Energy Recovery Heat Exchanger Performance in Building's Air Conditioning System

Nik Mohd Hafizzal Ab Razak

Department of Mechanical Engineering,
Universiti Teknologi PETRONAS,
31750 Bandar Seri Iskandar, Tronoh, Perak, Malaysia

Abstract

In this work, a thermal performance of a Z shape enthalpy heat exchanger utilising 70gsm Kraft paper as the heat and moisture transfer surface is studied. Effect of different inlet humidity ratio condition on the heat exchanger effectiveness and on the energy recovered by the heat exchanger has been the main focus of this investigation. A typical air conditioning cooling coil which incorporates an enthalpy heat exchanger has been modelled for tropical climate. Under test conditions, results have shown that the latent effectiveness and the moisture resistance coefficient have strong dependency on the inlet air humidity ratio. The latent effectiveness has been found to be strongly dependent on the moisture resistance coefficient rather than the convective mass transfer coefficient. Finally, annual energy analysis for Kuala Lumpur, Dubai, Miami, Rome and Sydney weather conditions have also shown that a significant amount of energy can be recovered since the Z shape enthalpy heat exchanger utilize the room exhaust air to pre-condition the ambient fresh air.

Keywords: Energy Recovery, Z shape Enthalpy Heat Exchanger, Latent effectiveness, Moisture transfer resistance

1 INTRODUCTION

The design in heating, ventilating and air conditioning (HVAC) systems for thermal comfort requires a great concern, so that an optimal level of indoor air quality (IAQ) and humidity may be reached and maintained to ensure a comfortable and healthy environment and to avoid condensation damage to the building envelope and furnishings. The usage of 100% fresh air and good ventilation in HVAC system are necessary especially when the requirement for fresh air in a building like hospital is not only for breathing purposes, but also for the prevention of deadly epidemic diseases like bird flu and SARs [1-2]. Due to this requirement, energy expenses for ventilation are very heavy. Zhang et al. [3] Ventilation air accounts for 20-40% of the cooling load for HVAC industry. The ratio can be even higher in hot and humid regions where latent load from fresh air is as heavy as 50% of the cooling load. For such system, it is essential to utilize energy recovery systems in order to create a reduction in energy consumption to this load. The operating principle of the energy recovery systems is to use the room exhaust air to pre-condition the ambient fresh air. As a result, a significant amount of energy is recovered which in return reduces the overall HVAC energy requirement.

Heat exchangers are devices that allow the exchange of heat between two fluids without allowing them to mix with each other. The heat transfer in a heat exchanger usually involves convection in each fluids and conduction through the wall separating the two fluids [4]. Z-shape Enthalpy heat exchanger is a device that can recover both sensible and latent energy by using a porous membrane as the heat and moisture transfer surface [3, 4]. This enthalpy heat exchanger is a static device that does not involve any moving parts and can be easily integrated into existing air conditioning system. Theoretically, it could save a large fraction of energy for cooling and dehumidifying the fresh air since cool air and dryness would be recovered from the exhaust stream to the fresh air in summer. Similarly in winter, the total heat exchangers could also save energy because exhaust moisture and heat can be recovered to save heating and humidification energy for fresh air.

2 METHODOLOGY

In this research, energy modelling is used to study the heat exchanger sensible and latent effectiveness variation by varying the inlet humidity ratio conditions. Determining the heat exchanger permeable surface moisture transfer resistance from direct measurements is difficult and sometimes expensive and complicated. As a result, mathematical models were developed using effectiveness-NTU method to predict the heat exchanger effectiveness.

2.1 Sensible Heat

To model sensible heat transfer in the heat exchanger, a mathematical model was developed based on Nusselt number and Reynolds number correlation of each air channel. The heat exchanger inlet face velocity was assumed to be 0.75m/s, and the respective air velocity at each air flow passage of the heat exchanger was 1.5m/s. By using the air velocity at each passage, Reynolds number at each flow passage was calculated based on Eq. (1). A Reynolds number value of 970 was estimated concluding that the flow was a laminar flow.

$$\mathrm{Re} = \frac{V d_{hyd}}{\upsilon} \quad (1)$$

Nusselt number value was estimated based on the Hausen's correlation for laminar flow in ducts under uniform heat flux [14].

$$\mathrm{Nu} = 8.235 + \frac{0.0068\left(\frac{d_{hy}}{L}\right)\mathrm{Re\,Pr}}{1 + 0.04\left[\left(\frac{d_{hy}}{L}\right)\mathrm{Re\,Pr}\right]^{\wedge}\left({}^{2}/_{3}\right)} \quad (2)$$

The convective heat transfer coefficient (hheat) is a function of Nu and is calculated as:

$$h_{heat} = \frac{\mathrm{Nu}\,k}{d_{hy}} \quad (3)$$

The total number of transfer units (NTU) is calculated as:

$$\mathrm{NTU}_s = \frac{A_{ht}\,U_s}{C_{min}} \quad (4)$$

The general form of the overall sensible heat transfer coefficient, Us is:

$$U_s = \left[\frac{1}{h_{h,heat}} + \frac{\delta}{k_{ther}} + \frac{1}{h_{c,heat}}\right]^{-1} \quad (5)$$

The thermal resistance (δ/k_{ther}) represents the conduction resistance of the 70gsm Kraft paper and is a function of the material type and thickness. This term was very small due to the small thickness of the paper (130μm) and could be ignored [7, 10-13]. The other two terms are the convective heat transfer resistance in the two streams which contributed the largest portion of the sensible heat transfer resistance (Us).

Similarly, the sensible effectiveness for counter flow, cross flow and parallel flow conditions are defined in Eq. (6), Eq. (7) and Eq. (8) respectively.

$$\varepsilon_{s,counter} = \frac{1 - \exp\left[-\mathrm{NTU}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]}{1 - \left(\frac{C_{min}}{C_{max}}\right)\exp\left[-\mathrm{NTU}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]} \quad (6)$$

$$\varepsilon_{s,cross} = 1 - \exp\left\{\frac{\mathrm{NTU}_s^{\,0.22}}{\frac{C_{min}}{C_{max}}}\left[\exp\left(-\frac{C_{min}}{C_{max}}\mathrm{NTU}_s^{\,0.78}\right) - 1\right]\right\} \quad (7)$$

$$\varepsilon_{s,parallel} = \frac{1 - \exp\left[-\mathrm{NTU}_s\left(1 + \frac{C_{min}}{C_{max}}\right)\right]}{\left(1 + \frac{C_{min}}{C_{max}}\right)} \quad (8)$$

Where the minimum/maximum stream heat capacity is:

$$C_{min} = \min(\dot{m}Cp\,|_{cold}\,, \dot{m}Cp\,|_{hot})$$

$$\text{or} \quad C_{max} = \max(\dot{m}Cp\,|_{cold}\,, \dot{m}Cp\,|_{hot})$$

The overall sensible effectiveness of the heat exchanger could be obtained by combining Eqs. (6) – (8):

$$\varepsilon_{s,combined} = \left(\frac{A_{counter}}{A_{ht}}\right)\varepsilon_{s,counter} + \left(\frac{A_{cross}}{A_{ht}}\right)\varepsilon_{s,cross} + \left(\frac{A_{parallel}}{A_{ht}}\right)\varepsilon_{s,parallel}$$

The counter flow area, for the current Z flow heat exchanger, was found to be equal to 36% of the total heat transfer area. Similarly, the cross flow and parallel flow areas were 43% and 21% of the total heat transfer area, respectively.

2.2 Latent Heat

A latent heat transfer model was, also developed to simulate moisture transfer using the convective mass transfer Sherwood correlation. The convective mass transfer coefficient can be obtained using the Chilton-Colburn analogy [15].

$$\mathrm{Sh} = \mathrm{Nu}\,\mathrm{Le}^{\frac{1}{3}} \quad (9)$$

Where Sherwood number is defined as:

$$\mathrm{Sh} = \frac{h_{mass}\,d_{hy}}{D_{va}} \quad (10)$$

By substituting Nusselt number from Eq. (3) and Sherwood number from Eq. (10) into Eq. (9), the convective mass transfer coefficient (hmass) can be defined as:

$$h_{mass} = \frac{h_{heat}}{C_{pa}}\,\mathrm{Le}^{\frac{1}{3}} \quad (11)$$

It has been found that Lewis number value was around 0.81 for the conditions studied [8, 11]. The number of latent transfer units (NTU_L) is:

$$\mathrm{NTU}_L = \frac{A_{ht}\,U_L}{\dot{m}_{min}} \quad (12)$$

The overall mass transfer coefficient (UL) is given by:

$$U_L = \left[\frac{1}{h_{h,mass}} + R_{paper} + \frac{1}{h_{c,mass}}\right]^{-1} \quad (13)$$

The moisture resistance of the porous paper is Rpaper and the other two terms are the convective mass transfer resistance at each flow stream. Similarly to the sensible effectiveness, latent effectiveness is calculated using the following correlations for different heat exchanger configurations [7].

$$\varepsilon_{L,counter} = \frac{1-\exp\left[-NTU_L\left(1-\frac{\dot{m}_{min}}{\dot{m}_{max}}\right)\right]}{1-\left(\frac{\dot{m}_{min}}{\dot{m}_{max}}\right)\exp\left[-NTU_L\left(1-\frac{\dot{m}_{min}}{\dot{m}_{max}}\right)\right]} \quad (14)$$

$$\varepsilon_{L,cross} = 1-\exp\left\{\frac{NTU_L^{\ 0.22}}{\frac{\dot{m}_{min}}{\dot{m}_{max}}}\left[\exp\left(-\frac{\dot{m}_{min}}{\dot{m}_{max}}NTU_L^{\ 0.78}\right)-1\right]\right\} \quad (15)$$

$$\varepsilon_{L,parallel} = \frac{1-\exp\left[-NTU_L\left(1+\frac{\dot{m}_{min}}{\dot{m}_{max}}\right)\right]}{\left(1+\frac{\dot{m}_{min}}{\dot{m}_{max}}\right)} \quad (16)$$

$$\varepsilon_{L,combined} = \left(\frac{A_{counter}}{A_{ht}}\right)\varepsilon_{L,counter} + \left(\frac{A_{cross}}{A_{ht}}\right)\varepsilon_{L,cross} + \left(\frac{A_{parallel}}{A_{ht}}\right)\varepsilon_{L,parallel} \quad (17)$$

2.3 Energy Model

An air conditioning cooling coil which supplies 100% fresh air coupled with an enthalpy heat exchanger was modelled as shown in Fig. 1. The analysis was conducted for an office space of 300m2, office height of 3.5m which operates 24 hours - seven days a week, and for an internal load of 20kW and occupant moisture generation of 0.04kg/hr per person. For energy modelling purposes, Kuala Lumpur weather data was used as the first variable due to the high latent loads. The table below shows the details for all initial parameter required in order to develop the energy model analysis.

Table 1: Room's initial parameter

Number of people inside room	30
U wall value	2kW/K
Wall's surface area	545m2
Office volume	1050m3
Air flow rate	1000L/s
Initial humidity ratio	0.00082kg/kg

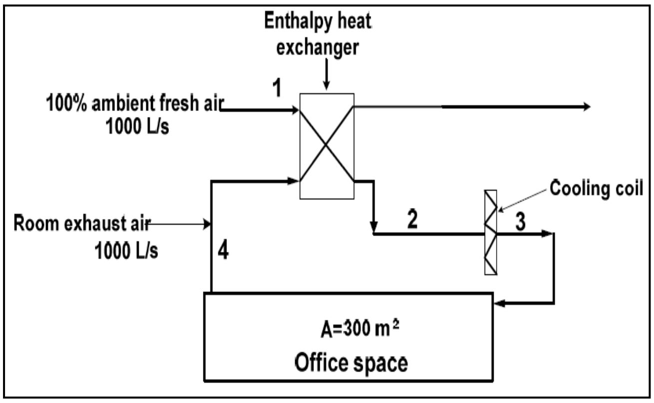

Figure 1: Air conditioning system coupled with enthalpy heat exchanger.

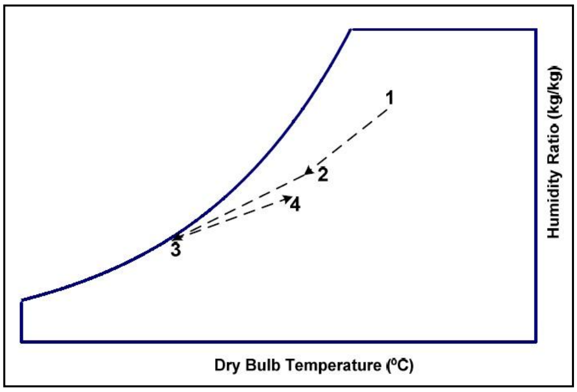

Figure 2: System psychometric

Process 1 to 2 in Fig.6 represents the cooling and dehumidification process inside the enthalpy heat exchanger. Point 2 indicates the air-on conditions (T2 and ω2) at the cooling coil inlet were calculated as follows:

$$T_2 = T_1 - \varepsilon_s\,(T_1 - T_4) \quad (18)$$

$$\omega_2 = \omega_1 - \varepsilon_L\,(\omega_1 - \omega_4) \quad (19)$$

The air discharged from the heat exchanger was cooled and dehumidified at the cooling coil (point 3). Cooling coil effectiveness value of 0.9 was used in this simulation, whereas the air-off humidity (point 3) was assumed to have a value of 0.95. The air-off temperature at this point depended on the cooling coil efficiency and it was calculated from:

$$T_3 = T_2 - \varepsilon_{Cooling\_coil}\left(T_2 - T_{Cooling\_coil}\right) \quad (20)$$

The simulation was carried out by reading Kuala Lumpur hourly weather data. The coil-off calculation was performed from the above equations based on the supplied air condition (air-on conditions). The cooling provided by air conditioner was calculated from:

$$Q_{cooling} = \dot{m}_{air}C_{p\,air}\left(T_{room} - T_{Cooling\_coil}\right) \quad (21)$$

The heat transfer through the room walls was calculated from:

$$Q_{heat} = AU_{office}(T_{ambient} - T_{room}) \quad (22)$$

The sensible cooling load provided by the air conditioner was considered to be the sum of the heat transferred via the walls and the heating load within the space:

$$Q_{cooling} = Q_{heat+}\,Q_{load} \quad (23)$$

Substituting equations. (21) and (22) into (23), the office temperature can be obtain as follow:

$$T_4 = \frac{AU_{office}T_1 + m_{air}C_{p\,air}T_3 + Q_{Load}}{m_{air}C_{p\,air} + AU} \quad (24)$$

Moisture content in the room at each time step will be calculated as follows:

$$m_{room\ moisture(kg)} = m^{i}_{occupant} + m^{i-1}_{room\ moisture} + m^{i}_{coil\ moisture} - m^{i}_{moist\ room\ exhaust}$$

Where m_{room} moisture is the room moisture content (kg), $m^{i}_{occupant}$ represent the occupant's total moisture generation at specified time step, $m^{i-1}_{room\ moisture}$ represents room moisture content at previous time step, $m^{i}_{coil\ moisture}$ is coil air off supplied moisture at specified time step, and $m^{i}_{moisture\ room\ exhaust}$ corresponds to moisture discharged from the room exhaust air at specified time step. The amount of energy recovered by the enthalpy heat exchanger will be calculated at each time step in accordance with eq. (25). The annual total energy recovered by the enthalpy heat exchanger will be summed for the modelled heat exchanger with variable moisture transfer resistance.

$$\mathrm{E}_{\mathrm{Annual\ recovered}} = \dot{\mathrm{m}}(\mathrm{H}_1 - \mathrm{H}_2) \qquad (25)$$

The developed sensible and latent mathematical models were used at each time step in the simulation to calculate the sensible and latent effectiveness. At the start of the calculations, the office air conditions were unknown. It was assumed that for the first time step, the room temperature and relative humidity were 22°C and 50% respectively. The coil air-off conditions were calculated from Eq. (20). The office air temperature and humidity ratio were then calculated from energy and mass balance of the modelled space using Eq. (24) and Eq. for room moisture content above.

For the next time step, the heat exchanger membrane moisture transfer resistance was calculated from relation Rpaper = 4168.3Δω based on values of the ambient air and the air leaving the office room. The heat exchanger effectiveness was calculated from the developed sensible and latent mathematical models. The air condition exiting the heat exchanger was obtained from the calculated heat exchanger effectiveness.

Then, the above calculations were repeated by using Dubai, Miami, Rome and Sydney weather data as inlet variable to study the impact of different humidity ratio has on the energy recovered. For the modelled system shown in Fig. 6, when the room temperature was less than 22°C, the air conditioner was turned off. As the result, the heat exchanger acted as a passive cooling device for the room with no energy required for cooling. The building thermal mass was not included in this model in order to simplify the calculation in this project.

2.4 *Spreadsheet Simulation*

Five desired weather data was obtained by using weather station software. Data for Kuala Lumpur, Dubai, Miami, Rome, and Sydney was then used in order to study the effect of different inlet air humidity ratio conditions on heat exchanger effectiveness, and on the energy recovered by similar heat exchanger for each individual city. Mathematical equations was then studied and used into spreadsheet in order to obtain all parameters required for each point as indicated in Fig.6.

3 RESULTS & DISCUSSION

Results from the annual energy modelling confirmed the earlier trends of the main parameters contributing to the heat exchanger effectiveness. Fig. 3 shows paper moisture transfer resistance and convective mass transfer coefficient variation for Kuala Lumpur weather data throughout the year. It can be seen that although the convective mass transfer variation was small, paper moisture transfer resistance variation was significant.

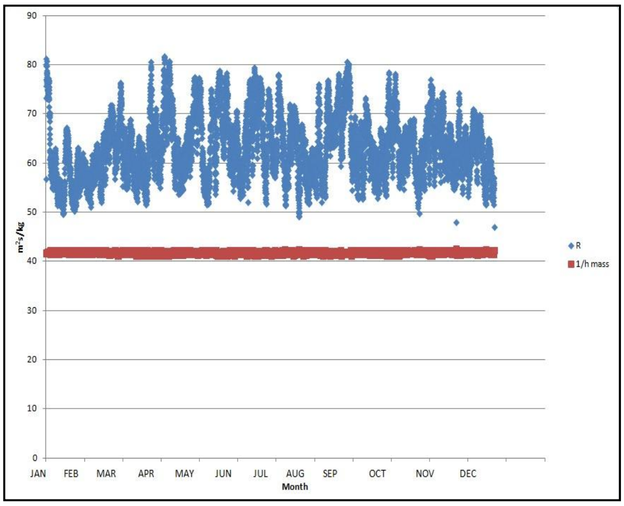

Figure 3: Variation history of paper moisture transfer resistance, (R) and convective mass transfer coefficient, (1/hmass) – Kuala Lumpur

The heat exchanger effectiveness variation is illustrated in Fig. 4. The figure shows that the sensible effectiveness variation was small and could be considered as a constant. However the latent effectiveness was significant which was attributed to the consequential variation of paper moisture transfer resistance.

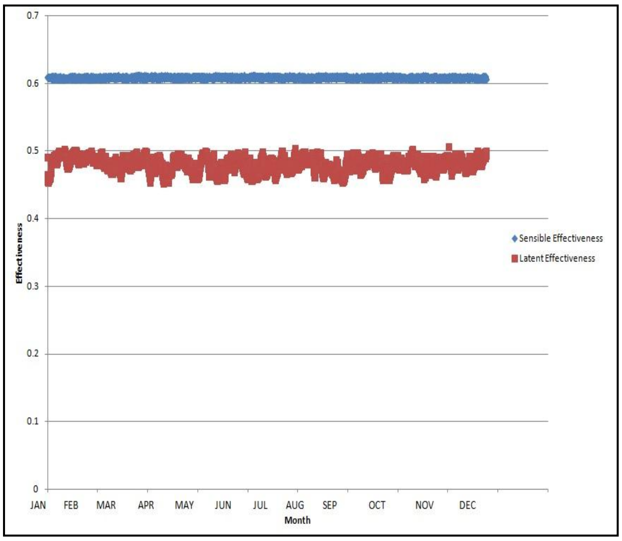

Figure 4: Heat exchanger sensible and latent effectiveness variation history– Kuala Lumpur

The energy consumption for cooling for the modelled system with Kuala Lumpur weather data was about 707 GJ which is considered high as shown in Fig. 5. This is due to the nature of the working hours of the modelled office. It was assumed that the modelled office operates 24 hours a week and the supplied fresh air flow rate was 1000L/s. Large amount of energy is required in order to cold and dry the ambient air especially in hot and humid weather like Kuala Lumpur. However, a significant large amount of energy was recovered since the cold and dryness was extracted from the room exhaust air to pre- condition the ambient fresh air.

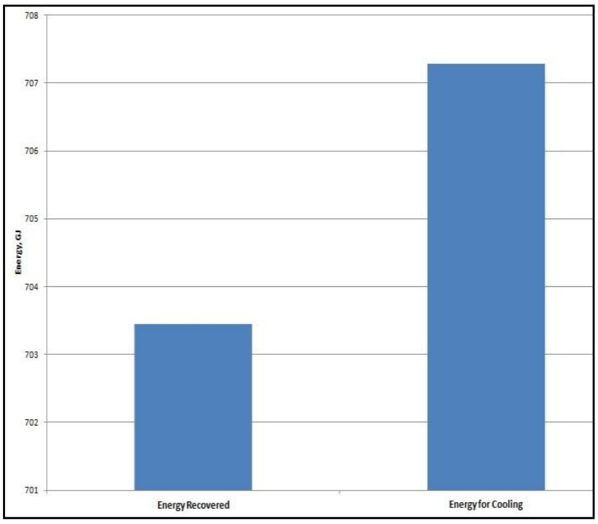

Figure 5: Comparison of the annual energy recovered and energy consumption– Kuala Lumpur

The variation history of paper moisture transfer resistance against convective mass transfer coefficient for another four different cities was observed to exhibit equal trait. Based on the graph obtained, the R-value for all four cities was observed to be slightly higher than 1/hmass was mainly due to the low porosity characteristic of the paper that was used as enthalpy heat exchanger. Results from the heat exchanger sensible and latent effectiveness variation history for the rest of the cities confirmed the earlier trends that R-value and latent effectiveness have strong dependency on the inlet air humidity ratio.

They exhibit the same pattern which is the sensible effectiveness remains almost constant when the inlet air conditions vary. However, heat exchanger latent effectiveness varies when the air inlets conditions change. This variation in sensible and latent effectiveness was caused by the thickness of the kraft paper that was used as permeable surface heat exchanger. Due to this thin paper, the conduction of heat through the paper becomes dominant which allows the energy recovered in all cities become easier.

3.1 Energy Analysis

The results of energy consumption for cooling and dehumidification process was found to be slightly higher than energy recovered under constant heat exchanger as shown in Fig. 6. This behavior applies for the first four cities which is Kuala Lumpur, Dubai, Miami, and Rome. For Kuala Lumpur, 708 GJ of energy was required to cold and dry the ambient air, and at the same time 703 GJ of energy can be recovered annually. As for Dubai, 360 GJ of energy was needed to condition the ambient fresh air, while 158 GJ of energy managed to be recovered annually. Different result was obtained for Sydney which has the lowest energy required for cooling since the weather is already cold and dry, thus less energy was needed to condition the ambient fresh air. This shows that more energy will be needed in order to cold and dry the ambient fresh air especially for hot and humid weather. At the same time, large amount of energy can be recovered by utilizing enthalpy heat exchanger since it can recover both sensible and latent effectiveness. Energy analysis presented in Fig. 6 proved that the actual energy consumed or recovered was dependent on the nature of the permeable surface moisture transfer resistance and on the inlet air condition.

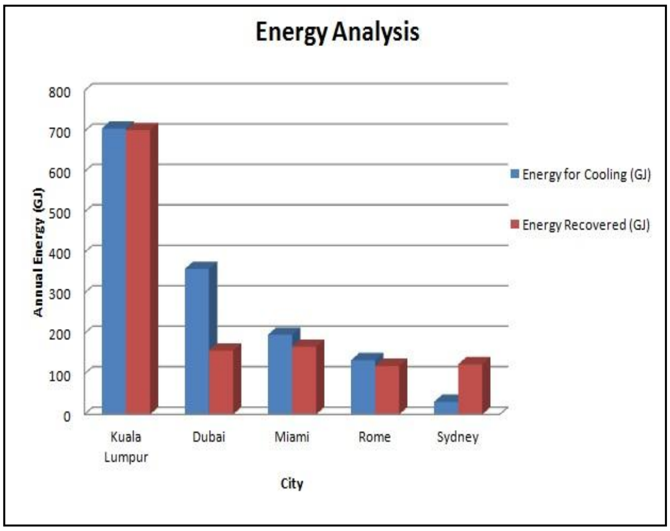

Figure 6: Comparison of the annual energy for- cooling consumption and annual energy recovered for 5cities

4 CONCLUSIONS

The performance of a Z flow heat exchanger utilizing 70gsm Kraft paper as heat and moisture transfer surface has been experimentally evaluated under constant air flow rate and different inlet air moisture content conditions. The performance has been determined in terms of both sensible and latent effectiveness. A mathematical model of the sensible and latent characteristics of the paper membrane heat exchanger has been developed. The developed mathematical model was integrated into the air conditioner cooling coil energy model to calculate the annual energy consumed and energy recovered by the heat exchanger under constant paper moisture transfer resistance.

It has been shown that the sensible effectiveness was near constant, whereas the latent effectiveness exhibited noticeable variation. This variation was attributed to the dependence of moisture resistance of the permeable surface on the variable inlet humidity ratios. In addition, the actual energy consumed and recovered was dependent on the nature of the permeable paper moisture transfer resistance. Energy calculations when Kuala Lumpur weather data was used shows significantly large amount of energy recovered since the cold and dryness was extracted from the room exhaust air to pre-condition the ambient fresh air. This shows the significant effect the inlet humidity ratio has on the energy analysis calculations of air conditioning systems coupled with enthalpy heat exchanger.

As conclusion, the objective of this project is achieved. The use of Z shape enthalpy heat exchanger can create a significant reduction in energy consumption for cooling and dehumidification process in any HVAC system since it uses the room exhaust air to pre-condition the ambient fresh air.

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