DTU Library

Active indoor air cleaning and heat recovery technology for energy saving of building ventilation

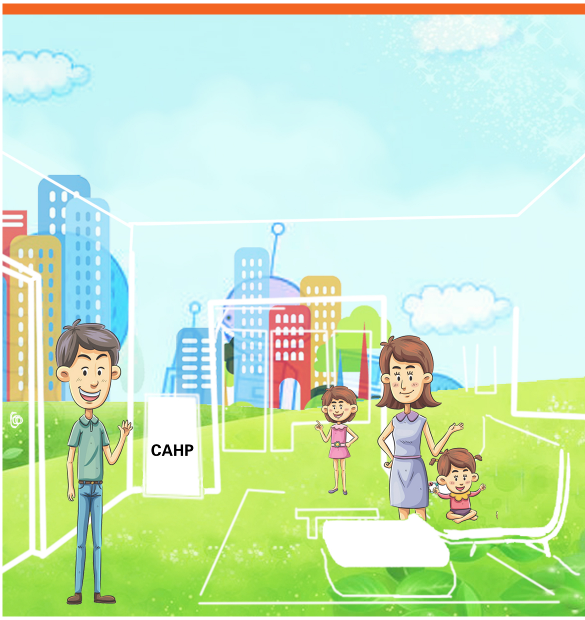
Nie, Jinzhe

Publication date:
2015

Document Version
Publisher's PDF, also known as Version of record

Active indoor air cleaning and heat recovery technology for energy saving of building ventilation

Jinzhe Nie



DTU Civil Engineering report R-333
June 2015

DTU Civil Engineering
Department of Civil Engineering

Active indoor air cleaning and heat recovery technology for energy saving of building ventilation

PhD Thesis by

Jinzhe Nie

International Centre for Indoor Environment and Energy

Department of Civil Engineering

Technical University of Denmark

May 2015

Preface

This PhD Thesis is the result of the study carried out at the International Centre for Indoor Environment and Energy, Department of Civil Engineering at the Technical University of Denmark from 2011 to 2014.

First of all I am very grateful to my supervisor, Lei Fang, for his guidance, patience, valuable suggestions, and friendly help to my life in Denmark. I would also like to thank him for giving me the opportunity to take part in the PhD project - it has been an enriching experience. I am also thankful to my co-supervisor Bjarne W Olesen who is the head of the International Centre for Indoor Environment and Energy for his help to my study and research in the centre. I would like to thank Professor Yufeng Zhang in Tianjin University who suggested me coming to DTU for PhD studying.

I appreciate Ole R. Hansen and Henning Grønbæk from Exhausto A/S who offered their knowledge and support on the development of the prototype CAHP. I also appreciate Reto M. Hummelshøj from COWI A/S for his advice on the development of the CAHP. I am also grateful to Dr. Ge Zhang from University of Science and Technology Beijing, for giving strong and useful suggestions on theoretical modelling of the silica gel rotor.

Special thanks to my family who have always encouraged me during my studies.

April 2015

Jinzhe Nie

Nomenclature

Symbols

C_i	VOC concentration in the air ($\mu g/m^3$)
G_i	VOC concentration on the adsorbent ($\mu g/g$)
$C_{w,i}$	VOC concentration in the air when it reaches equilibrium state with the adsorbent ($\mu g/m^3$)
d	Air speed in the flute section of the silica gel rotor (m/s)
f_d	Weight of the adsorbent per meter in one flute (kg/m)
A	Cross-sectional area of one flute section in the silica gel rotor (m^2)
t	Time (s)
x	Length (m)
α_m	Convective mass transfer coefficients (m/s)
P	Perimeter of the flute section (m)
H_i	Henry's law constant of the mass.
α	Heat transfer coefficient ($W/(m^2 \cdot K)$)
ρ_a	Density of the air (kg/m^3)
c	Specific heat of the air ($kJ/(kg \cdot K)$);
Re	Reynolds number.
T_k	Absolute temperature (K)
$G_{\max}$	Maximum content of VOC on adsorbent ($\mu g/g$)
Y	Moisture content in the air (g/kg)
W	Moisture adsorbed on the adsorbent (g/kg)
Y_W	Moisture content in the air which reached equilibrium state with adsorbent (g/kg)
K_Y	Mass transfer coefficient ($kg/m^2/s$)
f_m	Weight of substrate material in the direction along the airflow (kg/m)
q	Adsorption heat of moisture (J/kg)
h_v	Latent heat of vaporization of water (J/kg)
u	Uncertainty
u'	Relative uncertainty
Q	Heating or cooling load (kW)
V	Airflow rates (l/s)
T	Temperatures in Celsius degree (℃)
ρ	Air density(kg/m^3)
h	Enthalpy of refrigerant (kJ/kg)
η_{is}	Isentropic efficiency of the compressor.
E	Power or primary energy consumption (kW)
η	Efficiency
μ_{E-S}	Energy saving potential of the CAHP

G	Total energy consumption of CAHP or reference system in a whole season (kWh)
h_i	Number of hours (h)
dP_{si}	Pressure drop through the silica gel rotor (Pa)
φ_{ng}	Primary energy factor of the natural gas.

Subscripts

si	Silica gel rotor
reg	Regeneration air
pro	Process air
deh	Dehumidified air
sup	Air supplied to ventilated room
rec	Recirculation air
fre	Fresh air
ref	Reference system
$vent$	Ventilation
con	Condensing
eva	Evaporating
gb	Gas boiler
ng	Natural gas

Abbreviations

CAHP	Clean air heat pump
VOC	Volatile organic compound
PM	Fine particulate matter
HVAC	Heating ventilation and air-conditioning
COP	Coefficient of performance
UVGI	Ultraviolet germicidal irradiation
PCO	Photo catalytic oxidation
TVOC	Total volatile organic compound
SOA	Secondary organic aerosol
PTR-MS	Proton transfer reaction mass spectrometry
PD	Percentage dissatisfied
PHE	Plate heat exchanger
CAV	Constant air volume air-conditioning system
HFC	Hydro-fluoro-carbon
ODP	Ozone depletion potential
GWP	Global warming potential
DTU	Technical University of Denmark

Summary

An innovative clean air heat pump (CAHP) which integrated heating/cooling, dehumidification, ventilation, air cleaning and energy recovery in one unit was proposed. The CAHP was proposed based on the combination of desiccant rotor with heat pump, and was designed to maintain a comfortable, healthy indoor environment in normal office, commercial and residential buildings. The desiccant rotor was used for dehumidification and indoor air cleaning; the heat pump provided sensible heating/cooling and regeneration heat for the desiccant rotor.

A theoretical model of the CAHP was established with numerical equations. The theoretical model is used for predicting the volatile organic compound (VOC) removal and energy performance of the CAHP. The theoretical model was validated by experimental data. Validating results showed that the model could be used to predict the performance of CAHP. Numerical simulations were conducted to analyse and optimize the performance of the CAHP. Simulation results showed the CAHP could clean air borne contaminants effectively and could provide an energy efficient choice for ventilation.

Based on the theoretical analysis, a prototype unit of the CAHP was designed and developed. With the prototype unit, laboratory experimental studies were conducted to investigate its energy performance under different outdoor climates including cold, mild-cold, mild-hot and extremely hot and humid climates. The energy performance of the CAHP was then evaluated by comparing with conventional reference systems. The results showed that to keep same indoor air quality, the CAHP could save substantial amount of energy. For example, compared to conventional air source heat pump, the CAHP could save up to 55.93%, 36.83% and 32.33% of power for ventilation and air conditioning in a test room in summer of Copenhagen, Milan and Colombo. It can save 11.20%, 10.25% of power for ventilation and heating in the test room in winter of Copenhagen, Milan. If compared to a gas boiler system, the CAHP can save 46.86% and56.44% of primary energy use in Copenhagen and Milan respectively.

Overall, the CAHP can clean indoor air with a high VOCs removing efficiency and can hold heating/cooling load in an energy efficient way. The CAHP could be an energy efficient choice for ventilation systems to maintain a healthy, comfortable and productive indoor environment.

Key words: Air cleaning, Dehumidification, Silica gel rotor, Heat pump, Energy Performance

Resumé

En innovativ ren luft varmepumpe (RLV), som integrerer varme/køling, affugtning, ventilation, luftrensning og energiudnyttelse i én enhed blev foreslået. RLVen blev foreslået baseret på en kombination af en roterende entalpiveksler og en varmepumpe, og er designet til at opretholde et behageligt, sundt indeklima i kontorer og beboelsesejendomme. Entalpiveksleren blev brugt til affugtning og indendørs luftrensning. Varmepumpen leverede opvarmning/køling og regenerationsopvarmning til entalpiveksleren.

En teoretisk model af RLVen blev etableret med numeriske ligninger. Modellen bruges til at forudsige fjernelse af flygtige organiske forbindelser (VOC) og RLVens energimæssige ydeevne. Den teoretiske model blev valideret af eksperimentelle data. Resultaterne viste, at modellen kunne bruges til at forudsige RLVens ydeevne. Numeriske simuleringer blev udført for at analysere og optimere ydeevnen af RLVen. Simulationsresultaterne viste, at RLVen kunne rense luftbårne forureninger effektivt og kunne give en energieffektiv ventilation.

En prototype enhed af RLVen blev designet og udviklet, baseret på den teoretiske analyse. Med prototypen blev eksperimentelle laboratorieundersøgelser udført for at undersøge dens energimæssige ydeevne under forskellige udeklimaer, herunder koldt, middelkoldt, mildt, varmt og meget varmt og fugtigt klima. Den energimæssige ydeevne for RLVen blev derefter evalueret ved at sammenligne med konventionelle referencesystemer. Resultaterne viste, at for at holde samme indendørs luftkvalitet, kunne RLVen spare betydelige mængde energi. I forhold til en konventionel luft varmepumpe, kunne RLVen f.eks. spare op til 55,93%, 36,83% og 32,33% af elforbruget til ventilation og aircondition i et testrum om sommeren i hhv. København, Milano og Colombo. Det kan spare 13,16 % og 10,21 % af elforbruget til ventilation og opvarmning i et testrum om vinteren i hhv. København og Milano. Sammenlignet med en gaskedel, kan system RLVen spare 46,60 % og 56.16 % af det primære energiforbrug i hhv. København og Milano.

Alt i alt kan RLVen rense indeluft for VOC med en høj effektivitet og kan samtidig opvarme/køle med en energieffektiv metode. RLVen kunne være et konkurrencedygtigt valg til ventilationsanlæg til at reducere energiforbruget og opretholde et sund, komfortabel og produktivt indeklima.

Nøgleord: luftrensning, affugtning, silicagel rotor, varmepumper, energi ydeevne

Contents

Preface I
Nomenclature II
Summary IV
Resumé V
1 Background 1
1.1 Indoor air pollutants 2
1.1.1 Particulate pollutants 3
1.1.2 Biological contaminants 4
1.1.3 Molecular contaminants 4
1.1.4 Carbon dioxide 5
1.2 Energy use of building ventilation 5
1.2.1 Heat recovery 7
1.2.2 Temperature and humidity independent control 8
1.3 Indoor air purification 10
1.3.1 Catalytic oxidation 11
1.3.2 Ozone oxidation 12
1.3.3 Filtration 12
1.3.4 Plasma 13
1.3.5 Adsorption 14
1.4 Heat pump assisted solid desiccant cooling system 18
2 Introduction 22
2.1 Principle of the CAHP 22
2.1.1 Summer operation mode 22
2.1.2 Winter operation mode 23
2.2 Design of the CAHP 24
2.2.1 Design of the air system in summer mode 24
2.2.2 Design of the air system in winter mode 26
2.2.3 Design of the air system combining summer and winter mode 27
2.2.4 Design of the heat pump system 28
2.3 Control strategy of the CAHP 30

2.3.1 Control strategy in summer operation mode 30
2.3.2 Control strategy in winter operation mode 31
2.4 Expected performance of the CAHP 32
2.4.1 Summer operation mode 32
2.4.2 Winter operation mode 33
2.5 Research plan for the CAHP 34
3 Theoretical Study 35
3.1 Theoretical equations and solving methods 37
3.1.1 Sub-model of silica gel rotor for heat, moisture and VOC transfer 37
3.1.2 Sub-model for heat pump energy performance predication 44
3.2 Model validation 46
3.2.1 Air cleaning validation 46
3.2.2 Energy performance validation 49
3.3 Theoretical investigation on influence of outdoor air temperature and humidity ratio to CAHP performance 51
4 Experimental Study 56
4.1 The test room 56
4.2 CAHP prototype development 57
4.2.1 Silica gel rotor 57
4.2.2 Refrigerant 60
4.2.3 Compressor 62
4.2.4 Expansion valve 63
4.2.5 Condensers and evaporators 63
4.3 Prototype constructions 64
4.4 Experimental setup 65
4.5 Uncertainty analyses 68
4.6 Experimental Design 70
4.7 Data analysing methods 75
4.8 Results 80
4.8.1 Energy saving in summer climates 82
4.8.2 Energy Saving in winter climates 86
4.8.3 The annual energy saving 92

4.8.4 Extra energy use caused by the pressure drop of silica gel rotor in CAHP 93
4.9 Summary of experimental results 95
5 Discussions 97
5.1 Energy performance compared to reference system 97
5.1.1 Summer operation mode 98
5.1.2 Winter operation mode 98
5.1.3 Influence of outdoor air temperature and outdoor air heating load proportion to the energy saving of CAHP in winter mode 102
5.1.4 CAHP compared to reference systems with heat recovery in winter mode 104
5.2 Suggestion for further research and development of CAHP 111
5.2.1 Regeneration temperature 111
5.2.2 Validation for indoor air cleaning capacity of the CAHP 112
6 Conclusions 114
7 Reference 116
8 Appendix 127

List of Figures

Figure 1.1 Pictures of London smog disaster..2

Figure 1.2 Pictures of Los Angeles photochemical smog event..2

Figure 1.3 Pictures of Beijing haze (PM 2.5) event..2

Figure 1.4 Number of published journal articles (in English) about indoor air cleaning according to ISI web of Science (1993-2008) 11

Figure 1.5 Time course measurement of VOC levels (counts per second) downstream of the silica gel rotor when human bio-effluents served as pollutants; the top four VOCs were identified as 33: methanol, 59: acetone, 43: propene and 43/61: acetic acid 16

Figure 1.6 Time course measurement of VOC levels (counts per second) downstream of the silica gel rotor when carpet and linoleum served as pollutants; the top four VOCs were identified as 47: formic, acid (and/or ethanol), 59: acetone, 61: acetic acid and 107: xylenes...................................16

Figure 1.7 Perceived air quality (PD) in the test room at 23°C, 40%RH and $5h^{-1}$ACR without and with the dehumidifier using high temperature regeneration air...17

Figure 1.8 Odor intensity in the test room at 23°C, 40%RH and 5h-1ACR without and with the dehumidifier using high temperature regeneration air...17

Figure 1.9 Schematic diagram of regenerative silica gel rotor [89]..18

Figure 1.10 Schematic diagram of Pennington cycle for air conditioning [91]................................19

Figure 2.1 Principle of CAHP operation in summer mode..23

Figure 2.2 Principle of CAHP operation in winter mode 24

Figure 2.3 Schematic diagram of CAHP designed for summer operation mode...............................25

Figure 2.4 Schematic diagram of CAHP designed for winter operation mode 27

Figure 2.5 Schematic diagram of CAHP designed for both summer and winter modes...................28

Figure 2.6 Heat pump designed for CAHP operated in summer and winter modes 29

Figure 3.1 Flow chart of the simulation process of CAHP 36

Figure 3.2 Physical model of one silica gel rotor [89] 38

Figure 3.3 Comparisons of measured and calculated VOCs removing effect of silica gel rotor 49

Figure 3.4 Comparisons of simulated and experimental measured CAHP power consumption 51

Figure 3.5 Simulated power consumption of CAHP under different outdoor thermal climates 52

Figure 3.6 Relations between COP of the heat pump in CAHP and outdoor thermal climates 53

Figure 3.7 Toluene concentrations in the air delivered to ventilated room from CAHP and their relations with outdoor thermal climates 54

Figure 3.8 Concentrations of 1,2-dichloroethane in the air delivered to ventilated room from CAHP and their relations with outdoor thermal climates 54

Figure 4.1 Silica gel rotor selected for prototype CAHP [112] 59

Figure 4.2 Condensing pressure of different HFC refrigerants at temperature of 70°C 61

Figure 4.3 COP of different HFC refrigerants at recommended condensing and evaporating temperatures 61

Figure 4.4 Pictures of the heat pump and the silica gel rotor for prototype CAHP 65

Figure 4.5 Pictures of the test room for experimental measurements 66

Figure 4.6 Connections of CAHP with the existing air handling units and test room 67

Figure 4.7 Effectiveness of outdoor air humidity ratio to CAHP power consumption 69

Figure 4.8 Schematic diagram of the reference air source heat pump system operated in summer mode 76

Figure 4.9 Schematic diagram of the reference air source heat pump system operated in winter mode 77

Figure 4.10 Instantaneous energy use of CAHP and reference system in the three summer climate conditions of Copenhagen 84

Figure 4.11 Instantaneous energy use of CAHP and reference system in the four summer climate conditions of Milan 85

Figure 4.12 Instantaneous energy use of CAHP and reference system in the six summer climate conditions of Colombo 85

Figure 4.13 Instantaneous energy use of CAHP and reference air source heat pump in the two winter climate conditions of Copenhagen 88

Figure 4.14 Instantaneous energy use of CAHP and reference air source heat pump in the four winter climate conditions of Milan 88

Figure 4.15 Instantaneous primary energy use of CAHP and reference gas boiler in the two winter climate conditions of Copenhagen 90

Figure 4.16 Instantaneous primary energy use of CAHP and reference gas boiler in the four winter climate conditions of Milan 91

Figure 5.1 Simulation results of CAHP energy saving compared with air source heat pump under different outdoor climates and with different indoor occupants numbers 103

Figure 5.2 Simulation results of CAHP energy saving compared with gas boiler under different outdoor climates and with different indoor occupants occupants numbers 103

Figure 5.3 Energy saving of CAHP with 50% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit 107

Figure 5.4 Energy saving of CAHP with 50% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit 107

Figure 5.5 Energy saving of CAHP with 40% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit 108

Figure 5.6 Energy saving of CAHP with 40% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit 108

Figure 5.7 Energy saving of CAHP with 30% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit 109

Figure 5.8 Energy saving of CAHP with 30% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit 109

Figure 5.9 Energy saving of CAHP with 20% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit 110

Figure 5.10 Energy saving of CAHP with 20% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit 110

List of Tables

Table 3.1 Parameters of the adsorption characteristics of toluene and 1,2-dichloroethane on the adsorbent 47

Table 3.2 Properties and operating conditions of the silica gel rotor for air cleaning validation 48

Table 3.3 Measured and calculated VOCs removing effect of silica gel rotor and the deviation between the measured and calculated results 48

Table 3.4 Properties and operating conditions of the CAHP for energy performance validation 50

Table 3.5 Comparison of simulated results and experimental measured results of COP and power consumption of CAHP 50

Table 4.1 Geometry and thermal-physical properties of the test room 56

Table 4.2 Indoor thermal environment set-points and interior heat conduction 56

Table 4.3 Physical properties of the silica gel rotor used in the prototype CAHP 58

Table 4.4 Properties of the compressor selected for the prototype CAHP 62

Table 4.5 Sizes of condensers and evaporators of the prototype CAHP 63

Table 4.6 Models and accuracies of the measuring equipment in the prototype CAHP 67

Table 4.7 Subdivisions of summer and winter operating modes 70

Table 4.8 Summer climate data of Copenhagen 71

Table 4.9 Winter climate data of Copenhagen 71

Table 4.10 Summer climate data of Milan 71

Table 4.11 Winter climate data of Milan 71

Table 4.12 Climate data of Colombo 72

Table 4.13 Hygrothermal loads and supply air conditions calculated for summer climates 73

Table 4.14 Hygrothermal loads and supply air conditions calculated for winter climates 74

Table 4.15 Airflow rates of CAHP for different cities in summer 75

Table 4.16 Airflow rates of CAHP for different cities in winter 75

Table 4.17 Primary energy factors of power and natural gas in Copenhagen and Milan 78

Table 4.18 Outdoor and recirculation airflow rates of CAHP and reference system 79

Table 4.19 Thermal conditions of indoor and outdoor air during the experimental measurements .. 81

Table 4.20 Instantaneous use of the heat pump in CAHP in different cities and different summer climate classes 82

Table 4.21 Instantaneous energy use of the reference air source heat pump in different cities and different summer climate classes 83

Table 4.22 Instantaneous energy use of CAHP, reference system and energy saving of CAHP compared to reference system in different cities and different summer climates 83

Table 4.23 Total energy use of CAHP, reference system and energy saving of CAHP compared to reference system in summer of different cities 86

Table 4.24 Instantaneous energy use of the heat pump in CAHP and reference air source heat pump in different cities and different winter climate classes 87

Table 4.25 Instantaneous energy use of CAHP, reference air source heat pump and energy saving of CAHP compared air source heat pump in different cities winter climate classes 87

Table 4.26 Total energy use of CAHP, reference air source heat pump and energy saving of CAHP in experimental investigated winter climates of different cities 89

Table 4.27 Instantaneous primary energy use of CAHP, reference gas boiler and energy saving of CAHP in different cities and different winter climate classes 90

Table 4.28 Total primary energy use of CAHP, reference gas boiler and energy saving of CAHP compared to gas boiler in experimental investigated winter climates of Copenhagen and Milan..... 91

Table 4.29 Total energy use of CAHP, reference air source heat pump and energy saving of CAHP compared to reference system in whole year of different cities 92

Table 4.30 Comparisons on annual energy use of CAHP and reference system using air source heat pump in summer and gas boiler in winter.. 93

Table 4.31 Extra power consumptions caused by pressure drop of the silica gel rotor..................... 94

Table 4.32 Reevaluation on the energy saving of CAHP in different cities and different seasons compared to reference air source heat pump 94

Table 4.33 Reevaluation on the annual energy saving of CAHP in different cities compared to reference air source heat pump 95

Table 4.34 Reevaluation on annual energy saving of CAHP in different cities compared to reference system using air source heat pump in summer and gas boiler in winter... 95

Table 5.1 Instantaneous power consumption of CAHP, reference air source heat pump and energy saving of CAHP compared to reference air source heat pump in different cities and different winter climates 99

Table 5.2 Total power consumption of CAHP, reference air source heat pump and energy saving of CAHP compared to reference system in whole winter climates of Copenhagen and Milan........... 100

Table 5.3 Instantaneous primary energy use of CAHP, reference gas boiler and energy saving of CAHP compared to gas boiler in different cities and different winter climates.............................. 101

Table 5.4 Total energy use of CAHP, reference gas boiler and energy saving of CAHP compared to reference gas boiler in whole winter climate of Copenhagen and Milan.. 101

Table 8.1 Airflow rates measured in experiments for summer climates 127

Table 8.2 Temperatures measured in experiments for summer climates.. 127

Table 8.3 Humidity ratios measured in experiments for summer climates 128

Table 8.4 Airflow rates measured in experiments for winter climates 129

Table 8.5 Temperatures measured in experiments for winter climates 129

Table 8.6 Humidity ratios measured in experiments for winter climates 130

1 Background

With the rapid development of new and rebuilt constructions all over the world, indoor environment and energy use in buildings have drawn broad attentions in modern society. Indoor air quality is one of the most important aspects of indoor environment that affect occupants' comfort, health and working performance [1]-[4]. Traditionally, indoor air quality is controlled by ventilation which consumes up to 30% of energy in buildings [5]. This proportion can be even higher in future well-insulated and airproof low-energy buildings. Modern technologies of thermal insulation and airproof buildings have been highly developed to make it possible to limit the heat loss/gain between buildings and outdoor environment. In contrast to thermal insulation and airproof technology, ventilation has become the bottleneck on reducing the total energy use in buildings.

The total ventilation requirement of a building is determined by the indoor air quality requirement and indoor air pollution sources which are independent of the thermal insulation of buildings. Due to comfort and health concerns, the ventilation rate prescribed by the existing ventilation standards and guidelines [6][7] is in the range of 2.5 to 10 L/s per standard person. Many studies show that even 10 L/s per person of outdoor airflow rate is not sufficient to remove indoor air pollutants which can lead to the risk of SBS symptoms and short-term sick leaves [1]. An insufficient ventilation rate also decreases the productivity among occupants of office buildings [1]. However, further increases in the ventilation rate are hardly acceptable due to energy concerns.

On the other hand, the classical ventilation concept - which assumes that the outdoor air is clean, may not be valid anymore in most modern cities. Toxic gases and fine particles emitted from vehicles and industries are often introduced into indoors through ventilation. The London smog disasters (shown in Figure 1.1), Los Angeles photochemical smog event (shown in Figure 1.2) which happened in the middle of 20th century warned people that outdoor air can be harmful for indoor occupants. The haze which appears recently in many places of China, Singapore, India and other Asian countries (shown in Figure 1.3) has a huge impact on indoor environment. In these places, ventilation is becoming a pollutant source of indoor environment. In the ASHRAE standard 62.1-2013 [7], when the building is located in an area where the national standard or guideline for fine particulate matter (PM) or ozone is exceeded, particle filters or air cleaning devices shall be provided to clean the outdoor air at any location prior to its introduction to occupied spaces.

Figure 1.1 Pictures of London smog disaster

Figure 1.2 Pictures of Los Angeles photochemical smog event

Figure 1.3 Pictures of Beijing haze (PM 2.5) event

Hence, the best solution to decrease energy use of building ventilation and maintain a healthy and comfortable indoor environment is to develop energy efficient air purification technology to clean indoor air and use less outdoor air for ventilation.

1.1 Indoor air pollutants

Indoor air pollutants are normally classified to three categories including particulate pollutants, biological contaminants and molecular contaminants. Particulate pollutants are normally introduced to indoors from outdoor environment by ventilation or infiltration. Biological contaminants are mainly from dampness of indoor environment and ventilation systems. Molecular contaminants can be emitted from different sources such as building materials, furniture, appliances, ventilation systems and human activities. Particulate pollutants and biological contaminants have been well known as indoor air pollutants, and great efforts have been taken on the researches of particulate and biological contaminations cleaning. Molecular contaminant is a concept that was brought forward in the decade of 1980s, which means indoor air quality was extended to the control of chemical pollution control. Carbon dioxide may be another concern of indoor air pollutants. But, it is used usually as an indicator of indoor air quality [8], pure CO_2 below 3000 ppm is considered to be without negative influence on occupants' performance [9]-[11].

1.1.1 Particulate pollutants

For the indoor particulate pollutants, most of them are introduced from outdoor environment by ventilation systems or infiltration. Positive correlations between mortality and particle concentrations (especially ultra-fine particle concentrations) have been found in epidemiological studies [12]-[14]. These studies showed the importance of controlling the concentration of indoor ultra-fine particles as people spent 90% of their time indoors [15]. Normal medium efficiency particle filters cannot prevent most of the ultra-fine particles from entering indoors through ventilation systems. Although high efficiency filters can be used to remove fine particles, they also produce high pressure drop, which result in much higher electric power consumption for ventilation fans [16]. On the other hand, overdue filters may constitute pollutants sources of particles [17]-[20]. In the study by Bekö et al. [20], dirty dust filters had been identified as pollution sources that emit gas phase pollutants due to the oxidation effect of outdoor ozone. Hence, in some places where have polluted ambient environment, higher ventilation rate can lead to not only higher energy use but also higher indoor particle concentration. In another word, less outdoor air ventilation rate can be a solution in some ways to decrease indoor air particle concentration, but less ventilation rate will cause higher level of gas phase contaminant concentrations which emitted from indoor pollutant sources. If these gas phase contaminants can be cleaned with an energy efficient purification method, it could be a good solution for keeping healthy indoor environment with less energy use.

1.1.2 Biological contaminants

Biological contaminants include mold, house dust mites, bacteria, viruses, animal dander, cat saliva, and pollen. Many of the biological contaminants are small enough to be inhaled. Some molds and other biological contaminants can cause allergic reactions. Bacteria and viruses can cause infections. Mold can also cause infections. Studies have found increases in common symptoms such as coughing, wheezing and headaches in people who live at homes with dampness and visible mold growth. The growth of mold and other biological contaminants are normally caused by high humidity in indoor environment especially when condensation happens on the surfaces of indoor stuff and ventilation systems. To eliminate the indoor biological contaminants, keeping the room clean and all surfaces dust-free is important. Lower humidity ratio in indoor environment will prevent the growth of biological contaminants.

1.1.3 Molecular contaminants

The other type of indoor air pollutants is molecular contaminants, and it is the most difficult one to clean since they exist in gaseous phase. Volatile organic compounds (VOCs) are one type of molecular contaminants. The source of VOCs can be building materials, furniture and human activities. Building materials release a wide range of VOCs with high concentration, and they have greatest harm to occupants' comfort, health and working performance [21][22]. The most well-known VOC pollutants are formaldehyde and benzene homologues.

Formaldehyde was defined as carcinogens by World Health Organization [23]. It is mainly from adhesive for artificial wood, plywood, particleboard and other sheets. The adhesive will release formaldehyde when it is warmed up even in normal room temperature. The adhesive is then becoming a main source of formaldehyde. Paint on walls, doors, windows, furniture is the main source of benzene homologues including toluene, ethyl-benzene, xylene and other BTEX. The paint also releases formaldehyde. The formaldehyde concentration in general newly renovated buildings can exceed more than six times than the guideline in indoor environment standards and criteria. In some buildings, the concentrations of formaldehyde are likely to exceed guideline value for more than 40 times [24].

VOCs can stimulate occupants' tissues and organs including the ocular mucosal, nasal, throat, skin, face, neck, hands, upper and lower respiratory tract [25]. The influence of VOCs to humans belongs to acute response, and people tend to produce olfactory adaptation after a long time exposure [26]. But some sub-acute effects (such as headaches) will be more frequent and heavier with the increasing of exposure time to VOC pollutants [27]. Long-term exposure to formaldehyde may cause cancer of nose, mouth, throat, skin and digestive tract [23]. The releasing of VOCs from building materials and furniture can last many years, and indoor VOCs released from human activities can happen as long as the occupants enter the buildings. Therefore the indoor VOC concentrations should be controlled strictly. Great efforts have been taken on the research of air cleaning technologies which can remove VOCs from indoor air. The work proposed by this thesis is based on the research of indoor air VOCs purification.

1.1.4 Carbon dioxide

Carbon dioxide which is released by human activities may be another concern of indoor air pollutants, but the influence of carbon dioxide on indoor quality and human response is to be further studied. Carbon dioxide is normally used as an indicator of indoor air quality to show whether ventilation in a space is sufficient or not. No sufficient evidence in the published peer-reviewed literature showing that the levels of pure CO_2 below 5,000 ppm have significant negative effects on human health. Two studies [11][28] found negative effects of pure CO_2 at levels below 5000 ppm on the decision-making performance (at 2,500 ppm) and on the performance of office work (at 3000 ppm), but these results are still under validation.

Compared to traditional ventilation system which should keep indoor CO_2 concentration below 1000 ppm, ventilation system with indoor air cleaning can have a lower outdoor air ventilation rate. In this case, the indoor CO_2 concentration may increase, but the indoor air quality can be improved due to the removing of other indoor air pollutants. Thus, indoor air purification combining appropriate amount of outdoor air can improve indoor air quality and decrease building use.

1.2 Energy use of building ventilation

Indoor air pollutants can be removed by ventilation. With the outdoor air which has low concentrations of biological and molecular contaminants, ventilation can take indoor air pollutants away to keep a comfortable and healthy indoor environment. But the ventilation may increase indoor particle concentrations in the case when outdoor air is polluted by particles. Furthermore, ventilation consumes a large amount of energy. To keep a good indoor thermal environment, the outdoor air for ventilation normally needs to be cooled, dehumidified in summer and heated (may be humidified as well) in winter. With the requirement of cooling/heating and dehumidification/ humidification of the outdoor air supplied to buildings, substantial amount of energy will be cost. The energy use for building ventilation accounts as much as 30% of total building energy use [5]. As mentioned above, since the modern technologies of thermal insulation and airproof buildings have been highly developed to make it possible to limit the heat loss/gain between buildings and outdoor environment, the proportion of ventilation energy use can be even higher in future low energy buildings. Energy use for ventilation is becoming a bottleneck of energy conservation in buildings.

In civil buildings, ventilation is normally combined with the system for space heating and air-conditioning to constitute the heating, ventilation and air-conditioning (HVAC) system. Besides heating or cooling the ventilation air, HVAC system is also responsible for removing the heating or cooling load caused by heating gain/loss through the building envelope and the interior load. With the consolidation of the demand for thermal comfort, HVAC system has become an unavoidable asset, accounting for almost half of the energy consumed in buildings. In the study of Lombard et al. [29], the energy use of buildings was found to account for 40% and 37% of total energy use in USA and Europe Union. Within the building energy use, the proportions of HVAC systems reach 68% and 62% in USA and Europe Union respectively, which means the HVAC systems consume 23%-27% of total energy in USA and Europe Union. The trend of HVAC energy use still increase in future due predominantly to the growth in population, enhancement of building services and comfort levels together with the rise of occupants' time spent in buildings. The growing trend in building energy use will also continue during the coming years due to the expansion of built area and associated energy needs. Proliferation of energy use and CO_2 emissions on the built environment has made energy efficiency strategies a priority for energy policies, developing new building regulations and certification schemes. These policies include minimum energy requirements, maximum energy efficiency and exploring renewable energy. To satisfy these energy policies, more energy efficient HVAC systems are needed in future low energy buildings.

Great efforts have been taken on the research of energy efficient HVAC systems. Until now, the proposed technologies from these researches include heat recovery technology, temperature and humidity independent control systems, radiant heating/cooling, ground source heat pump, displacement ventilation, personalized ventilation system and so on. Among these building energy saving methods, heat recovery and temperature and humidity independent control attracted more attentions.

1.2.1 Heat recovery

To reduce energy use for ventilation, heat recovery technologies which transfer energy from indoor exhaust air to outdoor air supply have been widely used in ventilation systems. Sensible heat recovery ventilation technologies using plate heat exchangers have been well developed. The sensible heat recovery efficiency of a counter current plate heat exchanger can be as high as 90% [30]. Sensible heat recovery technologies are suitable for winter seasons or dry climates where sensible heating or cooling is the major hygrothermal load of a ventilation system. For hot and humid climates, the hygrothermal load of a ventilation system is mainly due to dehumidification (the latent load) which accounts for more than 70% of the total energy used to process the ventilation air [30]. This means a sensible heat exchanger (even if its temperature efficiency is 100%) can only recover a max of 30% of the total heat from the indoor exhaust air. Therefore, in order to save energy effectively for ventilation systems in hot and humid climate zones, heat recovery equipment should be able to recover not only sensible heat but also latent heat.

So far, the most commonly used total heat recovery technologies are based on rotary adsorption enthalpy exchangers. This type of enthalpy exchangers use desiccant rotors, which are heat exchange wheels coated with desiccant sorbent to achieve both sensible heat and moisture transfer. Due to the differences of temperature and moisture content between outdoor air supply and indoor exhaust air, heat and moisture can be transferred from outdoor air supply to indoor exhaust air or vice versa through the rotor. The enthalpy recovery efficiency of a rotary total heat exchanger was found to be in the range of 50% to 85% [31]. As it has much higher enthalpy recover efficiency than the plate heat exchanger, rotary enthalpy exchangers have been rapidly developed in recent years. However, studies have shown that gas-phase contaminants may transfer from the indoor exhaust air to outdoor air supply through a rotary enthalpy exchanger [32]-[34]. This transfer of contaminants can be caused by adsorption/desorption of contaminants, carrying over of the rotor, and leakage. Pejterson used a sensory method to assess a rotary enthalpy recovery unit, and found that the sensory pollution load from a rotary enthalpy exchanger was significant and it might constitute a severe pollution load in ventilation systems [32]. In another study conducted by Khoury et al. [33]

where SF6 was used as the monitored chemical, a loss of 30% of added gas was reported and explained by adsorption of SF6 on the rotor. Hult et al. [34] conducted field and chamber experiments to investigate the formaldehyde transfer in rotary residential energy recovery ventilators, and found that the formaldehyde transfer ratio was approximately 29% in field experiments and the ratios were between 10% and 29% in chamber tests.

The other total heat recovery technology use polymer membrane foils for heat and moisture transfer. Studies [35]-[46] have been conducted to investigate the heat and moisture transfer through polymer membranes. Zhang et al. [47] and Nie et al. [48] have studied on the permeation of VOCs and other pollutants through polymer membrane heat recovery units. The mass transfer of chemical pollutants can be much lower than rotary enthalpy recovery unit, but there can still be 10% of indoor air pollutants transferring from indoor exhaust air to outdoor air supply.

Heat pump can be another choice for heat recovering. With this heat recovery unit, indoor exhaust air instead of outdoor air can be used as heating/cooling source to get a higher coefficient of performance (COP) of the heat pump. This application is not used extensively due to the energy cost to transfer heat and moisture from indoor exhaust air to outdoor air supply or vice versa. But it can be a good choice in the case when there is request of no indoor air pollutant transferring from indoor exhaust air to outdoor air supply.

1.2.2 Temperature and humidity independent control

To keep comfortable indoor thermal environment in summer time, traditional air conditioning system use cooling coil to handle sensible and latent load of the ventilated room. The surface temperature of cooling coil is controlled lower than the dew point temperature of ventilation air. Water vapour in the ventilation air will then be condensed on the surface of the cooling coil when the air passes through the coil. Dehumidification and cooling is thus realized. In this process, the ventilation air is normally overcooled due to the mismatched sensible and latent cooling load. To keep constant indoor air temperature and humidity, the ventilation air should be reheated up before it is delivered to the air-conditioned room, and a large amount of energy will be wasted during the overcooling and reheating work. On the other hand, to handle the dehumidification load for ventilation by cooling coil, the evaporating temperature of the chiller should be lower than the dew point temperature as well. The low evaporating temperature in the traditional cooling system leads to a lower COP of chiller compared to an independent temperature and humidity control system where cooling coil undertakes only sensible cooling load. The huge energy use of coupled dehumidification and cooling system is becoming unacceptable as the energy use of buildings has shared a large proportion in total energy use [29]. The other concern is that with coupled dehumidification and cooling in cooling coil, the water vapour will be condensed to liquid water and provide opportunities to the growth of bacteria and mould. The bacteria and mould may pollute the ventilation air and decrease the efficiency of cooling coil or increase resistance of cooling coil in the air channels.

To make air conditioning system more energy efficient, temperature and humidity independent control (THIC) system has been introduced into air conditioning system. Studies on different types of THIC systems have been conducted including liquid/solid desiccant cooling, energy recovery units with a conventional cooling system to avoid excess cooling [49]-[53]. Zhang [53] has conducted studies on the energy performance of independent air dehumidification systems with energy recovery measures, and the results showed that the system of mechanical dehumidification with membrane total heat recovery consumes the least primary energy among the systems tested. In the case study conducted by Ling et al. [54], desiccant wheel assisted separate sensible and latent cooling air-conditioning systems were tested under the AHRI standard. The idea of applying divided condensers (or gas coolers) to refrigerant system was proposed, and the results showed that the COP of vapour compression cycles improves by 36 %, 61% to baseline R410A and CO_2 systems. The study of Chen et al. [55] designed and tested an independent dehumidification air-conditioning system with a hot water-driven liquid desiccant and a chiller that provides 18-21°C chilled water for an office building in Beijing, the results showed that the system tested saved about 30% cooling cost compared with conventional system. Ma et al. [56] has tested energy performance of a hybrid system, the results showed that the COP of the system tested was 44.5% higher than conventional vapour compression system at a latent load proportion of 30% and this improvement could be 73.8% at a 42% latent load proportion.

Heat recovery, temperature and humidity independent control technologies can somehow improve the energy efficiency of HVAC system, but the energy saving potential of the technologies is still limited. In some cases, it can even decrease the ventilation efficiency by contaminants transfer. If the heat recovery, temperature and humidity independent control technology can be further combined with indoor air cleaning technology, indoor environment can be maintained healthy and comfortable with less energy use.

1.3 Indoor air purification

Following the three different types of indoor air pollutants, indoor air purification also includes three categories. The purification of particulate pollutants is normally realized by filtration. Regularly replacing the filters will help improve the indoor air quality. The cleaning of biological contaminants is done by wiping out or vacuuming the dust on the surface of indoor stuffs, keeping relative low humidity ratio, avoiding condensation in the ventilation systems and the ventilated rooms. Compared to the particulate and biological contaminants, molecular contaminants including VOCs are more difficult to clean since they normally exist in gas phase. As mentioned above, the work proposed in this thesis is mainly focused on the purification of VOCs.

The literature review by Zhang et al. [57] summarized the numbers of published journal articles researching on each indoor air molecular contaminants purification method. These methods include catalytic oxidation, ozone-oxidation, filtration, plasma, sorption and ultraviolet germicidal irradiation (UVGI). Figure 1.4 gives the numbers of the articles researching on indoor air purification methods according to ISI Web of Science from 1993 to 2008. The studies and articles related to indoor air cleaning have increased rapidly since 1993 as indoor air quality has got more attentions. The relative effective VOCs cleaning technologies among the investigated are summarized to be catalytic oxidation, filtration with activated carbon, ozone-oxidation, plasma and adsorption.

The literature study by Zhang et al. [57] has also drawn the conclusions as followings:

1) None of the reviewed technologies was able to effectively remove all indoor pollutants and many were found to generate undesirable by-products during operation.

2) Filtration of particle pollutants and adsorption of gaseous pollutants were the most effective air cleaning technologies among the investigated, but there is insufficient information regarding long-term performance and proper maintenance.

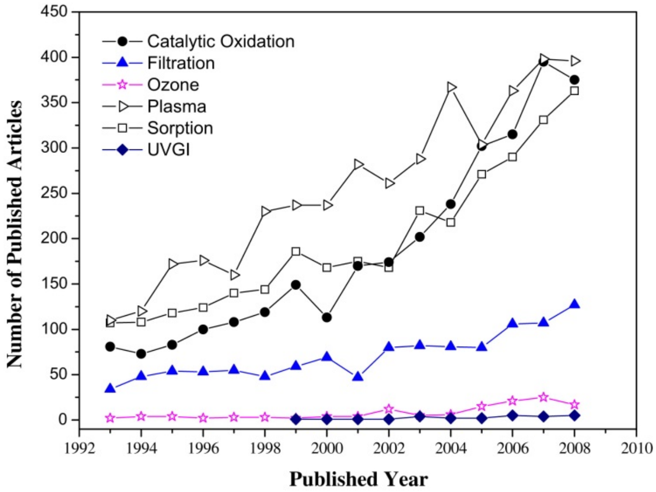


Figure 1.4 Number of published journal articles (in English) about indoor air cleaning according to ISI web of Science (1993-2008)

1.3.1 Catalytic oxidation

Catalytic oxidation is one common method for gaseous pollutants decomposition in many commercially available purifiers. It includes photo-catalytic oxidation and thermal catalytic oxidation. Most catalytic oxidation air cleaning studies focus on photo catalytic oxidation. The photo-catalytic oxidation (PCO) can degrade almost all contaminants such as aldehyde, aromatics, alkanes, olefins, halogenated hydrocarbons odor compounds and so on. The process is based on the decomposition of the pollutants on surface of catalyst under irradiation of ultraviolet light. Efficiency of these catalysts depends on the composition and surface area of the catalyst, humidity in the air, wavelength and intensity of the ultraviolet energy eradiated on the surface of the catalyst. The competitive adsorption effect between the contaminants and moisture was found to have a significant effect on the purification efficiency [58].Hybrid catalysts include combination of catalytic oxidation with ozone and adsorption materials were proposed to improve VOCs purification efficiency [59]-[61].

The main and biggest concern during catalytic oxidation operation is the production of by-products. PCO can generate by-products (formaldehyde, acetaldehyde etc.) that are even more harmful than what they have decomposed [62]-[65]. In most studies proposed until now, only a single compound was tested as indoor gas phase contaminant. However, indoor air contains numerous contaminants, the results of testing on one or a few contaminants may be misleading.Thus, catalytic oxidation is suggested to clean specific contaminants (e.g. in industrial environments) rather than nonindustrial indoor air with a wide spectra of contaminants.

1.3.2 Ozone oxidation

Ozone is an oxidant which can be used to clean indoor air due to its reaction with gas phase contaminants. Kwong et al. has combined ozone oxidation with porous adsorbent to take the advantage of oxidizing capability of ozone and reduce the residual ozone due to enhanced catalytic reaction in the porous structure. The ozone oxidation over porous materials is one of the biological treatments. Utilization of this method can meet industrial standards of decreasing total volatile organic compounds (TVOCs) concentration with lower investment cost than others which are currently in use. To make it possible to apply in non-industrial buildings, further development has to be performed [66]. Small, portable ozone generators have gone on sale in USA. This product is attached to shirt lapel or to necklace close to the breathing zone. Gaseous pollutants in breathed air are decomposed by oxidation using ozone and negative ions. The main concern of ozone oxidation is that high concentration of ozone in breathing area is even more harmful for health than inhalation of the same amount of gaseous pollutants without decomposition [67] , and the reaction of ozone with compounds such as terpens can produce potentially harmful secondary organic aerosol (SOA) in the ultra-fine and fine size ranges [68]. Cautions should be taken during the operation of ozone oxidation.

1.3.3 Filtration

Filtration is effective in removing particulate pollutants. Filters for different particulate sizes have been developed, and all of them have reported high efficiency in removing particles. However, no gas phase removing has been reported with normal filters which don’t contain adsorption materials [16]. As the indoor air cleaning in this thesis focuses on purification of molecular contaminants, the discussion of removing of particles won’t go to details. The combination of adsorbent materials (activated carbon) with filter has been studied by Bekö et al. [69][70] and some VOCs removal has been found. Removal of ozone has also been reported by Bekö et al. [20] and Zhao et al. [71] in the studies of filtration with activated carbon. Together with ozone reaction, ozone reaction products such as VOCs releasing from filters have been reported [72][73]. The odor and sensory pollution load from filters with active carbon have been found in the study of Bekö et al. [20][69][74] and

Hyttinen et al. [73]. Thus, mechanical filters can efficiently remove particles, but are not as effective for organic and inorganic chemical pollutants. The filters can even act as pollution source if they are not changed or maintained properly.

1.3.4 Plasma

Plasma is an indoor air purification method which uses ionization to decompose VOC chemicals. Gases can be ionized to positively charged particles and negatively charged particles. Air ionizers create charged air molecules upon the application of an energy source. By energetically either adding or removing an electron, air molecules are given a negative or positive charge (usually oxygen or nitrogen species, respectively). Three modes of ionization have been employed: photon ionization, nuclear ionization, and electronic ionization. Photon ionization uses a low-energy X-ray energy source to displace electrons from the gas molecules. Nuclear ionizers use polonium-210 radiation sources that emit alpha particles which then collide with the gas molecules and displace electrons. Molecules that lose electrons become positive ions. Neutral gas molecules rapidly capture these free electrons and become negative ions. These types of ion generators do not have electrodes, so deposits are not a concern. X-ray and nuclear sources must be carefully installed and controlled to avoid creating safety hazards. Charged particles form a plasma which keep overall neutral state. Plasma exist in oxidizing gases contain a large number of atomic oxygen, free radicals and other active substances which can oxidate harmful gases such as formaldehyde, benzene oxidation to carbon dioxide and water [75]. But the effectiveness of using plasma in ventilation system still needs to be investigated.

Plasma air cleaners have high efficiency e.g., within the range of 76–99% to remove particles [76]-[78], but it was found not efficient for removing gas-phase pollutants [76]. Combining plasma air cleaner with catalytic technology, VOCs such as toluene removal with high efficiency was observed [77]. Park et al. [76] combined plasma air cleaner with ultraviolet -catalytic technology, and realized improved removal efficiencies for formaldehyde, benzene, toluene and xylene. The VOCs removal efficiency of plasma-catalyst technology can be inhibited by humidity [78]. The main problem of plasma technology is the production of by-product pollutants such as NO_x and O_3 [76][79].

1.3.5 Adsorption

Adsorption is normally realized with porous materials. The capillary porosities between the particles of porous materials form free space which can adsorb chemicals. The porosities also make the desiccant material have huge specific surface area. The surface of desiccant material can reach several hundred square meters per gram [80], and the huge surface area makes them have strong adsorption ability. Porous materials can be used for dehumidification and adsorption of indoor air pollutants.

The adsorption materials include active carbon, molecular sieves, zeolites, silica gel and so on. Adsorption materials are widely used to adsorb moisture. The adsorption materials used mainly for dehumidification are called desiccant materials as well. Studies found that the adsorption materials can also adsorb gas phase pollutants other than water vapor [81]-[86]. Adsorption process is carried out on the phase interface of adsorbent and adsorbate. The molecules (or atoms, ions) existing in the phase interface will get a force vertical to the interface due to unbalanced attractive force from the molecules in the two phase bodies. The vertical force makes the molecules on the interface have additional energy compared to molecules in the phase body. To release the additional energy, and to achieve equilibrium state, the molecules on the interface will attract other molecules in the phase body. This will result in the molecule concentration difference in the interface layer and the phase body. That is how adsorption functions. Adsorption doesn't decompose or change VOCs, but it will make them adsorbed on the surfaces, thus to reduce its concentration in the air. The main problem of using commercial available sorbent material for indoor air cleaning is the short lifetime. Due to high concentration of moisture in air (usually 4 to 5 order-of -magnitudes higher than the concentration of VOCs), most of the surface of sorbents are occupied by H_2O molecules and very small space is left for adsorbing VOCs. Thus, any sorbents will be saturated in short time and loss the adsorption ability even if the air purifiers were not in operation unless the sorbents were sealed when the air purifiers are in backup mode. The active carbon, for example can be saturated in few hours with adsorbed moisture and contaminants. Of course, the adsorption life also depends on the amount of sorbents. Increase the amount of sorbents in an air purifier can increase the life time of adsorption but will increase the cost of the air purification unit, the noise level and greatly increase resistance of air resulting in an increase in the running cost by the increased fan power.

To overcome the short adsorption lifetime of the sorbents for air purification, the desiccant wheel with adsorption materials reactivated in real time was proposed for indoor air cleaning [87]-[89]. In the study of Fang et al. [88], the air cleaning efficiency of silica gel rotor was evaluated by PTR-MS and sensory assessment, and the results showed that the measured VOCs were removed effectively by the desiccant wheel with an average efficiency of 94% or higher; more than 80% of the sensory pollution load was removed and the percentage dissatisfied with the air quality decreased from 70% to 20%.

Figure 1.5 and Figure 1.6 show the air cleaning effect of a silica gel rotor. The experiment investigation conducted in the study of Fang et al. [88] used a commercially available rotary desiccant dehumidifier (a silica gel rotor used in a commercial dehumidifier) as the air cleaner. Two types of indoor air pollution sources including human bio-effluents and flooring materials were used in the experiment. The volatile organic compounds (VOCs) in the air were measured by a Proton-Transfer-Reaction-Mass Spectrometry (PTR-MS) gas analyzer. The results showed that almost all the measured VOCs were removed effectively when the air passed through the silica gel rotor. The results also showed that decreasing the regeneration heat of the rotor by 50% did not influence its air cleaning effect. Figure 1.7 and Figure 1.8 show the effectiveness of using silica gel rotor dehumidifier on perceived air quality and odor intensity in a test room. Compared to the other indoor air purification methods, there is no secondary pollutant or by-product emitted during the air cleaning process. However, such a high efficiency air cleaning technology of regenerative silica gel rotor has not been used for indoor air cleaning.

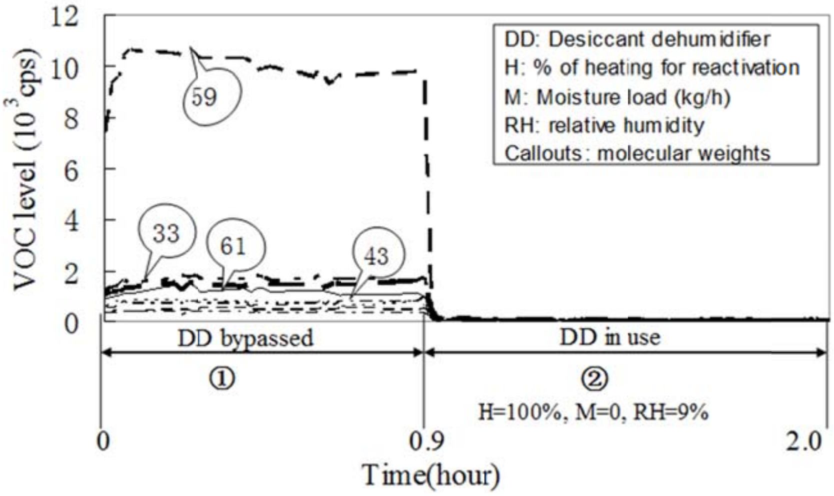


Figure 1.5 Time course measurement of VOC levels (counts per second) downstream of the silica gel rotor when human bio-effluents served as pollutants; the top four VOCs were identified as 33: methanol, 59: acetone, 43: propene and 43/61: acetic acid

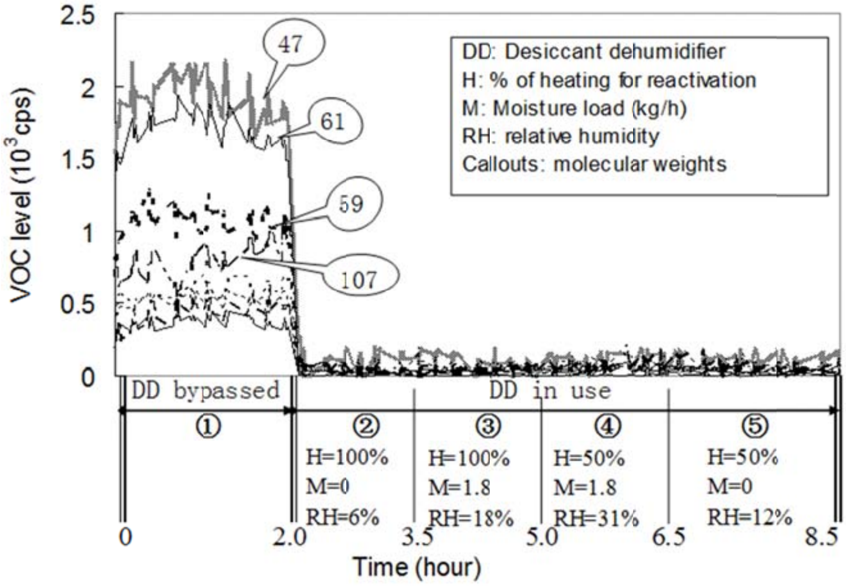


Figure 1.6 Time course measurement of VOC levels (counts per second) downstream of the silica gel rotor when carpet and linoleum served as pollutants; the top four VOCs were identified as 47: formic, acid (and/or ethanol), 59: acetone, 61: acetic acid and 107: xylenes

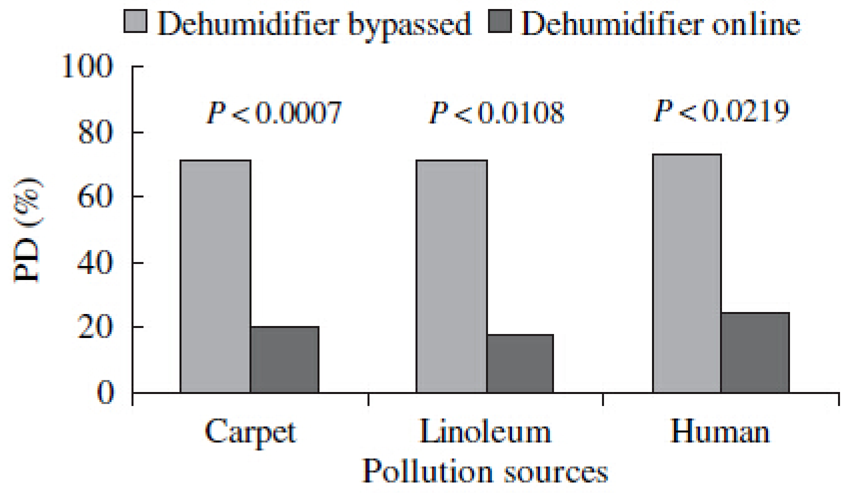


Figure 1.7 Perceived air quality (PD) in the test room at 23°C, 40%RH and 5h^{-1}ACR without and with the dehumidifier using high temperature regeneration air

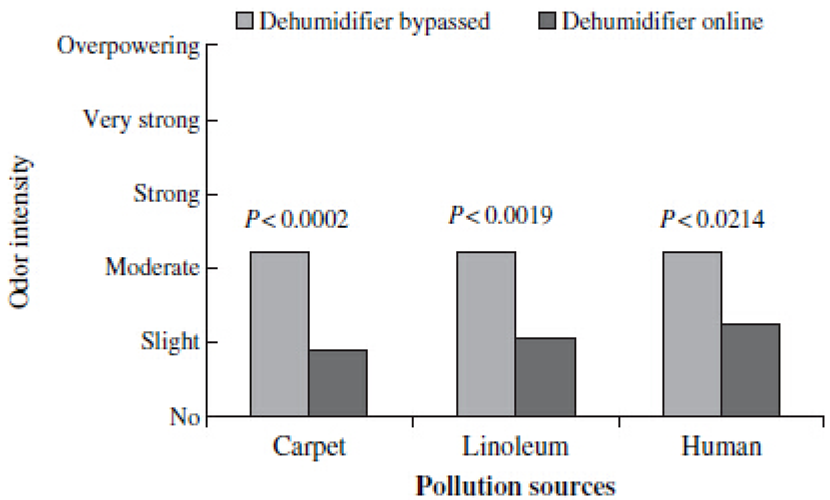


Figure 1.8 Odor intensity in the test room at 23°C, 40%RH and 5h-1ACR without and with the dehumidifier using high temperature regeneration air

The dehumidification and air cleaning capacity of regenerative silica gel rotor requires a certain amount of energy to get reactivated. Figure 1.9 gives the schematic of regenerative the silica gel rotor. The reactivation air needs normally high temperature, the consumption of energy for heating reactivation air is then the main barrier of using desiccant rotor for air cleaning in ventilation system.

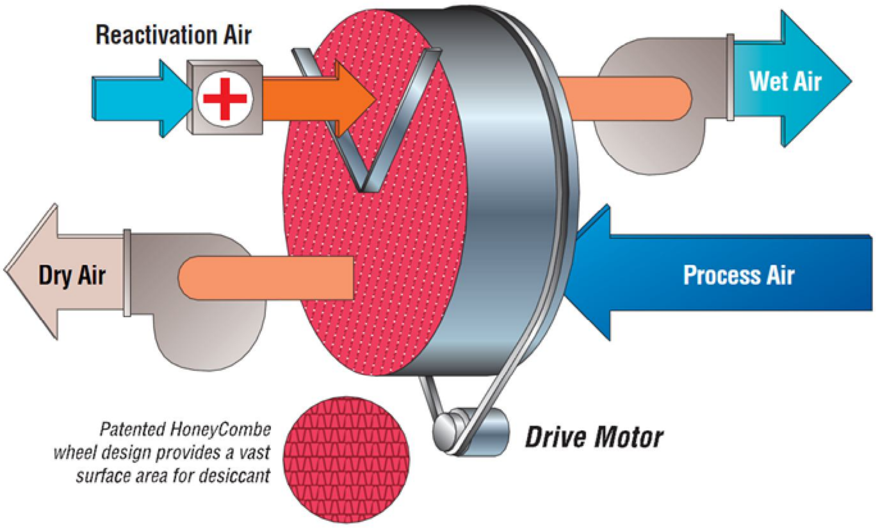


Figure 1.9 Schematic diagram of regenerative silica gel rotor [89]

To break through the barrier of regeneration heat for silica gel rotor, an innovative CAHP which combines silica gel rotor with heat pump is proposed in the study presented by this thesis. The CAHP was designed and developed based on the air cleaning capacity of silica gel desiccant rotor. In the CAHP, the condensing heat from the heat pump is used to regenerate the silica gel rotor. The CAHP has integrated air purification, dehumidification, cooling, heating and heat recovering in one unit. Details about the principles of the CAHP are given in following chapters.

1.4 Heat pump assisted solid desiccant cooling system

Regenerative desiccant wheel was originally designed for dehumidification known as regenerative rotary desiccant dehumidifier. This dehumidification technology has been used in ventilation and air-conditioning system since Pennington patented the first solid desiccant cooling system-Pennington cycle in 1955 [91]. One schematic diagram of the Pennington cycle is given in Figure 1.10. The outdoor air was firstly taken into the system by fan (13) and transferred moisture to the desiccant wheel (15), and then transferred heat to exhaust air through wheel (27). Before it is delivered to indoors, the outdoor air supply was cooled by cooling coil (38). Thus, the outdoor air is cooled, dehumidified and delivered to indoors to keep a comfortable indoor thermal environment. On the other side, indoor exhaust air was taken into the system through inlet (45) and cooled by cooling coil (48), and then got heat from outdoor air supply through wheel (27). Then indoor exhaust was heated by heating coil (53) and used to regenerate the wheel (15) which was statured with moisture adsorbed from outdoor air. After the regeneration, the indoor exhaust was rejected outside by fan (56). Since the dehumidification for outdoor air supply was done by the desiccant wheel (15). The liquid temperature in cooling coil (38) and cooling coil (48) could be higher than the cooling coil in vapor-compression refrigeration cycle.

The main drawbacks of Pennington cycle are followings.

1) The heating for regeneration air was done by electrical power which would decrease the energy efficiency of the system.

2) The contaminant transfer from indoor exhaust air to outdoor air supply through the wheel especially the adsorption wheel was not considered and it may constitute a severe pollutant load for outdoor air supply.

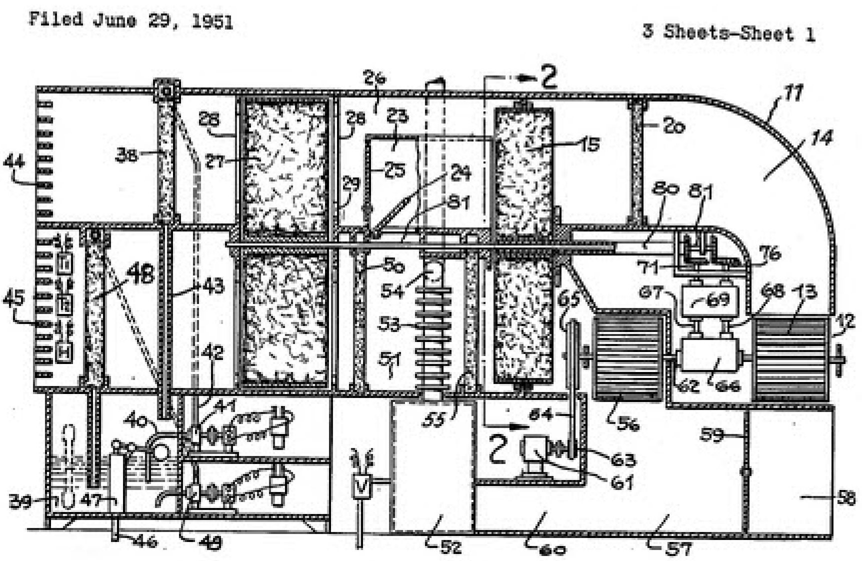


Figure 1.10 Schematic diagram of Pennington cycle for air conditioning [91]

Great efforts have been taken to study the solid desiccant cooling system after the Pennington cycle. Considering energy efficiency in HVAC system, the solid desiccant cooling system is only efficient when the desiccant rotor was regenerated with thermal sustainable energy such as solar energy [92]-[94], energy from co-generators [95][96], or waste heat [97]. But the using of solar energy, co-generator and waste heat is normally limited by regional and climate factors. To break the barrier, studies on heat pump assisted hybrid solid desiccant cooling system were brought forward [98]-[101]. However, there has not been a consistent conclusion on the energy efficiency of desiccant cooling systems due to the large variety in system configurations and operating conditions.

Subramanyam et al. [102] integrated a desiccant rotor into a traditional refrigeration air conditioning device. They concluded that the COP of the integrated system was 5% lower than the traditional system in the case where the supply air was not reheated and sent into conditioned space at dew point, and the COP of the integrated system was nearly double of the traditional air conditioning system if the supply air was reheated. On the condenser side of heat pump which assist solid desiccant cooling system, Jia et al. [98] and Hao et al. [101] have used an additional electrical heater before the desiccant rotor to get a high regeneration air temperature, but the additional electrical heater decreased the overall energy efficiency of the system. The studies of Zhang et al. [99] and Sheng et al. [100] used high temperature heat pump to achieve high regeneration temperature, but the COP of the heat pump was not high. Sheridan and Mitchell [103] proposed a solid desiccant cooling system, in which outdoor air at ambient state is mixed with the return air from the air-conditioned space. This air is dehumidified and heated using a desiccant dehumidifier and is subsequently cooled by an indirect evaporative cooler, which was a plate heat exchanger (PHE) device. Further cooling to the required entry is performed by the evaporator coil of the vapor compression unit. The desiccant dehumidifier has been assumed to be of the rotary wheel type and is regenerated by heat from the condenser of the vapor compression unit, plus an auxiliary heater. Solar cycle is used to provide auxiliary heat for the regeneration air. This means that the condenser operates at a lower temperature than in the case where regeneration heat is fully from the condensing heat, and as a result the vapor compression unit operates more efficiently. When the cooling load is predominately sensible, indirect evaporative cooling using the PHE is used. If the PHE is unable to meet the full load, it is augmented by the vapor compression unit. They reported that such a system saved energy compared with a vapor compression unit when the load had a high sensible fraction but vice-versa when the load had a high latent fraction. Jurinak et al. [104] used a direct evaporative cooling device instead of a plate heat exchanger or cooling coil to take the sensible load after the dehumidification of the descant wheel. They indicated that systems with improved dehumidifier, rotary sensible heat exchanger and direct evaporative cooling device could achieve seasonal COPs in the order of 1.1. Due to direct evaporative cooling after dehumidification, the outlet air humidity ratio from desiccant rotor should be lower that the supply air to the ventilated room, which results in much higher regeneration air temperature. These previous studies on solid desiccant cooling system show a fact that the desiccant cooling technique using silica gel rotor is not ideal in practice and didn’t get enough attentions either because of the great initial investment or due to inconsistent energy efficiency.

Besides the research on dehumidification of desiccant wheel, the studies of Fang et al. [88] and Zhang et al. [89] brought fresh blood for the desiccant wheel since they found the silica gel rotor can have effective VOCs purification capacity. The VOCs removal efficiency can reach more than 80% when it was operated as normal dehumidifier. As the adsorption of VOC contaminants occupies only a small proportion of the sorbent, there is almost no competition between the adsorption of water vapor and contaminants. Such studies strongly pushed forward the silica gel rotor being in use in ventilation system for air purification and dehumidification.

This thesis presents the design, development and test of a CAHP which combined silica gel rotor with heat pump. The CAHP integrated air purification, dehumidification, heating/cooling and energy recovery in one unit. It has connected the silica gel rotor with the heat pump to make full use of both heating and cooling from the condenser and evaporator of the heat pump. The CAHP also used the air purification and dehumidification capacity of silica gel rotor. With the CAHP, the indoor thermal environment and indoor air quality was expected to be maintained with an energy efficient method.

2 Introduction

The principle, design and operation mode of the CAHP are introduced as followings. The research plan and expected results are given and discussed as well.

2.1 Principle of the CAHP

The proposed CAHP combines silica gel rotor with heat pump to make full use of both heating and cooling from the condenser and evaporator of the heat pump. The design concept is to transfer the total energy output of the heat pump (both condenser and evaporator) into cooling, dehumidification, air purification for summer application and heating, air purification for winter application.

The CAHP was designed to be used for both summer and winter seasons when the thermal load of a ventilation system can be dehumidification, cooling or heating.

2.1.1 Summer operation mode

In summer operation mode of the CAHP, ventilation system supplies minimum levels of outdoor airflow and recirculates a large quantity of indoor air. The recirculated air is mixed with the outdoor air and then processed by the silica gel rotor where a large amount of moisture and indoor air pollutants are removed by the rotor. In this process, the mixed air is dehumidified, cleaned and warmed up. The mixed air will then be cooled by the evaporator of the heat pump. Finally, the cool, dry and clean air is delivered into the ventilated room to keep a healthy and comfortable indoor environment. On the other side of the heat pump, the regeneration air from outdoor is heated up by the condenser of the heat pump and used to regenerate the silica gel rotor. After the regenerating work, the regeneration air is rejected to outdoors together with the exhaust indoor air. Thus, the sensible heat, latent heat and air pollutants of indoors are removed to outdoors. Figure 2.1 shows the principle of CAHP operation in summer mode.

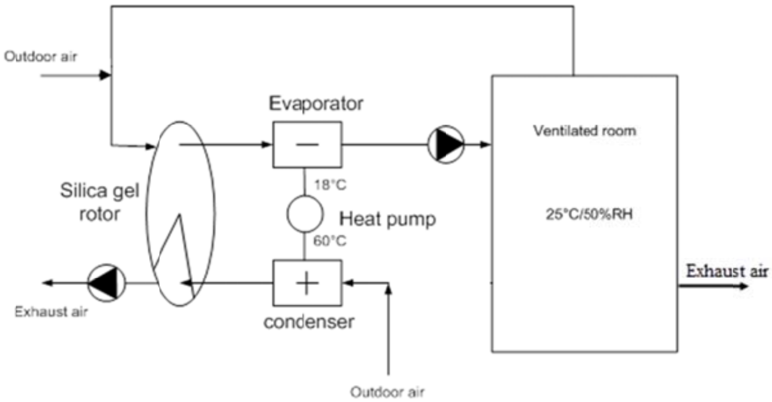


Figure 2.1 Principle of CAHP operation in summer mode

2.1.2 Winter operation mode

In winter operation mode of the CAHP, the recirculation air is cleaned by silica gel rotor as well. Different from the summer mode, the regeneration air temperature could be much lower due to the low outdoor air temperature and humidity ratio. The low regeneration temperature means that the rotor has a very limited ability for dehumidification which is just preferable for winter application since little moisture needs to be removed from indoor air. However, in previous study [88], it was found the low regeneration temperature does not significantly affect the air purification efficiency of the silica gel rotor. The recirculation air is, therefore, cleaned and slightly dehumidified, heated by the rotor in the winter operation mode. The heating effect of the rotor is also favorable for winter use. The recirculation air - after being cleaned, slightly dehumidified and heated by the rotor - is mixed with the outdoor air for ventilation (heated by the condenser of the heat pump) and then delivered into the room. The regeneration air for the silica gel rotor is outdoor air which is also heated by the condenser of the heat pump. After regenerating the rotor, the warm regeneration air is mixed with indoor exhaust air and rejected through the evaporator of the heat pump. The heat from the regeneration air (after regeneration work) and the indoor exhaust air is then recovered by the evaporator of the heat pump. Figure 2.2 gives the principle of CAHP operation in winter mode.

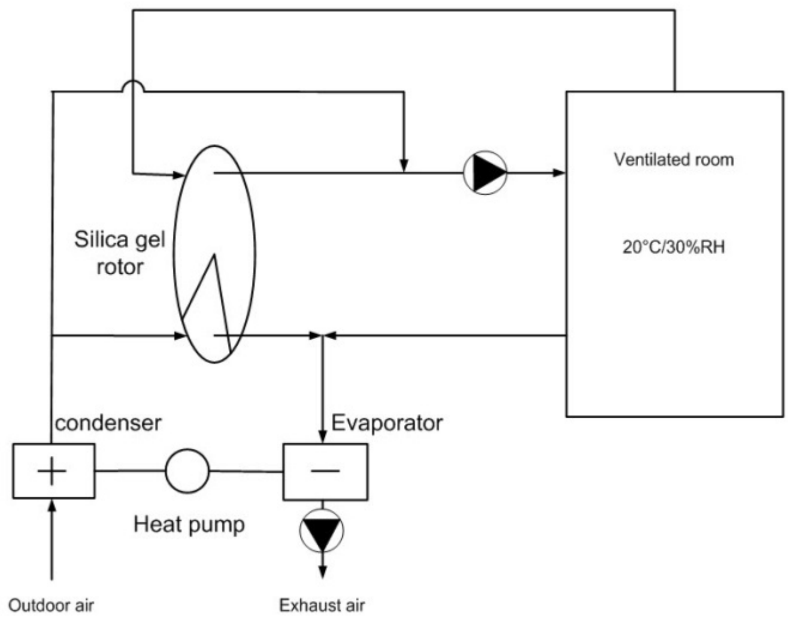


Figure 2.2 Principle of CAHP operation in winter mode

2.2 Design of the CAHP

The air and thermodynamic system of the CAHP is designed based on the principles mentioned above. The designed CAHP includes two evaporators and two condensers. One evaporator is used in summer mode and the other is used in winter mode. One condenser is used for heating the regeneration air, and the other is used to take away the excess heat generated from the heat pump (in summer) or used to heat up the outdoor air (in winter). In summer, outdoor air delivered to the ventilated room is dehumidified by the silica gel rotor and cooled by the evaporator of the heat pump, while in winter it is heated up by a condenser of the heat pump.

2.2.1 Design of the air system in summer mode

Figure 2.3 shows the schematic and air system design of the CAHP for summer operation mode. Calculation and theoretical studies before the CAHP design work found that, in most cases of summer, the condensing heat generated from the heat pump is more than what is required for regenerating the rotor. If all the condensing heat is used to regenerate the rotor, it may over dry the ventilation air and increase the cooling load of the evaporator and, in turn, increase the energy use of the CAHP. The dual-condensers design was adopted to control the heating for the regeneration air at the exact amount as demanded by dehumidification. In this dual-condensers design, the surplus condensing heat is rejected directly by the second condenser without feedback to the evaporator.

There are two air inlets and two air outlets in the CAHP as shown in Figure 2.3. The two inlets include one outdoor air inlet taking outdoor air from ambient and one indoor air inlet taking air returned from the ventilated room. The two outlets include one exhaust air outlet rejecting exhaust air to outdoor ambient and one supply air outlet delivering clean, dry and cool air to the ventilated room.

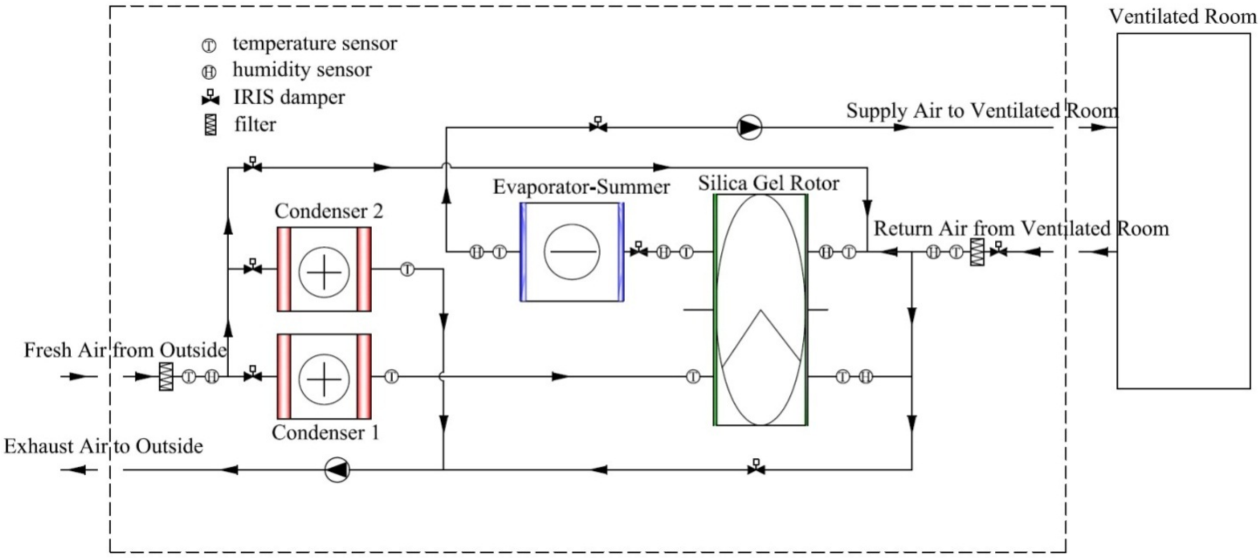


Figure 2.3 Schematic diagram of CAHP designed for summer operation mode

From this design, energy on both sides of the heat pump is used and contributed for air purification, dehumidification and cooling. The condensing heat is used to regenerate the rotor (removing latent heat and pollutants). The evaporating cooling is used to cool the process air (removing sensible heat). This design allows the heat pump to cool the indoor air in a very efficient way and consume less power compared to the conventional air source heat pump in which the heat from the condenser is rejected to ambient without being used. The system is working in a completely dry environment, no water condensation occurs on the evaporator of the heat pump since the moisture was removed by the silica gel rotor. This could effectively prevent the growth of mold and bacteria in the air-conditioning system. The CAHP system cleans a large quantity of recirculation air and removed pollutants from the air, which could be equivalent to a high outdoor air ventilation rate. In the study of Fang et al. [88], 80% of the cleaned recirculation air by the silica gel rotor could be used as outdoor air. Meanwhile, the air cleaning by the silica gel rotor takes place together with dehumidification, and it doesn’t consume extra energy. Therefore, such a system devoted to air cleaning can provide a large quantity of effective ventilation air without significantly extra energy use.

2.2.2 Design of the air system in winter mode

Figure 2.4 shows the schematic and air system design of the CAHP for winter operation mode. Similar with the summer mode, the CAHP also uses two condensers in the winter mode. In addition to regenerate the silica gel rotor, the surplus condensing heat in winter mode is used for heating the outdoor air supply delivered to the ventilated room. The evaporator is used to recover the total heat from the exhaust air and transfers them to the condensers. Since the dehumidification requirement in winter season is very low, the regenerating temperature (usually below 30°C) can be much lower than it is in summer. Therefore, the COP of the heat pump in CAHP for winter is usually higher than it is for summer due to the lower condensing pressure. During the regeneration process, part of the regeneration heat is transferred to warm up the recirculation air through the rotor. The rest of the regeneration heat is recovered by the evaporator of the heat pump. Such a winter mode design could keep all the heat in the ventilation system indoors without losing them from the exhaust air. When a ventilation system uses CAHP, the outdoor air requirement can be much lower than the conventional ventilation system due to the strong air cleaning ability of the CAHP. Thus the indoor air humidity could be slightly higher than the case using conventional ventilation system even though the silica gel rotor removes small amount of moisture when it is running at low regeneration temperature for air cleaning.

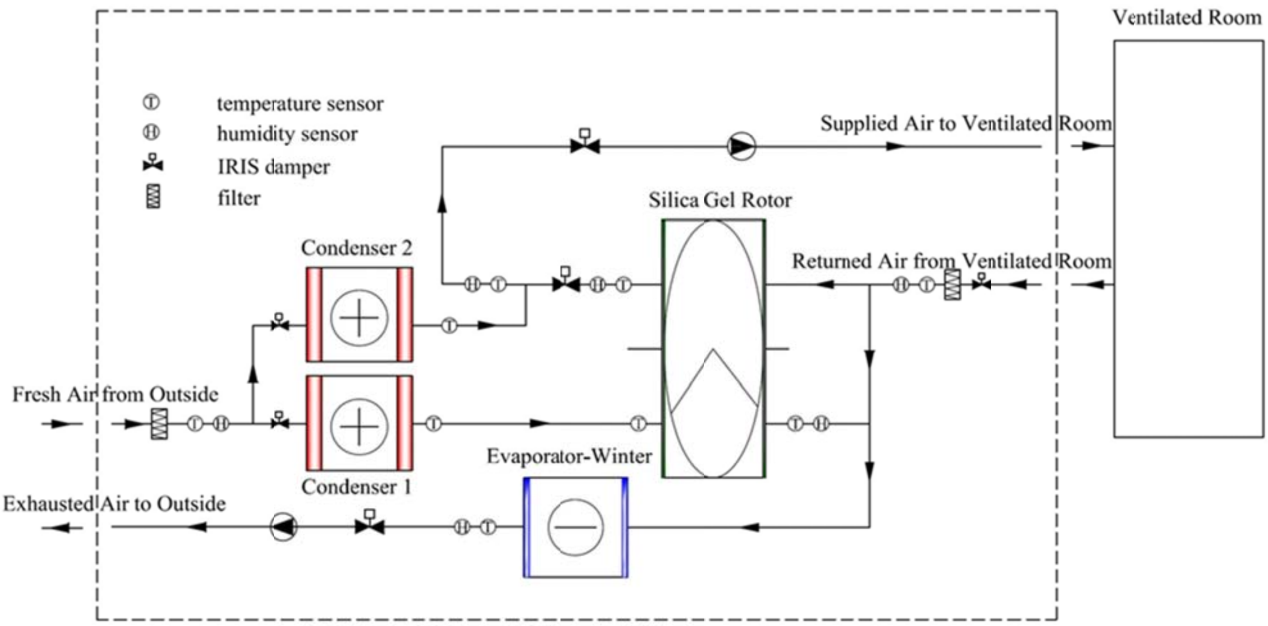


Figure 2.4 Schematic diagram of CAHP designed for winter operation mode

With this winter mode design, energy from the rejected air (exhaust air from the ventilated room and regeneration air) is recovered by the heat pump. Therefore, energy on both sides of the heat pump is fully used to heat the building. High clean air delivery rate without extra energy use can be realized. By proper control of the regeneration temperature, the silica gel rotor can be used for moisture control to prevent the risk of a damp building. Theoretical and experimental studies will be conducted to assess the energy performance of the CAHP.

2.2.3 Design of the air system combining summer and winter mode

In case where the proposed technology is used in the climate zone that requires heating in winter and cooling in summer, the design shown in Figure 2.5 can be used. In this design, two evaporators are used for the same heat pump, one is used in winter and the other is used in summer. The same condensers of the heat pump are used for both seasons but the outdoor air supplied to the ventilated room is switched by dampers. In winter, the outdoor air for ventilation is pre-heated by the condenser of the heat pump, while in summer; it is dehumidified, cooled by the silica gel rotor and evaporator of the heat pump.

In addition to the outdoor air supply, large quantity of indoor air is recirculated through the silica gel rotor where the recirculation air is cleaned and dehumidified by the rotor. The outdoor air joins the recirculation air to ventilate the room and to control indoor air temperature, humidity and air quality. In summer, the mixed air of outdoor and recirculation air after being processed by the rotor is too warm and is designed to be cooled by the evaporator of the heat pump before it is delivered into the ventilated room. In winter, such cooling is not necessary. The evaporator of the heat pump is then placed at the exhaust air position of the system to recover total heat of the air rejected from the system. Regeneration of the silica gel rotor uses outdoor air heated by one condenser of the heat pump. In summer, the warm air after regenerating the rotor and the exhaust air from the ventilated room is rejected directly to outdoor environment. In winter, the heat of the rejected air is recovered by the evaporator of the heat pump.

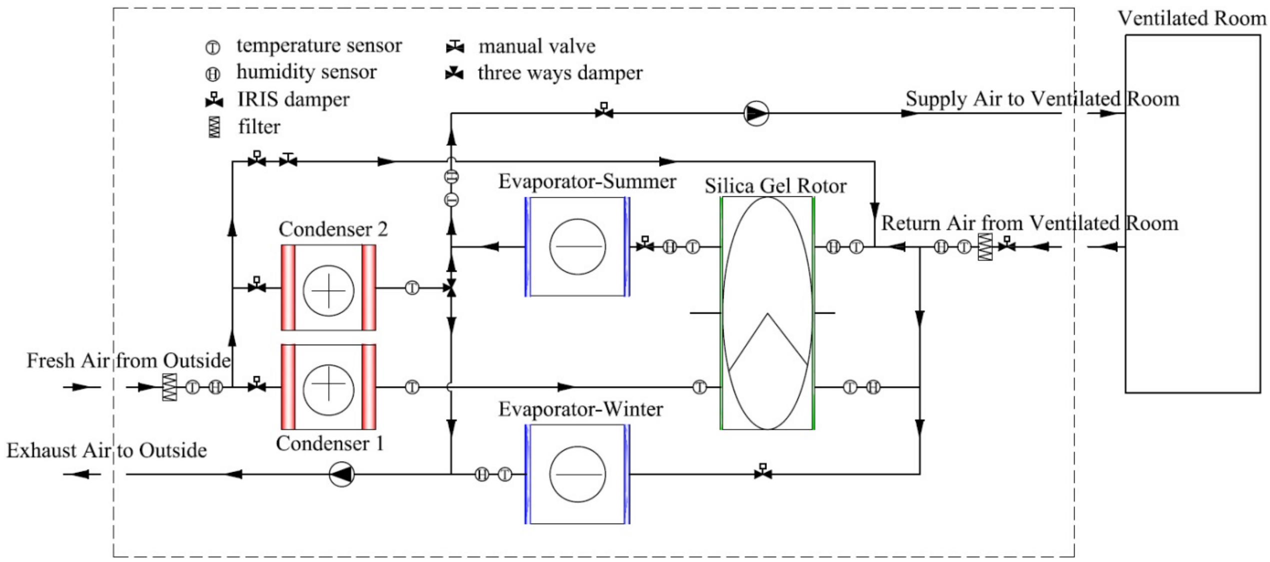
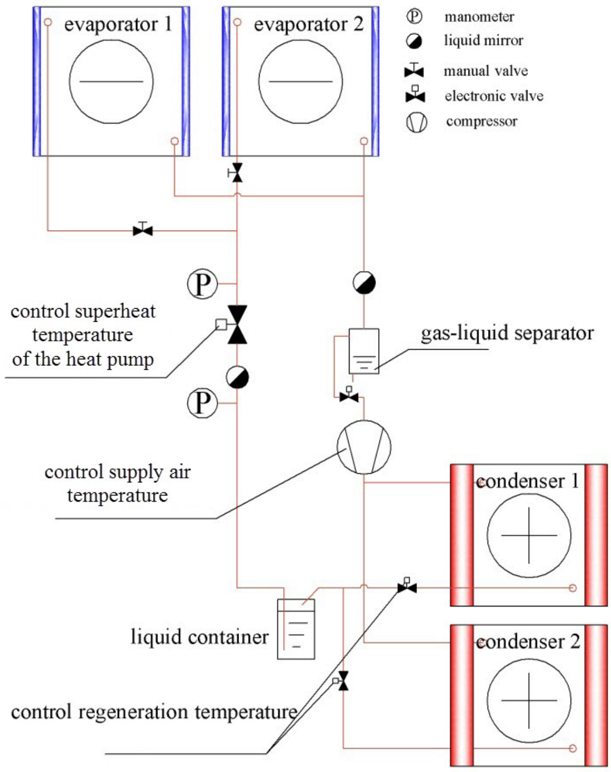


Figure 2.5 Schematic diagram of CAHP designed for both summer and winter modes

2.2.4 Design of the heat pump system

To meet the requirements of heating and cooling energy in the CAHP, a heat pump was designed to match the CAHP air system. The schematic diagram of the heat pump is given in Figure 2.6.



Refrigerant System Schematic Diagram

Figure 2.6 Heat pump designed for CAHP operated in summer and winter modes

As described in the air system of the CAHP, two evaporators are included in the heat pump. One is used in summer mode and the other is used in winter mode. Refrigerant to each evaporator is switched manually by turning on/off the valves connecting to the two evaporators when season changes. There are also two condensers in the heat pump for the CAHP. Condenser 1 is used for heating the air to regenerate the silica gel rotor, and condenser 2 has different functions in two seasons. In winter operation mode, condenser 2 is used to pre-heat outdoor air for ventilation, and in summer operation mode it is used to reject the surplus condensing heat to avoid over-heating the regeneration air in order to reduce the heat feedback to the evaporator and save power consumption. The refrigerant flow rate in each condenser is controlled by regulating the two valves that connected to the two condensers. By controlling the opening of the two valves, the distribution of the refrigerant is controlled and thus the regeneration air temperature can be controlled precisely.

A variable speed compressor was chosen for the heat pump. By controlling the speed of the compressor, the heat pump can adapt to different heating and cooling demand. Thus, indoor thermal environment can be controlled with minimum power consumption. A frequency inverter is used to match the speed variable compressor.

To maintain a constant superheat temperature in the evaporator when changing the speed of the compressor, an electronic expansion valve was selected to be used as the throttle in the heat pump for the CAHP. To ensure that the superheat temperature is independent of the speed of the compressor, the opening of the electronic expansion valve is controlled by the superheat temperature at the refrigerant outlet of the evaporator.

The speed control of the compressor and the control of refrigerant flow distribution in condenser 1 and 2 are the major control strategy of the CAHP. The speed of the compressor is designed to control the ventilation air temperature. The refrigerant flow distribution to the condensers is designed to control the ventilation air humidity ratio.

2.3 Control strategy of the CAHP

The control system of the CAHP was proposed to realize the temperature and humidity independent control of the supply air from the CAHP. The control strategy of CAHP includes two parts. One is regulating the compressor speed to control the supply air temperature. The other is regulating the opening degree of the valves connecting to the two condensers to realize humidity control of the supply air.

2.3.1 Control strategy in summer operation mode

In summer mode, the CAHP is controlled by regulating the speed of the compressor and the distribution of refrigerant between the two condensers. With this control strategy, independent control of ventilation air temperature and humidity can be achieved. The ventilation air temperature is controlled by a frequency invertor to regulate the speed of the compressor. When the measured temperature is higher than the set point, the compressor will speed up and vice versa. The humidity ratio of the ventilation air is controlled by regulating the opening degrees of the two valves connected to the condensers. When the measured humidity ratio is higher than the set point, more refrigerant will be distributed to the condenser 1 that will raise the temperature of regeneration air, and when the measured humidity ratio is lower than the set point, more refrigerant will be distributed to the condenser 2 that will reject more condensing heat from the heat pump and reduce the temperature of regeneration air. In this way, the dehumidification capacity of the silica gel rotor is controlled by the temperature of the regenerating air, and thus the ventilation air humidity ratio can be controlled.

The controls of the compressor and the two valves on the condensers coordinate with each other to fulfill the requirement of temperature and humidity ratio of the air delivered to the room ventilated.

2.3.2 Control strategy in winter operation mode

In winter operation mode, the ventilation air temperature is controlled by regulating the speed of the compressor as well. Different from summer mode, the strategy for controlling the compressor is in an opposite way as that in the summer mode. When the measured temperature of the ventilation air rises, the compressor speed will slow down and vice versa.

To avoid too high regeneration air temperature for the silica gel rotor, the refrigerant distributed to each of the two condensers is controlled by regulating the electrical valves connected to the two condensers. In winter mode, if there is no demand on humidity control of the ventilation air, the regeneration air temperature can be controlled at a constant level to keep a constant air cleaning capacity of the silica gel rotor. When measured regeneration air temperature reduces, more refrigerant will be distributed to the condenser 1 to heat up the regeneration air, and when the measured regeneration air temperature rises, more refrigerant will be distributed to the condenser 2 to pre-heat the outdoor air for ventilation. In the case if humidity control is demanded in winter operation mode of the CAHP, the supply air humidity ratio will be the reference parameter to control the opening degrees of the valves connecting to the condensers. When the measured humidity ratio is higher than the set point, more refrigerant will be distributed to the condenser 1 that will raise the temperature of regeneration air, and when the measured humidity ratio is lower than the set point, more refrigerant will be distributed to the condenser 2 that will pre-heat the outdoor air for ventilation and reduce the temperature of regeneration air.

By this control strategy, the temperature and humidity independent control of the ventilation air can be achieved as well.

2.4 Expected performance of the CAHP

The advantages and disadvantages of the CAHP compared to conventional air conditioning systems are discussed in this section.

2.4.1 Summer operation mode

In summer operation mode, the CAHP is expected to be more energy efficient for ventilation due to the following advantages:

1) With the high efficient air purification capacity of the silica gel rotor, much less outdoor air is used in the CAHP.

2) In summer when dehumidification is necessary, CAHP doesn't need to cool the air down to dew point and reheat it. The thermal process of the ventilation air in CAHP is more energy efficient than conventional dew point dehumidification.

3) Benefited from the non-dew point cooling, the evaporating temperature of the heat pump in the CAHP can be higher than the conventional air source heat pump, and thus increase the COP of the heat pump.

As any new technology, there are still some concerns on the energy performance of the CAHP when comparing it with conventional air conditioning system. The first concern is the regeneration air temperature of the CAHP. The second one is the pressure drop of the silica gel rotor.

1) Higher regeneration temperature is needed when outdoor air humidity ratio increases. Since the dehumidification is conducted by the silica gel rotor, higher dehumidification ability requires higher regeneration temperature. In the CAHP, the regeneration heat is provided by the condensing heat of the heat pump. Higher regeneration temperature will lead to higher condensing temperature and thus results in relative lower COP of the heat pump. In the conventional air conditioning system, dehumidification is done by cooling, and the condensing temperature is not significantly affected by outdoor air humidity ratio, thus its COP doesn't decrease much when outdoor air humidity ratio increase. The energy efficiency of the CAHP is more sensitive to the outdoor air humidity ratio. That could be one disadvantage of the CAHP compared to conventional air conditioning system. If the dehumidification capacity of the silica gel rotor could be reactivated by lower regeneration temperature in high humidity climates, the energy performance of the CAHP could increase significantly.

2) The additional pressure drop caused by the silica gel rotor could increase the power consumption of the ventilation fan. Such an impact should be included when evaluating the energy performance of the CAHP.

All these factors mentioned above can influence the energy performance of the CAHP. Theoretical and experimental studies were then conducted to investigate the energy performance of the CAHP.

2.4.2 Winter operation mode

The higher energy efficiency of the CAHP compared to a conventional heating system in winter mode is attributed to the followings advantages:

1) Much less outdoor air is used in the CAHP due to its high indoor air purification efficiency.
2) Since the evaporator of the heat pump takes heat from the regeneration air (after regeneration) and indoor exhaust air instead of outdoor air, the evaporating temperature can be higher than it is in the conventional air source heat pump. This increases the COP of the heat pump.

The main concerns of the CAHP operating in winter mode are the heat for regeneration air and the additional pressure drop caused by the silica gel rotor.

1) To clean indoor air, some additional heat should be used to regenerate the silica gel rotor. The regeneration heat is from the condensing heat of the heat pump. In winter operation mode, this additional heat can be used directly for space heating in buildings. Even though the additional regeneration heat can transfer to ventilation air through silica gel rotor or be recovered by the evaporator of the heat pump in CAHP, the heat recovering also consumes energy. Therefore, if the air purification can be achieved with less regeneration air, the energy performance of the CAHP in winter modes can be improved significantly. But this needs to be investigated further.
2) Same as the summer operating mode, the pressure drop caused by the silica gel rotor should be considered during the assessment of the energy performance of the CAHP.

Another issue needs to be discussed is the frosting and defrosting of the heat pump. Since the air system in the CAHP is different from the conventional air source heat pump, in winter operation model the outdoor air passes through condensers instead of evaporator to regenerate the silica gel rotor and to heat up the ventilated room, thus there won't be any frost built up in the outdoor air path. On the evaporator side, in winter operation mode, the evaporator is used to recover heat from the exhaust air (rejected indoor air and regeneration air). The temperatures of the two airflows are much higher than the outdoor air temperature. The risk of frosting is usually lower than conventional air source heat pump which uses outdoor air as heat source. However, when outdoor air temperature goes below 0 °C, the evaporating temperature maybe closed to outdoor air temperature in order to recover enough heat to the condenser. Thus the risk of frosting on the evaporator still exists. But the frost's building up speed will be much lower than the conventional air source heat pump due to that the temperature of air flowed to the evaporator is much higher than the outdoor air temperature. The defrosting work could be done by simply stop the heat pump and keep the exhaust fan running. The warm exhaust air will defrost the evaporator. In defrost mode, the regeneration air channel should be closed and leave only the indoor exhaust indoor air pass through the evaporator. This might lead to a slightly decrease of indoor temperature in short time but the defrost process does not consume extra energy. Since the evaporating temperature in the CAHP is much higher than the conventional air source heat pump, the defrosting time in the CAHP could be much shorter than the time used in conventional air source heat pump and the defrosting could also be less frequent.

2.5 Research plan for the CAHP

The research plan for the CAHP consists two parts: theoretical modelling and experimental investigation.

The theoretical modelling work includes a silica gel rotor model for heat, moisture and contaminant transfer prediction and a heat pump model for COP and energy use prediction of the CAHP.

In the theoretical study, a one-dimensional non-steady-state model was used with a set of one-dimensional partial differential equations for the silica gel rotor sub-model, the sub-model was written with MATLAB. The heat pump sub-model was built based on the COP calculation of the heat pump, and it was written with Microsoft-Excel.

The experimental investigation was planned to be mainly focused on the energy performance of the CAHP. Due to that the air cleaning capacity of silica gel rotor has been investigated in the study of Fang et al. [88], the air cleaning effect of CAHP was not investigated by experiment study in this thesis. The validation work for air cleaning effect of CAHP may be done as a future work.

For the experimental study, a prototype unit of the CAHP was developed including its control system. Relevant experimental setup for experimental investigation of the CAHP energy performance was developed as well.

3 Theoretical Study

To predict the performance of the CAHP under different climate conditions, a theoretical model for the CAHP was proposed. As mentioned above, the theoretical model consists of two parts, the silica gel rotor sub-model and the heat pump sub-model.

The silica gel rotor sub-model is used to simulate the heat, moisture and VOCs transfer in the silica gel rotor. With the sub-model, setting the airflow rates, thermal conditions and VOC concentrations of indoor, outdoor air and the target humidity ratio of supply air as input parameters, the temperature and VOC concentrations of dehumidified-cleaned air coming out from the silica gel rotor and the required regeneration air temperature can be calculated. With the simulation results from the silica gel rotor sub-model, setting the target supply air temperature as another input parameter, the energy use of the heat pump could be calculated by the heat pump sub-model. And thus, the air purification and energy performance of the CAHP can be simulated with the two sub-models. All the input and output parameters during the simulation process and their sequence are given in Figure 3.1.

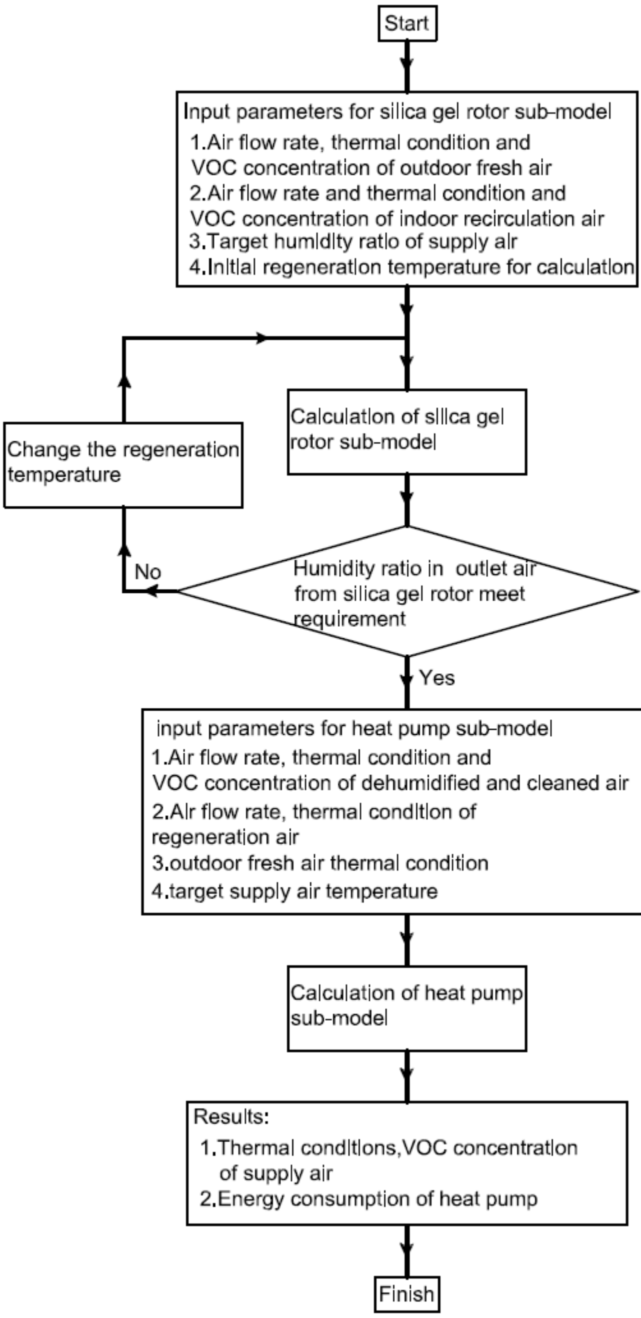


Figure 3.1 Flow chart of the simulation process of CAHP

The theoretical equations, the solving methods of the silica gel rotor sub-model and the heat pump sub-model are given as followings.

3.1 Theoretical equations and solving methods

The theoretical equations, the solving methods of the silica gel rotor sub-model and the heat pump sub-model are given as followings.

3.1.1 Sub-model of silica gel rotor for heat, moisture and VOC transfer

In the previous modeling work, the studies of Nia et al. [105] and Zhang et al. [106] modeled the heat and moisture transfer in silica gel rotors. The study of Nia et al. [105] simulated the combined heat and mass transfer process that occurred in a solid desiccant wheel having balanced regeneration and adsorption air streams with a transient and one-dimensional model. In the modeling work of Zhang et al. [106], the partial differential equations were converted into finite differential equations without uncoupling and were solved with Gauss-Jordan elimination method. The modeling work by Zhang et al. [89] proposed a one-dimensional transient model for the coupled heat, moisture and VOCs transfer in a silica gel rotor. The theoretical study provided a way to analyze the VOC pollutants transfer characters, such as the capability of the silica gel rotor for capturing VOC pollutants and the impact of regeneration temperature on the adsorption process.

From the previous models, the key questions for the sub-model of heat, moisture and VOCs transfer in silica gel rotor were summarized to be: how to select the heat and mass transfer equations and how to define the equilibrium relation of the mass concentration between the air and the adsorbent. In the model presented by this thesis, a one-dimensional non-steady-state model was used with a set of one-dimensional partial differential equations. Since the mass transfer mechanisms of VOCs through silica gel wheel are the same as that of moisture, the one-dimensional non-steady-state model was applied to both VOCs and moisture transfer, but with different adsorption characteristics.

In the theoretical sub-model, the heat conduction and mass diffusion in the air stream and solid materials along the axial direction of the rotor was neglected, the adsorption heat of VOCs was neglected because of its minor proportion compared to that of moisture. The competition of adsorption/desorption among VOCs and that between VOCs and moisture were neglected since the concentrations of VOCs in typical indoor air were much lower than that of moisture, and the adsorbed VOCs occupied only a minor portion of adsorption capacity of the rotor [88]. Since the study of this thesis is based on the air cleaning capacity of silica gel rotor which was found in the studies of Fang et al. [88] and Zhang et al. [89]. Most of the equations for mass transfer including VOCs transfer, moisture transfer and the heat transfer of the silica gel rotor were referred from the study of Zhang et al. [89], but with different size of silica gel rotor, different air flow rates and different thermal conditions of process and regeneration air.

To establish the theoretical model of the silica gel rotor, the physical model of the silica gel rotor is defined in Figure 3.2.

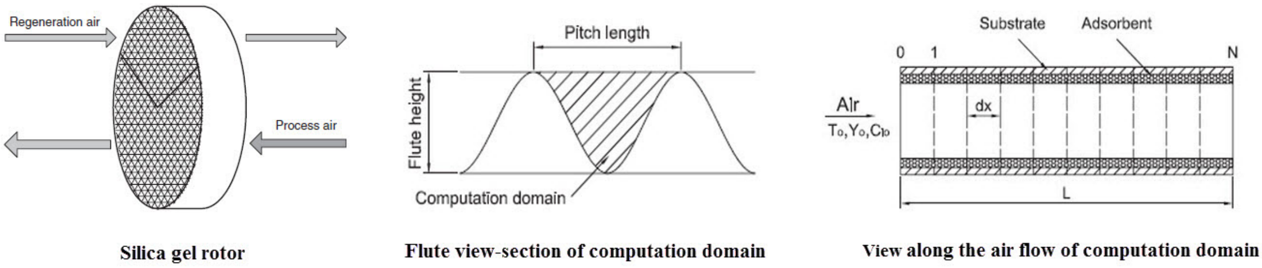


Figure 3.2 Physical model of one silica gel rotor [89]

3.1.1.1 VOCs transfer equations

With the assumptions above, the mass transfer of VOCs in the silica gel rotor was written as following equations.

The VOC mass conservation equation was written as:

$$\frac{\partial C_i}{\partial t} + d\frac{\partial C_i}{\partial x} + \frac{f_d}{A}\frac{\partial G_i}{\partial t} = 0, \quad i = 1, 2, \cdots \tag{1}$$

The VOC mass transfer rate equation was written as:

$$\frac{\partial G_i}{\partial t} = \frac{\alpha_m P}{f_d}(C_i - C_{w,i}), \quad i = 1, 2, \cdots \tag{2}$$

The VOC mass concentration equilibrium equation between air and the adsorbent can be written as:

$$G_i = H_i C_{w,i}, \quad i = 1, 2, \cdots \tag{3}$$

Where,

C_i is the VOC concentration in the air, $\mu g / m^3$;

G_i is the VOC concentration on the adsorbent, $\mu g / g$;

$C_{w,i}$ is the VOC concentration in the air when it reaches equilibrium state with the adsorbent, $\mu g / m^3$;

d is the air speed in the flute section of the silica gel rotor, m / s;

f_d is the weight of the adsorbent per meter in one flute, kg / m;

A is the cross-sectional area of one flute section in the silica gel rotor, m^2;

t is the time, s;

x is the length, m;

α_m is the convective mass transfer coefficients, m / s;

P is the perimeter of the flute section, m;

H_i is Henry's law constant of the mass.

The coefficient of convective mass transfer α_m is derived by analogy to the coefficient of convective heat transfer assuming Lewis number to be unity:

$$\alpha_m = \frac{\alpha}{\rho_a c_{pa}} \tag{4}$$

The coefficient of convective heat transfer of the air in the flute is given by the study of Hougen and Marshall [107].

$$\alpha = 0.683 \rho_a d \,\mathrm{Re}^{-0.51} \tag{5}$$

Where,

α is the heat transfer coefficient, $W/(m^2 \cdot K)$;

ρ_a is the density of the air, kg/m^3;

c_{pa} is the specific heat of the air, $kJ/(kg \cdot K)$;

Re is the Reynolds number.

The Henry's law constant of the VOC for adsorption was calculated with the equation below.

$$H_i = a_i e^{b_i T_{k-adsorption}} G_{i,\max} \tag{6}$$

Where,

$T_{k-adsorbentp\,tion}$ is the absolute temperature for adsorption, K;

a_i, b_i are the adsorption constant for VOC chemicals;

$G_{\max}$ is the maximum content of VOC on adsorbent. $\mu g/g$.

The equation 1, equation 2 and equation 3 form the VOC chemicals transfer between the air and the adsorbent. The equation 1 and equation 2 can be further written into apparent explicit finite differential equations as:

$$C_{i,k}^{n+1} = \begin{cases} C_{i,k}^{n} - \dfrac{d\Delta t}{\Delta x}(C_{i,k}^{n} - C_{i,0}) - \dfrac{f_d}{A}(G_{i,k}^{n+1} - G_{i,k}^{n}), \quad k = 1 \\ C_{i,k}^{n} - \dfrac{d\Delta t}{\Delta x}(C_{i,k}^{n} - C_{i,k-1}^{n}) - \dfrac{f_d}{A}(G_{i,k}^{n+1} - G_{i,k}^{n}), \ when\ k = 2,\cdots N \end{cases} \tag{7}$$

$$G_{i,k}^{n+1} = G_{i,k}^{n} + \frac{\alpha_m P \Delta t}{f_d}(C_{i,k}^{n} - C_{w,i,k}^{n}), \quad k = 1,\cdots N \tag{8}$$

3.1.1.2 Heat and moisture transfer equations

The heat and moisture transfer model was written based on the following equations.

The moisture conservation equation was written as:

$$\frac{\partial Y}{\partial t} + d\frac{\partial Y}{\partial x} + w_1\frac{\partial W}{\partial t} = 0 \tag{9}$$

Considering the condition in which the adsorbent is saturated with moisture, the equation 9 was further written to be：

$$\frac{\partial W}{\partial t} = \begin{cases} w_2(Y - Y_w)\ , \ when\ W < W_{\max}, or\ when W = W_{\max}\ and\ Y < Y_w \\ 0, \ when\ W = W_{\max}\ and\ Y > Y_w \end{cases} \tag{10}$$

The energy conservation equation was written as:

$$\frac{\partial T_{k-air}}{\partial t} + d\frac{\partial T_{k-air}}{\partial x} + w_3\frac{\partial T_{k-adsorbent}}{\partial t} = w_4\frac{\partial W}{\partial t} \tag{11}$$

The heat transfer equation on the adsorbent was written as:

$$\frac{\partial T_{k-adsorbent}}{\partial t} + w_5(T_{k-adsorbent} - T_{k-air}) + w_6(Y_w - Y) = 0 \tag{12}$$

Where,

Y is the moisture content in the air, g/kg;

W is the moisture adsorbed on the adsorbent, g/kg;

W_{max} is the maximum moisture adsorbed on the adsorbent, g/kg;

Y_W is the moisture content in the air which reached equilibrium state with adsorbent, g/kg;

T_{k-air} is the absolute temperature of air, K;

$T_{k-adsorbent}$ is the absolute temperature of adsorbent, K.

The coefficient w_1 - w_6 can be written as followings:

$$w_1 = \frac{f_d}{A\rho_a}$$

$$w_2 = \frac{K_Y P}{f_d}$$

$$w_3 = \frac{f_d(c_{pd} + Wc_{pl}) + f_m c_{pm}}{A\rho_a(c_{pa} + Yc_{pv})}$$

$$w_4 = \frac{K_Y Pq}{A\rho_a(c_{pa} + Yc_{pv})}$$

$$w_5 = \frac{\alpha P}{f_d(c_{pd} + Wc_{pl}) + f_m c_{pm}}$$

$$w_6 = \frac{\alpha P}{f_d(c_{pd} + Wc_{pl}) + f_m c_{pm}}$$

Where,

K_Y is the mass transfer coefficient, $kg/m^2/s$;

c_{pd} is the specific heat at constant pressure of the adsorbent, $J/kg/k$;

c_{pl} is the specific heat at constant pressure of moisture in the adsorbent, $J/kg/k$;

c_{pm} is the specific heat at constant pressure of substrate material, $J/kg/k$;

c_{pv} is the specific heat at constant pressure of moisture in the air, $J/kg/k$;

f_m is the weight of substrate material in the direction along the airflow, kg/m;

q is the adsorption heat of moisture, J/kg.

Referred to the study of San [108], the adsorption heat of moisture could be calculated with equation 13.

$$q = h_v(1.0 + 0.2843e^{-10.28W}) \tag{13}$$

Where,

h_v is the latent heat of vaporization of water, J/kg.

According to the study of Zhang et al. [106], the saturated vapor pressure in the air which achieved equilibrium with the adsorbent can be calculated with following equation.

$$\ln p_{ws} = 23.196 - \frac{3816.44}{T_w - 46.13} \tag{14}$$

The equations 10-12 form the integral heat and moisture transfer in the silica gel rotor. During the simulation process, all of the equations were written to be apparent explicit finite differential equations as followings.

$$W_k^{n+1} = \begin{cases} W_k^n + w_2 \Delta t (Y - Y_w) \ , \ \ when\, W < W_{\max}, \, or \ W = W_{\max} \ and \ Y < Y_w \\ W_k^n , \ \ when\, W = W_{\max} \ and \ Y > Y_w \end{cases} \tag{15}$$

$$Y_1^{n+1} = Y_1^n - \frac{d\Delta t}{\Delta x}(Y_1^n - Y_0) - w_1(W_1^{n+1} - W_1^n) \tag{16}$$

$$Y_k^{n+1} = Y_k^n - \frac{d\Delta t}{\Delta x}(Y_k^n - Y_{k-1}^n) - w_1(W_k^{n+1} - W_k^n), \quad k = 2, \cdots N \tag{17}$$

$$T_{w,k}^{n+1} = T_{w,k}^n - w_5 \Delta t (T_{w,k}^n - T_k^n) - w_6 \Delta t (Y_{w,k}^n - Y_k^n) \tag{18}$$

$$T_1^{n+1} = T_1^n - \frac{d\Delta t}{\Delta x}(T_1^n - T_0) - w_3(T_{w,1}^{n+1} - T_{w,1}^n) + w_4(W_1^{n+1} - W_1^n) \tag{19}$$

$$T_k^{n+1} = T_k^n - \frac{d\Delta t}{\Delta x}(T_k^n - T_{k-1}^n) - w_3(T_{w,k}^{n+1} - T_{w,k}^n) + w_4(W_k^{n+1} - W_k^n) \tag{20}$$

The heat, moisture and VOC transfer sub-model in the silica gel rotor was coded in MATLAB to solve the sub-model.

3.1.2 Sub-model for heat pump energy performance predication

The key issues for sub-model of heat pump energy use simulation were to get the coefficient of performance (COP) of the heat pump and the heating, cooling demand of the CAHP.

The heating, cooling demand could be calculated with equation 21 and equation 22:

$$Q_{heating} = V_{reg} * 3.6 * (T_{reg} - T_o) * \rho_{reg} * C_{reg} \tag{21}$$

$$Q_{cooling} = V_{pro} * (T_{\sup} - T_{deh}) * \rho_{pro} * C_{pro} \tag{22}$$

Where,

$Q_{heating}$, $Q_{cooling}$ are the heating and cooling demands for the heat pump, kW;

V_{reg}, V_{pro} are the airflow rates of the regeneration air and process air, m^3/h;

T_{reg}, $T_{outdoor}$, T_{deh}, $T_{\sup}$ are the temperatures of the regeneration air, outdoor air, dehumidified air and supply air, ℃;

ρ_{reg}, ρ_{pro} are the densities of the regeneration air, process air, kg/m^3;

C_{reg}, C_{pro} are the specific heat at constant pressure of regeneration air, process air, $kJ/kg/k$

The COP for cooling of the heat pump could be calculated with equation 23.

$$COP = (h_1 - h_4)/(h_2' - h_1) \tag{23}$$

Where,

h_1 is the refrigerant enthalpy at the inlet point to the compressor, kJ/kg;

h_4 is the refrigerant enthalpy at the inlet point to the evaporator, kJ/kg;

h_2' is the refrigerant enthalpy at the outlet point from the compressor, kJ/kg.

h_2' can be calculated with the equation below.

$$h_2' = h_1 + (h_2 - h_1)/\eta_{is} \tag{24}$$

Where,

h_2 is the enthalpy of refrigerant at condensing pressure and with equal isentropic as the refrigerant at h_1, kJ/kg;

η_{is} is the isentropic efficiency of the compressor.

With the COP of the heat pump and the heating, cooling demand for the heat pump, the energy use consumption of the heat pump in CAHP could be calculated with the following equation 25.

$$E = Max(Q_h/(COP+1), Q_c/COP) \tag{25}$$

Where,

E is the energy use of the heat pump, kW .

The heat pump sub-model was coded within Microsoft-Excel.

3.2 Model validation

The validation work for the theoretical model of the CAHP also includes two parts, validation of the air cleaning effect and validation of the energy performance. Both parts were validated by experimental measurements. The air cleaning effect simulation was validated by the VOCs concentration measurements from the air upstream and downstream of the silica gel rotor. The energy performance simulation was validated by the energy use measurements of the heat pump in the CAHP prototype unit.

3.2.1 Air cleaning validation

The sub-model for air cleaning effect was validated by the experimental results from the study of Fang et al. [88] and Zhang et al. [89]. Toluene and 1,2-dichloroethane were selected as chemicals which represented indoor air borne contaminants. During the validation, toluene and 1,2-dichloroethane were dosed with different rates to simulate air with different pollutant levels. The geometrical parameters and operating conditions of the silica gel rotor for validation are given in Table 3.2.

Firstly, the adsorption characteristics of selected chemicals (toluene and 1,2-dichloroethane) were determined. Referred to the study of Hines and Ghosh [84], the a,b and $G_{\max}$ for toluene and 1,2-dichloroethaneon the silica gel are given in the Table 3.1.

Table 3.1 Parameters of the adsorption characteristics of toluene and 1,2-dichloroethane on the adsorbent

Chemicals	a (m^3/g)	b (1/T)	G_{max} (mg/g)
Toluene	6.45×108	– 0.0718	294
1,2-dichloroethane	1.92×105	– 0.0496	455

For the experimental measurements, the concentrations of VOC chemicals were measured by PTR-MS. PTR-MS is a chemical ionization mass spectrometry technique that allows for on-line measurements of individual VOC at ppt levels. VOC chemicals are ionized via proton transfer reactions from H_3O^+ primary ions and mass spectrometry detected one mass unit higher than the neutral compound. PTR-MS detects all VOCs with a proton affinity higher than water. This includes all polyatomic volatile organic molecules with the exception of small aliphatic and cyclic hydrocarbons. The response of the PTR-MS instrument is linear in the ppt-to-ppm range. A major limitation of the PTR-MS technique is that compounds with the same molecular weight (i.e. isobaric and isomeric compounds) cannot be resolved. A commercial high-sensitivity PTR-MS instrument (PTR-MS-FDT-s, Ionicon GmbH, Innsbruck, Austria) was used for the measurements in the presented study. Detection limits of the PTR-MS for the VOCs measured were in the range of 20-100 ppt. Calibration factors for toluene were obtained using a dynamically diluted calibration gas standard (Apel-Riemer Environmental Inc., Denver, CO, USA) containing 1 ppm of toluene. The calibration factor for 1,2-dichloroethane was calculated using a procedure outlined in Sprung et al. [109]. The relative uncertainties for toluene and 1,2-dichloroethane measurements are ±20% and ±50% respectively.

For the experimental validation, the uncertainty of measured VOC chemical removal could be analyzed using Type-B uncertainty evaluations. The VOC removed by the silica gel rotor could be calculated with the equation below.

$$C_{i-removed} = C_{i-inlet} - C_{i-outlet} \tag{26}$$

Where, $C_{i-removed}$ is the VOC removed by the silica gel rotor, $\mu g / m^3$;

$C_{i-inlet}$, $C_{i-outlet}$ are the VOC concentration in the inlet air and outlet air, $\mu g / m^3$.

Thus, the uncertainty of VOC removal measured is based on the accuracy data of the PTR-MS instrument. The relative standard uncertainty of $C_{i-removed}$ (type-B) can be given by

$$u'(C_{i-removed}) = \frac{u(C_{i-removed})}{C_{i-removed}} = \sqrt{u'^2(C_{i-inlet}) + u'^2(C_{i-outlet})} \tag{27}$$

Where u' is the relative uncertainty;

u is the uncertainty.

With the equation 27, the relative uncertainty for the toluene and 1,2-dichloroethane measurements were calculated to be ±28% and ±71% respectively.

Table 3.2 Properties and operating conditions of the silica gel rotor for air cleaning validation

Geometrical parameters	
Radius (m) 0.225	Flute height (mm) 1.9
Thickness (m) 0.3	Pitch length (mm) 3.7
Sectional angle for regeneration air (°) 135	Total thickness of adsorbent and substrate (mm)0.15
Sectional angle for process air (°) 225	
Thermodynamic properties	
Specific heat of silica gel [J/(kg*°C)] 920	Specific heat of substrate [J/(kg*°C)] 900
Volume weight (kg/m^3) 200	Weight percentage of silica gel (%) 80
Operating parameters	
Revolution speed (rounds/h) 10	
Flow rate of process air (L/s) 160	Flow rate of regeneration air (L/s) 80
Humidity of process air (g/kg) 10	Temperature of process air (°C) 23
Humidity of regeneration air (g/kg) 10	Temperature of regeneration air (°C) 100

The simulation and experimental results are given and compared in Table 3.3 and Figure 3.3.

Table 3.3 Measured and calculated VOCs removing effect of silica gel rotor and the deviation between the measured and calculated results

Concentrations of chemicals in the inlet air to silica gel rotor (µg/m^3)		Chemicals removed by silica gel rotor (µg/m^3)		Deviation (%)
		measured values	calculated values	
Toluene	167	164.6	136	-17.38
	369	364	337	-7.42
	552	544.3	515	-5.38
	867	854	827	-3.16
	467	459	433	-5.66
1,2-dichloroethane	109	106	100.4	-5.28
	190	187	175	-6.42

Concentrations of chemicals in the inlet air to silica gel rotor (μg/m^3)	Chemicals removed by silica gel rotor (μg/m^3)		Deviation (%)
	measured values	calculated values	
289	282	267	-5.32
346	337	311	-7.72

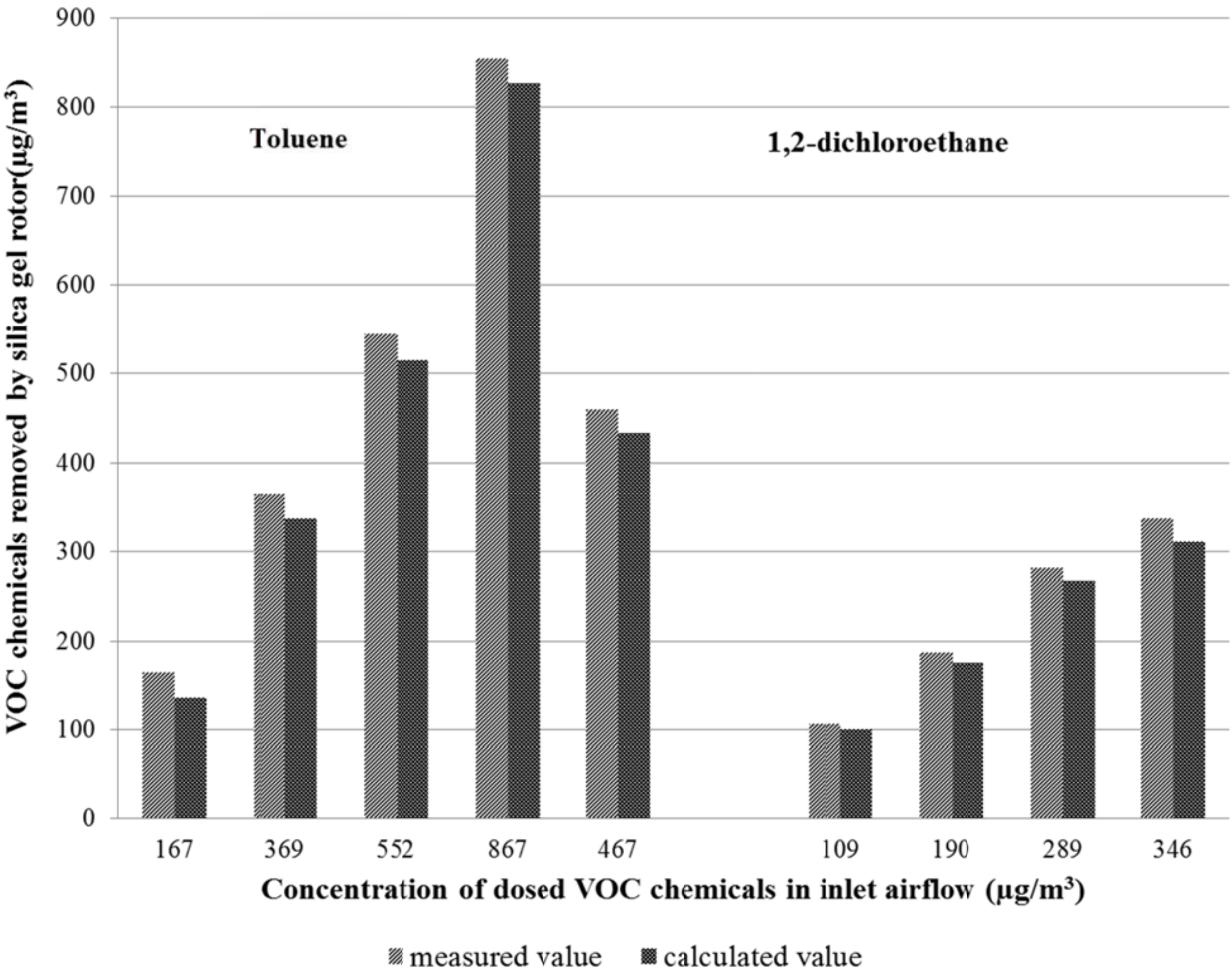


Figure 3.3 Comparisons of measured and calculated VOCs removing effect of silica gel rotor

The simulation results of chemicals cleaning effect were quite closed to the experimental results. The maximum deviation between the calculated and measured values of chemicals removed is 17.38% in the validation cases.

3.2.2 Energy performance validation

The theoretical model for energy performance simulation of the CAHP was validated by several experimental studies [110]. Details of the experimental study will be given in next section. The key input parameters during the simulation and experimental studies are given in Table 3.4. The refrigerant R134a was used in the heat pump of the numerical model and the experimental system of CAHP.

Table 3.4 Properties and operating conditions of the CAHP for energy performance validation

Airflow rates				
Indoor recirculation air (L/s) 190		Outdoor air (L/s) 60		Regeneration air (L/s) 125
Indoor thermal conditions				
Indoor air temperature (°C) 25		Indoor air humidity ratio (g/kg) 9.85		
Outdoor thermal conditions and air delivered to ventilated room				
Case No.	Outdoor air		Air delivered to ventilated room	
	Temperature (°C)	Humidity ratio (g/kg)	Temperature (°C)	Humidity ratio (g/kg)
1	25.5	11.9	18.92	8.91
2	30.5	12.7	18.24	8.91
3	33.0	13.6	17.89	8.91
4	36.1	15.1	17.93	9.17
5	38.0	17.1	17.65	9.17

The simulation and experimental results for all the cases are given and compared in Figure 3.4.

Table 3.5 Comparison of simulated results and experimental measured results of COP and power consumption of CAHP

Case No.	Simulation results of COP	Power consumption simulated (kW)	Measurement results of COP	Power consumption measured (kW)
1	4.61	0.83	5.02	0.76
2	3.69	1.26	4.00	1.16
3	3.27	1.51	3.28	1.51
4	3.06	1.55	2.85	1.66
5	2.91	1.68	2.50	1.96

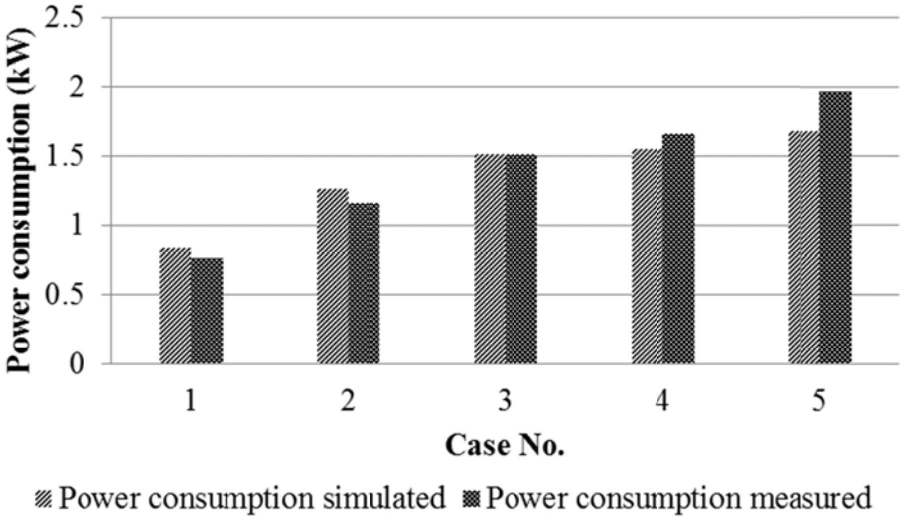


Figure 3.4 Comparisons of simulated and experimental measured CAHP power consumption

The simulation results of the energy use of the CAHP were closely around the experimental results. The maximum difference of the energy use between the simulation results and the experimental results was 14.1%. The energy use from the simulation results were higher than that of experimental results when the compressor was operated with relative low thermal loads, and they were lower than the experimental results when the compressor was operated with high thermal loads. This could be attributed to that the isentropic efficiency of the compressor varied when the thermal loads changed. Higher thermal load would lead to higher pressure difference between the refrigerant inlet and outlet points of the compressor, and thus led to lower entropy efficiency.

3.3 Theoretical investigation on influence of outdoor air temperature and humidity ratio to CAHP performance

The validating results showed that the theoretical model could be used to predict the air cleaning effect and energy performance of the CAHP. The effects of outdoor thermal environments on air cleaning and energy performances of the CAHP were then analyzed and discussed with the model.

In this section, a theoretical analysis was conducted to investigate the energy and air cleaning performance of the CAHP under different outdoor thermal climates with the numerical model. Since the CAHP was controlled with fixed supply air temperature and humidity ratio, the outdoor air thermal condition especially the humidity ratio will influence the regeneration air temperature of the silica gel rotor. The rotor will have different air cleaning efficiency under different regeneration air temperature. Thus, the outdoor air thermal climates affect not only the energy performance of CAHP but also the air cleaning effect of the silica gel rotor. The objective of the discussion was to reveal the energy and air cleaning performance of the CAHP under varied outdoor thermal climates.

During the analyzing, the outdoor temperatures were set to be from 26°C to 36°C with a step of 2°C, and the outdoor humidity ratios were set to be from 13g/kg to 21g/kg with a step of 2g/kg. Indoor air thermal environments were set to be at 25°C and 9.85g/kg of moisture content. The concentrations of toluene and 1,2-dichloroethanein indoor air were set to be 200µg/m^3 and 150µg/m^3. The concentrations of toluene and 1,2-dichloroethane in outdoor air were set to be 11µg/m^3 and 1µg/m^3. The indoor recirculation airflow, outdoor airflow supplied to the room and the regeneration airflow were set to be 190L/s, 60L/s, 125L/s respectively. The thermal conditions of the air delivered to the ventilated room were controlled at 18°C and 8.91g/kg of moisture content.

Inputting all the parameters above to the theoretical model of CAHP, the air cleaning effect and energy performance of CAHP could be simulated and analyzed. During the energy performance simulation, refrigerant R134a was used in the sub-model of heat pump. The results are given in Figure 3.5, Figure 3.6, Figure 3.7 and Figure 3.8.

The energy uses of the CAHP under different outdoor thermal climates are shown in Figure 3.5.

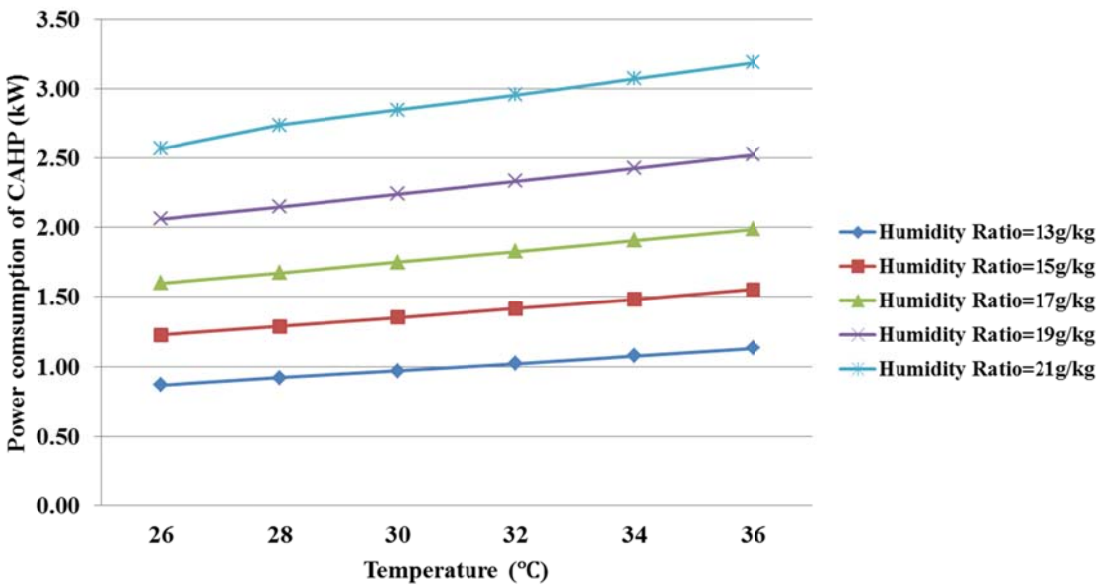


Figure 3.5 Simulated power consumption of CAHP under different outdoor thermal climates

Figure 3.5 shows that the energy use of CAHP increases with the increase of outdoor temperature and humidity ratio. This could be due to that the increase of outdoor temperature and humidity ratio lead to larger thermal load for the CAHP and relative lower COP of the heat pump. The relations of heat pump COP in the CAHP with the outdoor thermal climate conditions are given in Figure 3.6.

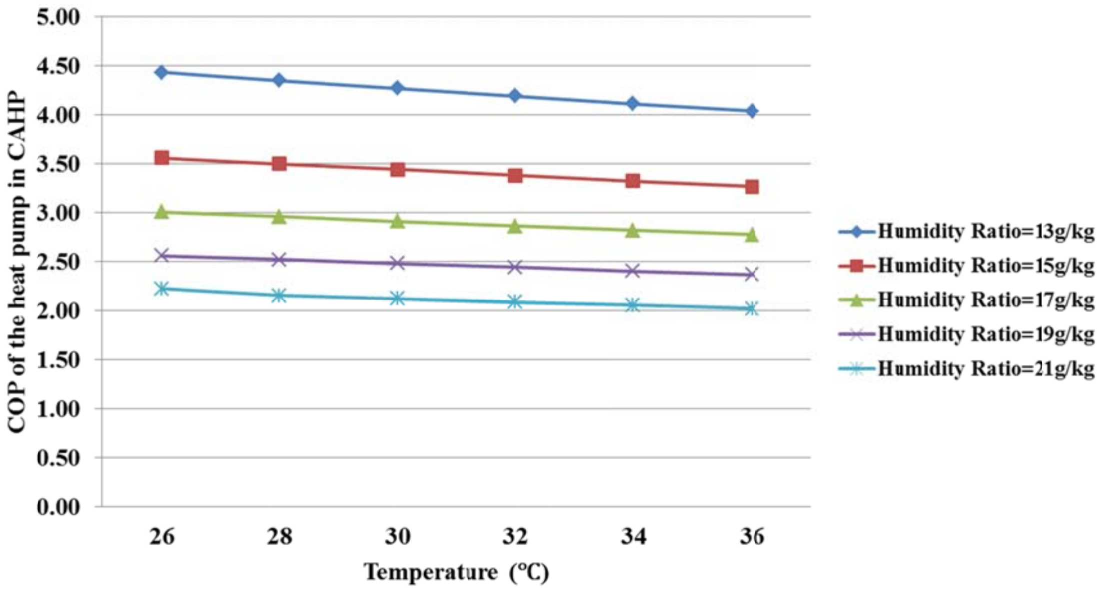


Figure 3.6 Relations between COP of the heat pump in CAHP and outdoor thermal climates

The increase of outdoor temperature and humidity ratio will lead to COP decrease of the heat pump in CAHP. This is because higher regeneration temperature is required when outdoor temperature and humidity ratio increase. With constant supply air temperature from the CAHP, higher regeneration temperature will lead to higher compression ratio of the compressor and lower COP of the heat pump. Both Figure 3.5 and Figure 3.6 show that the energy use of CAHP is more sensitive to the humidity ratio rather than the temperature of outdoor air. This is mainly because the regeneration temperature of the desiccant wheel is much more affected by outdoor air humidity than the temperature. The desiccant wheel requires much higher reactivation temperature at higher outdoor air humidity, which leads to increase of condensing temperature and decrease of COP in the heat pump.

The chemical concentrations in the supply air from the CAHP and their relations with outdoor thermal climates are given in Figure 3.7 and Figure 3.8.

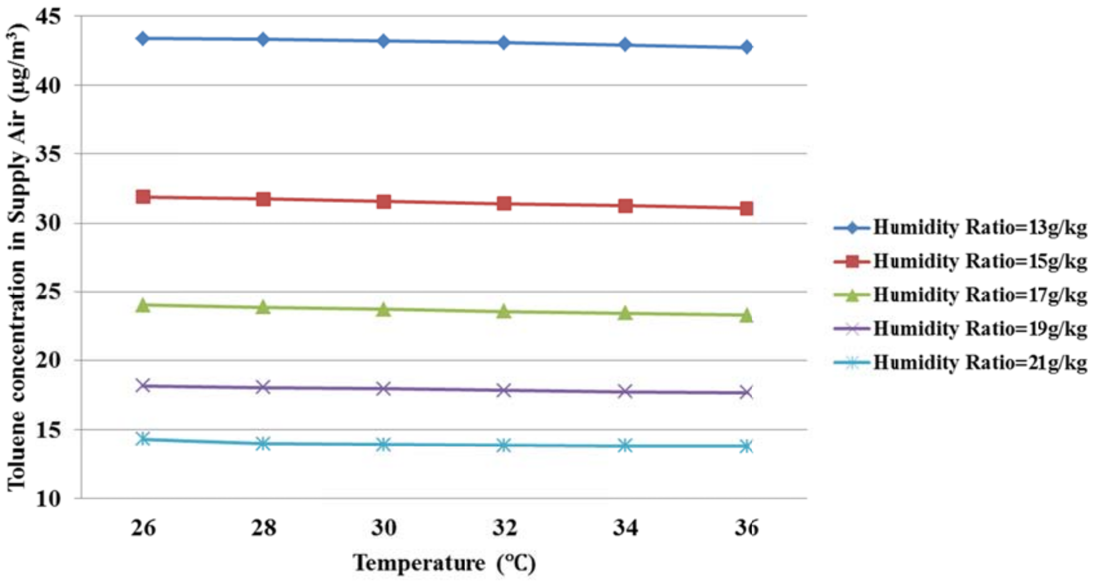


Figure 3.7 Toluene concentrations in the air delivered to ventilated room from CAHP and their relations with outdoor thermal climates

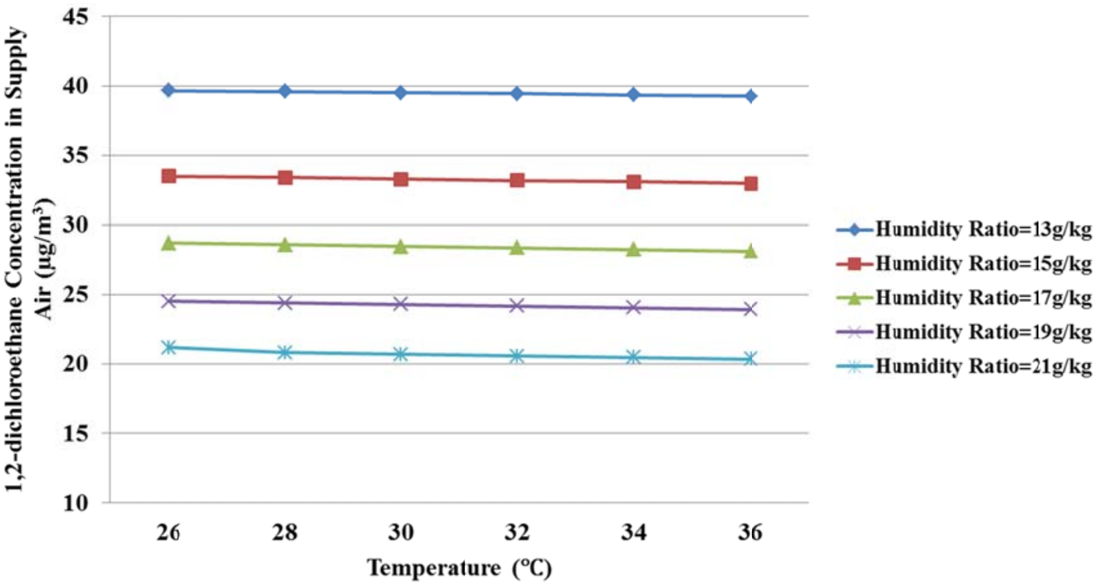


Figure 3.8 Concentrations of 1,2-dichloroethane in the air delivered to ventilated room from CAHP and their relations with outdoor thermal climates

The concentrations of dosed chemicals in the air delivered to ventilated room decrease significantly with increase of outdoor humidity ratio. With the increase of outdoor temperature, the concentrations of chemicals in the air delivered to ventilated room also decreased, but with a lower conspicuousness. This could be due to that the CAHP is controlled by supplying air with a relative fixed temperature and humidity ratio. Higher regeneration temperature is needed when outdoor temperature and humidity ratio increase, and higher regeneration temperature leads to higher air cleaning efficiency of the silica gel rotor.

Though the air cleaning effect was affected by outdoor thermal climate conditions, the lowest chemical cleaning efficiency of the CAHP was 65% in all the cases analyzed. In the analyzed cases, the lowest efficiency happened in the case which has outdoor temperature 26°C and 13g/kg of moisture content and for 1,2-dichloroethane cleaning. It means the CAHP can clean and remove VOC pollutants with an efficiency of at least 65%.

Within the outdoor thermal climates investigated, the COP of the heat pump in CAHP was in the range of 2.02-4.43. It is closed to normal heat pump. But combining with its strong air cleaning ability, the CAHP can be an energy efficient choice for ventilation system.

4 Experimental Study

Besides the theoretical modeling of the CAHP, a real prototype unit of the CAHP was needed for evaluating the performance of the ventilation system using CAHP technology. The development of a real prototype CAHP could also help to discover the practical problems and solutions in the design, development and optimizing the CAHP. In the study of this thesis, a prototype unit of the CAHP was developed. The capacity of the prototype CAHP was designed based on the ventilation requirement of a test room in Building 402 of Technical University of Denmark.

Experimental studies were conducted to investigate the performance of the CAHP operated in different climate conditions. So far, the experimental studies were mainly focused on the thermal process and energy performance of the CAHP. Thus, in this thesis, the experimental study will give the energy performance of the CAHP. The air cleaning performance of the whole CAHP in different outdoor thermal climate conditions will be measured in the future experiment.

4.1 The test room

The heating and cooling capacity of the CAHP was determined by the thermal load of the test room. Before the design of the prototype CAHP, an information collection work was conducted to get the geometry, thermal-physical properties and interior heat load of the test room, and all the information were used to calculate the hygrothermal load and ventilation rate of the test room. Geometry and thermal-physical properties of the test room are list in Table 4.1. Indoor thermal environment set-points and interior heat conduction in the test room are listed in Table 4.2.

Table 4.1 Geometry and thermal-physical properties of the test room

Parts of Envelope	Area (m^2)	Heat transfer coefficient (W/m^2*K)
Roof	72	0.20
External wall	21.6	0.25
External window	14.4	1.50
Interior walls and doors	72	2.00
Interior floor	72	0.20

Table 4.2 Indoor thermal environment set-points and interior heat conduction

Parameters		Value
Indoor temperature (°C)	Summer	25

Parameters		Value
	Winter	22
Indoor relative humidity (%)	Summer	50
	Winter	--
Occupants (persons)		10-15
Heat from lights (W)		43
Heat from computer (W)		210
Heat from projector (W)		250

Based on the thermal-physical information of the test room, the sensible and latent heat load of the test room in different outdoor thermal environments can be calculated. For example, taking the extremely hot climate in Copenhagen (32.1°C /38.6%RH) as reference outdoor thermal environment, the sensible cooling load of the test room was calculated to be 2.12kW, and the dehumidification load was calculated to be 1.02kg/h. Setting the temperature difference between supply air and indoor air at 7°C [111], the supply airflow rate was calculated to be 250L/s, and the supply air thermal condition was calculated to be 18°C with 8.91g/kg of moisture content. For the CAHP operation in other summer outdoor thermal climates, the supply airflow rate for the test room will be fixed at 250L/s but with variable supply air temperature and humidity ratio, i.e. a constant air volume air-conditioning system (CAV).

For winter operation mode, the same supply airflow rate was used as that in the summer mode. The supply air temperature and humidity ratio were also variable in different outdoor thermal conditions and different climate zones.

4.2 CAHP prototype development

For the test room, a prototype unit of the CAHP was designed and developed. The design and development of the CAHP prototype was based on the principles stated in section 2.1 “Principle of the CAHP”. All the components selected for the CAHP are introduced as followings.

4.2.1 Silica gel rotor

Silica gel rotor is the most important component of CAHP that functions as a core for indoor air cleaning and dehumidification. A silica gel rotor with dimensions of 454mm in diameter and 200mm in depth was selected to be built in the prototype CAHP. The silica gel rotor selected had a rated process airflow rate of 222L/s (800m^3/h) which was designed for an electrical power based dehumidifier (MLT 800) produced by “Munters A/S”. In the prototype CAHP, the rotor was used to process 250L/s of airflow. The main physical properties of the silica gel rotor used are given in Table 4.3.

Table 4.3 Physical properties of the silica gel rotor used in the prototype CAHP

<table>
<tr><th>Desiccant</th><th>HPS</th></tr>
<tr><td colspan="2">Geometrical parameters</td></tr>
<tr><td>Radius (m) 0.227</td><td>Flute height (mm) 1.9</td></tr>
<tr><td>Thickness (m) 0.2</td><td>Pitch length (mm) 3.7</td></tr>
<tr><td>Sectional angle for regeneration air (°) 120</td><td rowspan="2">Total thickness of adsorbent and substrate (mm)0.15</td></tr>
<tr><td>Sectional angle for process air (°) 240</td></tr>
<tr><td colspan="2">Thermodynamic properties</td></tr>
<tr><td>Specific heat of silica gel [J/(kg*°C)] 920</td><td>Specific heat of substrate [J/(kg*°C)] 900</td></tr>
<tr><td>Volume weight (kg/m^3) 200</td><td>Weight percentage of silica gel (%) 80</td></tr>
<tr><td colspan="2">Operating parameters</td></tr>
<tr><td colspan="2">Revolution speed (rounds/h) 11.6</td></tr>
</table>

The original design of the dehumidifier divided the silica gel rotor into three sections, i.e. process section, regeneration section and purge section. The application of this rotor in the CAHP merged the regeneration section and the purge section into one section for regeneration. The flow directions of process air and regeneration air in the silica gel rotor of CAHP are shown in Figure 4.1. The rotation speed of the rotor was 11.6 rounds per hour.

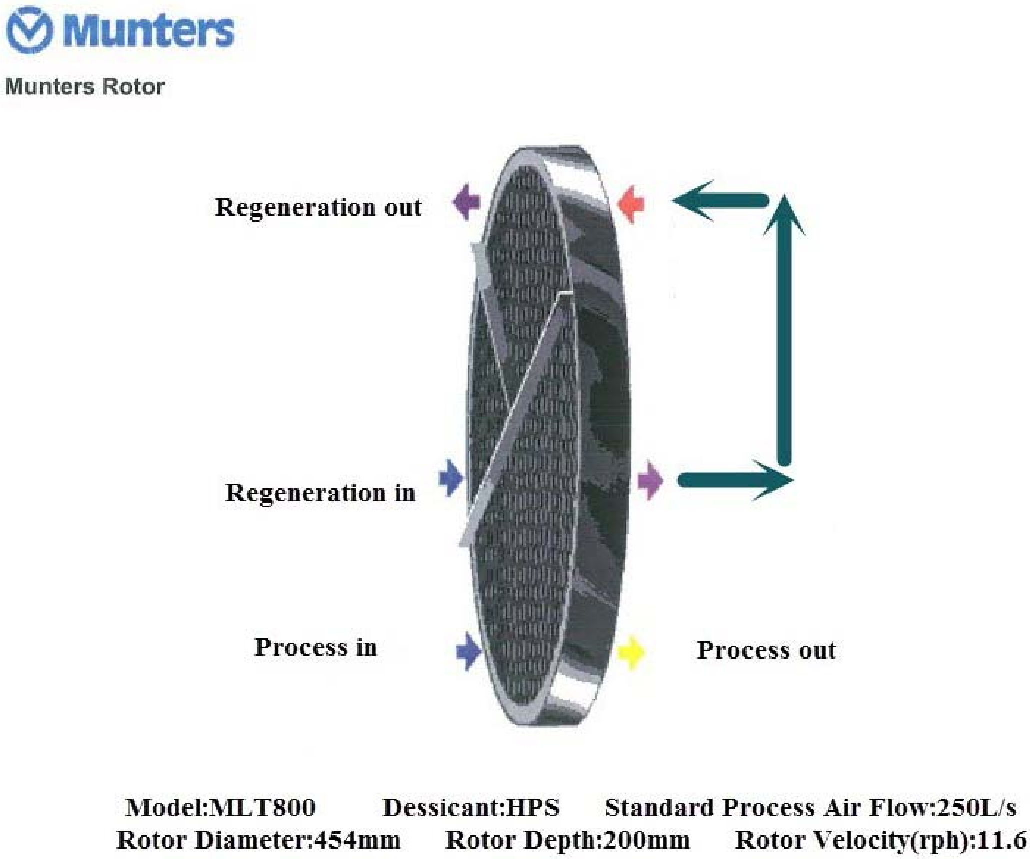


Figure 4.1 Silica gel rotor selected for prototype CAHP [112]

The theoretical model proposed in Chapter 3 was used to calculate the required regeneration temperature for dehumidification at different climate conditions. For Copenhagen, the humidity ratio of supply air was designed to be 8.91g/kg, and the requirement for regeneration temperature is calculated to be 44°C at the extremely hot climate condition of Copenhagen (32.1°C /38.6%RH). .

For the hot and humid climate zones e.g. in south Europe or south Asia, the regeneration temperature could be much higher than 44°C. One extremely hot and humid climate condition used in the experiment for testing the energy performance of the CAHP was the summer climate condition in Colombo (Capital of Sri Lanka) where the required regeneration air temperature was calculated to be 64°C.

4.2.2 Refrigerant

In summer operation mode, with the calculated supply airflow rate and humidity ratio, the regeneration air temperature for dehumidification under different outdoor climate conditions could be obtained, and they were varied from 44°C (for mild cold climate) to 64°C (for extremely hot and humid climate). To heat up the regeneration air to 64°C, the condensing temperature of the heat pump was designed to be 70°C. The evaporating temperature of the heat pump was determined mainly by the requirement of air temperature delivered to the ventilated room. Since the supply air temperature to the room was designed to be around 18°C (7°Clower than indoor air temperature), the evaporating temperature was designed to be 15°C to cool down the supply air to 18°C.

In winter operation mode, due to much lower or even without dehumidification requirement, the condensing temperature could be much lower than it is in summer mode. The evaporating temperature depends mainly on the rejecting air temperature from the evaporator. In the experimental design work, the minimum rejecting air temperature was design to be 0°C, and then the minimum evaporating temperature of the heat pump in the prototype CAHP was design to be -5°C. Thus, the maximum condensing temperature and the minimum evaporating temperature were determined to be 70°C and -5°C respectively. These two temperatures were used as the starting points for selecting the refrigerant.

To achieve the condensing and evaporating temperatures being 70°C and -5°C, different refrigerant candidates were compared with their condensing pressure and coefficient of performance. According to the environmental protection policy and criteria in Europe, only HFC refrigerants were selected to be candidates in this study. Natural refrigerants such as CO_2, NH_3 and HC also meet the environmental requirements. But they also have disadvantages such as too high requirement for technology of manufacturing heat pump of CO_2, toxicity of NH_3 and the flammability of HC. Natural refrigerants were not included in the refrigerant selection.

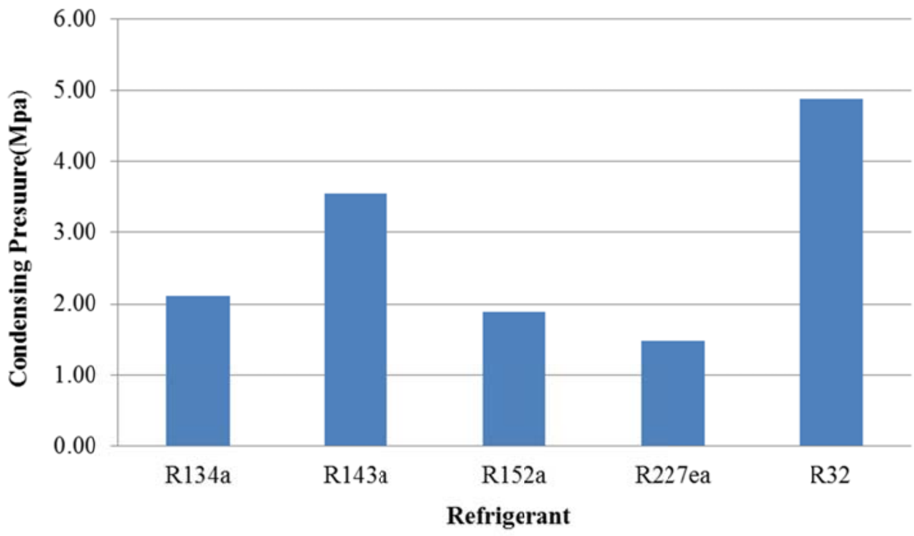


Figure 4.2 Condensing pressure of different HFC refrigerants at temperature of 70°C

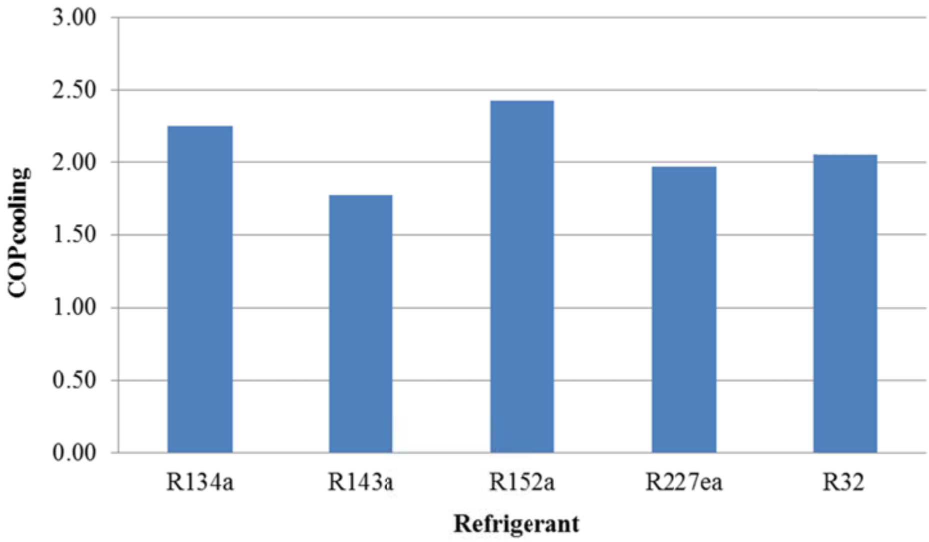


Figure 4.3 COP of different HFC refrigerants at recommended condensing and evaporating temperatures

During the selection, following standards were used as selection criteria.

1) Pressure at condensing temperature of 70°C should be lower than 2.5Mpa to avoid especial requirement on manufacturing the heat pump.
2) Pressure at evaporating temperature of -5°C should be higher than ambient pressure to avoid negative pressure in the refrigeration system

3) COP should be relatively high among the candidates.
4) The refrigerant should be un-combustible to ensure safety in laboratories.

Comparing the properties of different refrigerant candidates, refrigerant R152a and R134a are the two which have higher COP than the others. The thermal dynamic property of R152a is better than R134a. However, with the limitation of the laboratory safety conditions, R152a was not used in this study due to its combustibility. R134a meets all the requirements mentioned above. The Ozone Depletion Potential (ODP) value of R134a is 0 and its Global Warming Potential (GWP) value is 1300. It is non-combustible and meets the environmental standards for refrigerant in Denmark and Europe.

4.2.3 Compressor

With the selected refrigerant and the required condensing and evaporating temperature, the condensing and evaporating pressure for R134a in the summer design condition were validated to be 2.2Mpa (at 70°Ccondensing temperature) and 0.49Mpa (at 15°C evaporating temperature). Thus, the compression ratio of the compressor for CAHP was calculated to be 4.5.

With the maximum cooling load of the test room, e.g. the maximum designed cooling capacity for the CAHP, the refrigerant sucked to the compressor of the heat pump in CAHP was calculated to be with a flow rate of 8.11m^3/h. Based on the compression ratio and refrigerant flow rate calculated for the heat pump, a piston compressor “2GC-2.2”produced by Bizer was selected to be used in the CAHP prototype. The technical data of the compressor is listed in Table 4.4.

Table 4.4 Properties of the compressor selected for the prototype CAHP

Compressor: Bizer 2GC-2.2	
Displacement (1450RPM 50Hz)	7.58m3/h
Displacement (1750RPM 60Hz)	9.15 m3/h
Motor voltage	220-240V△/3/50Hz, 380-420VY/3/50Hz
Max operating current	4.7A
Max power consumption	2.7kW
Starting current	39A/22.5A (△/Y)
Weight	45kg
Max pressure (LP/HP)	19/28bar
Max compression ratio	9.3

The standard refrigerant displacement capacity of the selected compressor is $9.15m^3/h$, and the compression ratio is 9.3. Both of them fulfill the requirements calculated for the refrigerant flow rate and compression ratio. The maximum refrigerant pressure in the compressor is 2.8 MPa, and it is higher than the required pressure. The motor voltage of the compressor is 400V, and the input power frequency is variable from 20Hz to 80Hz. As described above, during the experiments, the input power frequency to the compressor was regulated by a frequency inverter to regulate the speed of the compressor.

4.2.4 Expansion valve

A solenoid valve “AKV10-6” produced by “Danfoss” was selected to be the throttle device in the heat pump. Besides the function for refrigerant expansion, the valve was also used to control the superheat temperature of refrigerant in the evaporator. During the experiments, the superheat temperature was controlled between 4°C and 6°C. A pressure sensor and a temperature sensor were connected to the outlet of the evaporator of heat pump. With the detected temperature and evaporating pressure, the superheat temperature could be calculated and used as a measured value for the expansion valve control. When the measured superheat temperature is higher than set value, the opening frequency of the solenoid valve will be adjusted upward. When the measured superheat temperature is lower than set value, the opening frequency of the solenoid valve will be adjusted downward. By regulating the opening frequency of the solenoid valve, the superheat temperature for refrigerant in evaporator can be well controlled.

4.2.5 Condensers and evaporators

The sizes of condensers and evaporators were calculated with software “Rrecalc ver.1.2.3” provided by company “Roenest”. The calculation was based on the airflow rates calculated in section 4.1 and the air temperatures supplied to the ventilation room. During the calculation, the condensing and evaporating temperature of refrigerant was set at 70°C and 15°C respectively, and the superheat temperature was set at 5°C as stated in the chapter above. The calculation results are listed in Table 4.5.

Table 4.5 Sizes of condensers and evaporators of the prototype CAHP

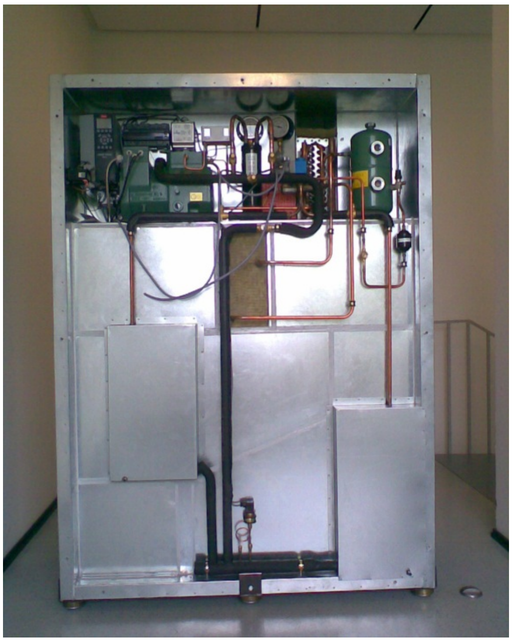
Heat exchangers	Length (mm)	Width (mm)	Rows

Heat exchangers	Length (mm)	Width (mm)	Rows
Evaporator for summer	500	500	8
Evaporator for winter	450	450	8
Condenser 1 for regeneration	300	250	5
Condenser 2 for excess heat (summer) or outdoor air-heating (winter)	300	250	5

To control the refrigerant flow rate to the two condensers, electronic control valves “EX5-U21” produced by “Emerson” were selected and connected to the outlet of condensers. As mentioned above, the two valves control the refrigerant flow rate to each condenser and thus the dehumidification capacity of the silica gel rotor. The controlling of the two valves will also influence the evaporating temperature of the heat pump. Therefore, the controlling of compressor speed should cooperate with the controlling of condenser valves’ opening degrees to realize required supply air temperature and humidity ratio. The control of the condenser valves and compressor speed are the key parameters for temperature and humidity independent control of the CAHP.

4.3 Prototype constructions

With all the key components selected, the prototype unit of CAHP was constructed. Figure 4.4 shows pictures of the heat pump and silica gel rotor in the prototype unit. The designed CAHP prototype was made by “Exhausto A/S” who is the industrial partner of the project.



heat pump

silica gel rotor

Figure 4.4 Pictures of the heat pump and the silica gel rotor for prototype CAHP

4.4 Experimental setup

The experiment was conducted in a test room at the International Center for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark (DTU). The test room is located on the 2nd floor of Building 402 at DTU. With the purpose of testing air handling units operating in different outdoor climates, the test room was equipped with a ventilation system which can simulate different outdoor climate conditions. The room also integrates many different types of ventilation terminals. One picture of this classroom is shown in Figure 4.5. The air delivering terminals chosen for testing the prototype CAHP were mixing ventilation diffusers. More details of the test room can be referred in the study of Fang et al. [113].

Figure 4.5 Pictures of the test room for experimental measurements

During the experiments, outdoor air was taken from the garden behind the building and delivered to the second floor of the building with air channels. There is an outdoor air handling unit to process the outdoor air to simulate different outdoor thermal climate conditions. Another air handling unit was used to simulate the cooling and heating load in the test room. Several electric humidifiers were used to simulate the latent load in the test room. Figure 4.6 gives the connections of the CAHP prototype with the existing air handling units and the test room.

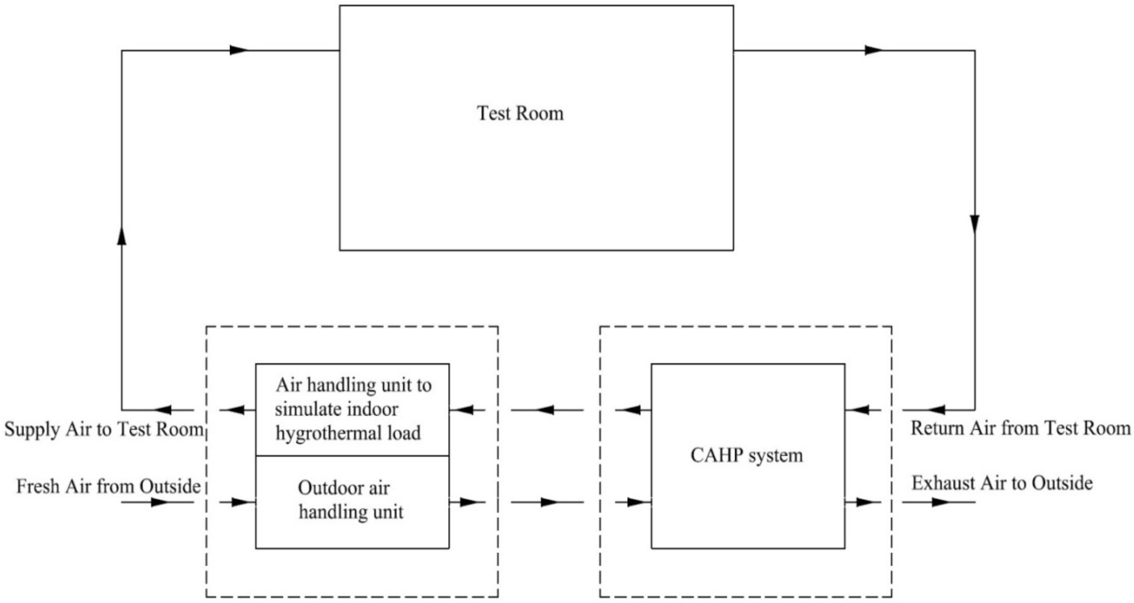


Figure 4.6 Connections of CAHP with the existing air handling units and test room

As described before, there are two inlets and two outlets in the CAHP system. During the experiments, outdoor air was taken from ambient, and recirculation air was taken from the test room. Exhaust air from the test room and the regeneration air of the CAHP were rejected to outdoor ambient. Large quantity of recirculation air and small amount of fresh air was processed by the CAHP and delivered to the test room.

The temperatures and humidity ratios of the air in the CAHP were measured with CAREL temperature and humidity sensors. The airflow rates were measured using IRIS differential-pressure transmitters. The energy use of the CAHP was measured and recorded by “Danfoss ADAP-KOLL Drive” frequency controller. The models of the sensors and their accuracies are listed in Table 4.6.

Table 4.6 Models and accuracies of the measuring equipment in the prototype CAHP

Measured objects	Measuring equipment	Measuring ranges	Accuracies
Temperature and Humidity Ratio	Carel DPDT010000 Duct Probe thermal meters	-20°C to 70°C 10% to 90%RH	±0.5°C at 25°C, ±0.9°C at -20°C to70°C ±3%RH at 25°C /50% RH, ±6%RH at -20°C to 70°C
Airflow rates	IRIS damper and Huba Control AG 669 pressure transmitters	From 0m^3/h -288 m^3/h to 0 m^3/h - 1260 m^3/h	±10% of the measuring range
Power consumption	Danfoss ADAP-KOLL Drive AKD-102	0kW to 2.7kW	±0.05kW

All the measured thermal conditions and flow rates of the air in the CAHP were logged to a computer by Agilent 34970A data logger.

4.5 Uncertainty analyses

From the accuracy data of the measurement instrument, the relative standard uncertainty of CAHP power consumption measurement caused by the “Danfoss ADAP-KOLL Drive AKD-102” was ±1.85%. However, the accuracy of temperature, humidity and airflow rate measurements could also cause the uncertainty of the power consumption measured. From the operation manual of the measurement instruments, the relative uncertainty of temperature and humidity ratio measurement was ±1%, ±6% respectively. The relative uncertainty of airflow rates measurement was ±10%.

From Figure 3.5, it can concluded that the outdoor air temperature has more significant influence on CAHP power consumption when outdoor humidity ratio is higher, and the relationship between CAHP power consumption and outdoor air temperature in the highest outdoor humidity ratio investigated case could be express with the following equation.

$$E_{CAHP} = 0.0604 T_{outdoot} + 1.024 \tag{28}$$

Where, E_{CAHP} is the energy use of CAHP, kW ;

$T_{outdoor}$ is the outdoor air temperature, ^{o}C .

The power consumption of CAHP has a linear relationship with the temperature. The relative uncertainty of measured power consumption caused by temperature measurement was thus calculated to be ±1%. The relationship between CAHP power consumption and airflow rate could also be expressed with a linear function, and then the relative uncertainties of measured power consumption caused by airflow rate measurement was ±10%. For the influence of the humidity ratio measurement, the relationship between the CAHP power consumption and the outdoor air humidity ratio was summarized and shown in Figure 4.7.

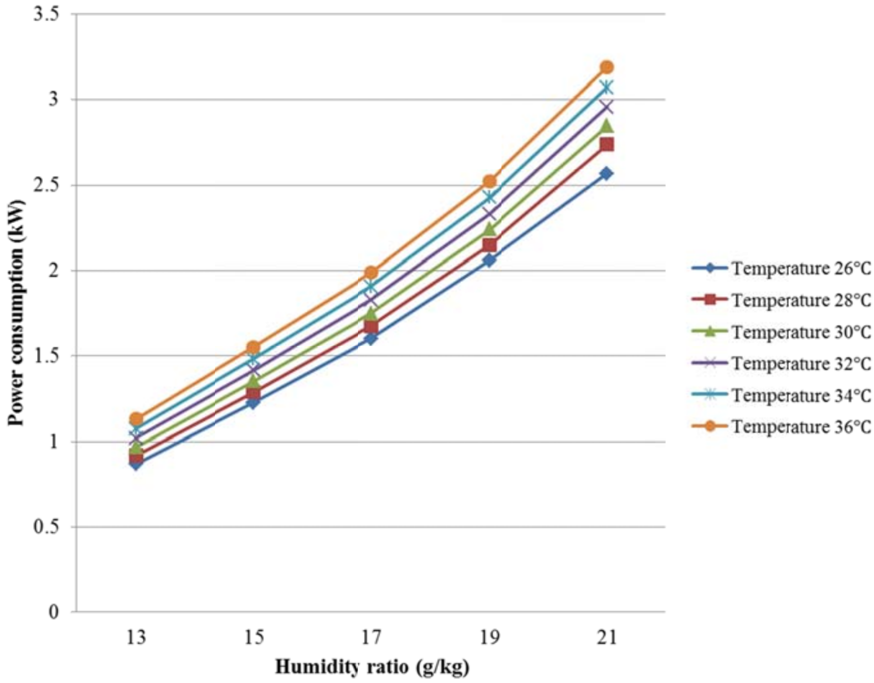


Figure 4.7 Effectiveness of outdoor air humidity ratio to CAHP power consumption

The maximum influence of humidity ratio measurement to power consumption happens when temperature is 36^{o}C. The relationship between CAHP power consumption with outdoor humidity ratio was expressed as following equation.

$$E_{CAHP} = 0.0029e^{0.1281Y} \tag{29}$$

Where, Y is the humidity ratio of the air, g/kg .

The relative uncertainty of CAHP power consumption caused by humidity ratio could then be calculated with following equation.

$$u'(E_{i-removed}) = 0.000371\,e^{0.1281\,u'(Y)} \tag{30}$$

Where, $u'(E_{i-removed})$ is the relative uncertainty of CAHP power consumption caused by humidity ratio measurement;

$u'(Y)$ is relative uncertainty of the humidity ratio measurement.

The relative uncertainty of CAHP power consumption caused by humidity ratio measurement was then calculated to be ±0.04%.

In conclusion, the relative uncertainty of CAHP power consumption measurement caused by power, temperature, humidity ratio and airflow rate measuring instruments were ±1.85%, ±1%, ±0.04% and ±10% respectively. The function of CAHP power consumption with the measured parameters all together is unknown. Thus, the maximum total relative uncertainty of CAHP power consumption measurement was calculated to be ±12.89% which sum all the relative uncertainties caused by the measuring instruments.

4.6 Experimental Design

The experiment was designed to validate the energy performance of the CAHP under different outdoor climate conditions, i.e. mild-cold, mild-hot and extremely hot and humid climates. Using the climate data of temperature and humidity for each hour of year 2002 provided by COWI, five typical outdoor climate classes plus one extreme condition in Copenhagen, Milan and Colombo were categorized for winter and summer seasons. They represent an average of the most probable outdoor conditions in which the CAHP could work during the whole year in each location.

The testing work assumed that the system was used only in office space, i.e. during normal office hours between 6:00 am to 6:00 pm. The classifications of summer and winter period were divided according to Table 4.7 for the three cities. With this assumption and classification, the outdoor air temperature, humidity and the number of hours of five categories in both summer and winter of the three cities were calculated and summarized in Table 4.8 -Table 4.12.

Table 4.7 Subdivisions of summer and winter operating modes

Location	Winter mode	Summer mode
Copenhagen	16th September – 30th April	1st May – 15th September
Milan	16th October – 15th April	16th April – 15th October
Colombo	-----	1st January – 31th December

During categorizing the outdoor climate conditions, the five classes were categorized by outdoor air temperatures, and the temperature of each class was the mean value in the range of the class. The corresponding moisture content of the class is the mean value of outdoor air humidity ratio in the same temperature class.

Table 4.8 Summer climate data of Copenhagen

Copenhagen - Summer			
	Temperature (°C)	Humidity Ratio (kg/kg)	Hours
Class 1	6.5	0.0057	76
Class 2	12.2	0.0075	655
Class 3	17.9	0.0086	808
Class 4	23.6	0.0096	231
Class 5	29.3	0.0105	23
Extreme case	32.1	0.0115	1

Table 4.9 Winter climate data of Copenhagen

Copenhagen - Winter			
	Temperature (°C)	Humidity Ratio (kg/kg)	Hours
Class 1	-16.69	0.00089	15
Class 2	-9.87	0.00163	62
Class 3	-3.05	0.00277	585
Class 4	3.77	0.00419	1303
Class 5	10.59	0.00577	400
Extreme case	-20.10	0.00060	1

Table 4.10 Summer climate data of Milan

Milan - Summer			
	Temperature (°C)	Humidity Ratio (kg/kg)	Hours
Class 1	10.5	0.0077	219
Class 2	15.5	0.0096	662
Class 3	20.5	0.0105	820
Class 4	25.5	0.0119	507
Class 5	30.5	0.0127	170
Extreme case	33.00	0.0136	1

Table 4.11 Winter climate data of Milan

Milan- Winter			
	Temperature (°C)	Humidity Ratio (kg/kg)	Hours
Class 1	-5.3	0.0025	98
Class 2	0.1	0.0035	623
Class 3	5.5	0.0049	898
Class 4	10.9	0.0065	550

Milan- Winter			
	Temperature (°C)	Humidity Ratio (kg/kg)	Hours
Class 5	16.3	0.0069	196
Extreme case	-8.00	0.0019	1

Table 4.12 Climate data of Colombo

Colombo			
	Temperature (°C)	Humidity Ratio (kg/kg)	Hours
Class 1	20.9	0.0143	80
Class 2	24.7	0.0176	1122
Class 3	28.5	0.0185	2847
Class 4	32.3	0.0180	692
Class 5	36.1	0.0151	3
Extreme case	38.0	0.0171	1

With the outdoor thermal climate conditions and the thermal physical properties of the test room listed in Table 4.1 and Table 4.2, the hygrothermal loads of the test room were calculated for different cities and different climate conditions. It is important to state that 15 persons were assumed working in the test room for calculating the ventilation rate and thermal load in the summer climates of Copenhagen and Milan, but 10 persons were assumed working in the test room when calculating the ventilation rate and thermal load in the summer climates of Colombo. Since the outdoor air temperature and humidity ratio of summer classes 1 to 3 in Copenhagen and classes 1 to 2 in Milan is low enough to balance the indoor cooling load, the buildings should be ventilated directly by outdoor air without running the CAHP. The energy performance of the CAHP in these outdoor climate conditions were not investigated in the experimental study.

For the winter climates, the outdoor air handling unit in the test room could only mimic outdoor climate with air temperature above 0°C since frost will accumulate on the cooling coil of the outdoor air handling unit when the temperature went below 0°C. Therefore, the experiment was conducted at two Copenhagen winter climate conditions - classes 4 and 5, and four Milan winter climate conditions - class 2, 3, 4 and 5. For Colombo, there is no winter season through the year and thus there is no need for space heating.

The hygrothermal loads of the test room and the hygrothermal conditions of the supply air delivered to the test room under different climate conditions were calculated and summarized in the following Table 4.12 and Table 4.13. In winter, the humidity ratio in the test room is not controlled.

Table 4.13 Hygrothermal loads and supply air conditions calculated for summer climates

Cities and Climate Classes	Indoor climate		Outdoor climate		Hygrothermal Load		Air delivered to ventilated room	
	Temperature (°C)	Humidity Ratio (g/kg)	Temperature (°C)	Humidity Ratio (g/kg)	Sensible Load (kW)	Latent Load(kg/h)	Temperature (°C)	Humidity Ratio (g/kg)
Copenhagen Class 4	25	9.85	23.6	9.6	1.76	1.02	19.18	8.91
Copenhagen Class 5	25	9.85	29.3	10.5	2.00	1.02	18.40	8.91
Copenhagen extreme	25	9.85	32.1	11.5	2.12	1.02	18.02	8.91
Milan Class 3	25	9.85	20.5	10.5	1.64	1.02	19.60	8.91
Milan Class 4	25	9.85	25.5	11.9	1.84	1.02	18.92	8.91
Milan Class 5	25	9.85	30.5	12.7	2.05	1.02	18.24	8.91
Milan Extreme	25	9.85	33	13.6	2.15	1.02	17.89	8.91
Colombo Class 1	25	9.85	20.9	14.3	1.34	0.68	20.18	9.17
Colombo Class 2	25	9.85	24.7	17.6	1.50	0.68	19.62	9.17
Colombo Class 3	25	9.85	28.5	18.5	1.66	0.68	19.06	9.17
Colombo Class 4	25	9.85	32.3	18	1.81	0.68	18.49	9.17
Colombo Class 5	25	9.85	36.1	15.1	1.97	0.68	17.93	9.17
Sri Lanka extreme	25	9.85	38	17.1	2.05	0.68	17.65	9.17

Table 4.14 Hygrothermal loads and supply air conditions calculated for winter climates

Cities and Climate Classes	Indoor climate		Outdoor climate		Hygrothermal Load		Air delivered to ventilated	
	Temperature (°C)	Humidity Ratio (g/kg)	Temperature (°C)	Humidity Ratio (g/kg)	Sensible Load (kW)	Latent Load (kg/h)	Temperature (°C)	Humidity Ratio (g/kg)
Copenhagen Class 5	22	4.89	10.59	5.77	-0.47	---	23.56	---
Copenhagen Class 4	22	4.89	3.77	4.19	-0.91	---	25.01	---
Milan Class 5	22	4.89	16.3	6.9	-0.24	---	22.78	---
Milan Class 4	22	4.89	10.9	6.5	-0.46	---	23.52	---
Milan Class 3	22	4.89	5.5	4.9	-0.68	---	24.25	---
Milan Class 2	22	4.89	0.1	3.5	-0.91	---	24.99	---

The calculated air flows including the outdoor airflow, recirculation airflow and other airflows of the CAHP operated in different seasons and different cities are listed in Table 4.15 and Table 4.16. According to the EU standard for ventilation [6], the fresh air in the CAHP system was designed to be 4L/s per person.

Table 4.15 Airflow rates of CAHP for different cities in summer

Cities	Recirculation air (L/s)	Outdoor air to test room (L/s)	Regeneration air (L/s)	Air for excess heat (L/s)	Exhaust air from room (L/s)
Copenhagen	190	60	125	120	60
Milano	190	60	125	120	60
Colombo	190	40	115	130	40

Table 4.16 Airflow rates of CAHP for different cities in winter

Cities	Recirculation air (L/s)	Outdoor air to test room (L/s)	Regeneration air (L/s)	Exhaust air from room (L/s)
Copenhagen	190	60	95	60
Milan	190	60	95	60

After the calculation, the experiments were conducted using the designed conditions. The measured data were then analyzed comparing with conventional heating or air-conditioning systems.

4.7 Data analysing methods

The energy use of the CAHP was measured under different outdoor climate conditions. To better evaluate the energy performance of the CAHP, the measured energy use of the prototype CAHP was compared to reference systems which were conventional heating or air-conditioning systems commonly used in existing buildings.

Two reference systems were assumed in the comparisons. One reference system is the air source heat pump used in both summer and winter seasons. In this reference system, outdoor air was used as cooling source in summer and heating source in winter. The compressor used in the reference heat pump system was variable speed compressor as used in the CAHP. The refrigerant used in the reference system was R134a as it was in the CAHP. Figure 4.8 and Figure 4.9 give schematic diagrams of the reference air source heat pump system.

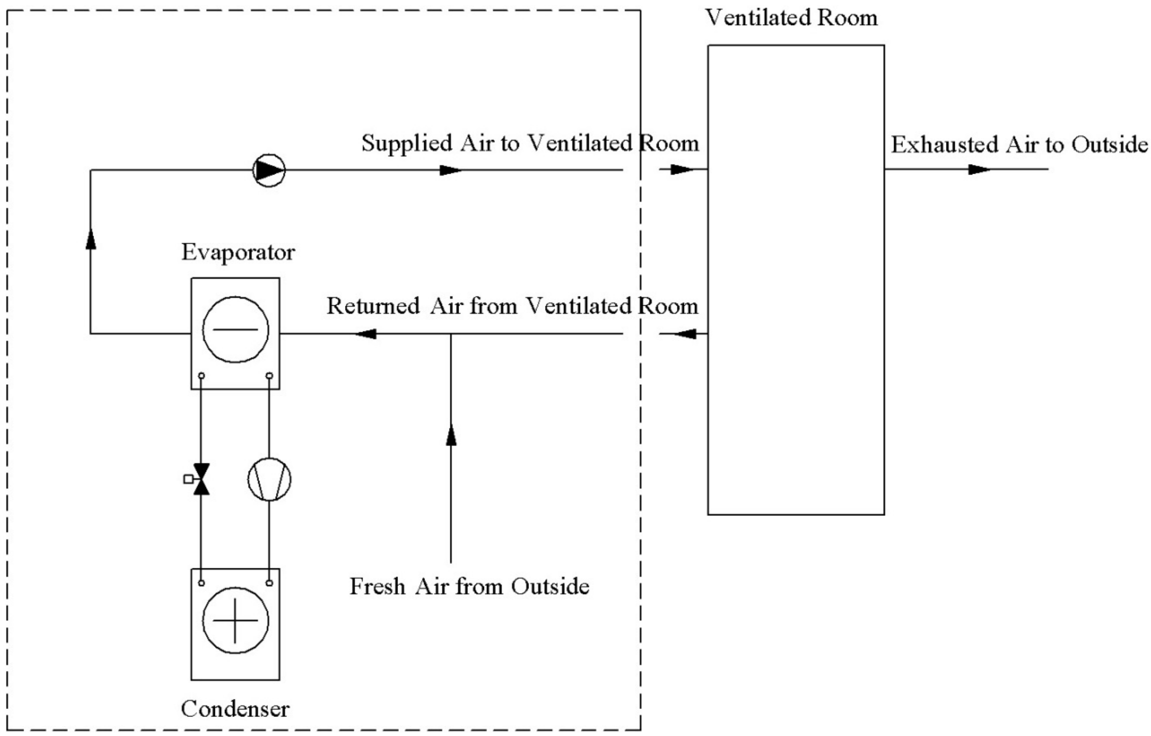


Figure 4.8 Schematic diagram of the reference air source heat pump system operated in summer mode

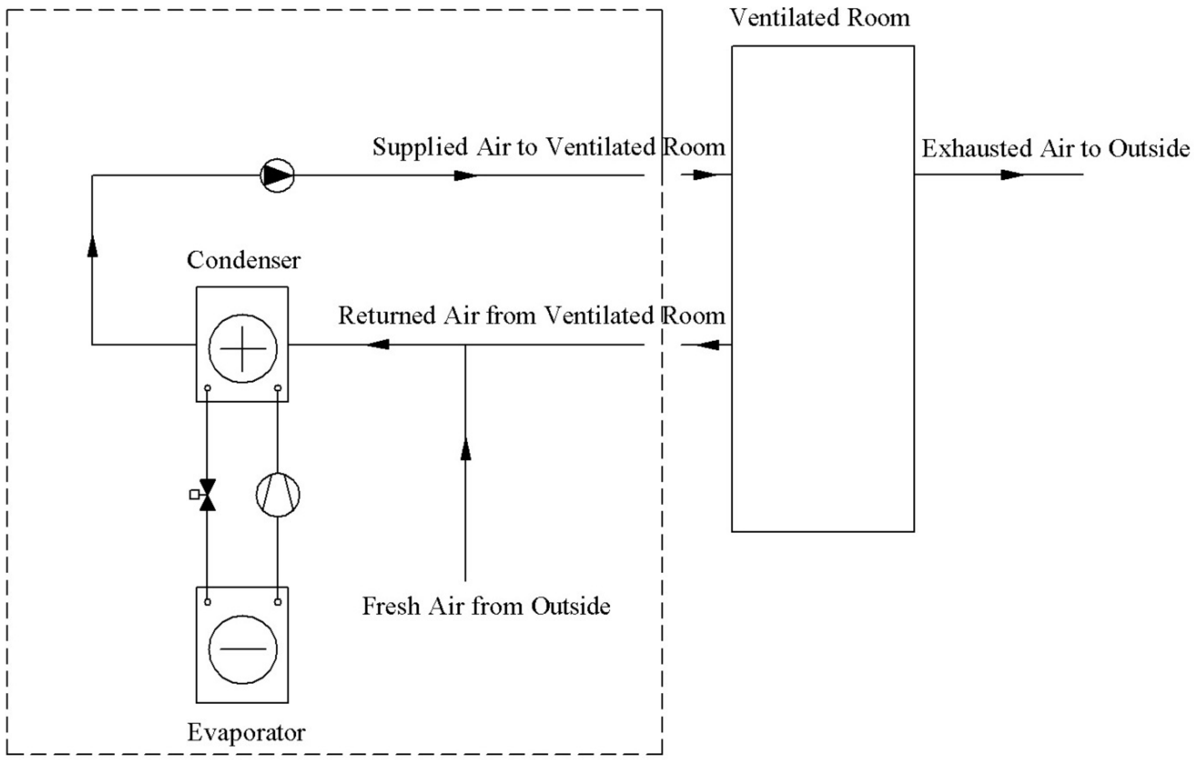


Figure 4.9 Schematic diagram of the reference air source heat pump system operated in winter mode

The indoor thermal loads of the test room are same for both the CAHP and the conventional air source heat pump systems. The COPs of the conventional air source heat pump were calculated with different condensing and evaporating temperatures in different classes of outdoor climates. During the COP calculations, the condensing and evaporating temperatures of the conventional air source heat pump were calculated referring to the measured air and refrigerant temperature difference in the CAHP. The calculating methods are shown in the following equations.

$$T_{con-ref} = T_{con-outlet-ref} + (T_{con-CAHP} - T_{con-outlet-CAHP}) \tag{31}$$

$$T_{eva-ref} = T_{eva-outlet-ref} + (T_{eva-CAHP} - T_{eva-outlet-CAHP}) \tag{32}$$

Where:

$T_{con-ref}$, $T_{eva-ref}$ are the calculated condensing and evaporating temperature of the reference air source heat pump, °C;

$T_{con-outlet-ref}$, $T_{eva-outlet-ref}$ are the designed air temperature at the outlet point of the condenser and evaporator of the reference air source heat pump, °C;

$T_{con-CAHP}$, $T_{eva-CAHP}$ are the measured condensing temperature and evaporating temperature of the heat pump in the CAHP system, °C;

$T_{con-outlet-CAHP}$, $T_{eva-outlet-CAHP}$ are the measure air temperature at the outlet point of the condenser and evaporator of the heat pump in the CAHP system, °C;

The entropy efficiency of the compressor in the reference air source heat pump system was assigned referring to the measured entropy efficiency of the compressor in the CAHP.

Besides the energy use comparisons with reference air source heat pump, the CAHP is futher compared with another reference system which use air source heat pump in summer but use gas boiler provide heating in winter. Due to that the energy sources of the CAHP and gas boiler are different, the comparions of CAHP with gas boiler in winter mode is based on the primary energy use. During the calculation, the overall efficiency of boiler and heat exchangers was assumed to be 81%. The primary energy factors of natural gas and power in Copenhagen and Milan was investigated and listed in Table 4.17 [114].

Table 4.17 Primary energy factors of power and natural gas in Copenhagen and Milan

Copenhagen (Denmark)		Milan (Italy)	
Natural Gas	Power	Natural Gas	Power
1	2.5	1	2.18

To ensure the comparisons between the CAHP and the reference systems impartial enough, the comparisons were made based on both systems providing same indoor air quality and same indoor thermal environment in the test room. Since the CAHP has a very strong ability on indoor air cleaning, the airflow rates of outdoor air and indoor recirculation air are quite different in the CAHP and the reference systems. Based on the previous study [88], 80% of indoor recirculated air is cleaned by the silica gel rotor and can be used to substitute for outdoor air.

The flow rate of outdoor air in the reference system should be equivalent to the flow rate of clean air delivered into the test room by the CAHP system and was calculated with the following equation.

$$V_{f-ref} = V_{f-CAHP} + \eta \times V_{r-CAHP} \tag{33}$$

Where:

$V_{fre-ref}$ is the outdoor fresh airflow rate in the reference system, l/s;

$V_{fre-CAHP}$ is the fresh airflow rate in the CAHP system, and here it was taken to be 4L/s per person;

η is the air cleaning efficiency of the silica gel rotor. Referring to the study of Fang et al. [88], the value was taken to be 0.8;

$V_{rec-CAHP}$ is the recirculation airflow rate in the CAHP system, l/s.

Thus, the outdoor airflow rates and the recirculation airflow rates of CAHP and the reference system were calculated as shown in Table 4.18.

Table 4.18 Outdoor and recirculation airflow rates of CAHP and reference system

Cities	Outdoor air (L/s)		Recirculation air (L/s)	
	CAHP	reference system	CAHP	reference system
summer				
Copenhagen	60	212	190	38
Milan	60	212	190	38
Colombo	40	192	190	38
winter				
Copenhagen	60	212	190	38
Milan	60	212	190	38

With the airflow rates in the reference system (Table 4.18) and the outdoor climate conditions listed in Table 4.8-Table 4.12, the hygrothermal load and the energy use of reference system can be calculated.

With the calculated energy use of reference systems and the measured energy use of the CAHP, the energy saving potential of the CAHP was thus estimated. Overall, the comparisons were made based on both systems providing the same indoor air quality and indoor thermal environment in the same test room. The hygrothermal conditions of the supply air delivered to the ventilated room in the reference system were equal to that in the CAHP system. The energy uses of the proposed

CAHP were measured with experimental studies. The energy uses of the reference conventional systems were calculated. During the calculation, the indoor air, outdoor air and supply air thermal conditions in the reference systems were designed referring to the measured values from the experimental study of the CAHP. In other words, the thermal parameters of indoor air, outdoor air and supply air in the two systems were equal.

After the measurements and calculations of energy use of the CAHP and the reference system, the energy saving ratio of the CAHP was calculated with the following equation.

$$\mu_{E-S} = (E_{referece} - E_{CAHP}) / E_{referece} \tag{34}$$

Where:

μ_{E-S} is the energy saving potential of the CAHP;

E_{CAHP} is the energy use of CAHP, kW;

$E_{referece}$ is the energy use of the reference system, kW.

4.8 Results

During the experiments, the indoor, outdoor thermal climate conditions listed in Table 4.13 and Table 4.14 were established stably by air handing units of the test room. The indoor thermal conditions were controlled stably around the set points e.g. 25°C with 9.85g/kg moisture content in summer modes and 22°C in winter modes.

Table 4.19 Thermal conditions of indoor and outdoor air during the experimental measurements

Cities and Climate Classes	Indoor Air				Outdoor Air			
	Temperature		Humidity Ratio		Temperature		Humidity Ratio	
	Mean (°C)	SD	Mean (g/kg)	SD	Mean(°C)	SD	Mean(g/kg)	SD
Copenhagen Summer Class 4	25.04	0.03	9.92	0.06	23.54	0.07	9.52	0.32
Copenhagen Summer Class 5	25.32	0.03	10.10	0.04	29.19	0.24	10.41	0.21
Copenhagen Summer extreme	26.26	0.03	9.98	0.13	31.88	0.29	11.14	0.43
Milan Summer Class 3	25.46	0.03	9.87	0.08	19.83	0.21	10.30	0.46
Milan Summer Class 4	25.83	0.04	9.98	0.06	25.78	0.11	11.68	0.42
Milan Summer Class 5	25.74	0.03	9.82	0.16	30.56	0.10	12.25	0.41
Milan Summer Extreme	25.94	0.11	10.10	0.04	32.66	0.20	13.64	0.39
Colombo Summer Class 1	25.08	0.03	9.84	0.10	20.63	0.30	14.01	0.24
Colombo Summer Class 2	25.06	0.03	9.79	0.17	24.98	0.22	17.49	0.57
Colombo Summer Class 3	25.36	0.09	9.90	0.16	28.37	0.32	18.18	0.30
Colombo Summer Class 4	24.99	0.05	9.80	0.18	32.52	0.36	18.30	0.31
Colombo Summer Class 5	25.70	0.07	9.98	0.16	36.03	0.11	15.09	0.35
Colombo Summer extreme	25.46	0.08	9.93	0.17	38.28	0.19	17.04	0.36
Copenhagen Winter Class 5	22.12	0.13	5.38	0.03	10.51	0.08	5.75	0.33
Copenhagen Winter Class 4	21.98	0.15	4.88	0.08	4.21	0.47	4.34	0.18
Milan Winter Class 5	21.82	0.19	5.63	0.14	15.98	0.07	6.59	0.46
Milan Winter Class 4	21.80	0.16	6.21	0.09	10.91	0.52	6.66	0.32
Milan Winter Class 3	22.04	0.04	4.80	0.07	5.62	0.43	4.86	0.19
Milan Winter Class 2	22.54	0.15	4.94	0.09	0.39	0.55	3.46	0.17

The airflow rates and the hygrothermal conditions of the supply air (air delivered to the ventilated room) were controlled closely around the calculated values listed in the Table 4.13 to Table 4.16. During the experimental measurements, all the data kept stable for at least one hour.

4.8.1 Energy saving in summer climates

The cooling capacity and energy use of the CAHP operating in summer modes were recorded and listed in Table 4.20. The COPs of the CAHP for cooling were thus calculated and listed. For the reference system, the COPs of the air source heat pump for cooling were calculated at first.With the calculated COPs and the heating/cooling, dehumidification load of the test room, the energy uses of the reference air source heat pump were calculated and listed in Table 4.21. As mentioned above, due to the strong air purification capacity of the CAHP, the requirement of outdoor fresh air in the CAHP system was much lower than it was in the reference system. Therefore, the total cooling loads (sensible cooling and dehumidification) of the CAHP and the reference system were different even though both systems provided same indoor air quality.

Table 4.20 Instantaneous use of the heat pump in CAHP in different cities and different summer climate classes

Cities and Climate Classes	CAHP		
	Cooling capacity (kW)	Power consumption (kW)	$COP_{cooling}$
Copenhagen Summer Class 4	2.45	0.53	4.63
Copenhagen Summer Class 5	3.60	0.85	4.25
Copenhagen Extreme Summer	4.38	1.22	3.58
Milan Summer Class 3	2.65	0.53	4.98
Milan Summer Class 4	3.77	0.76	4.94
Milan Summer Class 5	4.69	1.16	4.03
Milan Extreme Summer	5.13	1.51	3.41
Colombo Summer Class 1	3.42	0.99	3.47
Colombo Summer Class 2	4.78	1.77	2.70
Colombo Summer Class 3	5.25	2.08	2.53
Colombo Summer Class 4	5.04	1.72	2.92
Colombo Summer Class 5	5.13	1.66	3.09
Colombo Extreme Summer	5.39	1.96	2.75

The above results show that the COPs for cooling of the heat pump in CAHP varied from 3.6 to 4.6, from 3.4 to 5.0 and from 2.5 to 3.5 when the CAHP was operated in the summer climate conditions of Copenhagen, Milan and Colombo respectively.

For the reference system, the COP of the air source heat pump and its energy use for cooling under different outdoor climate conditions were calculated and listed in Table 4.21.

Table 4.21 Instantaneous energy use of the reference air source heat pump in different cities and different summer climate classes

Cities and Climate Classes	Reference air source heat pump		
	Cooling capacity (kW)	Power consumption (kW)	$COP_{cooling}$
Copenhagen Summer Class 4	3.95	1.32	2.99
Copenhagen Summer Class 5	5.97	2.13	2.81
Copenhagen Extreme Summer	7.26	3.06	2.37
Milan Summer Class 3	3.77	0.79	4.76
Milan Summer Class 4	5.82	1.34	4.35
Milan Summer Class 5	7.82	2.46	3.18
Milan Extreme Summer	9.08	3.28	2.77
Colombo Summer Class 1	5.28	1.10	4.80
Colombo Summer Class 2	8.11	2.29	3.55
Colombo Summer Class 3	9.37	3.06	3.06
Colombo Summer Class 4	10.36	3.65	2.84
Colombo Summer Class 5	9.79	4.11	2.38
Colombo Extreme Summer	11.21	5.56	2.02

Comparing the values of energy use listed in Table 4.20 and Table 4.21, the energy saving of the CAHP in the three regional summer climates were calculated and listed in Table 4.22.

Table 4.22 Instantaneous energy use of CAHP, reference system and energy saving of CAHP compared to reference system in different cities and different summer climates

Cities and Climate Classes	Power consumption (kW)		Energy saving
	CAHP	Reference system	CAHP to Reference
Copenhagen Summer Class 4	0.53	1.32	59.87%
Copenhagen Summer Class 5	0.85	2.13	60.10%
Copenhagen Extreme Summer	1.22	3.06	60.04%
Milan Summer Class 3	0.53	0.79	32.90%
Milan Summer Class 4	0.76	1.34	43.07%
Milan Summer Class 5	1.16	2.46	52.61%

Cities and Climate Classes	Power consumption (kW)		Energy saving
	CAHP	Reference system	CAHP to Reference
Milan Extreme Summer	1.51	3.28	54.15%
Colombo Summer Class 1	0.99	1.10	10.36%
Colombo Summer Class 2	1.77	2.29	22.65%
Colombo Summer Class 3	2.08	3.06	32.18%
Colombo Summer Class 4	1.72	3.65	52.79%
Colombo Summer Class 5	1.66	4.11	59.66%
Colombo Extreme Summer	1.96	5.56	64.73%

The comparisons of energy use between the CAHP and the reference air source heat pump system in summer mode in the three cities are demonstrated by histograms in Figure 4.10 to Figure 4.12.

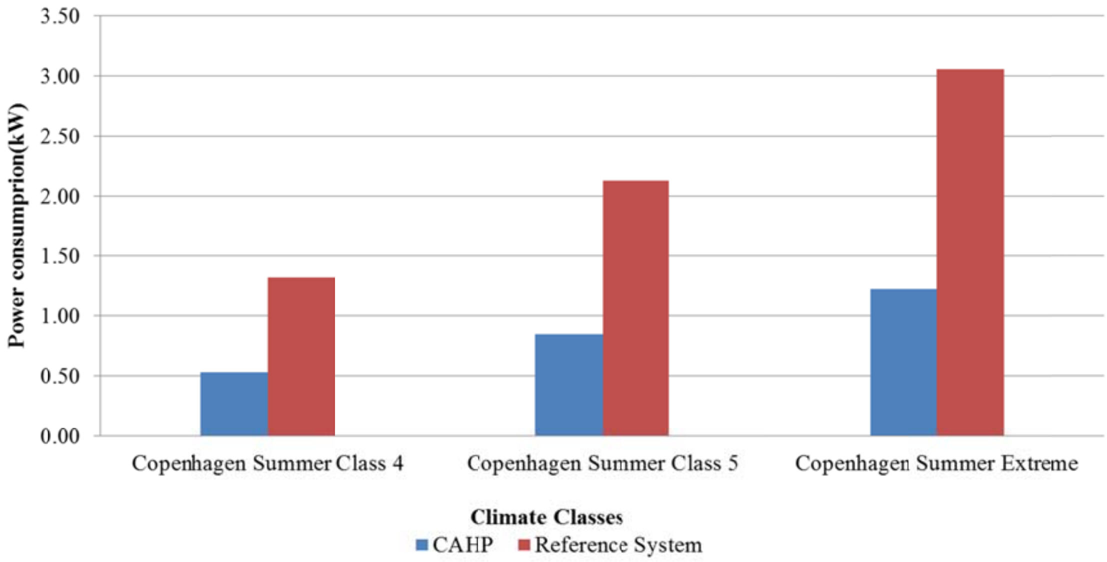


Figure 4.10 Instantaneous energy use of CAHP and reference system in the three summer climate conditions of Copenhagen

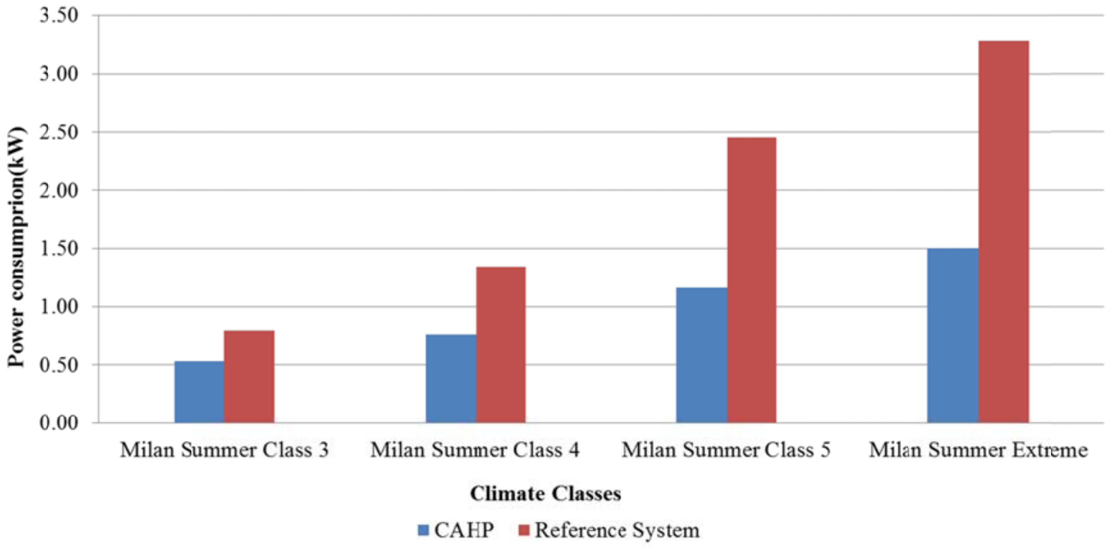


Figure 4.11 Instantaneous energy use of CAHP and reference system in the four summer climate conditions of Milan

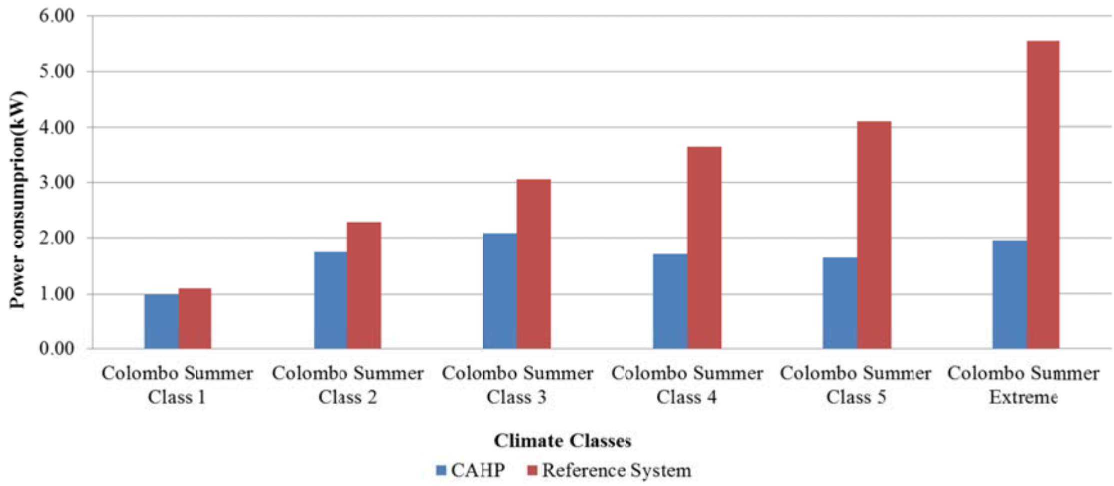


Figure 4.12 Instantaneous energy use of CAHP and reference system in the six summer climate conditions of Colombo

With the number of hours of different climate categories obtained in Table 4.8, Table 4.10 and Table 4.12, the energy use of CAHP system and reference air source heat pump in whole summer could be calculated with the following equation.

$$G = \sum_{i=1,2,\cdots\cdots} Q_i h_i \tag{35}$$

Where,

G is the total energy use of CAHP or reference system in a whole season, kWh ;

Q_i is the instantaneous energy use of CAHP or reference system in a category of outdoor thermal environment, kW ;

h_i is the number of hours in the category of outdoor thermal environment, h ;

Then the energy saving proportion of CAHP compared to reference air source heat pump in whole summer could be calculated, and the results are listed in Table 4.23.

Table 4.23 Total energy use of CAHP, reference system and energy saving of CAHP compared to reference system in summer of different cities

Cities and Seasons	Power consumption (kWh)		Energy saving
	CAHP	Reference System	CAHP to reference
Copenhagen Summer	143.15	356.99	59.90%
Milan Summer	1022.21	1749.97	41.59%
Colombo Summer	9170.06	13908.81	34.07%

The experiment results showed that the energy saving proportion of CAHP compared to the reference system varies from 34% to 60%. This means that for the same indoor air quality, CAHP can save more than 34% of energy use when it operates in summer of all the three different climate zones. In Copenhagen, it has the maximum energy saving potential of 60%.

4.8.2 Energy Saving in winter climates

The energy use of CAHP in winter operation mode is compared with two reference systems i.e. a reference air source heat pump and a gas boiler.

4.8.2.1 Energy use of CAHP in winter compared to reference air source heat pump

The energy use of the heat pump in the CAHP operated in winter climates were recorded and listed in Table 4.24. Together with the heating load (including heating the fresh air and the regeneration air) in different winter climates, the COPs for heating of the CAHP (COP_{heaing}) were calculated and listed in Table 4.24. For the reference system, the COPs of the air source heat pump were calculated in different classes of outdoor climates at first. With the calculated COPs, the energy uses of the reference air source heat pump were obtained and listed in Table 4.24. As it was in the summer operation mode, the requirement of outdoor fresh air for ventilation in the CAPH system was much lower than the reference system due to its strong air purification capacity. But some extra heat from the heat pump of the CAHP should be used to heat the regeneration air. Even though the regeneration heat was recovered by the evaporator of heat pump, it required extra energy for recovering the regeneration heat. Thus the heating load of CAHP was quite different with the reference system, and it could be higher or lower than the reference system.

Table 4.24 Instantaneous energy use of the heat pump in CAHP and reference air source heat pump in different cities and different winter climate classes

Cities and Climate Classes	Total heating load (kW)		Power consumption (kW)		$COP_{heating}$	
	CAHP	Air Source Heat Pump	CAHP	Air Source Heat Pump	CAHP	Air Source Heat Pump
Copenhagen Winter Class 5	3.45	3.50	0.67	0.72	5.12	4.85
Copenhagen Winter Class 4	5.06	5.20	1.34	1.40	3.78	3.72
Milan Winter Class 5	2.24	2.02	0.44	0.37	5.15	5.38
Milan Winter Class 4	3.32	3.43	0.64	0.69	5.20	4.99
Milan Winter Class 3	4.62	4.73	1.09	1.13	4.23	4.19
Milan Winter Class 2	5.48	6.25	1.66	2.14	3.30	2.92

The comparisons of energy use between the CAHP and the reference air source heat pump system in winter mode in the two cities are demonstrated by histograms in Figure 4.13 and Figure 4.14.

Table 4.25 Instantaneous energy use of CAHP, reference air source heat pump and energy saving of CAHP compared air source heat pump in different cities winter climate classes

Cities and Climate Classes	Power Consumption (kW)		Energy Saving
	CAHP	Air source heat pump	CAHP to air source heat pump
Copenhagen Winter Class 5	0.67	0.72	6.74%
Copenhagen Winter Class 4	1.34	1.40	4.06%
Milan Winter Class 5	0.44	0.37	-16.18%
Milan Winter Class 4	0.64	0.69	6.94%
Milan Winter Class 3	1.09	1.13	3.12%
Milan Winter Class 2	1.66	2.14	22.34%

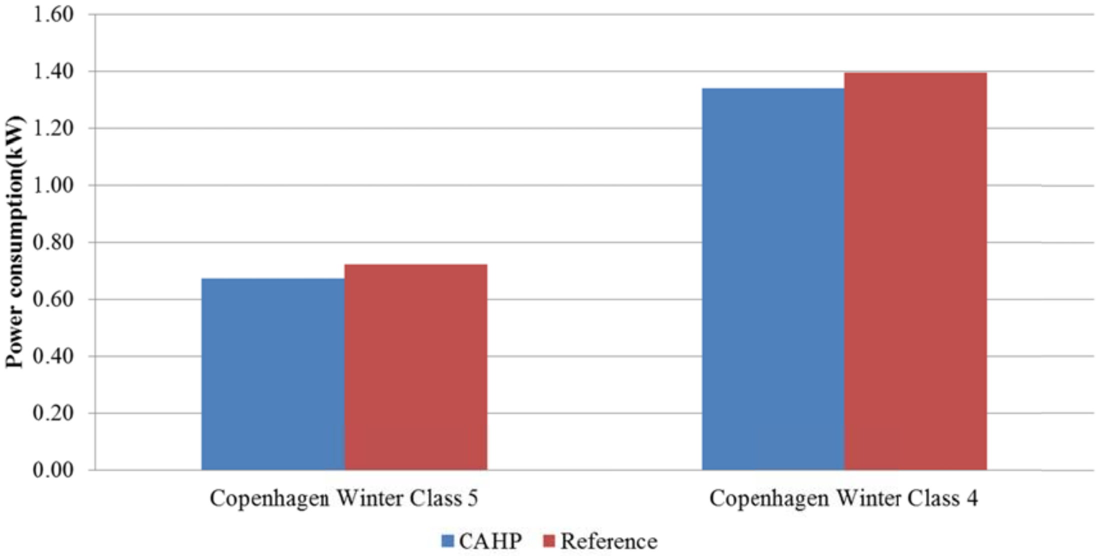


Figure 4.13 Instantaneous energy use of CAHP and reference air source heat pump in the two winter climate conditions of Copenhagen

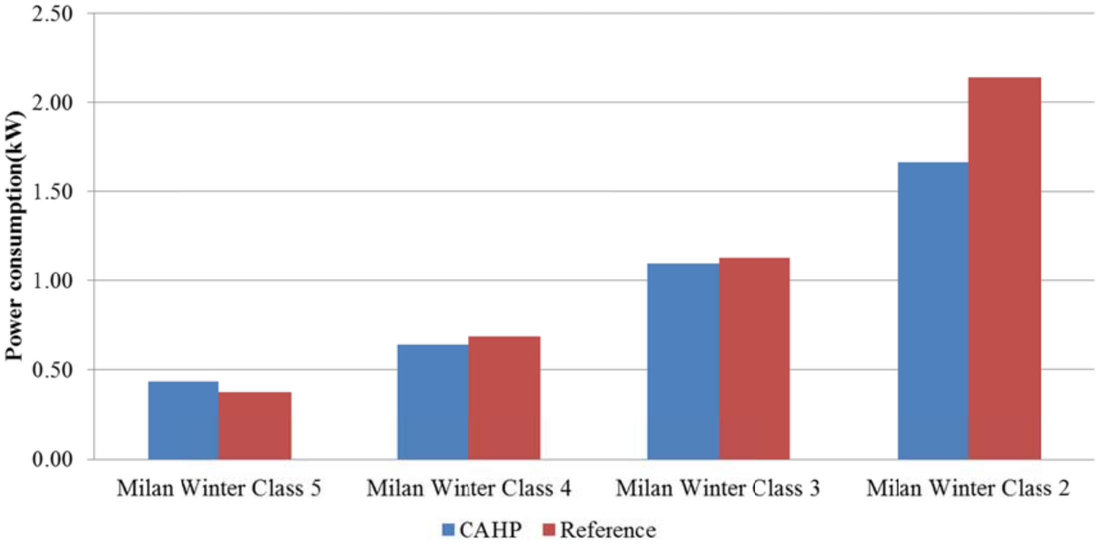


Figure 4.14 Instantaneous energy use of CAHP and reference air source heat pump in the four winter climate conditions of Milan

From the tables and figures above, it can be seen that the energy use of CAHP was lower than the reference air source heat pump system in most of the experimental investigated cases, and the energy conservation proportion depends partly on the outdoor thermal climates. Using the hourly data and the number of hours obtained in Table 4.9 and Table 4.11 for different climate classes of the two cities, energy use of the CAHP and reference air source heat pump systems in the whole experimental investigated winter climates of the two cities could be calculated and compared. The results are given in Table 4.26, and they show that the energy saving proportion of the CAHP compared to reference air source heat pump system varies from 4% to 12%. This means that for the same indoor quality, CAHP can save 4.43% of energy when it operates in the experimental investigated winter climates of Copenhagen, and it can save 12.28% of energy in the experimental investigated winter climates of Milan.

Table 4.26 Total energy use of CAHP, reference air source heat pump and energy saving of CAHP in experimental investigated winter climates of different cities

Cities and Seasons	Power consumption (kWh)		Energy saving
	CAHP	Air source heat pump	CAHP to air source heat pump
Copenhagen winter	2015.52	2108.87	4.43%
Milan winter	2454.96	2798.77	12.28%

4.8.2.2 Energy use of CAHP in winter compared to gas boiler

Besides the comparison of energy use between the CAHP and the air source heat pump, the energy use of CAHP in winter operation mode is further compared to a gas boiler.

In this comparison, the reference system in winter becomes a gas boiler which provides heating source in the ventilation system. The energy use of reference system was, therefore, converted to the consumption of natural gas. However, the CAHP and reference gas boiler used different energy sources in winter (CAHP used power, and gas boiler used natural gas), the primary energy was then used as the object for comparing the energy efficiency of the CAHP and the reference system. Considering that the primary energy factors of power and natural gas are different between Copenhagen and Milan, the primary energy use used by the CAHP and boiler were calculated with local primary energy factors listed in Table 4.17.

Since regenerating the silica gel rotor (for air cleaning) takes heat from the condenser of the heat pump, extra heating load for this regeneration heat should be fulfilled by the CAHP. On the other hand, the CAHP could save some amount of energy for heating outdoor airflow due to its high efficient indoor air cleaning. Thus, the heating load of the CAHP is different from the heating load of the reference gas boiler. It could be higher or lower than the reference gas boiler system.

With the heating load for the reference gas boiler and the primary energy factors of natural gas, the primary energy use of gas boiler operated in experimental investigated winter climates of Copenhagen and Milan could be calculated and compared to that of CAHP. Table 4.27 gives the comparison results.

Table 4.27 Instantaneous primary energy use of CAHP, reference gas boiler and energy saving of CAHP in different cities and different winter climate classes

Cities and Climate Classes	Primary energy use (kWh)		Energy saving
	CAHP	Gas Boiler	CAHP to Gas Boiler
Copenhagen Winter Class 5	1.68	4.32	61.05%
Copenhagen Winter Class 4	3.35	6.42	47.81%
Milan Winter Class 5	0.95	2.49	61.90%
Milan Winter Class 4	1.40	4.24	67.08%
Milan Winter Class 3	2.38	5.84	59.16%
Milan Winter Class 2	3.62	7.72	53.06%

The comparison of energy use between the CAHP and the reference gas boiler in winter mode in the two cities are demonstrated by histograms in Figure 4.15 and Figure 4.16.

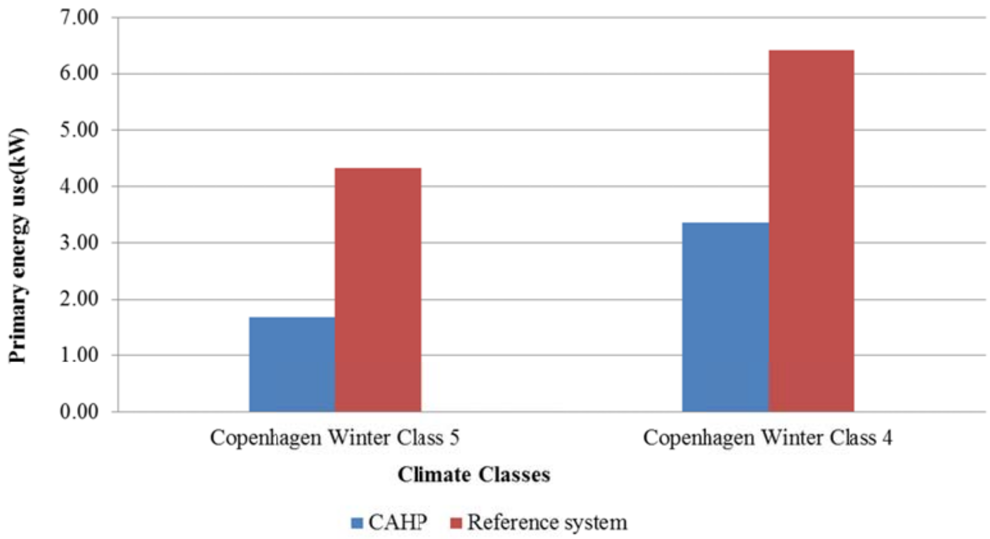


Figure 4.15 Instantaneous primary energy use of CAHP and reference gas boiler in the two winter climate conditions of Copenhagen

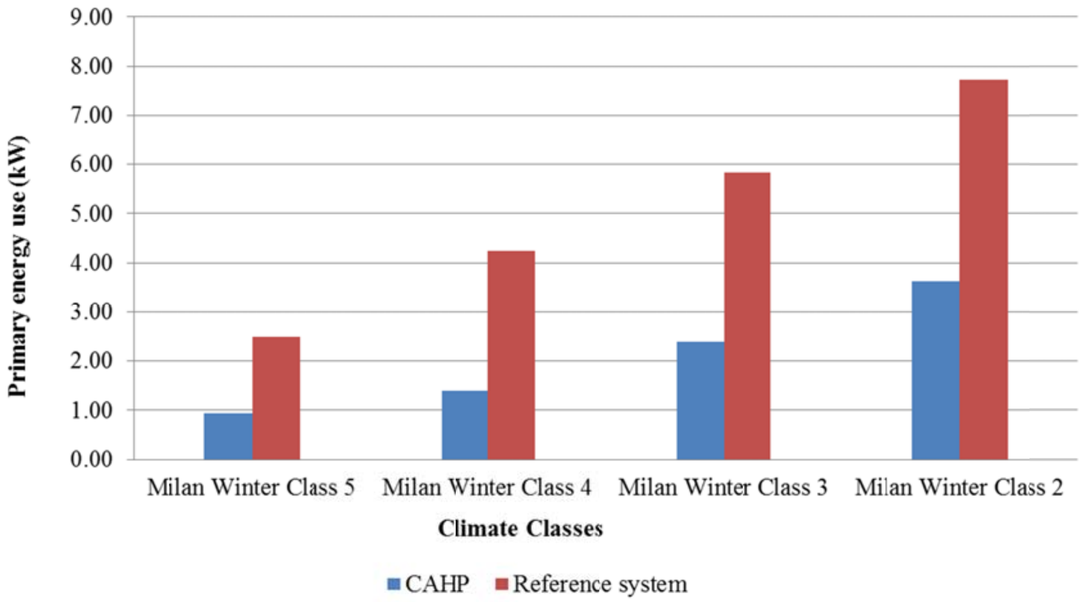


Figure 4.16 Instantaneous primary energy use of CAHP and reference gas boiler in the four winter climate conditions of Milan

The energy saving of CAHP compared to reference gas boiler in the experimental investigated winter climates of Copenhagen and Milan was calculated and shown in Table 4.28. In the calculation, the total energy use was weighted by the number of hours of each climate classes.

Table 4.28 Total primary energy use of CAHP, reference gas boiler and energy saving of CAHP compared to gas boiler in experimental investigated winter climates of Copenhagen and Milan

Cities	Primary energy use (kWh)		Energy saving
	CAHP	Gas boiler	CAHP to Gas Boiler
Copenhagen	5038.80	10094.27	50.08%
Milan	5351.82	12871.54	58.42%

Based on the consumption of primary energy, the measured energy saving using CAHP in winter varies from 50%- 58%.

Overall, the comparison results show that the CAHP system can also save energy compared to the reference gas boiler system in winter operation mode.

4.8.3 The annual energy saving

Based on the measured and calculated energy use of operating CAHP and reference system in summer and winter seasons, the energy use of the systems for the experimental investigated climates in whole year round of the three cities could be calculated and compared.

4.8.3.1 Energy use of CAHP compared to reference air source heat pump

The energy use of CAHP in the experimental investigated climates of whole year and the comparison with reference air source heat pump are listed in Table 4.29.

Table 4.29 Total energy use of CAHP, reference air source heat pump and energy saving of CAHP compared to reference system in whole year of different cities

Cities	Power consumption (kWh)		Energy saving
	CAHP	Air Source Heat Pump	CAHP to Air Source Heat Pump
Copenhagen	2158.67	2465.86	12.46%
Milan	3477.18	4548.74	23.56%
Colombo	9170.06	13908.81	34.07%

Based on the power consumption, the measured annual energy saving using CAHP compared to reference air source heat pump system varies from 12% to 34% in the three cities. In the extremely hot and humid climate as Colombo, it has the maximum energy saving potential with 34.07%. The CAHP can save 12.46% of power consumption when it is used in Copenhagen, and save 23.56% of power consumption in Milan. Though the energy saving proportions of using a CAHP in Copenhagen and Milan are much less than that in Colombo, the CAHP system can still save substantial amount of energy compared to the reference conventional air source heat pump system.

4.8.3.2 Primary energy use of CAHP compared to reference system using air source heat pump in summer and gas boiler in winter

The comparisons of primary energy use between the CAHP and the reference system which use air source heat pump in summer and gas boiler in winter are listed in Table 4.30. Due to that no heating is required in Colombo, the energy saving proportion of CAHP in in whole year of Colombo is same with the energy saving proportion in summer and not listed in the Table 4.30.

Table 4.30 Comparisons on annual energy use of CAHP and reference system using air source heat pump in summer and gas boiler in winter

Cities	Primary energy use (kWh)		Energy expense saving
	CAHP	Air Source Heat Pump and Gas Boiler	CAHP to Air Source Heat Pump and Gas Boiler
Copenhagen	5396.68	10986.74	50.88%
Milan	7580.24	16686.47	54.57%

Based on the consumption of primary energy, the measured annual energy expense saving using CAHP varies from 51% to 55% in the three cities. The CAHP can save 50.88% of energy use when it is used in Copenhagen, and save 54.57% of energy use in Milan. The CAHP system can save substantial amount of energy compared to the reference system which used air source heat pump in summer and use gas boiler in winter.

4.8.4 Extra energy use caused by the pressure drop of silica gel rotor in CAHP

In the results mentioned above, the comparisons of energy efficiency between the CAHP and the reference systems were made based on the power consumption of compressor of the CAHP. The power consumption of fans was not included in the comparison. The pressure drop caused by the silica gel rotor in CAHP could be a main concern since it will increase the power consumption of the fans and thus decrease the energy efficiency of the whole system.

From the data provided by the manufactory of the silica gel rotor, the pressure drop by the silica gel rotor is 200 Pa. With the process airflow rate through the silica gel rotor in different outdoor climate conditions and different cities, the extra power consumption caused by the pressure drop of the silica gel rotor can be calculated with the following equation. The results are listed in Table 4.31.

$$E_{si} = V_{pro} \times dP_{si} / \eta_{fan} /1000 \tag{36}$$

Where,

E_{si} is the extra power consumption caused by the pressure drop of the silica gel rotor, W;

V_{pro} is the process airflow rate through the silica gel rotor, l/s;

dP_{si} is the pressure drop through the silica gel rotor, Pa;

η_{fan} is the energy transfer efficiency of the fan, it is taken to be 0.9.

Table 4.31 Extra power consumptions caused by pressure drop of the silica gel rotor

Cities and Climate Classes	Process airflow rate (L/s)	Pressure drop (Pa)	Extra power consumption (kW)	Extra power consumption in the whole season (kWh)
Copenhagen Summer Class 4	250	200	0.06	14.17
Copenhagen Summer Class 5	250	200	0.06	
Copenhagen Summer extreme	250	200	0.06	
Milan Summer Class 3	250	200	0.06	83.22
Milan Summer Class 4	250	200	0.06	
Milan Summer Class 5	250	200	0.06	
Milan Summer Extreme	250	200	0.06	
Colombo Summer Class 1	230	200	0.05	242.52
Colombo Summer Class 2	230	200	0.05	
Colombo Summer Class 3	230	200	0.05	
Colombo Summer Class 4	230	200	0.05	
Colombo Summer Class 5	230	200	0.05	
Colombo Summer extreme	230	200	0.05	
Copenhagen Winter Class 5	190	200	0.04	71.90
Copenhagen Winter Class 4	190	200	0.04	
Milan Winter Class 5	190	200	0.04	95.72
Milan Winter Class 4	190	200	0.04	
Milan Winter Class 3	190	200	0.04	
Milan Winter Class 2	190	200	0.04	

The energy saving of the CAHP compared to reference air source heat pump in different cities and seasons were revaluated with the extra power consumption caused by the pressure drop of the silica gel rotor. The results are shown in the following Table 4.32. The reevaluated annual energy saving of the CAHP compared to the reference air source heat pump is shown in Table 4.33.

Table 4.32 Reevaluation on the energy saving of CAHP in different cities and different seasons compared to reference air source heat pump

Cities and Seasons	Power consumption (kWh)		Energy saving
	CAHP	Air Source Heat Pump	CAHP to Air Source Heat Pump
Copenhagen Summer	157.32	356.99	55.93%
Milan Summer	1105.43	1749.97	36.83%

Cities and Seasons	Power consumption (kWh)		Energy saving
	CAHP	Air Source Heat Pump	CAHP to Air Source Heat Pump
Colombo Summer	9412.58	13908.81	32.33%
Copenhagen Winter	2087.42	2108.87	1.02%
Milan Winter	2550.68	2798.77	8.86%

Table 4.33 Reevaluation on the annual energy saving of CAHP in different cities compared to reference air source heat pump

Cities	Power consumption(kWh)		Energy saving
	CAHP	Air Source Heat Pump	CAHP to Air Source Heat Pump
Copenhagen	2244.74	2465.86	8.97%
Milan	3656.12	4548.74	19.62%
Colombo	9412.58	13908.81	32.33%

The annual energy saving of the CAHP compared to reference system which use air source heat pump in summer and gas boiler in winter in different cities were revaluated with the extra power consumption caused by the pressure drop of the silica gel rotor. The results are shown in the following Table 4.34.

Table 4.34 Reevaluation on annual energy saving of CAHP in different cities compared to reference system using air source heat pump in summer and gas boiler in winter

Cities	Primary energy use (kWh)		Energy saving
	CAHP	Air Source Heat Pump and Gas Boiler	CAHP to Air Source Heat Pump and Gas Boiler
Copenhagen	5611.86	10986.74	48.92%
Milan	7970.33	16686.47	52.23%

Overall, the pressure drop of the silica gel rotor will slightly decrease the energy saving proportion of the CAHP (from 2% to 4%). But the CAHP can still save substantial amount of energy compared to reference systems.

4.9 Summary of experimental results

From the experimental studies on the energy performance of the CAHP, the following conclusions may be summarized.

In summer of Copenhagen, Milan and Colombo, the CAHP tested can save 55.93%, 36.83%, 32.33% of power consumption respectively compared to conventional air source heat pump for ventilation and air-conditioning in the office room tested. The power consumption of the CAHP in summer is sensitive to the outdoor air humidity ratio.

In winter operation mode, under the experimental investigated cases, the CAHP can save 1.02% and 8.86% of power consumption compared to reference air source heat pump systems in Copenhagen and Milan for ventilation and heating in the office room tested. The CAHP can save 48.92% and 52.23% of primary energy compared to reference gas boiler systems in Copenhagen and Milan for ventilation and heating in the office room tested.

From the aspects of energy saving and indoor air quality, the CAHP could be a competitive choice for ventilation systems to reduce energy use and maintain a healthy, comfortable and productive indoor environment.

5 Discussions

Indoor air quality should get more attentions than it has got in modern society as people spend more time indoors. The new developed materials and chemicals are polluting indoor air with new contaminants and with an unprecedented load. Particle polluted outdoor air is becoming another indoor air pollution source through the ventilation system and infiltration. With the complex indoor air pollutants, how to achieve healthy indoor air quality is becoming a main problem and challenge in the researching field of indoor climate. Another issue which is challenging the indoor climate research is the building energy efficiency. As mentioned above, ventilation accounts for 30% of total building energy use. Technologies which can clean indoor air with less energy are urgently needed in buildings.

The silica gel rotor was found to have significant effect on indoor air purification especially on the cleaning of VOCs which are the most important indoor air pollutants. However, such an efficient technology has not been used in ventilation systems. In the study presented by this thesis, the using of silica gel rotor for indoor air purification in ventilation system was explored. The new ventilation system termed CAHP was initially designed to clean indoor air with an energy efficient method. The CAHP was theoretical modelled and experimental investigated in the study proposed, and was found to have a significant improvement on the energy efficiency of building ventilation. As any new technology, there are still several problems which should be solved during the developing and promoting of the CAHP, the suggestions for further research and development of the CAHP are given as well.

5.1 Energy performance compared to reference system

The energy performance of the CAHP was compared to reference systems. The advantages and disadvantages of the CAHP compared to conventional air source heat pump and gas boiler should be discussed.

5.1.1 Summer operation mode

Experimental studies show that the CAHP can save energy compared to conventional system in all the investigated summer climates. The energy saving of the CAHP compared to the reference system in summer mode is attributed to much less outdoor fresh air was used in the CAHP due to its high efficient air purification capacity, the reducing of cooling load and increased evaporating temperature benefited from non-dew point cooling.

Even though the CAHP was demonstrated to be more energy efficient than reference air source heat pump for building ventilation in all the investigated summer climates, the energy saving proportion was found quite different among Copenhagen, Milan and Colombo. To make the CAHP more competitive, some improvement points are summarized from the experimental studies. In the location where the outdoor air humidity ratio was higher, the energy saving proportion became lower. It was most probably due to the power consumption of the CAHP was sensitive to the outdoor air humidity ratio. Higher regeneration temperature was needed when outdoor air humidity ratio increased. Increasing regeneration temperature required higher condensing temperature of the heat pump which reduced the COP of the heat pump. Lower COP of CAHP than reference air source heat pump can happen in high outdoor humidity ratio climates, but the CAHP can still save energy in these cases due to its strong indoor air purification capacity. If the dehumidification capacity of the silica gel rotor could be reactivated by lower regeneration temperature in high humidity climates, the energy saving proportion of the CAHP could increase significantly.

5.1.2 Winter operation mode

Experimental studies show that the CAHP can save energy compared to reference air source heat pump in five of the six experimental investigated winter climates, and the CAHP can save primary energy compared to reference gas boiler in all the experimental investigated winter climates. Overall, the CAHP can save energy in winter of Copenhagen and Milan. It is worth to use CAHP for building ventilation. The energy saving of the CAHP compared to the reference air source heat pump system in winter mode is attributed to that much less outdoor fresh air was used in the CAHP due to its high air purification efficient, and the relative higher evaporating temperature. The energy saving of the CAHP compared to reference gas boiler is attributed to less outdoor airflow rate, and high COP of the heat pump in CAHP. The calculation works found that, with the primary energy factors of power and natural gas in Denmark and Italy, the energy use of air source heat pump can be lower than gas boiler if its COP is higher than 2.5 in Copenhagen (or 2.18 in Milan). During the experimental investigation, the COP of the heat pump in CAHP was measured to be in the range of 3.3 to 5.2. Higher COP and lower outdoor airflow rate make the CAHP save substantial amount of energy expense than reference gas boiler.

But the experimental studies were conducted only in the winter climates which have outdoor temperature higher than $0^{o}C$ due to the limitation of outdoor air handling unit. To investigate the energy performance of the CAHP under colder winter climates (Copenhagen winter class 3, 2, 1, extreme class and Milan winter class 1, extreme class), simulations were conducted with the theoretical model developed in section 3. The results are listed in the following Table 5.1 and Table 5.3. To show the changing trend of energy saving proportion under different winter climates, the experimental measured energy saving of CAHP under Copenhagen winter class 5, class 4 and Milan winter class 5, class 4, class 3, class 2 are also list in these tables. The extra power consumption caused by the pressure drop of silica gel rotor is included in the calculation and comparison.

Table 5.1 Instantaneous power consumption of CAHP, reference air source heat pump and energy saving of CAHP compared to reference air source heat pump in different cities and different winter climates

Cities and Climate Classes	Power consumption (kW)		Energy Saving
	CAHP	Air Source Heat Pump	CAHP to Air Source Heat Pump
Copenhagen Winter Class 5	0.72	0.72	0.89%
Copenhagen Winter Class 4	1.38	1.40	1.04%
Copenhagen Winter Class 3	1.65	2.17	24.29%
Copenhagen Winter Class 2	2.27	3.19	28.63%
Copenhagen Winter Class 1	2.99	4.33	30.94%
Copenhagen Extreme Winter	3.38	5.00	32.43%
Milan Winter Class 5	0.48	0.37	-27.45%
Milan Winter Class 4	0.68	0.69	0.80%
Milan Winter Class 3	1.14	1.13	-0.62%
Milan Winter Class 2	1.70	2.14	20.36%
Milan Winter Class 1	1.84	2.49	25.93%
Milan Extreme Winter	2.08	2.89	27.93%

From the figure above, it can be seen that the energy saving of CAHP compared to reference air source heat pump increased significantly in winter with the decrease of outdoor air temperature. The total energy saving of CAHP compared to reference air source heat pump in whole winter could be much higher than the experimental measured results.

Table 5.2 Total power consumption of CAHP, reference air source heat pump and energy saving of CAHP compared to reference system in whole winter climates of Copenhagen and Milan

Cities and seasons	Power consumption (kWh)		Energy saving
	CAHP	Air Source Heat Pump	CAHP to Air Source Heat Pump
Copenhagen Winter	3239.60	3648.20	11.20%
Milan Winter	2733.51	3045.69	10.25%

Table 5.2 gives the total power consumption of CAHP, reference air source heat pump and energy saving of CAHP in whole winter climates of Copenhagen and Milan. The CAHP could save 11.20% and 10.25% of power consumption compared to reference air source heat pump in Copenhagen and Milan respectively.

Another issue needs to be sated is that during the power consumption calculation of the reference air source heat pump, the air temperature at the outlet point of the evaporator will go quite low to get enough heat if outdoor air temperature is low. For example, in the Copenhagen extreme winter class when outdoor air temperature is -20.10^{o}C, the air temperature at the evaporator outlet of the heat pump will go to -39^{o}C to fulfill the heating load. The difference of condensing and evaporating temperature will be more than 75^{o}C, and this will make the heat pump difficult to operate. In these cases, two-stage heat pump should be carried out for building ventilation and space heating, but it will increase the complexity of the system. With two-stage heat pump, energy saving can happen only if the average COP of the two stages is double of the single-stage heat pump, but this is difficult to realize. In the CAHP, regeneration air and indoor exhaust air instead of outdoor air are used as heat source, thus the air coming out from the evaporator could be much higher than the reference air source heat pump. In the extreme case of Copenhagen, the air temperature at the evaporator outlet was calculated to be -13^{o}C. As a result, the CAHP improved the operation reliability in winter, and it can be a competitive succedaneum of air source heat pump.

Table 5.3 gives the energy saving of CAHP compared to reference gas boiler in all the winter climates.

Table 5.3 Instantaneous primary energy use of CAHP, reference gas boiler and energy saving of CAHP compared to gas boiler in different cities and different winter climates

Cities and Climate Classes	Primary energy use (kW)		Energy Saving
	CAHP	Gas boiler	CAHP to Gas boiler
Copenhagen Winter Class 5	1.79	4.32	58.61%
Copenhagen Winter Class 4	3.46	6.42	46.17%
Copenhagen Winter Class 3	4.12	7.47	44.94%
Copenhagen Winter Class 2	5.69	9.51	40.19%
Copenhagen Winter Class 1	7.47	11.54	35.24%
Copenhagen Extreme Winter	8.45	12.56	32.72%
Milan Winter Class 5	1.04	2.49	58.21%
Milan Winter Class 4	1.49	4.24	64.91%
Milan Winter Class 3	2.48	5.84	57.59%
Milan Winter Class 2	3.72	7.72	51.87%
Milan Winter Class 1	4.02	8.14	50.63%
Milan Winter Class Extreme	4.54	8.95	49.31%

It can be seen that the energy saving of CAHP decreased with the decrease of outdoor air temperature. This is due to that the COP of the heat pump in CAHP decreased when outdoor air temperature get lower, but the efficiency of gas boiler didn’t change much with the variable of outdoor air temperature. Overall, the CAHP could save substantial amount of energy expense compared to reference gas boiler.

Table 5.4 Total energy use of CAHP, reference gas boiler and energy saving of CAHP compared to reference gas boiler in whole winter climate of Copenhagen and Milan

Cities and Seasons	Primary energy use (kWh)		Energy saving
	CAHP	Gas boiler	CAHP to Gas boiler
Copenhagen Winter	8099.01	15241.50	46.86%
Milan Winter	5959.05	13678.68	56.44%

Table 5.4 gives the energy saving of the CAHP compared to reference gas boiler operated in winter modes, and it shows that the CAHP could save 46.86%, 56.44% of primary energy in whole winter of Copenhagen and Milan respectively. The CAHP could be a competitive choice for building ventilation and indoor climate control.

To further improve the energy performance of the CAHP, potential improvement points are summarized from the experimental studies. In the CAHP, some extra energy is consumed to heat the regeneration air, and this may not be suitable in winter when heat could be used for space heating directly. To reduce the heat for regeneration, the regeneration air may have a lower airflow rate. In the conducted experiments, the airflow rate ratio of the regeneration air to process air was 1:2, and this was demonstrated to have high air purification efficiency by previous study [88][89][115]. The 1:2 airflow rate ratio set for regeneration air and process air in the CAHP was demonstrated to have less energy use than reference gas boiler in all the investigated winter climates. Compared to reference air source heat pump, the CAHP can save energy in whole winter of Copenhagen and Milan. However, if the air purification effect could be achieved with less regeneration air, the energy saving proportion of the CAHP in winter modes could be higher. But the influence of low regeneration airflow rate on the silica gel rotor's air cleaning ability needs to be investigated further.

5.1.3 Influence of outdoor air temperature and outdoor air heating load proportion to the energy saving of CAHP in winter mode

Due to that much of the heating produced by the heat pump of CAHP is used for heating the regeneration air, the proportion of outdoor air heating load in total building heating load may affect the energy saving of CAHP. Another factor which affects the energy saving of CAHP is the outdoor air temperature. With the theoretical model of CAHP and calculation modes of air source heat pump and gas boiler, the influence of outdoor air heating load proportion and outdoor air temperature to CAHP energy saving can be simulated.

Simulations were conducted to investigate the energy saving of CAHP compared with reference systems under different outdoor air temperature and different outdoor airflow rates. In the simulation, the building thermal physical properties were assumed as shown in Table 4.1. The heat transfer through the interior walls, floor and doors was neglected due to equal temperature inside the building. The outdoor air temperature was assumed in the range from 10^{o}C to -15^{o}C with steps of 1^{o}C, and the number of indoor occupants was assumed to be from 3 to 15 with steps of 3 persons. The changing of indoor occupants represents the changing of outdoor airflow rates and thus the changing of outdoor air heating load proportions.

The simulation results are shown in the following Figure 5.1 and Figure 5.2.

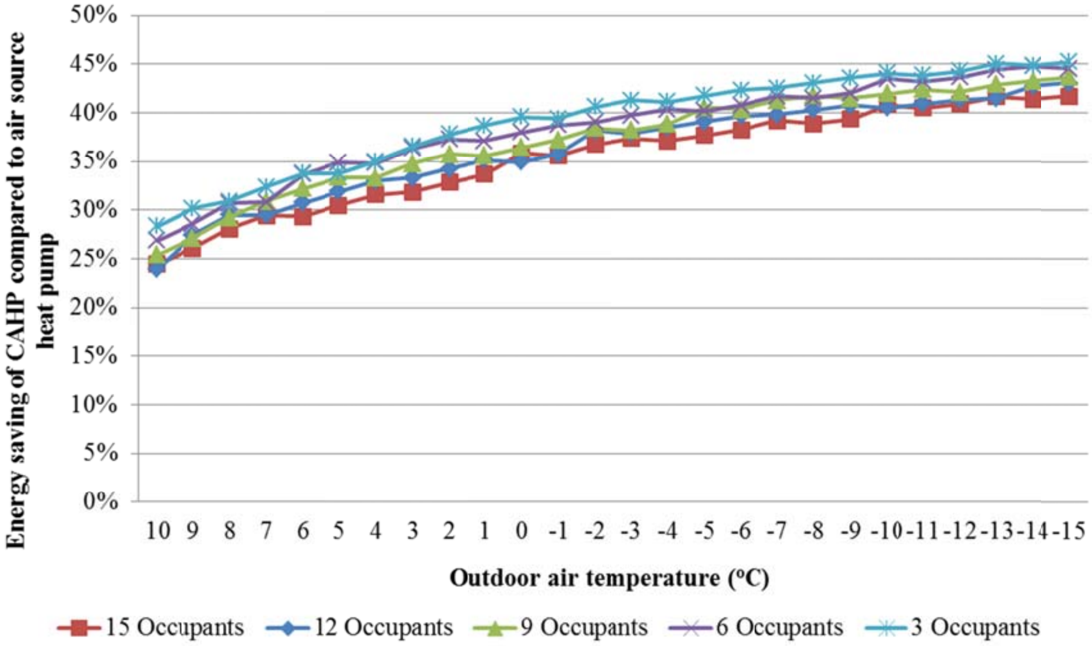


Figure 5.1 Simulation results of CAHP energy saving compared with air source heat pump under different outdoor climates and with different indoor occupants numbers

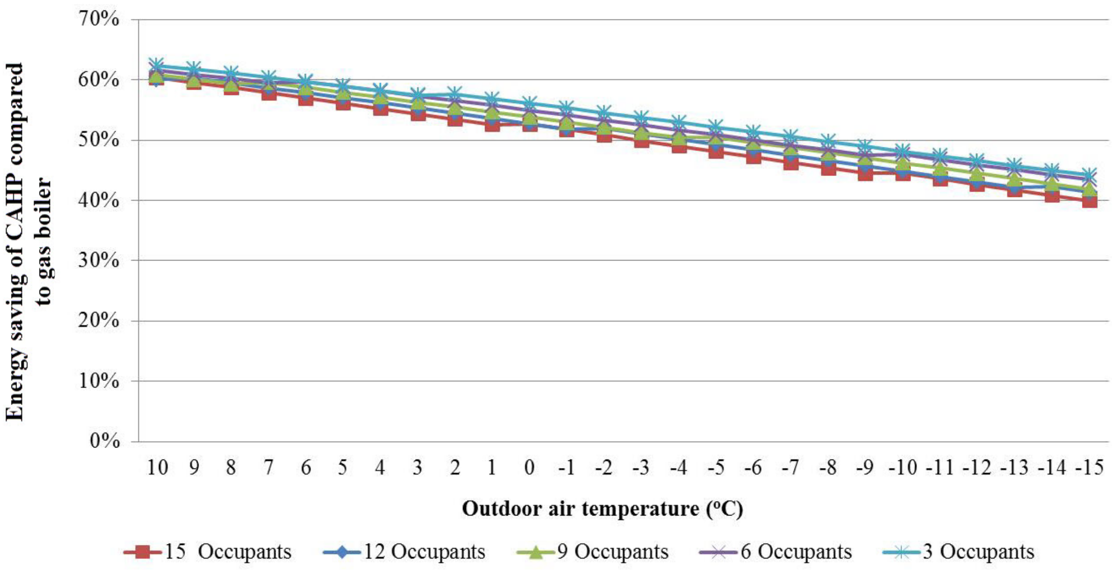


Figure 5.2 Simulation results of CAHP energy saving compared with gas boiler under different outdoor climates and with different indoor occupants occupants numbers

Figure 5.1 and Figure 5.2.show that the number of indoor occupants didn't affect the CAHP energy saving much. The energy saving of CAHP are almost equal with varied numbers of indoor occupants. The outdoor air temperature affects the energy saving of CAHP compared to either air source heat pump or gas boiler. But the trends are in oposite directions. This is consistent with the experimental measured results. Due to that the CAHP used indoor exhasut air and regeneration air (after regenerting work) as heating source, the COP of CAHP didn't reduce as much as air source heat pump did when outdoor air temperature decreased. The energy saving of CAHP compared with air source heat pump increased with the decrease of outdoor air temperature. But compared with gas boiler whose efficiency was almost constant under different outdoor air temperature, the energy saving of CAHP reduced when outdoor air temperature decreased.

5.1.4 CAHP compared to reference systems with heat recovery in winter mode

The sensible heat recovery unit has been popular used in the new built and renovated buildings of Copenhagen, Milan and other European cities. To better evaluate the performance of CAHP operated in winter climates, the CAHP should be further compared to reference systems with heat recovery unit. The comparisons were conducted by theoretical analyzing.

The energy use of the CAHP for winter operation mode can be written as:

$$E_{CAHP} = (Q_{indoor-CAHP} + Q_{vent-CAHP} + Q_{reg} - Q_{rotor-transfer}) / (COP_{heating-CAHP}) \tag{37}$$

Where,

E_{CAHP} is the energy use of the CAHP, kW ;

$Q_{indoor-CAHP}$ is the indoor thermal load of the test room for CAHP, kW ;

$Q_{vent-CAHP}$ is the heating load of outdoor air delivered to the test room for CAHP, kW ;

Q_{reg} is the heating load of regeneration air for CAHP, kW ;

$Q_{rotor-transfer}$ is the heat transferred from regeneration air to process air through silica gel rotor, kW ;

$COP_{heating-CAHP}$ is the COP for heating of the CAHP.

The energy use of the reference air source heat pump with heat recovery unit for winter operation mode can be written as:

$$E_{ref-ashp} = (Q_{indoor-ref} + Q_{vent-ref} - Q_{vent-ref-recovered}) / COP_{heating-ashp} \quad (38)$$

Where,

$E_{ref-ashp}$ is the energy use of the air source heat pump, kW ;

$Q_{indoor-ref}$ is the indoor thermal load of the test room for reference systems, kW ;

$Q_{vent-ref}$ is the heating load of outdoor air delivered to the test room for reference systems, kW ;

$Q_{vent-ref-recovered}$ is the heat recovered by the heat recovery unit, kW ;

$COP_{heating-ashp}$ is the COP for heating of the air source heat pump.

The $Q_{vent-ref-recovered}$ can be calculated as:

$$Q_{vent-ref-recovered} = Q_{vent-ref} * (t_{indoor} - 0) / (t_{indoor} - \min(0, t_{oudoor}))(1-\eta_{hr}) \quad (39)$$

η_{hr} is the efficiency of the heat recovery unit, the efficiency of cross-flow heat recovery unit can achieve 60%, the efficiency of rotary sensible heat recovery unit can achieve 70%, the efficiency of counter-flow heat recovery unit can achieve 80%.

The primary energy use of the reference gas boiler with heat recovery unit for winter operation mode can be written as:

$$E_{ref-gb} = (Q_{indoor-ref} + Q_{vent-ref} - Q_{vent-ref-recovered}) * \varphi_{ng} \quad (40)$$

Where,

E_{ref-gb} is the primary energy use of the gas boiler, kW ;

φ_{ng} is the primary energy factor of the natural gas.

To make the comparisons between the CAHP and air source heat pump impartial, the indoor thermal load for both systems should be equal, and the clean air deliver rate should be also equal in the two systems. Thus the outdoor air ventilation rate of the reference systems can be calculated with equation 33 mentioned above.

To better show the comparisons between the CAHP and the reference systems, a calculation example is given. In the example, the test room for the experimental study in chapter 4 was used. The outdoor air temperature was set to be $0^{o}C$. The outdoor airflow rates delivered to the ventilated room in the CAHP was 60L/s, the recirculation air in the CAHP was 190L/s. The outdoor airflow rates delivered to the ventilated room in the reference systems were 212L/s, the recirculation air in the CAHP was 38L/s.

With equations 37, 38, 39 and 40, the indoor thermal load for the CAHP was calculated to be 0.91kW. The heating load for the outdoor air in the CAHP was 1.60kW, and the heating load for the regeneration air in the CAHP was 3.45kW. With the silica gel rotor model, the heat transferred from the regeneration air to the process air was calculated to be 0.69kW. Thus, the total heating load of the heat pump in the CAHP was 5.27 kW. The indoor thermal load for the reference system was also 0.91kW, and the heating load for the outdoor air in the reference system was 5.65kW. With sensible heat recovery unit whose efficiency was 80%, the heating load for the outdoor air in the reference system could reduce to 1.13kW. Thus, the total heating load for the reference systems was 2.04kW. To realize energy saving of CAHP compared to air source heat pump with heat recovery, the COP of the CAHP should be 2.58 times of the COP of the air source heat pump. With the primary energy factors of power and natural gas in Copenhagen, the COP of the CAHP should be above 6.46 to achieve energy saving. The example shows how the CAHP can be more energy efficient than reference systems with counter-flow heat recovery unit, but this high requirement of COP of CAHP is difficult to be realized.

From the calculation, it was found that the heating for the regeneration air accounts for a big proportion of the total heating requirement for the heat pump in the CAHP. Decreasing regeneration airflow could be an effective method for decreasing the heating load of the CAHP and thus improve the energy efficiency of the CAHP. With the equations mentioned above, the energy saving of

CAHP with different regeneration airflow rates compared with reference systems with different heat recovery efficiency were analyzed and shown in the following figures.

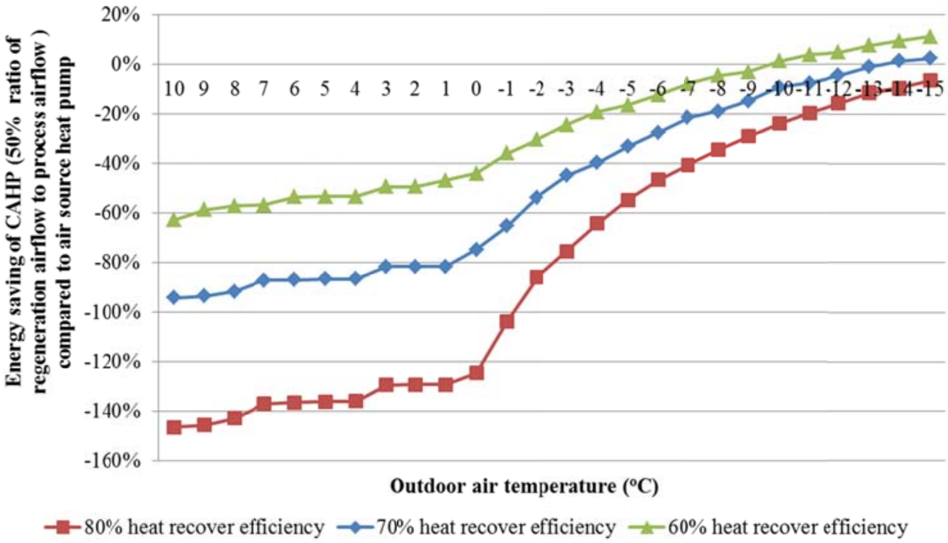


Figure 5.3 Energy saving of CAHP with 50% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit

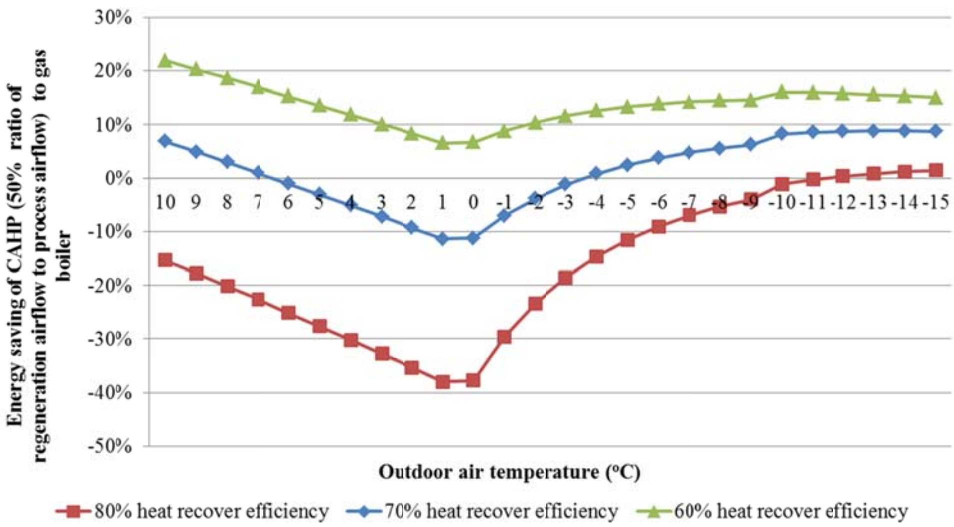


Figure 5.4 Energy saving of CAHP with 50% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit

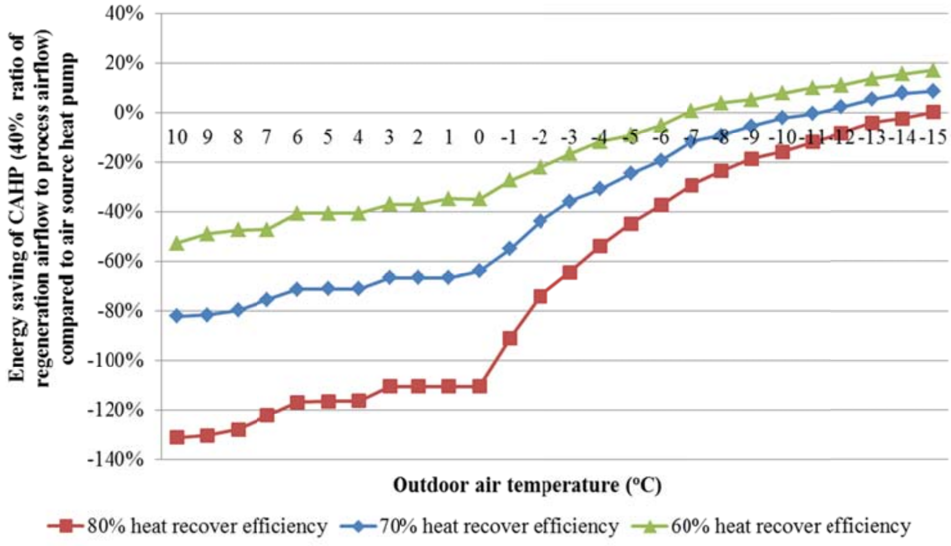


Figure 5.5 Energy saving of CAHP with 40% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit

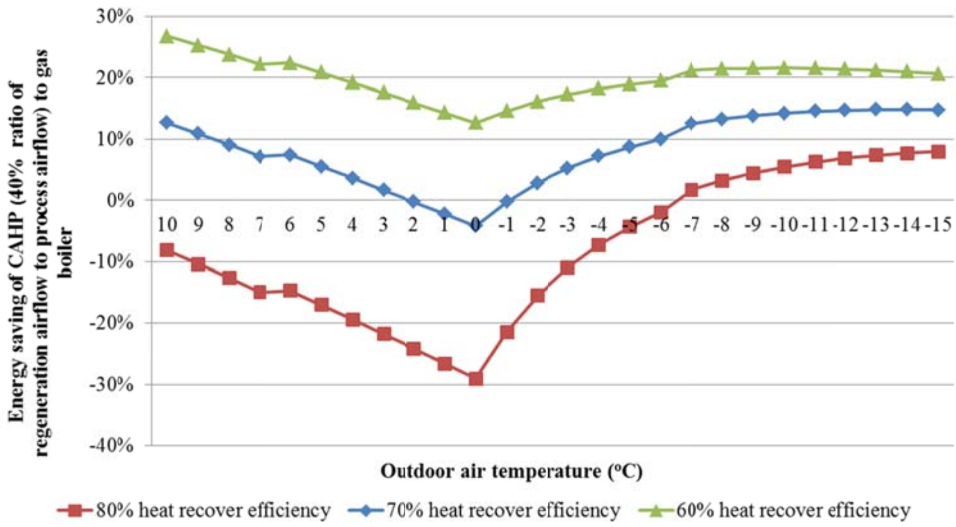


Figure 5.6 Energy saving of CAHP with 40% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit

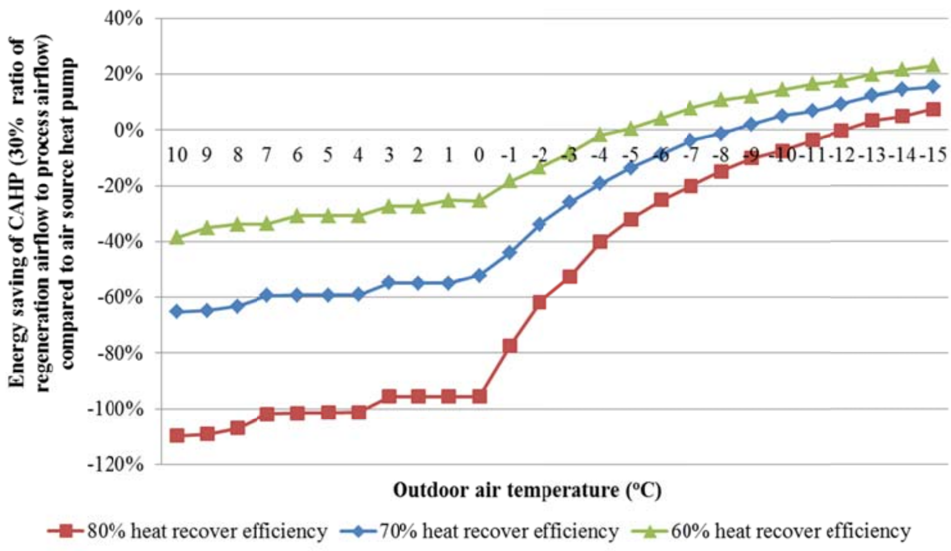


Figure 5.7 Energy saving of CAHP with 30% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit

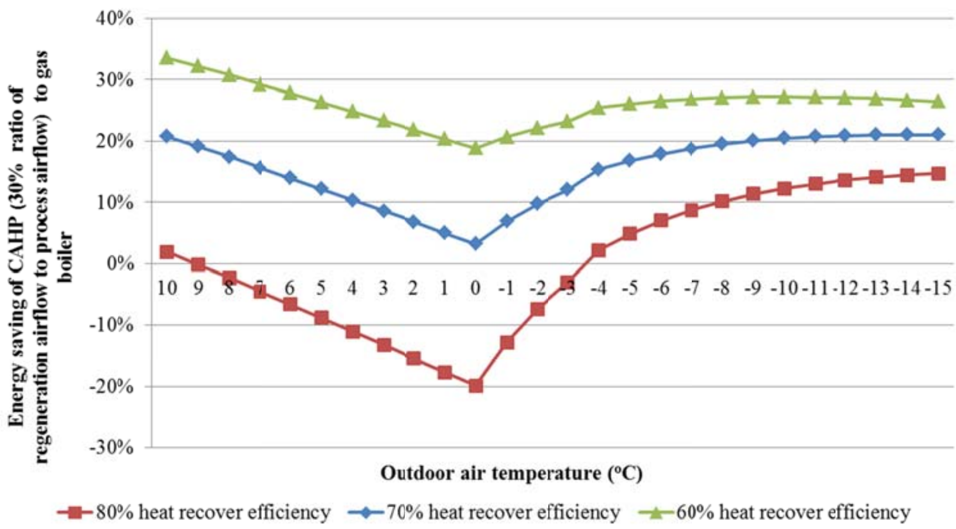


Figure 5.8 Energy saving of CAHP with 30% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit

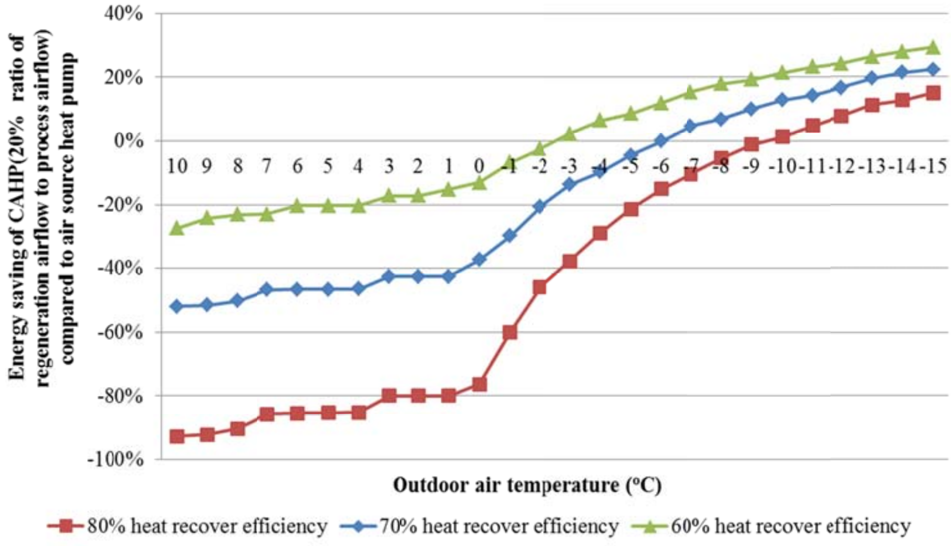


Figure 5.9 Energy saving of CAHP with 20% ratio of regeneration airflow to proceass airflow compared to air source heat pump with sensible heat recovery unit

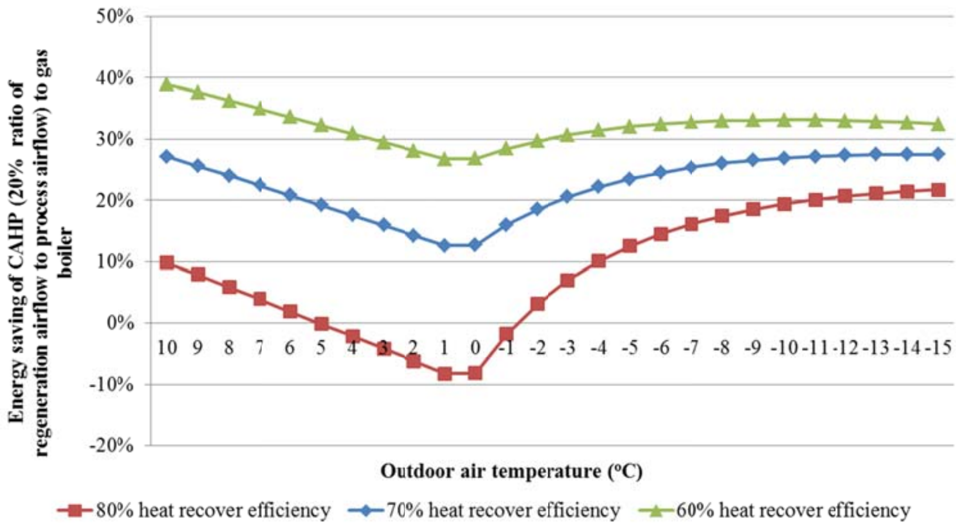


Figure 5.10 Energy saving of CAHP with 20% ratio of regeneration airflow to proceass airflow compared to gas boiler with sensible heat recovery unit

From the Figure 5.3, Figure 5.5, Figure 5.7 and Figure 5.9, it can be seen that the energy efficiency of CAHP was normally lower than air source heat pump with sensible heat recovery unit. But, as mentioned above, the air source heat pump use outdoor air as heating source. The air temperature at the outlet point of the evaporator will go quite low to get enough heat if outdoor air temperature is low. This will make the air source heat pump difficult to operate. The CAHP will use regeneration air and indoor exhaust air instead of outdoor air as heating source, thus the air coming out from the evaporator could be much higher than the reference air source heat pump. As a result, the CAHP improved the operation reliability in winter, and it can be a competitive succedaneum of air source heat pump.

Compared to gas boiler with cross-flow sensible heat recovery unit, the primary energy use of CAHP is lower in all the investigated cases. Compared to gas boiler with rotary sensible heat recovery unit, the primary energy use of CAHP is lower in most of the investigated cases. When the CAHP use 30% or 20% ratio of regeneration airflow rate to process airflow rate, it can save energy than gas boiler with cross-flow sensible heat recovery unit in all the investigated climates. But when the CAHP use 50% or 40% ratio of regeneration airflow rate to process airflow rate, it can save energy in most cases excepted when the outdoor air temperature was in the range from -3^{o}C to 6^{o}C, -1^{o}C to 2^{o}C respectively. Compared to gas boiler with counter-flow heat recovery unit, the CAHP didn't show significant energy conservation. For sure, decreasing the regeneration airflow rate will improve the energy performance of the CAHP, but its influence on the air cleaning ability of silica gel rotor needs to be studied further.

Another interesting thing is that the energy saving proportion of CAHP compared to reference systems shows turning points in the case when outdoor air temperature is 0^{o}C. This is mainly due to that pre-heating is needed to prevent frosting when outdoor air temperature is lower than 0^{o}C.

5.2 Suggestion for further research and development of CAHP

Based on the theoretical and experimental study on the innovative CAHP, the suggestions for further research and development of the CAHP are given as followings.

5.2.1 Regeneration temperature

As the dehumidification using silica gel rotor in the CAHP requires high regeneration temperature, this results in high condensing temperature of the heat pump. Even though the CAHP has been demonstrated to be more energy efficient than conventional ventilation systems in the experimental investigated cases, high condensing temperature could reduce the COP of the heat pump in CAHP. The solution for the high regeneration temperature could be either use low regeneration temperature rotor or select refrigerant that can provide higher COP than R134a at high condensing temperature. As mentioned above, the refrigerant used in the prototype CAHP is R134a which is commonly used in normal conventional heat pump. Though R134a meets all the requirements of condensing, evaporating pressure and have relative higher COP among the pure refrigerant candidates, the COP of the CAHP can be improved further if refrigerant which have better property in the required thermodynamic process was found. This could be a further research to improve the CAHP.

Low regeneration temperature silica gel rotor could reduce the condensing temperature of the heat pump and thus increase the energy efficiency of the CAHP. Low regeneration temperature silica gel desiccant rotor has been developed by “Munters A/S” with a big commercial available size. Shrinking the size of the low regeneration temperature silica gel rotor is not difficult, but is effect on indoor air VOCs purification needs further validation.

From the results above, it may be concluded that the CAHP is suitable for improving energy efficiency of ventilation systems in buildings. But the high regeneration temperature limits the energy saving of the CAHP since it decreases the COPs of the heat pump. Energy saving proportion of the CAHP will increase significantly if the dehumidification capacity of the silica gel rotor could be reactivated by low regeneration temperature or the high regeneration temperature could be fulfilled by more thermal efficient refrigerants. The research on low regeneration temperature silica gel rotor and more efficient refrigerant can be the key points on improving the energy performance of the CAHP.

5.2.2 Validation for indoor air cleaning capacity of the CAHP

The most important advantage of the CAHP was using silica gel rotor to clean indoor air and using less outdoor air for building ventilation. The indoor air cleaning was based on the VOCs removing capacity of the silica gel rotor. Theoretical and experimental studies have been conducted to investigate the VOCs removing capacity of the silica gel rotor, but experimental validation on indoor air purification of the CAHP has not been conducted. Since the control strategy of the CAHP was designed based on the thermal conditions of the air delivered to ventilated room, the regeneration temperature will change when outdoor thermal climate changes. It is necessary to test the indoor purification ability of the CAHP under different outdoor climates with experimental or field measurements.

6 Conclusions

An innovative CAHP which integrated heating, cooling, dehumidification, ventilation, air cleaning and energy recovery in one unit was proposed. The CAHP was designed based on the combination of silica gel rotor with heat pump. A theoretical model simulating the air cleaning effect and energy performance of the CAHP was established. The influence of outdoor thermal environments on air cleaning and energy performance of the CAHP were analyzed with the model. Laboratory experimental studies were conducted to investigate the energy performance of the CAHP. From the theoretical and experimental studies, following conclusions are drawn.

- Theoretical analyses show that the silica gel rotor is effective for removing VOCs from indoor air with a minimum efficiency of 65%. In average, 80% of the VOCs can be removed in all the investigated cases.
- Theoretical studies show that the outdoor humidity ratio plays an important role which influences the air cleaning and energy performance of the CAHP.
- Experimental studies show that the CAHP could save substantial amount of energy for ventilation systems compared to conventional systems.
- For the office room tested, the CAHP can save up to 55.93%, 36.83%, 32.33% of power consumption compared to conventional air source heat pump for ventilation and air-conditioning in summer climates of Copenhagen, Milan and Colombo.
- In winter operation mode, the CAHP can save 11.20% and 10.25% of power consumption compared to reference air source heat pump systems in Copenhagen and Milan for ventilation and heating in the office room tested.
- The CAHP can save 46.86% and 56.44% of primary energy compared to reference gas boiler in Copenhagen and Milan for ventilation and heating in the office room tested.
- Low regeneration temperature silica gel rotor can be used to further improve the energy performance of the CAHP. Research on using low regeneration temperature silica gel rotor in CAHP should be the further researched.
- Refrigerant which has higher COP than R134a within the condensing and evaporating temperature requirement of the CAHP could improve the energy performance of the CAHP, further research is suggested to be conducted on the more efficient refrigerant..
- Laboratory or field measurements are strongly suggested to validate the indoor air purification ability of the CAHP unit.

Overall, the CAHP can clean indoor air with a high VOCs removing efficiency. With the CAHP, less outdoor air can be used to ventilate buildings and rooms. Less outdoor air ventilation can decrease heating or cooling load for ventilation. Thus, energy conservation for ventilation in buildings can be realized. Less outdoor air ventilation can also decrease indoor particle concentrations which are introduced to indoors by ventilation. This is helpful to improve indoor air quality. From all the above, it may be concluded that the CAHP can be a competitive choice for indoor air cleaning and ventilation.

7 Reference

[1] P. Wargocki, J. Sundell, W. Bischof, G. Brundrett, P.O. Fanger, F. Gyntelberg, S.O. Hanssen, P. Harrison, A. Pickering, O. Seppänen, P. Wouters, Ventilation and health in non-industrial indoor environments, report from a European Multidisciplinary Scientific Consensus Meeting (EUROVEN), Indoor Air, 12(2)(2002) 113–128.

[2] D.P. Wyon, P. Wargocki, How indoor environment affects performance, Ashrae Journal, 55(3)(2013) 46–52.

[3] N. Nashihara, P. Wargocki, S. Tanabe, Cerebral blood flow, fatigue, mental effort, and task performance in offices with two different pollution loads, Building and Environment, 71(2014) 153–164.

[4] Z. Bakó-Biró, P. Wargocki, C.J. Weschler, P.O. Fanger, Effects of pollution from personal computers on perceived air quality, SBS symptoms and productivity in offices, Indoor Air, 14(3)(2004) 178–187.

[5] M. Orme, Energy impact of ventilation-estimates for the service and residential sectors, AIVC TN 49, AIVC, UK, 1998.

[6] CEN Report, CR 1752, “Ventilation for buildings-design criteria for the indoor environment”, European Committee for Standardization, 2001.

[7] ASHRAE, “Ventilation for acceptable indoor air quality”. ANSI/ASHRAE Standard 62.1-2013, American Society of Heating, Refrigerating and Air-Conditioning Engineers. Inc, Atlanta, 2013.

[8] M.V. Pettenkofer, Über den luftwechsel in wohngebäuden, Cotta’schen Buchhandlung, München, 1858.

[9] S. Stricker, M. Bourgeau, E. Fonberg, D. Parent, Physiological responses to elevated carbon dioxide levels in buildings, Indoor and Built Environment, 6(5)(1997) 301-308.

[10] O. Persson, C. Östberg, J. Pagels, A. Sebastian, Air contaminants in a submarine equipped with air independent propulsion. Journal of Environmental Monitoring, 8(11)(2006) 1111-1121.

[11] L. Kajtar, L. Herczeg, E. Lang, Influence of carbon-dioxide pollutant on human well-being and work intensity, In: Proceedings of Healthy Buildings, Lisboa, (1)(2006) 85-90.

[12] D.W. Dockery, C.A. Pope III, X. Xu, J.D. Spengler, J.H. Ware, M.E. Fay, Jr. B.G. Ferris, F.E. Speizer, An association between air pollution and mortality in six US cities, New England Journal of Medicine, 329(24)(1993) 1753–1759.

[13] C.A. Pope III, M.J. Thun, M.M. Namboodiri, D.W. Dockery, J.S. Evans, F.E. Speizer, Jr. C.W. Heath, Particulate air pollution as a predictor of mortality in a prospective study of US adults, American Journal of Respiratory and Critical Care Medicine, 151(3)(1995) 669–674.

[14] C.A. Pope III, R.T. Burnett, M.J. Thun, E.E. Calle, D. Krewski, K. Ito, G.D. Thurston, Lung cancer, cardiopulmonary mortality and long-term exposure to fine particulate air pollution, The Journal of American Medical Association, 287(9)(2002) 1132–1141.

[15] U.S. Environmental Protection Agency, Report to congress on indoor air quality: Volume 2. EPA/400/1-89/001C. Washington, DC. 1989.

[16] S. Batterman, C. Godwin, C.R. Jia, Long duration tests of room air filters in cigarette smokers' homes, Environmental Science & Technology, 39(2005) 7260–7268.

[17] H. Schleibinger, H. Rüden, Air filters from HVAC systems as possible source of volatile organic compounds (VOC) - laboratory and field assays, Atmospheric Environment, 33(28)(1999) 4571–4577.

[18] H.L. Higdon III, J.E. Graves, D.W. Blackhurst, W.R. Boone, Air quality within the incubator: will volatile organic compound (VOC) filters make a difference in in vitro fertilization? Fertility and Sterility, 80(3)(2003) 260.

[19] O.M. Alm, G. Clausen, P.O. Fanger, Exposure-response relationships for emissions from used ventilation filters, Proceeding of 6th International Conference on Healthy Buildings, Helsinki, Finland, 2(2000) 245–250.

[20] G. Bekö, O. Halás, G. Clausen, C.J. Weschler, Initial studies of oxidation processes on filter surfaces and their impact on perceived air quality, Indoor Air, 16(1)(2006) 56–64.

[21] ECA (European Concerted Action "Indoor Air Quality & its Impact on Man"), Determination of VOCs emitted from indoor materials and products. Inter laboratory comparison of small chamber measurements, Report No. 13 (EUR 15054 EN) Luxembourg: Office for Official Publications of the European Communities, 1993.

[22] O. Hov, A. Sorteberg, N. Schmidbaue, H. Lattila, S. Solberg, N.Z. Heidam, F. Stordal, D. Simpson, A. Lindskog, H. Areskoug, P. Oyola, European VOC emission estimates

evaluated by measurements and model calculations, Journal of Atmospheric Chemistry, 28(3)(1997) 173–193.

[23] WHO, Formaldehyde, 2-Butoxyethanol and 1-tert-Butoxy-2-propanol, IARC Monographs on the Evaluation of Carcinogenic Risks to Humans, 88(2004) 2-9.

[24] G. Zhang, Study on the air cleaning effect of silica gel rotors, Tianjin University, PhD Thesis, 2007.

[25] B. Berglund, The role of sensory reaction as guides for non-industrial indoor air quality. In: D. M. Weekes & R. B. Gammage (Eds.), The practitioners approach to indoor air quality investigations. Akron, Ohio: American Industrial Hygiene Association, (1990) 113–130.

[26] W.S. Cain, Perception of odor intensity and the time course of olfactory adaptation, ASHRAE Transactions, 80(1)(1974) 53–75.

[27] D. Otto, L. Molhave, G. Rose, H.K. Hudnell, D. House, Neurobehavioral and sensory irritant effects of controlled exposure to a complex mixture of volatile organic compounds, Neurotoxic ology and Teratology, 12(6)(1990) 649–652.

[28] U. Satish, M.J. Mendell, K. Shekhar, T. Hotchi, D. Sullivan, S. Streufert, W.J. Fisk, Is CO_2 an indoor pollutant? direct effects of low-to-moderate CO_2 concentrations on human decision-making performance, Environmental Health Perspectives, 120(12)(2012) 1671–1705.

[29] L.P. Lombard, J. Ortiz, C. Pout, A review on building energy consumption information, Energy and buildings, 40(3)(2008) 394–398.

[30] L. Fang, S. Yuan, J. Nie, Experimental evaluation of a total heat recovery unit with polymer membrane foils. Proceedings of the 8th International Symposium on Heating, Ventilation and Air Conditioning, Xi'an, China, 23(2014): 235–242.

[31] M. Rabbia, G. Dowse, Understanding energy wheels and energy recovery ventilation technology, Energy Recovery Ventilation. Carrier Corporation, Syracuse, New York, 2000.

[32] J.H. Pejtersron, Sensory air pollution caused by rotary heat exchangers. Proceeding of 7th International Conference on Indoor Air Quality and Climate, Nagoya, Japan 3(1996) 459–464.

[33] G.A. Khoury, S.N. Chang, D.A. Lessley, A.A. Abdelghani, A.C. Anderson, An investigation of re-entrainment of chemical fume hood exhaust air in a heat recovery unit, American Industrial Hygiene Association Journal, 49(2)(1988) 61–65.

[34] E.L. Hult, H. Willem, M.H. Sherman, Formaldehyde transfer in residential energy recovery ventilators, Building and Environment, 75(2014) 92–97.

[35] M.J. Costello, A.G. Fane, P.A. Hogan, R.W. Scholfield, The effect of shell side hydrodynamics on the performance of axial flow hollow fiber modules, Journal of Membrane Science, 80(1)(1993) 1–11.

[36] H. Fan, C.J. Simonson, R.W. Besant, W. Shang, Performance of a run-around system for HVAC heat and moisture transfer applications using cross-flow plate exchangers coupled with aqueous lithium bromide, HVAC&R Research, 12(2)(2006) 313–336.

[37] A. Gabelman, S. T. Hwang, Hollow fiber membrane contactors. Journal of Membrane Science, 159(1-2)(1999) 61–106.

[38] K.R. Kistler, E.L. Cussler, Membrane modules for building ventilation, Chemical Engineering Research and Design, 80(1)(2002) 53–64.

[39] M.D. Larson, R.W. Besant, C.J. Simonson, The effect of membrane deflections on flow rate in crossflow air-to-air exchangers, HVAC&R Research, 14(2)(2008) 275–288.

[40] J. Min, M. Su, Performance analysis of a membrane-based energy recovery ventilator: effects of membrane spacing and thickness on the ventilator performance, Journal of Membrane Science, 30(8–9)(2010) 991–997.

[41] J. Min, M. Su, Performance analysis of a membrane-based enthalpy exchanger: effects of the membrane properties on the exchanger performance, Journal of Membrane Science,348(1-2)(2010) 376–382.

[42] J.L. Niu, L.Z. Zhang, Membrane based enthalpy exchanger: material considerations and clarification of moisture resistance, Journal of Membrane Science, 189(2)(2001) 179–191.

[43] L.Z. Zhang, Y. Jiang, Heat and mass transfer in a membrane-based energy recovery ventilation, Journal of Membrane Science, 163(1)(1999) 29–38.

[44] L.Z. Zhang, Fabrication of a lithium chloride solution based composite supported liquid membrane and its moisture permeation analysis, Journal of Membrane Science, 276(1-2)(2006) 91–100.

[45] L.Z. Zhang, Y.Y. Wang, C.L. Wang, H. Xiang, Synthesis and characterization of a PVA/LiCl blend membrane for air dehumidification, Journal of Membrane Science, 308(1-2)(2008) 198–206.

[46] L.Z. Zhang, Heat and mass transfer in plate-fin enthalpy exchangers with different plate and fin materials, International Journal of Heat Mass Transfer, 52(11-12)(2009) 2704–2713.

[47] L.Z. Zhang, X.R. Zhang, Q.Z. Miao, L.X. Pei, Selective permeation of moisture and VOCs through polymer membranes used in total heat exchangers for indoor air ventilation, Indoor Air, 22(4)(2012) 321–330.

[48] J. Nie, J. Yang, L. Fang, X. Kong, Experimental evaluation of enthalpy efficiency and gas-phase contaminant transfer in an enthalpy recovery unit with polymer membrane foils, Science and Technology for the Built Environment, 21(2)(2015) 150-159.

[49] D.G. Waugaman, A. Kini, C.F. Kettleborough, A review of desiccant cooling systems, Journal of Energy Resources Technology, 115(1)(1993) 1–8.

[50] T.A. Ameel, K.G. Gee, B.D. Wood, Performance predictions of alternative, low cost absorbents for open-cycle absorption solar cooling, Solar Energy, 54(2)(1995) 65–73.

[51] C.S.K. Ahmed, P. Gandhisan, A.A. Al-Farayedhi, Simulation of a hybrid liquid desiccant based air-conditioning system, Applied Thermal Engineering, 17(2)(1997) 125–134.

[52] C.H. Liang, L.Z. Zhang, L.X. Pei, Independent air dehumidification with membrane-based total heat recovery: modeling and experimental validation, Journal of Refrigeration, 33 (2)(2010) 398–408.

[53] L.Z. Zhang, Energy performance of independent air dehumidification systems with energy recovery measures, Energy, 31(8–9)(2006) 1228–1242.

[54] J.Z. Ling, O. Kuwabara, Y. Hwang, R. Radermacher, Experimental evaluation and performance enhancement prediction of desiccant assisted separate sensible and latent cooling air-conditioning system, International Journal of Refrigeration, 34(4)(2011) 946–957.

[55] X.Y. Chen, Z. Li, Y. Jiang, K.Y. Qu, Field study on independent dehumidification air-conditioning system – II. Performance of the whole system, ASHRAE Transactions, 111(2)(2005) 277–284.

[56] Q. Ma, R.Z. Wang, Y.J. Dai, X.Q. Zhai, Performance analysis on a hybrid air conditioning system of a green building, Energy and Buildings, 38(5)(2006) 447–453.

[57] Y. Zhang, J. Mo, Y. Li, J. Sundell, P. Wargocki, J. Zhang, J.C. Little, R. Corsi, Q. Deng, M.H.K. Leung, L. Fang, W. Chen, J. Li, Y. Sun, Can commonly-used fan-driven air

cleaning technologies improve indoor air quality? A literature review, Atmospheric Environment, 45 (2011) 4329–4343.

[58] T.N. Obee, R.T. Brown, TiO_2 photo-catalysis for indoor air applications – effects of humidity and trace contaminant levels on the oxidation rates of formaldehyde, toluene, and 1,3-butadiene, Environmental Science & Technology, 29 (1995) 1223–1231.

[59] W.D. Ellis, P.V. Tometz, Room-temperature catalytic decomposition of ozone, Atmospheric Environment, 6 (1972) 70.

[60] C.W. Kwong, C.Y.H. Chao, K.S. Hui, M.P. Wan, Catalytic ozonation of toluene using zeolite and MCM-41 materials, Environmental Science & Technology, 42 (2008) 8504–8509.

[61] C.H. Ao, S.C. Lee, Enhancement effect of TiO_2 immobilized on activated carbon filter for the photo degradation of pollutants at typical indoor air level, Applied Catalysis B: Environmental, 44 (2003) 191–205.

[62] D.S. Muggli, J.T. McCue, J.L. Falconer, Mechanism of the photo-catalytic oxidation of ethanol on TiO_2, Journal of Catalysis, 173 (1998) 470–483.

[63] A.T. Hodgson, H. Destaillats, D.P. Sullivan, W.J. Fisk, Performance of ultraviolet photo-catalytic oxidation for indoor air cleaning applications, Indoor Air, 17(2007) 305–316.

[64] J.H. Mo, Y.P. Zhang, Q.J. Xu, Y.F. Zhu, J.J. Lamson, R.Y. Zhao, Determination and risk assessment of by-products resulting from photo-catalytic oxidation of toluene, Applied Catalysis B: Environmental, 89(2009) 570–576.

[65] L.J. Joaquin, J.H. Mo, Y.P. Zhang, Q.J. Xu, R.Y. Zhao Photo-catalytic purification of volatile organic compounds in indoor air: A literature review, Atmospheric Environment, 43(14)(2009) 2229–2246.

[66] C. Kwong, C.Y.H. Chao, K.S. Hui, M.P. Wan, Removal of VOCs from indoor environment by ozonation over different porous materials, Atmospheric Environment, 42 (2008) 2300–2311.

[67] T. J. Phillips, D. P. Bloudoff, P. L. Jenkins, and K. R. Stroud, Ozone emissions from a “personal air purifier”, Journal of Exposure Analysis and Environmental Epidemology, 9(6) (1999) 594–601.

[68] M.S. Waring, J.A. Siegel, R.L. Corsi, Ultrafine particle removal and generation by portable air cleaners, Atmospheric Environment, 42(2008) 5003–5014.

[69] G. Bekö, G. Clausen, C.J. Weschler, Sensory pollution from bag filters, carbon filters and combinations, Indoor Air, 18(2008) 27–36.

[70] T.A. Metts, S.A. Batterman, Effect of VOC loading on the ozone removal efficiency of activated carbon filters, Chemosphere, 62(2006) 34–44

[71] P. Zhao, J.A. Siegel, R.L. Corsi, Ozone removal by HVAC filters, Atmospheric Environment, 41(2007) 3151–3160.

[72] H. Schleibinger, H. Rüden, Air filters from HVAC systems as possible source of volatile organic compounds (VOC) - laboratory and field assays, Atmospheric Environment, 33(1999) 4571–4577.

[73] M.Hyttinen , P.Pasanen, M.Björkroth, P.Kalliokoski, Odors and volatile organic compounds released from ventilation filters, Atmospheric Environment, 41(19)(2007) 4029–403.

[74] G. Bekö, M.O. Fadeyi, G. Clausen, C.J. Weschler, Sensory pollution from bag-type fiberglass ventilation filters: conventional filter compared with filters containing various amounts of activated carbon, Building and Environment, 44(2009) 2114–2120.

[75] Stacy L. Daniels, On the ionization of air for removal of noxious effluvia (Air ionization of indoor environments for control of volatile and particulate contaminants with nonthermal plasmas generated by dielectric-barrier discharge), IEEE Transactions on Plasma Science, 30(4)(2002) 1471-1481.

[76] J.H. Park, J.H. Byeon, K.Y. Yoon, J. Hwang, Lab-scale test of a ventilation system including a dielectric barrier discharger and UV photo-catalyst filters for simultaneous removal of gaseous and particulate contaminants, Indoor Air, 18(2008) 44–50.

[77] J. Van Durme, J. Dewulf, W. Sysmans, C. Leys, H. Van Langenhove, Efficient toluene abatement in indoor air by a plasma catalytic hybrid system, Applied Catalysis B: Environmental, 74(2007) 161–169.

[78] J. Van Durme, J. Dewulf, K. Demeestere, C. Leys, H. Van Langenhove, Post-plasma catalytic technology for the removal of toluene from indoor air: effect of humidity, Applied Catalysis B: Environmental, 87(2009) 78–83.

[79] S. Futamura, A. H. Zhang, and T. Yamamoto, The dependence of nonthermal plasma behaviour of VOCs on their chemical structures, Journal of Electrostatic, 42(1–2)(1997) 51–62.

[80] ASHRAE, ASHRAE Handbook Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers. Inc., Atlanta, 2005, 22: 4.

[81] W. Chen, J.S. Zhang, and Z. Zhang, Performance of air cleaners for removing multiple volatile organic compounds in indoor air, ASHRAE Transaction, 111(1)(2005) 1101–1114.

[82] A. Wisthaler, P. Strøm-Tejsen, L. Fang, T.J. Arnaud, A. Hansel, T.D. Märk, D.P. Wyon, PTR-MS assessment of photo catalytic and sorption-based purification of recirculated cabin air during simulated 7-h flights with high passenger density, Environmental Science and Technology, 41(1)(2007) 229–234.

[83] A. Wisthaler, E.C. Apel, J. Bossmeyer, A. Hansel, W. Junkermann, R. Koppmann, R. Meier, K. Müller, S.J. Solomon, R. Steinbrecher, R. Tillmann, T. Brauers, Technical note: Inter comparison of formaldehyde measurements at the atmosphere simulation chamber SAPHIR, Atmospheric Chemistry and Physics, 8(2008) 2189–2200.

[84] A.L. Hines, T.K. Ghosh, Investigation of co-sorption of gases and vapors as a means to enhance indoor air quality – phase 2: Water vapor uptake and removal of chemical pollutants by solid adsorbents, Chicago, Report of Gas Research Institute: GRI-92/0157.2. (1993).

[85] A.L. Hines, T.K. Ghosh, S.K. Loyalka, R.C. Warder, Investigation of co-sorption of gases and vapors as a means to enhance indoor air quality – Phase 2: A summary of pollutant removal capabilities of solid and liquid desiccants from indoor air, Chicago, Report of Gas Research Institute GRI-92/0157.1.(1993).

[86] S.L. Kuo, A.L. Hines, Adsorption of chlorinated hydrocarbon pollutants on silica gel, Separation Science and Technology, 23(4–5)(1988) 293–303.

[87] M. Popescu, T.K. Ghosh, Dehumidification and simultaneous removal of selected pollutants from indoor air by a desiccant wheel using a 1M type desiccant, Journal of Solar Energy Engineering, 121(1)(1999) 1–13.

[88] L. Fang, G. Zhang, A. Wisthaler, Desiccant wheels as gas-phase absorption (GPA) air cleaners: evaluation by PTR-MS and sensory assessment, Indoor Air, 18(5)(2008) 375–385.

[89] G. Zhang, Y.F. Zhang, L. Fang, Theoretical study of simultaneous water and VOCs adsorption and desorption in a silica gel rotor, Indoor Air, 18(1)(2008) 37–43.

[90] Munters, Desiccant wheels-Advanced rotor technology, www.munters.com, 2011.

[91] N.A. Pennington, Humidity changer for air conditioning. U.S. Patent No.2700537, January 1955.

[92] P. Mavroudaki, C.B. Beggs, P.A. Sleigh, The potential for solar powered single-stage desiccant cooling in southern Europe, Applied Thermal Engineering, 22(3)(2002)1129–1140.

[93] P. Bourdoukan, E. Wurtz, P. Joubert, M. Sperandio, Potential of solar heat pipe vacuum collectors in the desiccant cooling process: modeling and experimental results, Solar Energy, 82(2008) 1209–1219.

[94] N. Enteria, H. Yoshino, A. Mochida, R. Takaki, A. Satake, R. Yoshie, T. Mita-mura, S. Baba, Construction and initial operation of the combined solar thermal and electric desiccant cooling system, Solar Energy, 83(2009) 1300–1311.

[95] G. Angrisani, F. Minichiello, C. Roselli, M. Sasso, Desiccant HVAC system driven by a micro-CHP: experimental analysis, Energy and Buildings, 42(2010) 2028–2035.

[96] G. Angrisani, F. Minichiello, C. Roselli, M. Sasso, Experimental investigation to optimise a desiccant HVAC system coupled to a small size co-generator, Applied Thermal Engineering, 31(2011) 506–512.

[97] J. Fischer, K. Mescher, B. Elkin, S.M. McCune, J. Gresham, High-performance schools, ASHRAE Journal, 49(5)(2007) 30–46.

[98] C.X. Jia, Y.J. Dai, J.Y. Wu, R.Z. Wang, Analysis on a hybrid desiccant air conditioning systems, Applied Thermal Engineering, 26(2006) 2393–2400.

[99] Y.F. Zhang, X.W. Hu, Z.S. Mao, L.L. Wei, C.M. Chen, H. Hao, Cycle performance of heat pump in desiccant cooling system, Journal of Chemical industry and Engineering, 20(2009) 2177–2182.

[100] Y. Sheng, Y. Zhang, N. Deng, L. Fang, J. Nie, L. Ma, Experimental analysis on performance of high temperature heat pump and desiccant wheel system, Energy and Buildings, 66(2013) 505–513.

[101] H. Hao, Y.F. Zhang, G.B. Zhou, Feasibility study of moderate or high temperature heat pump coupling to desiccant wheel hybrid system, Journal of Shenyang Jianzhu University, 24(4)(2008) 675–678.

[102] N. Subramanyam, M.P. Maiya, and M. Srinivasa, Application of desiccant rotor to control humidity in air-conditioning systems, Applied Thermal Engineering, 24(2004) 2777–2788.

[103] J.C. Sheridan, J.W. Mitchell, A hybrid solar desiccant cooling system, Solar Energy, 34(1985) 187–193.

[104] J.J. Jurinak, J.W. Mitchell, W.A. Beckman, Open-cycle desiccant air conditioning as an alternative to vapor compression cooling in residential applications, Journal of Solar Energy Engineering, 106(1984) 252–260.

[105] F.E. Nia, D. Paassen, M.H. Saidi, Modeling and simulation of desiccant wheel for air conditioning, Energy and Buildings, 38(10)(2006) 1230–1239.

[106] X.J. Zhang, Y.J. Dai, R.Z. Wang, A simulation study of heat and mass transfer in a honeycombed rotary desiccant dehumidifier, Applied Thermal Engineering, 23(2003) 989–1003.

[107] O.A. Hougen, W.R. Jr. Marshall, Adsorption from a fluid stream flowing through a stationary granular bed, Chemical. Engineering Progress, 43(4)(1947) 197–208.

[108] J.Y. San, Heat and mass transfer in a two-dimensional cross-flow regenerator with a solid conduction effect, International Journal of Heat and Mass Transfer, 36(3)(1993) 633–643.

[109] D. Sprung, C. Jost, T. Reiner, A. Hansel, A. Wisthaler, Acetone and acetonitrile in the tropical Indian Ocean boundary layer and free troposphere: aircraft-based inter comparison of APCIMS and PTR-MS measurements, Journal of Geophysical Research,106(D22)(2001) 28.511–28.528.

[110] J. Nie, L. Fang, B.W. Olesen, Experimental evaluation on energy performance of innovative clean air heat pump for indoor environment control in summer and winter seasons, Energy and Buildings, 84(2014) 641–650.

[111] C.E. Dorgan, J.S. Ellleson, Cold air distribution, ASHRAE Transaction, 12(3)(1988) 2004–2008.

[112] Munters, MLT800 product information, www.munters.com, 2011.

[113] L. Fang, A. Melikov, B.W. Olesen, HVAC Systems in a field laboratory for Indoor Climate Study, Proceedings of Healthy Buildings 2012 Conference, Session 7A.5, Brisbane.

[114] J. Kurnitski, Cost-Optimal and Nearly Zero-Energy Buildings, Springer, 2013.

[115] G. Molinaro, Theoretical calculation on energy reduction for ventilation on buildings by desiccant air conditioning integrated with air cleaning, Technical University of Denmark, Master thesis, 2010.

8 Appendix

The parameters including airflow rates, temperatures and humidity ratios at the test points shown in Figure 2.3 and Figure 2.4 are listed in the following tables. All the values are the average of the data recorded during the steady state period of tests.

Table 8.1 Airflow rates measured in experiments for summer climates

Cities and Climates	Return Air (L/s)	Fresh Air (L/s)	Cleaned Air (L/s)	Supply Air (L/s)	Regeneration Air (L/s)	Air for Excess Heat L/s)	Total Exhaust Air (L/s)
Copenhagen Summer Class 4	250.89	58.60	251.01	259.98	121.48	119.57	297.15
Copenhagen Summer Class 5	251.37	59.36	250.95	259.25	122.88	121.59	303.63
Copenhagen Summer Extreme	250.48	59.19	250.74	257.99	122.27	121.26	303.53
Milan Summer Class 3	248.35	60.34	247.95	256.35	123.20	122.36	302.74
Milan Summer Class 4	250.58	59.89	252.70	259.80	121.45	120.96	300.43
Milan Summer Class 5	248.78	59.55	248.60	255.14	122.03	121.73	301.08
Milan Summer Extreme	248.23	59.73	248.37	254.97	121.49	121.67	303.08
Colombo Summer Class 1	234.82	40.30	232.61	240.26	111.17	129.61	298.86
Colombo Summer Class 2	233.73	40.47	233.65	240.45	109.79	129.30	302.76
Colombo Summer Class 3	231.48	41.69	233.49	239.20	112.85	132.08	305.07
Colombo Summer Class 4	232.27	41.63	235.61	240.52	113.84	131.20	305.96
Colombo Summer Class 5	232.74	40.95	233.33	239.53	113.45	131.05	303.74
Colombo Summer Extreme	229.77	41.18	233.36	238.60	114.19	130.75	307.52

Table 8.2 Temperatures measured in experiments for summer climates

Cities and Climates	Return Air (°C)	Fresh Air (°C)	Mixed Air (°C)	Cleaned Air (°C)	Supply Air (°C)	Regeneration Air (°C)	Air after Regeneration (°C)	Air for Excess Heat (°C)
Copenhagen Summer Class 4	25.04	23.54	24.69	26.15	18.18	29.58	26.21	37.69
Copenhagen Summer Class 5	25.32	29.19	26.19	30.11	18.40	38.48	30.95	47.67

Cities and Climates	Return Air (°C)	Fresh Air (°C)	Mixed Air (°C)	Cleaned Air (°C)	Supply Air (°C)	Regeneration Air (°C)	Air after Regeneration (°C)	Air for Excess Heat (°C)
Copenhagen Summer Extreme	26.26	31.88	27.62	32.57	18.34	43.75	33.99	55.53
Milan Summer Class 3	25.46	19.83	24.04	27.28	18.56	35.61	28.32	23.29
Milan Summer Class 4	25.83	25.78	25.87	30.71	18.56	42.67	32.65	33.40
Milan Summer Class 5	25.74	30.56	26.81	33.41	18.04	49.34	35.63	44.45
Milan Summer Extreme	25.94	32.66	27.64	35.48	18.63	55.70	38.06	44.80
Colombo Summer Class 1	25.08	20.63	24.37	31.61	19.63	49.74	34.52	23.12
Colombo Summer Class 2	25.06	24.98	25.24	34.64	17.97	60.16	39.01	28.55
Colombo Summer Class 3	25.36	28.37	26.07	36.26	17.96	64.33	41.10	32.95
Colombo Summer Class 4	24.99	32.52	26.48	35.68	18.25	61.10	40.47	37.20
Colombo Summer Class 5	25.70	36.03	27.89	36.05	18.13	57.62	39.54	55.92
Colombo Summer Extreme	25.46	38.28	28.15	37.10	18.30	62.17	41.54	53.16

Table 8.3 Humidity ratios measured in experiments for summer climates

Cities and Climates	Return Air (g/kg)	Fresh Air (g/kg)	Mixed Air (g/kg)	Cleaned Air (g/kg)	Supply Air (g/kg)	Regeneration Air (g/kg)	Air after Regeneration (g/kg)	Air for Excess Heat (g/kg)
Copenhagen Summer Class 4	9.92	9.52	9.70	8.96	8.78	9.52	10.00	9.52
Copenhagen Summer Class 5	10.10	10.41	9.95	9.01	8.64	10.41	12.00	10.41
Copenhagen Summer Extreme	9.98	11.14	10.08	8.94	8.48	11.14	13.17	11.14
Milan Summer Class 3	9.87	10.30	9.92	8.75	8.63	10.30	11.58	10.30
Milan Summer Class 4	9.98	11.68	10.28	9.08	8.75	11.68	13.65	11.68
Milan Summer Class 5	9.82	12.25	10.15	8.77	8.34	12.25	14.99	12.25
Milan Summer Extreme	10.10	13.64	10.79	8.95	8.60	13.64	17.05	13.64
Colombo Summer Class 1	9.84	14.01	10.65	9.22	8.85	14.01	16.99	14.01
Colombo Summer Class 2	9.79	17.49	11.29	9.43	8.87	17.49	21.58	17.49
Colombo Summer Class 3	9.90	18.18	11.52	9.41	8.89	18.18	23.11	18.18

Cities and Climates	Return Air (g/kg)	Fresh Air (g/kg)	Mixed Air (g/kg)	Cleaned Air (g/kg)	Supply Air (g/kg)	Regeneration Air (g/kg)	Air after Regeneration (g/kg)	Air for Excess Heat (g/kg)
Colombo Summer Class 4	9.80	18.30	11.34	9.45	8.98	18.30	21.96	18.30
Colombo Summer Class 5	9.98	15.09	10.89	9.07	8.57	15.09	18.63	15.09
Colombo Summer Extreme	9.93	17.04	11.29	9.29	8.85	17.04	21.25	17.04

Table 8.4 Airflow rates measured in experiments for winter climates

Cities and Classes	Return air (L/s)	Cleaned air (L/s)	Fresh air (L/s)	Supply air (L/s)	Regeneration air (L/s)	Total Exhaust air (L/s)
Copenhagen Winter Class 5	256.03	190.42	64.92	256.93	94.54	176.25
Copenhagen Winter Class 4	257.77	195.42	63.60	261.87	93.93	172.68
Milan Winter Class 5	255.27	190.85	65.16	256.06	95.35	176.02
Milan Winter Class 4	255.15	191.40	63.81	256.23	93.08	173.26
Milan Winter Class 3	255.95	191.32	62.98	255.91	92.97	172.59
Milan Winter Class 2	249.94	194.77	59.76	263.89	95.38	173.29

Table 8.5 Temperatures measured in experiments for winter climates

Cities and Classes	Return Air (°C)	Cleaned Air (°C)	Fresh Air (°C)	Heated Fresh Air (°C)	Supply Air (°C)	Regeneration Air (°C)	Air after Regeneration (°C)	Total Exhaust Air (°C)
Copenhagen Winter Class 5	22.12	23.83	10.51	26.83	23.73	29.84	25.75	12.33
Copenhagen Winter Class 4	21.98	24.07	4.21	31.82	24.64	30.03	25.17	8.65
Milan Winter Class 5	21.82	23.52	15.98	22.98	22.78	30.31	25.99	16.45
Milan Winter Class 4	21.80	23.89	10.91	26.29	23.72	30.26	25.48	12.94
Milan Winter Class 3	22.04	23.72	5.62	31.08	24.37	29.53	25.51	9.53
Milan Winter Class 2	22.54	24.39	0.39	31.20	24.44	28.33	23.86	6.40

Table 8.6 Humidity ratios measured in experiments for winter climates

Cities and Classes	Return Air (g/kg)	Cleaned Air (g/kg)	Fresh Air (g/kg)	Heated Fresh Air (g/kg)	Supply Air (g/kg)	Regeneration Air (g/kg)	Air after Regeneration (g/kg)	Total Exhaust Air (g/kg)
Copenhagen Winter Class 5	5.38	4.88	5.75	5.75	5.12	5.75	6.58	5.19
Copenhagen Winter Class 4	4.88	4.25	4.34	4.34	4.37	4.34	5.77	4.50
Milan Winter Class 5	5.63	5.31	6.59	6.59	5.66	6.59	7.28	5.69
Milan Winter Class 4	6.21	5.74	6.66	6.66	6.02	6.66	7.64	6.05
Milan Winter Class 3	4.80	4.42	4.86	4.86	4.63	4.86	5.94	4.57
Milan Winter Class 2	4.94	4.23	3.46	3.46	4.16	3.46	5.24	4.15

An innovative clean air heat pump (CAHP) which integrated heating/cooling, dehumidification, ventilation, air cleaning and energy recovery in one unit was proposed in the thesis. The CAHP was designed to maintain a comfortable, healthy indoor environment in buildings. A theoretical model simulating the air cleaning effect and energy performance of the CAHP was established. Laboratory experimental studies were conducted to investigate the energy performance of the CAHP as well. The simulation and experimental studies showed that the CAHP can clean indoor air effectively and provide an energy efficient choice for building ventilation.

DTU Civil Engineering
Technical University of Denmark

Brovej, Building 118
2800 Kongens Lyngby
Tel. 45251700

www.byg.dtu.dk

ISBN 9788778774279
ISSN 1601-2917