Bjarne Saxhofs fund project

DEVELOPMENT OF NEW TOTAL HEAT RECOVERY VENTILATION TECHNOLOGY FOR ENERGY CONSERVATION IN BUILDINGS

Final Report

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Contents

Executive summary

The existing energy recovery equipment includes heat and/or mass transfer technology. The energy recovery equipment using heat transfer technology that exchanges heat between incoming outdoor air and exhaust indoor air can be made in a way that the inbound and outbound airflows are completely separated without mass transfer (e.g. using plate heat exchanger). The driving force for the heat transfer is the temperature difference between inbound and outbound air. Such energy recovery technology is mainly suitable to be applied in winter season when the temperature difference between indoor and outdoor is high and the energy recovery is mainly on the sensible heat of the exhaust air. In summer season, however, the temperature difference between indoor and outdoor is not very high and most part of the energy used for ventilation is not to decrease the temperature of incoming outdoor air but to remove moisture from the humid outdoor air for ventilation. In this case, the efficiency of energy recovery equipment using heat transfer technology is very low.

The existing total heat recovery technology usually involves in both heat and mass transfer between the inbound and outbound airflows. The mass transfer between the incoming outdoor airflow and the exhaust airflow normally include the transfer of both moisture and other chemical compositions (e.g. volatile organic compounds (VOCs) which are the major indoor air pollutants). Thus, this type of total heat recovery equipment has a fatal weakness that is transfers the pollutants from the exhaust air back to the room when recovering total heat. To avoid such pollutant transfer between the incoming ventilation air and the exhaust indoor air, it is essential to cut off their passage of mass transfer and leave the heat transfer only. One solution is to remove moisture from the incoming outdoor air by condensation instead of moisture transfer between the two airflows. This requires that the temperature of the exhaust air is low enough to extract the moisture in the incoming outdoor air to be condensed. However, in many cases, the temperature of the exhausted indoor air is too high to extract enough condensation out from the incoming outdoor airflow when the two airflows exchange heat. On the other hand, in order to balance the condensing heat released from the incoming outdoor air, the exhaust air must be able to absorb this condensing heat which is impossible to be achieved by the sensible heat gain of the exhaust air from indoor air temperature to outdoor air temperature. Even if the exhaust air is cooled down to its wet bulb temperature by evaporative cooling, the sensible heat gain of the exhaust air can only balance a small fraction of the condensing heat of the incoming outdoor air.

This study explored a new approach of heat exchange. The new approach over saturates the exhaust air into fog region and exchange heat with the incoming outdoor air. The fog in the exhaust air evaporates instantly inside the heat exchanger to absorb the condensing heat released from the incoming outdoor air. This successfully solved the problem of heat balance in the heat exchanger to decrease the total heat of the incoming outdoor air.

To prove the effectiveness of this approach, a prototype unit of such a heat recovery unit was developed in this project. The prototype unit included a counter flow sensible heat exchanger and a specially designed ultrasonic atomizer. The ultrasonic atomizer over saturates the exhaust air into fog region with very fine water droplets which can evaporate instantly. This evaporating heat well balanced the condensing heat form the incoming outdoor air and increased significantly the total heat recovery.

The experiments were conducted in climate chambers simulating different outdoor climates to evaluate the heat recovery effectiveness of the novel technology. The outdoor climates used in the experiments include warm and humid climate (e.g. climate in Shanghai), mild warm and dry climate (e.g. climate in Copenhagen) and hot and dry climate (e.g. climate in Las Vegas). The results of the experiments show excellent performance of the newly developed heat recovery technology. 70% enthalpy (total heat recovery) efficiency in warm and humid climate was observed which is the highest among all the total heat recovery technologies available in the market. Over 200% temperature (sensible heat recovery) efficiency was observed in mild warm and dry climate which means this technology can also be used for cooling in a ventilation system. For example, this heat recovery technique can be used to cool the outdoor air supply from 30 to 18 °C which is usful for air-conditioning with very low energy consumption.

The application of this technology is very much rely on the quality of water used to generate the fog in the exhaust air. The evaporation of the fog inside the heat exchanger may leave water scale inside the heat exchanger and decrease the efficiency of the heat transfer. Suitable water demineralization technology is required in the region where hard water is used to generate the fog. However, the water consumption in warm and humid region is very low because the condensed water from incoming outdoor air can be collected and used for the evaporative cooling.

After the experiments of this study, following conclusions may be drawn.

- In warm and dry climate, the temperature efficiency of this novel technology can reach 250 %; in warm and humid climate where dehumidification is need, the total heat recovery efficiency can reach 70%.
- The supply air temperature after heat recovery can be as low as 18.5 ℃ in this study under various climate conditions.
- Atomizing technology and water quality is important for the application of this total heat recovery technique in practice.

1. Introduction

Energy saving is part of the key issues, not only from the viewpoint of fuel consumption but also for the protection of global environment. It was pointed out by Perez-Lombard et al. that building energy consumption has increased with economic growth, building sectors expansion and spreading of heat, ventilation and air conditioning (HVAC) system. Large amount of energy is lost due to heating, air-conditioning and ventilation.

Energy has been defined as "that which makes things go". There is accounted for 1/3 of the total energy use in air conditioning in modern society.

On the other hand, with the improvement of living quality, people have paid growing attention on indoor air quality in buildings. The importance of energy efficiency in buildings has been realized in this society art beginning of 1970s, after that building began to develop in the direction of energy saving. In order to decrease heat loss from buildings, the maintenance structure became even closely by input airflow from doors and windows greatly reduced. Thus, due to the reduction of the air change rate in ventilation, the indoor air quality becomes poorer. Since the 20th century, people increasing their indoor time up to 90% of the day, especially the elderly and children. Sick Building Syndrome (SBS) caused by poor indoor air quality greatly affected physical and mental health of people, also the work efficiency. So that, the indoor air quality (IAQ) was becoming a big issue which was worried about nowadays society.

One of the promising methods is using heat recovery technique, which consists of total heat recovery. Especially in humidity climates, total heat recovery has much higher efficiency compared with the sensible heat recovery. Heat and moisture recovery, or the so called total heat exchangers, could save a large fraction of energy that is used for cooling and dehumidifying the fresh air during the summer period. In fact, it is found that 70-90% of the energy for fresh air treatment can be saved. The total heat recovery equipment can be classified into two categories: energy wheels and stationary total heat exchangers. If defined the technical in detail, there are various technologies to realize heat and moisture recovery: heat pipes, energy wheels, liquid descant systems, membrane based total heat exchangers and so on. [1]

In the design of air-conditioning, it is necessary to ensure enough inlet fresh air in order to meet the requirements of envelope indoor air quality. The overall design of ventilation system should focus on removing air pollution of the building. Standards of fresh air quantity have been constantly emphasis the importance of indoor air quality and requirements also paid attention on the tightness of building itself. To this end, "ASHRAE Standard 62 - 1989" had proposed two improvements: ① Increased 2-4 times of the original design of fresh air quantity; ②Maintained building air relative humidity at 30% -60 %. After discussion and research, "ASHRAE Standard 62-1999" has become more mature basis of the minimum fresh air in conditioning system, reasonably. In addition, the demand of fresh air has become even higher in some special occasions, such as hospitals, shopping malls, theaters, stadiums, conference rooms and other crowded places. In this case, the load of fresh air would be the major part of the overall system energy consumption. If the energy of the exhaust system can be utilized to pre-treated fresh air, while saving energy was undoubtedly a good thing.

The heat and mass transfer in novel total heat exchanger structures are continuously searched, both in cross-corrugated triangular ducts and in quasi-counter flow parallel plates.

1.1. Definition and concept

1.1.1. Heat recovery

Energy heat recovery in a ventilation system is a technology that transfers the energy (both heating or cooling) of the exhaust air to the supply air.

Technology of heat recovery is the energy recycle process of exchanging the energy in currently building or space air and using it to precondition the inlet fresh air which come from outdoors. During the summer conditions, the advantage of using energy recovery is the ability to reach the ASHRAE ventilation & energy standards, while improving indoor air quality.

Heat recovery technology, as expected, has proved the actual meaning of reducing energy cost and also allows for the indoor environment to maintain a relative humidity of a range of 40% to 50%. The only energy consumption is the power for the pressure drop of fan in the system.

With the variety of products on the market, efficiency is unquestionably going to the key of the problem by each kind of equipment. Some of these systems were known that the sensible heat exchange efficiency was higher as 70-80% while others was lower as 50%. [2]

Heat recovery system is an air-to-air heat or energy recovery system which recovers energy from a stream at a high temperature to a low temperature stream efficiently. Heat recovery systems are typically used for recovering the energy loss fraction. With the use of the recovery system, the energy is used to heat the incoming air instead of being lost. Actually, heat recovery systems typically recover about 60-95% of the heat in the exhaust air and improve the energy efficiency of buildings significantly.

1.1.2. Total heat recovery

The total heat recovery technology is a type of air-to-air heat exchanger that not only transfers sensible heat but also latent heat (refers to moisture transfer). This type of heat recovery devices is also called enthalpy recovery device.

1.1.3. Evaporative cooling

Evaporative cooling is the addition of water vapor into air, which causes significant lowering of the temperature of the air. The energy needed to evaporate the water is taken from the air in the form of sensible heat, which affects the temperature of the air, and converted into latent heat, the energy present in the water vapor component of the air, whilst the air remains at a constant enthalpy value. This conversion of sensible heat to latent heat is called an adiabatic process because it occurs at a constant enthalpy value. Evaporative cooling therefore causes a drop in the temperature of air proportional to the sensible heat drop and an increase in humidity proportional to the latent heat gain. [3]

In the return air side and before the sensible heat exchanger, a rigid media air cooler is utilized to cool the air stream. This particular type of evaporative cooler consists of rigid materials, which from the wetted surface. Moist air flows through the corrugations. Water enters the top of the evaporative cooler and flows by gravity through the wetted surface.

1.1.4. Indirect evaporative cooling

With indirect evaporative cooling, a secondary (scavenger) air stream is cooled by water. The cooled secondary air stream passes through a heat exchanger, where it cools the primary air stream. The cooled primary air stream is distributed to a blower.

Evaporative cooling is economical, effective, environmentally friendly, and healthy. Evaporative cooling is economical because of reduced chilled water cooling requirements for fresh air and increases existing equipment cooling capacities without adding mechanical cooling.

This technology can be used in all climates, not just in hot, dry climates. Because evaporative cooling does not use chlorofluorocarbons (CFCs), it is not conducive to ozone depletion. It is salubrious and comfortable because it can bring in outside air and exhaust stale air, smoke, odors, and germs without cross- contamination. [4]

1.2. Background of heat recovery technology

1.2.1. Liquid desiccant total heat exchanger

Fresh air has a prominent advantage compared with return air in discharging the indoor CO2 and VOC and especially diluting possible zyme or mildew, but increasing fresh air flow rate is restricted by the rapidly increasing energy consumption.

As one of the solutions to the contradiction between improving the indoor air quality and reducing the energy consumption for handling the fresh air, a new system for total heat recovery with liquid desiccant is newly being proposed. The main benefit of the liquid desiccant total heat recovery system is to improve the indoor air quality and reduce the energy consumption for inlet fresh air at same time without cross contamination. Combined with the fixed-plate heat exchanger, liquid desiccant total heat exchanger increases the capacity for humidification and dehumidification by adjusting the temperature of the desiccant.

Lithium bromide was used for general reagents in liquid desiccant technology. Lithium bromide had corrosion effect on the equipment. In addition, the lithium bromide solution would immerse into air during the processing. It would impact human health by breathing the air mixed with lithium bromide.

1.2.2. Heat pipe

A heat pipe is an inner wall with a layer which has a liquid-absorbent core sealed tube. The heat pipe heat exchanger is essentially a sensible heat transfer device. The heat pipe has its roots in the nuclear industry. To be the passive device, there are two main types of heat pipe – vertical and horizontal. A working fluid is employed to affect the heat transfer. The fluid was chosen to suit the requirement of temperature range and would typically be one of the common refrigerants. The heat exchanger is composed of rows of conductors. In operation, heat applied in the evaporation section (exhaust air duct) will cause the fluid to evaporate and travel to the condensation section (supply air duct) of the pipe. As the vapor condenses the heat is transferred and the vapor returns to a fluid, thus filling out the cycle. Fresh inlet air is not contact with the exhaust air directly, so that there is no pollution issue in heat pipe technique.

The results of studies, fully proved the heat pipe technology for air conditioning systems are feasible not only analysis from technical point of view or economic aspect. Considering from the standpoint of heat transfer efficiency, the sensible heat transfer efficiency can reach 45% to 65% by this technology. [5]

The heat recovery efficiency could reach 45% to 65%. In the existing technology, it was not a high-efficiency of sensible recovery.

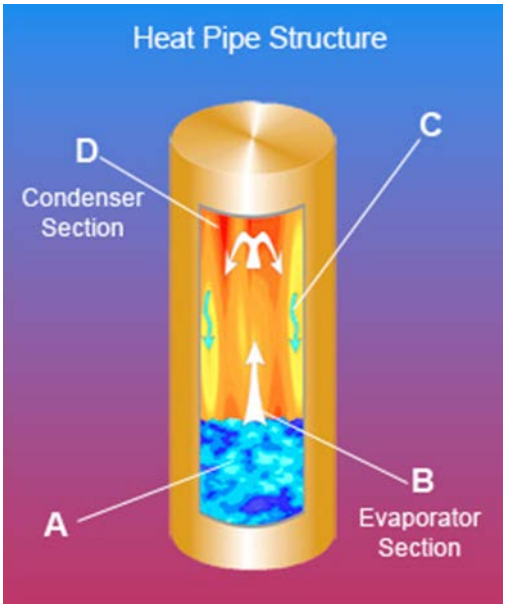


Fig.1.2.2 (1) Heat pipe recovery structure.

1.2.3. Papery finned plate heat exchanger

The rotor of the papery finned plate heat exchanger is pasted by corrugated paper and plate paper alternately. Moisture and the heat are moved by infiltration. The fresh air and the static pressure in exhaust ducts are highly required due to the use of the particular paper.

In the 1980s, many companies of Japan have done on a paper plate heat exchanger research and development. In the latter time, various products were marketed all over the world. In Europe and the United States, for large centralized air-conditioning system, paper heat exchanger had been extensively used for energy recovery.

With the higher pressure in the gap of paper fiber, the larger amount of leakage occurs which leads to reduced heat recovery efficiency. Worse still, this system is not available for devices with large airflow due to the limit of paper intensity and the design of the rotor.

1.2.4. Membrane total heat recovery

Membrane-based total heat exchanger has been fundamental equipment for heat and moisture recovery. The structure of the device is like a general plate heat exchanger with a metal foil. However, several unwanted impermeable gases such as CO2 are invoked as the plates.

A membrane should be a barrier to other unwanted gases like VOCs. In the laboratory, the most commonly used membranes, either was hydrophilic ones (PVP, PVA, PAM, Na (Alg), CS, CA, EC) or hydrophobic ones (PP and PDMS). The total heat exchanger is ideal to control indoor air quality, in addition to energy recovery.

The membrane technique is also extended to other directions. For example: air humidification and air dehumidification. It should be pointed out that indoor environmental control is crucial for fresh air ventilation. Membranes have been applied to total heat exchangers for a dozen of years. Numerous concerns have been spent on water vapor permeability of various membranes, however relatively less attention has been paid to the selectivity of moisture over VOCs through such membranes. Membrane total heat exchangers have being claimed to have the following advantages:

(1)It has no cross-over problems.

(2)It is stand still and has no moving parts.

(3)It is simple.

(4)Its performance deteriorates little over time.

(5)It is compact and high efficient.

(6) Both the sensible and the latent effectiveness can be as high as 90%.

However, recent study found that the effectiveness sensible and total heat recovery in a commercially available membrane total heat recovery unit was not was not as good as it is claimed (Fang et al. 2003). Another study found that the contaminants cross-over was as high as 8%.

1.2.5. Rotary wheel

Rotary wheel recovery consists of a rotor with permeable storage mass fitted in a casing which operates intermittently between a hot and cold fluid. With the heat and humidification absorption on the motor, the heat storage of the motor and the heat recovery from the exhaust during the moisture absorption, the recovered energy is in a position to transfer to the fresh air with high efficiency. Rotary wheel is the most widely used total heat recovery technology at present. However, cross contamination cannot be avoided due to the air leakage caused by the pressure difference between inlet and outlet air.

The rotor is driven by a motor with relatively low speed at 10 rpm and the exhaust air and fresh air is alternately passed through each section. On outlet and inlet air alternately passing through reversely, the heat is released to the rotor.

In the rotary wheel total heat recovery system, the rotary wheel also transfers the moisture as well as sensible energy between streams. The rotor dumps the excess moisture within the fresh air supply back to the atmosphere. In summer, the rotary wheel total heat recovery system is able to pre-cool and dehumidify the inlet fresh air while in the winter it can pre-heat and humidity the fresh air. The typical efficiency amounts to 70% as a result of reducing the energy consumption.

There are still several disadvantages of rotary wheel technology. For example, it adds to the first cost and the fan power to overcome its resistance, requires that the two air streams be adjacent to each other. In addition, it also requires that the air streams must be relatively clean and may require filtration. The rotating mechanism that requires it be periodically inspected and maintained, as does the cleaning of the fill medium and any filtering of air streams, in cold climates, there may an increase in service needs, results in some cross-contamination (mixing) of the two air streams, which occurs by carryover and leakage. [7]

Heat and moisture transfer in energy wheels during sorption, condensation, and frosting condition. [8] Rotary heat exchangers that transfer sensible energy have been used for many years in gas turbine plants to recover thermal energy from the exhaust gases, thereby increasing the overall plant thermal efficiency (Harper and Rohsenow, 1953, Shah,1981).

One problem that restricts the practical application of energy wheels is condensation and frosting. Energy wheels are often selected and operated to prevent condensation or frosting within the wheel. However, during hot and humid operating conditions, water may condense in the energy wheel, saturate the desiccant and finally run off.

A similar problem exists during cold weather operation where frost may build up in the energy wheel, restrict the air flow and reduce the effectiveness of the wheel.

The rotating wheel heat exchanger is consisted of a rotating cylinder filled with an air permeable material resulting in a larger surface area. This area is the transmitter for the sensible heat recovery. The wheel rotates picks up heat energy and drops it into the colder air stream by rotation of the wheel. Typical media used made up of polymer, aluminum, or synthetic fiber.

The enthalpy exchange is accomplished through the use of desiccants in rotary wheel technology. Transport of moisture through the desiccant adsorption process is mainly tempted by the difference in the regional pressure of vapor within the reverse air flow.

Enthalpy wheels are effective devices to recovery both latent and sensible heat, but there are many different types of construction. The most normal type of this equipment is established of the polymer and can be endured with a high pressure drop and shorter life. Other material of the wheel has a longer using life with much lower pressure drop capability of adaption. Rotary heat exchanger due to the different materials of runner regenerator can be divided into four types:

(1) ET Type: It makes of corrosion-resistant aluminum foils which coated with hygroscopic coating. This type of rotary heat exchanger has excellent moisture absorption and also can recovering sensible heat and latent heat simultaneously. The total heat recovery efficiency can reach to 70% to 90%.

(2) RT type: It makes of pure aluminum foil with no moisture absorption and mainly recovery of sensible heat.

(3) PT Type: It makes of corrosion-resistant aluminum foil with high temperature capability and only recovery sensible heat.

(4) KT type: It makes of corrosion-resistant aluminum foil and coats with a plastic layer. There is a strong resistance to corrosion, mainly recovery of sensible heat.

There is inevitably penetration through the cracks which are between the inlet fresh air and exhaust air in a rotary system, leading to air cross contamination. It can be explained from two aspects: First all, there are some gaps between the areas of inlet and outlet air, the occurrence of pressure difference can cause infiltration; on the other hand, the air remained in the honeycomb core space of rotation will stream from one side to another, which is the other reason of penetration. Even though rotary wheel has remarkable ability in the field of total heat recovery, also, there should be special considerations on colder climates to prevent wheel frosting. Systems can avoid frosting by modulating wheel speed, preheating the air, or stop/jogging the system. Anyway, it will reduce the total heat recovery efficiency or waste much energy on overcoming the additional pressure drop. Cross-contamination of the pollutions via the desiccant is also a pivotal problem which has been studied further.

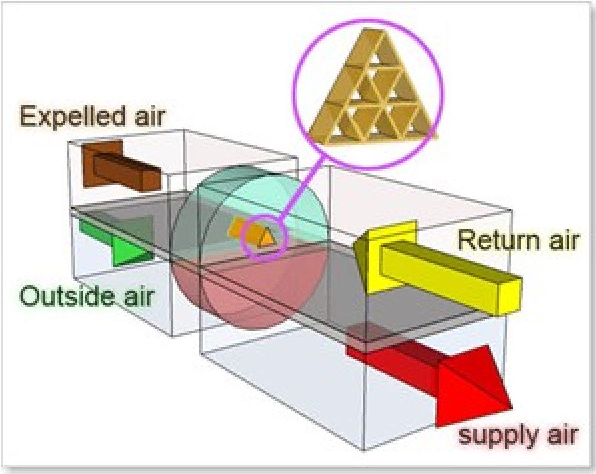


Fig.1.2.5 (1): Rotary wheel heat recovery structure.

1.3. Principle of the new total heat recovery technology

To reduce energy consumption for ventilation, heat recovery technologies have been widely paid strict attention. Recovery energy from exhaust air to inlet fresh air was used more and further in ventilation nowadays. Sensible heat recovery efficiency of ordinary plate heat exchanger could reach 90%. In the hot and humid areas, the main load of a ventilation system is discussed by dehumidification (the latent load) which occupied for more than 70% of the total energy. This means more than 70% of the total heat from the outlet air was wasted by a sensible heat exchanger (even though the temperature efficiency is 100%). Therefore, it is necessary to mention the importance of total heat recovery which meant not only sensible load has to be recycled but also the part of the latent load.

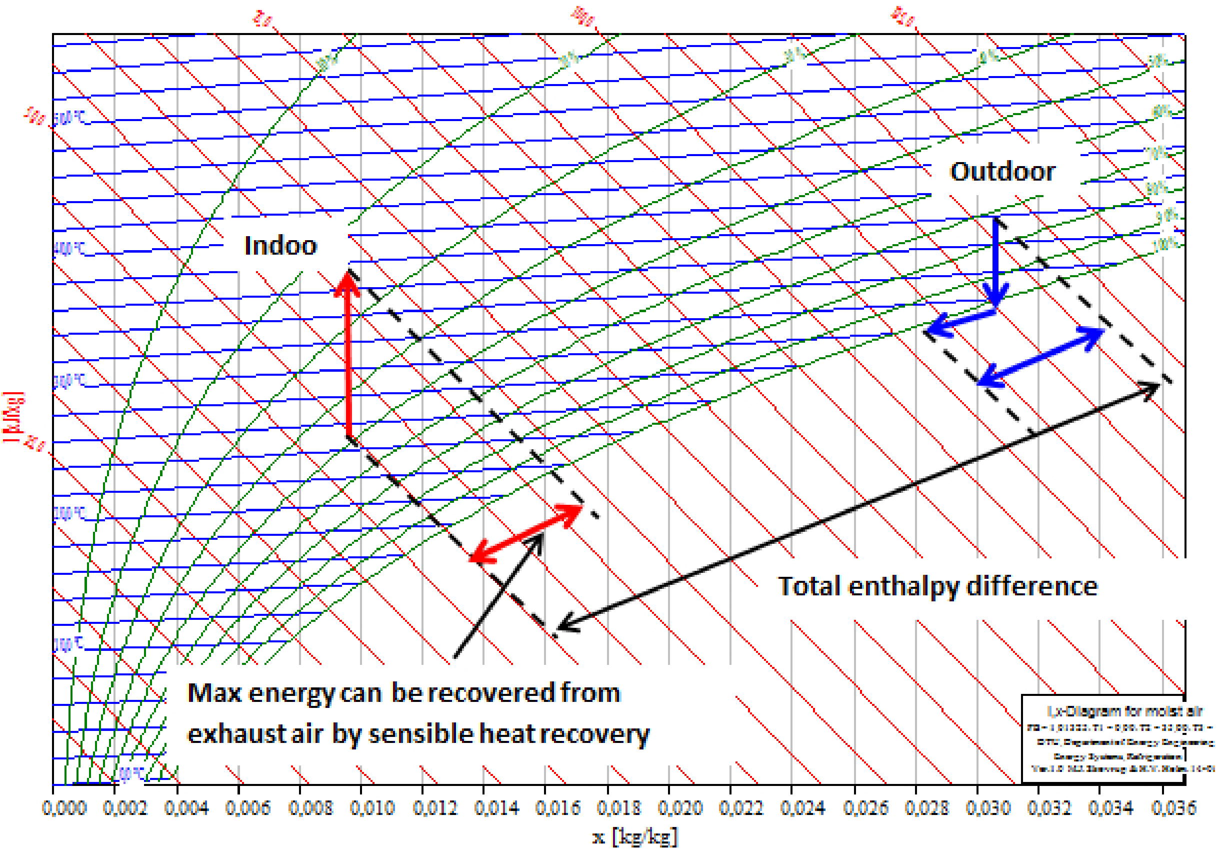


Fig 1.3 (1): Enthalpy difference between indoor and outdoor air and the max energy that can be recovered by a sensible heat exchanger in an example of a summer climate.

The existing energy recovery equipment includes heat and/or mass transfer technology. The energy recovery equipment using heat transfer technology that exchanges heat between incoming outdoor air and exhaust indoor air can be made in a way that the inbound and outbound airflows are completely separated without mass transfer (e.g. using plate heat exchanger). The driving force for the heat transfer is the temperature difference between inbound and outbound air. Such energy recovery technology is mainly suitable to be applied in winter season when the temperature difference between indoor and outdoor is high and the energy recovery is mainly on the sensible heat of the exhaust air. In summer season, however, the temperature difference between indoor and outdoor is not very high and most part of the energy used for ventilation is not to decrease the temperature of incoming outdoor air but to remove moisture from the humid outdoor air for ventilation. In this case, the efficiency of energy recovery equipment using heat transfer technology is very low. As an example, Figure 1.3 (1) shows how much heat can be recovered in a mild summer climate where outdoor condition is 38˚C/70%RH and indoor condition is 25˚C/50%RH. In this example, the max energy that can be recovered by heat transfer at the temperature difference between indoor and outdoor air is around 8kJ/kg (assuming 100% temperature efficiency of the heat recovery). Considering a realistic temperature efficiency of 80% for a good heat recovery unit, only 15% of the total heat for the ventilation is recovered. To recover more energy from the exhaust air, a total heat recovery equipment is required to recover the most part of the latent heat.

The purpose of this technology is to put in place an evaporative cooling device that can supersaturate the exhaust indoor air with moisture and to cool the air to its wet-bulb temperature before it exchanges heat with the incoming outdoor air. Since the cooling of the exhaust air is due to water evaporation, no additional energy is required for this adiabatic cooling process. The minimum air temperature of this cooling process is the wet-bulb temperature of the indoor air which is many degrees lower than its dry-bulb temperature. Evaporative cooling maximizes the temperature difference between the exhaust air and the incoming outdoor air (e.g. it can double the temperature difference as shows in Figure 1.3 (2)) and thus maximizes the heat recovery from the exhaust air. Meanwhile, the over saturated exhaust airflow will evaporate continuously inside the heat exchanger to further cool the ventilation airflow and extract moisture out of the airflow by condensation. Theoretically, it can recover 100% of the total enthalpy difference between indoor and outdoor air.

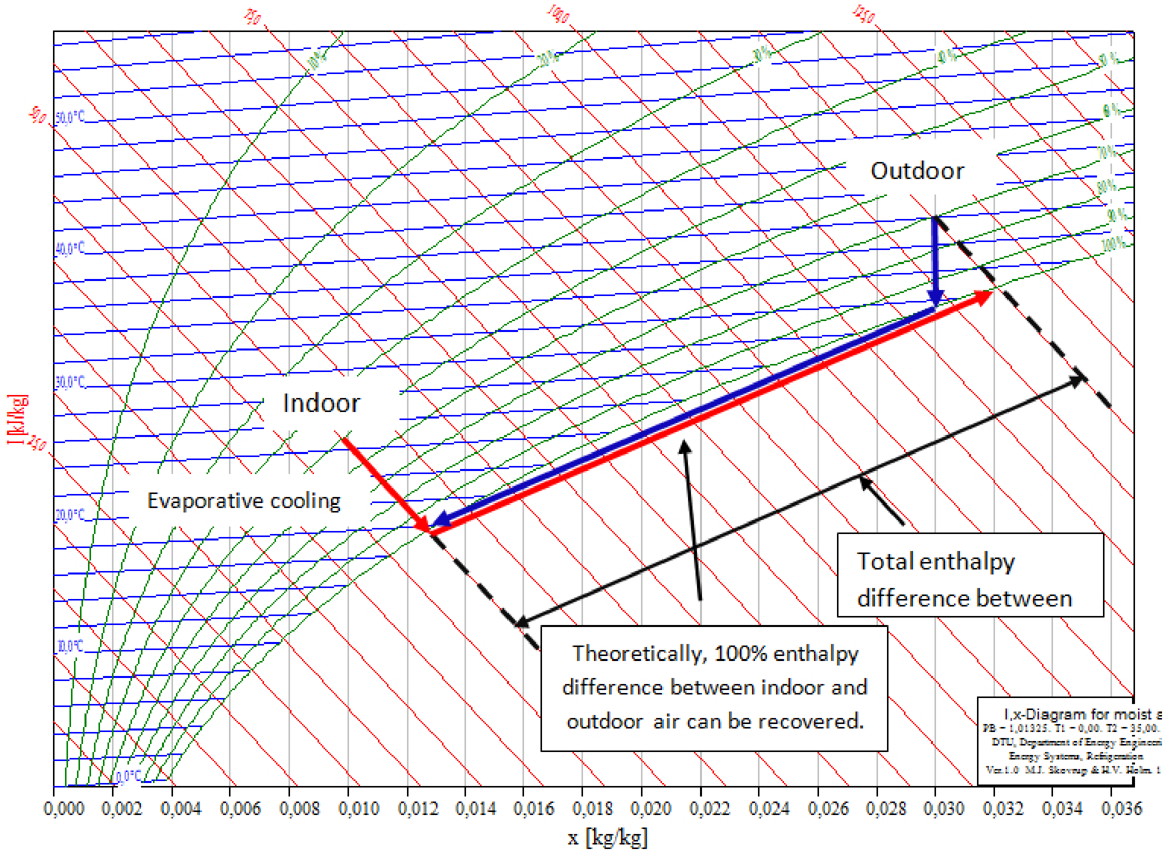


Fig 1.3 (2) Enthalpy difference between indoor and outdoor air and the max energy that can be recovered by the proposed total heat recover technology in an example of a summer climate.

The evaporative cooling device can be attached to an existing cross-flow or counter-flow heat exchanger (the heat exchanger without mass transfer) to enhance their enthalpy efficiency of heat recovery without a significant change in the existing design of the heat exchanger. It is expected that, with the help of the evaporative cooling device, the enthalpy efficiency of the existing heat recovery unit could be increased to 80% which mean that up 65% of the extra energy can be saved for ventilation in summer.

The evaporative cooling device comprises an ultrasonic water atomizer and an evaporation chamber. The ultrasonic water atomizer breaks water into very fine droplets (1-5 µm in diameter) which will vaporize rapidly in the evaporation chamber. The quick evaporation will cool the air temperature down to wet-bulb temperature and increases the temperature difference between the exhaust indoor air and incoming outdoor air by which it enhances the heat transfer of the heat recovery unit. Since the wet-bulb temperature of indoor air is less than the dew-point temperature of outdoor air, moisture in the incoming outdoor air for the ventilation will be condensed by which latent heat is removed from the ventilation airflow without mass transfer between the two airflows. The evaporative cooling device will be developed in the way that the air is over saturated in the evaporation chamber. Thus airflow leaving the evaporation chamber will carry large amount of fine water droplets when it enters the heat exchanger. The translucent water droplets carried by the airflow will evaporate continuously inside the heat exchanger and increase further the cooling effect.

In the following presentation of study, one manufactured total heat recovery unit was used as a reference unit and tested under laboratory conditions to evaluate its temperature, enthalpy efficiencies and other factors.

2. Method

2.1 Full-scale room

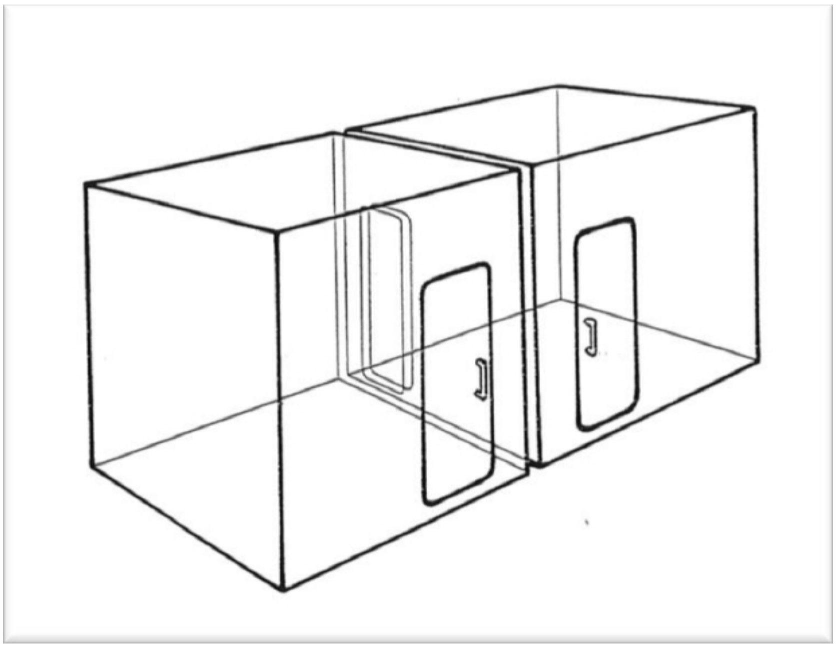


Fig 2.1 (1) Twin-chamber in ICIEE.

The experiment was performed at the International Centre for Indoor Environment and Energy at the Technical University of Denmark. The twin-chambers were a powerful tool for comparative studies of the effect of indoor climate on humans with regard to health and comfort.

Both chambers were approachable from a corridor getting through different doors. And chambers were connected by a stainless door which was used for crossing in a different space.

The diameter of the chamber was 3.6*2.5*2.5 m3. The material of walls and surface was stainless steel which could ensure minimum gas absorption and particulates on faces of the chamber. Walls were insulated well with transmission coefficient of 0.34W/m2 ºC. The floor was created by a perforated plate and a grid to walk on as showed in figure2.1 (2).The diameter of the ventilation holes situating on the plate was produced to achieve identical air flow rate from grids. Because of the gas tightness requirement, doors were sealed with silicon-rubber pipe. Other ducts and connection components were also designed for stainless material in order to keep cleaning and easier for maintenance.

These two chambers has own ventilation systems. Air change rate can be controlled from 0 to 60 air changes per hour. The air flow is supplied through a perforated floor which was similar to the displacement ventilation process. Exhaust air get out of the chamber by means of 4 openings located on the ceiling.

The air temperature in the chamber could be controlled from 5 ºC to 38 ºC with ±0.1ºC accuracy and humidity can be controlled between 20% RH and 90% RH with the accuracy of ±1% RH.

Data handling and processing system were used only for controlling the air temperature and humidity of the chambers. Temperature was measured by Pt 100 sensors and Ni 100 sensors. The steam nozzles which located in the duct closing to chamber were selected to be part of LICL- sensor measured humidity.

Fresh air was filtered by special filtering apparatus. After that, heating/cooling system would retreat by different units. Sometimes fasten electric heater would use for reheating the temperature of the air for particular demands.

In this project, twin-chamber was implemented to simulate disparate air condition. Chamber 1 was chosen for outdoor weather, and chamber 2 was used for modified indoor air condition during the experiment. The detail data of operational states would be recommended later.

Experimental installation was seated in chamber 2 which was using simulate air condition indoors. A hole was made on the door that between twin-chambers connecting a duct in order to transfer the moist air from chamber 1. The total heat recovery unit in this experiment was appeared as shown in figure2.2.1 (1).

2.2 Experiment system

2.2.1 Total heat recovery unit

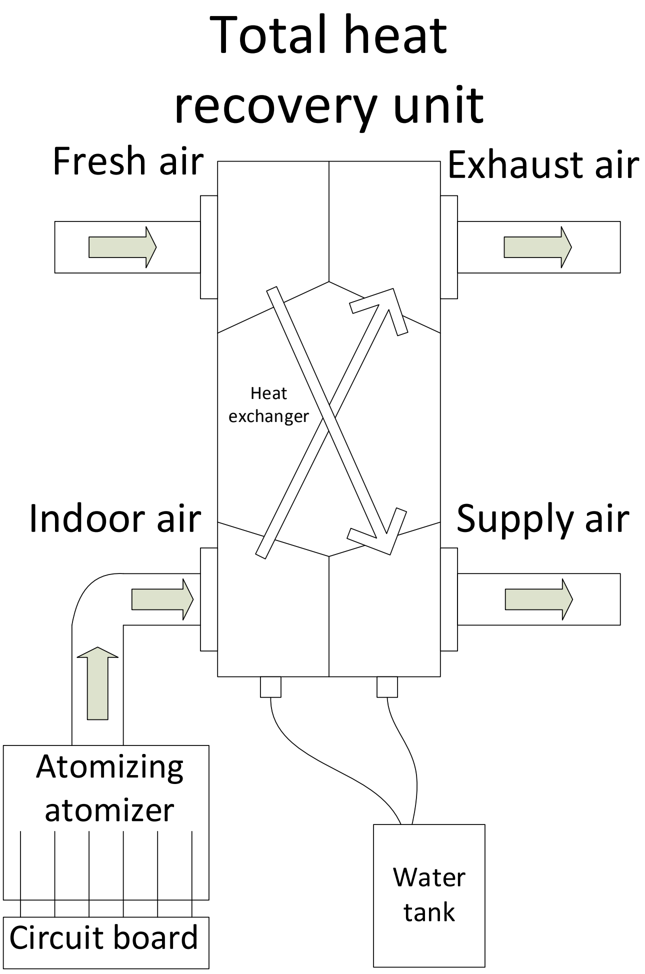


Fig 2.2.1 (1): Total heat recovery unit.

The total heat recovery unit was combined with several parts shown in fig 2.2.1 (1). As arrows given direction, fresh air inlet to the heat exchanger, after total heat transmission became supply air contributed to the room. On the other side, indoor return air is humidified by an ultrasonic atomizer. The air was over saturated by the atomizer before it went into heat exchanger. A water tank was connected to the two air channels of the heat exchangers to collect condensed water.

2.2.2 Layout of experiment table

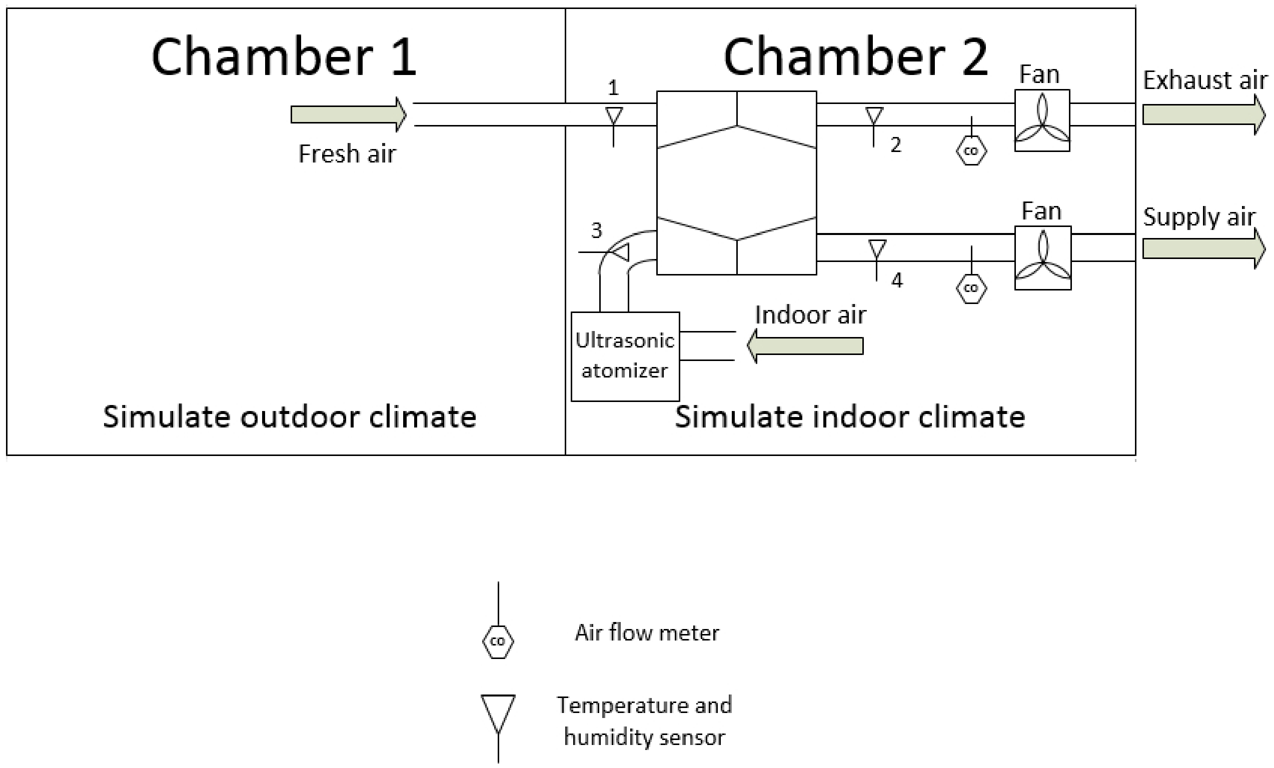


Fig2.2.2 (1) Total heat recovery experiment table.

The experimental setup is shown in Fig2.2.2(1). Chamber simulated outdoor climate and chamber 2 was used to simulate indoor climate. In chamber 2, a test rig was constructed to test the total heat recovery unit developed. The test includes the developed total heat recovery unit that connected the outdoor air (taken from chamber 1) and indoor air (taken directly from chamber 2) and both airflows were rejected through the exhaust in chamber 2. There were 4 temperature and humidity sensors located in the two inlets and the two outlet of the heat exchanger as shown the Fig 2.2.2(1). Two iris dampers were used to measure the air flows on both sides of the heat exchanger. A specially designed ultrasonic atomizer was used to moisturize and cool the exhaust indoor air that exchange heat in the heat exchanger to cool the outdoor air supply.

2.3 Calculation and formulas

The temperature, humidity, and enthalpy efficiencies were calculated using the following formulas.

Temperature efficiency:
$$\eta_{t,c} = \frac{t_{w,in} - t_{c,out}}{t_{w,in} - t_{c,in}}$$

Humidity efficiency:
$$\eta_{x,c} = \frac{X_{w,in} - X_{c,out}}{X_{w,in} - X_{c,in}}$$

Enthalpy efficiency:
$$\eta_{I,c} = \frac{I_{w,in} - I_{c,out}}{I_{w,in} - I_{c,in}}$$

where :

$\eta_{t,c}$	is the temperature efficiency;
$t_{w,in}$;$t_{c,in}$;$t_{w,in}$	are temperatures of supply, indoor and outdoor air;
$\eta_{x,c}$	is the humidity efficiency;
$X_{w,in}$;$X_{c,in}$;$X_{w,in}$	are humidity ratios of supply, indoor and outdoor air;
$\eta_{I,c}$	is the enthalpy efficiency;
$I_{w,in}$;$I_{c,in}$;$I_{w,in}$	are enthalpies of supply, indoor and outdoor air.

The enthalpies of air are calculated from the measured air temperature and humidity using the following formula.

$$E = 1.006t + 0.622(2501 + 1.84t) \times \frac{0.01\emptyset e^{\left(23.58 - \frac{4043}{t+273.15-37.58}\right)}}{P_0 - 0.01\emptyset e^{\left(23.58 - \frac{4043}{t+273.15-37.58}\right)}}$$

Where

E is enthalpy of the air, KJ/Kg;

t is temperature of the air, ºC;

$\emptyset$ is relative humidity of the air, %;

P_0 is barometric pressure of the air, Pa.

2.4 Equipment

Total heat recovery unit was constituted by a counter flow plate heat exchanger and an ultrasonic atomizer.

2.4.1 Ultrasonic atomizer

Fig 2.4.1 (1) Ultrasonic atomizer in this project.

Ultrasonic atomizer uses the vibration of ultrasonic high frequency resonator to break water into ultrafine droplets with the diameter of 1-5 microns in each. The ultrafine droplets can easily absorb heat from air and evaporate into water vapor. In this process, it cools the air down to its wet bulb temperature. The over saturated air is cooled to the wet bulb temperature and still carries a lot of ultrafine droplets. These droplets were further evaporated inside the heat exchanger when receiving heat transferred from the other side of the heat exchanger.

The feature of ultrasonic atomizer is introduced below:

(1) Humidification strength; wet uniform; high efficiency humidification; energy saving;
(2) Long life; humidity self-balancing; automatic protection of water.
(3) The disadvantage was that there were certain requirements of water quality.

2.4.2 Total heat recovery model

Plate heat recovery was the most common type of heat recovery equipment which is defined by the construction of its exchange. In this working unit, the surface of the plate heat exchanger is made of aluminum thin plates which are arranged together with different directions of internal airstreams. The plate was built with forms of corrugation. Plate heat exchanger provided high efficient heat recovery with the reason of its superior heat transfer coefficients, and counter-current flow types designing.

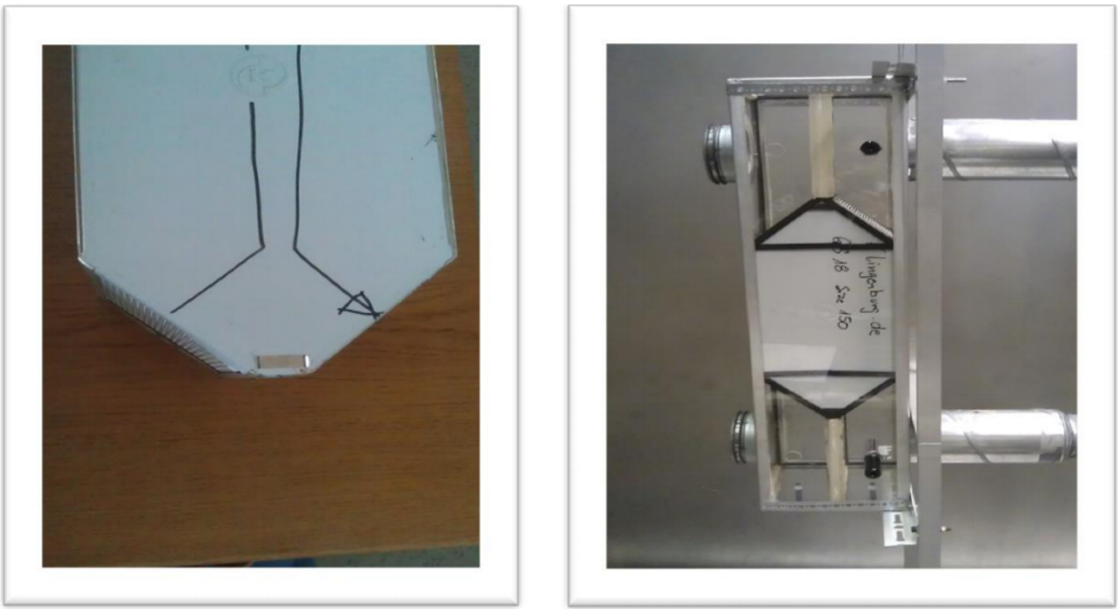


Fig 2.4.2 (1): Heat exchanger model

Counterflow plate heat exchanger type GS which produced by Klingenburg of Germany was selected for application in ventilation systems in this project.

It was designed as counterflow plate heat exchanger for recovery of sensible and latent heat set out in the air streams.

Exhaust air and supply air was totally separated. The air flows were through by each other along this parallel aluminium plates by the counterflow principle.It was the preferential function for any transfer of odours or humidity was excluded.

The material of the plates was non-corroding aluminium alloy. With the superior plate structures, 90% heat recovery efficiencies would be achieved readily.

It was also produced with unique edge-folding due to high tightness and stability.

High performances by simultaneous low pressure losses over the entire volumetric scope because of distribution of air flow through definite channeled guides.

Precision engineering with totally smooth outer surfaces was to ensure optimum sealing and perfect matching to the A/C systems.

Robust aluminium quality and sealing between plates was the cause of high level of impermeability.

The senior requirement of extremely hygienic would be reached through optimum drainage of all condensates.

Considering the requirement of pressure drop, GS18 which was minimum size in this series has been carried out with maximum 120m3/h air mass flow both in supply and exhaust.

In order to obtain optimal pressure tightness, a special outer-shell plate has been established by using of PMMA.(Figure2.4.2(1)) In addition, outer-shell plate was a convenient medium for experimental observation of special phenomena of mist at the inlet and outlet. The outer-shell plate was made of five sections, connecting with indoor air, outdoor air, return air, exhaust air and the heat recovery core part. As the figure shown, rubber strips were served as sealing material ensuring water proofing property under the pressure of the wind. Two small holes were created at the bottom of outer-shell plate which was used for discharging condensate timely.

2.4.3 VAISALA temperature and humidity transmitter

The HMP140A transmitters were reliable and easy to be used instruments for the measurement of relative humidity and temperature.

The transmitters could be ordered with a black cover with a local display which outputs relative humidity and temperature readings. The reading to be contained was chosen with a pushbutton on the cover and the desired temperature unit could be chosen with a jumper.

HMP140A transmitters measured relative humidity in the range of 0-100%RH and temperature from -40 to +60 degrees. The measurement was temperature compensated.

The durable plastic covered provides IP65 protection from dust and sprayed water. The HMP140A transmitters were therefore suitable for most indoor and outdoor applications, including those with high humidity like indoor swimming pools and so on. These versatile transmitters were also easy to set up and to use.

The HMP140A transmitters incorporated the HUMICAP180 sensor, the operation of which was based on the changes in its capacitance as the thin polymer film absorbs water molecules. The sensor was immune to most chemicals and was in an excellent long-term stability. The temperature was measured with a Pt 1000 sensors.

Fig 2.4.3 (1): VAISALA HMP140 temperature and humidity transmitter.

Platinum resistance temperature sensor was the point of metallic platinum resistance changes with temperature characteristics to measure the temperature.

Indicator was shown the resistance data of platinum resistance temperature corresponding value.

The measured temperature was equal to a range of temperature sensing element in the average temperature of the dielectric layer, when there was a temperature gradient in the measured medium.

Experiment was using four platinum resistance temperature detectors. Plate heat exchangers were installed in four entrances (labeled as "indoor inlet", "indoor outlet”, "outdoor inlet, "outdoor outlet”).

Four temperature detectors have before been corrected before use.

To ensure the tightness between temperature detectors and outer-shell plate, sealing tape has been chosen for connection in order to forbid the influence from outside air.

2.4.4 Fan

There are two fans installed on sides of exhaust air and supply air. The model number was K 150 XL which was produced by "Systemair" company in this experiment. The maximum airflow would be 702 M3/h with the weight of 4.2kg. The fan impeller speed was 2522 rpm and sound pressure level was 48.3 dB (A)

Fig2.4.4 (1): The fan (K150 XL) in this experiment.

2.4.5 Airflow meter

The airflow was measure by iris damper. The opening of the iris dampers were adjusted to have the pressure drop in the experimental conditions at 30 to 100 Pa. The airflow coefficient of the iris dampers were calibrated before the experiment to make sure that the airflow rates on both side of the heat exchanger were the same. MF-PD differential pressure transmitter was used to measure the pressure drop on the iris dampers during the experiment.

Fig 2.4.5 (1): PD pressure transmitter.

Fig 2.4.5 (2): Iris damper.

2.4.6 AGILENT data collection

The experimental process was monitored and controlled by Agilent data logger 34970A. Visual Engineering Environment (VEE) software was used to control the data logger and collecting the experimental data. This Windows-based application was designed to make it a snap to use computer for gathering and analyzing measurements. Use it to set up test, acquired and archived measurement data, and performed real-time display and analysis of the incoming measurements. A familiar spreadsheet environment made it easy to configure and control tests. Set up multiple graphics using strip charts, histograms, bar and scatter charts, individual channel results, and more(Figure 2.4.6).

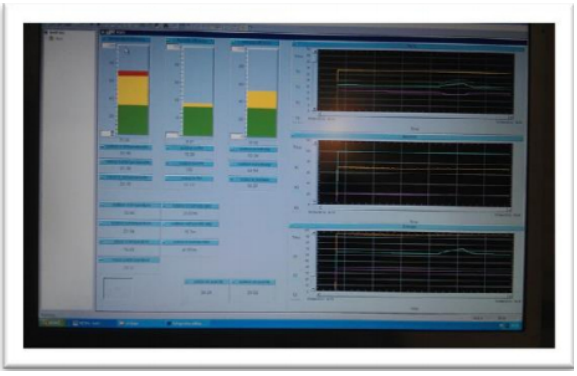


Fig 2.4.6 (1): Agilent data collection.

Fig 2.4.6 (2): Agilent data display panel.

2.5 Procedure

2.5.1 Experiment process

As mentioned, there was a trial test between the real experiments. The trial test continued for 30 min before each testing conditions. The air flow rate was determined by 25L/s which are the highest in the design.

For each weather condition, 30min was enough to stabilize the operating mode. In order to guarantee the real experiment smoothly, all the temperature and humidity deviation have been kept in 0.2°C and 1% respectively.

Before the experiment operation, pressure drop has been adjusted. The air flow rates of inlet and outlet have to keep the same during experiment process.

As the real experiment started, the ultrasonic atomizer has been connected with total heat recovery system for different numbers of resonators. For each of studied experimental conditions, between 4 and 10 resonators were chosen. Every 2 resonators would be a level to change the atomizing quantity which has inordinately helpful in this project. The other side of ultrasonic atomizer was associated with a water tank which should be filled and weighted before ultrasonic atomizer working. The weight of using water in the experimental process was the other important data to analyze the system consumption.

To make sure that the test condition was thermally stabilized, each condition was measured for 30 to 60 minutes after all the measured parameter reached to the steady state. However, only 10 min steady state data were used for the analysis.

The last step of the experiment was turning off all equipment and measuring the water weight. After that, returned two fans to keep the system running in arid condition that could dehumidifier the whole system.

2.5.2 Air flow rate

According to the tested principles of the ventilation parameters, air flow measuring points was arranged on a straight pipe. It was much easier to get accurate value when sensors were located in steady air flow part of the duct. Measuring sections were designed in the front of elbow and T-branch pipe (air movement direction relatively). The distance from these parts was greater than 2 times of the pipe diameter. If the positions were located in the back of these components, the dimensions were more than 4 to 5 times of the duct size.

There were 4 temperatures and humidity measuring points which were distributed in inlet and outlet position of the total heat exchanger. During the experimental period, temperature (t1, t2, t3, t4), humidity (d1, d2, d3, d4) at the inlet and outlet of both sides of the heat recovery unit were measured continuously at an interval of about 11 s.

After all the preparation work was ready, it was time to collect initial data as the first test in order to examine the preparations had been completed.

After the final inspection, ultrasonic atomizer could be turned on.

The experimental conditions were designed by consulting several summer city climates in the world.

Finally, four cities were chosen to simulate different weather conditions in this project. The principle of choice was a significant difference in those climates during summer period. Temperature, ratio humidity and air quantity were monitored as setting points.

2.5.3 Experiment conditions

2.5.3.1 Las Vegas

Table 1: Las Vegas experiment condition.

Outdoor:	40 ºC 11% RH		
Indoor:	25 ºC 30% RH		
Air quantity :	15L/s	20L/s	25L/s

5% of the Las Vegas weather in the summer condition was defined by 40 degrees with 11% humidity. It could be detected from the ASHRAE HANDBOOK. In this case, three different air volume levels were selected as the contrast conditions of each other.

2.5.3.2 Dubai

Table 2: Dubai experiment conditions.

Series 1:	Outdoor:	40°C 25% RH		
	Indoor:	25°C 50% RH		
	Air quantity:	15L/s	20L/s	25L/s
Series 2:	Outdoor:	40°C 25% RH		
	Indoor:	25°C 65% RH		
	Air quantity:	15L/s	20L/s	25L/s

In addition, it was important to include arid case for example Dubai where the summer temperature was around 40 degrees, but humidity was only 25%RH.There were two kinds of indoor air condition to measure by different humidity (50% and 65%).

2.5.3.3 Denmark

Table 3: Denmark climate —28 °C 40%RH.

Outdoor:	28 °C 40% RH

Indoor:	25 °C 50% RH		
Air quantity:	15L/s	20L/s	25L/s

Table 4: Denmark climate —30 °C 40%RH.

Series 1:	Outdoor:	30 °C 40% RH		
	Indoor:	25 °C 35% RH		
	Air quantity:	15L/s	20L/s	25L/s
Series 2:	Outdoor:	30 °C 40% RH		
	Indoor:	25 °C 50% RH		
	Air quantity:	15L/s	20L/s	25L/s
Series 3:	Outdoor:	30 °C 40% RH		
	Indoor:	25 °C 65% RH		
	Air quantity:	15L/s	20L/s	25L/s

Table 5: Denmark climate —32 °C 40%RH.

Outdoor:	32 °C 40% RH		
Indoor:	25 °C 50% RH		
Air quantity:	15L/s	20L/s	25L/s

Besides the extreme conditions of high temperature and low moisture, mild weather was also considered such as Denmark. Outdoor air temperature was set at three levels-28 ºC, 30 °C and 32 ºC. Indoor humidity was also consisted by 3 grades, increased from 35% to 65% every 15% as a ladder.

2.5.3.4 Shanghai

Table 6: Shanghai climate—38 °C 70%RH.

Series 1:	Outdoor:	38°C 70% RH		
	Indoor:	25°C 50% RH		
	Air quantity:	15L/s	20L/s	25L/s
Series 2:	Outdoor:	38°C 70% RH		
	Indoor:	28°C 65% RH		
	Air quantity:	15L/s	20L/s	25L/s

Table 7: Shanghai climate—35 °C 63%RH.

Series 1:	Outdoor:	35°C 63% RH		
	Indoor:	25°C 35% RH		
	Air quantity:	15L/s	20L/s	25L/s
Series 2:	Outdoor:	35°C 63% RH		
	Indoor:	25°C 50% RH		
	Air quantity:	15L/s	20L/s	25L/s
Series 3:	Outdoor:	35°C 63% RH		
	Indoor:	25°C 65% RH		
	Air quantity:	15L/s	20L/s	25L/s
Series 4:	Outdoor:	35°C 63% RH		
	Indoor:	28°C 65% RH		
	Air quantity:	15L/s	20L/s	25L/s

It was necessary in order to mention that several conditions of Shanghai were operated during the experiment. Extreme outside climate in Shanghai could reach 38 degrees with 70% RH in August, but normally it was around 35 degrees and 63% RH (ASHRAE HANDBOOK). As matching indoor environments, was defined by 25 degrees with three levels of humidity which were shown in the following tables.

2.6 Others work

2.6.1 Calibrations

2.6.1.1 Temperature and humidity sensors calibration

The temperature and humidity sensors (HMP130Y and HMP140) which produced by VAISALA company, were used to measure the temperature and humidity during the experimental process.

The HMK11 humidity calibrator has been designed for calibration and checking of humidity probes and transmitters. The functioning of the calibrator was based on the fact that certain salt solutions generate a specified relative humidity in the air above them.

There were 4 holes on the box which covering Vaisala probes and transmitters with 12mm, 13.5mm(include 2 holes) and 18.5mm diameters.

Liquor in the HMK 11 calibrator was lithium chloride LiCl (11%RH) and sodium chloride NaCl (75%RH).

Perform the calibration according to the following steps:

-Leave the HMK11 calibrator and the probe at the calibration site for at least 30 minutes before starting the procedure to make sure temperature of probe stable and close to the chamber temperature.

-insert a new probe into a suitable hole of the LiCl salt chamber. At shorter time as possible before open the hole when inserting the probe.

-please wait until the humidity reading stabilizes; it seems take about 30 minutes.

-Adjust the dry end(DRY, offset) which locate in humidity sensors to correspond to the value.

-Insert the thermometer into the hole of the NaCl salt chamber.

-Waiting around 30 minutes or even longer for stabilization time.

-Adjust the wet end (WET. Gain) to correspond to the value depending on the calibration table.

The error was ± 0.2 when the LiCl solution was used to calibrate the humidity, at the room temperature of 25 ℃. The temperature error was ± 0.1 with the calibration of NaCl solution, which satisfies the calibration requirement. The accuracy of the sensor may change during the experiment process due to being exposed to high humidity and extreme saturation condition in long term. Therefore, the calibration for the sensor should be repeated twice during the experiment to ensure the accuracy.

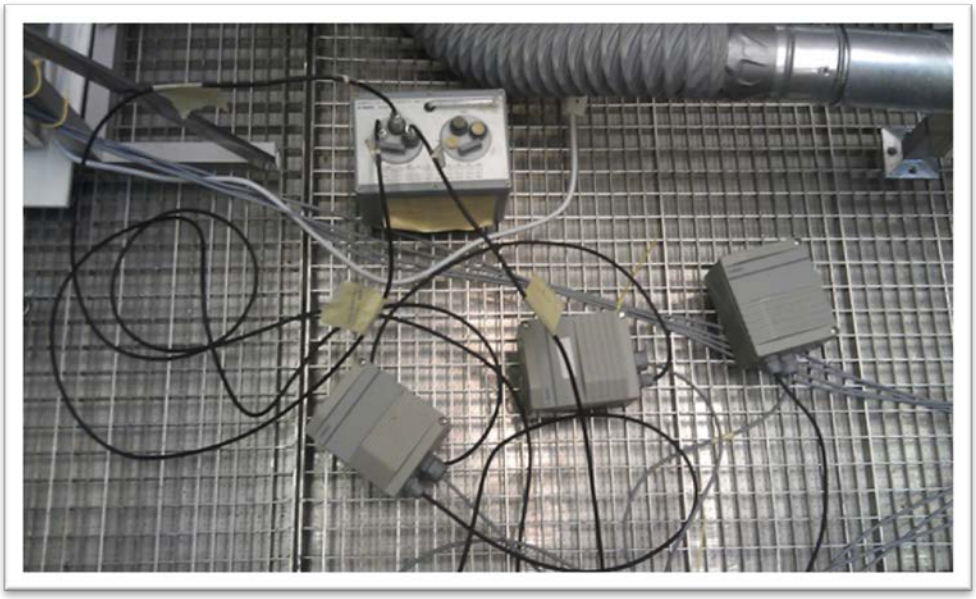


Fig2.6.1.1(1): Calibration sensors before experiment.

2.6.1.2. Airflow pressure calibration

The iris damper airflow meters were calibrated prior to the experiment to ensure the same airflow rates on both side of the heat exchanger. The calibration was conducted by connecting the two iris dampers in serial, measuring the pressure drop on each damper and adjusting the opening of the iris damper to make sure identical pressure drop on both iris damper. The identical flow coefficient was then used for the two iris dampers. FCO510 micromanometer was used for the calibration the calibration was conducted in the airflow ranging from 15 to 28 L/s to cover the range of airflow used in the experiment.

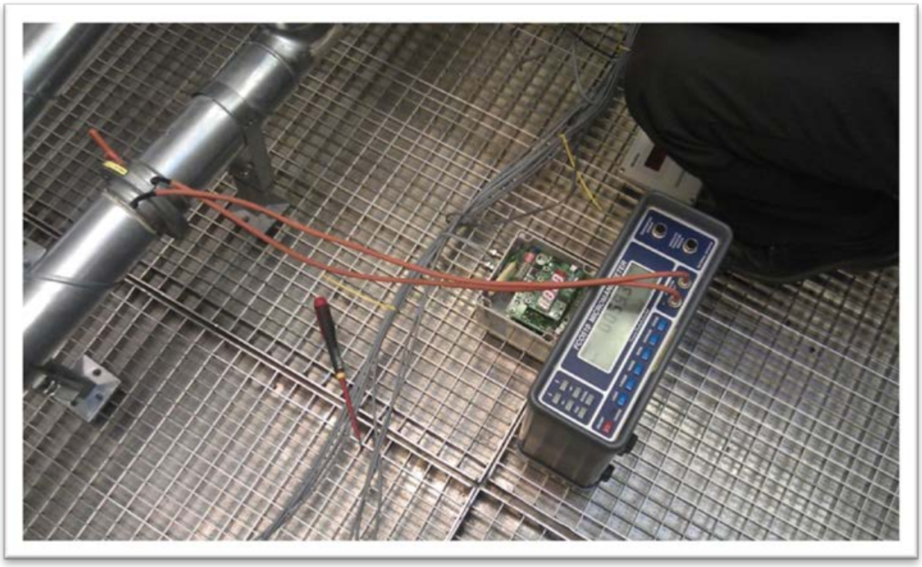


Fig2.6.1.2(1): Airflow calibration.

2.6.2 Smoke fuming

Tiny C 07 is built to be used with original look tiny-fluid. Use of other liquids will clog the vaporizer and void the warranty. Pour the Tiny-fluid into the fluid tank and firmly shut the lid. Check the cleanliness of the fluid tank, so that nothing dirt can be sucked in by the pump. Insert a new battery into the machine until it clichés into place. Press the start button on the lid of the machine. After around 1 second the unit generates fog as long as the button is held.

Depending on the analysis of results in this project, a special reason for the deviation would be different pressure between inlet and outlet systems. This phenomenon generated the leakproofness problem which was happened in the place between the heat exchanger and outer-shell. To be sure whether it has to gas leakage, carried out using C07 as the equipment to measure the direction of airflow. By detecting the smoke fuming experiment, it has the presence of leakage which reducing the efficiency and making deviations.

Fig 2.6.2 (1) : Tiny C 07 smoke fuming test.

2.6.3. Insulation

The temperature in the indoor test chamber was maintained constant at 25 °C and 28 °C for different simulated conditions. To avoid experimental error due to heat transfer between the air inside the heat exchanger and the ambient air, the test rig was insulated by polystyrene. It was made of polystyrene resins and other additives manufactured by the closed cellular structure of the sheet. It had the same low thermal conductivity (only 0.028W/MK) and excellent thermal performance and compression enduring, compressive strength of up to 220-500Kpa.

During preparing for experiment, not only the ducts and elbows had to insulate by special material, also the heat exchanger unit. The influence of heat transmission between inlet and outlet channels could result in huge error of the result. To avoid this error, polystyrene board was used between the two channels to eliminate the heat transfer.

2.6.4. Electronic driving circuit of ultra-sonic resonator

The commercialy available ultrasonic resonators were built together with the electronic driving circuits. In the pre-experiment, it was found that the heat generated by the driving circuits heated up the water that decreased the effect of evaporative cooling. To remove this heating effect, the electronic driving circuits were removed away from the water tank and were rebuilt. Each circuit board rack has been equipped with a mini fan to remove the heat generated by the circuits. Secondly, there were 10 groups of electronic panel connected with ultra-sonic resonators, all of them could be controlled independently. It means that the atomizing quantity can be controlled. Last but not least, for safety reason, all electrical system should be fed off in the experiment at the end of every day.

Figure 2.6.4 (1): Electronic driving circuit of ultra-sonic resonator.

3. Result

3.1. Las Vegas

3.1.1 Las Vegas outdoor condition: 40 °C with 11% RH.

Las Vegas indoor condition: 25 °C with 30%RH.

Table 8:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Las Vegas.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	40,6 °C	40,4 °C	39,9 °C
	Relative humidity:	11,3 %	11,3 %	11,7 %
	Air quantity:	15,4 L/s	20,1 L/s	25,2 L/s
Indoor:	Temperature:	25,0 °C	25,0 °C	25,1 °C
	Relative humidity:	30,5 %	30,4 %	30,6 %
	Air quantity:	15,4 L/s	20,5 L/s	25,1 L/s

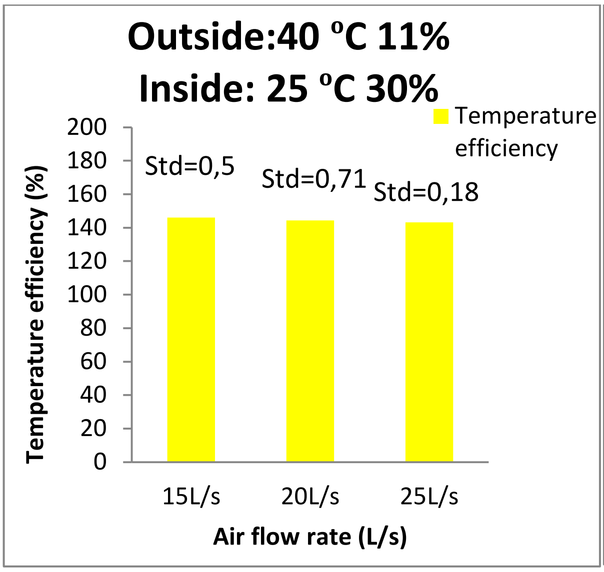
The chart below indicates the result of Las Vegas. The graph shows the trends in consumption in different airflow rates.

The first bar chart shows the temperature efficiency in the condition of 40 ºC and 11% RH outdoor climate and 20 ºC and 30% RH indoors at three levels of airflow rate, i.e. 5L/s, 15L/s and 25L/s. The figure shows average temperature efficiencies decreased with increasing airflow rate from low to high level However, the enthalpy efficiency showed an opposite trend that it increased with increasing airflow rate.

The supply temperature revealed a similar trend of growth as that of supply enthalpy, though the growth rate was not fast. Supply temperature rose slowly from 18 ºC in diverse air qualities, comparing with outside temperature which is always controlled by 40 ºC in this case. In addition, enthalpy of supply air is not affected by three levels airflow in this hyperthermia situation.

The deviations of two efficiencies have been discussed by software-Statistica. It can be observed that experimental results are relatively stable.

Through the statistical analysis, the otherness of data was not significant.



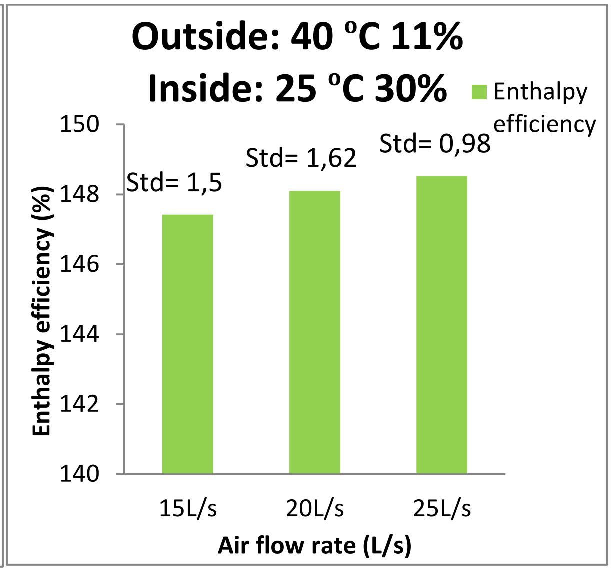
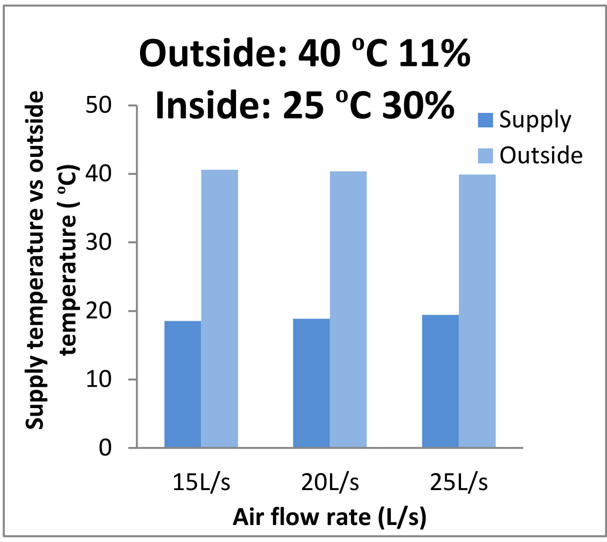


Figure3.1 (1): Las Vegas temperature efficiency. Figure3.1 (2): Las Vegas enthalpy efficiency.



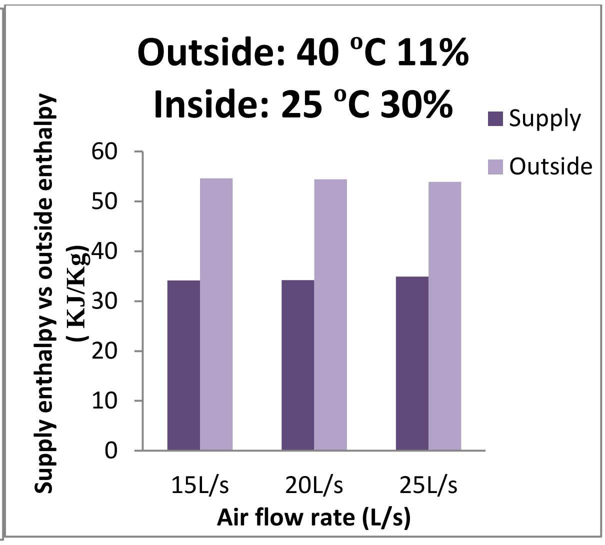


Figure3.1 (3): Las Vegas supply temperature. Figure3.1 (4): Las Vegas supply enthalpy.

Discussion:

From the chat 3.1 (3)and 3.1(4), outside temperature decreased from 40 to 18 ºC after total heat exchanger, dropped 22 degrees in summer condition. Corresponding to this temperature decrease, Enthalpy decreased 20 KJ/Kg between supply air and outside air. For these hot areas, the technique has obviously impacts which can effectively reduce the temperature supply air. The absolute heat recovery unit in this condition not only could recover the energy for ventilation, but also provide strong cooling capacity as the result shown that enthalpy efficiency was higher than 100%.

3.2. Dubai

3.2.1. Dubai outside condition: 40 °C with 25% RH.

Dubai inside conditions: 25 °C with 50%RH and 65%RH.

Table 9:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Dubai. The indoor relative humidity was designed as medium level.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	40,0 °C	40,0°C	40,1 °C
	Relative humidity:	25,0 %	25,0 %	25,2 %
	Air quantity:	15,6 L/s	20,4 L/s	25,1 L/s
Indoor:	Temperature:	25,1 °C	25,2 °C	25,0 °C
	Relative humidity:	50,3 %	51,5 %	50,018 %
	Air quantity:	15,5 L/s	20,1 L/s	25,1 L/s

Table 10:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Dubai. The indoor relative humidity was designed as high level.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	39,8, °C	40,0 °C	40,2 °C
	Relative humidity:	24,7 %	24,5 %	25,1 %
	Air quantity:	15,5 L/s	20,1 L/s	25,0 L/s
Indoor:	Temperature:	24,8 °C	24,7 °C	25,0 °C
	Relative humidity:	65,6 %	65,9 %	65,3 %
	Air quantity:	15,5 L/s	20,3 L/s	25,1 L/s

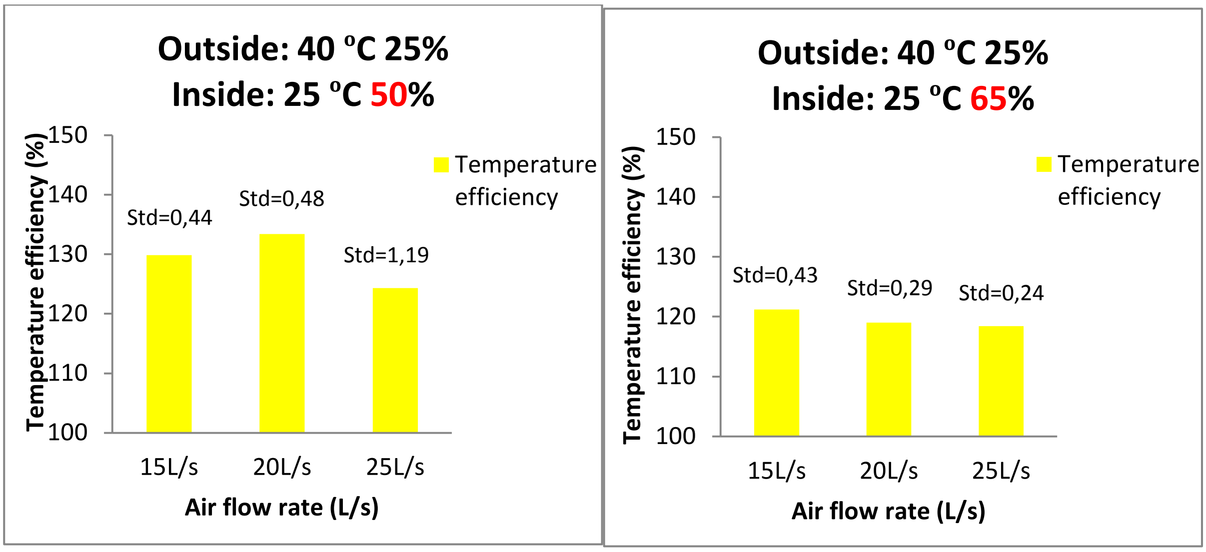


Fig 3.2 (1)Temperature efficiency in Dubai(50%).Fig3.2 (2)Temperature efficiency in Dubai(65%).

The bar charts compare the temperature efficiency proportion in two levels relative humidity of both 50% and 65% inside the building within three airflow rates between 15L/s and 25L/s. As can be observed in the chart, the percentage of indoor high humidity condition generally decreased with the rise of the air quantity while the average value was equaled 125 KJ/Kg with that of two conditions in Dubai.

When it comes to enthalpy efficiency (Figure 3.2 (1) and (2)), values of 65% RH inside building also keeps stable, accounting for 130% which is 50% higher than that of low indoor humidity situation.

However, the situation in the other two analysis data categories was relative uniformity. It was reported as 23 ºC in supply temperature bar charts. And supply enthalpy constituted roughly 55KJ/Kg in both 50% RH and 65% RH indoor humidity conditions.

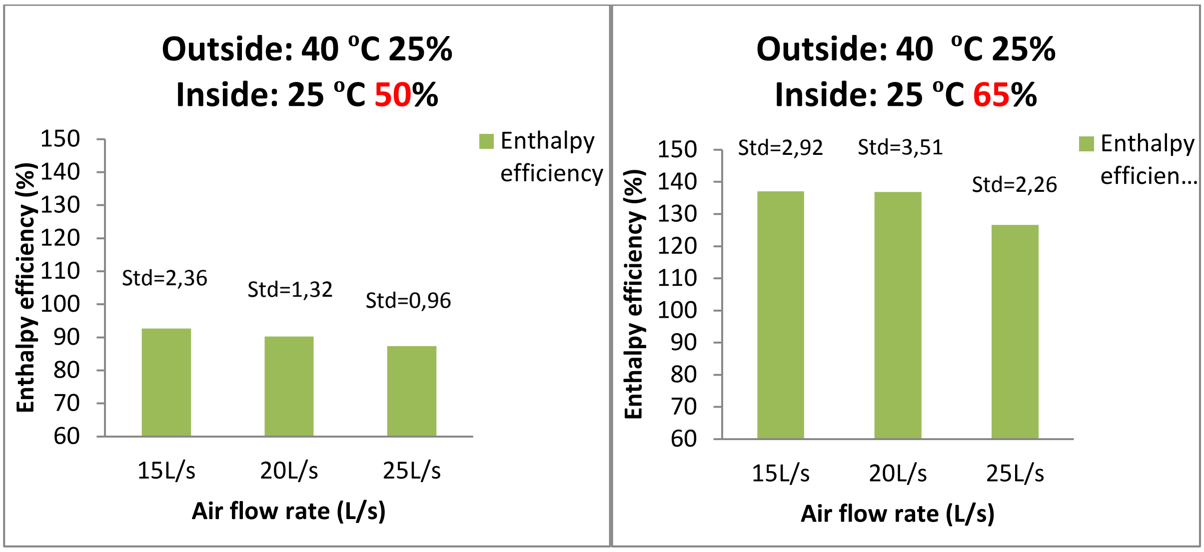


Figure 3.2(3) Enthalpy efficiency in Dubai (50%).Figure 3.2 (4) Enthalpy efficiency in Dubai (65%).

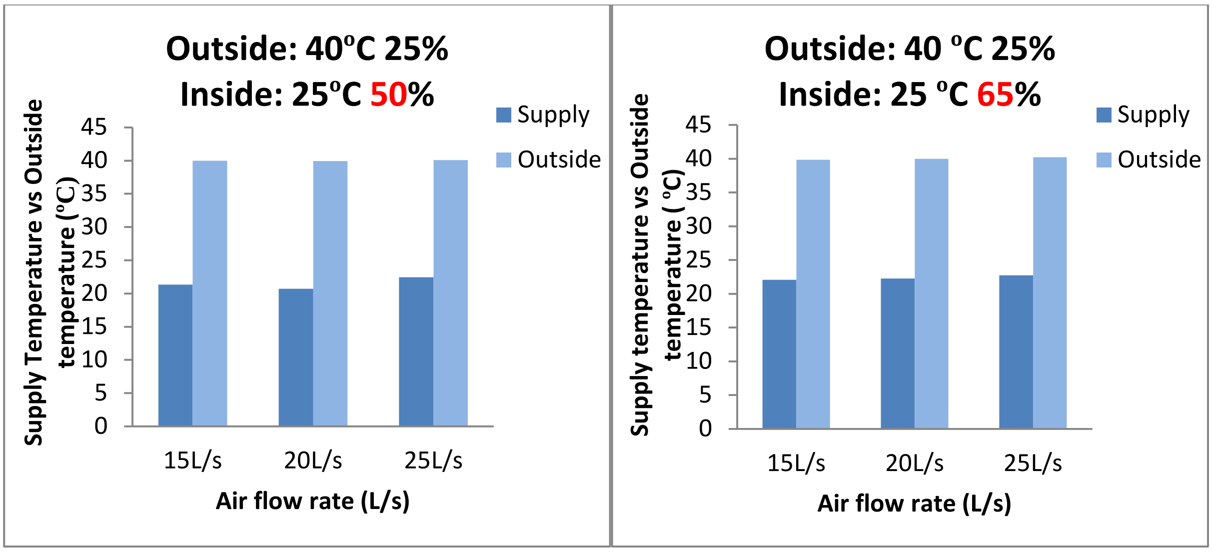
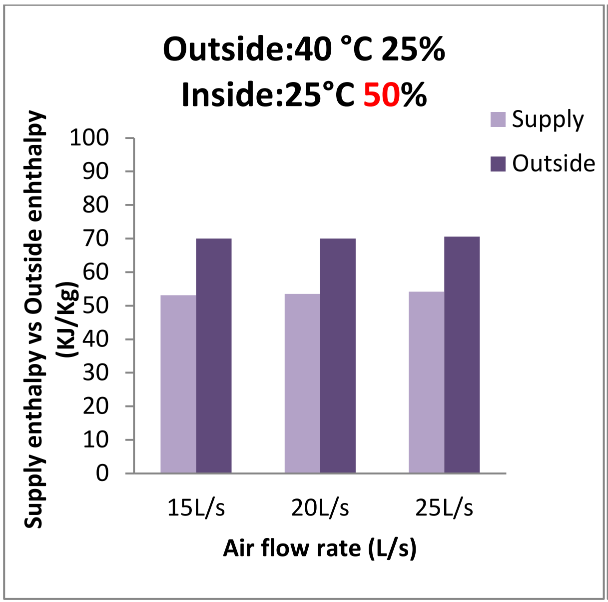


Figure 3.2(5)Temperature comparing in Dubai(50%). Figure 3.2(6)Temperature comparing in Dubai(65%).



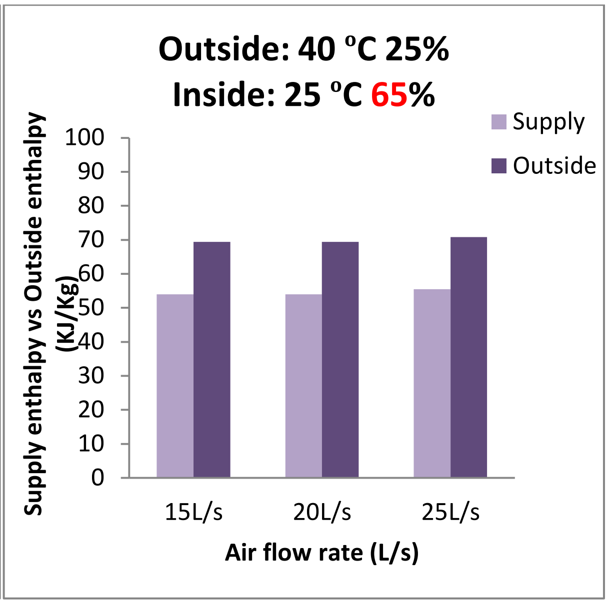


Figure 3.2 (7) Enthalpy comparing in Dubai (50%). Figure 3.2 (8) Enthalpy comparing in Dubai (65%).

Discussion:

The impact caused by the different indoor air humidity was studied by comparing the two conditions. It is revealed from the results that the inlet air temperature was not affected distinctly by increasing indoor air humidity, which means almost no more energy was recovered. Whereas the enthalpy efficiency at the indoor RH of 65% was approximately 50 % higher than when indoor air humidity was 50 %. The reason is that the indoor enthalpy value increased which results in a smaller denominator in the enthalpy efficiency equation and as a result the enthalpy efficiency increased.

3.3. Denmark

3.3.1 Denmark outdoor condition: 30 ºC with 40%RH.

Denmark indoor condition: 25 ºC with 35%RH.

Table11:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Denmark. The indoor relative humidity was designed as low level.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	30,1 °C	30,1 °C	30,2 °C
	Relative humidity:	40,0%	40,0 %	39,6 %
	Air quantity:	14,6 L/s	19,7 L/s	25,2 L/s
Indoor:	Temperature:	25,1, °C	25,1 °C	25,1 °C
	Relative humidity:	34,3 %	35,4 %	34,2 %
	Air quantity:	14,8 L/s	19,9 L/s	25,1 L/s

The temperature, humidity, and enthalpy efficiencies of the heat recovery unit measured at the three levels of indoor air quantity with the inside air condition of 25 ºC/35% RH and outdoor air of 30 ºC/40 % RH are shown in Figs3.3(1)and 3.3(2).

The figures show that the temperature efficiencies of the heat recovery unit were slightly over 240%. Enthalpy efficiencies were slightly lower than 80% also regardless of airflow rate during same air conditions both indoor and outdoor. The average supply temperature is comparable with 18 ºC which can reduce indoor temperature effectively. Comparing with outside simulation, 12 degrees differences of temperature are appeared in this special technical. Both temperature and enthalpy efficiencies and the supply air conditions were not influenced by airflow rate.

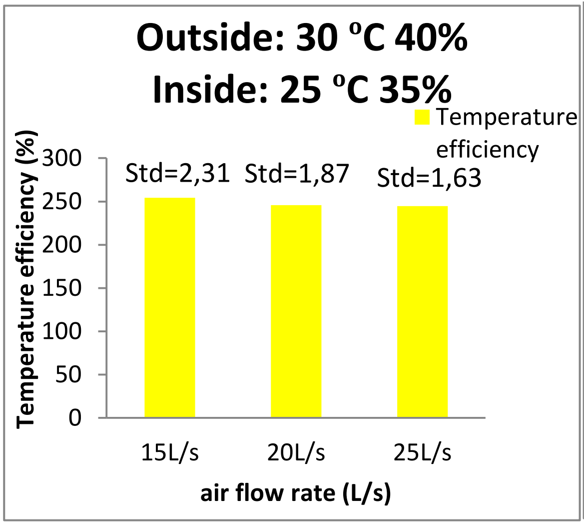


Figure3.3 (1): Temperature efficiency.

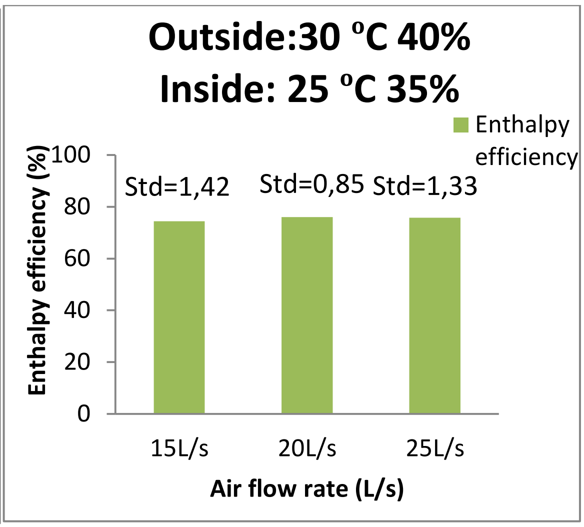


Figure3.3 (2): Enthalpy efficiency.

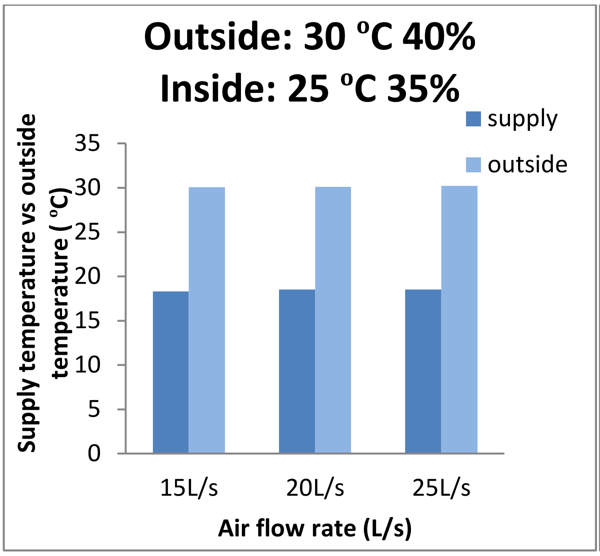


Figure3.3 (3): Supply temperature.

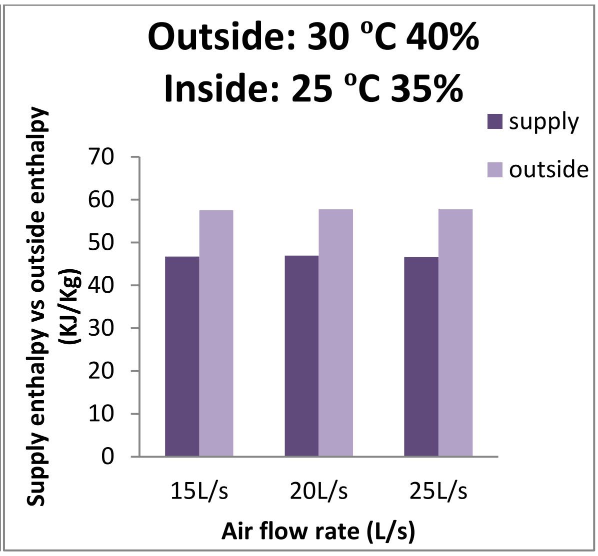


Figure3.3 (4): Supply enthalpy.

Discussion:

Obviously, this is a very ideal result, whether it from the temperature and enthalpy efficiency value, or the lower supply air temperature is discussed. Thus, it is concluded that the device can replace air conditioning to inlet cold and comfortable fresh air during the hot summer period in Denmark. It can be seen in psychometric chart that state point of indoor air is close to outdoor air point in this condition. It results in the denominator of enthalpy efficiency calculation formula becoming smaller, leading to a slightly higher efficiency by more than 75%.

3.3.2 Denmark outdoor condition: 30 ºC with 40%RH.

Denmark indoor condition: 25 ºC with 50%RH.

Table 12:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Denmark. The indoor relative humidity was designed as medium level.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	30,4 °C	30,3 °C	30,1 °C
	Relative humidity:	39,4 %	39,3 %	39,8 %
	Air quantity:	15,5 L/s	20,4 L/s	24,6 L/s
Indoor:	Temperature:	24,9 °C	25,1 °C	25,0 °C
	Relative humidity:	51,2 %	49,3 %	50,4 %
	Air quantity:	14,6 L/s	20,0 L/s	24,5 L/s

Fig 3.3(5) and (6) temperature and humidity efficiencies measured at three different airflow rate conditions with the outside simulation of 30 ºC 40% RH. The 50% relative humidity was defined by indoor case. In Denmark summer period, this is the most common condition due to bright sunlight caused by high temperature.

As Figure 3.3(5) showed that temperature efficiency was in the range of 195 KJ/Kg to 210KJ/Kg. The result assumed an increasing trend with the air quantity up. Based on the enthalpy efficiency calculation formula of the law, if the denominator which was outdoor enthalpy minus indoor enthalpy has been decreased, efficiency would be increased by huge range. It was stable both in supply temperature and enthalpy results. Average 20 degrees was also gratified to people working inside the building. In addition, 10 KJ/Kg decreasing was a big energy savings come from the moisture transfer during summer.

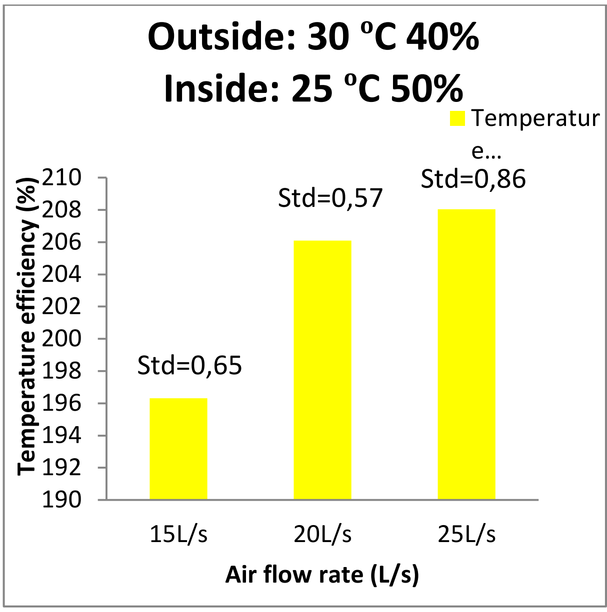


Figure3.3 (5): Temperature efficiency.

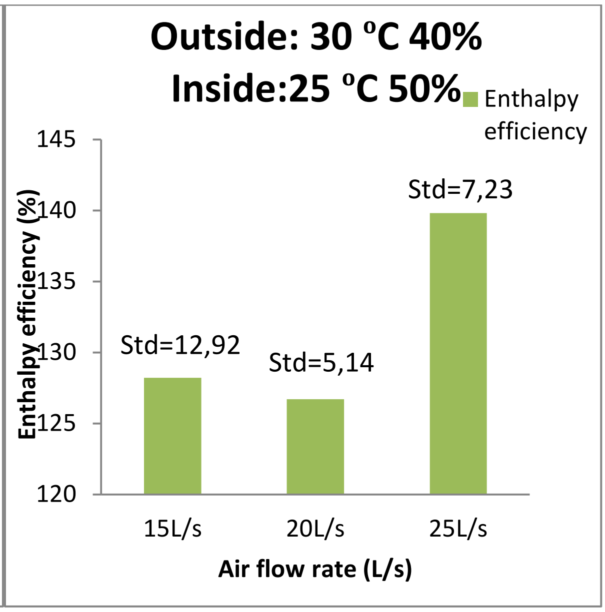


Figure3.3 (6): Enthalpy efficiency.

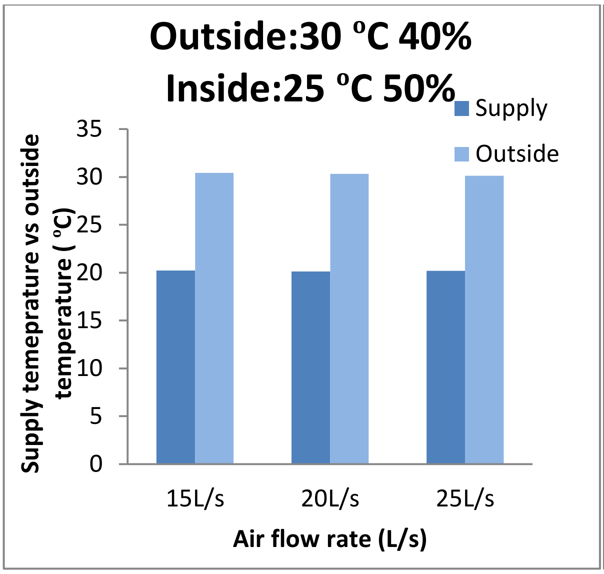


Figure3.3 (7): Supply temperature.

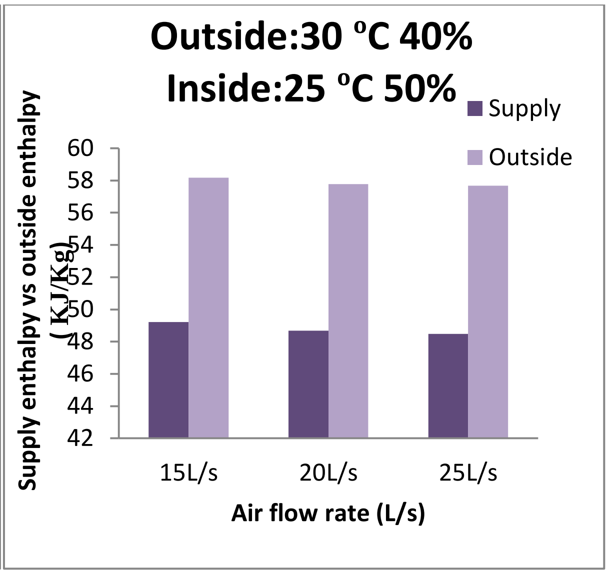


Figure3.3 (8): Supply enthalpy.

Discussion:

Higher deviations were appeared in fig3.3 (6) which was expression of enthalpy efficiencies. Due to the smaller SD-value between indoor and outdoor enthalpy, the denominator would be decreased. Thereby, efficiencies in green bar chart had biggish deviation leading to a big difference. No distinct difference was found in supply temperature when air quantity was changed.

Generally, 25L/s airflow rate would be the best situation in this weather condition with much enthalpy savings and almost 20 ºC supply air. To remind that it was necessary to provide sufficient water spray and not waste.

3.3.3 Denmark outdoor condition: 30 ºC with 40%RH.

Denmark indoor condition: 25 ºC with 65%RH.

Table13:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Denmark. The indoor relative humidity was designed as high level.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	30,2 °C	29,9 °C	30,0 °C
	Relative humidity:	39,7 %	39,5 %	39,1 %
	Air quantity:	15,0 L/s	19,9 L/s	25,0 L/s
Indoor:	Temperature:	25,0 °C	24,9 °C	24,9 °C
	Relative humidity:	64,4 %	64,7 %	64,2 %
	Air quantity:	15,0 L/s	20,3 L/s	24,5 L/s

Comparing with previous two experimental conditions, indoor high relative humidity with 65% RH is the characteristic in this simulation. The humidity and enthalpy efficiency was also studied at three air quantity conditions. The huge differences of enthalpy efficiency can be ignored because of similar values between two enthalpies in the part of the denominator. Supply temperature is 22 ºC which is 2 degrees more than previously condition.

There were the other two conditions presented which the outside temperature was 28 ºC and 32 ºC respectively. The indoor air was kept on 25 C with 50% RH for normally. As mentioned, the average supply temperature in Denmark analysis was around 20 ºC which could satisfy Danish requirement during summer.

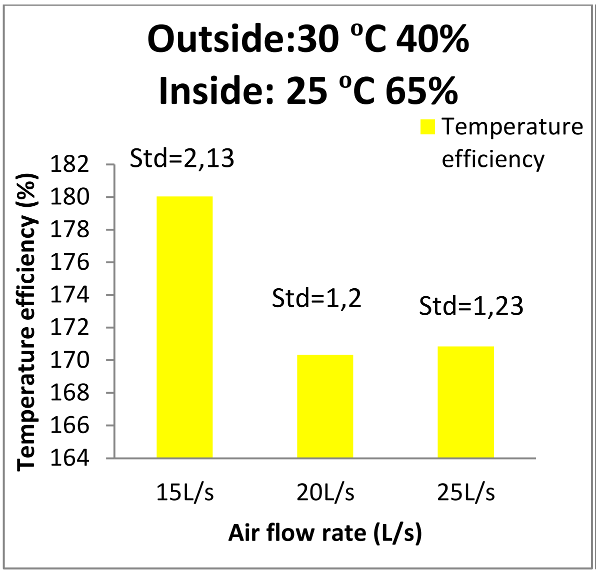


Figure 3.3 (9): Temperature efficiency.

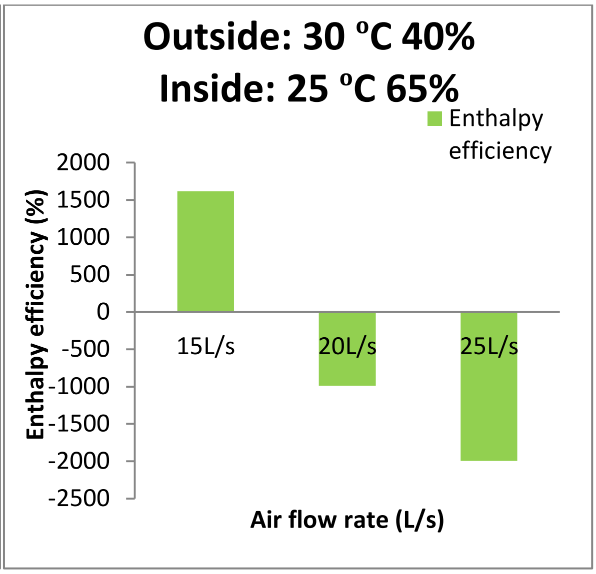


Figure 3.3 (10): Enthalpy efficiency.

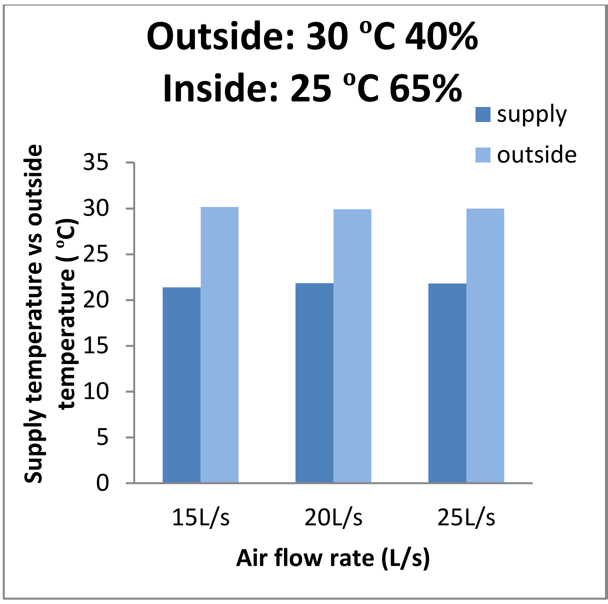


Figure 3.3 (11): Supply temperature.

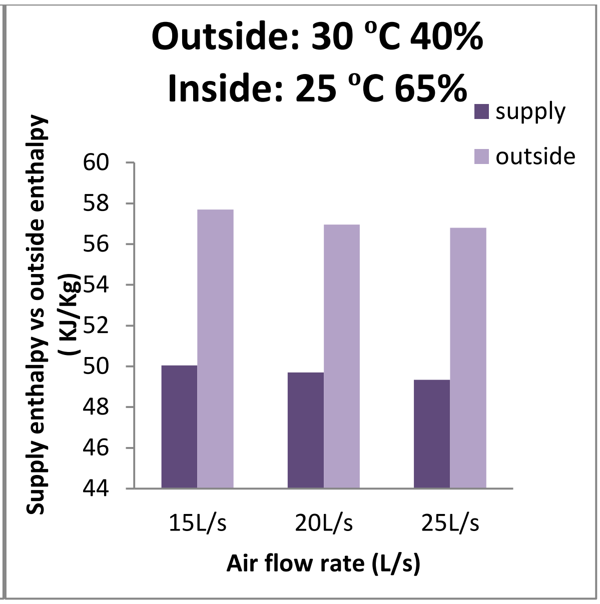


Figure 3.3 (12): Supply enthalpy.

Discussion:

By comparing the different air volume in the same indoor and outdoor climate condition of experiment, there was no definite influence came from changes in air quantity only based on this trial.

After the total heat recovery unit, fresh air could be supplied directly by 20 ºC. This indicated that the increase in air volume did not cause much more changing for two kinds of efficiencies and other results as expected. Both temperature and enthalpy efficiencies could not be influenced by any changes of outside climates besides it had a little effect on indoor relative humidity.

3.3.4 Denmark outdoor condition: 30 ºC with 40%RH.

Denmark indoor condition: 25 ºC.

Air quantity: 15L/s.

Table 14:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Denmark. The indoor relative humidity was designed by three levels condition.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	30,4 °C	29,9 °C	30,0 °C
	Relative humidity:	39,7 %	39,5 %	39,1 %
	Air quantity:	15,0 L/s	14,9 L/s	15,2 L/s
Indoor:	Temperature:	25,0 °C	24,9 °C	24,9 °C
	Relative humidity:	34,4 %	54,7 %	64,2 %
	Air quantity:	15,0 L/s	14,8 L/s	14,7 L/s

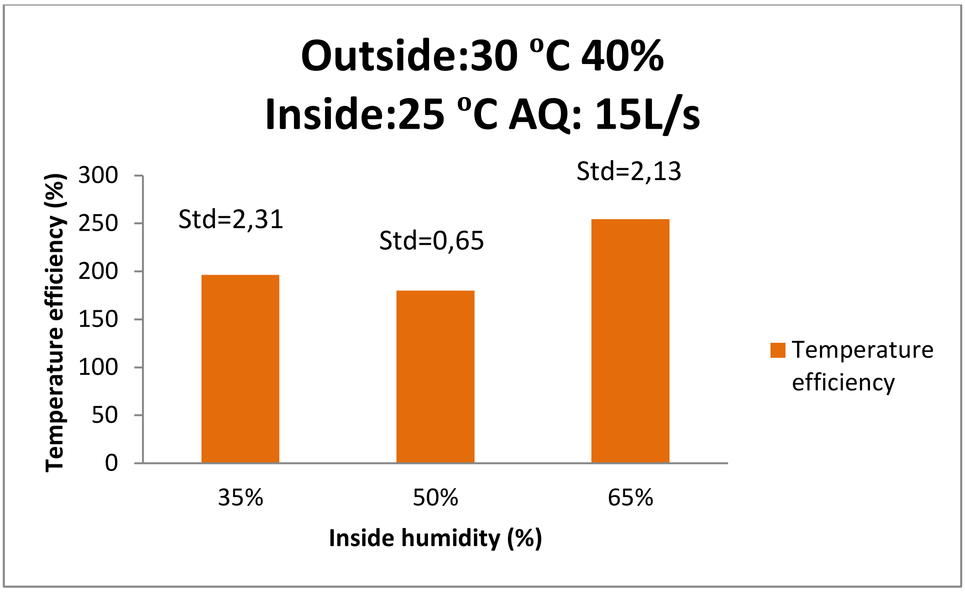


Figure 3.3 (13): Denmark experiment condition.

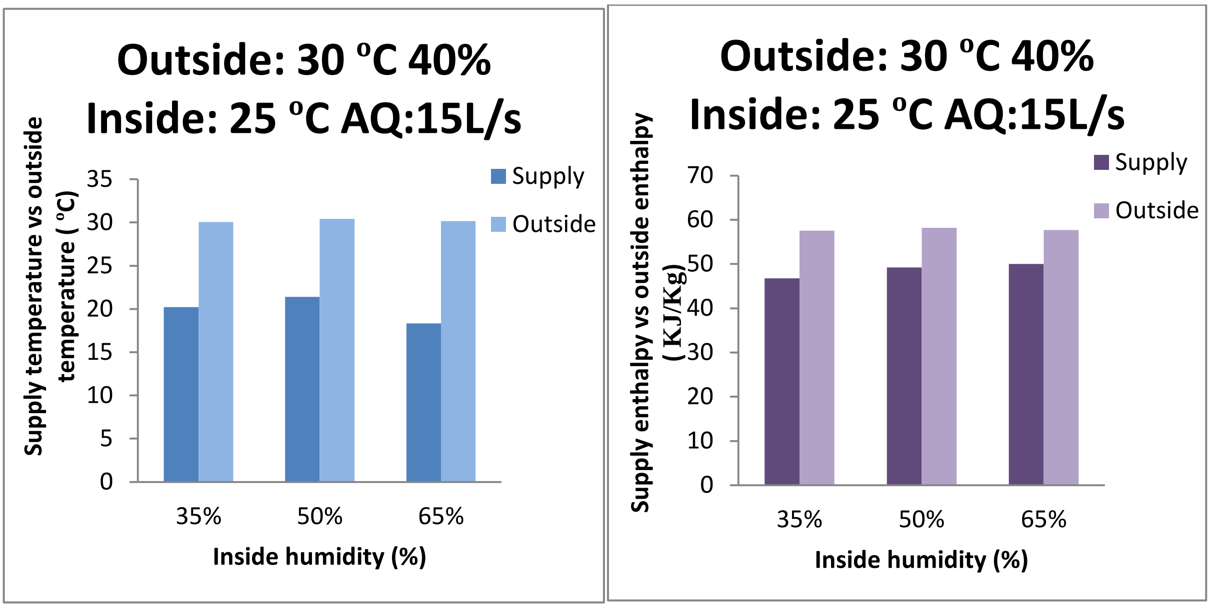


Figure 3.3 (14): Denmark experiment condition.

Fig3.3(13) revealed the temperature distribution in the condition of 30 ºC with 40% RH outside and 25 ºC with different indoor relative humidity while the air supply flow rate was set at 15 L/s.

However, the temperature in the supply air part was slightly different when inside humidity was operated at 35%, 50% and 65%. The results for the enthalpy were steady growth with the increasing of inside humidity when the air quantity was at 15 L/s, which value was approached 50 KJ/Kg. The supply temperature seems not follow a trend which was increased or decrease. The reason for this should be deviations of analysis data.

3.3.5 Denmark outdoor condition: 30 ºC with 40%RH.

Denmark indoor condition: 25 ºC.

Air quantity: 20L/s and 25L/s.

Table 15:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Denmark. The indoor relative humidity was designed by three levels condition with medium air quantity.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	30,2 °C	29,8 °C	30,4 °C
	Relative humidity:	39,9 %	39,4 %	39,8 %
	Air quantity:	15,2 L/s	14,7 L/s	15,2 L/s
Indoor:	Temperature:	25,2 °C	24,8 °C	24,7 °C
	Relative humidity:	34,2 %	54,9 %	64,8 %
	Air quantity:	20,0 L/s	19,9 L/s	19,9 L/s

Table 16:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under climate condition of Denmark. The indoor relative humidity was designed by three levels condition with high air quantity.

Location	Parameter	Condition 1	Condition 2	Condition 3
Outdoor:	Temperature:	30,4 °C	29,6 °C	30,2 °C
	Relative humidity:	39,8 %	39,5 %	39,5 %
	Air quantity:	15,0 L/s	14,9 L/s	15,2 L/s
Indoor:	Temperature:	25,4 °C	24,6 °C	24,6 °C
	Relative humidity:	34,7 %	54,5 %	64,4 %
	Air quantity:	25,0 L/s	24,8 L/s	24,7 L/s

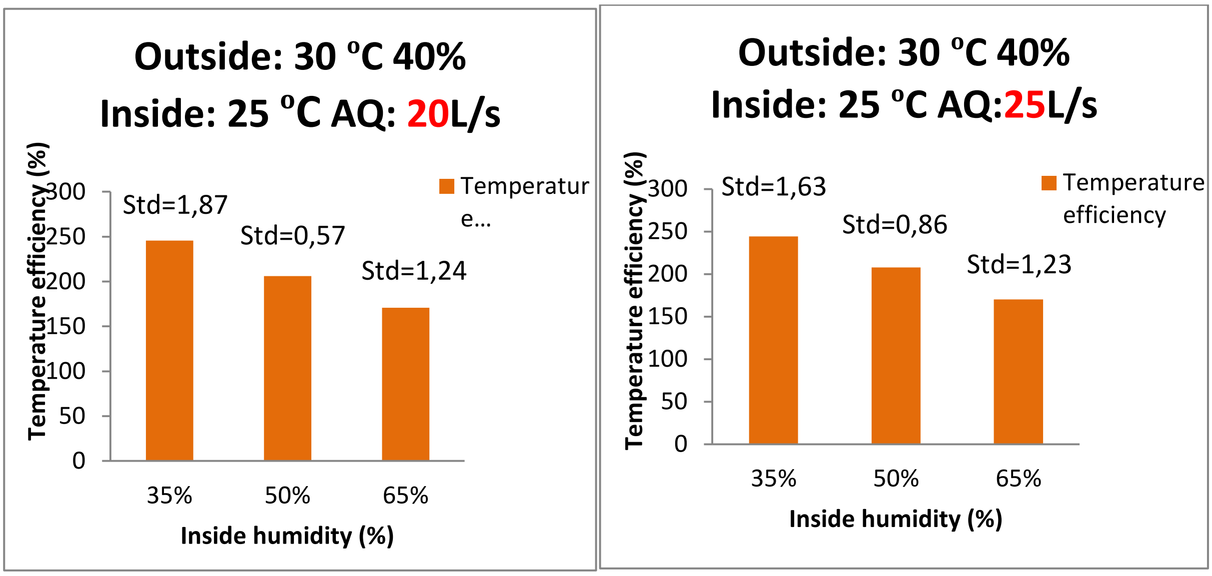


Figure 3.3(15): Denmark experiment condition.

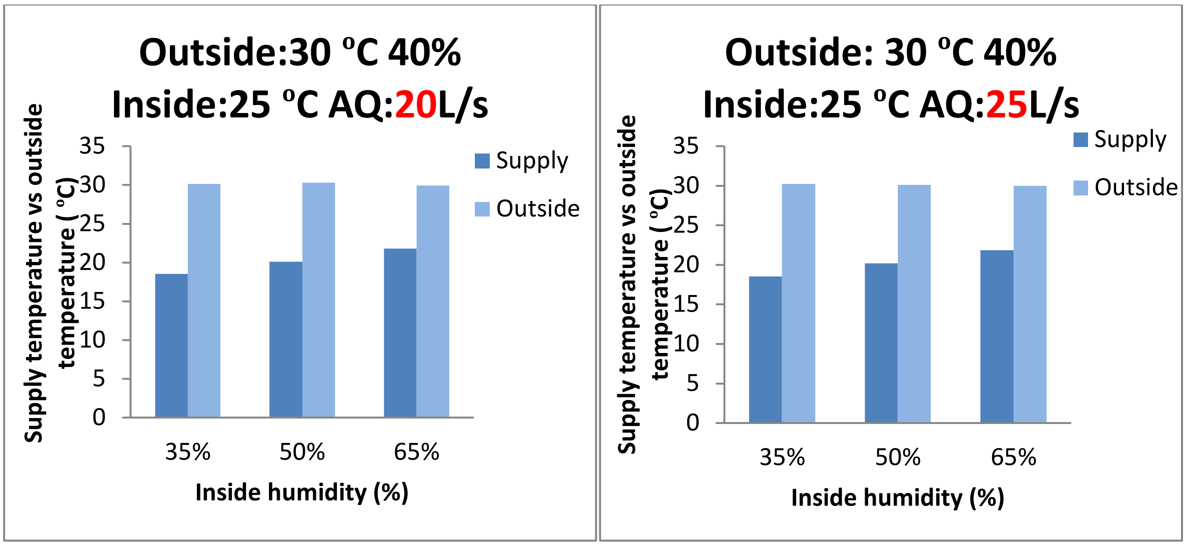


Figure 3.3(16): Denmark experiment condition.

For both airflow rates of 20 L/s and 25 L/s used in three relative humidity with the same outside climate presenting in figures. A clear impact of indoor humidity on both temperature and enthalpy efficiency can be observed. It can be possible to conclude that indoor humidity had a relatively smaller impact on the supply temperature distribution. The maximum supply temperature will be 23 degrees took place in higher relative humidity of 65%. It was well-understood that efficiency of temperature decreased by indoor humidity growing. At the same time, supply temperature has been rose by 2 degrees in each indoor RH variation. The explanation of the decreased temperature and enthalpy efficiency with the increase of indoor humidity could due to the reduced evaporation cooling with increase of indoor humidity.

3.3.6. Denmark outdoor condition: 40%RH.

Denmark indoor condition:25 °C with 50%RH.

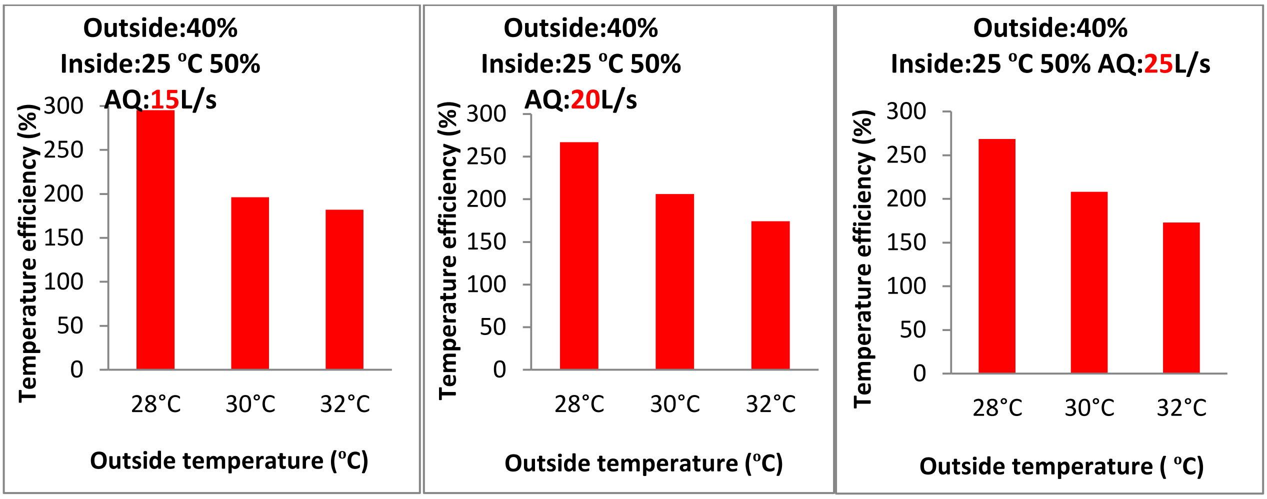


Figure3.3(17) Temperature efficiencies.

Fig3.3(17) compares temperature efficiency at three levels of airflow rate each with three levels of outdoor air temperature. The figure shows a clear pattern that the temperature efficiency decreased with increasing outdoor air temperature. Decrease airflow rate below 20 L/s slightly increase temperature efficiency. Above 20 L/s, airflow has little influence on temperature efficiency. However, enthalpy efficiency increased very much at low airflow of 15L/s when outdoor air temperature was 28°C (see Fig. 3.3(18)),. This is because the enthalpy difference between indoor air at 28°C/40%RH and outdoor air at 25°C/50%RH was very close. This makes enthalpy efficiency very sensitive to the change of temperature and humidity of supply air after the heat exchanger. Change the number of resonator in the ultrasonic atomizer has little influence on enthalpy efficiency

It could be seen in fig 3.3 (19), supply temperatures were quite similar in different outdoor climate and air quantity conditions reported by average value of 20 degrees.

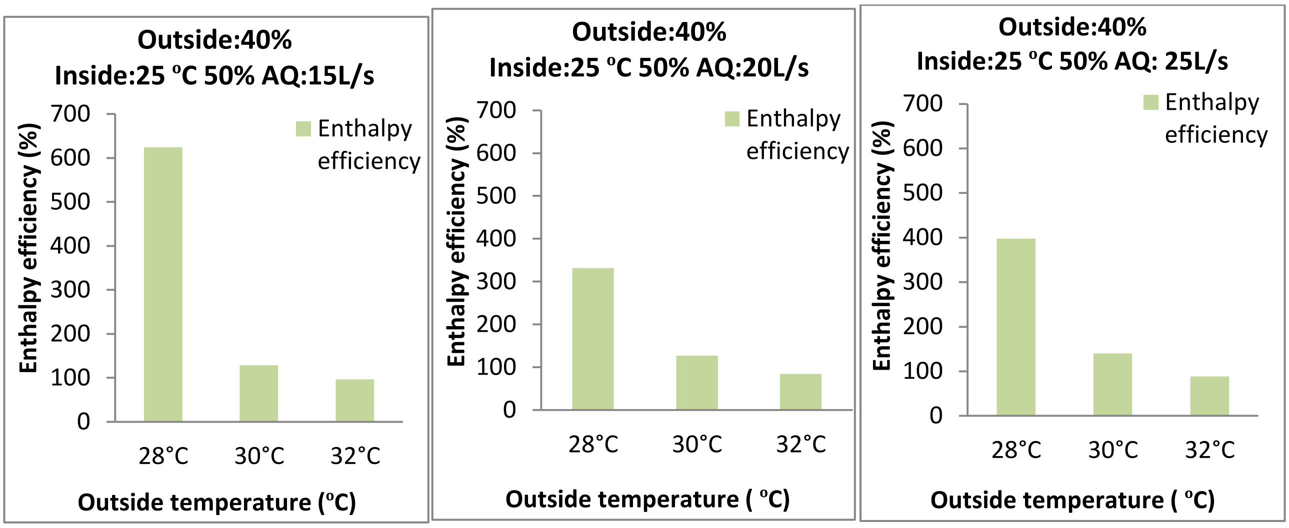


Figure 3.3 (18) Enthalpy efficiencies

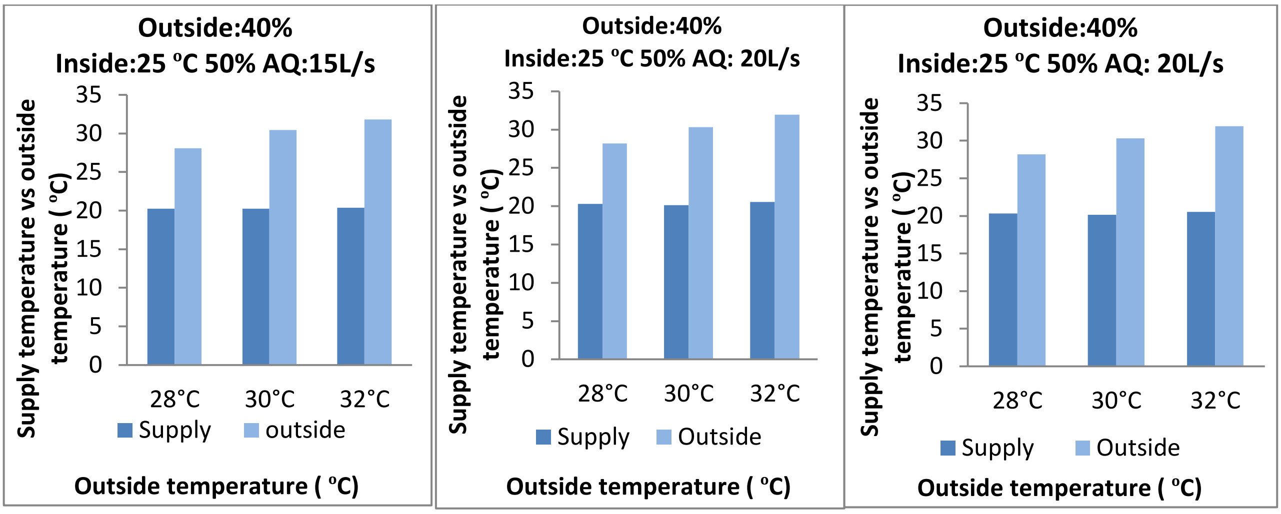


Figure3.3 (19): Supply temperatures.

3.4. Shanghai

3.4.1 Shanghai outdoor condition: 35 ºC with 63%RH.

Shanghai indoor condition: 25 ºC with 50% RH.

Table17:

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under designed climate condition of Shanghai. The indoor relative humidity was designed by medium level.

Location:	Parameter:	Condition 1	Condition 2	Condition 3
Outdoor	Temperature:	34,951 °C	35,056 °C	35,107 °C
	Relative humidity:	62,268 %	63,587 %	62,486 %
	Air quality:	15,74 L/s	20,196 L/s	25,09 L/s
Indoor	Temperature:	25,159 °C	25,226 °C	25,150 °C
	Relative humidity:	51,377 %	51,124 %	50,256 %
	Air quality:	15,753 L/s	20,215 L/s	24,96 L/s

A summary of the data representing the total monitoring period is shown in table 17. Urban heat island effect was increasingly seriously, the summer outdoor temperature rising year by year in Shanghai. On the other hand, recently, the relative humidity about July and August would reach 70%. In this test, 35 ºC and 63% RH was chosen for normal design pattern with indoor climate of 25 ºC 50% RH simulated for working environment.

Following 4 figures reported the result of the measurement by temperature and enthalpy efficiencies, supply temperature and supply enthalpy compared with external conditions respectively.

Fig 3.4 (1) showed the parameters of temperature efficiencies at airflow rate of 15L/s, 20L/s and 25L/s. Average temperature efficiency value was around 130% with the floating not over 10%. Green bar chart shows the enthalpy efficiency with maximum value of 70% appeared when airflow rate was 25L/s. Both temperature and enthalpy efficiencies were slightly increased with the increase of airflow rate. In practice, deviation control was the major of data credibility. Either temperature or enthalpy efficiency deviations was displayed under 1 that means the data was stable.

Comparing with outside high temperature 35 ºC, inlet air could achieve 21 degrees which balance indoor air heat load during the summer.

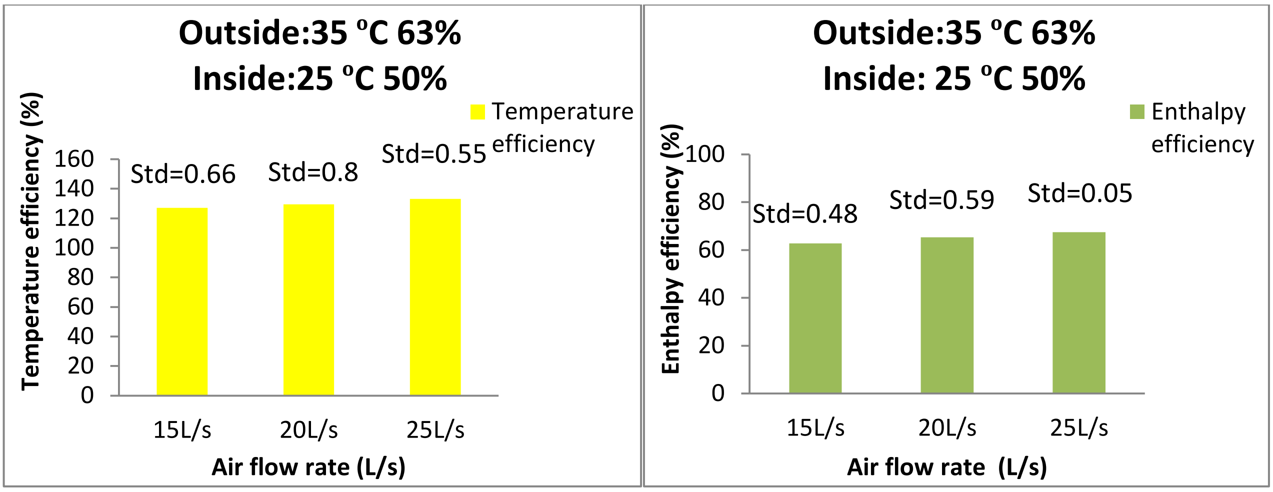


Figure3.4 (1): Temperature efficiency and enthalpy efficiency.

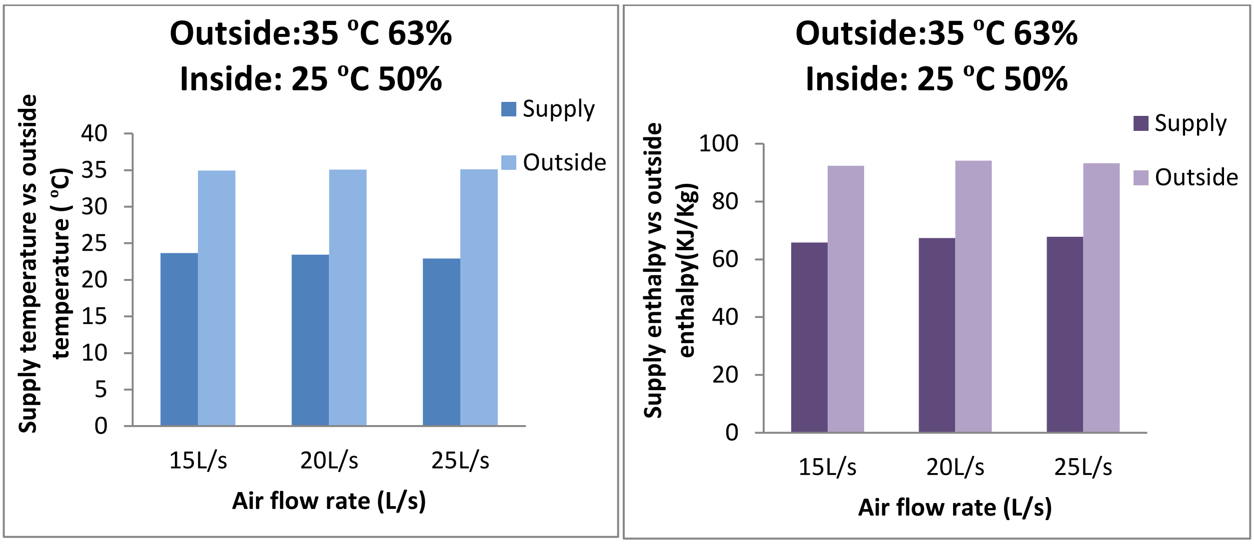


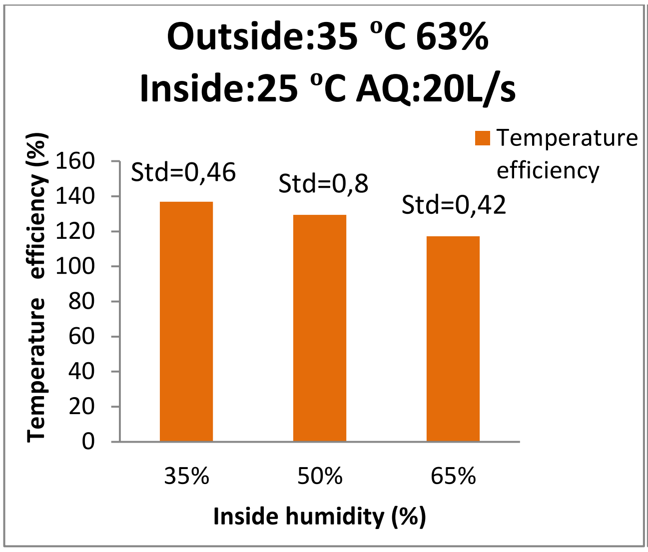
Figure3.4 (2): Supply temperature and supply enthalpy.

3.4.2. Shanghai outdoor condition: 35 ºC with 63%RH.

Shanghai indoor condition: 25 ºC.

Air quantity: 20L/s.

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under designed climate condition of Shanghai. The indoor relative humidity was designed by three levels with medium air quantity condition.



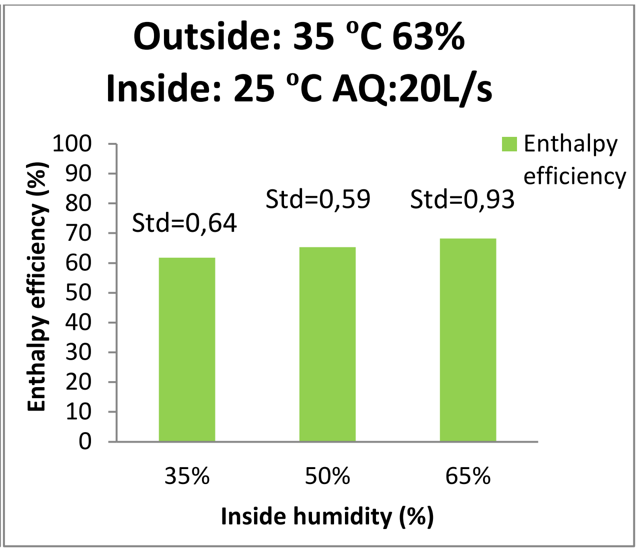


Figure3.4 (3) Temperature efficiency and enthalpy efficiency.

To analyze the relationship between experimental parameters and relative humidity, fig 3.4(3) was made to show the efficiencies values in three levels humidity ratios. The proportion of efficiency in the temperature and enthalpy was followed the different trend. The percentage of data from a steady growth was equal to 125% in temperature result and 65% in enthalpy bar chart respectively by this condition.

The measured data as presented in figures were used to calculate not only the supply temperature and enthalpy, also compared with the outdoor temperature and enthalpy parameters. Analysis tool STATISTICA was used for estimating data deviations which were one of the most important steps of the study, in view of the lack of reliable data on these consequences.

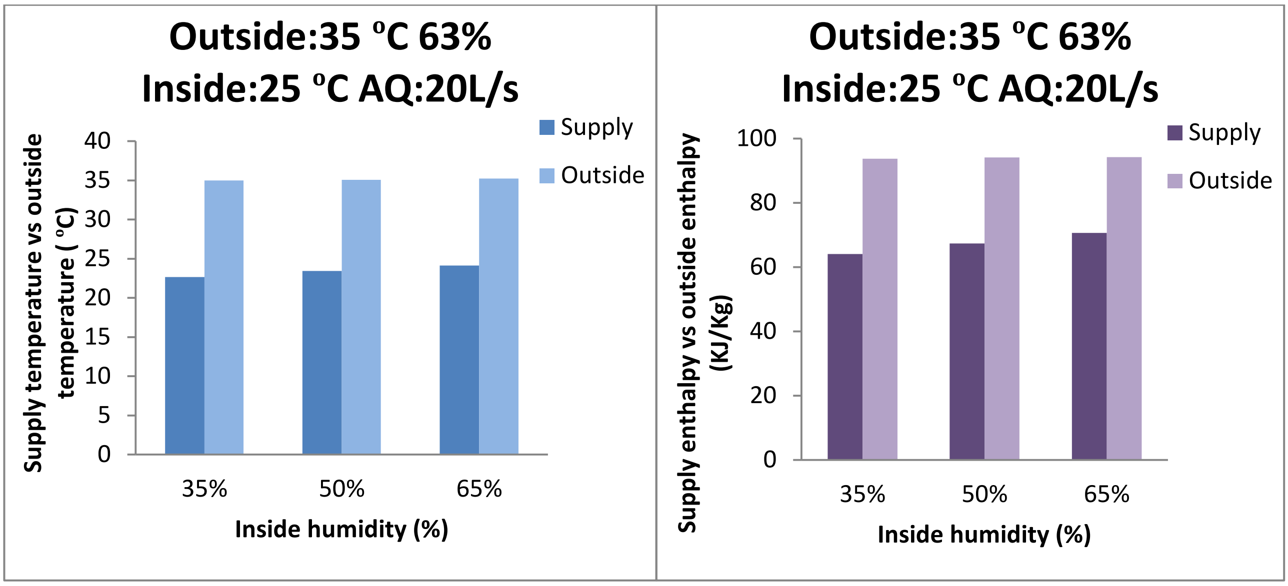


Figure 3.4(4): Supply temperature and supply enthalpy.

Discussion:

Only slight difference was found between the changing of indoor air humidity ratio, either in four bar charts. The first graph showed that there was a gradual decrease in study for temperature efficiency with inside humidity. This percentage gradually declines by approximately 10% for every 15% RH changing. Conversely, the second graph also showed that study stemming from interest increases with relative humidity inside the building. The percentage increases slowly till the maximum value impend 70% for enthalpy efficiency.

With the fully evaporation, thus wet recovery was much. Also, supply air temperature change was not apparent. The ideal condition should be in 50% RH for supply temperature was no higher than 24 ºC.

3.4.3. Shanghai outdoor condition: 35 ºC with 63%RH.

Shanghai indoor condition: 65%RH.

Air quantity: 15L/s.

Measured mean temperature, relative humidity and airflow rate established in the experiment testing the heat recovery unit under designed climate condition of Shanghai. The indoor temperature was designed by two levels with low air quantity condition.

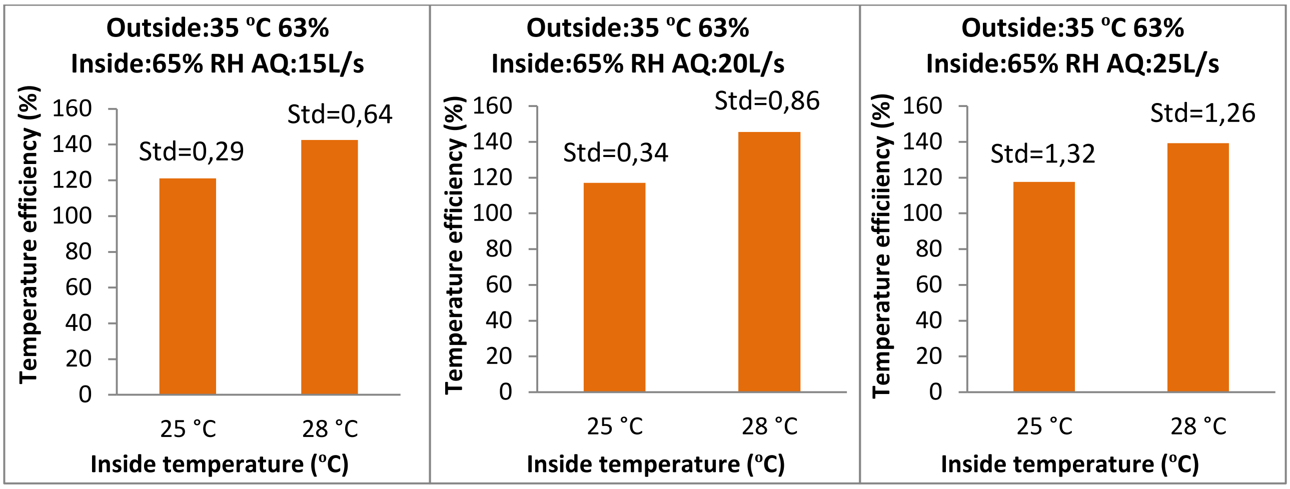


Figure 3.4 (5): Temperature efficiencies.

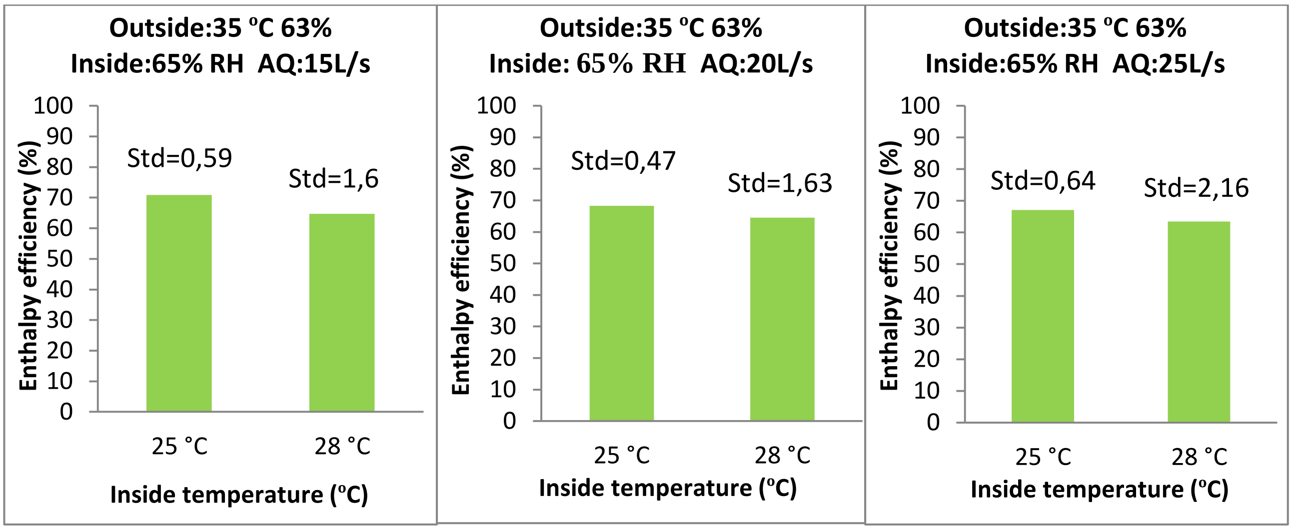


Figure3.4(6): Enthalpy efficiencies.

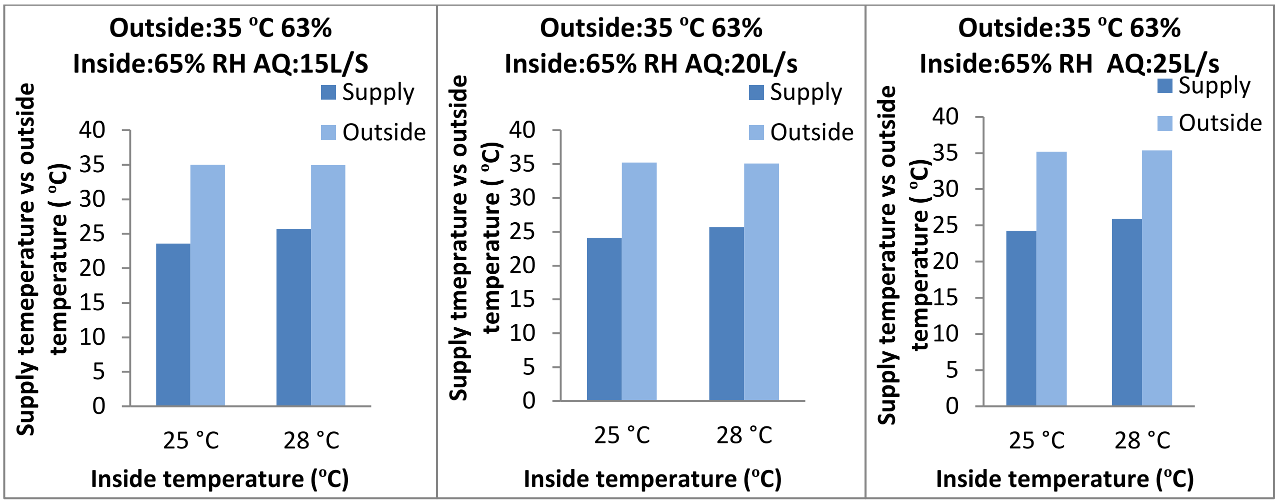


Figure3.4(7): Supply temperatures vs outside temperatures.

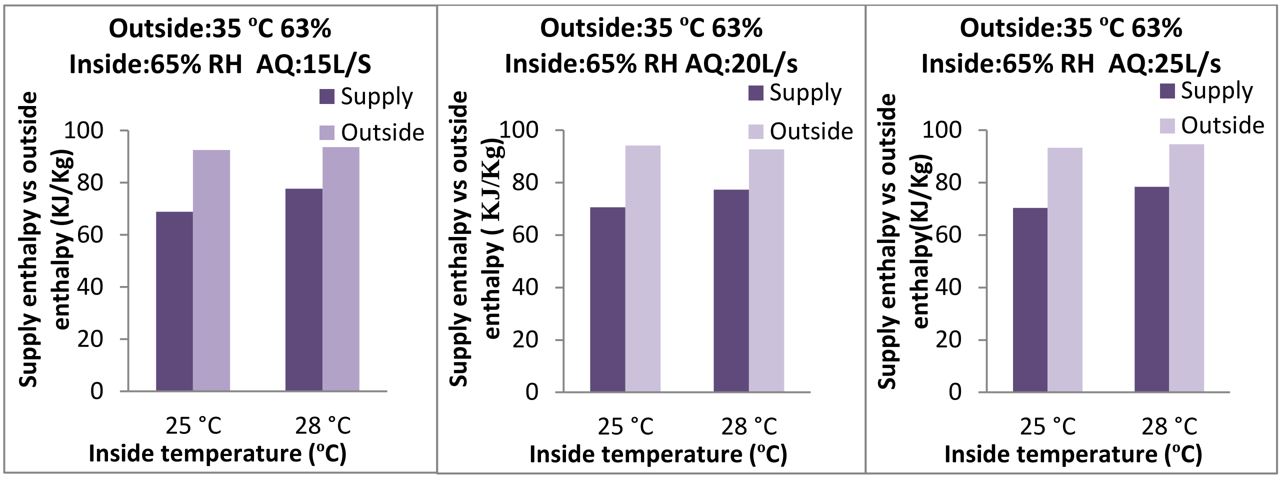


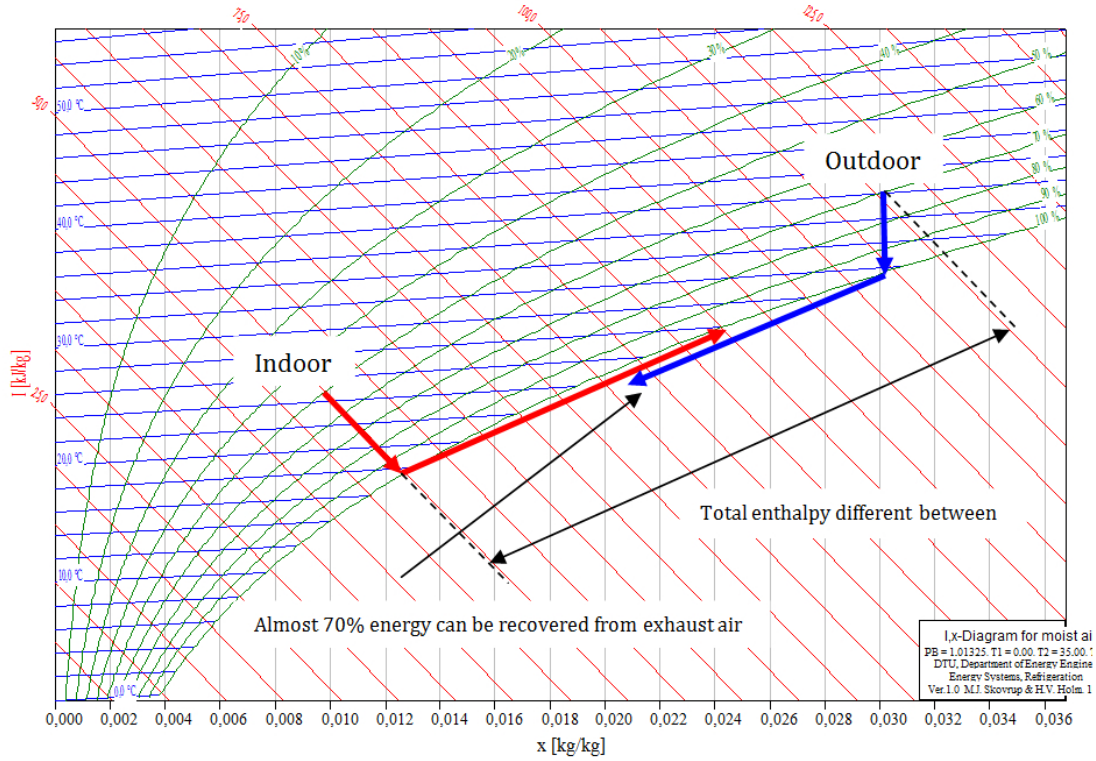
Figure3.4(8): Supply enthalpies vs outside enthalpies.

After summarizing of effects on distinction between indoor relative humidity, temperature of indoor return air was also considered for the other influence factor in this technical experiment. As showed in Fig.3.4(5) and Fig.3.4(6), all cases provided similar trend in each kinds of parameters. Two indoor air conditions had similar relative humidity with different air temperature, while the different values of enthalpy was almost 10 KJ/Kg in this case.

Fig.3.4(5) and Fig.3.4(6), showed the temperature and enthalpy efficiency at only two different indoor conditions (25 ºC and 28 ºC). Overall, most of the results included in the figure had similar tendencies at three levels air quantity during the experiments. Thus, airflow rate was not an important factor for research.

The highest enthalpy efficiency was appeared in 25 ºC for indoor air above 15L/s flow rate. From the perspective of the supply air temperature, 2 degrees diversity was shown in the study of each air volume under the different indoor temperature.

4. Discussion



4.1. Total heat recovery efficiency

Compare with ordinary plate heat exchanger, the efficiency of this technology has obvious improvement, make up the defects of traditional plate heat exchanger which can only recover sensible heat, to complete the process of total heat recovery.

Theoretically, the efficiency of this novel total heat recovery device could be 100 %, whereas the practical efficiency is around 70 % according to the testing (the measuring efficiency may float due to different indoor and outdoor conditions). One of the reasons that 30 % of the energy is not recovered is evaporation not being completely performed.

To a great extent, the evaporating process depends on the diameter of the water drop during the process. Sprinkling water (relatively large water drop) was tested in the first place during the evaporating process when it was found worse than the spraying water with smaller water drop.

Another reason that explains the enthalpy efficiency declines was the water temperature during the evaporating and chilling process. The water temperature might increase due to heat generated by the ultrasonic atomizer circuit, part of which was in the water. The increase of water temperature resulted in reducing the water evaporating.

The best place for water evaporating is close to the inlet of heat exchanger where the heat from inlet air is easier to be absorbed in evaporating process. The other advantage of the spraying water is that the water drop is light enough to avoid free falling and has adequate time to evaporate.

Compared to other refrigeration devices that require refrigerant, this novel device is more environmental friendly and energy saving.

4.2. The influence of water quality

Water quality is a major challenge affected atomization during actual situations.

Due to the limitations of spray equipment, if the evaporative cooling water contains calcium or other similar substance which prone to clogging of the nozzle spouts, over time, the nozzle will be blocked, unable to generate sufficient amount of fog, which decreasing the effectiveness of evaporative cooling, making the latent heat recovery efficiency reduced, total heat recovery outcomes is also declined. Water scale may also accumulates on the surface of the heat exchange plate and reduce the efficiency of heat transfer.

In this experiment, water spray is employed to distilled water, which effectively avoids the influence of impurities from the water spray volume.

In the future practical application of this technology, the water of spray needs to be purified, especially in some areas of poor water quality conditions. Water quality in some areas is relatively soft, perhaps can be used directly.

Purify water is necessary for energy and some chemical reagents. Then, it is necessary in order to calculate the energy consumption which is used for purifying water impurities in the art. Therefore, the method of purifying water is of particular importance. The approach can be also a significant branch of the future development of this technology research.

4.2. The influence of water temperature

As above mentioned, groundwater of Shanghai can be used directly as spray evaporative cooling water during summer condition. The other reason for this usage is relatively lower temperature of Shanghai groundwater. It is beneficial to use spraying water in the heat recovery technology. The lower evaporative cooling water temperature is, the higher heat transfer coefficient can be achieved.

This is requirements of spraying water quality used in total heat recovery technology and future directions of research.

4.3. The influence of ultrasonic atomizer

This experiment used ultrasonic atomizer to humidify the air in order to achieve the purpose of the total heat recovery.

The advantage of the ultrasonic atomizer is that it can produce mist with very fine water droplets. . Compared to nozzle spray atomizer, the total heat recovery using ultrasonic atomizer is more efficient. However, ultrasonic atomizer consumes more power compared to the nozzle spray atomizer. Atomizer using nozzle spray combine with ultrasonic resonator was reported that can produce very fine mist but consumed less energy for the atomizing. However, such technology has not been commercialized. Better atomizing technology is still under development.

The contrast of nozzles and ultrasonic humidification on energy consumption was examined and is reported in the following section.

4.4. Energy consumption of technology

Energy consumption of this device can be subdivided into the fan consumption, water purification, spray ultrasonic, condensate water recovered power and several aspects.

First, the energy consumption of fan is based on the amount of airflow rate. Secondly, depending on the method of water purification and above mentioned requirements for water, it can be calculated the expanding energy of water treatment. Again, the ultrasonic atomizer requires energy supply. Condensate recycling is the additional smaller energy wastage, pending further product development research in the future.

It is worthwhile to mention on energy consumption that parts of heat generate by ultrasonic resonator is input to evaporation water which leading to increase the water temperature. Thereby, it can reduce the latent heat recovery efficiency. The commercially available ultrasonic humidifier combines the ultrasonic resonator with its electronic driving circuit soaked together in water. The heat generated by the driving circuit greatly reduced the efficiency the total heat recovery. The present study found that remove this heat from the water lead to an improvement in the total heat recovery by 3%-5% or more.

Conclusion

Based on the present study, the following points may be concluded.

1. The total heat recovery efficiency of this novel technology can reach 70 %. It is environmental friendly and energy efficient.
2. The supply air temperature after heat recovery can be as low as 18.5 ℃ in this study under various climate conditions. Besides the total heat recovery, the refrigeration can be achieved under some conditions.
3. Atomizing technology and water quality is important for the application of this total heat recovery technique in practice. Further study should be performed to optimize the atomizing technology and investigate the water demineralization technology.

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