Plate flat finned tubes heat exchanger: Heat transfer and pressure drop modeling

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Abstract. Finned tubes heat exchanger is used on an innovative industrial solar dryer which consists of three main components namely, solar concentrators, thermal room and heat exchanger. In this drying system, the heat transfer fluid coming from solar concentrators field circulates in the fin-and-tube bundles to heat the air moving through this battery to be injected into the thermal room to dry clay bricks. In this paper, we present a modeling and parametric study of the plate flat fin-and-tube bundles considering the most used models in the evaluation of the heat transfer and air flow performances. First, we carry out with a validation of the proposed physical and numerical models based on previous experimental results. In the second part we focus on the variation effects of the different geometric parameters of the tube bundles on the drying temperature and the pressure drop in order to estimate the performance evaluation criterions. Numerical results have shown that big external tube diameter increases the performance evaluation index by 67% when it varies from 20mm to 35mm which promotes the thermo-flow performances of the heat exchanger. However, small transverse and longitudinal pitches present a higher air velocity at the minimum flow area and a stronger flow disturbance which ensure better air flow and heat transfer characteristics.

Keywords: heat transfer, pressure drop, finned tubes heat exchanger, modeling, performance evaluation criterions, solar drying.

1. Introduction

Many industrial applications include drying in their manufacturing processes such as textiles, phosphate, dairy processing, production of cement, waste water treatment, production of tiles and clay bricks, etc.

Several sources can provide the required energy for this industrial process namely, electricity, fossil fuels, natural gas, wood and solar. For many decades, solar radiation was used for drying agricultural products based on some traditional systems. However, its integration in the industrial sector is very limited due to the low temperatures values reached by the existing systems. Considering the depletion of natural resources in the near future leading the rise of oil prices, the integration of solar drying in industries is expected to become a necessity. Active, inactive, mixed and hybrid solar drying [1,2,3] are the essential modes used in the existing prototypes of solar dryers which have been designed and tested for the drying agricultural products [4, 5, 6, 7, 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].
In order to evaluate the thermo-flow characteristics of flat finned tube bundles, we carry out with a parametric study of different geometric parameters of the heat exchanger which is conducted based on a physical model established according to the most practical models in the performance evaluation of the air flow and heat transfer. The thermo-flow performance involves many geometric parameters namely fin pitch, fin oblique angle, fin thickness, tube pitches and tube external diameter. The variation effect of the transverse and longitudinal pitches as well as the tube outside diameter will be investigated in this based in order to optimize the architecture of similar heat exchangers for a predefined operating conditions.

2. Methodology

2.1. Air-side flow and heat transfer models

The modeling of thermo-flow characteristics of the flat finned tube heat exchanger (Fig.1) is performed based on the following assumptions:

- Steady-state and incompressible flow conditions
- Unchangeable air properties with the temperature
- Neglected radiation effects.

The heat transfers on the air side of the finned tube bundles in staggered configuration (Fig.2) is modelled based on the following dimensionless parameters [11, 12]:

The Reynolds Number:

\[ Re = \frac{\rho U_m D}{\mu} \]
with:

\[ U_m = \begin{cases} U(S_T/2a) & \text{if } 2a \leq 2b \\ U(S_T/2b) & \text{if } 2a \geq 2b \end{cases} \]  \hspace{1cm} (2)

\[ 2a = (S_T - D) - [(S_T - D)\delta.n_f] \]  \hspace{1cm} (3)

\[ b = (S_D - D) - [(S_T - D)\delta.n_f] \]  \hspace{1cm} (4)

Where, \( \rho, \mu, \lambda \), are the density, dynamic viscosity and thermal conductivity respectively.

The friction factor:

\[ f = 2 \frac{\Delta p}{\rho U_m^2} \]  \hspace{1cm} (5)

\( \Delta p \) is the pressure drop across the finned tube bundles.

The Nusselt Number:

\[ Nu = h \cdot D / \lambda \]  \hspace{1cm} (6)

where \( h \) is the average heat transfer coefficient which is defined by:
The logarithmic mean temperature difference between the tube wall and the air is defined as:

$$\Delta T_{lm} = \frac{T_o - T_i}{\ln(T_w - T_i/T_w - T_o)}$$

For different number of rows, several works have been conducted to estimate the average heat transfer coefficient and the pressure drop through the calculation of the Nusselt number and the friction factor respectively based on the geometric parameters of the finned tube bundles.

For the 4 rows plate flat finned tube bundles Kong et al [13] have established the correlations based on longitudinal and transvers pitches, and tube diameter as follows:

$$f = 662.3561 \text{Re}^{-0.6453} \left(\frac{S_T}{D}\right)^{-0.2801} \left(\frac{S_L}{D}\right)^{-0.3927}$$

$$Nu = 2.6653 \text{Re}^{0.3175} \left(\frac{S_T}{D}\right)^{-0.8732} \left(\frac{S_L}{D}\right)^{-0.5618}$$

with:

$$1 \leq S_T/D \leq 1.8 , 1 \leq S_T/D \leq 1.8, 800 \leq Re \leq 13000$$

Other correlations have been proposed [14], for 6 rows heat exchanger, in function of longitudinal and transverse pitches, tube diameter, hydraulic diameter, fin oblique angle, fin thickness and fin spacing which are given by the following equations:

$$Nu = 0.3313 \text{Re}^{0.575} (1 - \alpha)^{-0.0522} \left(\frac{F_S}{D_h}\right)^{1.3204} \left(\frac{\delta}{D_h}\right)^{0.0351} \left(\frac{D}{D_h}\right)^{-0.1002}$$

$$f = 7.655 \text{Re}^{-0.4927} (1 - \alpha)^{0.1441} \left(\frac{F_S}{D_h}\right)^{1.7383} \left(\frac{\delta}{D_h}\right)^{0.0004} \left(\frac{D}{D_h}\right)^{0.0922}$$
\[ D_h = 4A_c/P_c \]

With \( A_c \) is the minimum free flow area and \( P_c = A/L_2 \) is the perimeter at the minimum flow cross-section.

The application ranges of this correlations are:

\[ 0.5476 \leq \frac{F_x}{D_h} \leq 0.678, \ 0.013 \leq \frac{\delta}{D_h} \leq 0.332, \ 2.228 \leq \frac{D}{D_h} \leq 9.143 \text{ and } 220 \leq Re \leq 5500 \]

For a wider application ranges in term of number of rows from 2 to 7 rows and based on the total number of tubes, fin spacing, tube diameter, and the longitudinal and transverse pitches Xie et al [15] have proposed the following correlations:

\[ Nu = 1.565 \cdot Re^{0.3414} \left( \frac{F_x}{D} \right)^{-0.165} \left( \frac{S_T}{S_L} \right)^{0.0558} \]
\[ f = 20.713 \cdot Re^{-0.3489} \left( \frac{F_x}{D} \right)^{-0.1676} \left( \frac{S_T}{S_L} \right)^{0.6265} \]

With:

\[ 0.67 \leq V \leq 4m/s, \ 1000 \leq Re \leq 6000, \ 16 \leq D \leq 20mm, \ 38 \leq S_T \leq 46mm, \ 32 \leq S_L \leq 36mm \]

The overall performance estimation of the finned tube bundles is defined by the performance evaluation criterion \( P_{EC} \) which is calculated by the following equation:

\[ P_{EC} = \frac{Nu}{f^{1/3}} \]

3. Results and discussion

3.1. Simulations reliability

In order to validate the physical model, we compare our simulations with the experimental results presented in previous works. Table 1 illustrates the used parameters in our simulations. Figure 4 and 5 show that the simulations are reliable for more than 90% and 97% respectively for the Nusselt number and friction factor estimations. Kong et al [13] model was considered due to its wide application ranges.

3.2. Variation effects of diameter

The variation effect of the tube diameter was studied between \( D= 20mm \) and \( 35mm \) for a constant non-dimension pitches of \( 35 \ mm \) and an air velocity of \( 0.5 \ m/s \) in order to fulfil the application ranges of the physical model.

The increase of the tube diameter reduces the cross section of air flow which generates the swirl flows due to the flow separation at the tubes rear part. This disturbed air flow improves the thermo-flow performance, but increases the pressure drop.

Figures 6 and 7 confirm this concept and show that both the drying temperature and the pressure drop increase with increasing the tubes diameter.
Table 1: Parameters used in simulations

| Parameters                        | Values                        |
|----------------------------------|-------------------------------|
| Ambient temperature $T_i$        | 289.15 K                      |
| Tube wall temperature $T_w$      | 331.72 K                      |
| $S_T$ and $S_L$                  | 0.035 m                       |
| Air kinematic viscosity          | $19.82 \times 10^{-6}$        |
| Thermal conductivity of air      | 0.0229 W/m.K.K                |
| Air density                      | 1.217 kg/m$^2$                |
| Specific heat                    | 1009 J/kg K                   |
| Number of tubes per row ($N_T$)  | 6                             |
| Total number of tubes (N)        | 24                            |
| Tube diameter                    | 0.025 m                       |
| Air velocity (V)                 | 0.5 m/s                       |

Figure 4: Simulation validation for Nusselt number

Figure 5: Simulation validation for Friction factor

Figure 8 shows that the performance evaluation index increase with the tube external diameter which means that the heat transfers are more important than the pressure drop, and allows us to conclude that the wide diameter is of benefits to the heat exchanger performances.

It is important to note that the increase of the tube external diameter reduces the fin surface area so it will be necessary to make a techno-economic assessment to determine the tubes diameter.

3.3. Variation effects of transverse pitch

In order to understand how the dimensionless transverse pitch contributes in heat transfer performances, we studied the impact of this parameter on the outlet air temperature, the pressure drop and the performance evaluation factor.
Figures 9 and 10 show that both the outlet air temperature and the pressure drop decrease, respectively, by 3.2% and 31.68% with the transverse pitch increase from 25mm to 45mm. This can be explained by the reduced air velocity and local air flow at the minimum flow area for a wider transverse pitch which consequently reduce the Reynolds and Nusselt numbers as well as the friction factor. As a combination of the friction factor and the Nusselt number the performance evaluation index decrease for wider transverse pitch as shown in Figure 11.

3.4. Variation effects of longitudinal pitch

Finally, we investigate the effects of the longitudinal pitch on the heat transfer and the air flow at a constant diameter and transverse pitch. The dimensionless longitudinal pitch is varied from 25mm to 45mm to fulfil the application ranges of the considered model.
It can be seen from figures 12 and 13 that the drying temperature and the pressure drop decrease for a wider longitudinal pitch due to weakened disturbance of the air flow. Based on the same figure we notice that the air outlet temperature decreases by 1.5% while the pressure drop reduces by 20%.

Figure 14 illustrates that small longitudinal tube pitch is superior to the large one in the overall heat transfer performances of the finned tubes heat exchanger.

According to the performance evaluation index which decreases by 46% with the transverse pitch (Fig.8) and only 22% with the longitudinal pitch (Fig.14) we conclude that the impact of the transverse pitch variation is more significant than that of the longitudinal pitch on the heat exchanger performances.
Figure 12: Temperature variation with longitudinal pitch

Figure 13: Pressure drop variation with longitudinal pitch

Figure 14: Variation of $P_{EC}$ with longitudinal pitch

4. Conclusion
Parametric study of the thermo-flow characteristics of the flat finned tube bundles is performed at various transverse and longitudinal pitches as well as different tube's external diameter. Based on the conducted simulations we concluded that a big diameter is of benefits to the air flow and heat transfer performances which is traduced by the increase of the performance evaluation index by 67% when the tube's external diameter varies from 20mm to 35mm.
Small transverse and longitudinal pitches present a higher air velocity at the minimum flow area and a stronger flow disturbance which ensure better thermo-flow characteristics. The proposed physical model and simulations allow us to evaluate the air flow and heat transfer performances in order to optimize the geometric parameters of the flat finned tubes heat exchanger. This study can be developed by considering the variations effects of different geometric parameters of fins namely the fin spacing and the fin thickness to achieve a complete analysis of the thermo-flow performances of the finned tube bundles.

It's important to note that all the simulations are conducted at a constant tubes wall temperature which is difficult to ensure when the heat exchanger is coupled with an intermittent heat production system as solar concentrators which require the evaluation of the thermo-flow performances of this heat exchanger at various heat transfer fluid temperatures to get a deeper analysis of the heat transfer performances in order to integrate this solar energy system in the industrial drying process of clay bricks.

5. References

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