Model-based optimal control coupled with experimental and numerical analysis for performance of energy recovery ventilator in HVAC system

Huynh, Nam Khoa

2018

Huynh, N. K. (2018). Model-based optimal control coupled with experimental and numerical analysis for performance of energy recovery ventilator in HVAC system. Doctoral thesis, Nanyang Technological University, Singapore.

<http://hdl.handle.net/10356/73095>

<https://doi.org/10.32657/10356/73095>

**NANYANG
TECHNOLOGICAL
UNIVERSITY**

SINGAPORE

**MODEL-BASED OPTIMAL CONTROL COUPLED
WITH EXPERIMENTAL AND NUMERICAL
ANALYSIS FOR PERFORMANCE OF
ENERGY RECOVERY VENTILATOR
IN HVAC SYSTEM**

HUYNH NAM KHOA

**INTERDISCIPLINARY GRADUATE SCHOOL
ENERGY RESEARCH INSTITUTE @ NTU (ERI@N)**

2017

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ENERGY RECOVERY VENTILATOR
IN HVAC SYSTEM**

HUYNH NAM KHOA

Interdisciplinary Graduate School
Energy Research Institute @ NTU (ERI@N)

A thesis submitted to the Nanyang Technological
University in partial fulfilment of the requirement for
the degree of
Doctor of Philosophy

2017

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research and has not been submitted for a higher degree to any other University or Institution.

14 August 2017
Date

.....
Huynh Nam Khoa

Abstract

Recently, the demand for energy conservation management attracts much attention, due to the depletion of energy resources and the environmental impacts caused by the increase in energy consumption, especially in tropical countries, such as Singapore. Generally, building sector, which is one of the most important economic sectors, accounts for approximately 40% of total national energy demand in Singapore. As well known, the heating, ventilation, and air-conditioning (HVAC) systems provide thermal comfort for occupants in buildings. However, they consume up to 50% of the total energy usage in the buildings. Therefore, in order to improve the energy efficiency of the HVAC systems, numerous technologies are developed, including the novel configuration, mechanical design, and advanced control algorithm. In particular, the heat or energy recovery is one of the key energy-efficient technologies, which shows potential to overcome the increase in energy consumption in buildings without reduction of the indoor air quality. However, the literature search reveals that only sensible heat is investigated for the conventional heat recovery system, while the latent heat recovery is not studied well. Furthermore, few studies were carried out for the recovery system in the tropics, where little effort was made on the control aspect, which is potentially a strategy for considerable improvement in the efficiency of energy recovery devices.

In order to address the issues mentioned above, the present project is conducted to improve overall energy efficiency in HVAC recovery system by computational fluid dynamic (CFD) analysis, experimental validation, and optimal control scheme on important parameters of the energy recovery ventilation (ERV). Throughout the present project, several novelties are contributed to the relevant community, as detailed below.

As the first achievement made in this thesis, a novel model for analysis of ERV as a component of HVAC system is developed both mathematically and

experimentally, in order to investigate the performance of ERV sensible and latent energy subject to tropical climate conditions. The three-dimensional ERV model with consideration of a semi-permeable membrane is comprehensively investigated for analysis of critical parameters, including the velocity, temperature, and humidity of supply and exhaust airflows, for the improvement of energy efficiency. Subsequently, in order to examine the predictive capability of the mathematical ERV model, the present numerical results are validated by comparison with the experimental data, in which good agreements are achieved. Through the numerical and experimental results, it is demonstrated that the developed membrane-based recovery system is capable of substantial energy savings in buildings, where the sensible and latent effectiveness could be achieved up to 80% and 70%, respectively.

As the second achievement, a control model is developed to optimize the operation of HVAC recovery components via a dynamic grey-box ERV model through MATLAB/Simulink environment, since very few examinations of control-based recovery systems are found in the open literature. In order to enhance the numerical model, the experimental investigation of the prototype system, consisting of the membrane-based ERV and two (fresh/exhaust) airflows as well as a sensor system, is conducted for control model identification process. The temperature and carbon dioxide (CO₂) concentration level are incorporated into the control model for development of an optimized model predictive control (MPC) strategy for the total energy consumption of building, which maintains the indoor air quality and thermal comfort for occupants. The results show that the zone temperature is regulated better in MPC controller, and the energy consumption of HVAC with MPC controller is considerably less than that of HVAC with PI controller.

Acknowledgements

I would like to express my great gratitude towards my main supervisor, Associate Professor Li Hua from School of Mechanical and Aerospace Engineering (MAE), and my co-supervisor Professor Soh Yeng Chai from School of Electrical and Electronic Engineering (EEE), Nanyang Technological University. They not only helped me to touch deeper knowledge in my research, but also provided a comfortable working environment with their kindness and patience. Without their supports, I would never have been able to finish this thesis.

I am deeply grateful to Professor Xie Lihua for his valuable advice as my mentor. I would like to express my special gratitude to Assoc. Prof. Cai Wenjian for offering me the equipment to carry out the experiment work. I highly appreciate Dr Wang Xinli from Process Instrumentation Lab EEE for his support to set up the first experiment; and Jiang Chaoyang, and Mustafa Khalid Masood for their help to solve the difficulties in the beginning period of my project.

I would like to thank Mr Lew Sui Leung Thomas, Assistant Manager of Material lab 2, who always makes the convenience for me to carry out my experiments in the Lab; and 10 Final year project and master students, who worked with me on this project. I also want to say thanks to my three housemates in my PhD journey of four years; they are my big brothers who share with me the concern, cheers, and sadness when the journey goes up and down. I learnt from them the human kindness and the strong passion for science and research, which became my foundation values.

I would like to thank my lab mates: Goh Kek Boon and Zhou Xiaoli; my seniors: Tran Ngoc Phu and Tong Shanshan; my good-friend: Ruby Hoa, who gave me valuable advice on doing research and useful comments on my thesis draft.

A very special gratitude goes to Nanyang Technological University, Interdisciplinary Graduate School (IGS), and Energy Research Institute @ NTU

(ERI@N), and ABB Pte. Ltd. for offering me the special opportunity to accomplish my PhD. The work was support by the funding of Nanyang Research Scholarship, Singapore's National Research Foundation under Grant NRF2011NRF-CRP001-090, partially supported by ABB Pte. Ltd and ERI@N.

Last but not least, I would like to send my deepest gratitude to my parents, my older brother, and my family, who always encourage me even in my worst time. Their love is the biggest gift of my life.

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Abbreviations

AHU	Air Handling Unit
BES	Building Energy Simulation
CFD	Computational Fluid Dynamics
COP	Coefficients of Performance
DOE	Department of Energy
ERV	Energy Recovery Ventilator
FEM	Finite Element Method
HRV	Heat Recovery Ventilator
IWEC	International Weather for Energy Calculations
LabVIEW	Laboratory Virtual Instrument Engineering Workbench
MPC	Model Predictive Control
NTU	Number of Transfer Units
HVAC	Heating, Ventilation, and Air Conditioning System
RC	Resistance Capacitance
TRNSYS	TRansient SYstems Simulation

Nomenclature

A_{tot}	total heat transfer surface area of energy exchanger (m^2)
C	thermal capacitance (J/K)
C_{CO2}	concentration of carbon dioxide (ppm)
c_p	specific heat (kJ/(kg.K))
D	diffusivity (m^2/s)
D_H	hydraulic diameter (m)
E	energy (W)
E_S	sensible energy saving (W)
E_L	latent energy saving (W)
E_{tot}	total energy saving (W)
h	convective heat transfer coefficient ($kWm^{-2}K^{-1}$); enthalpy (kJ/kg)
h_{fg}	enthalpy of vaporization (kJ/kg)
H	specific enthalpy (kJ/kg)
H^*	ratio of latent to sensible energy differences
k	thermal conductivity (W/(m.K))
m	mass flowrate of air (kg/s)
N_{people}	number of people
NTU	number of transfer units
Nu	Nusselt number
p	pressure (Pa)
P_{WS}	water vapor saturation pressure (Pa)
P_A	atmospheric pressure (Pa)
q	heat flux (kW/m^2)
Q	total heat transfer (kW)

R	thermal resistance (K/W)
Re	Reynolds number
RH	relative humidity (%)
R_{plate}	plate mass transfer resistance (m^2s/kg)
t	times (s)
T	temperature ($^{\circ}C$)
U_s	overall sensible heat transfer coefficient
v	velocity (m/s)
V	volumetric flow rate (m^3/h)
W	humidity ratio (kg/kg)

Greek Letters

ε	effectiveness (%)
ε_s	sensible heat transfer effectiveness (%)
ε_L	latent heat transfer effectiveness (%)
ε_H	total heat transfer effectiveness (%)
δ	thickness (mm)
ρ	density of air (kg/m^3)
μ	dynamic viscosity ($kgm^{-1}s^{-1}$)

Subscripts

a	air
ai	air inlet
chw	chilled water
e	exhaust
EA	exhaust air
f	fresh

i	in
int	internal compornent
m	membrane
o	out
OA	outdoor air
SA	supply air
RA	return air
tot	total
vent	ventilation
win	windows

Chapter 1

Introduction

This chapter presents an overview of the increase in energy consumption and greenhouse gas emission in the building sector, especially in tropical climate. Due to the high temperature and humidity of tropical weather condition, the electricity consumption in HVAC system is a major energy burden, accounting for 50% of the energy usage in buildings in Singapore. Therefore, a number of studies have been carried out to enhance building energy efficiency, in which energy recovery technology shows a promising potential to decrease energy demand of HVAC system. However, it has not been thoroughly investigated for tropical climate. Hence, this study aims to develop a modeling platform for performance analysis and operation optimization of an energy recovery device coupled with HVAC system, to save power consumption and improve the indoor air quality along with thermal comfort for occupants. In the last section, the introductory chapter states the scopes of the project, followed by the organization and a quick summary of the whole thesis.

1.1. Background

In this section, an overall analysis on the energy consumption of building sector is presented. As part of an attempt to save energy, the study also emphasizes the impacts of greenhouse gas emissions and how to devise an energy efficiency strategy specifically through recovery technologies in HVAC system.

1.1.1. Building energy consumption and greenhouse gas emissions

Recently, the energy consumption of the world steadily increases with a high rate as a consequence of economic development and population explosion [1, 2]. This situation raises concern about the depletion of energy resources and environmental issues. The energy usage is classified into three main economic sectors: building, transportation, and industry. The building sector constitutes approximately 41% of national energy demand in United States [3], 39% in the United Kingdom [4], and 20-40% in developed countries [1]. Statistics from U.S Department of Energy (D.O.E.) shows that the heating, ventilation and air conditioning (HVAC) system, which plays a major role in maintaining the thermal comfort of occupants, representing the largest energy consumption in both residential and commercial buildings, followed by lighting and water heating [5], as shown in Figure 1.1.

From the viewpoint of environmental impacts, the building segment composes a significant proportion of energy related emissions. It is estimated that buildings contributed one-third of global greenhouse gas emissions [6]. The current high level of greenhouse gas emissions is the major reason for global warming, climate change, and ozone layer depletion. Thus, the central problem nowadays is to establish a highly energy-efficient system in the existing and new buildings, which can reduce the energy consumption and CO₂ emissions significantly.

Figure 1.1 The end-use electricity consumption in the building sector in the US [5]

1.1.2. Building energy efficiency strategies

To address the problems of high energy requirement and the associated environmental concern in building, numerous technologies were developed to improve the energy efficiency of the HVAC system, including configuration, mechanical design, and control algorithm for the evaporative cooling, ground-coupled HVAC, thermal storage, and heat recovery systems [7, 8]. In these approaches, heat and energy recovery system are sustainable ways to decrease energy demand for heating, cooling, air conditioning system and to improve the indoor air quality considerably [9].

Heat and energy recovery system is defined as a system that recovers the sensible and latent energy from the exhaust air, to pre-process the outdoor air through heat transfer and moisture diffusion phenomena [10]. The sensible heat is the energy which is used for cooling or heating the air from the outdoor temperature to the desired indoor temperature. The latent heat is the energy which is used to dehumidify or humidify the outdoor air to the expected relative humidity.

With the alarming rise in energy consumption in the building sector, various control techniques and optimization strategies were implemented in building automation and control system [11]. Due to the simple and low-cost operation, the classical control techniques such as the on/off controller and proportional integral derivative (PID) controller are still very prevalent in existing buildings [12]. However, in the long term, these controllers are costly because they operate at a non-optimal efficiency. Moreover, with the modern cooling and heating technologies, the simple control shows the incapability of handling the operation, resulting in inconsistent performance in HVAC system. Therefore, there is a high potential for employing an advanced control method to optimize operation on system level and save an enormous amount of energy. In recent years, numerous advanced control algorithms have been designed and implemented to overcome the inherent issues of classical HVAC control system, from the hard control techniques, such as optimal, adaptive, robust, and predictive control schemes, to soft control techniques, such as neural network, fuzzy logic, and genetic algorithm [13]. Due to various advantages, the model predictive control (MPC) approach is particularly emphasized. MPC control utilizes a system model with predictable action and the ability to handle constraints with disturbances. Moreover, the cost function of MPC control is designed to achieve multiple objectives [14].

1.1.3. Recovery technologies for saving energy and improving indoor air quality in HVAC system

One of the main approaches to reduce thermal loss in building is increasing the building airtightness with envelope insulation [15]. However, this method possibly leads to poor indoor air quality, which negatively affects the occupants' health. An adequate fresh air ventilation is required and which account for nearly 68% of total moisture load in air-conditioning system [16]. For these reasons, heat recovery system (HRS) is applied as an effective approach to reuse thermal loss and preserve indoor air quality in safe level of pollution concentration. In HRS, the building exhaust air is used as a heat source or heat sink to pre-process

fresh air by transferring thermal energy as heating or cooling, depending on the temperature difference of outdoor and indoor air.

Heat recovery systems have many types of thermal transfer from exhaust air to incoming air. Generally, a typical HRS consists of three parts: core of heat exchanger, inlet or outlet air duct, and air blower. The effectiveness of recent HRS possibly gained 60-90% [17], which shows a promising potential to decrease energy demand of HVAC system in buildings.

1.2. Motivation

As an urbanized city with tropical weather, Singapore's buildings are responsible for nearly 40% of the national energy consumption [18]. Singapore has a tropical climate without distinct seasons, with the characteristics of high temperature, high humidity, and abundant rainfall throughout the year. As recorded by National Environment Agency (NEA), Singapore has an average annual temperature of 29°C and an average annual relative humidity of 84.3% [19]. To maintain the thermal comfort, the HVAC system is widely used in almost all buildings in Singapore. Because of high temperature and high humidity climate, electricity consumption in HVAC system is a major energy burden, accounting for 50% of total energy usage of the building sector in Singapore [18].

Apart from that, indoor air pollution is another issue, and usually leads to major health problems, including eyes, nose and throat irritation, headache, sneezing, and coughing. If the occupied room is inadequately ventilated, the level of indoor air contaminants, such as carbon dioxide (CO₂) and carbon monoxide (CO), becomes considerably higher than that of outdoor air, resulting in the discomfort for occupants, who often spend most of time inside the building. Hence, the index of indoor air quality (IAQ) is necessarily employed to characterize the indoor air quality in both residence and office environment. In order to improve the IAQ, the ventilation rate of buildings is usually increased to dilute the contaminated indoor air. However, the increase of the ventilation rate significantly consumes much more energy to condition a new fresh air into the

buildings. For example, in hot and humid climate, energy for cooling and dehumidifying fresh air accounts for approximately 20~40% of total HVAC energy consumption, in which dehumidification process contributes mostly to the energy utilization [16]. In the conventional air-conditioning system, the fresh air is cooled down to the dew point temperature for condensation and dehumidification, when it travels through the cooling coil. The air is then reheated to the desired condition, such as a temperature at 21~25⁰C with 40~60% relative humidity [20]. Therefore, the building consumes a huge part of electricity to maintain the indoor air quality, which causes a high load on the energy system.

In recent years, air-to-air enthalpy recovery technology has played a key role to overcome this problem and gained much attention from researchers. However, in the conventional recovery technology for the building application, the sensible energy was recovered through heat transfer, but the latent energy was neglected. It was called heat recovery system (HRV) and was used widely in cold climate countries where the difference between indoor and outdoor temperature is significant. HRV core is often made from the metallic material. Lately, with the development of membrane technology, Zhang *et al.* [21] proposed the membrane-based energy recovery ventilator (ERV) which has the capability to exchange the sensible and latent heat simultaneously with high effectiveness. Consequently, the energy load of HVAC systems is reduced substantially.

Up to now, the literature review has showed that some fundamental works were conducted on the applicability of heat or energy recovery system. They pointed out that recovery system was an efficient and sustainable way to solve the increase in energy consumption and indoor air quality problem. However, only a few studies on the membrane-based recovery system were conducted in tropical countries such as Singapore, Malaysia, and Thailand. In addition, few studies paid attention to control aspect which is potential for considerable improvement in efficiency of energy recovery system.

1.3. Objective and scope

The objective of this study is to develop a modeling platform for investigation of the energy saving and the thermal comfort maximizing, in which the performance of ERV is simulated first in hot and humid tropical climate. A dynamic model is then constructed for energy transfer in recovery system. Finally, an optimal control strategy for ERV is developed and integrated into the whole building control system. Therefore, the working scope of the present Ph.D. dissertation is detailed below.

- *Simulation of ERV in hot and humid tropical climate*

The performance of a membrane-based energy recovery ventilator is simulated via both numerical and semi-analytical models. The heat and moisture transfer effectiveness is then examined for the improvement of energy efficiency. The energy recovery efficiency of the membrane-based system is also thoroughly investigated by the mathematical models, especially subject to the typical weather conditions in tropical countries.

- *Construction of the dynamic model for energy transfer in recovery system*

The present dynamic model is mathematically constructed by systemically experimental studies through a prototype system, which consists of membrane-based ERV, two (fresh/exhaust) airflows and a sensory system, to examine the influence of several crucial parameters on ERV performance, such as temperature, humidity and velocity. In addition, the mathematical model is also validated via a comparison between present numerical results and present experimental data.

- *Development of an optimal ERV control strategy integrated into the whole building control system*

A dynamic grey-box ERV control model is identified, based on the above ERV simulation and experiment-based dynamic model. The temperature, humidity, and CO₂ concentration level are incorporated into the control model for the development of an optimized predictive control strategy, which is utilized to minimize the total energy consumption of buildings,

and to maintain the thermal comfort and indoor air quality for the occupants simultaneously.

1.4. Thesis structure

The thesis is composed of seven chapters, and each chapter consists of several sections for a systematic organization, which is briefed below.

Chapter 1 introduces an overview of the increase in world energy consumption and environmental concerns. Recent building energy efficiency strategies are presented, especially the energy recovery technology. The motivation, objective and scopes of the whole project are also addressed in this chapter.

A literature review on the HVAC and ERV systems is systematically conducted in *Chapter 2*. The classification of air-to-air heat exchangers, which includes the most common types such as heat pipe, rotary wheel, and fixed-plate heat exchanger, is presented. Experimental investigations, mathematical models, as well as control techniques for ERV and HVAC system are thoroughly reviewed and discussed.

In *chapter 3*, a numerical model of ERV is developed based on the governing equations of energy recovery process, and solved by CFD method. The simulation model of ERV is validated based on previous studies and examined under various configurations and working conditions. The effects of critical parameters such as airflow velocity, outdoor air temperature, and humidity ratio to the performance of ERV are systematically investigated.

Chapter 4 demonstrates the experimental setup of the energy recovery system and the performance analysis of ERV. A sensor system with data acquisition component is also equipped to measure significant parameters and manage data for further analysis, which is utilized to validate the theoretical model developed in Chapter 3.

Chapter 5 proposes appropriate control models of ERV and HVAC components such as air handling unit and cooling coil. Afterward, those models

are incorporated into a thermal dynamic building model, which is developed for the implementation of a model-based control algorithm in the following chapter.

In *chapter 6*, the model predictive control (MPC) strategy is presented as an advanced controller which has a capability of handling operation constraints and disturbances such as weather and occupancy, while achieving multiple objectives. Two optimized MPC strategies are designed and implemented to the ERV integrated into HVAC system, to maintain the indoor CO₂ concentration, zone temperature in a thermal comfort range, as well as minimize energy consumption.

Chapter 7 discusses the significant results, contributions and draws conclusions of the whole thesis. Recommendations for future research work with recommendations are proposed in this last chapter.

Chapter 2

Literature Review

In this chapter, previous studies in heat and energy exchangers are intensively analyzed and categorized to obtain a full review of literature. At first, air-to-air heat exchangers are classified into three most common types: heat pipe, rotary wheel, and fixed-plate heat exchanger based on the construction and characteristics. Then the fundamental background on air-to-air heat exchangers, such as thermal and moisture transfer mechanisms, sensible and latent effectiveness, is presented and discussed. Meanwhile, thorough literature reviews of numerical and experimental studies, as well as recent control strategies for ERV and HVAC are systematically conducted. Finally, the research gaps, which construct the background problem of this study, are pointed out. This chapter also presents the significant contributions of the Thesis.

2.1. Classification of air-to-air heat exchangers

Depending on the applications, air-to-air heat exchanger is generally categorized into two groups: heat recovery ventilator (HRV) and energy recovery ventilator (ERV) [22]. HRV, which transfers only sensible energy, is commonly used in cold climate where the difference of temperature between outdoor and indoor air is large. Furthermore, in humid regions, latent energy could be recovered by transferring the moisture between airstreams. Therefore, ERV, which transfers both sensible and latent energy simultaneously, is widely used in commercial buildings in tropics.

HRV and ERV are the important part of energy recovery system. In particular, air-to-air energy recovery devices are classified into various types, configurations, and airflow arrangements. Over the last decade, the most common types of heat exchangers were utilized to recover energy between supply air and exhaust air such as heat pipe, rotary air-to-air energy exchangers, and fixed-plate heat exchangers [23]. Each of heat exchanger types has specific advantages, disadvantages, performance parameters, and applications, which are discussed in follow subsections.

2.1.1. Heat pipe heat exchanger

Heat pipe heat exchanger utilizes the latent heat of vaporization to transfer heat over a long distance [24]. It involves self-seal tubes which contain suitable working fluid. The heat pipe working fluid operates in a closed-loop evaporation or condensation cycle as illustrated in Figure 2.1.

Figure 2.1 Heat pipe heat exchanger [24]

The hot airstreams go through the evaporation section of the heat pipe and the heat vaporizes the working fluid. Due to a vapor pressure gradient, the vapor moves to condenser section, condenses, and releases the latent energy of evaporation. The condensed fluid returns to evaporation part by capillary action of the wick or gravitational force, thus completing the cycle. The heat pipe recovery technology has some advantages such as no moving parts, no cross contamination, and no external electric power requirement. However, the typical efficiency of heat pipe heat exchangers is quite low. El-Baky and Mohamed [25] conducted an experiment to study the thermal performance of heat pipe system. Fresh airstream and exhaust airstream which were controlled in the temperature from 32-40⁰C were connected to heat pipe recovery. They found that the effectiveness for both evaporator and condenser section was about 48%. Heat pipe recovery system was suitable for utilization in naturally ventilated buildings [26].

2.1.2. Rotary wheel

A rotary air-to-air energy exchanger consists of a cylinder which is coated with a permeable medium and operated by a motor [27]. Adjacent supply and exhaust airstream pass through half of the wheel as shown in Figure 2.2 [23]. The advantage of the rotary wheel is the capability to recover sensible and latent energy simultaneously with high efficiency. However, a main drawback of a recovery wheel is cross-contamination between two airstreams due to leakage

and carryover. This challenge limits the application of rotary wheel in health care buildings in which uncontaminated air is of utmost importance.

Figure 2.2 Rotary wheel recovery [28]

2.1.3. Fixed-plate heat exchanger

Apart from the heat pipe heat exchanger and rotary wheel, mentioned in subsections 2.1.1 and 2.1.2, the fixed-plate heat exchanger is the most common construction of heat exchanger devices [29]. Typically, it is associated with various advantages, including no moving part, little cross-contamination, simplicity to integrate into existing conditioning systems, and the ability to transfer both sensible heat and latent heat with high efficiency. Notably, the core of the fixed-plate heat exchanger consists of thin plates, which are stacked together to separate the internal exhaust and fresh airstreams, as shown in Figure 2.3 [23]. These thin plates' surface can have a form of smooth or corrugation plates. They perform as exchanger surfaces, in which both thermal and latent energy are transferred between two airstreams. These two different airstreams pass through each other under the airflow configurations including cross flow, counter current flow, and concurrent flow [29].

It is very encouraging that the current sensible heat exchanger can recover 50-80% of waste energy [17]. The performance of fixed-plated heat exchanger is significantly affected by plate materials [30, 31], airflow pattern [32, 33], and

physical configuration of exchanger (plate thickness, pitch, and orientation of corrugation) [34]

Currently, it is noticed that a majority of the investigations focused on materials and structural designs of the heat exchanger core, however, few studies paid attention to the optimal control problem. In earlier studies, the thin plate was made from metal such as aluminum, steel alloy, and copper. By using these materials, only the sensible energy is transferred from the warm airstream through the plates into the cooler airstream. Manz and Huber [35] carried out the experiments and simulations to examine the heat recovery effects of the duct/heat exchanger ventilation unit by adding the aluminum fins on the exchanger surface. They figured out that the temperature efficiency of approximately 70% could be obtained.

Figure 2.3 Typical fixed plate heat exchanger [23]

In recent years, membrane material attracts more attention in the development of the heat exchanger core [36]. Heat exchanger core which is made from permeable membrane such as, cellulose, polymer, and other synthetic materials, has the ability to transfer both thermal energy and water vapor from one airstream to another, and thus the enthalpy energy could be recovered. With reference to this, Zhang and Jiang [21] presented a detail heat and mass transfer model for ERV, with a porous hydrophilic membrane core. They found that the

heat exchanger with membrane core and the counter flow air arrangement outperformed the metal and paper core heat exchanger.

From the literature search, it reveals that most studies were on the fundamental levels, and focused on the material and structure aspects of energy recovery devices [37-40]. These analyses pointed out that the membrane-based heat recovery exchanger could be utilized as an effective way of recovering the energy in the air conditioning system. However, few studies paid attention to the optimal control problem, which is a promising approach to give a significant improvement to heat or energy recovery ventilator efficiency.

2.2. Experimental investigation of heat exchangers

Air-to-air heat exchanger can be categorized into two main groups: sensible heat recovery ventilator (SHRV) and energy recovery ventilator (ERV). This part will review experimental studies on these two types of recovery ventilator.

2.2.1. Sensible heat recovery ventilator

The sensible heat recovery ventilator (SHRV) is the device which reclaims the thermal energy from exhaust stale indoor air to pre-condition the incoming fresh air. The core of SHRV is made from metal, thus it cannot transfer the moisture between two airstreams. SHRV is broadly used in cold climate countries where the sensible energy is dominant.

Kotcioglu *et al.* [41] conducted an experiment to calculate the effectiveness of a cross-flow sensible heat recovery ventilator. The experimental procedure consisted of measuring the supply and exhaust air temperatures in order to examine the effects of the winglets which were inserted into the cross-flow plate type SHRV. The number of transfer units (NTU) method was used to analyze the performance of plate heat exchanger. They found that theoretical effectiveness was in a good agreement with experimental results, which was very high and desirable from economical saving.

Kim *et al.* [42] carried out a study to examine the influence of SHR and ERV in two high-rise residential buildings in Korea in aspect of energy saving. They pointed out that the ERV achieved slightly more energy saving than the SHR under various operation schedules. More energy was saved when the hours of operation of both recovery ventilators increased.

Nguyen *et al.* [43] conducted an experiment to investigate the overall performance of a sensible heat recovery combined with a ventilation system in winter season. Four types of heat recovery system were built and compared with each other as illustrated in the Figure 2.4. In Type A, the indoor air was directly exhausted to the environment. At the same time, the air conditioning system was used to heat the cold supply air and introduce to the indoor space. No heat recovery was implemented in this case. In Type B, a separate sensible heat recovery was utilized to recover the thermal energy from the exhaust air to the incoming indoor air. In type C, the heat pump was integrated with the ventilation system into a single unit which mixed the indoor exhaust air with the fresh outdoor air. Type D utilized the cross flow sensible heat exchanger to transfer the thermal energy from exhaust air to the ambient fresh air. Then, the pre-heated supply air was mixed with the return air. Their experiment results showed that, in case the sensible energy recovery was utilized, the energy consumption of the compressor was lower due to the pre-heating effects of the air. For the experiment with a ventilation ratio of 23%, the net coefficient of performance (net COP) recorded the highest value of 3.28 in Type D, in which the sensible heat recovery was integrated with heat pump system. Compared with the base case (type A), the integration of heat recovery system in type D improved the net COP of 13.9%, indicating that type D is the most efficient system.

Figure 2.4 Configuration of four types of heat recovery system [43]

2.2.2. Enthalpy recovery ventilator (ERV)

Zhang and Niu [44] conducted a simulation to compare the energy saving impacts of ERV and SHR in the humid weather of Hong Kong. The simulation result showed that in humid weather, the ERV achieved greater impact on energy saving. With the obvious advantages of membrane based fixed-plate energy exchanger in a humid climate, numerous experiments were established to get the insights of enthalpy exchanger performance. Major studies focused on the material innovation and the shape structure of the membrane. Nasif *et al.* [45] carried out experimental studies to investigate the performance of an energy heat exchanger in terms of both sensible and latent effectiveness. 60gsm Kraft paper was utilized as the surface to transfer heat and moisture concurrently. The enthalpy recovery had the Z shape configuration as showed in Figure 2.5 and 49 air passages in each exhaust and supply airstream. They found that the air conditioning system integrated with this air-to-air heat exchanger saved the energy amount of 4% and 8% compared to traditional air conditioning system in

moderate climate and humid climate, respectively. Nasif *et al.* [46] showed that, the sensible effectiveness was stable, while the latent effectiveness presented obvious variation. The variation heavily depended on the permeable surface resistance.

Figure 2.5 Plastic frame of flow channels in heat exchanger [45]

Mardiana and Riffat [47] developed an energy recovery system which utilized cellulose paper as the material for transferable core. The experimental investigation of the performance of enthalpy exchanger indicated that the

efficiency of 66% and 59% was possible for sensible and latent effectiveness, respectively. The recovered energy was obtained up to 167W at 3.0m/s air velocity.

In order to solve the low latent effectiveness problem, Zhang [39] constructed two different plate-fin cores for enthalpy exchanger and studied heat and moisture transfer in the plate-fin core at the same time. The conventional core was made from paper fin and paper plate, and the new core was made from paper fin and membrane plate which utilized the composite supported liquid membrane as material. By experimental analyses, Zhang showed that the latent effectiveness of the new core was 60% higher than the conventional core due to high moisture diffusivity in the new core.

Fernandez *et al.* [48] conducted an experimental analysis of enthalpy exchanger in cold climate. The air-to-air enthalpy recovery had the sensible polymer core as showed in Figure 2.6 and operated in balanced ventilation condition. The study indicated that the relative humidity of fresh air decreased from 95% to 34% and thermal efficiency was approximately 80%.

Figure 2.6 Enthalpy recovery unit [48]

2.3. Mathematical modeling of heat exchangers

The fundamental physics of energy recovery ventilator are the difference of moisture holding capacity of air and heat and mass transfer phenomena. The

holding capacity increases dramatically with the rise of temperature. Due to the difference of temperature between two airstreams, when the hot inlet air went through the ERV, it transferred heat energy to the cold exhaust air. The exhaust air was heated to the temperature that was approximately the same as the outdoor air temperature. Hence, moisture retention of the exhaust air was increased. The moisture from the humid inlet air easily diffused across the membrane into the dryer exhaust air. Two of these processes occurred simultaneously. As a result, the hot and humid incoming air was colder and dryer before it came to the cooling unit of HVAC system. These behaviors were simplified in Figure 2.7.

Figure 2.7 Simplified principle of a counter flow ERV in hot and humid region

2.3.1. Heat and mass transfer

A heat transfer process is a transmission of the thermal energy from one airstream to another because of the temperature difference [49]. The heat transfer depends on the magnitude of the temperature difference and the heat conductivity of the membrane material. The description of theoretical model of energy exchangers was simplified as shown in Figure 2.8

Figure 2.8 Description of a theoretical model of energy exchanger

The heat transfer occurs through these models:

- Convective heat transfer occurs on the hot side of the surface, due to air movement.
- Heat conduction transfers through the membrane due to the physical contact. The thermal conductivity of the membrane is an important parameter to evaluate the heat conduction.
- Convective heat transfer occurs on the cool side of the surface, due to air movement.

The heat balance equation for conduction and convection [50] can be written as

$$\delta_{is} \rho c_p \frac{\partial T}{\partial t} + \nabla \cdot (-k \nabla T) = Q - \rho c_p u \cdot \nabla T \quad (2.1)$$

where T is temperature, k is thermal conductivity, c_p is the capacity, and Q is heat source. In a steady state problem, the equation (2.1) is simplified as:

$$\nabla \cdot (-k \nabla T) = -\rho c_p u \cdot \nabla T \quad (2.2)$$

On the other hand, mass transfer is the net mass movement from one airstream to another. The physical process of mass transfer is analogous to heat transfer. The performance of mass transfer depends on the difference of moisture concentration and the characteristics of the membrane. The mass transfer process happens through three models: convective mass transfer occurs on the dry side,

convective mass transfer occurs on the humid side, and mass diffuses through the membrane.

The mass balance equation for convection and diffusion via the ERV membrane is shown as follows,

$$\delta_{ts} \frac{\partial c}{\partial t} + \nabla \cdot (-D \nabla c + cu) = R \quad (2.3)$$

where c is the concentration of species (moisture), D denotes diffusion coefficient and R is a reaction rate expression for the species (water moisture). The reaction rate is zero because there is only moisture in the air, therefore no reaction occurs in this simulation. The steady state problem is applicable and the equation (2.3) is simplified as:

$$\nabla \cdot (-D \nabla c + cu) = 0 \quad (2.4)$$

2.3.2. The performance parameters of heat exchanger

The ERV performance is estimated by the capability to transfer the sensible and latent energy of a recovery system. The effectiveness is calculated by the ratio between the actual energy recovered and maximum possible energy that can be recovered. There are 3 types of effectiveness: sensible heat transfer effectiveness (ε_s), latent heat transfer effectiveness (ε_L), and total heat transfer effectiveness (ε_H), which are given below [51],

$$\varepsilon_s = \frac{m_s c_{ps} (T_{si} - T_{so}) + m_e c_{pe} (T_{eo} - T_{ei})}{2(m c_p)_{\min} (T_{si} - T_{ei})} \quad (2.5)$$

$$\varepsilon_L = \frac{m_s h_{fg} (W_{si} - W_{so}) + m_e h_{fg} (W_{eo} - W_{ei})}{2m_{\min} h_{fg} (W_{si} - W_{ei})} \quad (2.6)$$

$$\varepsilon_H = \frac{m_s (H_{si} - H_{so}) + m_e (H_{eo} - H_{ei})}{2m_{\min} (H_{si} - H_{ei})} \quad (2.7)$$

where subscripts s , e , i , o denote supply, exhaust, inlet, and outlet of airstream, ρ is density, c_p is specific heat, v is air velocity in the core, T is temperature,

W is humidity ratio of the air, and H is the specific enthalpy of air and calculated by:

$$H = c_p T + W(2501 + 1.86T) \quad (2.8)$$

Zhang [52] stated that, at constant flow properties and equal supply and exhaust rate, the equations (2.5), (2.6), and (2.7) are simplified as follows,

$$\varepsilon_s = \frac{T_{si} - T_{so}}{T_{si} - T_{ei}} \quad (2.9)$$

$$\varepsilon_L = \frac{W_{si} - W_{so}}{W_{si} - W_{ei}} \quad (2.10)$$

$$\varepsilon_H = \frac{H_{si} - H_{so}}{H_{si} - H_{ei}} \quad (2.11)$$

The total effectiveness (ε_H) was calculated as a combination of ε_L and ε_s as follows [53],

$$\varepsilon_H = \frac{\varepsilon_s - \varepsilon_L H^*}{1 + H^*} \quad (2.12)$$

$$H^* = \frac{2501(W_{si} - W_{ei})}{c_{pa}(T_{si} - T_{ei})} \approx 2501 \frac{\Delta W}{\Delta T} \quad (2.13)$$

where H^* is ratio of latent to sensible energy differences between inlets of two airstreams. Above equations showed that total effectiveness was not a simple average of latent and sensible effectiveness. It was a function of the ratio of humidity and temperature difference and sensible and latent effectiveness.

2.3.3. Number of transfer units (NTU) model

NTU is a popular method to evaluate the performance of a heat exchanger [49, 50]. The number of transfer units is a dimensionless parameter which is the combination of heat transfer area, overall heat transfer coefficient, fluid flow rate and heat capacity as follows,

$$NTU = \frac{UA}{mc_p} \quad (2.14)$$

2.3.3.1. Sensible heat

The mathematic model was developed for the only sensible heat exchanger to estimate the sensible heat transfer efficiency. Nusselt number and Reynold number correlations were utilized to calculate the heat transfer coefficient. The number of transfer units (NTU) is defined as:

$$NTU_s = \frac{U_s A_{tot}}{C_{min}} \quad (2.15)$$

where A_{tot} is the total heat transfer surface area of enthalpy recovery, C is the heat capacity ratio, U_s is overall sensible heat transfer coefficient, which is calculated below,

$$U_s = \left[\frac{1}{h_s} + \frac{\delta_p}{\lambda_p} + \frac{1}{h_e} \right]^{-1} \quad (2.16)$$

where h_s and h_e is the convective heat transfer coefficient of supply and exhaust air, respectively.

The empirical correlations for parallel-plate sensible heat exchanger is presented in [50] with counter, concurrent and cross flow arrangements as follows,

$$\epsilon_{s,counter} = \frac{1 - \exp[-NTU_s(1-C)]}{1 - C \exp[-NTU_s(1-C)]} \quad (2.17)$$

$$\epsilon_{s,co-current} = \frac{1 - \exp[-NTU_s(1+C)]}{1 + C} \quad (2.18)$$

$$\epsilon_{s,cross} = 1 - \exp \left[\frac{\exp(-NTU_s^{0.78} C) - 1}{NTU_s^{-0.22} C} \right] \quad (2.19)$$

where

$$C = \frac{C_{\min}}{C_{\max}} = \frac{(mc_p)_{\min}}{(mc_p)_{\max}} \quad (2.20)$$

2.3.3.2. Latent heat

By analogy of heat transfer, the mass transfer mathematical model was also developed based on the number of transfer units for moisture [49, 54, 55]:

$$NTU_L = \frac{A_{tot} U_L}{m_{\min}} \quad (2.21)$$

The overall mass transfer coefficient is calculated as:

$$U_L = \left[\frac{1}{k_s} + R_m + \frac{1}{k_e} \right]^{-1} \quad (2.22)$$

The moisture diffusive transfer resistance of membrane is R_m , which is co-determined by thickness and diffusivity of membrane, operating conditions and membrane material [36]. Two other terms k_s and k_e are the convective mass transfer coefficient of the supply and exhaust air side.

Similarly, the latent effectiveness, which is analogous to sensible effectiveness, is calculated for different flow arrangements as follows:

$$\varepsilon_{L,counter} = \frac{1 - \exp[-NTU_L (1 - M)]}{1 - M \exp[-NTU_L (1 - M)]} \quad (2.23)$$

$$\varepsilon_{L,co-current} = \frac{1 - \exp[-NTU_L (1 + M)]}{(1 + M)} \quad (2.24)$$

$$\varepsilon_{L,cross} = 1 - \exp \left[\frac{\exp(-NTU_L^{0.78} M) - 1}{NTU_L^{-0.22} M} \right] \quad (2.25)$$

where

$$M = \frac{m_{\min}}{m_{\max}} \quad (2.26)$$

The ε -NTU method for heat and mass exchanger was developed as in the above formulas to analyze the performance of heat recovery system. This method is reliable, inexpensive, and faster than the experimental method. Zhang [51, 53] showed that the average discrepancies are 7.3% and 8.6% between mathematical and experimental results for sensible and latent effectiveness, respectively. However, in some particular operating conditions and designs of the HRV/ERV system, ε -NTU cannot provide deep insights into mechanisms of heat and mass transfer. It is proved that the numerical method became an effective tool to investigate energy recovery performance in such system. CFD simulation was an effective approach for detailed design of the HRV/ERV system under different operating conditions, flow configurations, climate seasons, and membrane properties.

2.3.4. Numerical model

To optimize the operation of heat recovery system, a number of studies were conducted by developing numerical models [56], finding novel materials and new structures of membrane [21, 38, 57]. Zhang and Jiang [21] presented a numerical model of temperature and humidity transfer through membrane-based heat exchanger. By finite difference method, they found that the membrane area was not well utilized with the cross-flow arrangement. A theoretical model was developed by Niu and Zhang [38] to study the thermal and vapor transfer mechanism of a cross flow ERV. Different types of materials were used for investigation. They showed that the sensible effectiveness was slightly influenced by operating conditions, whereas the latent effectiveness was considerably affected by materials and operating conditions.

Min and Su [57] built a numerical model to analyze the performance of a membrane-based ERV. They studied the impacts of the ERV core parameters such as the membrane spacing and thickness on the ERV effectiveness. They found that the membrane parameters and the fan power have significant effects on both the sensible and latent effectiveness.

Recently, Huynh *et al.* [58] carried out a comprehensive study on ERV with semi-permeable membrane. The three-dimensional ERV model is developed in CFD simulation software to solve the heat and mass balance equation by finite volume method. The CFD results showed that, the heat and mass effectiveness could be gained up to 75 and 65%, respectively.

2.4. Control strategies

2.4.1. Control techniques for HVAC system

With the significant increase in energy consumption in buildings, energy saving strategies play a key role in energy policies in many countries. Therefore, the development of efficient control methods in HVAC system is particularly important. Various control methods which included classical, hard and soft controls were proposed for HVAC system [13, 59, 60].

Due to the simplicity and low cost, the classical control techniques such as the on/off controller and proportional integral derivative (PID) controller are still very popular in existing buildings. However, they are inadequate to control moving processes with time delays or nonlinear dynamic system with time varying disturbances. The on/off control showed that the overshoot in the controlled signal was not avoided. Although the PID controllers improved the situation, inappropriate gain values could made the whole system unstable [61]. Nowadays, with the development in computing and communication techniques, it is feasible to implement the advance controllers such as optimal and predictive methods to overcome the common issues of HVAC systems.

In model predictive control (MPC), a system model, which is utilized as a state estimator, is developed and incorporated into the optimization problem. The optimal output is calculated by minimizing the objective function with respect to the constraints of actuators. Pappa *et al.*, [62] evaluated the performance of neural model predictive controller and the result is compared with standard PID controller in real time control of a heat exchanger with counter flow arrangement. The feed-forward neural network is utilized for training and prediction of the heat

exchanger behavior. The study showed that the response of the nonlinear predictive controller was faster and more stable than PID controller. In general, the model predictive control was regarded as the most promising predictive control in buildings because of its ability to handle constraints and disturbances, to use cost functions for multi objectives and to integrate energy conservation strategies in the controller formulation.

2.4.2. Control techniques for ERV

As mentioned above, various parameters affect the energy saving performance of ERV such as airflow rate, membrane characteristics, indoor and outdoor environmental conditions, enthalpy efficiency of the exchanger, and so on. Some investigations were conducted to analyze applicability and energy saving aspects of ERV [44, 63, 64]. These studies proved that the employment of the energy exchanger reduced the heating energy consumption considerably; however, it could increase cooling energy demand due to some specific outdoor conditions. Fauchoux *et al.* [65] carried out a study to determine the impact of an energy wheel in four North America cities. He found that the uncontrolled ERV (energy wheel) had a negligible impact on the indoor relative humidity, energy consumption in mild and humid climate (Vancouver, British Columbia). Rasouli [66, 67] studied the impacts of ERV on annual energy consumption by conducting energy simulation by TRNSYS [68]. The simulation results indicated that the integration of ERV was ineffective when the enthalpy of indoor air was higher than that of outdoor air, and the cooling energy consumption increased by 5% with uncontrolled ERV. Therefore, an appropriate controller is necessary to avoid the ERV consuming more energy than its recoverable ability. Moreover, a suitable controller for ERV could reduce the cooling energy consumption considerably [44, 66].

A temperature-based control strategy for ERV is proposed by Zhang and Niu [44] for Hong Kong climate. According to psychometric chart, they classified the weather data into 6 regions and compared the performance between the ERV and

HRV in each region. The result presented that the annual power consumption for cooling and heating system decreased when the indoor air quality kept unchanged. However, the authors were unable to calculate the potential saving and they did not consider the effect of latent energy recovery.

Rasouli *et al.* [66, 67] developed an optimum control strategy which depended on an operating factor H^* . This factor was defined as the ratio of the latent and sensible effectiveness of ERV. If the operating factor H^* was equal to one (sensible effectiveness was equal to latent effectiveness), the optimum operating condition at ambient air had higher enthalpy or temperature than indoor air. The control strategy showed that it was possible to save up to 20% annual energy for cooling. Nevertheless, this control strategy used ideal energy consumption and didn't consider the impact of supply air condition on controlling ERV.

2.5. Remarks

This chapter provides a completely review of published works on heat exchangers in the aspects of experimental and numerical investigation, model development, and optimization control. The energy recovery technology is a sustainable solution to solve the problem of increase in building energy consumption without reduction of indoor air quality. The MPC control algorithm is highlighted because of its advantages in dynamic model control. However, it is recognized that few studies have been carried out for the recovery system in tropical climates, such as Singapore, Malaysia, and Thailand. In addition, few researchers have paid attention to control aspect, which is indeed potential for considerable improvement in energy recovery device efficiency. As such, several possible contributions arising from the present project are proposed below:

- Investigation of the performance of ERV in various operated conditions, especially in tropical climates under balanced and unbalanced airflow rate.
- Construction of a dynamic model of the energy recovery ventilator.

- Development of an optimal control scheme for ERV model integrated into the HVAC control system via MPC algorithm.

In the next chapter, the development of numerical model for energy exchanger is presented in detail.

Chapter 3

Development of Computational Fluid Dynamic (CFD) model of ERV and analysis of thermal performance

In this chapter, the numerical model is obtained based on the governing equations of energy recovery processes via heat and moisture transfer between two airstreams. Besides, the computational fluid dynamics (CFD) technique and finite volume method are adopted to simulate the operation of ERV in various configurations and working conditions to investigate the performance of energy recovery system. Mesh generation and mesh independence tests are carried out to decide whether a mesh quality is fine enough for accurate results, yet ensures reasonable computational time. Moreover, previous studies are employed to validate the results from CFD model. The impacts of balanced flow velocity, outdoor air temperature, humidity ratio on the effectiveness of ERV are systematically examined. In addition, the permeability of membrane and the channel height between two membrane layers are also taken into consideration for performance analysis.

3.1. CFD simulation

Over the last decade, with the development of computational technology, the numerical analysis method attracts much attention. Computational fluid dynamics (CFD) simulation, which applies the numerical method, is widely utilized to solve the fluid flow problem. Because the effectiveness correlates strongly with the capability to recover energy, the performance of ERV is intensively investigated by the theoretical model which is based on CFD and finite element method (FEM). In this part, a simulation model of ERV is built in the intensive CFD software ANSYS\FLUENT [69] which was used and validated by numerous engineers and scientists. In addition, the simulation model will support designing a process for the physical model for experimental analysis in the future work.

3.1.1. ERV geometry model

The three-dimensional CAD drawing of ERV with the quasi-counter-flow arrangement is generated in SolidWorks, then imported to ANSYS workbench as illustrated in Figure 3.1. The model consists of supply and exhaust airflow channels, and membrane layers. The supply and exhaust airstreams travel along the parallel membrane layers in cross, counter-current, and cross flow arrangement (quasi-counter-flow) and transfer heat and moisture via the membrane. In the counter flow arrangement, the supply and exhaust airflows move in the channel in the opposite direction. In the cross-flow arrangement, the supply and exhaust airflow move in the channel in perpendicular direction. Moreover, there are 8 plastic ribs which are attached on the membrane to support air channel and introduce the air into cross-counter direction. Due to the symmetric geometry, the computational domain, which includes half of the volume of supply airstream, a membrane, and half of the volume of exhaust airstream, is utilized for simulation as illustrated in Figure 3.2.

Figure 3.1 3D CFD model of ERV in ANSYS FLUENT

Figure 3.2 The symmetric model of a quasi-counter-flow ERV with membrane core

The 3D CFD model of ERV in ANSYS is designed with basic dimensions as shown in Table 3.1. The properties of air, the conditions of supply and exhaust airflows are presented in Tables 3.2 and 3.3.

Table 3.1 ERV dimensions

Length (mm)	x	109
Width (mm)	y	175
Height (mm)	h	227
Air channel (mm)	d	2.5
Number of supply channel	n_s	50
Number of exhaust channel	n_e	50
Membrane thickness (mm)	δ	0.1
Thermal conductivity (W/m*K)	k	0.13

Table 3.2 Properties of the air

Diffusivity water to air (m²/s)	D_{wa}	2.5E-5
Dynamic viscosity (kgm⁻¹s⁻¹)	μ	1.86E-5
Specific heat (kJ/(kg.K))	c_p	1.005

Table 3.3 The conditions of supply and exhaust airflows

Supply dry bulb temperature (°C)	T_{sdb}	34
Supply wet bulb temperature (°C)	T_{swb}	30.9
Supply relative humidity (%)	RH	80
Humidity ratio (kg H₂O/kg dry air)	W	0.02729
Exhaust dry bulb temperature (°C)	T_{edb}	25
Exhaust wet bulb temperature (°C)	T_{ewb}	17.9
Exhaust relative humidity (%)	RH	50
Humidity ratio (kg H₂O/kg dry air)	W	0.0105
Diffusivity water to air (m²/s)	D_{wa}	2.5E-5

3.1.2. CFD configurations

Several assumptions about the ERV physical model are made during the simulation process:

- The ERV operates under a steady state condition.
- The membrane has constant heat conductivity and constant moisture diffusivity.
- The air is incompressible.
- The Reynold number is calculated from ERV dimensions, air properties, and internal air velocity 0.5-2.5 m/s as follow [50]

$$Re = \frac{\rho v D_H}{\mu}$$

Due to relative low air velocity and small channel pitch, it is found that the Reynolds number at the inlet channel has maximum value 800, much less than 2300. Therefore, the laminar flow is modelled during the simulation.

3.1.2.1. Governing equations

The conjugate heat transfer and laminar flow model is employed in the simulation to model the heat transfer and fluid flow inside the ERV channel. Mass transfer equations are utilized to model the diffusion in the membrane and convection in air domain. These models are coupled and solved together with energy balance equations in 3D domain.

Fundamental equations of the fluid motion are the conservation laws of momentum, mass, and energy for incompressible flow. These three fundamental equations make up a well-known system of Navier-Stoke equation, which is the mathematical background of CFD program. Conservation of momentum equation is

$$\rho \frac{\partial u}{\partial t} = \nabla \left[\mu \left(\nabla u + (\nabla u)^T \right) \right] - \rho (u \cdot \nabla) u - \nabla p \quad (3.1)$$

The mass continuity equation is

$$\rho \nabla \cdot u = 0 \quad (3.2)$$

where u is velocity vector, ρ is density, μ is dynamic viscosity, and p is pressure. Due to the steady state condition, the time-dependent parameter is eliminated, the equation (3.1) is simplified as follows:

$$\rho(u \cdot \nabla)u + \nabla \left[p - \mu(\nabla u + (\nabla u)^T) \right] = 0 \quad (3.3)$$

The governing equation of heat transfer in the fluid is:

$$\rho c_p u \cdot \nabla T = \nabla \cdot (k \nabla T) + Q \quad (3.4)$$

where c_p is heat capacity, T is temperature, k is fluid thermal conductivity, and Q is heat source. The simulation is implemented by the conjugate heat transfer model in steady state condition. Therefore, no heat source or heat sink is involved in the model.

The governing equations for mass transfer in ERV with steady state condition are shown as

$$\nabla \cdot (-D \nabla c) + u \cdot \nabla c = R \quad (3.5)$$

where D denotes diffusion coefficient, c is the concentration of species, and R is a reaction rate expression for the species. The reaction rate is zero because there is only moisture in the air, therefore no reaction occurs in this simulation.

3.1.2.2. Boundary condition

No slip condition is assumed at the membrane surface, which means the velocity of fluid at the wall of membrane is zero. No infiltration or cross flow appears on the membrane wall boundary and the symmetric boundary (the center of supply or exhaust air channel). The boundary condition is as follows:

$$u \cdot n = 0 \quad (3.6)$$

$$t \cdot \left(-p + \mu(\nabla u + (\nabla u)^T) \right) n = 0 \quad (3.7)$$

where n and t are normal vector and tangential vector to the boundary, respectively.

It is assumed that the Dirichlet condition for pressure exists at the outlet channels of the ERV and the viscosity is small, therefore

$$p = p_0 \quad (3.8)$$

$$\mu(\nabla u + (\nabla u)^T)n = 0 \quad (3.9)$$

It is also assumed that zero heat and mass flux is imposed on symmetric boundary and at the center of fresh and exhaust air, leading to

$$-n \cdot (-k \nabla T) = 0 \quad (3.10)$$

$$-n \cdot (-D \nabla c + uc) = 0 \quad (3.11)$$

The continuity of heat and mass flux is assumed to take place at the membrane interface; as a result, the boundary condition is as follows,

$$-n \cdot (q_1 - q_2) = 0 \quad (3.12)$$

$$-n \cdot (N_1 - N_2) = 0 \quad (3.13)$$

where q and N are heat flux and mass flux, respectively.

3.1.2.3. Mesh quality and numerical convergence analysis

In this simulation model, the user-controlled mesh is applied since the difference in the width of the membrane, the air channel, and the membrane thickness is considerably large. Due to the geometry of the model, the tetrahedron and hexahedron elements are utilized in all domains. Figure 3.3 shows the final mesh adopted for ERV model.

Mesh independence analysis is implemented to determine the optimal mesh size that provides the desired results with numerical convergence. Five meshing configurations with different mesh sizes of the domains, which are called the coarse (case 1), normal (case 2), fine (case 3), finer (case 4), and finest mesh (case 5), are investigated at the operating conditions, as shown in Table 3.4. The mesh refinement is carried out along the channel height and membrane thickness directions. As listed in the Table 3.4, the temperature and humidity ratio of the outlet of supply airflow are solved as key variables for comparison.

Table 3.4 Mesh independence analysis

Case	Number of elements	Number of nodes	Temperature (°C)	Relative error of Temperature (%)	Humidity ratio (kg/kg)	Relative error of Humidity Ratio (%)
1	38,262	31,130	31.93	3.02	0.02130	2.31
2	70,147	62,260	32.43	1.50	0.02159	1.01
3	165,802	155,650	32.76	0.51	0.02170	0.49
4	261,457	249,040	32.86	0.18	0.02176	0.24
5	325,227	311,300	32.92	0	0.02181	0

In Table 3.4, the relative errors between the cases 4 (finer mesh) and 5 (finest mesh) are 0.18% and 0.24% for temperature and humidity ratio of the supply airflow, respectively. It is concluded that refining the mesh from the finer to finest models has a negligible impact on the key variables. Therefore, the finer mesh model consisting of 261,457 nodes and 249,040 elements, is chosen to get reasonably accurate results with acceptable computational time. The quality of the final mesh model is in good range of skewness and orthogonal distributions.

The numerical convergence criterion is set with the tolerance of 10^{-5} , and all residuals after less than 300 iterations are satisfied by the criterion. Hybrid initialization is adopted for solution initialization.

Figure 3.3 Final mesh for the ERV model

3.2. Model Validation

The studies by Zhang [52] and Al-Waked *et al.* [70] adopted the similar model dimension and properties but the method of solving the mass transfer problem was different to the current model. This study utilizes the porous zone options under cell zone condition to model both the heat and mass transfer problem.

Simulation results are validated by two studies in literature of Zhang [52] and Al-Waked *et al.* [70] under balanced flow velocities. Sensible and latent effectiveness as well as pressure drop are the key criteria in the model validation. The effectiveness is calculated as shown equations (2.9), (2.10).

The current model is firstly validated by the study of Zhang [52] as showed in Figure 3.4. It can be observed that the overall trend for both models is similar in the sense that when velocity increases, the effectiveness decreases, which is a proven trend according to previous studies [52, 70]. The maximum deviation in these two models are 6% and 9% for sensible and latent effectiveness, respectively; and greater deviation occurs at increasing speed. As velocity increases, less heat and mass should be able to exchange through the membrane which results in a lower permeability. The investigation of the permeability value at each velocity scenario requires a separate set of experiments. In addition, the simulation model by Zhang [52] only solved the heat transfer problem while the mass transfer problem was indirectly solved through heat-mass transfer analogy using mathematical models. This deviates from the current model where both heat and mass are coupled and solved by FLUENT simulation.

Figure 3.4 Model validation through effectiveness [52]

Figure 3.5 shows the effect of increasing velocity on pressure drop as part of the model validation. As velocity increases, pressure drop also increases. Pressure drop is highly dependent on velocity, which is supported by previous studies [52, 70].

Figure 3.5 Model validation through pressure drop [52]

The final model to be validated with is the study of Al-Waked *et al.* [70]. The trend in Figure 3.6 is analogous for both models but there is a greater deviation at increasing velocity and could be explained due to the assumption of keeping through-plane permeability constant. The maximum deviation for this model validation are 7% and 12% for the sensible and latent effectiveness, respectively. This higher latent effectiveness deviation could be explained by the fact that Al-Waked *et al.* [70] modelled the membrane as a solid thin wall instead of the current model which utilizes the porous zone option available under the cell zone condition.

Figure 3.6 Model validation 2 through effectiveness [70]

The pressure drop increases as velocity increases in Figure 3.7 is an analogous to the previous model validation as shown in Figure 3.5.

Figure 3.7 Model validation 2 through pressure drop [70]

The heat and mass transfer profile at the symmetric plane of both exhaust and supply airflows at velocities 1 m/s are illustrated in Figures 3.8-11. The heat and mass transfer profiles show noticeable similarities with a slight variation. Therefore, most heat and mass transfer occur at the centre section of ERV core generally. This is well expected because the centre section consists of the counter-flow component which allows for better heat and mass transfer than the two sections at the side which consist of the cross-flow component. In addition, the first variation in contour of Figure 3.8 occurs nearer to the inlet compared to Figure 3.10 where it occurs slightly further away from the inlet. This could signify the ability of the porous zone option in FLUENT being able to capture the moisture resistance as it is greater than the thermal resistance. Similar results can be seen in Figure 3.9 and Figure 3.11. Hence, previous studies appreciated the use of CFD for such application and upon the model validations by others works [52, 70], the current model setup is valid for further investigation.

Figure 3.8 Temperature contours of exhaust airstream on the symmetric plane

Figure 3.9 Temperature contours of supply airstream on the symmetric plane

Figure 3.10 Mass contours of exhaust airstream on the symmetric plane

Figure 3.11 Mass contours of supply airstream on the symmetric plane

3.3. CFD simulation results and discussions

3.3.1. Effect of balanced flow velocity

Figure 3.12 shows the trend on both the sensible and latent effectiveness of the ERV with 8 ribs used in the experimental set-up and it clearly shows that the latent effectiveness decreases at a much faster rate as compared to sensible

effectiveness. This could be explained by the fact that the moisture resistance is higher than the thermal resistance in the membrane. With the addition of the 8 ribs, the ribs help in aligning the flow more uniformly which aid in a better heat and mass transfer but concurrently, the effect of the moisture resistance and thermal resistance become much more significant. Hence, at a higher velocity, moisture faces more difficulty to be transferred over as compared to heat.

Figure 3.13 shows the pressure drop of the ERV with 8 ribs. The pressure drop for this particular ERV with 8 ribs shows a much greater magnitude compared to the ERV without ribs. At high velocity, the pressure drop for the ERV with 8 ribs increases exponentially at a much higher rate. The fan power consumption is proportional to the pressure drop. The addition of the 8 ribs would significantly increase the fan power consumption and thus, increase the operating cost of the ERV as a trade-off while having a better performance in terms of sensible and latent effectiveness due to a better heat and mass transfer.

Figure 3.12 Effectiveness with balanced flow velocity

Figure 3.13 Pressure drop with balanced flow velocity

3.3.2. Effect of temperature and humidity ratio of outdoor air

It can be observed in both Figures 3.14 and 3.15 that, little to no variation when varying either the outdoor air temperature or the outdoor air humidity is shown. A very slight increase is observed in both sensible and latent effectiveness, which could be assumed to have no effect. This could be highly attributed to the effects of the addition of the 8 ribs which aid in both the heat and mass transfer through more uniform distribution of the flow.

Figure 3.14 Variation with outdoor air temperature

Figure 3.15 Variation with outdoor air humidity

3.3.3. Effect of through-plane permeability

A similar trend is observed in Table 3.5 through different materials of through-plane permeability compared with ERV. As permeability increases, the sensible and latent effectiveness increases. Pressure drop decreases from an increasing through-plane permeability. However, the overall pressure drop for the ERV with 8 ribs increases significantly due to the addition of the ribs. Hence, fan power consumption also significantly increases even with greater through-plane permeability.

Table 3.5 Permeability of materials in ERV with 8 ribs

Through-plane permeability (m ²)	Sensible	Latent	Pressure drop
1.22e-13 (GB)	82.7	65.8	10.41
8.33e-12 (PP)	91.6	81.0	7.87
3.80e-11 (PET)	97.0	92.7	6.9
4.70e-11 (PE)	97.7	94.2	6.79

3.3.4. Effect of channel pitch

Similar results are shown in Table 3.6 , where ERV with and without ribs are compared with each other. The pressure drop is significantly higher due to the addition of the ribs. Hence, for ERV with 8 ribs, with a small channel pitch, it has a better performance, but the fan power consumption significantly increases due to the higher pressure drop.

Table 3.6 Analysis of ERV channel pitch variations

Channel pitch (mm)	Sensible	Latent	Pressure drop
2 mm	95.5	89.4	10.31
2.5 mm	91.6	81.0	7.87
3 mm	87.1	73.1	6.45

3.3.5. Velocity contours

Figure 3.16 shows the velocity contours at the symmetric surfaces of the ERV in simulation, when the velocity inlet is 2 m/s. It is observed that the airflows travel along the guided channel at various velocities, where the air velocity at the middle part of the ERV is decreased due to the width expansion of airflow. Moreover, heat and mass transfer ability is increased in lower velocity of airflow, especially in counter flow arrangement. The flow contours also explain the advantages of the diamond shape geometry of ERV which is easy for sealing the device and possible for maximizing the effectiveness.

(a)

(b)

Figure 3.16 Velocity contours at the (a) symmetric surface on top (b) symmetric surface at bottom

3.4. Remarks

In this chapter, the ERV theoretical model is derived from the balance equations of mass, momentum, and energy. The computational fluid dynamics (CFD) technique and finite element method (FEM) is adopted to solve the model numerically by the commercial software ANSYS\FLUENT, which is chosen as one of most powerful CFD tools to solve the governing equations efficiently.

Two previous studies are employed to validate the obtained ERV model with a good agreement of results. The performance of ERV is thoroughly examined under various configurations and working conditions such as balanced airflow velocity, air temperature and humidity, the permeability of membrane, and air channel height variation. The results show that, under tropical weather condition, both sensible and latent energy can be recovered with high effectiveness, leading to the considerable energy saving in HVAC system.

In the following chapter, the experimental setup of ERV system is carried out to analyze the performance of energy exchanger and validate the mathematical model.

Chapter 4

Experimental analysis of ERV thermal performance

This chapter presents the experimental setup of the energy recovery system and investigates the thermal performance of ERV under real weather condition. In addition, the experimental results are also used to validate the theoretical model developed in Chapter 3. The stacked membrane-based core with thin channels is explored as the key component in the ERV system, where a specialized temperature and humidity control chamber is utilized to reproduce the outdoor air condition of a tropical climate. Various types of transmitters coupled with Data Acquisition (DAQ) system are equipped into the experiment to measure the important parameters accurately and to manage the data for further analysis. Investigations into the energy saving performance of the membrane-based ERV system are conducted subject to different environmental conditions, including temperature, humidity, and air velocity.

4.1. Experimental design and setup

In order to analyze the performance of a membrane-based total energy exchanger in various environmental conditions and validate the simulation result, an experiment is conducted, as illustrated in Figure 4.1. The experimental model comprises of an energy exchanger core, two blowers, an environmental chamber, and two main wind tunnels. Fresh air is drawn from the outdoor environment whose condition heavily depends on outdoor weather. Therefore, the fresh air is presented in an environmental chamber first. The temperature and humidity of supply air are varied by the dehumidifier and heater in the chamber before the air travels through the core of ERV and exchanges energy. The performance of ERV is analyzed under various conditions of temperature and humidity to investigate the dependence of ERV performance on inlet condition. In addition, two blowers at the supply and exhaust air inlets are variable. Therefore, the supply and exhaust airflow rates are controlled at various velocities to explore the effect of balanced and unbalanced airflow rates.

Figure 4.1 The schematic drawing of experimental model of ERV

4.1.1. Experimental test rig

Based on the experimental design described in Section 4.1, a field-based testing, which utilizes a membrane-based ERV device with cross-counter flow arrangement, is carried out at Process Instrumentation Lab and Material Lab 2, Nanyang Technological University. The membrane-based ERV is implemented, instead of the typical desiccant wheel because of its advantages, including little cross-contamination, no moving parts, and simplicity; thereby, allowing the integration of the ERV into an existing air conditioning system, in order to efficiently transfer both the latent and sensible heat.

The entire experimental setup for performance analysis of membrane-based energy recovery system is illustrated in Figure 4.2. The test rig consists of an energy recovery device with two membrane-based cores, two air tunnels with different air conditions, two controllable air blowers, a specialized temperature and humidity control chamber, and various types of sensors coupled with data acquisition (DAQ) system. The heater and humidifier chamber are utilized to reproduce the outdoor air condition by regulating air temperature and humidity to desired setpoints before introducing the air through the ERV system. There are two flat bracket heaters with total power of 2kW, which are stably controlled by Carel IR33 professional controller. Besides, the ultrasonic humidifier, which has a capacity of 6 L/hour and an integrated controller, is employed to add sufficient amount of moisture into the fresh air, to reach relative humidity setpoints. Moreover, the air duct is covered by 20mm thick layers of polystyrene insulation to reduce the effect of the surrounding environment. The whole experiment is constructed in Material Lab 2 with stable temperature and humidity condition. Therefore, the generated heat flux and moisture can be efficiently adjusted based on the temperature and humidity setpoints. The ERV operation under various inlet air conditions from the temperature and humidity control chamber is carried out to investigate the ERV performance, for example, very hot and very humid air conditions, mild air condition, hot and less humid air conditions.

Figure 4.2 Experimental setup for the analysis of membrane-based recovery system, which comprises of hot and humid airflow from outdoor, cold and dry return airflow from the indoor environment, various types of sensor system, and NI DAQ to collect data.

The most important part of the total heat exchanger in this experiment is the membrane core inside ERV unit. There are 2 membrane cores with diamond shape, dimension 109mm x 175mm x 227.5mm and thickness 250mm as illustrated in Figure 4.3. Each core of the exchanger consists of 100 thin membranes which are stacked together by a plastic frame with 2.5mm thickness. The exhaust and fresh air are divided into internal airstreams which flow in cross-counter-cross arrangement between thin plates as shown in Figure 4.4. This airflow arrangement incorporates the advantages of high energy exchange efficiency of counter flow and easy fabrication of cross flow. The thermal energy and latent energy are transferred through heat and moisture exchanger surfaces.

Figure 4.3 Dimension of ERV membrane core unit at front view and top view

Figure 4.4 Airflow arrangement inside the membrane core of ERV unit

The air controlling units include two variable-speed blowers with control panels to control the air flow rate through the supply and exhaust duct. The supply air blower introduces hot and humid fresh air from the outdoor environment through membrane core to indoor environment. Similarly, the exhaust blower also pulls out cool and dry air from the indoor environment through membrane core to outdoor. Because of the gradient of temperature and humidity, both airflows will exchange enthalpy energy, but not get mixed inside the core of the energy recovery system.

In addition, humidity and temperature sensors, differential pressure transmitters, air velocity transmitters are set up at four sides of the ERV to measure the real-time data of the air condition. The specification, measurement procedure, as well as data acquisition system will be discussed in the next section.

4.1.2. Instrumentation

Investigation of the performance of energy recovery system requires various complicated parameter measurements. To achieve reliable data, the instrumentations such as transmitters, sensors, and DAQ system are carefully chosen and compatible with HVAC application. All measurement devices with associated accuracies utilized in the experiment are shown in Table 4.1.

Table 4.1 Measurement instruments with associated accuracies.

Instrument	Parameter	Measurement Range	Accuracy
Temperature/Humidity transmitter – EE21	Air temperature	–40 to 60 °C	±0.3 °C
	Air relative humidity	0 to 100%	±2% RH
Gems 5226 – Differential Pressure transmitter	Pressure	-50 to 50 Pa	±1Pa
Air velocity transducer – TSI 6455	Air velocity	0.125 to 50 m/s	±2%
Kestrel 3500 Pocket Weather Meter (hand held)	Temperature	-29 to 70 °C	±0.5 °C
	Relative humidity	5 to 95%	±3% RH
	Air velocity	0.6 to 40 m/s	±3%
OWL Wireless Electricity Monitor	Power (kW)		
	Accumulated energy (kWh)		
	Energy cost		
Multifunction DAQ NI USB-6002	Data Acquisition	8 input channels, ADC 16 bit, sample rate 50kS/s 2 Analog outputs, 16 bit	

Temperature and humidity EE21 transmitter is developed for precision measurement of humidity and temperature. The humidity and temperature sensor

of this transmitter is fully encapsulated and protected by a proprietary coating. This allows the sensor to be in environments such as soft sprayed mist, chlorine, bromine ammonium, and other such environments might face. This sensor can also handle high to near condensing environment. Inside the transmitter, there is an electronic circuit board and a pin connector which can be configured with a computer using a cable to change temperature scaling and do humidity calibration. In the experiment, four humidity/temperature transmitters are used to collect the data of the Outside Air, Supply Air, Return Air, and Exhaust Air streams through sensors.

In order to determine the airflow rate, air velocity transducer TSI 8455 is utilized as a main device for measurement. The transducer is ideal for both temporary and permanent installation in the air duct, at the same position of temperature and humidity transmitter. The velocity range and accuracy of the transducer is 0.125-50m/s and $\pm 2\%$ measured range, respectively. This wide range and high accuracy can fit every measurement demand in the experiment. The signal output is sent out as either a current or voltage, which is suitable for a variety of data acquisition systems. Moreover, the transducer contains an onboard LCD which can show real-time velocity with fast response time.

Besides, the Kestrel 3500 weather meter is a convenient handheld device which provides a complete range of weather measurements: air temperature, humidity, wind speeds, barometric phenomena, etc. The accuracy of the weather meter is in an acceptable range as shown in Table 4.1.

The pressure drop of recovery system is measured on both air sides: supply air and exhaust air by differential pressure transmitter Gems 5226. On the supply airside, the pressure difference is determined between the outdoor airstream to recovery unit and ambient air of indoor environment. On exhaust airside, the pressure difference is determined between return airstream to recovery unit and ambient outdoor air environment. These pressure drops, which are compensated by two blowers on supply and exhaust airside, are main sources of energy consumption in energy recovery system.

4.2. Experimental procedure

As mentioned in the above section, the whole experiment test rig located in Process instrumentation lab and Material lab 2 of NTU where there are steady cold and dry air conditions. Firstly, the energy recovery ventilator is operated for a while to pull out old air inside the core and ensure consistent conditions for the measurement system. Then, the speeds of two blowers are adjusted to achieve balanced airflow rate working condition for the energy recovery system. All performance assessments are carried out at a balanced flow rate in the range of 70-200 m³/h (equivalent to air velocity from 1.5 to 3 m/s in pipe). Heater with PID controller and humidifier are turned ON to add thermal energy and moisture stream into the inlet air to simulate various outdoor air conditions for analysis. Temperature setpoints of the system are adjusted in the range of 28-35°C and humidity ratio difference between the outdoor air and return air is adjusted in the range of 4-8g/kg. Depend on low or high setpoints of temperature and humidity, the temperature and humidity control chamber needs 10-30 minutes to achieve a steady state working condition. Temperature and humidity measurement is conducted at many different points in the experimental rig and real-time data is shown in the data acquisition (DAQ) system. Therefore, the operator easily observes whether the system reaches a steady state condition then records measurement data.

4.3. Data acquisition system

The operational data observed from the sensors, which are equipped with the recovery device, is collected by the data acquisition system, and analyzed further. The data acquisition system based on NI LabVIEW (Laboratory Virtual Instrument Engineering Workbench), a programming software from National Instruments, is utilized to acquire various operation data for the experiment. Integrating all the tools that engineers and scientists need to build a wide range of applications, LabVIEW is a development environment for problem solving,

accelerated productivity, and continual innovation. LabVIEW also supports thousands of hardware devices, including sensors and the DAQ.

In the experiment, national instrument USB-6002 is the DAQ to acquire data for LabVIEW program. All of 8 analog input channels in NI USB-6002 (AI0 to AI7) are utilized for temperature and humidity ratio data collection from 4 transmitters as illustrated in Figure 4.5.

Figure 4.5 DAQ NI USB-6002 with pin outs and connections

In order to monitor the real-time performance of recovery system and record data for analysis, a computer program as an interface between DAQ and operator is created in LabVIEW as illustrated in Figure 4.6. Electronic signal (current 4-20mA or voltage 0-10V) from a variety of transmitters is processed and converted to required data such as air temperature, humidity, velocity, and pressure drop. Moreover, the program also incorporates few complicated formulas to derive expected data, such as humidity ratio, enthalpy energy, and effectiveness of the system. All recorded data is reflected as real-time monitoring of virtual

instruments (VI) as illustrated in Figure 4.7 and automatically captured in a database for further investigation.

Figure 4.6 LabVIEW program for monitoring real-time data and logging database

Figure 4.7 Virtual instruments for monitoring real-time data of air temperature, humidity ratio, and effectiveness

4.4. Experimental evaluation

The sensible and latent effectiveness, and total recovered energy are the critical operating performance parameters of an energy recovery system.

4.4.1. Relative humidity and humidity ratio

Relative humidity (RH) is the ratio of water vapor present in the air at the partial pressure to that at the saturation pressure at a certain temperature, expressed in percentage. Relative humidity depends on temperature and air pressure at a given condition, but it does not reflect the amount of vapor occurring in the airstream. As such, even though data from the humidity transmitter is relative humidity, another appropriate parameter, which is humidity ratio, is employed to investigate the performance of moisture transfer in ERV.

On the other hand, humidity ratio is calculated as the mass of water vapor in the humid air (in kilogram) per unit mass of dry air (in kilogram). As such, the actual amount of vapor occurs in outdoor and return airstream can be measured to investigate the performance of the ERV system. As mentioned above, the humidity transmitter is only capable of recording the relative humidity value. Hence, a series of calculation are utilized to convert relative humidity to humidity ratio as follows [50]:

$$P_{ws} = e^{\left(\frac{77.345 + 0.0057T - \frac{7235}{T}}{T^{8.2}} \right)} \quad (4.1)$$

$$\text{Relative Humidity (RH)} = \frac{P_w}{P_{ws}} * 100\% \quad (4.2)$$

$$\text{Humidity Ratio (W)} = \frac{0.62198P_w}{(P_A - P_w)} \quad (4.3)$$

where T (°C) is air temperature, P_{ws} (Pa) is water vapor saturation pressure, P_w (Pa) is actual vapor pressure, P_A (Pa) is atmospheric pressure which is at 101325 (Pa), and humidity ratio is denoted as W (kg/kg).

4.4.2. Analysis of ERV performance

Effectiveness is a good indicator to evaluate the performance of ERV system in terms of capability to recover waste energy from exhaust airstream. The effectiveness is calculated by the ratio between the actual energy recovered and maximum energy that can be recovered. There are 3 types of effectiveness [51]:

- Sensible heat transfer effectiveness (ε_S)
- Latent heat transfer effectiveness (ε_L)
- Total heat transfer effectiveness (ε_H)

Sensible effectiveness is the ratio of the amount of thermal energy transferred between exhaust and supply airstream to the maximum amount of thermal energy transferable. Therefore, sensible effectiveness can be calculated as follows:

$$\varepsilon_S = \frac{m_s c_{ps} (T_{OA} - T_{SA}) + m_e c_{pe} (T_{EA} - T_{RA})}{2m_{\min} c_p (T_{OA} - T_{RA})} \quad (4.4)$$

where m_s is the mass flow rate of the supply airstream, m_e is the mass flow rate of the exhaust airstream, c_p is specific heat of air, T_{OA} , T_{SA} , T_{EA} , T_{RA} are the temperatures of the outdoor air, supply air, exhaust air, and return air, respectively.

Latent effectiveness is the ratio of the amount of latent energy transferred between two airstreams to the maximum amount of latent energy possibly transferred. Therefore, latent effectiveness can be calculated as follows:

$$\varepsilon_L = \frac{m_s h_{fg} (W_{OA} - W_{SA}) + m_e h_{fg} (W_{EA} - W_{RA})}{2m_{\min} h_{fg} (W_{OA} - W_{RA})} \quad (4.5)$$

where h_{fg} is enthalpy of vaporization, the W_{OA} , W_{SA} , W_{EA} , W_{RA} are the humidity ratio of the outdoor air, supply air, exhaust air, and return air, respectively.

Since both sensible and latent energy involve in recovery process, the enthalpy which is defined as total energy in a thermodynamic system is evaluated. The specific enthalpy denoted by H (kJ/kg) can be calculated as [50]:

$$H = 1005T + W(2501 + 1.86T) \quad (4.6)$$

Enthalpy effectiveness is defined as the ratio of the amount of enthalpy transferred to the maximum amount of enthalpy that can be transferred. The enthalpy effectiveness is evaluated as follows:

$$\varepsilon_H = \frac{m_s(H_{OA} - H_{SA}) + m_e(H_{EA} - H_{RA})}{2m_{\min}(H_{OA} - H_{RA})} \quad (4.7)$$

where H_{OA} , H_{SA} , H_{EA} , H_{RA} are the specific enthalpy of the outdoor air, supply air, exhaust air, and return air, respectively.

Under balanced airflow condition, equations (4.4), (4.5), and (4.7) can be simplified as follows:

$$\varepsilon_S = \frac{T_{OA} - T_{SA}}{T_{OA} - T_{RA}} \quad (4.8)$$

$$\varepsilon_L = \frac{W_{OA} - W_{SA}}{W_{OA} - W_{RA}} \quad (4.9)$$

$$\varepsilon_H = \frac{H_{OA} - H_{SA}}{H_{OA} - H_{RA}} \quad (4.10)$$

4.4.3. Energy saving in ERV

The effectiveness is usually expressed in percentage of the ratio of the actual amount of recovered energy to maximum energy that can be recovered between two airstreams. Hence, the absolute amount of sensible, latent, and enthalpy energy recovered by ERV system is examined further to obtain insights of the saving capability and cost saving by the recovery system. The sensible and latent energy saving can be formulated as follows,

Amount of sensible energy saving (E_S)

$$E_S = c_p \rho V (T_{OA} - T_{SA}) \quad (4.11)$$

where c_p is the specific heat of air, ρ (kg/m³) is density of air, V (m³/h) is volumetric flow rate, T_{OA} is the temperature of outdoor air, and T_{SA} is the temperature of supply air.

Amount of latent energy saving (E_L)

$$E_L = h_{fg} \rho V (W_{OA} - W_{SA}) \quad (4.12)$$

where h_{fg} (kJ/kg) is the latent heat of vaporization of water, ρ (kg/m³) is density of air, V (m³/h) is volumetric flow rate, W_{OA} and W_{SA} are humidity ratio of outdoor and supply air, respectively.

Total energy saving (E_{tot}) is calculated as follows,

$$E_{tot} = E_S + E_L \quad (4.13)$$

4.4.4. Uncertainty analysis

The uncertainty analysis is carried out to correlate the uncertainty of instruments and the repeated measurements. In the experiment, the air temperature and humidity are measured by EE21 transmitter with an accuracy of $\pm 0.3\%$ and $\pm 2\%$, respectively. The air velocity is measured by transducer TSI 6455 with the calibration error with in $\pm 2\%$. The pressure drop is measured by differential pressure transmitter Gems 5226 with an accuracy of $\pm 1\%$. Moreover, the uncertainty can occur when conducting the repeated measurements. The estimated standard deviation for a series of N measurements can be expressed [71]:

$$SD = \sqrt{\frac{\sum_{i=1}^N (X_i - \bar{X})^2}{N-1}} \quad (4.14)$$

where X_i is the result of i th measurement and $\bar{X}$ is the mean of N considered results. The uncertainties associated with respective error bars are presented in the experimental results section.

4.5. Experimental results and discussions

Effects of temperature, humidity, velocity on sensible and latent effectiveness, and energy saving of recovery system are investigated in this section.

4.5.1. Effect of temperature on sensible effectiveness

Figure 4.8 shows the impact of temperature on sensible effectiveness at different air velocities. As a general trend, the sensible effectiveness slightly increases with the growth of outdoor air temperature, at different airflow rates. For example, when the temperature increases from 28 to 35 °C, the sensible effectiveness also increases about 70 to 79% at the velocity of 1.5 m/s, while there is only 5% growth from 63 to 68% at the velocity of 3 m/s. This improvement result is probably caused from the enhancement of heat transfer ability via the membrane, due to the increase in the temperature gradient between outdoor and return airflows. Furthermore, the sensible effectiveness slightly decreases when the velocity increases. For example, if the air velocity increases from 1.5 to 3m/s, the sensible effectiveness decreases from 70 to 63% at 28 °C, and reduces from 79 to 68% at 35 °C . Due to the increase in air velocity, the travel time of air through the core is declined, leading to the reduction of energy exchange between the supply and exhaust airstreams. Consequently, the sensible effectiveness is reduced.

Figure 4.8 Impacts of outdoor air temperature on sensible effectiveness at different air velocities

The effect of temperature on sensible effectiveness at different humidity ratio is illustrated in Figure 4.9. It is shown that the humidity ratio has a negligible impact on sensible effectiveness at the same temperature. It is explained that, the increase of humidity ratio does not affect the thermal energy of outdoor airstream. Therefore, the amount of thermal energy transferred via membrane is not affected.

Figure 4.9 Impacts of outdoor air temperature and humidity ratio on sensible effectiveness (air velocity = 1.5m/s)

The effect of outdoor air temperature on sensible energy recovered at different air velocities (1.5 – 3 m/s) is shown in Figure 4.10. When outdoor air temperature increases, the thermal energy of airstream also increases proportionally. Therefore, the amount of energy recovered by thermal transfer increases significantly from 163W to 293W at 1.5 m/s, and 338W to 700W at 3m/s.

Figure 4.10 Impacts of outdoor air temperature on sensible energy recovered at different air velocities

4.5.2. Effect of humidity ratio on latent effectiveness

Effect of outdoor air humidity ratio on latent effectiveness is analyzed and illustrated in Figure 4.11. When the humidity ratio difference between outdoor air and return air increases from 2 to 6g/kg, latent effectiveness slightly decreases from 77 to 72% at velocity 1.5m/s and 71 to 62% at velocity 3m/s.

Figure 4.11 Impacts of outdoor air humidity ratio on latent effectiveness at different air velocity

The effect of humidity ratio on latent effectiveness at different temperature is illustrated in Figure 4.12. It is shown that the temperature has small impact within 3% on latent effectiveness at the same humidity ratio difference. It is explained that, the increase of temperature does not affect to latent energy of outdoor airstream. Therefore, the amount of latent energy transferred via membrane is not affected.

Figure 4.12 Impacts of outdoor air humidity ratio on latent effectiveness at different outdoor air temperature (velocity = 3 m/s)

The effect of outdoor air humidity on latent energy recovered at different air velocities (1.5 – 3 m/s) is shown in Figure 4.13. When outdoor air humidity ratio increases, the latent energy of airstream also increases proportionally. Therefore, the amount of energy recovered by moisture transfer increases significantly from 147W to 372W at 1.5 m/s and 254W to 643W at 3m/s.

Figure 4.13 Impacts of outdoor air humidity ratio on latent energy recovered at different air velocities

4.5.3. Effect of velocity on sensible and latent effectiveness

The effect of velocity on sensible and latent effectiveness is illustrated in the Figure 4.14. The results show that the performance of ERV depends heavily on the velocity. When the velocity increases from 1.5 m/s to 3m/s, the sensible and latent effectiveness decreases from 77% to 70% and from 72% to 64%, respectively. The sensible effectiveness is always higher than the latent effectiveness because the thermal resistance of the membrane is a very small value, thus thermal energy can transfer via membrane easily while the mass transfer resistance of membrane is quite high due to the characteristics of membrane material. Besides, when the velocity increases, the time of the air existing in the core is decreased. Therefore, the time for the airstream to exchange energy is also reduced, causing the effectiveness decrease consequently.

Figure 4.14 Effect of balanced velocity on sensible, latent and enthalpy effectiveness

4.5.4. Pressure drop in ERV

Pressure drop on both supply and exhaust airsides of the experiment is measured and illustrated in Figure 4.15. It is observed that when velocity increases from 1.5 – 3m/s, the differential pressure on both sides increase significantly from 7 Pa to 20 Pa. This pressure drop is quite small because the membraned-base core is designed very well with 100 channels each side, which is appropriate for airflow rate of 70-200 m³/h.

Figure 4.15 Effect of balanced velocity on pressure drop of supply and exhaust airflow

4.5.5. Recovery of energy by ERV

Sensible, latent, and total energy saving at different velocities, temperature, and humidity ratio are thoughtfully investigated in this section and illustrated in Figures 4.16-19. Minimum energy saving 300W is occurred at minimum outdoor air temperature (28 °C) and humidity ratio (14g/kg) as in Figure 4.16. Maximum energy saving 1300W is achieved at maximum outdoor air temperature (35 °C) and humidity ratio (18g/kg) as in Figure 4.19.

Figure 4.16 Sensible, latent, and total energy saving at different velocities ($T_{OA} = 28\text{ }^{\circ}\text{C}$, $HR_{OA} = 14\text{ g/kg}$)

Figure 4.17 Sensible, latent, and total energy saving at different velocities ($T_{OA} = 28\text{ }^{\circ}\text{C}$, $HR_{OA} = 16\text{ g/kg}$)

Figure 4.18 Sensible, latent, and total energy saving at different velocities ($T_{OA} = 35\text{ }^{\circ}\text{C}$, $HR_{OA} = 16\text{ g/kg}$)

Figure 4.19 Sensible, latent, and total energy saving at different velocities ($T_{OA} = 35\text{ }^{\circ}\text{C}$, $HR_{OA} = 18\text{ g/kg}$)

4.5.6. Comparison between experimental and numerical results

The CFD model is developed from the geometry, material properties, boundary condition of the experimental setup. Moreover, both numerical and experimental models are studied under the similar operating conditions of tropical climate such as high temperature and humidity. The ERV performance results is compared between two models as shown in Figure 4.20. Both the sensible and latent effectiveness decrease with the increase of the inlet velocity from 1.5-3m/s. The sensible effectiveness of CFD model is higher than that of experimental result from 3.9-6.4%, which could be explained by the ideal working condition of simulation compare to the heat loss in experimental setup. The latent effectiveness of the CFD result is also higher than experimental result by 3.5-5%, which could be explained by the corrugation in the membrane and the cross contaminant of experimental setup. It is concluded that a good agreement between numerical and experimental results is achieved. The numerical model could be employed to further investigate the key parameters, which affect the recovery performance of ERV.

Figure 4.20 Comparison of the sensible and latent effectiveness between the numerical and experimental results

4.6. Remarks

This chapter presents experimental setup of energy recovery system to analyze thermal performance of ERV and validate the simulation results. At first, the requirements for experiment design are specifically determined. A heat exchanger comprised of a membrane-based core is the key equipment of the study. Then, the experiment facilities such as temperature and humidity control chamber, sensor and DAQ systems are constructed to reproduce the outdoor air condition in tropical climate and to measure, collect, and store data for further investigation. A number of tests are conducted under various supply air conditions such as temperature, humidity, and velocity. A parametric analysis is also carried out to investigate the energy saving performance of membrane-based energy exchanger. Compared with the simulation results in Chapter 3, the experiment shows a similar trend of the impacts of airflow conditions on ERV performance, where the sensible and latent effectiveness can gain up to 80% and 70%, respectively. The maximum recovered energy 1300W is occurred at outdoor air temperature 35°C and humidity ratio 18g/kg.

Chapter 5

Development of a novel ERV model coupled with HVAC system for optimization

To achieve the objectives of saving energy consumption in buildings and improving indoor air quality by energy recovery technology, an appropriate control strategy for ERV in HVAC system is required. Modeling is the necessary initial stage to implement the model-based control strategy for optimization of energy consumption in buildings. Modeling methods are typically categorized into three classes of model: physical based (white-box) model, data driven (black-box) model, and hybrid (grey-box) model. This chapter firstly reviews the modeling method for HVAC system, then presents the model identification of building thermal dynamic and HVAC components such as air handling unit, chiller, cooling coil, and ERV unit.

5.1. Background

Modeling of HVAC system is essential to optimize the energy usage efficiency and the indoor air quality of the building. An appropriate model is necessary to implement control method and acquire accurate results. Typically, the modeling methods are classified into three types including physical based model (white-box), data driven model (black-box), and hybrid model (grey-box).

Physical based model (white-box)

In physical based approach, the model is developed based on the governing equations and the knowledge of the physical principles of underlying process. The development of physical based model requires detailed understanding of system components and significant efforts. Therefore, the model has good generalization abilities and is widely used in building energy simulation software [72]. The physical based method is mostly used for modeling building thermal dynamic and HVAC components, which include the primary and secondary components.

A lot of mature white box software tools, including EnergyPlus [73], ESP-r [74], TRNSYS [75] were widely used to analyze building energy consumption and determine the optimize operation schemes. The input data such as weather conditions, building conditions, and HVAC components' descriptions need to be obtained from design plan, manufacture catalog, which is not always available. The calculation in the physical based model which consists of vastly governing equation systems is significantly time-consuming. That major barrier of the white box building model restricted its implementation to an online model based control and optimization.

Data driven model (black-box)

In data-driven approach, the data of system performance is collected over a certain period under specific tests. The correlation between input and output is

derived by mathematical methods, including statistical models, frequency domain, data mining algorithms, and state-space identification [76, 77]. The detailed knowledge of the system is not required in this approach. However, under different conditions, the model needs to be trained by the measured data to enable the prediction of building behavior.

The data-driven model is widely employed in recent studies to identify the building control model, which assists to optimize the energy usage in building and minimize expense. It is easy to build computational efficiency. However, the huge data training is required in specific building operation conditions is necessary to identify an appropriate model. Sometimes, the data-driven model suffers huge calculation error because the operation condition is not covered in training data set.

Hybrid model (grey-box)

In grey-box approach, the model structure is developed via the white-box method, in which the complex process is simplified to model with unidentified parameters [78]. These parameters are estimated based on the appropriate identification algorithms with system performance measurement. Moreover, less effort and time are required to train the simplified physical model. This hybrid method takes the advantages of the good generalization capabilities of the physical based model and the high accuracy within training domain of the data driven model [79, 80]. However, the knowledge of physics process and the availability of system performance data for model training are necessary.

5.2. Building energy simulation and co-simulation process

Whole building energy simulation (BES) models play a significant role in design and operation optimization of buildings. They may be utilized to predict the energy consumption, the cost-effectiveness, and the environmental comfort in the design stage of building. Moreover, in the operational stage, BES models are implemented for onsite performance optimization of building operation.

5.2.1. Building energy simulation

Various building energy simulation tools are developed and validated for building modeling, simulation, and control. These tools based on the heat balance equations and energy conservation to predict the energy consumption which is required to maintain the thermal comfort in the building (e.g. zone temperature, humidity, and lighting). The influences of external disturbance such as outdoor weather, solar radiation, occupancy, and infiltration are incorporated into the calculation equations. These calculations are generally carried out in the period of hours, days, weeks, months, or a whole year. The most mature tools that will be discussed are TRNSYS, ESP-r, and EnergyPlus.

TRNSYS [81] is a Transient Systems Simulation program which utilizes component-based system approach to handle the input data as well as the output of simulations. The modular structure provides the flexibility and facilitates the addition model which is not composed in standard TRNSYS library. The program is best suited to analyze the systems such as renewable energy system, HVAC systems, solar system, and co-generation systems.

ESP-r [74] is a general-purpose modeling tool for building performance simulation, including thermal, moisture, visual and acoustic. ESP-r is powerful in modeling different building physics but inflexible for simulating of novel electrical, mechanical, and co-generation systems. The program requires specific information of building, HVAC system, components, and schedules.

EnergyPlus [73, 82] is an advanced whole building energy simulation engine, which is developed for simulation of building thermal behavior, HVAC, lighting system, ventilation, and other energy flow in a building. EnergyPlus implements detailed building physics for air, moisture, and heat transfer; supports a large set of HVACs' component model; combines heat and mass transfer models; simulates sub-hourly time steps to handle fast system dynamics and control strategies; and generates standard summary and detail output reports.

Energy Plus is a console-based program, which reads the input file such as building description and weather data, and writes the output file such as energy

consumption report and zones temperature to the text file. Therefore, several comprehensive graphical interfaces are developed for Energy Plus, such as OpenStudio [83] which is an open source integrated platform. OpenStudio provides a convenient environment for carrying out integrated whole-building energy analysis.

EnergyPlus utilizes the modular-based approach, which is well-organized and facilitated to add new features or link to other programs. This approach provides the ability to simulate a variety of building types, HVAC components, weather conditions and complex configurations. It also supports both standard and innovative system configurations such as renewable-energy unit, solar system, and energy recovery unit. The program has been extensively validated by various studies and tests [84, 85], which demonstrates that EnergyPlus model is reasonable to present the buildings thermal dynamic, analyze energy efficiency as well as incorporate innovative recovery unit into the whole HVAC system. However, because of the complexity, the model is not suitable for predictive control in the optimization process. Therefore, there is a need to develop a simplified and good accuracy model which is utilized for control design.

5.2.2. Co-simulation platform

While building simulation tools can well present the behavior of buildings and accurately calculate building energy consumption, they cannot perform advanced control techniques and optimization process, which are possibly developed in MATLAB environment. Thus, there is a need for the end-to-end co-simulation platform for building energy-efficiency control. Bernal *et al.* [86] developed a co-simulation toolbox, MLE+, which utilized the building simulation model with the advantages of the controller and optimization design capabilities from MATLAB environment.

MLE+ allows different tools running simultaneously and exchanging information via a co-simulation platform. The graphic front-end of the program streamlines the necessary parameters between building model and controller, as

well as provides the control development process, in which the building model is presented in EnergyPlus, and the controller design is carried out in MATLAB or Simulink. The MLE+ control interfaces toolboxes with building models is illustrated in Figure 5.1. Various toolboxes in MATLAB such as Model Predictive Control toolbox, System Identification, and Optimization Toolbox can be implemented conveniently in the co-simulation process.

Figure 5.1 MLE+ interfaces toolboxes with building models [86]

5.3. Building thermal dynamics model

The thermal dynamics model is developed based on the thermal mass of building material, building structure, and the indoor and outdoor air condition. The high accuracy of the model is crucial to predict and maintain the indoor air condition in the thermal comfort level.

The modeling objective is to develop a dynamic model which is less complicated to incorporate to the controller and optimization design (particularly model predictive control), yet ensures reasonable accuracy to represent the dynamic behavior of building thermal performance. The physical based approach is adopted to model: heat transfer by conduction and convection, solar radiation on the exterior and interior surface, internal gain (equipment, occupancy) on interior surfaces.

5.3.1. Building thermal model

There are several assumptions that need to be considered to develop the building thermal model:

- The zone air is well-mixed. Therefore, the temperature, humidity, and CO₂ concentration in one zone have a uniform distribution.
- The air density in each zone is constant and not affected by temperature or humidity change.
- No air infiltration and exfiltration are considered, which means the building has balanced ventilation between inlet airflow and outlet airflow.

Due to the requirement of a sufficiently simplified model for implementation of MPC controller, the physical thermal model of building in [87, 88] is adopted to develop the room temperature model. Based on the energy balanced equation of zone air, the room temperature is calculated as follows:

$$m_{\text{air,zone}} c_{pa} \frac{dT_{\text{room}}}{dt} = Q_{\text{vent}} + Q_{\text{cool}} + Q_{\text{int}} + \sum_j Q_{\text{wall},j} + \sum_j Q_{\text{win},j} \quad (5.1)$$

Heat energy stored in the zone air is shown in the left side of the energy balance equation. The right side is the sum of heat energy due to ventilation air (Q_{vent}); heat energy of cooling air (Q_{cool}) to maintain the thermal comfort in building; heat of internal components (Q_{int}) including occupants, equipment, and lighting; and heat flow from the building envelope, including wall (Q_{wall}) and windows (Q_{win}). Each thermal component in the right side is calculated as follows:

$$Q_{\text{vent}} = \sum m_{\text{vent}} c_{pa} (T_{\text{ai}} - T_{\text{room},j}) \quad (5.2)$$

$$Q_{\text{cool}} = \sum m_{\text{cool}} c_{pa} (T_{\text{sa}} - T_{\text{room},j}) \quad (5.3)$$

$$Q_{\text{int}} = cN_{\text{people}} \quad (5.4)$$

$$Q_{\text{wall}} = \sum A_w h_w (T_w - T_{\text{room},j}) \quad (5.5)$$

where m_{vent} is mass flow of ventilation air, T_{ai} is temperature of air inlet (K), T_{room} is temperature of room (K), c_{pa} is specific heat of room air ($kJ/kg K$), m_{cool} is mass flow of cool air from cooling coil, T_{sa} is supply air temperature (K), N_{people} is number of people, A_{wall} is surface area of wall (m^2), h_w is average heat transfer coefficient of wall ($kW/m^2 K$), and T_w is inside surface temperature of wall (K).

5.3.2. RC networks method

The building envelope includes external wall, internal wall, floor, and roof, which are boundaries between the environments outside and inside the building. The interaction between the building envelope and indoor air will determine the actual thermal demand and building thermal condition. The analysis of these dynamic interactions is critical to calculate power consumption of the building and thermal comfort for occupants. In order to obtain the whole thermal dynamics model, the behavior of wall temperature and interaction between zone and envelope needs to be determined. Wall temperature is calculated by means of energy balance between indoor and outdoor surfaces.

The Resistances Capacitances (RC) networks [89] are commonly adopted to construct the thermal model of transient heat flow through the wall of the building. The resistance and capacitance are carefully evaluated to model the effects of conduction and convection between the air zones and the wall, as well as the long-wave radiation. Gouda *et al.* [89, 90] showed that three resistors two capacitors (3R2C) network model achieved the best compromises of model accuracy and model complexity. The thermal capacitance and resistance incorporated in thermal system in the building are similar to those in an electrical network, which is illustrated in Figure 5.2. The temperatures of the indoor zone, outdoor air, an interior surface, and exterior surface are represented as the voltage at node T_i , T_o , $T_{w,i}$, and $T_{w,o}$, respectively. The coefficients of heat convection between zone and surface, and heat conduction between nodes is characterized by resistors $R_{w,i}$, $R_{w,m}$, and $R_{w,e}$ where subscripts w, i, m, e denote wall, interior,

middle, and exterior. The thermal capacitances of zone air and wall surface are modelled by capacitors $C_{w,e}$ and $C_{w,i}$. The heat fluxes between the thermal nodes are represented by the current between the nodes in RC networks.

Figure 5.2 Three resistors two capacitors (3R2C) analogue heat transfer model

Heat transfer between the wall surfaces, which is shown in Figure 5.2, is presented below:

$$C_{w,e} \frac{dT_{w,e}}{dt} = \frac{(T_o - T_{w,e})}{R_{w,e}} - \frac{(T_{w,e} - T_{w,i})}{R_{w,m}} \quad (5.6)$$

$$C_{w,i} \frac{dT_{w,i}}{dt} = \frac{(T_{w,e} - T_{w,i})}{R_{w,m}} - \frac{(T_{w,i} - T_r)}{R_{w,i}} \quad (5.7)$$

where, T is the temperature of zone or wall surface (K), C is the total thermal capacitance of layer (kJ/K), and R is thermal resistance (K/W)

The 3R2C network method is implemented to the three zones building as illustrated in Figure 5.3. The main nodes are zone air temperature nodes, outdoor air temperature nodes, and surface temperature nodes. The RC model is simplified here for demonstration purpose.

Figure 5.3 The RC networks model for 3 zones building

All the parameters in the thermal network model are determined by the geometry of building, the physical properties of materials, and the knowledge of construction models. The detailed description of the building including geometry, materials properties, operation schedules, and HVAC components are interpreted in the text-based file of EnergyPlus. Typical weather data for specific location is obtained from available weather database or direct measurement and prediction of the sensor system in the building. Moreover, sensors network of building detects the occupancy scheme in zones easily. All the description of building, weather database, and occupancy scheme are treated as input of building energy simulation program EnergyPlus which adopts the physical based method [91]. The parameter calculation in RC network model is carried out by EnergyPlus by solving the linearized differential equation set. The post-processed data from EnergyPlus including thermal capacities, heat conduction coefficients, heat convection coefficients, and energy flux is exchanged with MATLAB through a co-simulation platform. The thermal model in MATLAB is converted to state-

space model for an advanced controller such as model predictive control (MPC) as described below:

$$x(k + 1) = Ax(k) + Bu(k) + Ew(k) \quad (5.8)$$

$$y(k) = Cx(k) \quad (5.9)$$

where $x(k)$ is state vector containing all value of zone air temperature, interior and exterior surface temperature of walls; $u(k)$ is input vector including heating or cooling heat flux, and radiation energy; $w(k)$ is disturbance vector including weather disturbance (ambient temperature, solar radiation) and internal disturbance (number of occupants, equipment operation); $y(k)$ is the output of system which is the room temperature at time k ; matrix A , B , C , D with appropriate size is obtained from solving the energy balance equations in EnergyPlus.

5.4. HVAC components model

The HVAC system is a complicated system which typically comprises of subsystems such as energy recovery ventilator (ERV), air handling unit (AHU) with cooling or heating coil, fan and pump, duct and pipe. A typical HVAC model with three zones building is simplified as illustrated in Figure 5.4. The model of each subsystem can be identified by techniques such as white-box, black-box, or grey-box modeling. Then subsystem models are combined to obtain a full HVAC model which is necessary for the implementation of energy efficiency strategies, advanced controller development, and operation optimization purposes. In this part, a typical HVAC system including ERV and AHU with the cooling coil is demonstrated by physical based modeling approach. The accurate models of these components are critical to investigate the effects of the advanced controller such as model predictive control (MPC) on total energy consumption and optimization operation schemes for the whole system.

Figure 5.4 A typical HVAC model with three zones building

5.4.1. ERV unit model

The ERV unit exchanges the sensible and latent energy between the supply air from the outdoor environment and exhaust air from building zones. In tropical climates, the outdoor weather at daytime is usually very hot and humid, while the indoor air is maintained cool and dry for thermal comfort of the occupants. Therefore, when two airflows go through the ERV unit, the cold and dry exhaust air will pre-cool and dehumidify the hot and humid supply air. Through the energy exchange process, the energy consumption to condition the hot and humid supply air is reduced. Moreover, the ERV unit introduces the fresh air which is necessary to dilute the concentration of the contaminant in enclosed zone, then keep the indoor air quality in a suitable range. The amount of required fresh air mostly depends on the number of occupants inside the building, air quality of the outdoor environment, and the volume of building zone.

The white box model of ERV unit is developed based on the energy balance equations for the supply and exhaust airstreams. ERV unit comprises of a plate-type core and two fans for supply and exhaust airstreams. Figure 5.5 illustrated the energy balance equations of ERV.

Figure 5.5 The energy balance equations of ERV unit

The ERV model inputs are two mass flow rates of outdoor and return air and two temperatures of outdoor and return air. The balanced flowrate of outdoor and return air is studied in this case. The model outputs are the temperature of supply and exhaust air at the outlets of ERV.

$$\frac{dT_{ea}}{dt} = \frac{1}{C_{am}} \left[m_{ea} C_{pa} (T_{ra} - T_{ea}) - (UA) \left(\frac{T_{ra} + T_{ea}}{2} - \frac{T_{oa} + T_{sa}}{2} \right) \right] \quad (5.10)$$

$$\frac{dT_{sa}}{dt} = \frac{1}{C_{am}} \left[(UA) \left(\frac{T_{ra} + T_{ea}}{2} - \frac{T_{oa} + T_{sa}}{2} \right) - m_{sa} C_{pa} (T_{sa} - T_{oa}) \right] \quad (5.11)$$

where C_{am} is thermal capacitance of air and membrane (J/K), T is air temperature (K), m is air mass flowrate (kg/s), C_{pa} is specific heat of air at constant pressure (J/kgK), UA is overall conduction heat transfer coefficient (W/K), and the subscripts oa, sa, ra, ea denote the outdoor air (from outdoor environment), supply air (to the zone), return air (from the zone), and exhaust air (to outdoor), respectively.

5.4.2. Cooling coil model

As well known, the cooling coil unit in HVAC systems is an essential component, which removes the thermal energy from the zone air by forcing the air through the cooling coil and entering the conditioned space. Through this process, the thermal energy is transferred between the air loop and the chilled water loop, thus

the occupants comfort is maintained simultaneously. Existing cooling coil models in the open literature are classified from simple linear approximation to complicated nonlinear model, which is impractical to implement the advanced controls, such as the model predictive control. Yao-Wen *et al.* [92] developed a simple, yet accurate model of cooling coil, based on heat transfer principle and energy balance equations. This model is validated and resulted in satisfied accuracy in real time control and system optimization.

The diagram of a typical cooling coil in air handling unit is illustrated in Figure 5.6. The heat transfer happens between two loops: chilled water loop and air loop, which is forced by the chilled water pump and air supply fan. In the water loop, the cool water with temperature T_{chw} flows into the cooling coil with mass flow rate m_{chw} . In the air loop, the hot air with temperature T_a is forced to pass through the cooling coil with the mass flow rate m_a . At the cooling coil, the heat flux of hot air transfers to cool water by three main processes: the convection in chilled water, conduction at the interface, and convection in hot air. Based on heat transfer theory and energy balance equations, the cooling load is calculated by the overall heat resistance as follows:

$$Q = \frac{T_a - T_{chw}}{R_{cool}} \quad (5.12)$$

where R_{cool} is the overall heat resistance of cooling coil, T_a is the temperature of process air, and T_{chw} is the temperature of chilled water.

Figure 5.6 The schematic of a typical cooling coil in air handling unit

The overall heat resistance R_{cool} comprises of the resistance of three components: air convection, interface conduction, and chilled water convection. The interface of cooling coil usually is fabricated from high conductive materials (copper or aluminum), the resistance of interface conduction is thus relatively small and possible to be discarded in calculation. The overall heat resistance is calculated as follows:

$$R_{cool} = R_a + R_{chw} \quad (5.13)$$

where R_a is the thermal resistance of air convection, and R_{chw} is the resistance of chilled water convection.

Heat transfer rate between the process air and the interface is influenced by the characteristics of interface and moving air, such as geometry of interface, interface material, velocity of process air, and temperature difference between air and water. In cooling coil, the cool water and process air are driven by pumps and air fans, thus the forced heat transfer mechanism is considered for calculating

the heat load. Through dimensionless analysis, the following equation is proposed [93]:

$$\frac{hD}{k} = C \left(\frac{D\rho v}{\mu} \right)^e \left(\frac{c_p \mu}{k} \right)^f \quad (5.14)$$

In steady flow, the fluid density ρ and flow rate v are assumed as constants. Subsequently, the mass flow rate (product of ρv) is constants. Equation (5.14) is rewritten bellow:

$$h = C \left(\frac{4m}{\pi \mu D} \right)^e \left(\frac{c_p \mu}{k} \right)^f \frac{k}{D} \quad (5.15)$$

where h is heat transfer coefficient, D is diameter of passage, m is mass flow rate, ρ is fluid density, v is fluid flow rate, c_p is fluid specific heat, μ is viscosity, and k is thermal conductivity.

The cooling load transfers between chilled water and process air in equation (5.12) can be written as:

$$Q = \frac{(T_a - T_{chw})}{\frac{1}{h_a A_a} + \frac{1}{h_{chw} A_{chw}}} \quad (5.16)$$

where h_a, h_{chw} are heat transfer coefficients of process air convection and chilled water convection which calculated by equation (5.15); A_a, A_{chw} are transfer areas of process air and chilled water, respectively.

5.4.3. Dynamic model of CO₂ concentration

This control approach aims to provide the correct level of fresh air to the indoor air via the HVAC system. Carbon dioxide (CO₂) level is often utilized as an indicator of pollution in the building. A typical room will have a CO₂ setpoint about of 700-1000 ppm, higher than the CO₂ level of outdoor air for about 600ppm. By adjusting the fresh air ventilation rate while retaining the acceptable air quality, model-based for fresh air control is a possible strategy for greater energy saving compared to the constant rate ventilation system that keeps a maximum air rate for maximum occupancy in buildings.

The ERV delivers the fresh air from outdoor to the occupied zone to dilute the contaminated air and decrease the CO₂ concentration based on the real-time

CO₂ level of the building. The total occupants of a small office are 32 people. The working hours is from 8:00 to 18:00, thus the occupancy is almost full at that time, except the break time from 12:00 to 13:00 as shown in Figure 5.7 [94]. According to initial value and model of CO₂ concentration of room, the ERV delivers proper fresh airflow to the occupied zone to maintain the CO₂ concentration in a limit range. The energy consumption is reduced because of the decrease in cooling energy consumption and fan energy consumption.

Figure 5.7 Occupancy scheme in small office building one typical working day [94]

An accurate dynamic model is crucial for the development and optimization of model predictive control. The model has to present the external disturbances, such as variation of temperature, humidity, and air mass flow rate. The reason behind is that outdoor weather condition is usually harsh in hot and humid regions, which would have serious impacts on system performance.

The models are built on the bellow assumptions:

- The fresh air flows to the indoor zone equals the exhaust air flows to outdoor;
- The zone air is well mixed, which means that the concentration of CO₂ and temperature inside one zone are constant.

Concentration level of CO₂ is determined by an equation that balances all the flows that characterize the air inside the room [95]

$$V \frac{dC_{CO_2}}{dt} = C_{in} - C_{out} + C_{occ} \quad (5.17)$$

where $C_{in} = \dot{m}_{air} C_{CO_2,i}$, $C_{out} = \dot{m}_{air} C_{CO_2}$, $C_{occ} = g_{CO_2} N_{people}$

We discretize the balance equations through the backward Euler Method. The difference equation:

$$C_{CO_2}(t_{k+1}) = C_{CO_2}(t_k) + \frac{\dot{m}_{air} \Delta t}{V} (C_{CO_2}(t) - C_{CO_2,i}) + \frac{g_{CO_2} \Delta t}{V} N_{people}(t_k) \quad (5.18)$$

where $\Delta t = t_{k+1} - t_k$ is defined as the considered timesteps.

The identified dynamic model is formulated by MATLAB\Simulink, which is a commercial software for modeling, simulating and analyzing dynamic system [96], in order to implement the dynamic model. Simulink provides customizable blocks and MPC toolbox, which is utilized for controlling system analysis conveniently [97]. Then, the dynamic model of CO₂ concentration module will be built and developed for the implementation process.

5.5. Remarks

This chapter presents three modeling approaches in HVAC system which are essential to identify the model for implementation of advanced controller and optimization method. Based on the availability of measurement data sets and detailed understanding of the physical principle, the appropriate approach is employed for model development. The whole building thermal dynamic model is developed based on the thermal mass of building material, building structure, and indoor and outdoor air conditions. The 3R2C network approach is adopted to construct the thermal model of transient heat flow through the building envelope. The model accuracy is satisfied to predict cost-effectiveness and maintain the indoor air condition at the thermal comfort level. The model of

HVAC subsystems such as ERV unit, and AHU with cooling coil, are identified by the physical based modeling approach. Then the thermal dynamic model of buildings and HVAC subsystem models are combined to obtain a full HVAC model which is necessary for the implementation of energy efficiency strategies, advanced controller development, and operation optimization purposes.

In the next chapter, the effects of the advanced controller such as model predictive control (MPC) on total energy consumption and thermal comfort are investigated for the whole system. The air quality based control and temperature-based control are demonstrated and validated to achieve the objectives of saving energy consumption in buildings and improving indoor air quality by energy recovery technology.

Chapter 6

Model Predictive Control (MPC) optimization for energy recovery system

The full building model, which comprises of building thermal dynamic model and HVAC subsystem models, is obtained and successfully validated in Chapter 5. In this Chapter, the model predictive control strategy is presented as an advanced controller which has the capability of handling measured or unmeasured disturbances and operation constraints, and achieving multiple objectives of control concurrently. The optimized MPC strategy is implemented in the ERV integrated with HVAC system to maintain the indoor CO₂ concentration, and zone temperature in a thermal comfort range, as well as to minimize energy consumption. The air quality based control strategy and temperature based control strategy are proposed and thoroughly investigated in this chapter.

6.1. MPC and the application in buildings

MPC is proven as a successful approach, which is widely used by numerous industrial applications especially in process control technology [98]. The controller utilizes a system model to predict the future behavior of the control plants over a finite horizon and generates the control vector for the system to minimize the cost function. MPC has the capabilities of handling multivariable with constraints and disturbances, coupling with a dynamic system, and using a cost function to achieve multiple objectives. In the standard scheme of implicit MPC, an optimization problem is formulated at each time step, and solved online over a determined future window. The outputs of this problem are optimal inputs and theoretical behaviors of the studied plant with respect to an identified model. Basically, an open-loop control is set at each sampling time and only the first control input applied to the system. The entire process which based on the new control input and shifting the considered window is repeated. This receding horizon approach introduces feedback into the system.

The typical optimization problem for an MPC approach could be formulated as follows:

$$\underset{u_0, u_1, \dots, u_{N-1}}{\text{minimize}} \quad J = p(x_N) + \sum_{i=0}^{N-1} q(x(k), u(k), r(k)) \quad (6.1) \text{ Cost function}$$

$$\text{Subject to} \quad x_0 = x(0) \quad (6.2) \text{ Current state}$$

$$x_{k+1} = f(x(k), u(k), d(k)) \quad (6.3) \text{ Dynamic model}$$

$$(x(k), u(k)) \in X_k \times U_k \quad (6.4) \text{ Constraints}$$

where u is the controlled inputs, x is the system states, d is the internal and external disturbances, f is the dynamic equation which describe the plant. The system state x and controlled input u are subject to the operation constraints X_k and U_k . A flowchart of MPC calculation is illustrated in Figure 6.1.

Figure 6.1 Flowchart of MPC calculation

In the implementation of this control strategy to HVAC system, ERV will be integrated with fresh air demand ventilation and the whole HVAC system under tropical climate conditions as illustrated in Figure 6.2. The disturbances are also modeled, including occupancy activities and weather condition. The prediction will be utilized in the computation of control vector. The most importance controlled signals in the HVAC system include the zone temperature, humidity, and the mass flow rate of fresh air through the ERV. Apart from that, the constraints of MPC should be defined clearly. For example, the temperature

of the zone will be maintained within a certain thermal comfort band; the supply airflow rate is also operated in the given range.

The cost function describing the performance target of the controller is an essential part of the MPC design. In this simulation, the cost function takes the form of minimizing energy consumption by minimizing the total fresh air flow volume when the indoor air quality is kept in a suitable range. The trade-off must be defined between them by determining the weight numbers of the objectives in the cost function form.

Figure 6.2 Model predictive control for HVAC system in building

In our HVAC system, the ventilation system can be divided into two fundamental parts: the first part regulates the amount of fresh air coming into the room, and the second part determines the temperature of the cool air going into the room. The amount of fresh air, coming from the ventilation units, is regulated by a damper.

6.2. The implementation of MPC to HVAC systems by co-simulation

EnergyPlus incorporates the detailed model of a building, HVAC components, weather forecast, and operation schedules in the description text-based file, which is associated with the calculation for building energy consumption, based

on the energy balance equation and fundamental principles of heat and mass transfer. EnergyPlus is widely validated as an efficient program in the prediction of the thermodynamic behavior of various building types. However, the development of EnergyPlus is not aimed at controller design, then the limited controller such as on-off, schedule, and rule-based control is integrated into the program. Moreover, MATLAB is a multi purpose environment which specializes in modeling, designing, and implementing from the basic to the advanced controller to the building [99, 100]. Model predictive control is one of the comparative controllers, which can deal with constraints, predictions, and disturbances to achieve multi objectives in terms of cost function which is very complicated or impossible to perform if presented in rule-based control. However, the MPC strategy requires to develop an appropriate model which can capture the behavior of thermodynamic of buildings, the HVAC components, and possible disturbances. The development of the building model in MATLAB environment is a very challenging and time-consuming task. Therefore, incorporation of the thermodynamic model which developed in EnergyPlus to MATLAB attracts considerable attention.

Much effort in co-simulation is made to design the advanced controller in MATLAB environment and incorporate it into the EnergyPlus for energy efficiency investigation. MLE+ [86], Building Controls Virtual Test Bed (BCVTB) [101], and OpenBuild [102] are popular toolboxes used for the purpose of developing a co-simulation platform between MATLAB and EnergyPlus. Gorecki *et al.* [102] developed OpenBuild toolbox which was based on MLE+ to provide an integrated platform where the key functions are building modeler, simulation analysis, and communication between the applications. The building model in EnergyPlus is reconstructed, linearized, and presented as a state-space thermal dynamic model, which is ready for implementation of the controller. Moreover, the disturbances such as occupancy, weather data, and internal heat gain are also incorporated into the state-space model. The HVAC component models are developed in line with controller design in MATLAB environment.

The toolbox also supports data exchange mechanism between the applications such as thermal control signal from MATLAB and energy consumption calculation from EnergyPlus, which creates a closed-loop operation for co-simulation.

6.2.1. Building thermodynamic model

A reliable building model is the foremost importance in co-simulation to attain an accurate control result. The building model is specifically described in description file of EnergyPlus, which includes the building materials, the detailed geometry, and the facing direction of the building. In the simulation, a standard small office model which is recommended by ASHRAE [103] is demonstrated in Figure 6.3. The International Weather for Energy Calculations (IWEC) database of Singapore which stores the hourly values of meteorological data such as temperature, humidity ratio, solar radiation is utilized as the input weather data file for EnergyPlus [104]. The mean daily maximum and minimum temperature of each month in the TMY is shown in Figure 6.4. The highest temperature of the hottest month (April) and the coolest month (December) has a small difference of 2°C. Therefore, it is acceptable to use the weather of a single day or single week to represent the typical weather in Singapore. The weather input is possible to obtain from the on-site measurement to calibrate the IWEC data. The schedules for occupancy, lighting, and equipment in office are incorporated in the description file of EnergyPlus.

Figure 6.3 Three zones building model in Sketchup

Figure 6.4 The daily maximum and minimum temperature and relative humidity of Singapore in 12 months [19]

The RC network model which is described in subsection 5.3.2 is implemented to obtain the thermodynamic model of the above building. The temperature of each zone and surface is equivalent to the voltage of a node in the electrical network. The heat flux of convection and conduction are equivalent to the current between the nodes. The transfer coefficients of conduction and convection are the resistance, and the thermal capacity is presented as the capacitance in RC networks. The computation of parameters such as resistance and capacitance is carried out by different algorithm in EnergyPlus. The energy balance equation is applied to each node in RC networks to attain differential equations which represents for each node. The system of equations is linearized and discretized to obtain a state-space thermodynamic model which is suited with the advanced controller in MATLAB especially the Model predictive control.

6.2.2. HVAC model

HVAC is a complex system which comprises of multi components such as cooling coil, heating coil, cooling tower, energy recovery device, air handling unit, fan, and pump, etc. In EnergyPlus, each component of HVAC system is described in detail, which is not suitable for control purpose. Moreover, the co-simulation platform is not possible to handle the operation of low-level actuators, which is probably the optimal output of controller in MATLAB. Therefore, the HVAC system is developed in MATLAB environment with the heat flux to the zones or surfaces is treated as the input of controllers.

6.2.3. Controllers

The HVAC model coupled with the thermodynamic model of the building establish a full dynamic system which is ready to implement an optimal controller such as Model predictive control (MPC). Based on the obtained model, the MPC calculates optimal output trajectory to achieve the multi objectives function while respecting the constraints. Temperature, humidity ratio, CO₂ concentration, and the minimum and maximum limitation of actuators are usually the important

constraints of the controller. The objective function is usually minimum energy consumption or minimum operating cost of the system.

There are two controllers which are investigated: CO₂-MPC aims to keep the CO₂ concentration within comfort bounds; and T-MPC aims to maintain the temperature of the zones within the comfort level while minimizing energy consumption.

6.3. Fresh air based control strategy

6.3.1. MPC design

In this control strategy, ERV will be integrated with demand fresh air ventilation and the whole HVAC system under tropical climate conditions. For the objective function, we choose integral cost function, linear dynamic and linear constraints model for optimization problem. CO₂ concentration is the criteria to evaluate the indoor air quality. CO₂-MPC will aim to keep the concentration within comfort bounds while minimizing energy consumption. Figure 6.5 shows the flowchart to implement MPC to control the indoor CO₂ concentration.

Figure 6.5 The flowchart to implement the CO₂-MPC into building

Because the obtained model is nonlinear, the linearization needs to be carried out by approximating with a state-space model:

$$x_{CO_2}(k+1) = ax_{CO_2}(k) + bu_{CO_2}(k) + ew_{CO_2}(k)$$

$$y_{CO_2}(k) = x_{CO_2}(k)$$

where

$$x_{CO_2}(k) = C_{CO_2}(k) - C_{CO_{2,i}} = \Delta CO_2;$$

$y(t) = C_{CO_2}(t) - C_{CO_{2,i}} = \Delta CO_2$ is the difference of CO₂ between the concentration inside and outside the room;

$u(t) = \dot{m}_{air}(C_{CO_2}(t) - C_{CO_{2,i}})$ is the input (where $C_{CO_{2,i}}$ is the CO₂ concentration of the outdoor air, which is assumed as a constant value of

400ppm). These inputs will regulate the valve opening percentage of both inlet and outlet duct;

$w_{CO_2}(k) = N_{people}(k)$ is disturbance.

Formulation of the cost function and constraints for CO2-MPC:

$$\min_{u_{CO_2}(t), \dots, u_{CO_2}(t+N-1)} \sum_{k=t}^{t+N-1} u_{CO_2}(k)$$

$$\begin{aligned} u_{CO_2}^{\min}(k) &\leq u_{CO_2}(k) \leq u_{CO_2}^{\max}(k) \\ 0 &\leq y_{CO_2}(k) \leq y_{CO_2}^{\max}(k) \end{aligned}$$

where

$$\begin{aligned} u_{CO_2}^{\min}(k) &= \dot{m}_{air}^{\min} \cdot x_{CO_2}(k) \\ u_{CO_2}^{\max}(k) &= \dot{m}_{air}^{\max} \cdot x_{CO_2}(k) \\ y_{CO_2}^{\max}(k) &= C_{CO_2}^{\max} - C_{CO_2,i} \end{aligned}$$

with N is the prediction horizon and $C_{CO_2}^{\max}$ is the desired upper bound of the CO₂ concentration (850ppm)

At each sampling time, the constraints are defined as:

$$\begin{aligned} G_{u,CO_2} u(k) + G_{x,CO_2} x(k) &\leq H_{u,CO_2} \\ G_{y,CO_2} y_{CO_2}(k) &\leq H_{y,CO_2} \end{aligned}$$

where

$$\begin{aligned} G_{u,CO_2} &= \begin{bmatrix} -1 \\ 1 \end{bmatrix} \quad G_{x,CO_2} = \begin{bmatrix} -1 & 1 \\ 1 & -1 \end{bmatrix} \quad G_{y,CO_2} = \begin{bmatrix} -1 \\ 1 \end{bmatrix} \\ H_{u,CO_2} &= \begin{bmatrix} -u_{\min} \\ u_{\max} \end{bmatrix} \quad H_{y,CO_2} = \begin{bmatrix} -y_{\min} \\ y_{\max} \end{bmatrix} \end{aligned}$$

Consequently, the air mass flow rate $\dot{m}_{air}$ is calculated as follows

$$\dot{m}_{air}(k) = \frac{u_{CO_2}(k)}{C_{CO_2}(k) - C_{CO_2,i}}$$

6.3.2. Results and discussions

In this section, the results of some preliminary control strategies are presented, followed by the discussion and explanation of the results.

The above CO₂ concentration model is implemented in MATLAB with the detailed constraints for input and output of the building as follows:

$$0 \text{ (kg/s)} \leq \dot{m}_{air} \leq 0.5 \text{ (kg/s)}$$

$$400 \text{ ppm} \leq C_{CO_2} \leq 850 \text{ ppm}$$

Many tests are run in different setpoints and occupancy forecasts to test the control potential of MPC in consideration about constraints and energy consumption.

- Test 1: Compare MPC and PID
 - Design the PID control for the model
 - CO₂ setpoint reference is 600ppm
 - 8 hours period (from 8h00 to 20h00), normal occupancy

The results are shown in Figure 6.6
- Test 2: MPC with different occupancy schemes
 - CO₂ setpoint reference is 700ppm
 - 8 hours period (from 8h00 to 20h00), high and low occupancy

The results are shown in Figure 6.7
- Test 3: Investigate the effects of different reference setpoints of MPC
 - CO₂ setpoint reference is changed to 650ppm, 750ppm, and 850ppm
 - 8 hours period (from 8h00 to 20h00), normal occupancy

The results are shown in Figure 6.8

Figure 6.6 Simulation on controller with different setpoints

(a) MPC – 600ppm (b) PID – 600ppm (c) MPC - 800ppm (d)PID-800ppm

Figure 6.7 MPC controller with (a) low occupancy (b) high occupancy

Figure 6.8 MPC control with upper bounds for CO₂ concentration (a) 650ppm, (b) 750ppm, (c) 850ppm

In all cases, the CO₂-MPC controller regulated the output with respect to the indoor air quality requirement. The output of CO₂ concentration always stays under the upper constraints. The simulation results illustrate the significant potential of MPC in dealing with nonlinearities, handling constraints, and performing offset free tracking for multiple control objectives.

6.4. Temperature-based control strategy

In this control strategy, the mean temperature of each zone is performed as the control output, which is maintained in a suitable comfort range by the control algorithm. Two main control methods including PI control and MPC are implemented in three zones building model coupled with HVAC system which is presented as a typical office building. The effects of disturbance signal such as

weather, occupancy, and internal equipment on control responses as well as results are thoroughly investigated in this section.

6.4.1. Basic model without HVAC control

In based-case model, the three-zone building model is simulated in EnergyPlus without the HVAC system to analyze the thermal dynamic behavior of zones and building under Singapore weather condition. The temperatures of 3 zones are shown in Figure 6.9 in a simulation with a period of 7 days. Generally, the zone temperatures are very hot without the HVAC system, from about 36 to 45 °C. The heat gain sources mainly come from the power solar radiation, occupants, lighting system, and internal equipment heat gain. The zone temperatures on daytime are higher than night time up to 8°C, because of the hot weather of tropical climate at noon. Furthermore, on Saturday and Sunday, the operation of internal equipment, lighting, and the number of occupant is substantially decreased as shown in the schedule in Figure 6.10. Therefore, the zone temperatures are only warmed up to 38 °C, the difference of temperature at daytime and nighttime is less significant.

The thermal building model is performed in good response to the heat gain by weather, occupants, lighting, and interior equipment. In next section, the PI and MPC controller are implemented to HVAC system to remove the heat and maintain the zone temperatures in suitable setpoints in comfort range.

Figure 6.9 Temperature of 3 zones in a building without HVAC system

Figure 6.10 Internal heat gain schemes in 7 days of lighting, occupancy, and internal equipment of building

6.4.2. Temperature-based PI control strategy

The PI controller is implemented in the cooling system of the office building, which consists of three thermal zones under Singapore climate. The simulation period of one typical week is carried out to investigate the effect of PI controller on the temperatures of the three zones, as illustrated in Figures 6.11 and 6.12. Each timestep of simulation is 15 minutes, and there are 672 timesteps in total for seven days. The schedules of internal heat gain due to the lighting, occupancy, and internal equipment are employed, as shown in Figure 6.9.

Under the tropical climate, the temperature of outdoor is usually higher than that of indoor environment for the whole year. In particular, the outdoor temperature at daytime can reach up to 33°C, where the air-conditioning system is essential to condition the hot air before the air flows into the building. Figure 6.11 illustrates the zone temperature outputs of the three zones under the PI controller in HVAC system. In the case of the temperature setpoint as 24°C, a slight fluctuation in temperature zones is observed in Figure 6.11, especially in Zones 1 and 3 in daytime. It could be explained by the fact that the areas of Zones 1 and 3 are much bigger than Zone 2, thus they absorb more solar heat gain and internal heat from occupants and equipment. Consequently, the output control signals of those zones fluctuate further by the change of disturbances, such as outdoor weather.

Figure 6.12 illustrates the effect of PI controller to regulate the temperature of three thermal zones with higher reference setpoint of 26°C. In general, the responses of controller are handled very well around the reference setpoint. The thermal comfort of the conditioned zones is satisfied.

Figure 6.11 Temperature of outdoor air and three zones in building with PI control in 7 days, reference setpoint = 24 °C

Figure 6.12 Temperature of outdoor air and three zones in building with PI control 7 days, reference setpoint = 26 °C

Figure 6.13 shows the power inputs of the cooling system to maintain the zones' temperature at a setpoint of 24°C. The power inputs are regulated by PI controller for a typical week from 1st July to 7th July under Singapore weather. In the first period, the power inputs increased significantly to 40kW to initiate the control state. This is considered as overshoot state which is common in PI controller. The overshoot value raises the issues of overloaded utilization of the cooling system, which caused severe impacts on the operation of equipment. In the following period, the cooling power inputs are sustainably regulated based on the heat gain of solar radiation and internal equipment operation. For example, the power inputs of Zones 1 and 3 vary about 4-15kW for nighttime and daytime highest load. Power inputs of Zone 2 vary about 2-5kW because of a smaller area of Zone 2 compared to other zones. On Sunday, the power inputs of three zones decrease considerably. It could be explained by the fact that the operation schedules of lighting, occupancy, and internal equipment on weekend are much less than those on weekdays.

Figure 6.14 presents the power inputs of the cooling system to maintain the zones' temperature at a higher setpoint of 26°C. Similarly, the power inputs are regulated by PI controller under Singapore weather. To maintain the zone temperature, the input signals of Zones 1 and 3 also overshoot to 35kW at the initiation stage, afterward, the inputs varied stably about 3-13kW for nighttime and daytime peak. The power inputs of Zone 2 varied about 2-5kW due to a smaller area of Zone 2. The power inputs, in this case, are parallel and slightly less than the power inputs in case of reference setpoint of 24 °C. Generally, the PI controller regulates the output zone temperatures very well which follow closely the comfort temperature setpoints.

Figure 6.13 Power consumption in three zones in building with PI controller for 7 days, reference setpoint = 24 °C

Figure 6.14 Power consumption in three zones in building with PI controller for 7 days, reference setpoint = 26 °C

The effects of PI controller on the zone temperatures and corresponding energy power inputs are thoughtfully investigated in the simulation period of one typical working day under Singapore weather condition. The temperature outputs and power inputs are illustrated in Figures 6.15 and 6.16. The reference temperature setpoint of controller is 24 °C which is standard indoor temperature for office in Singapore. The outdoor temperature at noontime can reach up to 30°C. Figure 6.15 shows that, three output responses are closely satisfied the setpoint at 24°C, even though there is a minor fluctuation in temperatures of Zones 1 and 3 at daytime. The power inputs of Zones 1 and 3 increase from 4kW to 16kW when working day starts from 8:00 to end of day at 18:00. An internal heat gain scheme for a typical day is shown in Figure 6.17. Similarly, the power input requirement of Zone 2 is considerably smaller because of the small area. Generally, the PI controller handles the output responses of the system very well.

Figure 6.15 Temperature of outdoor air and three zones in the building with PI controller in 24 hours, reference setpoint = 24 °C

Figure 6.16 Power consumption in three zones in the building with PI controller in 24 hours, reference setpoint = 24 °C

Figure 6.17 Internal heat gain schemes in 24 hours of lighting, occupancy, and internal equipment of building.

6.4.3. Temperature-based MPC control strategy

The MPC controller is implemented to the cooling system of the office building which consists of three thermal zones under Singapore climate. The simulation period of a typical week is carried out to investigate the effect of the MPC controller to three zone temperatures as illustrated in Figures 6.19 and 6.20, for the temperature constraints of 22-24°C and 22-26°C in daytime 8:00 to 18:00, respectively. The constraints are loosened to 17-29°C in unoccupied time from 18:00 to 8:00 of the following day. Each timestep for simulation is 15 minutes, and there are 672 timesteps in total for seven days. The internal heat gain schemes of lighting, occupancy, and internal equipment shown in Figure 6.18 are employed as the disturbances of the controller.

Figure 6.18 Internal heat gain schemes in 7 days of lighting, occupancy, and internal equipment of the building

Due to the hot and humid climate in tropics, the outdoor temperature is usually higher than the indoor temperature for the whole year. The outdoor temperature

at daytime can reach up to 33°C, where the air-conditioning system is essential to remove the thermal energy from the hot air before providing it into the building.

Figure 6.19 illustrates the zone temperature outputs of three zones under implementation of MPC controller in the HVAC system. The output responses are maintained strictly in the constraint bounds with maximum temperature of 24°C. Since the MPC objective function is designed to minimize the energy usage, the zones' temperature is regulated to track the upper constraint of the controller. Consequently, the minimum cooling energy is required to balance thermal heat gains of the zones.

It can be observed in Figure 6.19 that the zone temperature fluctuates considerably in the beginning stage of control, as a result of lack of initiation process for the controller. In the next stage, the control signals of three zones response steadily to the disturbances by satisfying the constraint strictly. Through those behaviors, the thermal comfort in controlled zones is obtained in the daytime of 8:00 to 18:00. At unoccupied time, even though the outdoor temperature is lower than the upper constraint of 29°C, the heat gain of lighting and internal equipment still warms up the building zones especially in Zone 1. However, the zone temperatures are still in constraint bounds most of the time. In general, the MPC controller regulates the power inputs to achieve excellent quality outputs of three zone temperatures. It could be explained that the MPC employs the model of building and predictive algorithm to deal with various disturbances which are usually the big challenges of control.

The temperature upper constraints are extended to 26°C to examine the effects of constraints to response signals. The correspond temperatures of three zones and outdoor temperature are illustrated in Figure 6.20. It can be observed that, there are less fluctuation at initiation stage compared to the case of 24°C upper constraint. Moreover, the temperature of Zone 1 and 3 are regulated faster to the constraint at the initiation stage. Afterward, the temperatures of all zones are well-tuned to satisfy the thermal comfort in the daytime and maintain within the extended constraints during unoccupied time.

Figure 6.19 Temperature of outdoor air and three zones in building with MPC in 7 days, temperature constraint is 22-24°C, the upper constraint of cooling power is 25kW

Figure 6.20 Temperature of outdoor air and three zones in building with MPC in 7 days, temperature constraint is 22-26°C, the upper constraint of cooling power is 25kW

The power consumption in three zones in the building with MPC controller in 1 typical week is illustrated in Figures 6.21-22 for the constraints of 22-24°C and 22-26°C, respectively. The constraint for cooling power input of each zone is set at maximum of 25kW. In the initiation stage, the cooling power inputs of Zone 1 and 3 increase immediately to the maximum value of 25kW to quickly remove the thermal storage inside the zones. In the case of temperature constraint 22-24°C, this cooling load is not powerful enough to maintain the control responses less than 24°C in the initiation stage as observed in Figure 6.19. In the case of temperature constraint 22-26°C, the zone temperatures are less overshoot than in the previous case as observed in Figure 6.20. In both cases, after the initiation stage, three cooling power inputs are regulated within the constraint to generate well-response temperatures for three zones in building. At weekend, because there are no occupants in the building, the cooling load mainly come from the interior equipment, lighting, and solar radiation which reach the maximum load around 5kW at noon.

The effects of MPC controller on the zone temperatures and corresponding energy power inputs are thoughtfully investigated in the simulation period of one typical working day under Singapore weather condition. The temperature outputs and power inputs are illustrated in Figure 6.23 and Figure 6.24. The upper constraint of temperature is 26 °C and the outdoor temperature at noon time can reach 30°C. Figure 6.23 shows that, three output responses are more stable than PI controller, and closely satisfied the upper constraint of 26°C in occupied hours and 31°C in unoccupied hours. An internal heat gain scheme for a typical day is shown in Figure 6.17. Similarly, the power input requirement of Zone 2 is considerably smaller than Zones 1 and 3 because of the smaller space. Generally, the MPC controller handles the output responses of the system better and more stable than the PI controller.

Figure 6.21 Power consumption in three zones in the building with MPC in 7 days, temperature constraint is 22-24°C, the upper constraint of cooling power is 25kW

Figure 6.22 Power consumption in three zones in the building with MPC in 7 days, temperature constraint is 22-26°C, the upper constraint of cooling power is 25kW

Figure 6.23 Temperature of outdoor air and three zones in the building with MPC in 24 hours, temperature constraint 22-26°C

Figure 6.24 Power consumption in three zones in the building with MPC in 24 hours, temperature constraint 22-26°C

6.5. Remarks

This chapter presents the overview the model predictive control (MPC) strategy and the implementation of MPC in HVAC system to optimize energy usage and maintain the thermal comfort for occupants. MPC is introduced as an advanced controller which can handle various disturbances and operation constraints, and helps to achieve multiple objectives of control. In the building, the main disturbances which affect to the quality of control system considerably are weather condition and occupancy. The optimized MPC strategy is implemented in HVAC system coupled with ERV to maintain indoor CO₂ concentration, and zone temperature in a thermal comfort range, as well as to minimize energy consumption. The air quality based control strategy is carried out based on the indoor CO₂ concentration criterion. Temperature-based control strategy for PI controller and MPC controller are proposed and intensively investigated. The results show that the zone temperature is regulated better in MPC controller, and the power inputs are managed within the capability of HVAC system by constraint bounds. The energy consumption of HVAC with MPC controller is considerably less than that of HVAC with PI controller.

Chapter 7

Conclusions and Recommendations

The present project is conducted to improve overall energy efficiency in HVAC energy recovery system by CFD analysis, experimental validation, control model development, and optimal control scheme on important parameters of the membrane-based ERV. This chapter summarizes the noteworthy conclusions from the present work. Thereafter, the recommendations in model development, experimental setup, significant control variables, objective function design, and optimization method are taken into consideration for future work.

7.1. Conclusions from the present work

Due to the increase in energy consumption and the environmental impacts caused by the building sector, the present project is conducted to improve overall energy efficiency in HVAC system by incorporating the energy recovery system. Throughout the present project, several conclusions are detailed below:

- In order to investigate the performance of ERV sensible and latent energy subject to tropical climate conditions, a novel model for modeling of ERV as a component of HVAC is developed both mathematically and experimentally. The CFD technique and finite element method are adopted to develop a mathematical model of a semi-permeable membrane ERV. Besides, mesh generation and mesh independence tests are carried out to determine a reasonable mesh for both accuracy and computational time. The previous studies are also employed to validate the results from CFD model successfully. Finally, the three-dimensional ERV model is comprehensively investigated for the analysis of critical parameters, including velocity, temperature, humidity of supply and exhaust air flows, for the improvement of energy efficiency.
- Subsequently, to examine the predictive capability of the mathematical model, an experimental model of the energy recovery system is set up in a real building. The stacked membrane-based core with thin channels is explored as the key component in the ERV system, where a specialized heat and humidity control chamber is utilized to reproduce the outdoor air condition of a tropical climate. Various types of transmitters coupled with Data Acquisition system are equipped into the experiment to measure the important parameters accurately and to manage the data for further analysis. Investigations for the energy saving performance of the membrane-based ERV system are conducted subject to different environmental conditions, including temperature, humidity, and air velocity. The previous numerical results are validated by direct comparison with the experimental data, in which good agreements are achieved.

Through the numerical and experimental results, it is demonstrated that the developed membrane-based recovery system is capable of substantial energy saving in buildings, where the sensible and latent effectiveness could be achieved up to 80% and 70%, respectively.

- To achieve the objectives of saving energy consumption in building and improving indoor air quality by energy recovery technology, an optimized control strategy for ERV in HVAC system is a promising approach. Modeling is a prior necessary stage to implement the model-based control strategy for optimization of energy consumption in building. Firstly, the whole building thermal dynamic model is developed based on the thermal mass of building material, building structure, and the indoor and outdoor air conditions. The 3R2C network approach is adopted to construct the thermal model of transient heat flow through the building envelope. Beside the building thermal dynamic model, HVAC is a complex system comprising of components such as air handling unit, chiller, cooling coil, and ERV unit, which are identified by the physical-based modeling approach. The controlled models of HVAC components are developed in MATLAB/Simulink environment, for optimization purpose in the overall operation of the HVAC system.
- The thermal dynamic model of buildings and HVAC subsystem models are combined to obtain a full HVAC model which is necessary for the implementation of energy efficiency strategies, advanced controller development, and operation optimization purposes. The model predictive control strategy is presented as an advanced controller which has the capabilities of handling various disturbances and operation constraints, while achieving multiple objectives of control. The MPC strategies is developed in MATLAB, which is a specialized environment for modeling, designing, and implementing the basic to advanced controller to the building. However, the development of the building model in MATLAB environment is a very challenging and time-consuming task. In addition,

EnergyPlus is broadly validated as an efficient program in the prediction of the thermodynamic behavior of various building types, and in the estimation of total energy consumption. Therefore, a co-simulation platform is adopted to incorporate the behavior of thermodynamic of buildings, the HVAC components, and possible disturbances. Singapore weather data and typical occupancy scheme as main disturbances are incorporated into the MPC controller. The air quality based control strategy and temperature based control strategy are proposed and thoroughly investigated. The results of comparing this controller with other controllers proves that, the proposed MPC strategy well maintains the indoor CO₂ concentration, and zone temperature within a thermal comfort range, as well as optimizes power usage in building.

7.2. Recommendations for future work

There are several recommendations in model development, experimental setup, objective function design, and optimization method which are considered for future work as follows:

- The three-dimensional CFD model of ERV is developed in CFD simulation software ANSYS\FLUENT to investigate the performance of energy recovery device, which is examined and fabricated for experimental model. Therefore, the dimensions of 3D CFD model such as length, width, thickness are the same as the dimensions of a real model, which may limit the examination in different geometry of ERV. In CFD environment, the effects of ERV geometry can be thoroughly evaluated by varying the dimension of length, width, and thickness of air channel, as well as the airflow arrangement, which can lead to an improvement in both heat and mass exchange efficiency for ERV.
- In experimental work, with a limited space, the whole experiment of energy recovery system is carried out in a tight environment, where both fresh and

return airflows are taken from a same space. The advantage of this setup is the stable environment conditions, such as the temperature and relative humidity always vary around 23°C and 50%, respectively. However, if the experiment is conducted in a long time, the indoor temperature and relative humidity will increase continuously since the environment controlled chamber keeps adding the heat and moisture, for generating the desired setpoint of inlet air. Moreover, in a tight place, the supply and return airstreams are possible to mix with each other, which affects the performance evaluation of energy recovery system. Therefore, it is recommended to carry out the experiment in a large space with separate duct for outdoor, supply, return and exhaust airflows. The outdoor air should be taken from outdoor environment in different weather conditions in a long time to examine thoroughly the effects of air conditions on the performance of ERV system.

- The dynamic models of cooling coil, ERV unit, and thermal dynamic building which are developed in this work are essential to establish an optimization scheme by an advanced controller. Besides, the dynamic models of other HVAC equipment such as active chilled beam cooling and liquid desiccant dehumidification system can be developed and straightforwardly incorporated into HVAC system in MATLAB environment. The new HVAC system model can be employed to investigate the effects of new components to overall energy consumption or indoor air quality of the building.
- The Model predictive control is a promising approach, which is based on the building dynamic model to predict the behavior of the plant over receding horizon and generate the control inputs such as the temperature setpoint, mass flowrate, valve opening for the actuators of HVAC system. In this work, MPC well handles the actuator constraints and the disturbances such as weather, occupancy, interior equipment and then minimizes a cost function to achieve the objectives of saving energy

consumption and maintaining the indoor air quality. There are several methods which are favorable for being employed in MPC for energy usage optimization such as incorporation of the electricity tariff schemes, and load shifting at on-peak and off-peak time by a chilled water storage. The weather condition to apply the outcomes of this research is also possible to extend to other climate zones besides the tropics.

Publications arising from the Thesis

Journal

1. **Nam Khoa Huynh**, Hua Li, Yeng Chai Soh, Wenjian Cai, “*Performance characterization of membrane-based energy recovery system*”, *Procedia Engineering* 2017 (Accepted)
2. **Nam Khoa Huynh**, Hua Li, Yeng Chai Soh, Wenjian Cai, “*Characterizing performance of an energy recovery system in the tropics via numerical and experimental methods*”, *Building and Environment* (Submitted)
3. **Nam Khoa Huynh**, Hua Li, Yeng Chai Soh, Wenjian Cai, “*Model-based control for energy recovery ventilator incorporated in HVAC system*”, *Energy and Buildings* (in preparation)

Conference

4. **Nam Khoa Huynh**, Hua Li, Yeng Chai Soh, Wenjian Cai, “*Modelling of membrane-based energy recovery system*”, *International Conference on Materials for Advanced Technologies* 2017
5. **Nam Khoa Huynh**, Hua Li, Yeng Chai Soh, Wenjian Cai, “*Numerical analysis and model-based control of energy recovery ventilator in HVAC system*”, *2016 World Congress on Sustainable Technologies (WCST)*, IEEE, 2016.

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