HEAT PIPE HEAT EXCHANGER AND ITS POTENTIAL TO ENERGY RECOVERY IN THE TROPICS

by

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The heat recovery by the heat pipe heat exchangers was studied in the tropics. Heat pipe heat exchangers with two, four, six, and eight numbers of rows were examined for this purpose. The coil face velocity was set at 2 m/s and the temperature of return air was kept at 24 °C in this study. The performance of the heat pipe heat exchangers was recorded during the one week of operation (168 hours) to examine the performance data. Then, the collected data from the one week of operation were used to estimate the amount of energy recovered by the heat pipe heat exchangers annually. The effect of the inside design temperature and the coil face velocity on the energy recovery for a typical heat pipe heat exchanger was also investigated. In addition, heat pipe heat exchangers were simulated based on the effectiveness-number of heat transfer units method, and their theoretical values for the thermal performance were compared with the experimental results.

Key words: *energy recovery, heat pipe heat exchanger, tropics, effectiveness-number of heat transfer units method*

Introduction

Ventilation systems are designed to exhaust the conditioned air from inside of a structure, and replace the air with up to 100% fresh outdoor air, which must then be treated to bring the air within the designed indoor thermal condition. The exchange of indoor air for outdoor air represents a considerable wastage of energy, especially in places, which need a higher rate of the air change per hour.

In hot and humid tropical countries such as Malaysia, heating, ventilating, and air-conditioning (HVAC) systems are installed in most of the buildings to provide a comfortable indoor environment. Consequently, the operating power expenses of air conditioning and mechanical ventilating systems for buildings account for the considerable amount of the total power bill. Therefore, a major part of the coolness can be recovered if heat recovery technologies are incorporated into the air conditioning systems. Moreover, the worldwide emphasis on energy conservation because of the decline of energy resources such as gas and oil and environmental protection has made engineers to look for new environmental friendly energy saving technologies and devices to reduce the energy consumption of the air conditioning systems.

Heat pipe heat exchangers (HPHX) have the potential of accomplishing this objective. An HPHX is a heat exchanger consists of externally-finned sealed pipes with a working fluid. The heat exchanger is divided into two sections, namely the evaporator and condenser sections

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for the heat exchange between two air flows. The HPHX has a number of advantageous over conventional heat exchangers such as external power is not needed, no cross-contamination between two air flows, easy manufacturing and easy maintenance [1, 2]. Therefore, HPHX are recommended for systems, in which fresh and exhaust air should not be mixed such as surgery rooms in hospitals and chemical and biological laboratories. For mentioned spaces, the exhaust air is not allowed to be mixed with the exhaust air. Therefore, the possible coolness recovery from the exhaust air is significant. The HPHX are useful for coolness recovery in air conditioning systems in tropical countries, where the incoming fresh air at the high temperature can be pre-cooled by the cool exhaust air stream before entering the chilled water coil.

The HPHX have been extensively used as an energy saving device in air conditioning systems in western countries [2, 3]. Heat recovery capability of the HPHX was reported in some investigations. A HPHX was tested for energy savings for an air conditioning system in the subtropical Florida climates [4]. The HPHX was installed between the warm return air and cold supply air. In terms of energy savings, the findings revealed that the average saving of 75% over the years 1985 to 1986. The effect of a HPHX on the energy consumption and the peak demand of an existing air conditioning system were simulated for operating yearly [5, 6]. During the summer, analysis was carried out with and without indirect evaporation cooling and the findings suggested that by installing a HPHX in the system, 22.1 kW of electrical energy could be saved [5]. From the simulation results, the retrofitted system could pay for itself in 10 months [5, 6]. The thermal performance of a HPHX for heat recovery purposes in air conditioning systems that used fully fresh air was also investigated in the controlled condition [7]. In this study, the effect of different fresh air temperature and ratios of mass flow rate between return and fresh air was investigated. Moisture removal characteristic of HPHX in humid climates was examined by Abtahi *et al.* [8]. The HPHX was placed between the warm return air and cold supply air and the heat recovery was accomplished with the supply air reheat and return air pre-cooled. Hence, pre-cooling reduced the sensible cooling fraction and the dehumidification capacity of the system was improved.

In another research conducted in subtropical climates, the application of a HPHX to reduce the energy use and peak demand in the buildings that required to be kept at 50% relative humidity (RH) and below were studied at an art museum in St. Petersburg, Fla., USA [9]. The study showed that by using the HPHX, the dehumidification capability of the heating, ventilation and air conditioning system was enhanced and considerable amount of coolness was saved.

The effectiveness of HPHX was examined in the naturally-ventilated tropical buildings [5, 10]. The thermal performance of a wickless HPHX was simulated applying a computer source code and the temperature distribution across the HPHX, its thermal performance, and overall effectiveness of HPHX were examined in the study. In another research, the BIN weather data (BIN weather data is the average of the actual data that has been measured over extended periods) was applied to examine the impact of the HPHX in the energy use of an air conditioning system [11, 12]. Based on the prediction results, it showed that the retrofitting on the existing air conditioning system could pay for itself in less than twelve months.

In the earlier paper [13], a base line performance characteristic study of a typical 8-row HPHX was investigated in a controlled condition as the source of outside air. The empirical performance expressions were then used in a year-round operation estimation for an operating theater using the typical meteorological year data for Kuala Lumpur, Malaysia [14].

Literature review indicated that despite many practical operations of HPHX, the complete practical research for HPHX used in the tropics is limited, at least in Malaysia, Singapore, Indonesia, and Brunei. To this end, this study has been carried out and the main objective is to study the effect of the HPHX as the coolness recovery technology in the air conditioning systems operating in the tropics. The HPHX with two, four, six, and eight numbers of rows were studied experimentally. The effect of inside design temperature and face velocity on the energy recovery by a typical HPHX was also examined. Based on the thermal performance and collected data, the amount of net energy recovery by the HPHX was estimated for the entire year. The system was a fully fresh air system and there was no mixing between fresh and re-circulated air. The tests were performed in Kuala Lumpur, Malaysia, as the representative of a tropical country.

Energy consumption in the air conditioning process

The amount of energy saved by the HPHX can be explained by using a psychrometric chart. Figure 1 shows the typical air conditioning process, in which point A presents the outdoor air. In the conventional air handling process, the air is overcooled from point A to point C, and then re-heated to the point D.

$$\mathrm{A} \xrightarrow{\text{Cooling \& dehumidification}} \mathrm{C} \xrightarrow{\text{Heating}} \mathrm{D}$$

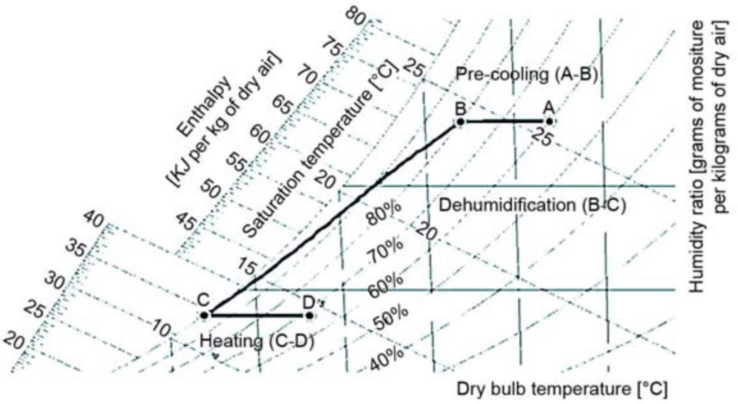

Figure 1. Simple psychrometric processes for a typical HVAC system with an added heat pipe heat exchanger

From fig. 1 the cooling load required in the conventional air handling process is:

$$Q = \dot{m}(h_\mathrm{A} - h_\mathrm{C}) \tag{1}$$

For the coolness recovery, the evaporator section of the heat pipe heat exchanger can be placed at the fresh air inlet to pre-cool the fresh air while the condenser section can be placed at the exhaust air outlet. Based on fig. 1, the fresh air at point A is pre-cooled at the evaporator section of HPHX to point B. The moisture extraction process occurs later at the cooling coil from point B to point C. After that, the heating process takes place from point C to point D. By using this system, the free pre-cooling can actually be achieved (process A to B).

$$\mathrm{A} \xrightarrow{\text{Pre-cooling}} \mathrm{B} \xrightarrow{\text{Cooling \& dehumidification}} \mathrm{C} \xrightarrow{\text{Heating}} \mathrm{D}$$

By doing this, the cooling duty for the initial system is re-distributed with the intention that the latent cooling capability of the conventional cooling coil can also be improved.

Therefore, the required cooling load in the cooling coil will be:

$$Q^{'} = \dot{m}(h_\mathrm{B} - h_\mathrm{C}) \tag{2}$$

The cooling energy recovered by the evaporator section of the installed HPHX:

$$\Delta Q = Q - Q^{'} = \dot{m}(h_{\mathrm{A}} - h_{\mathrm{B}}) \tag{3}$$

Experimental description

Experimental set-up

The detailed specifications of the HPHX examined in the preset study are listed in tab. 1. The design parameters of the HPHX in terms of dimensions were fabricated based on the available space in the ducting system. The total length of each tube is 1020 mm and the length of 420 mm in both the evaporator and condenser sections, and 180 mm for the adiabatic section. The heat exchanger surface areas on the evaporator and condenser sections of HPHX are the same indicated in fig. 2 [2]. The heat pipe tubes have been arranged in staggered form, since this configuration is an efficient form in terms of heat transfer and three sheets of stainless steel wire mesh are pressed alongside the internal tube wall as a wick structure.

Table 1. Design specifications of the HPHX [2, 15]

HPHX dimensions	420 mm wide, 350 mm high
Number of rows	Two, four, six and eight rows of 11 tubes, OD: 13.4 mm, ID: 12.7 mm
Tubes arrangement	Staggered
Centre-to-centre tube spacing	Transverse: 31.75 mm. Longitudinal: 27.5 mm
Fin	Aluminum corrugated, wavy plate, 12 fin per inch, fin thickness: 0.15 mm
Wick structure	Three layers of stainless steel wire mesh, 100 mesh per inch
Working fluid	R-134a, (HFC refrigerant), 110% fluid charge

One of the important parameters in HPHX performance is the working fluid filling ratio inside the tubes. Based on the importance of the working fluid filling ratio in the tubes, the just mentioned parameter has been studied in details. Therefore, a separate experimental setup was established and the effect of different filling ratios was examined. The tests were run at different evaporator inlet temperatures and face velocities to find out the optimum filling ratio for the working fluid [15]. The tests showed that the optimum performance of the HPHX was obtained at a filling ratio slightly more than the amount required to saturate the heat pipe tubes, *i. e.* 110%. Note that the R-134a was used as the working fluid for the HPHX.

After fabrication of the HPHX, the HPHX were placed in the climate chamber. The climate chamber used in the current research consists of an HPHX, a fan coil unit, an electric heater and a variable speed fan. The measurement sections, the HPHX, supply and return ducts are insulated by using the fiber glass to minimize intervening heat transfer with the ambient environment. Figure 2 illustrates the schematic drawing of the system and HPHX.

The HPHX are fixed in the test rig such that they pre-cool the ambient air in the evaporator section of HPHX. The air then passes through the chilled water coil and the electric heater to be heated to the required temperature. The two air flows through the HPHX are such that it operates in the counter-flow arrangement. The ambient air is drawn through the HPHX evaporator by a centrifugal fan installed inside the fan coil unit (FCU). The FCU consists of two units of (9.390 kW in total cooling duty) cooling coils and a centrifugal fan to drive the air to the room. In addition, a 5.5 kW (maximum power) electric heater with a control box operating at on/off position is situated after the FCU to keep the needed indoor air temperature. The dry bulb temperature (DBT) of the air at measuring stations is recorded using the resistance temperature detector (RTD) sensors and a RH sensor is applied in each measuring station in place of wet and DBT measurements at the measuring stations [12]. The output signals from these sensors have been measured and converted to °C and RH (%) via a data-logging system. The measured data are analyzed by DASYLab software, (DASYLab is an icon-based data acquisition software used for data analysis). All RTD sensors are calibrated with the temperature calibrator and Pt100 in the temperature range of 0 °C to 50 °C with the accuracy of ±0.05 °C. The RH sensors are calibrated to the accuracy of ±2%. The temperatures of two RTD sensors in every measuring station are averaged, and the findings are indicated as T_1, T_2, T_3, and T_4 as shown in fig. 2 [2]. The pressure drop caused by the HPHX is measured using a pressure transmitter. The pressure transmitter is calibrated to the accuracy of ±0.7 Pa.

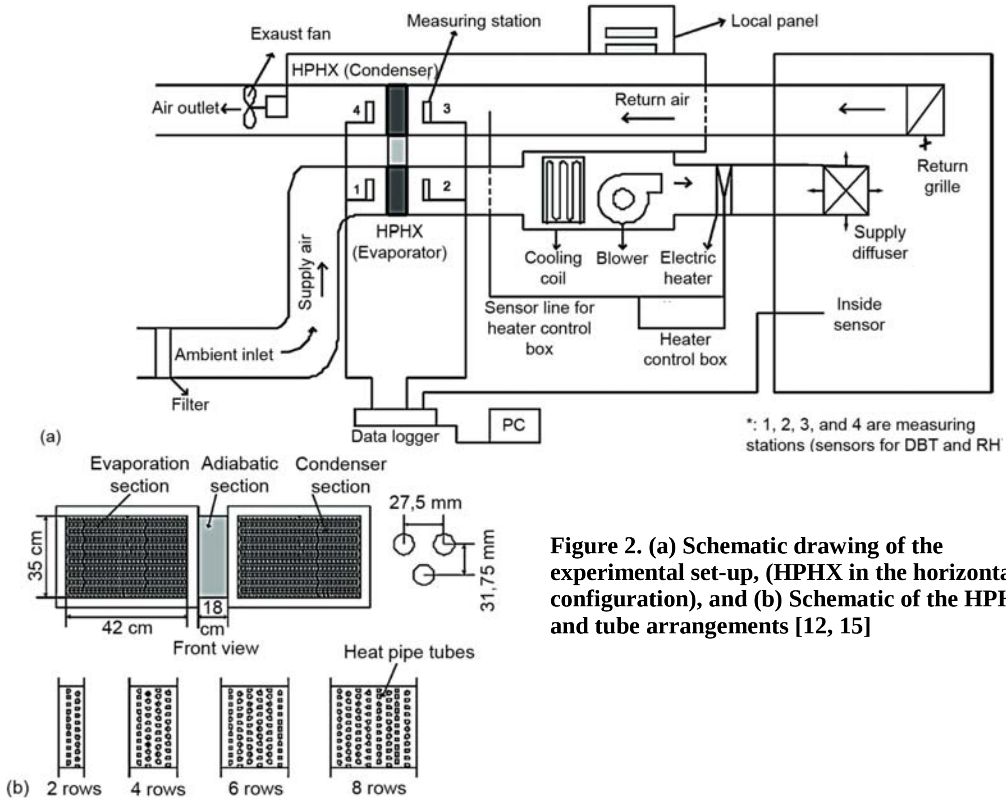

Figure 2. (a) Schematic drawing of the experimental set-up, (HPHX in the horizontal configuration), and (b) Schematic of the HPHX and tube arrangements [12, 15]

The mass flow rate in the system is measured by a multi-Pitot-tube digital meter consisting of sixteen holes in every measuring cross section. The sixteen holes are equally distributed in the cross section of measuring point to take the average static and total pressure to examine the mass flow rate using a micro-manometer with the accuracy of ±3%. In the current work, the multi-Pitot-tube meter is fixed at the outlet of the supply diffuser to record and generate the needed coil face velocity [2].

Test conditions

In the first part of the experiments, the effect of number of rows on the energy recovery by HPHX was investigated. For this purpose, four HPHX with two, four, six, and eight numbers of rows were installed in the system separately, and the performance of them was monitored during the one week of operation (168 hours) as the representative week for the year. The coil face velocity on the HPHX evaporator section was fixed at 2 m/s, and the re-circulated air temperature was controlled at 24 °C as the representative temperature for the inside air.

The effect of inside design temperature on the energy recovery by the typical HPHX was then studied. The inside temperature was set at 22 °C for this situation. The system was operated for one week (*i. e.* 168 hours) to examine the performance of the system in this situation, and the effect of reducing the inside design temperature from 24 °C to 22 °C was discussed.

Moreover, the effect of the evaporator coil face velocity on energy recovery by a typical HPHX was also examined. Three face velocities at 2 m/s, 2.2 m/s, and 2.5 m/s as the approximations of what could commonly happen in practice were established on the evaporator section. The system was run for one week for every coil face velocity, and the results were presented. In all tests examined, HPHX experience directly the ambient tropical air in the evaporator section as the fresh air.

Experimental effectiveness calculation procedure

The effectiveness of the HPHX in eq. (4) is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate [16]:

$$\varepsilon = \frac{Q_{\mathrm{act}}}{Q_{\mathrm{max}}} \tag{4}$$

For the equal mass flow rate in the evaporator and condenser sections of the HPHX:

– for $C_{\mathrm{e}} > C_{\mathrm{c}}$

$$\varepsilon = \frac{T_{\mathrm{c,out}} - T_{\mathrm{c,in}}}{T_{\mathrm{e,in}} - T_{\mathrm{c,in}}} \tag{5}$$

– for $C_{\mathrm{e}} < C_{\mathrm{c}}$

$$\varepsilon = \frac{T_{\mathrm{e,in}} - T_{\mathrm{e,out}}}{T_{\mathrm{e,in}} - T_{\mathrm{c,in}}} \tag{6}$$

In order to determine the impact of HPHX on the energy recovery, the temperature and RH for every air state must be recorded. For this reason, the following assumptions were made for the measurements [13]:

– the HPHX are completely insulated and no outside energy is supplied into or lost from it,
– no air leakage occurs within the HPHX and ducting system and the mass flow rate in evaporator and condenser sections is the same,
– the energy transfer rates in the two sections of the HPHX are the same and opposite, and the heat transfer between the air flows in the ducts and the ambient environment is insignificant, and
– even air flow and properties entering the HPHX, and the air is properly mixed at each measuring point.

Simulation of the HPHX

The method of effectiveness number of transfer units was employed to predict the thermal performance of HPHX in the present research [17]. In the evaporator and condenser sections of the HPHX, the hot and cold fluids are in cross flow with the vapor inside the heat pipes.

However, due to the vapor inside the heat pipe is almost at the constant temperature, its specific heat, c_p and capacity rate, C_v, become by definition, equal to infinity, and consequently, $C_e/C_v = C_c/C_v = 0$. Hence, the effectiveness-NTU equations for a single-row HPHX are [18, 19]:

For the evaporator section:

$$\varepsilon_{\mathrm{e}1} = 1 - \exp(-NTU)_{\mathrm{e}} \tag{7}$$

For the condenser section:

$$\varepsilon_{\mathrm{c}1} = 1 - \exp(-NTU)_{\mathrm{c}} \tag{8}$$

The NTU has the following values for the evaporator and condenser sections, respectively:

$$NTU = \frac{UA}{C} \tag{9}$$

$$C = \dot{m} c_p \tag{10}$$

For a HPHX with n rows of heat pipes in the flow direction, the effectiveness-NTU equations are [18, 19]:

For the evaporation section:

$$\varepsilon_{en} = \frac{\left(\dfrac{1-\dfrac{C_e}{C_v}\varepsilon_{\mathrm{e}1}}{1-\varepsilon_{\mathrm{e}1}}\right)^n - 1}{\left(\dfrac{1-\dfrac{C_{\mathrm{e}}}{C_v}\varepsilon_{\mathrm{e}1}}{1-\varepsilon_{\mathrm{e}1}}\right)^n - \dfrac{C_{\mathrm{e}}}{C_v}} \tag{11}$$

For the condenser section:

$$\varepsilon_{cn} = \frac{\left(\dfrac{1-\dfrac{C_{\mathrm{c}}}{C_v}\varepsilon_{\mathrm{c}1}}{1-\varepsilon_{\mathrm{e}1}}\right)^n - 1}{\left(\dfrac{1-\dfrac{C_{\mathrm{e}}}{C_v}\varepsilon_{\mathrm{c}1}}{1-\varepsilon_{\mathrm{c}1}}\right)^n - \dfrac{C_{\mathrm{c}}}{C_v}} \tag{12}$$

For $C_e/C_v = 0$ and $C_c/C_v = 0$ the expressions will be in the form:

$$\varepsilon_{en} = 1 - (1 - \varepsilon_{\mathrm{e}1})^n \tag{13}$$

and

$$\varepsilon_{\mathrm{c}n} = 1 - (1 - \varepsilon_{\mathrm{c}1})^n \tag{14}$$

The overall effectiveness of the heat exchanger is then written:

– if $C_e > C_c$

$$\varepsilon_o = \left(\frac{1}{\varepsilon_{cn}} + \frac{\dfrac{C_c}{C_e}}{\varepsilon_{en}} \right)^{-1} \tag{15}$$

– if $C_c > C_e$

$$\varepsilon_o = \left(\frac{1}{\varepsilon_{en}} + \frac{\dfrac{C_e}{C_c}}{\varepsilon_{cn}} \right)^{-1} \tag{16}$$

where ε_o is the overall effectiveness of the HPHX. After calculation of the overall effectiveness from eq. (15) or (16), the outlet temperature of evaporator could be computed from the expression:

$$T_{e,outlet} = T_{e,inlet} - \varepsilon_o (T_{e,inlet} - T_{c,inlet}) \tag{17}$$

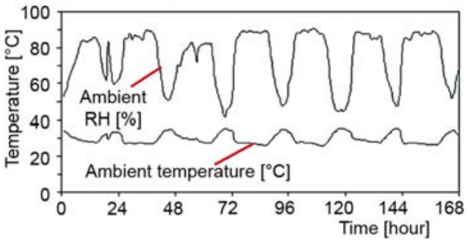

Figure 3. The ambient temperature [°C] and RH [%] during the test for the evaporator inlet air

Results and discussion

Test results

It can be seen from fig. 3 that the average recorded ambient RH and the temperature throughout the one week (168 hours) of data acquisition. The RH is in the range of 36-93% and the ambient temperature is ranging between 25 °C and 35 °C. The low temperature is accompanied by a high RH, and the high temperature is accompanied by the low RH as observed in fig. 3.

Figures 4 and 5 show the average measured temperature and the RH of the measuring points for a typical six-rows HPHX, respectively. The temperature and RH before and after the evaporator and condenser sections are shown in figs. (4) and (5).

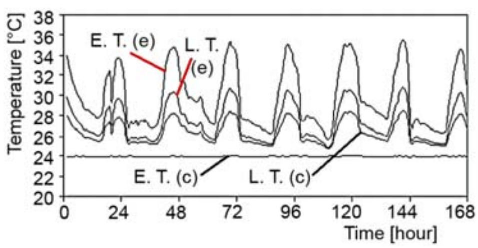

Figure 4. Average measured temperature [°C] of measuring stations during the test for the typical six-rows HPHX

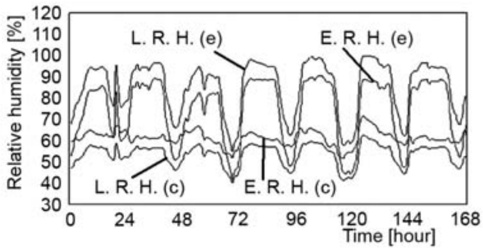

Figure 5. Average measured RH [%] of measuring stations during the test for the typical six-rows HPHX

Table 2. Maximum and minimum effectiveness for the HPHX during the tests

Number of rows	Minimum effectiveness [%]	Maximum effectiveness [%]
2	24.5	27.5
4	39	41.5
6	43.6	47.1
8	42	51.6

From the experimental results, it is evident that the effectiveness of the HPHX is varying with the ambient temperature's variation. Table 2 shows the maximum and minimum amount of effectiveness during the one week of operation for the HPHX. It is evident that the maximum and minimum values of effectiveness do not change significantly when the number of rows increases from six to eight. Therefore, the present result suggests that six-rows HPHX is the optimized HPHX design for operating in the tropical climates.

As illustrated in figs. 6 and 7, by recovering the energy in the pre-cooling section (*i. e.* the evaporator) of the two-rows HPHX, the total amount of 2885 kWh per year can be saved, and this amount can be increased up to 7023 kWh per year by the employing of eight-rows HPHX. Moreover, if the energy in the condenser section can be recovered, the total amount of 14046 kWh per year can be saved for the eight-rows HPHX. The amount of the extra power required for the added pressure drop due to the HPHX is also considered for the estimation of energy recovery by the HPHX.

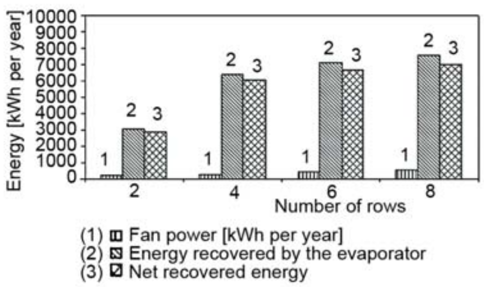

Figure 6. Energy recovered by the evaporator section of the HPHX for the whole year (8760 hours)

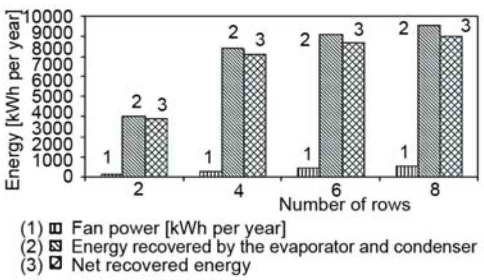

Figure 7. Energy recovered by the evaporator and condenser sections of the HPHX for the whole year (8760 hours)

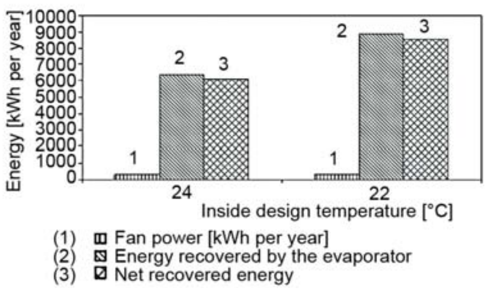

Figure 8. Effect of the inside design temperature on energy recovery by the evaporator section for a typical four-rows HPHX

The effect of inside design temperature on energy recovery by the HPHX was also studied. For this reason, the inside temperature was decreased from 24 °C to 22 °C. Again, the system was operated for 168 hours to examine the thermal performance and the temperature drop in the evaporation section for this situation. After the estimation of energy recovery for the entire year, as shown in fig. 8, it was revealed that by decreasing the inside design temperature about 2 °C, the energy recovery by the evaporator section of the typical four rows HPHX was enhanced from 6103 kWh per year to 8540 kWh per year. In another word, the energy recovery has been increased about 40%. Data processing for this situation showed that by reducing the inside tem-

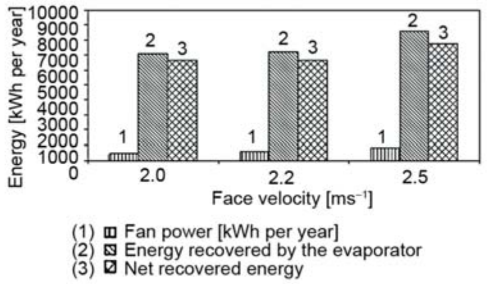

Figure 9. Effect of the coil face velocity on energy recovery by the evaporator section for a typical six-rows HPHX

perature from 24 °C to 22 °C, the maximum temperature in the HPHX was increased. Therefore, the temperature drop in the evaporator section has been increased.

To study the effect of the coil face velocity on energy recovery of a typical six-rows HPHX, three coil face velocities were chosen. In fact, by increasing the face velocity from 2 m/s to 2.5 m/s, the volumetric flow rate will be increased from 0.294 m^3/s to 0.367 m^3/s, and the amount of energy recovery is expected to increase. However, two negative points of the high coil face velocity are the higher pressure drop in the coil and the decrease of the effectiveness can happen. Therefore, by increasing the coil face velocity, the power requirement of the fan to overcome to the pressure drop caused by the HPHX can be increased. Figure 9 shows the effect of the coil face velocity on energy recovery. As can be seen from fig. 9, the net recovered energy by the evaporation section has been increased by the coil face velocity.

Comparison of experimental and theoretical results

Two comparisons between experimental data and theoretical values for the effectiveness are made. First comparison is made for the mean effectiveness for the HPHX with different number of rows. As illustrated in fig. 10, there is a good agreement between experimental data and theoretical values with the maximum deviation of 11%. In another comparison, the experimental data are compared with theoretical values for different coil face velocities for the six-rows HPHX as the representative of the HPHX indicated in fig. 11. It is also shown that experimental results agree well with theoretical values with the maximum deviation of 5%.

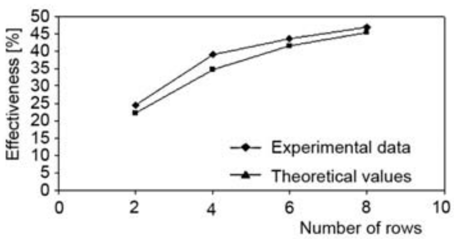

Figure 10. Comparison of the mean experimental and theoretical effectiveness for HPHX

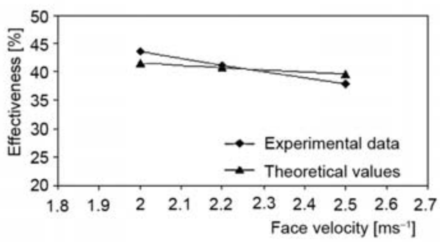

Figure 11. Comparison of the experimental and theoretical effectiveness at different coil face velocities for a typical six-rows HPHX

In addition, the computed outlet temperature for the evaporator section from eq. (17) is compared with the experimental results in figs. 12 and 13. The experimental results are compared with the simulation findings for the four and eight-rows HPHX as representatives. From figs. 12 and 13, the experimental results from the research are found to be in good agreement to those acquired from the simulation with a deviation less than 2%.

Conclusions

In this research, the effect of HPHX on energy recovery was successfully investigated experimentally in tropical climates. Four different HPHX, namely two, four, six, and eight

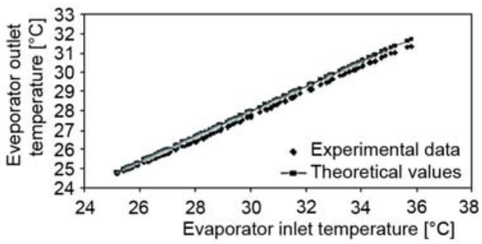

Figure 12. Experimental and theoretical comparisons of the evaporator outlet temperature [°C] with the evaporator inlet temperature [°C] for four-rows HPHX

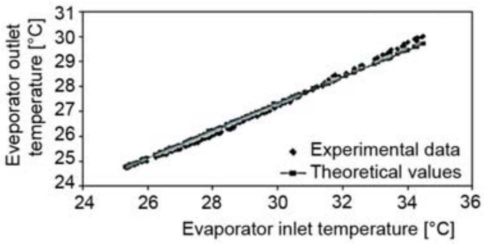

Figure 13. Experimental and theoretical comparisons of the evaporator outlet temperature [°C] with the evaporator inlet temperature [°C] for eight-rows HPHX

rows, were installed in the system, and the tests performed for duration of one week (168 hours) as the representative week of the year. The evaporator section of HPHX experienced the ambient air during the operation, and the return air was set at 24 °C as the representative inside temperature for this purpose. The impact of the inside temperature (*i. e.* the return air) on energy recovery of a typical HPHX was also examined by reducing the inside temperature from 24 °C to 22 °C. In addition, the influence of coil face velocity on the thermal performance and energy recovery for a typical HPHX was examined. For this purpose, three coil face velocities, namely 2 m/s, 2.2 m/s, and 2.5 m/s were tested. The entire tests for all situations were conducted for a one-week of operation. Based on the collected data, by recovering the energy in pre-cooling section (*i. e.* the evaporator) of the two-rows HPHX, the total amount of 2885 kWh per year can be saved and this amount can be increased up to 7023 kWh per year by employing a eight-rows HPHX. Furthermore, if the energy in the condenser section can be recovered, the total amount of 14046 kWh per year can be saved for the eight-rows HPHX. It is also found that by decreasing the inside design temperature at about 2 °C, the energy recovery of a typical HPHX can be enhanced up to 40%. From the experimental results, it is found that there is a decrease in the effectiveness and an increase in the pressure drop in the HPHX by increasing the coil face velocity, even though the net energy recovered by the HPHX is enhanced.

A simulation was also carried out on the HPHX, and the experimental and theoretical results for the effectiveness and the evaporator outlet temperature were compared. It is revealed that there is a good agreement between the experimental and simulation results.

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Nomenclature

A	– heat transfer area, [m^2]
c_p	– specific heat of the ambient air, [Jkg^{-1}K^{-1}]
C	– heat capacity, [Js^{-1}K^{-1}]
h	– specific enthalpy, [kJkg^{-1}]
$\dot{m}$	– mass flow rate, [kgs^{-1}]
n	– number of rows, [–]

Q – heat transfer rate, [W]
T – temperature, [°C]
U – heat exchanger heat transfer coefficient, [$Wm^{-2}°C^{-1}$]

Greek symbols

ε – effectiveness, [%]

Subscripts

act – actual
c – condenser
e – evaporator
max – maximum
O – overall
v – vapor
n – number rows

Acronyms

DBT – dry bulb temperature, [°C]
E. RH. – entering air RH [%]
E. T. – entering air temperature, [°C]
FCU – fan coil unit
HPHX – heat pipe heat exchanger
L. RH. – leaving air RH, [%]
L. T. – leaving air temperature, [°C]
NTU – number of heat transfer units (= UA/C), [–]
RH – relative humidity, [%]
RTD – resistance temperature detector

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