began to work on the Stirling engine [9]. Stirling engines have been used in different aerospace applications [10,11]. However, a large amount of this research is focused on performance, prediction and optimization. Most analyses in steady or dynamic state draw from mathematical models derived from the ideal gas equation, differentiated by system considerations such as dead volume, compressibility factor, adiabatic sections, and the use of different design variables [12–15].

In order to study the operative behavior and the sizing of a Stirling

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ANEXO J. A NUMERICAL STUDY OF THE INFLUENCE OF DESIGN VARIABLE INTERACTIONS ON THE PERFORMANCE OF A STIRLING ENGINE SYSTEM

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Nomenclature			
α	phase angle, degree	V_{CD}	cooler dead volume, cm ³
A_{HT}	transversal area, cm ²	V_{CS}	compression zone dead volume, cm ³
BHI	ideal heat input, W	V_{EX}	expansion zone dead volume, cm ³
BP	basic power, W	V_{HD}	heater dead volume, cm ³
C_v	heat capacity, J/kgK	V_R	regenerator dead volume, cm ³
D	diameter, cm	V_{ratio}	volume ratio, cm ³
f	regenerator friction factor	V_{SW}	swept volume, cm ³
F	fanning friction factor	V_{rpm}	engine speed, rpm
FCT	time fraction	W	mass flow, kg/s
F_r	friction factor	WP	losses by pressure drops, W
G	mass flux, g/cm ²	ΔT_{MX}	maximum temperature difference, K
KMX	regenerator matrix conduction constant, W/mK		
L	length, cm	Greek	
MF_L	mechanic friction losses, W	α	alpha configuration
MSH	mesh, threads per centimeter	β	beta configuration
N	Number	γ	gamma configuration
NP	net power, W	φ	interaction grade of the variable pair
NTU	number of transfer units	Φ	indicator of overall system interaction
Nu	Nusselt number	η_k	thermal efficiency response surface
Pr	Prandtl number	ρ	density, kg/cm ³
P_S	piston stroke, cm		
Q	heat, W	Subscripts	
$Q_{MX,k}$	regenerator matrix heat losses, W	C	cooler
Q_S	static heat losses, W	CYL	cylinder
Q_{TS}	regenerator overheating losses, W	H	heater
Re	Reynolds number	k	conduction
Rt	thermal resistance, K/W	PC	piston crown
T_C	compression temperature, K	R	regenerator
T_{Crank}	crankshaft temperature, K	T	tubes
T_H	expansion temperature, K		
THW	wire thickness, cm		

motor, different mathematical models with different complexity and real process representativeness levels have been proposed. These models are differentiated and classified by the considerations and number of variables involved which determine the computational requirements for their solution and are classified as zero, first, second, or third-order models. The zero-order model was proposed by West [16]; it determines engine power as a function of 4 parameters: mean pressure, swept volume, engine frequency, and the Beale number. This model is presented in actual works such as a gamma-configuration low temperature differential (LTD) Stirling engine research by B. Kongtragool and S. Wongwises [17]. The first-order model is based on a thermodynamic analysis using equations of state, energy and/or mass balances; such models are also called first-order simulators [18]. The second-order models are first-law thermodynamic analyses, and they consider energy loss in the system due to heat and pressure drops in exchangers, cylinder conduction, and mechanical friction, depending on the system configuration [12,19]. The third-order model draws from the second-order model and adds computational fluid dynamics (CFD) analysis; thus, the computational requirements are very high [20].

Table 1 shows the studies and tools that have been used to determine the thermodynamic behavior of the Stirling engine, and how their design variables can be optimized.

It can be observed that the study of the Stirling engine variables has followed the conventional individual exploration method (the remainder of the variables remain fixed), or that methods with complex algorithms (artificial intelligence) have been used when compared with previous works. However, studies on the degree of influence and interaction between the variables, and a hierarchy between them have not been conducted. Hence, this paper proposes a new methodology to determine qualitatively and quantitatively the degree of influence of

the different variables, as well as their interaction between each other, achieving, thus a categorization. Furthermore, this work sets a critical path for the optimized design of a Stirling motor.

2. Description of the system

The system in this study (Fig. 1) is an Alpha-type Stirling engine comprising of a rod-crank mechanism, a heater, a regenerator, and a cooler. The cylinders (D_{HCYL}) are distanced by the phase angle (α). The displacement piston is called stroke (P_S), which, along with the cylinder diameter, defines the swept volume (V_{SW}). A crown with length L_{PC} is attached to the piston; it resists high temperatures and reduces thermal losses. The union between the cylinder and the heater adds dead volume to the expansion zone (V_{EX}). The heat transfer area and the dead volume added by the heater are defined by the number of parallel tubes (N_{HT}), length (L_{HT}) and the internal diameter of the tubes (D_{HT}). The regenerator parameters are the diameter D_R and the length L_R . Its matrix is composed of a pile of metallic mesh defined by the number of wires per centimeter (MSH) and their thickness (THW). The cooler is shell and tube type. The working fluid resides inside the cooler tubes and the cooling fluid is outside the cooler tubes. The number of tubes (N_{CT}), their internal diameter (D_{CT}), and their length (L_{CT}) define the heat transfer area and the dead volume added to the system. Finally, the compression cylinder and the cooler are attached through a connection element, which adds dead volume to this zone (V_{CS}). Table 2 presents the initial values of the design variables to be studied, which is set based on a previous design of an Alpha-type Stirling motor prototype [27]. The exploration process and the study of the interactions initiate with this, ensuring that the results obtained from the numerical simulation are consistent values with physical significance.

ANEXO J. A NUMERICAL STUDY OF THE INFLUENCE OF DESIGN VARIABLE INTERACTIONS ON THE PERFORMANCE OF A STIRLING ENGINE SYSTEM

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Table 1
Recent models for Stirling engine prediction and optimization.

Author(s)	Enginetype	Model	Order	Number of variables	Approach	Parametric study type
W. Beale, 1977 [21]	–	–	0	4	Prediction	Heuristic model to obtain net power
I. Urieli, 1977 [22]	–	Steady State	1	+15	Prediction	A thesis work to obtain the net power and behavior of a Stirling engine using Schmidt thermodynamic analysis
B. Kongtragool, 2006 [15]	–	Steady State	1	+25	Prediction	A theoretical Stirling cycle thermodynamics analysis with dead volume influence to explain how variation of dead volume affects net power
L. Scollo, P. Valdez, J. Barón, 2008 [23]	α	Steady State	1	+15	Experimental	Design and construction of a Stirling engine with a compressor block exploring different compression temperatures within a cooling process
C. Roldan, P. Pieretti, & L. Solorzano, 2010 [24]	γ	Steady State	1	+15	Prediction	A conceptual and basic design of a Stirling engine prototype using an ideal Stirling cycle
L. Yaqi, H. Yaling, & W. Weiwei, 2011 [25]	–	FTT	1	+30	Optimization	A high temperature differential solar powered Stirling engine finite-time thermodynamic (FTT) analysis
I. Tlili & S. Musmar, 2013 [19]	γ	Steady State	2	+40	Prediction	A thermodynamic evaluation of a solar Stirling engine with Ross-Yoke mechanism
M. Ahmadi, et al., 2013 [26]	α	Steady State	2	+50	Prediction	A solar Stirling system design, simulation and thermal analysis embedded on a Solar Dish Design program applied to SOLO 161 V
S. Costa et al., 2014 [27]	γ	Steady State	2	+30	Optimization	An experimental and theoretical study to improve the regenerator using a CFD-based finite volume method
W. Chen, Y. Yang, & J. I. Salazar, 2015 [28]	γ	CFD	3	+50	Prediction	A CFD parametric study on the performance of a low-temperature-differential gamma type Stirling engine
M. Babaelahi & H. Sayyaadi, 2015 [29]	β	Steady State	2	+60	Optimization	A polytropic analysis of a Stirling engine with various losses applied to a GPU-3 Stirling engine
R. Gheith et al., 2015 [30]	γ	Steady State	1	+15	Optimization	An experimental and theoretical study to improve the heater of a high temperature differential Stirling engine
M. Hooshang et al., 2015 [31]	α	Neural networks	–	+35	Optimization	Improvement procedure for a Stirling engine based on a multi-layer perceptron applied on an ST500 Stirling engine
Z. Luo et al., 2016 [32]	β	Genetic algorithm	–	+25	Optimization	A Multi-objective study for a GPU3 Stirling engine with TOPSIS and SAW methods
W. Jan & P. Marek, 2016 [18]	–	Steady State	2	+20	Prediction	A second order model approach to a CFD analysis using a Ross-Yoke mechanism
R. Li, L. Grosu & W. Li, 2017 [33]	β, γ	Steady State	2	+20	Prediction	A polytropic model to predict the performance of a Stirling engine exploring the operation variables
A. Almajri et al., 2017 [20]	α	CFD	3	+50	Optimization	Modelling and parametric performance study based on a 3D CFD analysis
G. Xiao et al., 2017 [34]	β	Steady State	2	+30	Optimization	An approach to combine a second and a third order model to improve the design parameter of a GPU-3 prototype
G. Carrillo Caballero et al., 2017 [35]	–	Steady State	2	+30	Optimization	An optimization of a Dish Stirling system using multi-objective techniques to improve receiver parameters
D. Shendage, S. Kodare & S. Bapat, 2011 [36]	β	Steady State	2	+30	Prediction + experimental	A theoretical and experimental study of a rhombic mechanics engine with hydrogen as working fluid; the crank angle was explored for optimization
D. Shendage, S. Kodare & S. Bapat, 2017 [37]	β	Steady State	2	+30	Prediction	The piston stroke, diameter of cylinder, regenerator diameter and length, heater and cooler dead volume were explored on individual ways for optimization

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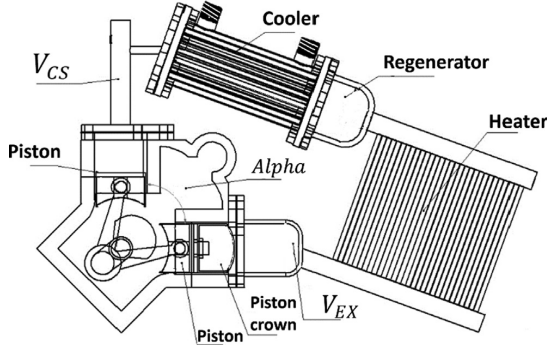


Fig. 1. Alpha-type Stirling engine prototype.

Table 2
Initial variables values on the design point.

Variable	Design point	Units
Piston stroke (P_S)	6.2	cm
Diameter of cylinder (D_{HCVL})	6.6	cm
Phase angle (α)	90	degrees
Length of piston crown (L_{PC})	4.359	cm
Length of heater tubes (L_{HT})	16.612	cm
Diameter of heater tubes (D_{HT})	0.3048	cm
Number of heater tubes (N_{HT})	30	–
Expansion side dead volume (V_{HS})	12.389	cm ³
Regenerator length (L_R)	5	cm
Regenerator diameter (D_R)	6.5	cm
Mesh (MSH)	60	wires/cm
Wire thickness (THW)	0.0041	cm
Length of cooler tubes (L_{CT})	21	cm
Diameter of cooler tubes (D_{CT})	0.2134	cm
Number of cooler tubes (N_{CT})	85	–
Compression side dead volume (V_{CS})	5.785	cm ³

3. Thermal model

The performance of the Stirling engine shown in Fig. 1 is calculated using a second-order numerical model, following the flow diagram shown in Fig. 2. The net power and efficiency are calculated using the method described by Martini [9] and a numerical model. The losses due to mechanical friction, working fluid pressure drops, as well as the main engine thermal losses within the high temperature areas are accounted as shown in Fig. 3. Previous works have either used the Stirling engine mathematical model of Martini or at least, a part of it for their results [19,25,29,30].

The thermodynamic analysis of the engine is conducted considering the volumes of the following components: expansion cylinder (V_{HCVL}), expansion cylinder-heater joint (V_{EX}), heater (V_{HD}), regenerator (V_R), compression cylinder-cooler joint (V_{CS}), cooler (V_{CD}), and compression cylinder (V_{CCYL}). The expansion (T_H), regeneration (T_R), and compression (T_C) temperatures remain constant. However, the pressure is the same in all components and it varies sinusoidally due the crank angle (θ).

Other considerations in the model are: an isothermal analysis is performed for each of the 5 volumes, the variation of the gas volume is sinusoidal; ideal gas law as the equation of state, isothermal expansion and compression; the working fluid has no mass loss; heat transfer and fluid-dynamic losses in exchanger connections are null; the effects of gas inertia are null and the temperature of the walls of the heater and cooler tubes are constant. The cylinder cavity, piston and piston crown have the same profile. The piston crown is a perfect cylinder.

Furthermore, the analyzed energy losses are: thermal conduction in the crown of the piston and cylinder wall, thermal conduction in the wall and matrix of the regenerator towards the cooler, inefficiency and

temperature oscillation of the regenerator matrix, radiation emission from the face of the piston to the air in the crankcase, mechanical motor friction and pressure drop of the working fluid in the heater, regenerative cooler. The equations are presented in an SI file.

This numerical model was validated by Chen and Griffin [38], who reported that by correcting the gas friction with a 2.9 factor, the error in the calculation of power and efficiency would be reduced by $\pm 10\%$. The numeric model was also validated with the experimental results of the GPU-3 engine thereby obtaining a 5.7% and 8.2% average error for the heat input and net power respectively [39].

4. Method for the definition of degree of interaction and its hierarchy

In order to classify and quantify the importance of each interaction, a second-order numerical simulator was used to carry out explorations by pairs of variables with system efficiency as the response parameter. That is, if two out of the 16 system variables analyzed are considered independent, while the rest are fixed variables, then the simulator calculation may be expressed as a two-variable function:

$$f(i, j) = \eta_s \quad (1)$$

where i and j are the vectors of the two selected variables, starting and ending in the lower and upper limits of the normalized exploration ranges, obtaining η_s as a response surface (Fig. 4), showing the efficiency of the system points, and denoting the maximum efficiency zones.

In order to determine the maximum efficiency points in the response surface (η_s), the optimal relationship between the interacting variables must be determined as shown in Fig. 4a and b with orange dashed line.

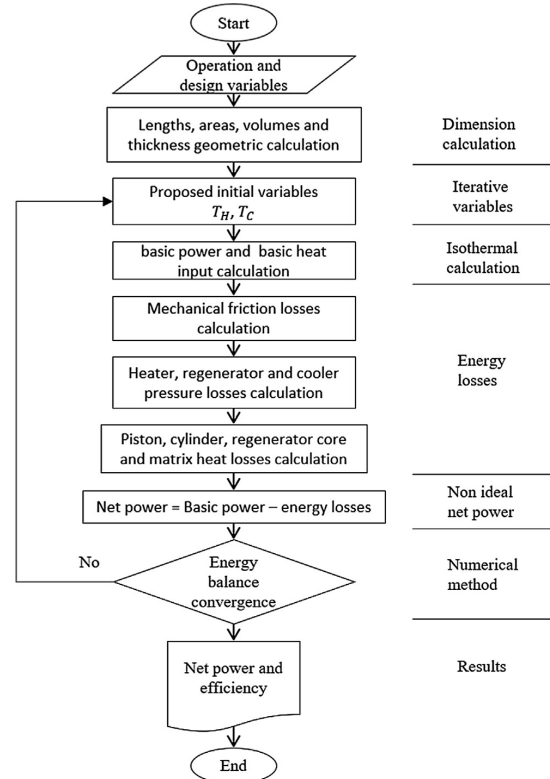


Fig. 2. Flow diagram of the second-order numerical simulation in MATLAB.

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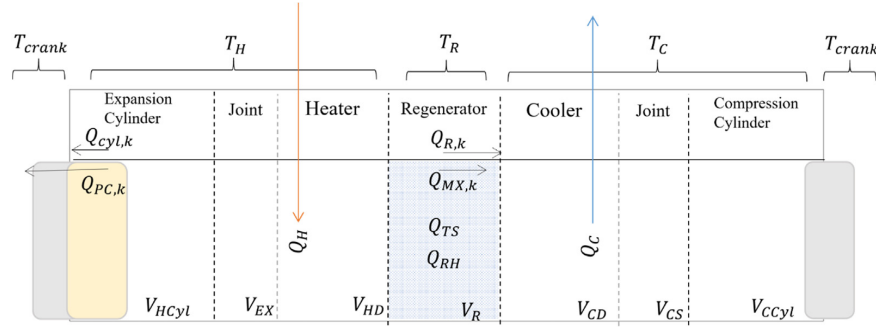


Fig. 3. Component temperatures, volumes and conduction heat losses.

This relationship is linearized using the least-squares method that results in the following equation:

$$j_{\eta} = i_{\eta} * \varphi + b_0 \quad (2)$$

where j_{η} is the dimensionless value that the second selected variable must take in order to guarantee the maximum efficiency as a function of the first selected variable i_{η} . Therefore, the slope φ is an indicator that allows the identification of the degree of interaction between the selected variables.

Using the absolute value of the slope $|\varphi|$ (equation 23) obtained from the interaction of variables 1 and 2 (Fig. 4b), it was proposed to categorize the interactions as shown in Fig. 5. The four categories proposed are the following: Strong (A), Significant (B), Weak (C), and Null (D). If the slope (φ) is in a null zone (D), it means that the values of only one variable are predominant over the maximum efficiency; therefore, there is no interaction. If the slope (φ) is in a Strong (A) zone, it means that there are specific values between both variables that allow achieving maximum efficiency; therefore, the interaction is strong.

Once classified, the variables were given integer values according to $|\varphi|$, developing a matrix (with $\varphi_{x,y}$ values), named Global Scheme of Interactions. On the other hand, all φ resulting from the interactions of

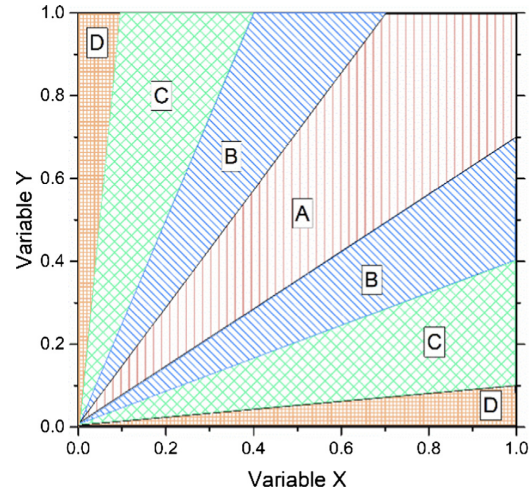


Fig. 5. Interaction categories zone.

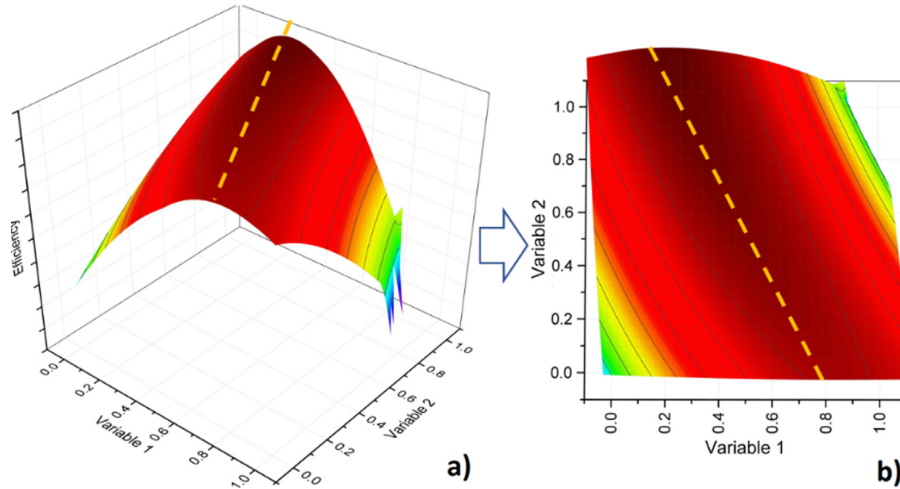


Fig. 4. Response surface η of two random variables: (a) Isometric view, (b) Top view.

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the x variable with the 15 remaining y variables of the system are added:

$$\Phi_x = \sum_{y=1}^{y=15} \varphi(x, y) \quad (3)$$

where Φ is the indicator of overall system interaction. It denotes the influence of the interaction of one variable with the rest of the variables; i.e., how such variable affects other variables or equipment. Table 3 shows the predetermined values and exploration ranges of each of the studied variables.

5. Analysis and discussion of results

A simultaneous exploration was carried out by pairs of variables in a second-order numerical simulation using the main design variables of an Alpha-type Stirling Engine in order to obtain the response surfaces of the system efficiency. Such surfaces were studied and analyzed based on the proposed methodology which determines if the variables under consideration have a joint effect on the efficiency and, in turn, on the extent of such effect.

The scheme in Fig. 6 shows a summary of the 240 response surfaces of the 16 selected design variables which are based on the interaction categories and are represented abstractly. The variables are grouped by the equipment: engine, heater, regenerator and cooler. The arrows below show a qualitative, top-bottom variable hierarchy with the piston stroke (P_S) as the dot marked variable with the most joint effects or significant interactions with the rest of the system variables. However, the dashed marked box shows how the piston crown length (L_{PC}) does not have a joint effect with the rest of the system variables.

Fig. 7 shows the variable hierarchy developed by the Φ indicator of each variable; where Φ is determined by the amount and extent of the joint effects of a variable with the rest of the system variables. It was noted that piston stroke (P_S) is the variable with the highest influence on the design of the system since it interacts with 11 variables and out of which 7 are classified as strong. This is due to the fact that P_S directly affects the swept volume, mass flow and maximum basic power. Therefore, modifying this variable affects the performance of the rest of the equipment, and a value that optimizes its performance must be determined. In contrast, variables M_{SH} , V_{HS} , L_{HT} , and L_{PC} indicated in Fig. 7, which belong to groups G5 and G6 can be manipulated without thinking of a joint effect with other variables since their influence over the system is virtually individual. In the search of a better system performance, the hierarchy found suggests the order in which variables must be explored. Variables from group G2 can exchange their hierarchical order among them, as well as those of groups G4 and G5 without affecting the exploration.

A conventional parametric study which shows the influence of individual variables over system efficiency (Fig. 8) was developed using the same second-order model with the conditions of Table 3. Such influence is determined from the percentage difference between maximum and minimum efficiency found in the exploration range of each variable. Piston crown length (L_{PC}) shows an effect on the system efficiency (dashed arrow) without a joint effect with other variables. This means that it does not have an effect on other components, but it must be considered in the design as it plays a significant role in heat loss reduction within the expansion zone. The regenerator diameter (D_R) impacts the operational behavior of the system significantly due to its direct influence on pressure losses which results in a system with poor efficiency; i.e., a very small diameter mass flow throttling, increasing pressure losses exponentially.

In contrast, a conventional parametric study shows either the operational behavior of the system trend, or whether there are inflection points in maximum or minimum operational parameters within a limited design group. This kind of exploration offers a partial maximum efficiency value as the rest of the variables remain fixed. It also allows

the sensitization of the operational functioning of the system to a certain extent. As a result, the simultaneous exploration of variable pairs (proposed methodology) generates additional information as the response surfaces, the global interactions scheme, and the variable hierarchy are carried out. In this methodology, the response surfaces show the system efficiency behavior as a function of two selected variables denoting the maximum efficiency zones with multiple viable design points, the optimal relationship between those variables, and the degree of influence of one of the variables over the other. Furthermore, with the analysis of these simultaneous explorations, conclusions are more representative of the behavior of the system. However, the global interaction scheme depicts abstract response surfaces and the degree of influence of variables indicating how one variable affects the other system components and which interactions should be analyzed in depth. A hierarchy of variables provides the order in which these must be fixed and/or explored in order to obtain a high performance design from an iterative process. Such interpretation allows for recommendations which may be used in the definition of objective functions in an optimization process or method.

Fig. 6 presents a detailed analysis of the most significant response surfaces of the global interactions scheme, as well as the causes and effects of system variables. The trend of the involved thermodynamic phenomena is discussed using two design points of the analyzed response surfaces. The most significant interactions of the equipment of the engine are shown below.

5.1. Significant interactions in the engine

Fig. 9 shows the response surface resulting from the simultaneous variation of variables D_{HCVL} and P_S ; it also shows that there is a relation between these variables to maintain maximum system efficiency. This relation is a function of a determined swept volume and mass flow. Two design points (A and B) were selected to show the causes that influence efficiency variation within the maximum efficiency zone; one on quadrant $D_{HCVL} < P_S$ and the other on quadrant $D_{HCVL} > P_S$. Fig. 10 shows that both design points have similar values in swept volume ($\sim 176 \text{ cm}^3$), mass flow ($\sim 109 \text{ g/s}$) and temperature ($\sim 950 \text{ K}$). However, the net power has a difference of 62 W.

Both points operate with similar swept volumes and with identical heat exchangers, thus, the mass flow and energy losses by pressure drops are similar in both cases (Fig. 11). Therefore, the power increase is due to a heat loss decrease in the expansion zone with a difference of 132 W between both design points. The behavior of the optimal relation of these variables presents a great similarity to the constant efficiency lines as they are determined by the same effects which are discussed

Table 3
Variable exploration ranges and design point.

Variable	Exploration range		Units
	Min.	Max.	
Piston stroke (P_S)	4.5	8	cm
Diameter of cylinder (D_{HCVL})	4.5	8	cm
Phase angle (α)	30	150	degrees
Length of piston crown (L_{PC})	3	7	cm
Length of heater tubes (L_{HT})	10	35	cm
Diameter of heater tubes (D_{HT})	0.14	0.45	cm
Number of heater tubes (N_{HT})	10	50	–
Expansion side dead volume (V_{HS})	10	120	cm^3
Regenerator length (L_R)	3	8	cm
Regenerator diameter (D_R)	3	8	cm
Mesh (MSH)	40	80	wires/cm
Wire thickness (TH_W)	0.0020	0.0040	cm
Length of cooler tubes (L_{CT})	10	50	cm
Diameter of cooler tubes (D_{CT})	0.14	0.45	cm
Number of cooler tubes (N_{CT})	45	120	–
Compression side dead volume (V_{CS})	10	120	cm^3

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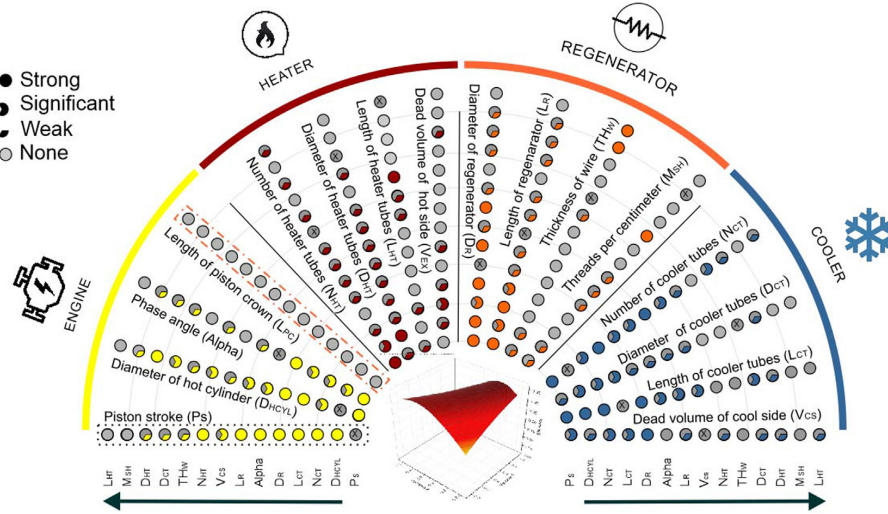


Fig. 6. Global scheme of the system variable interactions.

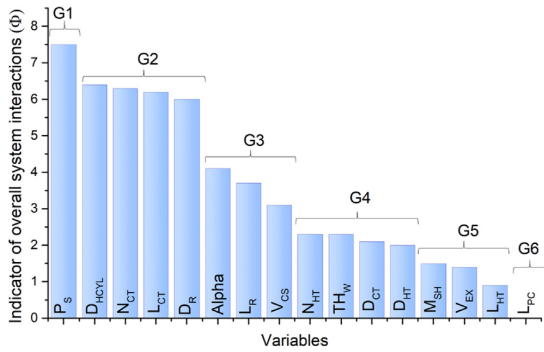


Fig. 7. Design variable interaction hierarchy.

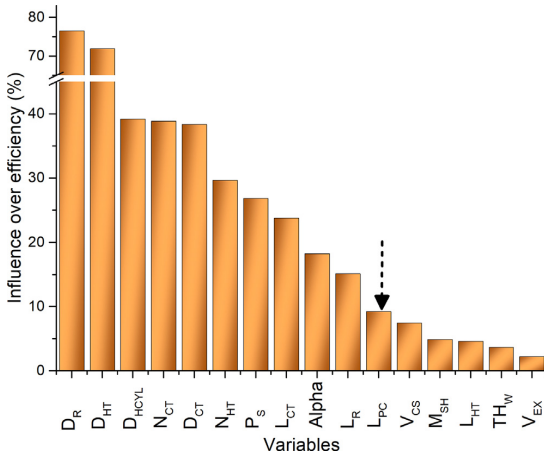


Fig. 8. Influence of individual variables over system efficiency.

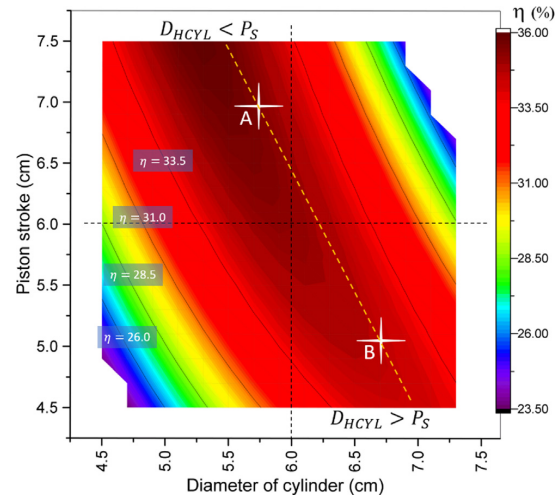


Fig. 9. Efficiency response surface of cylinder diameter vs piston stroke, where A and B are the selected design points to analyze on the line that indicates the optimal relation of the variables.

below. This decrease occurs partly due to cylinder diameter reduction, which decreases heat losses through the piston crown. However, when the length of the piston stroke increases, the length of the cylinder must also increase, thereby decreasing the heat losses through their transversal section towards the motor crankcase.

Variables P_s and $D_{H_{CYL}}$ have a high degree of interaction with the rest of the system and define its maximum basic power; therefore, it is suggested to fix these variables first for system optimization. For simulation purposes, it is recommended to use a $P_s > D_{H_{CYL}}$ ratio for heat loss reduction in the expansion zone (Q_{LOSS}).

When the phase angle ($Alpha$) tends to zero degrees, a high-pressure difference is obtained but the mass flow is minimal thereby preventing heat transfer. Conversely, $Alpha \approx 180^\circ$ allows for maximum mass flow, but the pressure difference is minimal which decreases work generation capacity. Therefore, the optimal compromise between the mass flow and the pressure differences is at $Alpha \approx 90^\circ$ for cases in which heat

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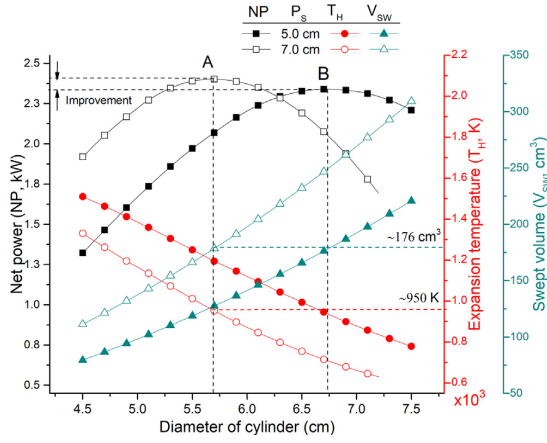


Fig. 10. Analysis of the influence of D_{HCYL} and P_3 over the net power, expansion temperature, and swept volume for design points A (7 cm) and B (5 cm).

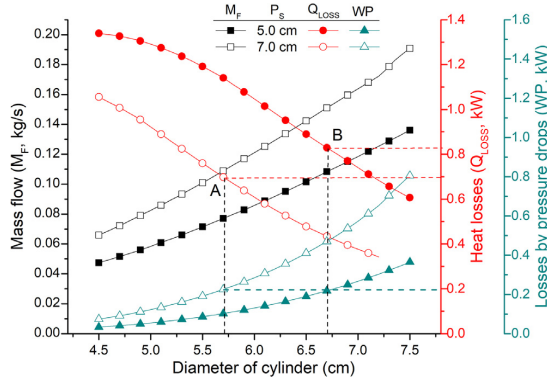


Fig. 11. Analysis of the influence of D_{HCYL} and P_3 over mass flow, thermal and pressure drop losses for design points A (7 cm) and B (5 cm); an energy trade-off is observed between thermal and pressure losses.

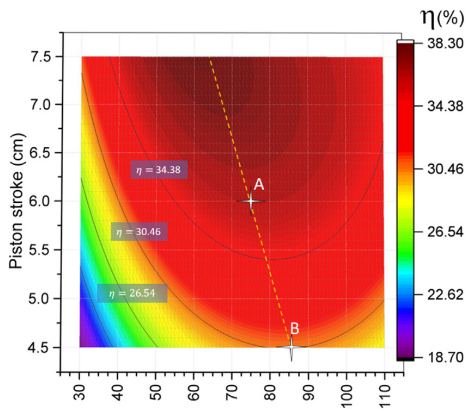


Fig. 12. Efficiency response surface of the phase angle with piston stroke, where A and B are the selected design points to analyze on the line that indicates the optimal relation of the variables.

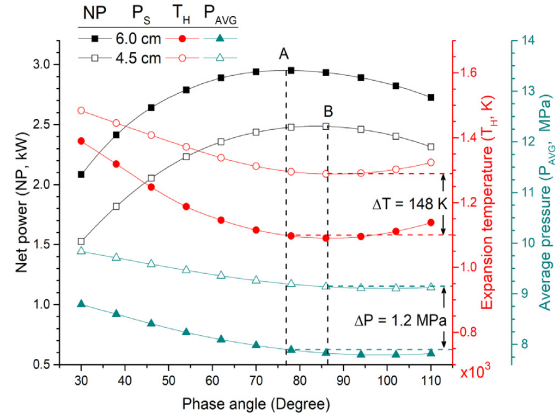


Fig. 13. Analysis of the influence of α and P_3 over net power, expansion temperature and average pressure for design points A (6 cm) and B (4.5 cm).

input is sufficient and can compensate among other parameters, the heat, pressure and mechanical friction losses.

Fig. 12 shows the response surface of the interaction of variables P_3 and α ; the orange line indicates the relationship (ϕ) between them to stay in the maximum efficiency zone. When the swept volume increases due to the increase of P_3 , the mass flow must be reduced in order to maintain a high expansion temperature. If the heat supplied is not enough for the swept volume, the phase angle must be reduced in order to increase the pressure difference of the cycle.

Fig. 13 shows how point B with a phase angle of 86° presents a net power of 2485 W; its temperature and average pressure are higher than point A by 13% and 15.5% respectively. However, point A has a greater power (2951 W) since this design has 211.5 cm^3 cylinders and point B has 158.6 cm^3 cylinders; hence, a higher volume is capable of generating more work if the heat supplied allows it as shown in Fig. 13.

When the supplied heat is insufficient, decreasing the phase angle allows engines with high swept volumes to operate efficiently whereby working at a higher temperature and average pressure (Fig. 14). In contrast, with an adequate heat supply, the phase angle tends to 90° , thereby operating the engine at a high temperature and an average pressure. P-V work increases power and efficiency. In addition to this, in solar applications, phase angle reduction may improve the system performance when the energy provided is insufficient. Moreover, it

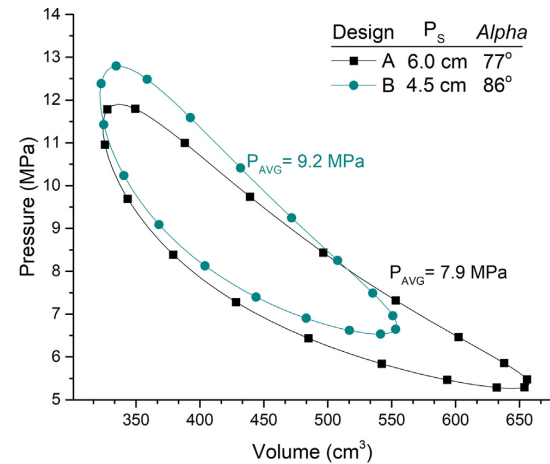


Fig. 14. P-V curve of design points A (6 cm) and B (4.5 cm).

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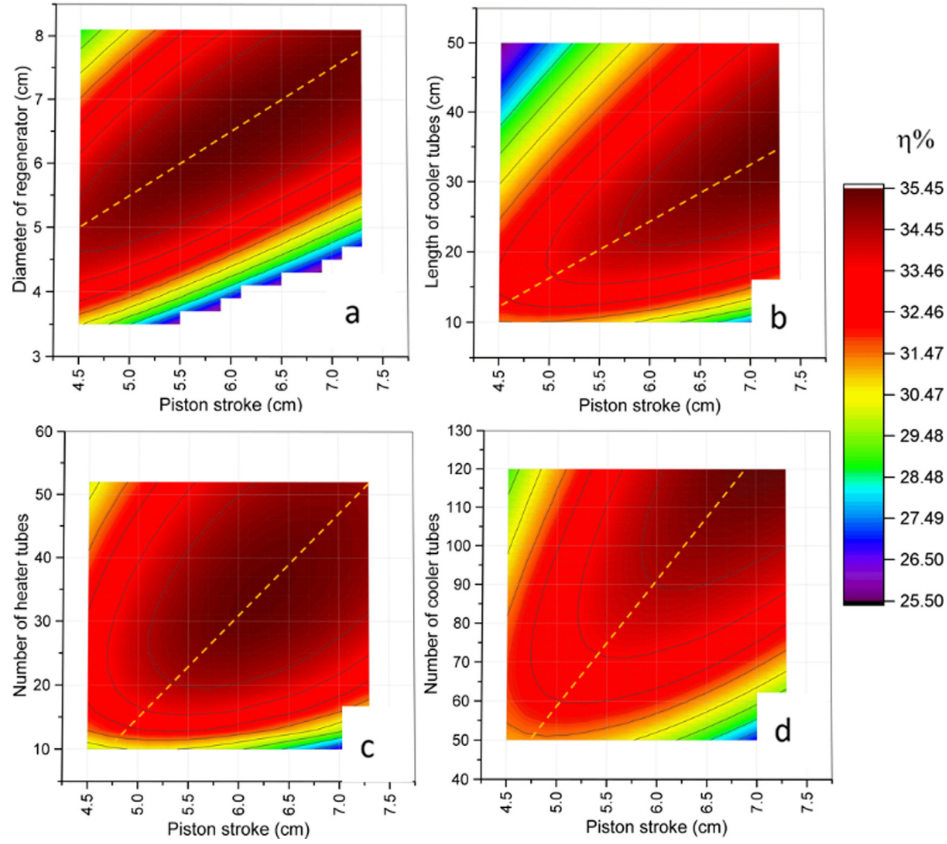


Fig. 15. Important interactions of the piston stroke with the system variables, where the orange lines indicate the optimal relation of the variables determined by the proposed method. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

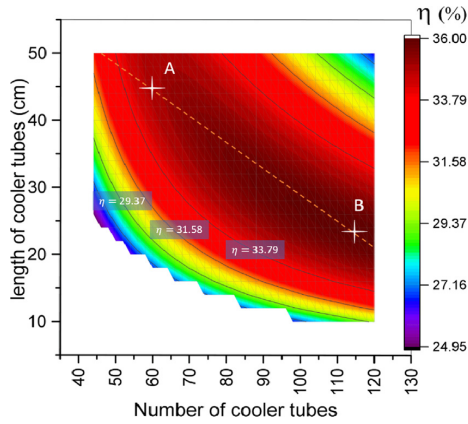


Fig. 16. Efficiency response surface of the number and length of cooler tubes, where A and B are the selected design points to analyze on the line that indicates the optimal relation of the variables.

allows the receiver to operate at a point of low thermal losses.

As seen in the global map of interactions (Fig. 6), there are multiple pairs with strong and significant effects in the engine equipment; these are summarized in Fig. 15. Fig. 15a shows that when the stroke is increased, it is necessary to increase the regenerator diameter to decrease

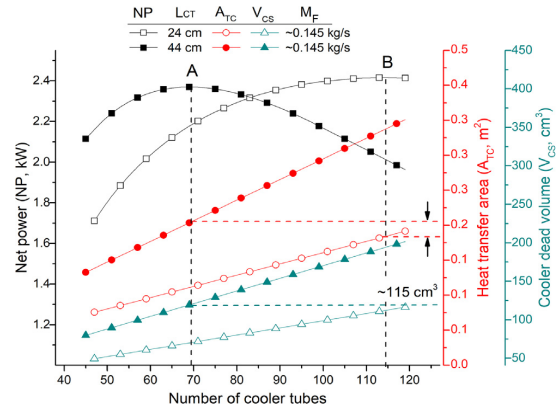


Fig. 17. Analysis of the influence of N_{TC} and L_{TC} over the net power, transfer area and dead volume for design points A (44 cm) and B (24 cm).

energy losses by pressure drops (WP). It can also be observed that increasing the stroke may require increasing the number or the length of the cooler tubes in order to decrease pressure and heat drop losses through conduction in the expansion zone (Fig. 15d and b). On the other hand, the increased P_3 may also require increasing the number of heater tubes (N_{TH}) to decrease WP and improve heat transfer in the

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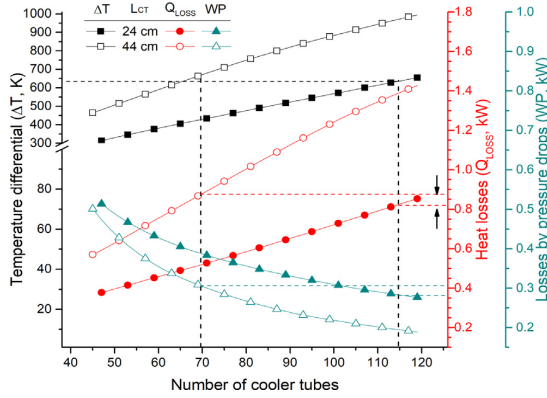


Fig. 18. Analysis of the interaction of N_{TC} and L_{TC} over the temperature difference, and thermal and pressure drop losses for design points A (44 cm) and B (24 cm). An energy trade-off is observed between thermal and pressure losses.

expansion zone (Fig. 15c). Other phenomena can be discussed if the interactions mentioned are analyzed in-depth as shown in Figs. 9–14.

5.2. Significant interaction in the shell and tube cooler

The diameter (D_{CT}), number (N_{CT}) and length (L_{CT}) of the tubes define the dead volume of the cooler and the heat transfer area. Fig. 16 shows the response surface of efficiency for values higher than 24%, resulting from the interaction of variables N_{TC} and L_{TC} . The dotted

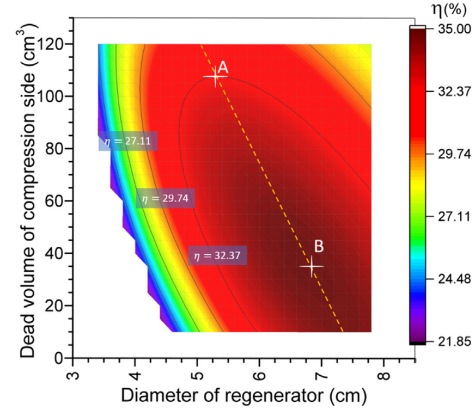


Fig. 20. Efficiency response surface of regenerator diameter with dead volume of compression side, where A and B are the selected design points to analyze on the line that indicates the optimal relation of the variables.

orange line indicates the linear approximation of the relationship among these variables which allows obtaining maximum efficiency with a $\pm 2.8\%$ variation. Results indicate that there is a length that allows for reaching maximum efficiency given a number of tubes, and there is an almost linear proportion for this parameter interaction. Points A and B are marked in the figure for the analysis of the phenomena involved.

The net power between points A and B have a difference of only

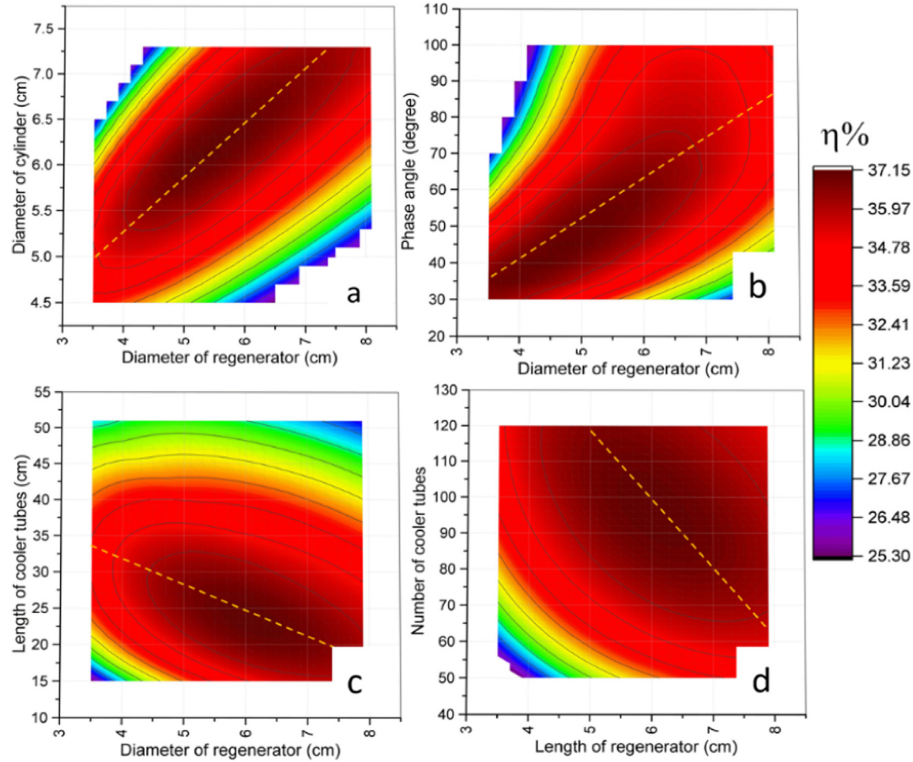


Fig. 19. Significant interactions of the shell and tube cooler, where the orange lines indicate the optimal relation of the variables determined by the proposed method. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

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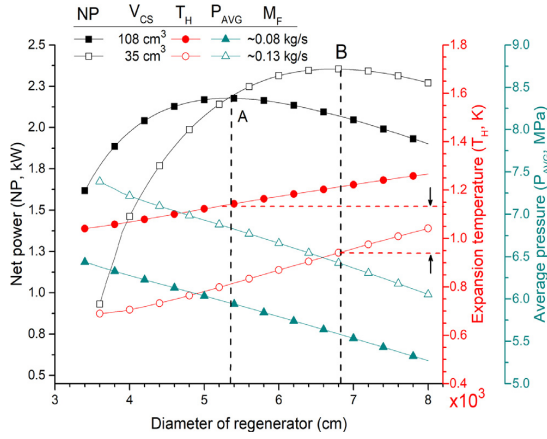


Fig. 21. Analysis of the influence of D_R and V_{CS} over net power, expansion temperature and average pressure for design points A (108 cm³) and B (35 cm³).

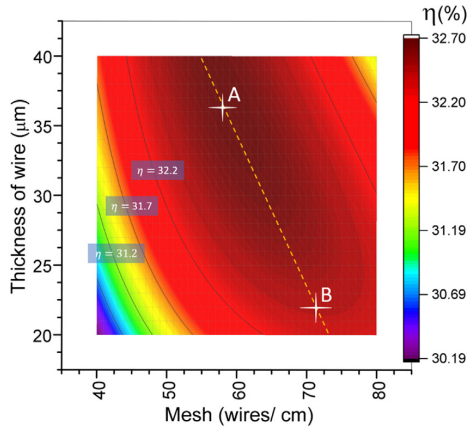


Fig. 22. Efficiency response surface of the mesh with thickness of the wires, where A and B are the selected design points to analyze on the line that indicates the optimal relation of the variables.

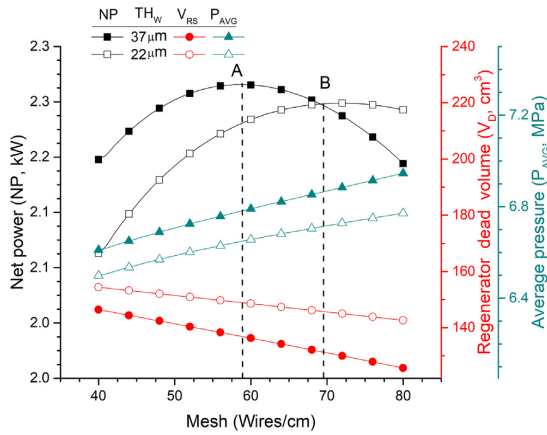


Fig. 23. Analysis of the influence of MSH and TH_W over net power, regenerator dead volume and average pressure for design points A (37 μm) and B (22 μm).

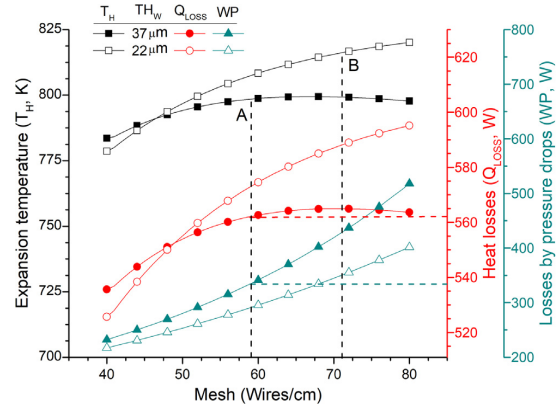


Fig. 24. Analysis of the influence of MSH and TH_W over expansion temperature, heat and pressure drop losses for design points A (37 μm) and B (22 μm). An energy trade-off is observed between thermal and pressure losses.

50 W (Fig. 17) because their efficiencies, mass flows, and dead volumes are similar ($\sim 34.90\%$, ~ 145 g/s and ~ 115 cm³ respectively). However, the heat transfer of point B is 7.5% less than point A. This area reduction causes point B to operate at a temperature difference slightly lower than point A, which reduces thermal losses (Fig. 18). Point B presents a greater transversal area for the fluid inside the cooler with a higher amount of tubes, offering lower resistance and less pressure drop losses. Under these conditions, point B presents a lower thermal and pressure drop losses than point A due to a lower temperature difference and a greater transversal area.

On the other hand, although efficiency in point A is lower, it has no significant decrease since the higher difference in temperature has a positive influence over pressure-volume (P-V) work; therefore, it compensates higher thermal and pressure drop losses in comparison with point B.

It is then possible to indicate whether there is an interaction between N_{TC} and L_{TC} variables over the difference of temperatures, thermal and pressure drop losses. When the number of tubes is low, the positive influence of the high difference in temperature over the P-V work predominates over the thermal and pressure drop losses. When the number of tubes increases, the decrease of temperatures causes the effects of thermal and pressure drop losses to predominate over efficiency.

For design purposes, it is important to analyze these phenomena in order to achieve both balance and the highest efficiency (dotted orange line in Fig. 16).

Fig. 19 shows other important interactions of the shell and tube cooler with other system variables. The optimal relation of the number vs the diameter of the cooler tubes was observed, thereby allowing a partial improvement in the performance of the equipment (Fig. 19a). Given these operation and design conditions, Fig. 19b shows the relation obtained from parameters L_{TC} and L_R which presents an optimal zone around the point found in $L_{TC} = 24.5$ cm and $L_R = 6.2$ cm, although the zone with a greater area shows efficiencies higher than 32%. The interaction of the number of tubes against the phase angle shows that angles less than 90° improves the performance of the system for these conditions. It can also be optimized as a function of the minimum number of tubes for maximum efficiency (Fig. 19b). Finally, the inverse relation between D_R and N_{TC} determined by the efficiency decreases, given the increase of WP and of the dead volume in the compression zone as shown in Fig. 19d.

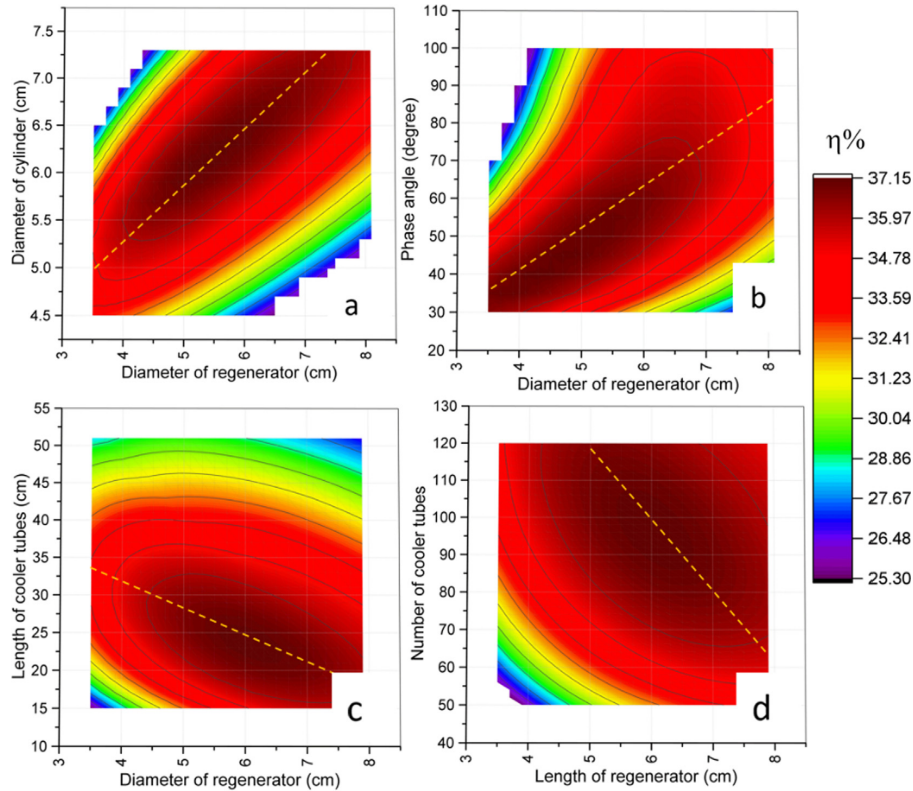


Fig. 25. Significant regenerator interactions, where the orange lines indicate the optimal relation of the variables determined by the proposed method. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

5.3. Significant interactions in the regenerator

The dead volume of the compression side (V_{CS}) affects system performance directly since it reduces the average pressure and mass flow. However, when the global interactions scheme is analyzed, it can be seen that V_{CS} has a strong interaction with the regenerator diameter (D_R), indicating that there is a relationship between these variables which influence system efficiency. Fig. 20 shows the response surface of efficiencies that results from the interaction of variables D_R and V_{CS} , where a yellow dashed line represents the optimal relationship of such variables. For a given D_R , it is noted that increasing V_{CS} could improve system efficiency. Two design points (A and B) were selected within the zone of the maximum efficiency values to analyze the phenomena involved.

Fig. 21 shows that points A and B have a power difference of 235 W despite being selected from the optimal relation. With diameters less than 5.4 cm, the 108 cm³ of added dead volume benefits the system power because as V_{CS} increases, the mass flow decreases, and the expansion temperature of the cycle and P-V work increases considerably despite the average pressure decrease. Under these conditions, the interaction of the extra dead volume V_{CS} over the temperature is predominant over the effect of the average pressure. Conversely, for diameters over 5.4 cm, the improvement occurs by reducing V_{CS} to 35 cm³ which increases the average pressure. Under these conditions, the effect of the average pressure is predominant over the effects of the temperature.

With the analysis of variable interaction, it is possible to conclude that the relationship between them is due to the influence they have over expansion temperature and average pressure. Furthermore, there

is an amount of additional dead volume that optimizes the system, balances expansion temperature and average pressure.

The matrix of the regenerator is composed by a stack of metallic meshes characterized by a number of threads per centimeter (MSH) and their thickness (THW). The variable hierarchy (Fig. 7) shows that MSH and THW have a low Φ indicator; therefore, these variables do not have a great impact over other system variables. However, the scheme of global interactions (Fig. 6) shows that those variables present a strong interaction among them. Fig. 22 shows the response surface resulting from the interactions of variables MSH and THW , where two points (A and B) were selected within the optimal relationship of such variables and represented with a dotted orange line. Fig. 23 shows the power from point A, which is used as a thick wire (37 μ m). The amount of added dead volume is decreased by the regenerator (V_R), whereby increasing the average pressure of the system. In addition, the reduction of V_R causes a decrease in expansion temperature, thereby decreasing the thermal losses. We can observe in Fig. 24 the design in point A having a lower dead volume; it has less than 20 K and 30 W in the expansion temperature and thermal losses respectively. On the other hand, the increase of the thermal and pressure drop losses were caused by the increase of mass flow and the reduction of the transversal area, which occurs when the number of threads per centimeter increases.

Using thick wires (40–30 μ m) with a few threads per centimeter reduces dead volume which increases the average pressure. Such an increase has more influence on efficiency than expansion temperature decrease; on the other hand, using thin wires and many threads per centimeter increases dead volume, and P-V works by increasing the expansion temperature. Therefore, the maximum efficiency is obtained when the influences of temperature and pressure are balanced. The Φ

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indicators of these variables have low values which represent a low variation of the efficiency in the response surface. Therefore, these variables must be the last to be optimized as suggested in the variable hierarchy.

Fig. 25 shows other important interactions of the regenerator equipment where the optimal variable relations and their best performance areas are indicated. Fig. 25a shows that the swept volume is higher; hence, it is necessary to increase the regenerator diameter to reduce energy losses by pressure drops. Similarly, as was discussed in Fig. 12, decreasing α allows improving heat transfer when the source is insufficient (Fig. 25b). Fig. 25c and d show the maximum efficiency zones determined by the variables of the regenerator equipment against the cooler-regenerator equipment. It can be seen that for values between $30\text{ cm} > L_{TC} > 20\text{ cm}$, the efficiency is over 35%, denoting that this variable has a predominant effect. Variables N_{TC} and L_R have an effect with similar proportions, thus, representing a strong interaction between them. The maximum efficiency zones allow selecting different designs that present almost the same performance in the system although with different objective functions.

6. Conclusions

The study of variable interactions from the 240 response surfaces showed that piston stroke has a greater number of joint effects or significant interactions with the rest of the variables and followed by the cylinder diameter. The number and length of the cooler tubes are the pair of variables with the second-highest hierarchy and their optimal relationship is limited by heat transfer area, added dead volume, and the compromise between thermal and pressure losses. On the other hand, the piston crown did not present any considerable interaction; however, it has an important influence on the efficiency of the system as it decreases thermal losses in the expansion zone.

Moreover, for a determined swept volume, the piston stroke must be greater than the cylinder diameter in order to reduce thermal losses in the expansion zone. Phase angles less than 90° present benefits when the heat input is insufficient since mass flow reduction allows increasing expansion temperature and P-V work. It is possible to determine the amount of V_{CS} required to make the system more efficient with the interactions of the compression zone dead volume.

Although the interaction of MSH and TH_w resulted strong, it was observed from the hierarchy that variation in the efficiencies of such interaction is below 3%.

The proposed methodology for the analysis of design variable interactions shows that there are effects with a joint influence on efficiency, as well as the conditions under which the effects of a variable predominate over the effects of another. The proposed indicator of the overall system interaction Φ represents how a variable has influence over the other system variables whereby constituting a hierarchy of design variables for a Stirling engine.

CRedit authorship contribution statement

Saúl Islas: Conceptualization, Data curation, Formal analysis, Methodology, Visualization, Writing - review & editing, Writing - original draft. **Ricardo Beltrán-Chacón:** Data curation, Funding acquisition, Methodology, Project administration, Visualization, Writing - review & editing, Writing - original draft. **Nicolás Velázquez:** Funding acquisition, Writing - review & editing, Writing - original draft. **Daniel Leal-Chávez:** Investigation, Methodology. **R. López-Zavala:** Resources, Validation. **J.A. Aguilar-Jiménez:** Software, Supervision.

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.

Appendix A. Supplementary material

Supplementary data to this article can be found online at <https://doi.org/10.1016/j.applthermaleng.2020.115039>.

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**K. Modelo de administración sustentable para
microrredes aisladas: caso de estudio
Puertecitos, México**

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Modelo de administración sustentable para microrredes aisladas: caso de estudio Puertecitos, México

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Abstract. Las microrredes con energías renovables nos permiten desarrollar sistemas de generación eléctrica sustentables, con los cuales se puede combatir la marginación de las comunidades aisladas. En este trabajo primeramente se presenta la metodología que se sigue para la detección de las comunidades en estado de marginación y se realiza un análisis de los diferentes modelos de gestión energética, posteriormente se muestra cómo se lleva a cabo el desarrollo de sistemas de generación eléctrica sustentables con microrredes para comunidades aisladas. El modelo de organización de Sociedad Cooperativa con Participación Estatal fue desarrollado e implementado en la comunidad de Puertecitos, Ensenada, Baja California, México. La sostenibilidad de los Sistemas Eléctricos Sustentables de las comunidades aisladas depende directamente de un modelo de gestión energética que considere los aspectos económicos, sociales y culturales de la comunidad, así como del compromiso de las instituciones gubernamentales involucradas.

Keywords: Microrred, Energías Renovables, Comunidades aisladas.

1 Introducción

Para conseguir un crecimiento equitativo de la población es indispensable incorporar el sector rural y urbano-marginal al proceso de desarrollo del país. Ambos grupos marginales necesitan de las mismas posibilidades y servicios básicos para desarrollar sus capacidades sociales y económicas. Sin embargo, muchas de estas poblaciones se encuentran en regiones aisladas sin acceso a energía eléctrica y servicios de salud, los cuales son básicos y necesarios para contar con una calidad de vida adecuada. Claro está que los países más electrificados son los más desarrollados y, por ende, los que gozan de una mejor posición social y económica. En estos países se tiene la convicción de que la energía eléctrica es una herramienta que requiere la población para el desarrollo de sus actividades productivas y mejora de su calidad de vida, atendiendo sus necesidades de comunicación, alumbrado y principalmente para el desarrollo de sus actividades agropecuarias, artesanales, comerciales e industriales [1], sin dejar de lado la conservación de alimentos y confort térmico.

La electrificación rural en México es un problema nacional que se debe de atacar. La Comisión Federal de Electricidad (CFE) estima que en México existen poco más de millón y medio de personas sin acceso a la electricidad [2]; si bien no es un porcentaje alto, sí representa una gran cantidad de habitantes sin este servicio público. El principal factor que predomina en esta inaccesibilidad eléctrica es el lugar remoto en que se encuentran las comunidades, ya que para la CFE no le es económicamente viable llevar el servicio a estos pequeños pueblos, dada la gran inversión que tendría que hacer por la ampliación de la red.

Dados los problemas que se tienen al tratar de llevar energía eléctrica a todos los rincones del planeta, en la actualidad se han desarrollado tecnologías capaces de producir este servicio en el mismo lugar donde se necesita, es decir, instalar generadores de energía en el centro de consumo, sin la necesidad de una interconexión a la red de transmisión eléctrica [3]. A esto se le llama electrificación descentralizada o generación distribuida. El sistema más comúnmente adoptado consiste en la instalación de generadores eléctricos en base a combustibles fósiles, como lo son la gasolina y el diésel ya que, de primera instancia, el costo de inversión inicial es bajo, pero el asociado a la operación es muy elevado [4], lo cual, en conjunto con las condiciones económicas de las poblaciones marginadas, vuelve imposible su operación continua. Un claro ejemplo de esto es la comunidad del Ejido Delicias, ubicada en el tramo carretero San Felipe-Puertecitos, en el estado de Baja California, México. Donde se cuenta con una población de 80 personas (20 viviendas) y su asentamiento se encuentra a 41 km de distancia de la Red Eléctrica Nacional (REN), obligándolos a producir su propia energía para satisfacer necesidades básicas, como lo son conservación de alimentos, alumbrado, aire acondicionado, entre otros. Cada vivienda aporta alrededor de \$450 MX (aproximadamente 23 USD, 24 €) mensuales sólo por dos horas diarias de electricidad, lo cual se destina únicamente a la compra de diésel para la operación de un generador eléctrico.

Es fácil comprender lo costoso que es generar energía eléctrica con combustibles fósiles durante 24 horas del día en regiones aisladas. Otras fuentes de energía muy importantes son las energías renovables, las cuales aprovechan los recursos presentes en el medio ambiente para su conversión energética, como lo son la radiación solar, viento, energía potencial de masas de agua, calor geotérmico, entre otras. Estas, a diferencia de los combustibles fósiles, no producen contaminantes cuando se utilizan, por eso también se les conoce como energías verdes.

Actualmente existe la tecnología para generar energía eléctrica a partir de las energías renovables, pero el éxito de su implementación depende en gran medida del modelo de gestión energética, ya que se han instalado muchos sistemas de generación eléctrica aislados basados en energías renovables y han fracasado por no implementar un modelo de gestión energética que considere los aspectos culturales, sociales y económicos de las comunidades.

Se han desarrollado modelos de gestión donde el gobierno (municipal, estatal o federal) o gobierno en conjunto con las comunidades se encargan de la administración y operación y mantenimiento del sistema de generación eléctrica de las comunidades [5], pero no se ha logrado desarrollar un modelo de gestión sostenible a largo plazo. Akin-yele et al [6], analizando proyectos de electrificación rural fallidos en Nigeria, proponen un modelo de implementación de microrredes aisladas basado en los principios

fundamentales de la sostenibilidad y experiencias propias de los autores. En el modelo se toma en cuenta la situación social, técnica, económica, ambiental y política, con el fin de lograr microrredes sustentables. Domenech et al [7] realizaron la instalación de microrredes en comunidades remotas de Perú utilizando distintas fuentes de energías renovables, donde la tecnología adecuada en cada área de acuerdo a sus necesidades y potencial energético fue evaluada. Este proyecto llevó electrificación a 58 casas, un centro de salud, una escuela y otros consumidores. La operación y mantenimiento de los sistemas fue realizada en conjunto con un modelo de administración comunal especialmente planteado para esta comunidad, basado en las características de la población.

Para resolver la problemática descrita, en este trabajo se propone satisfacer las necesidades de energía eléctrica de las comunidades aisladas con microrredes que utilizan energías renovables y un modelo de gestión comunitario, lo cual fue implementado en la comunidad de Puertecitos, Ensenada, Baja California, México.

2 Metodología para la detección de comunidades e implementación de microrredes

Con el enfoque de utilizar las energías renovables para combatir la marginación de las comunidades aisladas y después de hacer un estudio de viabilidad técnico económica para la electrificación de dichas comunidades, comparando las opciones; 1) extender la red eléctrica convencional, 2) generación eléctrica con diesel y 3) microrred con energías renovables, se encontró que, si la comunidad está a 35 km o más alejada de la red eléctrica convencional, la mejor opción es la microrred [8]. En la Fig. 1 se muestra la secuencia que se sigue para la detección de las comunidades en estado de marginación y el desarrollo de sistemas de generación eléctrica sustentables con microrredes, así como su modelo de gestión energética.

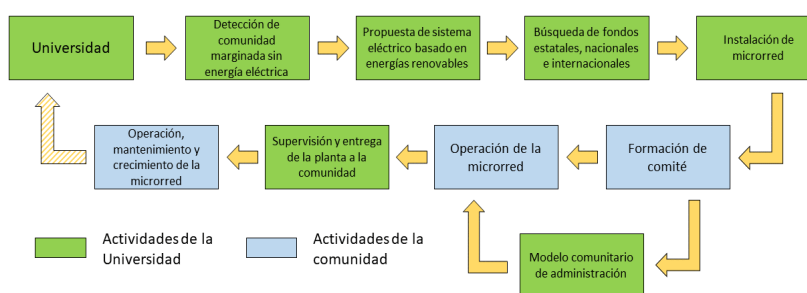


Fig. 1. Metodología para el desarrollo del sistema de generación eléctrica sustentable.

La metodología descrita en la Fig. 1 está compuesta por tres etapas:

1. Detección de comunidades: La comunidad ideal para la instalación de una microrred sería aquella que se encuentre alejada de la red eléctrica convencional, deben vivir familias que se encuentren organizadas y que tenga un recurso natural explotable o

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una actividad productiva que pueda detonarse con la existencia de un servicio eléctrico de calidad. Habitualmente son comunidades sin servicios de electricidad, agua potable, drenaje, recolección de basura, transporte y telefonía. En esta primera etapa se realiza un estudio de viabilidad técnica y económica de las diferentes opciones para la electrificación de las comunidades aisladas y se justifica el Sistema de Generación Eléctrica Sustentable.

2. Diseño e implementación de la microrred: Considerando la disponibilidad de los recursos energéticos renovables, la posible demanda de energía eléctrica, las tecnologías y recursos económicos disponibles, el poder de pago de las familias, aceptación de la tecnología por parte de los miembros de la comunidad, entre otros, se procede al dimensionamiento e implementación de la microrred.
3. Desarrollo del modelo de gestión para sistemas aislados: Una vez localizada la comunidad en estado de marginación y conseguir los recursos económicos para el Sistema Eléctrico Sustentable, se procede a realizar un estudio social, económico y cultural de la comunidad para desarrollar el modelo de gestión energética y el diseño de la microrred de acuerdo a las características particulares de la comunidad. En esta etapa se realiza las siguientes actividades:

- Visita guiada a la comunidad.
- Encuesta socioeconómica.
- Encuesta de opinión.
- Base de datos sobre las condiciones socioeconómicas de los habitantes y aparatos eléctricos disponibles.

3 Modelos de gestión para sistemas aislados

Los modelos de gestión para sistemas aislados actuales están basados en los siguientes *paradigmas de la sostenibilidad* [5, 9]:

- La asistencia técnica como criterio único de sostenibilidad
- La rentabilidad económica.
- La opción multicriterio o integradora

Con los paradigmas de la sostenibilidad se busca consolidar lo técnico, definir lo económico, ser responsable con el medio ambiente y fortalecer lo social (empoderar a las comunidades).

Los *modelos de gestión energética* que se han implementado en sistemas aislados hasta ahora, tienen las siguientes características:

- De tipo Municipal, no están capacitados para el manejo de servicios, se politizan rápidamente y se pierden los objetivos iniciales.
- De tipo Estatal, generalmente se incurre en paternalismo, se hacen inversiones sin análisis realistas (social, económico, técnico) y esta propenso a los cambios políticos.

- De tipo Comunal, es de todos y es de nadie, no existen responsabilidades claras ni reglas establecidas. Si no se desarrolla un buen modelo, con el tiempo la comunidad se desorganiza y pierde interés.
- De tipo Cooperativo, se convierte en el botín para algunos grupos internos, no cuenta con un marco legal claro.
- De tipo Privado, es más manejable, se puede adaptar al marco legal vigente, hay experiencias sostenibles, pero se encárese el servicio.

Los *criterios básicos* que se aplican en los modelos de gestión son:

- Se aprovecha los recursos locales (naturales y humanos).
- Se define la propiedad y la instancia de administración.
- Evaluación de la capacidad y voluntad de pago.
- Diagnóstico, consenso, participación, toma de decisiones.
- Se promueve el criterio empresarial en el manejo de energía (tarifas, contratos, etc.).
- Se incentiva el desarrollo de habilidades y destrezas (capacitación).
- Se fortalece la organización local.
- Se desarrolla el concepto de uso racional de energía.

Para la implementación de los modelos de gestión energética se desarrollan los siguientes *instrumentos*:

- Esquema tarifario
- Reglamentos sobre el servicio
- Contrato usuario – empresa
- Manual de operación
- Capacitación (usuarios, operadores, autoridades)
- Fiscalización

Ya que la finalidad es proporcionar un sistema para la generación de energía renovable que ayude al desarrollo económico y social de la comunidad aislada y que perdure con el paso de los años, fue necesario plantear un esquema de administración y operación y mantenimiento de la microrred en el que la universidad pueda heredar el proyecto a la población y fungir solamente como asesor técnico. A este esquema se le llamó “Modelo de Administración Comunitario Autosustentable” y se aplicó a la comunidad de Puertecitos y su microrred.

Esto permitió que la universidad no interviniera en la administración, operación y mantenimiento de la microrred, y que dejara planteado el modelo para que la misma población, propietarios del proyecto, la mantengan en óptimas condiciones. Eso no significa que la universidad no aporte con la asesoría técnica y administrativa si es que la comunidad la necesitara, ya que la institución educativa cuenta con la facultad de dar soporte en futuros incrementos de capacidad instalada, rediseño de circuitos, modificaciones del banco de baterías y actualizaciones al modelo de administración comunitario, entre otros aspectos técnicos.

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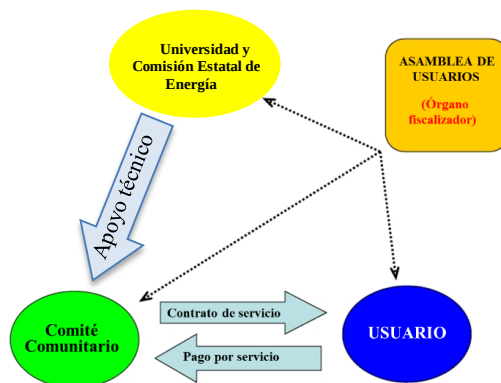


Fig. 2. Esquema de modelo de gestión.

4 Caso de estudio: Microrred de Puertecitos, México

La comunidad de Puertecitos, Ensenada, Baja California, México ($30^{\circ}21'19.7''$ N, $114^{\circ}38'26.3''$ O), es un pueblo pesquero aislado de la REN (Fig. 2), donde habitan alrededor de 20 familias con un alto índice de marginación social ya que, al no contar con los servicios básicos de electrificación, agua potable y salud, las condiciones de vida son muy deplorables y, en conjunto con las altas temperaturas ambientales de la temporada de verano de hasta 50°C , la necesidad de aire acondicionado, conservación de alimentos y medicamentos se vuelve primordial. Es aquí donde el Centro de Estudios de las Energías Renovables (CEENER) del Instituto de Ingeniería de la Universidad Autónoma de Baja California (UABC), en conjunto con distintas instituciones públicas y privadas, nacionales e internacionales, llevaron a cabo la instalación de una microrred con base a energías renovables para proporcionar energía eléctrica a la sección de la comunidad más necesitada.



Fig. 3. Comunidad de Puertecitos, Ensenada, Baja California, México.

Una vez localizada la comunidad en estado de marginación y conseguir los recursos económicos para el Sistema Eléctrico Sustentable de Puertecitos, se procedió a realizar un estudio social, económico y cultural de la comunidad para desarrollar el modelo de gestión energética y el diseño de la microrred de acuerdo a las características particulares de la comunidad de Puertecitos. Considerando la disponibilidad de los recursos energéticos renovables, la posible demanda de energía eléctrica, las tecnologías y recursos económicos disponibles, el poder de pago de las familias, aceptación de la tecnología por parte de los miembros de la comunidad, entre otros, se procedió al dimensionamiento de la microrred y al desarrollo del modelo administrativo y operativo de la misma. En la tabla 1 y Fig. 4 y 5 se muestran las características de los componentes de la microrred o Sistema Eléctrico Sustentable de Puertecitos.

Table 1. Componentes de la microrred instalada en Puertecitos, B. C., México.

Equipo	Capacidad
Sistema fotovoltaico Solartec	55 kW
Aerogenerador Liten	5 kW
Generador diésel de 480V CA	75 kVA
Banco de baterías selladas de 1500 AH c/u	522 kWh
Controlador de carga solar con MPPT	55 kW
Controlador de carga eólico	7 kW
Inversor eólico	7 kW
Inversor principal @480V CA	100 kVA
Transformador principal	112 kVA
Transformador cuarto de control	15 kVA
Transformadores reductores	50 kVA
Interfaz de monitoreo	N/A

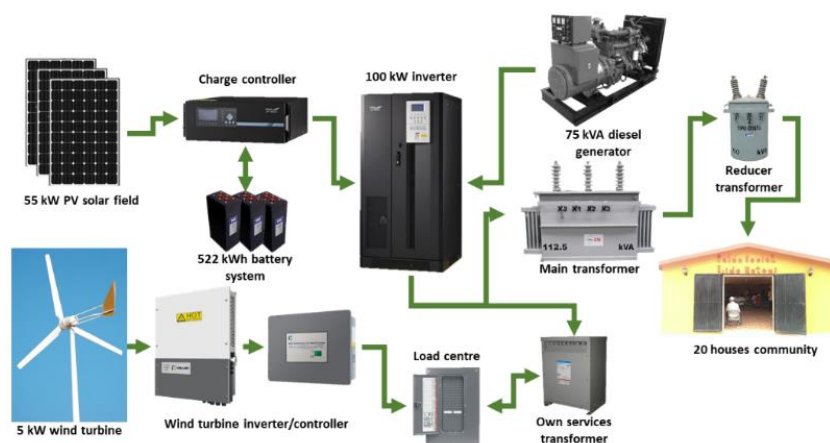


Fig. 4. Diagrama esquemático de la configuración de la microrred.



Fig. 5. Fotografía de la microrred de Puertecitos.

4.1 Modelo de gestión propuesto para la comunidad de Puertecitos

El propósito del modelo de gestión o modelo de organización es tener una administración, operación, mantenimiento, e inclusive crecimiento sostenible del Sistema de Generación Eléctrica Sustentable de Puertecitos. Para lograr lo anterior fue necesario realizar un estudio socioeconómico y cultural de la comunidad para desarrollar el modelo de gestión energética de acuerdo a las características particulares de la comunidad. Con dicho estudio se tomó en cuenta todos los aspectos sociales al instalar la planta de generación eléctrica con energías renovables y se promovió su aceptación de forma sostenible, considerando sobre todo los siguientes aspectos:

- Situación actual de la comunidad.
- Condiciones socioeconómicas de los hogares.
- Generación y consumo de energía eléctrica.
- Posición de la comunidad con respecto a la prestación del servicio de electricidad.
- Análisis FODA.

El modelo de organización propuesto para el Sistema de Generación Eléctrica Sustentable de Puertecitos es bajo la figura jurídica de Sociedad Cooperativa de Participación Estatal. Las principales razones para seleccionar dicho modelo son:

- Cuenta con una base legal ya definida lo que acelera la planeación de actividades en el corto plazo y facilita la resolución de conflictos en el largo plazo.
- Permite el registro de la Sociedad en convocatorias públicas para solicitud de fondos gubernamentales.
- Fomenta la participación comunitaria sin dejar de lado atribuciones que le permitan al gobierno estatal (Comisión Estatal de Energía) asegurar el buen funcionamiento de la Sociedad y el servicio de electricidad.
- Permite el manejo de recursos por medio de una cuenta bancaria con cheques controlables y tarjetas de débito limitadas.

No es posible presentar en este artículo la totalidad de los documentos desarrollados, por lo que se enfocará en algunos aspectos de la constitución y reglamento del modelo de organización propuesto.

4.2 Constitución

- El servicio de energía eléctrica se otorgará bajo la figura jurídica de Sociedad Cooperativa de Participación Estatal.
- Esto permitirá que la Comisión Estatal de Energía (CEE) y la Comunidad puedan desarrollar sus actividades de manera conjunta y con una base legal definida en la Ley de Sociedades Cooperativas.
- La Comunidad desempeñaría las funciones requeridas para mantener el servicio de electricidad funcionando y la CEE en conjunto con la Universidad (UABC) las de soporte y fiscalización.

El acta constitutiva consta de 90 artículos que hablan de los siguientes aspectos:

- Constitución, domicilio, duración y objeto
- Los socios
- Capital social
- Entradas y salidas de recursos económicos
- Registro de actividades, contabilidad y del ejercicio social
- Administración, representación y auditoría
- Fiscalización
- Disolución y liquidación
- Disposiciones generales

4.3 Reglamento

El objetivo del reglamento es regular y asegurar el aprovechamiento, mantenimiento, reparación y mejoramiento del servicio de energía eléctrica, en beneficio de los habitantes de la localidad.

La aplicación del presente reglamento les corresponde a las siguientes autoridades:

- Al presidente del Consejo Administrativo
- A la Asamblea
- A las autoridades fiscalizadoras

El reglamento comprende los siguientes apartados:

- Definiciones
- Condiciones para la entrega del servicio
- Aspectos técnicos del servicio
- Derechos y obligaciones del usuario
- Derechos y obligaciones de la distribuidora
- Condiciones comerciales

5 Resultados

Antes de la puesta en marcha del sistema, existía un gran escepticismo por el proyecto debido a las malas experiencias adquiridas con autoridades locales. Sin embargo, cuando se inició el suministro de energía eléctrica, la población cambió repentinamente de actitud. Fue entonces cuando se logró concluir el ciclo metodológico presentado en la Fig. 1, entregando el proyecto al comité de la comunidad.

Ya que no se contaba con un servicio de energía eléctrica continua y confiable en la comunidad, al iniciar operaciones la microrred no se esperaba que se utilizaran una gran cantidad de equipos electrodomésticos en los hogares de los habitantes y que, a medida que avanzara el tiempo, se fueran adquiriendo hasta el límite máximo establecido.

En la Fig. 6 se muestra la cantidad de energía eléctrica suministrada por la microrred a la comunidad durante todo un año completo de mediciones. Para tal registro se utilizó el equipo Fluke 1735 Power Logger. El inicio operativo de la microrred fue en febrero, un mes en el que las temperaturas ambientales no eran tan elevadas, por lo que no era necesario el encendido del aire acondicionado y, por ende, la demanda de energía de la comunidad no superaba los 9 kW. A medida que avanzaban los meses la demanda se incrementaba debido a distintos factores, como lo es el incremento de la temperatura ambiental y la adquisición de nuevo equipos electrodomésticos. Para el mes de mayo a julio las temperaturas presentes en la comunidad obligaban a tener encendidos los equipos de aire acondicionado, por lo que la demanda se incrementó en gran medida, al grado de que la generación eléctrica basada en energías renovables no era suficiente para soportar periodos de 24 horas continuas de suministro, provocando colapsos energéticos. Cada vez eran más prolongados los periodos sin energía eléctrica a medida que avanzaban los días, hasta que en el mes de agosto se detuvo su operación completa para evitar dañar al banco de baterías por las sobredescargas.

Para el mes de diciembre el sistema se reestableció, y en la actualidad sigue operando con normalidad teniendo un mejor control con la generación-demanda-almacenamiento.

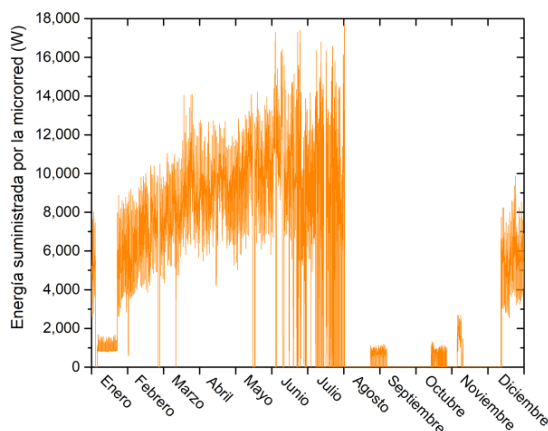


Fig. 6. Energía eléctrica suministrada por la microrred durante un año.

Por otro lado, el modelo de administración sustentable propuesto cumplió con las expectativas; durante los cuatro años operativos el comité realiza sus actividades de medición del consumo energético, facturación, cobro y mantenimiento del sistema, creando una bolsa económica para futuras expansiones y/o reemplazos de equipos.

6 Conclusiones

La instalación de Sistemas Eléctricos Sustentables con microrredes, se puede utilizar como punta de lanza para el desarrollo social y económico de las comunidades aisladas en estado de marginación. La sostenibilidad de los Sistemas Eléctricos Sustentables de las comunidades aisladas depende directamente de un modelo de gestión energética que considere los aspectos económicos, sociales y culturales de la comunidad, así como del compromiso de las instituciones gubernamentales involucradas. El reto más importante que se tiene en la electrificación de las comunidades aisladas no está en la parte técnica, la sostenibilidad depende mucho de los aspectos sociales y culturales de las comunidades.

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