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CLEAN-AIR Heat Pump

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

(Task 1)
Final Report

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1. SUMMARY

This report summarizes task 1 of the Clean Air Heat Pump project – modelling and simulation on energy savings when using the clean air heat pump for ventilation, air cleaning and energy recovery. The total energy consumption of the proposed ventilation systems using clean air heat pump technology was calculated by a theoretical model and compared with the reference ventilation systems (conventional ventilation systems). The energy compared between the two systems includes energy used for heating, cooling and fan. The simulation and energy saving calculation was made for the application of the clean air heat pump in three typical climate conditions, i.e. mild-cold, mild-hot and hot & wet 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.

For the Danish climate (the mild cold climate), the calculations show that the ventilation system using clean air heat pump technology can save up to 42% of energy cost in winter compared to the conventional ventilation system. The energy saving in summer can 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 is, therefore, estimated that more than 35% annual energy saving for ventilation is expected in Denmark using the clean air heat pump ventilation technology.

For the mild hot climate, e.g. the Italian climate, the calculations show that up to 63% of the energy saving can be achieved in summer season. For the winter mode, 17% reduction of the energy cost can be expected for the domestic use. For industrial use, the energy cost of the clean air heat pump may not be favourable due to the industrial price of gas in Italy is much lower than the domestic price.

For the extremely hot and humid climate, the clean air heat pump has the maximum ability of the energy saving for ventilation. The calculations showed that annual energy saving of using the clean air heat pump for ventilation in Sri Lanka is 62%.

In general, the clean air heat pump system is suitable for ventilation in all kind of climates around the world except for the hot and dry climate. The annual energy saving is expected in the range between 30% and 60% depending on the climate. **It is worth noting that the calculated energy reduction of a ventilation system using the clean air heat pump technology was an extra saving compared to a ventilation system that equipped with the high efficiency counter flow heat recovery equipment with a temperature efficiency of 80%. Based on this simulation, it can be concluded that the energy saving of the clean air heat pump for ventilation is remarkable. Therefore, the technology is highly recommended provided that this simulation results are further validated by experiments.**

2. INTRODUCTION

2.1 Background

Ventilation is used for providing acceptable indoor air quality to achieve healthy, comfortable and productive indoor environment. In most buildings, ventilation accounts for as much as 30% of energy consumption. This proportion can be high even in 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 the outdoor environment. In contrast to thermal insulation and airproof technology, ventilation has become the bottleneck on reducing the total energy consumption in buildings.

The total ventilation requirement of a building is determined by the indoor air quality requirement and indoor air pollution sources that are independent of the thermal insulation and airproof of buildings. Due to comfort and health concerns, the ventilation rate prescribed by the existing ventilation standards and guidelines [1,2] 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 [3]. An insufficient ventilation rate also decreases productivity among occupants of office buildings [3]. 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 in most modern cities. Toxic gases and ultra fine particles emitted from vehicles and industries are the major pollutants that are introduced into indoors through ventilation. A positive correlation between mortality and particle concentration (especially ultra fine particle concentration) has been found in a number of epidemiological studies [4,5,6], which shows the importance of controlling the concentration of ultra fine particles. Most indoor ultra fine particles come from the outdoors through ventilation (including infiltration). Normal particle filters cannot stop most of the ultra fine particles. Although the HEPA filter can be used to remove fine particles, it also produces a high pressure drop, which results in much higher electric power consumption for the ventilation fan. Hence, the best solution to decrease energy consumption for ventilation and maintain a healthy and comfortable indoor environment is to develop energy efficient air purification technology to clean the indoor air and use less outdoor air for ventilation.

2.2 The Proposed Technology

The proposed research project is to study and demonstrate a new ventilation approach that can achieve the above mentioned goal. The new approach is proposed based on the results of recent studies [7,8] on air purification technology performed by the International Centre for Indoor Environment and Energy (ICIEE) at DTU. Studies by ICIEE on different air purification technologies show that the regenerative silica gel rotor is the best candidate for indoor air purification among different air purification techniques. Apart from its high air purification efficiency, the regenerative silica gel rotor does not produce any by-products during the air purification process. That is its biggest advantage compared to other air purification technologies using oxidation principles (e.g. photo-catalytic oxidation, plasma oxidation, ozone oxidation, etc.) since all air purifiers using an oxidation process produce by-products that can be irritable or even toxic.

The regenerative silica gel rotor requires a certain amount of heat energy to regenerate the rotor, which is the major barrier for the technique being used in practice. To break the barrier and allow this air cleaning technology to be applicable in replacing ventilation and saving energy, the proposed new ventilation technology combines the silica gel rotor with a heat pump. The novel design of the proposed technology is in the method used in connecting the silica gel rotor and the heat pump to make full use of both heating and cooling from the condenser and evaporator of the heat pump we gave a name to this technology "clean air heat pump". Figure 2.1 to Figure 2.3 show the designs for summer or winter use only, and for use in both summer and winter seasons. The design concept is to transfer the total energy output of the heat pump (both condenser and evaporator) into cooling for summer application and heating for winter application, and leave the ventilation and air purification free of energy consumption. The design also recovers heat energy from the exhaust ventilation air. Therefore, the design combines heating, cooling, ventilation, air-cleaning and energy recovery into one unit to minimize the energy consumption for HVAC in buildings. The technology has further advantages for the Danish energy strategies in the future when wind energy becomes the major energy source and especially suitable for the ventilation in the future active buildings that requires super low energy consumption. This report summarises the theoretical calculation of energy consumption of the clean air heat pump for both summer and winter in three different typical climate zones, i.e. mild-cold, mild-hot and extremely hot & wet climates.

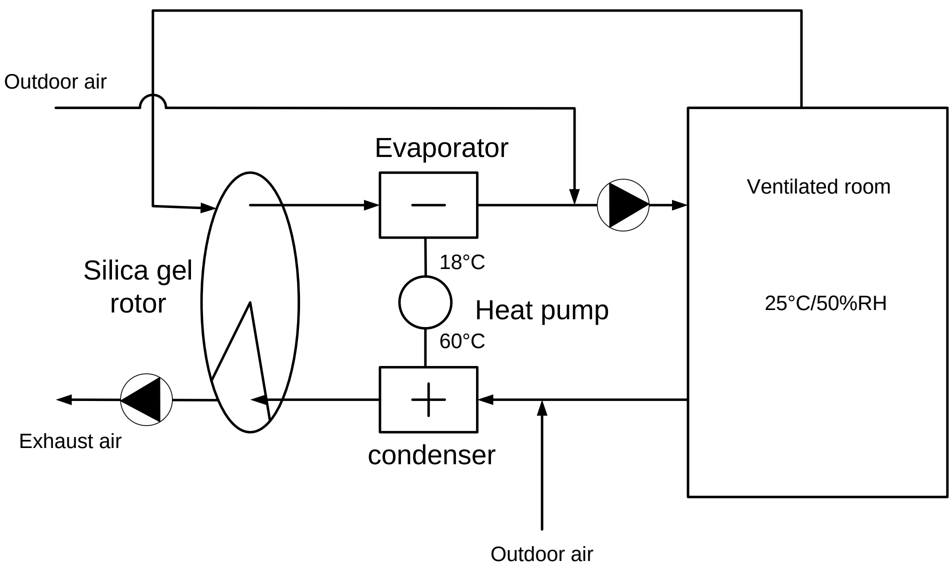

Figure 2.1. Schematic diagram of the clean air heat pump for summer use.

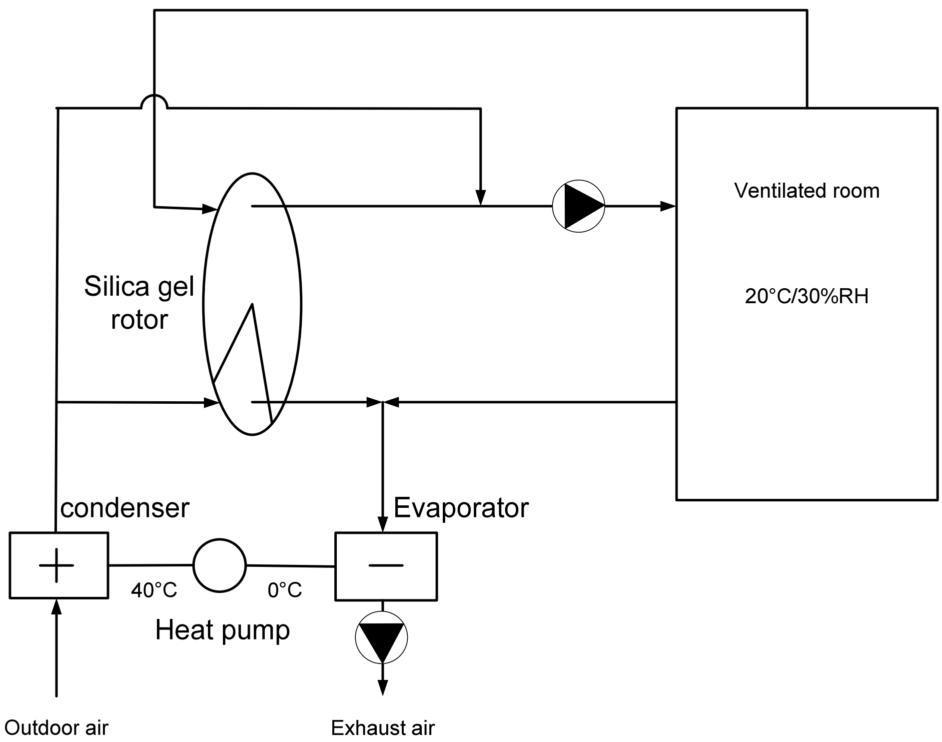

Figure 2.2. Schematic diagram of the clean air heat pump for winter use.

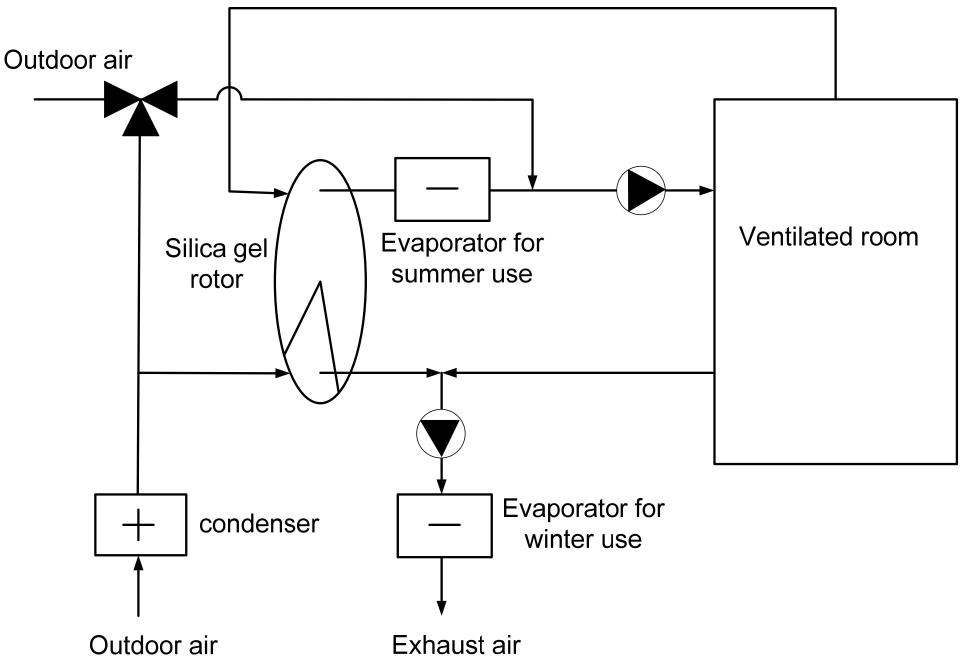

Figure 2.3. Schematic diagram of the clean air heat pump for both summer and winter uses.

3. METHODS

3.1 Model structure for the simulation

A theoretical model was built for calculating the energy consumption of a clean air heat pump. The model was established using an Excel spread sheet that joints two sub-models – a silica gel rotor model simulating the heat and mass transfer of a silica gel rotor and a heat pump model simulating the thermodynamic process of a heat pump. The model of silica gel rotor [8] was developed by Matlab. The heat pump model was developed by a commercially available mass and energy balance software – THERMOFLEX. Figure 3.1 shows the structure of the clean air heat pump model. The input of the model were outdoor air temperature and humidity, indoor air temperature and humidity, outdoor air supply rate, regenerating temperature of the silica gel rotor and the evaporating/condensing temperature of the heat pump. The output of the model was the energy consumption of the clean air heat pump.

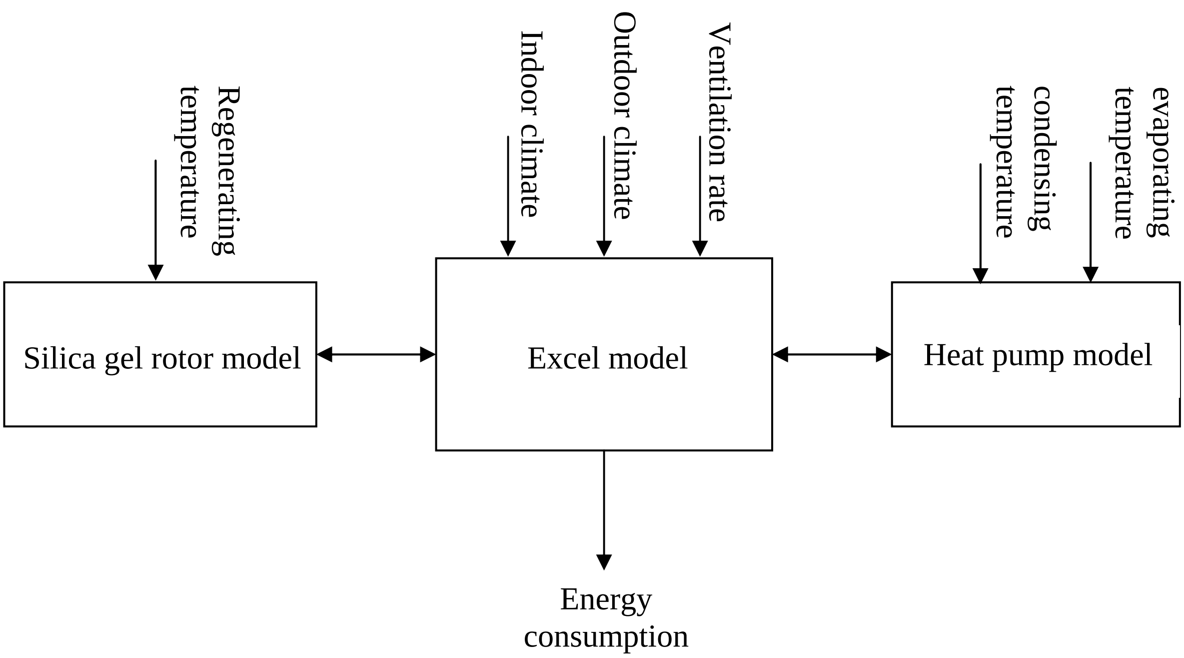

Figure 3.1. Structure of the clean air heat pump model

3.2 Model of the silica gel rotor

The silica gel rotor is a wheel which removes moisture and gaseous pollutants from air by desiccant material that attracts and holds water vapor and VOCs. The primary desiccant used is Titanium Silica Gel that is an adsorbent. Water and pollutants are adsorbed by the desiccant on the rotor. Munters has developed a patented method for manufacturing Titanium Silica Gel in a HoneyCombe® wheel form, which results in a strong and stable structure [9]. Because Titanium Silica Gel is a solid, insoluble desiccant, it is not possible to “wash out” the desiccant from the wheel. This means no special precautions are required even when it is exposed to air at 100% relative humidity.

A regenerative desiccant wheel was used for the “clean air heat pump” modelling. Figure 3.2 shows the principle of the wheel. Air passes through the wheel and comes in contact with the desiccant. The wheel rotates slowly (5 to 10 rph) between two airstreams. The process airstream, the one being dehumidified and cleaned, gives off its moisture and the gaseous pollutants to the desiccant. The process air is thus dry and clean when it leaves the wheel. The rotor, laden of VOCs and moisture, rotates slowly into a second, smaller airstream which is heated. This smaller exhaust airstream, called the reactivation air, warms the desiccant. The silica gel gives off its moisture and the gas phase pollutants, which are then carried away by the reactivation air. The newly dried and clean desiccant material is rotated back into the process airstream where it again begins to adsorb moisture and clean the process air.

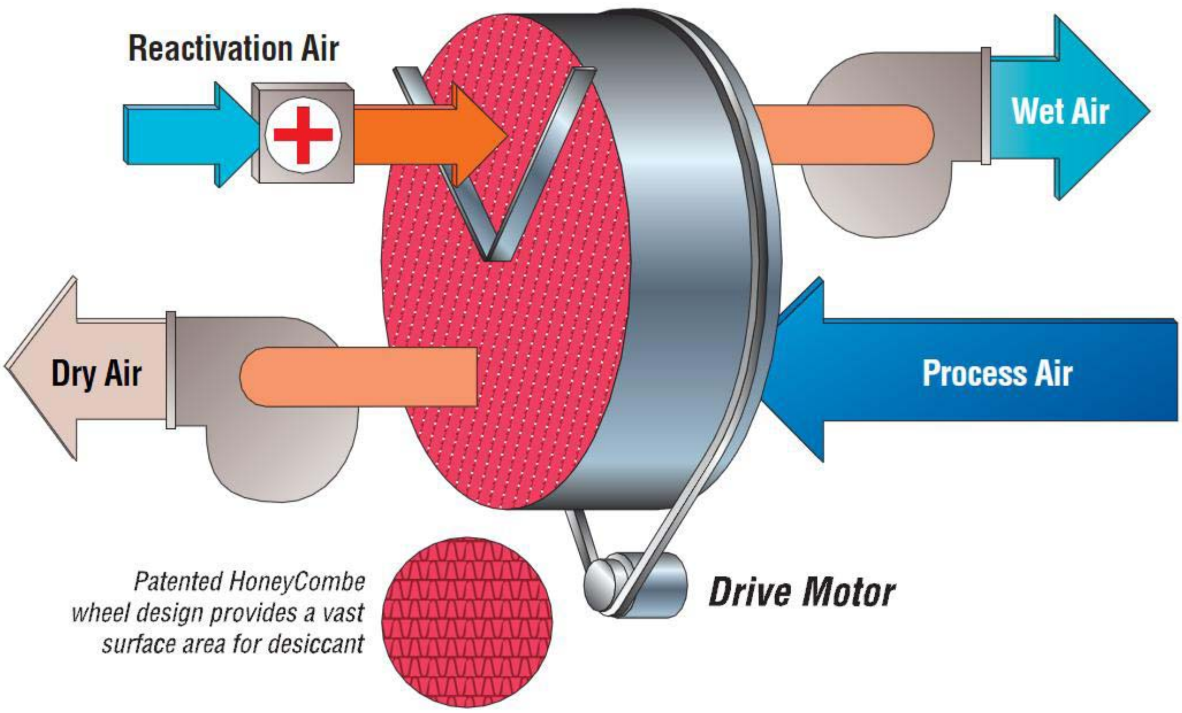

Figure 3.2. Scheme of the MUNTERS silica gel rotor [24].

To simulate the functioning of the regenerative silica gel rotor, a program developed by Zhang et al. in 2007 [8] was used. This theoretical model is based on the silica gel rotor MUNTERS ML 690. According to previous experimental studies [7], the competition between VOCs and moisture in the adsorption and desorption process was assumed to be negligible in the study used to make the model. The following assumptions were also made before making the model:

- The heat conduction and mass diffusion in the airstream and solid materials along the axial direction of the rotor are neglected.

- The adsorption heat of VOCs is neglected because of its minor proportion to that of moisture.

With these assumptions, the heat, moisture and VOC transfer in the silica gel rotor during both the adsorption and desorption process can be described with a one-dimensional transient coupled heat and mass transfer model. The model was built as a MATLAB script.

A series of experiments were conducted to assess the validity of the model when toluene and 1,2-dichloroethane were dosed at a series of concentrations. The time-mean state of the two VOCs at the outlet of the regeneration side and the inlet and outlet of the process side were measured in real time. The calculated concentration difference between the inlet and outlet of the process air section was quite close to the measured value at all the inlet concentration for the two VOCs [8].

This model simulates a number of revolutions of the wheel. After each revolution it provides the properties of the outgoing air. It also assumes that the wheel at the beginning is clean and at the ambient properties; then, revolution after revolution, it converges at the conditions of heat and mass balance among the ingoing and outgoing airstreams. To reach constant values, it has been proved that iterations should be at least 15, and for this reason 15 revolutions have been assumed as the running cycle number for each simulation.

In this work, only the temperature and the humidity after and before the regeneration and the process part of the rotor were used, assuming that the clean effect of the silica gel rotor was already demonstrated in the previous mentioned study. The theoretical model presented has been thus very useful to calculate the thermodynamic properties of the air after the wheel. Each simulation takes from 40 minutes, depending on the processor speed of the computer used. Up to six computers have been used at the same time to reduce the total time for all simulations.

Table 3.1 summarizes the properties of the silica gel rotor (MUNTERS ML690) used to verify the model built by Zhang et al. and used in this project for all simulations.

Table 3.1. Geometrical parameters of Munters ML 690

MUNTER ML		
Thickness	0.30	m
Radius	0.225	m
Rotation speed	0.167	rpm
Process angle	270°	
Regeneration angle	90°	
Nominal process airflow	690	m^3/h

3.3 Model of the heat pump

The software called THERMOFLEX was used to model the heat pump. It was used to evaluate the COP of the heat pump under different load conditions. THERMOFLEX is a mass and energy balance tool developed for the modelling of thermal systems. The simplest possible cycle was modelled, including compressor, motor, condenser, expansion valve and evaporator (Figure 3.3). Different simulations have been carried out, for different evaporating and condensing pressures.

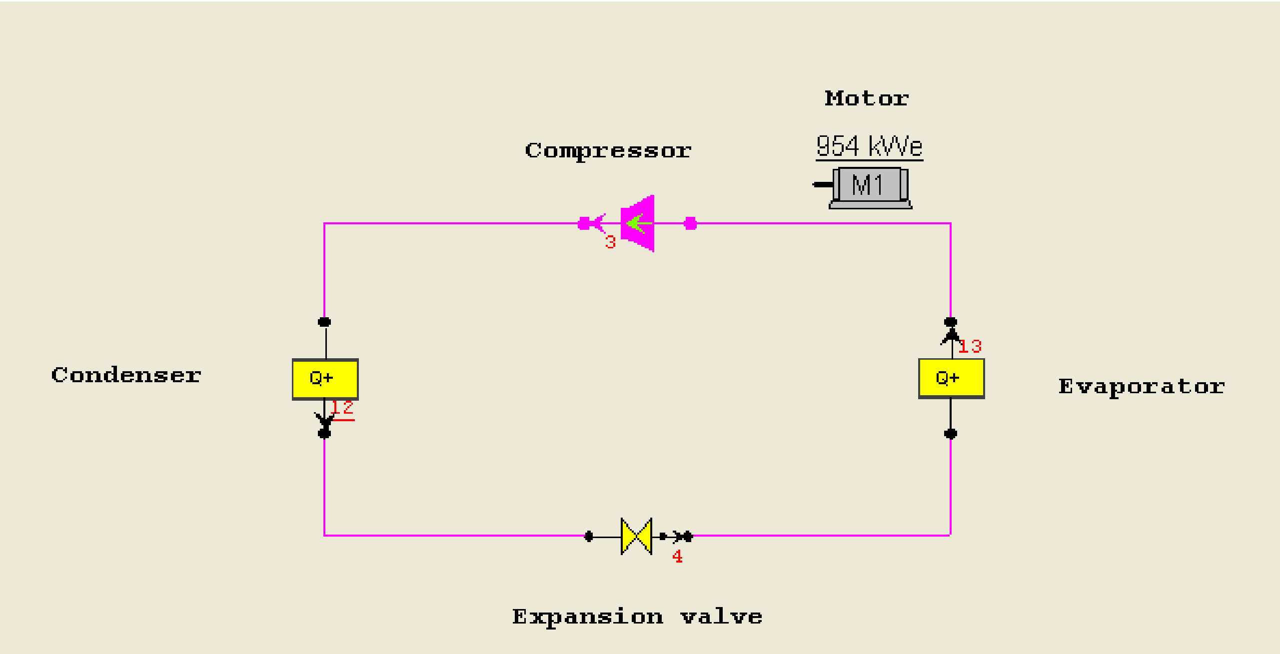

Figure 3.3 Screenshot of the heat pump model developed using THERMOFLEX.

The following four assumptions were made for the heat pump:

- The compressor of the heat pump equipped with a frequency controller that regulates the speed of the compressor.
- The refrigerant assumed is R134a.
- An isentropic efficiency of 85% was assumed for the cycle.
- The engine and the mechanical efficiency assumed for the compression system are 90% and 95% respectively.

It was assumed that a control system automatically regulates the speed of the compressor according to the room air temperature in order to obtain the heating/cooling capacity desired. For instance, in the winter mode, when the outdoor temperature is very low, the compressor works at the maximum speed, while when the outside temperature is higher it reduces the speed of the compressor in order to save energy. During the winter season, if the outside temperature is very cold, the heat recovered at the evaporator may not be enough to heat the air at the condenser. A post heating coil has been thus designed after the condenser to provide sufficient heating to the reactivation air. The post heating coil was designed after the condenser and not before in order to keep the minimum gap between the condensing and evaporating pressures. This operation allows having a higher COP.

Moreover, during the winter season, it has been assumed that the evaporator can only recover the heat from the exhausted air until the temperature of the air after the evaporator reaches 0˚C. This temperature cannot be lower because the risk of ice that may be formed outside the evaporator.

4. ASSUMPTIONS FOR THE SIMULATION

4.1 Clean air heat pump ventilation system

The calculation was performed for both summer and winter seasons using summer and winter mode of the clean air heat pump design as show in Figure 2.1 and 2.2.

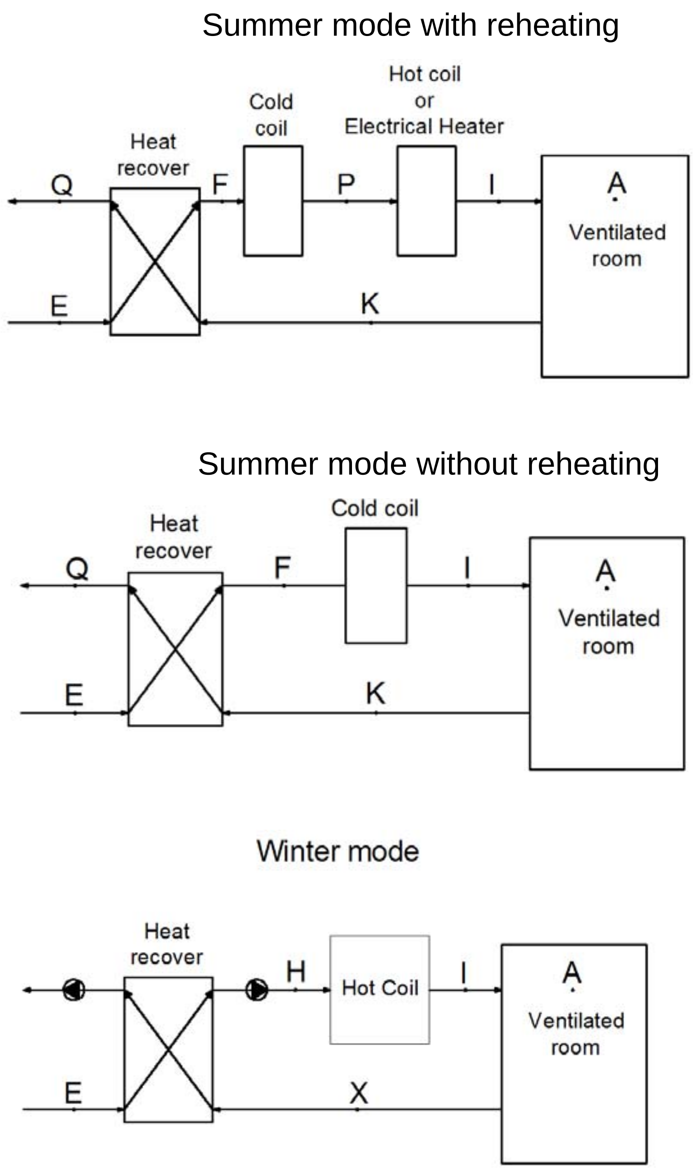

Figure 4.1. The reference ventilation systems.

4.2 Reference ventilation system

To calculate the energy saving of the clean air heat pump, three conventional ventilation systems with heat recovery units were used as the references. The heat recovery unit was assumed to have a counter flow heat exchanger with a high temperature efficiency of 80%. Figure 4.1 shows the three reference ventilation systems. The reference systems include two summer mode systems (one with and one without reheating) and one winter mode system.

4.3 Indoor climate conditions and ventilation rate

The assumption of indoor air temperature and humidity in summer was 25˚C and 50%RH. The indoor air temperature in winter was assumed at 22 ˚C. The relative humidity in winter was not specified but was an output of the model to check if it was controlled in an acceptable range. Moisture load indoors was assumed to be 50g/h per person. The outdoor air supply rate for the reference system was determined according to the European Ventilation Guideline (CEN 1752)[2] as shown in Table 4.1 and 4.2. To achieve the best indoor air quality as defined by CEN 1752, category A was chosen for calculating the outdoor air supply rate. The same airflow rate was also used for clean air heat pump ventilation system. Since the clean air heat pump can remove more than 85% of the pollutants in the air processed as shown in the previous study [7], the required outdoor air supply rate for the clean air heat pump ventilation system can be reduced by 85%. Based on this result, the ventilation airflow for the clean air heat pump system was determined as 80% recirculation and 20% outdoor air. In this arrangement, the out air supply used by the clean air heat pump ventilation systems still fulfilled the minimum outdoor air requirement of CEN 1752 (Category C for occupants only) but would achieve better indoor air quality of Category A. Table 4.3 and 4.4 show the airflow rates and the building area used for the simulation.

Table 4.1. Ventilation requirement prescribed by CEN 1752 assuming both building materials and occupants are the indoor pollution sources

SINGLE OFFICE - Required ventilation rate for comfort		
CATEGORY A	2	$l / s / m^2_{floor}$
CATEGORY B	1.4	$l / s / m^2_{floor}$
CATEGORY C	0.8	$l / s / m^2_{floor}$

Table 4.2. Ventilation requirement prescribed by CEN 1752 assuming occupants are the only indoor air pollution source

SINGLE OFFICE - Required ventilation rate ignoring the building as a pollution source		
CATEGORY A	1	$l / s / m^2_{floor}$
CATEGORY B	0.7	$l / s / m^2_{floor}$
CATEGORY C	0.4	$l / s / m^2_{floor}$

Table 4.3. Airflow rates and the building area used for the winter mode simulation

	Clean air heat pump system	Reference system
Building area (m^2)	120	120
Outdoor airflow (L/s)	48	240
Recirculation airflow (L/s)	192	0
Total supply airflow (L/s)	240	240

Table 4.4. Airflow rates and the building area used for the summer mode simulation

	Clean air heat pump system	Reference system
Building area (m^2)	90	90
Outdoor airflow (L/s)	36	180
Recirculation airflow (L/s)	144	0
Total supply airflow (L/s)	180	180

4.4 Outdoor climate conditions

The energy saving of the clean air heat pump was calculated for three typical climate conditions, i.e. mild-cold, mild-hot and extremely hot & wet 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. Since the silica gel rotor model requires long time to calculate for one condition, we simplified the climate conditions for summer or winter operating mode into five classes respectively. Table 4.5 shows the operating period of summer and winter mode for the three cities. Table 4.6 to 4.10 show the outdoor air temperature, humidity ratio and the number of hours in each class used for the calculation of the three cities in both summer and winter seasons.

Table 4.5. The time period of winter and summer mode used for the energy calculations in the three cities

LOCATION	WINTER MODE	SUMMER MODE
Denmark	16th September – 30th April	1st May – 15th September
Italy	16th October – 15th April	16th April – 15th October
Sri Lanka	-----	1st January – 31th December

Table 4.6. The five classes of climate data used for the simulation of the Danish summer

DENMARK - 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	24
Extreme case	32.1	0.0115	---

Table 4.7. The five classes of climate data used for the simulation of the Danish winter

DENMARK - WINTER			
	T [°C]	x [kg_w/kg_a]	Hours
1st class	-16.69	0.00089	16
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	---

Table 4.8. The five classes of climate data used for the simulation of the Italian summer

ITALY - SUMMER			
	T [°C]	x [kg_w/kg_a]	Hours
1st class	10.5	0.0077	219
2nd class	15.5	0.0096	662
3rd class	20.5	0.0105	820
4th class	25.5	0.0119	507
5th class	30.5	0.0127	171
Extreme case	33.00	0.0136	---

Table 4.9. The five classes of climate data used for the simulation of the Italian winter

ITALY - WINTER			
	T [°C]	x [kg_w/kg_a]	Hours
1st class	-5.3	0.0025	99
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	---

Table 4.10. The five classes of climate data used for the simulation of the Sri Lanka climate

SRI LANKA			
	T [°C]	x [kg_w/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	4
Extreme case	38.0	0.0171	---

5. RESULTS OF THE SIMULATION

Since the clean air heat pump uses mainly electrical power, while most of the existing heating systems use gas as heating power, the comparison of energy consumption between clean air heat pump and reference systems should be made in the way that they are comparable. The best way is to compare the cost of energy consumption for the two systems. The results of the calculation are presented by comparing the energy cost of the ventilation systems using the clean air heat pump technology and the conventional heating/cooling + heat recovery technology.

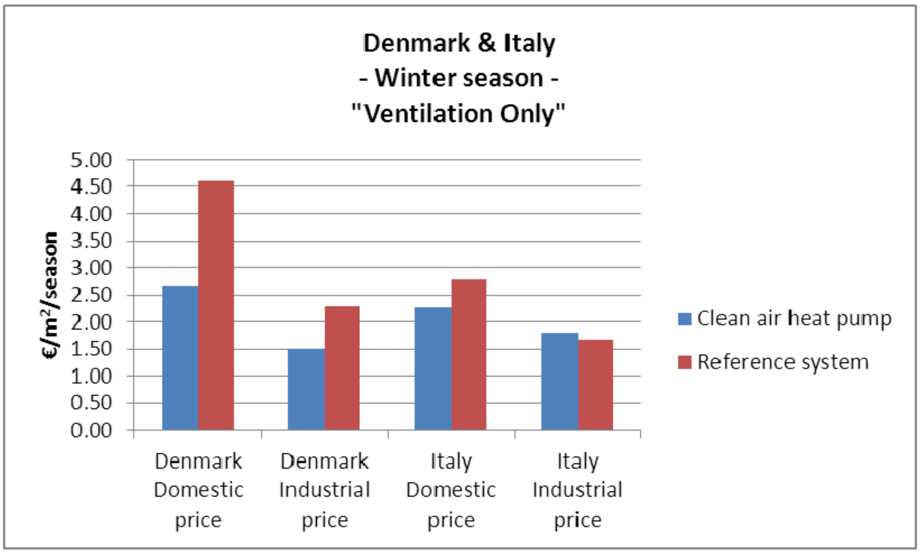

Figure 5.1. Comparison of the engery cost per square meter between clean air heat pump and the reference ventilation systems in winter season (Danish and Italian climate).

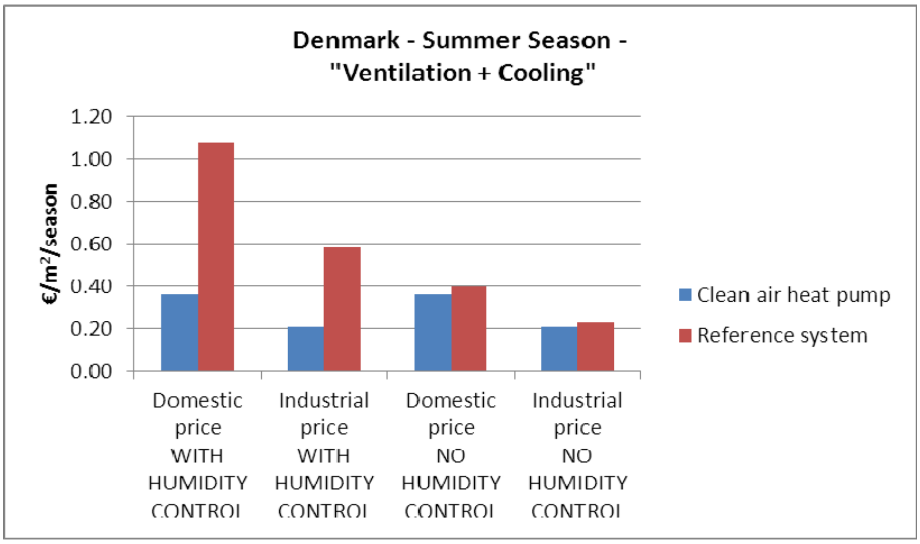

Figure 5.2. Comparison of the engery cost per square meter between clean air heat pump and the reference ventilation and cooling systems in summer season (Danish climate).

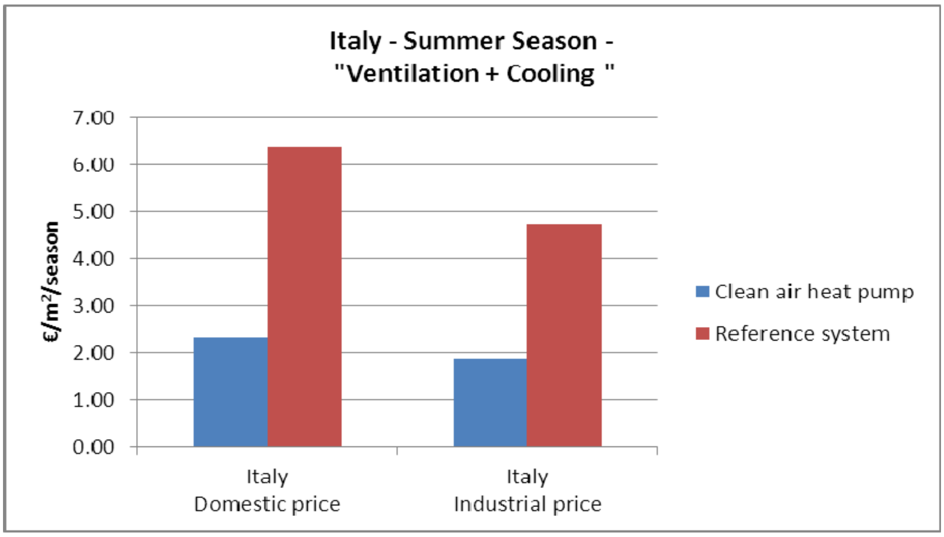

Figure 5.3. Comparison of the engery cost per square meter between clean air heat pump and the reference ventilation and cooling systems in summer season (Italian climate).

Since the energy price in Sri Lanka was not available and considering that only summer ventilation mode is used in Sri Lanka, electricity consumptions of both clean air heat pump system and the reference system were used for the comparison.

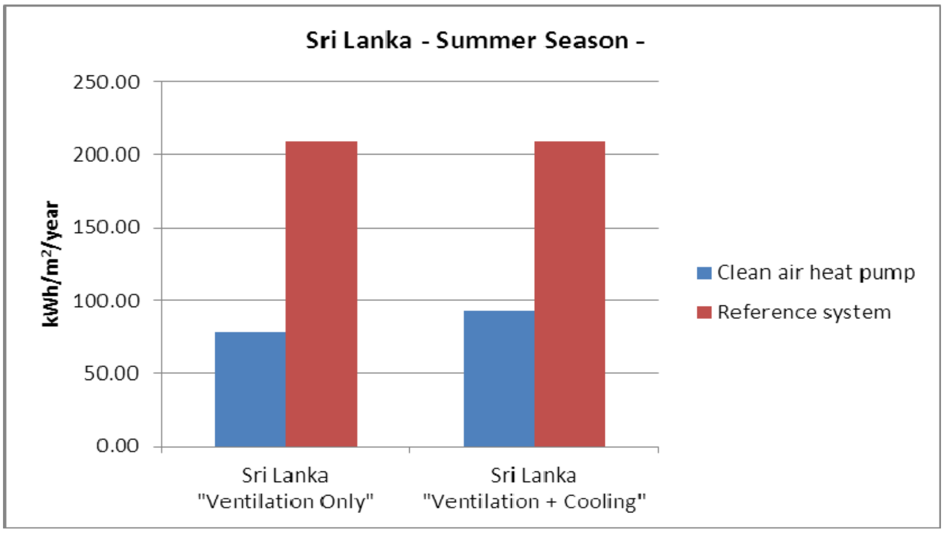

Figure 5.4. Comparison of the annual engery consumption per square meter between clean air heat pump and the reference ventilation and cooling systems in Sri Lanka climate.

6. CONCLUSIONS

For the Danish climate (the mild cold climate), the calculations show that the ventilation system using clean air heat pump technology can save up to 42% of energy cost in winter compared to the conventional ventilation system. The energy saving in summer can be as high as 66% for the ventilation system with humidity control and 9% for the ventilation system without 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 is, therefore, estimated that more than 35% annual energy saving for ventilation is expected in Denmark using the clean air heat pump ventilation technology.

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For the extremely hot and humid climate, the clean air heat pump has the maximum ability of the energy saving for ventilation. The calculations showed that annual energy saving of using the clean air heat pump for ventilation in Sri Lanka is 62%.

In general, the clean air heat pump system is suitable for ventilation in all kind of climates around the world except for the hot and dry climate. The annual energy saving is expected in the range between 30% and 60% depending on the climate. **It is worth noting that the calculated energy reduction of a ventilation system using the clean air heat pump technology was an extra saving compared to a ventilation system that equipped with the high efficiency counter flow heat recovery equipment with a temperature efficiency of 80%. Based on this simulation, it can be concluded that the energy saving of the clean air heat pump for ventilation is remarkable.**

One concern on the use of desiccant rotor in winter season is the indoor air humidity. Since desiccant rotor removes some moisture from the air, one could worry about too low indoor humidity may occur when using desiccant rotor to process the recirculation air. The simulation shows that such worry is not necessary. In winter mode, the regeneration temperature of the silica gel rotor is much lower than it is used in summer because of the low moisture load to be removed. Usually, the low indoor humidity in winter season is mainly due to the dry outdoor environment. The higher outdoor air used for ventilation the lower indoor humidity becomes. Since the clean air heat pump ventilation system requires much less outdoor air supply compared to the conventional ventilation system, the indoor relative humidity is 5% increased on average when the clean air heat pump ventilation system is used even if the silica gel rotor removes some of the moisture from the air. If moisture control of indoor air is needed, the clean air heat pump is the best choice for ventilation since the humidity of indoor air can be easily controlled by adjusting the regeneration temperature of the silica gel rotor. The results of this project show that the clean air heat pump has a positive impact on indoor humidity in both winter and summer. It gives the ventilation system a very energy efficient measure to control indoor humidity.

Except for energy saving and moisture control, the effect of desiccant rotor on air cleaning is another advantage. Compared to other air cleaning technology, such as photocatalytic and plasma air cleaner etc, the air cleaning using silica gel rotor produces no by-products in the processed air, which is safe and healthy for the occupants in the ventilated buildings. Based on this simulation study and all the other available information, the clean air heat pump is a highly recommended technology for ventilation provided that this simulation results are further validated by experiments.

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8. APPENDICES

Energy calculation and cost analysis for the application of the clean air heat pump in Denmark, Italy and Sri Lanka

8.1 Application in Denmark

8.1.1 Winter application

For the Danish winter application, the calculation was performed by two models:

a) The clean air heat pump supplies isothermal airflow to the ventilated room (denoted as "ventilation only"). In this model, the clean air heat pump mainly used to clean the indoor air and warm up the supplied outdoor air for ventilation.
b) The clean air heat pump supplies airflow with the temperature high than the room temperature (denoted as "ventilation + heating"). In this model, the clean air heat pump provides heating to the ventilated room in addition to the "ventilation only" function.

Calculation using ventilation only model

The calculations are summarized in Table 8.1 to 8.4. Table 8.1 shows the input values assumed in the calculation; Table 8.2 shows devices used to process the air in both the clean-air heat pump and the reference system; Table 8.3 shows their energy consumptions of clean air heat pump and reference system; Table 8.4 shows the calculated cost reduction using the clean air heat pump compared to the reference system.

Table 8.1. The input values used for the Danish winter “Ventilation Only” model.

DENMARK WINTER “VENTILATION ONLY”		
	Desiccant-wheel cycle	Reference system
Supplied Temperature	22°C	22°C
Room air Temperature	22°C	22°C
Room Area	120 m^2	120 m^2
Outdoor Airflow	48 l/s	240 l/s
Outdoor air temperature and humidity	Shown in table 4.7	Shown in table 4.7
Recirculation Airflow	192 l/s	0 l/s
Supplied Airflow	240 l/s	240 l/s
Total sensible load considered	0 kW	0 kW
Total latent load considered	0.42 kW	0.42 kW
Sensible load covered by the ventilation system	0 kW	0 kW
Latent load covered by the ventilation system	0.42 kW	0.42 kW
Humidity control	NO	NO

Table 8.2. Devices used in ventilation systems simulated by the Danish winter "Ventilation Only” model.

DENMARK WINTER “VENTILATION ONLY”	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Post hot coil	Hot coil
Silica gel wheel	
Evaporator	

Table 8.3. Energy consumption obtained with the Danish winter "Ventilation Only" model.

	CLEAN-AIR HEAT PUMP			REFERENCE SYSTEM	
	Post hot coil	Compressor	Fans	Hot coil	Fans
Energy consumption per season [kWh/season]	100	948	389	4172	493
Area [m^2]	120				
Energy consumption per m^2 [kWh/m^2/season]	0.83	7.90	3.24	34.77	4.11

Table 8.4. Cost reduction for the Danish winter "Only Ventilation" model.

	Open - Cycle Desiccant Wheel System	Traditional Air Conditioning System	Cost Reduction
Domestic price	2.67 €/season/m2	4.60 €/season/m2	**41.94%**
Industrial price	1.51 €/season/m2	2.28 €/season/m2	**33.68%**

Calculation using ventilation + heating model

Two levels of regeneration temperature (30°C and 40°C) were used in the calculation in order to understand if it is more energy efficient to increase the regeneration temperature. The sensible heating demand was fixed at 2 kW for a 120 square meters building that is more then what the ventilation system can supply with the minimum supplied airflow used. A radiator system in the room was assumed to supply the difference between heating demand in the room and the heating supplied by the ventilation system. Therefore, increasing or decreasing the heating demand in the room did not influence the calculated energy consumption of the ventilation system. The tables below summarize the assumptions and the calculation.

Table 8.5 Main values used for the Danish winter "Ventilation + Heating" model.

DENMARK WINTER "VENTILATION + HEATING"			
	Desiccant wheel cycle 30°C regeneration temperature	Desiccant wheel cycle 40°C regeneration temperature	Reference system
Supplied Temperature	≈ 24.7 °C	≈ 27.9 °C	22°C
Indoor air Temperature	22°C	22 °C	22°C
Building Area	120 m^2	120 m^2	120 m^2
Outdoor Airflow	48 l/s	48 l/s	240 l/s
Outdoor air temperature and humidity	Shown in table 4.7	Shown in table 4.7	Shown in table 4.7
Recirculation Airflow	192 l/s	192 l/s	0 l/s
Supplied Airflow	240 l/s	240 l/s	240 l/s
Total sensible heating demand assumed	2 kW	2 kW	2 kW
Total latent load assumed	0.42 kW	0.42 kW	0.42 kW
Sensible heating covered by the ventilation system	≈ 0.77 kW	≈ 1.65 kW	0 kW
Latent load covered by the ventilation system	0.42 kW	0.42 kW	0.42 kW
Sensible heating covered by other systems	Radiators ≈ 1.23 kW	Radiators ≈ 0.35 kW	Radiators 2 kW
Humidity control	NO	NO	NO

Table 8.6 Devices used in ventilation systems simulated by the Danish winter "Ventilation Only" model.

DENMARK WINTER "VENTILATION ONLY"	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Post hot coil	Hot coil
Silica gel wheel	
Evaporator	

Table 8.7 Energy consumption obtained with the Danish winter "Ventilation + Heating" model.

		Clean-air heat pump				Reference system		
		Post hot coil	Compressor	Fan	Radiators system	Hot coil	Fan	Radiators system
Energy consumption per season [kWh/season]	Reg. T 30°C	568	1360	465	3709	4172	535	6074
	Reg. T 40°C	1257	2294	466	990			
Area [m²]		120						
Energy consumption per m² [kWh/m²/season]	Reg. T 30°C	4.73	11.33	3.87	30.91	34.8	4.46	50.62
	Reg. T 40°C	10.47	19.12	3.88	8.25			

Table 8.8 Cost reduction for the Danish winter "Ventilation + Heating" model.

	Rotor system 30°C Reg. T	Rotor system 40°C Reg. T	Traditional air conditioning system	Cost reduction 30°C Reg. T	Cost reduction 40°C Reg. T
Domestic price	7.27 €/season/m^2	7.30 €/season/m^2	10.00 €/season/m^2	**27.32%**	**26.98%**
Industrial price	3.79 €/season/m^2	3.97 €/season/m^2	4.86 €/season/m^2	**22.02%**	**18.24%**

8.1.2 Summer application

For the Danish summer application, the calculation was also performed by two models. In both case, the clean air heat pump had the same design. It performed air cleaning, cooling and dehumidification control. However, two reference ventilation systems were used for the comparison of energy consumption. The two reference ventilation systems were:

a) A ventilation system without humidity control. The reference ventilation system cools the outdoor air without condensation and reheating before the air supplied into the room. The consequence was that the indoor air temperature can be controlled but indoor humidity was not controlled. In general when using the reference ventilation system, the indoor air humidity follows outdoor humidity and was higher than the compared case using clean air heat pump system for ventilation. Thus the reference ventilation system must maintain lower indoor air temperature in order to compare the two ventilation system that provided the indoor thermal environments with the same sensation.
b) A ventilation system with humidity control. The reference ventilation system cools and dehumidifies the air by condensation. Reheating was used to control the supply air temperature to be the same as air temperature supplied by the clean air heat pump system. Thus both systems maintained the same indoor air temperature and humidity.

Calculation using the reference ventilation system without humidity control model

The calculations are summarized in Table 8.9 to 8.12. Table 8.9 shows the input values assumed in the calculation; Table 8.10 shows devices used to process the air in both the clean-air heat pump and the reference system; Table 8.11 shows their energy consumptions of clean air heat pump and reference system; Table 8.12 shows the calculated cost reduction using the clean air heat pump compared to the reference system.

Table 8.9. Main values used for the Danish summer "Ventilation + Cooling - reference system without humidity control" model.

DENMARK SUMMER "VENTILATION + COOLING - REFERENCE SYSTEM WITHOUT HUMIDITY CONTROL"		
	Desiccant-wheel cycle	**Reference system**
Supplied Temperature	17 °C	15 °C
Indoor air Temperature	25 °C	23 °C
Building Area	90 m^2	90 m^2
Outdoor Airflow	36 l/s	180 l/s
Outdoor air temperature and humidity	Shown in table 4.6	Shown in table 4.6
Recirculation Airflow	144 l/s	0 l/s
Supplied Airflow	180 l/s	180 l/s
Total sensible cooling demand assumed	1.70 kW	1.70 kW
Total latent load assumed	0.32 kW	0.32 kW
Sensible cooling load covered by the ventilation system	1.70 kW	1.70 kW
Latent load covered by the ventilation system	0.32 kW	0.32 kW
Humidity control	NO	NO

Table 8.10. Devices used in systems simulated with the Danish summer "Ventilation + Cooling - reference system without humidity control" model.

DENMARK SUMMER "VENTILATION + COOLING - REFERENCE SYSTEM WITHOUT HUMIDITY CONTROL"	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Silica gel wheel	Cold coil
Evaporator	

Table 8.11. Energy consumption obtained with the Danish summer "Ventilation + Cooling - reference system without humidity control" model.

	CLEAN-AIR HEAT PUMP		REFERENCE SYSTEM	
	Compressor	Fan	Cold coil	Fan
Energy consumption per season [kWh/season]	87	55	100	55
Area [m²]	90			
Energy consumption per m² [kWh/m²/season]	0.96	0.61	1.11	0.61

Table 8.12. Cost reduction for the Danish summer "Ventilation + cooling - reference system without humidity control" model.

	Open - Cycle Desiccant Wheel System	Traditional Air Conditioning System	Cost Reduction
Domestic price	0.36 €/season/m²	0.40 €/season/m²	**8.78%**
Industrial price	0.21 €/season/m²	0.23 €/season/m²	**8.78%**

Note: In these models no gas is used, the cost reduction using either industrial or domestic price is the same.

Calculation using the reference ventilation system with humidity control model

The calculations are summarized in Table 8.13 to 8.16. Table 8.13 shows the input values assumed in the calculation; Table 8.14 shows devices used to process the air in both the clean-air heat pump and the reference system; Table 8.15 shows their energy consumptions of clean air heat pump and reference system; Table 8.16 shows the calculated cost reduction using the clean air heat pump compared to the reference system.

Table 8.13. Main values used for the Danish summer “Ventilation + Cooling - reference system with humidity control” model.

DENMARK SUMMER “VENTILATION + COOLING - REFERENCE SYSTEM WITH HUMIDITY CONTROL”		
	Desiccant-wheel cycle	**Reference system**
Supplied Temperature	17 °C	17 °C
Inside Temperature	25 °C	25 °C
Building Area	90 m^2	90 m^2
Outdoor Airflow	36 l/s	180 l/s
Outdoor air temperature and humidity	Shown in table 4.6	Shown in table 4.6
Recirculation Airflow	144 l/s	0 l/s
Supplied Airflow	180 l/s	180 l/s
Total sensible cooling demand assumed	1.70 kW	1.70 kW
Total latent load assumed	0.32 kW	0.32 kW
Sensible cooling load covered by the ventilation system	1.70 kW	1.70 kW
Latent load covered by the ventilation system	0.32 kW	0.32 kW
Humidity control	YES	YES

Table 8.14. Devices used in systems simulated with the Danish summer "Ventilation + Cooling - reference system with humidity control” model.

DENMARK SUMMER “VENTILATION + COOLING - REFERENCE SYSTEM WITH HUMIDITY CONTROL”	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Silica gel wheel	Cold coil
Evaporator	Hot Coil

Table 8.15. Energy consumption obtained with the Danish summer "Ventilation + Cooling - reference system with humidity control" model.

	CLEAN-AIR HEAT PUMP		REFERENCE SYSTEM		
	Compressor	Fan	Cold coil	Hot coil	Fan
Energy consumption per season [kWh/season]	87	55	275	251	43
Area [m²]	90				
Energy consumption per m² [kWh/m²/season]	0.96	0.60	2.79	3.05	0.48

Table 8.16. Cost reduction for the Danish summer "Ventilation + Cooling - reference system with humidity control" model.

	Open - Cycle Desiccant Wheel System	Traditional Air Conditioning System	Cost Reduction
Domestic price	0.36 €/season/m²	1.08 €/season/m²	**66.30%**
Industrial price	0.21 €/season/m²	0.58 €/season/m²	**64.59%**

8.2 Application in Italy

8.2.1 Winter application

The ventilation only model was used for the calculation on the winter application in Italy.

The input values assumed in the calculation and the devices used to process the air in both the clean-air heat pump and the reference system were the same as used in the Danish winter application except for the input values of outdoor air temperature and humidity as shown in Table 8.17. Table 8.19 shows the energy consumptions of clean air heat pump and reference system; Table 8.20 shows the calculated cost reduction using the clean air heat pump compared to the reference system.

Table 8.17. The input values used for the Italian winter “Ventilation Only” model.

ITALIAN WINTER “VENTILATION ONLY”		
	Desiccant-wheel cycle	Reference system
Supplied Temperature	22°C	22°C
Room air Temperature	22°C	22°C
Room Area	120 m^2	120 m^2
Outdoor Airflow	48 l/s	240 l/s
Outdoor air temperature and humidity	Shown in table 4.9	Shown in table 4.9
Recirculation Airflow	192 l/s	0 l/s
Supplied Airflow	240 l/s	240 l/s
Total sensible load considered	0 kW	0 kW
Total latent load considered	0.42 kW	0.42 kW
Sensible load covered by the ventilation system	0 kW	0 kW
Latent load covered by the ventilation system	0.42 kW	0.42 kW
Humidity control	NO	NO

Table 8.18. Devices used in ventilation systems simulated by the Italian winter "Ventilation Only" model.

ITALIAN WINTER "VENTILATION ONLY"	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Post hot coil	Hot coil
Silica gel wheel	
Evaporator	

Table 8.19. Energy consumption obtained with the Italian winter "Ventilation Only" model.

	CLEAN-AIR HEAT PUMP			REFERENCE SYSTEM	
	Compressor	Post hot coil	Fan	Hot coil	Fan
Energy consumption per season [kWh/season]	747	9	462	3045	550
Area [m^2]	90				
Energy consumption per m^2 [$kWh/m^2/season$]	6.22	0.07	3.85	25.37	4.58

Table 8.20. Cost reduction for the Italian winter "Ventilation Only" model.

	Open - Cycle Desiccant Wheel System	Traditional Air Conditioning System	Cost Reduction
Domestic price	2.26 €/season/m^2	2.78 €/season/m^2	**18.54%**
Industrial price	1.81 €/season/m^2	1.68 €/season/m^2	**-7.32%**

8.2.2 Summer application

Since the climate in Italy is quite humid, the ventilation system must provide dehumidification to control the indoor humidity to be too high. Thus the calculation for the Italian summer application was only made for the comparison between the clean air heat pump ventilation system and reference ventilation system with humidity control.

The input values assumed in the calculation and the devices used to process the air in both the clean-air heat pump and the reference system were the same as used in the Danish summer application **(calculation using the reference ventilation system with humidity control model)** except for the input values of outdoor air temperature and humidity as shown in Table 8.21. Table 8.23 shows the energy consumptions of the clean air heat pump and reference system; Table 8.24 shows the calculated cost reduction using the clean air heat pump compared to the reference system.

Table 8.21. Main values used for the Italian summer "Ventilation + Cooling - reference system with humidity control" model.

ITALIAN SUMMER "VENTILATION + COOLING - REFERENCE SYSTEM WITH HUMIDITY CONTROL"		
	Desiccant-wheel cycle	**Reference system**
Supplied Temperature	17 °C	17 °C
Inside Temperature	25 °C	25 °C
Building Area	90 m^2	90 m^2
Outdoor Airflow	36 l/s	180 l/s
Outdoor air temperature and humidity	Shown in table 4.8	Shown in table 4.8
Recirculation Airflow	144 l/s	0 l/s
Supplied Airflow	180 l/s	180 l/s
Total sensible cooling demand assumed	1.70 kW	1.70 kW
Total latent load assumed	0.32 kW	0.32 kW
Sensible cooling load covered by the ventilation system	1.70 kW	1.70 kW
Latent load covered by the ventilation system	0.32 kW	0.32 kW
Humidity control	YES	YES

Table 8.22. Devices used in systems simulated with the Italian summer "Ventilation + Cooling - reference system with humidity control” model.

ITALIAN SUMMER “VENTILATION + COOLING - REFERENCE SYSTEM WITH HUMIDITY CONTROL”	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Silica gel wheel	Cold coil
Evaporator	Hot Coil

Table 8.23. Energy consumption obtained with the Italian summer "Ventilation + Cooling” model.

	CLEAN-AIR HEAT PUMP		REFERENCE SYSTEM		
	Compressor	Fan	Cold coil	Hot coil	Fan
Energy consumption per season [kWh/season]	724	220	1819	1614	252
Area [m^2]	90				
Energy consumption per m^2 [kWh/m^2/season]	8.04	2.45	20.21	17.93	2.80

Table 8.24. Cost reduction for the Italian summer "Ventilation + Cooling" model.

	Open - Cycle Desiccant Wheel System	Traditional Air Conditioning System	Cost Reduction
Domestic price	2.35 €/season/m^2	6.39 €/season/m^2	**63.24%**
Industrial price	1.88 €/season/m^2	4.73 €/season/m^2	**60.29%**

8.3 Application in Sri Lanka

Since the climate in Sri Lanka is always hot, only the summer application was simulated. The calculations were performed by two models:

a) The clean air heat pump supplies isothermal airflow to the ventilated room (denoted as "ventilation only"). In this model, the clean air heat pump mainly used to clean the indoor air and cool down the supplied outdoor air for ventilation.
b) The clean air heat pump supplies airflow with the temperature lower than the room temperature (denoted as "ventilation + cooling"). In this model, the clean air heat pump provides cooling to the ventilated room in addition to the "ventilation only" function.

Calculation using ventilation only model

In the "Ventilation Only" case, the evaporator load is low because of the high supply air temperature (25°C). Moreover the high regeneration temperatures require that the heat pump contributed high condensing heat. In some conditions, it was not economy to use the heat pump to provide the entire regenerating heat and, for this reason, a post electrical heater was proposed after the condenser of the heat pump. In this operation mode, the heat load in the room has to be removed by a separated cooling system when using the clean air heat pump ventilation system.

The calculations are summarized in Table 8.25 to 8.28. Table 8.25 shows the input values assumed in the calculation; Table 8.26 shows devices used to process the air in both the clean-air heat pump and the reference system; Table 8.27 shows their energy consumptions of clean air heat pump and reference system; Table 8.28 shows the calculated energy reduction using the clean air heat pump compared to the reference system.

Table 8.25. Main values used for the Sri Lankan summer “Ventilation Only” model.

SRI LANKA SUMMER “VENTILATION ONLY”		
	Desiccant-wheel cycle	**Reference system**
Supplied Temperature	25 °C	17 °C
Inside Temperature	25 °C	25 °C
Building Area	90 m^2	90 m^2
Outdoor Airflow	36 l/s	180 l/s
Outdoor air temperature and humidity	Shown in table 4.10	Shown in table 4.10
Recirculation Airflow	144 l/s	0 l/s
Supplied Airflow	180 l/s	180 l/s
Total sensible load considered	1.70 kW	1.70 kW
Total latent load considered	0.32 kW	0.32 kW
Sensible load covered with the system	0 kW	1.70 kW
Latent load covered with the system	0.32 kW	0.32 kW
Sensible load covered with other systems	Fancoils system 1.70 kW	NO
Humidity control	YES	YES

Table 8.26. Devices used in systems simulated with the Sri Lankan summer "Ventilation Only” model.

SRI LANKA SUMMER “VENTILATION ONLY”	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Post hot coil	Hot coil
Silica gel wheel	Cold coil
Evaporator	

Table 8.27. Energy consumption obtained with the Sri Lankan summer "Ventilation Only" model.

	CLEAN-AIR HEAT PUMP				REFERENCE SYSTEM		
	Post electrical heater	Compressor	Fan	Fancoil system	Electrical heater	Hot coil	Fan
Energy consumption per season [kWh/season]	323	2867	625	3257	4192	13865	252
Area [m^2]	90						
Energy consumption per m^2 [kWh/m^2/season]	3.6	31.9	36.2	6.9	46.6	154.1	8.9

In this case, for both systems, it has been assumed to use electricity to heat up the air. Since no gas is used, it is possible to compare the two systems using the total energy consumption. Table 8.28 shows the energy reduction of the clean air heat pump ventilation system compared to the reference ventilation system.

Table 8.28. Energy reduction for the Sri Lankan summer " Ventilation Only" model.

	Open - Cycle Desiccant Wheel System	Traditional Air Conditioning System	Energy Reduction
Energy consumption	78.58 kWh/m^2/year	209.50 kWh/m^2/year	**62.49%**

Calculation using ventilation + cooling model

In this mode the clean air heat pump ventilation system provided cooling to balance the internal heat load. The input values assumed in the calculation and the devices used to process the air in both the clean-air heat pump and the reference system were the same as used in the Italian summer application **(calculation using the reference ventilation system with humidity control model)** except for the input values of outdoor air temperature and humidity as shown in Table 8.29. Table 8.31 shows the energy consumptions of clean air heat pump and reference system; Table 8.32 shows the calculated energy reduction using the clean air heat pump compared to the reference system.

Table 8.29. Main values used for the Sri Lankan summer "Ventilation + Cooling - reference system with humidity control" model.

SRI LANKAN SUMMER "VENTILATION + COOLING - REFERENCE SYSTEM WITH HUMIDITY CONTROL"		
	Desiccant-wheel cycle	**Reference system**
Supplied Temperature	17 °C	17 °C
Inside Temperature	25 °C	25 °C
Building Area	90 m^2	90 m^2
Outdoor Airflow	36 l/s	180 l/s
Outdoor air temperature and humidity	Shown in table 4.10	Shown in table 4.10
Recirculation Airflow	144 l/s	0 l/s
Supplied Airflow	180 l/s	180 l/s
Total sensible cooling demand assumed	1.70 kW	1.70 kW
Total latent load assumed	0.32 kW	0.32 kW
Sensible cooling load covered by the ventilation system	1.70 kW	1.70 kW
Latent load covered by the ventilation system	0.32 kW	0.32 kW
Humidity control	YES	YES

Table 8.30. Devices used in systems simulated with the Sri Lankan summer "Ventilation + Cooling - reference system with humidity control" model.

SEI LANKAN SUMMER "VENTILATION + COOLING - REFERENCE SYSTEM WITH HUMIDITY CONTROL"	
Desiccant-wheel system	**Reference system**
Condenser	Heat recover
Silica gel wheel	Cold coil
Evaporator	Hot Coil

Table 8.31. Energy consumption obtained with the Sri Lankan summer "Ventilation + Cooling" model.

	CLEAN-AIR HEAT PUMP		REFERENCE SYSTEM		
	Compressor	Fan	Cold coil	Hot coil	Fan
Energy consumption per season [kWh/season]	7280	1100	13865	4192	798
Area [m^2]	90				
Energy consumption per m^2 [kWh/m^2/season]	80.9	12.2	154.1	46.6	8.9

Table 8.32. Energy reduction for the Sri Lankan summer "Ventilation + Cooling" model.

	Open - Cycle Desiccant Wheel System	Traditional Air Conditioning System	Energy Reduction
Energy consumption	93.11 kWh/m^2/year	209.50 kWh/m^2/year	**55.56%**

Note: both the clean air heat pump ventilation system and the reference ventilation system use electricity. The energy consumption can be compared directly.