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Adaptive Radiative Collectors and Emitters (AD-RCE) to improve the efficiency of heat pumps

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Abstract:

A Radiative Collector and Emitter (RCE) is a device which combines solar collection and radiative cooling functionalities to provide both heat and cold from renewable sources. In solar collection mode, fluids are heated up using the incoming solar radiation. In radiative cooling mode, it takes advantage of the atmospheric window transparency to dissipate infrared radiation towards the outer space at nights, allowing to cool down fluids circulating through it. However, the heat production of the RCE is about 10 times higher than the cold production, and the cold water can only be produced a few degrees below ambient temperature (4-8 ºC). An evolution of the RCE, the adaptive RCE (ad-RCE) is capable to adapt its behaviour to the energy requirements, producing either heat or cold during daytime, as well as cold during night-time. To further enhance the cooling potential, we suggest coupling the ad-RCE with a compression heat pump (HP) that utilizes the cold produced by the ad-RCE as a heat sink for the condenser. In this study, we numerically estimate the performance of a water-to-water compression heat pump coupled with an ad-RCE. Our results indicate a yearly average improvement of the coefficient of performance (COP) of 3.89%, which translates to an annual electricity savings of 3.70%.

Keywords:
radiative cooling, solar thermal collection, adaptive energy production, renewable energy, heat-pump, COP.

1. Introduction

The building sector is considered to have a significant impact on energy consumption, with estimates indicating that it accounts for 40% of final energy consumption in Europe and generates 36% of CO_2 emissions [1]. Space conditioning, including DHW, cooling, and heating, constitutes the largest share of 80% of the total energy consumption in buildings. This trend is expected to continue, with rising global temperatures and heat waves leading to increase the energy consumption for refrigeration in households. Unfortunately, this will contribute to a vicious cycle of CO_2 emissions, further exacerbating the problem.

While renewable energies have become more important in recent years, cooling comfort is still achieved through electricity consumption [1]. However, new technologies based on radiative cooling have emerged in recent years, enabling the production of cold in a renewable way [2–4]. Radiative cooling (RC) is the process by which terrestrial bodies reduce their surface temperature by emitting infrared radiation towards outer space, taking advantage of the transparency of the infrared atmospheric window at certain wavelengths (7-14 µm). The low effective temperature of the space makes it possible to cool down below ambient temperature [5]. At the beginning, it was only possible to achieve RC during the night, in absence of solar radiation. New materials, called Daytime Radiative Cooling (DRC) materials, have been developed in the last decade, which enable the achievement of RC during daytime hours [6–11]. These materials reflect most of the incident solar radiation [10–12], thus the surface is able to cool down. This phenomenon, which is known as all-day radiative cooling, has been achieved using low-cost materials in recent years [13, 14].

Radiative coolers present low cooling rates, between 20 and 80 W/m^2 with peak values of 120 W/m^2 [15], which represent an order of magnitude lower than those that can be achieved in solar heating, representing a limitation of this type of technology. In order to make the technology attractive to markets, Vall et al. [15]

introduced the Radiative Collector and Emitter (RCE) as a novel technology that integrates, in a single device, solar collection and radiative cooling to produce both heat and cold through renewable means. The RCE operates by circulating a heat transfer fluid through pipes that are in contact with a radiative surface. During the day, the radiative surface heats up and transfers heat to the fluid. At night, the heat from the fluid is transmitted back to the radiative surface and radiated away. By using this technology, the dependence on fossil fuels can be greatly reduced, leading to a smaller carbon footprint.

The integration of radiative cooling systems with the built environment is still a field under study. One of the proposal is that the low power rates of cold generated in the RCE can be used to improve the performance of a water-water heat pump by increasing its coefficient of performance (COP), thereby decreasing electricity consumption. Goldstein et al [16] showed electricity savings up to 21% in an office building when a heat pump was combined with radiative cooling. In 2023, Vilà et al. [17] presented a study that simulated different configurations of RCE coupled with heat pumps, where the RCE produced cold during the night in combination with the heat pump, while during the day the RCE was decoupled and used to produce DHW independently by means of solar heating. The study examined various cities and climates and found an improvement in the system performance. However, the hot water production during solar heating mode far exceeded the domestic hot water (DHW) demand.

To minimize the underutilization of the solar collection mode while increasing the production of cold water, we propose an evolution of the RCE, the adaptive RCE (ad-RCE). The RCE takes advantage of the above-mentioned DRC materials to modify its behaviour to produce hot or cold water during the day and cold water during the night, as required. By reducing the solar heating mode to 1 or 2 hours, the ad-RCE can optimize cold production during the day.

This study builds upon the previous research conducted by Vilà et al. by implementing the advanced version ad-RCE. The aim of this study is to present a preliminary evaluation of the improvement of the COP of a water-water heat pump coupled with an ad-RCE in the condenser's side. In this study, we evaluate the performance of the combined system ad-RCE+HP integrated in a hotel in Brisbane (Australia) to cover the cold and DHW requirements of the building. The results are compared with a reference case of a conventional heat pump, and the potential energy savings are discussed.

2. Methodology

2.1. Description of the configurations

2.1.1. Studied case: ad-RCE+HP

An ad-RCE was coupled to the condenser of a water-to-water compression heat pump (**Figure *1***). The proposed configuration consisted of two modes of operation. In the first mode, the radiative cooling mode, the heat transfer fluid was initially pre-cooled through an air-water heat exchanger where its temperature decreases close to ambient temperature. The fluid was then circulated through the ad-RCE circuit, where it was further cooled, and finally, it was used as the condenser heat sink. In the second mode of operation, the heating mode, the ad-RCE field was disconnected from the heat pump. During the day, the heat transfer fluid was circulated through the ad-RCE field, heating up the fluid, and directly supplying the DHW demand.

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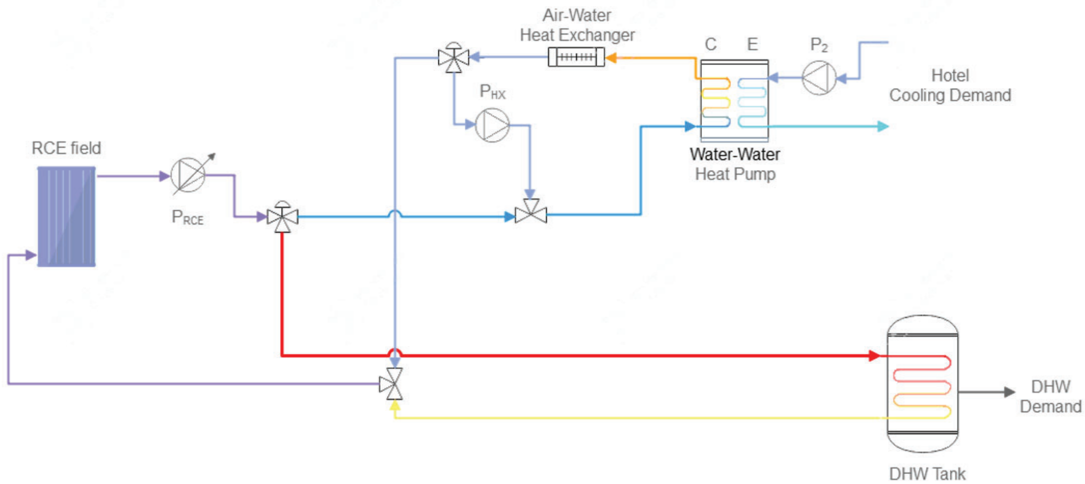

Figure 1. Conceptual scheme of the proposed configuration of an ad-RCE field with a water-to-water heat pump.

2.1.2. Reference case: water-to-water heat pump

The conventional cooling circuit, which employs a water-to-water heat pump, was used as the reference case to assess the improvement provided by the ad-RCE (**Figure *2***). The condenser side comprised a closed circuit with a single speed pump and an air-water heat exchanger. The heat transfer fluid was cooled down to temperatures close to ambient in the heat exchanger (5 ºC above the ambient), releasing the heat of condensation in the heat pump to the surrounding air.

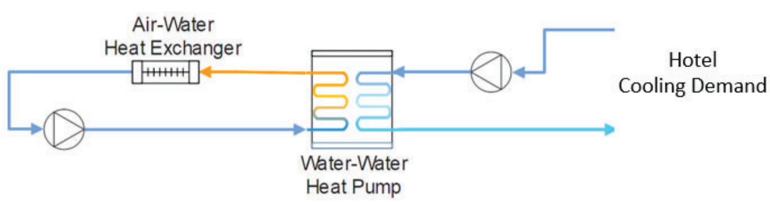

Figure 2. Conceptual scheme of the reference case of conventional water-to-water heat pump.

2.2. Building Load Demands and simulation condition

The studied facilities were designed to meet the load demand for cooling and DHW of a small hotel located in Brisbane (Australia). To determine the energy demands, numerical simulations were conducted using EnergyPlus software [33]. The building models used in the simulation were adapted from those published by the US Department of Energy (DOE). The small hotel consisted of a rectangular floor plant spanning 4 floors, with a net conditioning area of 4,013.6 m^2 and a roof area equivalent to 1,003.4 m^2 (**Figure *3***). The hotel's facades were oriented towards each of the four cardinal points, and the set point temperature was set at 25 ºC. Brisbane's climate is classified as Cfa in the Koppen-Geigger classification, which is characterized by relatively high temperatures and evenly distributed precipitation throughout the year.

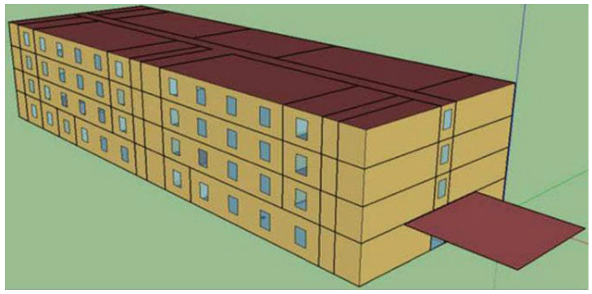

Figure 3. Small hotel simulated in EnergyPlus.

Prior to evaluate the integration of the heat pump system, an extensive energy analysis was conducted through a year-long numerical simulation to assess the building's energy requirements. Results were organized on a monthly basis. As shown in **Figure *4***, the cooling demand greatly surpasses the DHW demand in almost all the months, except for the months of June, July, and August in Brisbane, which is worth noting that are the winter months in Australia. January, December, and February have peak values of cooling demand, exceeding 30,000 kWh. In contrast, the DHW demand remained barely constant throughout the year, with an average of around 10,000 kWh per month. This finding emphasizes the importance of focusing on cooling demand in the design and implementation of the system.

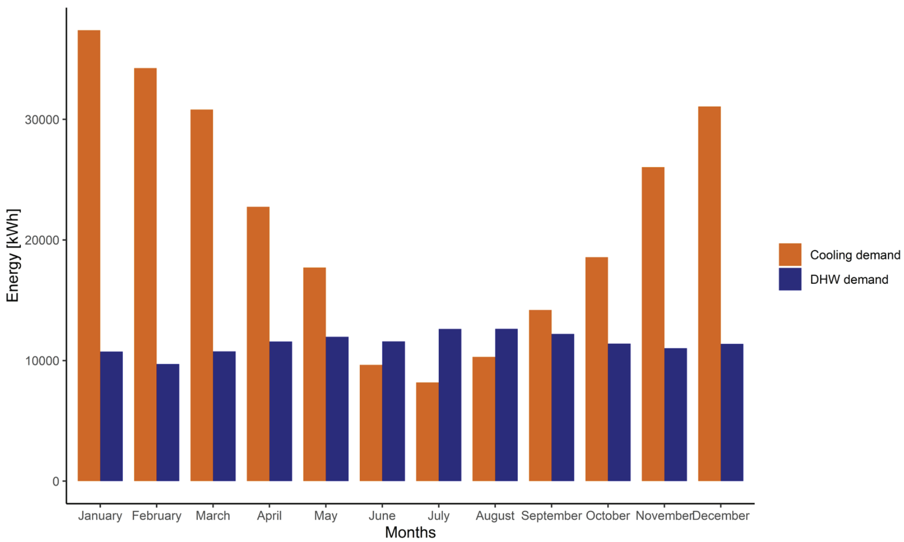

Figure 4. Yearly energy loads (cooling and DHW) in a small hotel in Brisbane.

2.3. Sizing

In the present study, an ad-RCE was considered which could switch its optical properties perfectly between each of the two modes. During the solar heating mode, the devices absorbed the totality of the incoming solar radiation while blocking the long-infrared radiation. During the radiative cooling mode, the device was able to reflect the totality of the incoming solar radiation, while perfectly emitting in the 7-14 μm range (atmospheric window). For the calculations, the following assumptions were also made:

- A steady-state model was used.
- The efficiency of the ad-RCE accounted for the conductive and convective losses.
- The tilt angle for the ad-RCE was assumed horizontal, maximizing the cold production.

The number of ad-RCE installed on the horizontal roof was determined by the required water flowrate for the case when the maximum cooling was demanded. 349 ad-RCE were used which corresponded to a total surface (A) of 698 m^2, representing 69.5% of the total roof area. All the months produced the required DHW from 11 a.m. to 12 a.m., except from April to September, when an additional hour was added (from 11 a.m. to 13 a.m.); the remaining hours were dedicated to radiative cooling mode.

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The heat pump was sized so that its evaporator could match the peak value of the year demand, which corresponded to 93.9 kW. The flowrate for the RC mode was set to 5.84 $kg \cdot s^{-1}$, while a lower flowrate of 4.17 kg/s was set for the solar heating mode. As both modes had different design parameters, the different flowrates were represented by a variable speed pump in the diagram.

2.4. Heating Mode

To determine the DHW production, the hourly Global Horizontal Irradiance (GHI) data was used along with the efficiency of the ad-RCE in solar collection mode, which is calculated using Eq. 1. This method of calculating solar collector efficiency is widely accepted and it takes into account various inefficiencies of the system, including optical and thermal losses. In line with previous research [15], an annual average efficiency η_{sc} of 0.6 was used in the calculations.

$$P_{solar,net}\left[\frac{W}{m^2}\right] = GHI \cdot \eta_{sc} \tag{1}$$

Once the average thermal power for each hour of the year was calculated, the next step was to determine the amount of energy produced. This was done by multiplying the average thermal power by the time step of 1 hour. To calculate the daily DHW production, the values obtained for each hour of the day were integrated over the course of a day. This allowed for a more accurate determination of the DHW production on a daily basis. The monthly production was then calculated by integrating the daily production values for a month. The formula used to calculate monthly production is presented in Eq. 2.

$$E_{solar,month}\left[\frac{Wh}{month}\right] = A \cdot \sum_{month} P_{solar,net} \cdot \Delta t \tag{2}$$

2.5. Cooling Mode

The maximum achievable RC power was determined using Eq. 3, which takes into account the infrared atmospheric radiation (Q_{atm}) and the absorbed radiation from the Sun (Q_{sun}) - both available in the weather data file- the radiation emitted on the surface of the ad-RCE (Q_s) – expressed in Eq.4., and the conductive and convective heat transfer. The approximation was made that the conductive and convective transfer were included in the efficiency parameter of the ad-RCE and therefore were assumed to be zero in Eq. 3. To determine the useful cooling power (Eq. 5), an efficiency of 60% was used.

$$Q_{net}[W] = [Q_s(T_s) - Q_{atm}(T_{atm}) - Q_{sun}(T_{sun}) - Q_{cond} - Q_{conv}] \cdot A \tag{3}$$

$$Q_s\left[\frac{W}{m^2}\right] = \varepsilon_s \sigma T^4 \tag{4}$$

$$P_{RC,net}[W] = Q_{net} \cdot \eta_{RC} \tag{5}$$

The temperature at the outlet of the ad-RCE (T_{out}) was calculated with the Eq. 6, where T_{in} represents the inlet temperature, $\dot{m}_{in}$ is the inlet flowrate and C_p is the water specific heat. The temperature at the inlet (T_{in}) was assumed to be equal to the pre-cooled temperature in the heat exchanger, which was set 5 ºC above the ambient temperature.

$$T_{out}[°C] = T_{in} - \frac{P_{RC,net}}{\dot{m}_{in} \cdot C_p} \tag{6}$$

A correlation was developed to model the performance of a water-to-water heat pump in TRNSYS. The correlation, expressed in Eq. 7, calculates the coefficient of performance (COP) of the heat pump as a function of the inlet temperature of the condenser (T_{cond}), assuming a constant evaporator temperature of 12 ºC.

$$COP = 7.45 - 0.116 \cdot T_{cond} - 0.00234 \cdot T_{cond}^2 + 0.0000852 \cdot T_{cond}^3 - 6.67 \cdot 10^{-7} \cdot T_{cond}^4 \tag{7}$$

The *COP* was calculated, for each time-step, both for the reference configuration and the ad-RCE+HP configuration. In the ad-RCE+HP configuration T_{cond} was set to be the same temperature as the outlet of the RCE (T_{out}). In the reference case, T_{cond} was assumed to be equal to the temperature at the outlet of the heat exchanger (5 ºC above the ambient temperature). The electrical energy of the compressor (E_{comp}) was determined using the energy supplied by the evaporator (E_{evap}, cooling demand) and the COP (Eq. 8).

$$E_{comp}[Wh] = \frac{E_{evap}}{COP} \tag{8}$$

3. Results and discussion

3.1. Heating Analysis

In **Figure *5***, the DHW production from the ad-RCE and the DHW demand of a small hotel are presented. As mentioned earlier, the DHW production is limited to two hours per day from April to September and one hour per day during the remaining months. On average, the system can produce 12,233 kWh of DHW per month. From April to September, the DHW production from the ad-RCE is higher than the hotel's demand. During the rest of the year, although the demand exceeds the potential production, the system has been sized to ensure a coverage of at least 80% of the demand, with actual coverage ranging from 85.25% to 95.52%.

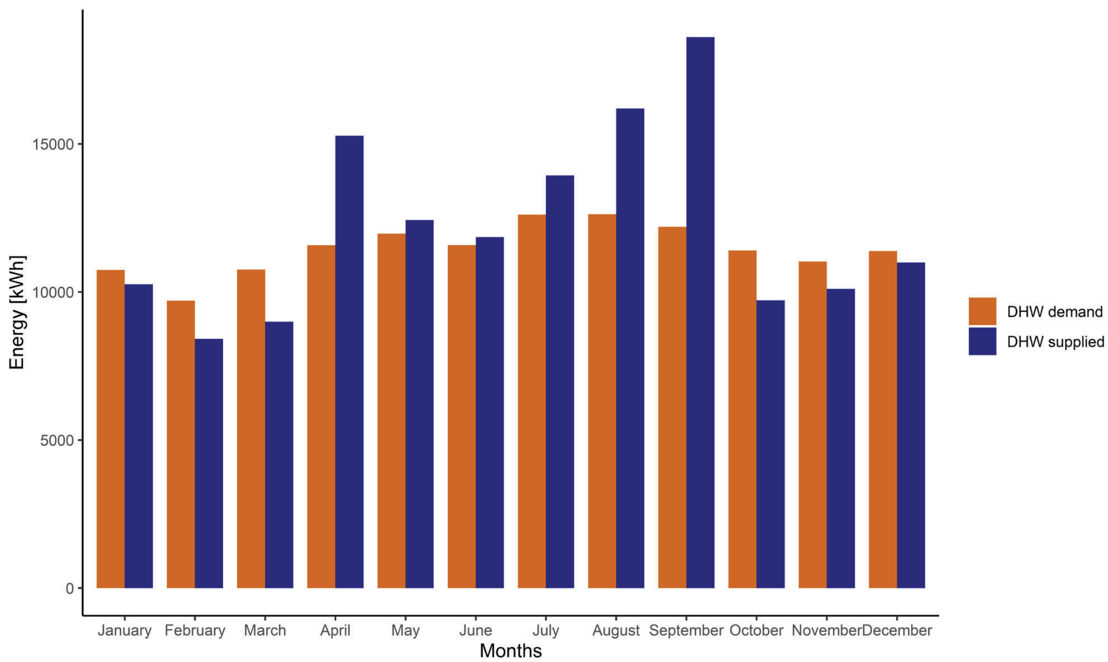

Figure 5. DHW energy loads in a small hotel in Brisbane and DHW produced in the ad-RCE.

3.2. Cooling Analysis

Through the implementation of radiative cooling, the average reduction in the temperature at the condenser inlet of the heat pump is found to be 1.4 ºC. The highest decrease is recorded in June with a value of 1.49 ºC, while the lowest reduction is observed in May, with a value of 1.31 ºC. A graphical representation of the monthly average temperature decrease at the condenser inlet is presented in **Figure *6***.

Additionally, the integration of the ad-RCE system with the heat pump leads to an increase in the yearly average coefficient of performance (COP) from 4.15 to 4.42 (**Figure *7***). The greatest improvements are observed during the winter months in Brisbane (**Figure *6.*** Temperature difference between inlet and outlet temperature in the ad-RCE over a year (orange) and improvement of the COP in the new configuration (blue). Cooling of water by more than 1.3 ºC is achieved in all the months.). The average increase in performance is 3.89%, with relatively constant values throughout the year ranging from 3.64% to 4.10%, as demonstrated in **Figure 6**.

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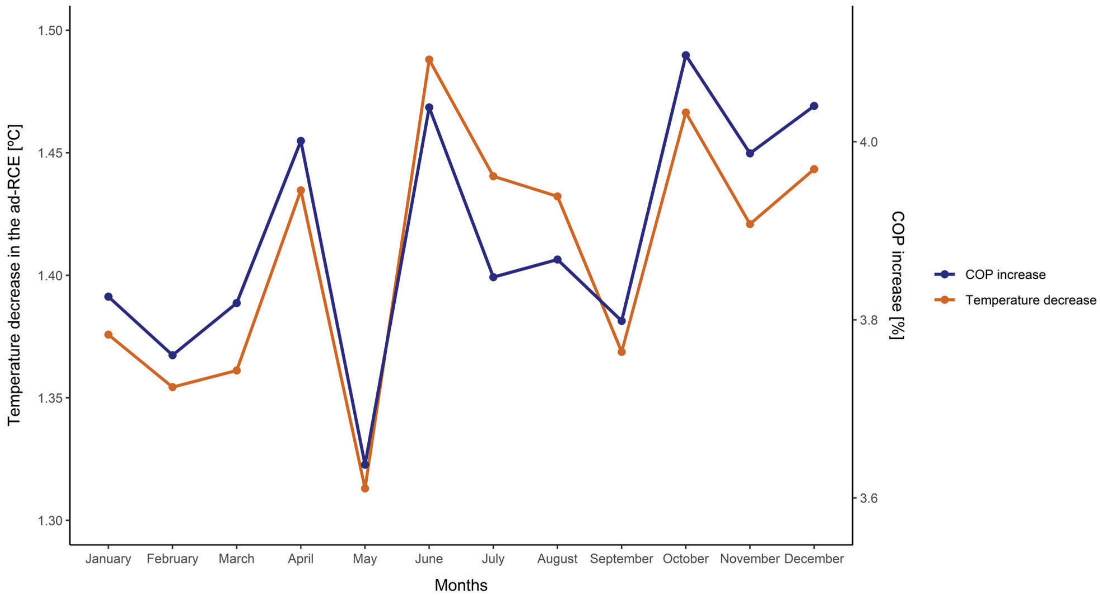

Figure 6. Temperature difference between inlet and outlet temperature in the ad-RCE over a year (orange) and improvement of the COP in the new configuration (blue). Cooling of water by more than 1.3 ºC is achieved in all the months.

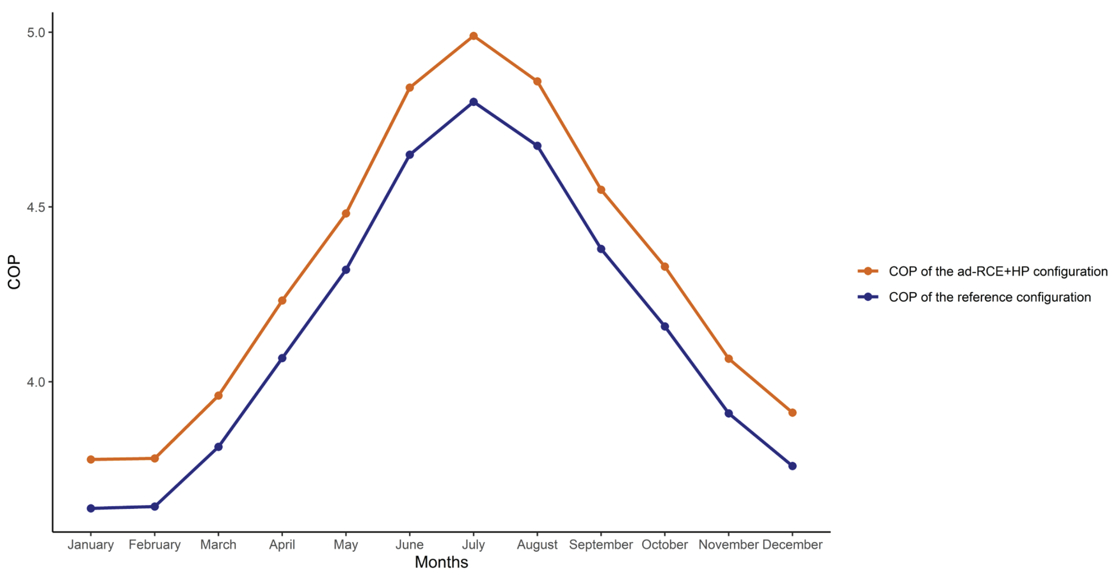

Figure 7. Evolution of the COP of the reference configuration and the studied configuration.

The integration of the ad-RCE with the heat pump system resulted in estimated total electricity savings of 2,533.3 kWh per year, which represents a reduction of 3.7%, the greatest savings are found during the summer months in Brisbane (**Figure *8***). This finding is consistent with literature reports which suggest that electricity consumption of a heat pump is reduced by approximately 3.5% for every degree Celsius reduction in the condenser's temperature [18]. In our study, the average reduction in the inlet temperature achieved with the ad-RCE is 1.4 ºC, resulting in a slightly lower condenser temperature. The monthly energy savings remained relatively constant throughout the year, ranging between 3.34% and 3.92%, which could contribute to a decrease in fossil fuel dependency of the system.

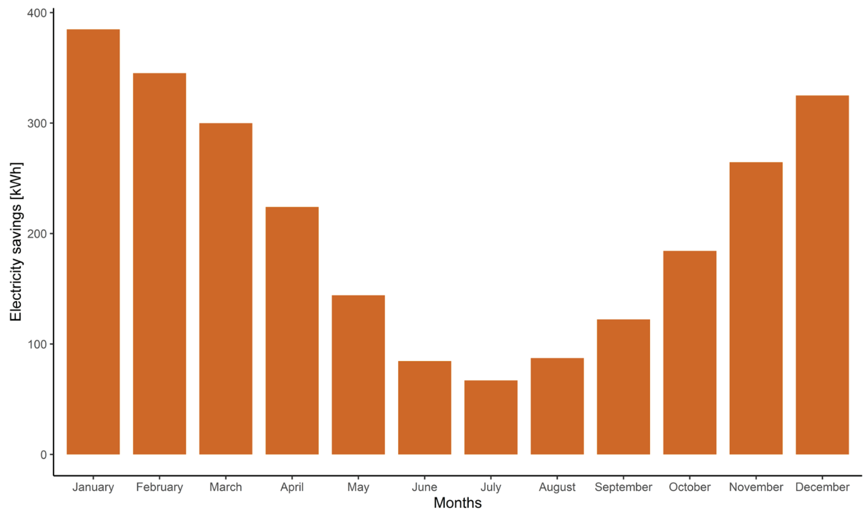

Figure 8. Electricity savings in the ad-RCE+HP configuration.

These results are preliminary estimates, and future work involves developing a validated ad-RCE model, similar to the RCE model previously developed [19] – and implementing effective control strategies to maximize energy savings. Additionally, a promising approach to improve energy savings would be to develop an efficient control system that anticipates the energy demand and operates the ad-RCE during optimal radiative cooling conditions. Another potential strategy is to integrate storage tanks, such as sensible heat water tanks and latent heat phase change materials tanks, to store the cold produced by the ad-RCE during favourable radiative cooling periods for later use during less favourable weather conditions. These strategies will be investigated in detail in the following stages of our research.

4. Conclusions

In this paper, we present a numerical analysis of the performance improvement of a water-to-water compression heat pump when combined with an adaptive Radiative Collector and Emitter (ad-RCE). The ad-RCE is an advancement of the Radiative Collector and Emitter (RCE), a device capable of producing hot water by solar thermal collection during the day and cold water by radiative cooling at night. The ad-RCE is a redesigned version of the RCE that offers users the option of selecting either daytime radiative cooling mode or solar collection mode, depending on the daily energy demands. By coupling the heat pump with the ad-RCE, higher COPs can be achieved compared to conventional heat pump reference systems. Our results demonstrate an average yearly COP improvement of 3.89% in a small hotel in Brisbane (Australia), leading to annual electricity savings of 3.70%.

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Nomenclature

$\boldsymbol{A}$	Radiator/Absorber surface	(m^2)
$\boldsymbol{E_{solar,month}}$	Net monthly heating energy thanks to solar collection	($Wh/m^2/month$)
$\boldsymbol{E_{cool,month}}$	Net monthly cooling energy thanks to RC	($Wh/m^2/month$)
$\boldsymbol{Q_{atm}}$	Absorbed infrared radiation from atmosphere	(W)
$\boldsymbol{Q_{cond}}$	Conduction heat power	(W)
$\boldsymbol{Q_{conv}}$	Convective heat power	(W)
$\boldsymbol{Q_{net}}$	Net balance radiation power	(W)
$\boldsymbol{Q_{s}}$	Infrared radiation power emitted by a radiative surface	(W)
$\boldsymbol{Q_{sun}}$	Incident solar radiation power	(W)
$\boldsymbol{T_{atm}}$	Ambient and atmosphere temperature	(K)
$\boldsymbol{T_{s}}$	Surface temperature	(K)
$\boldsymbol{\sigma}$	Stefan-Boltzmann's constant: $5.6704 \cdot 10^{-8}$	($W/m^2 \cdot K^4$)
$\boldsymbol{\varepsilon_{s}}$	Surface emissivity	(-)

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