Effect of Tube Diameter on The Design of Heat Exchanger in Solar Drying system

Shaymaa Husham Abdulmalek\textsuperscript{1} Morteza Khalaji Assadi\textsuperscript{1} Hussain H. Al-Kayiem\textsuperscript{1} Ali Ahmed Gitan\textsuperscript{2}

\textsuperscript{1}Universiti Teknologi PETRONAS, Solar Thermal Advanced Research Center, Bandar Seri Iskandar, Perak, Malaysia.

\textsuperscript{2}Tikrit University, Faculty of Engineering, Mechanical Engineering Department, Tikrit, Iraq.

morteza.assadi@utp.edu.my

Abstract. The drying of agriculture product consumes a huge fossil fuel rates that demand to find an alternative source of sustainable environmental friendly energy such as solar energy. This work presents the difference between using solar heat source and electrical heater in terms of design aspect. A circular-finned tube bank heat exchanger is considered against an electrical heater used as a heat generator to regenerate silica gel in solar assisted desiccant drying system. The impact of tube diameter on the heat transfer area was investigated for both the heat exchanger and the electrical heater. The fin performance was investigated by determining fin effectiveness and fin efficiency. A mathematical model was developed using MATLAB to describe the forced convection heat transfer between hot water supplied by evacuated solar collector with 70 °C and ambient air flow over heat exchanger finned tubes. The results revealed that the increasing of tube diameter augments the heat transfer area of both heat exchanger and electrical heater. The highest of fin efficiency was around 0.745 and the lowest was around 0.687 while the fin effectiveness was found to be around 0.998.

Keywords: (solar energy, air- water heat exchanger, silica gel, drying)

1. Introduction

High thermal energy storages are gained from evacuated tube solar collectors that use water to transfer heat to another system through an energy converter such as heat exchanger \cite{1-3}. Several different types of heat exchangers have been developed for different applications being the shell-and-tube and plate-fin the most commonly utilized configurations. Because many heat exchanger arrangements involve multiple rows of tubes in a fluid cross flow, the heat transfer characteristics for tube banks are of important practical interest. The gas flowing on the outside impinges perpendicularly on these tubes \cite{5}. With constant fluid velocity, this type of flow gives rise to an increase in turbulence compared with the case where the gas flows along the tubes and parallel to them. Heat exchangers are extensively used in diverse industrial processes nowadays. Many previous researches have presented different designs of heat exchanger used in different applications such as reactivation the silica gel as shown in Figure1 by using many sources of heat for example, solar energy \cite{6, 7}. In an industrial application, the silica gel is regenerated by using fossil full like electrical heater at 100 °C. A desiccant system was used to dry crushed oil palm fronds \cite{6, 8}. Solar energy was used to regenerate the silica gel through a heat exchanger. A theatrical analysis was carried out with an optimization of gas-to-gas heat exchanger with a non-constant cross sectional area \cite{9}. The results obtained from the modeling demonstrate the premise that it is possible to realize designs for heat exchangers that are highly exergy-efficient and very cheap. Some
researchers studied the performance of heat exchanger starting from the CFD simulations at the microscale and was thus proposed. A set of control volumes that include the finned surfaces of both the cold and hot sides (separated by a wall) were detected [10-12]. This work aims to investigate the effect of diameter of tube bank heat exchanger on the heat transfer area of heat exchanger in comparison with an electrical heater for the purpose of desiccant regeneration in solar assisted desiccant drying system.

Figure 1. Desiccant wheel air process

2. Heat exchanger input and mathematical model

The concept of the circular finned tub bank heat exchanger is shown in Figure 2 was adopted for building the mathematical model. The input parameters required for the mathematical model is defined in Table 1.

Figure 2. Tube bank water-air heat exchanger

Table 1. Input Parameters for Heat Exchanger Design

| Parameter | $T_{at}$ (k) | $D_o$ (inch) | $T_{ad}$ (k) | $T_{wi}$ (k) | $S$ (W/m²) | $\eta_c$ (%) | $F$ | $C_r$ | $N_r$ | $N_f$ | $A$ (m²) | $N_{row}$ | $N_{col}$ |
|-----------|--------------|---------------|--------------|--------------|-------------|--------------|-----|------|------|------|----------|---------|----------|
| value     | 303          | 0.375         | 328          | 343          | 800         | 0.95         | 0.95| 15   | 4    | 5    | 2        | 2       |

The heat gained from solar radiation ($S$) through solar collector is calculated by Eq. (1).
\[ Q_{\text{act}} = \eta \cdot A \cdot S \]  
(1)

The energy balance between air and water is modeled in equations (2-4).

\[ Q_{\text{act}} = Q_w = Q_a \]  
(2)

Heat absorbed by air, is

\[ Q_a = m_a \cdot C_p_a \cdot (T_{a1} - T_{a0}) \]  
(3)

Which as same as the heat released from water,

\[ Q_w = m_w \cdot C_p_w \cdot (T_{wi} - T_{wo}) \]  
(4)

To predict the heat transfer from the bank of tubes to air flow, heat exchanger roles may be applied. Hence, Reynolds number, as the key parameter in the internal forced convection heat transfer is a prominent factor. For the present geometry, \( Re \) for the tube water flow,

\[ \frac{\rho_w \cdot V_w \cdot D_t}{\mu_w} \]  
(5)

The Nusselt number for internal flow inside circular tube is given by Equation (6) and (8) for laminar and turbulent flow, respectively [6].

\[ Nu_w = 4.364 \left( 1 + \left( \frac{Gz}{29.6} \right)^{\frac{1}{5}} \right)^{\frac{1}{2}} \left( 1 + \left( \frac{Gz}{10000} \right)^{\frac{1}{5}} \right)^{\frac{1}{2}} \left( 1 + \left( \frac{Gz}{10000} \right)^{\frac{1}{5}} \right)^{\frac{1}{2}} \right)^{\frac{1}{2}} \text{ For } Re_w \leq 2300 \]  
(6)

\[ Nu_w = 0.023 \, Re_w^{0.8} \, Pr_w^{0.3} \text{ For } Re_w > 2300 \]  
(7)

Where:

\[ Gz = \left( \frac{\mu_l}{4} \right) \left( \frac{L}{D_{\text{Re}w} \cdot Pr_w} \right)^{-1} \]  
(8)

The Nusselt number of external flow over hot tube bank array is given by equations (9-14) [6].

\[ Nu_a = 0.9 \, C_n \, Re_a^{0.4} \, Pr_a^{0.36} \left( \frac{Pr_a}{Pr_{as}} \right)^{0.25} \text{ For } 1 < Re_a < 10^2 \]  
(9)

\[ Nu_a = 0.52 \, C_n \, Re_a^{0.5} \, Pr_a^{0.36} \left( \frac{Pr_a}{Pr_{as}} \right)^{0.25} \text{ For } 10^2 < Re_a < 10^3 \]  
(10)

\[ Nu_a = 0.27 \, C_n \, Re_a^{0.63} \, Pr_a^{0.36} \left( \frac{Pr_a}{Pr_{as}} \right)^{0.25} \text{ For } 10^3 < Re_a < 2 \times 10^5 \]  
(11)

\[ Nu_a = 0.033 \, C_n \, Re_a^{0.8} \, Pr_a^{0.4} \left( \frac{Pr_a}{Pr_{as}} \right)^{0.25} \text{ For } 2 \times 10^5 < Re_a < 2 \times 10^6 \]  
(12)

\[ \text{Where } Re_a = \frac{\rho_a \cdot V_a \cdot D_t}{\mu_a} \]  

All air properties are taken at film temperature defined as \( T_f = \frac{T_{a1} + T_{a0}}{2} \) by considering surface temperature \( T_s = \frac{T_{wct} + T_{wo}}{2} \) except \( Pr_{as} \), which is at surface temperature. The overall heat transfer coefficient \( (U) \) of heat exchanger is determined as follows.

\[ U = \frac{\rho_l \cdot D_t \cdot (R_w + R_a + R_s)}{1} \]  
(13)
Where $R_w$, $R_s$ and $R_a$ are thermal resistances of water side, solid wall and air side, respectively and they are given by

$$R_w = \frac{1}{h_w d_w}, \quad R_s = \frac{\ln(h_w)}{2\pi L k_s} \quad \text{and} \quad R_a = \frac{1}{h_a d_a} \quad (14)$$

Where $h_w = \frac{Nu_w k_w}{D_i}$, $h_a = \frac{Nu_a k_a}{D_o}$ and the thermal conductivity of copper ($k_s = 400 \text{ W/m.K}$).

For unmixed flow heat exchanger, the temperature changes in logarithmic trend. The temperature difference can be calculated by using the logarithmic mean temperature difference (LMTD) as in the following equation.

$$\text{LMTD} = \frac{(T_{wi} - T_{ao}) - (T_{wo} - T_{ai})}{\ln\left(\frac{(T_{wi} - T_{ao})}{(T_{wo} - T_{ai})}\right)} \quad (15)$$

Equation 15 can be used for counter and parallel flow heat exchangers while for cross flow heat exchanger, it needs to use a correction factor ($F$). The total heat transfer area of heat exchanger is calculated as follows.

$$A = \frac{Q_{act}}{UF(LMTD)} \quad (16)$$

The fin performance can be determined by calculating fin efficiency and effectiveness.

$$\eta_f = \frac{\tanh(MLc)}{MLc} \quad (17)$$

Where $M = \left(\frac{h_p f}{K_s A_t}\right)^{0.5}$ and the critical length ($Lc = L + \frac{L}{2}$).

$$\epsilon = 1 - \left(\frac{h_a}{A}\right) (1 - \eta_f) \quad (18)$$

The MATLAB algorithm is divided into two main parts. First part is related to heat exchanger calculations which aim to predict the length of heat exchanger from heat transfer area. Starting with initial value of heat exchanger length ($L = 0.5 \text{ m}$) and set of input data (Table 1), the MATLAB code solves in closed iteration loop the value of heat transfer area as modeled in Eq. (3 – 16). A convergence criterion is adopted to predict the heat exchanger pipe length by comparing the old value with its new value. Second part is about the electrical heater calculations which performed separately out of the iteration loop to determine the heat transfer area of the heater.

3. **Result and discussion**

The water-air heat exchanger for regeneration purpose of silica gel material at 55°C was designed and analyzed. The aim of this work is to use renewable energy instead of fossil fuel in desiccant regeneration process. Usually the silica gel is regenerated by using electrical heater of 100 °C surface temperature. In the current work, a heat exchanger was designed to replace the electrical heater by solar energy. The results showed that the design of tube bank heat exchanger was almost the same in terms of heat transfer area with the electrical heater as illustrated in Figure 3 and Figure 4. The total and finned areas of heat exchanger were influenced by changing the diameter of the tube as shown in Figure 3. Generally, an increasing of tube diameter led to an increase in both areas. This gives an indication of that to minimize
the size of heat exchanger, a reduction in heat transfer area is required by selecting smaller diameter of 0.375 inch that provides the same outlet air temperature from the heater of around 55 °C. The finned area was slightly less than total area and the unfinned area represented that difference. The fin efficiency as a measurement of fin performance was investigated as well in this work. Figure 5 represents the variation of fin efficiency with the outlet diameter of heat exchanger tube as a design input parameter. The efficiency increases with the increasing of the diameter until reaches its maximum value of around 74.5% at a diameter value of around 20.3 mm. By further increasing of diameter, the efficiency degrades. The fin effectiveness has been considered to show the performance of fin and it was found to be around 0.998 which is almost unity. This means any additional increase in fin diameter will not enhance the heat transfer.

Figure 3. The effect of outer diameter on total area and finned area of heat exchanger

Figure 4. The effect of outer diameter on the area of the electrical heater

Figure 5. The effect of outer diameter on fin efficiency of heat exchanger
4. Conclusion

In conclusion, this work gives a clear indication about the use of solar energy to regenerate the silica gel at a suitable temperature. The results illustrate the effect of tube diameter of tube bank water-air heat exchanger on different designs and operation conditions. It was found that the heat exchanger at diameter of 0.375 inch can provide hot air at 55°C to regenerate the silica gel. Also, the fin efficiency was in the range of (0.745 - 0.687) while the fin effectiveness was found to be around 0.998.

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