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Multi-stage desiccant cooling system for moderate climate

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ABSTRACT

This paper presents a numerical study of a novel, multi-stage desiccant air conditioning system designed for moderate climates. The proposed system is based on multi-stage cooling process through the Maisotsenko Cycle (M–Cycle), regenerative heat and mass exchangers (pre-cooling and post-cooling) combined with a desiccant wheel. The performance of the system was analysed numerically with original ε–NTU models and it was compared with a typical solution based on a desiccant wheel and the Maisotsenko cycle. It was found that multi-stage system obtains lower supply airflow temperatures and higher moisture content decrease in desiccant wheel as compared to typical M-Cycle desiccant system. Moreover, this solution allows a decrease in the regeneration airflow temperature to 40 °C maintaining the same or lower supply airflow temperature in moderate climatic conditions. Using multi-stage cooling, the proposed system was able to attain a thermal COP (defined as amount of cooling capacity obtained by the unit divided by required heating capacity) of up to 4.0. Due to this performance the proposed system has high application potential in moderate climates.

1. Introduction

Cooling energy is necessary in most of residential, commercial and industrial buildings because it provides an efficient working environment and comfortable daily life. However, an adequate amount of energy has to be utilized to provide the thermal comfort during summer season, when high outdoor temperatures are observed. To meet these requirements, air conditioning (AC) systems consume about 6% of the overall energy consumed by buildings in 2016 [1]. Nevertheless, it has to be mentioned that across the countries and regions the trends of cooling energy usage varies a lot, for example in USA it equals approximately 10% [1]. Mechanical vapor-compression systems are generally used to cool the airflow to the proper temperature level [1]. Due to the fact that these cooling devices consume a lot of electrical energy to provide indoor comfortable conditions, researchers are searching for the alternative cooling methods to minimize the energy consumption and the operational costs [2]. Currently, the indirect evaporative cooling technology, which is considered as the most effective for this type of solutions, are dew-point units based on the Maisotsenko cycle (M-Cycle) which are continuously being improved [3,4].

As an example recently, Peng Xu et al. [5] experimentally investigated a novel dew point cooler equipped with super-performance hydrophilic material layer, complex structure of the heat exchanger and intermittent water supply system. The device performance was compared to a typical commercial dew point air cooler (M30) under five different climatic conditions in the laboratory. The results show that for the same inlet airflow parameters, the new prototype obtains the lower product air temperature, a higher COP (Coefficient of Performance) values and higher wet-bulb and dew-point effectiveness.

Sohani et al. [6] also studied the design of dew point evaporative coolers at various climatic conditions. Nevertheless, other key performance indicators were considered: life-cycle costs, annual water consumption and the annual average of the COP values. The analysis show that in very hot and dry climate, the counter-regenerative configuration performs the best. On the other hand, in the rest of climates the cross-flow configuration was a better alternative. Hamoon Jafarian et al. [7] developed a precise model of the dew-point indirect evaporative cooler. They carried out an optimization of the system by determining optimum values of channel length, channel gap, inlet air velocity and return to intake air ratio for different cities. It allowed to maximize the average coefficient of performance and minimize the specific area of the cooler, simultaneously. Through this development, the system to

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Nomenclature

A	rotor desiccant section area [m^2]
a	assimilation of total cooling loads factor [-]
c_p	specific heat capacity of moist air [J/(kg·K)]
COP	thermal coefficient of performance [–]
C	heat capacity rate of fluid [W/K]
D_h	hydraulic diameter of flow passages [m]
e	additional electricity demand factor [–]
f	fluid friction coefficient [–]
F	surface area [m^2]
G	mass flow rate of moist air [kg/s]
g_c	proportionality of constant in Newtons second law of motion, $g_c = 1$ [–]
H	Height [m]
i	specific enthalpy of the moist air [J/kg]
L, l	streamwise length of cooler [m]
M	water vapor mass transfer rate [kg/s]
N	theoretical fan power [kW]
P	regeneration power [kW]
p_b	atmospheric air pressure [kPa]
R	desiccant wheel radius [m]
RH	relative humidity [%]
t	temperature [°C]
q	specific heat [kJ/kg]
Q	rate of heat transfer [W]
u	airflow axial velocity [m/s]
V	volumetric airflow rate [m^3/s]
W	moisture content of desiccant material [kg/kg]
x	humidity ratio of moist air [kg/kg]
X	coordinate along airflow direction [m]
Y	heat exchanger width [m]
Z_3	coordinate along the process airflow direction: desiccant wheel [m]
Z_4	coordinate along the regeneration airflow direction: desiccant wheel [m]
τ	time [s]
$\Delta\tau_3$	adsorption stage duration [s]
$\Delta\tau_4$	desorption stage duration [s]
$\Delta\tau_0$	time of one revolution of the wheel $\Delta\tau_0 = \Delta\tau_3 + \Delta\tau_4$ [s]
Φ_0	system cooling power [kW]
η	fan efficiency [–]
Ω	rotor desiccant rotation velocity [1/s]

Special characters

α	convective heat transfer coefficient
β	mass transfer coefficient [W/(m^2 K)]
δ	thickness [kg/(m^2 s)]
ρ	air density [m]
σ	surface wettability factor, $\sigma \in (0.0–1.0)$ [kg/m^3]

Non dimensional coordinates

Le	Lewis Number $\mathrm{Le} = \alpha/(\beta c_p)$ [–]
NTU	number of transfer unitsNTU $= \alpha F/(G c_p)$
NTU_d^*	Number of transfer units for rotor desiccant $\mathrm{NTU}_d^* = \alpha F_d \Delta\tau_o/(M_d\, c_d)$. [–]
Re	Reynolds number [–]
$\overline{X}$	$\overline{X} = X/L_X$–relative X coordinate [–]
$\overline{Z}$	$\overline{Z} = Z/L_Z$ relative Z coordinate [–]
$\overline{\tau}$	non-dimensional time $\overline{\tau} = \tau/\tau_0$ [–]
j	wheel revolution counter ($j = (1\ldots n)$) [–]

Subscripts

0	initial condition [–]
1	primary airflow [–]
2	working airflow
3	Product airflow
4	regeneration airflow
cond	heat transfer by thermal conduction
d	desiccant material
DH	desiccant wheel
DW	dry-bulb temperature
E	ambient airflow parameters
g	water vapour
int	internal airflow parameters
HMX1	heat and mass exchanger no 1
HMX2	heat and mass exchanger no 2
i	inlet
o	outlet
p	plate surface
S	sensible heat flow
L	latent heat flow
sorp	sorption
Sys1	referenced to system 1
Sys2	referenced to system 2
WB	wet-bulb temperature
•	referenced to the elementary plate surface
*	referenced to assimilated cooling loads in air-conditioning zone

reached its full potential for diverse climatic conditions.

One of the solutions which is considered as most promising for moderate and humid climate are desiccant systems [8], based on the combination of a desiccant unit (solid or liquid), which dehumidifies the air and an indirect evaporative air cooler which cools the air to the appropriate temperature level without adding any moisture. Desiccant cooling systems in different configurations with an indirect evaporative air cooling have been investigated by many authors in the past. Kashif Shahzad et al. [9] compared a solid desiccant dehumidifier integrated with a cross flow Maisotsenko cycle heat and mass exchanger (HMX) with a traditional desiccant air conditioning system integrated with a direct evaporative cooler under different operating conditions and the same process and return airflow rates. It was found out that for the same supply airflow and regenerative airflow parameters, first system was approximately 60–65% more efficient than the second one. Jinzhe Nie et al. [10] modelled and analyzed a solid desiccant wheel combined with a heat pump (HP-SDC). The results clearly prove that HP-SDC is characterized by more efficient energy performance as compared to traditional cooling systems.

Yasser Abbassi et al. [11] studied desiccant cooling systems in three configurations where solar energy was used for regeneration of the desiccant. The systems differ in the air parameters entering the dehumidifier. These are: fresh air, conditioned air and a mixture of the two. Moreover, different number of desiccant wheels and possibility of additional heat recovery implementation were taken into consideration. It was found out that in the ventilation and recirculation mode, the COP value is higher for single-stage than for a double-stage system. On the other hand, additional heat recovery application improves the double-stage system more than the system with only one dehumidifier used. Rang Tu and Yunho Hwang [12] established the efficient configurations for desiccant cooling systems with usage of different heat sources. The results allow to design the wheel according to selected heat source,