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Experimental study for hybrid humidification–dehumidification water desalination and air conditioning system

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HIGHLIGHTS

- A new hybrid humidification–dehumidification and air conditioning system is proposed and investigated.
- The proposed system keeps the function of the air conditioning system and uses it in water desalination.
- The effects of system operating parameters on system performance were evaluated and correlated.

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ABSTRACT

An experimental study of the performance of a hybrid humidification–dehumidification water desalination and air conditioning system using vapor compression refrigeration cycle is presented and investigated. A test rig is designed and constructed to study the performance under different operating parameters (air flow rate, air inlet temperature, specific humidity and evaporator saturation temperature). The effects of these operating parameters on fresh (desalinated) water production rate, refrigeration capacity, compressor work per kilogram of fresh water, mass transfer coefficient and supply air conditions to conditioned space (air temperature and relative humidity) are investigated and analyzed. The results show the enhancement of the fresh water production rate, the refrigeration capacity and the compressor work per kilogram of fresh water with increasing air specific humidity and air mass flow rate. The supply air temperature and relative humidity increase remarkably with increasing fresh water rate. Experimental correlations for fresh water production rate, refrigeration capacity and compressor work per kilogram of fresh water in terms of all studied parameters are deduced and presented within accepted error.

1. Introduction

There is a lack of drinkable water in most of hot and humid environment regions around the world. At the same time, these regions need air conditioning systems for thermal comfort. Nowadays, many of the countries depend on water desalination systems to supply their water demands. Atmospheric water vapor processing (AWVP) is a recent technology of fresh water production, especially for hot and humid climates. Nevertheless, produced fresh water is slightly compared to the present methods; it is a choice to be studied for low water demand regions [1]. Aly et al. [2] studied theoretically and experimentally the performance of the mechanical vapor compression (MVC) desalination system. Siqueiros and Holland [3] proposed desalination systems operated by heat pumps (mechanical vapor compression and/or absorption machines) as a compact and less cost structure that has first been used for the dry areas in north of Mexico. Additionally, the reasonable economic potential is given as compared to reverse osmosis technology. Hawlader [4] studied and described a novel solar-assisted heat pump desalination system and a good water production was obtained. Slesarenko [5] suggested incorporating heat pumps as a source of heat energy for seawater desalination plants. Two plants were proposed: desalination plant with compression heat pump operated with R12 and a steam and water cycle plant. Al-Juwayhel et al. [6] used a combined vapor compression heat pump with a single effect evaporator desalination system for atmospheric water vapor condensation on the evaporator surface. The performance of a new type of a humidification–dehumidification desalination unit driven by mechanical vapor compression pump was mathematically analyzed by Gao et al. [7]. Yuan et al. [8] presented an integrative unit for air-conditioning and desalination driven by vapor compression heat pump on basis of direct humidification–dehumidification process. Performance study of a combined heat pump (HP) with a dehumidification process to produce fresh water from the atmospheric air was analyzed by Habeebullah [9]. A

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Nomenclature

A	Total surface area, m^2
A_{min}	Minimum air flow area, m^2
A_o	Total coil surface area, m^2
D_h	Coil hydraulic diameter ($D_h = 4L_d A_{min}/A_o$), m
h_m	Mass transfer coefficient, m/s
$h_{fg,0\ °C}$	Latent heat of evaporation of water at 0 °C, kJ/kg
i	Specific enthalpy, kJ/kg
L_d	Coil depth, m
$\dot{m}$	Mass flow rate, kg/s
$\dot{m}_{steam}$	Steam mass flow rate, kg/s
$\dot{Q}_{ref}$	Refrigeration capacity, kW
$\dot{Q}_{ref,\ l}$	Refrigeration capacity–latent part, kW
P	Gauge pressure, Pa
RH	Air relative humidity, dimensionless
Re_{Dh}	Reynolds number based on hydraulic diameter
t	Temperature, °C
t_{amb}	Equivalent dry bulb temperature of inlet state, °C
u_a	Air velocity, m/s
w	Air specific humidity, $g_{\ water}/kg_{\ dry\ air}$
$\dot{W}_c$	Actual compressor work, kW

Greek symbols

η_c	Overall compressor efficiency
μ	Dynamic viscosity, N s/m^2
$\rho_{v,a}$	Moist air water vapor density, kg/m^3
$\rho_{v,su}$	Water vapor density at evaporator surface, kg/m^3

Subscript

a	Air
c	Compressor
ev	Evaporator
i	Inlet
o	Outlet
r	Refrigerant
s	Evaporator saturation temperature
w	Fresh water

patented layer freezing based technology which is scalable and coupled with a heat pump to switch freezes water from seawater in the evaporator and melts the ice in the subsequent phase when it serves as a condenser that was discussed by Rane and Padiya [10]. Heat pumps using agent R12 or water and vapor to be used as a source of heat energy for seawater desalination were introduced by Jinzeng and Huang [11]. An experimental evaluation of a two-stage technique to improve the humidification–dehumidification process in fresh water production from brackish water was presented by Zamen et al. [12]. The investigation of the potential for heat recovery from Multi Stage Flash (MSF) desalination plant hot distillate water to power an Organic Rankine Cycle (ORC), comparing R134a and R245fa refrigerants as the working fluid addressed by Al-Weshahi et al. [13]. Theoretical study of a simple solar still coupled to a compression heat pump was presented by Halima et al. [14]. The mathematical model has been developed using mass and heat balance. A new concept of produced water purification by humidification–dehumidification (HD) process in which low-temperature energy sources, such as co-produced geothermal energy or solar energy that could be used to drive the water desalination process was developed by Xinhua Li [15]. An open air–vapor compression refrigeration system for both air-conditioning and desalination on ship cooled by seawater was presented by Houa et al. [16]. An experimental investigation for dehumidification process of a wavy-finned-tube direct expansion cooling coil under humid condition was carried out by Huzayyin et al. [17]. Shen et al. [18] presented a comprehensive analysis of a single-effect mechanical vapor compression (MVC) desalination system using water injected twin screw compressors. The operational characteristics of the twin screw compressor including inlet volume flow rate, compressor pressure ratio and mass fraction of the injected water were investigated. Ghazal et al. [19] presented an experimental investigation for the performance of a solar humidification prototype that is suitable for using in humidification–dehumidification desalination (HDD) systems. Attia [20] introduced a new proposed system that depends on the optimization of utilizing the heat flow of the heat pump system to increase the whole system efficiency. Al-Ansari et al. [21] modeled and analyzed a single effect evaporation desalination process combined with adsorption heat pump (ADVC) in terms of designed and operational system parameters. Nafey et al. [22] presented a numerical investigation of a humidification–dehumidification desalination (HDD) process using solar energy. Nafey et al. [23] presented an experimental investigation for desalination system based on humidification–dehumidification desalination (HDD) technique using solar energy at the weather conditions of Suez City, Egypt.

According to authors' review, there is a shortage in the utilization of air conditioning systems in fresh water production by incorporating humidification–dehumidification desalination to the system with keeping the function of air conditioning systems (maintain human thermal comfort). Therefore, the present study introduces hybrid air conditioning and humidification–dehumidification desalination system to produce fresh water and satisfy the required thermal comfort conditions inside the conditioned space. In the study, the effects of the different system operating conditions (air mass flow rate, evaporator air inlet temperature, evaporator air inlet specific humidity and the evaporator saturation temperature) on the supply air conditions to conditioned space, desalinated water production rate, mass transfer coefficient, evaporator refrigeration capacity and compressor work per kilogram of fresh water are experimentally investigated to evaluate the feasibility of the system at the different operating conditions.

2. Experimental setup description and instrumentation

The experimental setup was designed to be consists of four independent main loops (refrigerant, humid air, fresh water and sea water loops) to enable the investigation of the effects of the operating conditions on the conditioned space-supplied air conditions, fresh water production rate, refrigeration capacity, compressor work per kilogram of fresh water and mass transfer coefficient.

2.1. Experimental setup description

Fig. 1 illustrates a schematic diagram of the experimental setup. The main system components are air blower (1), air heaters (2), steam boiler (10), humidifier (steam distributer) (3), evaporator (cooler and dehumidifier) (7). The setup includes the measuring and instrumentation devices to measure the different parameters needed for the study (temperatures, relative humidities, and air flow rates). The system consists of four loops, one closed loop for refrigerant, and the others are open loops for air, fresh water and sea water. In the refrigerant loop, refrigerant R134a is used. As shown in Fig. 1, the loop consists of compressor, oil separator condenser, liquid receiver, expansion device (automatic expansion valve, AEV), filter drier, heat exchanger, evaporator and suction line accumulator. The evaporator is a wavy-finned tube coil and it consists of three tube rows in staggered arrangement with 304.8 mm × 228.6 mm face area as shown in Fig. 2. The evaporator dimensions and specifications are given in Table 1. Air is flowing over evaporator surface passing through its fins where the cooling and dehumidification process is conducted. The evaporator pressure and corresponding saturation temperature varies with changing inlet air conditions and mass flow rate to verify system balance [24,25].

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