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Gray-box identification of thermal transfer coefficients of desiccant wheels

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ABSTRACT

Dynamic models are needed for automatic control and online fault detection and diagnosis of desiccant wheels used in solar cooling systems. The parameters of these models need to be experimentally identified during operation. A model based on the heat and mass transfer equations, given in state-space representation, is developed. Using this model, a gray-box approach is developed to identify experimentally the thermal transfer coefficients. The Nusselt number and the heat and mass transfer coefficients have been obtained by defining the unknown parameters of the model as a function of a single parameter, which itself depends on the heat transfer coefficient, and by expressing the other parameters as a function of the geometrical and physical properties of the wheel.

The difference between the values of the thermal transfer coefficients given in literature and those obtained by this method is less than 10%. Therefore, the gray-box identification method may be adopted as a tool for experimental estimation of the thermal coefficients under various design and operation conditions for the desiccant wheels.

1. Introduction

Mechanical compression is the most widely used solution for air conditioning. While it shows a relatively high level of performance compared to other solutions such as absorption chillers, it has two major disadvantages. On one hand, the power consumption of the mechanical compressor is not negligible and increases the peak demand in summer. On the other hand, refrigerants have a negative impact on the atmosphere and contribute to global warming.

Several cooling solutions use solar energy, a renewable source that has the advantage of being available when the cooling demand is high. The best known techniques are cooling by absorption and desiccation by adsorption as in the adsorption open cycle [1].

A desiccant cooling unit is shown in Fig. 1. It is a thermally driven open cycle based on evaporative cooling and adsorption. The cycle operates as follows: outside air (1) is dehumidified in the desiccant wheel and its temperature increases due to adsorption (2). Then it is cooled in the sensible heat exchanger (3). After that, it is cooled and humidified by the humidifier (4) and finally it is introduced into the building. The return air (5) is cooled by humidification (6) then passed through the rotary heat exchanger (7) which cools the inlet air (3). This air is then heated by a heat source, such as a heat exchanger, a solar collector or a gas burner. The warm air (8) is then used to regenerate the desiccant wheel by removing the humidity before leaving the air-handling unit (9).

Understanding the transport phenomena occurring in the rotary desiccant wheel has become important in designing and controlling the desiccant air units. Many studies have focused on the performance estimation of the desiccant air units (DAU) using computer simulation software such as TRNSYS [2,3]. The modeling of the desiccant wheel was based on analytical and numerical methods such as the analogy theory [4] and the finite difference method [5]. Mathematical models of the heat and mass transfer processes in the desiccant wheels have been developed. Some of these investigations considered one-dimensional transient models for coupled heat and mass transfer between the solid desiccant and the air streams in the wheel [6,7]. A few studies used a mathematical model for the heat and mass transfer process that occurs during the sorption of water vapors in the channels of the dehumidifier, considering both the gas-side and the solid-side resistances for the heat and mass transfer [8].

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Nomenclature

a	channel height [m]
b	channel width [m]
A	adsorption potential [J mol^{-1}]
A_c	air pass-through area of the channel (taken perpendicular to the axis of the wheel) [m^2]
A_d	cross-section area for the desiccant layer in a channel [m^2]
A_g	cross-section area for air flow [m^2]
c	isobaric specific heat [J kg^{-1} K^{-1}]
d_t	thickness of the desiccant coating [m]
D_h	hydraulic diameter of a channel [m]
h	heat transfer coefficient [W m^{-2} K^{-1}]
h_m	mass transfer coefficient [kg m^{-2} s^{-1}]
κ	thermal conductivity [W m^{-1} K^{-1}]
Le	Lewis number [–]
Nu	Nusselt number [–]
Nu_T	Nusselt number for thermally fully developed laminar flow with temperature boundary condition [–]
Nu_F	Nusselt number for thermally fully developed laminar flow with heat flux boundary condition [–]
Re	Reynolds number [–]
Pr	Prandtl number [–]
Gz	Graetz number [–]
P	pressure [Pa]
Q_{sor}	adsorption heat [kJ kg^{-1}]
R	gas constant [kJ kmol^{-1} K^{-1}]
t	time [s]
T	temperature [°C]
U	air velocity in the channel [m s^{-1}]
υ	kinematic viscosity [N s m^{-2}]
L	depth of the rotor [m]
w	water content of the desiccant material [kg kg^{-1}]

Greek notations

φ	relative humidity [%]
ρ	density [kg m^{-3}]
ω	humidity ratio of the air [kg kg^{-1}]
ω_s	humidity ratio of the air in equilibrium with the desiccant at saturation [kg kg^{-1}]

Subscripts

d	desiccant
r	regeneration
g	gas (air)
i	inlet
o	outlet
s	saturation
v	water vapor

The heat and mass transfer coefficients between the fluid flow and the porous materials in the desiccant wheels were studied theoretically in order to express the temperature and the humidity as a function of the properties of the desiccant channel coating material, the velocity of the airflow through the wheel and the rotational speed of the wheel [8,9].

A literature review shows that heat and mass coefficients for desiccant wheels were not measured directly. Correlations given in literature for heat transfer coefficients in channels, obtained in laboratory conditions, were used in the models of the wheel. The differences between the methods reside in the models describing the phenomena in the channels of the wheel [9–11]. These models, also called white-box models (*white* because the parameters with physical significance have values known before the construction of the model) have been developed for simulation purposes. They are not applicable to control purposes when the model needs to correspond to the wheel under the actual operating conditions. The models used for control purposes reproduce the input–output relation without using the governing physical laws. These models, also called black-box models (*black* because the parameters without physical significance, having unknown values at the construction of the model, are estimated by measurements) have the disadvantage that the values of the parameters depend on the operating point for which the identification was performed. Consequently, the parameters experimentally identified for one domain (i.e. a specific operating point) are not generally valid for another domain [12,13].

The present work is an attempt to combine the advantages of the white box and the black box approaches: a priori values for some parameters and the observed values for the others, respectively. The dynamic model of the wheel, obtained from physical considerations, is put in a state-space representation. Some of its parameters are given from known physical laws or geometric considerations (white box approach), while the rest of them are obtained by fitting the model to the experimental data (black box approach). The gray-box approach