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Development and Performance Investigation of Energy Recovery System in Tropical Climate

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ABSTRACT: *With the global primary energy consumption and carbon dioxide emissions elevating at 2% and 1.7% annually, it is critical to install energy recovery systems in buildings for better energy conservation. Due to limited research on the energy recovery system in the tropical climate, this study presents the development and performance investigation of an energy recovery system in the tropical climate region. The hydrophilic polymeric membrane of the heat exchanger core was developed and organised in a cross-flow manner. Performance investigation was carried out for several operating parameters, temperature, relative humidity and air velocity. It was found that there were negative relationships between air velocity and efficiency, temperature and humidity ratio differences with increasing residence time. Ranges of latent and sensible efficiencies were 42%–74% and 45%–78%, respectively. The highest sensible energy recovered was 18 kW at the temperature intake of 40°C. One-way ANOVA showed that the air velocity significantly affects sensible and latent efficiencies at different temperature intakes.*

Keywords: Efficiency, energy recovery system, recovered energy, statistical analysis, hot-humid climate

1. INTRODUCTION

The global primary energy consumption has elevated from 75% in 1980 to 85% in 2012, with an average annual increase of 2%.[1] A study in Malaysia indicates that about 19% of the total energy consumed was from residential buildings. The increasing use of air conditioners further raises energy consumption.[2] Concerns have been raised over the potential supply difficulties, depletion energy resources and expedited negative environmental impacts due to this increasing energy demand. Thus, one of the priorities of the global energy policy makers is to enhance energy efficiency in buildings.[3]

Energy recovery system is widely employed in buildings to improve energy efficiency and ventilation. Energy recovery system reclaims sensible and latent energies from the stale exhaust air via heat exchangers using the induced fresh air from the ventilation processes.[4] Numerous research works have been done on the application of energy recovery systems in the heating, ventilation and air conditioning (HVAC) system.[5–8] However, research on the performance of the stand-alone energy recovery system should be emphasised to increase the efficiency of the integrated energy recovery system with the HVAC system.

Numerous studies using simulation tools have been conducted to assess energy recovery, but these studies were purely theoretical with little experimental validation. Wu et al. used transient systems simulation tool (TRNSYS) to evaluate energy recovery in an office building in Chingqing, China during the cool and hot seasons.[9] Fouih et al. modelled a heat recovery ventilator using the software TRNSYS and characterised its annual performance when integrated into residential and commercial low-energy buildings in France.[10] They found that the adequacy of using a heat recovery ventilation (HRV) system depends on the building type, the heating load and ventilation device. Fan and Ito analysed the energy conservation performance of a real office space with an energy recovery ventilator (ERV) and an air-conditioning system in Japan.[11] They applied a computational fluid dynamics software with the building energy simulation software, which provided a more complete and accurate information of the air flow distribution and thermal performance in an office space.

Various investigations have been carried out on the energy recovery system in the cold climate, which were mostly in the summer and winter seasons. Liu et al. investigated the performance of quasi-counter-flow membrane energy exchanger by developing an analytical model to predict heat and moisture transfers for low operating temperatures in cold climates.[12] Yang et al. conducted the performance and energy saving analyses of an energy recovery ventilator with an air-conditioning system in both cold and hot seasons in China.[5] Fan et al. carried out a field study on the thermal performance of integrated ERV packaged with air-conditioners in Japan.[13]

In contrast, limited research on the energy recovery system in the tropical climate has been conducted. The few research works available are given in the following. Nasif et al. carried out performance testing and energy analysis on a Z-type enthalpy heat exchanger in an air-conditioning system using the software HPRate.[6] Masitah et al. performed heat transfer and effectiveness analyses for a cross-flow heat exchanger.[14] Zafirah and Mardiana studied the performance of an energy recovery system with changing operating parameters.[15] Yau and Ahmadzadehtalatapeh applied a heat pipe heat exchanger (HPHX) to study the effects of the inside-design temperature and the coil-face velocity on energy recovery.[7] Zhang and Zhang found that over 80% of the energy during the operational hours of an air conditioning system can be reduced by applying the HPHX in the air handler.[16]

A large research gap exists between research result and experimental validations. Little is also known about the effectiveness of the energy recovery system in the hot tropical climate. Thus, this study focuses on the development and performance investigation of an energy recovery system in the tropical climate zone, where hot and humid climate conditions warrant the use of energy recovery systems. This research applies a cross-flow heat exchanger with a series of parallel equilateral triangular ducts (i.e., the energy recovery system) to increase the recovery efficiency of sensible and latent energies. Therefore, the primary objective of this research is to determine the efficiencies and recovered energies of the energy recovery system at different operating parameters in the tropical climate region.

2. EXPERIMENTAL

2.1 Experimental Setup

An energy recovery system (or the heat exchanger) prototype has been set up in an insulated test room (dimensions of 2 m in width, 2 m in height and 2 m in length constructed using 0.045-m polyurethane foam sandwich panels) as shown in Figure 1. Two centrifugal fans were installed in the supply and exhaust ducts of the energy recovery system for an even distribution of air velocity. Heaters and humidifiers were installed at the inlet duct to create heat and moisture differences across the heat exchanger. A portable air-conditioner unit was applied to control the temperature intake for the cold air stream.

2.2 Development of Energy Recovery System

Figure 2 shows the development of the energy recovery system. The 0.001-m thick aluminium plates and the 0.025-m thick polystyrene sandwich panels covered the heat exchanger core, which has the dimensions of 0.25 m in length, 0.25 m in width and 0.1 m in height with two separate air ducts of diameters 0.185 m. The core consists of layers of the hydrophilic polymeric membrane for the absorption of moisture. The layers are arranged in a cross-flow manner (Figure 1). This configuration has the capability to transfer heat and moisture simultaneously.

It is made of cellulose paper with corrugated structures with several internal channels to increase the surface area (Figure 3).

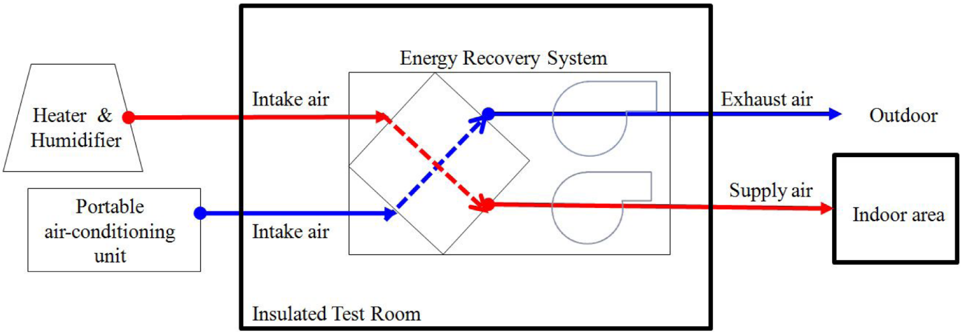

Figure 1: Experimental setup.

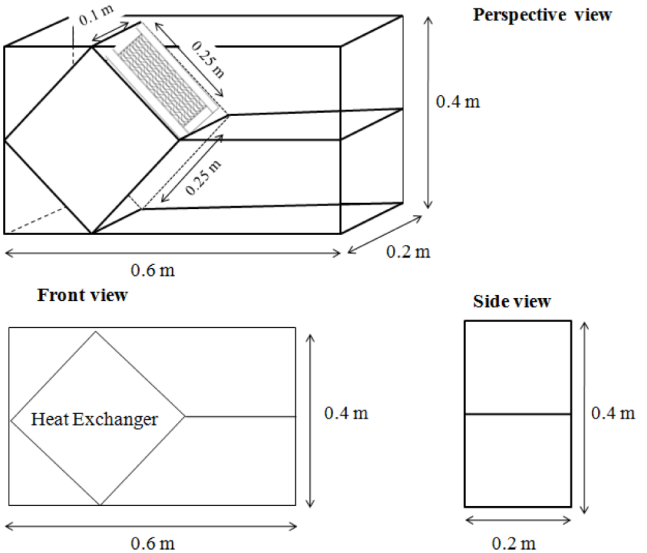

Figure 2: Development of the energy recovery system.

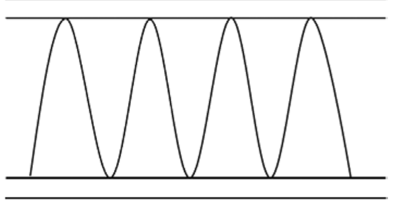

Figure 3: Corrugated structure of the plate fin channels.

2.3 Experimental Instrumentation

The operating parameters (temperature, relative humidity and air velocity) of the hot and cold air streams were measured to determine the performance of the energy recovery system. Temperature and relative humidity were measured using the HD9817T1 temperature and humidity transmitters with a temperature accuracy of ± 0.21°C and a humidity accuracy of ± 2.5%. To ensure even air velocities across the heat exchanger core, Digi-Sense 20250-16 hot-wire thermoanemometer with an accuracy of ± 0.05 m s^{-1} was used. Data were recorded for 2 h, using the datataker DT80 with the DtUsb software. To ensure that measurements were only made during steady-state conditions, data were recorded after 20 min of operation. Inlet air conditions were controlled at the temperatures of 28°C, 31°C, 34°C and 40°C with the air velocities of 1 m s^{-1}, 2 m s^{-1} and 3 m s^{-1}. These conditions are similar to the study by Zafirah and Mardiana as well as Mardiana-Idayu and Riffat.[15,17] These conditions were chosen because they are the common monthly-averaged dry bulb temperatures for several cities in the hot-humid climate zone.[18,19]

2.4 Performance Investigation

Temperature and relative humidity profiles were analysed to evaluate the performance of the energy recovery system. The efficiency and recovered energy were also calculated. Statistical analysis was conducted to assess the effects of temperature, relative humidity and air velocity to the efficiency of energy recovery system.

2.4.1 Temperature and relative humidity profile

The temperature difference between the intake and supply air were calculated using Equation 1. The temperature difference was used to determine the energy transfer efficiency:

$$\Delta T = T_{h,in} - T_{h,sup} \tag{1}$$

where,

ΔT = temperature difference
$T_{h,in}$ = temperature of intake air in the hot air stream (°C)
$T_{h,sup}$ = temperature of supply air in the hot air stream (°C)

Relative humidity difference between the intake and supply air was calculated using Equation 2 and was used to determine the mass transfer efficiency:

$$\Delta \omega = \omega_{h,in} - \omega_{h,sup} \tag{2}$$

where,
$\Delta\omega$ = humidity ratio difference
$\omega_{h,in}$ = humidity ratio of intake air in the hot air stream (g kg^{-1})
$\omega_{h,sup}$ = humidity ratio of supply air in the hot air stream (g kg^{-1})

2.4.2 Efficiency

According to the American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE) standard, sensible efficiency is determined by the temperature difference between supply air and intake air in the hot air stream to the temperature difference between the intake air in the cold and hot air streams.[16] The sensible efficiency was calculated using Equation 3:

$$\epsilon_s = \frac{T_{h,sup} - T_{h,in}}{T_{c,in} - T_{h,in}} \tag{3}$$

where,
ϵ_s = sensible efficiency (%)
$T_{c,in}$ = temperature of intake air in cold air stream (°C)

Latent efficiency was determined using the moisture difference between the supply air and the intake air in the hot air stream and the moisture difference of the intake air in the cold and hot air streams. Latent efficiency was calculated using Equation 4:

$$\epsilon_L = \frac{\omega_{h,sup} - \omega_{h,in}}{\omega_{c,in} - \omega_{h,in}} \tag{4}$$

where,
ϵ_L = latent efficiency (%)
$\omega_{c,in}$ = humidity ratio of intake air in the cold air stream (g kg^{-1})

2.4.3 Recovered energy

Sensible recovered energy was calculated using Equation 5 without considering heat conduction and energy transfer through the heat exchanger case:

$$q_s = m_a C_{pa} \Delta T \tag{5}$$

where,
q_s = sensible recovered energy (kW)
m_a = mass flow rate of air (kg s^{-1})
C_{pa} = specific heat of air (kJ kg^{-1} K^{-1})

Latent recovered energy was determined by the change of the humidity content of the air by using Equation 6:

$$q_L = m_a h_{fg} \Delta\omega \tag{6}$$

where,
q_L = latent recovered energy (kW)
h_{fg} = enthalpy of evaporation (kJ kg^{-1})

2.5 Statistical Analysis

One-way analysis of variance (ANOVA) was applied using the software IBM SPSS Statistics version 20. It was used to determine if there are statistically significant differences between the means of two or more independent groups of air velocities. Assumptions of ANOVA were tested before running the statistical analysis using test of homogeneity of variance, Levene's test and test of normality.

3. RESULTS AND DISCUSSION

3.1 Temperature and Relative Humidity Profile

Figure 4 depicts the variation of temperature and humidity ratio differences with temperature intake. There are positive relationships between temperature intake as well as temperature and humidity ratio differences. The high-temperature intake (40°C) caused larger temperature and humidity ratio differences than the low-temperature intake (28°C). The temperature intake of 40°C at the air velocity of 1 m s^{-1} produced the highest temperature and humidity ratio differences of 12°C and 20 g kg^{-1}. As the temperature intake increased, the humidity ratio and temperature differences increased. Due to the increase in moisture transfer resistance, the humidity ratio increased with the high-temperature intake.[17]

Figure 4 shows that the largest temperature and humidity ratio difference reported at 12°C and 20 g kg^{-1} at 40°C at 1 m s^{-1}. The smallest temperature and humidity ratio differences were 2°C and 3 g kg^{-1} at 28°C at 3 m s^{-1}. The low air velocity of 1 m s^{-1} were more efficient than the high air velocity of 3 m s^{-1} due to the prolonged residence time.[17] Hence, heat and mass transfers are enhanced at low air velocities.

3.2 Efficiency

Figure 5 and Figure 6 show the variations of the average latent and sensible efficiencies with temperature intake. An uptrend of efficiency can be seen when temperature intake increased. Average sensible efficiency increased from 67%

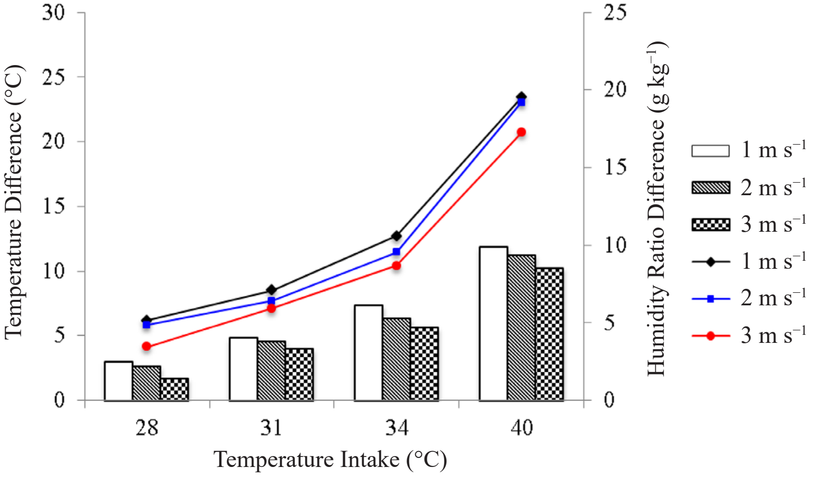

Figure 4: Temperature intake with temperature difference and humidity ratio difference.

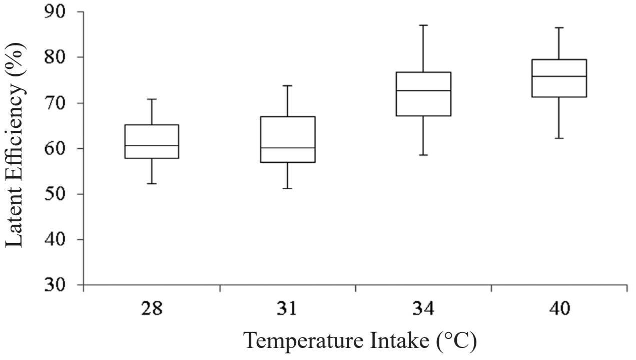

Figure 5: Variation of average latent efficiency with temperature intake.

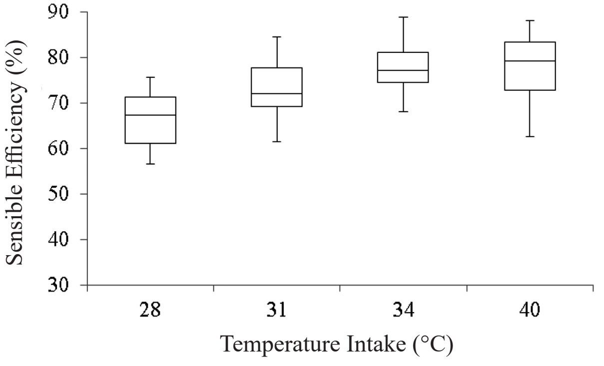

Figure 6: Variation of average sensible efficiency with temperature intake.

at 28°C to 78% at 40°C whereas average latent efficiency increased from 61% at 28°C to 75% at 40°C. Both average latent and sensible efficiencies increased with raised temperature intake. Due to the high humidity ratio, latent efficiencies increased with temperature intake.[20]

Average latent and sensible efficiencies with air velocity are presented in Figure 7 and Figure 8. Both latent and sensible efficiencies exhibited the same uptrend characteristics with temperature intake. Maximum average latent and sensible efficiencies were recorded at 74% and 78% of the temperature intake 40°C at the air velocity 1 m s^{-1}. Temperature intake of 28°C resulted in the lowest average latent and sensible efficiencies of 42% and 45%, respectively, at the air velocity 3 m s^{-1}. In comparison with Zafirah and Mardiana, they applied a 0.2 m by 0.2 m by 0.1 m fixed-plate heat exchanger to achieve the highest efficiency of 84.6% at

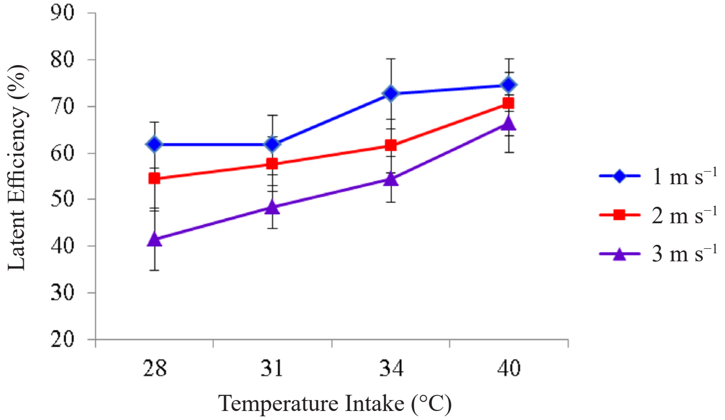

Figure 7: Average latent efficiency of energy recovery system with temperature intake.

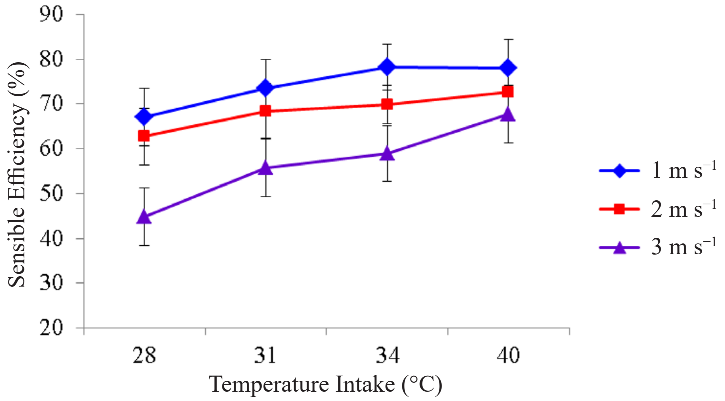

Figure 8: Average sensible efficiency of energy recovery system with temperature intake.

the temperature intake of 40°C and the air velocity of 1 m s^{-1}. They reported the lowest efficiency of 57.8% at the temperature intake of 31°C at the air velocity 3 m s^{-1}.[15] Mardiana-Idayu and Riffat noted the highest sensible and latent effectiveness of 66% and 58% at 1 m s^{-1} and lowest sensible and latent effectiveness of 48% and 26% at 3 m s^{-1} using a diamond-shaped fixed-plate enthalpy recovery core.[17] Hence, the results show that as air velocity increases, sensible and latent efficiencies diminish accordingly. As latent efficiency involves heat and mass recoveries, they are lower than sensible efficiency. However, both sensible and latent efficiencies drop when air velocity rises. This decrease is due to the small transfer units that occur at high air velocity.[21]

3.3 Recovered Energy

Figure 9 and Figure 10 display the average recovered energy with temperature intake. Increased air velocity results in greater recovered energy. It is found that the highest latent and sensible recovered energy were 3 kW and 17 kW at the air velocity 3 m s^{-1}. The lowest latent and sensible energies recovered were 0.3 kW and 6 kW at the air velocity 1 m s^{-1}. Recovered latent energy was less than recovered sensible energy with air velocity as a result of mass and heat transfers.

3.4 Statistical Analysis

One-way ANOVA is used to determine the statistically significant differences between the means of efficiencies caused by changing air velocities. Table 1 lists the one-way ANOVA results for air velocity at various temperature intakes. It can be seen that the p-values for the temperature intake were smaller than 0.05. Thus, air velocity significantly affects sensible and latent efficiencies with temperature intake.

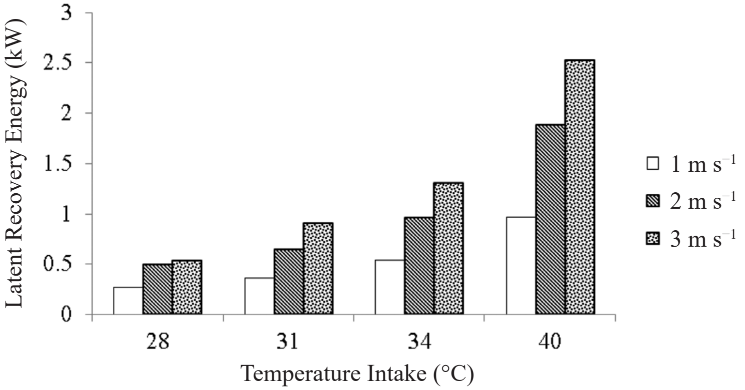

Figure 9: Average latent recovered energy with temperature intake.

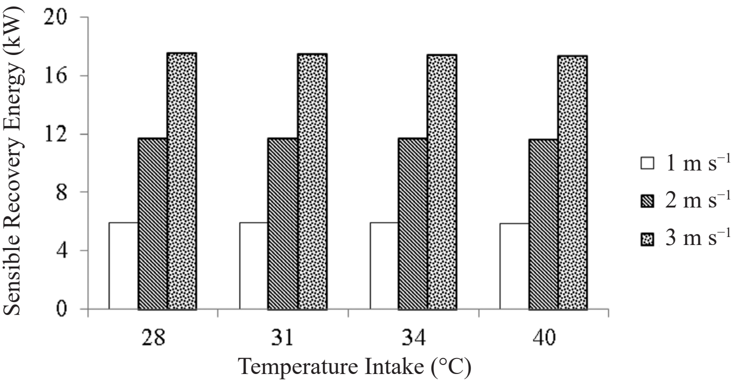

Figure 10: Average sensible recovered energy with temperature intake.

Table 1: Results from one-way ANOVA – air velocity with various temperature intakes.

	Source		SS	df	MS	F	P
28°C	Sensible	Between group	9625.44	2	4812.72	122.53	<0.05
		Within group	4124.18	105	39.28		
		Total	13749.62	107			
	Latent	Between group	7332.11	2	3666.06	94.12	<0.05
		Within group	4089.72	105	38.95		
		Total	11421.83	107			
31°C	Sensible	Between group	5569.70	2	2784.85	71.63	<0.05
		Within group	4082.46	105	38.88		
		Total	9652.16	107			
	Latent	Between group	3222.61	2	1611.30	49.71	<0.05
		Within group	3403.25	105	32.41		
		Total	6625.86	107			
34°C	Sensible	Between group	6451.15	2	3225.57	116.88	<0.05
		Within group	2897.84	105	27.60		
		Total	9348.99	107			
	Latent	Between group	5966.48	2	2983.22	78.63	<0.05
		Within group	3983.75	105	37.94		
		Total	9950.20	107			
40°C	Sensible	Between group	1951.87	2	975.93	23.99	<0.05
		Within group	4272.38	105	40.69		
		Total	6224.25	107			
	Latent	Between group	1422.27	2	711.14	18.62	<0.05
		Within group	4009.97	105	38.19		
		Total	5432.24	107			

Notes: df = degrees of freedom, SS = sum of squares, MS = mean of squares, F = statistical test, P = statistical value

4. CONCLUSION

The performance of the energy recovery system was investigated at different operating parameters of temperature, relative humidity and air velocity. It was found that the average sensible and latent efficiencies reduced from 78% to 45% and 74% to 42%, respectively, with increased air velocity. The highest sensible energy recovered was 18 kW at the temperature intake of 40°C. One-way ANOVA revealed that the air velocity significantly affects both sensible and latent efficiencies at different temperature intakes. To extend the application of the energy recovery system to buildings, a thorough study of fresh air flow rate, life span and thermal performance should be considered.

5. ACKNOWLEDGEMENTS

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