**Energy and exergy study of the integrated adsorption-absorption system driven by transient heat sources for cooling and desalination**

Ramy H. Mohammeda,b, Ali Radwanc,d, Ahmed Rezke, Abdul Ghani Olabic\*, Vikas Sharmae, Abul Kalam Hossaine, Abed Alaswade, Mohammad Ali Abdelkareemc,e\*

a Department of Mechanical Power Engineering, Zagazig University, Zagazig 44519, Egypt

b Multiphysics Energy Research Center (MERC), College of Engineering, University of Missouri-Columbia, Columbia, MO 65211, USA

cSustainable Energy & Power Systems Research Centre, RISE, University of Sharjah, P.O. Box 27272, Sharjah, United Arab Emirates

dMechanical power engineering department, Mansoura university, Egypt.

eEnergy and Bioproducts Research Institute (EBRI), College of Engineering and Physical Science, Aston University, Birmingham B4 7ET, UK

eChemical Engineering Department, Minia University, Elminia, Egypt

\*Corresponding authors

**Abstract**

The scarcity of clean water and the lack of sustainable cooling systems are continuously pressing. Although many technologies are well-established, such as vapour compression for refrigeration and reverse osmosis for desalination, they are energy-intensive and conventional refrigeration technology utilising working fluids of long-lasting ozone-depleting and greenhouse effects. Alternatively, adsorption and absorption technologies can meet such demands, and they are the most feasible to utilise the waste and renewable heat abundant in many locations. Therefore, this paper computationally studies the emerging integrated adsorption-absorption system for cooling cum desalination employing transient waste heat sources of various waveform characteristics. A previously validated computational model for the adsorption subsystem was coupled with a thermodynamic model for the absorption subsystem and experimental heat profiles obtained from an internal combustion engine. The energy and exergy analysis of the integrated system utilising the actual heat source from an internal combustion engine and predefined waveforms were undertaken and benchmarked against that operated under steady heat sources. The integrated system operated with a relatively low exergy efficiency in the absorption cycle of up to 15.33%. The adsorption bottoming cycle successfully utilised the heat from the absorption subsystem at a relatively higher exergy efficiency of up to 42.69%. The execution of a transient heat source of sinusoidal waveform enhanced the water production by up to 30% and the cooling of absorption and adsorption subsystems by 24% and 15%, respectively. However, admitting realistic waveforms of an internal combustion engine showed marginal differences compared to the steady heat sources owing to their high frequencies and small amplitudes.

**Keywords:** Adsorption; Absorption cycle; Refrigeration; Desalination; Transient heat loads; waste heat recovery.

**Nomenclature**

|  |  |
| --- | --- |
| A | Amplitude |
| C | Specific heat, J/kg K |
| COP | Coefficient of Performance, – |
| E | Exergy, J |
| h | Enthalpy, J/kg |
| Hads | Heat of adsorption, J/mol |
| hfg | Latent heat of evaporation, J/kg |
| M | Mass, kg |
|  | Mass flow rate, kg/s |
| s | Entropy, J/K |
| SECWP | Specific exergy of the cyclic clean water productivity, W/kg |
| SCP | Specific Cooling Power, W/kg |
| SCWP | Specific clean water production, kg/kg |
| SHP | Specific Heating Power, W/kg |
| T | Temperature, K |
| To | Ambient temperature, K |
| t | Time, s |
| W | Adsorption uptake, kg/kg or Work, J |
| **Greek symbols** | |
| η | Efficiency, % |
| **Subscripts** | |
| AB | Absorption |
| AD | Adsorption |
| co | Condenser |
| des | Desorption |
| ev | Evaporator |
| Hex | Heat exchanger |
| in | Inlet |
| out | Outlet |
| P | Pump |
| sg | Silica gel |
| w | Water |

# Introduction

Freshwater, sustainable cooling, a clean environment, and green energy are the main pillars of sustainable development worldwide [1]. However, the continuous world population growth stands in achieving sustainable development because it [aggravates the problem](https://context.reverso.net/%D8%A7%D9%84%D8%AA%D8%B1%D8%AC%D9%85%D8%A9/%D8%A7%D9%84%D8%A5%D9%86%D8%AC%D9%84%D9%8A%D8%B2%D9%8A%D8%A9-%D8%A7%D9%84%D8%B9%D8%B1%D8%A8%D9%8A%D8%A9/aggravated+the+problem)s of water scarcity, energy shortage, and carbon emissions nexus [2]. Salty water (i.e., seawater and brackish water) is the secondary source of fresh water because of its abundance. However, desalinating salty water to obtain fresh water is expected to expand further in the following decades to meet the increasing global demand. Besides, meeting the cooling demand for indoor thermal comfort or food preservation is of increasing concern from energy consumption and sustainability viewpoints, particularly in hot climates. Therefore, freshwater and cooling technologies are coupled with [renewable and waste energy resources](https://www.sciencedirect.com/topics/engineering/renewable-energy-resources) to reduce the dependency on grid electricity, reduce the carbon footprint, and foster several sustainable development goals (SDG): SDG-6 “clean water and sanitation”, SDG-11 “sustainable cities and communities” and climate action [3,4].

The vapour compression cycle is the predominant technology to provide cooling by far [5]. However, cooling using conventional vapour compression cycles consumes much energy and significantly contributes to global warming potential [6]. Therefore, heat-driven cooling systems (i.e., sorption systems) that can utilise low-temperature waste and solar heat are feasible alternatives to supersede and retrofit the well-established vapour compression cycles [7]. Sorption systems, consequently, including physical adsorption, chemical absorption, and composited adsorption/absorption, have received increasing attention due to their low maintenance cost, easy operation, low environmental impact and low energy consumption, as the majority of energy demanded can be met via waste, underground or solar heat [8–11]. Besides, the adsorption cycle can provide fresh water along with cooling with minimal technical complications [12]. This literature review summarises the recent studies that emphasised solar-driven and waste heat-driven sorption cooling systems.

* 1. **Solar-driven sorption cycles**

Recently, several solar-driven sorption systems have been investigated. For instance, Du et al. [13] studied the optimisation of evacuated-tube solar collectors to provide heat for adsorption desalination to maximise its economic benefits. They achieved a 20% saving in freshwater unit cost by optimising the collector area. [Olkis](https://www.sciencedirect.com/science/article/pii/S0360544221011907#!) et al. [14] utilised solar energy to power an adsorption desalination cycle that used advanced ion gutilizedial as an adsorbent. In that study, solar collectors of 7000 m2 provided fresh water sufficient to meet the demand for 160 persons in Sicily, Italy. Alelyani et al. [[19]](https://www.sciencedirect.com/science/article/pii/S0196890421002892#b0095)  studied an adsorption cooling cycle driven by an evacuated tube solar collector at a COP of 0.15. [Vallès](https://www.sciencedirect.com/science/article/pii/S1359431119351695#!) et al. [10] studied a solar thermally driven Li-Br/water absorption cycle for heat pump and cooling. A heat source of 40°C could drive the studied system to deliver hot water at 55°C and a COP of 0.45.

[Mehrpooya](https://www.sciencedirect.com/science/article/pii/S0196890418309749#!) et al. [15] studied integrated concentrated solar power collectors, absorption refrigeration, and Multi-effect desalination (MED) systems for simultaneous freshwater production and cooling production. The integrated system delivered electrical power of 4632 kW, refrigeration power of 820.8 kW, and freshwater of 82,000 kg/hr, and the overall exergy efficiency was 66%. Further, Bai et al. [16] studied the cooling cum desalination performance of an adsorption cycle using multi-walled carbon-nanotube-embedded zeolite 13X/CaCl2 composite adsorbent; the reported specific daily water production (SDWP) was 18 m3/ton adsorbent/day, and the specific cooling power (SCP) was 490 W/kg. Youssef et al. [17] showed that the adsorption cycle using MOF CPO-27Ni adsorbent can produce 22.8 m3/ton-adsorbent/day and achieve SCP of 760 kW/ton-adsorbent. Also, Ali et al. [18] coupled an adsorption desalination cycle with a humidification/dehumidification (HDH) and ejectors; the system was powered by solar energy. The plant produced a daily freshwater of about 98.41 m3/ton at the cost of US$0.48/m3 with a gained output ratio (GOR) of 2.75. Capocelli et al. [19] conducted a numerical study for an integrated adsorption/HDH system. The plant yielded a daily freshwater of 30 m3 at a GOR value of 7.0 using extractions stages. Furthermore, Qasem and Zubair [20] studied an integrated system of adsorption desalination and air-heated HDH cycles. This plant produced fresh water at the cost of US$6.4/m3 with a GOR of 7.8. Elbassoussi et al. [21] proposed integration between the adsorption cooling cycle and a water-heated HDH. The system produced water at the cost of US$11.5/m3, while the GOR was about 2.5. Qasem et al. [22] coupled an absorption refrigeration cycle with an HDH cycle. Freshwater and cooling power of 1145 L/h and 62.45 TR were produced. It was found that the system could give a GOR value of 4.54, a COP of 1.29, and a drinkable water cost of 2.89 US$/m3.

* 1. **Waste heat-driven sorption cycles**

There has been an increasing interest in utilising the waste heat from heat-intensive industries and internal combustion engines to reduce the carbon footprint. In the cement industry, for example, Laazaar and Boutammachte [23] reported that heat losses by convection and radiation in the rotary kiln and clinker cooler account for about 26% of the thermal energy utilised by the plant. Waste heat from internal combustion engines and other heat-intensive industries like steel and glass can be virtually recovered via several approaches, such as driving the organic Rankine cycle (ORC), Kalina cycle, steam generators, hybrid pneumatic power systems, organic Rankine cycle (ORC) or multigeneration by integrating more of these cycles [24]. Hossain et al. [25] studied trigeneration utilising internal combustion engines integrated with a vapour absorption refrigeration system for remote/rural areas in India at an overall efficiency of 86.2%. Another technological advancement was generating distilled water using multiple-effect distillation powered by engine waste heat [25]. The waste heat from the exhaust gases of a Bi-fueled engine was recovered to drive the high-temperature Kalina cycle (HTKC) and the organic Rankine cycle (ORC) [24,26]. It was reported that the HTKC and ORC cycles could cut their respective fuel usage by 20%–30% and 8%–10%. Besides, HTKC performed more substantially than the ORC [24].

Wan et al. [27] theoretically evaluated the performance of the conventional [combined power cycle integrated with an adsorption cooling system for heat recovery.](https://www.sciencedirect.com/topics/engineering/combined-cycle-power-plant) Both thermal and economic analyses were undertaken, and the study concluded that the cooling cycle has the largest desorption and cooling capacity at a desorption temperature of 60 °C. The reported [payback period](https://www.sciencedirect.com/topics/engineering/payback-time) of the system was around 2.34 at this adsorption temperature. Pan et al. [28] experimentally evaluated the performance of Silica gel-water adsorption refrigeration operated by the low-temperature waste heat below 80 °C driving hot water temperature. This heat was obtained from data centre waste heat. They concluded that the adsorption chiller could be well operated under heat source temperatures of 51.4 to 61.3 °C, with the COP of 0.285–0.477 and SCP (specific cooling power) of 71.2–108.4 W/kg and producing 18.8–22.4 °C chilled water. Furthermore, they utilised the waste heat from the glass, and liquid crystal display panel manufacturing, COP of 0.317–0.483 and SCP of 95.2–146.7 W/kg were obtained at a heat source temperature of 56.2–74.9 °C and chilled water outlet temperature of 14.2–16.8 °C. These experimental results clarified the feasibility of using the adsorption chillers with low-grade waste heat. Gupta and Ishwar Puri [29] theoretically evaluated utilising the waste heat of data centres and high-performance clusters to drive a silica gel/water adsorption chiller. The proposed strategy’s thermal, economic, and environmental benefits were compared against a baseline infrastructure without waste heat utilisation. They concluded that the proposed implementation saves energy by 22.5%, and the annual CO2 emissions are reduced by 104 tons. Following the same concept of using waste heat to produce cooling, Hamdy et al. [26] conducted a comprehensive literature review to explore the adsorption cooling systems driven by waste heat from automobiles. However, the engine’s use to deliver cooling and clean water has not been investigated. In addition, the effect of the instabilities of the heat source has not been explored.

Given the literature above, it can be concluded that integrating sorption cycles (adsorption or absorption) with other thermal cycles is a promising avenue to increasing each system’s performance and reducing the integrated systems’ overall irreversibility. On the one hand, adsorption cycles can be driven by low-grade heat that other technologies cannot exploit to generate cooling and (or) clean water from highly saline water but generally at relatively low energy conversion efficiency. On the other hand, the absorption cycle could produce cooling power at COP higher than adsorption cycles, but it has limited operating conditions, primarily to avoid the crystallisation problem. Therefore, hybridising both technologies hold a strong potential to boost the integrated systems’ overall performance. On reflection, Almohammadi and Harby [30] developed an integrated absorption-adsorption system powered by solar energy utilising evacuated tube solar collectors and studied its performance throughout the year. The performance of this system was compared with a standalone adsorption and absorption cycle. As a result, the integrated system produced a cooling capacity of 19.86 kW at a COP of 1.17, which was higher than the individually operated ABS and ADS by 154.42% and 59.74%.

Having concluded the above, hybridising adsorption and absorption technologies and driving them using waste heat from internal combustion engines (ICE) for cooling cum water desalination is an application area that has been overlooked, despite the widespread use of stationary ICE for power generation. Such an integrated technology poses a solution for the scarce sustainable cooling and water desalination in various geographical locations, including arid locations and, for example, the marine industry. However, the waste heat generated from ICE is highly intermittent, unlike the widely investigated solar heat and heat from data centres for each system or their integration. Therefore, this work’s critical contribution (i.e., novelty) is to understand the energy conversion potential of the integrated adsorption-absorption systems –via exergy analysis– and their performance –via energy analysis– when operating under transient heat sources of different characteristics. Accordingly, the objectives of this work are to (1) develop a computational model to imitate the integrated adsorption and absorption subsystems; (2) experimentally measure the transient heat generated from an ICE’s exhaust gasses and couple it with the simulation model; (3) undertake the energy and exergy analysis for the integrated system when operating under steady heat sources of various temperatures and predefined transient heat sources of different characteristics (frequency and amplitude) to develop an insight into the system operation under actual transient heat wasted from an ICE at several mean temperature values. The integrated system can be installed in the rural and remote areas to recover the waste heat of ICE, diesel energy, or power generation cycles installed to generate electricity, and hence improve the overall energy efficiency.

# Methodology

## System description

### Adsorption-absorption integrated system

The integrated system investigated in this work consists of a silica gel adsorption cooling cum desalination cycle and a single-effect LiBr-water absorption cooling cycle, as shown in Figure 1. The exhaust gas waste heat of an internal combustion engine drives the absorption subsystem, where a heat exchanger is employed to transfer the heat from the exhaust gas stream, and the working fluid (i.e., hot water stream) flows in the absorption subsystem’s generator (1-2). The absorption cooling subsystem consists of four components: absorber, AB-generator, AB-evaporator, and AB-condenser. LiBr/water is used as a working fluid because LiBr (absorbent) is non-volatile, widely available and water suitable for the chilling application owing to its high latent heat.

A hot stream (state 1) enters the generator to increase the temperature of the stream (8) and generate water vapour that flows to the condenser. The water leaves the AB-condenser as a saturated liquid directed to the AB-evaporator through an expansion valve. As a result, the AB-evaporator meets the cooling demand, and the water inside the AB-evaporator evaporates and flows to the absorber (state 15), where a strong solution enters at state (11) and exits at state (6). This stream is pumped to the generator after exchanging heat with the weak solution in a solution heat exchanger installed between the absorber and the generator.

The adsorption cooling cum desalination subsystem consists of four components: AD-evaporator, AD-condenser, and two adsorption beds. The silica gel/water pair was employed as an adsorbent because it is affordable and thermochemically stable. The AD-evaporator is filled with seawater (salinity of 30 ppt) which evaporates employing chilled water follow and low-pressure conditions in the AD-evaporator. This vapour flows to the adsorption bed and undergoes an adsorption process (states 21 and 22), where silica gel absorbs and creates low pressure inside the AD-evaporator. At the same time, cooling fluid is used to remove the heat of adsorption and facilitate the adsorption process. Once the adsorption process is completed, the valve between the adsorption bed and AD-evaporator is closed, and the hot water exit from the AB-generator (state 2) is employed to regenerate the adsorbed water vapour and build up the pressure in the adsorbent bed. When the pressure inside the desorption bed exceeds the AD-condenser pressure, the valve between the AD-condenser and desorption bed is opened, allowing the vapour to enter the AD-condenser (state 20), where it condenses and collected as freshwater (state 23). Two beds are used to alternate between the adsorption and desorption processes and ensure continuous operation. Due to the seawater evaporation in the AD-evaporator, the salinity of seawater increases over time. When this salinity reaches a particular value, the brine (i.e., high saline water) is drained, and the AD-evaporator is filled with seawater again. Previous studies have shown that the evaporation rate is almost constant for seawater salinity up to 220 ppt [31]. Therefore, such an integrated system can produce cooling and freshwater simultaneously.

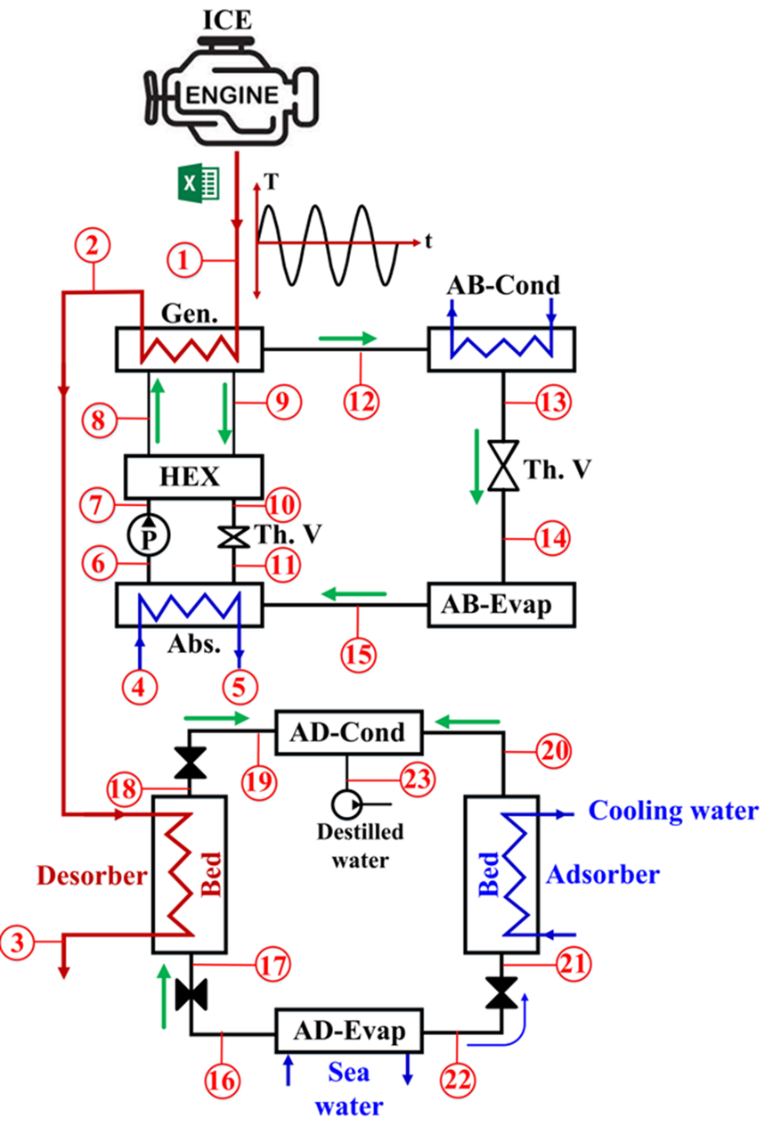


Figure 1 - Schematic of the integrated adsorption-absorption cooling and desalination system

### Internal combustion engine subsystem

This study employed a two-cylinder water-cooled Yanmar 2TNV70 compression ignition engine as a modular ICE to experimentally acquire the actual transient exhaust gas waste-heat profiles. As explained above, such heat is transferred to the adsorption-absorption integrated system via a heat exchanger. The engine was connected to a GUNT CT-300 test stand, as shown in Figure 2; their specifications are shown in table 1. An Asynchronous motor was connected to the engine to apply different loads, hence varying the waste heat profile, as shown in Figure 2.

The engine was fueled using standard diesel fuel (B7). Several performance parameters were collected: speed, torque, air intake, air temperature, fuel inlet temperature, exhaust gas temperature, coolant water inlet and outlet temperatures, oil temperature, and the fuel volumetric consumption rate. The transient data were recorded using LabVIEW coupled with GUNT-CT300 software.

Table 1 – Specification engine test rig

|  |  |
| --- | --- |
| Manufacturer | Yanmar |
| Model | 2TNV70 |
| No. of cylinders | 2 |
| Type of injection | Indirect |
| Cooling system | Water-cooled |
| Bore × Stroke | 70 × 74 [mm] |
| Displacement | 0.570 [L] |
| Rated power | 7.5 kW at 2500 rpm |

The exhaust gas stream from the ICE is used to heat water via a heat exchanger. Then the hot water is used to drive the AB/AD cycles. The flow rate of the water is controlled to obtain the required driving temperature.

|  |  |  |
| --- | --- | --- |
|  | | |
| 1. Settling tank | 2. Air filter | 3. Cooling water control valve |
| 4. Fuel pump | 5. Filling valve | 6. Drain Valve |
| 7. Bypass | 8. Fuel measurement | 9. Diesel return line |
| 10. Fuel filter | 11. Shut off valve | 12. Asynchronous motor |
|  | 13. Combustion engine |  |

Figure 2 – GUNT CT-300, the modular experimental ICE heat source

## The integrated system modelling

The simulation model of the adsorption subsystem consisted of four sub-models, each imitating a component’s heat and mass transfer performance: AD-evaporator, AD-condenser, adsorption-bed, and desorption-bed. In addition, the adsorption subsystem model was coupled with a thermodynamic model that imitates the absorption cooling subsystem. Figure 3 demonstrates the programming flow chart of the numerically integrated sub-models to determine the system’s dynamic performance, including the energy and exergy efficiencies.

Diagram

Description automatically generated

Figure 3 - Flow chart of the computational model

MATLAB® platform was employed to solve the governing equations considering the variation of the thermophysical properties of the working fluids by employing REFRPOP 9.1® [32]. Various waste heat temperature profiles were coupled with the model via Microsoft excel to determine the temporal heating water inlet temperatures. The assumptions below were considered in alignment with other relevant research [18,33].

1. The working substances are stable.
2. Pressure loss in all heat exchangers, as well as pipelines, is negligible.
3. The saturated vapour exits the evaporators, and saturated liquid leaves the condensers.
4. LiBr-water solution is homogenous.
5. The components are perfectly insulated, meaning zero heat losses to the ambient.

### Governing equations

#### Adsorption cooling and desalination subsystem

Equations 1-3 present the energy balance equations for the adsorbent bed, evaporator, and condenser [34]. The term on the left-hand side of each equation represents the dynamic (i.e., time-dependent) temperature change by means of water vapour adsorption/evaporation in adsorbent bed/AD-evaporator and desorption/condensation in adsorbent bed/AD-condenser interconnection. The first term on the right-hand side represents the heat of adsorption and desorption rate in the adsorbent bed, the evaporation heat rate, and the condensation heat rate. The last term on the right-hand side is the heat added to or rejected from the component. Equation 4 presents the intra-cycle water mass balance in the evaporator, considering no saline water in-flow condition to the evaporator during the adsorption/evaporation process. The saline water makeup occurs inter-cycles and equals the amount of clean water produced from the condenser during the desorption/condensation process, as in equation 5, and when it reaches the maximum salinity, as mentioned above.

|  |  |  |
| --- | --- | --- |
|  |  | (1) |
|  |  | (2) |
|  |  | (3) |
|  |  | (4) |
|  |  | (5) |

Where is the mass (kg), is the specific heat capacity (J/kg/k); is the water uptake (kg of water/kg of silica gel), is the temperature (K), is the mass flow rate (kg/s); is the latent heat of evaporation (J/kg); is the heat of adsorption (J/kg). The detailed models of the evaporator, condenser, and adsorption beds, including the governing equations for determining the heat transfer coefficients, their variation during the operation due to varying the working fluid thermophysical properties, and the computational code, were discussed in detail by Rezk [35].

Equations 6-8 present the specific exergy rate for the adsorbent bed, evaporator, and water production, respectively, neglecting the kinetic and potential energy of the clean water produce stream and considering the desorbed water from the adsorbent bed is fully condensed.

|  |  |  |
| --- | --- | --- |
|  |  | (6) |
|  |  | (7) |
|  |  | (8) |

Where is the change of energy rate (W); is the enthalpy (J/kg); is the ambient temperature (K).

#### Absorption cooling subsystem

Equations 9-14 present the energy balance across each component in the absorption cycle. Equation 9 represents the rate of energy used to pump the working fluid from the state (6) to the state (7). Equations 10-14 are the energy balance across each component in the absorption cycle. The numbers correspond to the states shown in Figure 1.

|  |  |  |
| --- | --- | --- |
|  |  | (9) |
|  |  | (10) |
|  |  | (11) |
|  |  | (12) |
|  |  | (13) |
|  |  | (14) |

Equations 15-17 present the specific exergy rate for the absorption cycle’s generator, pump and evaporator [36,37].

|  |  |  |
| --- | --- | --- |
|  |  | (15) |
|  |  | (16) |
|  |  | (17) |

#### Performance assessment

The integrated system cyclic performance was assessed in terms of the coefficient-of-performance (COP), specific cooling power (SCP), specific heating power (SHP), specific clean water production (SCWP), and exergy efficiency (), as shown in equations 18-26. The specific performance indicators were determined per unit mass of silica gel employed in the system.

|  |  |  |  |
| --- | --- | --- | --- |
|  | |  | (18) |
|  |  | | (19) |
|  |  | | (20) |
|  |  | | (21) |
|  |  | | (22) |
|  |  | | (23) |
|  |  | | (24) |
|  |  | | (25) |
|  |  | | (26) |

The exergy destruction rate was estimated by integrating the incremental destruction over the cycle time, similar to Eqs. 18-26. Further, the exergy efficiency was used to determine the effectiveness of a system relative to its performance in reversible conditions. Higher exergy efficiency reflects higher energy conversion quality in the system, which consequently makes the system more sustainable. It was also used to ensure the system did not violate the first and second laws of thermodynamics.

## Transient heat sources

This study investigated several predefined transient heat source profiles alongside the actual transient heat profile generated from the ICE at several mean temperatures. The predefined heat source profiles were sinusoidal at various amplitudes and frequencies. As shown in Figure 4, amplitude (A) is the maximum increase in the cyclic wave from the mean temperature value, and frequency (*f*) is the rate at which the wave is completed. The cycle time for the adsorption subsystem was maintained at 480 s, including a 30 s switching time. Equations 24 and 25 show the frequency () and the normalised frequency (), as a ratio to the adsorption cycle time.

|  |  |  |  |
| --- | --- | --- | --- |
|  |  | | (27) |
|  |  | | (28) |
| (a) | | (b) | |

Figure 4 – Demonstration and terminologies of various predefined heat source temperature profiles

# Results and discussion

This section presents the validation of the developed simulation models for the adsorption and absorption subsystems. Following the establishment of the model validation, the influence of the transient heat source temperature profiles on the integrated system performance was investigated and benchmarked against its performance under steady heat sources of various temperatures. Table 2 shows the investigated cases and the corresponding heat source’s characteristics.

Table 2 - The investigated heat source profiles

|  |  |  |  |
| --- | --- | --- | --- |
| **Case No.** | **Mean temperature, [ºC]** | **Amplitude, A, [ºC]** |  |
| 1 | 89.1 ºC | 5 | 1 |
| 2 | 2.5 |
| 3 | 5 |
| 4 | 10 |
| 5 | 10 | 1 |
| 6 | 2.5 |
| 7 | 5 |
| 8 | 10 |
| 9 | 70 ºC | Steady heat source | |
| 10 | 80 ºC |
| 11 | 90 ºC |
| 12 | 90 ºC | ICE Actual profile | |
| 13 | 80 ºC |
| 14 | 70 ºC |

## Model validation

Each of the adsorption and absorption sub-models was individually validated, as there are no experimental measurements of a similar integrated plant. The validation of the adsorption system submodel was already examined in previously published works by Rezk et al. [34,35,38]. For the absorption subsystem, Table 3 compares results obtained from the present model and those published by Konwar and Gogoi [39], where a good agreement was observed as the maximum error was 1.53%. The submodels were carefully coupled to ensure the results’ reliability and suitability to determine the integrated system performance.

Table 3 Comparison of components capacities of the present model and those determined by Knowar and Gogoi [39] at the same operating conditions

|  |  |  |  |
| --- | --- | --- | --- |
| **Component** | **Konwar and Gogoi** [39] | **Present study** | **% Error** |
| Generator (kW) | 245.35 | 247.15 | 0.73 |
| Condenser (kW) | 168.46 | 165.88 | 1.53 |
| Evaporator (kW) | 300.0 | 300.0 | 0.00 |
| Absorber (kW) | 384.72 | 380.55 | 1.08 |

## The influence of steady heat source temperature

The influence of the steady heat source temperature on the integrated system performance was investigated, while the cooling water and chilled water inlet temperature remained at 30 ºC and 11 ºC, respectively. Figure 5-a shows the temporal temperature variations of the adsorption bed, desorption bed, and AD-evaporator when the integrated system was driven using various steady heat source temperatures: 70, 80, and 90 ºC. Generally, increasing the heat source temperature while maintaining the cooling and chilled water temperatures increases the driving temperature lift leading to higher water vapour cyclic uptake. Such enhanced cyclic uptake decreased the mean AD-evaporation temperature from 7.2 ºC to 6.3 ºC (12.5% reduction) when the heat source temperature increased from 70 to 90 ºC. These results agree with the previously reported by Rezk et al. [34].

Figure 5-b presents the instantaneous variation of the cyclic specific clean water production at various steady heat source temperatures: 70, 80, and 90 ºC. It was observed that increasing the heat source temperature enhances the cyclic specific clean water production significantly owing to the enhanced cyclic water vapour uptake. For example, increasing the hot source temperature from 70 ºC to 90 ºC increased the cyclic specific water production from 26.8 kg/kgsg to 54.3×10-3 kg/kgsg.

|  |
| --- |
|  |
| (a) |
|  |
| (b) |

Figure 5 - The influence of heat source temperature on the adsorption system’s (a) temperature profile and (b) specific clean water production.

Figure 6 shows the steady-state cyclic specific cooling power (SCP), specific heat power (SHP), coefficient of performance (COP), and specific clean water production (SCWP) when the integrated system operates using steady heat sources at different temperatures. It was noticed that SCP, SHP, and SCWP decreased with decreasing the heat source temperature for both the adsorption and absorption cycle. However, decreasing the heat source temperature enhances the COP for the absorption subsystem and, more noticeably, for the adsorption subsystem. The increased COP in the adsorption system was due to the decrease in the temperature difference between the generator and AB-evaporator, which is acceptable from the thermodynamic point of view [40]. Furthermore, decreasing the heat source temperature in the adsorption subsystem reduces the heat stored grade in the adsorbed bed during the desorption process hence relatively speeding its cooling during the subsequent cooling during the adsorption process resulting in higher cooling-to-heating ratio during the cycle compared to that of higher heat source temperatures. Such an effect of thermal inertia is due to the high specific heat of the adsorbent packed into the adsorbent bed heat exchanger, as reported by Elsheniti et al. [41].

On the one hand, increasing the heat source temperature from 70 ºC to 90 ºC increased the cyclic SCP from 0.24 to 0.32 kW/kg (33.3% increase) for the adsorption subsystem and from 0.23 to 0.41 kW/kg (78.2% increase) for the absorption subsystem, respectively. On the other hand, the COP decreased from 0.69 to 0.63 (8.6% reduction) and from 0.81 to 0.78 (3.7% reduction) for the adsorption and absorption subsystems when the heat source temperature changed from 70 ºC to 90 ºC, respectively. Furthermore, increasing the heat source temperature from 70 ºC to 90 ºC increases the SHP from 0.35 to 0.6 kW/kg for the adsorption subsystem and from 0.28 to 0.52 kW/kg for the absorption subsystem. In addition, the SCWP improves by a factor of 2.0 when the heat source temperature increases from 70 ºC to 90 ºC.

|  |  |
| --- | --- |
| (a) | (b) |
| (c) | (d) |

Figure 6 - The influence of steady heat source temperatures on the cyclic (a) specific cooling power, (b) specific heating power, (c) COP, and (d) specific clean water production.

Figure 7 shows the exergetic efficiency for the adsorption and absorption subsystems and specific exergy of the cyclic clean water productivity (SECWP) at various heat source temperatures. Typically, the absorption subsystem operates at a higher temperature heat source and shows a higher temperature difference between their high- and low-temperature reservoirs than the adsorption subsystem, as the hot water stream temperature degrades while flowing from the absorption to the adsorption subsystem. These conditions result in more exergy destruction in the absorption subsystem than in the adsorption subsystem, the most in the generator and desorption bed. Such difference in the exergy destruction led to higher exergy efficiency in the adsorption subsystem compared to the absorption subsystem. For instance, the exergy efficiency for the adsorption subsystem was 22.4% higher than that for the absorption subsystem across the investigated range of steady heat source temperatures.

The ratio between the cooling and heating powers in the adsorption subsystem increased more notably compared to that for the absorption subsystem by reducing the heat source temperature across the investigated range, as can be observed in the COP trends in Figure 6-c; this thermodynamically caused more notable increase in the exergy efficiency for the adsorption subsystem compared to that for the absorption subsystem. On reflection, increasing the heat source temperature from 70 ºC to 90 ºC decreases the exergy efficiency from 42.6 % to 33.4% (9.2 efficiency points) for the adsorption cycle and from 15.3% to 14.8% (0.5 efficiency points) for the absorption cycle.

Increasing the heat source temperature increases the water vapour offtake rate during the desorption process () (i.e., the specific clean water production) and the desorbed water vapour temperature. While maintaining the cooling water flow rate and temperature in the condenser, increasing the desorbed water vapour temperature increases the enthalpy (), change the entropy () (i.e., irreversibility) and the production rate () of clean water. It thermodynamically increases the SECWP according to equation 8, primarily due to the elevated heat. For example, increasing the heat source temperature from 70 ºC to 90 ºC increased the SECWP by 104.5%, from 26.6 W/kgsg to 54.4 W/kgsg. These results agree with Cao and Chung [42].

Further, detailed energetic and exergetic analyses of the system components at various heat source temperatures are depicted in table 4. It was noticed that the rate of exergy in both adsorption and absorption components increases with increasing the heat source temperature.

|  |  |
| --- | --- |
| (a) | (b) |

Figure 7 – The influence of steady heat source temperatures on the (a) exergy efficiency of cooling production, and (b) specific exergy associated with water production.

Table 4 – The system’s energetic and exergetic analysis at the component level under various constant heat source temperatures.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | **Parameter** | **Tavg = 70 ºC** | **Tavg = 80 ºC** | **Tavg = 90 ºC** |
| **Adsorption cycle AD** | Cooling power [kW] | 220.73 | 291.16 | 342.89 |
| Heating power [kW] | 317.63 | 441.22 | 543.13 |
| Condenser power [kW] | 152.53 | 235.24 | 294.39 |
| COP [ - ] | 0.69 | 0.66 | 0.63 |
| SCP [ - ] | 0.25 | 0.33 | 0.38 |
| Water production | 23.88 | 38.24 | 48.68 |
| Rate of exergy for condenser [kW] | 4.69 | 7.74 | 10.60 |
| Rate of exergy for evaporator [kW] | 14.16 | 19.76 | 24.26 |
| Rate of exergy for desorber [kW] | 33.24 | 53.15 | 72.66 |
| Exergy efficiency AD [%] | 42.61 | 37.18 | 33.39 |
| **Absorption cycle, AB** | Cooling power [kW] | 208.97 | 297.89 | 368.89 |
| Condenser power [kW] | 218.54 | 313.84 | 391.25 |
| Generator power [kW] | 257.90 | 371.87 | 467.89 |
| Absorber power [kW] | 248.34 | 355.92 | 445.54 |
| COP [ - ] | 0.81 | 0.80 | 0.79 |
| Rate of exergy in, ABS [kW] | 31.68 | 53.64 | 76.27 |
| Rate of exergy out, ABS [kW] | 4.86 | 8.16 | 11.34 |
| Exergy efficiency AB [%] | 15.33 | 15.21 | 14.87 |

## The influence of transient heat source

The heat source temperature is transient in real-life heat recovery applications, such as solar, ICE, and industrial processes. For instance, in the most investigated solar thermal application, the harnessed heat changes significantly during the day due to variations in weather conditions, including solar radiation, wind speed, ambient temperature, cloud effect, and dust. However, the gained hot water temperature pattern from a solar thermal collector usually follows the solar irradiance pattern for a clear sky condition, which is almost a sinusoidal waveform. In ICE, the exhaust gas temperature also fluctuates at higher frequencies, smaller amplitude, and undefined waveform than solar heat. The ICE waste heat characteristics usually vary according to engine load and speed.

This work investigated the effect of utilising transient heat sources at various predefined sinusoidal waveforms, as characterised in table 2 were employed, on the integrated system’s performance as a stepping-stone to understanding the effect of realistic transient heat sources (i.e., IEC waste heat). The cooling and chilled water inlet temperatures were maintained at 30 ºC and 11 ºC, respectively. Figure 8 shows the effect of varying the heat source temperature profiles on the adsorption/desorption temperatures: steady at 90 ºC, sinusoidal at A=5 & =10, and sinusoidal at A=5 & =10. The intermittent heat source temperature resulted in a notable fluctuation in the desorption bed temperature but damped to a much lower amplitude and shifted peak because of the high thermal capacity of the desorption bed, predominantly silica gel. However, no monitored fluctuation in the adsorption bed and AD-evaporator temperatures primarily due to the steady heat transfer fluid’s (i.e., cooling water) temperatures, hence the steady adsorption/evaporation process.

|  |  |
| --- | --- |
| (a) | (b) |
| (c) | (d) |

Figure 8 – Variation of the heat source temperature, adsorption temperature, desorption temperature and evaporator temperature (a) at *f’* =0, A= 0ºC; (b) at *f’* = 10, A= 5ºC; (c) at *f’* =10, A= 10ºC; (d) zoomed view for a certain period at *f’* =10, A= 10 ºC.

Figure 9 shows the temporal variations of the desorption/adsorption temperatures and the corresponding water productivity utilising a sinusoidal heat source with different characteristics. The frequency of the heat source influenced the fluctuations in the desorption temperature profiles and the SCWP. The higher the frequency, the higher the fluctuation of the desorption temperatures and the SCWP were, with a more notable effect at high amplitudes.

|  |  |
| --- | --- |
|  |  |
| (a)  A= 5 ºC | (b)  A= 10 ºC |
| (c)  A= 5 ºC | (d)  A= 10 ºC |

Figure 9 - The influence of sinusoidal heat source at different amplitudes and frequencies on (a, b) cycle temperature profiles; (c, d) water production.

Figure 10 shows the influence of utilising a transient heat source of a sinusoidal waveform at different characteristics on the overall cyclic performance of the integrated system. Although the sinusoidal transient heat source alternate around the steady heat source line, it develops different energy levels when utilised in the integrated system and is more notable in the adsorption subsystem. It was observed that the temperature ramping-up time was longer than the ramping-down time, while this deficit reduces later in the desorption process. For instance, the ramping-down time from 75.18 oC (*t* =140 s) to 73.07 oC (*t* =175) was 35 sec, while the ramping-up time to 78.93 oC (*t* =235) was 60 sec at =5 & A=10; the ramping-down time from 75.66 oC (*t* =165 s) to 74.12 oC (*t* =185) was 20 sec, while the ramping-up time to 77.48 oC (*t* =215) was 30 sec at =10 & A=10. The longer ramping time potentially increases the rate of desorption in the adsorption subsystem and generation in the absorption subsystem but significantly depends on the heat source’s frequency, amplitude, and starting/ending of the heat wave, as can be perceived from the SCWP in Figure 10 c-d, where the level of water production at the end of the cycle varied from the steady heat source. The higher desorbed water level will be followed by a higher magnitude of adsorbed water vapour, resulting in more cooling [30].

As a result, for the adsorption subsystem, the maximum SCP and SHP values of 0.387 kW/kg and 0.618 kW/kg at A=5 and 0.391kW/kg and 0.63 kW/kg at A=10 were found at =5 and higher than those for steady state by 0.383% and 0.608%, respectively. The SCWP peaked accordingly at =5 and showed the values of 55.2 x10-3 and 56.2 x10-3 kgw/kgsilica gel at A=5 and A=10, respectively. For the absorption subsystem, the maximum SCP and SHP values of 0.41 kW/kg and 0.529 kW/kg at A=5 and 0.42 kW/kg and 0.536 kW/kg at A=10 were found at a frequency of =2.5 and higher than those for steady state by 0.412% and 0.522%, respectively. The resulting COP varied marginally compared to the steady heat source, specifically for the absorption subsystem. The COP for the adsorption subsystem peaked at =1 at values of 0.638 and 0.645 at A=5 and A=10. For the peak SCP and SHP at =5, the COP was 0.62 and 0.626 at A=5 and A=10, respectively, compared to 0.63 for a steady state. Therefore, it can be concluded that the transient heat source can potentially be designed to gain better cooling and water production with little change in the energy conversion efficiency (i.e., COP).

Figure 10-e shows the exergetic efficiency for the adsorption and absorption systems at the studied amplitudes and frequency values. Generally, the energy efficiency in the adsorption subsystem is higher than that for the absorption subsystem, as the grade of heating in the former was lower, leading to less energy destruction than in the latter. The variation in the exergy efficiency was marginal for the absorption subsystem across the investigated range of amplitudes and frequencies. Nevertheless, the adsorption subsystem attained the highest exergy efficiency at frequency values of 1 and 10. It is thermodynamically because of the higher cooling-to-heating ratio at these frequencies, as observed in COP values. For instance, the adsorption subsystem attained the maximum exergy efficiency of 33.8 at =1 and A=5, while the minimum exergy efficiency was 32.1 at =2.5 and A=10.

|  |  |
| --- | --- |
| (a) | (b) |
| (c) | (d) |
| (e) | (f) |

Figure 10 - The influence of predefined transient heat source of different amplitudes and frequencies on the cyclic (a) specific cooling power; (b) specific heating power; (c) COP; (d) exergy efficiency; (e) specific clean water production; (f) specific exergy of clean water production.

Even though the system developed the most cumulative desorption heat during the entire cycle at =5, the SECWP trend did not follow the cyclic cumulative heat, unlike the steady heat source, as the rate of change of this heat imposed a degree of irreversibility quantified by the term “”. The magnitude of the specific exergy was observed to be a strong function of the wave’s magnitude and frequency, compounded. For example, the highest SECWP was 10x10-3 kW/kgsg at =1 and A=10, while the maximum SHP was observed at =5 and A=10. Furthermore, the minimum SECWP was 8.8x10-3 kW/kgsg at =10 and A=10, while the maximum SHP was observed at =1 and A=10.

## The influence of utilising realistic engine waste heat

The impact of utilising a realistic transient heat source from the exhaust gas of an ICE was investigated. Three waves were employed, each at a different mean temperature: 70, 80 and 90 °C. The mean temperature was changed by varying the flow rate of the heat transfer fluid (i.e., water) that carries the heat from the exhaust gas stream via a heat exchanger to drive the generator of the absorption topping subsystem. As shown in Figure 11, the frequency of the transient heat source is relatively high, which was damped further while flowing in the generator and the desorption bed. Therefore, the temporal temperature and water production profile are like steady heat sources with marginal differences in desorption bed temperature, water production and evaporator temperature. Such a variation is due to the execution of heat energy carried by the heat wave, but it was more noticeable in the sinusoidal waveform of much lower frequency.

|  |  |
| --- | --- |
| (a) | (b) |

Figure 11 - The effect of ICE real waste heat temperature of various mean values on (a) temperature profiles in the adsorption system and (b) specific water production profiles.

Figure 12 shows the influence of utilising a transient heat source from an ICE on the overall cyclic performance of the integrated system. Like the steady state heat source, SCP, SHP, and SCWP decreased with decreasing in the heat source mean temperature for both the adsorption and absorption cycle, while decreasing the heat source mean temperature enhanced the COP for the absorption and more noticeably for the adsorption subsystem. As a result, increasing the heat source temperature from 70 ºC to 90 ºC increased the cyclic SCP from 0.239 to 0.384 kW/kg (60% increase) for the adsorption subsystem and from 0.226 to 0.414 kW/kg (83.2 % increase) for the absorption subsystem, respectively. Besides, the COP decreased from 0.692 to 0.628 (9.2% reduction) and from 0.811 to 0.788 (3% reduction) for the adsorption and absorption subsystems when the heat source temperature changed from 70 ºC to 90 ºC, respectively. Increasing the heat source temperature from 70 ºC to 90 ºC increased the SHP from 0.346 to 0.611 kW/kg for the adsorption subsystem and from 0.278 to 0.525 kW/kg for the absorption subsystem. The SCWP was improved by 114% when the heat source temperature increased from 70 ºC to 90 ºC.

A marginal variation was observed between the cyclic performance of the integrated system operating under a steady heat source and transient heat from an ICE, primarily due to the heat wave’s high frequency and relatively small amplitudes. Such marginal differences are populated in table 5.

Table 5 - The cyclic performance of the integrated system utilising transient heat from an ICE compared to a steady heat source

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **T**  **[°C]** | **Waveform** | **Adsorption** | | | **Absorption** | | | **Total** |
| **SCP**  **[kW/kg]** | **SHP**  **[kW/kg]** | **COP**  **[--]** | **SCP**  **[kW/kg]** | **SHP**  **[kW/kg]** | **COP**  **[--]** | **SCWP×10-3**  **[kgw/kgads]** |
| 90 | Steady | 0.383 | 0.607 | 0.631 | 0.412 | 0.523 | 0.789 | 54.389 |
| ICE | 0.384 | 0.611 | 0.628 | 0.414 | 0.525 | 0.788 | 54.790 |
| 80 | Steady | 0.325 | 0.493 | 0.660 | 0.333 | 0.415 | 0.801 | 42.727 |
| ICE | 0.319 | 0.485 | 0.658 | 0.326 | 0.406 | 0.802 | 41.865 |
| 70 | Steady | 0.247 | 0.355 | 0.694 | 0.233 | 0.288 | 0.810 | 26.680 |
| ICE | 0.239 | 0.346 | 0.692 | 0.226 | 0.278 | 0.811 | 25.605 |

|  |  |
| --- | --- |
| (a) | (b) |
| (c) | (d) |

Figure 12 - The effect of ICE real waste heat temperature of various mean values on the cyclic (a) specific cooling power; (b) specific heating power; (c) COP; (d) exergy efficiency

Figure 13 shows the exergetic efficiency for the adsorption and absorption subsystems and specific exergy of the cyclic clean water productivity (SECWP) when utilising the ICE waste heat at various mean temperatures. It was observed that the exergy efficiency in the adsorption subsystem was higher than the absorption subsystem due to the higher operating temperature of the latter, causing more significant exergy destruction. In agreement with Cao et al. [42], increasing the mean temperature increased the clean water production rate, its enthalpy and entropy, increasing the specific exergy of the clean water produced while increasing the transient heat source mean temperature. For instance, Increasing the transient heat source mean temperature from 70 °C to 90 ºC resulted in a decrease of the exergetic efficiency from 42.6% to 33.1% (9.5 efficiency points) for the adsorption cycle and from 15.3% to 14.8% for the absorption cycle respectively (0.5 efficiency points). Further, increasing the heat source temperature from 70 °C to 90 ºC increased the SECWP from 3.06 ×10-3 kW/kgsg to 9.14×10-3 kW/kgsg. The observed variations between the exergy efficiency and SECWP for the integrated system while utilising the transient and steady heat sources are populated in table 6.

Table 6 - The exergies of the integrated system utilising transient heat from an ICE compared to a steady heat source

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **T**  **[°C]** | **Waveform** | **Exergy Efficiency**  **[ % ]** | | **SECWP×10-3**  **[kW/kg]** |
| **Adsorption** | **Absorption** |
| 90 | Steady | 33.38 | 14.86 | 9.06 |
| ICE | 33.15 | 14.85 | 9.14 |
| 80 | Steady | 37.18 | 15.21 | 6.23 |
| ICE | 37.28 | 15.23 | 5.93 |
| 70 | Steady | 42.61 | 15.33 | 3.23 |
| ICE | 42.69 | 15.32 | 3.06 |

|  |  |
| --- | --- |
| (a) | (b) |

Figure 13 – The effect of ICE real waste heat temperature of various mean values on the cyclic (a) specific clean water production; (b) specific exergy of clean water production

## Economic study

A simplified comparative study was undertaken between utilising the engine’s waste heat to (i) directly operate the absorption-desorption integrated cycle for cooling and desalination presented in this study and (ii) indirectly operating a conventional vapour compression cooling unit, as depicted in Figure 14. In the benchmark system, the waste heat from the engine is used to drive Organic Rankine Cycle (ORC). Such an ORC drives a vapour compression cycle (VCC) with a relatively high COP of 4.1. It is noteworthy that utilising VCC with lower COP will be less economically efficient. The energy cost was estimated using the energy price in UAE as an example of the geographical location that demands cooling cum water desalination. Other locations will be relatively comparative. In the case of utilising 100% of the waste heat for cooling via the ORC cycle, the freshwater can be considered an additional benefit of the adsorption-absorption integrated system.

Table 6 shows the comparison between the two systems. It can be concluded that using the absorption-desorption integrated system can produce more cooling capacity by 39.1% in addition to the distilled water. To estimate the cost savings from this method compared with the system presented in Fig. 13, the benefit of extra cooling capacity, 39.1%, is converted to electricity in kWh. Therefore, the absorption-desorption integrated system can provide a benefit of 716.4 $/day due to the gained cooling and freshwater compared to the ICE-ORC-VC integrated system shown in Figure 14.

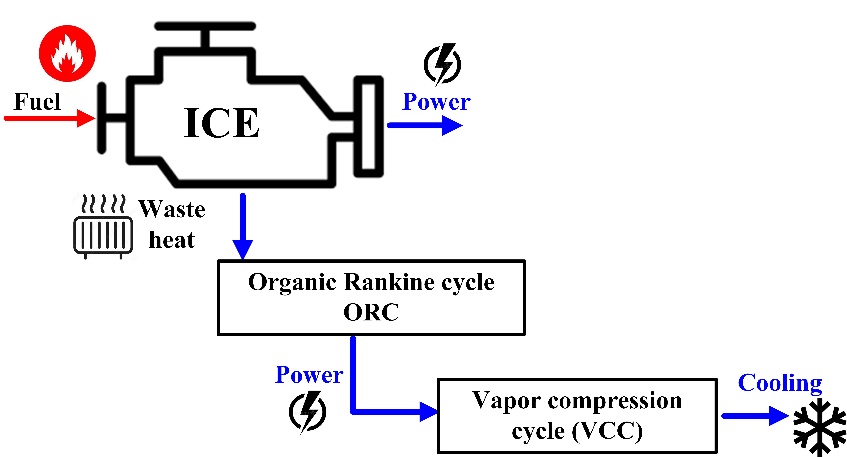


Figure 14 - A proposed cooling system uses ORC for operating the VCC for cooling purposes.

Table 6 - Benefit analysis of using the Adsorption-absorption system of this study compared to the benchmark system depicted in Figure 14

|  |  |  |
| --- | --- | --- |
|  | **Parameter** | **Tavg = 90 ºC** |
| **Ads. cycle** | Cooling power\_Ads., kW | 342.89 |
| Heating power from the heat engine, kW | 543.13 |
| COP (Ads.) | 0.63 |
| Water\_Production | 48.68 |
| **Abs. cycle** | Cooling power\_Abs., kW | 368.89 |
| Q\_generator\_abs. from the heat engine, kW | 467.89 |
| COP (Abs.) | 0.79 |
| **Ads-Abd overall energy conversion** | Overall cooling capacity | 711.78 |
| Overall heating | 1011.02 |
| COP | 0.70 |
| **ICE-ORC-VCC** | Total available heat from the waste heat, kW | 1011.02 |
| Assumed heat sink temperature, ºC | 25.00 |
| Assumed ORC efficiency [43]. | 0.1235 |
| Estimated electric power generation, kW | 124.8 |
| Assumed VCC COP [44]. | 4.1 |
| Attained Cooling Capacity driven by ORC, kW | 511.9 |
| **Benefit** | Increase in cooling capacity due to the use of this study, kW (%) | 199.9 kW (39%) |
| Attained freshwater production kg/cycle | 48.68 |
| Number of cycles per day | (3600×24)/480 = 180 |
| Total saved electricity in kWh | 199.9×180/4.1= 8776 |
| The cost of saved electricity, $0.081/kWh [45]. | 8776×0.081= $ 710.8 |
| Produced water in m3/day | 48.68×180 = 8.762 |
| Cost of produced water using seawater reverse osmosis. Total cost is 1.35 $/m3 and O&M cost is 0.64 $/m3 [46]. | 1.35×8.762= 11.8 $ |
| 0.64× 8.762 = 5.6 $ |
| Total saving due to the use of Ads/Abs system | 5.6+710.8 =716.4 $/day |

# Conclusions and prospects

## Conclusion

This study aimed to understand the energy conversion potential —via energy and exergy analysis— of the integrated adsorption-absorption systems for cooling cum desalination when operating under transient heat sources of different characteristics. A computational model was developed to imitate the integrated system, and the actual transient heat source of an ICE was experimentally acquired and coupled with the model alongside a group of steady and predefined transient heat sources of different characteristics. The finding of this study can be concluded as below.

* Increasing the steady heat source temperature elevated the driving temperature left, and cyclic water vapour uptake enhanced the SCP, SHP and SCWP. As a result, the maximum SCP, SHP and SCWP were 0.383 kW/kg, 0.607 kW/kg and 54.389 x10-3 kgw/kgads for the absorption subsystem and 0.233 kW/kg, and 0.412 kW/kg for the adsorption subsystem when operated at 90 °C. Nevertheless, the higher the steady heat source temperature negatively affected the COP due to the system’s thermal inertia. Therefore, the maximum COP was 0.694 and 0.811 for absorption and adsorption subsystems when operated at 70 °C.
* The exergy efficiency of the adsorption subsystem was, on average, 18.5 % higher than that of the absorption subsystem due to the exergy destruction that occurred at high temperatures. Besides, the maximum exergy efficiency was 42.61 and 15.33 when operated using 70 °C heat due to the enhanced cooling-to-heating ratio for adsorption and absorption cycles, respectively. The higher water production and elevated temperature increased the SECWP to 9.06 kW/kgsg at 90 °C.
* Admitting transient heat of a sinusoidal waveform developed different cumulative energy levels and was more notable in the adsorption subsystem due to the deficit between the ramping-up and ramping-down times of the heat waves in the desorption bed. It was a strong function of the frequency; the maximum SCP and SHP values of 0.387 kW/kg and 0.618 kW/kg at A=5 and 0.391kW/kg and 0.63 kW/kg at A=10 were found at =5 and higher than those for steady state by 0.383% and 0.608%, respectively. However, the COP for the adsorption subsystem peaked at =1 at values of 0.638 and 0.645 at A=5 and A=10, but there were marginal changes in the absorption subsystem.
* Admitting transient heat in the form of a sinusoidal waveform caused marginal variation in the exergy efficiency for the absorption subsystem. However, the adsorption subsystem showed the highest exergy efficiency of 33.8 and 32.1 at = 1 and 10 due to the higher cooling-to-heating ratios. The rate change of the heat source imposed a degree of irreversibility, causing the SECWP trend not to follow the cyclic cumulative heat, as the highest SECWP was 10x10-3 kW/kgsg at =1 and A=10, while the maximum SHP was observed at =5 and A=10.
* Using realistic transient heat from an ICE showed a marginal difference compared to utilising steady heat sources for adsorption and absorption subsystems, primarily because of the high frequency and low amplitudes of the heat waves damped by the system’s thermal capacity.
* Adsorption-absorption integrated system showed better financial benefit than an alternative benchmark system comprising ORC and conventional VCC utilising the exact amount and grade of heat wasted from the internal combustion engine. Such financial benefit was quantified by 716.4 $/day.

## Prospect

The developed understanding in this work is a step toward further work concerning utilising various waste heat sources. An emphasis will be on maximising the gains from the transient heat sources by potentially optimising the waveform of the transient heat source.

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