Building Energy Simulation of A Run-Around Membrane Energy Exchanger (RAMEE)

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ABSTRACT

The main objective of this thesis is to investigate the energetic, economic and environmental impact of utilizing a novel Run-Around Membrane Energy Exchanger (RAMEE) in building HVAC systems. The RAMEE is an energy recovery ventilator that transfers heat and moisture between the exhaust air and the fresh outdoor ventilation air to reduce the energy required to condition the ventilation air. The RAMEE consists of two exchangers made of water vapor permeable membranes coupled with an aqueous salt solution.

In order to examine the energy savings with the RAMEE, two different buildings (an office building and a health-care facility) were simulated using TRNSYS computer program in four different climatic conditions, i.e., cold-dry, cool-humid, hot-humid and hot-dry represented by Saskatoon, Chicago, Miami and Phoenix, respectively. It was found that the RAMEE significantly reduces the heating energy consumption in cold climates (Saskatoon and Chicago), especially in the hospital where the required ventilation rate is much higher than in the office building. On the other hand, the results showed that the RAMEE must be carefully controlled in summer to minimize the cooling energy consumption.

The application of the RAMEE in an office building reduces the annual heating energy by 30% to 40% in cold climates (Saskatoon and Chicago) and the annual cooling energy by 8% to 15% in hot climates (Miami and Phoenix). It also reduces the size of heating equipment by 25% in cold climates, and the size of cooling equipment by 5% to 10% in hot climates. The payback period of the RAMEE depends on the air pressure drop across the exchangers. For a practical pressure drop of 2 cm of water across each exchanger, the payback of the RAMEE is 2 years in cold climates and 4 to 5 years in hot climates. The total annual energy saved with the RAMEE (including heating, cooling and fan energy) is found to be 30%, 28%, 5% and 10% in Saskatoon, Chicago, Miami and Phoenix, respectively.

In the hospital, the RAMEE reduces the annual heating energy by 58% to 66% in cold climates, and the annual cooling energy by 10% to 18% in hot climates. When a RAMEE is used, the heating system can be downsized by 45% in cold climates and the cooling system can be downsized by 25% in hot climates. For a practical range of air pressure drop across the exchangers, the payback of the RAMEE is immediate in cold climates and 1 to 3 years in hot climates. The payback period in the hospital is, on average, 2 years faster than in the office building). The total annual energy saved with RAMEE is found to be 48%, 45%, 8% and 17% in Saskatoon, Chicago, Miami and Phoenix, respectively. The emission of greenhouse gases (in terms of CO_2-equivalent) can be reduced by 25% in cold climates and 11% in hot climates due to the lower energy use when employing a RAMEE.

ACKNOWLEDGMENTS

I owe my deepest gratitude to my supervisor Carey Simonson who stood by my side for more than 2 years of completion of my master studies. Your perpetual energy, encouragement and support motivated me to gain an understanding of problem solving not only for my thesis subject but also as a lifelong skill. I would also like to thank my co-supervisor Robert Besant whose knowledge and patience made the research life rewarding and smooth for me.

This research would not have been possible without the financial support from the Natural Science and Engineering Research Council of Canada (NSERC) and Venmar CES Inc, Saskatoon. Also, I would like to thank American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE) and George Ira Hanson scholarship in energy research for their valuable award on my research project. I appreciate your support.

Dedication

I dedicate this thesis to my family. Thank you for your unwavering love and support during my life.

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NOMENCLATURE

Acronyms

ACH	Air Change per Hour
AHRI	Air-Conditioning, Heating and Refrigeration Institute
ANN	Artificial Neural Network
ASHRAE	American Society of Heating, Refrigerating and Air-conditioning Engineers
CAV	Constant Air Volume
CBECS	Commercial Buildings Energy Consumption Survey
EPA	Environment Protection Agency
ERV	Energy Recovery Ventilator;
HVAC	Heating, Ventilation and Air-Conditioning;
IAQ	Indoor Air Quality;
LAMEE	Liquid-to-Air Membrane Energy Exchanger
LCEA	Life-Cycle Environmental Assessment
LCC	Life-Cycle Cost
PNL	Pacific Northwest Lab
RAMEE	Run-Around Membrane Energy Exchanger
RH	Relative Humidity
TESS	Thermal Energy System Specialists
TMY	Typical Meteorological Year
US	The United States
VAV	Variable Air Volume

Symbols

A	Membrane surface area in the exchanger (m^2)
C_p	Specific heat capacity of air (J/kg.K)
Cr*	Ratio of salt solution heat capacity to that of the air (dimensionless group)
H*	Operating condition factor (dimensionless group)
h	Enthalpy of air (J/kg dry air)
h_{fg}	Enthalpy of phase change (J/kg)
$\dot{m}_a$	Mass flow rate (kg/s)

NTU	Number of transfer units (dimensionless group)
NTU_m	Number of mass transfer units (dimensionless group)
OC	Operational costs ($US)
PBP	Payback period (yr)
Q	Energy transfer via the RAMEE system (W)
$Q_{sens,rec}$	Sensible heat recovery (J/s)
$Q_{lat,rec}$	Latent energy recovery (J/s)
R	Bypass fraction (dimensionless)
RH	Relative Humidity (%)
T	Temperature (°C) (°F)
U	Overall convective heat transfer coefficient (W/m^2 K)
U'	Overall convective mass transfer coefficient (kg/m^2 s)
w	Humidity ratio (kg water/kg dry air)
ε	Effectiveness (%)

<u>Subscripts</u>

air	Refers to the air properties
ave	Average
exh,in	The exhaust air at the inlet of the energy exchanger, i.e., indoor air
oa	Outdoor ventilation air
opt	Optimal
out	Refers to outdoor condition (temperature, humidity ratio or enthalpy)
in	Refers to indoor condition (temperature, humidity ratio or enthalpy)
l	Latent
s	Sensible
sol	Refers to the solution properties
sup,in	The supply air at the inlet of the energy exchanger, i.e., outdoor air
sup,out	The supply air at the outlet of the energy exchanger
sup	The air supplied to the conditioned space
t	Total

CHAPTER 1

INTRODUCTION

1.1. An Overview on Ventilation and Energy Recovery in Buildings

In order to maintain an acceptable Indoor Air Quality (IAQ) that affects occupants' health and productivity (Fang et al. 2000; Kosonen and Tan 2004; Seppänen and Fisk 2005), HVAC-related organizations have set standards that specify the minimum required ventilation rate depending on the type of buildings and occupancy (e.g., ASHRAE 2008; ASHRAE 2010). Higher ventilation rates improve the IAQ by diluting pollutants such as airborne particles and volatile organic compounds. On the other hand, studies have shown that higher ventilation rates increase the building energy consumption in a majority of cases, especially during the heating season (McDowell et al. 2003; Brandemuehl and Braun 1999; Fauchoux 2006; Orme 2001; Rey and Velasco 2000). For instance, McDowell et al. (2003) showed that increasing the ventilation rate of a building in Washington D.C. from 0 to 10 l/s.person (corresponding to 0.37 Air Change per Hour, ACH) increases the annual energy consumption by 14%.

Energy Recovery Ventilators (ERVs), which transfer energy between exhaust and supply airstreams, have been used to reduce the energy consumption associated with conditioning the ventilation air. In general, ERVs can be divided into two groups: heat recovery systems which transfer only sensible heat, and heat and moisture recovery systems which transfer both sensible and latent energy. Some research has been conducted to study the applicability and benefits of heat recovery systems (Zhong and Kang 2009; Dhital et al. 1995; Manz et al. 2000) and heat and moisture recovery systems (Zhang and Niu 2001; Zhou et al. 2007; Fauchoux et l. 2007; Liu et al. 2009). These studies have shown that ERVs decrease the annual heating energy consumption significantly; however, they require a proper control during the summer (Mumma 2001; Zhang and Niu 2001). Liu et al. (2009) studied the applicability and energy savings with enthalpy exchangers employed in five Chinese cities. Their study was limited to heating season only, and the results showed that the heating energy could be reduced by 20% when an ERV with total effectiveness of 75% was employed. Zhou et al. (2007) simulated an ERV system in two locations with different climatic conditions in China using a dynamic building simulation model (EnergyPlus 2007). They reported that an ERV reduces the energy consumption during the winter; however, in summer, the operation of ERV in a cold climate (Beijing) was uneconomical when the cooling set-point was above 24°C.

All the available ERVs that are mentioned above require adjacent installation of the supply and exhaust ducts which usually imposes higher ducting costs. In addition, contaminant carryover in rotary wheels and cross-flow leakage of air through seals are concerns in some types of buildings such as health care facilities and laboratories. The extra ducting cost and the contaminant transfer could be avoided if the exhaust and supply air ducts were separated.A Run-Around Membrane Energy Exchanger (RAMEE) which is capable to transfer both heat and moisture between remote supply and exhaust ducts could minimize these problems and is the focus of this thesis. An overview of the RAMEE is presented in the next section.

1.2. Run-Around Membrane Energy Exchanger (RAMEE)

A Run-Around Membrane Energy Exchanger (RAMEE) was proposed as a novel energy recovery system. Figure 1.1 shows a schematic of the RAMEE. The RAMEE consists of two separate exchangers that are located into supply and exhaust air ducts of the building. Semi-permeable membranes used in each exchanger allow the transfer of both heat and water vapor simultaneously, so both sensible and latent energy recovery is achieved. In addition, compared to other enthalpy recovery systems such as energy wheels, the application of the RAMEE system is more feasible in retrofit applications when the supply and exhaust ducts are not adjacent.

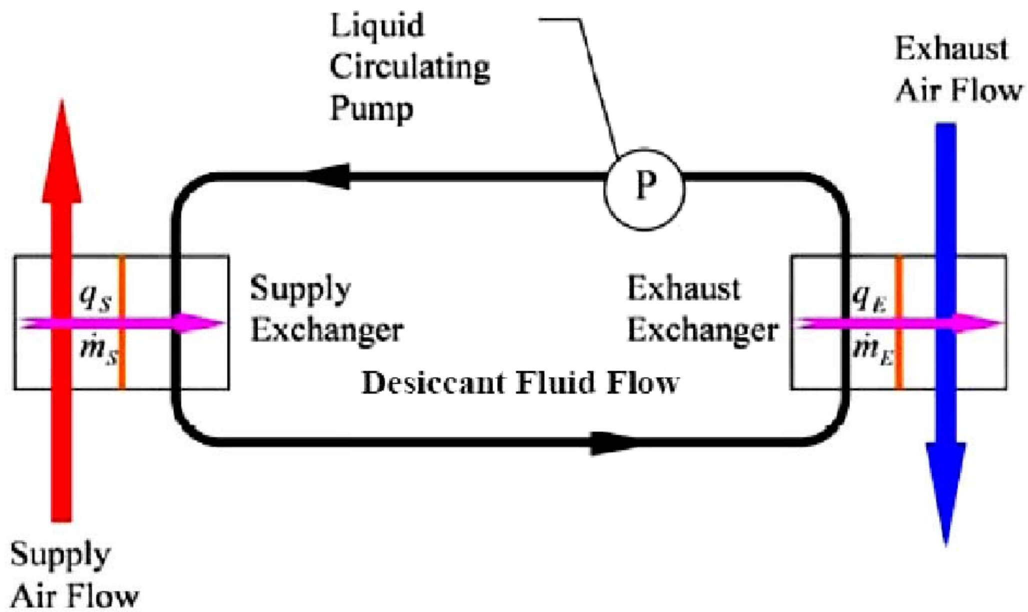


Figure 1.1 Schematic diagram of a RAMEE (Vali 2009)

As is shown in Figure 1.1, the RAMEE system consists of two separate exchangers and a desiccant fluid flow as the coupling liquid is pumped between two exchangers. The RAMEE system preconditions the supply air as it passes through the supply exchanger. During the summer when the outdoor air is warm and humid, the desiccant salt solution gains heat and moisture from the supply air stream in the supply exchanger and releases the heat and moisture to the exhaust air stream in the exhaust exchanger. During winter operation, the RAMEE system transfers both heat and moisture from the warm-humid exhaust air to cold and dry supply air. One of the main advantages of the RAMEE is the capability of working under a wide range of solution flow rates which results in different sensible and latent effectiveness values at any specific outdoor condition. For some particular outdoor conditions, either heat or moisture transfer might not be desired; e.g., in cool-humid summer days when the outdoor temperature is lower than the indoor, but the humidity ratio is higher, the maximum moisture transfer which reduces dehumidification load is preferred while the sensible heat transfer should be kept at the minimum possible rate. For such conditions, the maximum net energy transfer can be obtained by adjusting the appropriate solution flow rate.

Previous graduate students have (a) developed numerical models of the RAMEE (Erb et al. 2009; Seyed-Ahmadi et al. 2009a and 2009b; Vali et al. 2009; Hemingson 2010), (b) built and tested experimental prototypes (Mahmud et al. 2010; Erb et al. 2009, Beriault 2010), (c) studied the crystalization risk of the salt solution (Afshin et al. 2010), and (d) trained artificial neural networks to predict the system performance at different conditions (Akbari 2010). The following section describes the contribution of this thesis in the completion of the RAMEE project

1.3. Thesis Objectives and Overview

As mentioned previously, much research has been done on the RAMEE; however, these studies have not addressed the application of RAMEEs in buildings. In addition, no universal control strategy of ERVs is found in the literature. In this thesis, the results on the study of optimal control strategy of ERVs are presented. Thereafter, the energy savings and life-cycle-cost analysis of the RAMEE when it is operating in different buildings and climates is quantified using TRNSYS building energy simulation program. An office building and a hospital are the two selected buildings for this study as office buildings account for the largest fraction of US commercial buildings (EIA 2003) and hospitals are the second most energy-intensive US commercial buildings (EIA 2003). To examine the RAMEE's performance in heating and cooling energy, these buildings are simulated in four different locations representing different climates, i.e., Saskatoon (cold-dry), Chicago (cool-humid), Miami (hot-humid) and Phoenix (hot-dry). The main objectives of this M.Sc research are to determine:

- An optimal control strategy for the RAMEE.
- The life-cycle cost (LCC) and energy savings of the RAMEE in different buildings and climates.

These objectives have been met and the results are described in four research manuscripts as listed below: manuscripts 1 and 2 present the optimal control of ERVs, and manuscripts 3 and 4 present two case studies for the application of a RAMEE in buildings (an office building and a health-care facility).

1- **Rasouli, M**., C.J. Simonson and R.W. Besant. 2010. Applicability and optimum control strategy of energy recovery ventilators in different climatic conditions, *Energy and Buildings* 42(9): 1376-1385.

2- **Rasouli, M**., C.J. Simonson and R.W. Besant. 2010. Optimization of energy recovery ventilators and their impact on energy and comfort, submitted to *Journal of Energy* (July 7).

3- **Rasouli, M**., S. Akbari, H. Hemingson, R.W. Besant and C.J. Simonson. 2010. Application of a run-around membrane energy exchanger in an office building HVAC system, accepted for publication in *ASHRAE Transactions* (December 2010)

4- **Rasouli, M**., S. Akbari, C.J. Simonson and R.W. Besant. 2010. Analysis of a health-care facility HVAC system equipped with a run-around membrane energy exchanger, submitted to *Energy and Buildings* (December 2010)

This thesis is organized such that each of the major chapters (chapters 2, 3 and 4) include one of the listed manuscripts. Due to the similarities between manuscript #1 and #2, and the fact that manuscript #2 describes the control strategy of manuscript #1 and compares it to other control alternatives in the literature, only manuscript #2 is included in the main body of the thesis. Manuscript #1 is attached as an appendix (Appendix B). Manuscripts 3 and 4 are two case studies of a RAMEE that is controlled based on an optimal strategy presented in manuscript 2. Each chapter starts with a brief overview on the focus of the chapter, the connection of the chapter with the following and/or previous chapters and the contribution of each author in the completion of the research work. In chapter 5, the main conclusions of the thesis are highlighted and some recommendations for future work are made.

Appendix C includes the copyright permissions from the publisher of manuscript #1 and the co-authors who contributed to manuscripts # 2, 3 and 4.

CHAPTER 2

CONTROL OF ERVS

2.1. Overview of Chapter 2

Previous research has shown that the operation of an ERV in summer should be controlled, otherwise it may increase the cooling energy consumption. However, no universal control strategy was found in the literature. In this chapter, which includes manuscript # 2 (optimal control of energy recovery ventilators and their impact on energy and comfort), a general strategy to control the operation of ERVs that applies to all ERVs whether they transfer heat only or both heat and moisture is presented. In order to study the optimal control, equations relating the cooling energy consumption to ERV's operation are derived. Using the MATLAB (MATLAB 2006) optimization tool, these equations are minimized and the ERV's operating condition which results in minimum energy use is obtained. The proposed control strategy is compared to other controls in the literature (including manuscript #1) using TRNSYS (Klein 2000) modeling of an ERV operating under different controls in a building. The control strategy developed in this chapter will be used in Chapters 3 and 4 to control the RAMEE operation in an office building and a health-care facility.

Manuscript #2: Optimal Control of Energy Recovery Ventilators and Their Impact on Energy and Comfort

M. Rasouli, C.J. Simonson and R.W. Besant

2.2. Abstract

Concern over providing thermal comfort for occupants while minimizing associated energy consumption has raised attention towards optimizing HVAC equipment. Energy Recovery Ventilators (ERVs) transfer energy between conditioned exhaust air and outdoor ventilation air to reduce the energy demand of HVAC system. In this paper, based on minimization of HVAC system energy consumption, an optimal strategy to control the operation of an ERV is concluded and compared to other control alternatives in the literature. The optimum control depends on the ERV's latent to sensible effectiveness ratio and requires part-load operation of ERV for hot-dry outdoor conditions and full-load operation for particular cool-humid outdoor conditions at which the ERV can decrease the enthalpy of outdoor ventilation air. Potential energy savings with an optimized system are investigated by TRNSYS simulations of an office building in four North American cities as representatives of major climates. The results show that an ERV can lead to significant annual heating energy saving (about 35% in cold climate) and annual cooling energy saving (up to 20%) provided the ERV has the capability to transfer moisture and is properly controlled. Also, occupants' thermal comfort can be improved during the winter since employing ERVs humidifies cold-dry outdoor air.

2.3. Introduction

ERVs reduce the energy consumption associated with conditioning ventilation air by transferring heat (and moisture) between conditioned exhaust air and outdoor ventilation air. They can be divided into two general groups: i.e., heat recovery systems which transfer only sensible heat, and heat and moisture recovery systems which transfer both sensible and latent energy. Some research has been conducted to study the applicability and benefits of heat recovery systems (Zhong and Kang 2009; Manz et al. 2000) and heat and moisture recovery systems (Mumma 2001; Zhang and Niu 2001; Zhou et al. 2007; Fauchoux et al. 2007; Liu et al. 2010; Rasouli et al. 2010a) showing that ERVs can decrease the annual heating energy consumption significantly. ERVs can also reduce the cooling energy consumption, but they require to be properly controlled. Control of ERVs during cooling season is important, because cooling energy consumption may actually increase when and un-controlled ERV is used in an HVAC system (Mumma 2001; Zhang and Niu 2001; Rasouli et al. 2010a).

Zhang and Niu (2001) studied the applicability of heat and moisture recovery systems in Hong Kong and showed that an ERV controlled by a temperature-based control during the summer may significantly reduce the annual cooling energy consumption. Mumma (2001) used a control strategy for summer operation of enthalpy wheels employed in dedicated outdoor air systems where the ERV was stopped when the outdoor enthalpy was lower than the indoor air enthalpy while the outdoor humidity was higher than the humidity ratio of the air supplied to the conditioned space. Rasouli et al. (2010a) studied the applicability of ERVs in different climates in North America and proposed a control strategy which was dependent on the ERV's sensible to latent effectiveness ratio. Their control strategy allowed the ERV to operate for outdoor temperatures greater than the indoor, and a specific portion of cool-humid outdoor condition at which net energy is transferred from supply air to the exhaust stream. They also showed that a heat and moisture recovery system with 75% sensible and latent effectiveness may save 30% of the annual heating energy in cold climate and 20% of the annual cooling energy in hot climate when operating under their proposed control strategy.

The purpose of this paper is to conclude an optimal control method for ERVs that will minimize energy consumption of HVAC system. In this approach, the equations describing the actual cooling energy consumption in a practical HVAC system are developed, and the ERV operation, i.e. off, partly-on (part-load) or fully-on (full-load), that minimizes the cooling energy at any given outdoor condition is specified using MATLAB optimization tool. Such optimal strategy is compared to controls proposed in the literature, and the impact on energy and comfort is investigated for ERVs operating in a practical range of sensible and latent effectiveness in different climates.

2.4. HVAC System and Air-Conditioning Process

Figure 2.1 schematically shows a typical HVAC system consisting of a cooling/heating unit equipped with an energy recovery system.

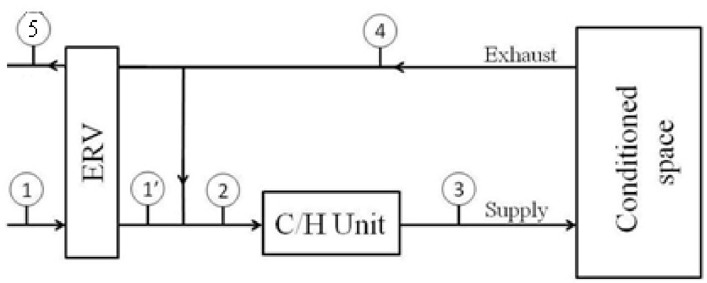


Figure 2.1 A typical HVAC system equipped with an ERV

During the operation of HVAC system when fresh outdoor ventilation air is required, the ERV transfers heat (and moisture) between exhaust air (state 4, Figure 2.1) and outdoor ventilation air (state 1). The ventilation air leaving the ERV (state 1') is mixed with the return air adiabatically. During the winter, the mixture (at state 2) is heated in the heating unit and then supplied to the space at space 3. But, during the summer, the mixture (state 2) enters the cooling unit and completes a sensible cooling process if it is dry enough to provide a satisfactory indoor humidity. But, dehumidification may be required if the humidity at state (2) is too high.

Depending on the type of heating or cooling process needed to condition the space, the psychrometric chart can be divided into 3 main regions as shown in Figure 2.2; i.e. low temperatures that require a heating system, high temperature/humidity condition which needs mechanical cooling, and low temperatures and humidities where economized cooling (i.e., 100% outdoor air system and possibly some mechanical cooling) is available. In this figure, state (4) represents summer indoor comfort condition and state (3) is the condition of air supplied to the space during the cooling season.

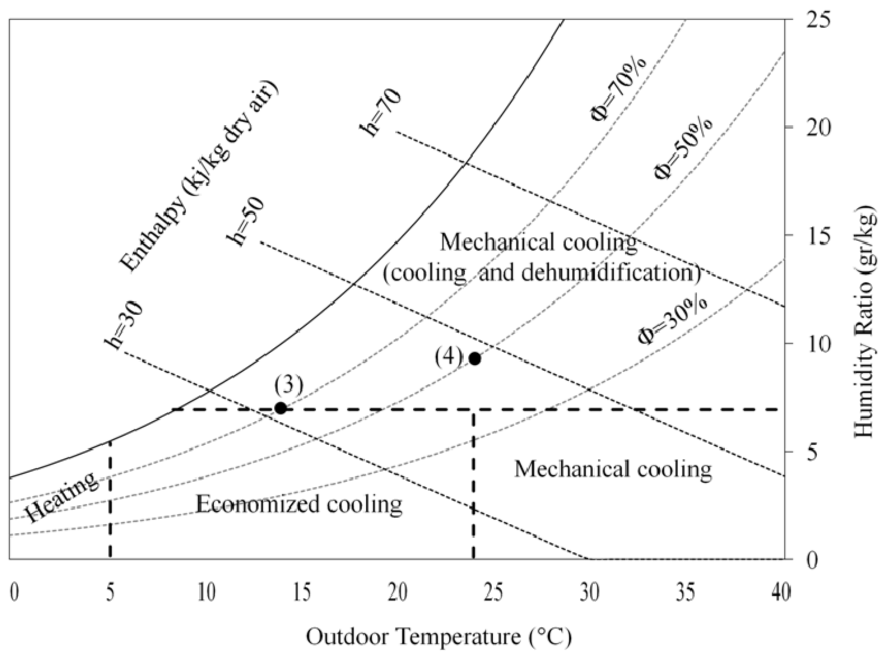


Figure 2.2 Air conditioning process required in each psychrometric region

For outdoor humidities higher than the supply humidity ratio (region above the dashed line crossing state 3 in Figure 2.2), dehumidification is certainly necessary to keep the indoor humidity at the desired level. The air-conditioning process in such conditions, as is schematically shown on the psychrometric chart in Figure 2.3(a), requires sensible cooling, dehumidification and reheat. But, for hot-dry condition (outdoor conditions with higher temperature than the indoor and lower humidity than the supply, as shown in Figure 2.2), ERV operation cools the outdoor air, but may impose an unnecessary dehumidification in the cooling system. This is schematically shown in Figure 2.3b.

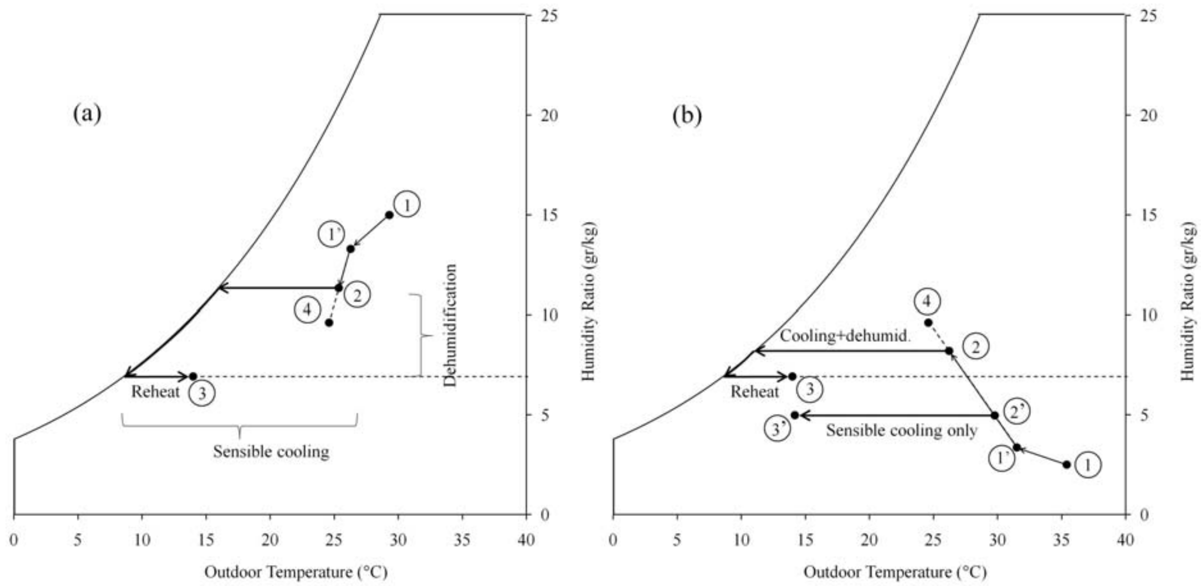


Figure 2.3 Air-conditioning process for different outdoor conditions during the cooling season; (a) humid outdoor condition and (b) hot-dry outdoor condition

2.5.Present Control Strategies

Several studies have indicated that the utilization of ERVs during the winter can reduce the annual heating energy significantly by heating (and humidifying) the cold (and dry) outdoor air (e.g., Zhong and Kang 2009; Zhou et al. 2007; Liu et al. 2010; Rasouli et al. 2010a). Thus, the ERV should be operated at the maximum capacity during the heating season. However, summer operation of the ERV for specific outdoor conditions is found to be unbeneficial (causing higher cooling energy consumptions), and therefore the ERV needs to be controlled during such conditions (Mumma 2001; Zhang and Niu 2001; Rasouli et al. 2010a). In this section, an overview of three main present control strategies is presented, and the outdoor conditions at which each strategy lets the ERV operate during the summer is hatched in Figure 2.4. In this figure, states (3) and (4) present the condition of air supplied to the space and the indoor condition, respectively.

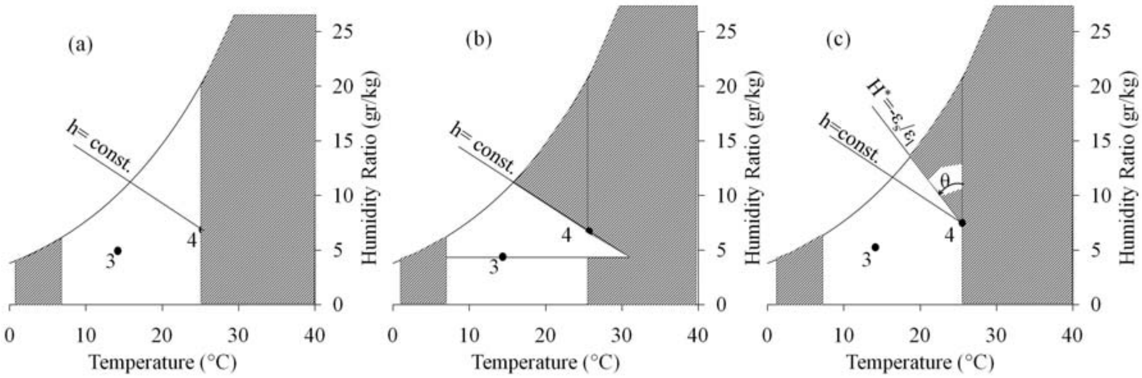


Figure 2.4 The outdoor condition in which each control strategy recommends the ERV operation, (a) temperature-based control, (b) Mumma (2001), and (c) Rasouli et al. (2010a)

Temperature-based control, as shown in Figure 2.4(a), is the most common way to control ERV summer operation. In this control strategy, the ERV is off (i.e., transfers no heat or moisture between supply and exhaust air streams) for outdoor temperatures lower than the indoor when cooling is required. Mumma (2001) suggested a strategy to control enthalpy wheels which is schematically shown in Figure 2.4(b). This control strategy allows the ERV to operate when (i) the outdoor enthalpy is greater than the indoor or (ii) the humidity ratio of outdoor air is lower than that of the supply air and outdoor temperature is greater than the indoor temperature. Rasouli et al. (2010a) proposed a control strategy which predicts higher cooling energy savings compared to temperature-based control. They showed that an ERV has the maximum saving potential when operating for outdoor temperatures higher than the indoor and a specific portion of cool-humid condition which is shown in Figure 2.4(c).

Rasouli et al. (2010a) showed that ERV control can be simplified by defining an operating condition factor, which represents the ratio of latent to sensible energy potentials of inlet airstreams (H^* defined as Equation (2-1) (Simonson and Besant 1999)), the ERV should be operated when either outdoor temperature is greater than the indoor temperature, or cool-humid outdoor condition that satisfy the condition specified by Equation (2-2).

$$H^* = \frac{h_{fg}\Delta w}{C_p \Delta T} \cong 2500 \frac{w_{\sup,in} - w_{exh,in}}{T_{\sup,in} - T_{exh,in}} \tag{2-1}$$

$$H^* \le \frac{-\varepsilon_s}{\varepsilon_l} \tag{2-2}$$

Angle θ can be defined as:

$$\theta = \tan^{-1}\left(\frac{\varepsilon_l}{\varepsilon_s}\right) \tag{2-3}$$

For example, θ=0 is applied for sensible-only heat exchangers (ε_l=0) and the ERV should be operated only for outdoor temperatures higher than the indoor. As the latent to sensible effectiveness ratio increases, the hatched region covers a larger area in cool-humid condition. For a specific case of ε_s=ε_l, H^*=-1 which closely follows on a line of constant enthalpy that crosses the summer indoor condition has to be applied as the boundary of the hatched region. For this case, the ERV operates when either the enthalpy or the temperature of outdoor air is greater than that of the indoor air.

The influence of moisture transfer is not considered in the definition of temperature-based strategy, and the decision regarding operation of ERV is made considering the temperature difference as the only mechanism of energy transfer. Mumma's control strategy (Mumma 2001) may be limited to specific types of heat and moisture recovery systems (e.g., energy wheels operated in dedicated outdoor air systems) since it does not consider the different capabilities that different ERVs might have in transferring heat versus moisture. On the other hands, Rasouli's control strategy (Rasouli et al. 2010a) is based on ideal energy consumptions (instead of practical energy demand), and it does not consider the influence that condition of the air supplied to the space might have on ERV control. In the next section, equations describing practical cooling energy consumption of an actual HVAC system are developed and the optimal ERV operation (off, full-load or part-load) that results in minimum energy demand is concluded.

2.6.New Optimization Method

2.6.1. Air-Conditioning Process for Humid Outdoor Condition

As mentioned previously (section 2), for outdoor conditions with humidity ratios higher than the supply humidity ratio, the cooling process requires sensible cooling, dehumidification and reheat (as shown in Figure 2.3(a)). This process is completed at the cooling unit and the required cooling power for such air-conditioning process (assuming no cost for the energy consumed during the reheat process based on the requirement in ASHRAE standard 90-2004 (ASHRAE 2004c)) can be described as Equation (2-4):

$$q_{cooling} = \dot{m}_{\sup}\left[C_p(T_2 - T_{dew@w3}) + h_{fg}(w_2 - w_3)\right] \tag{2-4}$$

Where,

$$T_2 = \frac{\dot{m}_{oa}T_{1'} + (\dot{m}_{\sup} - \dot{m}_{oa})T_4}{\dot{m}_{\sup}} \tag{2-5}$$

$$w_2 = \frac{\dot{m}_{oa}w_{1'} + (\dot{m}_{\sup} - \dot{m}_{oa})w_4}{\dot{m}_{\sup}} \tag{2-6}$$

$$T_{1'} = T_1 + \varepsilon_s(T_4 - T_1) \tag{2-7}$$

$$w_{1'} = w_1 + \varepsilon_l(w_4 - w_1) \tag{2-8}$$

For given outdoor, indoor and supply conditions, i.e. states 1, 4 and 3 respectively, the coil cooling load given in Equation (2-4) is a function of the ventilation air and supply air mass flow rates and the sensible and latent effectivenesses of the ERV. Assuming a constant ventilation flow rate (that satisfies ASHRAE standard 62 (ASHRAE 2004a)) and knowing that the supply flow rate is already determined considering the building cooling load (not a variable in Equations (2-5) and (2-6)), the coil cooling load is a function of ERV sensible and latent effectiveness.

2.6.2. Air-Conditioning Process for Hot-Dry Outdoor Condition

For hot-dry outdoor conditions (lower humidity than the supply and higher temperature than the indoor) the need for dehumidification depends on the condition of the mixture of outdoor air and return air (state 2, Figure 2.3b). When the mixture is dry enough, it can complete a sensible-only cooling process and be supplied to the space with no need for dehumidification (process 1→1'→2'→3' in Figure 2.3b). On the other hand, if the humidity of the mixture is higher than the supply humidity, cooling, dehumidification and reheat process will be required (process 1→1'→2→3 in Figure 2.3b). Therefore, the coil cooling load for hot-dry outdoor condition can be specified as Equation (2-9).

$$q_{cooling} = \dot{m}_{\sup}\left\{C_p(T_2 - T_{\sup}) + A\left[C_p(T_{\sup} - T_{dew@w3}) + h_{fg}(w_2 - w_3)\right]\right\} \tag{2-9}$$

Where,

$$A = \begin{cases} 0 & \text{for: } w_2 < w_3 \\ 1 & \text{for: } w_2 \geq w_3 \end{cases} \tag{2-10}$$

As defined in Equation (2-6), w_2 is a function of ventilation and supply air flow rate and the ERV sensible and latent effectiveness. Depending on the humidity ratio at state (2), for mixtures with humidities lower than the supply ($0 < w_2 < w_3$), "A" returns a zero value which gives a sensible-only cooling load in Equation (2-9), where it becomes "1" for $w_2 \geq w_3$

2.6.3. Part-Load Operating Condition of ERV

An economizer could be employed for outdoor conditions with lower humidity and temperature than the indoor (economized cooling condition shown in Figure 2.2). For such conditions, mixing the outdoor air with return air heats and humidifies the cool-dry outdoor air. This is not beneficial and in such condition, the economizer introduces 100% outdoor air to meet a portion (or all) of the building cooling load. But, for lower temperatures when the outdoor temperature falls below a certain limit (i.e., supply temperature) 100% outdoor air would overcool the building, and the outdoor air has to be mixed with return air to keep the indoor at desired temperature. Mumma (2001) showed that for outdoor temperatures lower than the supply temperature, the ratio of outdoor air to supply air flow rate is inversely proportional to the temperature difference between the outdoor and the indoor air. For such condition, heating/cooling system and the ERV are off and the energy balance can be expressed as Equation (2-11):

$$\dot{m}_{oa} C_p (T_4 - T_1) = \dot{m}_{sup} C_p (T_4 - T_{sup}) \tag{2-11}$$

And this can be re-arranged as Equation (2-12):

$$\frac{\dot{m}_{oa}}{\dot{m}_{sup}} = \frac{T_4 - T_{sup}}{T_4 - T_1} \tag{2-12}$$

This relationship is presented in Figure 2.5 (Ke and Mumma 1997).

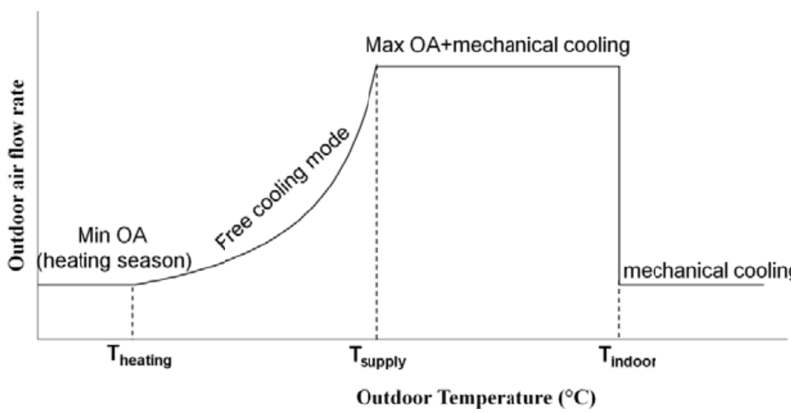


Figure 2.5 Outdoor air flow rate during economized cooling mode

So, the line connecting T_{sup} and T_{min} is a hyperbolic curve (Ke and Mumma 1997) and not a straight line as it is usually plotted in the literature (e.g., Janu et al. 1995). M_{oa} is set to the minimum possible (ventilation requirement) when the outdoor temperature falls below the critical temperature that the heating system starts operating.

During the economized cooling mode (shown in Figure 2.2), the ERV is off operation and the economizer introduces 100% outdoor air to minimize the HVAC load, as described. Instead of adjusting the outdoor air to return air ratio by an economizer, an ERV can be operated in a part-load condition to meet the desired supply temperature. This part-load operation brings the outdoor air to desired supply temperature (Simonson et al. 2000a). Adjusting the wheel speed (in energy wheels), the flow rate of one of the fluid streams or by-passing a fraction of the ventilation air are some strategies which can be applied for part-load operation of an ERV. It should be noted that, ERV full operation may heat the outdoor air to a temperature greater than the desired supply temperature which causes a demand for cooling.

Assuming a HVAC system with no recirculation (100% outdoor air) and net zero energy consumption in the cooling/heating unit during the part-load operation, the energy balance for the control volume shown in Figure 2.6 can be written as:

$$q_c = \dot{m}_{\sup} C_p (T_5 - T_1) \tag{2-13}$$

Where, q_c is the building cooling load removed by the supply air.

$$q_c = \dot{m}_{\sup} C_p (T_4 - T_3) \tag{2-14}$$

And, T_5 is the temperature of the exhaust air leaving the ERV.

$$T_5 = T_4 + \varepsilon_s (T_1 - T_4) \tag{2-15}$$

The substitution of Equations (2-14) and (2-15) into Equation (2-13) gives:

$$\dot{m}_{\sup} C_p (T_4 - T_3) = \dot{m}_{\sup} C_p (T_4 + \varepsilon_s (T_1 - T_4) - T_1) \tag{2-16}$$

And, assuming T_3 as $T_{\sup}$, T_4 as T_{in} and T_1 as T_{out}, Equation (2-16) can be simplified to give the ERV effectiveness during the part-load operation:

$$\varepsilon_s = \frac{T_{\sup} - T_{out}}{T_{in} - T_{out}} \tag{2-17}$$

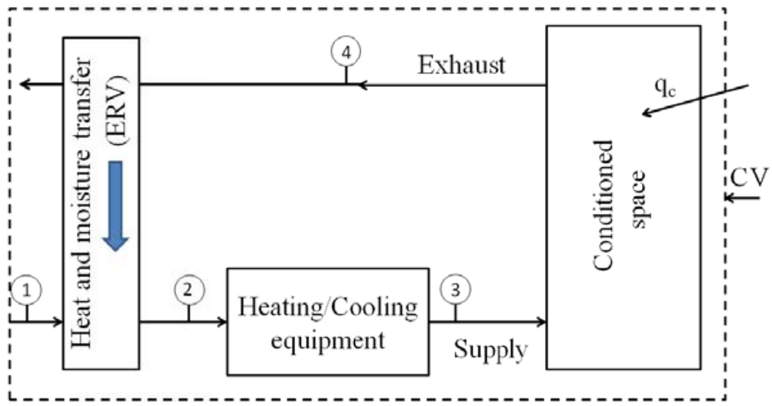


Figure 2.6 Schematic view of a 100% outdoor air HVAC system

Assuming a constant indoor and supply temperature during the cooling season, Equation (2-17) gives the ERV effectiveness during the part-load operation as a homographic function of outdoor temperature. This is valid only for a range of outdoor temperatures lower than the supply temperature and higher than the critical temperature that the heating system comes to operation.

Either of the above mentioned strategies (economizer control or part-load ERV operation) can be applied during the cooling season when the outdoor temperature is below the desired supply temperature. For both cases, a net zero energy is consumed by the HVAC system to condition the air, but the fan power is higher when employing the economizer approach due to the introduction of higher ventilation flow rates (which causes a higher pressure drop) compared to ERV approach.

2.6.4. Optimization Results

For given indoor condition, supply condition, supply flow rate and ventilation air flow rate, the ERV effectiveness which minimizes coil load (Equations (2-4) and (2-9)) is computed using MATLAB optimization tool. The result for a wide range of outdoor temperatures and humidities is presented on the psychrometric chart in Figure 2.7.

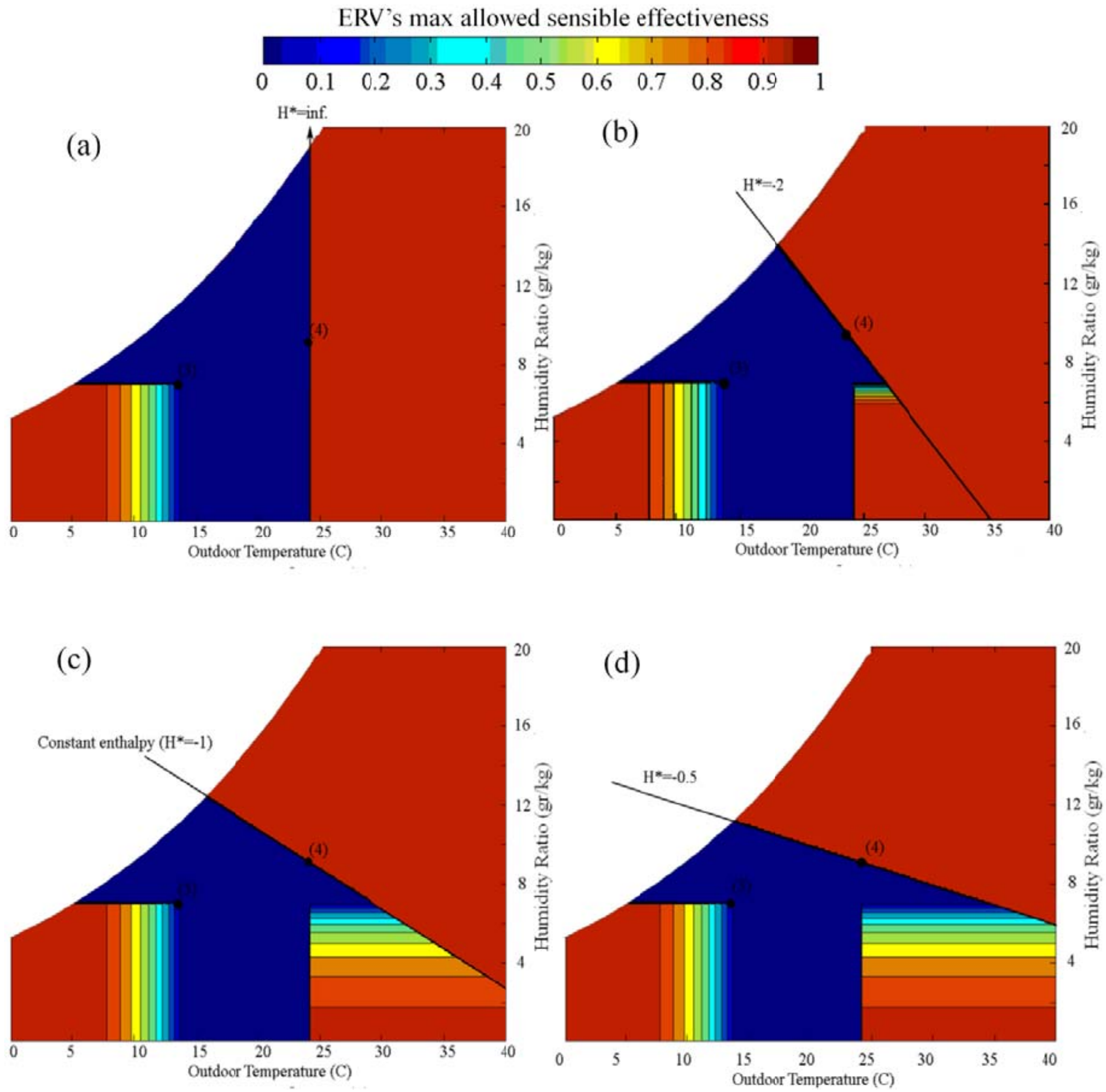


Figure 2.7 Optimum strategies to control ERVs with different sensible and latent effectiveness, (a) ε_s=0.8, ε_l=0 ($\varepsilon_s/\varepsilon_l=\infty$), (b) ε_s=0.8, ε_l=0.4 ($\varepsilon_s/\varepsilon_l$=2), (c) ε_s=0.8, ε_l=0.8 ($\varepsilon_s/\varepsilon_l$=1) and (d) ε_s=0.4, ε_l=0.8 ($\varepsilon_s/\varepsilon_l$=0.5)

The color bar shows the maximum allowable sensible effectiveness at which the ERV could be operated. "0" stands for the case that ERV should not transfer heat (and moisture) between the two air streams, where "1" means that the ERV can operate with 100% sensible effectiveness (if possible). Values within this range present the maximum allowable sensible effectiveness at which an ERV should be operated. For example, the yellow areas present that the ERV's sensible effectiveness should not exceed 65%. Therefore, if the ERVS's sensible effectiveness is higher than 65%, it should be operated in part-load condition to maintain the sensible effectiveness at 65%. It's assumed that the part-load operation of an ERV decreases the sensible and latent effectiveness with the same ratio. This means that the ratio of sensible to latent effectiveness remains constant during the part-load operation.

The optimization results presented in Figure 2.7 show that the temperature-based control is appropriate only for sensible-only ERVs since this strategy allows the ERV operation for outdoor temperatures greater than the indoor, only (Figure 2.7a). As the latent to sensible effectiveness ratio increases, the ERV should fully operate for a larger portion of cool-humid outdoor condition and this agrees with the control strategy proposed by Rasouli et al (2010a) which suggests a variable angle strategy depending on the ERVs latent to sensible effectiveness ratio. Also, Mumma's control strategy (Mumma 2001) should be applied only for ERVs with sensible to latent effectiveness ratios equal to unity (similar to the case presented in Figure 2.7c). The part-load operation of ERV for hot-dry outdoor condition is not addressed in previous research and is to prevent overhumidifying the dry ventilation air to humidities higher than the supply (which imposes an unnecessary dehumidification).

2.7.Discussion of Control Alternatives and Energy Savings with the Optimal Control

2.7.1. Model Description

A 10-storey office building with total floor area of 28,800 m^2, representing 3.34% of the existing U.S. office (Briggs et al. 1987), is selected and simulated in four North American cities representing different climatic conditions, i.e., Saskatoon (Saskatchewan, Canada), Chicago (Illinois), Miami (Florida) and Phoenix (Arizona). Chicago, Miami and Phoenix are selected based on Brigg's climatic classification which suggests these locations as representatives of cool-humid, hot-humid and hot-dry climate, respectively (Briggs et al. 2003). Saskatoon is also representing a cold climate considering the significant fraction of a year in heating season. The building description is taken from a study carried out at Pacific Northwest National Lab and includes the building parameters required for an energy analysis. The building has about 30 W/m^2 of internal heat gains and an occupant density of 5 Person/100 m^2 which requires outdoor ventilation air at the rate of 0.5 ACH (11.3 m^3/s), limited to occupied hours, to meet ASHRAE Standard 62-2004 (ASHRAE 2004a). The ventilation rate is reduced to 50% and 25% of design flow rate on Saturdays and Sundays, respectively. The indoor temperature during occupied hours is maintained within ASHRAE comfort zone (i.e., 24°C in summer and 22°C in winter) (ASHRAE 2004b) and is kept above 15°C during unoccupied hours. Also, dehumidification is provided to prevent indoor humidities higher than an upper limit (i.e., 60% RH) during the occupied hours in summer and no humidification is provided during the winter. This reasonably presents a practical cooling and heating system operating in a majority of office buildings in North America. More details about the building can be found in section 2.2, Appendix B.

The thermal system (including the HVAC system, ERV and the building) is simulated using TRNSYS building energy simulation tool (Klein 2000) equipped with the Second version of TESS libraries (Thornton et al. 2009) working in conjunction with MATLAB programming language (MATLAB 2006).

2.7.2. Comparison of Control Strategies

For a 90% sensible and 60% latent effectiveness ERV, the present control strategies, i.e., optimum strategy presented in the paper, temperature-based control, Mumma's control for enthalpy wheels (Mumma 2001) and the control strategy proposed by Rasouli et al (2010a), are compared to the base case when no ERV is in operation.

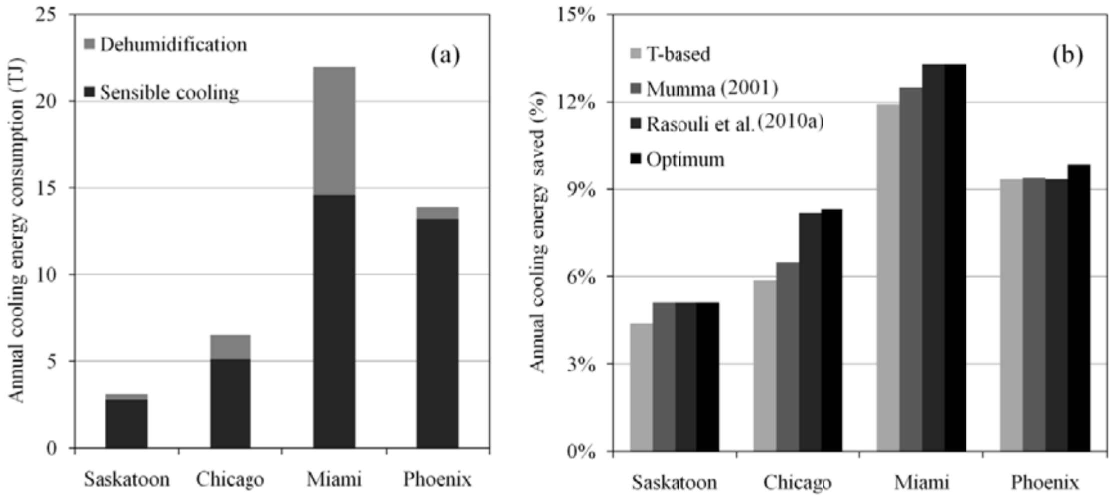


Figure 2.8 (a) Annual cooling energy consumption for each location without an ERV, and (b) annual cooling energy saving with an ERV (90% sensible and 60% latent effectiveness) controlled with different strategies

Considering the results presented in Figure 2.8, the optimum control can predict the savings higher than present control strategies. Among the present controls, the control strategy presented by Rasouli et al. (2010a) gives higher savings compared to other alternatives; however, it does not require a complicated part-load control in hot-dry outdoor condition. Employing the appropriate control is more critical for humid climate (Miami and Chicago) and the extra savings with optimal control is more significant than dry climate (Phoenix and Saskatoon). In conclusion, part-load operation of ERV in hot-dry outdoor condition (as required by optimal control) may not lead to a significant saving (except in hot-dry climate) and Rasouli's control may reasonably meet the saving potentials with no need for a complicated part-load control in hot-dry condition. For hot-dry climate, a temperature-based control which is easier to design may be applied, and it is shown that such control leads to a negligible reduction in saving (about 0.3% less saving) compared to the optimized system.

2.7.3. Energy Saving During Heating Season

As discussed previously, the ERV heats and humidifies the cold and dry outdoor air when the building requires heating; therefore, the ERV operates at the maximum capacity and requires no control. In this section, energy savings by employing ERVs with sensible effectivenesses in practical range of 55% to 95% are investigated. This is the range recommended by ASHRAE standard 90.1 (ASHRAE 2004c) since it requires ERVs with effectivenesses greater than or equal to 55%. In the base case, no ERV was in operation and the heating equipment had to meet building and ventilation loads. Figure 2.9a presents the building annual heating energy consumption when no ERV is employed. The savings with ERVs for each location (except for Miami due to insignificant heating energy consumption) within a range of sensible effectiveness are presented in Figure 2.9b.

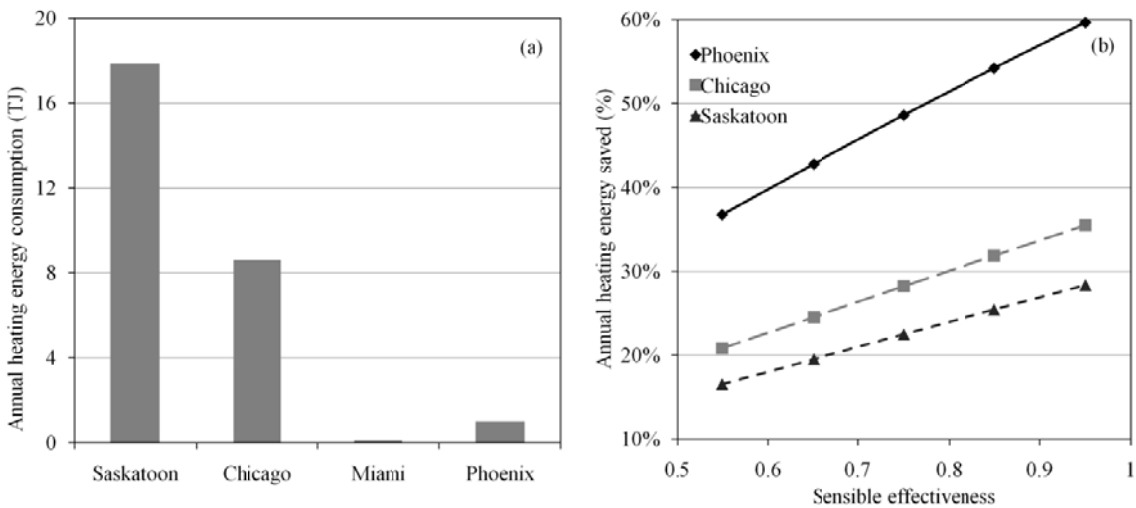


Figure 2.9 Impact of ERV on heating energy saving, (a) annual heating energy consumption in each location without an ERV and (b) annual heating energy saved with ERV

As presented in Figure 2.9, the amount of annual heating energy saved with ERV is more significant in cold climate (Saskatoon and Chicago) than hot climate (Phoenix). About 30% of the annual heating energy could be saved in cold climate by employing ERVs with 80% sensible effectiveness.

A notable impact that the ERV has on the indoor environmental condition is the increase of indoor humidity when humidification is not provided by the HVAC system during the heating season. For a majority of office buildings, the heating system is designed to meet the indoor temperature set point by applying a sensible-only heating process to the air supplied to the space. When employing ERVs with the capability of moisture transfer, the water vapor is transferred from the air exhausted from conditioned space to the cold-dry outdoor ventilation air. This process increases indoor relative humidity and improves occupant's comfort. Studies have discussed the undesirable effect of dry environmental condition on intensity of dryness, aching eyes and nose-related symptoms which weakens the comfort (Tham 2004). Green (1974) carried out an experimental research of students in schools with or without humidification, and reported that the absenteeism decreases in school when indoor RH increased to between 20 and 40%. It was concluded that absenteeism decreased by 3–9% for each percentage point increase in RH. In this study, the average indoor RH during the heating season was computed for each location (except Miami due to negligible need for heating). The average indoor RH without ERV was found 15.5% for Saskatoon and Chicago and 17.1% for Phoenix. When employing an ERV with 75% sensible and latent effectiveness, the indoor RH was increased by 1.2%, 2.5% and 2% in Saskatoon, Chicago and Phoenix, respectively. The impact of higher indoor RH on occupants' comfort and reduction of absenteeism may be used for ERVs life cycle cost assessments.

2.7.4. Energy Saving During Cooling Season

Annual cooling energy savings in different locations when employing ERVs in practical range of sensible and latent effectiveness under optimum control strategy are presented in Figure 2.10.

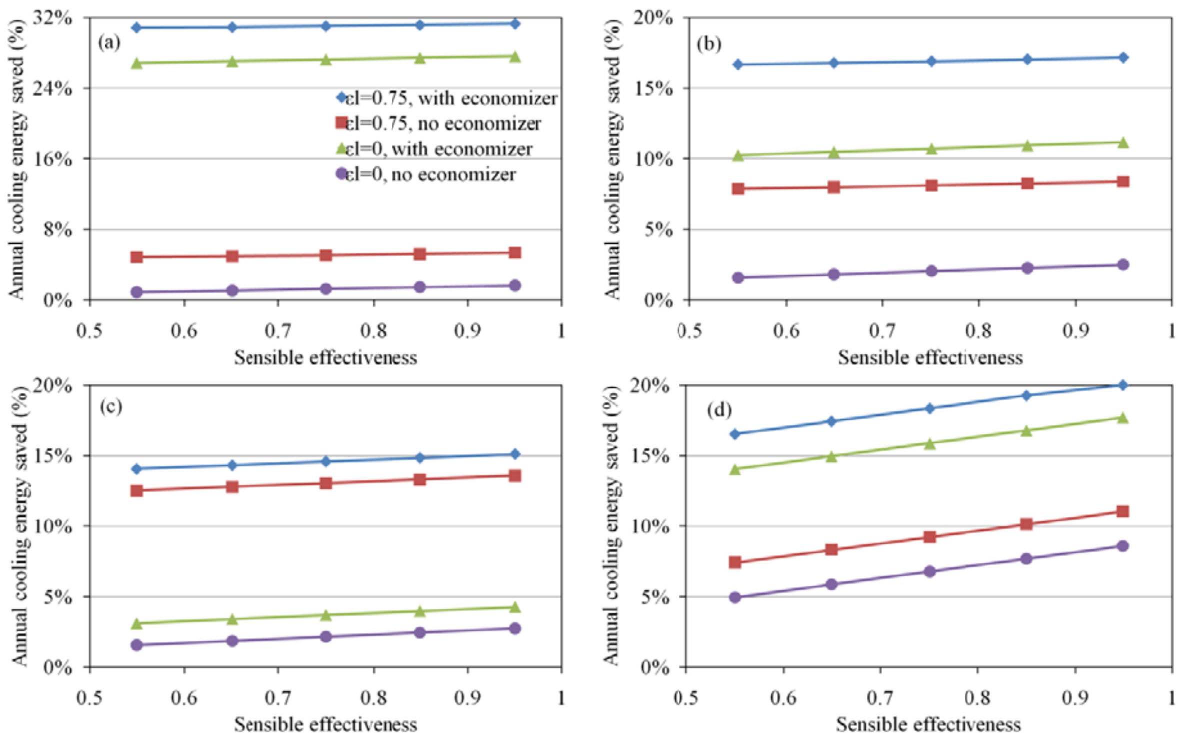


Figure 2.10 Impact of ERV on annual cooling energy saving using the optimal control strategy, (a) Saskatoon, (b) Chicago, (c) Miami and (d) Phoenix

For humid climates (Miami and Chicago), a significant difference between a heat recovery system (ε_l=0) and heat and moisture recovery systems is observed which presents the importance of moisture transfer in humid climates. For instance, a sensible-only exchanger can save up to 4% in Miami and Chicago, where the savings can be increased by 10% in Chicago and 15% in Miami when a heat and moisture recovery system is employed. For Phoenix, as representative of hot and dry climate, the change of sensible effectiveness is found to have more impact on energy savings than the latent effectiveness. Compared to sensible-only ERV, the more humid the outdoor conditions, the more superior is the heat and moisture recovery system in reducing the annual cooling energy consumption.

It should be noted that Commercial Building Energy Consumption Survey (CBECS) has reported an average energy intensity of $46.9*10^3$ (BTU/ft^2.year) (≈533 MJ/m^2.year) for HVAC system energy consumption in US office buildings (EIA 2003). In this research, the energy intensity of the studied office building varies depending on the climate and is found

726, 526, 780 and 519 (MJ/m^2.year) for Saskatoon, Chicago, Miami and Phoenix, respectively. Miami and Saskatoon represent extremely hot or cold weather conditions, therefore the results show a higher energy intensity for these two locations. The simulated office building in Phoenix and Chicago has energy intensities close to the CBECS reported average value due to the mild weather condition in these locations. On average, these results are reasonably close to the expected values reported by CBECS.

2.8. Conclusions

In this paper, an optimum control strategy for ERVs that minimizes the energy consumption of HVAC systems was introduced. The optimum control strategy was developed based on cooling load minimization and compared to the three present control strategies described in the paper. The optimization results agree with a control strategy proposed by Rasouli et al. (2010a) in that the operation of ERV for cool-humid outdoor condition depends on ERV sensible to latent effectiveness ratio. Their proposed control was dependent on operating condition factor, H^*, ranging from $-\infty$ to $+\infty$ and presenting latent to sensible energy potentail of inlet airstreams. Depending on the latent to sensible effectiveness ratio, the operation of ERV in cooling season should be limited to specific outdoor conditions within a certain range of operating condition factors described in the paper. For example, the optimum operating condition for an ERV with equal sensible and latent effectiveness values is when the outdoor air has higher enthalpy or higher temperatures than the indoor. But, it was found that the optimum control necessitates a part-load operation for hot-dry outdoor condition which was not addressed in previous literature, however, such part-load control leads to a negligible saving when applied. The impact of an optimized ERV on annual cooling and heating energy consumption was studied by conducting TRNSYS simulations of a 10-storey office building in four North American locations representing major climatic conditions (i.e., Saskatoon with a cold and dry climate, Chicago with a cool and humid climate, Miami with a hot and humid climate and Phoenix with hot and dry climate). Depending on the climate and ERV effectiveness, an ERV with capability of moisture recovery may reduce the annual heating energy consumption by 35% in cold climate. When the ERV operated under the proposed optimum control, up to 20% annual cooling energy was saved in hot climate. The savings in humid climates (Chicago and Miami) were found more significant than elsewhere since the moisture transfer in ERV could reduce the dehumidification load dramatically.

CHAPTER 3

APPLICATION OF A RAMEE IN AN OFFICE BUILDING HVAC SYSTEM

3.1. Overview of Chapter 3

This chapter includes manuscript # 3 that studies the application of a RAMEE in an office building HVAC system. The chapter begins with an overview of the RAMEE system and a summary of the related research work conducted by former graduate students in the RAMEE research group at the University of Saskatchewan. The office building, HVAC system and a RAMEE that is operating under optimal control (described in Chapter 2) are simulated in four different climatic conditions using TRNSYS computer program. Yearly simulations are run to investigate the impact of the addition of the RAMEE to a base HVAC system that is not equipped with any types of ERVs. The results present the impact of a RAMEE on energy savings, the size of heating and cooling equipment and life-cycle cost of the HVAC system.

Three graduate students contributed to the completion of this research work. H. Hemingson (Hemingson 2010) developed and modified the numerical solution of heat and mass transfer in the RAMEE. S. Akbari used the results of this numerical solution to produce an artificial neural network. This neural network is able to predict the optimal operating condition of the RAMEE based on the control strategy presented in Chapter 2. My contribution to this research was to (a) develop the computer models of the HVAC system, the RAMEE (Appendices A1, A2 and A3) and the building, (b) run the simulations for different cases and climates (c) post-process the results and (d) organize the data and write the paper.

Manuscript#3: application of a run-around membrane energy exchanger in an office building hvac system

M. Rasouli, S. Akbari, H. Hemingson, R.W. Besant and C.J. Simonson

3.2. Abstract

A Run-Around Membrane Energy Exchanger (RAMEE) has been introduced in the literature as a novel energy recovery system that transfers heat and moisture between the ventilation and exhaust air. The RAMEE consists of two separate (supply and exhaust) flat-plate exchangers made of water vapor permeable membranes, and coupled with an aqueous salt solution. In this paper, the application of a RAMEE in an HVAC system is investigated. The paper discusses the dependency of RAMEE performance on ventilation air and salt solution flow rates and indoor and outdoor air conditions and describes how to control the RAMEE in different operating conditions (summer, winter and part-load). An Artificial Neural Network (ANN) that is able to predict the optimal system performance was developed in previous research. The ANN results are used for TRNSYS computer simulation of the RAMEE system when operating in an office building in four different climates. The results show up to 43% heating energy saving in cold climates, and up to 15% cooling energy saving in hot climates. Cost analysis proves the important role of pressure drop across the exchangers in life cycle cost, and predicts payback period ranging from 2 to 5 years for the RAMEE.

3.3.Introduction

Recent research has presented a strong relationship between indoor air quality (IAQ) and occupants' productivity (Fang et al. 2000; Kosonen and Tan 2004; Seppänen and Fisk 2005). On the other hand, studies have indicated a higher demand for energy when a higher ventilation flow is introduced to a conditioned space (Brandemuehl and Braun 1999; Orme 2001; McDowell et al. 2003). Therefore, HVAC system operating conditions and equipment sizes should be optimized to provide a satisfactory level of productivity and thermal comfort while HVAC energy consumption is minimized.

Energy Recovery Ventilators (ERVs) reduce the energy required to condition ventilation air by transferring heat (and moisture) between conditioned exhaust air and outdoor ventilation air. The pre-conditioning of this outdoor air reduces the energy required by HVAC systems, while thermal comfort is satisfied. In general, ERVs can be divided into two groups: i.e., heat recovery systems which transfer only sensible heat, and heat and moisture recovery systems (also called energy exchangers) which transfer both sensible and latent energy. Heat pipes, flat plate heat exchangers and rotary heat wheels only transfer heat between the supply and exhaust airstreams, however, they are common due to their low pressure drop and convenient maintenance (Besant and Simonson 2003). The main disadvantage of heat recovery systems is that they cannot transfer moisture. Energy wheels and permeable flat plate exchangers can transfer both heat and moisture. For example, an energy wheel coated with a desiccant can transfer both heat and moisture between two air streams (Simonson and Besant 1997; Simonson 2007). Flat plate exchangers constructed with water permeable membranes can transfer heat and moisture between the airstreams (Zhang and Jiang 1999).

All above mentioned devices require that the supply and exhaust ducts to be side-by-side which usually imposes higher ducting costs. In addition, contaminant carryover in rotary wheels and cross-flow leakage of air through seals are concerns in some types of buildings such as health care facilities and laboratories. The extra ducting cost and the contaminant transfer could be avoided if the exhaust and supply air ducts were separated. In this paper, a literature review on a novel Run-Around Membrane Energy Exchanger (RAMEE) which is capable of transferring both heat and moisture between remote supply and exhaust ducts is presented. Since the performance of a RAMEE depends on the ventilation air and salt solution flow rates and indoor and outdoor air conditions, which continuously change throughout the year, appropriate control of the RAMEE system is needed. Therefore, an investigation on the optimum operation of a RAMEE during summer, winter and part-load conditions is conducted. As a case study, an office building equipped with a RAMEE is simulated in different climates using the TRNSYS (Klein 2000) computer program, and the potential cooling and heating energy savings are presented. A Life Cycle Cost Analysis (LCCA) is performed over a 15-year period to study the economics of the RAMEE system compared to a conventional HVAC system with no energy recovery.

3.4. Run-Around Membrane Energy Exchanger (RAMEE)

In this section, an overview of the literature is presented to introduce the RAMEE. A schematic of exchangers and the flow diagrams of a HVAC system equipped with a RAMEE is described. The parameters affecting the RAMEE effectiveness are discussed.

3.4.1. Exchanger Design

A Run-Around Membrane Energy Exchanger (RAMEE), shown in Figure 3.1a, which exchanges both heat and water vapor between the exhaust air and un-conditioned outdoor ventilation air has been proposed to overcome the limitations of currently available ERVs (Fan et al. 2005). The RAMEE system consists of two separate exchangers with a salt solution coupling liquid that is pumped in a closed loop between the two exchangers. Each exchanger, which is called a liquid to air membrane energy exchanger (LAMEE), is a flat plate energy exchanger constructed with vapor permeable membranes that allow the transmission of water vapor but not liquid water. The salt solution loop couples these two LAMEEs in the RAMEE, and the air and salt solution may flow in cross flow (Seyed-

Ahmadi et al. 2009a and 2009b), counter/cross flow (Mahmud et al. 2009; Vali et al. 2009) or counter flow (Hemingson et al. 2010) arrangement through each of the two LAMEEs placed into the supply and exhaust streams. However, the flow arrangement that combines high performance with practical header design is the cross/counter flow arrangement as shown in Figure 3.1(b). It should be noted that the numerical simulation results of a counter flow LAMEE are used in this paper (Vali et al. 2009). However, the manufacturer may consider a counter/cross flow due to the limitation in separating the flow inlets. The numerical model of the RAMEE system shows that a good cross/counter flow will reduce the RAMEE effectiveness by less than 2% compared to a counter flow design at the same operating condition (Vali 2009).

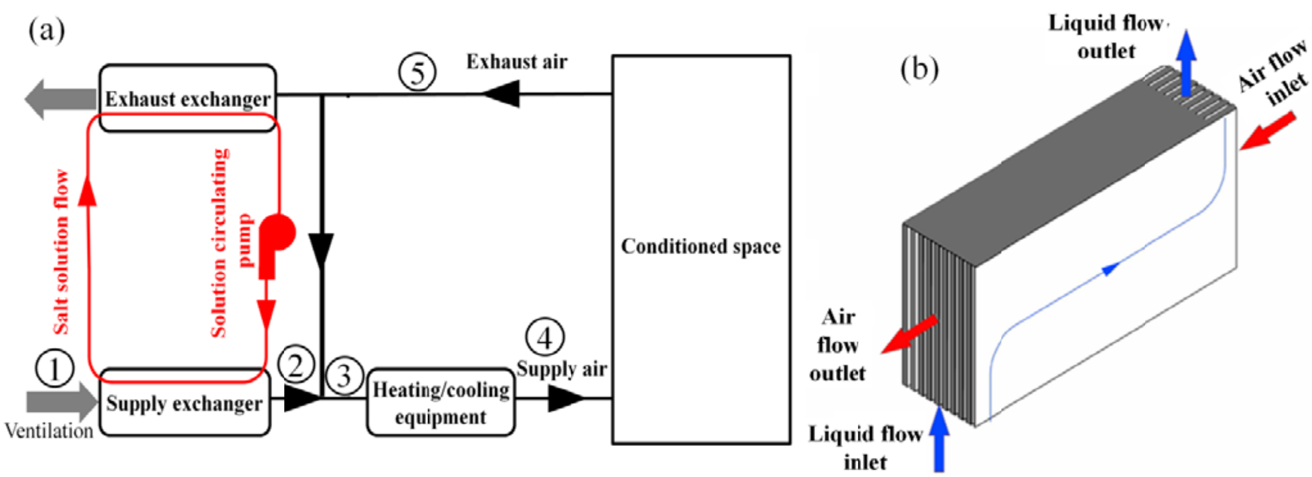


Figure 3.1 Schematic diagram of a (a) HVAC system equipped with a RAMEE, and (b) air and solution flow in a LAMEE

The RAMEE system uses the exhaust air to precondition the ventilation air and decreases the energy consumption and the size of the heating/cooling equipment. For example, during the summer when the outdoor air is warm and humid, the desiccant salt solution gains heat and moisture from the ventilation air stream in the supply exchanger. The solution is then pumped into the exhaust exchanger where it releases this heat and moisture to the exhaust air stream. This loop cools and dehumidifies the outdoor ventilation air in summer. During winter, the salt solution gains heat and water vapor from the conditioned exhaust air when passing through the exhaust exchanger. This solution then releases both heat and moisture while it flows through the supply exchanger and thus pre-conditions (i.e., heats and humidifies) the ventilation air before it enters to the heating equipment.

3.4.2. System Performance

Based on the numerical model developed in previous research (Vali et al. 2009; Hemingson et al. 2010) for a RAMEE system with equal supply and exhaust air flow rates, the RAMEE effectiveness in transferring heat (ε_s), moisture (ε_l) and enthalpy (ε_t) is a function of three dimensionless groups, i.e., NTU (number of heat transfer units), NTU_m (number of mass transfer units) and Cr* (ratio of salt solution heat capacity to that of the air) as defined below:

$$\varepsilon_s = \frac{T_1 - T_2}{T_1 - T_5} \tag{3-1}$$

$$\varepsilon_l = \frac{W_1 - W_2}{W_1 - W_5} \tag{3-2}$$

$$\varepsilon_t = \frac{h_1 - h_2}{h_1 - h_5} \tag{3-3}$$

$$NTU = \frac{UA}{\dot{m}_{air} C_{p,air}} \tag{3-4}$$

$$NTU_m = \frac{U'A}{\dot{m}_{air}} \tag{3-5}$$

$$Cr^* = \frac{\dot{m}_{sol} C_{p,sol}}{\dot{m}_{air} C_{p,air}} \tag{3-6}$$

In addition, the system performance strongly depends on the condition of outdoor ventilation air, and slightly depends on the indoor air conditions which might vary between summer and winter indoor set-points (Hemingson et al. 2010).

3.4.2.1. Impact of NTU and Cr* on RAMEE Performance

Equation (3-4) shows that NTU is directly related to the heat exchange surface area of each exchanger and represents the size of the RAMEE. The higher the NTU, the higher the effectiveness (shown in Figure 3.2(a)) (Hemingson et al. 2010). Cr* characterizes thermal capacity rate of the liquid flow compared to the thermal capacity rate of the air flow in the

RAMEE system and is similar to Cr used in the literature to describe the thermal capacity rate ratio for run around heat exchangers (Vali 2010). As shown in Figure 3.2(b), effectiveness increases from zero as Cr* increases from zero until it reaches the peak value. The optimum Cr* at which the peak performance is achieved depends on the type of ERV. For instance, the maximum effectiveness of a run-around heat and moisture recovery system operating at the AHRI summer test conditions (AHRI 2005) occurs approximately at Cr*=3 (for equal supply and exhaust air flow rates), while a run-around heat recovery system has its peak effectiveness at Cr=1 (London and Kays 1951).

Hemingson et al. (2010) used a numerical model to predict the RAMEE effectivenesses in different outdoor conditions and these results showed good agreement with heat transfer theory. They indicated that the RAMEE effectiveness increases with NTU (as shown in Figure 3.2(a)) and it follows the same trend as expected by analytical solutions and empirical correlations (e.g., Zhang and Niu 2002; Incropera and DeWitt 2002). The system has a significantly higher sensible effectiveness and slightly higher latent effectiveness when its NTU is increased. Also, increasing NTU_m leads to a considerable increase in latent effectiveness and a slight increase in sensible effectiveness. The system performance varies with Cr* until it reaches the optimal value where the peak performance is achieved. This is schematically shown in Figure 3.2(b) for a specific outdoor condition. It should be noted that the dependency of the RAMEE effectiveness on Cr* varies with outdoor condition and is discussed in the next section.

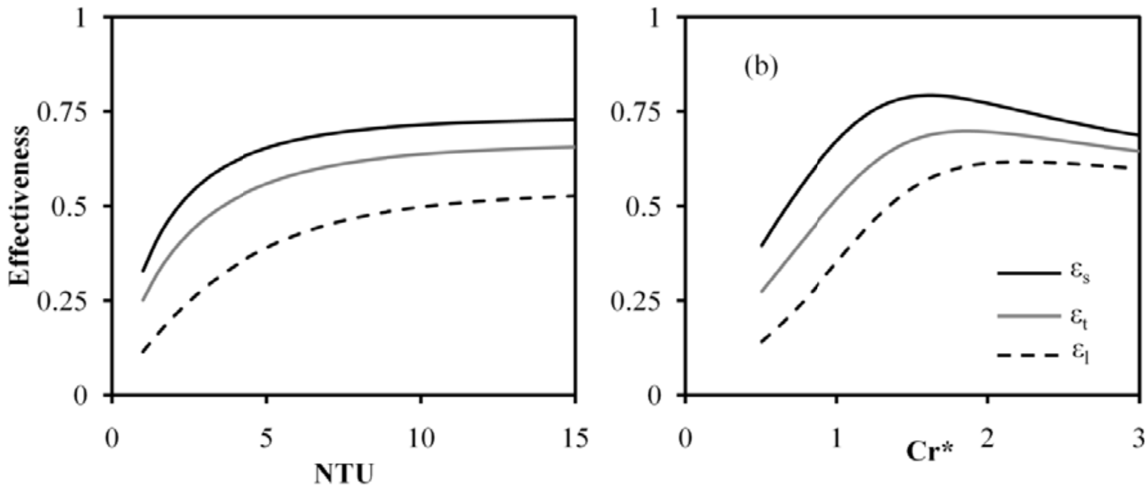


Figure 3.2 Variation of RAMEE effectiveness as a function of NTU and Cr* for outdoor condition at 5°C and 5 g/kg and indoor condition at 22°C and 9.3 g/kg (a) NTU (at Cr*=1.3) and (b) Cr* (at NTU=10)

3.4.2.2. Impact of Indoor and Outdoor Conditions on RAMEE Performance

Hemingson et al. (2010) showed the influence of outdoor air temperature and humidity on the effectiveness of the RAMEE. The main reason for the dependency of RAMEE effectiveness on outdoor conditions is the impact that outdoor temperature and humidity will have on the liquid desiccant and the fact that heat and moisture transfer are coupled. The moisture transfer between the two fluid streams in each LAMEE releases/absorbs phase change energy and increases/decreases the desiccant temperature and consequently the sensible effectiveness. The change in desiccant temperature and humidity due to heat and moisture transfer affects the latent effectiveness of the system as well. Hemingson et al. (2010) concluded that as the temperature difference between outdoor and indoor air increases (either summer or winter), the latent effectiveness increases. Also, the greater the humidity ratio difference between the indoor and outdoor air, the higher the heat transfer. Figure 3.3 presents the RAMEE effectiveness for five different outdoor conditions for NTU=10 where the summer/winter indoor conditions are chosen from the AHRI test conditions (AHRI 2005).

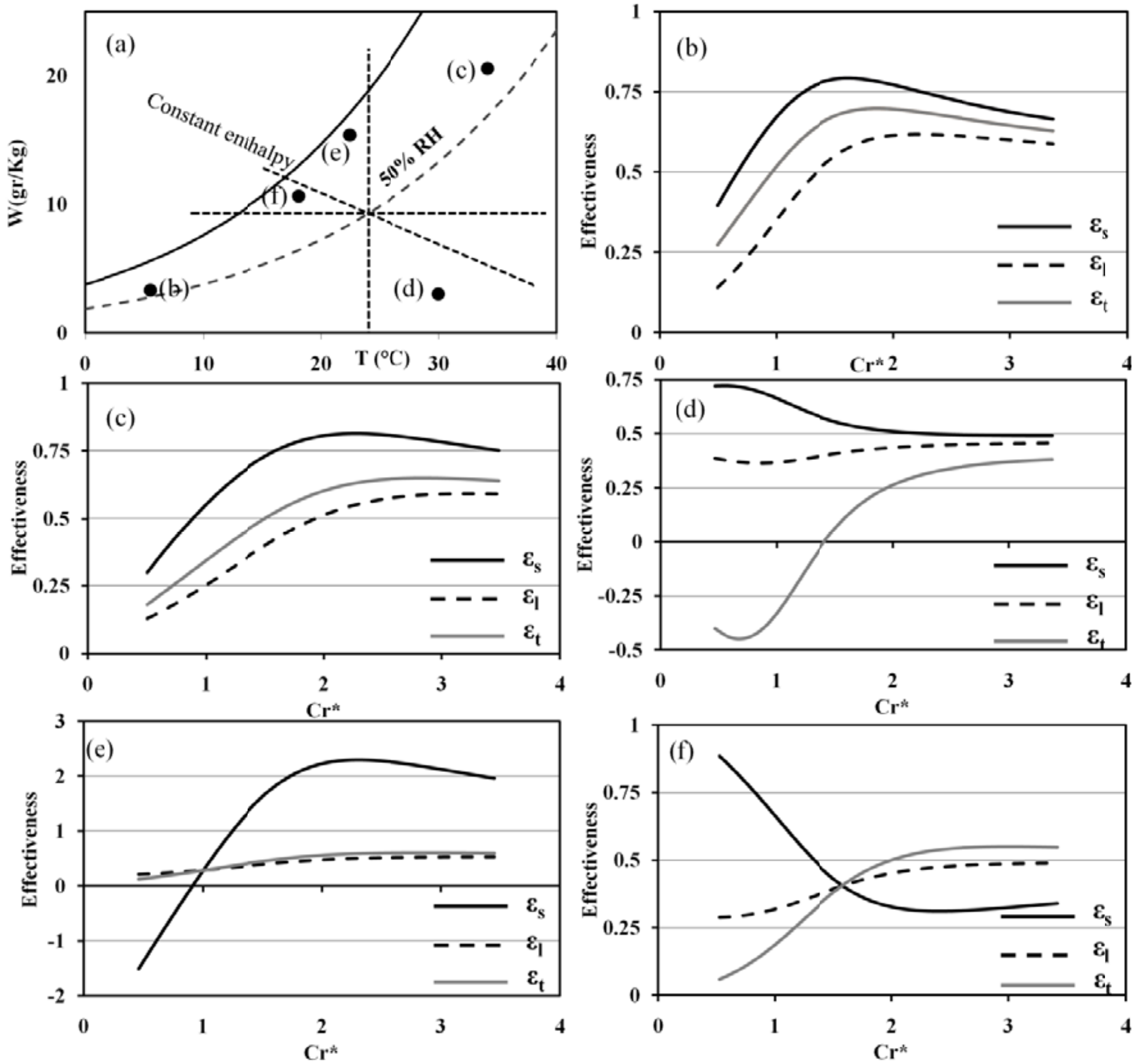


Figure 3.3 RAMEE effectiveness versus Cr* for five different outdoor conditions (NTU=10) (a) the psychrometric chart, (b) cold-dry (5°C and 5 g/kg), (c) hot-humid (35°C, 20g/kg), (d) hot-dry (30°C, 2g/kg), (e) cool-humid, high enthalpy (22°C, 15g/kg), and (f) cool-humid, low enthalpy (19°C, 10g/kg)

Figure 3.3 shows that the optimal Cr* (i.e., the Cr* at which the maximum RAMEE effectiveness is achieved) varies significantly with outdoor condition. Cr*, as defined in Equation (3-3), depends on ventilation air and salt solution flow rates. For a given building, where ventilation rates are maintained at a constant rate specified by standards (e.g., ASHRAE 2004a; ASHRAE 2008), Cr* remains only a function of the salt solution flow rate. Therefore, the salt solution flow rate should be controlled to give the optimal Cr* at all outdoor conditions.

Regarding the impact of indoor condition, Hemingson et al. (2010) found that changing the indoor conditions between summer and winter indoor temperature and humidity set-points has a minimal impact on RAMEE performance (about 0.3% change in total effectiveness).

3.5. RAMEE Control

As mentioned in the previous section, the RAMEE effectiveness depends on NTU, Cr* and indoor and outdoor air conditions. Among these variables, only NTU and Cr* are controllable and the optimal operation of the RAMEE system requires proper control of these variables. The design NTU is determined during the exchanger design and manufacturing process. But, it can be changed by changing the ventilation air flow rate (e.g., bypassing a fraction of ventilation air) during the operation of the RAMEE. The operating Cr* can be controlled via adjustment of salt solution or ventilation air flow rates by the operator during the operation of the RAMEE.

NTU represents the size of the RAMEE system, and the greater the NTU, the higher the effectiveness. On the other hand, increasing the size of the system increases the manufacturing costs (Teke et al. 2010). Therefore, NTU should be large enough to give a reasonable effectiveness, but not extremely large which may cause excessive production cost. A design NTU of 10 is found feasible in the literature (e.g., Teke et al. 2010) and is used for this study. However, it may change as the ventilation rate might change during the operation of the RAMEE. The following sub-sections discuss the appropriate control of Cr* and NTU to achieve the optimal performance of the RAMEE system in different operating conditions (i.e., summer, winter and part-load).

3.5.1. Heating Season (Winter)

When the outdoor temperature is lower than the HVAC system indoor set-point, and the internal heat loads and solar radiation gains do not satisfy the space heating demand, the heating system needs to be operated. Due to a low outdoor air temperatures and moisture content, conditioning the outdoor ventilation air during cold weather requires heating and possibly humidification.

Previous research (Simonson 2007; Fauchoux et al. 2007; Liu et al. 2009; Rasouli et al. 2010) has studied the savings using different types of ERVs in various climates and have found that the operation of ERV is beneficial especially for cold weather conditions. For instance, Rasouli et al. (2010a) simulated an office building in different climates and showed that ERVs with sensible effectiveness values in the range of 55%-95% may save 15-30% of annual heating energy for buildings in cold climates. They showed that in a typical office building in the US, the sensible heating accounts for most (about 96%) of the annual HVAC heating energy consumption while humidification accounts for less than 4% of the annual heating energy when the goal is to maintain an indoor humidity of 30% RH. Since humidification energy is small and many buildings don't have humidification system, the focus on the winter is to reduce the sensible heating energy.

As shown in Figure 3.3, the Cr* at which maximum sensible, latent and total effectiveness occur depends on the outdoor conditions. As indicated by Rasouli et al. (2010a), minimizing the sensible heating load of the HVAC system is the main concern during the winter, therefore, the optimal Cr* is the Cr* at which the sensible effectiveness is maximum (Cr* of about 1.5 in Figure 3.3(b)). Applying such an optimal Cr* does not sacrifice the latent effectiveness, and gives a latent effectiveness that is only slightly lower than its peak value. The moisture transfer from exhaust air to the outdoor ventilation air should improve the indoor humidity during the winter when outdoor air is mostly dry and humidification is not provided by the HVAC system. Studies have shown that absenteeism in schools and offices may be reduced when the indoor humidity is increased in the winter (Green 1974; Tham 2004).

3.5.2. Cooling Season (Summer)

Research on ERVs in the cooling season has shown that reducing the annual cooling energy requires proper control of the ERV (Rasouli et al. 2010b; Mumma 2001; Zhang and Niu 2001). In general, the present control strategies can be categorized into two groups: (i) temperature-based controls which allow the ERV to operate only if the outdoor air temperature is greater than the indoor air, and (ii) enthalpy-based controls which allow the ERV to operate only if it can reduce the enthalpy of outdoor air. Rasouli et al. (2010b) compared the present control strategies and proposed an optimal ERV control. Based on their results, an ERV should be operated only if it can reduce the enthalpy of outdoor ventilation air, and the greater the reduction of outdoor air enthalpy the lower the coil cooling load. Therefore, as defined in Equation (3-3), the RAMEE system should be operated at maximum absolute total effectiveness when the outdoor enthalpy is greater than the indoor, and should have minimum (and negative) total effectiveness when the outdoor enthalpy is lower than the indoor.

For a better explanation, refer to the performance of the RAMEE in four different summer outdoor conditions presented in Figure 3.3(c), (d), (e), and (f). For cases (c) and (e), where the outdoor enthalpy is greater than the indoor enthalpy, the RAMEE should be operated at maximum positive total effectiveness (i.e., Cr* of about 2.5). Such Cr* maximizes both heat and moisture transfer (cooling and dehumidification) for the hot-humid case (Figure 3.3(c)). But, it maximizes the moisture transfer (dehumidification) and minimizes the heat transfer (heating) for the cool-humid case (Figure 3.3(e)). When the outdoor enthalpy is lower than the indoor enthalpy and the cooling is still required, the RAMEE should be operated only if a negative total effectiveness can be achieved by adjusting the appropriate Cr*. Therefore, for case (d), the RAMEE should be operated at Cr* of about 0.8 where the minimum (and negative) total effectiveness is achieved. Such Cr* maximizes the heat transfer (cooling) and minimizes the undesirable moisture transfer (humidification). In case (f), however, the RAMEE should be turned off, because no Cr* value gives negative total effectiveness values.

3.5.3. Economizer

During the heating and cooling season, HVAC system energy consumption increases as the outdoor ventilation rate increases (Brandemuehl and Braun 1999; McDowell et al. 2003). Therefore the outdoor air flow is typically maintained at the minimum rate that satisfies ASHRAE ventilation standard requirements (ASHRAE 2004a). However, during cool summer days when the internal loads and solar gains necessitate the operation of the cooling system, free cooling can be provided by increasing the outdoor air flow rate. In such outdoor conditions, the RAMEE should be turned off (to prevent heating of the cool outdoor air) and an economizer should be employed to introduce 100% outdoor air to meet a portion (or all) of the building cooling load. This will reduce (or even eliminate) the cooling load and improves the indoor air quality. Seem and House (2010) introduced a strategy to control economizers based on minimization of coil cooling load. Their results showed that the outdoor ventilation flow should be increased when the outdoor enthalpy and outdoor temperature are lower than the indoor. In practice, the introduction of 100% outdoor air when the outdoor temperature is slightly lower than the indoor temperature may not be beneficial, because the additional fan power may exceed the cooling energy savings. Therefore, in this paper, 100% outdoor air is provided when the outdoor enthalpy is lower than the indoor enthalpy and the outdoor temperature is between 14°C and 20°C. To prevent thermal discomfort, if the outdoor temperature falls below 14°C, a fraction of the exhaust air is recirculated and mixed with the outdoor air to maintain minimum of 14°C supply temperature.

3.5.4. Part-Load Operation

During cool summer days when the outdoor temperature is lower than the indoor temperature, a cooling system might be still required to meet the internal heat loads and solar radiation gain. The supply temperature is determined based on the building cooling load and the required ventilation air flow rate. In case the outdoor temperature is below the required supply temperature, the outdoor air needs to be heated up to the desired supply temperature. As an alternative, an ERV could be operated to heat the ventilation air, however, full-load operation of the ERV may overheat the outdoor air to temperatures greater than the desired supply temperature. This require the cooling of overheated air, and in such conditions, the ERV should be operated in part-load operating condition (i.e., not in full capacity of transferring heat and moisture).

Depending on the type of ERV, different methods can be used to adjust the effectiveness to the desired value. For example, adjusting the wheel speed for energy wheels, decreasing the flow rate of the fluid streams (ventilation or exhaust) or by-passing a fraction of the ventilation air can give the required effectiveness for other ERVs. For the RAMEE system, considering the parameters affecting the system effectiveness, adjusting NTU or Cr* are the two available strategies to control the part-load operation. Considering Equations (3-4) and (4-6), Cr* and NTU are functions of salt solution and ventilation air flow rate, therefore the system effectiveness could be changed by changing the flow rate of any of these two streams. Between the two available options, adjusting NTU is simpler because the RAMEE effectiveness is more predictable with changing NTU (i.e., effectiveness increases with NTU), but the effectiveness has a complex behavior with changing Cr* as shown in Figure 3.3 By-passing a fraction of ventilation air, as shown schematically in Figure 3.4, decreases the heat transfer from exhaust air to the cool ventilation air and prevents overheating. The bypass fraction should be adjusted carefully to give the desired supply temperature.

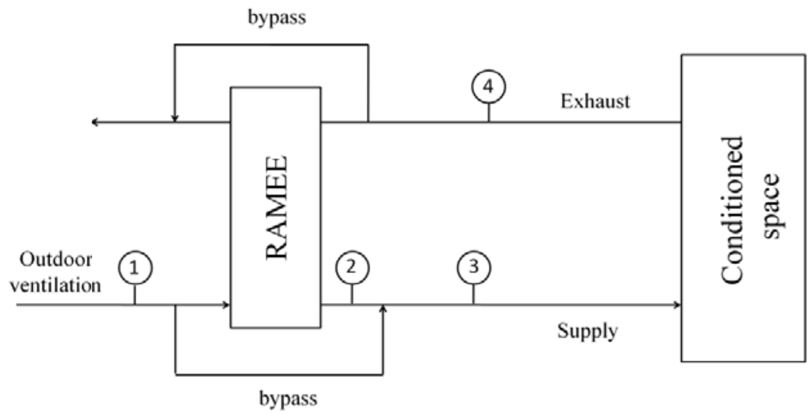


Figure 3.4 Schematic of the RAMEE system operating under part-load condition

For given indoor and outdoor conditions and a known ventilation rate (i.e., minimum standard requirement), the condition at state (3) is a function of RAMEE effectiveness, and the fraction of ventilation air bypassing the RAMEE:

$$T_3 = \frac{\dot{m}_{bypass}.T_1 + \dot{m}_2.T_2}{\dot{m}_{bypass} + \dot{m}_2} \tag{3-7}$$

Where T_2 is the condition of outdoor air leaving the RAMEE and can be stated as:

$$T_2 = T_1 - \varepsilon_s(T_1 - T_4) \tag{3-8}$$

And, assuming no air leakage in the exchangers:

$$\dot{m}_{bypass} + \dot{m}_2 = \dot{m}_1 = Ventilation\ rate\ (constant) \tag{3-9}$$

The condition at state (3) can be specified as a function of RAMEE effectiveness and bypass fraction by substitution of Equations (3-8) and (3-9) into Equation (3-7):

$$T_3 = T_1 + \varepsilon_s(1 - R)(T_4 - T_1) \tag{3-10}$$

where, R is the bypass fraction and is defined as:

$$R = \frac{\dot{m}_{bypass}}{\dot{m}_1} \tag{3-11}$$

Equation (3-13) can be re-arranged to determine the bypass fraction:

$$R = 1 - \frac{T_3 - T_1}{\varepsilon_s(T_4 - T_1)} \tag{3-12}$$

Equation (3-12) determines the by-pass fraction as a function of indoor, outdoor and supply temperature and the RAMEE optimal sensible effectiveness at the given operating condition. In conclusion, the operation of RAMEE in different outdoor conditions is shown on the psychrometric chart in Figure 3.5. States (3) and (4) refer to the condition of the supply air and indoor air, respectively.

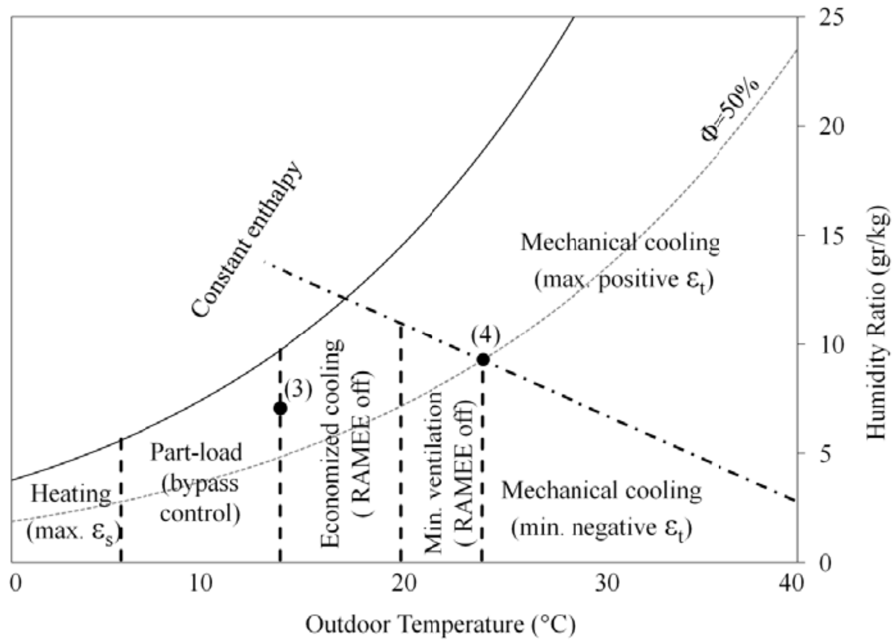


Figure 3.5 Operating condition of the RAMEE system in different outdoor condition

3.6.Model Specification

3.6.1. Building Description

The RAMEE system is simulated in a 10-storey office building with total floor area of 28,800 m^2 (310,000 ft^2), representing 3.34% of the existing U.S. office buildings (Briggs et al. 1987). The building description is taken from a study carried out at Pacific Northwest National Lab and includes the building parameters required for an energy analysis. The original building is constructed in Fort Worth, Texas, and only has about 2 cm (0.8 in) of insulation which gives a thermal resistance of 0.78 m^2 K/W (4.43 h ft^2 °F/BTU). In order to have a building that could fairly represent a typical building in different locations, walls, slabfloor and roof are improved by adding insulation layers. Walls are made of light weight concrete, an insulation layer and gypsum board that gives a total thermal resistance of 2.72 m^2 K/W (15.45 h ft^2 °F/BTU). The roof is made of built up roofing, insulation and aluminum siding that gives a total thermal resistance of 3.64 m^2 K/W (20.68 h ft^2 °F/BTU) and the slab thermal resistance is 3.45 m^2 K/W (19.60 h ft^2 °F/BTU). The windows are changed from single pane (as specified in the original PNL report) to double pane windows. The building has about 30 W/m^2 (9.5 BTU/h ft^2) of internal heat gains based on PNL report. An occupant density of 5 People/100 m^2 ($\approx$ 0.47 people/100 ft^2) is assumed that gives an outdoor ventilation air flow rate of 0.5 ACH (11.3 m^3/s; 24,000 CFM), limited to occupied hours (7am to 9pm), to meet the ASHRAE ventilation requirement (ASHRAE 2004a).

3.6.2. HVAC System

The cooling system operating in the described building is a variable air volume HVAC system (VAV HVAC) that supplies air at 14°C (57.2°F) or higher when the building is occupied. The RAMEE system pre-conditions the ventilation air, and the cooling unit completes the air-conditioning process and provides the supply air at the required temperature and humidity to maintain the indoor conditions at the average ASHRAE comfort temperature (i.e., 24°C (75.2°F) in summer) (ASHRAE 2004b). The cooling system may sensibly cool the supply air if it is dry enough to provide a satisfactory indoor humidity, but dehumidification is provided to prevent indoor humidity ratios above 12 g/kg (0.012 lb/lb) (about 64% RH at specified indoor temperature).

The heating system consists of radiators that operate with hot water (natural convection) and are installed inside the building. The radiant heating system mainly addresses the building loads and maintains an indoor temperature of 22°C (71.6°C) in the winter (ASHRAE 2004b). Outdoor ventilation air is provided when the building is occupied and the RAMEE system along with an auxiliary heating system heats the ventilation air up to

14°C (57.2°F) to prevent thermal discomfort. During unoccupied hours, no ventilation air is provided, and the radiant heating system does not operate unless the indoor temperature falls below 15°C (59°F).

The outdoor ventilation rate is maintained at the minimum standard requirement (i.e., 0.5 ACH) when the building is occupied, unless economized cooling is available. During economizer operation, the outdoor ventilation rate can increase up to 4 ACH. The ventilation rate is reduced to 50% and 25% of the design flow rate on Saturdays and Sundays due to lower occupancy, respectively.

3.6.3. Climatic Conditions

The described office building is studied in Saskatoon (Saskatchewan, cold-dry climate), Chicago (Illinois, cool-humid climate), Miami (Florida, hot-humid climate) and Phoenix (Arizona, hot-dry climate) as the four North American cities that represent different climate zones (Briggs et al. 2003). Figure 3.6 shows the yearly distribution of outdoor conditions for each location in three main regions on the psychrometric chart; i.e., Region 1 includes low outdoor temperatures when heating is required (i.e., the HVAC system is in heating mode), region 2 includes outdoor conditions when economized cooling is available (lower temperature and lower enthalpy than the indoor), and region 3 includes high temperature and humidity outdoor conditions where cooling and possibly dehumidification is required. The pie graph associated with each building location presents the fraction of a year that the HVAC system operates in each specific region. Typical Meteorological Year (TMY 2 weather data format) (Marion and Urban 1995) which contains typical hourly weather data required for yearly building energy analysis is used for this study.

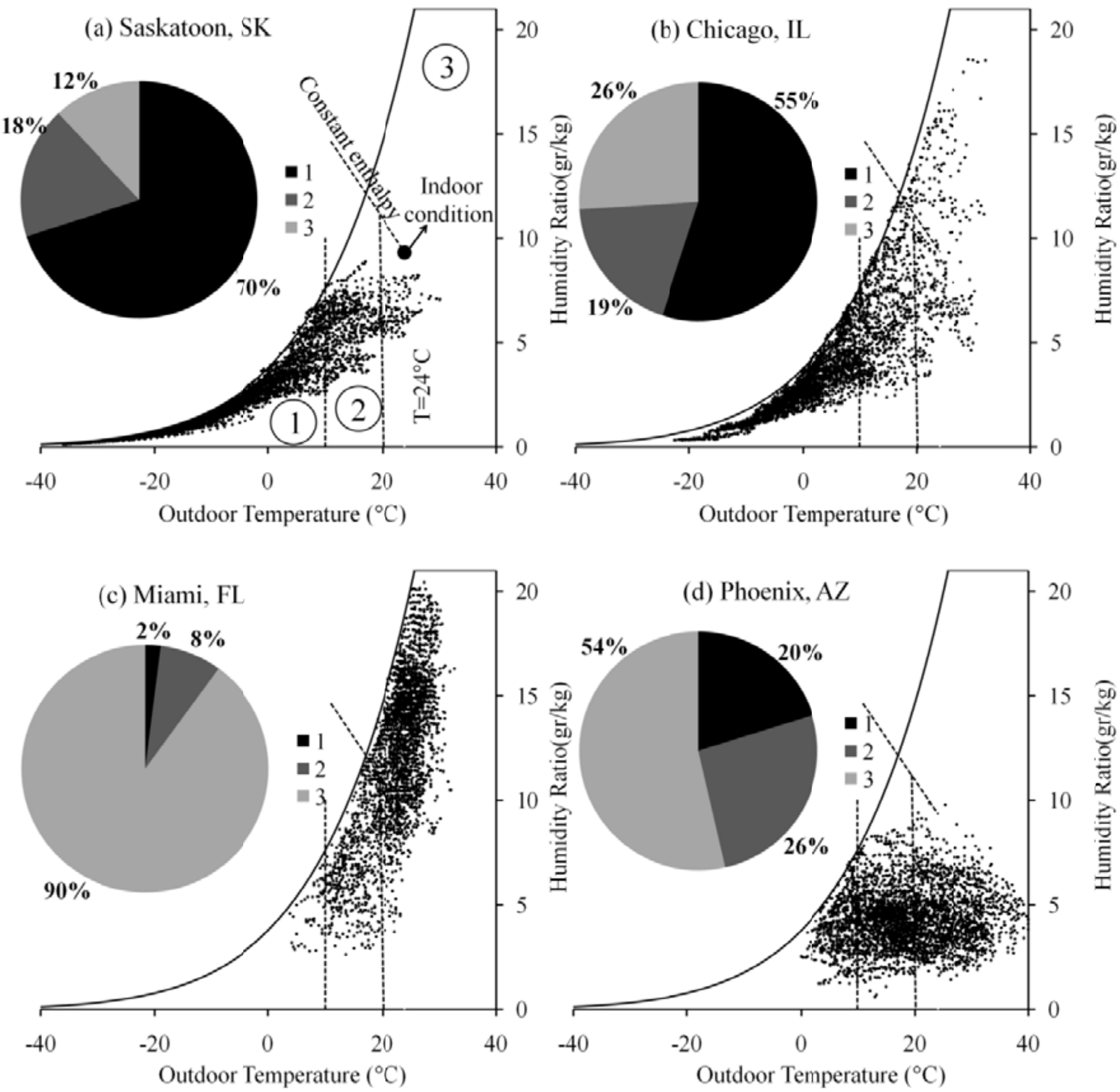


Figure 3.6 TMY2 yearly distribution of hourly outdoor conditions and HVAC system operation when heating is required (1), economized cooling is available (2) and cooling is required (3) in (a) Saskatoon, (b) Chicago, (c) Miami and (d) Phoenix

3.6.4. Simulation Program

The numerical solution of heat and mass transfer in the RAMEE system for steady-state and balanced air flow rates was developed in previous research (Fan et al. 2005; Vali et al. 2009; Hemingson et al. 2010). Akbari et al. (2010) developed an Artificial Neural Network (ANN) that is able to predict the RAMEE performance. The neural network was subjected to direct pattern search optimization algorithm that is able to find the optimal operating Cr* at any given condition. The thermal system (including the HVAC system, RAMEE and the building) is simulated using the TRNSYS building energy simulation tool (Klein 2000) equipped with the Second version of TESS libraries (Thornton et al. 2009) working in conjunction with MATLAB programming language (MATLAB 2006). Figure 3.7 schematically shows the dataflow between the TRNSYS model and the ANN.

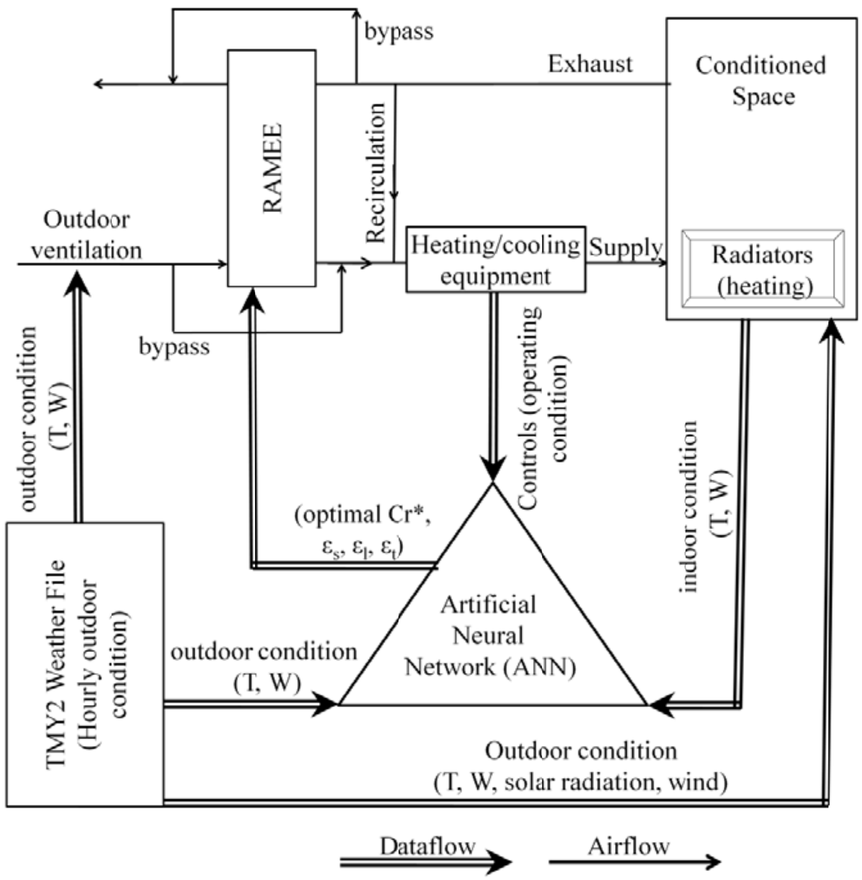


Figure 3.7 Schematics of the dataflow between the TRNSYS model and the ANN

At any specific hour, the TRNSYS simulation gives the hourly building loads based on internal loads, infiltration rate and outdoor condition (temperature, humidity, solar radiation, wind, etc.). Assuming that the RAMEE system is not employed, the condition and flow rate of the supply air to the conditioned space that meets the space loads, minimum ventilation requirement, and the indoor comfort conditions, and the hourly heating/cooling loads are calculated. Based on the indoor and outdoor conditions the TRNSYS model (and the assumed NTU of 10), the ANN predicts the optimal Cr* and the sensible and latent effectiveness associated with such optimal Cr*. The sensible and latent effectiveness are input to the TRNSYS model of the RAMEE system. The operation of RAMEE system under specified effectivenesses preconditions the outdoor ventilation air and reduces the heating/cooling loads. It should be noted that the operation of RAMEE may slightly change the indoor condition compared to the base case. Such a change in indoor condition can affect the system effectiveness and requires iterations to determine the modified system effectiveness based on new indoor conditions. Iterations between the TRNSYS and ANN models are not conducted here, because typical variations in indoor conditions may change the RAMEE effectiveness by less than 0.3% (Hemingson et al. 2010).

3.7.Results and Discussions

In this section, the TRNSYS simulation results of the RAMEE employed in different climates are presented. The results mainly focus on the impact of the RAMEE on annual energy consumption and equipment sizes for both heating and cooling seasons at each location. As mentioned before, the ANN predicts the hourly optimal Cr* at which the RAMEE system should operate to have the peak performance. The optimal Cr* varies from hour to hour as the outdoor (and possibly indoor) conditions change. Figure 3.8 shows the hourly values of optimal Cr* during one year in each location.

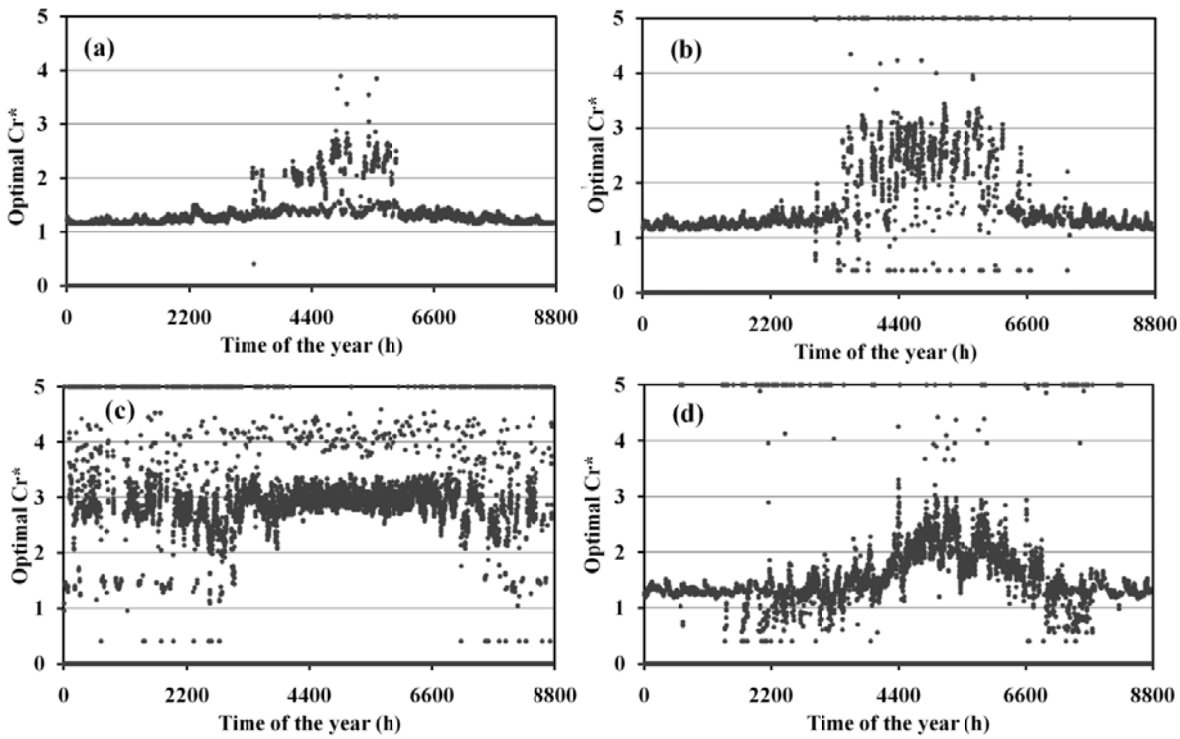


Figure 3.8 Yearly variation of hourly optimal Cr* values for different climatic conditions, (a) Saskatoon, (b) Chicago, (c) Miami, and (d) Phoenix

As is shown in Figure 3.8, the optimal Cr* (Cr^*_{opt}) is higher in the summer than in the winter. For cold climates (Saskatoon and Chicago), the average Cr^*_{opt} is close to 1.2, where for Miami as representative of hot and humid climate, the optimal hourly Cr* is close to 3 for most of the year. As shown in Equation (3-6), Cr* is a function of ventilation air and salt solution flow rates. Having the ventilation rate set at the minimum ASHRAE requirement, the solution flow rate has to be controlled to achieve the optimal Cr*. In the next sections, the annual cooling and heating energy saved due to the use of the RAMEE when operating under hourly optimal Cr* is presented.

3.7.1. Heating Season

The results for annual heating saving and reduction in the size of heating system when the RAMEE is operating under hourly optimal Cr* (i.e., the Cr* that gives the maximum sensible effectiveness) are presented in Figure 3.9.

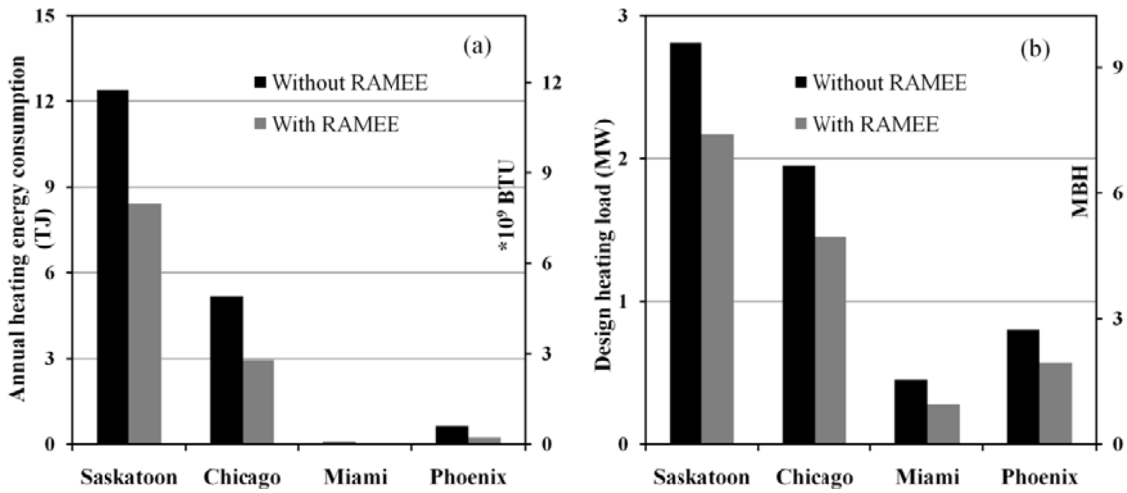


Figure 3.9 Impact of the RAMEE on (a) annual heating energy consumption and (b) the size of heating equipment

The simulation results presented in Figure 3.9, indicate that the operation of the RAMEE under optimal Cr* leads to 32%, 43%, 74% and 63% annual heating energy saving in Saskatoon, Chicago, Miami and Phoenix, respectively. The size of heating equipment is also reduced by 23%, 26%, 38% and 29% in Saskatoon, Chicago, Miami and Phoenix, respectively. The results obtained from a series of TRNSYS simulations of constant effectiveness ERVs indicated that such savings could be achieved if a constant effectiveness ERV with sensible effectiveness of about 77% was employed in the same building during the heating season.

3.7.2. Cooling Season

The results from the TRNSYS simulation of the RAMEE operating in the office building during the cooling season are presented in Figure 3.10. The results show that the RAMEE with economizer reduces the annual cooling energy by 39%, 21%, 8% and 15% in Saskatoon, Chicago, Miami and Phoenix, respectively. The cooling energy saved in Saskatoon (cold climate) is mostly due to the presence of economizer, which saves about 30% of the cooling energy, rather than the RAMEE itself, which saves about 9% of annual cooling energy. This is because Saskatoon represents a cold climate and free cooling is available for a majority of the time in cooling season (Figure 3.6). On the other hand, the savings with the RAMEE account for the majority of cooling energy saved in Miami and the RAMEE system alone reduces the cooling energy by 7%, and adding an economizer results in an additional 1% energy saving. The size of the cooling equipment is reduced by 5% in Saskatoon and Phoenix, and by 10% in Miami and remains unchanged in Chicago.

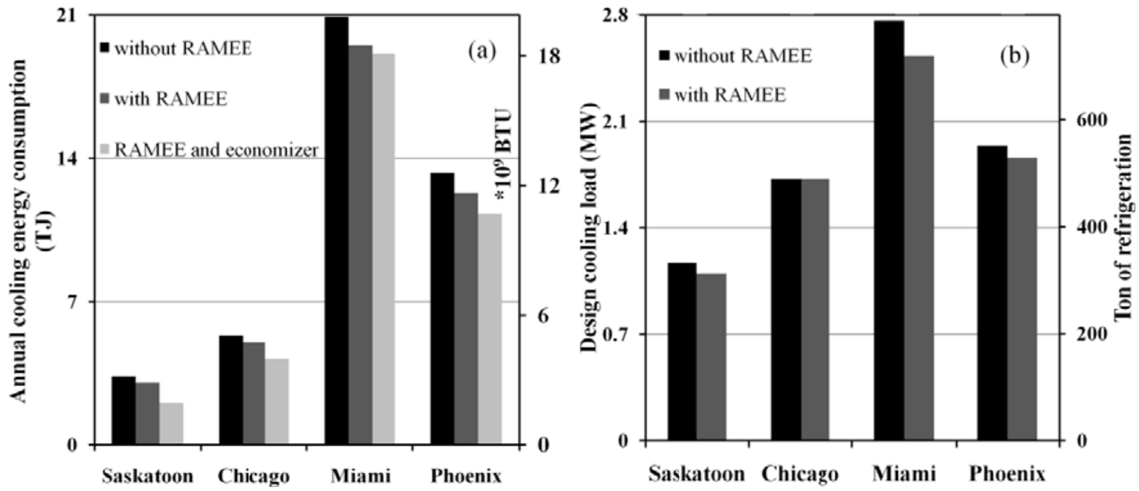


Figure 3.10 Impact of the RAMEE system on (a) annual cooling energy consumption and (b) the size of cooling equipment

It should be noted that Commercial Building Energy Consumption Survey (CBECS) has reported an average energy intensity of 533 MJ/m^2.year (46.9 Thousand BTU/ft^2.year) for HVAC system energy consumption in US office buildings (EIA 2003). In this research, the energy intensity of the studied office building varies depending on the climate and for the base case it is found 555, 300, 304 and 237 MJ/m^2.year (48.8, 26.4, 26.7 and 20.9 Thousand BTU/ft^2.year) for Saskatoon, Chicago, Miami and Phoenix, respectively. These results are lower than the CBECS average value except for Saskatoon. This could be due to the fact that existing buildings may have equipments operating at lower efficiencies compared to the high-efficient heating and cooling equipments used in this paper (i.e., boiler with 88% nominal combustion efficiency, cooling unit with COP of 3 and fans of 60% efficiency). In addition, as mentioned previously, the building envelope was improved by using double pane glasses (instead of single pane glasses that are used in the original building) and adding 10 cm (4 in) and 15 cm (6 in) of insulation to walls and roof, respectively. Having the RAMEE and an economizer employed in the office building, the total energy intensity was reduced by 30%, 32%, 5% and 12% in Saskatoon, Chicago, Miami and Phoenix, respectively.

3.8.Control Based on Average Cr* Values

For any specific outdoor condition, the implementation of optimal Cr* requires an accurate control of salt solution flow rate to achieve the desired Cr* value. As shown in Figure 3.8, a scatter variation of optimal hourly Cr* between 1 and 5 is observed; however, the optimal Cr* stays fairly constant during each season. For example in Chicago, the optimal Cr* fluctuates around an average value of 1.2 during the winter and increases to about 2.4 during the summer. Therefore, it may be possible to use a constant salt solution flow rate (Cr* value) during each season (or during the entire year) rather than having the Cr* value change every hour. Table 3.1 shows the seasonal and yearly weighted averaged values of Cr* for each location for the office building and its associated standard deviation. The standard deviation is higher for cooling season as the optimal Cr* has a more scatter variation with Cr* in summer (shown in Figure 3.8). The weighted average Cr* is defined as:

$$Cr^*_{ave} = \frac{\sum_{i=1}^{8760} Cr^*_{opt,i} \cdot Q_i}{\sum_{i=1}^{8760} \cdot Q_i} \tag{3-13}$$

Where: $Cr^*_{opt,i}$ and Q_i are the optimal Cr* and energy transfer via the RAMEE system (positive values for both heating and cooling) at i^{th} hour, respectively.

When employing the seasonal average Cr* value, the Cr* switches between the heating and cooling set-points according to the season. But, with the yearly average value, the RAMEE system operates with constant Cr* throughout the year.

Table 3.1 Seasonal and yearly weighted average Cr* and associated standard deviation for the office building in each location

	Seasonal average Cr*		**Yearly average Cr***
	Winter (heating)	Summer (cooling)	Heating and cooling
Saskatoon	1.21±0.05	2.19±0.17	1.22±0.29
Chicago	1.24±0.05	2.41±0.31	1.30±0.46
Miami	1.43±0.01	2.91±0.38	2.90±0.41
Phoenix	1.29±0.02	1.76±0.51	1.62±0.54

Table 3.2 presents the annual cooling and heating energy savings when the RAMEE system operates under specified average Cr* values. In order to highlight the effect of implementing average Cr* values on RAMEE savings, the energy savings with economizer are not included in the cooling savings.

Table 3.2 Annual energy saved with the RAMEE system operating with selected average Cr* values

	Annual heating energy saved			**Annual cooling energy saved**		
	Optimal Cr*	Seasonal Cr*	Yearly Cr*	Optimal Cr*	Seasonal Cr*	Yearly Cr*
Saskatoon	32%	32%	32%	9%	9%	8%
Chicago	43%	43%	43%	6%	6%	5%
Miami	74%	74%	67%	7%	7%	7%
Phoenix	63%	62%	61%	8%	8%	7%

Based on the results obtained from the TRNSYS simulation of the studied office building (Table 3.2), the annual cooling and heating energy savings are nearly the same whether hourly or average Cr* values are used. Such an insignificant change in annual energy savings can be explained by considering the behavior of the RAMEE effectiveness as a function of Cr* presented in Figure 3.3. As shown in the figure, changing the Cr* around the optimal value does not influence the RAMEE effectiveness significantly (sensible effectiveness in the winter and total effectiveness in the summer). For instance, for typical summer conditions presented in Figures 3.3c and 3.3e, the total effectiveness is fairly constant for Cr* values ranging from 2 to 3. Therefore, applying an average Cr* value instead of the hourly optimal value does not reduce the total effectiveness and consequently the cooling energy saved significantly. As an advantage of implementing yearly average Cr* value, there is no need to vary the salt solution flow rate as seasons change; however, a negligible reduction in annual savings is observed compared to seasonal average Cr* approach.

3.9. Life Cycle Cost Analysis (LCCA)

A Life Cycle Cost Analysis (LCCA) of the RAMEE system is performed to study the system from an economic point of view. The LCCA is carried out for three different alternatives; i.e., the base case where the VAV HVAC system is not equipped with an economizer or ERV, the second alternative that is the VAV HVAC system equipped with the RAMEE, and a case where the HVAC system is equipped with an economizer and the RAMEE. The LCCA is carried out over a 15-year life cycle and the present value method (all expenses converted to the present equivalent value) is used. The LCCA includes capital costs and operation costs. The capital costs (or investment costs) include all the expenses before the project begins to operate and includes the cost of heating and cooling equipment, supply and exhaust fans and the RAMEE. The operational costs are defined as all the expenses that occur during the operation of the system throughout its life cycle and include the energy costs to run the HVAC equipments. The main assumptions for this LCCA approach are: no demolition cost or residual value for the alternative systems, and no extra cost for the maintenance of the RAMEE system. RSMeans Mechanical Cost Data (Mossman et al. 2010) that includes the required information about HVAC system equipment cost is used to estimate the investment costs. Also, the local energy prices in each city are used to calculate the operational costs.

A gas-fired boiler with nominal efficiency of 88% is selected as the heating unit (to satisfies the minimum combustion efficiency of 80% required by ASHRAE standard 90.1: ASHRAE 2004c). RSMeans Mechanical Cost Data (Mossman et al. 2010) suggests an average investment cost of about \$68.3/KW (\$20/MBH) for cast-iron gas-fired boilers operating in the range of power outputs required for the studied building. An air-cooled air conditioning unit with coefficient of performance (COP) of 3 is selected as the cooling unit (to satisfies ASHRAE standard 90.1 minimum requirement of 2.78 COP; ASHRAE 2004c). The capital cost of the cooling unit based on RSMeans Mechanical Cost Data (Mossman et al. 2010) for direct-expansion water chillers is considered to be on average 171\$/KW (\$600/ton). Centrifugal type HVAC fans that cost \$851/m^3/s (\$0.4/CFM) are used for the LCCA in this study. RSMeans Mechanical Cost Data (Mossman et al. 2010) estimates an investment cost of about \$1.5/CFM for energy wheels, however, technical papers in the field of air-to-air energy recovery ventilators (e.g., Besant and Simonson 2000; Turpin 2000) have expected the manufacturing cost of an ERV as high as \$5/CFM. In this paper, the investment cost of the RAMEE is considered \$3/CFM.

Table 3.3 compares the capital costs for different alternatives. It should be noted that the addition of an economizer to an HVAC system does not change the design heating load. Also, the design cooling load occurs at high temperature outdoor conditions that is out of the economizer's operating range; therefore, the design cooling load remains unchanged when an economizer is employed. The capacity of supply and exhaust fans is similar for all three alternatives. Therefore, the investment cost of RAMEE is similar to the case which RAMEE works with an economizer. In Tables 3.3 and 3.4, for simplification, Alt. 1 refers to the base case HVAC system that is not equipped with a RAMEE, Alt. 2 refers to the HVAC system equipped with a RAMEE and Alt. 3 refers to the HVAC system equipped with a RAMEE and an economizer. Table 3.3 Summary of equipment capacity and HVAC equipment costs for the selected office building

Table 3.3 Summary of equipment capacity and HVAC equipment costs for three system alternatives for the selected office building

		Saskatoon		Chicago		Miami		Phoenix	
		Alt. 1	**Alt. 2 Alt. 3**	**Alt. 1**	**Alt. 2 Alt. 3**	**Alt. 1**	**Alt. 2 Alt. 3**	**Alt. 1**	**Alt. 2 Alt. 3**
Equipment size	Heating system, KW	2814	2169	1948	1453	449	279	799	569
	Cooling system, KW	1168	1104	1720	1720	2757	2532	1941	1857
	Fan capacity, m^3/s	90	90	95	95	96	96	127	127
Equipment cost	Heating system, Thousand $US	192.2	148.1	133	99.2	30.7	19.1	54.6	38.9
	Cooling system, Thousand $US	199.2	188.4	293.4	293.4	470.4	432	331.2	316.8
	Cost of fans, Thousand $US	76	76	80.4	80.4	81.6	81.6	108	108
	Cost of RAMEE, Thousand $US	0	72	0	72	0	72	0	72
	Total investment, Thousand $US ($US/m^2)	**467.4 (16.2)**	**484.5 (16.8)**	**506.8 (17.6)**	**545.0 (18.9)**	**582.7 (20.2)**	**604.7 (21.0)**	**493.8 (17.1)**	**535.7 (18.6)**

Table 3.4 shows the comparison of three alternatives in operational costs of heating and cooling equipment and the fan energy consumption excluding the pressure drop across the RAMEE system. The fan power is a function of air flow rate, the pressure drop in the supply and exhaust ducting and the fan efficiency. The pressure drop across the ducting system and fan efficiency are assumed to be 4 in. water and 60%, respectively.

Table 3.4 Summary of annual energy consumption and energy cost of different alternatives excluding the fan energy consumption due to the pressure drop in the RAMEE

	Saskatoon			**Chicago**			**Miami**			**Phoenix**		
	Alt. 1	**Alt. 2**	**Alt. 3**	**Alt. 1**	**Alt. 2**	**Alt. 3**	**Alt. 1**	**Alt. 2**	**Alt. 3**	**Alt. 1**	**Alt. 2**	**Alt. 3**
Heating energy (TJ/year)	14.1	9.6	9.6	5.9	3.3	3.3	0.1	0.02	0.02	0.7	0.3	0.3
Cooling energy (TJ/year)	1.1	1.0	0.7	1.8	1.7	1.4	7.0	6.5	6.4	4.4	4.1	3.8
Fans (TJ/year)	0.80	0.80	0.89	0.99	0.99	1.08	1.74	1.74	1.78	1.76	1.76	1.86
Natural gas, Thousand m^3/year	373	253	253	156	88	88	2	0.6	0.6	19	7	7
Natural gas, Thousand $US/year	21.4	14.9	14.9	39.8	23.6	23.6	0.6	0.5	0.5	6.8	2.8	2.8
Electricity, TJ/year	1.92	1.81	1.58	2.77	2.66	2.48	8.71	8.24	8.15	6.19	5.86	5.63
Electricity, Thousand $US/year	73.9	69.9	66.0	154.5	148.2	147.8	103.3	98.2	97.9	109	103.2	100.7
Total energy cost, $US/year	**95.3**	**84.8**	**80.9**	**194.3**	**171.8**	**171.4**	**103.9**	**98.7**	**98.4**	**115.8**	**106.0**	**103.5**

Although the RAMEE system reduces the energy consumption of heating and cooling equipment, it imposes an extra pressure drop that increases the energy consumed by the fan(s). Therefore, the life cycle cost of the RAMEE system will be dependent upon the pressure drop across the exchangers. Figure 3.11 summarizes the LCCA for three alternatives in different locations as a function of pressure drop across each LAMEE. As expected, the greater the pressure drop across the exchangers, the higher the life cycle cost.

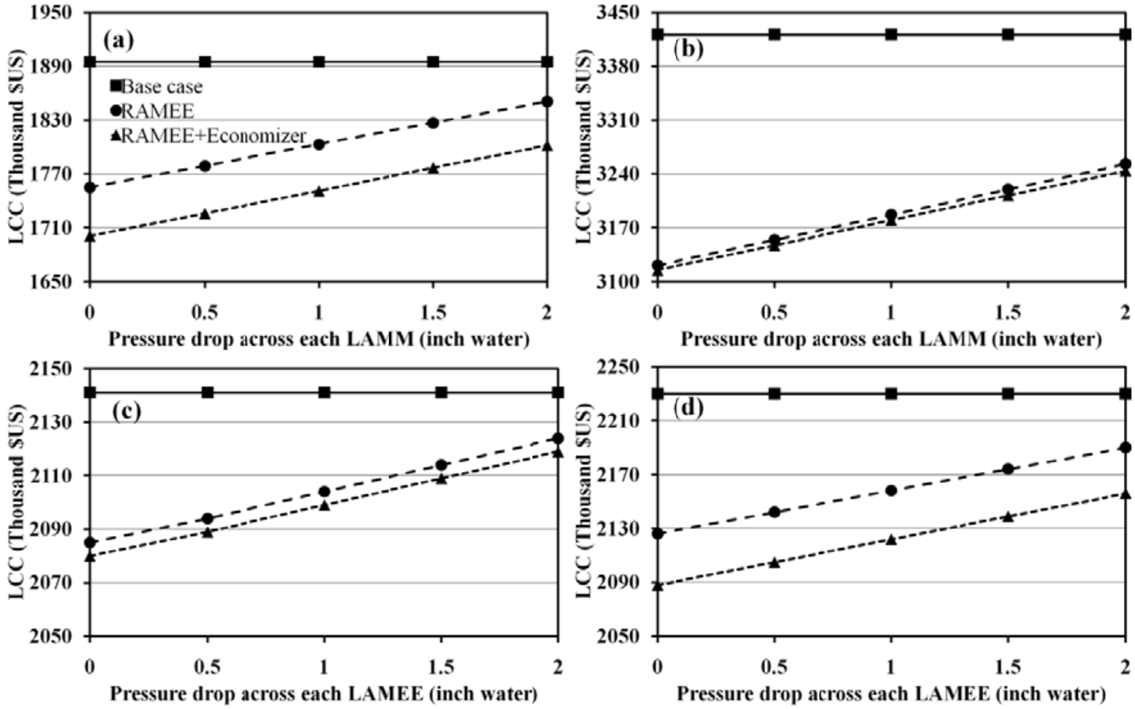


Figure 3.11 LCC of the three alternative systems as a function of pressure drop across the RAMEE system in (a) Saskatoon, (b) Chicago, (c) Miami and (d) Phoenix

Payback Period (PBP) is a measure to determine the amount of time it takes the consumer to recover the extra investment cost to purchase the high-efficient alternative as a result of lower operation cost (Rosenquist et al. 2004). The PBP, as defined in Equation (3-14), is the ratio of extra investment cost to purchase the more efficient option to the decrease in annual operation costs.

$$PBP = \frac{\Delta IC}{\Delta OC} \tag{3-14}$$

Where, IC and OC stand for investment costs and operational costs, respectively. The PBP of employing alternative 3 (i.e., the RAMEE system along with an economizer) in different locations, assuming a total pressure drop of 0.8 in. water across each exchanger (as expected by the manufacturer) is presented in Table 3.5.

Table 3.5 Payback period of RAMEE and economizer in different locations

	Saskatoon	**Chicago**	**Miami**	**Phoenix**
PBP(years)	1.8	2.0	4.8	4.0

3.10. Conclusions

The operation of a Run-Around Membrane Energy Exchanger (RAMEE) that is able to transfer heat and moisture between outdoor ventilation and building exhaust air is described in this paper. The RAMEE control varies depending on outdoor condition and whether the building needs heating or cooling. When the HVAC system is on heating mode, the RAMEE operates with maximum sensible effectiveness. However, a fraction of ventilation air should be bypassed if the full-load operation at maximum sensible effectiveness overheats the outdoor air (also called part-load operation). When the HVAC system is in the cooling mode, the RAMEE should operate with maximum total effectiveness. Using an Artificial Neural Network (ANN) that is trained based on a numerical solution of heat and moisture transfer in the RAMEE, the optimal system performance (optimal hourly Cr* and associated sensible and latent effectiveness) is predicted when the RAMEE system operates in a 10-storey office building. This building represents 3.34% of US office building stock, and is simulated using the TRNSYS computer program in four different North American locations representing major climatic conditions; i.e., Saskatoon (cold and dry), Chicago (cool and humid), Phoenix (hot and dry) and Miami (hot and humid). The simulation results showed 32% and 43% annual heating energy saving in Saskatoon and Chicago as representatives of cold climate. During the cooling season, the RAMEE operates under maximum absolute total effectiveness (to maximize the reduction of outdoor air enthalpy) and results in about 8% and 15% cooling energy saving when it operates along with an economizer in Miami and Phoenix as hot climates. Since the application of hourly optimal Cr* requires an accurate control of the salt solution flow rate and causes a transient response, the impact of applying average seasonal and yearly Cr* was studied. The results show that operating the system under seasonal average Cr* (i.e., constant salt solution flow rate throughout each season) that switches between cooling and heating season set points has a minimal impact on energy savings. The life cycle cost analysis showed that the pressure drop across the exchangers plays an important role in payback of the RAMEE system. Based on manufacturer's estimation on RAMEE's pressure drop, the payback period of the RAMEE system was found to be about 2 years in cold climates and 4 to 5 years in hot climates.

CHAPTER 4

APPLICATION OF A RAMEE IN A HEALTH-CARE FACILITY HVAC SYSTEM

4.1. Overview of Chapter 4

In this chapter, the impact of a RAMEE on a health-care facility HVAC system is investigated as the second RAMEE case study. This chapter is the fourth manuscript (energetic, economics and environmental analysis of a health-care facility HVAC system equipped with a run-around membrane energy exchanger) that is submitted for peer review and has a similar organization as manuscript #3. This chapter begins with an overview of previous research on RAMEE (sections 4.3 and 4.5.1 and 4.5.2), followed by a summary of Chapter 2 on controlling the RAMEE in different conditions (section 4.5.2). The reader may skip these sections if they have already obtained the necessary information about the RAMEE in the previous chapter.

The RAMEE operates under the optimal control of ERVs (described in Chapter 2), and a neural network is used to predict the system effectiveness for optimal operation. The hospital building, the RAMEE and the HVAC system are simulated in TRNSYS, and the results on the impact of RAMEE on (a) cooling and heating energy consumption, (b) size of HVAC equipment, (c) life-cycle cost of the HVAC system and (d) emission of greenhouse gases is presented. The results for the two case studies (i.e. the office building described in Chapter 3 and the hospital described in this chapter) are compared in section 4.6.5.

My contribution in this research was to (a) simulate the RAMEE, the hospital building and the HVAC system in TRNSYS and (b) post-process the results and write the paper. The neural network developed by S. Akbari was used to predict RAMEE’s optimal operating condition.

Manuscript #4: Energetic, economics and environmental analysis of a health-care facility hvac system equipped with a run-around membrane energy exchanger

M. Rasouli, S. Akbari, C.J. Simonson and R.W. Besant

4.2. Abstract

Run-Around Membrane Energy Exchanger (RAMEE) is a novel heat and moisture recovery system that consists of two separate supply and exhaust exchangers coupled with an aqueous salt solution flow. The salt solution transfers energy (heat and moisture) in a closed loop between outdoor ventilation air and the exhaust air from buildings. The system performance is a function of the flow rate of the salt solution and ventilation air and the outdoor air conditions. The dependency of system performance on the solution flow rate and the outdoor conditions requires adjustment of the appropriate flow rate which gives the optimal system performance at any specific outdoor condition. In this paper, the RAMEE is simulated for a hospital building in four different climates using TRNSYS and MATLAB computer programs. The steady-state RAMEE can reduce the annual heating energy by 60% in cold climates and annual cooling energy by 15% to 20% in hot climates. The RAMEE has an immediate payback in cold climates and a 1 to 3-year payback in hot climates depending on the pressure drop across the exchangers. Finally, the RAMEE reduces greenhouse gas emission (CO_2- equivalent) by 25% and 10% in cold climates and hot climates, respectively.

4.3. Introduction

Energy Recovery Ventilators (ERVs) have been widely used to reduce the energy required to condition the ventilation air. ERVs transfer heat (heat recovery systems) or heat and moisture (energy recovery systems) between conditioned exhaust air and outdoor ventilation air. Heat pipes, fixed-plate heat exchangers and heat wheels are examples of the heat recovery systems, and energy wheels coated with desiccant (Simonson and Besant 1997) and flat-plate exchangers made of water permeable membranes (Zhang and Jiang 1999) are examples of energy recovery systems. The main disadvantage of present ERVs is that some are unable to transfer moisture. Also, they all require a side-by-side installation of the supply and exhaust ducts. This may impose a higher ducting cost for adjacent installation of the supply and exhaust ducts. Adjacent air inlet and exhaust increases the probability of contaminant transfer from exhaust air to the supply air, especially for polluted spaces (e.g., some laboratories) and highly-sensitive areas (e.g., surgery room).

A novel Run-Around Membrane Energy Exchanger (RAMEE) that consists of two separate supply and exhaust exchangers was presented by Fan et al. (2005). For this system, each exchanger is a flat-plate energy exchanger constructed with water vapor permeable membranes that allow the transfer of heat and water vapor. Such a system is suitable for retrofitting buildings even where the supply and exhaust ducts are not adjacent. Research has been done on (a) developing numerical models of the RAMEE (Seyed-Ahmadi et al. 2009a and 2009b; Vali et al. 2009; Hemingson 2010), (b) predicting the system performance at different conditions using an artificial neural network (Akbari 2010) (c) investigating the crystallization risk of the salt solution (Afshin et al. 2010) and (d) obtaining experimental data on RAMEE performance for two prototypes (Mahmud et al. 2009; Erb et al. 2009).

ASHRAE Standard 170-2008 (ASHRAE 2008), ventilation of health-care facilities, has recommended much higher rates of outdoor air flow compared to ASHRAE 62-2010 (ASHRAE 2010) for ventilation rates of other types of buildings. For example, a typical office building may require about 0.5 ACH ventilation air (Rasouli et al. 2010a), while a minimum outdoor air change of 2 to 6 ACH is recommended for health-care facilities. The energy consumption due to conditioning of ventilation air increases as the ventilation rate increases (McDowell et al. 2003; Brandemuehl and Braun 1999; Omre 2001). For instance, McDowell et al. (2003) showed that, without energy recovery, increasing the ventilation rate of a building in Washington D.C. from 0 to 10 l/s.person (corresponding to about 0.37 ACH) increases the annual energy consumption of the HVAC system by 14%. This result is in a good agreement with Commercial Building Energy Consumption Survey (CBECS) in 2003 (EIA 2003) that reported that health-care facilities were the second highest energy-intense commercial buildings with 1472 MJ/ m^2.year HVAC system energy consumption. This is 2.8 times higher than the average HVAC energy consumption in US office buildings (i.e., 533 MJ/m^2.year) (EIA 2003). Although the ventilation energy is very significant in hospitals, most of the recent research has focused on energy-saving technologies in office spaces, residential buildings and educational facilities. Rasouli et al. (2010c) studied the application of a RAMEE in an office building HVAC system. The TRNSYS simulation of the RAMEE showed savings of about 30 to 40% for heating energy in cold climates (Saskatoon and Chicago) and 8 to 15% for cooling energy in hot climates (Miami and Phoenix). This paper presents the energy saving with a RAMEE for a hospital building (as the second case study of the RAMEE). An overview of the RAMEE is presented and the findings of Rasouli et al. (2010c) regarding the control and operation of the RAMEE are implemented when it operates in a hospital building. This paper presents the energy savings, Life-Cycle Cost (LCC) analysis and Life Cycle Environmental Assessment (LCEA) of the RAMEE in the hospital over a 15-year life-cycle for four different climates.

4.4.Model Description

A 3-storey hospital with total floor area of 3150 m^2 is chosen for this study. The thermal resistances of walls, roof and the floor are 2.72, 3.64 and 3.45 (m^2.K/W), respectively. The building has double-glazed windows, about 31 (W/m^2) of internal heat gains (includes lighting, cooking and equipment loads based on CBECS data, EIA 2003) and an occupant density of 5 People/100 m^2. A variable air volume HVAC system is considered for the building that maintains the indoor temperature within ASHRAE comfort zone (i.e., 24°C in summer and 22°C in winter, ASHRAE 2004a), and the indoor humidity below 60%

RH. The day-time (6:00-22:00) ventilation rate is set at 2 ACH as an average rate recommended by ASHRAE ventilation standard for different spaces in health-care facilities (ASHRAE 2008) and is reduced to 1.3 ACH for the rest of the day (22:00-6:00) when a lower occupancy is expected. A total air change rate of 3 times the ventilation rate is always maintained for the space (as recommended by ASHRAE for most of health-care spaces, ASHRAE 2008).

The building is simulated in Saskatoon (Saskatchewan, Canada), Chicago (Illinois), Miami (Florida) and Phoenix (Arizona) as the four North American cities which represent different climatic conditions. Chicago, Miami and Phoenix are chosen as representatives of cool-humid, hot-humid and hot-dry climates, respectively, based on Briggs et al. (2003) climatic classifications for building energy analysis. Saskatoon is chosen to represent a cold climate because heating is required for a large fraction of a year (Rasouli et al. 2010c).

4.5. Run-Around Membrane Energy Exchanger (RAMEE)

4.5.1. Overview

Figure 4.1 schematically presents a HVAC system equipped with a RAMEE.

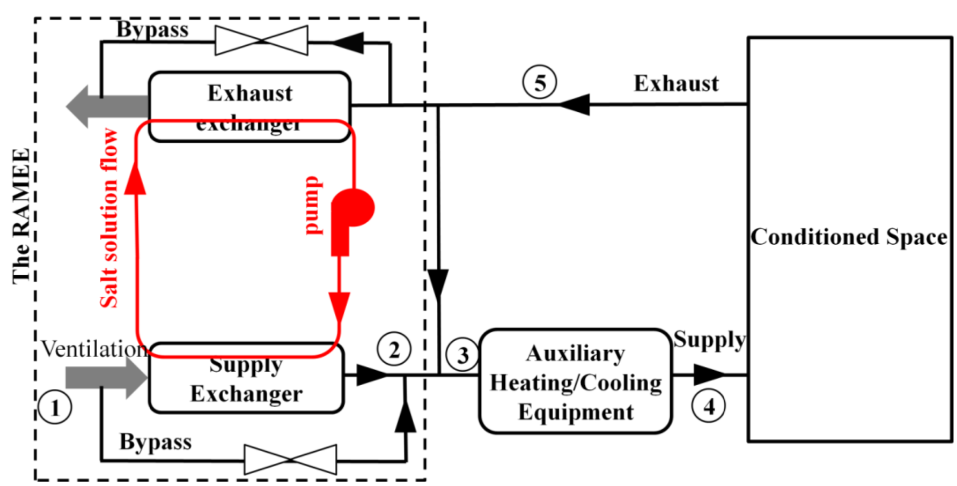


Figure 4.1 Schematic view of a HVAC system equipped with a RAMEE

The RAMEE shown in Figure 4.1 consists of two separate exchangers located in supply and exhaust ducts. Each exchanger is a flat-plate, liquid-to-air membrane energy exchanger (LAMEE) that is made using water vapor permeable membranes. The LAMEEs are coupled with an aqueous salt solution that is pumped in a closed loop and transfers both heat and moisture between the exhaust and ventilation airstreams. Such a design has the capability of transferring both heat and moisture in new and retrofit applications where the ducts are not adjacent.

During the winter, the mixture of outdoor ventilation air and the return air is heated by the heating system up to the desired supply temperature. In the absence of the RAMEE, the ventilation air temperature is equal to the outdoor temperature. But, the RAMEE transfers energy (heat and moisture) from the exhaust air to the supply air. Such an energy transfer increases the ventilation air temperature and consequently lowers the energy consumption of the heating system. During the summer, the mixture of outdoor ventilation air and the return is cooled and also dehumidified if the humidity of the mixture (state 3) is unable to maintain the indoor humidity within comfort zone (i.e., below 60% RH; ASHRAE 2004a). The operation of the RAMEE in summer transfers heat and moisture from warm-humid outdoor air to the cool-dry exhaust air. This reduces the enthalpy of the ventilation air and consequently decreases the cooling energy for the auxiliary cooling system. The air and salt solution can flow in counter flow, cross flow or counter/cross flow arrangements through each LAMEE. A counter flow RAMEE is studied in this paper.

4.5.2. System Performance, Controls and Operation

The effectiveness of a RAMEE for transferring heat (ε_s), moisture (ε_l) and enthalpy (ε_t) is mainly a function of three dimensionless groups defined in Equations (4-4)-(4-6), indoor and outdoor air conditions and the air/salt solution flow arrangement. Figure 4.2

illustrates the dependency of RAMEE effectiveness on NTU, Cr* and outdoor conditions for some specific conditions.

$$\varepsilon_s = \frac{T_1 - T_2}{T_1 - T_5} \tag{4-1}$$

$$\varepsilon_l = \frac{W_1 - W_2}{W_1 - W_5} \tag{4-2}$$

$$\varepsilon_t = \frac{h_1 - h_2}{h_1 - h_5} \tag{4-3}$$

$$NTU = \frac{UA}{\dot{m}_{air} C_{p,air}} \tag{4-4}$$

$$NTU_m = \frac{U^{'}A}{\dot{m}_{air}} \tag{4-5}$$

$$Cr^* = \frac{\dot{m}_{sol} C_{p,sol}}{\dot{m}_{air} C_{p,air}} \tag{4-6}$$

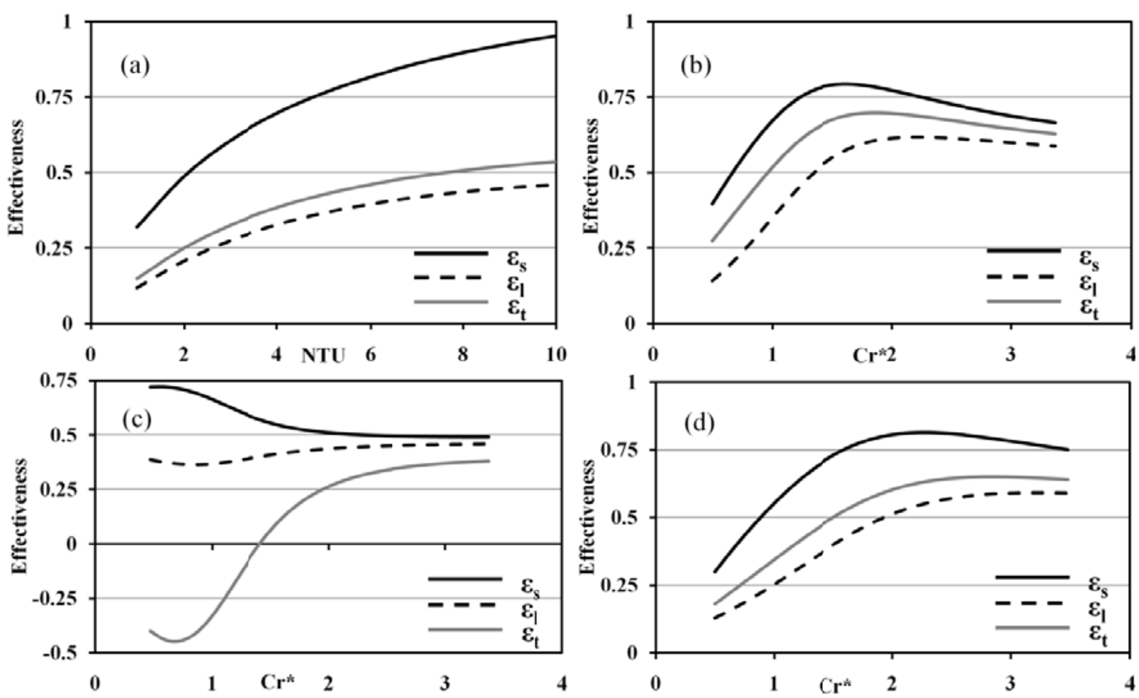


Figure 4.2 RAMEE effectiveness (a) as a function of NTU at Cr*=2, and as a function of Cr* and outdoor air conditions at (b) cold, (c) hot-low enthalpy and (d) hot-high enthalpy outdoor conditions with NTU=10

Similar to the other types of ERVs, NTU is directly proportional to the surface area or the size of the RAMEE. Hemingson et al. (2010) showed that RAMEE effectiveness increases with NTU (as shown in Figure 4.2a) and follows a similar trend expected by other references (e.g., Zhang and Niu 2002; Incropera and DeWitt 2002). By increasing NTU, the sensible effectiveness increases significantly and the latent effectiveness increases slightly. Also, a considerable increase in latent effectiveness may be obtained by increasing NTU_m to a larger value. A design NTU of 10 may be feasible for ERVs (e.g., Teke et al. 2010), therefore, it is used for this study. As well, NTU will increase when the night-time ventilation rate is lower than the day-time.

Hemingson (2010) found that the variation of indoor conditions between the heating and cooling indoor set-points has a minimal impact on the RAMEE effectiveness, and may change the total effectiveness by 0.3%. But, the dependency of RAMEE effectiveness on outdoor air conditions is more significant which is due to the impact that the outdoor temperature and humidity have on the liquid desiccant and the fact that heat and moisture transfer are coupled in the RAMEE (Hemingson et al. 2010). A greater temperature difference between outdoor and indoor air (either summer or winter) improves the RAMEE moisture transfer. Also, the RAMEE heat transfer increases as the humidity ratio difference between indoor and outdoor air increases. Figures 2b, 2c and 2d present the RAMEE effectiveness as a function of Cr* in different outdoor conditions and NTU=10. As shown in these figures, the Cr* at which the peak effectiveness is achieved (Cr^*_{opt}) varies depending on the outdoor conditions. Therefore, at any given outdoor condition, the Cr* should be controlled so that the maximum effectiveness is achieved. Rasouli et al. (2010c) studied the operation of the RAMEE in different outdoor conditions in an office building and showed that the strategy of controlling the Cr^*_{opt} depends on RAMEE's operating condition (heating, cooling and part-load operation). Figure 4.3 presents the TMY operating conditions of the RAMEE for one year in different locations.

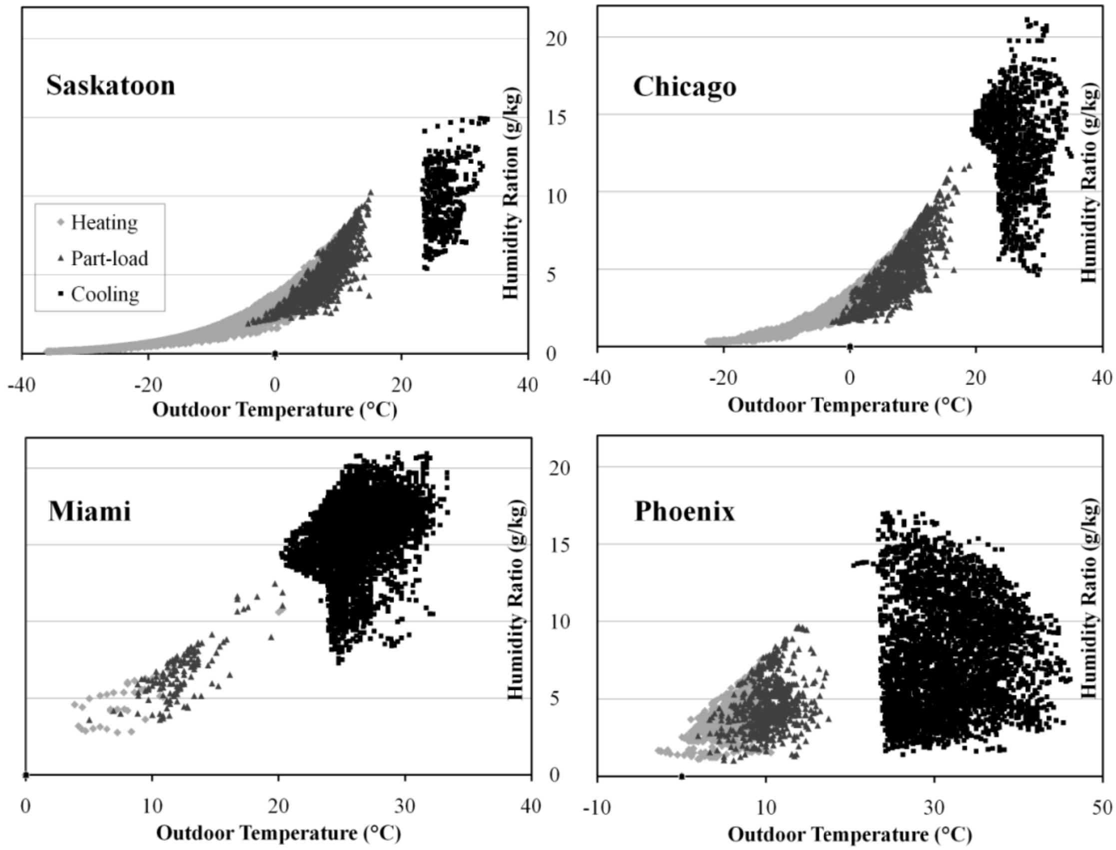


Figure 4.3 Operating conditions of the RAMEE for different locations for the hospital building in one year

As an explanation of Figure 4.3, the RAMEE heats the ventilation air in a full-load or part-load operation for low outdoor temperatures. The RAMEE should be off or by-passed when the building needs cooling while both outdoor temperature and enthalpy are lower than that of the indoor air (Rasouli et al. 2010b). The operation of RAMEE in such conditions heats and humidifies the cool outdoor air and increases the cooling energy consumption. When either the outdoor temperature or the outdoor enthalpy is greater than that of the indoor air, the RAMEE should be operated to reduce the temperature or the enthalpy of the ventilation air. Rasouli et al. (2010c) found that in order to optimize the operation of the RAMEE, the Cr* needs to be controlled under a different strategy as the RAMEE's operating condition changes. Table 4.1 summarizes the required control strategy to achieve optimal performance of the RAMEE.

Table 4.1 Cr* control strategy and definitions of Cr^*_{opt} for optimal performance of the RAMEE for different steady-state operating conditions

RAMEE's operating condition:	Heating	Cooling ($h_{out}>h_{in}$)	Cooling ($h_{out}<h_{in}$)	Part-load
Cr^*_{opt} is the Cr* at which:	ε_s is maximum	ε_t is maximum and positive	ε_t is minimum and negative	ε_s is maximum and bypass fraction of : $R = \frac{\dot{m}_{bypass}}{\dot{m}_{ventilation}} = 1 - \frac{T_4 - T_1}{\varepsilon_s(T_5 - T_1)}$

The numerical solution of heat and mass transfer in the RAMEE for steady-state operation was developed in previous research (Fan et al. 2005; Vali et al 2009; Hemingson et al. 2010). Based on the numerical solution of the counter flow RAMEE, Akbari et al. (2010) developed an optimization Artificial Neural Network (ANN) using MATLAB neural network toolbox (MATLAB 2006). For given RAMEE operating condition, NTU and indoor and outdoor conditions, the ANN is able to predict the Cr^*_{opt} and the associated effectivenesses. Figure 4.4 shows the variation of hourly Cr^*_{opt} during a TMY of operation of the RAMEE in each location. Cr* of zero refers to RAMEE's being off operation that means the conditions specified in Table 4.1 are not satisfied.

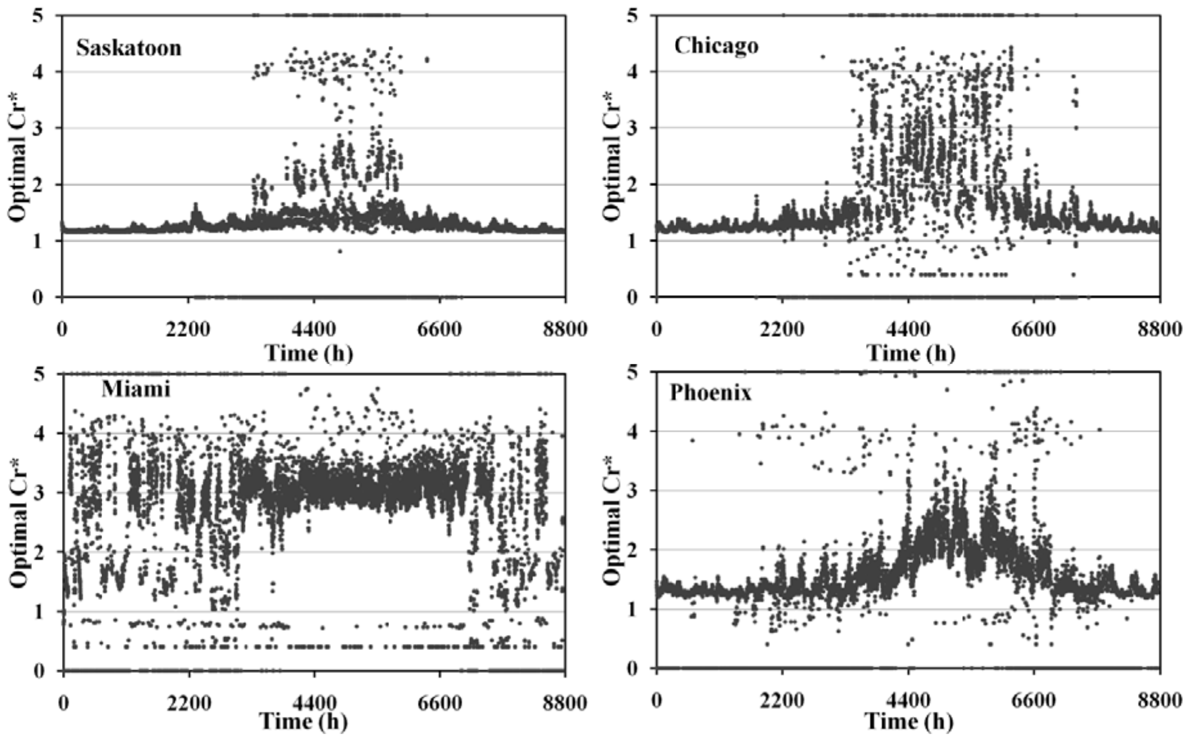


Figure 4.4 Yearly variation of the hourly Cr* for optimal operation of the RAMEE

Figure 4.4 shows less scatter variation of Cr^*_{opt} during the winter compared to the summer. For all climates (except for Miami), the Cr^*_{opt} varies between 1.2 and 1.3 during the heating season, and increases up to 4 during the summer. It should be noted that Cr* is a function of ventilation rate and salt solution flow rate (Equation 4-6). Since the air flow rate is typically set based on minimum standard ventilation requirement, the solution flow rate remains the controllable variable to achieve Cr^*_{opt}.

As mentioned previously, the effectiveness of the RAMEE depends on the outdoor/indoor air conditions, the operating Cr* and the ventilation air flow rate. All these variables (except for NTU that switches between day-time and night-time values) show scatter during a year which may results in a variation of the effectiveness. The ANN can determine the effectiveness of the RAMEE based on the specified outdoor conditions (Figure 4.3), indoor conditions (ASHRAE 2004a) and operating Cr* (Figure 4.4). Figure 4.5 shows the variation of sensible and latent effectiveness for different locations for one typical year.

The effectiveness values are bounded between 0 and 1 in Figure 4.5. However, the effectiveness of RAMEE may exceed 100% at specific operating conditions. These mostly occur when the energy transfer via the RAMEE is not very significant (Hemingson 2010).

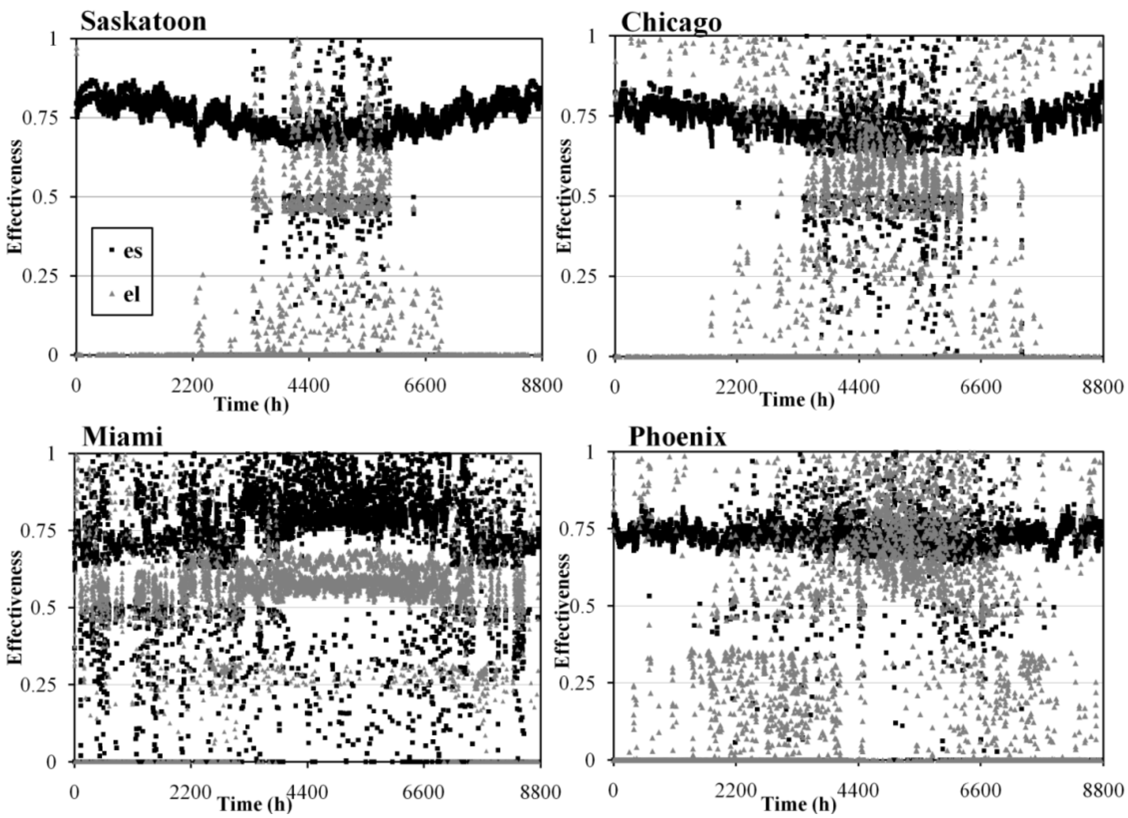


Figure 4.5 Variation of the RAMEE steady-state sensible and latent effectiveness for a TMY in different locations ($Cr^*=Cr^*_{opt}$)

The hourly effectiveness values are inputs to the TRNSYS (Klein 2000) model of the RAMEE. The thermal system (including the HVAC system, RAMEE and the building) is simulated using the TRNSYS building energy simulation program equipped with TESS libraries (Thornton et al. 2009).

In order to quantify an average operating effectiveness for the RAMEE, the hourly effectiveness values are weighted by the associated hourly net energy transfer via the RAMEE. Table 4.2 presents the average sensible and latent effectiveness of the RAMEE throughout a year in different locations.

Table 4.2 Average sensible and latent effectiveness of the RAMEE

	Saskatoon	Chicago	Miami	Phoenix
Average sensible effectiveness	0.78	0.76	0.86	0.73
Average latent effectiveness	0.70	0.59	0.59	0.58

4.6. Results

In this section, the following assumptions are made regarding the RAMEE and the HVAC system unless otherwise stated: The HVAC system consists of a gas-fired boiler with efficiency of 88% and a direct-expansion water chiller with a COP of 3 which satisfies ASHRAE Standard 90.1-2004 minimum boiler efficiency of 80% and chiller COP of 2.78 (ASHRAE 2004b). Fan efficiency is assumed to be 60% and air pressure drop of the HVAC system and each LAMEE are assumed to be 10 cm and 2 cm of water, respectively. The RAMEE operates under hourly Cr^*_{opt} and design NTU of 10.

4.6.1. Energy

Figure 4.6 shows the simulation results for the impact of RAMEE on annual heating and cooling energy consumption in the hospital compared to the case of no energy recovery. The RAMEE saves 58%, 66%, 90% and 83% of annual heating energy in Saskatoon, Chicago, Miami and Phoenix, respectively. Also, it saves 4%, 10%, 18% and 15% of the annual cooling energy in Saskatoon, Chicago, Miami and Phoenix, respectively. The cooling energy saved in cold climate (Saskatoon and Chicago) is not very significant since the internal loads (not the ventilation load) account for the larger portion of the cooling load.

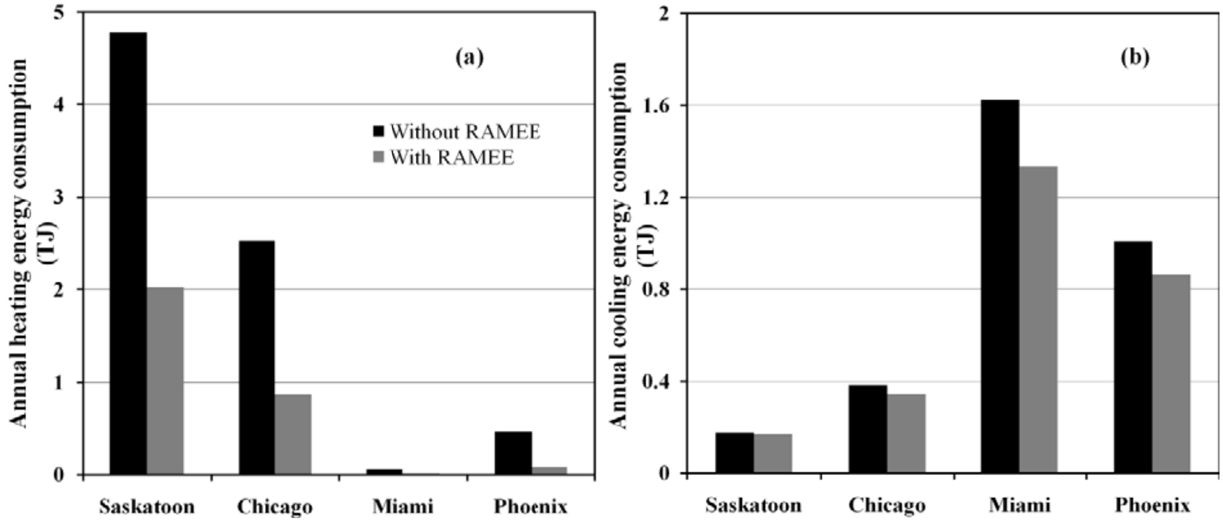


Figure 4.6 The impact of RAMEE on annual energy consumption for (a) heating and (b) cooling

Figure 4.7 presents the impact of the RAMEE on the size of HVAC equipment compared to the case of no energy recovery. The size of heating equipment can be reduced by about 45% in cold climates and 65% in hot climates. Also, the cooling system can be downsized by about 25% in all climates.

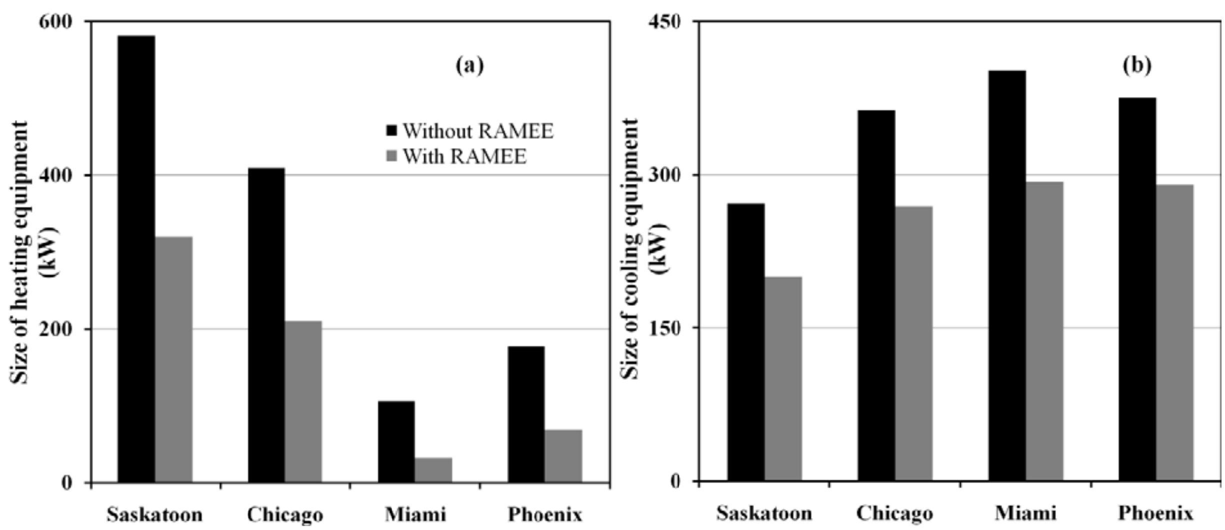


Figure 4.7 The impact of RAMEE on the capacity of HVAC equipment for (a) heating and (b) cooling

CBECS reported inpatient health-care facilities to have the second highest energy intensity among US commercial buildings with an average total energy intensity of 2830 (MJ/m^2.year) in 2003. The HVAC system energy consumption accounted for 52% of the total energy use which gives an average HVAC energy intensity of 1472 MJ/m^2.year. Thus the HVAC energy intensity of inpatient health-care facilities was much higher than the total energy intensity of educational facilities (944 MJ/m^2.year) or office buildings (1055 MJ/m^2.year). In this research, the HVAC system for the studied hospital has an energy intensity of 1730, 1100, 739 and 672 MJ/m^2.year with no energy recovery in Saskatoon, Chicago, Miami and Phoenix, respectively (giving an average of 1060 MJ/m^2.year). By employing the RAMEE, the total energy intensities will be reduced by 48%, 45%, 8% and 17% in Saskatoon, Chicago, Miami and Phoenix, respectively.

It should be noted that the underestimating of annual energy consumption using the computer simulation (compared to CBECS reported values) might be mostly due to the energy-saving envelope (well-insulated walls and roofs and double-glazed windows) considered for the simulated building compared to the data obtained from the US office building categorization. In addition, the following assumptions are made for this research which may cause underestimation of energy consumption in computer simulation compared to real buildings: (1) high-efficiency heating and cooling systems (combustion efficiency of 88% and chiller COP of 3), (2) zero heat loss and leakage from equipment and ducting, and (3) running a VAV HVAC system in the building (instead of a less-efficient CAV system; Yao et al. 2007).

4.6.2. Control based on an Operating Averaged Cr*

As discussed in section 3.2, the optimal operation of the RAMEE requires an accurate control of the salt solution flow rate (giving the Cr^*_{opt}). Rasouli et al. (2010c) showed that the RAMEE may be operated in an office building using an average seasonal or yearly Cr* value with no significant impact on energy savings (i.e., less than 2% for most climates). The advantage of operating the RAMEE using an average Cr* is that there is no need for an accurate control of salt solution flow rate for each slight change of outdoor condition. In this section, the impact of applying an average seasonal or yearly Cr* value on energy saving with the RAMEE in the hospital is studied. Table 4.3 shows the seasonal and yearly averaged Cr* weighted by hourly energy transfer via the RAMEE and the associated standard deviation. Table 4.4 presents the annual cooling and heating energy savings when the RAMEE system operates under specified average Cr* values.

Table 4.3 Seasonal and yearly weighted average Cr* and associated standard deviation for the hospital building

	Seasonal average Cr*		Yearly average Cr*
Location	**Winter (heating)**	**Summer (cooling)**	**Heating and cooling**
Saskatoon	1.21±0.09	2.25±0.35	1.22±0.53
Chicago	1.26±0.12	2.78±0.42	1.37±0.70
Miami	1.52±0.28	3.07±0.52	2.99±0.71
Phoenix	1.31±0.13	1.88±0.47	1.64±0.55

Table 4.4 Annual energy saved with the RAMEE system operating with selected Cr* values

	Annual heating energy saved			Annual cooling energy saved		
Location	**Optimal Cr***	**Seasonal Cr***	**Yearly Cr***	**Optimal Cr***	**Seasonal Cr***	**Yearly Cr***
Saskatoon	58%	58%	58%	4%	4%	3%
Chicago	66%	66%	65%	10%	9%	7%
Miami	90%	90%	83%	18%	18%	18%
Phoenix	83%	83%	81%	15%	14%	14%

Compared to using the optimal Cr*, the results in Table 4.4 show that the energy savings slightly reduce by using a yearly average Cr*, however the reduction in energy savings is negligible with the averaged seasonal Cr* values. The RAMEE may operate under seasonal or yearly average Cr* with no significant loss of energy.

4.6.3. Life Cycle Cost (LCC) Analysis

LCC analysis is known as a very good measure to evaluate and compare different available alternatives in terms of expenses associated with each system during the life-cycle. The life-cycle of a system includes its production, operation, demolition and disposal. The two alternative systems in this research are: (1) A VAV HVAC system that is not equipped with any energy recovery systems, and (2) A VAV HVAC system that is equipped with the RAMEE. The cost analysis is conducted over a 15-year life-cycle for both systems. For this LCC study, only those expenses that are not equal for the two alternatives need to be considered. These costs can be categorized as capital costs, that have to be invested before the project begins to operate, and operational costs that include all the expenses during the operation of the system (i.e., maintenance and energy).

The capital costs include the cost of the HVAC system that consists of a cast-iron gas-fired boiler ($68.3/kW), a direct expansion water chiller ($227/kW) and Centrifugal type HVAC fans ($851/m^3/s). These costs are based on RSMeans Mechanical Cost Data 2010 (Mossman et al. 2010). The cost of the RAMEE, as an ERV, is considered to be $3/CFM ($6357/m^3/s) as recommended by technical papers in the field of air-to-air energy exchangers (e.g., Besant and Simonson 2000; Turpin 2000). Also, a zero residual value is assumed as the worth of the HVAC system at the end of its life-cycle. The operational costs include the cost of the energy consumed by the heating/cooling equipment and the fans and the maintenance cost. Assuming equal maintenance costs for both alternatives, the operational cost will only include the cost of energy. The energy rates may vary depending on the location and the energy source. In this study, natural gas and electricity are assumed to be those for the energy sources for heating and cooling, respectively. The gas-fired boiler using natural gas produces combustion heat at 37.8 MJ/m^3 (CRC handbook of chemistry and physics 1977). Figure 4.8 presents the comparison of capital costs and operational costs for the two alternatives.

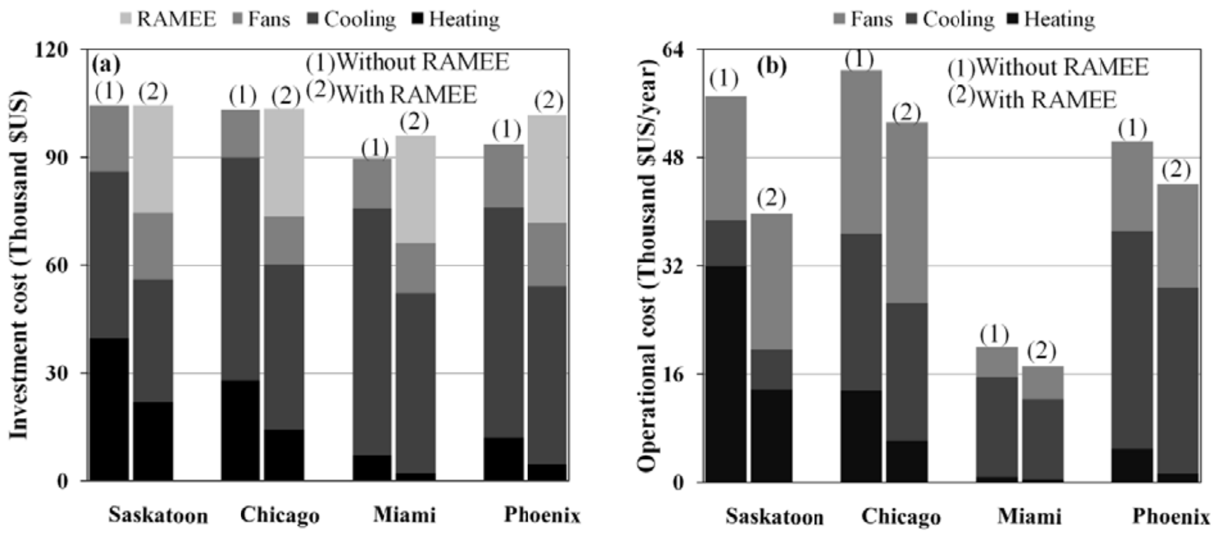


Figure 4.8 Life-cycle cost analysis results (a) capital costs and (b) operational costs for the HVAC system (1) without the RAMEE and (2) with the RAMEE

It can be seen in Figure 4.8a that the RAMEE can be purchased at no net extra cost for cold climates (Saskatoon and Chicago) due to the money saved by downsizing the heating/cooling equipment. This means that the payback of the RAMEE in cold climate is instant (immediate payback) and the energy savings during the RAMEE's life-cycle are achieved with no extra investment. On the other hand, an HVAC system equipped with a RAMEE has a higher capital cost in hot climates (Miami and Phoenix shown in Figure 4.8a). The operational cost of the RAMEE depends on air pressure drop across each LAMEE. Figure 4.8b is plotted based on air pressure drop of 2 cm of water across each LAMEE. Increasing the RAMEE's pressure drop decreases the energy savings of the RAMEE due to higher fan energy. In cold climates (Saskatoon and Chicago), increasing the LAMEE's pressure drop reduces the energy saved, but the payback period will remain zero for air pressure drops within 0 to 5 cm of water across each LAMEE (expected range by manufacturer). On the other hand, increasing the LAMEE's pressure drop increases the payback period in hot climates (Miami and Phoenix). Figure 4.9 presents the payback period of the RAMEE in Miami and Phoenix as a function of the pressure drop across each LAMEE.

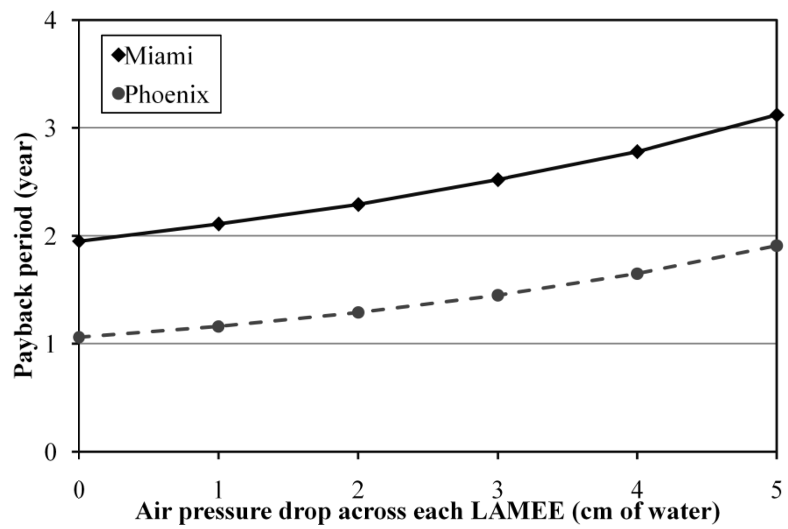


Figure 4.9 Payback period of the RAMEE in Miami and Phoenix as a function of pressure drop across each LAMEE

Based on the results presented in Figure 4.9, the RAMEE system will have a payback period of 1-2 years in Phoenix and 2-3 years in Miami. Table 4.5 summarizes the life cycle cost (including investment and operation costs) of two alternative systems over 15 years of operation assuming 2 cm of water pressure drop across each LAMEE.

Table 4.5 LCC (including capital and operational costs) of the two HVAC system alternatives for a 15-year life-cycle

	Saskatoon	Chicago	Miami	Phoenix
Without RAMEE (Thousands of $US)	961	1018	389	850
With RAMEE (Thousands $US)	701	901	354	763
% saving	27%	11%	9%	10%

4.6.4. Life Cycle Environmental Assessment (LCEA)

Similar to the life cycle cost analysis that addresses the expenses associated with a project during its life-cycle (including production, operation and disposal), life cycle environmental assessment (LCEA) deals with the impact of a system on the environment. Both approaches are similar in that they study the system over its life cycle rather than making a decision based on just the capital cost; however, they are different in their measuring metrics (i.e., money for LCC and environment for LCEA) (Nyman and Simonson 2004). In this paper, the environmental impact of the two systems, i.e. VAV HVAC systems with and without the RAMEE on greenhouse gas emissions and climate change is studied. The tons of CO_2- equivalent emission is used to represent the climate change since CO_2 is the main greenhouse gas.

The mass of greenhouse gases emitted during the combustion of natural gas depends on the fuel composition and this may vary slightly from location to location. However, an average value is used for both US and Canada based on the data obtained from Canada's Clean and Renewable Energy Research Centre (Aube 2001). On the other hand, due to the variety of resources that different utilities use to generate electricity (e.g., hydro, nuclear, fossil fuel, etc.), the greenhouse gas emissions due to electricity consumption varies dramatically for different locations. Table 4.6 presents the amount of emitted greenhouse gases associated with consuming natural gas and electricity in the different locations studied in this paper. Data obtained from Canada's Clean and Renewable Energy Research Centre (Aube 2001) and US Environmental Protection Agency (EPA 2010) are used to produce the results shown in this table. CO_2-equivalent is calculated using weighting factors (also called Global Warming Potential, GWP) of CO_2, N_2O and CH_4 as 1, 310 and 21, respectively (Aube 2001).

Table 4.6 The greenhouse gas emission due to electricity and natural gas consumption in different locations

	Natural gas				Electricity			
	CO_2 (t/TJ)	N_2O (kg/TJ)	CH_4 (kg/TJ)	**CO_2-equivalent (t/TJ)**	CO_2 (t/TJ)	N_2O (kg/TJ)	CH_4 (kg/TJ)	**CO_2-equivalent (t/TJ)**
Saskatoon	49.68	0.52	1.1	**49.86**	-	-	-	**234**
Chicago	49.68	0.52	1.1	**49.86**	194	3.2	2.3	**195.04**
Miami	49.68	0.52	1.1	**49.86**	166	2.1	5.8	**166.77**
Phoenix	49.68	0.52	1.1	**49.86**	165	2.3	2.2	**165.76**

Nyman and Simonson (2004) studied the LCA of air-handling units with and without energy recovery systems and found that the emission of greenhouse gases during their operation in a 20-year life-cycle was typically 20 to 40 times greater than the emissions occurred during the manufacturing process of the units. Therefore, the LCA in this paper takes the environmental impacts of the systems during the operation only. Figure 4.10 compares the annual equivalent CO_2 emission by the HVAC system with and without the RAMEE for the hospital in different locations.

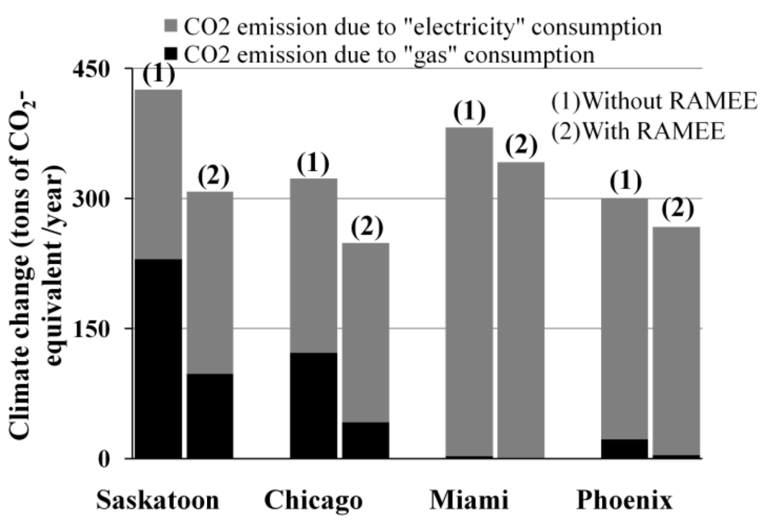


Figure 4.10 Annual equivalent emission of CO_2 from the hospital building with and without the RAMEE

Figure 4.10 demonstrates the positive impact of energy recovery when a RAMEE is used to reduce the emission of greenhouse gases. By employing the RAMEE, the emission of CO_2-equivalent from the hospital building HVAC system can be reduced by about 25% and 10% in cold climates and hot climates, respectively. A typical mature tree absorbs CO_2 at a rate of 21.6 kg/year (McAliney 1993), and a new medium size car emits 3.3 tons of CO_2 per year (traveling 20,000 km/year, using regular gas with an automatic transmission; Natural Resources Canada 2010). Therefore, the carbon offset by purchasing the RAMEE for the hospital building is equal to planting 5450, 3440, 1850 and 1490 trees or removing 36, 23, 12 and 10 cars off the road in Saskatoon, Chicago, Miami and Phoenix, respectively.

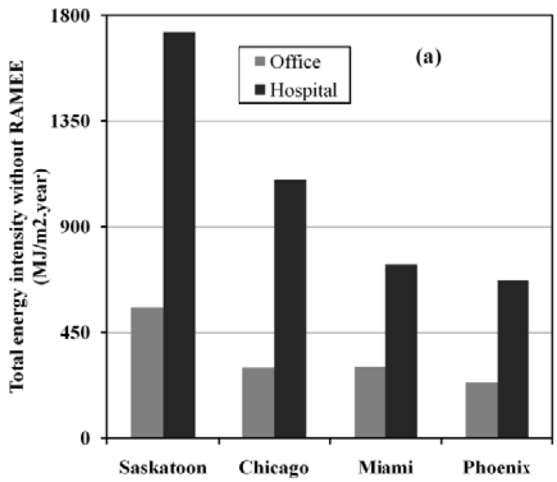
4.6.5. Comparison of Two Case Studies

Rasouli et al. (2010c) studied the application of a RAMEE in an office building HVAC system simulated for different climates. In this section, a comparison of results between the two case studies of the RAMEE (i.e., the office building and the hospital) is presented. Table 4.7 summarizes the differences and similarities between the characteristics of the two cases.

Table 4.7 Summary of the characteristics of each case study

Area:	Office: 28800 m^2, 10-storey Hospital: 3150 m^2, 3-storey	**Heating system:**	Office: Radiator heating Hospital: VAV HVAC
Building envelope:	Similar, described in section 4.4	**Cooling system:**	Similar, VAV HVAC
Operation schedule:	Office: 6:00-22:00 Hospital: day-time: 6:00-22:00; night-time: 22:00-6:00	**Min. required total air change**	Office: N/A Hospital: 6 ACH day-time; 4 ACH night-time
Ventilation rate:	Office: 0.5 ACH Hospital: 2 ACH day-time; 1.3 ACH night-time	**Indoor RH**	Similar, below 60% when building is occupied
Indoor set-point temperature:	Office: 24°C at summer day-time; 22°C at winter day-time; 15°C night-time Hospital: 24°C in summer, 22°C in winter	**RAMEE's control and operating condition**	Similar, refer to Tables 4.1
Efficiency and pressure drop of HVAC equipment	Similar, specified in section 4.6	**Internal loads:**	Similar in loads but different operation schedules

Figure 4.11a presents the comparison of the total annual energy intensity for the buildings in different climates. The results show that the total energy intensity in the hospital without the RAMEE is 3.7, 3.1, 2.4 and 2.8 times greater than the office building in Saskatoon, Chicago, Miami and Phoenix, respectively. As a comparison to the TRNSYS simulation results, CBECS (EIA 2003) has reported 2.8 times higher HVAC energy intensity in hospitals compared to office buildings in US in 2003 (i.e., 1472 MJ/m^2 in hospitals versus 533 MJ/m^2 in office buildings). Figure 4.11b shows the energy savings with RAMEE (including heating, cooling and fan energy) that is 48%, 45%, 8% and 17% in the hospital, and 30%, 28%, 5% and 10% in the office building in Saskatoon, Chicago, Miami and Phoenix, respectively.



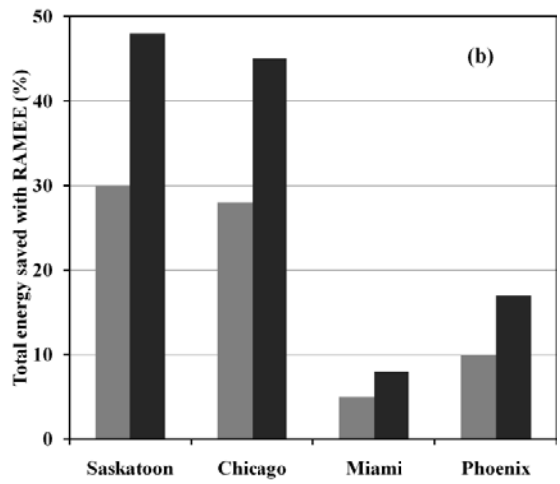


Figure 4.11 Comparison of (a) energy intensity of the HVAC system without the RAMEE and (b) energy saved with the RAMEE for two case studies in different climate

Table 4.8 shows the comparison of operating average yearly Cr* for the two case studies. The results show that the average Cr* is very close for both buildings in each location. Therefore, average Cr* seems to be a climate-dependent parameter (not a building-dependent parameter).

Table 4.8 Comparison of weighted average yearly Cr* in for the two case studies in different locations

	Saskatoon	**Chicago**	**Miami**	**Phoenix**
Office	1.22	1.30	2.90	1.62
Hospital	1.22	1.37	2.99	1.64

Assuming similar air pressure drop of 2 cm of water across each LAMEE, the payback of the RAMEE in the hospital is about 2 years faster than the office building. In cold climates (Saskatoon and Chicago), the payback is immediate for hospitals and takes 1.8 to 2 years for office buildings. In hot climates (Miami and Phoenix), the payback may take 1.5 to 2.5 years for hospitals and about 4 to 4.8 years for office buildings.

4.7. Conclusions

The steady-state operation of a Run-Around Membrane Energy Exchanger (RAMEE) that transfers heat and moisture between outdoor ventilation and building exhaust air is described in the paper. The RAMEE effectiveness varies depending on outdoor conditions, indoor conditions, ventilation air flow rate (represented by NTU) and salt solution flow rate (represented by Cr*). The RAMEE effectiveness can be optimized by changing these parameters; however, the salt solution flow rate is the only controllable variable for a given building in a given location. During the winter, the RAMEE should operate at the Cr* which gives maximum sensible effectiveness. While in the summer, the RAMEE should be operated at the Cr* resulting in maximum reduction of outdoor air enthalpy. The RAMEE is simulated in a hospital building using TRNSYS computer program joint with an Artificial Neural Network (ANN) that predicts the optimal salt solution flow rate (corresponding to Cr^*_{opt}). The hospital building is simulated in four different climates, i.e., Saskatoon (cold and dry), Chicago (cool and humid), Phoenix (hot and dry) and Miami (hot and humid). The simulation results showed about 58% to 65% annual heating energy saving in cold climates and 15% to 20% annual cooling energy saving in hot climates. Since the application of hourly optimal Cr* requires an accurate control of the salt solution flow rate, the impact of applying average seasonal and yearly Cr* values was studied. Also, the results show that operating the system under seasonal or yearly average Cr* (that vary depending on the location) has a minimal impact on energy savings compared to the case that hourly optimal Cr* is applied. The life cycle analysis results showed that the payback of the RAMEE is immediate in cold climates and reduces the equivalent emission of CO_2 (corresponding to the climate change) by 25%. In hot climates, the payback may take up to 2 to 3 years, and the RAMEE reduces the equivalent emission of CO_2 by 10%.

CHAPTER 5

CONCLUSIONS

A Run-Around Membrane Energy Exchanger (RAMEE) is a heat and moisture recovery system which consists of two separate exchangers that are coupled with an aqueous salt solution. In this thesis, the TRNSYS computer program was used to study the impact of adding a RAMEE to conventional HVAC systems. The objectives of the thesis were to determine an appropriate control strategy for the RAMEE, annual energy savings with a RAMEE in different climates and buildings (i.e., an office building and a health-care facility), and to perform RAMEE’s life-cycle cost and life-cycle environmental assessment.

5.1. Conclusions

Previous research has proven the necessity of controlling energy recovery ventilators (ERVs) during the cooling season, however, there was no universal control strategy found in the literature that is applicable to all types of ERVs. Therefore, this research undertook studying an optimal control strategy of ERVs. The proposed optimal control was dependent on latent to sensible effectiveness ratio of the ERV, and limited the summer operation of an ERV to the cases where (a) the ERV can reduce the temperature of the ventilation air, or (b) it can reduce the enthalpy of the ventilation air. This control was tested using TRNSYS modeling of an office building in different climates and resulted in higher cooling energy savings when compared to other controls available in the literature.

The results of the ERVs’ optimal control were applied to a RAMEE as a variable effectiveness ERV. The salt solution flow rate (as the fluid coupling the two exchangers) was found to be the key parameter to control the operation of a RAMEE. The salt solution flow rate is represented by Cr* that is the ratio of the solution heat capacity rate to the air heat capacity rate. The optimal Cr* (corresponding to the optimal salt solution flow rate) could have a scatter variation during a year as the outdoor conditions change. This requires a complex control of salt solution flow rate by the operator or controller. However, it was shown that applying an average seasonal or yearly Cr* did not reduce the energy savings significantly. The results indicate that the average Cr* is a climate-dependent parameter, not a building-dependent parameter. For instance, in Saskatoon, the yearly average Cr* was found to be 1.22 for both the office building and the hospital. But in Miami, the yearly average Cr* was 2.90 for the office building and 2.99 for the hospital.

As the first case study on investigating the energy savings with a controlled RAMEE, a 10-storey office building with an outdoor ventilation rate of 0.5 ACH was selected. The chosen building represented 3.34% of the US office building stock. However, the insulation thickness of the walls and roof were increased to 10 cm and 15 cm, respectively, so that it more closely represents a typical office building in all climates. The results showed that the RAMEE can save 32% and 43% of the annual heating energy in Saskatoon and Chicago (cold climates), respectively, and reduce the capacity of the heating system by about 25% in these locations. These reductions in energy consumption and the size of the heating equipment give a payback period of about 2 years for the RAMEE in cold climates. On the other hand, The RAMEE can save about 10% of the annual cooling energy in Miami and Phoenix (hot climates), respectively. Also, the cooling equipment can be downsized by 10% in Miami and 5% in Phoenix. The payback period for the RAMEE in hot climates is 4 to 5 years.

As the second case study, the RAMEE was simulated in a 3-storey health-care facility that has the same envelope as the office building, but higher ventilation rates (i.e., 2ACH day-time and 1.3 ACH night-time versus 0.5ACH day-time and 0 ACH night-time for the office building). The results showed that the RAMEE reduces the annual heating energy consumption by 58% and 66% in Saskatoon and Chicago (cold climates), respectively. It also downsizes the heating equipment by about 45% in these locations. This amount of saving results in an immediate payback of the RAMEE in the hospital building. In the hot climates (Miami and Phoenix), the RAMEE reduces the annual cooling energy by 18% and the size of cooling system by 25%. This gives a payback of 1 to 3 years for RAMEE in hot climates. Regarding the environmental impact of the RAMEE, it can reduce the emission of greenhouse gases by 25% in cold climates and 10% in hot climates. This is equal to planting 5450, 3440, 1850 and 1490 trees or removing 36, 23, 12 and 10 cars off the road as the RAMEE is being installed for a hospital building in Saskatoon, Chicago, Miami and Phoenix, respectively.

As a comparison of the two case studies, the total energy savings with a RAMEE (including heating, cooling and fan energy) in Saskatoon, Chicago, Miami and Phoenix was found to be 48%, 45%, 8% and 17% in the hospital, and 30%, 28%, 5% and 10% in the office building, respectively. On average, for all climates, the payback period for the RAMEE in the hospital building is 2 years lower than in the office building due to a higher ventilation rate that gives higher energy savings potential.

5.2. Future Work

This study has taken a step in the direction of estimating the impact of a RAMEE on energy, economics and environment. There are some limitations encountered with the applicability of the results of this research, such as: considering RAMEE at steady-state operation, NTU of 10, no maintenance cost for the RAMEE, extreme climates, local energy rates, etc. The following recommendations are made for future work to provide generalized and more accurate results:

- For hot-humid outdoor conditions, the air conditioning process requires cooling and dehumidification that needs a low operating temperature of the refrigerant fluid. In the presence of a RAMEE, the refrigerant's operating temperature can be increased that results in a higher COP for the chiller and consequently lower electricity consumption. As a future work, the impact of RAMEE on the COP of the chiller and corresponding energy savings should be studied.

- The application of a RAMEE in an active HVAC system could be a useful future study. There is a good research work done by Bergero and Chiari (2010) on the performance of liquid desiccants in active HVAC systems that can be referred to for more details.
- Depending on the initial conditions and other design parameters, it may take the RAMEE several minutes to several hours to reach the steady-state condition. In case the transient model of the RAMEE is developed, it's recommended to study a transient RAMEE and compare the results to the present steady-state RAMEE.
- A series of sensitivity studies should be performed to investigate the impact of the following parameters on energy saving, life-cycle cost and payback period of a RAMEE. Such sensitivity studies may include the following parameters and will allow us to generalize the results for a variety of RAMEE operating conditions, building types and locations:
  - RAMEE's maintenance cost and investment cost
  - Energy rates (natural gas and electricity)
  - RAMEE's operating NTU
  - Ventilation rate
  - Building envelope (insulation, windows, air tightness, etc.)
- The life-cycle cost of the RAMEE could be studied including the impact of a RAMEE on reducing the emission of greenhouse gases. Although the building owner is not

responsible for the expenses related to offsetting the greenhouse gases, such a study can present the benefit of using RAMEE in larger scales (global or national).

- The control strategies proposed in Chapter 2 require accurate measurements of the enthalpy (indoor air and outdoor air) that requires temperature and humidity sensors. Humidity sensors are more expensive and usually have considerable errors associated with their measurements, especially in more humid conditions. The savings that one can get using less accurate and more expensive humidity sensors may not be significant in all climates. Taylor and Cheng (2010) studied different strategies that are currently used to control economizers. They discussed that an ideal control strategy that may appear to provide large energy savings may actually increase the energy use due to sensor errors or may not be able to save significant amount of energy in all climates. A suggestion for future work is to study whether the proposed ERV controls (Chapter 2) can be simplified and become more specific for each certain climate.

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APPENDIX A
ALGORYTHMS AND MATLAB COMPUTER CODES

Appendix A includes the flowchart and MATLAB codes for the RAMEE and HVAC system

A.1. The Base HVAC System of the Office Building (No Energy Recovery System or Economizer Employed)

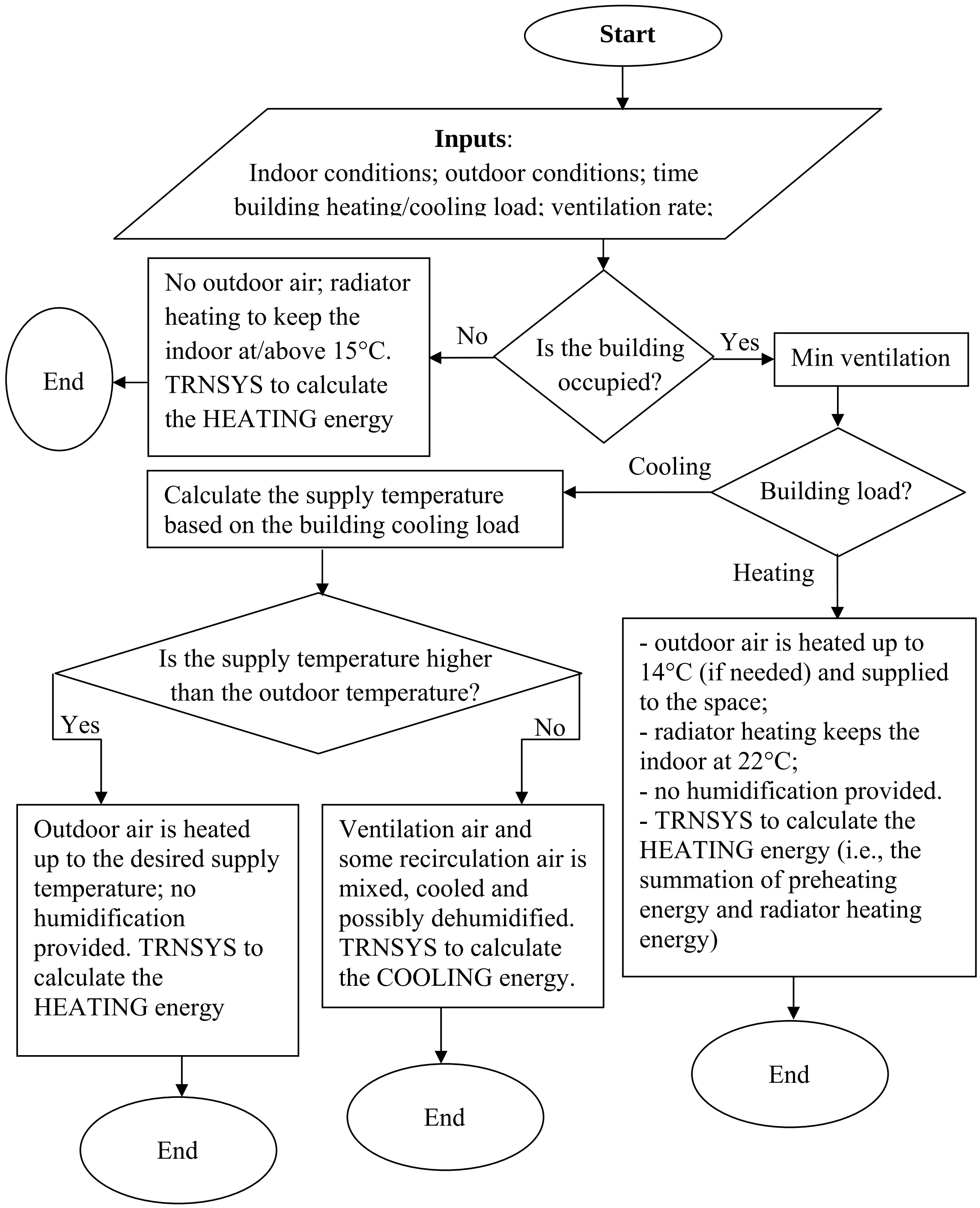

Figure A.1. Flowchart of the base HVAC system operating in the office building

```
=================================== Setting the given values==============================
Mvent=50000;                  %Standard requirement for ventilation (kg/hr)
Cp=1.02;                      %Specific heat capacity of air (kJ/kg K)
hfg=2500;                     %Enthalpy of phase change (kJ/kg)
Winset=0.012;                 %Indoor maximum allowed humidity ratio (kg/kg)
======Receiving the inputs from other TRNSYS components (i.e. the building and the weather data file)====
hr=trnInputs(1);              %Hour (input from TMY2 weather data file)
To=trnInputs(2);              %Outdoor temperature (C) (input from TMY2 weather data file)
Wo=trnInputs(3);              %Outdoor humidity ratio (kg/kg) (input from TMY2 weather data file)
Ti=trnInputs(4);              %Indoor temperature (C) (input from previous iteration at current hr)
Wi=trnInputs(5);              %Indoor humidity ratio (kg/kg) (input from previous iteration at current hr)
qh=trnInputs(6);              %Building heating load (kJ/hr) (space load, only. input from the building)
qc=trnInputs(7);              %Building cooling load (kJ/hr) (space load, only. input from the building)
Wif=trnInputs(8);             %Indoor humidity ratio at zero supply air flow rate (kg/kg) (input from the
                              building)
P=trnInputs(9);               %Local atmospheric pressure (atm) (input from TMY2 weather data file)
=================================== The main body===================================
                    (1)  Check the building occupancy to determine the ventilation rate
if(mod(hr,24)<22&&mod(hr,24)>5) %Is the building occupied?
  Mvent=Mvent;                %Yes! Provide the required ventilation
else
  Mvent=0;                    %No! No outdoor ventilation air provided
end
                              (2)  If the building needs heating
if(qh>0)                      %Heating
  Msup=Mvent;                 %Supply of minimum ventilation air
  Moa=Mvent;
  Tsup=max(14,To);            %Supply temperatures not lower than 14(C)
  Wsup=Wo;                    %Supply humidity equal to outdoor humidity
  Wmix=Wo;
  coilsens=Msup*Cp*(Tsup-To); %Calculation of the sensible heating load
  coillat=0;                  %No latent load during the heating!
  reheat=0;                   %Reheat is zero during the heating!
end
                              (3)  If the building needs cooling
if(qc>0)                      %Cooling
  Msup=max(Mvent,qc/(10*Cp)); %Calculation of supply air flow rate
  Wsupmax=min(0.012,Winset-86400*1.2*(Wif-Winset)/Msup);
                              %Calculation of maximum allowed supply humidity ratio to prevent indoor
                              humidities greater than 0.012 (kg/kg)
```

```
    Moa=Mvent;                      %Specifying the outdoor ventilation rate
    Tsup=Ti-qc/(Cp*Msup);           %Calculation of the supply air temperature
    Tmix=(Moa*To+(Msup-Moa)*Ti)/Msup;
                                    %Properties of the mixture of the return air and the ventilation air
    Wmix=(Moa*Wo+(Msup-Moa)*Wi)/Msup;
    if(Tsup>Tmix)                   %sensible heating if the supply temperature is greater than the mixture
                                    temperature
      Wsup=Wmix;
      coilsens=Msup*Cp*(Tsup-Tmix);
      coillat=0;
      reheat=0;
    else                            %cooling
      T=Tsup+273.15;                %Dew point calculation (ASHRAE Fundamental 2009, Chapter 1)
      pws=exp(-5800.2206/T+1.3914993-T*4.8640239e-2+T^2*4.176476e-5-T^3*1.445209e-
8+6.545967*log(T));
      Ws=0.621945*(pws/(101325*P-pws));
      Wsup=min(Ws,Wsupmax);
      pw=101.325*P*Wsup/(0.621945+Wsup);
      a=log(pw);
      Tdew=6.54+14.526*a+0.7389*a^2+0.09486*a^3+0.4569*(pw)^0.1984;
      if(Wsup>=Wmix)                %Sensible cooling only, if the mixture has lower humidity ratio than the
max                                         allowed humidity ratio)
        Wsup=Wmix;
        coilsens=Msup*Cp*(Tsup-Tmix);
        coillat=0;
        reheat=0;
      else
        coilsens=Msup*Cp*(Tdew-Tmix);   %Sensible cooling and dehumidification
        coillat=Msup*hfg*(Wsup-Wmix);
        reheat=Msup*Cp*(Tsup-Tdew);
      end
    end
end
```

(4) If the internal loads and the envelope heat loss balance out

```
if(qh+qc==0)                        %No building load (the internal gains balance the heat loss)
      Msup=Mvent;                   %Supply of ventilation requirement
      Tsup=Ti;                      %Supply at indoor temperature
      Wsup=Wo;                      %Supply at outdoor humidity
      Moa=Mvent;
      coilsens=Msup*Cp*(Ti-To);     %Heating load calculation
```

```
    coillat=0;
    reheat=0;
end
===========Sending the results to other TRNSYS components (i.e., the building and the printer)=========
trnOutputs(1)=Msup;           %Supply air mass flow rate (kg/kg) (input to the building)
trnOutputs(2)=Tsup;           %Supply air temperature (C) (input to the building)
trnOutputs(3)=Wsup;           %Supply air humidity ratio (kg/kg) (input to the building)
trnOutputs(4)=Moa;            %Outdoor ventilation rate (kg/hr) (input to the printer)
trnOutputs(5)=coilsens;       %Sensible heating/cooling load (kJ/hr) (input to the building)
trnOutputs(6)=coillat;        %Latent load (kJ/hr) (input to the building)
trnOutputs(7)=reheat;         %Reheat (kJ/hr)(input to the building)

mFileErrorCode=0
return
```

A.2. The HVAC System Equipped with a RAMEE in the Office Building (No Economizer Employed)

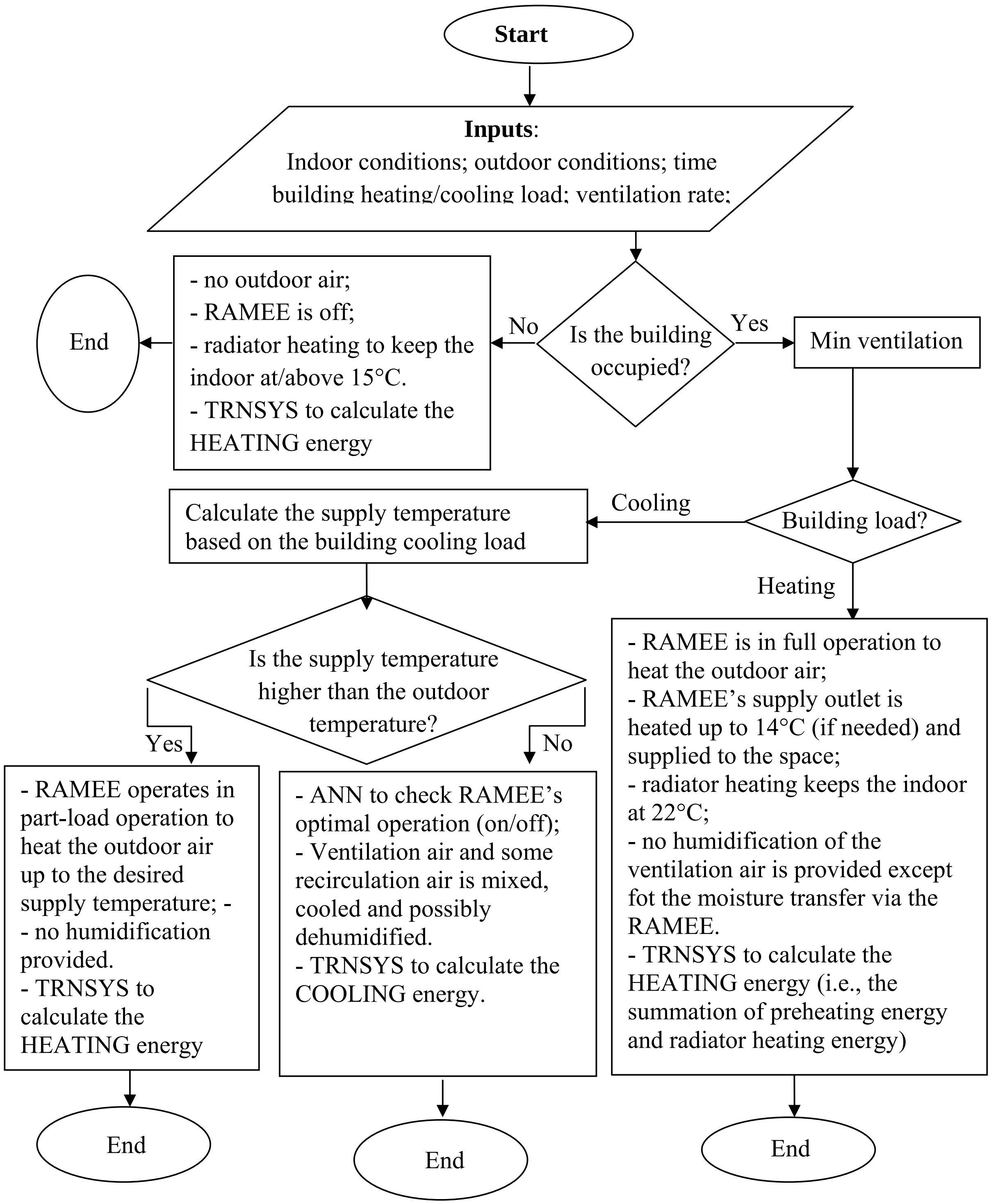

Figure A.2. Flowchart of the office building HVAC system equipped with a RAMEE

```
================================== Setting the given values===============================
Mvent=50000;                 %Standard requirement for ventilation (kg/hr)
Cp=1.02;                     %Specific heat capacity of air (kJ/kg K)
hfg=2500;                    %Enthalpy of phase change (kJ/kg)
Winset=0.012;                %Indoor maximum allowed humidity ratio (kg/kg)
===Receiving the inputs from other TRNSYS components (i.e. the building, the ANN and the weather data
file)===
hr=trnInputs(1);             %Hour (input from TMY2 weather data file)
To=trnInputs(2);             %Outdoor temperature (C) (input from TMY2 weather data file)
Wo=trnInputs(3);             %Outdoor humidity ratio (kg/kg) (input from TMY2 weather data file)
Ti=trnInputs(4);             %Indoor temperature (C) (input from previous iteration at current time step)
Wi=trnInputs(5);             %Indoor humidity ratio (kg/kg) (input from previous iteration at current hr)
qh=trnInputs(6);             %Building heating load (kJ/hr) (space load, only. input from the building)
qc=trnInputs(7);             %Building cooling load (kJ/hr) (space load, only. input from the building)
es=trnInputs(8);             %RAMEE sensible effectiveness (input from ANN)
el=trnInputs(9);             %RAMEE latent effectiveness (input from ANN)
et=trnInputs(10);            %RAMEE total effectiveness (input from ANN)
dts=trnInputs(11);           %The change in air temperature across the supply LAMEE (input from
ANN)
dws=trnInputs(12);           %The change in air humidity ratio across the supply LAMEE (input from
ANN)
Wif=trnInputs(13);           %Indoor humidity ratio at zero supply air flow rate (kg/kg) (input from the
                             building)
P=trnInputs(14);             %Local atmospheric pressure (atm) (input from TMY2 weather data file)
================================== The main body===================================
                    (1)  Check the building occupancy to determine the ventilation rate
if(mod(hr,24)<22&&mod(hr,24)>5)%Is the building occupied?
  occ=1;                     %Yes! Provide the required ventilation
  Mvent=Mvent;
else
  occ=0;                     %No! No outdoor ventilation air provided and the RAMEE is off
  Mvent=0;
end
                             (2)  If the building needs heating
if(qh>0)                     %Heating
  Msup=Mvent;                %Minimum ventilation rate during the heating
  Tso=To-dts;                %RAMEE increases the temperature of ventilation air
  Wso=Wo-dws;                %RAMEE increases the humidity of ventilation air
  Wmix=Wso;
  Tsup=max(Tso,14);          %The air is supplied to the space with a temperature not lower than 14
```

```
    Wsup=Wso;                       %No change in humidity of the air after leaving the supply LAMEE
    coilsens=Msup*Cp*(Tsup-Tso); %Calculation of the sensible heating load
    coillat=0;
end

ho=Cp*To+hfg*Wo;                    %Outdoor enthalpy
hi=Cp*Ti+hfg*Wi;                    %Indoor enthalpy
```

(3) If the building needs cooling

```
if(qc>0)                            %Cooling
    Msup=max(Mvent,qc/(10*Cp));  %Calculation of supply air flow rate
    Wsupmax=min(0.012,Winset-86400*1.2*(Wif-Winset)/Msup);
                                    %Calculation of maximum allowed supply humidity ratio to prevent indoor
                                    humidities greater than 0.012 (kg/kg)
    Tsup=Ti-qc/(Cp*Msup);           %Calculation of the supply air temperature
    Moa=Mvent;
      if(To<Tsup)                   %Part-load operation
        Moa=Mvent;
        Tsup=max(14,Tsup);
        Msup=max(Mvent,qc/(Cp*(Ti-Tsup)));
        es=min((Msup*(Tsup-Ti)+Moa*(Ti-To))/(Moa*(Ti-To)),es);
        A=0.5;
        Tso=To-es*(To-Ti);
        Wso=Wo-es*(Wo-Wi);
        Wmix=(Moa*Wso+(Msup-Moa)*Wi)/Msup;
        Wsup=Wmix;
        Tmix=(Moa*Tso+(Msup-Moa)*Ti)/Msup;
        coilsens=Msup*Cp*(Tsup-Tmix);
        coillat=0;
      else
        T=Tsup+273.15;
          pws=exp(-5800.2206/T+1.3914993-T*4.8640239e-2+T^2*4.176476e-5-T^3*1.445209e-
          8+6.545967*log(T));
        Ws=0.621945*(pws/(101325*P-pws));
        Wsup=min(Ws,Wsupmax);
        pw=101.325*P*Wsup/(0.621945+Wsup);
        a=log(pw);
        Tdew=6.54+14.526*a+0.7389*a^2+0.09486*a^3+0.4569*(pw)^0.1984;
        if(((ho>hi&&et<0&&To<Ti)||(ho<hi&&et>0&&To<Ti))||(dts==0))
                                    %Control based on outdoor/indoor enthalpy
          A=0;
```

```
        else
            qcoil=@(A)Msup*Cp*(((Moa*(To-A*dts)+(Msup-Moa)*Ti)/Msup)-Tsup)+min(floor(((Moa*(Wo-A*dws)+(Msup-Moa)*Wi)/Msup)/Wsup),1)*Msup*(Cp*(Tsup-Tdew)+hfg*(((Moa*(Wo-A*dws)+(Msup-Moa)*Wi)/Msup)-Wsup));          %Cooling load minimization function
            A=fminbnd(qcoil,0,1);
        end
            Tmix=(Moa*(To-A*dts)+(Msup-Moa)*Ti)/Msup;
            Wmix=(Moa*(Wo-A*dws)+(Msup-Moa)*Wi)/Msup;
        if(Wsup>=Wmix)          %Sensible cooling only, if the humidity ratio at the outlet of supply LAMEE
                                is belowe the maximum limit
            Wsup=Wmix;
            coilsens=Msup*Cp*(Tsup-Tmix);
            coillat=0;
            reheat=0;
        else                    %Sensible cooling and dehumidification
            coilsens=Msup*Cp*(Tdew-Tmix);
            coillat=Msup*hfg*(Wsup-Wmix);
            reheat=Msup*Cp*(Tsup-Tdew);
        end
    end
end
                        (4)  If the internal loads and the envelope heat loss balance out
if(qh+qc==0)                %No building load (the internal gains balance the heat loss)
    A=occ;                  %The operation of RAMEE depends on whether the building is occupied or not
    Msup=Mvent;             %If occupied, minimum ventilation is provided
    Tsup=Ti;
    Moa=Mvent;
    Tso=To-A*dts;
    Wso=Wo-A*dws;
    Wsup=Wso;
    Tmix=Tso;
    Wmix=Wso;
    coilsens=Msup*Cp*(Tsup-Tmix);
    coillat=0;
    reheat=0;
end
==========Sending the results to other TRNSYS components (i.e., the building and the printer)=========
trnOutputs(1)=Msup;         %Supply air mass flow rate (kg/kg) (input to the building)
trnOutputs(2)=Tsup;         %Supply air temperature (C) (input to the building)
```

```
trnOutputs(3)=Wsup;            %Supply air humidity ratio (kg/kg) (input to the building)
trnOutputs(4)=Moa;             %Outdoor ventilation rate (kg/hr) (input to the printer)
trnOutputs(5)=coilsens;        %Sensible heating/cooling load (kJ/hr) (input to the building)
trnOutputs(6)=coillat;         %Latent load (kJ/hr) (input to the building)
trnOutputs(7)=reheat;          %Reheat (kJ/hr)(input to the building)
trnOutputs(8)=A;               %RAMEE operation (0, 0.5 or 1; input to the printer)

mFileErrorCode=0
return
```

A.3. The HVAC System Equipped with a RAMEE and an Economizer in the Office Building

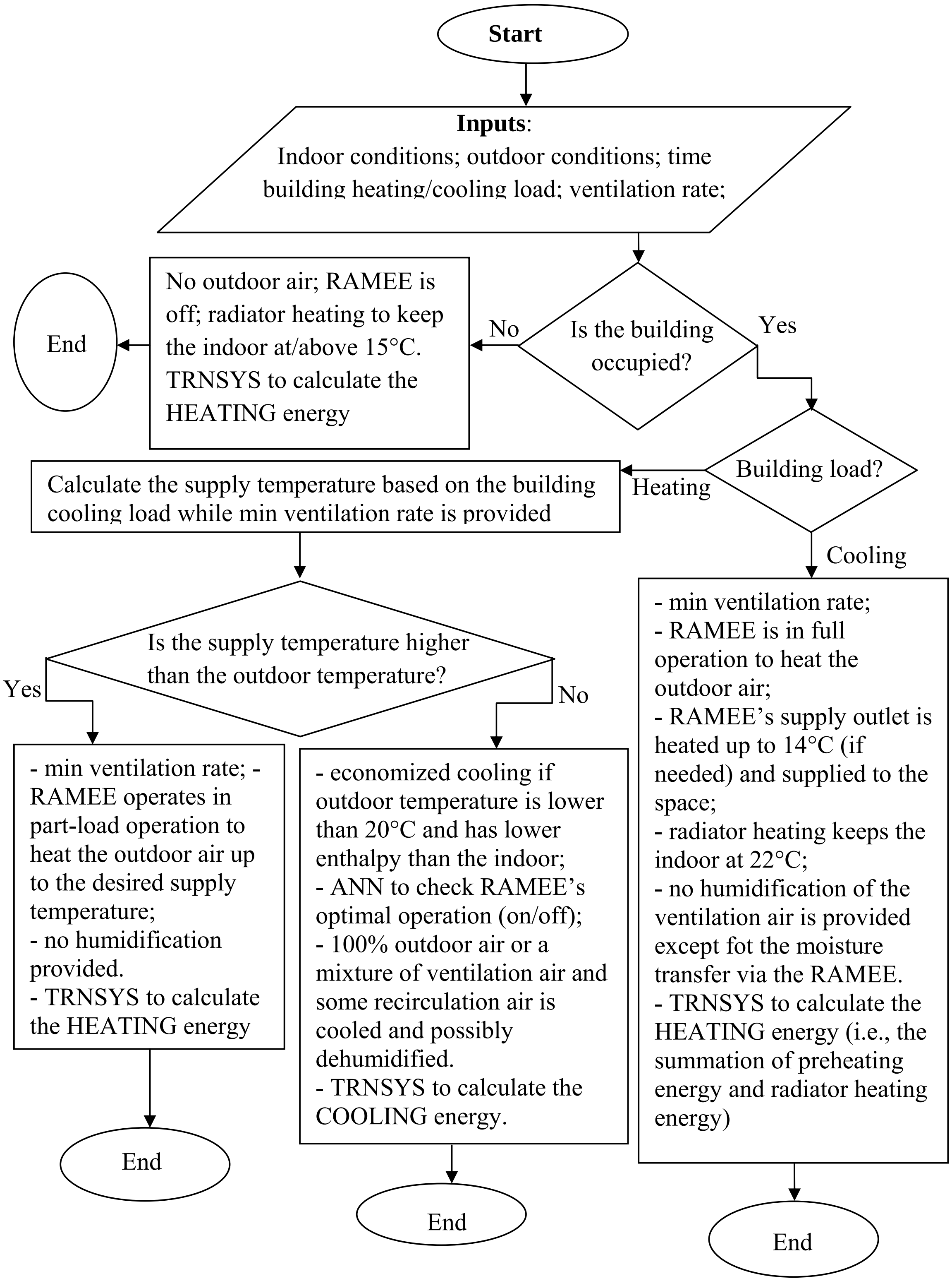

Figure A.3. Flowchart of the office building HVAC system equipped with a RAMEE and an economizer

```
================================= Setting the given values=============================
Mlim=400000                      %Maximum ducting capacity (kg/hr)
Mvent=50000;                     %Standard requirement for ventilation (kg/hr)
Cp=1.02;                         %Specific heat capacity of air (kJ/kg K)
hfg=2500;                        %Enthalpy of phase change (kJ/kg)
Winset=0.012;                    %Indoor maximum allowed humidity ratio (kg/kg)
===Receiving the inputs from other TRNSYS components (i.e. the building, the ANN and the weather data
file)===
hr=trnInputs(1);                 %Hour (input from TMY2 weather data file)
To=trnInputs(2);                 %Outdoor temperature (C) (input from TMY2 weather data file)
Wo=trnInputs(3);                 %Outdoor humidity ratio (kg/kg) (input from TMY2 weather data file)
Ti=trnInputs(4);                 %Indoor temperature (C) (input from previous iteration at current time step)
Wi=trnInputs(5);                 %Indoor humidity ratio (kg/kg) (input from previous iteration at current hr)
qh=trnInputs(6);                 %Building heating load (kJ/hr) (space load, only. input from the building)
qc=trnInputs(7);                 %Building cooling load (kJ/hr) (space load, only. input from the building)
es=trnInputs(8);                 %RAMEE sensible effectiveness (input from ANN)
el=trnInputs(9);                 %RAMEE latent effectiveness (input from ANN)
et=trnInputs(10);                %RAMEE total effectiveness (input from ANN)
dts=trnInputs(11);               %The change in air temperature across the supply LAMEE (input from
ANN)
dws=trnInputs(12);               %The change in air humidity ratio across the supply LAMEE (input from
ANN)
Wif=trnInputs(13);               %Indoor humidity ratio at zero supply air flow rate (kg/kg) (input from the
                                 building)
P=trnInputs(14);                 %Local atmospheric pressure (atm) (input from TMY2 weather data file)
A=trnInputs(15);                 %The operating condition of the RAMEE(input from APPENDIX B)

 ================================= The main body==================================
                    (1)  Check the building occupancy to determine the ventilation rate
if(mod(hr,24)<22&&mod(hr,24)>5)%Is the building occupied?
  occ=1;                         %Yes! Provide the required ventilation
  Mvent=Mvent;
else
  occ=0;                         %No! No outdoor ventilation air provided and the RAMEE is off
  Mvent=0;
end
                                 (2)  If the building needs heating
if(qh>0)                         %Heating
  Msup=Mvent;                    %Minimum ventilation rate during the heating
  Tso=To-dts;                    %RAMEE increases the temperature of ventilation air
  Wso=Wo-dws;                    %RAMEE increases the humidity of ventilation air
```

```
    Wmix=Wso;
    Tsup=max(Tso,14);              %The air is supplied to the space with a temperature not lower than 14
    Wsup=Wso;                      %No change in humidity of the air after leaving the supply LAMEE
    coilsens=Msup*Cp*(Tsup-Tso);   %Calculation of the sensible heating load
    coillat=0;
end

ho=Cp*To+hfg*Wo;                   %Outdoor enthalpy
hi=Cp*Ti+hfg*Wi;                   %Indoor enthalpy
                                       (3)  If the building needs cooling
if(qc>0)                           %Cooling
    Msup=max(Mvent,qc/(10*Cp));
    Wsupmax=min(0.012,Winset-86400*1.2*(Wif-Winset)/Msup);
    Tsup=Ti-qc/(Cp*Msup);
        if(To<Tsup)                %Part-load operation of RAMEE (minimum outdoor ventilation rate)
            Moa=Mvent;
            Tsup=max(14,Tsup);
            Msup=max(Mvent,qc/(Cp*(Ti-Tsup)));
            es=min((Msup*(Tsup-Ti)+Moa*(Ti-To))/(Moa*(Ti-To)),es);
            A=0.5;
            Tso=To-es*(To-Ti);
            Wso=Wo-es*(Wo-Wi);
            Wmix=(Moa*Wso+(Msup-Moa)*Wi)/Msup;
            Wsup=Wmix;
            Tmix=(Moa*Tso+(Msup-Moa)*Ti)/Msup;
            coilsens=Msup*Cp*(Tsup-Tmix);
            coillat=0;
        else
            if(A>0)                %Minimum outdoor ventilation if the RAMEE operates during the cooling
                                   season
                Moa=Mvent;
                Msup=max(Mvent,qc/(10*Cp));
                Tsup=Ti-qc/(Cp*Msup);
            else                   %Economized cooling if the RAMEE is off , the outdoor temperature is
                                   below 20C and outdoor enthalpy is lower than the indoor
                if(To<20&&ho<hi)
                    Tsup=max(14,To);
                    Msup=min(Mlim,max(Mvent,qc/(Cp*(Ti-Tsup))));
                    Moa=Msup*(Tsup-Ti)/(To-Ti);
                else
```

```
            Moa=Mvent;
            Msup=max(Mvent,qc/(10*Cp));
            Tsup=Ti-qc/(Cp*Msup);
         end
      end
      T=Tsup+273.15;
      pws=exp(-5800.2206/T+1.3914993-T*4.8640239e-2+T^2*4.176476e-5-T^3*1.445209e-
8+6.545967*log(T));
      Ws=0.621945*(pws/(101325*P-pws));
      Wsup=min(Ws,Wsupmax);
      pw=101.325*P*Wsup/(0.621945+Wsup);
      a=log(pw);
      Tdew=6.54+14.526*a+0.7389*a^2+0.09486*a^3+0.4569*(pw)^0.1984;
      Tmix=(Moa*(To-A*dts)+(Msup-Moa)*Ti)/Msup;
      Wmix=(Moa*(Wo-A*dws)+(Msup-Moa)*Wi)/Msup;
      Wsup=min(Ws,min(Wmix,Wsupmax));
      if(Wsup>=Wmix)
         Wsup=Wmix;
         coilsens=Msup*Cp*(Tsup-Tmix);
         coillat=0;
         reheat=0;
      else
         coilsens=Msup*Cp*(Tdew-Tmix);
         coillat=Msup*hfg*(Wsup-Wmix);
         reheat=Msup*Cp*(Tsup-Tdew);
      end
   end
end
```

(4) If the internal loads and the envelope heat loss balance out

```
if(qh+qc==0)
   A=occ;
   Msup=Mvent;
   Tsup=Ti;
   Moa=Mvent;
   Tso=To-A*dts;
   Wso=Wo-A*dws;
   Wsup=Wso;
   Tmix=Tso;
   Wmix=Wso;
```

```
        coilsens=Msup*Cp*(Tsup-Tmix);
        coillat=0;
        reheat=0;
 end

trnOutputs(1)=Msup;
trnOutputs(2)=Tsup;
trnOutputs(3)=Wsup;
trnOutputs(4)=Moa;
trnOutputs(5)=coilsens;
trnOutputs(6)=coillat;
trnOutputs(7)=reheat;
trnOutputs(8)=A;
trnOutputs(9)=Wsupmax;
trnOutputs(10)=Wmix;

mFileErrorCode=0
return
```

APPENDIX B

Applicability and Optimum Control Strategy of Energy Recovery Ventilators in Different Climatic Conditions

Mohammad Rasouli, Carey J. Simonson, Robert W. Besant

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Abstract

Energy Recovery Ventilators (ERVs) transfer energy between the air exhausted from building and the outdoor supply air to reduce the energy consumption associated with the conditioning of ventilation air. In this paper, the applicability of ERVs with sensible and latent effectiveness values in a practical range is studied using TRNSYS simulation program. The impact of ERV on annual cooling and heating energy consumption is investigated by modeling a 10-storey office building in four American cities as representatives of major climatic conditions. The results show that heat and moisture recovery can lead to a significant reduction in the annual heating energy consumption (i.e., up to 40%, which is 5% higher than heat recovery). Also, an ERV with the capability of moisture recovery may reduce the annual cooling energy consumption by 20% provided the ERV is properly controlled. Since the un-controlled operation of ERVs during the summer may increase the cooling energy consumption, an optimum control strategy is developed and verified in the paper. This optimum control strategy depends on ERV's latent to sensible effectiveness ratio. For instance, an ERV with equal sensible and latent effectiveness should be operated when either the outdoor enthalpy or temperature is greater than that of the indoor air.

Keywords: Energy recovery ventilator; Energy consumption; Control strategy; Climatic condition; Psychrometric

1. Introduction

Due to concerns over Indoor Air Quality (IAQ) and occupants' health, HVAC-related organizations have set standards that specify the minimum required ventilation rate depending on the type of buildings and occupancy[1-2]. Higher ventilation rates improve the IAQ by diluting pollutants such as airborne particles and volatile organic compounds. On the other hand, studies have shown that higher ventilation rates increase the building energy consumption in a majority of cases, especially during the heating season [3-7]. Therefore, more energy is required to provide the space with more outdoor ventilation air and consequently better IAQ.

Air-to-air energy exchangers which transfer energy between exhaust and supply airstreams were proposed as a solution to reduce the energy consumption associated with conditioning the ventilation air. In general, air-to-air energy exchangers can be divided into two groups: i.e., heat recovery systems which transfer only sensible heat, and heat and moisture recovery systems which transfer both sensible and latent energy. Some research has been conducted to study the applicability and beneficial aspects of heat recovery systems [8-10] and heat and moisture recovery systems [11-14]. These studies have shown that the utilization of an ERV decreases the annual heating energy consumption significantly; however, it may lead to higher cooling energy demands for particular outdoor conditions during the summer [15]. This demand for higher cooling energy mainly occurs during summer days when the outdoor temperature is lower than the indoor temperature and while cooling is still required to meet the internal loads and solar radiation gains. If an ERV is operated under such conditions, it may heat the supply air above the desired supply temperature and lead to a higher cooling load.

Liu et al. [14] studied the applicability and energy savings with enthalpy exchangers employed in five Chinese cities. Their study was limited to heating season only, and the results showed that the heating energy could be reduced by 20% when an ERV with 75% total effectiveness was employed. Zhou et al. [12] simulated an ERV system in two locations with different climatic conditions in China using EnergyPlus, a dynamic building simulation model. They reported that the application of ERV reduces the energy consumption during the winter, however, ERV operation in cold climate (Beijing) was uneconomical when the cooling set-point was above 24°C. Fauchoux [13] presented the undesirable impact of an uncontrolled ERV (energy wheel) on cooling loads in mild and cold climates (Vancouver and Saskatoon, respectively). The results showed that cooling energy consumption can be reduced by applying a temperature-based control strategy. Zhang et al. [11] studied the applicability of heat and moisture recovery systems in Hong Kong. They classified the psychometric chart into six regions based on outdoor temperature and humidity and illustrated that by turning the ERV off in a region bounded by cooling and heating set-point temperatures (called neutral ventilation region), the operation of ERV reduces both annual heating and cooling energy consumption. Mumma [16] used a control strategy for enthalpy wheels employed in dedicated outdoor air systems. Their control scheme did not allow the operation of ERV when the outdoor enthalpy was lower than the indoor air while the outdoor humidity was higher than the humidity of the air supplied to the conditioned space. Simonson et al. [15,17] experimentally validated two strategies to control energy wheels by applying an operating condition factor [18] which presented the ratio of latent to sensible energy potential of inlet airstreams. Since this factor presents both mechanisms of energy transfer in an ERV, i.e. latent and sensible, it is included as a control option in this study beside other alternatives proposed in the literature [11,16].

Although some research has been conducted on applicability of ERVs, it has been limited to specific climates, particular types of energy recovery systems or has not included the study of optimum control for energy recovery systems in cooling mode. The purpose of this study is to investigate the annual energy savings with the use of ERVs for a practical range of sensible and latent effectiveness. Also, an optimum strategy to control the operation of ERVs in cooling season which predicts the maximum cooling energy savings is introduced.

2. Model description

2.1. Software

A TRNSYS model is used as the building energy simulation tool in this paper. TRNSYS [19] is a FORTRAN-based transient system simulation program which is designed to solve complex thermal systems by breaking them down into less complicated components. The main advantage of TRNSYS is the capability of solving each thermal component independently. Then, the components are coupled to solve the main thermal system [20]. Thermal Energy System Specialists, TESS, is one of the major developers of TRNSYS component libraries and TRNSYS 16 and the Second version of TESS libraries are used in this study [21].

2.2. Building description

The building used for this study is chosen from a set of buildings known as the US office building stock. The U.S. office building stock has been classified into 25 categories based on a study carried out by researchers at the Pacific Northwest national Laboratory, PNL. As such, each of these buildings represents a specific percentage of the US office building stock as determined by a Commercial Building Energy Consumption Survey

(CBECS) [22]. The details for this set of office buildings have been described by Briggs et al. [23] and Crawley et al. [24].

A 10-storey office building with total floor area of 28,800 m^2, representing 3.34% of the existing U.S. office floor area (in 1995) [23], is selected for this study. All of the floors are to be occupied and have to be conditioned, except for six elevators that operate in the building and are considered as an unconditioned single zone. The building description is taken from the PNL studies [23] and includes several building parameters required for an energy analysis. These parameters are defined as a building template in TESS loads and structure library [21] and are used in this paper. For this building, lighting, occupancy and receptacle have maximum intensity of 23.1 (W/m^2), 2.4 (Person/100 m^2) and 6.6 (W/m^2), respectively. Fig. 1 presents the hourly schedule of the fractional internal loads and HVAC system operation with respect to the peak values.

TESS component type 571 [21] was used to calculate the building infiltration rate. This component determines the infiltration rate and infiltration heat loss/gain as a function of the wind speed, indoor and outdoor temperatures, ambient pressure and humidity at each time step.

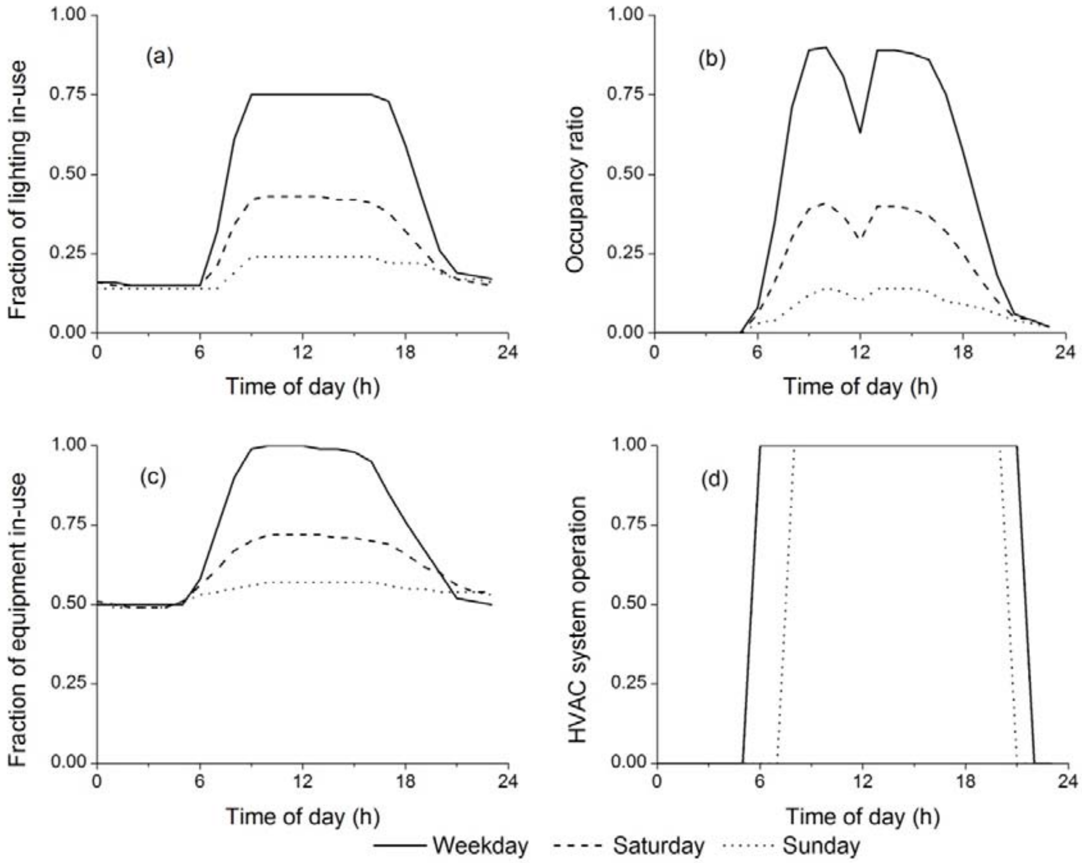

Fig. 2. Schedules for (a) lighting, (b) occupancy, (c) receptacle load and (d) the HVAC system operation in the 10-storey office building

2.3.HVAC System

TRNSYS user has the option to define desired heating and cooling temperature set-points so that the program will calculate the required energy rates to meet the building and ventilation loads. In this way, the design of the HVAC system and the determination of condition of the air supplied to the space (which meets the building loads) are not required and the program is used to calculate the required power if the HVAC system has to be designed. This model simplification is not expected to affect the accuracy of results since the focus of this study is the determination of optimum control strategy for the cooling mode and annual energy savings based on comparisons of the ideal energy consumptions.

The indoor condition, as presented in Table 1, is set considering ASHRAE recommendations on thermal comfort for occupants in summer and winter [25].

Table 1. Design indoor conditions

	Summer		Winter	
	occupied	**unoccupied**	**occupied**	**unoccupied**
Temperature (°C)	24	No control	22	T ≥15
Relative Humidity (%)	50	No control	30	No control

2.4. Ventilation

Outdoor ventilation air at a constant rate of $13m^3/s$ (0.6 ACH), limited to occupied hours shown in Fig. 1-b, is supplied to the building to meet ASHRAE Standard 62-2004 [1] requirements. Considering the lower occupancy of the building during weekends, the ventilation rate is reduced to 50% and 25% of the design flow rate on Saturdays and Sundays, respectively. The outdoor ventilation air flow rate is determined considering the effective parameters, such as area, occupant density and the building type.

3. Psychrometrics

Whether an ERV should be operated or stopped depends on several factors such as, the indoor and outdoor conditions and whether the building requires auxiliary heating or cooling energy. In this section, these different scenarios are presented by dividing the psychrometric chart into sub-regions that establish the conditions when the ERV needs to be controlled.

By selecting the summer indoor comfort condition (24°C, 50% RH) as a reference summer indoor condition (point a, Fig. 2), the psychrometric chart can be divided into four areas based on the outdoor temperature and humidity ratio. Furthermore, the area with lower outdoor temperatures and humidities than the indoor air can be also divided into 3 regions:

very cold which requires heating (i.e., region 1), cool which needs no heating or cooling (e.g. spring or fall, and shown as region 2) and moderate which needs a cooling system (e.g. cool summer days and shown as region 3). All six regions are shown in Fig. 2.

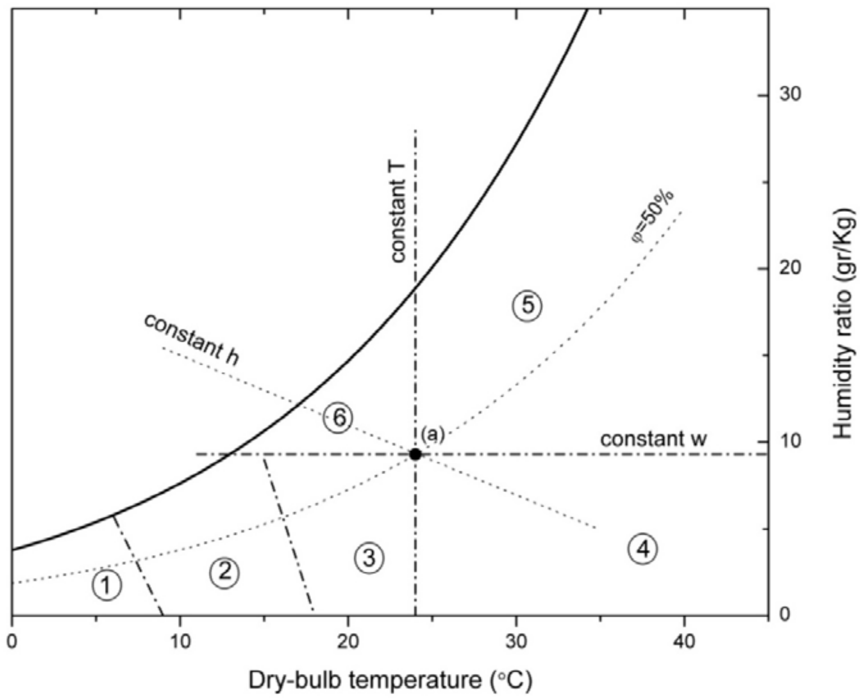

Fig. 3. Schematic view of the 6 different psychrometric chart regions

Region 1(heating season): When outdoor temperature is lower than the heating set point, and the internal heat gains do not satisfy the space heating demand, the heating system needs to be operated. As is shown in Fig. 2, due to low temperatures and humidities in region (1), conditioning of outdoor supply air requires heating and humidification. The operation of ERV for such outdoor conditions transfers both heat and moisture from exhaust airstream to the outdoor supply air. The ERV operation is beneficial for this region and it should be operated at its maximum capacity for heat and moisture recovery. Therefore, during the occupation period for region (1), a control signal ensures that the energy exchanger is operating. It should be noted that solar radiation heat gains and internal heat sources affect the temperature at which the heating system may come into operation. These gains vary during the day and from day to day for each location. As such, the line defining the boundary of region 1 (shown as a dashed line in Fig. 2) should not be thought of a fixed temperature.

Region 2 (no heating or cooling): Includes the conditions under which the building heat loss due to conduction and ventilation balances the internal loads and solar radiation gains. In such a case, which mostly occur during the spring and fall, no heating or cooling is required and the ERV is off. The introduction of cool outdoor ventilation air directly to the space balances the internal and solar heat gains. An economizer can be employed for such operating conditions, but the study of energy savings associated with employing economizers is out of the scope of this study.

Regions 3, 6 (moderate cooling season): During moderately cool summer days when the outdoor temperature is lower than the cooling set point, but the internal loads and solar radiation gains are significant, a cooling system is required. Such conditions can be divided into two regions based on humidity ratio; i.e., cool-dry (region 3) and cool-humid (region 6). The ERV should be off when outdoor condition falls into region 3, since the ERV may heat and humidify the cool and dry outdoor supply air thus increasing the cooling and dehumidification loads for the cooling system. Operation of ERV for region 6 will heat and dehumidify the ventilation air. This can be beneficial when dehumidification in ERV is greater than the heating (assuming similar energy cost as for sensible cooling and dehumidification), so the operation of the ERV during the cooling season when the outdoor condition falls into regions 3 and 6 should be controlled to prevent the increase of cooling energy consumption. The best strategy to control the ERV operation for these regions is proposed in this paper.

Regions 4 and 5 (cooling season): High outdoor temperature and gains from solar and internal heat sources necessitate the operation of the cooling system. The outdoor conditions during hot summer days can also be divided into two regions: i.e. hot-dry (region 4) and hot-humid (region 5). ERV operation in these regions cools the outdoor air and reduces the sensible cooling load. An ERV with capability of moisture transfer, in addition to sensible cooling, will dehumidify the outdoor air for region 5.

4. Climatic conditions

4.1. Weather data

Typical Meteorological Year (TMY 2 weather data format) [26] contains typical hourly weather data required for yearly building energy analysis. These data, which are compatible with TRNSYS models, were obtained from the National Renewable Energy Laboratory [27] and used for different locations described in section 4.2. TRNSYS interpolates the hourly weather data for time steps smaller than 1 hour.

4.2. Choice of representative cities

Briggs et al. [28] developed a new climate classification method to be used for building energy analyse. Eight climatic zones were suggested based on a temperature-based classification ranging from subarctic to very hot. In addition, these zones were divided into three humidity-based subdivisions, i.e. humid, marine and dry. The combination of temperature-based and humidity-based classification resulted in 17 climatic zones and sample cities were introduced as representatives of each climate. Table 2 presents a summary of the climatic zones studied in this paper, followed by the representative American city, dry bulb (DB) and wet bulb (WB) temperatures for heating and cooling seasons.

Table 2. Major climatic zones and representative American cities used in this study [28,29]

Climate type	Hot-Humid	Hot-Dry	Cool-Humid	Cold-Dry
US representative	Miami, FL	Phoenix, AZ	Chicago, IL	Helena, MT
Elevation (m)	9	337	205	1179
Latitude	25.82N	33.44 N	41.99N	46.61N
Heating-DB (°C)	10.9	5.2	-16.6	-22.3
Cooling-DB (°C)	32.6	42.3	31.6	31.8
Cooling-WB (°C)	25.3	21.0	23.0	15.9

Fig. 3 presents the hourly TMY2 weather data on the psychrometric chart and the distribution of outdoor condition in different regions for one year in Phoenix during the period that HVAC system operates (Fig. 1-d). Fig. 4 presents the fraction of outdoor conditions that fall into each psychrometric region for all locations.

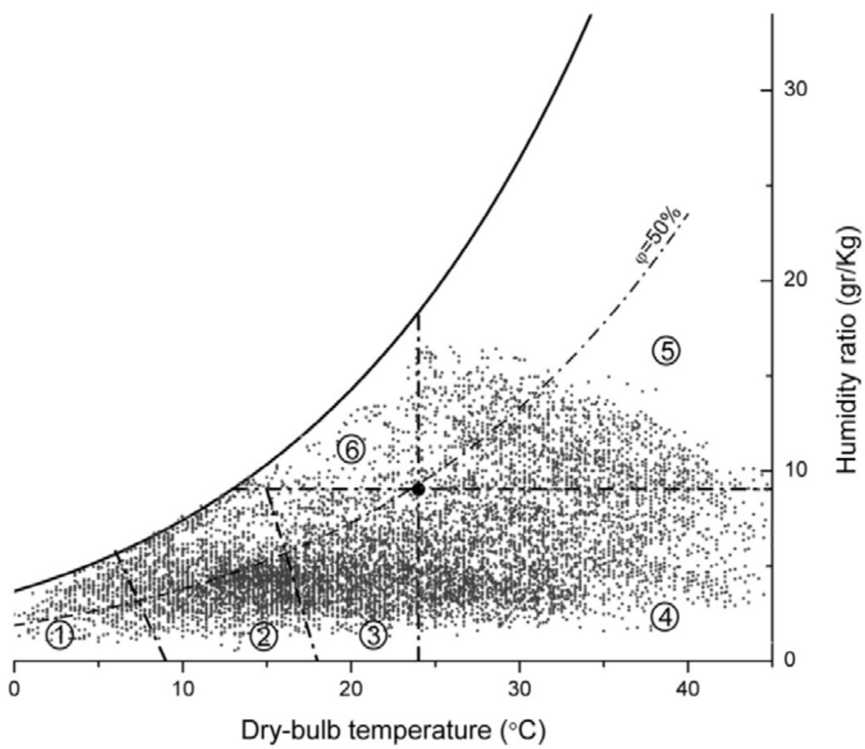

Fig. 3. Hourly outdoor conditions for one year in Phoenix

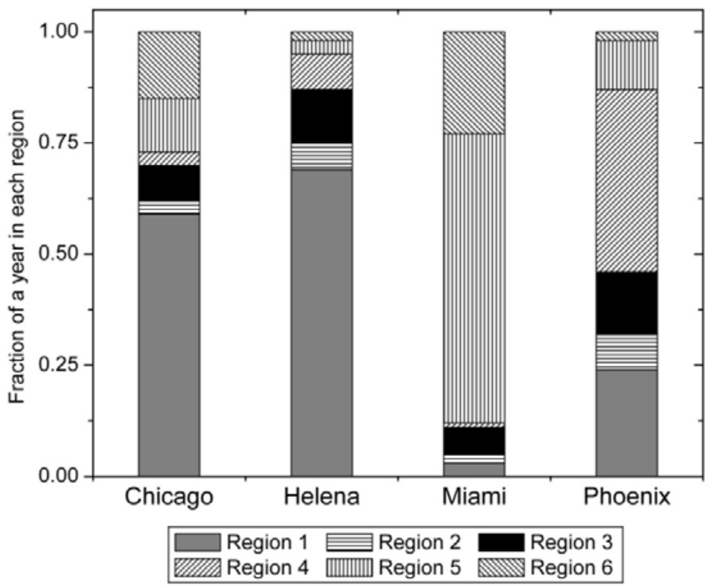

Fig. 4. Yearly distributions of outdoor conditions for different psychrometric chart regions for each city

Good agreement is observed when comparing Brigg's [28] classification for each city presented in Table 2 and the data obtained from standard TMY2 weather data for each location presented in Fig. 4. Miami represents a hot-humid climate since 88% of outdoor conditions fall into regions 5 and 6. Phoenix has about 53% of the time in a year in hot regions (regions 4 and 5) and it represents a hot-dry climate. The heating system is active for 69% of the time in a year in Helena which represents a cold climate, and Chicago represents a cool climate with 25% of the time in a year in moderate conditions (regions 2, 3 and 6), and 59% in cold condition (region1).

It is also shown that the outdoor conditions fall into regions 3 and 6 for a significant fraction of the cooling season in all locations. These are the regions in which the operation of ERV should be controlled. Regions 3 and 6 account for about 60% of the cooling season in cold climates (Helena and Chicago), and 25% of the cooling season in hot climates (Miami and Phoenix).

5. Results and discussions

A series of yearly simulations were run to investigate the energy saved by employing ERVs in the 10-storey office building for four selected cities located in different climates. As the base case, no ERV was in operation and the cooling or heating equipment had to meet building and ventilation loads. For the heating season, which requires no ERV control, the savings with ERVs in a practical range of sensible and latent effectiveness (recommended by ASHRAE Standard 90.1[30]) are determined. For the cooling season, the undesirable impact of an un-controlled ERV on cooling energy consumption is presented. Afterwards, the best strategy to control the operation of ERVs during the cooling mode is studied and compared to the case of temperature-based control. Finally, for a practical range of sensible and latent effectiveness, the annual cooling energy savings with ERVs operating under the best control strategy are presented.

Fig. 5 presents the results for annual cooling and heating energy consumption for the base case when no ERV is employed.

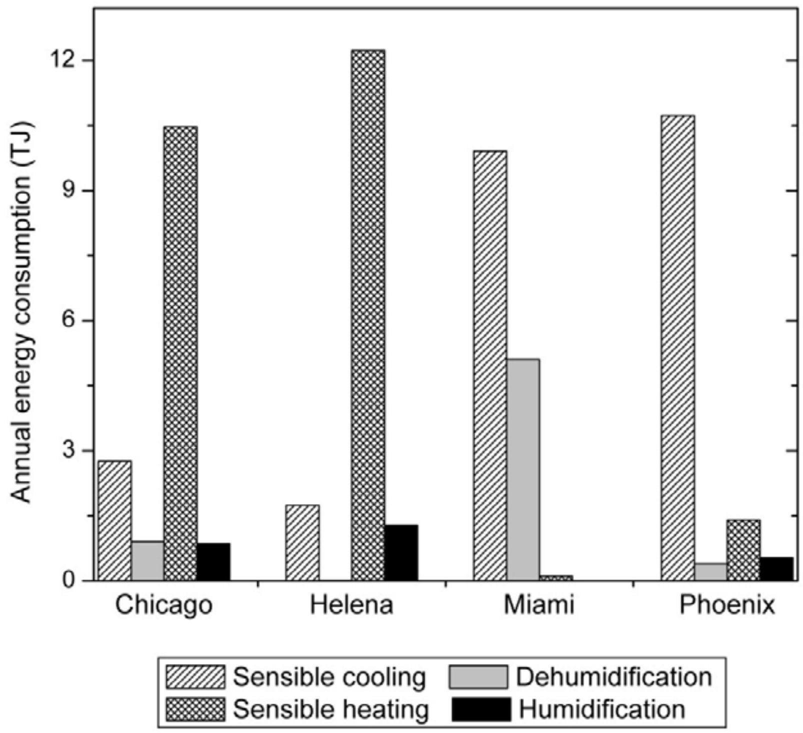

Fig. 5. Annual energy consumption in each location for the base case (no ERV)

As shown in Fig. 5, sensible heating accounts for a majority (about 90%) of the total heating energy consumption in cold climates (Chicago and Helena). Therefore, it is expected that sensible heat recovery will have the largest impact on the energy saved during the winter. During the summer, the energy required to dehumidify the space is a significant part of the total cooling energy consumption in humid climates (Chicago and Miami), and both sensible heat recovery and moisture transfer are expected to be important.

5.1. Heating season

As discussed previously, the ERV heats and humidifies the cold and dry supply outdoor air when the building requires heating; therefore, no ERV control is required during the heating season. In this section, energy savings by employing ERVs with sensible and latent effectiveness values in practical range of 55% to 95% are investigated. This is the range recommended by ASHRAE standard 90.1 [30] since it requires ERVs with effectivenesses greater than or equal to 55%. The case of sensible only heat exchanger (ε_l=0) is also studied to present the impact of moisture recovery on energy savings.

Fig.6 presents the results for annual heating energy savings with ERVs. Due to insignificancy of heating energy consumption in Miami (as shown in Fig.5), it is not included in the study of ERV applications during the winter.

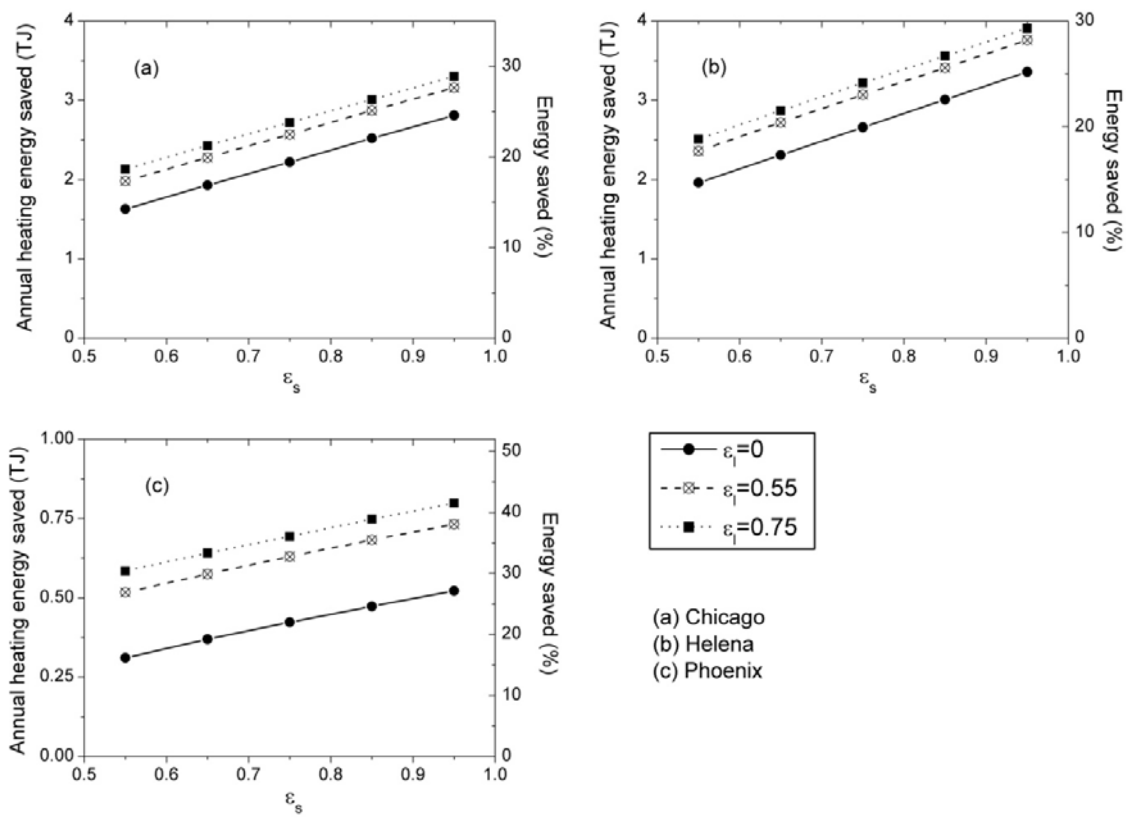

Fig. 6. Annual heating energy saved by employing ERVs in the practical range of effectiveness, (a) Chicago, (b) Helena and (c) Phoenix

As presented in Fig.6, the total heating energy saved with ERV is more significant in cold climate (Chicago and Helena) than Phoenix in hot climate. Also, the increase of both sensible and latent effectiveness leads to higher energy up to 30% in cold climate. The possibility of moisture recovery in dry climates (Phoenix and Helena) which decreases the humidification load has more considerable impact on energy savings than humid climates (Chicago). It is also shown that the impact of sensible effectiveness on energy savings in winter is more significant in cold climates, since the sensible heating load accounts for the majority of total heating load (about 90% as shown in Fig.5).

It should be noted that 60% of the buildings listed in the U.S office building set [23] have envelopes with lower overall U-values (i.e. they have better insulation) than the office building studied in this paper. For buildings with lower building heating loads, the ventilation load accounts for a larger portion of the total heating energy consumption, and application of ERVs can lead to higher percentages of energy saving (compared to the savings presented in Fig.6).

5.2. Cooling season

5.2.1 Study of un-controlled operation

In order to present the undesirable impact of un-controlled operation of ERV on energy consumption, the results of the base case (i.e., no ERV shown in Fig.5) are compared to the case that the ERV is in operation but not controlled (i.e., continuous operation along with the cooling system). The results for the annual cooling energy saving for two different ERVs are presented in Fig.7.

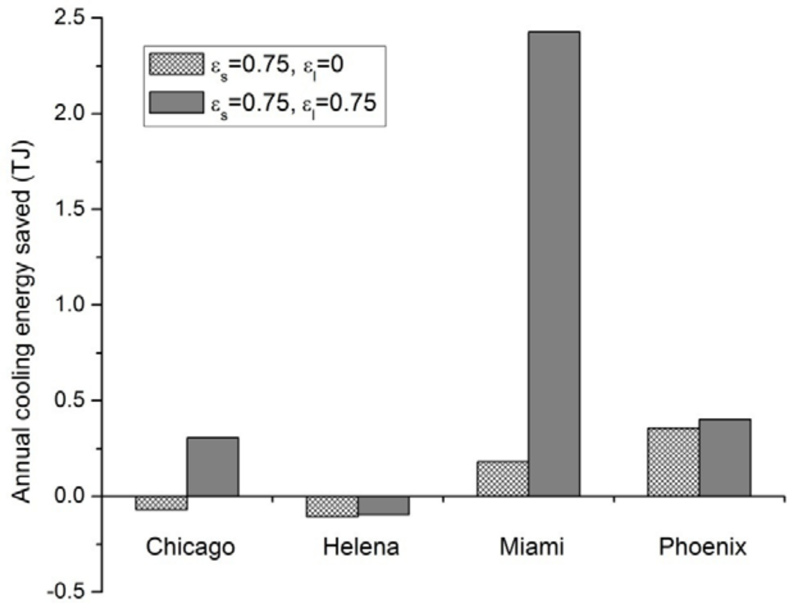

Fig. 7. Annual cooling energy saved in each city by employing an un-controlled ERV

Negative values for energy savings, as shown in Fig.7, indicate that the un-controlled operation of ERV in cool and cold climates increases the cooling energy consumption. However the uncontrolled operation of ERV can save energy in hot climates as shown in Fig. 7, it may not be the maximum potential saving achievable with an energy recovery system.

As discussed previously, the operation of ERVs for specific outdoor condition (regions 3 and 6, Fig.2) may heat and/or humidify the ventilation air. Therefore, control strategies should be applied to prevent the operation of ERVs when it transfers energy from exhaust air to the outdoor supply air during these times when it adds to the cooling load.

5.2.2 Control alternatives

Temperature-based control (T-based) is the most common strategy to control energy recovery systems. This control allows the operation of ERVs only when the outdoor temperature is greater than the indoor temperature. Studies have shown that the operation of ERV under this control strategy cools the outdoor supply air and consequently reduces the sensible cooling energy required in the cooling unit [5,11,16].

However T-based control properly determines the condition at which reduction of sensible cooling energy can be achieved, it may not be able to predict the maximum potential saving since it does not consider the impact of moisture recovery. In this part, a theta-based control which considers both heat and moisture transfer in ERV is introduced, and the savings with this control is compared to present temperature-based strategy.

Theta-based control: Fig. 8 schematically shows the process applied to cool-humid air when passing through an operating ERV.

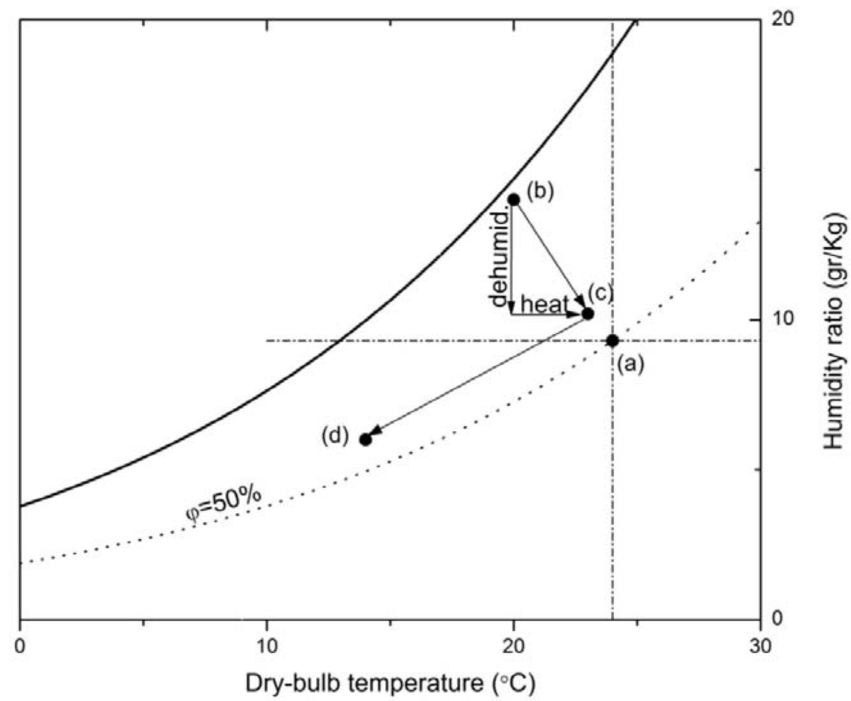

Fig. 8. Heating and dehumidification of cool-humid outdoor air during the operating of ERV

The operation of ERV in this region heats and dehumidifies the ventilation air and brings the outdoor air to state c (i.e. process b→c in Fig. 8). This increases the sensible energy cooling load in the cooling unit, but reduces the dehumidification load. As the ventilation air leaves the ERV, it has to be cooled and dehumidified to state d (i.e. process c→d) using auxiliary energy provided by the HVAC system. Therefore, the operation of ERV in cool-humid region should be controlled and limited to the conditions when the dehumidification in ERV is greater than the heating (assuming equal costs for cooling and dehumidification). This can be mathematically expressed as:

$$|Q_{lat,rec}| \geq |Q_{sens,rec}| \tag{1}$$

$$|\dot{m}_a h_{fg}(w_{sup,in} - w_{sup,out})| \geq |\dot{m}_a C_p(T_{sup,in} - T_{sup,out})| \tag{2}$$

But,

$$\varepsilon_l = \frac{w_{sup,in} - w_{sup,out}}{w_{sup,in} - w_{exh,in}} \tag{3}$$

$$\varepsilon_s = \frac{T_{sup,in} - T_{sup,out}}{T_{sup,in} - T_{exh,in}} \tag{4}$$

And, the substitution of equations (3) and (4) into equation (2) results in:

$$\frac{h_{fg}\varepsilon_l}{C_p\varepsilon_s}\left|\frac{w_{sup,in} - w_{exh,in}}{T_{sup,in} - T_{exh,in}}\right| \geq 1 \tag{5}$$

Considering that the heat and moisture transfer in the ERV do not occur in the same direction in cool-humid condition, equation (5) can be rearranged to give:

$$\frac{w_{sup,in} - w_{exh,in}}{T_{sup,in} - T_{exh,in}} \leq \frac{C_p}{h_{fg}} \cdot \frac{-\varepsilon_s}{\varepsilon_l} \tag{6}$$

For a given ERV (known sensible and latent effectiveness) and design indoor condition, equation (6) determines the outdoor condition (temperature and humidity ratio) under which dehumidification of ventilation air in ERV is greater than the heating. During such conditions, a net positive energy is transferred from the supply air to the exhaust and the thermal power required to condition the ventilation air is decreased. As an example, the load of the cooling system for 20°C and 70% RH outdoor condition (24°C and 50% RH indoor) when employing different ERVs is presented in Fig. 9.

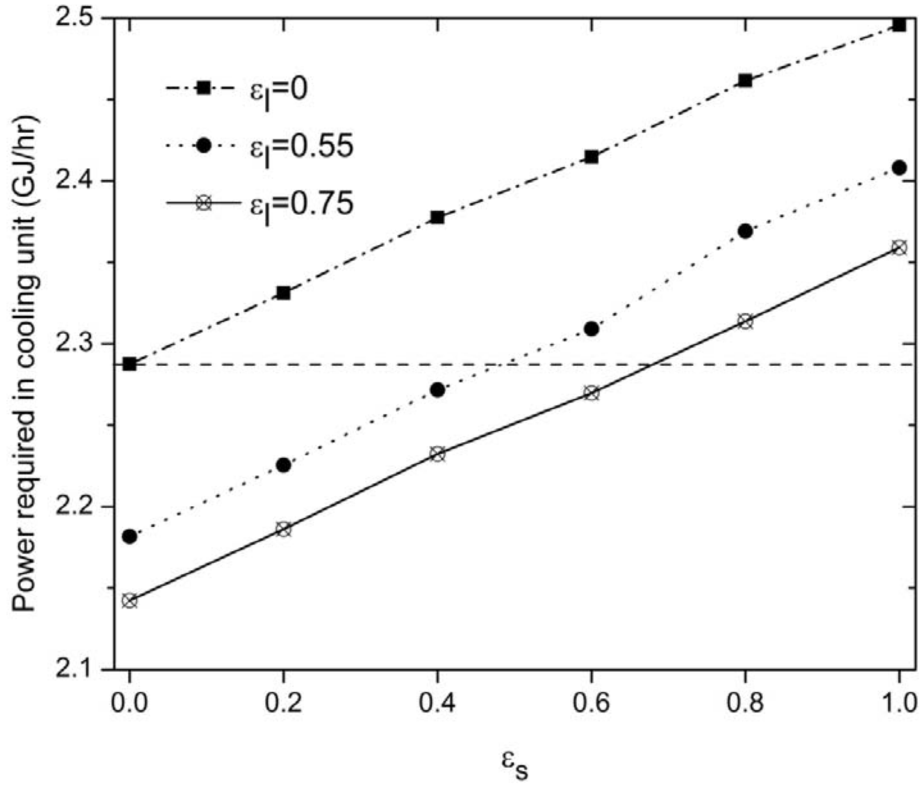

Fig. 9. Cooling load when employing different ERVs at 20°C and 70% RH outdoor condition

The dashed line shows the power required in the cooling system when no ERV is in operation. As shown in the figure, for a certain range of sensible and latent effectiveness, the cooling power can be reduced by employing an ERV. However, the cooling system may require higher power when employing ERVs with sensible and latent effectiveness values within a specific range. Only ERVs with sensible and latent effectiveness values which satisfy inequality given in equation (6) can reduce the cooling load at any given cool-humid outdoor condition.

It should also be noted that the outdoor condition at 20°C and 70%RH has a lower enthalpy than the indoor; however, the cooling load can be reduced for a specific range of sensible and latent effectiveness. This is in contrast with enthalpy based controls [16] which do not allow the operation of ERV when outdoor enthalpy is lower than the indoor.

Simonson and Besant [18] presented an operating condition factor which represents the ratio of latent to sensible energy potentials of inlet airstreams and is defined as:

$$H^* = \frac{h_{fg}\Delta w}{C_p \Delta T} \cong 2500 \frac{w_{sup,in} - w_{exh,in}}{T_{sup,in} - T_{exh,in}} \tag{7}$$

By applying the definition given in equation (7), equation (6) can be simplified as:

$$H^* \leq \frac{-\varepsilon_s}{\varepsilon_l} \tag{8}$$

The lines of constant H^* are shown in Fig. 10 and the hatched region is the condition which satisfies equation (8).

The hatched region is bounded by two legs diverging from summer indoor condition; one is fixed at constant temperature (i.e., summer indoor temperature) and the other produces an angle of θ which can be defined as:

$$\theta = tan^{-1}\left(\frac{\varepsilon_l}{\varepsilon_s}\right) = tan^{-1}\left(-\frac{h_{fg}\Delta w}{C_p \Delta T}\right) = tan^{-1}(-H^*) \tag{9}$$

The operation of ERV when the outdoor condition falls into the hatched region decreases the dehumidification load of the cooling unit more than it increases the sensible cooling load, and a net energy is transferred from supply air to the exhaust. Clearly, the ERV has the maximum potential savings when it operates for both outdoor temperatures greater than the indoor and the hatched region shown in Fig. 10.

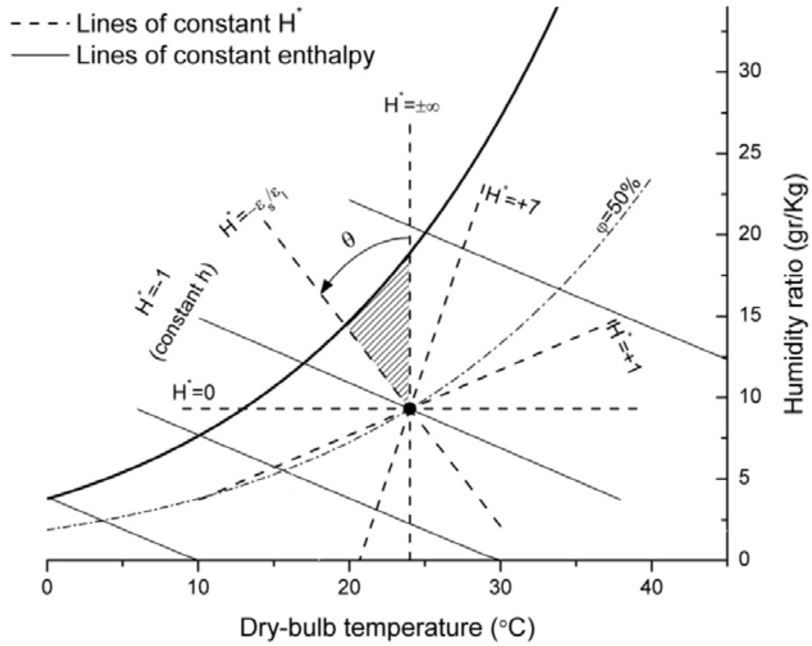

Fig. 10. Schematic view of H^*-constant lines and the region satisfying equation (8) on psychrometric chart

As is shown in equation (9), angle θ depends on the latent to sensible effectiveness ratio. For sensible-only heat exchangers (ε_l=0), the maximum saving is achieved when θ=0° is applied. This means ERV should not operate for outdoor temperatures lower than the indoor. For ERVs with equal sensible and latent effectiveness values, the maximum energy saving is achieved when θ=45°. For this angle, the hatched region (shown in Fig. 10) is bounded by a line of constant enthalpy and a line of constant temperature which pass through the indoor condition. This means that such an ERV should operate when either the outdoor temperature or enthalpy is greater than that of the indoor air. In general, as the latent effectiveness increases (more capability to dehumidify in ERV), θ increases and ERV can operate for a wider range of outdoor conditions in cool-humid region.

A series of computer simulations were run to investigate the validity of the theory of this optimum control strategy. For three different ERVs, angle θ was varied from 0° to 180°

to determine the angle at which the maximum saving was achieved. For each ERV, it was expected that the maximum saving occurs when $\theta = tan^{-1}\left(\frac{\varepsilon_l}{\varepsilon_s}\right)$ was applied. Fig. 11 presents the savings with different ERVs at any particular angle of θ ranging from 0° to 180°. The saving at θ=0° is due to the operation of ERV in outdoor conditions with higher temperature than the indoor.

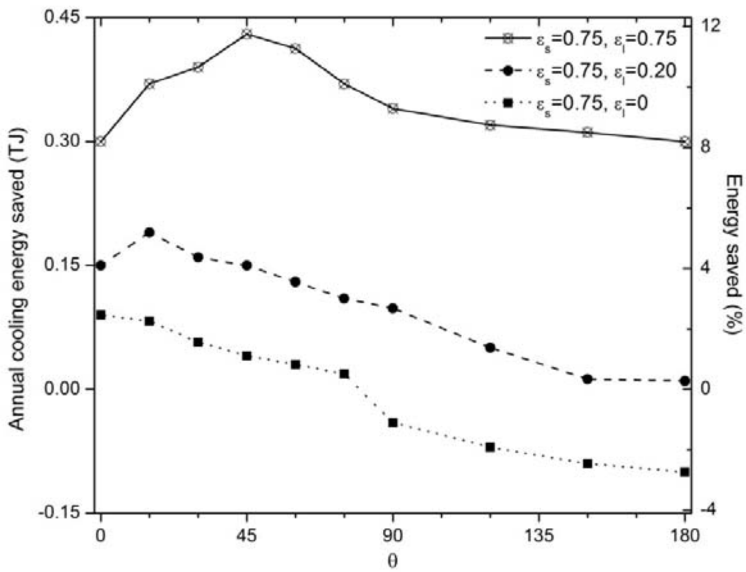

Fig.11. Variation of annual cooling energy saving with θ for three different ERVs in Chicago

Good agreement is observed between simulation results and the theory. For a sensible-only heat exchanger, the maximum saving is achieved when θ=0° and the operation of ERV should be limited to outdoor temperatures higher than the indoor. This is similar to T-based control and should be applied for sensible-only ERVs. Also, applying θ=180° - which corresponds to uncontrolled operation of ERV- increases the annual cooling energy consumption (negative energy saving).

For an ERV with equal sensible and latent effectiveness values (i.e. ε_s=0.75 and ε_l=0.75), the maximum energy saving is achieved when θ=45°. This means that such an ERV should be in operation when the outdoor condition has higher temperature or enthalpy than the indoor.

For the case of ε_s=0.75 and ε_l=0.20, optimum θ is found to be 15° ($tan^{-1}\left(\frac{0.20}{0.75}\right) = 15°$). For outdoor conditions confined between θ=15° and θ=45°, however the outdoor enthalpy is higher than the indoor, the operation of ERV increases the energy consumption. Again, this is in contrast with enthalpy-based control strategy [16] that always lets the ERV operate when outdoor enthalpy is greater than the indoor.

It should be noted that Mumma's enthalpy-based control strategy [16] does not allow the operation of ERV when the outdoor conditions fall within a triangle in hot-dry region confined by three lines; i.e., a line of constant temperature crossing the indoor condition, a line of constant enthalpy crossing the indoor condition and a line of constant humidity ratio crossing dew point temperature of the air supplied to the conditioned space. In a real HVAC system, such outdoor air needs to be dehumidified since the humidity ratio is greater than the supply humidity ratio; whereas, an ideal HAVC system may apply a sensible-only cooling process for such outdoor conditions. Since an ideal HVAC system (which does not consider the supply condition) is applied in this paper, the above mentioned triangle is not considered. It is worth mentioning that the outdoor conditions do not fall in the triangle except for hot and dry climates (e.g., Phoenix). Also, the ERV won't transfer a significant amount of energy in this region due to small temperature (and humidity) differences between indoor and outdoor air. Therefore, control of the ERV many not be critical in this region.

5.2.3 Energy savings with the optimum control

Fig. 12 presents the power required in the cooling system during one working day when different strategies are applied to control the operation of the ERV. The ERV has sensible and latent effectiveness of 75% and the building is located in Chicago.

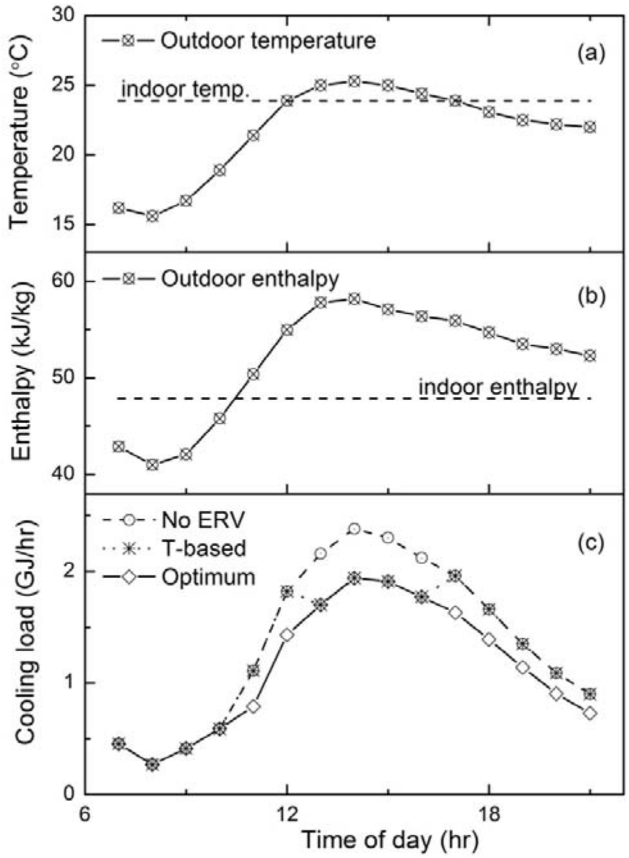

Fig. 12. Savings with an ERV in a summer day in Chicago; (a) outdoor temperature, (b) outdoor enthalpy and (c) cooling load with different control strategies

Based on the optimum control, such ERV with equal sensible and latent effectiveness should be operated when the outdoor enthalpy or temperature is greater than the indoor. But, with T-based control, the ERV operates only for outdoor temperatures higher than the indoor. As shown in Fig. 12, both of the controls don't allow the ERV operation for the first hours when both outdoor enthalpy and temperature are lower than the indoor (7:00-11:00). From 11:00 to 13:00 when the outdoor temperature is still below the indoor temperature, the optimum control lets the ERV operate since the outdoor enthalpy is higher than the indoor. This leads to a reduction in cooling power in this time period with optimum control. From 13:00 to 17:00, both outdoor temperature and enthalpy are higher than the indoor and both controls make the ERV operate to reduce the cooling load. For the rest of the day (17:00-

21:00), the outdoor temperature goes below the indoor, but the enthalpy still stays above. The application of optimum contol allows the operation of the ERV for this time period which reduces the required cooling power. As shown in the figure, T-based control is unable to meet all the potential savings compared to the proposed optimum control.

Fig. 12 presented the savings with an ERV for one typical summer day in Chicago. Annual cooling energy savings in different locations when employing ERVs in practical range of sensible and latent effectiveness under optimum control strategy (optimal θ) are presented in Fig. 13. As is shown in Fig. 5, the cooling energy consumption in Helena as representative of cold climate is found insignificant, therefore it is not included in this figure. It should be noted that an optimum θ associated with latent to sensible effectiveness ratio (which satisfies equation (8)) is applied to control the ERVs presented in Fig. 13.

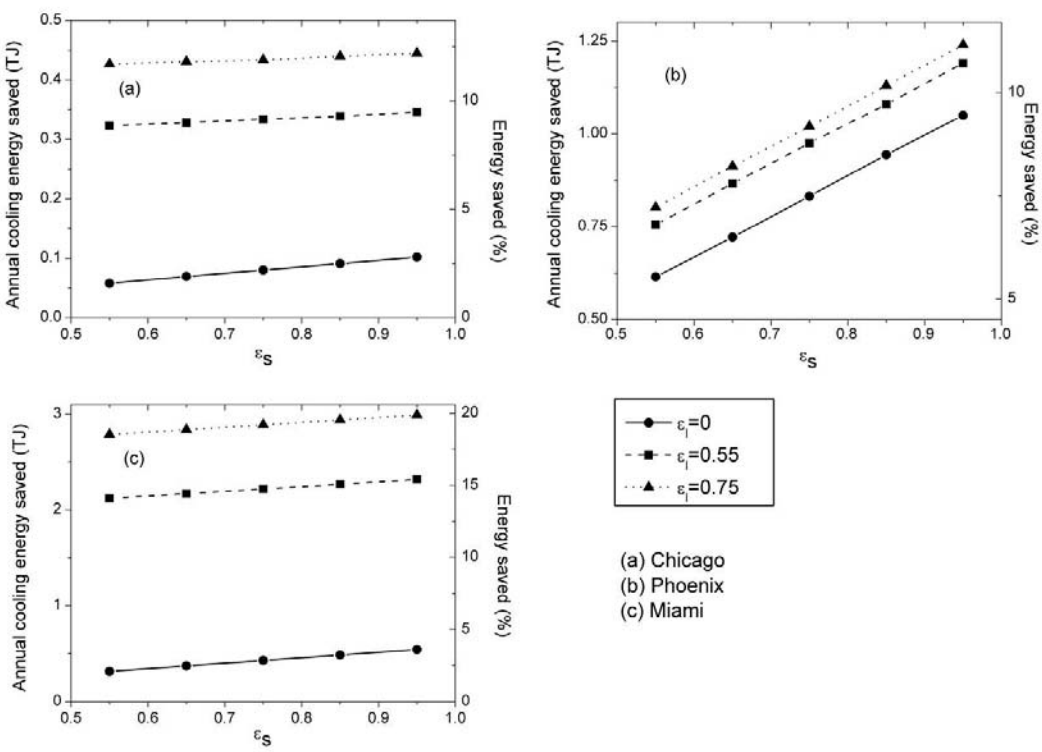

Fig. 13. Annual cooling energy saving for ERVs operating under optimum control strategy, (a) Chicago, (b) Phoenix and (c) Miami

For humid climates (Miami and Chicago), a significant difference between a heat recovery system (ε_l=0) and heat and moisture recovery systems is observed which presents the importance of moisture transfer in humid climates. For instance, a sensible-only exchanger can save up to 5% in Miami and Chicago, where the savings can be increased by 10% in Chicago and 15% in Miami when a heat and moisture recovery system is employed. For Phoenix, as representative of hot and dry climate, the change of sensible effectiveness is found to have more impact on energy savings than the latent effectiveness. Compared to sensible-only ERV, the more humid the outdoor conditions, the more superior is the heat and moisture recovery system in reducing the annual cooling energy consumption.

It should be noted that more than 55% of the buildings listed in U.S office building categorization [23] have lower overall U-values (better insulation) and lower receptacle and lighting loads. For such buildings with lower building cooling loads, the ventilation load accounts for a larger portion of total cooling energy consumption and application of ERV results in higher percentages of cooling energy saved (compared to the savings presented in Fig.13).

6. Conclusions

The impact of energy recovery ventilators (ERVs) on annual cooling and heating energy consumption was studied by conducting TRNSYS simulations. As a representative of the US office building stock, a 10-storey office building was simulated in four US cities representing four different climatic conditions (i.e., Helena with a cold and dry climate, Chicago with a cool and humid climate, Miami with a hot and humid climate and Phoenix with hot and dry climate). Results showed that depending on the climate and system effectiveness, the operation of ERV with capability of moisture recovery reduces the annual heating energy consumption by 40% during heating season. This is about 5% higher than the energy saved with heat recovery systems. The simulation results for the cooling season indicated that uncontrolled operation of ERV may increase the cooling energy consumption by 5%. An optimum control strategy which considered energy savings with both heat and moisture recovery was proposed and compared with temperature-based control. This optimum control was dependent on operating condition factor, H^*, ranging from $-\infty$ to $+\infty$ and presenting latent to sensible energy potentail of inlet airstreams. Depending on the latent to sensible effectiveness ratio, the operation of ERV in cooling season should be limited to specific outdoor conditions within a certain range of operating condition factors described in the paper. For example, the optimum operating condition for an ERV with equal sensible and latent effectiveness values is when the outdoor air has higher enthalpy or higher temperatures than the indoor. The simulation results, in a good agreement with theory, indicated that an ERV may operate for a wider range of cool-humid outdoor condition when it has higher latent effectiveness. When the ERV operated under the proposed optimum control, up to 20% annual cooling energy was saved depending on location and ERV effectiveness. The savings in humid climates (Chicago and Miami) were found more significant than elsewhere since the moisture transfer in ERV could reduce the dehumidification load dramatically.

Acknowledgements

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Nomenclature

Symbols

$Q_{sens,rec}$	Sensible heat recovery (J)
$Q_{lat,rec}$	Latent energy recovery (J)
RH	Relative Humidity (%)
T	Temperature (°C)
C_p	Specific heat capacity of air (J/kg.K)
h_{fg}	Enthalpy of phase change (J/kg)
h	Enthalpy of air (J/kg dry air)
$\dot{m}_a\dot{m}_a$	Mass flow rate of air (kg/s)
w	Humidity ratio (kg water/kg dry air)
ε	Effectiveness (%)

Subscripts

s	Sensible
l	Latent
sup,in	the supply air at the inlet of the energy exchanger, i.e., outdoor air
sup,out	the supply air at the outlet of the energy exchanger
exh,in	the exhaust air at the inlet of the energy exchanger, i.e., indoor air

Acronyms

ERV	Energy Recovery Ventilator
IAQ	Indoor Air Quality
HVAC	Heating, Ventilation and Air Conditioning
ASHRAE	American Society of Heating, Refrigerating and Air conditioning Engineers
PNL	Pacific Northwest national Lab
CBECS	Commercial Building Energy Consumption Survey

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