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Concentration solar dryer water-to-air heat exchanger: Modeling and parametric studies

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ABSTRACT

In this paper we present a modeling and parametric studies of a water-to-air heat exchanger. This exchanger is formed of a fan blowing the air to be heated through a battery of smooth tubes where the hot water—coming from solar concentrators—circulates. The heated air is injected into a thermal room to dry the clay bricks.

In the first part, we study the most used models in the estimation of the heat transfer and air flow pressure drop across a tube bundle, and subsequently calculate the required transmitted power to the air.

In the second part, we focus on the parametric study of the influence of the different geometric parameters of the exchanger on the heat flow rate, the air outlet temperature, the pressure drop and the requested transferred power to the air. The considered parameters are: The water heat flow rate, the heat exchanger compactness, the rows arrangement, the tube diameter, the transverse pitch, the total number of tubes, the number of rows and the air velocity.

Simulations have shown that the heat exchanger performance could be improved essentially throughout the design and manufacturing process by modifying the different geometrical parameters and filling certain conditions.

Introduction

Drying is an essential process in many industrial applications such as textiles, phosphate, dairy processing, production of cement, waste water treatment, production of tiles and clay bricks, etc.

The energy required for drying can be provided from various sources, namely, electricity, fossil fuels, natural gas, wood and solar. The use of solar radiation for drying exists since ancient times. However, it has not been widely integrated in the industrial sector. Considering the depletion of natural energetic resources in the near future leading the rise of oil prices, solar drying is expected to become a necessity in the near future.

Several prototypes of solar dryers have been built and tested primarily for drying agricultural products [1], [2], [3] using four main modes namely, active, inactive, mixed and hybrid solar drying [4], [5]. There are also some studies on solar drying of phosphates [6], sludge [7], and of solid waste [8].

The cross flow heat exchanger is one of the most effective devices in the heat recovery for its numerous advantages, namely: the large amount of heat which can be transferred from a small exchange area, the simplicity of design and manufacturing, wide operating temperature range, the ability to control high heat flux at different temperature levels [9], [10].

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Nomenclature

A_{min}	Minimum free flow area, m^2
A_{surf}	Tube wall outside surface area, m^2
D	Tube diameter, m
f	Friction factor for smooth tubes
G_{max}	Maximum mass velocity, kg/m^2 s
G	Mass velocity$G=\rho V$, kg/m^2 s
h_{fluide}	Convective heat transfer coefficient of the HTF, W/m^2 K
$\overline{h}$	Average convective heat transfer coefficient, W/m^2 K
k_{fluid}	fluid thermal conductivity, W/m K
k_{tube}	Tube thermal conductivity, W/m K
k_{air}	Air thermal conductivity, W/m K
l_{tube}	Tube length, m
L	Total tubes length, m
$\dot{m}$	Air mass flow rate, kg/s
N_T	Number of tubes per row
N_L	Number of rows
$\overline{Nu}_D$	Average Nusselt number
Δp	Pressure drop, Pa
p_1	Air Pressure in the bank inlet, Pa
$Re_{D,max}$	Reynolds number at the maximum velocity
r_1	inner radius of tube, m
r_2	Outer radius of tube, m
S_T	Transverse pitch, m
S_L	Longitudinal pitch, m
S_D	Diagonal pitch, m
S_D	Dimensionless diagonal pitch
S_T	Dimensionless transverse pitch
S_L	Dimensionless longitudinal pitch
T_s	Surface temperature, K
T_m	Average temperature of the Heat transfer fluid, K
T_i	Inlet temperature of air, K
T_o	Outlet temperature of air, K
T_{h_1}	Heat transfer fluid temperature, K
T_{S_1}	Inside surface temperature of tubes, K
T_{S_2}	Outside surface temperature of tubes, K
ΔT_{lm}	Log-mean temperature difference, K
Pr_s	Prandtl number at the surface temperature
Pr	Prandtl number
u_m	Maximum HTF flow velocity, m/s
U_{app}	Approach velocity, m/s
V	Air velocity, m/s
V_{max}	Maximum air velocity in the minimum flow area, m/s
W	Transmitted Power, Watts
ρ	Air density, kg/m^2
ρ_1	Air density in the bank inlet, kg/m^3
ρ_2	Air density in the bank outlet, kg/m^3
σ	Ratio of minimum free flow area to frontal area $\sigma = \frac{A_{min}}{A_{Fr}}$
μ_w	Air viscosity at the wall temperature, N s/m^2
μ	Average free-stream viscosity, N s/m^2
χ	Correction factor
ν_1	Kinematic viscosity in the bank inlet, m^2/s
ν_2	Kinematic viscosity in the bank outlet, m^2/s
ν	Kinematic viscosity of air, m^2/s
ν_m	Kinematic viscosity of air, m^2/s
c_p	Specific heat, J/kg K

In this study, we investigate the modeling and simulation of heat transfer and pressure drop in flow across a tube bank within a water-to-air heat exchanger using in an industrial dryer of clay brick. This exchanger is formed of a fan blowing air across the tube bank where the heated water—coming from a solar concentrator—circulates.

First, we explored the different models adopted for the modeling of heat transfers and pressure drop, then we will lead a parametric study to show the variation influence of the different parameters of the exchanger on the heat flux, the outlet air temperature, the pressure drop, and the required transmitted power to move the air across the tube bundle.

Methodology

Heat transfer modeling

The equations governing the heat transfers in the heat exchanger can be modeled in two parts:

- The heat transfers from the HTF (water inside the tubes) to the outside surface of the tubes (Fig. 1).
- The heat transfer process between the outside wall and the air crossing the tube bundle.

Two types of tubes arrangement are considered: aligned and staggered arrangement. These arrangements are characterized by the dimensionless transverse, longitudinal, and diagonal pitches (Fig. 2).

In order to simplify the calculations, we adopted the following assumptions:

- We consider the steady-state and incompressible flow conditions.
- We neglect the change air on air properties with the temperature
- We neglect the radiation effects.

First part

Convective heat transfer from HTF to the inner surface of tube. In the case of a fluid passes through a circular tube, the velocity and temperature profiles at a given axial location might be estimated as being uniform and parabolic (Fig. 3) [12].

On the other hand, we can define the average temperature T_m as a practical reference temperature for internal flows, consequently, the Newton's law can be expressed as [12].

$$q_{r_1} = h_{fluid}(T_s - T_m) \quad (1)$$