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EXPERIMENTAL INVESTIGATION OF THE THERMAL PERFORMANCE OF AN AIR-TO-AIR HORIZONTAL HEAT PIPE HEAT EXCHANGER

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ABSTRACT

An experimental study has been performed to investigate the thermal performance of a 2-row copper-R134a horizontal heat pipe heat exchanger in different working conditions. During the tests, dry bulb temperature of evaporator inlet has been controlled at 27-35 $^{\circ}C$ while the condenser inlet dry bulb temperature was kept constant at about 24 $^{\circ}C$. The influence of face velocity on the performance of heat pipe heat exchanger was also studied. The thermal resistance of the single heat pipe identical to the heat pipes embedded in the heat pipe heat exchanger was estimated using the recommended mathematical correlations and compared to the experimental results. The investigation has been carried out for 30 experiments and the results are presented in the present paper.

Keyword: Experimental study, Heat pipe heat exchanger, Thermal performance.

NOMENCLATURE

A	Heat transfer area (m^2)
c_p	Specific heat of the ambient air ($J/(kgK)$)
D	Diameter (m)
DBT	Dry bulb temperature ($^{\circ}C$)
EBR	Energy balance ratio
F.C	Fluid charge (%)
HPHX	Heat pipe heat exchanger
h	Specific enthalpy (kJ/kg)
k	Thermal conductivity ($W/(mK)$)
L	Length (m)
$\dot{m}$	Mass flow rate (kg/s)
Q	Heat transfer rate (W)
R	Thermal resistance (K/W)
T	Temperature ($^{\circ}C$)
ΔT	Temperature difference (K)
Greek symbols	
ε	Effectiveness
η	Surface effectiveness
ζ	Porosity
α	Heat transfer coefficient (W/m^2K)
Subscripts	
act	Actual
e	Evaporator
eff	Effective
e.s	External surface
i	Inner
max	Maximum
min	Minimum
o	Outer
s	Solid phase
s.t	Single tube
sen	Sensible
tot	Total
w	Wall

1. INTRODUCTION

Heat pipe heat exchangers (HPHXs) have many advantages such as high heat recovery effectiveness, no moving part, compactness, no external power, and high reliability. Therefore, because of considerable advantages, heat pipe heat exchangers have been extensively employed in different industry such as energy engineering and air conditioning systems. One of the important applications of heat pipe heat exchangers is the recovery of energy from exhaust air in the buildings. Because of dehumidification and energy saving aspects of heat pipe heat exchangers, application of this recovery device can lead to a vast amount of energy saving.

A number of investigations have been conducted to obtain the thermal performance of HPHXs. Because of simplicity of thermosyphones, most of the researchers used a thermosyphone to investigate the effect of different parameters on the HPHX thermal performance. Mostafa and Mousa (2007) conducted a research using a horizontal HPHX having four layers of 100 mesh brass screen and R-11 as the working fluid. Two streams of fresh and exhaust air was connected with heat HPHX to study the thermal performance at different mass flow rate between two sections of HPHX with different temperature conditions. It was found that the temperature change of fresh and return air are increasing with the increase of inlet temperature of fresh air. Noie (2006) studied the thermal performance of an air-to-air thermosyphone heat exchanger. The experimental results were compared with theoretical results. It was found that the minimum effectiveness of thermosyphone took place with equal mass flow rate in evaporator and condenser side. Yau and Tucker (2003) investigated the thermal performance of a six-row thermosyphone HPHX operating in tropical buildings.

In this paper, the thermal performance of a horizontal heat pipe heat exchanger consisting of 22 tubes arranged in two rows in the test rig was investigated. The variable parameters, which were being altered, were the face velocity and evaporator inlet air temperature. Besides, the thermal resistance of a single heat pipe identical to the heat pipes embedded in the HPHX was determined by available models and compared with the experimental results.

2. HORIZONTAL HEAT PIPE HEAT EXCHANGER

The HPHX was consisted of 22 copper tubes in 2 rows of 11 tubes. The tubes are in staggered position and 3 layer of stainless steel 100 wire mesh were pressed against the internal tube wall as a wick structure. The R-134a was used as the working fluid and was charged to the amount required to saturate the wick. The total length of each tube is 1020 mm, with 420 mm in the evaporator and condenser sections and 180 mm for the adiabatic section. The heat pipe tubes had 12 fins per inch, each 0.15 mm thickness. The detailed specification of HPHX is listed in Table 1.

Table 1 Design specification of the HPHX

Dimensions	420 mm wide, 350 mm high, and 53 mm deep
Number of coil rows	2 rows of 11 tubes, OD:13.4 mm, ID:12.7 mm
Tubes arrangement	Staggered
entre-to-centre	Transverse: 31.75 mm, Longitudinal: 27.5 mm
Fin	Aluminum corrugated, wavy plate, 12 fin per inch, fin thickness: 0.15mm
Wick structure	3 layers of stainless steel wire mesh, 100 mesh per inch

3. EXPERIMENTAL APPARATUSE AND TEST PROCEDURE

In order to study the thermal performance of the horizontal HPHX a test rig consisting of an air duct of $0.42 \times 0.35\, m^2$, HPHX, an electric heater and a variable speed fan was set-up as illustrated in Figure 1.

The dry bulb temperatures (DBT) of the air at measuring points were measured using RTD sensors and the output signals from these sensors were recorded via a data logging system. DASYlab software was employed to analyze the data. All the RTD sensors were calibrated in the range of 10 to 50 $^{\circ}C$ against a laboratory Dry-Well calibrator having an accuracy of +/-0.1 $^{\circ}C$. Velocity was measured by a digital flow meter having the accuracy of +/- 0.015 m/s by carrying out the velocity traverse through 25 point grid in the testing duct based on standard (ANSI/ASHRAE standard 41.2-1987) requirements.

4. MATHEMATICAL MODELE OF THE THERMAL RESISTANCE IN THE SINGLE HEAT PIPE

A single heat pipe similar to the heat pipes under test was considered for the simulation in the present study. A thermal resistance network is generally used to predict the thermal performance of the heat pipe. In this thermal resistance network, every component of the heat pipe is modeled by a relevant thermal resistance (Peterson, 1994; Dunn & Reay, 1978). Three main thermal resistances including air-side (fins), wall resistance and wick structure resistance for evaporator and condenser sections were determined to estimate the total thermal resistance of the single saturated heat pipe.

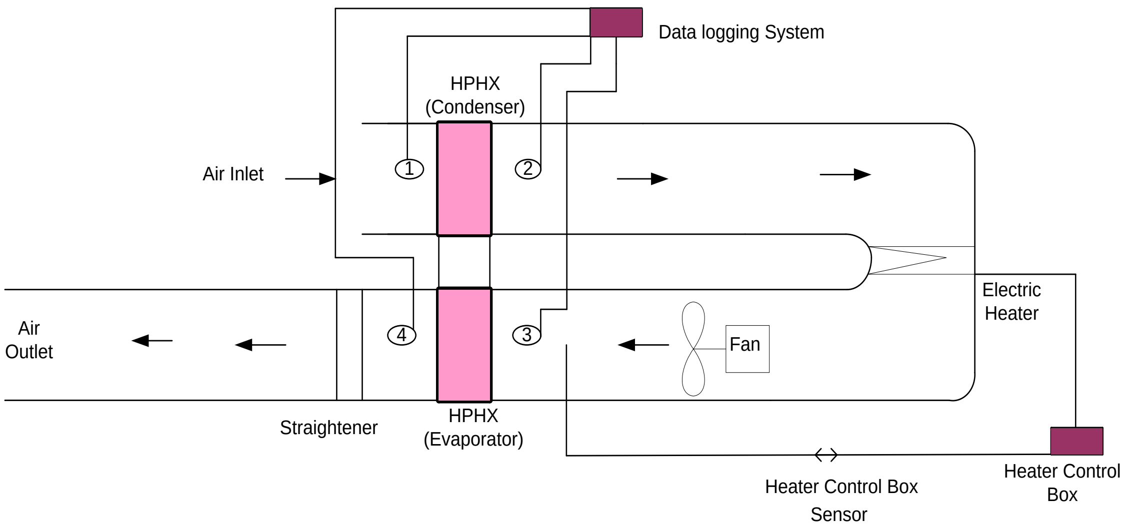

Figure 1 Schematic diagram of test set-up (top view)

4.1 External Surface Thermal Resistance

The mean heat transfer coefficient of the external surface ($\alpha_{e.s}$) was estimated for the air normal to a staggered finned tube bank (Azad *et al.*, 1985). Then, the related thermal resistance was estimated using (1).

$$R_{e.s} = \frac{1}{\eta_{e.s}\alpha_{e.s}A_{e.s}} \tag{1}$$

4.2 Wall Thermal Resistance

The heat pipe wall thermal resistance was calculated from the following equation:

$$R_w = \frac{\ln(D_o / D_i)_w}{2\pi L k_w} \tag{2}$$

4.3 Wick Structure Thermal Resistance

Regarding the available information about the wick structure of the heat pipes and existing effective thermal conductivity models for the wire screen wick structures, the recommended correlation (Li & Peterson, 2006) for predicting the effective thermal conductivity of wick structure was employed and is given in (3).

$$K_{eff} = \frac{(1-\zeta)K_s}{1+11\zeta} \tag{3}$$

Then, the thermal resistance of wick structure was determined from the following equation (Peterson, 1994):

$$R_{wick} = \frac{\ln(D_o / D_i)_{wick}}{2\pi L k_{eff}} \tag{4}$$

All the resistances were determined for the evaporator and condenser sections separately and the sum of them was considered as the total thermal resistance of the heat pipe.

Estimation of the experimental thermal resistance of a single heat pipe was determined by measuring the temperature difference across the evaporator and condenser side HPHX (ΔT) and the heat flux of the single heat pipe ($Q_{s.t}$). The average heat transfer for a single tube was estimated from the total heat transfer between two sections of the heat exchanger. Then, the total resistance for a single heat pipe was determined from the Fourier's law.

$$R_{s.t} = \frac{\Delta T}{Q_{s.t}} \tag{5}$$

5. EXPERIMENTAL EFFECTIVENESS CALCULATION PROCEDURE

The effectiveness is judged to be the most relevant in characterizing the performance of a HPHX (ANSI/ASHRAE Standard 84-1991; Yau, 2007; Guo *et al.*, 1998). The effectiveness ε as in (6) is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate.

$$\varepsilon = \frac{Q_{act}}{Q_{\max}} \tag{6}$$

And

$$Q_{act} = \dot{m}_e (h_3 - h_4) \tag{7}$$

And

$$Q_{\max} = \dot{m}_{\min}(h_3 - h_1) \tag{8}$$

From (6)-(8), the total effectiveness can be determined by:

$$\varepsilon_{tot} = \frac{\dot{m}_e}{\dot{m}_{\min}}\left(\frac{h_3 - h_4}{h_3 - h_1}\right) \tag{9}$$

Where h_4 is the evaporator outlet air specific enthalpy (state 4), h_3 is the evaporator inlet air specific enthalpy (state 3) and h_1 is the condenser inlet air specific enthalpy (state 1), see Figure 1. $\dot{m}_e$ is the mass flow rate of the evaporator side and $\dot{m}_{\min}$ is the smaller of the mass flow rate in evaporator and condenser side. However, in the present research the mass flow rate is equal in the evaporator and condenser side. Therefore, (9) can be simplified to:

$$\varepsilon_{tot} = \left(\frac{h_3 - h_4}{h_3 - h_1}\right) \tag{10}$$

Equation (10) can also be used to determine the sensible effectiveness with the exception that dry bulb temperature T is used instead of specific enthalpy and the sensible effectiveness can be represented by:

$$\varepsilon_{sen} = \left(\frac{T_3 - T_4}{T_3 - T_1}\right) \tag{11}$$

6. RESULTS AND DISCUSION

Many factors influence the thermal performance of a HPHX. In this study, the operating parameters investigated were:

- Face velocity values in the range of 1.5 m/s-2.5 m/s with 0.2 increments.
- Evaporator inlet DBT values of 27 $^\circ C$, 29 $^\circ C$, 31 $^\circ C$, 33 $^\circ C$ and 35 $^\circ C$.

Therefore, there was a total of 30 (6×5) experimental runs. For simplicity in presentation and discussion of the data, data are grouped according to the above nominal values, but because of some difficulties in achieving precise control of parameters, there was an inevitable departure from run-to-run compared with target values listed.

The following sub-sections describe the performance results as well as the comparison of theoretical and experimental thermal resistance for the single heat pipe and error analysis. The performance results are discussed first in sub-section 6.1 and the comparison of results and error analysis will be given later in sub-sections 6.2 and 6.3.

6.1 Performance Results

The energy balance ratio (EBR) as indicated in Table 2 is defined as the energy extracted from the air passing through the evaporator divided by energy transferred to the condenser. It can be seen from Table 2 that EBR values for most of the test examined are not equal to one (EBR should be equal to unity if all the energy extracted from the air passing through the evaporator was transferred to air passing through the condenser).
Three possible reasons for the EBR deviation from unity are possible intervening heat transfer between evaporator and condenser, possible heat transfer to the ambient air because of insufficient insulation and no uniform temperature profile in measuring stations because of the not well-mixed air. The intervening heat transfer test was conducted with zero fluid charge to check any possible intervening heat transfer between evaporator and condenser side and any possible heat transfer to surrounding air. It was found that for all the tests, the maximum temperature difference between the air stream in evaporator and condenser section was 0.1 $^{\circ}C$.Therefore, the heat transfer through conduction of tubes and heat transfer to the surrounding air were negligible. It was obvious that not well-mixed air at measuring points caused deviation of the EBR from unity.The sensible effective range was tabulated in Table 3 for the velocity range of study. Moreover, for convenient the influence of evaporator inlet DBT on sensible effectiveness of HPHX was shown in Figure 2.

Table 2 Energy balance ratio (EBR) between evaporator and condenser sections of HPHX
$EBR = (T_3 - T_4)/(T_2 - T_1)$

	Evaporator inlet DBT ($^{\circ}C$)				
	27	29	31	33	35
Face velocity	EBR				
1.5 m/s	1.13	1.08	1.08	1.2	1.03
1.7 m/s	0.98	1.03	1.1	1.15	1.11
1.9 m/s	1.06	0.94	1.17	1.14	1.16
2.1 m/s	1.11	1.11	1.15	1.21	1.12
2.3 m/s	0.97	1.08	1.19	1.18	1.1
2.5 m/s	1.05	0.99	1.15	1.1	1.11

As it can be seen from Figure 2, the sensible effectiveness was almost constant in the temperature range of study. Therefore, it can be found that the effectiveness was not influenced by the evaporator inlet DBT in the range studied. The influence of higher DBT on the sensible effectiveness was not studied since the main objective of the present research was to find the optimum sensible effectiveness in the temperature range of 27–35 $^{\circ}C$ for the purpose of future research.
Figure 3 shows the effect of face velocity on the evaporator outlet temperature. It can be seen that the outlet temperature increases slightly as velocity increases. Moreover, Figure 4 shows the influence of face velocity on the sensible effectiveness. The sensible effectiveness by the form of (11) suggests that the effectiveness is not the function of mass flow rate since the mass flow rate is equal in the evaporator and condenser sections. The small decrease in the effectiveness and also evaporator outlet temperature increase may be attributed to the fact that by increasing the face velocity or mass flow rate, the duration time to exchange heat is insufficient and consequently, heat transfer between air and tubes decreases.

6.2 Comparison of the Theoretical and Experimental Thermal Resistances

The predicted thermal resistance for the single saturated heat pipe experiencing the test condition was compared to the experimental results in Figures 5-7. As it can be seen from Figures 5-7, the agreement between the experimental and theoretical results increases as the face velocity is increased.

Table 3 Sensible effectiveness range (%), (min-max)

Face velocity (m/s)	Effectiveness range (min-max)
1.5	26.04-28.8
1.7	24.77-27.44
1.9	22.23-26.22
2.1	22.23-25.13
2.3	20.37-23.93
2.5	19.33-23.15

6.3 Error Analysis

An error analysis was conducted for some representative experimental runs. The uncertainties of energy balance ratio (EBR) and sensible effectiveness were calculated by using the root-sum-square (RSS) method. Table 4 shows the uncertainties for energy balance ratio and sensible effectiveness in some representative experimental runs.

Table 4 Uncertainty for energy balance ratio (EBR) and sensible effectiveness for some representative tests

Representative tests for uncertainties	Energy Balance Ratio (EBR)	Sensible Effectiveness
face velocity=2.1m/s and evaporator inlet temperature=31 $^{\circ}C$	+/-12.15%	+/-8.79%
face velocity=1.7m/s and evaporator inlet temperature=31 $^{\circ}C$	+/-10.9%	+/-7.75%

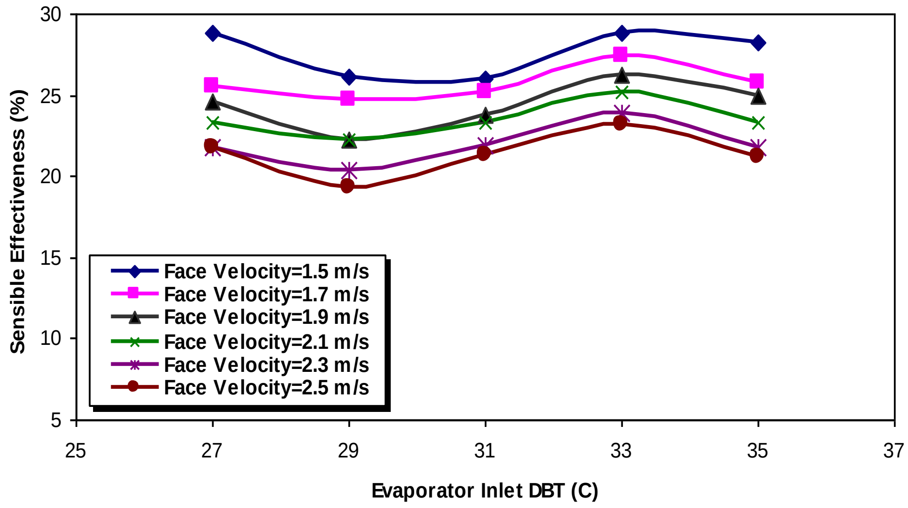

Figure 2 Sensible effectiveness versus evaporator inlet DBT.

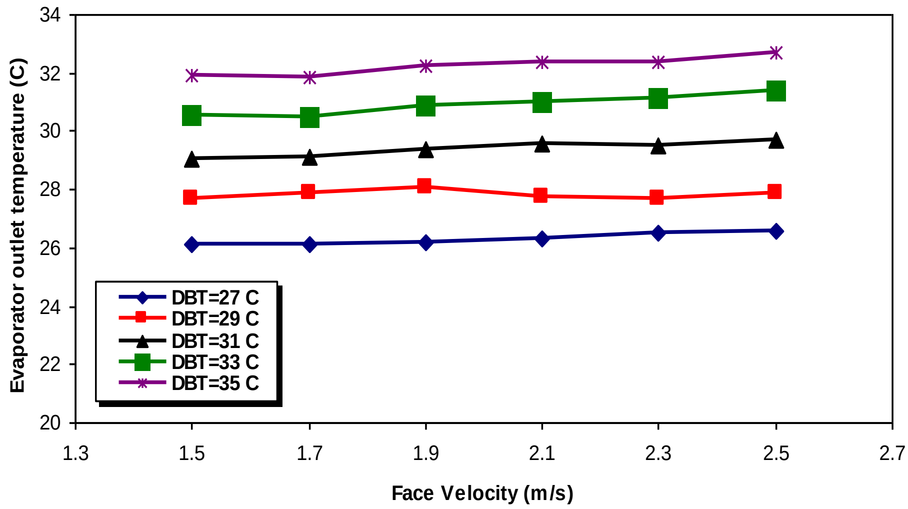

Figure 3 Evaporator outlet temperature versus face velocity.

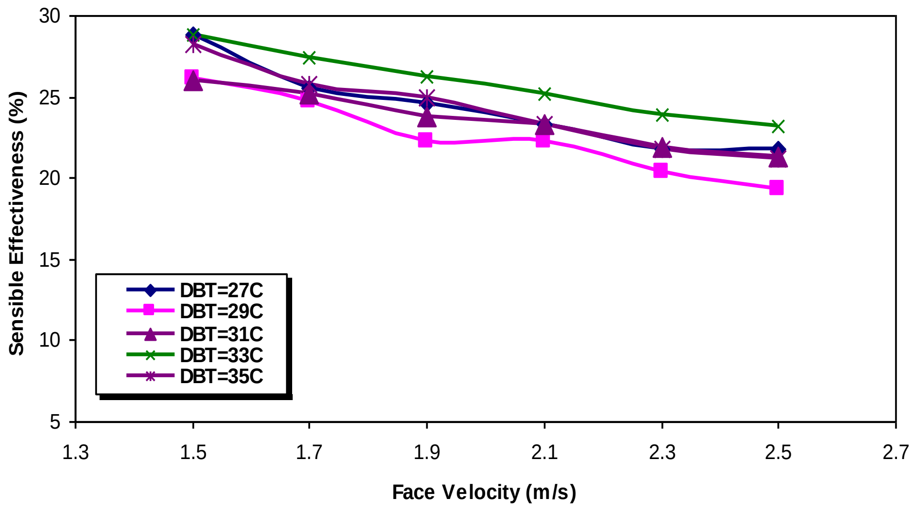

Figure 4 Sensible effectiveness versus face velocity.

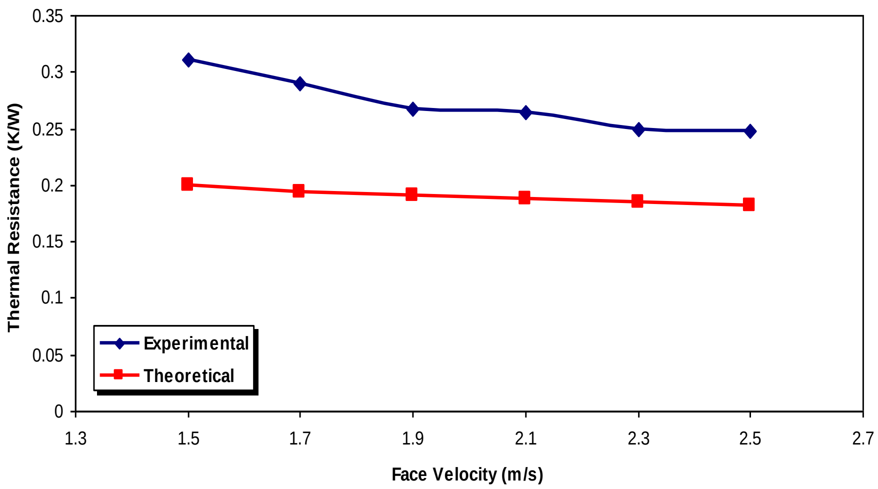

Figure 5 Comparison of experimental and theoretical thermal resistance at evaporator inlet DBT 31 $^{\circ}C$.

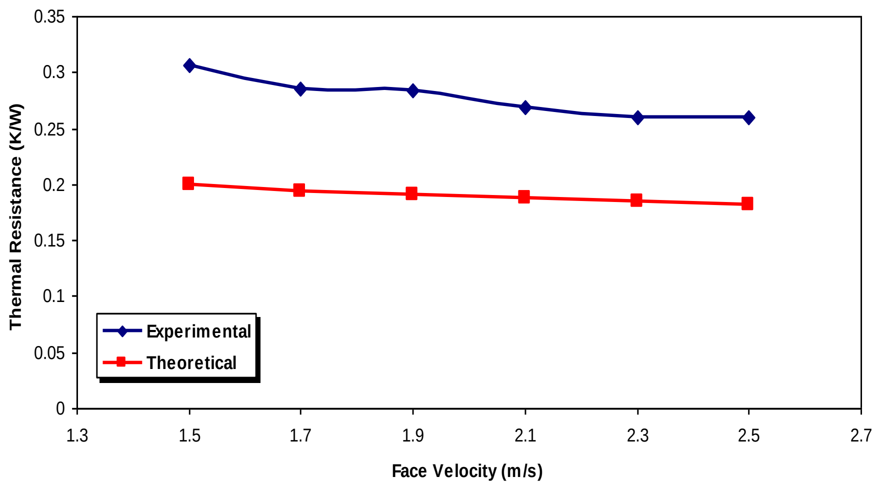

Figure 6 Comparison of experimental and theoretical thermal resistance at evaporator inlet DBT 33 $^{\circ}C$.

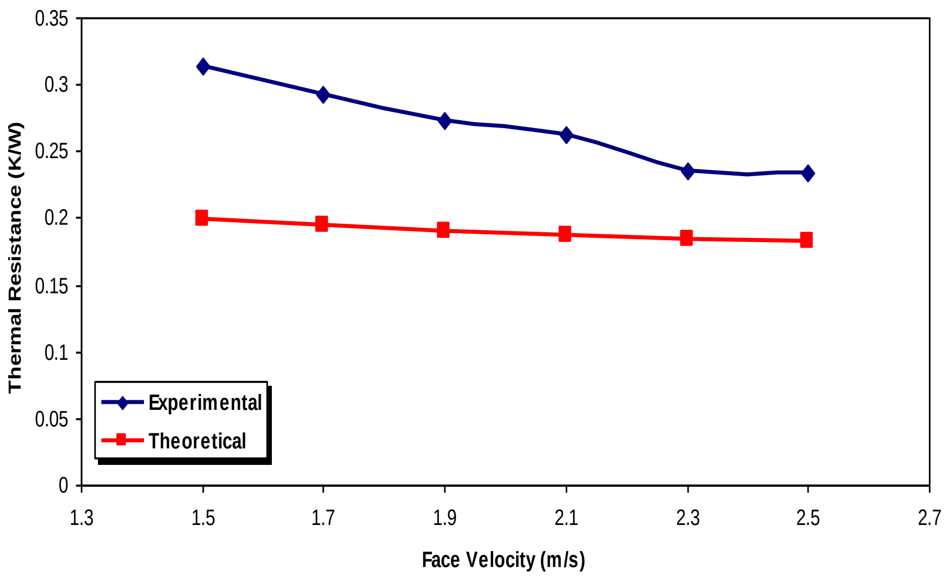

Figure 7 Comparison of experimental and theoretical thermal resistance at evaporator inlet DBT 35 $^{\circ}C$.

7. CONCLUSION

In the present research, the effect of parameters such as air velocity and evaporator inlet temperature on the performance of a typical 2-row air-to-air HPHX was investigated. The HPHX was tested with different face velocities at 27-35 $^\circ C$ evaporator inlet temperature. It was found that for all the cases studied, the sensible effectiveness slightly decreased as face velocity increased. Moreover, the experimental results showed that in the temperature range of study the sensible effectiveness was nearly constant and evaporator inlet air temperature has no major effect on the HPHX effectiveness. A theoretical simulation was made on the thermal resistance of the single heat pipe experiencing the test condition. The comparison of experimental and theoretical results showed that the agreement between experimental and theoretical results agreement becomes better as velocity increased.

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REFRENCES

Mostafa, A.A., Mousa, M.M., 2007. Heat pipe heat exchanger for heat recovery in air conditioning, Applied Thermal Engineering, Volume 27, pp. 795-801.

Noie, S.H., 2006, Investigation of thermal performance of an air-to-air thermosyphon heat exchanger using ε–NTU method, Applied Thermal Engineering, Volume 26, pp. 559-567.

Yau, Y.H., Tucker, A.S., 2003. Thermal performance study of a wet six-row heat-pipe heat exchanger operating in tropical buildings, International Journal of Energy Research., Volume 27, pp. 187-202.

ANSI/ASHRAE, Standard Methods for Laboratory Airflow Measurement, ASHRAE, Atlanta, Standard 41.2-1987 (RA 92).

Peterson, G.P., 1994. An Introduction to Heat Pipes: Modeling, Testing, and Applications, John Wiley & Sons, New York.

Dunn, P., Reay, D.A., 1978. Heat Pipes, Pergamon Press, Oxford, London.

Azad, E., Mohammadieh, F., Moztarzadeh, F., 1985. Thermal performance of heat pipe recovery systems, Journal of Heat Recovery Systems, Volume 5, pp. 561-570.

Li, C., Peterson, G.P., 2006. The effective thermal conductivity of wire screen, International Journal of Heat and Mass Transfer, Volume 49, pp. 4095-4105.

ANSI/ASHRAE, Method of Testing Air-to-Air Heat Exchangers, ASHRAE, Atlanta, Standard 84-1991.

Yau, Y.H., 2007. Experimental thermal performance study of an inclined heat pipe heat exchanger operating in high humid tropical HVAC systems, International Journal of Refrigeration, Volume 30, pp. 1143-1152.

Guo, P., Ciepliski, D.L., Besant, R.W., 1998. A testing and HVAC design methodology for air-to-air heat pipe heat exchangers, HVAC & R Research, Volume 4, pp. 3-26.