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CO₂-Assisted Compression-Adsorption Hybrid for Cooling and Desalination

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Abstract

This paper presents a novel compression-adsorption hybrid that symbiotically combines adsorption and CO₂ compression cooling devices. The seemingly low efficiency of each cycle individually is overcome by an amalgamation with the other. Hence, both heat and water vapour refrigerant mass are recovered for continuous cooling and desalination. Two different configurations are presented. The first configuration deals with a two-stage heat recovery system. At the first stage, heat is recovered from the compressed carbon dioxide to drive the adsorption device. The second stage heat recovery system internally exchanges heat between the low pressure and high pressure refrigerants of the CO₂ cycle. The second configuration is proposed with an additional third-stage heat recovery from the gas cooler to the high pressure evaporator of the adsorption cycle. The water vapour mass is recovered from bed-to-bed adsorption at relatively higher pressure. A detailed thermodynamic framework is presented to simulate the performances in terms of COP (coefficient of performance), SCP (specific cooling power), SDWP (specific daily water production), PR (performance ratio) and OCR (overall conversion ratio). It is found that the overall COP is improved by more than 60% as compared to the conventional CO₂ cycle, and in addition, the system generates 12.7 m³ of desalinated water per tonne of silica gel per day as extra benefits. Furthermore, both the heat and mass recoveries improve the overall conversion ratio, which is almost double as compared to the conventional CO₂ cycle.

Keywords: Adsorption, CO₂ cycle, Cooling, Desalination, Heat recovery, COP, OCR.

1. Introduction

Global energy consumption was increased remarkably from 371×10^5 to 850×10^5 TWh in the last century due to rapid population growth and technological advancement, with 15% of the global electrical energy being consumed for cooling applications. However, the usage of renewable energy is being increased. Fossil fuels provide most (about 78%) of the consumed energy [1-2] resulting in a significant increase in global carbon emissions [3]. The method of waste heat recovery can offer a sustainable solution to this problem as it requires no additional burning of fossil fuels. The conventional vapour compression cooling (VCC) system contradicts the concepts of sustainability as it consumes electrical energy and uses HCFC-based refrigerants. Hence, the VCC system significantly impacts on global warming potential [4]. The Montreal and Kyoto protocols stipulated the phasing out of CFC and HCFC refrigerants and promoted the policies for sustainability and global warming potential (GWP) reduction [5-6]. Table 1 shows a comparison of ODP (ozone layer depletion potential) and GWP for various refrigerants. It can be seen that sustainability in the fields of refrigeration and cooling implies the unavoidable use of natural refrigerants such as ammonia, water and carbon dioxide [7, 8]. However the aforementioned natural substances have their own drawbacks. For example ammonia is toxic and flammable [9]. In contrast, carbon dioxide is considered as environmentally friendly (GWP = 1 and ODP = 0) and non flammable [10]. Therefore, CO₂ can be employed as a suitable refrigerant for refrigeration or heat pump purposes. It should be noted here that the CO₂-assisted cooling cycle has some drawbacks due to its low critical temperature (304 K) and higher operating pressure. The operating problems of the CO₂ cycle due to low critical temperature can be effectively circumvented by transcritical operation and replacing the condenser with a gas cooler [11]. The transcritical system is operated at extremely high pressure and temperature, which eventually decreases its COP [12]. If the system is used in a tropical climate where the ambient temperature ranges

from 25 to 40°C, a significant amount of energy supplied to the system will be wasted through the gas cooling process. The COP of the CO₂ based VCC can be improved by utilising the heat energy generated from the CO₂ compression process with the rejection of heat from the gas cooler effectively. The heat removal is possible by supplying cooling water to the gas cooler, which can be performed either by circulating cooling water through the tubes or by evaporation method. It should be noted here that the heat energy removal from the pressurized CO₂ refrigerant is varied from 60°C to 90°C. This low grade heat can be utilised to drive a thermally-driven cycle for cooling and desalination applications.

The utilisation of thermal energy to drive an adsorption device has drawn substantial research attentions. For example, Ali and Chakraborty in 2015 proposed an adsorption cooling system for automotive air-conditioning using exhaust energy from the engine [13]. El-Sharkawy et al. [14] presented a theoretical investigation on the performance of solar powered silica gel/water based adsorption cooling system for the climate conditions of the Middle East region. In another study Liu and Leong [15] analytically proposed a cogeneration system that incorporates a solid oxide fuel cell and a zeolite/water adsorption chiller to investigate the performance of the combined system for different operating conditions and design parameters. Ng et al. [16] investigated the performance of an adsorption cooling and desalination system with low temperature heat input from flat plate solar collectors.

Recent published research on the transcritical carbon dioxide cycle show that waste heat generated from the compressor outlet can also be used to drive an adsorption cooling system. Table 2 shows a list of experimental studies for the compressor discharge pressure, and temperature with gas cooler outlet temperature. The discharge pressure of carbon dioxide cycle varies from 75 to 150 bar, whereas the discharge temperature ranges between 100 - 150°C. It reveals that an adsorption cooling and desalination system can be driven by waste heat from the CO₂ cycle. From experimental data [17-22] as furnished in Table 2, it can be

seen that the adsorption bed can be thermally compressed by the thermal energy from the compressor outlet.

For better explanation, we propose here two configurations. The first configuration deals with the heat recoveries between the compressed CO₂ and the heating source of adsorption cycle (four-bed, two-evaporator systems). The second configuration comprises i) the heat extraction from the compressor of CO₂ outlet to generate heating power for adsorption cycle, and ii) the supply of heat from the gas cooler to the evaporator of adsorption cycle. The second arrangement provides two benefits; one is for the production of more desalting water and the other is for decreasing the gas cooler outlet temperature. Therefore, the COP of CO₂ compression chiller is improved. The internal heat recovery exploits the benefits of i) longer adsorption period, ii) shorter desorption period and iii) continuous production of cooling and desalination.

This study aims to amalgamate CO₂ and adsorption cycles as a single unit for the utilisation of waste heat to produce more cooling energy and desalinated water. The energetic performances of the overall co-generation system are calculated in terms of coefficient of performance (COP), specific cooling power (SCP), specific daily water production (SDWP) and overall conversion ratio (OCR).

2. Description of hybrid compression-adsorption cooling and desalination system

Figs. 1 and 2 show the schematic diagrams of the hybrid compression-adsorption cycles (CAC). Here two schemes of the proposed cooling cum desalination systems are presented. The CAC comprises a transcritical carbon dioxide compression cooling system, a four-bed two-evaporator adsorption cooling and desalination cycle and two heat recovery systems. The main components of the carbon dioxide cooling system are an evaporator (Evp-1), a

compressor (Comp), a gas cooler (GC) and an expansion valve (Exp). The refrigerant carbon dioxide changes its phase from liquid to vapour due to latent heat of evaporation generated from the supply of chilled water through the copper tubes in Evp-1. Thereafter, the refrigerant vapour is compressed in the compressor. Due to compression, the temperature and pressure of CO₂ are increased. After compression, the thermal energy contained in refrigerant has to be purged before it is refluxed back to the evaporator through the expansion valve. The conventional CO₂-assisted cooling system employs a gas cooler to release the heat to the sink. The heat generated from the transcritical CO₂ cycle can be utilised as the driving heat source to run the thermal compressor of the adsorption system.

Configuration 1 as shown in Figure 1 proposes a two-stage heat recovery system. At first, the superheated carbon dioxide flows through the tubes in the water heating system (GN), and the heat generated from CO₂ compression is transferred into the water. The hot water is stored in the GN. At the end of the desorption process of adsorption cycle, the hot water temperature decreases and returns to the GN, where it absorbs heat from the carbon dioxide again for driving the next adsorption cycle. The GN provides thermal inertia to the proposed hybrid system when the heat energy requirement is varied. The second stage heat recovery (PH) is a concentric tube type counter flow heat exchanger. Due to cooling load at Evp-1, CO₂ is evaporated at low temperature and pressure, and flows through the tubes in PH. On the other hand, the hot CO₂ from GN goes through the annular side of PH. Due to the temperature difference between these two refrigerant streams, the low temperature CO₂ vapour generating from Evp-1 becomes superheated before entering the compressor. Therefore, the temperature of CO₂ refrigerant at compressor inlet is increased without consuming additional electrical energy for the mechanical compressor. At the same time, the carbon dioxide temperature at the inlet of the gas-cooler is decreased, which decreases the heat rejection rate from the gas cooler (GC). In the previous studies, Sanchez et al. [22, 23]

found that the superheating of refrigerant decreased its mass flow rate, which increased the power consumption of the compressor increases. Therefore, the COP of the CO₂ cycle was also decreased.

Configuration 2 as shown in Figure 2 proposes a three stage heat recovery system. The first and the second stage heat recovery for both Configuration 1 and Configuration 2 are the same. The third stage heat recovery system placed at the outlet of the GC exchanges heat between CO₂ (flowing through tubes) and saline water of Evp-3 (Figure 2). Due to additional heat transfer, the temperature of carbon dioxide is further decreased before passing through the expansion valve. The pressure of Evp-3 is increased. Therefore, the cooling capacity of the carbon dioxide cycle and the SDWP of adsorption cycle will be improved. To exploit the benefits of longer adsorption and shorter desorption periods for continuous cooling and desalination, a four-bed two-evaporator adsorption cooling and desalination system is proposed. It consists of two evaporators (i.e. Evp-2 and Evp-3), four adsorption beds (Bed-1, Bed-2, Bed-3 and Bed-4) and one condenser (Cond). These are shown in both Figs. 1 and 2. The electromagnetic vacuum valves connect the adsorption beds with the condenser and the evaporators. Both condenser and evaporators are shell-and-tube heat exchangers, whereas the adsorption beds comprise finned-tube heat exchanger. The adsorbent materials are housed between the fins of the heat exchangers. Raw seawater is charged in Evp-2 and Evp-3. For continuous cooling purposes, the low pressure evaporator or Evp-2 is operated at the pressure of ≈ 1 kPa. The chilled water inlet temperature is maintained at 14.8°C. Due to cooling effects, a small amount of desalination effect is obtained from Evp-2. For Configuration 1, water is circulated through the tubes inside Evp-3 at 30 °C as additional cooling loads. On the other hand, CO₂ refrigerant delivered from the GC is passed through the tubes in Evp-3 (for Configuration 2). Evp-3 is operated at relatively higher pressure as compared to Evp-2 (Figure 2).

The cyclic operation of four-bed two-evaporator system starts with the connection of beds to Evp-2 for the low pressure adsorption. Followed by this process, the adsorption bed connects with Evp-3 for high pressure adsorption. The duration of high and low pressure adsorption processes are considered equal, i.e., $t_{ads}^{lp} = t_{ads}^{hp}$. As the adsorption process is exothermic, cooling water is circulated through the adsorption bed to maintain the lowest possible temperature and pressure. After the adsorption phase, the valve connecting the bed and Evp-3 is closed and the adsorption bed is preheated. During the preheating/pre-cooling period, the bed is isolated from the condenser and the evaporator. In addition, the hot water from GN is circulated through the bed. As a result, the pressure of the bed increases. The condenser (Cond) is connected to the corresponding sorption beds during desorption period by opening the valves. Hot water is supplied through the desorption bed to regenerate the water vapour. The desorbed water vapour then goes to the condenser and changes its phase to liquid, which is collected as fresh water in potable water tank (PWT).

In this paper, an operating timetable of the hybrid system is proposed to achieve the shortest possible desorption and longer adsorption periods for continuous cooling and desalination effects. The desorption time is shortened to restrict the connection of more than one desorption beds with the condenser. Bed-1 is started when $t = 0$, i.e., there is no time lag. The cyclic operation starts with three different lag times for Bed-2, Bed-3 and Bed-4 such that more than one switching period is not occurred at the same time. The constraints on the adsorption time (t_{ads}) and desorption time (t_{des}) are required to ensure the homogenous cyclic operation. The proposed timetable is shown in Figure 3. For adequate precooling and preheating at the given heating/cooling water flow rate, the switching time is considered 40 s according to manufacturer's recommendation [24]. The adsorption periods for a given desorption period are given by

$$(t_{ads}^{lp} + t_{ads}^{hp}) + 2t_{sw} = 3t_{des} \quad (1)$$

where, t_{ads}^{lp} , t_{ads}^{hp} , t_{sw} and t_{des} indicate the low pressure adsorption period, the high pressure adsorption period, the switching period for precooling or preheating and the desorption period. The full cycle time is expressed by $t_{cycle} = t_{ads}^{lp} + t_{ads}^{hp} + 2t_{sw} + t_{des}$. The time schemes of adsorption based cooling and desalination are shown in Fig. 3.

3. Thermodynamic Framework of Hybrid Cooling and Desalination System

A thermodynamic framework of the proposed compression-adsorption hybrid cooling and desalination system is developed from the rigour of thermophysical properties of CO₂ and experimentally measured adsorption isotherms and kinetics data of a silica gel and water system. This thermodynamic modelling connects the mass and energy balances in all major components of the system.

3.1 Carbon dioxide cooling cycle

3.1.1 Evaporator

The evaporator of carbon dioxide cooling cycle is a shell-and-tube heat exchanger. The chilled water is circulated through the tube sides of the evaporator and delivers the heat of evaporation to the refrigerant carbon dioxide. As a result, carbon dioxide evaporates from the shell side of the evaporator. The energy balance of the evaporator is given by

$$\left[(Mc_p)_{CO_2}^{Evap,1} + (Mc_p)_{HX}^{Evap,1} \right] \frac{dT^{Evap,1}}{dt} = (\dot{m}c_p)_{chw}^{Evap,1} (T_{Evap,1}^{chw,in} - T_{Evap,1}^{chw,out}) - \dot{m}_{CO_2} h_g + \dot{m}_{CO_2} h_{out}^{exp}, \quad (2)$$

where $(Mc_p)_{CO_2}^{Evap,1}$ and $(Mc_p)_{HX}^{Evap,1}$ are the heat capacities of the heat exchanger tube and the carbon dioxide of the evaporator. The first term on the right hand side defines the cooling capacity of carbon dioxide cycle which produces from the sensible cooling of the chilled water, the second term is the latent heat of evaporation of carbon dioxide that leaves the evaporator, and the last term denotes the total enthalpy of liquid carbon dioxide refluxed from

the gas cooler through the expansion valve. The chilled water temperature at different lengths of tube is calculated by applying the energy balance on the water control volume,

$$(\rho c_p)_{Ev,1}^{chw} \frac{\partial T_{Ev,1}^{chw}}{\partial t} = -(\rho c_p)_{Ev,1}^{chw} \frac{\partial T_{Ev,1}^{chw}}{\partial z} + \lambda_{chw} \frac{\partial^2 T_{Ev,1}^{chw}}{\partial z^2} - \left(\frac{UA}{V} \right)_{Ev,1}^{chw} (T_{Ev,1}^{chw} - T^{Ev}) \quad (3)$$

where u , ρ and V defines the velocity, the density and the volume of chilled water. z is the distance along the direction of chilled water flow in the tube. Boundary conditions of the equation are $T_{Ev,1}^{chw}(z=0, t) = T_{Ev,1}^{chw, in}$ and $T_{Ev,1}^{chw}(z=L^{tube}, t) = 0$. L^{tube} is the length of the tube.

3.1.2 Compressor

In one of the previous studies, semi-hermetic compressor modelling is introduced for handling the superheat (ΔT) [23]. The higher ΔT increases the power consumption to the compressor and decreases the COP. From experimental data [23], a higher ΔT is observed experimentally at lower evaporating temperature. Correlations for calculating volumetric and compression efficiencies, the increase in superheating temperature (ΔT) and discharge temperature are also provided for evaporation temperatures ranging from -17°C to 0°C [23]. From experimental trends [23], it is also observed that the superheating ΔT of CO_2 decreases at higher evaporating temperature. Therefore the lower ΔT improves the COP. In the present analysis, the evaporator temperature is maintained approximately at 8°C or 10°C for air conditioning applications. Therefore, the superheating effects at the compressor inlet can be neglected for simplicity.

The mass flow rate of the compressor is expressed by

$$\dot{m}_{\text{CO}_2} = V_s \eta_v N \rho_{suction} \quad (4)$$

where η_v is the volumetric efficiency, $\dot{m}_{\text{CO}_2}$ indicates the refrigerant flowrate, $\rho_{suction}$ defines the density of the refrigerant inlet to the compressor, V_s and N are the swept

volume and the RPM of the compressor. The volumetric efficiency of the compressor is given by [25],

$$\eta_v = 0.9207 - 0.0756r + 0.0018r^2, \quad (5)$$

where the compression ratio (r) stands for the ratio of suction and discharge pressures of the compressor, as written as, $r = P_{Comp}^{in} / P_{Comp}^{out}$, where P_{Comp}^{in} and P_{Comp}^{out} are the suction and the discharge pressures of the compressor. The isentropic efficiency (η_{is}) of the compressor is calculated as [25],

$$\eta_{is} = -0.26 + 0.7952r - 0.2803r^2 + 0.0414r^3 - 0.0022r^4. \quad (6)$$

The isentropic efficiency can also be defined as $\eta_{is} = \frac{h_{Comp}^{act} - h_{Comp}^{in}}{h_{Comp}^{ideal} - h_{Comp}^{in}}$. Here h_{Comp}^{in} is the enthalpy of refrigerant at suction temperatures and pressures of compressor, h_{Comp}^{ideal} defines the enthalpy of refrigerant at discharge temperature and pressure considering no adiabatic loss, whereas h_{Comp}^{act} indicates the enthalpy of discharged refrigerant considering the isentropic efficiency.

The work done by the compressor can be estimated as

$$W_{comp} = \dot{m}_{CO_2} (h_{Comp}^{act} - h_{Comp}^{in}) \quad (7)$$

Power consumed by the compressor is calculated by,

$$W_{comp}^{is} = \dot{m}_{CO_2} (h_{Comp}^{ideal} - h_{Comp}^{in}) \quad (8)$$

3.1.3 Gas cooler (GC)

The GC is a concentric tube counter-flow heat exchanger, where CO_2 flows through the annulus and the cooling water flows along the inner tube. Therefore, water absorbs heat from CO_2 and transfers the energy to the surroundings. The energy balance is given by

$$\left[(M c_p)_{CO_2}^{GC} + (M c_p)_{HX}^{GC} \right] \frac{dT^{GC}}{dt} = \dot{m}_{CO_2} (h_g^{CO_2, in} - h_g^{CO_2, out}) - (\dot{m} c_p)_{cw}^{GC} (T_{out}^{cw} - T_{in}^{cw}) \quad (9)$$

where $(Mc_p)_{CO_2}^{GC}$ and $(Mc_p)_{HX}^{GC}$ are the heat capacities of carbon dioxide and the heat exchanger tubes of the gas cooler. The first term on the right hand side defines the amount of heat rejected by carbon dioxide and the second term represents the heat extracted by the cooling water. The outlet temperature of cooling water from gas cooler is calculated by,

$$(\rho c_p)_{cw}^{GC} \frac{\partial T_{cw}^{GC}}{\partial t} = -(u \rho c_p)_{cw}^{GC} \frac{\partial T_{cw}^{GC}}{\partial x} + \lambda_{cw}^{GC} \frac{\partial^2 T_{cw}^{GC}}{\partial x^2} - \left(\frac{UA}{V} \right)_{cw}^{GC} (T_{cw}^{GC} - T^{GC}) \quad (10)$$

where U and A are the overall heat transfer coefficient and surface area, respectively. u , ρ and V define the velocity, density and volume of the cooling water in tubes. x is the distance along the direction of cooling water flow in the tube. The boundary conditions are

$$T_{cw}^{GC}(x=0, t) = T_{cw}^{in} \text{ and } \frac{\partial T_{cw}^{GC}}{\partial x}(x=L^{tube}, t) = 0. \quad L^{tube} \text{ is the length of the gas cooler tube.}$$

3.1.4 Expansion Device

The expansion process of carbon dioxide cycle is considered as an isenthalpic process. The energy balance of the expansion device can be written as

$$h_{out}^{exp} = h_{in}^{exp} = h_g^{CO_2, out}(T_{out}^{CO_2}, P_{out}^{CO_2}) \quad (11)$$

3.1.5 Pressure Losses in the Carbon Dioxide Cycle

The pressure drops in the generator and the gas cooler are calculated from the Darcy-Weisbach equation for single-phase flow [26],

$$\Delta P = f \cdot \frac{L}{d_i} \cdot \frac{\rho V^2}{2} \quad (12)$$

where f is the Darcy friction factor, L is the length of the heat exchanger tube, d is the hydraulic diameter of the tube, ρ is the density of the fluid and V is the mean flow velocity. For the generator and the pre-heaters, the Darcy friction factors are calculated to be 0.02 and 0.03, respectively depending on the tube dimensions and mass flow rates of the refrigerant.

The Darcy friction factor is calculated by $f = 0.25 \left[\log \left(\frac{e}{D} + \frac{5.74}{\text{Re}^{0.9}} \right) \right]^{-2}$, where $\text{Re} \left(= \frac{\rho V D}{\mu} \right)$

is the Reynolds number, ρ is the density and μ defines the viscosity. D indicates the inner diameter of the tube and e ($= 0.0015$ mm) is the surface roughness of the pipe.

3.2 Heat Recovery System

3.2.1 Generator

The generator is the first stage heat recovery of the hybrid system which extracts heat from the superheated carbon dioxide of CO₂ cycle. The energy balance of the generator (thermal storage system) is given by

$$\left[(mc_p)^{hw} + (mc_p)^{HX} \right] \frac{dT^{GN}}{dt} = \dot{m}_{CO_2} (h_{in}^{GN,CO_2} - h_{out}^{GN,CO_2}) - \dot{m}_{hw} (h_{out}^{GN,hw} - h_{in}^{GN,hw}), \quad (13)$$

where $(Mc_p)^{hw}$ and $(Mc_p)^{HX}$ are the heat capacities of the water stored in the generator and the heat exchanger. The first term on the right hand side defines the heat addition by carbon dioxide and the second term is the heat absorption by water. The outlet temperature of CO₂ from the GN is calculated as

$$T_{out}^{GN,CO_2} = T^{GN} + (T_{in}^{GN,CO_2} - T^{GN}) \exp \left\{ \frac{-(UA)^{GN}}{(\dot{m}c_p)_{CO_2}} \right\} \quad (14)$$

where U is the overall heat transfer coefficient of the heat exchanger of the generator. It depends on (i) the flow rates of carbon dioxide and water, (ii) the size of the generator and (iii) the convective heat transfer coefficients at the carbon dioxide and water sides.

3.2.2 Preheater

The second stage heat recovery system is a preheater (PH), and internally exchanges heat between the low pressure saturated carbon dioxide vapor outlet of Evp-1 and the high pressure superheated carbon dioxide outlet of the generator. The high pressure carbon dioxide

flows through the annulus and the low pressure carbon dioxide flows in the reverse direction along the inner tube. The energy balance of the preheater is given by

$$\left[(Mc_p)_{CO_2}^{PH} + (Mc_p)_{HX}^{PH} \right] \frac{dT^{PH}}{dt} = \dot{m}_{CO_2} (h_{PH,hp}^{CO_2,in} - h_{PH,hp}^{CO_2,out}) - \dot{m}_{CO_2} (h_{PH,lp}^{CO_2,out} - h_{PH,lp}^{CO_2,in}), \quad (15)$$

where $(Mc_p)_{CO_2}^{PH}$ and $(Mc_p)_{HX}^{PH}$ are the heat capacities of carbon dioxide and heat exchanger tubes of the preheater. The first term on the right hand side of Eq. (15) defines the amount of heat rejected by the high pressure carbon dioxide stream, and the second term represents the heat extracted by the low pressure carbon dioxide stream. The outlet temperature of low pressure carbon dioxide is given by

$$T_{out}^{GN,CO_2} = T^{GN} + (T_{in}^{GN,CO_2} - T^{GN}) \exp \left\{ \frac{-(UA)^{GN}}{(\dot{m}c_p)_{CO_2}} \right\}. \quad (16)$$

3.3 Dual-evaporator adsorption cooling and desalination system

3.3.1 Evaporators

The adsorption system consists of two evaporators. Sea water is evaporated from Evp-2 and Evp-3. The evaporation from Evp-2 occurs at low pressure due to external cooling load of low temperature heat source and is adsorbed on to the porous adsorbent packed in Bed-1, Bed-2, Bed-3 or Bed-4 according to the operating principles of four-bed-two-evaporator system as shown in Figure 3. On the other hand, Evp-3 works at higher pressure due to load from the high temperature heat source e.g. the circulation of water at 30°C (for Configuration 1) or the flow of CO₂ from gas cooler (Configuration 2). The temperatures of both evaporators (Evap-2 and Evap-3) are calculated by the following energy balance equations:

$$\left[M_{sw} c_p (T_{sw}, X_{sw}) + M_{HX} c_p (T_{HX}, X_{HX}) \right] \frac{dT^{Evap,2}}{dt} = \dot{m}_p^{chw} (T_{chw}^{in} - T_{chw}^{out}) - M^{ads} h_{fg} (T_{sw}, X_{sw}) \frac{dq}{dt}, \quad (17)$$

$$+ M^{ads} h_f (T_{sw}, X_{sw}) (\dot{m}_{sw2}^{in} - \dot{m}_{br2}^{out})$$

$$\begin{aligned} & \left[M_{sw} c_p (T_{sw}, X_{sw}) + M_{HX} c_{p, HX} \right] \frac{dT^{Ev,3}}{dt} = \dot{m} c_p^{cw} (T_{cw/CO_2}^{in} - T_{cw/CO_2}^{out}) - M^{ads} h_{fg} (T_{sw}, X_{sw}) \frac{dq}{dt}, \\ & + M^{ads} h_f (T_{sw}, X_{sw}) (\dot{m}_{sw3}^{in} - \dot{m}_{br3}^{out}) \end{aligned} \quad (18)$$

where the first and the second terms of the left hand side are the heat capacities of sea water and heat exchanger of the evaporators for Eqs. (17) and (18). On the right side, the first term represents the sensible energy for the evaporation supplied by the flow of warm water or CO₂, the second term indicates the amount of latent heat generated by the evaporation of sea water and the third term defines the summation of sensible heat added by the feed sea water and the heat released by the concentrated discharged brine solution. The term $\frac{dq}{dt}$ shows the rate of water vapour uptake by the adsorbent materials. The rate of adsorption or desorption $\frac{dq}{dt}$ as a function of pressure and temperature is calculated by the linear driving force model [27, 28], i.e.,

$$\frac{dq}{dt} = \frac{D_{so} \exp\left(\frac{-E_a}{RT}\right)}{R_p^2} (q^* - q), \quad (19)$$

where D_{so} is a pre-exponential diffusion factor, E_a represents the activation energy, R is the universal gas constant, R_p is the average radius of the adsorbent grains and q is the instantaneous water vapor uptake and offtake in the adsorption beds. q^* is the amount of adsorbed adsorbate under equilibrium condition, and is calculated from the Dubinin-Astakov (D-A) equation as

$$q^* = q^o \exp\left\{-\left(\frac{RT}{E} \ln \frac{P_s}{P}\right)^n\right\} \quad (20)$$

where, q^o is the limiting uptake of the adsorbate molecules, E indicates the characteristic energy, P_s is the saturation pressure, P defines the pressure of the adsorption/desorption bed

and n generally stands for the surface-structural heterogeneity factor [28-31]. The chilled water temperature for Configuration 1 at different lengths of evaporator tube is calculated by

$$(\rho c_P)_{chw} \frac{\partial T^{chw}}{\partial t} = -(\rho c_P)_{chw} \frac{\partial T^{chw}}{\partial z} + \lambda_{chw} \frac{\partial^2 T^{chw}}{\partial z^2} - \left(\frac{UA}{V} \right)_{chw} (T^{chw} - T^{Evap}) \quad (21)$$

where u , ρ and V define the velocity, the density and the volume of the chilled water. z is the distance along the direction of chilled water flow in the tube. The boundary conditions of the

equation (20) are $T^{chw}(z=0, t) = T_{chw}^{in}$ and $\frac{dT^{chw}}{dz}(z=L^{tube}, t) = 0$. L^{tube} is the length of the

tube. The CO_2 outlet temperature of Evp-3 for Configuration 2 is calculated as

$$T_{Evap-3}^{CO_2, out} = T^{Evap} + (T_{Evap-3}^{CO_2, in} - T^{Evap-3}) \exp \left\{ \frac{-(UA)^{Evap-3}}{(\dot{m} c_P)_{CO_2}} \right\}, \quad (22)$$

where U and A are the overall heat transfer coefficient and surface area of the heat exchanger of Evp-3, respectively. Due to evaporation, the sea water level in both high and low pressure evaporator decreases and the concentration of the brine solution increases. It is assumed that the brine solution is discharged when the concentration of the solution is 1.5 times greater than that of the feed sea water [32]. Once the brine solution is discharged, the sea water is loaded again. The mass balances of sea water for Evaporators 2 and 3 (Evp-2 and Evp-3) are given by

$$\frac{dm^{sw2}}{dt} = \dot{m}_{sw2}^{in} - \dot{m}_{br2}^{out} - M^{ads} \frac{dq}{dt} \quad (23)$$

$$\frac{dm^{sw3}}{dt} = \dot{m}_{sw3}^{in} - \dot{m}_{br3}^{out} - M^{ads} \frac{dq}{dt} \quad (24)$$

where, m^{sw} is the amount of sea water in the evaporator, $\dot{m}_{sw}^{in}$ is the rate of feed sea water and $\dot{m}_{br}^{out}$ is the mass flow rate of concentrated brine rejected from the evaporator. The salinity of the sea water in the evaporator is calculated by the salt balances, i.e.,

$$m^{sw2} \frac{dX_{sw2}}{dt} = \dot{m}_{sw2}^{in} X_{sw2}^{in} - \dot{m}_{br2}^{out} X_{sw2} - X_D M^{ads} \frac{dq}{dt} \quad (25)$$

$$m^{sw3} \frac{dX_{sw3}}{dt} = \dot{m}_{sw3}^{in} X_{sw3}^{in} - \dot{m}_{br3}^{out} X_{sw3} - X_D M^{ads} \frac{dq}{dt} \quad (26)$$

where, X_{sw2} and X_{sw3} define the salinities of seawater in Evp-2 and Evp-3. The left hand side indicates the rate of change of salinity. For both Eqs. (25) and (26), the first term of the right hand side indicates the addition of salt by the feed sea water, the second term shows the removal of salt by the discharged brine and the final term is the rejection of evaporated sea water.

3.3.2 Adsorption Beds

The evaporated sea water is adsorbed into the pores of the solid adsorbent of the adsorption beds. As the adsorption beds are connected to the evaporator during the adsorption process, the pressure in the adsorption bed increases and become nearly equal to the evaporator pressure. Since the adsorption is an exothermic process, isosteric heat is released, and is purged by the flow of cooling water through the beds. Simultaneously, the hot water produced in the GN flows through the heat exchanger tubes of the desorption beds to desorb water vapor from adsorbent materials. During the desorption process, the bed pressure decreases and finally is nearly equivalent to the condenser pressure under equilibrium conditions. The temperatures of the beds are calculated by the following energy balance equation, i.e.,

$$[(Mc_P)_{ads} + (Mc_P)_{HX} + (Mc_P)_{abe}] \frac{dT_{bed}^k}{dt} = M^{ads} \frac{dq^k}{dt} Q_{st} + (\dot{m} c_p)_{cw/hw}^k (T_{cw/hw,in}^k - T_{cw/hw,out}^k) \quad (27)$$

where $(Mc_P)_{ads}$, $(Mc_P)_{HX}$, $(Mc_P)_{abe}$ and Q_{st} are the thermal masses of the adsorbent, heat exchanger and the adsorbed water vapor and the isosteric heat of adsorption. The first term on the right hand side of Eq. (27) denotes the addition (adsorption process) or the removal (desorption process) of energy due to the isosteric heat of adsorption and the second term

represents the sensible energy exchanged by the cooling or heating fluids. The terms $T_{cw/hw,in}$ and $T_{cw/hw,out}$ indicate the inlet/outlet temperatures of heat transfer fluid and the superscript “ k ” denotes the status of the bed (adsorption/desorption). The temperature of the fluid is calculated at different segments of the heat exchangers tubes, and is given by

$$(\rho c_p)_{cw/hw}^{bed} \frac{\partial T_{cw/hw}^k}{\partial t} = -u_f^k (\rho c_p)_{cw/hw}^k \frac{\partial T_f^k}{\partial z} + \lambda_{cw/hw}^{bed} \frac{\partial^2 T_f^k}{\partial z^2} - \left(\frac{UA}{V} \right)_{cw/hw}^k (T_{cw/hw}^k - T^{bed}) \quad (28)$$

z is the distance along the direction of the flow of heat transfer fluid. The boundary conditions of the fluid flows are

$$\text{During adsorption: } T_{cw/hw}^k(z=0, t) = T_{cw}^{in} \text{ and } \frac{\partial T_{cw/hw}^k}{\partial z}(z=L^{tube}, t) = 0.$$

$$\text{During desorption: } T_{hw}^k(z=0, t) = T_{hw}^{in} \text{ and } \frac{\partial T_{cw/hw}^k}{\partial z}(z=L^{tube}, t) = 0.$$

3.3.3 Condenser

The regenerated water vapor flows to the condenser from the corresponding desorption beds. The latent heat of condensation is delivered to the cooling water flowing through the heat exchanger tubes. The cooling water purges the heat to the environment. After condensation, the condensate is stored as desalinated water in the potable water chamber. The temperature of the condenser is calculated by

$$\left[(Mc_p)_{dw} + (Mc_p)_{HX} \right] \frac{dT^{Cond}}{dt} = -M^{ads} h_{fg} \frac{dq}{dt} - \dot{m} c_p^{cw} (T_{cw}^{out} - T_{cw}^{in}) + M^{ads} \frac{dq}{dt} h_f \quad (29)$$

The left hand side of Eq. (29) shows the heat capacities of the stored condensed water and the heat exchanger. The first term on the right hand side represents the latent heat added by the water vapour regenerated from the desorption beds, the second term represents the heat rejection by the flow of cooling water. The cooling water temperature at different length of condenser tube is calculated by applying the energy balance on the water control volume, i.e.,

$$(\rho c_p)_{cw} \frac{\partial T^{cw}}{\partial t} = -(\rho c_p)_{cw} \frac{\partial T^{cw}}{\partial z} + \lambda_{cw} \frac{\partial^2 T^{cw}}{\partial z^2} - \left(\frac{UA}{V} \right)_{cw} (T^{cw} - T^{Cond}) \quad (30)$$

where V is the volume of heat-exchanger tubes and z is the distance along the direction of cooling water flow in the condenser tube. The boundary conditions of the equation are

$$T^{cw}(z=0, t) = T_{cw}^{in} \quad \text{and} \quad \frac{dT^{chw}}{dz}(z=L^{tube}, t) = 0. \quad L^{tube} \quad \text{is the length of the condenser tube.}$$

3.4 Overall Heat Transfer Coefficients

The overall heat transfer coefficients of heat exchangers for the heat recovery system are calculated by [33],

$$U_{overall} = \frac{1}{h_i} + \frac{r_o \ln \frac{r_o}{r_i}}{k} + \frac{r_o}{r_i} \frac{1}{h_o} \quad (31)$$

where h_i and h_o are the convective heat transfer coefficients of the fluids flowing through the tube and shell sides of the heat exchanger, respectively. These parameters depend on i) the mass flow rate of the heat transfer fluid, and ii) the dimensions and sizes of heat exchanger tubes. r_i and r_o are the inner and outer radius of the heat exchanger tubes, k is the thermal conductivity of the tube material. The heat fluxes of Evp-2 and Evp-3 are calculated for the low pressure evaporation of seawater, and are predicted by the modified Rohsenow correlation as [34],

$$T_w - T = \left(\frac{C_{sf} h_{fg} \text{Pr}_l^5}{C_{pf}} \right) \left[\frac{\dot{q}}{\mu_l h_{fg} \sqrt{g(\rho_l - \rho_g)}} \right]^{0.33} \left(\frac{P}{P_{atm}} \right)^{0.293} \left(\frac{A_{wetted}}{A_{base}} \right)^{-0.0984} \quad (32)$$

The Nusselt film condensation correlation is applied to calculate the heat transfer coefficient for the condensation of water vapour and is written as [33]

$$h = 0.433 \left[\frac{\rho_l g k^3 (\rho_l - \rho_v) \left[h_{fg} + \frac{3}{8} C_{pl} (T - T_w) \right]}{D \mu (T - T_w)} \right] \quad (33)$$

3.5 Performance Parameters

The cooling capacity of the carbon dioxide cycle is measured by

$$Q_{cool}^C = (\dot{m} c_p)_{chw} (T_{chw}^{Evap,1,in} - T_{chw}^{Evap,1,out}) \quad (34)$$

The COP of the carbon dioxide cooling cycle is calculated as the ratio of the cooling capacity to the power consumed by the mechanical compressor and is expressed by

$$COP_C = \frac{Q_{cool}^C}{P_{comp}} \quad (35)$$

The cooling capacity of adsorption system is obtained from Evp-2 and the cycle average cooling capacity is estimated as

$$Q_{cool}^A = \frac{1}{t_{cycle}} \int_0^{t_{cycle}} (\dot{m} c_p)_{chw} (T_{chw}^{Evap-2,in} - T_{chw}^{Evap-2,out}) dt \quad (36)$$

Heat energy released from the condenser of the adsorption device is given by

$$Q_{cond}^A = \frac{1}{t_{cycle}} \int_0^{t_{cycle}} (\dot{m} c_p)_{cw} (T_{cw}^{cond,out} - T_{cw}^{in}) dt \quad (37)$$

where t_{cycle} is the cycle time and comprises one adsorption period, one desorption period and two switching intervals (when the adsorption or desorption bed changes their roles). The cycle average heat energy required for the regeneration of water vapor and the amount of energy rejected during adsorption process can be calculated as,

$$Q_{Des} = \frac{1}{t_{cycle}} \int_0^{t_{cycle}} (\dot{m} c_p)_{hw} (T_{hw}^{in} - T_{hw}^{out}) dt, \text{ and} \quad (38)$$

$$Q_{Ads} = \frac{1}{t_{cycle}} \int_0^{t_{cycle}} (\dot{m}c_p)_{cw} (T_{cw}^{in} - T_{cw}^{out}) dt \quad (39)$$

The cooling performance of the proposed system is represented in terms of SCP (specific cooling power) and is calculated as

$$SCP = \frac{Q_{Evap-2}}{n \times M^{Ads}} \quad (40)$$

where M^{Ads} is the mass of adsorbents per bed and n is the number of beds. On the other hand, the desalination performance is presented in terms of SDWP (specific daily water production) and PR (performance ratio). The specific daily water production is given by

$$SDWP = \int_0^{t_{cycle}} \frac{(\dot{m}c_p)_{cw} (T_{cw}^{out} - T_{cw}^{in}) \tau}{h_{fg} M^{Ads}} dt \quad (41)$$

where, τ is the number of cycle per day. The performance ratio (PR) is expressed by

$$PR = \frac{1}{t_{cycle}} \int_0^{t_{cycle}} \frac{\dot{m}_{dw} h_{fg} (T^{cond})}{(\dot{m}c_p)_{hw} (T_{hw}^{in} - T_{hw}^{out})} dt \quad (42)$$

As the cooling effect of the adsorption cooling and desalination system is obtained from the waste energy of CO₂ heat pump system, the overall COP is calculated as

$$COP_o = \frac{Q_{cool}^C + Q_{cool}^A}{W_{comp}^{iso}} \quad (43)$$

Considering both cooling and desalination performances of the hybrid system, the overall conversion ratio is given by,

$$OCR = \frac{\sum_{i=1}^2 Q_{Evap} + Q_{cond}}{Q_{des}} \quad (44)$$

4. Results and Discussion

The performances of the proposed compression-adsorption hybrid system are analytically estimated in terms of COP, SCP, SDWP, PR and OCR. The energy balance equations of the

proposed compression-adsorption hybrid are solved numerically. The water and CO₂ properties under saturated and superheated conditions are coupled with the lumped-parameter-based equations of CO₂ compressor, condenser, adsorber, desorber and evaporator. The computed values of the key system variables are constantly updated by numerical integration with time. The calculation procedures are commenced with initialization of variables, system geometries and adsorption characteristics of adsorbent–adsorbate systems. Thermodynamic models are solved numerically in a MATLAB platform with the absolute and relative error tolerances of 1×10^{-5} . The parameters required to simulate the overall system are furnished in Table 3. The amount of water vapour uptakes and offtakes on silica gel adsorbents are obtained from experimentally measured isotherms and kinetics data. These data are also added in the simulation process. Only the initial conditions are specified in the problem and the total system is allowed to operate from transient to cyclic steady-state conditions. As the cycle is operated in a batched manner, i.e., streams of hot and cold coolant are alternated between beds.

Mayekawa [35] manufactured compressors with model no C2HT for CO₂ based heat pump applications. We employed here the specifications of C2HT compressor for our simulation, and these are provided in Table 3. The operating conditions of the compressor are vital for the proposed hybrid system. The compressor discharge pressure is found higher than the critical pressure (= 73.9 bar) of CO₂. Figure 4 shows the cycle-averaged hot water temperature generated in the storage tank and power consumed by the compressor for different compression ratios. It is observed that the increase in hot water temperature is proportional to the compression power. The hot water temperature is increased with the increase of compression ratio. Previous studies on adsorption cooling and desalination system suggest that the system performances are deteriorated when the driving heat source temperature is less than 70°C or more than 85°C [36-37]. Hence, for the efficient operation of

adsorption cooling and desalination systems, the compression ratio should be between 2.5 to 4. On the other hand, the power consumption of the compressor is raised due to the increase in compression ratio. Therefore, the COP of carbon dioxide cycle is decreased. The volumetric and isentropic efficiencies of the compressor also influence the performances of the proposed system, as the discharge pressure and temperature are dependent on these efficiencies. Figure 5 shows the volumetric and isentropic efficiencies of the compressor at various compression ratios. It can be seen that the isentropic efficiency of the compressor decreases vigorously when the compression ratio is less than 2 or more than 7. According to the scope of this study, the compression ratio is varied from 2.5 to 4 for the required hot water temperatures to drive the adsorption cycle. The volumetric efficiency is found between 0.65 and 0.75 and the isentropic efficiency is obtained between 0.52 and 0.54 for adsorption cooling cum desalination operating conditions.

Thermodynamic behaviours for the conventional and hybrid systems of both configurations 1 and 2 are studied and simulated by using adsorption, isotherms, kinetics and experimentally confirmed some technical and thermophysical properties data as listed in Table 3. The operating conditions of the proposed hybrid system are considered according to the previously reported studies on carbon dioxide heat pump and adsorption desalination system [33, 35, 37 – 41]. For the compression ratio of 3.64, the hot water inlet temperature is calculated as 85°C. The inlet temperatures of cooling and chilled water are considered 30°C and 14.8°C, respectively (standard conditions). Figure 6(a) explains the thermodynamic behaviours of conventional transcritical carbon dioxide cooling cycle. The thermodynamic processes such as evaporation, compression, gas cooling and expansion are indicated by processes 1→2, 2→3, 3→4 and 4→1, respectively. The evaporation occurs at 10°C and 45 bar. Following that the refrigerant temperature and pressure increase up to 100°C and 165 bar during the compression process. After the gas cooling process, the refrigerant temperature

decreases to 64°C while the pressure remains at 165 bar. The thermodynamic processes of the proposed hybrid system employing configuration-1 are shown in Figure 6(b). Here process 1→2 shows the evaporation. Process 2→3 indicates the second stage of heat recovery, where the saturated vapour outlet of Evp-1 is isobarically heated by recovering heat from the compressed and superheated refrigerant at the outlet of GN (Figure 1). The degree of superheat is achieved by this process. The superheated carbon dioxide is compressed during the process 3→4 and the temperature and the pressure are reached to 130°C and 165 bar. Process 4→5 is the first stage heat recovery process when the compressed refrigerant gives up a fraction of its total heat to produce hot water in the GN. The temperature reduces to 96°C after the first heat recovery process. The second stage heat recovery process is shown as 5→6, when the heat from the high pressure refrigerant stream is transferred to the low pressure side. As a result, the temperature of the high pressure refrigerant decreases up to 78°C. After the second stage heat recovery, the CO₂ refrigerant goes through the gas cooling process (6→7 as shown in Figure 6(b)). It reduces the temperature from 78°C to 51°C. Hence, 7→1 defines the expansion process. Figure 6(c) shows the thermodynamic process for the second configuration of the proposed compression-adsorption hybrid system. It shows that the evaporation occurs at 8 °C and 43 bar (1→2). During the second stage of heat recovery (2→3), the suction temperature of the compressor increases to 39°C. Due to CO₂ compression in the compressor (3→4), the discharge temperature and pressure become 128°C and 157 bar. Later, the refrigerant temperatures are decreased up to 92°C and 75°C with respect to the first (4→5), the second (5→6) stages of heat recovery processes. The temperature of CO₂ is further decreased up to 50°C due to gas cooling process (6→7). Configuration 2 has an additional stage of heat recovery (7→8). During this (7→8) process the CO₂ from the outlet of gas cooler flows through the heat exchanger tubes of Evp-3. As a result, the evaporation rates of sea water are increased and the temperature of CO₂ is decreased from 50°C to 38°C.

Employing the simulation results, it can be seen that the cooling capacities of conventional system, Configuration 1 and Configuration 2 are 136 kW, 151 kW and 192 kW. For the considered compressor specifications and operating conditions, the refrigerant flow rates of conventional system, Configuration 1 and Configuration 2 are simulated to be 1.88 kg/s, 1.35 kg/s and 1.3 kg/s, respectively. Moreover, the work done by the compressor on refrigerant for the conventional system, configuration-1 and configuration-2 are calculated as 97 kW, 93 kW and 90kW, respectively.

The Dühring diagram of the proposed hybrid system is depicted in Figure 7 to show the pressurisation effect in the adsorption bed by the high pressure evaporator (from 1 to 2 or a to b). The temperature of the adsorption bed increases slightly (from 2 to 3 or b to c) due to quick pressurisation process as compared to the slow cooling process. Due to cooling effects in Evp-2, the temperature of the bed drops. At the beginning of the preheating process, the bed pressure is reduced as the valve between the bed and the high pressure evaporator Evp-3 is closed. Therefore, a slight decrease in the uptake is found. The preheating process ranges from 3 to 4 (or c to d for configuration -2) and the uptake is remained constant. At the beginning of desorption, the sudden depressurisation in the bed results in the pressure swing desorption. The desorption process (4 – 5 or d – e) is followed by the pre-cooling (5 – 1 or e – a). The cycle competes with (i) the low-pressure adsorption (1- 2 or a - b) process for cooling process and (ii) the high pressure adsorption for mainly desalination purposes. Hence, the adsorber pressure for Configuration 2 is increased from 4 kPa to 11.30 kPa. In contrast, it increases from 2.7 kPa to 10 kPa (for Configuration 1).

The transient water production rate for both Configurations 1 and 2 are shown in Figure 8. It is found that both the peak and minimum water production rates of configuration-2 are higher compared to Configuration-1. The peak water production rate of Configuration 1 and Configuration 2 are 1.84 L/min and 2 L/min, respectively. On the other hand, the minimum

water production rates of the proposed configurations are 0.2 L/min and 0.5 L/min. The higher water production rate from Configuration 2 is due to the pressurisation effect generated at the third stage heat recovery system. It recovers heat from the carbon dioxide outlet of the gas cooler, which increases the pressure of Evp-3. The performances of the conventional carbon dioxide system in terms of COP (considering the operating conditions as furnished in Table 3) are compared with the previously published experimental results of the transcritical carbon dioxide cooling cycle [42-45]. These results are furnished in Table 4. Employing the CO₂ cooling model, the COP is found to be 1.4. The experimental results show that for different operating conditions, the COP of the transcritical CO₂ cooling cycle is varied from 1.2 to 1.4. The simulation results agree well with the experimental data [40]. Table 5 shows the performances of the hybrid (both configurations 1 and 2), adsorption cooling cum desalination and conventional CO₂ cooling systems in terms COP, SCP, SDWP, PR and OCR. It can be seen that both Configurations 1 and 2 outperform the conventional adsorption and CO₂ cooling system in terms of COP. The overall conversion ratio is found higher for the second configuration of the hybrid system, which is 2.8 as compared to the CO₂ cooling cycle (OCR = 1.4). It should be noted here that the cooling and desalination effects from the adsorption cycle is produced without consuming any additional electrical energy. Therefore, the overall OCR is found very high. The overall COPs for Configuration 1 and Configuration 2 are found to be 1.79 and 2.25, respectively. These are 28% and 61% higher than the COP of the conventional CO₂ cycle. In addition, the SCP for the hybrid system is higher than the conventional adsorption chiller. Due to the effects of third stage heat recovery, the amount of water vapour transfer per adsorption-desorption cycle for Configuration 2 is the highest. Therefore, the SDWP and the PR of Configuration 2 are found 30% and 10% more than those of Configuration 1. The second configuration of compression-adsorption hybrid can be used to recover the most waste energy.

A parametric study is conducted here to investigate the effects of various operating conditions on the performances of the proposed system. Each parameter is varied within a practical range and the performances are calculated, when the other conditions remain unchanged. Figure 9 exhibits the effects of compression ratio on COP and SDWP of the hybrid system (Configuration 2). It shows that COP decreases with the increase in compression ratio. In this study it is found that the compressor consumes more power when the compression ratio increases (according to Figure 4). As a result, the COP of the carbon dioxide cycle is decreased. In addition, if the compression ratio increases, the hot water temperature in the GN is also increased (Fig. 4). The higher hot water temperature leads to better cooling and desalination effects of the adsorption cycle. The SDWP is also increased. Considering both COP and desalination performances, the optimum compression ratio can be chosen as 3.3.

Figure 10 shows the effects of cooling water inlet temperatures on the COP of CO₂ cycle of the proposed hybrid system for various compression ratios. Considering the climate of tropical regions, the cooling water temperature is varied from 24°C to 40°C. The lower cooling water temperature results in the lower CO₂ temperature at gas cooler outlet. Therefore, the highest COP is obtained at the lowest cooling water temperature and compression ratio. It is found that the highest possible COP of CO₂ cycle (~2.59) can be achieved at the cooling water temperature and compression ratio (CR) of 24°C and 2.5, respectively. On the other hand, the lowest COP (~1.76) is obtained at the cooling water temperature and CR of 40°C and 4. Figure 11 shows the effects of cooling water temperature on SDWP of the hybrid system for various compression ratios. The SDWP is found higher for higher compression ratio. From the simulation study, it is found that the SDWP provides less impact on the CR when CR > 3.5. In addition, the cooling water temperature at a fixed CR shows here two conflicting effects on SDWP. The higher cooling water temperature

causes the higher carbon dioxide outlet temperature from the gas cooler, which also improves the pressurization effects in Evp-3. Therefore, the SDWP increases. Secondly, the higher cooling water temperature provides the lower condensation rates, which decreases the amount of SDWP. For each compression ratio, there is an optimum cooling water temperature at which the maximum SDWP is obtained. For the higher the compression ratios, the optimum cooling water temperature is found lower. The SDWP is obtained to be 13.2 m³ per tonne of silica gel for the CR and the cooling water temperature of 4 and 26°C, respectively. As can be observed from Figure 11 that for the lower CR (such as 2.5), the SDWP increases with the increase in cooling water temperature due to the heat recovery between the outlet CO₂ temperature of gas cooler and the saline water in EVP-3.

Figure 12 shows the effects of cycle times on the COP of hybrid system for different CR. The overall COP decreases with the increase in CR due to the increase in heat rejection rate. The COP also drops at higher cycle time. This is due to the fact that at higher cycle times, the amount to unused sensible heat during desorption period is increased. The effect of cycle time on SDWP at various compression ratios is displayed in Figure 13. It shows that the SDWP increases with the increase in cycle time and reaches a maximum value. This occurs due to the fact that at higher cycle time, the amount of water per adsorption-desorption cycle is increased. At very long cycle time, the difference between uptake and offtake (Δq) will not be increased. If the desorption period is too long, some water vapour molecules are adsorbed in the porous silica gel, which on the other hand decreases the value of Δq .

5. Conclusions

This study presents a comprehensive thermodynamic framework for compression-adsorption hybrid cooling and desalination system. It symbiotically combines the transcritical CO₂ cooling cycle with adsorption assisted cooling and desalination system for higher SDWP. The heat recovery system recuperates heat from the compressed CO₂ refrigerant to produce

hot water for driving the adsorption system. The second stage of heat recovery internally exchanges heat between the carbon dioxide extracting from the GN and the CO₂ vapour evaporated from the evaporator. In this paper, the utilization of heat recovery for the production of cooling and water is discussed. The heat recovery decreases the gas cooler temperature of CO₂ cycle as an extra benefit for tropical climate conditions. The performances are estimated numerically and the main findings of this study are as follows:

- Chilled water inlet and outlet temperatures for the CO₂ cycle of conventional system, Configuration 1 and Configuration 2 are maintained 14.8 °C and 10 °C, respectively for the cooling capacities of 136 kW, 151 kW and 192 kW. On the other hand, the power consumption by the conventional, first configuration and second configuration are 97 kW, 93 kW and 90 kW, respectively.
- The first configuration improves the COP of CO₂ cycle by 28% and produces 9.86 m³ of fresh water per tonne of silica gel per day.
- The overall COP is improved about 61% as compared with the CO₂ cycle by the second configuration. Along with the cooling capacity the proposed hybrid system also provides 12.66 m³ of fresh water per ton of silica gel per day according to the proposed system configurations.
- The overall conversion ratio of Configuration 1 and Configuration 2 are 2.21 and 2.81 respectively. These are 58% and 100% higher than the COP of the conventional CO₂ cycle.

From the present study, it is concluded that the proposed hybrid system offers an efficient solution for cooling and desalination purposes in the tropics.

Nomenclature

Symbol

A	Area	m^2	
c_p	Specific heat	$\text{kJ/kg}\cdot\text{K}$	
h_{fg}	Latent heat of evaporation	kJ/kg	
h	Enthalpy/heat transfer coefficient	kJ/kg , $\text{kW/m}^2\text{ K}$	
K_o	Pre-exponential coefficient	$1/\text{Pa}$	
M	Mass	kg	
$\dot{m}$	Mass flow rate	kg/s	
P	Pressure	Pa	
T	Temperature	$^{\circ}\text{C}$	
t	Time	s	
U	Overall heat transfer coefficient	$\text{W/m}^2\cdot\text{K}$	
V	Volume	m^3	
Q	Heat/cold energy	kW	
q	Uptake	kg/kg	
ρ	Density	kg/m^3	
λ	Thermal conductivity	$\text{kW/m}\cdot\text{K}$	
X	Salinity	$\text{g of salt/kg of solution}$	
Subscripts			Superscripts
ads	Adsorption	sw	Seawater
des	Desorption	br	Brine
chw	Chilled water	Ads	adsorbents
cw	Cooling water	abe	adsorbate
hw	Hot water	Evp	Evaporator
dw	Distilled water	$cond$	Condenser
$cycle$	Cycle time	bed	Sorption reactor
f	Fluid	$Comp$	Compressor
k	Adsorption/desorption	PH	Preheater
is	isentropic	GN	Generator
in	inlet	GC	Gas cooler
out	outlet	exp	Expansion valve

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Figure 13: Effects of cycle time and CR on SDWP at the heating source and sink temperatures of 85°C and 30°C (Second configuration).

CO ₂ Cycle		Adsorption Cycle	
Evp-1	Evaporator-1	Evp-2	Evaporator-2
PH	Preheater	Evp-3	Evaporator-3
Comp	Compressor	Bed-1,2,3,4	4 adsorption Beds
GN	Hot water generator	Cond	Condenser
GC	Gas cooler	PWT	Potable water tank

Fig. 1

CO ₂ Cycle		Adsorption Cycle	
Evp-1	Evaporator-1	Evp-2	Evaporator-2
PH	Preheater	Evp-3	Evaporator-3
Comp	Compressor	Bed-1,2,3,4	4 adsorption Beds
GN	Hot water generator	Cond	Condenser
GC	Gas cooler	PWT	Potable water tank

Fig. 2

Time	Preparation	Cycle					
Bed 1	L. A	H. A	P H	D.	P C	L. A	H. A
Bed 2	L.	L. A	H. A	P H	D.	P C	L.A
Bed 3	L.	D.	P C	L. A	H. A	P H	D.
Bed 4	L.	D.	P C	L. A	H. A	P H

Fig. 3

Fig. 4

Fig. 5

Fig. 6

Fig.7

Fig.8

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Table 1

Type of Ref.	CFC (Abolished refrigerant)			HCFC (Regulated refrigerant)	HFC (Substitute refrigerant)				Natural refrigerant		
Ref	R11	R12	R502	R22	R134A	R404A	R407	R410A	R717	R744	R600a
ODP	1.0	1.0	0.334	0.055	0	0	0	0	0	0	0
GWP	4750	10900	4590	1810	1430	3922	1650	2088	>1	1	4

Table 2

Ref.	Description	Discharge Pressure (bar)	Discharge Temperature (°C)	Gas cooler outlet temperature (°C)
[17]	<ul style="list-style-type: none"> Refrigeration system. Experimental study. COP measured at: <ol style="list-style-type: none"> various evaporator temperatures. various discharge pressure. various gas cooler outlet temperatures. 	75 to 100 bar	-----	31 to 40°C
[18]	<ul style="list-style-type: none"> Heat pump system. Experimental study. A correlation of optimum discharge pressure is presented for: <ol style="list-style-type: none"> various gas cooler outlet temperatures. 	75 to 130 bar	-----	20 to 50°C
[19]	<ul style="list-style-type: none"> Heat pump system with internal heat exchanger. Experimental study. COP measured at: <ol style="list-style-type: none"> various gas cooler inlet temperatures. various discharge pressures. 	75 to 120 bar	60-90°C	-----
[20]	<ul style="list-style-type: none"> Heat pump system with two throttle valves. Experimental and Analytical study. Investigated the relation between optimum heat rejection pressure and <ol style="list-style-type: none"> gas cooler outlet temperature. evaporating temperature. 	75 to 135 bar	-----	33 to 45°C

	iii. compressor performance.			
[21]	<ul style="list-style-type: none"> • Mayekawa presented their commercial heat pump UNIMO ww. • Installed at EPRI (Electric Power Research Institute) 	145 bar	180°C	75°C
[22]	<ul style="list-style-type: none"> • Refrigeration system with internal heat exchanger. • Experimental study. • Discharge temperature and COP measured for different internal heat exchanger locations 	75-105 bar	100-150°C	-----

Table 3

Parameters of Adsorption Isotherms				
Adsorbent – adsorbate pairs	q^m (kg/kg)	E (kJ/mol)	n	ΔH (kJ/kg)
Silica gel – water [40,41]	0.592	3.105	1.1	2800
Kinetics Parameters				
Adsorbent – adsorbate pairs	D_{so} (m ² /s)	E_a (kJ/mol)	R_p (m)	
Silica gel – water [40,41]	2.54×10^{-4}	42	0.4×10^{-3}	
Other values used in the simulation				
Compressor [35]		$(UA)_{ads}^{bed}$ 10.43 kW/K		
Model	C2HT			
Refrigerant	CO ₂	Evaporator [33,37,38,39,40,41]		
Bore×Stroke	45 × 45 (mm ²)	$(Mc_p)_{HX}$ 25.44 kJ/K		
Displacement vol.	8.33 (m ³ /h) (at 970 rpm)			
	12.5 (m ³ /h) (at 1,450 rpm)	$\dot{m}_{chw}$ 1.42 kg/s		
Sorption reactors [33,37,38,39,40,41]		$(UA)_{evp}$ 6 kW/K		
M^{Ads}	36 kg			
$(Mc_p)_{HX}$	284.4 kJ/K	Condenser [33,37,38,39,40,41]		
$(C_p)_{ads}$	0.921 kW/K	$(Mc_p)_{HX}$ 18.61 kJ/K		
$\dot{m}_{hw}$	0.8 kg/s	$(Mc_p)_{dw}$ 40.18 kJ/K		
$\dot{m}_{cw}$	0.8 kg/s	$(UA)_{cond}$ 13.5 kW/K		
$(UA)_{des}^{bed}$	10 kW/K			

Table 4

Model/Ref	Description	COP
Conv (This model)	<ul style="list-style-type: none"> • Conventional CO₂ cycle. • Produce cooling effect and heat simultaneously. • Produced heat is used to drive an adsorption cooling and desalination system. • Compressor suction and discharge pressures are 43 bar and 157 bar. 	1.4
[40]	<ul style="list-style-type: none"> • CO₂ cooling cycle. • Experimental investigation. • Effect of internal heat exchanger is discussed. • Air cooled gas cooler. • Air temperature to gas cooler is 35 °C. 	1.4
[42]	<ul style="list-style-type: none"> • CO₂ refrigeration cycle. • Experimental investigation. • Performance of internal heat exchanger is investigated. • Compressor discharge pressure and temperature are 90 bar and 117°C. • Gas cooler temperature is 33°C. 	1.37
[44]	<ul style="list-style-type: none"> • CO₂ transcritical refrigeration cycle. • Numerical study. • Combined scroll expander-compressor is considered. • Compressor discharge pressure is 100 bar. • Gas cooler outlet temperature is 35°C. 	1.24
[45]	<ul style="list-style-type: none"> • Super-critical CO₂ refrigeration cycle. • Experimental analysis. • Compressor discharge pressure is 100 bar. • Gas cooler outlet temperature is 30°C. 	1.2

Table 5

Configuration	COP _O	COP _C	SCP _A	SDWP _A	OCR
Conv CO ₂ cycle	-----	1.40	-----	-----	1.40
Con-1	1.79	1.62	32	9.86	2.21
Con-2	2.25	2.12	24	12.66	2.81
ADC [37,38, 39]	0.32	-----	23	8.70	-----
<p>Notes: Con-1: Hybrid system with two stage heat recovery (preheater and generator). Con-2: Hybrid system with three stage heat recovery (preheater, generator and Evp-3). Conv : Conventional CO₂ compression cooling system. ADC : Adsorption cooling and desalination system.</p> <p>Units: SCP_A: Ton of Refrigeration per tonne of adsorbent material. SDWP_A: m³ of water per tonne of adsorbent material per day.</p>					