ELFORSK
Project nr. 343-008

CLEAN-AIR Heat Pump

- Reduceret energiforbrug til ventilation af bygninger ved luftrensning integreret med luft varmepumpe

(Task 2)

Final Report

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COWI A/S
EXHAUSTO A/S

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1 Summary

This report summarizes task 2 of the Clean Air Heat Pump (CAHP) project - experimental validation and demonstration on the energy savings capacity of the CAHP for heating, air-conditioning and ventilation. A prototype unit of the CAHP with air handling capacity of 250 L/s was developed for the test. The prototype unit includes a specially designed heat pump with two condensers, two evaporators and one silica gel rotor. The dual condensers and evaporators design was made to minimize the energy consumption of the CAHP operating in both summer and winter climate conditions. It also makes it easier to switch between different operation modes.

The prototype unit of the CAHP was tested in the laboratory at DTU. The test lab has an air handling system that can simulate climates of different seasons and climate zones. To make it comparable with the simulation made in Task 1 of the project, climates of the three cities i.e. Copenhagen (Denmark), Milan (Italy) and Colombo (Sir Lanka) were used for evaluating the energy performance of CAHP. The three cities cover mild-cold, mild-hot and hot & humid outdoor climates. Airflows of these climate conditions were established in the test room as outdoor air supplied to the CAHP and real energy consumptions of the CAHP were measured when it was operated at these climate conditions.

For the Danish climate (the mild cold climate), the measured energy saving of the CAHP system compared to the conventional heating/air-conditioning and ventilation system was around 59% in summer and 49% in winter. Considering that most of the heating systems in Denmark were not electric driven, the energy saving of the CAHP in winter was further compared by the cost of energy. Compared to a conventional heating and ventilation system which was gas driven, the CAHP system saved 25% in the cost of energy. Since the Danish summer climate is very mild, over 80% of the yearly energy consumption for ventilation is used during winter season. It is, therefore, estimated that more than 29% annual energy cost saving for heating/air-conditioning and ventilation is expected in Denmark using the CAHP ventilation technology.

For the mild hot climate, e.g. the Italian climate, the measured energy saving of the CAHP was around 40% in summer season. For winter season, 22% reduction of the energy cost was expected.

For the extremely hot and humid climate, e.g. Sri Lanka, cooling is required over the year. The energy saving was only measured for cooling and ventilation. The results showed that 30% of power saving could be achieved.

In general, the laboratory tests showed that CAHP technology is suitable for heating/air-conditioning and ventilation in all kinds of climates around the world except for the hot and dry climate. The energy saving is expected in the range between 25% and 60% depending on the climate. This measured energy saving is little lower than it was calculated by the numerical simulation in phase 1 of the project. The reasons are discussed in the chapter of Discussion. **It is worth noting that the reference system that was used to compare for the energy consumption with the CAHP included an efficient energy recovery system for ventilation. Compared to**

such a ventilation system, the CAHP system could still save substantial amount of energy. Therefore, the technology is highly recommended provided that its air cleaning function is further validated by experiments.

2 Introduction

Following Task 1 of the project, a prototype unit of the Clean Air Heat Pump (CAHP) was developed and tested in Task 2. Task 1 performed modeling and simulation of the CAHP on energy saving for heating/air-conditioning and ventilation, air cleaning and energy recovery. The total energy consumption of the CAHP system was calculated by a theoretical model and compared with the reference heating/air-conditioning and ventilation systems (conventional systems). The energy consumption comparison between the two systems included energy used for heating, cooling and fans. The simulation and energy saving calculation was made for the application of the CAHP in three typical climate conditions, i.e. mild-cold, mild-hot and hot & humid climates. Real climate data recorded from three cities in 2002 was used for the calculation. The three cities were Copenhagen (Denmark), Milan (Italy) and Colombo (Sir Lanka) which represent the above three typical climate zones. The following results were obtained from the simulation.

1. For the Danish climate (the mild cold climate), the calculations showed that the ventilation system using CAHP technology could save up to 42% of energy cost in winter compared to the conventional ventilation system. The energy saving in summer could be as high as 66% for the ventilation system with humidity control and 9% for the ventilation system without the requirement of humidity control. Since the Danish summer climate is very mild, over 80% of the yearly energy consumption for ventilation is used during winter season. It was, therefore, estimated that more than 35% annual energy saving for ventilation was expected in Denmark using the CAHP ventilation technology.
2. For the mild hot climate, e.g. the Italian climate, the calculations showed that up to 63% of the energy saving could be achieved in summer season. For the winter mode, 17% reduction of the energy cost could be expected for the domestic use. For industrial use, the energy cost of the CAHP might not be favorable due to the industrial price of gas in Italy was too much lower than the price of electricity.
3. For the extremely hot and humid climate, the CAHP has the maximum ability of the energy saving for ventilation. The calculations showed that annual energy saving of using the CAHP for ventilation in Sri Lanka was 62%.

In general, from the simulation results, the CAHP system was suitable for ventilation in all kinds of climates around the world except for the hot and dry climate. The annual energy saving was expected in the range between 30% and 60% depending on the climate. Based on the simulation, it was concluded that the energy saving of the CAHP for ventilation was remarkable. Therefore, the technology was highly recommended provided that the simulation results are validated by experiments.

In Task 2 of the project, a prototype unit of the CAHP with air handling capacity of 250 L/s was developed. The energy performance of the prototype CAHP was tested in the laboratory at different climate conditions. The measured energy performance of the CAHP was used to validate the results of the theoretical simulation made in Task 1 of the project. This report presents the design of the prototype CAHP, the design of the experiments testing energy performance of the CAHP and the results obtained from the experiments.

3 Method of the Study

To study the energy performance of the CAHP, a prototype unit of the CAHP was designed and developed. The process of air-conditioning and energy consumption of the prototype unit were measured in the laboratory at different climatic conditions to verify its ability on indoor climate control and energy saving.

3.1 Principle of the CAHP

The design principle of the CAHP has been described in the report of Task 1 of this project. Figure 3.1 shows the schematic diagram of the design.

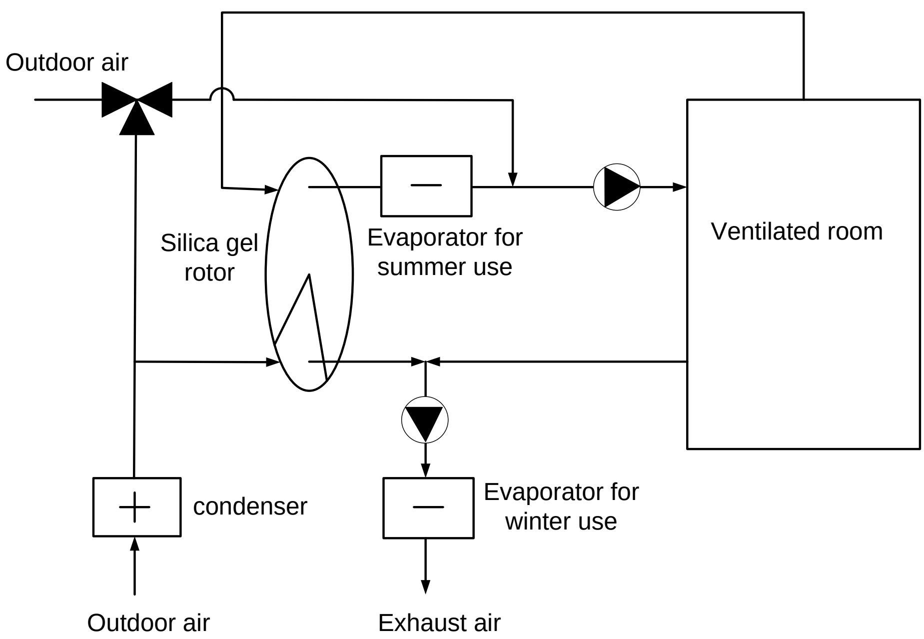


Figure 3.1 Schematic diagram of CAHP for summer and winter

This design combined the application of the CAHP for both summer and winter seasons. In the design, two evaporators are used in one heat pump, one is used in winter and the other is used in summer. One condenser is used in the heat pump for both seasons. Fresh outdoor air supplied to the ventilated room is switched by a three way damper to select whether it is heated by the condenser or not. In winter, the fresh outdoor air supplied to the ventilated room is pre-heated by the condenser of the heat pump, while in summer, it is taken directly from outside without pre-heating. In addition to the fresh outdoor air supply, large quantity of indoor air is recirculated through a silica gel rotor where it is cleaned and dehumidified by the rotor. The fresh outdoor air joins the cleaned recirculating air to ventilate the room and to control indoor air temperature, humidity and air quality. In summer, the fresh outdoor supply air and the recirculating air after being processed by the rotor are too warm and are cooled by the evaporator of the heat pump before they are delivered into the ventilated room. In winter, such cooling is not necessary. The evaporator of the heat pump is then placed at the exhaust 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 the condenser of the heat pump. In summer, the warm air after regenerating the rotor and the exhaust air from the ventilated room are rejected directly to outdoor, while in winter, total heat of the rejected air is recovered by the evaporator of the heat pump.

3.2 Design of the prototype CAHP

During developing the prototype unit of the CAHP, the same principle was used as described above. Small changes were made on the design of the heat pump to make it more energy efficient and controllable. Temperature and humidity sensors were installed in various locations of the CAHP for monitoring the operational performance of each component. The detailed designs of the CAHP are illustrated below separately in summer and winter mode.

3.2.1 Air system designed for summer mode

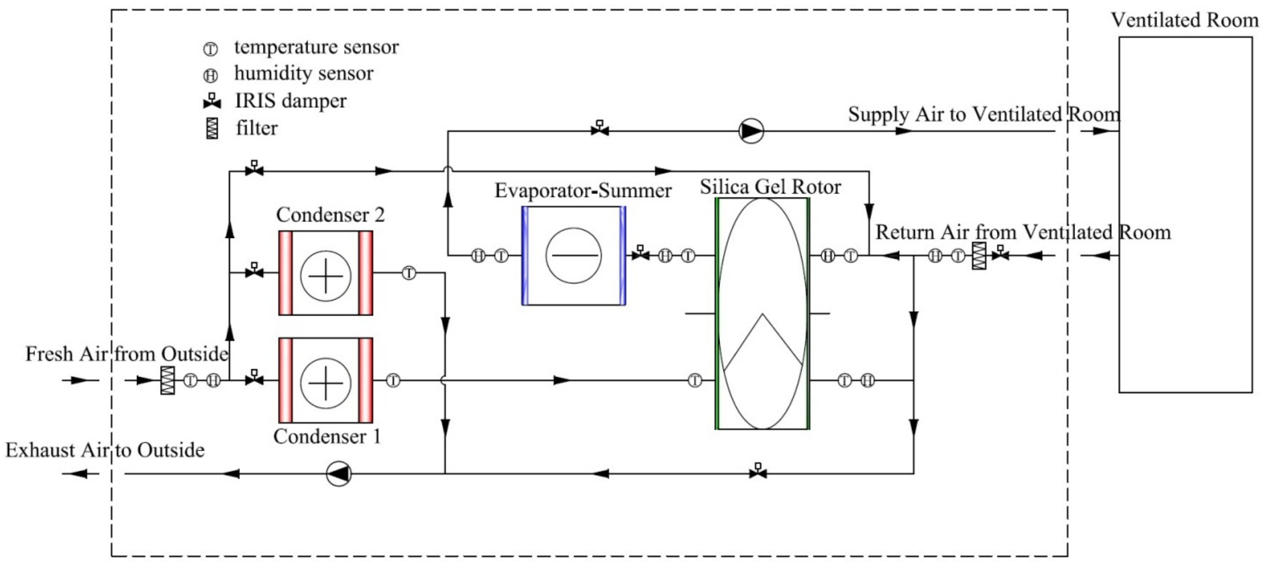


Figure 3.2 Air system of prototype CAHP for summer

In summer mode, the design is illustrated by Figure 3.2. There are two inlets and two outlets in the CAHP. The two inlets are the outdoor air inlet taken fresh air from outdoors and indoor air inlet taken air returned from the room. The two outlets are the exhaust air outlet and ventilation air supply outlet to the room. As shown in Figure 3.2, two condensers were included in the final design of the CAHP for the summer mode. The dual-condenser design was adopted to control the heating of the regeneration air at the exact amount as demanded by dehumidification. The calculation in task 1 found that, in most cases, the condensing heat is more than what is required for regenerating the rotor. If all the condensing heat is used for regenerating the rotor, it may over dry the ventilation air and increase the cooling load of the evaporator and, in turn, increase the speed of the compressor which results in a higher energy consumption. In this dual-condenser design, the extra condensing heat is rejected directly by the second condenser without feedback to the evaporator. This design also makes the temperature control of the CAHP independent of the humidity control and keeps the compressor always running at the minimum speed required to minimize power consumption.

In summer mode, the CAHP is controlled by regulating the speed of the compressor and the distribution of refrigerant between the two condensers to achieve independent control of ventilation air temperature and humidity. The control strategy is that the ventilation air temperature is controlled by a frequency invertor to modulating the speed of the compressor. The humidity of the ventilation air is controlled by modulating the opening of the two valves connected to the two condensers to control the temperature of the regenerating air in order to regulate the dehumidification capacity of the silica gel rotor.

3.2.2 Air systems designed for winter mode

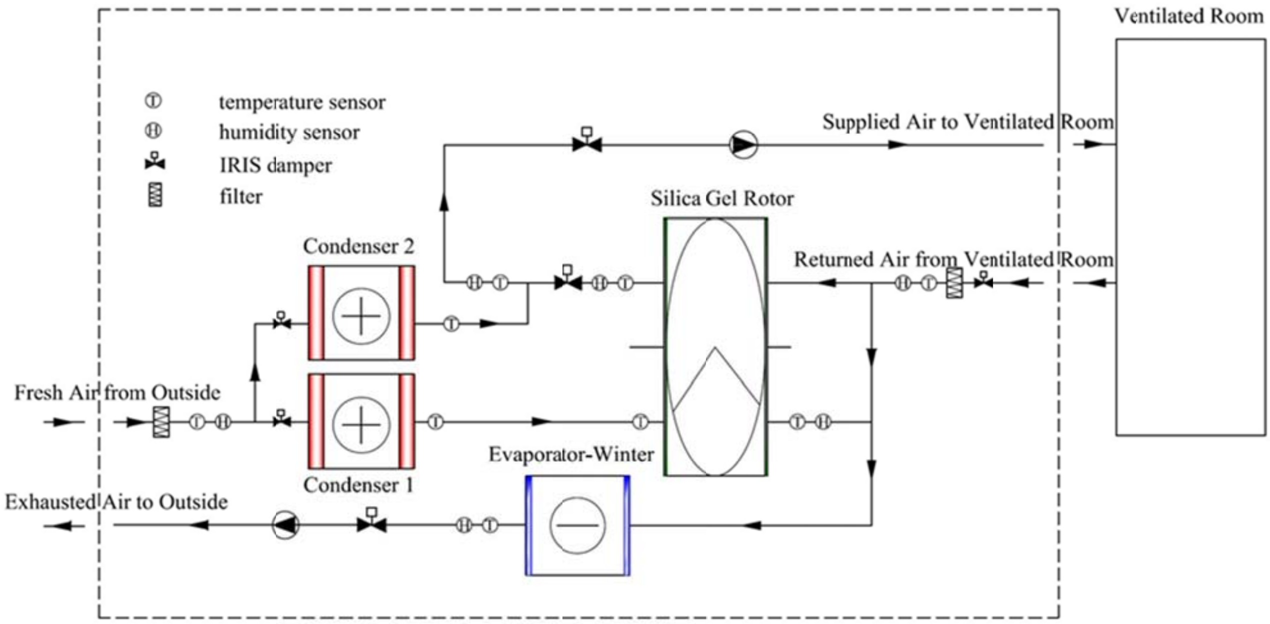


Figure 3.3 Air system of prototype CAHP for winter

The winter mode design of the CAHP is illustrated by Figure 3.3. There are also two inlets and outlets of the CAHP in winter design as it is in summer design. Similar to the summer mode, the CAHP also uses two condensers in 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 to ventilate the room. The evaporator is used to collect 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 winter is usually higher than it is in summer. Part of the regenerating heat is transferred to warm up the recirculation air through the rotor. The rest of the regenerating 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 even though the silica gel rotor removes small amount of moisture when it is running at low regeneration temperature for air cleaning.

The control strategy of the CAHP in winter mode is to control the room air temperature by regulating the speed of the compressor. To avoid too high air temperature for regenerating the silica gel rotor, the refrigerant distribution between the two condensers is controlled by modulating the two control valves connected to each of the condensers. In winter mode, the regenerating air temperature is controlled by condenser 1 at a constant level in a range between 25 to 30°C. Therefore, the control of the compressor and the two regulating valves has to be coordinated to fulfill the requirement of both room air temperature and the regeneration temperature. As mentioned above, the ventilation system using CAHP requires much lower outdoor air. Occasionally, dehumidification in the ventilated room using CAHP may be needed. This can be controlled easily by slightly raising the regenerating air temperature (e.g open the control valve of condenser 1).

To achieve the above process of air handling in the CAHP, a dual-condenser and dual-evaporator heat pump with variable compressor speed control was designed and developed. Figure 3.4 shows the principle of this heat pump.

3.2.3 Heat pump designed for prototype CAHP

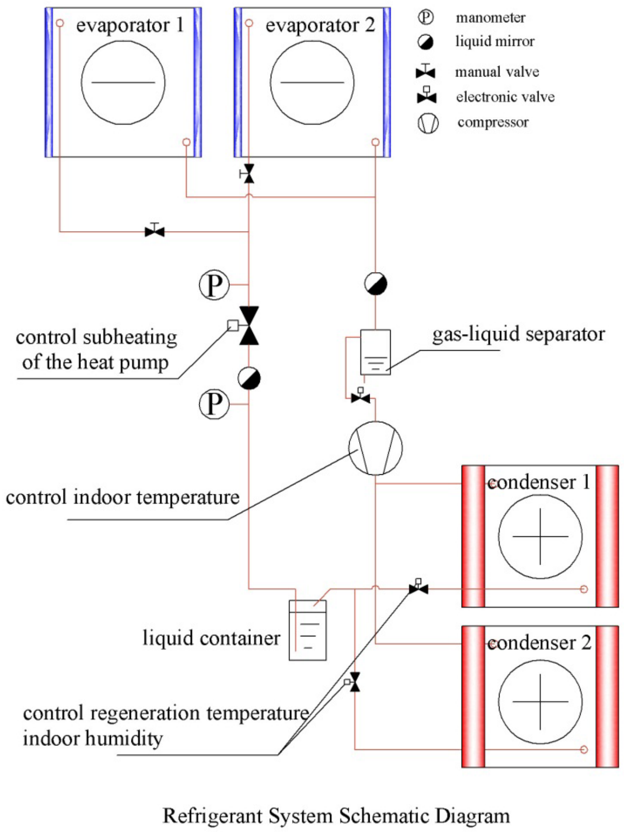


Figure 3.4 Refrigerant system of prototype CAHP for summer and winter

As described in the air systems of the CAHP, two evaporators were designed in the CAHP. One is used in summer operating mode and the other is used in winter. Refrigerant to each evaporator is switched manually or automatically by the valves connected to the evaporators when season changes. Among the two condensers, condenser 1 is used for heating the air for regenerating the silica gel rotor; condenser 2 has different functions in two seasons. In winter season, condenser 2 is used to pre-heat fresh outdoor air for ventilation. In summer season, condenser 2 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 the modulating valves that connected to the condensers. By controlling the opening of the two valves, the distribution of the refrigerant to the two condensers can be controlled very precisely.

The variable speed control of the compressor ensures that the heat pump can adapt to different heating and cooling demand and control the indoor climate with minimum power consumption. A speed variable piston compressor is selected in designing and developing the prototype CAHP. A frequency inverter is used to modulate the speed of the compressor. The speed control of the compressor and the regulating of refrigerant flow rate in condenser 1 and 2 are the major control strategy of the CAHP for indoor climate control and energy conservation.

To maintain a constant supper heating temperature in the evaporator when the speed of the compressor is regulated, an electronic expansion valve was selected to be used as throttle. The opening of the electronic expansion valve is controlled by the supper heating temperature to make sure that the supper heating temperature is independent on the speed of the compressor.

3.3 The test room

The capacity of the CAHP was determined by the size of the test room and its thermal performance. These information were collected for calculating the hygrothermal load of the test room and its ventilation requirement. Geometry and thermal performance of the test room is list in Table 3.1. Thermal environment and thermal load in the test room used in the experiment is listed in Table 3.2.

Table 3.1 Geometry of the test room and its heat transfer coefficients

Parts of Envelope	Area (m^2)	Heat transfer coefficient (W/m^2*K)	In contact with outside (Y/N)
Roof	72	0.20	Y
External Wall	21.6	0.25	Y
External Window	14.4	1.5	Y
Interior Walls and doors	72	2.0	N
Floor	72	2.0	N

Table 3.2 Assumptions of the indoor climate and thermal load in the test room

Parameters		Unit	Value
indoor temperature	summer	(℃)	25
	winter		22
indoor relative humidity	summer	(%)	50
	winter		--
occupants		p	10-15
heat from lights		W	43
heat from computer		W	210
heat from projector		W	250

Based on the above information of the test room and the data of the extremely hot climate in Copenhagen (32.1℃/38.6%RH), the ventilation rate, sensible and latent heat load was calculated for the design of the CAHP. Under the extremely hot climate in Copenhagen, the sensible heat load in the test room is 2.12kW, dehumidification load is 1.02kg/h. Taken 7°C difference between supply air and room air temperature [1], the supply airflow rate was calculated to be 250l/s and the supply air humidity was calculated to be 8.91g/kg.

3.4 Selection of components for the prototype CAHP

3.4.1 Silica gel rotor

With the airflow rate of 250l/s, a silica gel rotor produced by Munters with dimensions of 454mm in diameter and 200mm in depth was selected to be used in the prototype CAHP. The flow directions of process air and regeneration in the silica gel rotor are shown in Figure 3.5. The rotation speed of the rotor is 11.6 rounds per hour. With this rotor, a regenerating temperature of 44°C is required at the extremely hot climate conditions in Copenhagen.

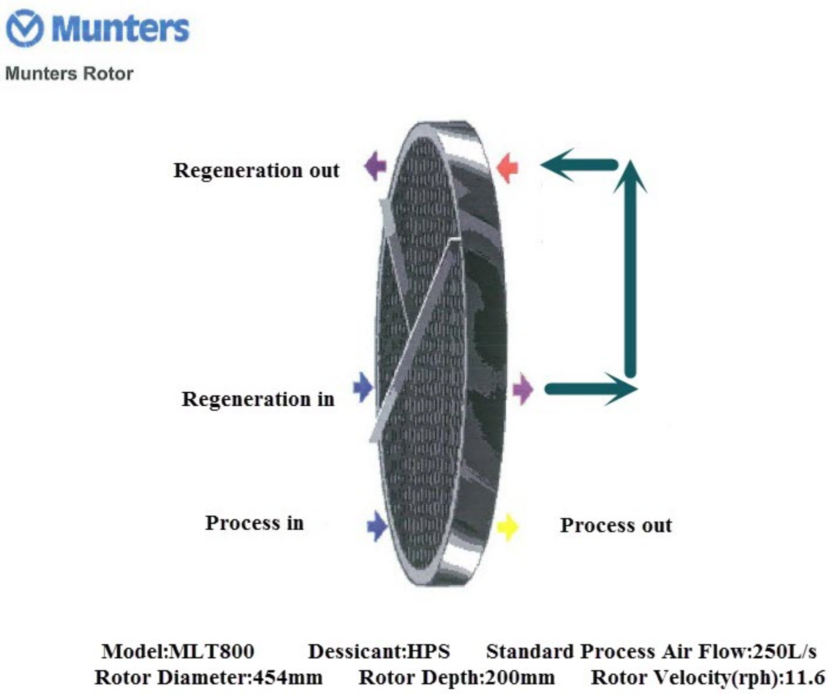


Figure 3.5 Silica gel rotor selected for prototype CAHP

3.4.2 Refrigerant

With the calculated airflow rate and target outlet humidity ratio of process air, considering different outdoor climate zones, the regeneration air temperature for dehumidification could be calculated. The results showed that the regeneration air temperature was variable from 44℃ to 64℃ from mild cold climate to extremely hot climate zones. To heat up the regeneration air to 64°C, the condensing temperature of the heat pump was designed to be 70°C. Meanwhile, the supply air temperature to the room was designed to be around 18℃ (7℃ lower than indoor temperature) as described above. To cool down the supply air to 18°C, the evaporating temperature should be 15°C or lower. Therefore, the condensing and evaporating temperature were design to be 70℃ and 15℃ for the extremely hot and humid climate zone.

To achieve the condensing and evaporating temperatures of 70℃ and 15℃ respectively, different candidates of HFC refrigerant were compared with their condensing pressure and coefficient of performance (COP).

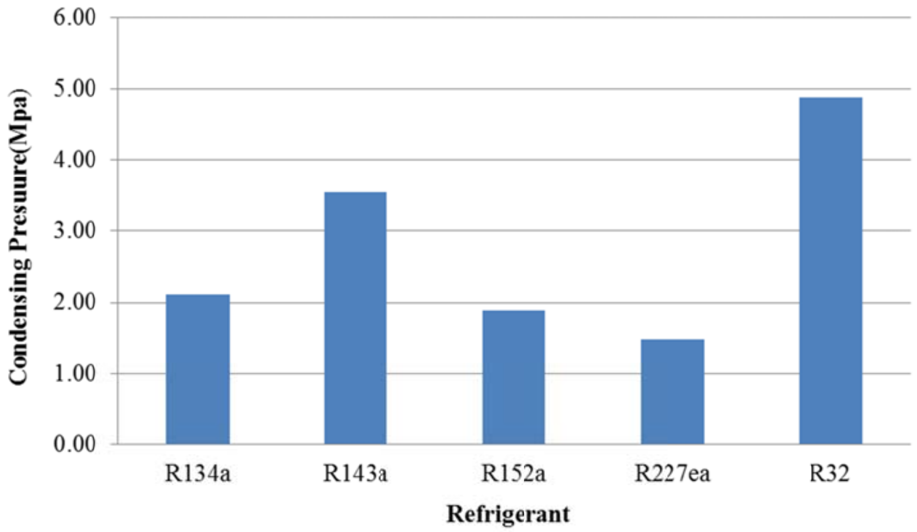


Figure 3.6 Condensing pressure of different HFC refrigerant at temperature of 70℃

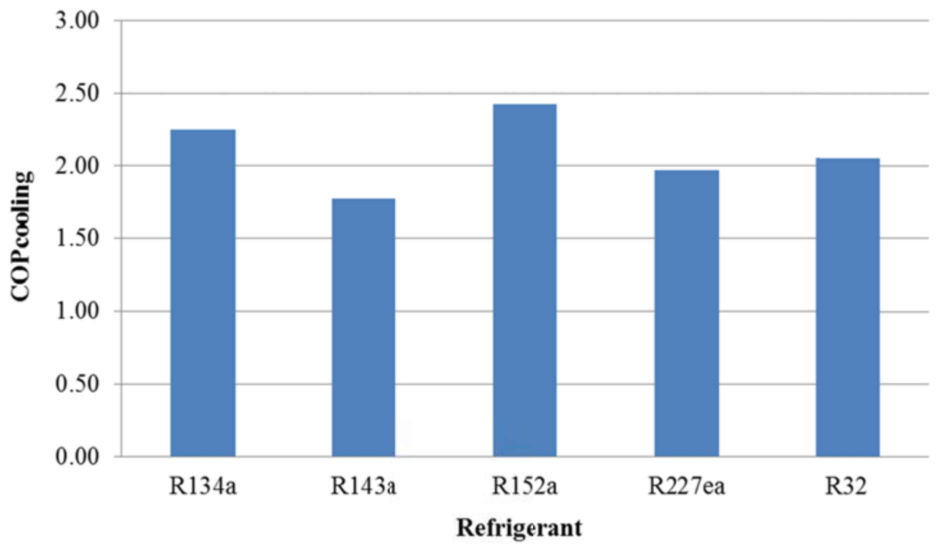


Figure 3.7 COP of different HFC refrigerant at recommended condensing and evaporating temperatures

The selecting criteria of the refrigerant were the followings.

1. Condensing pressure at 70℃ should be lower than 2.5Mpa to avoid especial requirement on manufacturing the heat pump.
2. COP should be relatively higher among the candidates.
3. The refrigerant should be un-combustible to be used safely in laboratories and buildings.

Comparing the properties of different refrigerants, the refrigerant R134a was selected to be used in the prototype of CAHP since it fulfilled all the above selecting criteria for the refrigerant. The Ozone Depletion Potential (ODP) value of R134a is 0 and its Global Warming Potential (GWP) value is 1300. It is un-combustible and fulfills all the requirements for refrigerant in Denmark and Europe.

3.4.3 Compressor

With the selected refrigerant and the required condensing and evaporating temperature, the condensing and evaporating pressure in the designed condition were 2.2Mpa and 0.49Mpa respectively. To fulfill these requirements, the compression ratio of the compressor for CAHP should be 4.5 or higher.

The refrigerant flow rate at the inlet of the compressor was calculated by cooling load of the evaporator of the CAHP. It required a refrigerant flow rate of 8.11m^3/h.

With the compression ratio and refrigerant flow rate calculated for the CAHP, a piston compressor “2GC-2.2”produced by Bizer was selected for the heat pump. The technical data of the compressor are listed in Figure 3.8. The refrigerant displacement capacity of the selected compressor was 9.15m^3/h, and the compression ratio was 9.3 which fulfilled the requirements of the calculations about refrigerant displacement and compression ratio. The motor voltage of the compressor is 400V, and the frequency of the power input is variable from 20HZ to 80HZ. As described above, during the experiments, the frequency of power input to the compressor was modulated by a frequency inverter to regulate the speed of the compressor.

Piston compressors

Data Sheet: 2GC-2.2

Abb. ähnlich / Fig. similar, © Bitzer

- Leaflet: KP-100 (50Hz / SI)
- Leaflet: KP-105 (60Hz / IP)
- Operating Instruction: KB-100
- Spare Part List: KE-120
- Manufacturers Declaration // Decl. of Conformity // Decl. of Incorporation: KC-001
- Manufacturers Declaration // Decl. of Conformity // Decl. of Incorporation: KC-100
- Manufacturers Declaration // Decl. of Conformity // Decl. of Incorporation: KC-102
- Technical Information:
  KT-122 KT-140 KT-150 KT-410 KT-420 KT-500 KT-510 KT-602 KT-660
- Show dimensional drawing
- Download CAD drawing
- Exploded View
- Maintenance Instruction: KW-100

Technical Data		
	SI	IP
Motor version	1+2	1+2
Displacement (1450 RPM 50Hz)	7.58 m^3/h	267.7 CFH
Displacement (1750 RPM 60Hz)	9.15 m^3/h	323.1 CFH
No. of cylinder x bore x stroke	2 x 41 mm x 33 mm	2 x 1.61 inch x 1.3 inch
Motor code	40S	4SU
Motor voltage (more on request)	220..240V Δ/3/50Hz 380..420V Y/3/50Hz	265..290V Δ/3/60Hz 440..480V Y/3/60Hz
Max operating current	4.7A	4.8A
Maximum power consumption	2.7 kW	2.7 kW
Starting current (Rotor locked)	39.0A / 22.5A (Δ/Y)	39.0A / 22.5A (Δ/Y)
Enclosure class	■ IP65	■ IP65
Weight	45 kg	99 lb
Max. pressure (LP/HP)	19 / 28 bar	275 / 400 psi
Connection suction line	16 mm	5/8"
Connection discharge line	12 mm	1/2"
Oil charge	1.00 dm^3	35.2 fl oz
Crankcase heater (self-control)	□ max. 60 W	□ max. 60 W
Oil type R134a // R407A/C/F // R404A // R507A	□ t_c<55°C: BSE32 □ t_c>55°C: BSE55	□ t_c<130°F: BSE32 □ t_c>130°F: BSE55
Oil type R22 R22 (R12 // R502)	■ B5.2	■ B5.2
Motor protection	■ SE-B1	■ SE-B1
Discharge shut-off valve	■	■
Suction shut-off valve	■	■
Additional fan	□	□
Vibration dampers	■	■
Sound power level (+5°C / 50°C)	64.0 dB(A) @ 50Hz	66.5 dB(A) @ 60Hz
Sound power level (-10°C / 45°C)	63.0 dB(A) @ 50Hz	65.5 dB(A) @ 60Hz
Sound power level (-35°C / 40°C)	63.5 dB(A) @ 50Hz	66.0 dB(A) @ 60Hz
Sound pressure level @ 1m (+5°C / 50°C)	56.0 dB(A) @ 50Hz	58.5 dB(A) @ 60Hz
Sound pressure level @ 1m (-10°C / 45°C)	55.0 dB(A) @ 50Hz	57.5 dB(A) @ 60Hz
Sound pressure level @ 1m (-35°C / 40°C)	55.5 dB(A) @ 50Hz	58.0 dB(A) @ 60Hz
■ Standard □ Option (1) 230V/1/50Hz+60Hz		
// Subject to change //		

Figure 3.8 Compressor selected for prototype CAHP

3.4.4 Expansion valve

A solenoid valve “AKV10-6” produced by Danfoss was selected to be the throttle valve in the heat pump. Except for the expansion function, the valve was also used to control the supper-heating temperature. During the experiments, the supper-heating temperature was controlled at 4℃-6℃. 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 supper-heating temperature could be calculated by the controller of the expansion valve. By regulating the time proportion on opening of the solenoid valve, the supper-heating temperature was controlled precisely.

3.4.5 Condensers and Evaporators

The sizes of condensers and evaporators were calculated with software Rrecalc ver.1.2.3 provided by company “Roen est”. The calculation was based on the airflow rates calculated in chapter 3.3 and the air temperatures at different points calculated in task 1 of the project. During the calculation, the condensing and evaporating temperature of refrigerant was set at 70℃ and 15 ℃ respectively, and the supper heating temperature was set at 5℃ as stated in the chapter above.

The calculation results are listed in Table 3.3.

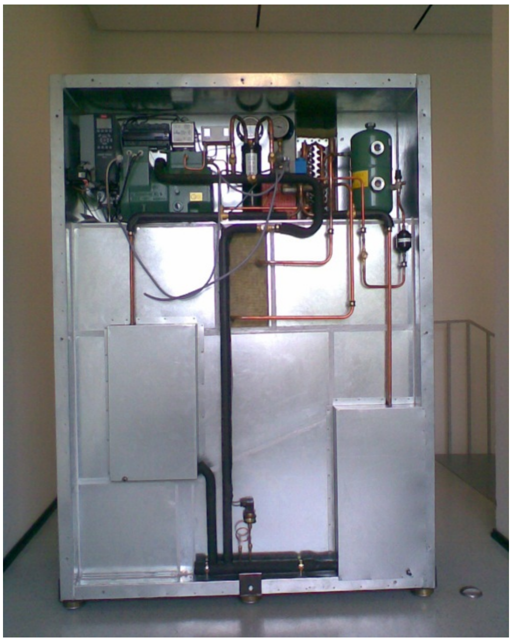
Table 3.3 Sizes of condensers and evaporators for prototype CAHP

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 pre-heat(winter)	300	250	5

To control the refrigerant distribution between the two condensers, electronic control valve“EX5-U21” produced by Emerson was selected.

3.5 Construction of prototype

After all the key components were selected, the prototype unit of CAHP was constructed. Figure 3.9 shows pictures of the heat pump and silica gel rotor in the prototype unit.

heat pump silica gel rotor

Figure 3.9 Pictures of heat pump and silica gel rotor for prototype CAHP

3.6 Experimental setup

The experiment was conducted in a test room of International Center for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark (DTU). The test room is located in 2nd floor of Building 402 at DTU [2]. The test room is equipped with a ventilation system that can simulate different outdoor climate conditions for testing an air handling unit. The room also integrates many different types of air terminals for ventilation. One picture of this classroom is shown in Figure 3.10. The air deliver terminals chosen for testing the prototype of CAHP were the diffusers for mixing ventilation.

Figure 3.10 Pictures of the test room for experiments

During the experiments, fresh air was taken from the garden of the lab and passed through a channel to the second floor of the building. There was an outdoor air handling unit to process the outdoor air to simulate different outdoor hygrothermal climates. 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.

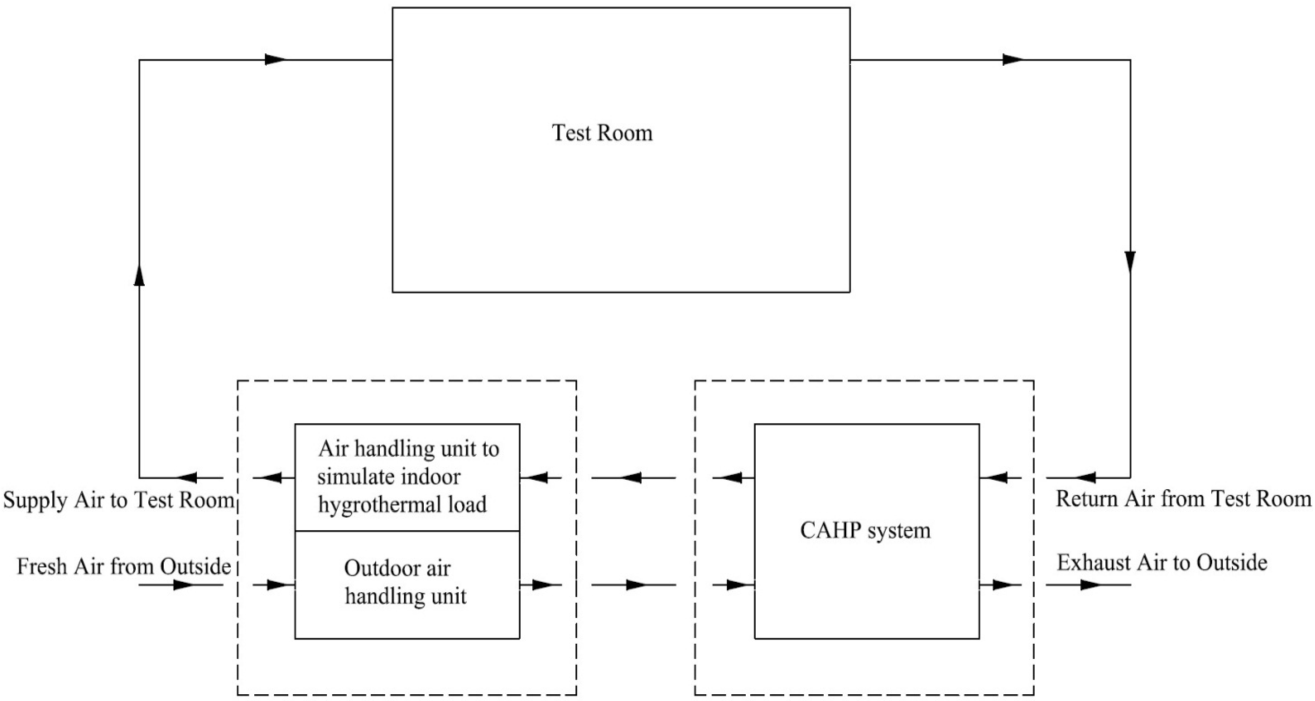


Figure 3.11 Connections from CAHP to the existing air handling units and test room

Figure 3.11 Connections from CAHP to the existing air handling units and test room. In this setup, the recirculation air was taken from the test room and fresh air was taken from the outdoor air unit of the test room. Exhaust air from the classroom and CAHP was rejected to outdoor. Recirculation air and small amount of fresh air after cleaning and hygrothermal processing was delivered to the test room.

3.7 Design of experiments

The experiment in task 2 was designed to validate the simulation work performed in task 1 of the project using the same climate conditions as used in the simulation, 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, Milano 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 simulation in task 1 assumed that the system was used only in office space, i.e. during normal office hours between 6:00am to 6:00 pm. The classification of summer and winter period was divided according to Table 3.4 for the three cities. With this assumption and classification, the outdoor air temperature, humidity and the number of hours of the five categories in both summer and winter of the three cities were calculated and summarized in Table 3.5-Table 3.9.

Table 3.4 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

In category 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 ratio of the class is the mean value in the same temperature class.

Table 3.5 Summer climate data for Copenhagen.

Copenhagen - Summer			
	T (°C)	x (kg_s/kg_a)	Hours
1st class	6.5	0.0057	76
2nd class	12.2	0.0075	655
3rd class	17.9	0.0086	808
4th class	23.6	0.0096	231
5th class	29.3	0.0105	23
Extreme case	32.1	0.0115	1

Table 3.6 Winter climate data for Copenhagen.

Copenhagen - Winter			
	T (°C)	x (kg_s/kg_a)	Hours
1st class	-16.69	0.00089	15
2nd class	-9.87	0.00163	62
3rd class	-3.05	0.00277	585
4th class	3.77	0.00419	1303
5th class	10.59	0.00577	400
Extreme case	-20.10	0.00060	1

Table 3.7 Summer climate data for Milan.

Milan - Summer			
	T (°C)	x (kg_s/kg_a)	Hours
1st class	10.5	0.0077	219
2nd class	15.5	0.0096	662
3rd class	20.5	0.0105	820

Milan - Summer			
	T (°C)	x (kg_s/kg_a)	Hours
4th class	25.5	0.0119	507
5th class	30.5	0.0127	170
Extreme case	33.00	0.0136	1

Table 3.8 Winter climate data for Milan.

Milan- Winter			
	T (°C)	x (kg_s/kg_a)	Hours
1st class	-5.3	0.0025	98
2nd class	0.1	0.0035	623
3rd class	5.5	0.0049	898
4th class	10.9	0.0065	550
5th class	16.3	0.0069	196
Extreme case	-8.00	0.0019	1

Table 3.9 Climate data for Colombo.

Colombo			
	T (°C)	x (kg_s/kg_a)	Hours
1st class	20.9	0.0143	80
2nd class	24.7	0.0176	1122
3rd class	28.5	0.0185	2847
4th class	32.3	0.0180	692
5th class	36.1	0.0151	3
Extreme case	38.0	0.0171	1

With the outdoor climate classification and the thermal performance of the test room listed in Table 3.1 and Table 3.2, the hygrothermal load of the test room was calculated for different cities and different seasons. It is important to state that in summer, 15 persons were assumed in the test room when calculating the ventilation rate and thermal load in the climates of Copenhagen and Milan and 10 persons were assumed in the test room when calculating the ventilation rate and thermal load in the climate of Colombo. Since the outdoor air temperature and humidity of class 1 to 3 in Danish summer and class 1 to 2 in Italian summer is low enough to be used for ventilation to balance the indoor cooling load, the buildings should be ventilated directly by outdoor air without running CAHP. These climate conditions were not included in the experiment.

For the winter climate, the test facility could only mimic outdoor climate with the air temperature above 0°C since frost forms on the cooling coil of the air handling system of the test room when the temperature of the cooling coil went below 0°C. Therefore, the experiment was conducted at two

Danish winter climate conditions - class 4 and 5, and four Italian winter climate conditions - class 2, 3, 4 and 5. For Sri Lanka, there is no need for heating in winter season.

The hygrothermal load in the test room and the hygrothermal conditions of the supply air to the test room were calculated and summarized in Table 3.10and Table 3.11.

Table 3.10 Hygrothermal load and supply air condition calculated for summer

Cities and Climate Classes	indoor climate		outdoor climate		Hygrothermal Load		Supply Air	
	Temperature (℃)	Humidity Ratio(g/kg)	Temperature (℃)	Humidity Ratio(g/kg)	Sensible Load(kW)	Latent Load(kg/h)	Temperature (℃)	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 3.11 Hygrothermal load and supply air condition calculated for winter

Cities and Climate Classes	indoor climate		outdoor climate		Hygrothermal Load		Supply Air	
	Temperature (℃)	Humidity Ratio(g/kg)	Temperature (℃)	Humidity Ratio(g/kg)	Sensible Load(kW)	Latent Load(kg/h)	Temperature (℃)	Humidity Ratio(g/kg)
Copenhagen Class 5	22	4.89	10.59	5.77	-0.47	---	23.56	---
Copenhagen Class 4	22	4.89	0	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 in different seasons and different cities are listed in Table 3.12 and Table 3.13.

Table 3.12 Air-flow rates for different cities in summer

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

Table 3.13 Air-flow rates for different cities in winter

Cities	Recirculation air	Fresh air to classroom	Regeneration air	Exhaust air from room
	L/s	L/s	L/s	L/s
Copenhagen	190	60	95	60
Milan	190	60	95	60

After the above calculation, the experiments were conducted using the calculated conditions. All parameters of the air-conditioning process in the CAHP were logged by Agilent 34970A data logger.

3.8 Assumption of Reference System

In task 2, the energy consumption of the CAHP was measured under different climate conditions. The measured energy consumption of the CAHP was compared to a reference system which was a conventional heating or air-conditioning system commonly used in existing buildings. The energy saving potential of the CAHP was thus estimated. Energy consumption of the reference system was calculated based on the following assumptions.

1. Summer: In summer, the reference system was assumed to be air source heat pump which use outside air as the cooling source. The COP of the heat pump was calculated with different condensing and evaporating temperatures in different cities and different classes of outdoor climates. During the calculation, the entropy efficiency of compressor was referred to the entropy efficiency measured for the compressor in the CAHP.
2. Winter: In winter, the reference system was assumed to be a gas boiler with heat recovered ventilation unit. From the previous study [3], the boiler efficiency was assumed at 82%, and the heat recover efficiency was 60%.

Since the CAHP has a very strong ability on air cleaning, the comparison of energy consumption between CAHP and the reference system was made assuming that both systems provide same indoor air quality. Based on the previous study [4], 80% of recirculated air in the CAHP system is cleaned and can be used to substitute for outdoor air.

According to the EU standard for ventilation [5], the fresh air in the CAHP system was designed to be 4L/(s*p). The flow rate of fresh outdoor air in the reference system was equivalent to the flow rate of clean air delivered into the test room by the CAHP system and was calculated by the following equation.

$$Q_{f\text{-}ref}=Q_{f\text{-}CAHP}+0.8*Q_{rec\text{-}CAHP}$$

Where:

$Q_{f\text{-}ref}$ is the fresh airflow rate in the reference system;

$Q_{f\text{-}CAHP}$ is the fresh airflow rate in the CAHP system;

$Q_{rec\text{-}CAHP}$ is the recirculation airflow rate in the CAHP system.

Thus, the outdoor airflow rates and the recirculation airflow rates of CAHP and the reference system were selected as shown in Table 3.14.

Table 3.14 Air flow of CAHP and reference system

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

With the airflow rate in reference system (Table 3.14) and the outdoor climate conditions listed in Table 3.5-Table 3.9, the hygrothermal load and the energy consumption of reference system can be calculated.

4 Results

During the experiment, the indoor and outdoor climates, the air flow rate and the hygrothermal conditions of the supply air were controlled in accordance with the calculated values listed on Table 3.11 – 3.13 and stabilized for at least one hour. All the process parameters of the CAHP listed on Table 3.11 – 3.13 and the energy consumption of compressor were measured and recoded.

4.1 Energy saving in summer condition

At first, the energy consumption of the heat pump of CAHP in summer was recorded and is listed in Table 4.1. The COP of the heat pump for cooling ($COP_{cooling}$) is also calculated and listed.

Table 4.1 Hourly energy consumption of heat pump of CAHP in different cities and different categories of summer climates

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

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

For the reference system, the energy consumption and the COP of the heat pump for cooling ($COP_{cooling}$) is also calculated and listed in Table 4.2.

Table 4.2 Hourly energy consumption of heat pump of reference system in different cities and different categories of summer climates

Cities and Climate Classes	Reference System		
	cooling capacity (kWh/h)	Energy Consumption Heat Pump(kWh/h)	COPcooling
Copenhagen Class 4 Summer	3.95	**1.32**	2.99
Copenhagen Class 5 Summer	5.38	**1.89**	2.84
Copenhagen extreme Summer	6.40	**2.62**	2.44
Milan Class 3 Summer	3.77	**0.79**	4.76
Milan Class 4 Summer	5.82	**1.34**	4.35
Milan Class 5 Summer	7.08	**2.20**	3.22
Milan Extreme Summer	8.06	**2.84**	2.84
Colombo Class 1 Summer	5.28	**1.10**	4.80
Colombo Class 2 Summer	8.11	**2.29**	3.55
Colombo Class 3 Summer	8.95	**2.89**	3.10
Colombo Class 4 Summer	9.32	**3.20**	2.91
Colombo Class 5 Summer	8.37	**3.32**	2.52
Colombo extreme Summer	9.44	**4.39**	2.15

Comparing the values of energy consumption listed in table 4.1 and 4.2, the energy saving of the CAHP in the three regional summer climates were calculated and listed in Table 4.3.

Table 4.3 Hourly energy consumption of CAHP, reference system and energy saving of CAHP compared to reference system in different cities and different categories of summer climates

Cities and Climate Classes	Energy Consumption(kWh/h)		Energy Saving
	CAHP System	Reference System	CAHP to Reference
Copenhagen Class 4 Summer	0.53	1.32	**59.87%**
Copenhagen Class 5 Summer	0.85	1.89	**55.20%**
Copenhagen extreme Summer	1.22	2.62	**53.42%**
Milan Class 3 Summer	0.53	0.79	**32.90%**
Milan Class 4 Summer	0.76	1.34	**43.07%**
Milan Class 5 Summer	1.16	2.20	**47.07%**
Milan Extreme Summer	1.51	2.84	**46.99%**
Colombo Class 1 Summer	0.99	1.10	**10.36%**
Colombo Class 2 Summer	1.77	2.29	**22.65%**
Colombo Class 3 Summer	2.08	2.89	**28.15%**
Colombo Class 4 Summer	1.72	3.20	**46.10%**
Colombo Class 5 Summer	1.66	3.32	**50.04%**
Colombo extreme Summer	1.96	4.39	**55.35%**

The comparison of energy saving between CAHP and the reference system in summer mode in the three cities are demonstrated by histograms in Figure 4.1 to 4.3.

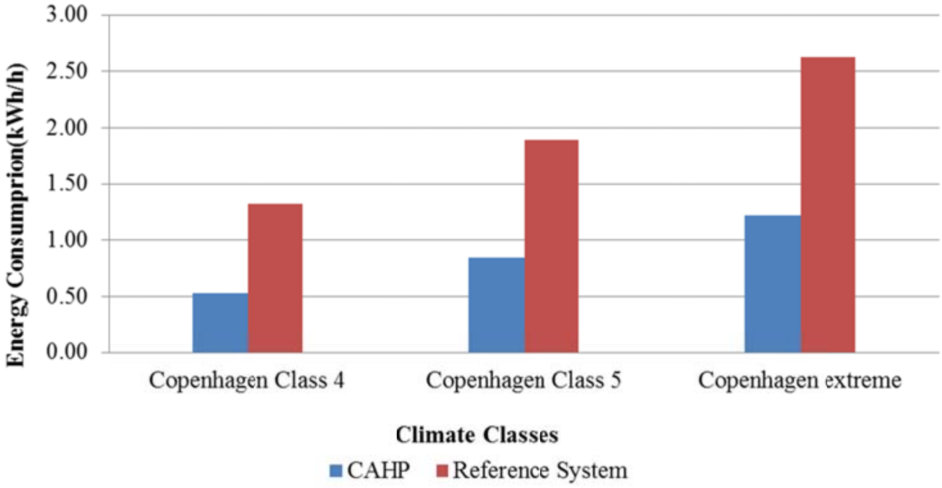


Figure 4.1 Hourly energy consumption of CAHP and reference system in the three summer climate categories in Copenhagen

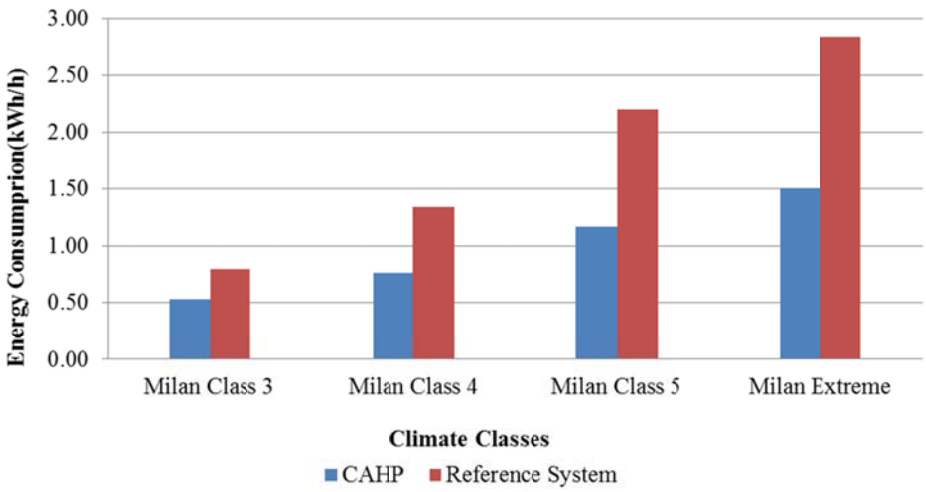


Figure 4.2 Hourly energy consumption of CAHP and reference system in the four summer climate categories in Milan

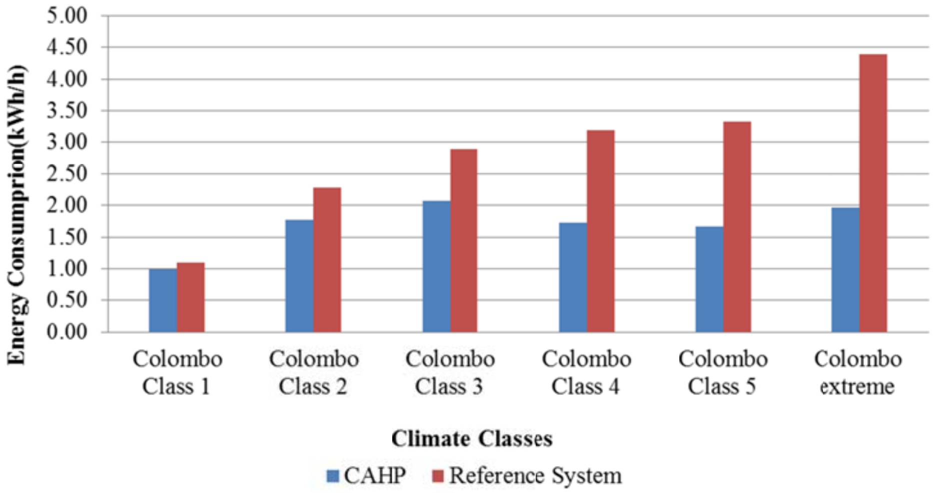


Figure 4.3 Hourly energy consumption of CAHP and reference system in the six summer climate categories in Colombo

With the number of hours of different climate categories, the power consumption of CAHP system and reference system could be calculated. The energy consumption and energy saving proportion are listed in Table 4.4-Table 4.6.

Table 4.4 Total energy consumption of CAHP, reference system and energy saving of CAHP compared to reference system in whole summer of Copenhagen

Cities	Energy consumption(kWh/m^2)		Energy saving
	CAHP	Reference System	CAHP to Reference
Copenhagen	1.99	4.88	59.24%

Table 4.5 Total Energy consumption of CAHP, reference system and energy saving of CAHP compared to reference system in whole summer of Milan

Cities	Energy consumption (kWh/m^2)		Energy saving
	CAHP	Reference System	CAHP to Reference
Milan	14.20	23.69	40.08%

Table 4.6 Total Energy consumption of CAHP, reference system and energy saving of CAHP compared to reference system whole summer of Colombo

Cities	Energy consumption (kWh/m^2)		Energy saving
	CAHP	Reference System	CAHP to Reference
Colombo	127.36	181.98	30.01%

The results of this experiment showed that the energy saving proportion of CAHP to the reference system varies from 30%- 59%. This means that for the same indoor quality, CAHP can save more than 30% of energy consumption when it operates in all the three difference climate zones in summer. In Copenhagen, it has the maximum energy saving potential of 59%.

4.2 Energy Saving in winter condition

The energy consumption of the heat pump in winter was recorded at different classes of climates in Copenhagen and Milan. Together with heating capacities recoded during the experiment, the COP of the heat pump for heating ($COP_{heating}$) was calculated and listed in Table 4.7.

Table 4.7 Hourly energy consumption of heat pump of CAHP in different cities and different categories of winter climates

Cities	CAHP System		
	Capacity	Energy Consumption	$COP_{heating}$
	Heating (kWh/h)	Heat Pump (kWh/h)	
Copenhagen Class 5 Winter	3.45	0.67	5.12
Copenhagen Class 4 Winter	5.06	1.34	3.78
Milan Class 5 Winter	2.24	0.44	5.15
Milan Class 4 Winter	3.32	0.64	5.20
Milan Class 3 Winter	4.62	1.09	4.23
Milan Class 2 Winter	5.48	1.66	3.30

The heating COP of the heat pump in winter varies from 3.8 to 5.1 in Copenhagen winter climate and from 3.3 to 5.2 in Milan winter climate.

The reference system in winter was a gas boiler with heat recovery in the ventilation system. The energy consumption was, therefore, converted to the consumption of natural gas as shown in Table 4.8.

Table 4.8 Hourly energy consumption of gas boiler of reference system in different cities and different categories of winter climates

Cities	Reference system	
	Capacity	Energy Consumption
	Heating (kWh/h)	Gas Boiler (m^3/h)
Copenhagen Class 5 Winter	1.73	0.26
Copenhagen Class 4 Winter	2.49	0.37
Milan Class 5 Winter	1.14	0.17
Milan Class 4 Winter	1.78	0.26
Milan Class 3 Winter	2.22	0.33
Milan Class 2 Winter	2.86	0.43

Since part of the heating capacity of the CAHP was used for regenerating the silica gel rotor (for air cleaning), the effective heating capacity for ventilating and heating of the room was the heat capacity of the reference system. Compared to the heating capacity of the reference system, the time weighted power consumption of the CAHP was around 49% less than heat energy required for heating and ventilation. However, the CAHP system and reference system used different energy sources in winter (CAHP used electricity and the reference system used natural gas), the cost of energy was then used for comparing the energy consumption of CAHP and the reference system. Considering that the price of electricity and gas are also different between Copenhagen and Milano, the cost of the measured energy consumptions in the experiment used by the CAHP were calculated with local energy prices.

The energy prices in Copenhagen, Milan and the measured energy saving are listed in Table 4.9-Table 4.10.

Table 4.9 Different energy prices in different cities

Copenhagen		Milan	
Gas	Electricity	Gas	Electricity
1.15 €/m^3	0.25 €/kWh	0.85 €/m^3	0.20 €/kWh

Table 4.10 Hourly energy consumption of CAHP, reference systems in price and energy saving of CAHP compared to reference system in different cities and different winter climates

Cities	Expense(€/h)		
	CAHP system	Reference system	Energy saving
Copenhagen Class 5 Winter	0.17	0.30	43.25%
Copenhagen Class 4 Winter	0.34	0.43	21.23%
Milan Class 5 Winter	0.09	0.14	39.49%
Milan Class 4 Winter	0.13	0.23	43.13%
Milan Class 3 Winter	0.22	0.28	22.05%
Milan Class 2 Winter	0.33	0.36	8.14%

The comparison of energy saving between CAHP and the reference system in winter mode in the two cities are demonstrated by histograms in Figure 4.4 and Figure 4.5 .

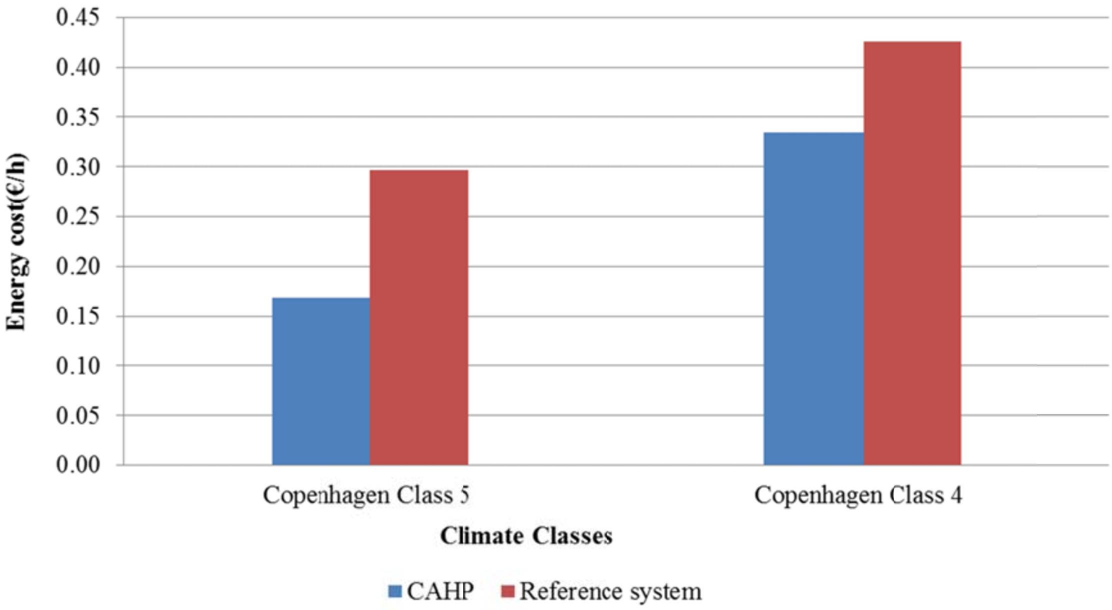


Figure 4.4 Hourly energy consumption of CAHP and reference system in price in the two winter climate categories in Copenhagen

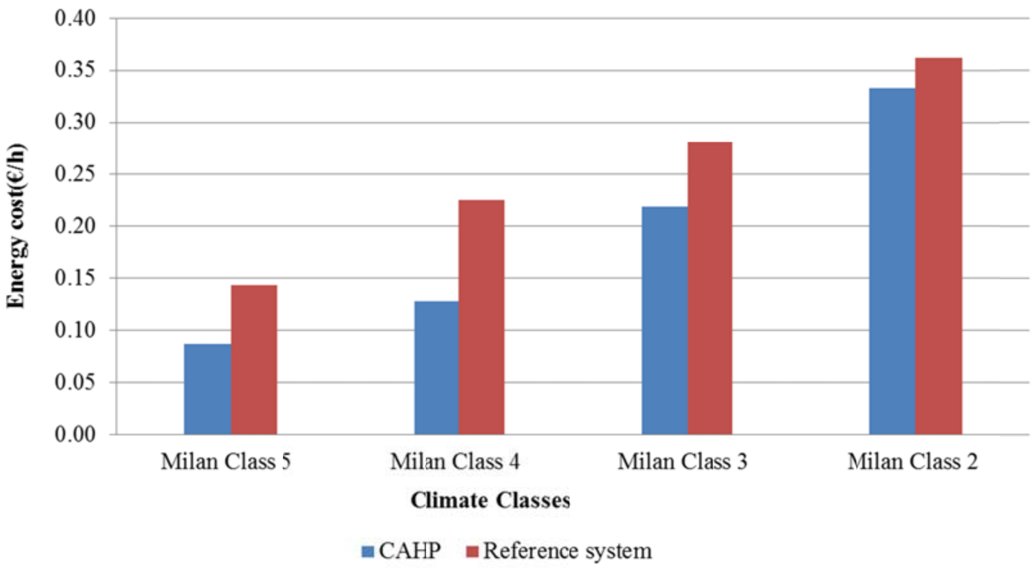


Figure 4.5 Hourly energy consumption of CAHP and reference system in price in the four winter climate categories in Milan

The saving of energy cost in winter Copenhagen was calculated and shown in Table 4.11. In the calculation, the energy consumption was weighted by the number of hours of each climate categories.

Table 4.11 Total energy consumption of CAHP, reference system in price and energy saving of CAHP compared to reference system in whole winter climate of Copenhagen

Cities	Expense(€/m^2)		Expense saving
	CAHP System	Reference System	CAHP to Reference
Copenhagen	7.00	9.35	25.11%

Table 4.12 Total energy consumption of CAHP, reference system in price and energy saving of CAHP compared to reference system in whole winter climate of Milan

Cities	Expense(€/m^2)		Expense saving
	CAHP System	Reference System	CAHP to Reference
Milan	6.82	8.74	21.99%

Based on the cost of energy, the measured energy saving using CAHP in winter season varies from 22%- 25%.

4.3 The annual energy saving

Based on the energy cost of operating CAHP and reference system calculated in summer and winter seasons, the energy cost for the whole year in the three cities could be calculated and compared. The energy consumption and energy saving proportion are listed in Table 4.13. During the calculation, the electricity price in Colombo was investigated and a price of 0.14 €/kWh was used.

Table 4.13 Total energy consumption of CAHP, reference system in price and energy saving of CAHP compared to reference system in whole year of different cities

Cities	Energy consumption(€/m^2)		Energy saving
	CAHP	Reference System	CAHP to Reference
Copenhagen	7.50	10.57	29.07%
Milan	9.66	13.48	28.33%
Colombo	17.83	25.48	30.01%

Based on the cost of energy, the measured energy saving using CAHP varies from 28%- 30% for the whole year in the three cities. The energy cost saving potential doesn't change a lot from one city to another.

5 Discussion

After testing on the prototype CAHP at different climate conditions, the energy performance of the CAHP was found different depending on the climates and seasonal operating mode.

Energy saving potential in summer of different cities: The energy saving proportion was found quite different among Copenhagen, Milan and Colombo in summer season. The results of the experiments showed that when the outdoor humidity ratio was higher, the energy saving proportion became lower. The reason could be that higher regenerating air temperature of the silica gel rotor was required when outdoor air humidity increased. Increasing regenerating temperature requires higher condensing temperature of the heat pump which reduced the COP of the heat pump and increased the energy consumption. In the reference system, dehumidification is done by cooling coil. The condensing temperature is not significantly affected by outdoor humidity ratio and the COP of reference system didn't change as much as it did in CAHP system when outdoor humidity ratio changes. Thus the energy saving potential of CAHP is little sensitive to the humidity ratio of outdoor air.

Energy saving potential in winter of different cities: The energy saving proportion doesn't change much between Copenhagen and Milan in winter seasons. The reason could be that the regenerating temperature of the silica gel was independent of the outdoor air temperature in winter mode. In winter mode, CAHP was not used to control indoor humidity. The regeneration temperature was set at a constant level to keep the air cleaning capacity of the silica gel rotor. Therefore, the regenerating temperature was not affected by outdoor humidity ratio. On the other hand, the experiments at Copenhagen and Milan climate were conducted with the minimum outdoor temperature above 0 ℃. Thus the energy saving potential did not change too much when it was operated in Copenhagen and Milan climates. In reality, the outdoor air temperature in Copenhagen could be lower than it is in Milan. More energy is expected to be saved by the CAHP when it is operated in cold winter climate in Copenhagen since the lower outdoor air temperature, the higher energy consumption for ventilation while the CAHP requires less outdoor air for ventilation. Therefore, the energy saving potential of the CAHP in Copenhagen climate in winter may be under estimated by the experiment. Higher energy saving potential in winter season is expected.

The energy saving potential measured in task 2 of the project was slightly lower than the simulation results of task 1. The following reasons could explain the difference.

1. The regeneration air flow in task 1 was assumed to be 25% of the process air. This was 50% lower than it was used in the experiments in task 2. In the experiments of task 2, the regeneration airflow was set at 50% of the process air to keep the airflow balance between the regeneration angel and process angel in the silica gel rotor. The higher regeneration airflow might lead to a higher energy consumption of CAHP when the regenerating air

temperatures were the same. This could be one of the reasons that lead to the winter energy saving proportions in task 2 lower than it was calculated in task 1.

2. In task 1, the regeneration air was assumed to be partly from exhaust air from the ventilated room in summer. But in task 2, to keep higher air cleaning capacity of the silica gel rotor, the regeneration air was pure outdoor air. Outdoor air is more humid than indoor air in summer and, therefore, higher regeneration temperature was required to reactive the silica gel rotor. This could lead to higher energy consumption of CAHP in summer, which could be another reason to explain the energy saving proportions calculated in task 2 was lower than it was in task 1.
3. The air cleaning efficiency of silica gel rotor was assumed to be 100% in task1 while it was assumed to be 80% in task 2 which should be more realistic. This could be the third reason that led to the difference.

Since the results obtained from task 2 were based on real measurements of the CAHP at the real climate conditions established in the lab, the results should be more reliable than the results obtained in task 1 which was the results of numerical simulation based on many assumptions.

6 Conclusions

A prototype unit of CAHP was designed, developed and tested in task 2 of the project. Energy consumption of the prototype CAHP was measured under different outdoor climates of different locations. To calculate energy saving potential of the CAHP, a reference system was assumed and used for comparison.

The results of the experiments showed that the CAHP saved substantial amount of energy.

1. In summer season in Copenhagen, the CAHP can save 59% of energy consumption for air-conditioning and ventilation. In winter, the energy saving proportion in price can be up to 25%.
2. In summer season in Milan, the CAHP can save 40% energy consumption for cooling, dehumidification and ventilation. In winter, the energy saving proportion in price can be up to 22%.
3. In Colombo, the CAHP can save 30% of electricity compared to reference conventional air-conditioning and ventilation system.
4. The annual saving on the energy cost for all the three climate regions was estimated at around 30%.

The experiment in task 2 validated the energy saving potential of the CAHP. Apart from energy saving, the CAHP should also provide better and controlled indoor air quality. This should be validated by the experiment in task 3 of the project.

7 References

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[5] CEN Report, 2001, CR 1752, "Ventilation for Buildings – Design Criteria for the Indoor Environment", European Committee for Standardization.

8 Appendices

The parameters including airflow rates, temperatures and humidity ratios at the test points shown in Figure 3.3 and Figure 3.4 are listed in the Table 8.1-

Table 8.6. 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 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 Class 5	256.03	190.42	64.92	256.93	94.54	176.25
Copenhagen Class 4	257.77	195.42	63.60	261.87	93.93	172.68
Milan Class 5	255.27	190.85	65.16	256.06	95.35	176.02
Milan Class 4	255.15	191.40	63.81	256.23	93.08	173.26
Milan Class 3	255.95	191.32	62.98	255.91	92.97	172.59
Milan Class 2	249.94	194.77	59.76	263.89	95.38	173.29

Table 8.2 Temperatures measured in experiments for winter climates

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

Table 8.3 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 Class 5	5.38	4.88	5.75	5.75	5.12	5.75	6.58	5.19
Copenhagen Class 4	4.88	4.25	4.34	4.34	4.37	4.34	5.77	4.50
Milan Class 5	5.63	5.31	6.59	6.59	5.66	6.59	7.28	5.69
Milan Class 4	6.21	5.74	6.66	6.66	6.02	6.66	7.64	6.05
Milan Class 3	4.80	4.42	4.86	4.86	4.63	4.86	5.94	4.57
Milan Class 2	4.94	4.23	3.46	3.46	4.16	3.46	5.24	4.15

Table 8.4 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 Class 4	250.89	58.60	251.01	259.98	121.48	119.57	297.15
Copenhagen Class 5	251.37	59.36	250.95	259.25	122.88	121.59	303.63
Copenhagen extreme	250.48	59.19	250.74	257.99	122.27	121.26	303.53
Milan Class 3	248.35	60.34	247.95	256.35	123.20	122.36	302.74
Milan Class 4	250.58	59.89	252.70	259.80	121.45	120.96	300.43
Milan Class 5	248.78	59.55	248.60	255.14	122.03	121.73	301.08
Milan Extreme	248.23	59.73	248.37	254.97	121.49	121.67	303.08
Colombo Class 1	234.82	40.30	232.61	240.26	111.17	129.61	298.86
Colombo Class 2	233.73	40.47	233.65	240.45	109.79	129.30	302.76
Colombo Class 3	231.48	41.69	233.49	239.20	112.85	132.08	305.07
Colombo Class 4	232.27	41.63	235.61	240.52	113.84	131.20	305.96
Colombo Class 5	232.74	40.95	233.33	239.53	113.45	131.05	303.74
Colombo extreme	229.77	41.18	233.36	238.60	114.19	130.75	307.52

Table 8.5 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 Class 4	25.04	23.54	24.69	26.15	18.18	29.58	26.21	37.69
Copenhagen Class 5	25.32	29.19	26.19	30.11	18.40	38.48	30.95	47.67
Copenhagen extreme	26.26	31.88	27.62	32.57	18.34	43.75	33.99	55.53
Milan Class 3	25.46	19.83	24.04	27.28	18.56	35.61	28.32	23.29
Milan Class 4	25.83	25.78	25.87	30.71	18.56	42.67	32.65	33.40
Milan Class 5	25.74	30.56	26.81	33.41	18.04	49.34	35.63	44.45
Milan Extreme	25.94	32.66	27.64	35.48	18.63	55.70	38.06	44.80
Colombo Class 1	25.08	20.63	24.37	31.61	19.63	49.74	34.52	23.12
Colombo Class 2	25.06	24.98	25.24	34.64	17.97	60.16	39.01	28.55
Colombo Class 3	25.36	28.37	26.07	36.26	17.96	64.33	41.10	32.95
Colombo Class 4	24.99	32.52	26.48	35.68	18.25	61.10	40.47	37.20
Colombo Class 5	25.70	36.03	27.89	36.05	18.13	57.62	39.54	55.92
Colombo extreme	25.46	38.28	28.15	37.10	18.30	62.17	41.54	53.16

Table 8.6 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 Class 4	9.92	9.52	9.70	8.96	8.78	9.52	10.00	9.52
Copenhagen Class 5	10.10	10.41	9.95	9.01	8.64	10.41	12.00	10.41
Copenhagen extreme	9.98	11.14	10.08	8.94	8.48	11.14	13.17	11.14
Milan Class 3	9.87	10.30	9.92	8.75	8.63	10.30	11.58	10.30
Milan Class 4	9.98	11.68	10.28	9.08	8.75	11.68	13.65	11.68
Milan Class 5	9.82	12.25	10.15	8.77	8.34	12.25	14.99	12.25
Milan Extreme	10.10	13.64	10.79	8.95	8.60	13.64	17.05	13.64
Colombo Class 1	9.84	14.01	10.65	9.22	8.85	14.01	16.99	14.01
Colombo Class 2	9.79	17.49	11.29	9.43	8.87	17.49	21.58	17.49
Colombo Class 3	9.90	18.18	11.52	9.41	8.89	18.18	23.11	18.18
Colombo Class 4	9.80	18.30	11.34	9.45	8.98	18.30	21.96	18.30
Colombo Class 5	9.98	15.09	10.89	9.07	8.57	15.09	18.63	15.09
Colombo extreme	9.93	17.04	11.29	9.29	8.85	17.04	21.25	17.04