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Membrane heat exchanger in HVAC energy recovery systems, systems energy analysis

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ABSTRACT

The thermal performance of an enthalpy/membrane heat exchanger is experimentally investigated. The heat exchanger utilizes a 60gsm Kraft paper as the heat and moisture transfer surface for HVAC energy recovery. The heat exchanger sensible, latent and total effectiveness have been determined through temperature and moisture content measurements. The annual energy consumption of an air conditioner coupled with an enthalpy/membrane heat exchanger is also studied and compared with a conventional air conditioning cycle using in-house modified HPRate software. The heat exchanger effectiveness are used as thermal performance indicators and incorporated in the modified software. Energy analysis showed that an air conditioning system coupled with a membrane heat exchanger consumes less energy than a conventional air conditioning system in hot and humid climates where the latent load is high. It has been shown that in humid climate a saving of up to 8% in annual energy consumption can be achieved when membrane heat exchanger is used instead of a conventional HVAC system.

1. Introduction

The requirement for improved indoor air quality has become a great concern worldwide [1–4] necessitating the use of 100% fresh air in HVAC systems for certain buildings, resulting in a significant increase in the building cooling/heating loads [5,6]. For such systems, it is essential to utilize energy recovery systems to reduce 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 substantial amount of energy is recovered which in return reduces the overall HVAC energy requirement.

The development of energy recovery systems over the last few decades has led to improved performance and capability in recovering both sensible and latent energy, where the latent load constitutes a large fraction of the total thermal load in the HVAC system [6,7]. Enthalpy heat exchangers that utilize a porous membrane as the heat and moisture transfer surface is one device that can recover both sensible and latent energy. The primary advantages of a membrane system are: it is simple to construct, it is a static device that does not involve any moving parts and can be easily integrated into existing air conditioning systems.

Within the last decade, researchers have been active in studying the performance of the membrane heat exchanger in HVAC systems. The majority of studies were performed on cross-flow heat exchangers. Furthermore, extensive research has been performed on modelling cross flow membrane heat exchangers utilizing computational fluid dynamics (CFD) simulation to investigate the heat exchanger performance and analyse the heat and moisture transfer mechanism [8–12].

The promising energy savings associated with latent recovery has attracted many researchers to perform theoretical and experimental energy analysis on the membrane heat exchanger to examine the membrane heat exchanger energy recovery performance.

Zhang and Niu [13] performed an energy saving comparison between a sensible heat exchanger and membrane/enthalpy heat exchanger in Hong Kong. Their computer simulation of HVAC energy recovery has been performed by fixing the air conditioner air design set point conditions in order to simplify the simulation. However in real air conditioning cycles, conditions of the air exiting the evaporator and condenser change according to the ambient air conditions. Their simulation results have shown that membrane

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Nomenclature

Nomenclature

A	area (m^2)
C_p	specific heat capacity (kJ/kg K)
E	energy
$\dot{m}$	mass flow rate (kg/s)
h	enthalpy (kJ/kg)
h_{fg}	enthalpy of evaporation (kJ/kg)
Q_s	heat (kW)
T	temperature (°C)
U	heat transfer coefficient (kW/m^2 K)

Greek letters

ε	effectiveness
ω	humidity ratio (kg/kg dry air)

Subscripts

A	air
c	cold air stream
Cond	condenser
e	exhaust air
evap	evaporator
h	hot air stream
i	inlet
L	latent
o	outlet
s	supply air
S	sensible
tot	total

heat exchanger performed well in humid weather conditions when compared with the sensible heat exchanger.

Zhang [6] conducted a simulation of an air conditioning system which uses 100% fresh air to determine the efficiency of an air conditioner that incorporates different energy recovery systems including an enthalpy heat exchanger. Zhang's energy analysis shows that a system that incorporates an enthalpy heat exchanger consumes less energy than other energy recovery system. Zhang also found that enthalpy heat exchangers consume significantly less energy than a system that uses 100% fresh air without an energy recovery device.

Zhang et al. [7] developed a theoretical thermodynamic model of an air conditioner that incorporates membrane-based heat exchanger air dehumidification system. The thermodynamic modelling of the system was performed through hour-by-hour analysis in a humid region in China. Their results show that utilizing membrane heat exchanger in HVAC system provides energy saving of up to 33% compared to systems without membrane heat exchanger.

Liu et al. [14] investigated the applicability and energy performance of an enthalpy heat exchanger in different climate zones in China. They derived latent and sensible weighted coefficients. These coefficients were used as a benchmark to evaluate the suitability of the heat exchanger (sensible or latent) for certain design climate conditions in China. They also investigated the performance of the heat exchanger by using EnergyPlus software.

They reported that the energy saving performance of the heat exchanger is dependent on the outdoor design conditions, total efficiency, fan power consumption and fresh air change.

Delfani et al. [15] performed experimental investigations to evaluate the effect of an enthalpy heat exchanger on reducing the air moisture content before it enters the cooling coil. They performed their experiment in different weather conditions in Iran for a period of 60 min. They discovered that the introduction of the membrane heat exchanger contributed significantly to the reduction of the latent load.

Zhou et al. [16] proposed the term named "the ratio of energy recovered to the energy supplied by the utilized HVAC system". They also investigated the energy recovery system performance under different indoor temperature set points for different weather conditions using EnergyPlus software. They found that the proposed ratio is linearly proportional with indoor temperature set points. They also noted that utilizing energy recovery system in Shanghai is beneficial throughout the year. However the performance of the heat exchanger could be adversely affected by increasing the indoor set point.

Liang et al. [17] modelled an air conditioner that incorporates a membrane heat exchanger and validated the modelled system performance against measurements. They reported that under hot and humid conditions an air conditioner coupled with membrane heat exchanger is more robust than an air conditioner with no energy recovery system. They concluded that incorporating the membrane heat exchanger in HVAC system improves the air conditioner performance significantly.

The most accurate way to determine the performance of an air conditioner is to use a calorimeter measurement. However, these measurements are costly and time consuming [18]. Hence, it is better to perform research on heat pumps based on computer simulation programs which are based on energy and thermodynamic equations of the refrigeration and air cycles and performing energy balance for the system.

The standard AS/NZS 3823.3 [19] specifies a method of performance evaluation using a computer simulation tool such as HPRate.

HPRate is a performance rating tool that evaluates the performance of vapour compression air conditioning cycles [18]. The program predicts the steady state performance of electrically driven, vapour compression, air-to-air heat pumps in both heating and cooling modes. It consists of a FORTRAN model of the heat pump components.

Several studies have been conducted on air conditioners to assess the accuracy of HPRate in terms of its cooling capacity, power consumption and energy efficient ratio prediction. For standard air conditioner operation, the predictions of air conditioner operation were found to have deviations from measured conditions of 2.6% in cooling capacity, 1.7% in power consumption and 2.8% for EER [18].

However, HPRate is able to predict the air conditioner performance only. Therefore, the need arises for a method for assessing the performance of various combinations of energy recovery devices with a standard air conditioner under varying operating conditions throughout the year.

In this research, the enthalpy heat exchanger under investigation is a Z-flow configuration heat exchanger utilizing 98 μm thick 60gsm Kraft paper as the heat and moisture transfer surface as shown in Fig. 1. The Z-flow configuration provides a counter flow arrangement over most of the transfer surface. Therefore, heat and moisture transfer will be maximized relative to the ideal counter flow arrangement over a substantial part of the heat exchanger surface.

The HPRate code is modified to model an air conditioner which includes the effect of adding a sensible and latent heat recovery heat exchanger to a conventional air conditioning system. The modified HPRate code is combined with a model of an office space in order to determine the transient operating states of the heat exchanger/air conditioner throughout the year.

HPRate is used to evaluate the power consumption on a 5 min time step throughout the year. The HPRate model is used to simulate off design performance so that the annual performance of a combined cooling/heating system could be determined. The heat exchanger effectiveness is incorporated in a model of an air

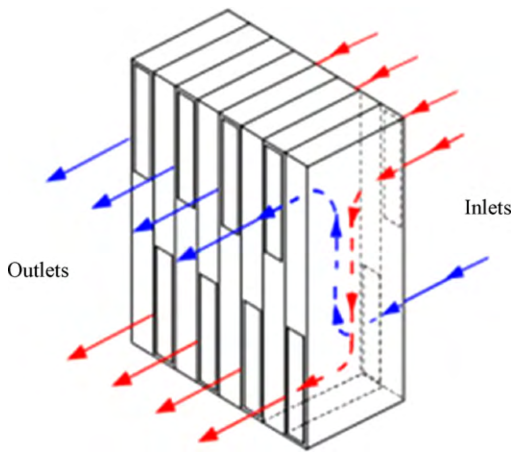

Fig. 1. Z type flow heat exchanger.

conditioner with pre-conditioning membrane heat exchanger to investigate the annual energy consumption of the system.

The modified HPRate code is then used to model a conventional air conditioning system that operates based on mixing fresh air with room exhaust air which is widely used in air conditioning.

2. Experimental setup

A test rig, similar to the one shown in Fig. 2, has been set up under laboratory conditions to evaluate the performance of the enthalpy heat exchanger. The performance is evaluated by calculating the sensible, latent and total effectiveness of the heat exchanger from flow rate and wet and dry bulb temperature measurements at the inlets and outlets of the heat exchanger. The experimental rig consists of two separate ducts. Heaters are used in the inlet of the top duct to vary the heat difference between the two streams and centrifugal fans located at the duct inlets provide the flow for the two air streams. Steam is also injected in the top hot inlet stream to create a moisture gradient across the heat exchanger.

The heat exchanger has an overall dimension of 0.6 m × 0.72 m × 0.3 m and constructed of 98 plastic frame flow passages with 49-inlet air passages in each of the air streams. Each flow passage is designed with a 'Z' shape flow configuration. The heat and moisture transfer surface is made of 98 μm thick 60gsm Kraft paper. The plastic frames are assembled together in an alternating pattern and the passage of the cold air stream is laterally inverted from the passage direction of the hot air stream, as shown in Fig. 3.

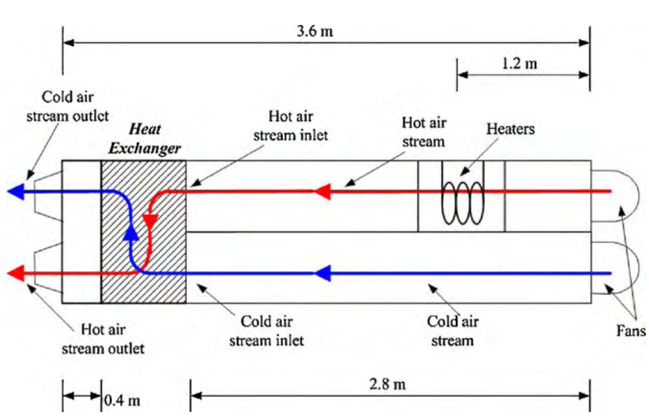

Fig. 2. Experimental rig for testing Z-flow heat exchanger.

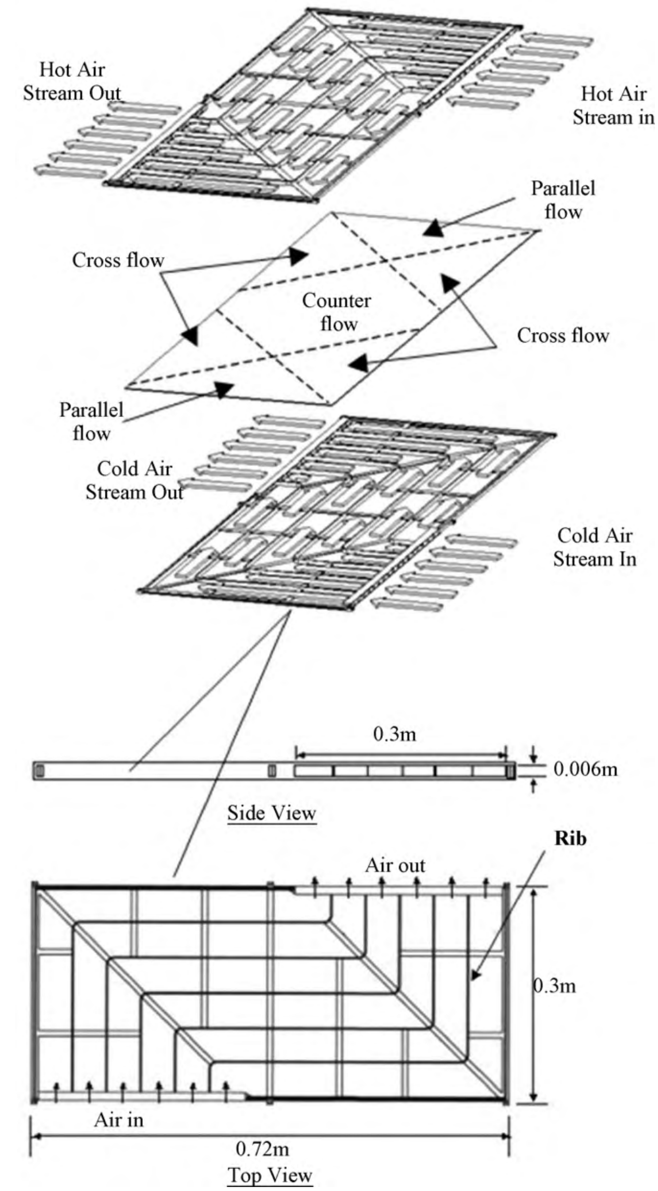

Fig. 3. Heat exchanger plastic frame flow channels.

At the heat exchanger inlet streams, the air dry bulb temperature and relative humidity are measured to obtain the moisture content. The 'T' type thermocouples are utilized to measure the air temperature at 18 different points across the air flow ducts before the exchanger. Nine thermocouples are located at each of the hot and cold inlets. Thermocouples are uniformly distributed across both of the inlet and outlet ducts. Each duct is divided into equal area segments and a sensor placed in the middle of each segment. The relative humidity of the inlet air is obtained by using a humidity probe located at the same locations across the duct as the temperature measurements in the two streams.

At the outlet section of the heat exchanger, a sampling tree is used to measure outlet temperature and moisture conditions (Fig. 4). Two thermocouples are installed inside the sampling tree outlet tube to measure average wet and dry bulb temperatures; thereby the moisture content of the outlet air is obtained, details of the airflow sampling system are shown in Fig. 5. The sampling tree is needed to ensure that the correct average temperature and moisture fraction are determined from the non-uniform outlet streams. The air velocity is measured using a hot-wire anemometer. The

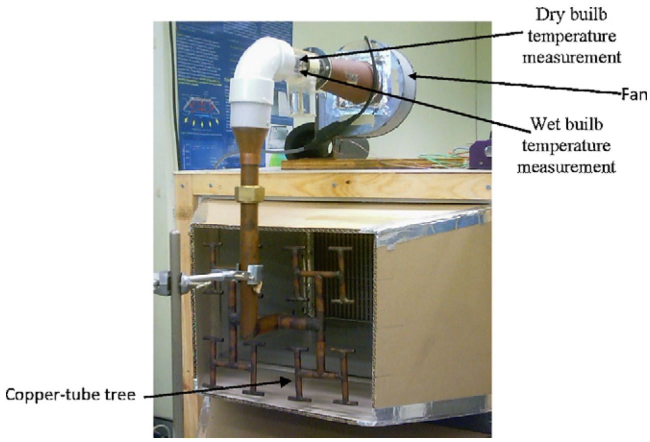

Fig. 4. Airflow sampling system installed at the heat exchanger outlet streams.

velocity is measured at 9 points across the duct outlets and the readings are averaged to obtain the air velocity in each stream.

2.1. Performance parameters

The performance of an air conditioner using an energy recovery system is influenced by local meteorological conditions in addition to the cooling and heating loads of the building. In order to evaluate the performance of the heat exchanger experimentally, the sensible, latent and total effectiveness are calculated using Eqs. (1)–(3) respectively.

$$\varepsilon_S = \frac{\dot{m}_s C_p(T_{hi} - T_{ho}) + \dot{m}_e C_p(T_{co} - T_{ci})}{2\dot{m}_{\min} C_p(T_{hi} - T_{ci})} \quad (1)$$

$$\varepsilon_L = \frac{\dot{m}_s h_{fg}(\omega_{hi} - \omega_{ho}) + \dot{m}_e h_{fg}(\omega_{co} - \omega_{ci})}{2\dot{m}_{\min} h_{fg}(\omega_{hi} - \omega_{ci})} \quad (2)$$

$$\varepsilon_{tot} = \frac{\dot{m}_s(h_{hi} - h_{ho}) + \dot{m}_e(h_{co} - h_{ci})}{2\dot{m}_{\min}(h_{hi} - h_{ci})} \quad (3)$$

Great care is taken to ensure that the measured data satisfies both sensible and latent energy balances between the two moist air streams passing through the heat exchanger. An error margin of 1.9–8.13% on sensible energy balance and 1.1–8.82% on latent energy balance between the hot and cold stream is recorded in this experiment which is considered to be acceptable.

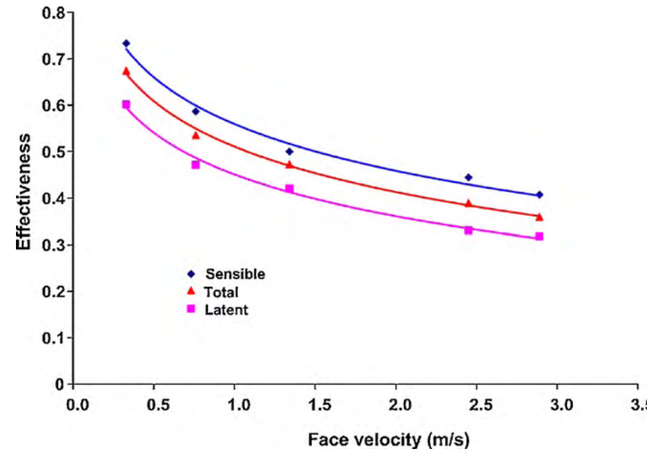

Fig. 6. Experimental sensible, latent and total effectiveness for 60gsm paper heat exchanger.

2.2. Effectiveness and pressure loss

The effectiveness of the heat exchanger that resulted from the current experiment is shown in Fig. 6. The measurements are performed for air face velocity ranging from 0.3 m/s to 2.89 m/s. The operating air speed reflects the amount of time the air resides within the heat exchanger. For this range of air velocities the larger the resident time, the higher the heat transfer and effectiveness. Therefore, as the air velocity increases, the effectiveness declines.

The pressure drop across the heat exchanger has been measured because it will contribute to the additional fan power. The static pressure difference is measured between the inlet and outlet for both streams using static pressure tapping points drilled through the test rig walls and connected to the manometer tubes located 50 mm from the heat exchanger inlets and outlets. From the pressure measurements at the heat exchanger inlet and outlet streams, Fig. 7 shows that the pressure drop is proportional to the air velocity.

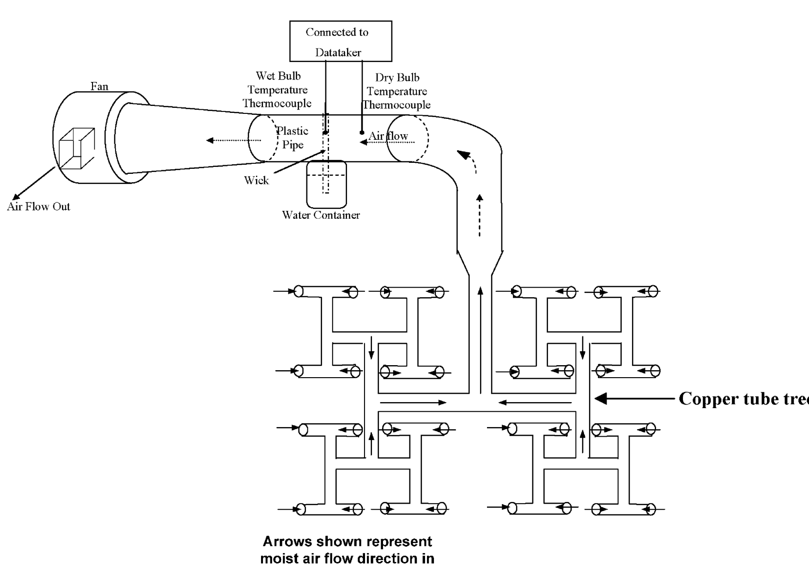

Fig. 5. Details of the airflow sampling system.

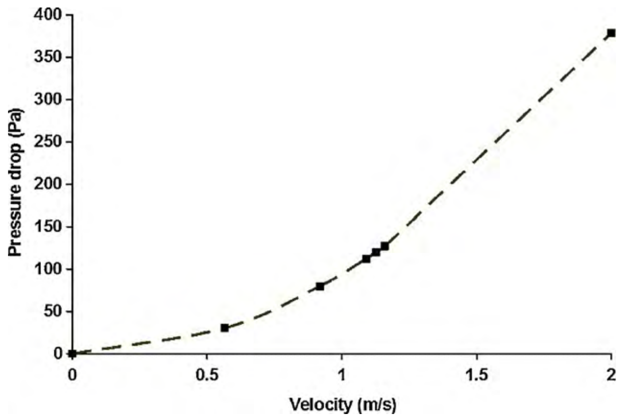

Fig. 7. Pressure drop measurements.

3. HPRate code development

The HPRate model is modified to include the membrane heat exchanger in the system. The original HPRate code was designed to quantify the performance of air conditioner according to the standard AS/NZS 3823.3 [19] rating point conditions. In the code developed here, HPRate has been extended to model air conditioner performance throughout a year for variable operating conditions specified by ambient temperature and humidity in standard typical meteorological year weather files for the location of interest [20].

The new code reads the hourly weather data (dry and wet bulb temperature) for any city around the globe presented in the typical meteorological year (TMY) format for evaluation of an air conditioner coupled with enthalpy energy recovery system.

The modified HPRate code reads the hourly weather data and interpolates for shorter time steps to achieve high sensitivity in the system modelling. Under this simulation, the modified HPRate code loops through 8760 h of the weather data and at each hour the weather data is interpolated into 5 min time steps. As the energy recovered is significantly affected by the heat exchanger effectiveness, the measured heat exchanger effectiveness is used to obtain the air-on conditions supplied to the condenser and evaporator. For given outdoor conditions, the annual energy consumed by the air conditioner to cool/heat any room can be determined.

Two systems are studied, the first is an air conditioning system coupled with an enthalpy heat exchanger. The second system is a conventional air conditioning system which operates based on mixing of 35% fresh air with 65% room exhaust air.

For both systems 1000 L/s air flow is supplied to the evaporator and 1500 L/s is supplied to the condenser. For the enthalpy heat exchanger system 1000 L/s room exhaust air is mixed with 500 L/s ambient fresh and supplied to the condenser coil. For the conventional system 350 L/s room exhaust air is mixed with 1150 L/s ambient fresh air and supplied to the condenser.

The analysis is conducted for an office space of 300 m^2. The standard AS 1668.2 [4] requires that 10 L/s of fresh air is to be supplied for each occupant. Moisture generation is assumed to be 0.04 kg/h per occupant.

If the air-on temperature entering the air conditioner is less than 18 °C, the operation mode of the heat pump will be changed to heating mode. However, if the air-on temperature is higher than 24 °C, the operation mode is switched to cooling mode.

In situations when the temperature of the supplied air to the air conditioner is between 24 °C and 18 °C, the air conditioner compressor is turned off and the heat exchanger acts as a passive cooling or heating device for the room. Since the air conditioner compressor is not operating under these conditions, the simulated results in terms of energy consumption will be the energy consumed to operate the fans only. If the air conditioner is turned off most of the time, then the energy recovered will be high. This is because cooling and heating can be achieved without operating the compressor which consumes a large amount of energy.

To enable HPRate to execute the above tasks, new subroutines have been developed and the HPRate code has been modified and included in the simulation package. The developed subroutines incorporate the enthalpy heat exchanger effectiveness. The hourly weather temperature and humidity are read and interpolated in the subroutines into 5 min time steps and the air conditions exiting the heat exchanger are determined from Eqs. (4)–(7). Based on these equations, the air conditioner inlet air conditions to the coils are obtained.

$$T_{evap} = T_a - \varepsilon_s(T_a - T_{room}) \tag{4}$$

$$T_{cond} = T_{room} + \varepsilon_s(T_a - T_{room}) \tag{5}$$

$$\omega_{evap} = \omega_a - \varepsilon_L(\omega_a - \omega_{room}) \tag{6}$$

$$\omega_{cond} = \omega_{room} + \varepsilon_L(\omega_a - \omega_{room}) \tag{7}$$

In order to model a conventional air conditioner that operates based on mixing of 35% fresh air with 65% room exhaust air, the developed subroutines calculate the air conditioner air-on conditions from the following equations:

$$\omega_{air\ on} = 0.65\omega_{room} + 0.35\omega_{ambient} \tag{8}$$

$$h_{room} = 1.005T_{room} + \omega_{room}(2501 + 1.83T_{room}) \tag{9}$$

$$h_{ambient} = 1.005T_{ambient} + \omega_{ambient}(2501 + (1.83T_{ambient}) \tag{10}$$

$$h_{air\ on} = 0.65h_{room} + 0.35h_{ambient} \tag{11}$$

Hence, the air conditioner air-on temperature is obtained as follows:

$$T_{air\ on} = \frac{h_{air\ on} - (2501\omega_{air,on})}{1.005 + (1.83\omega_{air,on})} \tag{12}$$

Similarly, the air-on temperature supplied to the condenser coil based on the air mixing conditions (for the heat exchanger system, 1000 L/s room exhaust air mixed with 500 L/s ambient fresh air and for the conventional system, 350 L/s room exhaust air mixed with 1150 L/s ambient fresh air) is calculated.

HPRate code shall read the data from the above codes and perform compressor, evaporator and condenser performance calculations iteratively and performs the simulation based on the supplied air conditions (air-on conditions). This analysis is continued throughout the year using the five-minute time step weather data. The office temperature is calculated from energy balance on the space according to the following Eqs. (13)–(16).

The cooling/heating provided by air conditioner:

$$Q_{cooling/heating} = m_{air}C_{pair}(T_{room} - T_{evap}) \tag{13}$$

The heat transfer through the walls is:

$$Q_{heat} = AU_{office}(T_{ambient} - T_{room}) \tag{14}$$

The cooling load provided by air conditioner is also the sum of the heat transferred via walls and the heating load within the space:

$$Q_{cooling} = Q_{heat} + Q_{load} \tag{15}$$

Substituting Eqs. (13) and (14) into Eq. (15), the space temperature is obtained:

$$T_{room} = \frac{AUT_{ambient} + m_{air}C_{p\,air}T_{evap} + Q_{Load}}{m_{air}C_{pair} + AU} \tag{16}$$

3.1. Assumptions and limitations

- Transient (cyclic or frosting/defrosting) effects are not considered and the program has physically based the heat transfer models for single and two phase refrigerant regions of fin-and-tube air to refrigerant heat exchangers. Parallel and series refrigerant circuiting is evaluated and air-side dehumidification and evaporator sensible heat supply are calculated.
- It is assumed that the system is charged with the correct amount of refrigerant for the specified operating conditions. This is because the user cannot specify the refrigerant charge in the current model.
- To enable the code to perform the first calculation where the room air conditions are unknown, it is assumed that for the first 5 min the room temperature and relative humidity are 24 °C and 50% respectively.
- The building thermal mass is not included in this model.
- The power required from the fan to overcome the pressure drop in the heat exchanger is included in the model.

4. System energy analysis

The annual energy consumption for the investigated systems is studied and a comparison on the annual energy use for cooling and heating of each system in different locations is performed. Then, the energy consumption of an air conditioner for a range of enthalpy heat exchanger face areas is evaluated.

4.1. Simulation performance

HPRate simulation is performed for Sydney and Kuala Lumpur weather conditions. The weather in Sydney is moderate, while it is hot and humid in Kuala Lumpur. The simulation is performed for an air conditioning system coupled with an enthalpy heat exchanger and a conventional air conditioning system based on air mixing. In cases where the air temperature entering the evaporator is between 24 and 18 °C, the compressor is switched off and the heat exchanger or the air mixing zone will then act as a passive cooling or heating device for the room. Under these conditions when the air conditioner compressor is not operating, the simulated result in terms of energy consumption will be the energy used to operate the evaporator and condenser fans only. The analysis is conducted for an office space of 300 m^2 with operating hours from 9 am till 6 pm and for an internal load of 2 kW and occupant moisture generation of 0.04 kg/h per person.

In this simulation the air volumetric flow rate supplied to the evaporator is 1000 L/s and 1500 L/s is supplied to the condenser and the refrigerant used is R22. The enthalpy heat exchanger inlet stream face area is 3.3 m^2 and the air face velocity of the heat exchanger is 0.3 m/s. The enthalpy heat exchanger sensible, total and latent effectiveness for an air face velocity of 0.3 m/s are 0.71, 0.66 and 0.61 respectively. The effectiveness is obtained from the 60gsm paper heat exchanger effectiveness curves shown in Fig. 6.

HPRate modelled evaporator and condenser specifications are according to the industrial standard models. The evaporator consists of four refrigerant plain tubes made of copper with a frontal area of 0.5 m^2 and a wavy fin surface made of aluminium. The condenser consists of five refrigerant plain copper tubes with a frontal area of 0.6 m^2 and a wavy fin surface made of aluminium. HPRate compressor refrigeration capacity and power are based on the condenser and the evaporator temperatures.

4.2. Annual energy analysis—Sydney

Energy consumption of the air conditioner obtained from the above simulation using Sydney hourly weather data is presented

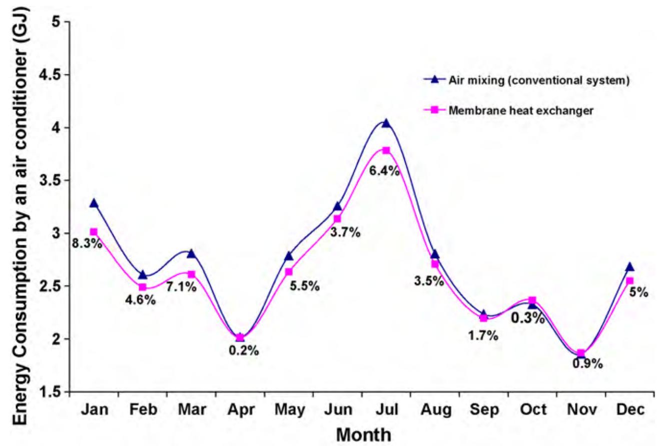

Fig. 8. Sydney monthly energy consumption for reverse cycle air conditioner (figures show percentage energy difference between the two systems).

in Fig. 8. It also shows that the air conditioning system that utilizes an enthalpy heat exchanger consumes less energy than the conventional air conditioning system that operates based on air mixing. The enthalpy air conditioner has achieved lower operating cost while simultaneously providing 100% fresh air.

When the weather is hot and humid in summer and the sensible and latent cooling loads are high, the amount of energy consumed by enthalpy heat exchanger system is 5%, 8.3% and 4.6% less in December, January and February respectively than a conventional air conditioning system. Similarly in March the air conditioning system coupled with an enthalpy heat exchanger consumes less energy than the conventional system. Whereas in April, the amount of energy consumed recorded its lowest values, the energy consumption for both systems is almost the same due to the moderate weather.

The energy consumption started to increase in winter season (from May to July) as the weather became colder and heating load became higher. Nonetheless, the system coupled with enthalpy exchanger system consumes 6.4% less energy than the conventional reverse cycle air conditioning system. When spring season began, the energy consumption decreases and the air conditioning system coupled with enthalpy heat exchanger continue to consume less energy. However, the energy consumption difference between both systems is less in the winter heating season than in the summer cooling season.

Seasonal energy analysis shows that the energy saving recorded by an air conditioning system coupled with an enthalpy heat exchanger in winter season is 4.7% less than conventional air conditioning system. In summer the humidity and temperature increase in Sydney, hence, the heat exchanger acts as both an energy recovery and dehumidifying tool which will reduce the latent load.

Consequently, in summer, energy consumption of an air conditioning system coupled with enthalpy heat exchanger is 6.2% less than the conventional system as shown in Fig. 9. This shows the importance of utilizing the enthalpy heat exchanger in an air conditioning system as an energy recovery and dehumidifying tool to reduce the latent load while simultaneously providing 100% fresh air.

4.3. Annual energy analysis—Kuala Lumpur

In a tropical climate like Kuala Lumpur, the weather is hot and humid throughout the year and the latent load is high. Fig. 10 shows the annual monthly energy consumption is almost the same throughout the year. However, it can be seen that the air conditioning system coupled with an enthalpy heat exchanger consumes less

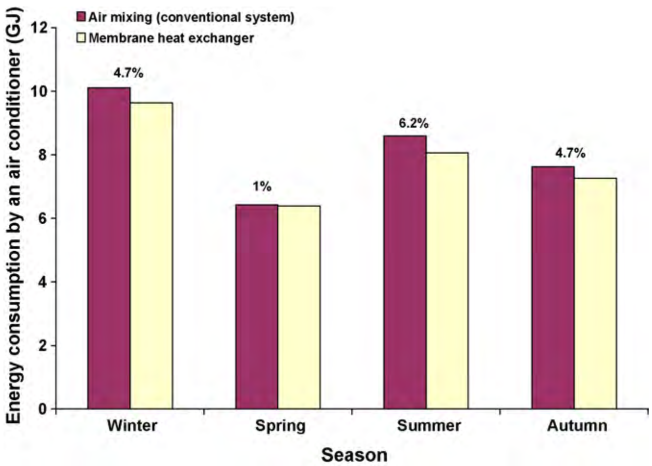

Fig. 9. Sydney seasonal energy consumption for reverse cycle air conditioning systems (figures show the energy difference between the two systems).

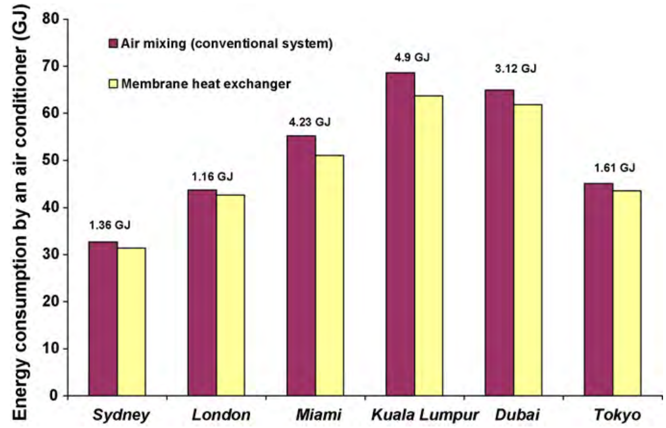

Fig. 11. Annual energy consumption for reverse cycle air conditioner (figures show the energy difference between the two systems).

energy than the conventional air conditioning system. The enthalpy exchanger system consumes 5.7–9.0% less energy than the conventional system resulting in energy savings throughout the year. This is due to the hot and humid climate in Kuala Lumpur throughout the year, where the amount of energy required to dehumidify the air by an air conditioner is large. Hence, utilizing an enthalpy heat exchanger to dehumidify the air before it enters the air conditioning system will contribute significantly in reducing the latent load, resulting in energy savings.

4.4. Annual energy analysis—worldwide

HPRate simulation and energy analysis are performed on different cities such as London, Miami, Tokyo and Dubai.

The summary of the total annual energy analysis shown in Fig. 11 illustrates that the highest annual energy consumption recorded is in Kuala Lumpur. Where, using enthalpy heat exchanger system resulted in 4.9 GJ energy saving in comparison with the conventional air conditioning system. In Miami, utilizing enthalpy heat exchanger in an air conditioning system has recorded 4.23 GJ energy saving. In Dubai, due to the hot and humid climate in spring, summer and autumn, the enthalpy heat exchanger system annual energy consumption is 3.12 GJ less than the conventional

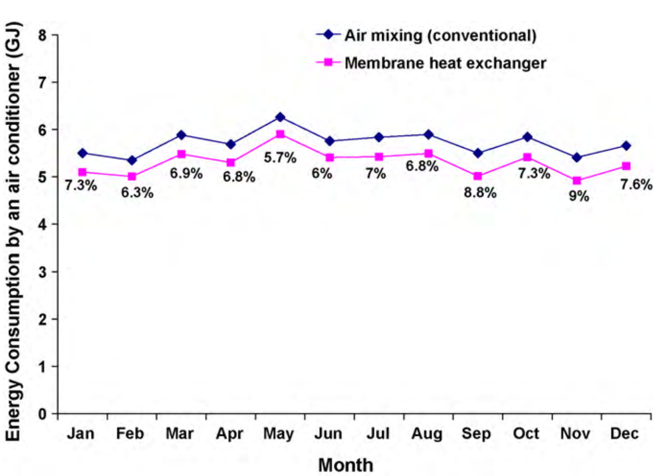

Fig. 10. Kuala Lumpur monthly energy consumption for reverse cycle air conditioner (figures show the percentage energy difference between the two systems).

system. In Tokyo, the annual energy savings is 1.61 GJ. Although the annual energy consumption in Sydney is the lowest in comparison with other cities, the air conditioning system coupled with an enthalpy heat exchanger consumes 1.36 GJ less than the conventional air conditioning system. In London, the annual energy consumption is relatively high due to the cold climate. Nevertheless, an air conditioning system coupled with an enthalpy exchanger consumes 1.16 GJ less than the conventional air conditioning system.

The above energy analysis showed that an air conditioning system coupled with an enthalpy heat exchanger performed well in terms of energy consumption in comparison with conventional air conditioning system in all locations investigated. In addition to the sensible energy recovered, the enthalpy heat exchanger also decreases energy consumption in hot and humid climate by reducing the latent load where the heat exchanger dehumidifies the air before it enters the air conditioning system, causing a decrease in energy consumption. Hence, the decrease in energy consumption is higher in hot and humid climates like Miami, Kuala Lumpur and Dubai. This shows the importance of reducing the latent load to achieve lower energy consumption.

4.5. Effect of heat exchanger face area on energy consumption

To study the effect of varying the heat exchanger face area on energy consumption, Kuala Lumpur weather data is used as a bench mark to perform this investigation since enthalpy heat exchanger performs well and consumes less energy in a hot and humid climate.

The energy savings is calculated as the difference between the energy consumption of an air conditioner that incorporates an enthalpy heat exchanger and a conventional air conditioner that operates based on air mixing

$$E_{saving} = E_{Enthalpy\,exchanger\,system} - E_{conventional\,system} \quad (17)$$

The area ratio (A_{ratio}), shown in Fig. 12, represents the ratio of the enthalpy heat exchanger face area to the face area of the evaporator coil (0.5 m^2). As the enthalpy heat exchanger face area increases, it can be seen from Fig. 12 the amount of energy saved increases. Increasing the heat exchanger face area causes the air velocity to decrease. Therefore, the heat exchanger effectiveness increases.

It can be seen that a substantial amount of energy is saved when the enthalpy heat exchanger is incorporated in an air conditioner especially in tropical climates. In addition to the energy saving, an air conditioner coupled with an enthalpy heat exchanger also

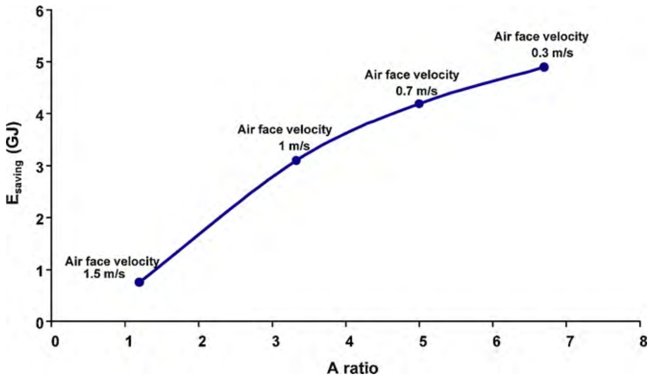

Fig. 12. Effect of changing enthalpy heat exchanger face area on annual energy saving in Kuala Lumpur (air face velocity indicated on top of each point).

has the advantage of providing 100% fresh air which significantly improves indoor air quality.

5. Conclusion

The performance of Z type flow enthalpy/membrane heat exchanger utilizing 60gsm Kraft paper as heat moisture transfer surface has been experimentally evaluated. The performance is determined in terms of both sensible and latent effectiveness.

The effective utilization and annual performance of an air conditioner coupled with enthalpy heat exchanger is investigated in relation to a conventional air conditioning system that operates based on mixing of fresh air with the room exhaust air.

The annual performance investigation is achieved by developing a modified version of HPRate software which models a vapour compression, electrically driven, air-to-air reverse cycle heat pump. The HPRate modified version reads the yearly weather data of different cities around the globe, and incorporates the measured enthalpy heat exchanger effectiveness functions.

Energy analysis shows that an air conditioning system coupled with an enthalpy heat exchanger contributed significantly in reducing the latent load and consumed 8% less energy throughout the year than the conventional air conditioning system in tropical climates. Whereas, in a moderate climate, systems coupled with enthalpy heat exchanger consumed 4% less energy than the conventional air conditioning system.

These results show the significant contribution of the enthalpy heat exchanger in reducing the latent load in hot and humid climates and the substantial energy savings achieved in comparison with conventional air conditioning systems. In addition, enthalpy heat exchanger has the advantage of providing 100% fresh air.

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References

[1] T. Paul, D. Sree, H. Aglan, Effect of mechanically induced ventilation on the indoor air quality of building envelopes, Energy and Buildings 42 (2010) 326–332.
[2] P.M. Bluyssen, Towards an integrative approach of improving indoor air quality, Building and Environment 44 (2009) 1980–1989.
[3] R. Kosonen, F. Tan, The effect of perceived indoor air quality on productivity loss, Energy and Buildings 36 (2004) 981–986.
[4] Standard AS 1668.2, The use of mechanical ventilation and air-conditioning in buildings—mechanical ventilation for acceptable indoor-air quality, Standards Australia (1991).
[5] A. Nguyen, Y. Kim, Y. Shin, Experimental study of sensible heat recovery of heat pump during heating and ventilation, International Journal of Refrigeration 28 (2005) 242–252.
[6] L.Z. Zhang, Energy performance of independent air dehumidification systems with energy recovery measures, Energy 31 (8–9) (2006) 1228–1242.
[7] L.Z. Zhang, D.S. Zhu, X.H. Deng, B. Hua, Thermodynamic modelling of a novel air dehumidification system, Energy and Buildings 37 (3) (2005) 279–286.
[8] L.Z. Zhang, C. Liang, L. Pei, Heat and moisture transfer in application scale parallel-plates enthalpy exchangers with novel membrane materials, Membrane Science 325 (2010) 672–682.
[9] L.Z. Zhang, F. Xiao, Simulation heat and moisture transfer through a composite supported liquid membrane, International Journal of Heat and Mass Transfer 51 (2008) 2179–2189.
[10] J.L. Niu, L.Z. Zhang, Membrane-based Enthalpy Exchanger: material considerations and clarification of moisture resistance, Membrane Science 189 (2001) 179–191.
[11] L.Z. Zhang, Heat and mass transfer in plate-fin enthalpy exchangers with different plate and fin materials, International Journal of Heat and Mass Transfer 52 (2009) 2704–2713.
[12] L.Z. Zhang, Convective mass transport in cross-corrugated membrane exchanger, Membrane Science 260 (2005) 75–83.
[13] L.Z. Zhang, J.L. Niu, Energy requirements for conditioning fresh air and the long-term savings with a membrane-based energy recovery ventilator in Hong Kong, Energy 26 (2001) 119–135.
[14] J. Liu, W. Li, J. Liu, B. Wang, Efficiency of energy recovery ventilator with various weathers and its energy saving performance in residential apartment, Energy and Buildings 42 (2010) 43–49.
[15] S. Delfani, H. Pasdarshahri, M. Karami, Experimental investigation of dehumidification process in cooling coil by utilizing air-to-air heat exchanger in humid climate in Iran, Energy and Buildings 42 (2010) 822–827.
[16] Y.P. Zhou, J.Y. Wu, R.Z. Wang, Performance of energy recovery ventilator with various weather and temperature set-point, Energy and Buildings 39 (2007) 1202–1210.
[17] C.H. Liang, L.Z. Zhang, L.X. Pei, Independent air dehumidification with membrane-based total heat recovery: modelling and experimental validation, International Journal of Refrigeration 33 (2010) 398–408.
[18] G.L. Morrison, Air conditioner performance rating, in: Proceedings of Asia Pacific Economic Co-operation Conference, Air Conditioning—The Next 5 years, Sydney, Australia, 2004, http://solar1.mech.unsw.edu.au/glm/papers/HPRATE_apec2004.pdf.
[19] Standard AS/NZS 3823.3, Performance of electrical appliances–Air conditioners and heat pumps. Part 3: Calculation of performance for minimum energy performance standard (MEPS) requirements, Standards Australia (2002).
[20] G.L. Morrison, A. Litvak, Condensed Solar Radiation Data Base for Australia, Report 1988/FMT/1, The University of New South Wales, Australia, 1988.