TECHNOLOGY ASSESSMENT OF DEDICATED OUTDOOR AIR SYSTEMS FOR AIR-CONDITIONED BUILDINGS IN TROPICAL CLIMATE

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**A THESIS SUBMITTED AS A PART OF THE REQUIREMENTS
FOR THE DEGREE OF MASTER OF ENGINEERING
IN ENERGY TECHNOLOGY AND MANAGEMENT**

**THE JOINT GRADUATE SCHOOL OF ENERGY AND ENVIRONMENT
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ABSTRACT

Ventilation is necessary for good indoor air quality (IAQ) in air-conditioned buildings. However, the ventilation causes a substantial load and the resulting energy consumption of the air-conditioning system, particularly for air-conditioned buildings in tropical region. To address this issue, this paper investigates the dedicated outdoor air system (DOAS) for the energy-efficient ventilation and also for the cost effectiveness. In the study, performance of various DOAS configurations is evaluated for a typical office building, hotel building and department store under the hot and humid climate of Thailand. The results from the simulations using TRNSYS software show that DOAS of the combination of a cooling coil, a run-around coil and an enthalpy wheel is the most energy efficient. It offers the smallest total air-conditioning loads in compliance with Thai's ventilation standard requirements and also provides the most cost effectiveness when the life time is 15 years with interest rate and escalation rate at 7% and 3%, respectively.

Keyword*:* Dedicated outdoor air system, heat recovery, energy simulation, ventilation, life cycle costing

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CHAPTER 1
INTRODUCTION

1.1 Rationale

Air ventilation is essential for air-conditioned buildings. The fresh air is drawn from outside into a building to dilute or remove pollutants that occurred from air-conditioned space such as carbon dioxide (CO_2) emitted form occupants, volatile organic compounds (VOC's) including pathogen generated by natural and human [1]. While the fresh air is drawing into the building, the stale air in the building is exhausting out in a simultaneous time.

Air ventilation must comply with standards to maintain good indoor air quality (IAQ). The worldwide accepted standard (ASHARE standard 62-2001) identifies that "air-conditioning system has to intake the fresh outdoor air into the air-conditioned space 20 CFM to control a relative humidity in the range of 30-65% and to maintain CO_2 concentration not exceed 1,000 ppm [2].

For conventional air-conditioning, fresh outdoor air is combined with return air or "**recirculated air**" before entering into the air handling unit (AHU) to condition the air and then distribute to the desired space which is shown in Figure 1.1.

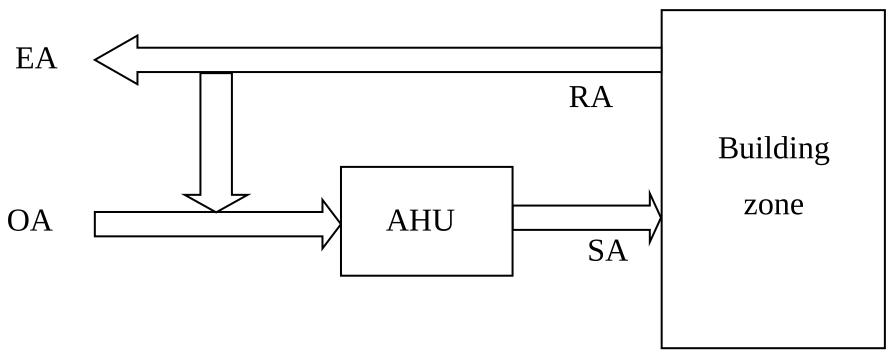


Figure 1.1 The conventional air-conditioning system.

There are currently 2 types of air-conditioning system at the present has for delivering air from AHU: (i) Variable Air Volume (VAV), and (ii) Constant Air Volume (CAV). Both 2 systems can be controlled the indoor air at a desired temperature only (e.g. 25ºC) and have a quite not good ventilation performance. In VAV system, cooling load and demand of air ventilation are not related to each other that make this system require high rates of air ventilation to meet the requirements of the ventilation standards. In CAV system, humidity inside the building is difficult to maintain which may be leaded to undesired condition supporting to growth of allergenic and pathogenic organisms such as bacteria, fungi, cootie and etc. [3]. Especially, in a condition that the sensible heat is low but the latent heat still high. Under this condition, the system has to supply the air with a higher temperature resulting in an ability of moisture decreasing. Moreover, Thailand is situated in a tropical region and has a hot and humid climate over the year hence an intake of the fresh outdoor air to provide a good IAQ causes a substantial load and resulting in energy consumption of the air-conditioning system due to the intake air has a high moisture and temperature or has high enthalpy.

According to energy reports from more than two thousand commercial buildings in Thailand, the air-conditioning is responsible for 50-60% of the total building energy consumption [4]. Energy conservation has undertaken in air-conditioning system to curb the increasing electricity demand of the buildings and the sector as a whole. However, among various measures, it is found that turning-off the ventilation system now become a choice and is implemented in several buildings to reduce the tremendous load from the outdoor air with an expectation of using air infiltration for the ventilation instead. This practice was made with a misunderstanding and no concern on the merit of the air ventilation. A turning-off the ventilation system although can be saved energy but the adverse consequence is a bad IAQ which has a side effect to occupant such as lethargy, headache, lack of concentration, runny nose, dry throat. These symptoms call "Sick Building Syndrome: SBS [1]". As mention above, the drawback of the conventional air-conditioning system can be conclude are as follows; (i) Energy performance (intensive energy used system), (ii) Moisture problem and (iii) Bad IAQ (when turn-off a ventilation system).

A dedicated outdoor air system (DOAS) is designed to eliminate the drawbacks of the conventional air-conditioning system. The concept of using air-conditioning with dedicated make-up air unit has an initiative since 1986 by Gershon Meckler and published more than 30 years [5]. DOAS nowadays is coming with an interest in many researchers such as Stanley Mumma, Kurt Shank, Timothy McDowell and Steven Emmerich that studied the DOAS system and found that the DOAS system can provide good ventilation and energy performance [6, 7].

Under the constraints that the air-conditioning system has for energy performance and ability in ventilation for providing the good IAQ. The analysis of the suitable system is necessary including the versatile technology of DOAS which have different advantage and disadvantage. The difference in technology will be resulted in ventilation and energy performances. This thesis emphasized on the study to identify the air-conditioning and ventilation system that qualified the constraints under the hot and humid climate

1.2 Literature Review

Ayuyuen, S. [8] studied the way to enhance the energy efficiency of an air-conditioning system by using silica gel and a heat exchanger to reduce the enthalpy of fresh air before intake into an air-conditioning system, and analyzed with mathematical equations for calculating this working system for comparison with obtained energy demand data for air-condition system form real office buildings. From the experiment, it was found that the result of the mathematical analysis can reduce the annual energy demand of air-conditioning system from 5,347,481 kWh to 4,129,812 kWh.

Lekkham, C. and Maneewattana, T. [9] studied and compared 9 configurations of DOAS that were simulated by an energy simulation program (EnergyPlus) to obtain annual energy consumption. The simulation result showed that the suitable configurations of DOAS are passive desiccant wheel, a sensible wheel, and a cooling coil which can reduce energy consumption 25-30% when compare with conventional air-conditioning system.

Mumma, S.A. [10] studied an air-conditioning system that integrated DOAS with parallel general air-condition equipment in each space and also compared a conventional air-conditioning system with an air-conditioning system with DOAS. The main point can be summarized and illustrated as: (1) Separation of outdoor air management system from cooling system in each space can provide the suitable ventilation; (2) Conditioning of air via DOAS is able to remove some part of sensible and latent load in the building with cost effective; and (3) Employing radiant ceiling to remove sensible load provide thermal comfort to the occupant.

Mumma, S.A. and Shank K.M. [6] analyzed the annual energy consumption of an air- conditioning system used to manage the outdoor air at the condition of added outside air to system of 4.7 m^3/s, terminal cooling unit, dry bulb temperature of 13^{o}C, and dew point temperature of 7^{o}C.

Air-conditioning can divided by different configurations into 6 types, including: (1) Conventional cooling, heating and humidification; (2) Conventional cooling, heating, humidification arrangement with run around heat recovery; (3) Total energy wheel heat recovery with conventional cooling and reheat; (4) Total energy wheel heat recovery and run-around coil with conventional cooling and reheat; (5) DOAS: Total energy wheel heat recovery, conventional cooling and sensible wheel; and (6) Conventional all-air VAV system. According to the results, DOAS has the lowest annual energy consumption.

McDowell, T.P. and Emmerich, S.J. [7] studied air-conditioning systems with a different configurations equipped with water source heat pumps (WSHP) air-conditioning system for 2-story office buildings located in different climates in 5 states in the USA. Annual energy consumption of each system is analyzed via TRNSYS simulation program and found that DOAS composed of preheat coil, cooling coil, and enthalpy wheel, could reduce annual energy consumption 14-37% and DOAS systems that composed of preheat coil, cooling coil, enthalpy wheel, and sensible wheel could reduce annual energy consumption of 21-38%.

Deng, S., Lua, J. and Jeong, J.W [11] simulated the DOAS that is comprised of enthalpy wheel, cooling coil and preheat coil operating with the AHU in parallel and series under different climates in 7 states in the USA. The simulation results found that the using of DOAS operate parallel with the AHU provided the energy saving 4-10%.

Murphy, J. [12] studied the delivery of supply air form DOAS and can be separated as: i) Delivering conditioned outdoor air directly to the space and ii) Delivering conditioned outdoor air to the local terminal units. From the study found that a delivering of dried and cold air (conditioned outdoor air) directly to the space allowing the local units to be sized for less air flow and less cooling capacity.

1.3 Research Objectives

The main objective of this study is to determine a configuration of dedicated outdoor air systems (DOAS) suitable for ventilation in air-conditioned buildings in tropical hot and humid climate. The DOAS shall provide sufficient outdoor air to achieve the minimum requirements of the indoor air quality according to the local standard of air-conditioning and ventilation system.

The specific objectives in this study are as follows:

- to review configurations and technologies of the DOAS that are currently used in air-conditioned buildings in various locations and climates,
- to evaluate the characteristics and energy performances of the selected DOAS when they are operating under hot and humid climate of tropical regions,

- to evaluate the cost-effectiveness of different DOAS configurations in terms of life cycle cost.

1.4 Scope of Research Work

For this thesis study, the suitable DOAS configuration was identified through the TRNSYS simulation program, which analyzed the energy consumption and cost effectiveness in term of life cycle cost (LCC). The scopes of the research work are as follows:

- The studied ventilation system was DOAS that had a controlling of moisture by cooling to make the moisture in the air condense when the air temperature below the dew point temperature.
- The study of ventilation and IAQ considered the conditioned space in the building only.
- A modeling of building models (office, hotel and department store) by using a TRNSYS simulation program for additional information to facilitate the analysis of a suitable system for each building type.
- The evaluation criteria of the analysis of a suitable system were the system energy consumption and cost effectiveness only.
- The analysis of the system energy consumption was obtained from the systems simulated by the TRNSYS simulation program.
- The analysis of the system performance was studied under the hot and humid climate by using weather data from the metrological area of Bangkok only.

CHAPTER 2
THEORIES

2.1 Weather Data

Bangkok is situated at latitude 13.45ºN longitude 100.31ºE close to the equator in the tropical region which has hot and humid climate throughout the year. From the metrological station of Bangkok, the weather data is hourly recorded in year 2000 can be plotted in the psychometric chart that is shown in Figure 2.1.

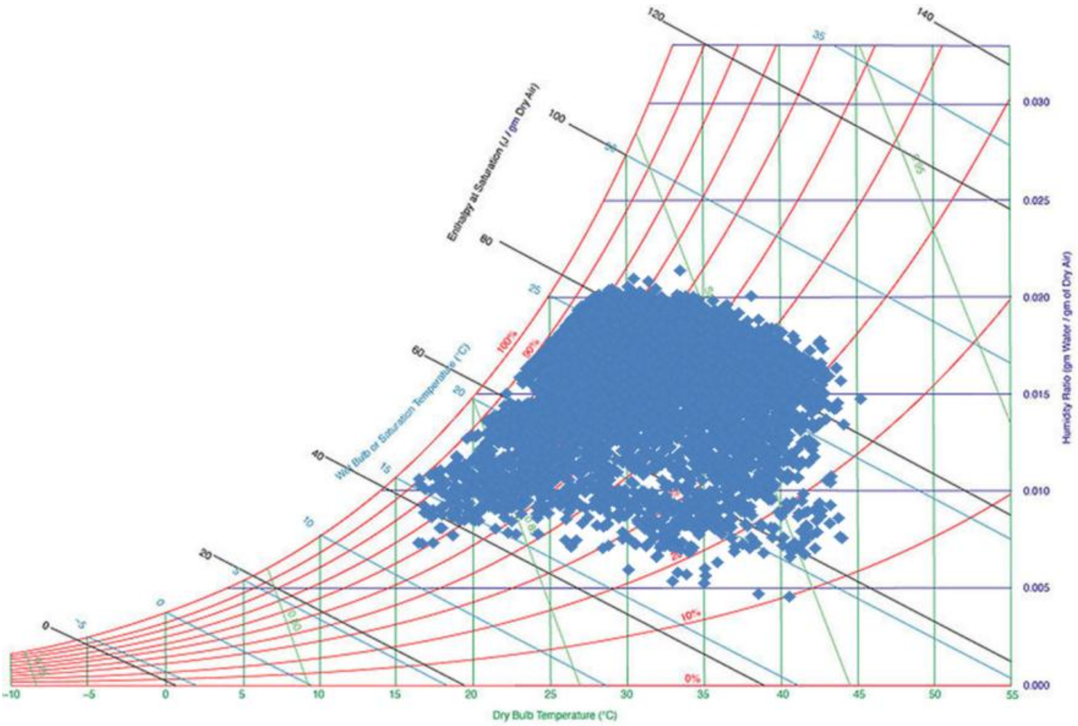


Figure 2.1 Weather data of Bangkok.

Figure 2.1 exhibits a plot of the hourly temperature and humidity of the ambient air recorded at the station. The air has the dry-bulb temperature in a range of 18-40°C and humidity ratio from 8-20 g_w/kg_{da}.

Generally, the weather data can be categorized into 4 periods: (1) cold and dry period, (2) hot and dry period, (3) hot and humid period, and (4) late rain period: cold and humid period. The dry-bulb temperature and humidity ratio record for each period is illustrated in Table 2.1.

Table 2.1 The weather data in each period.

period	Dry-Bulb Temperature				Humidity Ratio			
	Max	Min	Average	SD	Max	Min	Average	SD
Cold and Dry	38.2	16.9	27.5	4.5	19.1	5.4	13.7	2.6
Hot and Dry	40.7	21.3	29.4	4.1	21.6	7.5	17.3	2.1
Hot and Humid	28.8	23.4	29.1	3.5	20.0	13.8	17.6	1.3
Late Rain	39.0	22.8	28.7	3.6	20.5	12.3	17.7	1.2

From the Table 2.1 shows the cold and dry period is the season that has a low dry-bulb temperature and humidity ratio. This period occurs early in the year (1 Jan – 15 Feb) and the end of year (1 Nov – 31 Dec). Hot and dry period is the season that has a dry-bulb temperature and humidity ratio higher than cold and dry period and occur since the mid of February until the end of May (16 Feb – 31 May). This period is the longest period in the year. Hot and humid period is the season that occur since the mid of the year until the mid of August (1 June – 15 Aug). When compared with hot and dry period, a dry-bulb temperature is close to hot and dry period but humidity ratio is higher. For late rain period, this period occur after hot and humid period (16 Aug – 31 Sept). This period has a dry-bulb temperature close to cold and dry period but humidity ratio is higher.

2.2 Designed Indoor Conditions

Conditions inside the building should be designed to achieve thermal comfort and comply with the accepted standard. The accepted standard (ASHARE standard 62-2001) identifies that “inside the air-conditioned space must be controlled relative humidity not exceed 65% for protecting a growth of allergenic and pathogenic organisms that effect to a respiratory system of occupants [2]. For designed indoor condition standard of Thailand is illustrated in Table 2.2

Table 2.2 Designed parameters for any building types [13].

Suggestion of Designed Condition	Dry Bulb Temperature (ºC)	Relative Humidity (%RH)
Office Building	24	55
Department Store Building	25	55
Hotel Building	24	55

Not only thermal comfort but also IAQ should be realized for designing conditioned spaces, especially ventilation rates that are sufficient for maintaining CO_2 concentrations in the space conditioned below 1000 ppm. Engineering experience and field studies indicate that an outdoor air supply of about 20 CFM per person is very likely to provide acceptable perceived IAQ. Therefore, in this thesis define the designed indoor condition at 25°C with ventilation rate 20 CFM for all building types.

2.3 Dedicated Outdoor Air System (DOAS)

Dedicated Outdoor Air System (DOAS) is designed for enhancing the ventilation performance. DOAS is an air-conditioning system that consists of two parallel systems. First system, dedicated make-up air unit which bring the fresh outdoor air into the inside space include manage with sensible and latent heat of fresh outdoor air and occurred latent heat in each space. Other system, cooling unit which manage with occurred sensible heat in each space. This DOAS system is shown in Figure 2.2 and with this system, humidity is easier to maintain and assure that air ventilation also meet the standard.

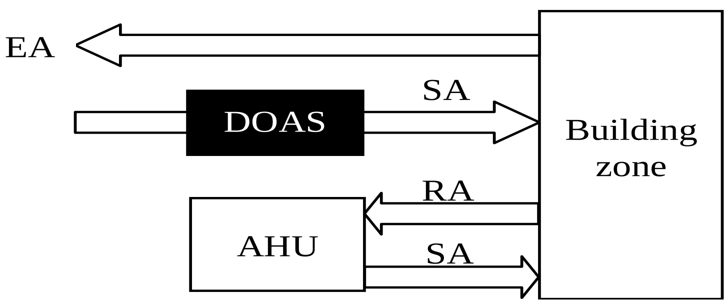


Figure 2.2 Basic arrangement of DOAS.

2.4 Delivering of Outdoor Air

The DOAS can be designed to deliver the conditioned outdoor air through the dedicated make-up air unit as 2 types: (i) Delivering conditioned outdoor air directly to the space, and (ii) Delivering conditioned outdoor air to the local terminal units [12, 14].

2.4.1 Delivering Conditioned Outdoor Air Directly to the Space

Delivering conditioned outdoor air directly to the space is shown in Figure 2.3. This DOAS system has dedicated make-up air unit which has duty to condition fresh outdoor air and directly convey into the conditioned space via duct system while conditioning of air in each space may have a specific device, such as fan coil, blower coil or air split system. The advantages and disadvantages of this approach are listed in Table 2.3 below.

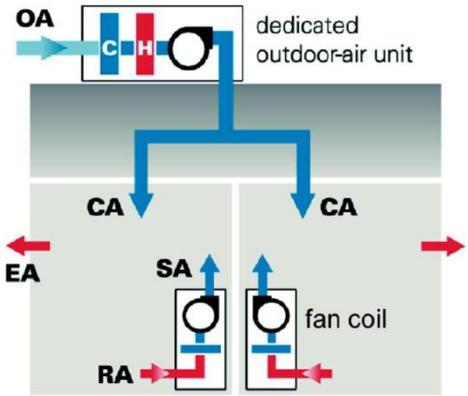


Figure 2.3 Delivering conditioned outdoor air directly to the space [14].

Table 2.3 The advantages and disadvantages of delivering conditioned outdoor air directly to the space [9].

Advantage	Disadvantage
1. Separation of ventilation air and recirculated air can reduce the fan size.	1. Required duct systems for outdoor air and recirculated air that lead to increase in initial costs.
2. Delivering conditioned outdoor air directly to the space can take part of sensible load and latent load of the terminal units allowing the local units to be sized for less cooling capacity.	2. Separation of duct system may be lead to incomplete mixing of conditioned outdoor air with recirculated air.

2.4.2 Delivering Conditioned Outdoor Air to the Local Terminal Unit

Delivering conditioned outdoor air to the local terminal units is shown in Figure 2.4. This DOAS system has dedicated make-up air unit which has duty to condition fresh outdoor air and directly convey to mix with recirculated air before entering into the air conditioner to supply to conditioned space. The advantages and disadvantages of this approach are listed in Table 2.4 below.

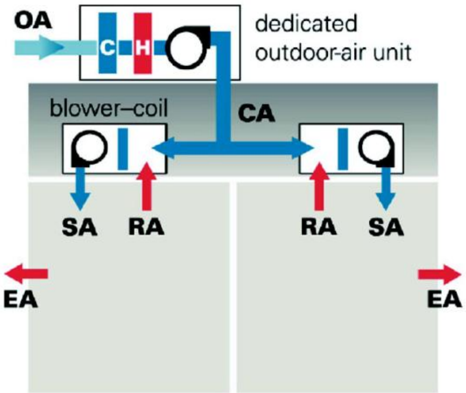


Figure 2.4 Delivering conditioned outdoor air to the local terminal unit [14].

Table 2.4 The advantages and disadvantages of delivering conditioned outdoor air to the local terminal units [14].

Advantage	Disadvantage
1. Duct system can be used together between outdoor air and recirculated air that can reduce initial cost.	1. AHU or air conditioner in specific space has to be active over the time when air ventilation is needed.
2. A combination of outdoor air and recirculated air at AHU or air conditioner in specific space make it easy for both fluids of air completely mix.	2. Lose the benefit from conditioning the outdoor air by dedicated make-up air unit. The dried and cold air is not directly supplied into the space hence the sensible load in the space is unchanged.

2.5 Equipment in DOAS for Providing Energy Savings

The main component of DOAS is a cooling coil that can reduce the dry-bulb temperature and dehumidify the air before delivering into the system by any supply method. For developing the ability of energy saving of DOAS and enhance the system performance lead to an integration of heat recovery device in DOAS configurations [15, 16].

2.5.1 Coil Energy Recovery Loops (Runaround coils)

Coil energy recovery loops shown in Figure 2.5 are composed of finned tube water coils located between the flow directions of supply and exhaust airstreams of the building or the process. The coils are connected in a closed loop that has pipe for counter flowing and contain of heat transfer fluids (typically water or an antifreeze solution) which the pipe is connected to the pump. This system operates for sensible heat recovery only. Energy transfer is seasonally reversible. The supply air is preheated when the outdoor air is cooler than the exhaust air and precooled when the outdoor air is warmer.

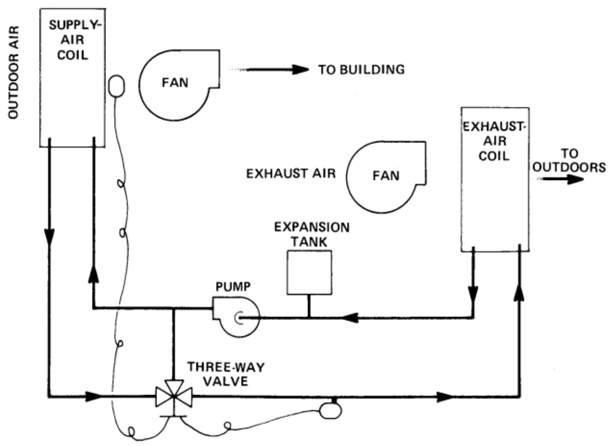


Figure 2.5 Coil energy recovery loops [15].

System Characteristics of Coil Energy Recovery Loops

Coil energy recovery loops have quite good flexibility and proper for industrial applications. The system can simultaneously transfer of energy between several sources and uses. The necessary equipment of this system is expansion tank which allow fluids able to expand or contract. For closed expansion, ethylene glycol is recommended which provide minimum oxidation.

The use of integrated runarounds with variable loads should be combined with the coil energy recovery loops simulation for building energy simulations to achieve maximum benefits. For selection the main component and designing the importance factor such as coils, air velocities, and pressure drops should be done following manufacturer’s design curves and performance data.

Construction Materials of Coil Energy Recovery Loops

The selection of construction material for coil energy recovery loops requires coils that are suitable for environmental and operating condition that are exposed. The important factor for selecting coils is operating temperature, pressure drop, corrosion, and air properties of airstreams. A permanent fin-to-tube bond should be used when operate at above 200°C. Furthermore, the effects of condensable gases and other adverse factors also that may require special coils construction or coating.

Thermal Transfer Fluids for Coil Energy Recovery Loops

Selected thermal transfer fluids for coil energy recovery loops depend on the application and temperature of the airstreams for both the supply side and the exhaust side. Typically, ethylene glycol is used as inhibitor freezing protection of water in solution. Heat transfer fluid manufacturers and chemical suppliers usually recommend appropriate fluids.

Freeze Protection

Moisture in the air often freezes in the exhaust air passage, So that is necessary to install three-way valves to control and maintain the temperature of the solution entering the exhaust coil at -1°C or above to prevent the freezing of the exhaust air and also to ensure that the supplied air temperature from the supply air coil is not too much. Moreover, some of the warmer solution is conveyed for bypassing around the supply air coil to make this condition stable.

Effectiveness of Coil Energy Recovery Loops

Moisture in the air is unable to transfer to other airstreams with coil energy recovery loops. However, cooling load can be reduced by reduction of exhaust air temperature via evaporation. For the best cost-effective operation, typical effectiveness values is interval 45 to 65% at equal airflow rates and no condensation condition, the highest effectiveness is not always provide the greatest net cost saving. The outside air temperature is generally influence the sensible heat effectiveness of a coil energy recovery loop. However, when the cooling capacity is controlled the sensible heat effectiveness will be decreased.

Maintenance of Coil Energy Recovery Loops

Coil energy recovery loops require less maintenance due to having a few moving parts in the system, which are pumps and three-way valve only. However, for operation process at the appropriate condition the air must be filtered before, coils surface regularly clean, the pump and valve maintained, and heat transfer fluids should be replaced or refilled with in the periodic time suggested by heat transfer fluid manufacturers and chemical suppliers

Cross-Contamination of Coil Energy Recovery Loops

The airstreams between the supply side and the exhaust side are completely separated, which leads to elimination of carryover and leakage on cross-contamination problem.

The Advantages and Disadvantages of Coil Energy Recovery Loops

The advantages and disadvantages of coil energy recovery loops or runaround coils are shown in Table 2.5.

Table 2.5 The advantages and disadvantages of coil energy recovery loops [16].

Advantage	Disadvantage
1. Flexible for arrangement in any position.	1. Freezing can occur when water is used as the heat transfer fluid.
2. All functions can be controlled by pump on-off cycling or valve control.	2. To achieve effective fluid flow and heat transfer, air trapped in any equipment, such as coils, pump, and piping must be vented.
3. Cost of equipment is relatively inexpensive.	3. There are water, ethylene glycol or refrigerant costs.

2.5.2 Heat Pipe Heat Exchanger

A heat pipe heat exchanger is a passive device that can recover large amounts of energy. Outside appearance of heat pipe heat exchanger is similar to plate-finned water or steam coil but differ at the tube is not interconnected and separated as evaporator and condenser sections by partition plate, as shown in Figure 2.6.

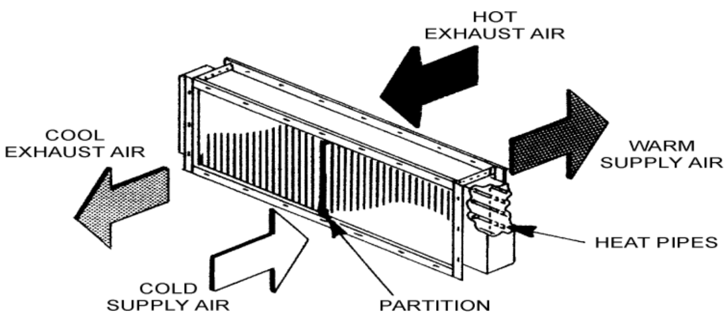


Figure 2.6 Heat pipe heat exchanger [15].

Hot air and cold air pass through the evaporator side and the condenser side of the exchanger respectively. Heat pipe heat exchangers are sensible heat transfer devices, but latent heat can be transferred with condensation on the fins, resulting in making a better performance.

Figure 2.7 illustrates a heat pipe tube that is fabricated with an integral capillary wick structure and permanent sealed, the inside of the tube is evacuated and filled with a suitable working fluid. Temperature requirement of application depend of contained working fluid typically is refrigerant but can be fluorocarbons, water, and other compounds for special temperature.

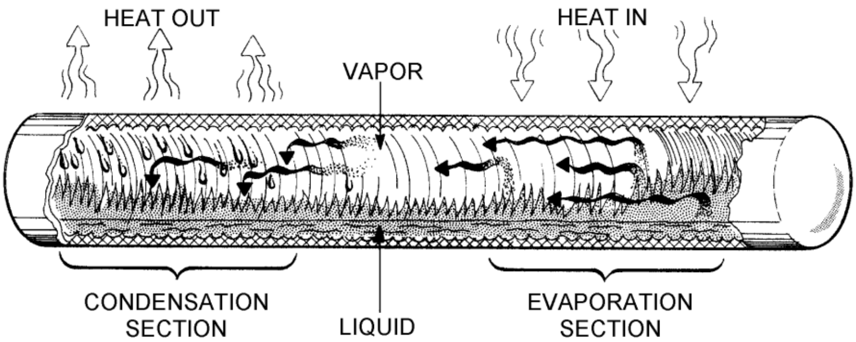


Figure 2.7 Heat pipe tube [15].

Principle Operation of Heat Pipe Heat Exchanger

Hot air enters the exchanger at the evaporation side and vaporizes the working fluid in the heat pipe tube, and then the differential pressure drives the vapor into the condensation side of the exchanger, making the vapor condense and release latent heat. Then, the condensed fluid flows back into the evaporator, where it is revaporized, thus completing the cycle. Therefore, the working fluid is continuously operated in closed loop as long as there is a differential temperature to drive the process.

In actuality, around the surface tube and at the working fluid, there is a small temperature drop. However, energy transferring in heat pipe heat exchanger is considered as isothermal. Heat transfer capacity of heat pipe depends on such factors as wick design, tube diameter, working fluid, and tube orientation.

Construction Materials of Heat Pipe Heat Exchanger

Copper and aluminum are used for making heat pipes and fins specific to aluminum in conventional air-conditioning system. Heat pipe heat exchangers that use at exhaust temperatures below 220°C often use aluminum for making tube and fin include coat with inexpensive protective coating rather than coat with an expensive metal which do not have an effect on thermal performance or less while at the condition exhaust temperatures above 220°C, steel are used for making tube and fin. The fin should be coated with aluminum for preventing oxide.

Operating Temperature Range of Heat Pipe Heat Exchanger

For long-term applications, the selection of the proper working fluid should be considered. The suited working fluid should have high latent heat of vaporization, a high surface tension, and a low liquid viscosity and must be thermally stable over operation period. Moreover, the selected working fluid must not be decomposed at operating temperature if working fluids decompose, noncondensable gases will be occurred and lead to decline heat exchange ability.

Performance of Heat Pipe Heat Exchanger

Design and orientation have an effect on heat pipe heat transfer capacity. Figure 2.8 presents heat pipe heat exchanger effectiveness which show the relationship between face velocities and rows of tubes found that the number of rows of tube is direct proportional to effectiveness and the face velocities are inverse proportional to effectiveness. The effectiveness of a counterflow heat pipe heat exchanger depends on the total number of rows. For example, two units in series has an effectiveness equal to a single unit when has a same total number of rows. However, Series units are often used because it easy for handling, cleaning, and maintenance.

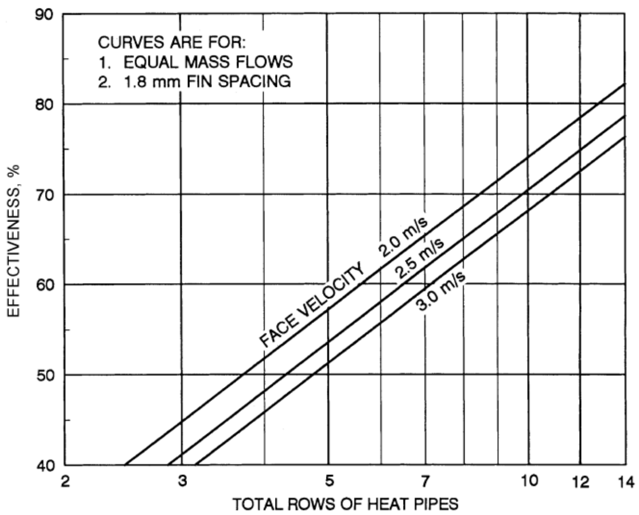


Figure 2.8 Heat pipe heat exchanger effectiveness [15].

The diameter of the pipe has an effect on the heat pipe. Heat transfer capacity increases roughly with the square of the inside diameter of the pipe. For example, when compared a 25 mm inside diameter with the heat transfer capacity of a 25 mm inside diameter more than a 16 mm inside diameter 2.5 times approximately because the diameter of the pipe relate to amount of flowing airstream that make heat can transfer better.

The heat transfer limit of the heat pipe heat exchanger is independent of the length of the tube, excluding very short heat pipes. For example, a 1 m long heat pipe has heat transfer capacity as same as a 2 m long heat pipe when the diameter is identical because the 2 m heat pipe has twice the external heat transfer surface area of the 1 m pipe, it will reach its capacity limit sooner thus, the length of the tube cannot increase heat transfer capacity. Nevertheless, heat transfer capacity can be improved by reconfiguration the tube and addition the number of rows of the tubes but make heat pipe shorter resulting in getting the same surface area but improve the system performance.

Designing of the fin and the spacing inside the heat pipe heat exchanger are based on the dirtiness of the two airstreams that relate to cleaning and maintenance requirements. Therefore, in designing not only acceptable pressure drop should be realized but also facilitate to cleaning and maintenance.

Controlling of Heat Pipe Heat Exchanger

The changing of the heat pipe’s slope can control the amount of heat transfer. The performance of the heat pipe can be regulated by operating on a slope of heat pipe with setting the condensation side below or above the horizontal resulting in improving or retarding the condensed working fluid flow back to the evaporator side respectively.

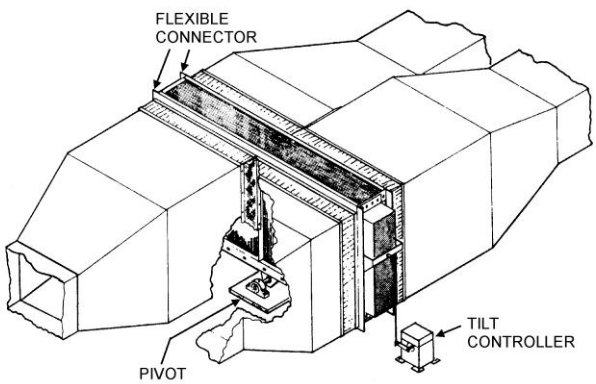


Figure 2.9 Heat pipe heat exchanger with tilt control [15].

Figure 2.9 illustrates a heat pipe with a tilt control in which the practical pivot is installed at the center of the heat pipe, connected with the temperature-controlled actuator at either side of the heat pipe, and attached to the flexible connector plate at the ductwork, allowing free tilting for a maximum of 6°.

The installation of tilt control can provide three features following the list below: (1) to change the operation mode or change the direction of heat transfer when the weather changes, following the seasons; (2) to avoid overheating and maintaining desired temperature of supplied air, which implies performance modulation; and (3) to prevent frost formation when outdoor air temperature is low by reducing effectiveness that makes the exhaust air temperature stay higher than the dew point temperature.

Cross-Contamination of Heat Pipe Heat Exchanger

Cross-contamination generally does not occur with the heat pipe heat exchanger when the differential pressure between the air streams does not exceeding 12 kPa. Cross-contamination can be protected by installing vented partition between the airstreams. In case of an exhaust duct is attached to the partition space, any leakage is released into the space between two ducts and exhaust into outside.

The Advantages and Disadvantages of Heat Pipe Heat Exchangers

The advantages and disadvantages of heat pipe heat exchangers are shown in Table 2.6.

Table 2.6 The advantages and disadvantages of heat pipe heat exchanger [16].

Advantage	Disadvantage
1. There are no moving parts.	1. The coils are more expensive than coil energy recovery loops.
2. Compression tank not required.	2. Difficult to control and expensive.
3. Not sensitive to damage due to freezing conditions.	3. The physical arrangement of the coils must enable gravity drainage of the condensed refrigerant.

2.5.3 Rotary Air to Air Energy Exchanger

Rotary air to air energy exchanger or rotary energy wheel is an energy exchanger device which has a rotatable cylinder filled with an air-permeable medium having a large internal surface area. Total heat (latent heat plus sensible heat) or sensible heat only can be selected to recover by heat transfer media. Rotary Energy Wheel is halved to receive air that use for exchange energy and exhaust air that release to outside and each air stream flow through one-half the exchanger in a counter flow pattern and is shown in Figure 2.10.

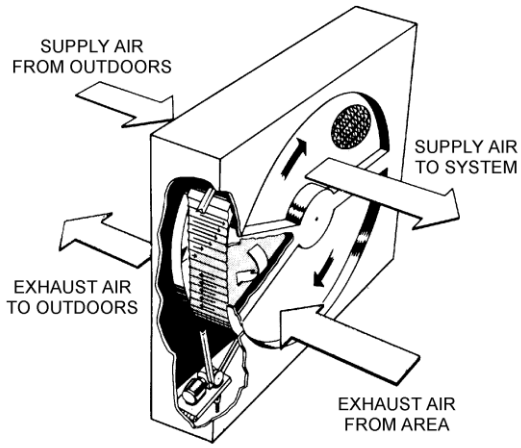


Figure 2.10 Rotary air to air energy exchanger [15].

Sensible heat will be transferred to an energy exchanger's heat transfer media in which heat is transferred from higher temperature airstreams and released to lower temperature airstreams. Latent heat can be transferred via heat transfer media as two forms: (i) Condenses moisture from the higher humidity ratio airstream which release heat (occurred because the medium temperature is below its dew point), and (ii) Releasing the moisture through evaporation into the lower humidity ratio airstream, moist air therefore is dried while drier air is humidified. In total heat transfer, both of sensible and latent heat transfer occur simultaneously due to rotary air to air energy exchanger as counterflow airstream and has a small diameter that make this device required less area but can provide high transfer effectiveness.

Construction of Rotary Air to Air Energy Exchanger

Clinical factors that influence the choosing of materials for the casing, rotor structure, and medium of a rotary energy exchanger are supply air properties, exhaust air temperature, dew point, and air contaminants. The usual structures are aluminum, steel, and also polymers for main structure, casing, and rotary part. Heat transfer media fabricated from metal, mineral, and synthesis material which can provide a good random flow or directionally oriented flow.

Random Flow Media

Random flow media is fabricated by knitting wire into an open woven cloth or corrugated mesh to get a desired layer configuration. Aluminum mesh is packed into pie-shaped which commonly used for comfort ventilation systems. For high-temperature and corrosive applications, Stainless steel mesh is proper. When compared random flow media with directionally oriented media, random flow media require a larger surface area to get desired airflow and pressure drop. This media type should only be used with filtered airstreams because they plug easily.

Directionally Oriented Media

Directionally-oriented media have several geometric configurations. The most common is composed of small (1.5 to 2 mm) air passages which is located parallel with airflow's direction. The shape is not influence a performance of air passages, triangular, hexagonal, or other are similar performance. For low temperature application, aluminum foil, paper, plastic, and synthesis materials are commonly used while stainless steel and ceramics are used for high temperatures and corrosive conditions.

Media surface areas exposed to airflow depend on the type of medium and physical configuration. The areas normally vary from 300 to over 3300 m^2/m^3. Media can be classified according to their ability that is sensible heat transferable or total heat transferable. Sensible heat transferable media are common made form aluminum, copper, and stainless steel while total heat transferable media are common made from various materials and treated with a desiccant such as zeolites, molecular sieves, silica gels, activated alumina, titanium silicate, synthetic polymers, lithium chloride and to each types have a different moisture recovery characteristics that are shown in Figure 2.11.

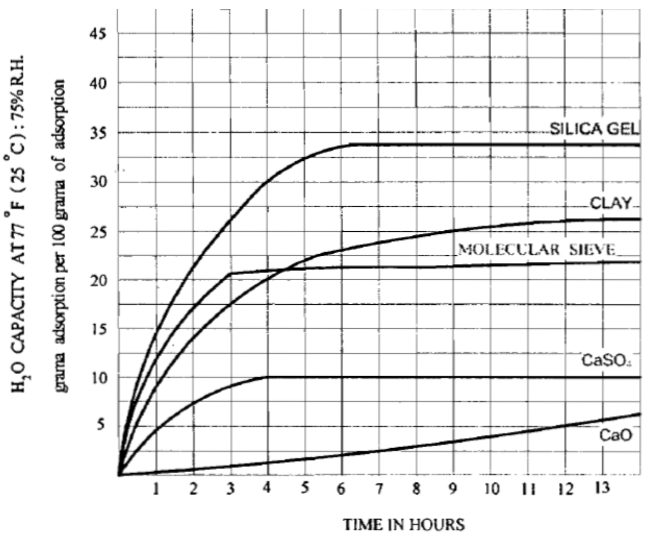


Figure 2.11 Moisture recovery characteristic of any desiccant [8].

Controlling of Rotary Air to Air Energy Exchanger

There are two common methods to control the rotary air to air energy exchanger. The first, controlling of air that used for exchange energy which called “supply air bypass control”, the amount of supply air that pass through the wheel is measured discharge temperature by installing sensor and amount of air is regulated to get the desire temperature. The other one method is regulation of the energy recovery rate rotary air to air energy exchanger by adjusting the speed of wheel revolving. Energy transfer ability is directly proportional to the wheel rotational speed, when the wheel rotational speed is high the energy transfer ability is increase.

Maintenance of Rotary Air to Air Energy Exchanger

Maintenance should be strictly done following the manufacturer’s recommendations to achieve the best performance. The maintenance procedures are: (1) The medium should be cleaned following the manufacturer’s instructions when strain, dust and other material are adhered at the medium; (2) Speed control motor should be frequently inspected and maintained more than other motor especially brushes should be replaced and also a commutator should be periodically turned on-off; (3) Regularly inspect belt and chain are available or unavailable; and (4) Some equipment part should be spared and replaced according to the manufacturer’s recommendations.

Cross-Contamination

Cross-contamination normally occurs in all rotary air to air energy exchangers or other means of mixing air between air used for exchange energy and exhaust air released to outdoors of rotary air to air energy exchangers. This is occurs by two mechanisms, carryover and leakage.

When air is flowed into the rotary medium and mixed with the other sides airstream of medium that is carryover mechanism while leakage occurred by differential pressure between airstream in intake side and exhaust side, air at high pressure side flow to low pressure side. For leakage, Increasing of exhaust air by installing blower can be reduced cross-contamination. For carryover problem, can be improved and reduced to less than 0.1% by installing purge section on the exchanger.

Carryover of rotary air to air energy exchanger is directly proportional to the wheel rotational speed and the porous volume of the medium when uninstalling a purge section. For installing blower, usually is located at the exit of the exchanger, should be realized leakage, purge, and carryover airflows factor to define size of blower.

Advantages and Disadvantages of Rotary Air to Air Energy Exchanger

The advantages and disadvantages of rotary air to air energy exchangers are shown in Table 2.3.

Table 2.7 The advantages and disadvantages of rotary air to air energy exchanger [16].

Advantage	Disadvantage
1. There is no water or refrigerant charge.	1. Required high maintenance for some driving systems.
2. Ease in controlling process.	2. The medium in the rotor is sensitive to clogging.
3. There is no freezing due to heat-transfer fluid.	3. Effective air quality and performance can be reduced by leakage between airstreams.

2.6 The Workings of Cooling Coils in DOAS

When investigating the DOAS system for the space that is designed, the dry-bulb temperature and relative humidity ratio are at 25ºC and 55%RH, respectively (office building) according to the air-conditioning and ventilation standard of Thailand. In the case of deliver the outdoor air directly into the space with the supply air temperature at designed indoor condition. It found that the weather data of Bangkok can be separated as 4 regions, as shown in Figure 2.12.

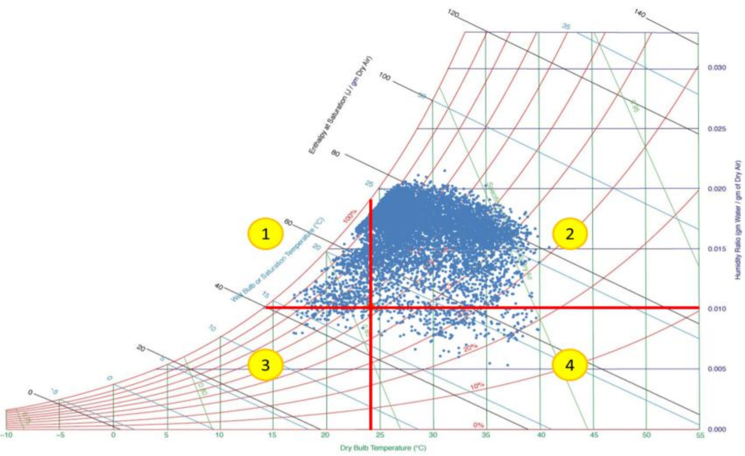


Figure 2.12 The weather data of Bangkok separated by process of air conditioning.

Figure 2.12 shows that the weather data can be segregated by basic process of air conditioning as follows: Region 1 Heating and dehumidify, Region 2 Cooling and dehumidify, Region 3 Heating and humidify, and Region 4 Cooling and humidify.

It should be noted that most of the processes are cooling and dehumidify. Therefore, the main component of DOAS for Thailand is cooling coil as previous mention.

CHAPTER 3
METHODOLOGY

This chapter describes the working plan and methodology of the thesis study to obtain the suitable internal configuration of DOAS under the hot and humid climate of Thailand. The working plan of this thesis can be separated into 4 main parts: (1) defining of building models, (2) selection of internal DOAS configuration, (3) system simulation by using TRNSYS simulation program, and (4) life cycle costing for evaluation of cost effectiveness.

3.1 Defining of Building Models

The building models in this study are composed of 3 building model types, which are office, hotel and department store. All types of the building models that are created for simulation have the same shape and dimension, width 40 m, length 40 m and height 4 m which this building model is shown in Figure 3.1. However, the parameters that reflect to characteristics of any building model types are differ as the Table 3.1. For designing of any parameters of each building model such as overall thermal transfer value (OTTV), roof thermal transfer value (RTTV) and etc. These parameters are designed according to the standard of Thailand which is called “Building Energy Code of Thailand” [17] and are illustrated in Table 3.1.

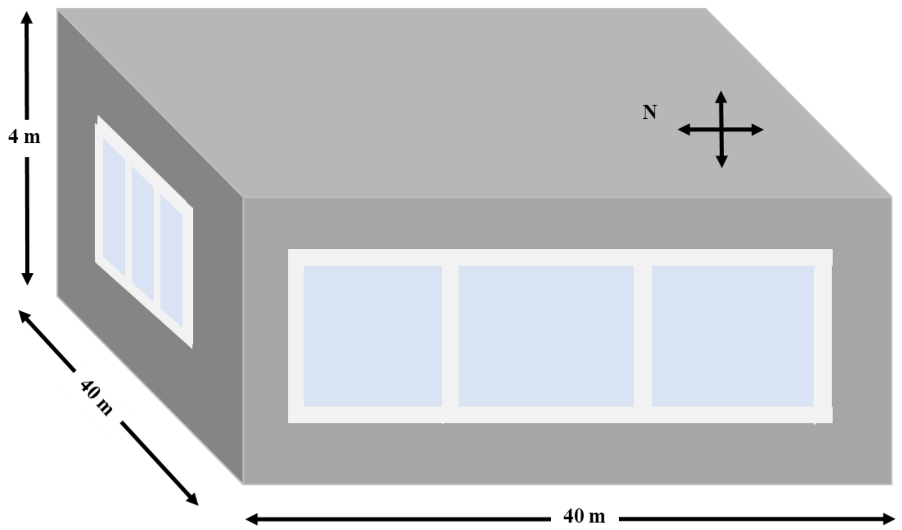


Figure 3.1 The building model.

Table 3.1 Parameters of building model types.

Parameters	Building Type		
	Office Building	Department Store Building	Hotel Building
Opaque Wall			
U-value of roof ($W/m^2.K$)	0.589		
U-value of Wall ($W/m^2.K$)	2.965		
U-value of floor ($W/m^2.K$)	3.448		
Glazing Window			
Window to wall ratio (WWR)	0.35	0.15	0.35
U-value of window ($W/m^2.K$)	5.73		
SHGC	0.53		
Ventilation Air			
Ventilation air flow rate (CFM/person)	20		
Building's load			
Occupant density (person/100 m^2)	10	30	10
Sensible load in from occupant (W/person)	75	90	65
Latent load in from occupant (W/person)	75	95	55
Light power density (W/m^2)	10	19	13
Equipment power density (W/m^2)	12	17	14

a) Office Building

The level of use of this building type quite stable and is usually used from 8.00 am to 6.00 pm. The using time of this building type can be defined as 2 period times; i) weekday and ii) weekend. The occupant density, level of lighting usage and level of equipment usage vary with time of the in any period that illustrate in Figures 3.2- 3.4. Cooling load from any factors will be had value follows the parameters in Table 3.1.

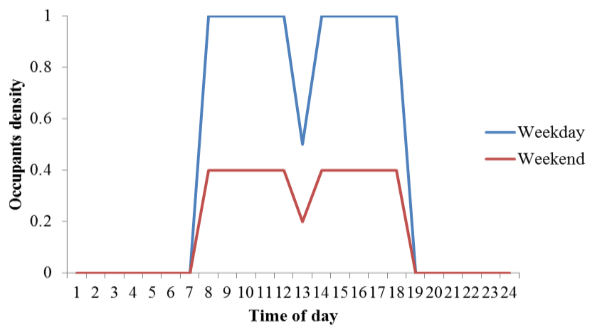


Figure 3.2 The occupant density of office building.

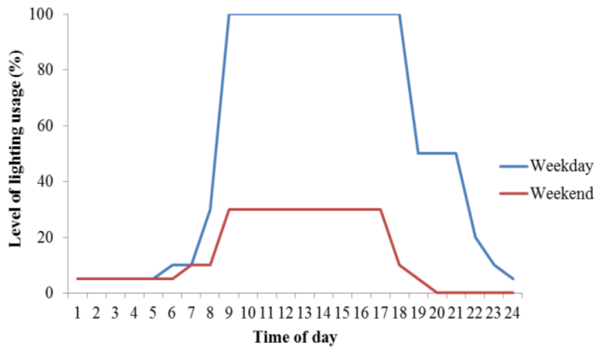


Figure 3.3 The level of lighting usage of office building.

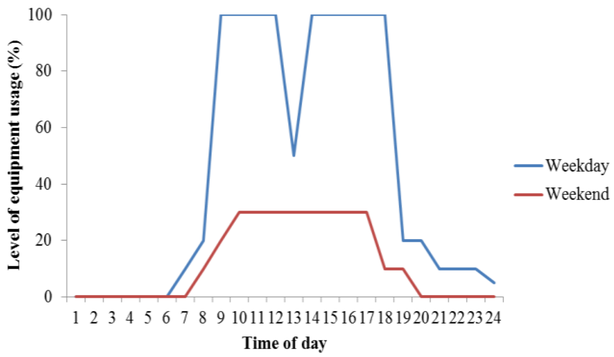


Figure 3.4 The level of equipment usage of office building.

b) Hotel Building

The time of usage of this building type can be defined as 2 periods: i) weekdays and ii) weekends. The occupant density, level of lighting usage and level of equipment usage vary with time of the in any period that illustrate in Figures 3.5- 3.7. The using of this building type usually uses at the night time and useless in a daytime. Cooling load from any factors will be had value follows the parameters in Table 3.1.

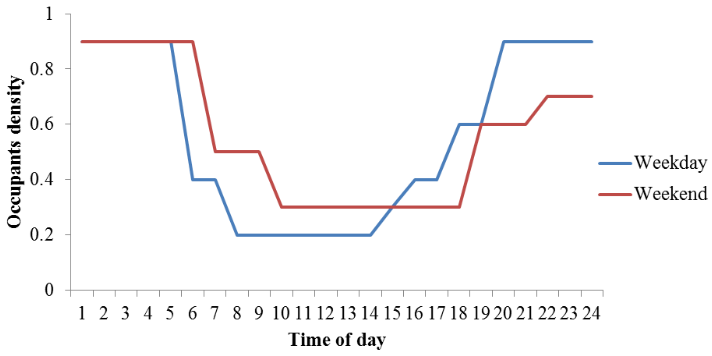


Figure 3.5 The occupant density of hotel building.

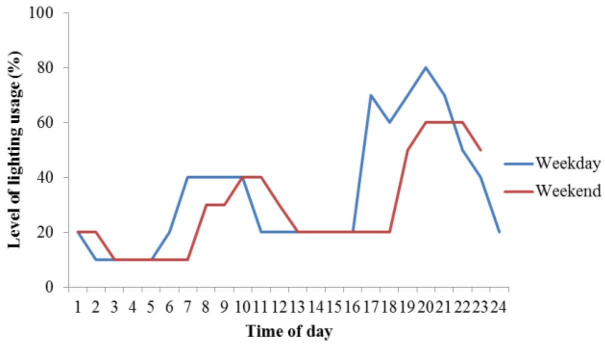


Figure 3.6 The level of lighting usage of hotel building.

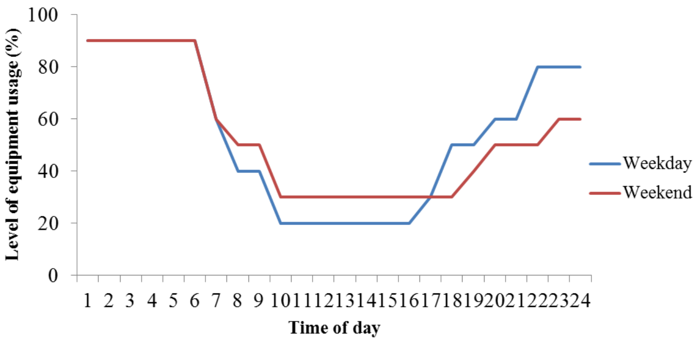


Figure 3.7 The level of equipment usage of hotel building.

c) Department Store Building

Although, the time of usage of this building type can be defined in the same way as for office building type, which are weekdays and weekends, for department store building, the occupant density is quite crowded in the evenings of weekdays and always crowded over weekends. The level of lighting usage and level of equipment is illustrated in Figures 3.8- 3.10. Cooling load from any factors will be had value follows the parameters in Table 3.1.

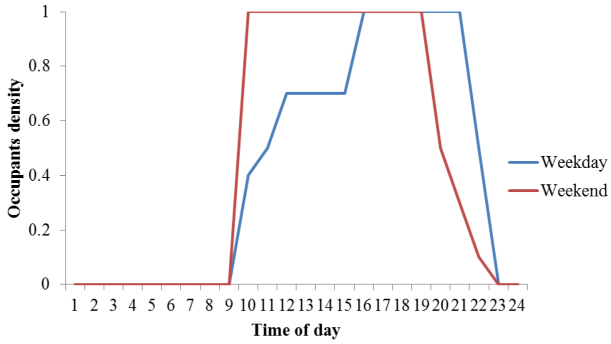


Figure 3.8 The occupant density of department store building.

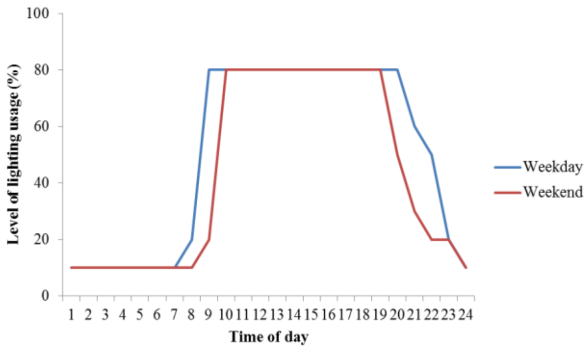


Figure 3.9 The level of lighting usage of department store building.

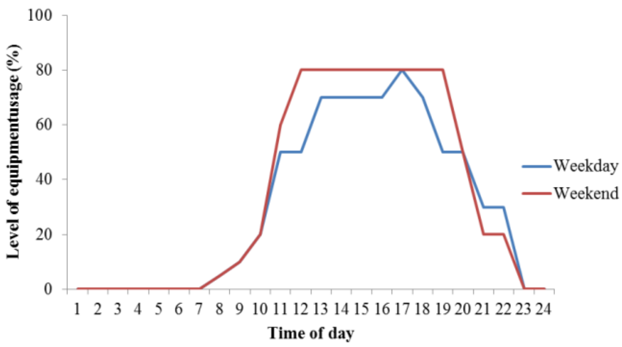


Figure 3.10 The level of equipment usage of department store building**.**

3.2 Selection of Internal DOAS Configuration

This study, considers 5 DOAS configurations. A designing of DOAS can be designed by an integration of heat recovery equipment such as rotary enthalpy wheel, rotary sensible wheel and runaround coil heating with the main component that is cooling coil as mentioned in Section 2.6. The DOAS possess their different characteristics and performances. The 5 DOAS configurations have the principle operations as follows.

a) Configuration 1: Cooling coil unit (FC)

This configuration is the simplest system, which is comprised of only a single unit of cooling coil. The outdoor air will passes through the cooling coil and is cooled and dehumidified to reach the desired condition. Figure 3.11 shows this DOAS configuration mentioned.

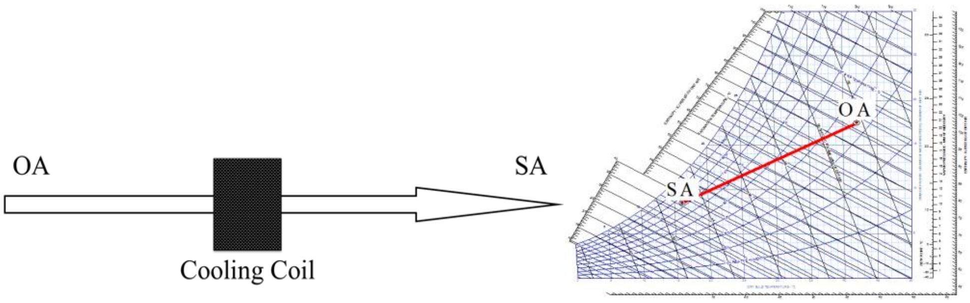


Figure 3.11 Cooling coil unit.

b) Configuration 2: Cooling coil with heat recovery runaround coil (RC)

Apart from a cooling coil unit, this configuration equips runaround coils or heat pipe with the cooling coils. The outdoor air is precooled by the first runaround coil before it enters the cooling coil. The air then passed through the second runaround coil for reheating purpose. Figure 3.12 illustrates this DOAS configuration.

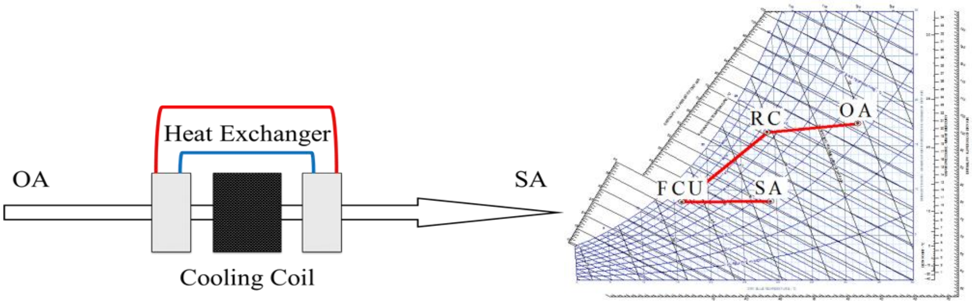


Figure 3.12 Cooling coil integrated with heat recovery runaround coil.

c) Configuration 3: Cooling coils with rotary energy wheel (EW)

This configuration is comprised of a cooling coil and a rotary air to air energy exchanger. The rotary energy wheel is used to transfer the cool air in the exhaust air to precool the fresh air intake before entering to the cooling coil unit. This technique can help reduce the cooling energy demand at the cooling coil. This configuration is shown in Fig. 3.13.

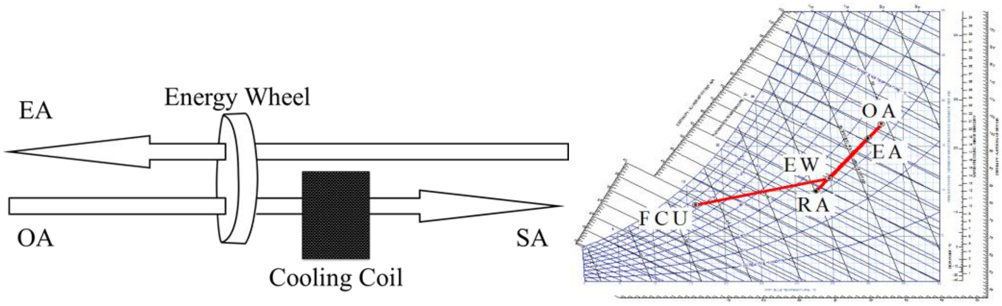


Figure 3.13 Cooling coil integrated with rotary energy wheel.

d) **Configuration 4: Cooling coils with rotary energy wheel and sensible wheel (EW+SW)**

This DOAS configuration is developed by integrating a rotary sensible wheel with the DOAS configuration 3. Figure 3.14 illustrates the arrangement of the equipment.

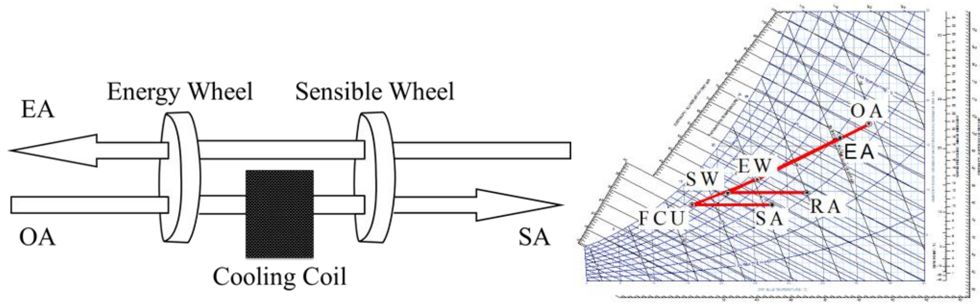


Figure 3.14 Cooling coil integrated with rotary energy wheel and sensible wheel**.**

e) **Configuration 5: Cooling coil with runaround coil and rotary energy wheel (EW+RC)**

This is the combination of the DOAS configurations 2 and 3. Rotary energy wheel together with the runaround coil are used to enhance the system performance. Figure 3.15 shows the arrangement of the equipment of this configuration.

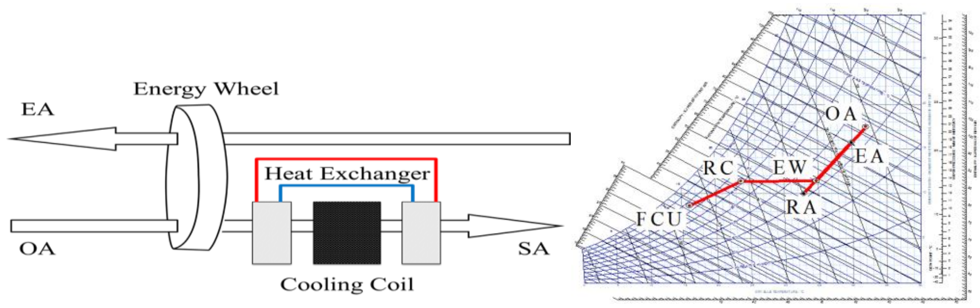


Figure 3.15 Cooling coil integrated with runaround coil and rotary energy wheel.

3.3 System Simulation Using TRNSYS Simulation Program

The program used to analyze the air-conditioning and ventilation systems is the TRNSYS simulation program. The full name is TRaNsient SYStem Simulation Program. This program is the useful software that worldwide used for any kind of system simulation. TRaNsient SYStem Simulation Program was developed by university of Wisconsin which is the agency studied in particular energy research. Functions in TRNSYS can be categorized by working characteristic as 2 parts.

Part 1 TRNBuil Program is a sub-program of TRNSYS used for modeling a house or a building. The defined parameters can be modified over the time. Moreover, this function can be completely presented the any desires parameters such as room temperature and thermal comfort index (PMV).

Part 2 TRNSYS Simulation Studio is the main program used for creating and simulating of any system. TRNSYS Simulation Studio has a standard component model for several systems, such as heating system, lighting system, wind energy system, including air-conditioning and ventilation, which are related to this study.

The DOAS configurations that are designed as mentioned in Section 3.2 will be equipped with any of the building model types mentioned in Section 3.1 in parallel by TRNSYS simulation program. The building characteristic parameters include weather data of Bangkok over the year as mentioned in Section 2.1 will be used to facilitate the analysis to obtain the annual energy consumption and the overall system performance including the parameter at any condition such as inlet temperature, outlet temperature of AHU, DOAS, humidity ratio at any position and etc. the standard component that were used for simulation of system in this study are shown in Table 3.2

Table 3.2 The TRNSYS modules used in the simulation [18].

Device	TRNSYS module	Description
Cooling coil	Type 752	This component models a simple cooling coil where the air is cooled as it passes across a coil containing a working fluid (typically water or a refrigerant).
Run-around coil	Type 689	This component models a run-around coil for air applications. A run-around coil is a device that

		transfers sensible heat from an inlet air stream to a secondary air stream. In many applications the secondary air stream is the original inlet.
Enthalpy wheel	Type 667	This component models an air to air heat recovery device in which two air streams are passed near each other so that both energy and possibly moisture may be transferred between the streams.
Sensible wheel	Type 760	This component models an air to air heat recovery device in which two air streams are passed near each other but only sensible heat can be transferred between the streams.
Building	Type 56	This component models the thermal behavior of a building having at least one thermal zone.
Weather	Type 99	This component serves the main purpose of reading weather data at regular time intervals from a data file and processing the solar radiation data.

3.4 Life Cycle Costing for Evaluation of Cost Effectiveness

Life cycle costs (LCC) are cradle-grave costs summarized as an economic model of evaluating the alternatives for equipment and projects. The LCC economic model provides good economic business for hood engineering proposals.

The definition of LCC according to the National Institute of Standard and Technology (NIST) Handbook 135, 1995 edition is: “The total discounted dollar cost of owning, operating, maintaining and disposing of a building or building system” over a period of time, or simply to defined as LCC being the summations of total cost estimates from inception to disposal for both equipment and projects. The time value of money, depreciation and taxed must be included in LCC calculation therefore the summarized LCC would result in the net present value (NPV) or annual worth (AW) formats. Life cycle cost analysis (LCCA) is an economic evaluation technique that determines the total cost or annual cost of owning and operating a facility over period of time.

Life cycle cost analysis (LCCA) is required to demonstrate that operational savings are sufficient to justify the investment costs. The usefulness of LCCA is not to determine a total cost of a project alternative but to determine the ability to compare the cost of project alternatives and to decide which alternative result the best value per money unit spent.

The implementation of LCCA can be applied at any level of the design process and can also be an effective tool for existing building system evaluation. The LCC equation has three variables which are relevant cost of ownership. The period of time over the relevant cost and the discount rate that is used to convert the future costs into present day costs [19, 20, 21].

Terminology and Basic Economic Principles for LCCA

a) Costs

- Initial investment costs are all costs before the occupation of the facility.
- Operational costs are annual costs, excluding maintenance and repair costs, involved in the operation of the facility. Most of these costs are related to building utilities and custodial services.
- Maintenance costs are scheduled costs associated with the upkeep of the facility.
- Repair costs are unanticipated expenditures that are required to prolong the life of a building system without replacing the system.
- Replacement costs are anticipated expenditures for major building system components required to maintain the operation of a facility.
- Residual value is the net value or net worth of a building at the end of the LCCA study period, which can be positive or negative depending on it costs or it values.

b) Time

- A study period is the period of time over which all expenses, including ownership and operations, are evaluated. The study period can be separated into two phases: the planning/construction period and the service period.

c) Discount Rate

- The discount rate is the interest rate a central bank charges depository institutions that borrow reserves from it, which reflecting the investor’s time value of money.
- There two steps for the discount rate. The first one is the real discount rates which excludes the rate of escalation. The second one is the nominal discount rate which includes the rate of escalation.

e) Cost Escalation

- The changes in the costs or prices of specific goods or services in a given economy over a period. This is similar to the concepts of inflation and deflation except that escalation is specific to an item or class of items (not as general in nature), it is often not primarily driven by changes in the money supply, and it tends to be less sustained. While escalation includes general inflation related to the money supply, it is also driven by changes in technology, practices, and particularly supply-demand imbalances that are specific to a good or service in a given economy.

d) Time Value of Money

- Present value is the value of a given data of a future payment or series of future payments discounted to reflect the time value of money and other factors, such as investment risk. It is the time-equipment value past, present or future cash flows of the beginning of the base year.
- To calculate the present value, the parameters as mentioned above are involved: recurring costs which are costs that occur every year over the time span of the study period, discount rate, escalation rate and the period of time project.
- To calculate the present value of future recurring costs, accounting for both the discount and escalation rates, the following formula is applied:

$$PV = \mathrm{A}\left(\frac{1+e^n}{d-e}\right)\left(1-\frac{(1+e)^n}{(1+d)^n}\right) \qquad (3.1)$$

- To distribute the present value to be an annual value accounting for both the discount and escalation rates, the following formula is applied:

$$A = \mathrm{PV}\left(\frac{d(1+d)^n}{(1+d)^n-1}\right) \qquad (3.2)$$

where PV = present value, A = amount of annual cost, d = real time discount rate, e = escalation rate, and n = time expressed as number of years.

This study defines the parameters involved in the evaluation of cost effectiveness according to Table 3.3 below.

Table 3.3 The parameters for evaluation of cost effectiveness.

Parameter	Value
1. Project time period	15 years
2. Discount rate	7%
3. Cost escalation rate	3%

The criteria for the evaluation of the cost effectiveness of the alternative choices are the consideration of the smallest payment choice over the time period of a project.

CHAPTER 4

RESULTS AND DISCUSSIONS

This chapter reports the simulation results of the five DOAS configurations when they are used with three commercial building types i.e. office, hotel, and department store, under the hot and humid climate of Thailand. The analysis is made in comparison with the conventional all-air system.

4.1 DOASs with the Office Building

4.1.1 Cooling load

Annual simulations of the DOASs were performed to determine the hourly cooling load and the indoor condition of the air-conditioned space. The results of a hot and humid day are chosen for discussions. Figure 4.1 shows the solar radiation and ambient air condition on the selected day of July, 4th.

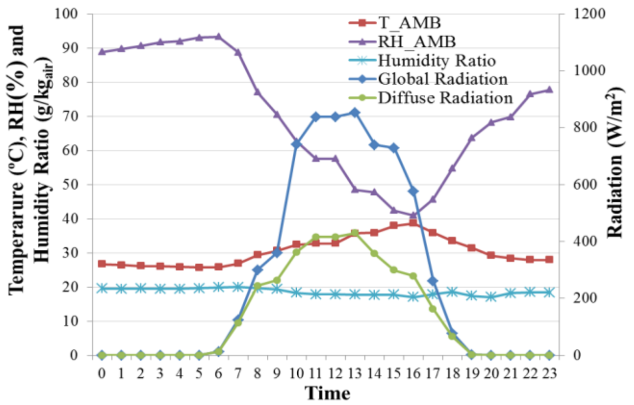
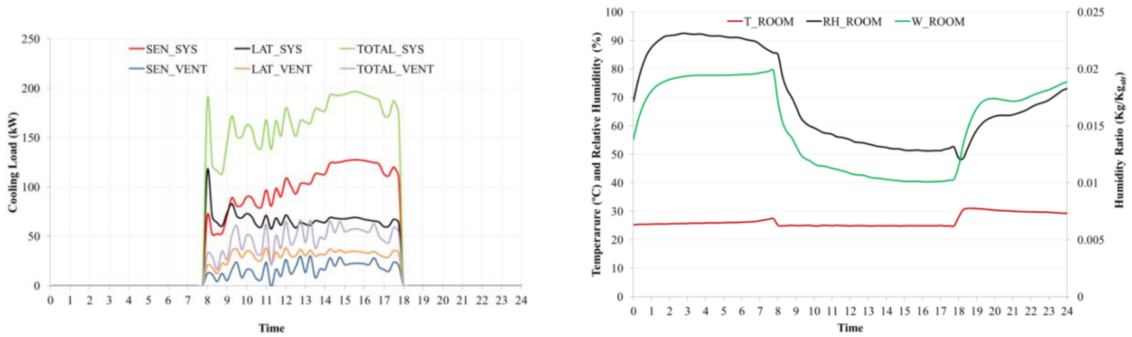


Figure 4.1 The solar radiation and ambient air condition on a hot and humid day (4th July).

It was observed that the solar radiation rise from 6:00 and peaked at 850 W/m^2 by noon. The ambient air temperature varies within 28-38°C while the humidity ratio is rather constant at 19 g_w/kg_{da} throughout the day.

Figure 4.2(a) shows the simulation results of the cooling load of the conventional air-conditioning system (AC). The plot illustrates the system and ventilation loads segregated to the sensible and latent parts. It was observed that the system load is relatively high when the system starts running at 8:00. In this period (about one hour after the startup), the system has to remove the accumulated heat in the space from the previous day. Considering Figure 4.2(b), the indoor air temperature drops from 29ºC to the set point temperature of 25ºC and the humidity ratio reduces from 20 g_w/kg_{da} to 18 g_w/kg_{da}. The system load rises again in the late afternoon from the increasing sensible heat from the space and ventilation. In the plot, while the sensible system load varies from 50-125 kW_{th}; the latent load is rather constant at 65 kW_{th}. For the ventilation, the load varies during 25-65 kW_{th} and is dominated by the latent heat.



(a) Cooling load (b) Space conditions

Figure 4.2 Simulation results of the all-air air-conditioning system (Office).

In Figure 4.2(b), it is observed that the relative humidity of the room air is higher than 60% till 9:00 and then gradually decreases to 50% at the end of the office hour. The average value of the relative humidity is 56.8% (or 0.0112 kg_w/kg_{da}) during the system operation (8:00-18:00). In the Figure, the increase of the humidity of the indoor air after 18:00 (no air-conditioning) is the result of the air infiltration.

For a space, the cooling load can be separated into two categories;

- the internal heat gained from occupants, lighting system, and equipment, and
- the external heat gained from the building envelope influenced by the solar radiation and temperature of the ambient air.

As illustrated in Figure 4.3, the cooling load from the building envelope and the ventilation are the two major components contributing 27.5% and 26.7% of the total cooling load.

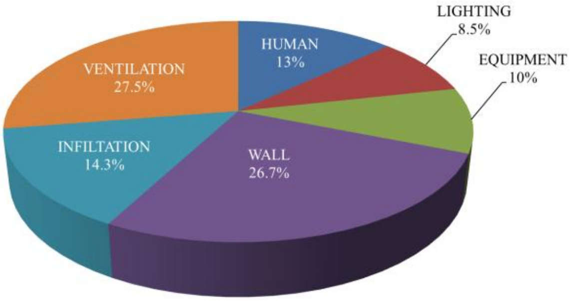
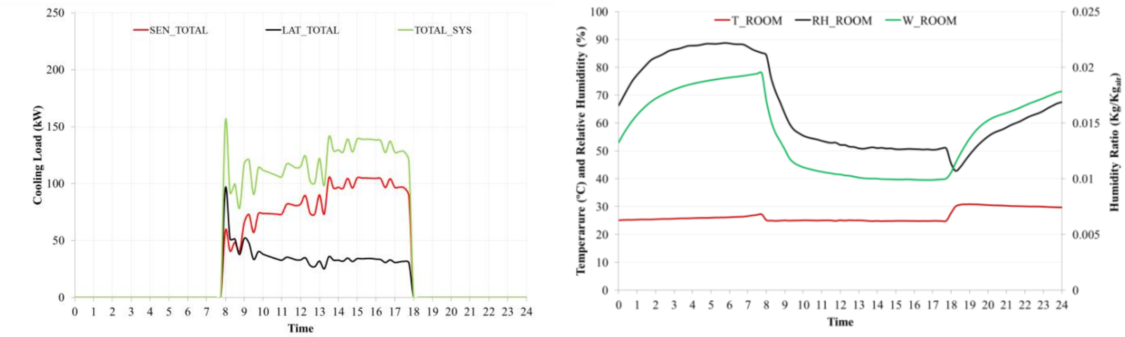


Figure 4.3 The cooling load share of the modeled office building.

Figure 4.4 shows the cooling load and the space condition when the ventilation system is turned off. It is observed that the cooling load varies from 80 to 160 kW_{th}; decreasing by 28%. The average relative humidity of the indoor air is 50.4% (0.0099 kg_w/kg_{da}). Although the cooling load can be reduced dramatically, it should be noted that the indoor air quality does not comply with the requirements of the air-conditioning standard.



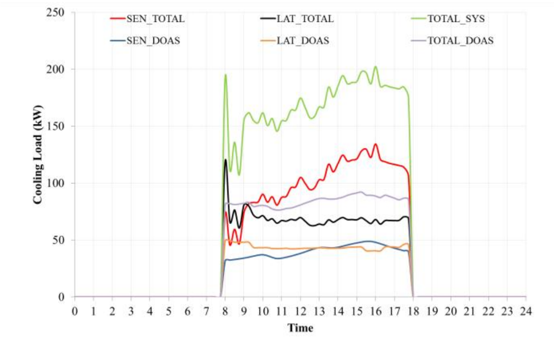
(a) Cooling load (b) Space conditions

Figure 4.4 Simulation results of the all-air air-conditioning system (Office with no ventilation).

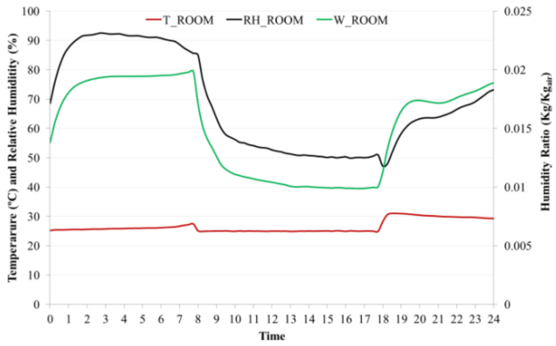
4.1.2 Air-Conditioning by the DOAS

In the analysis, all DOAS (configurations #1-#5) supply sufficient ventilation air into the space, while the terminal unit in the space maintains the room air temperature at 25ºC. The comparisons among the five DOAS configurations and with the conventional system are made in terms of the room air humidity and energy consumption by the system.

Figure 4.5 shows the cooling load and space condition of DOAS #1 that the outdoor air (OA) unit comprises only a cooling coil. It is observed that DOAS #1 has a higher total system load than the conventional system. The OA unit takes the load of 75-90 kW_{th} that is 30-45% larger than the ventilation load. This is because the DOAS cools and dehumidifies the outdoor air and then supplies it at 12°C into the space. The DOAS thus shares part of the space sensible load; resulting a smaller size of the space terminal unit. The average relative humidity of the indoor air is 54.8% (0.0110 kg_w/kg_{da}), lower than that of the conventional system.



(a) Cooling load

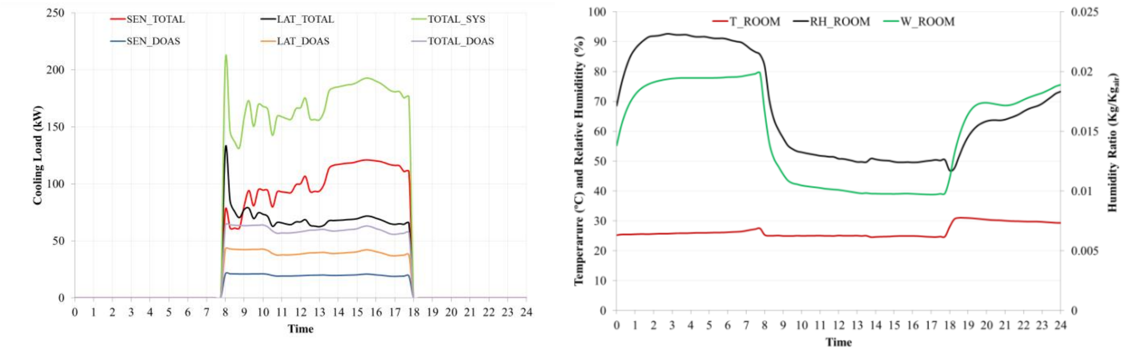


(b) Space conditions

Figure 4.5 DOAS #1 (FC) with the office building

Figure 4.6 shows the cooling load and space condition when the DOAS #2 (FC) is used in the office. For this configuration, the leaving temperature of the outdoor air from the cooling coil is still equal to 12ºC, but the air has to be reheated before entering the room by a reheating coil. It can be seen that the load at the OA unit is 55-65 kW_{th} smaller than that of DOAS #1 by 27-28%. However, the total system load of the two DOAS configurations is quite comparable; the load at the terminal unit of DOAS #2 is larger than that of DOAS #1.

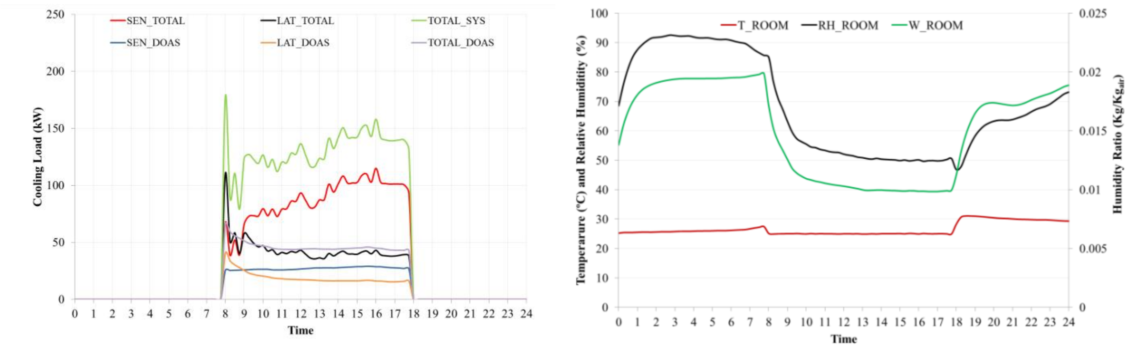
For DOAS #2, the relative humidity in the space is 52.9% (0.0104 kg_w/kg_{da}), slightly lower than that provided by DOAS #1.



(a) Cooling load (b) Space conditions

Figure 4.6 DOAS #2 (RC) with the office building.

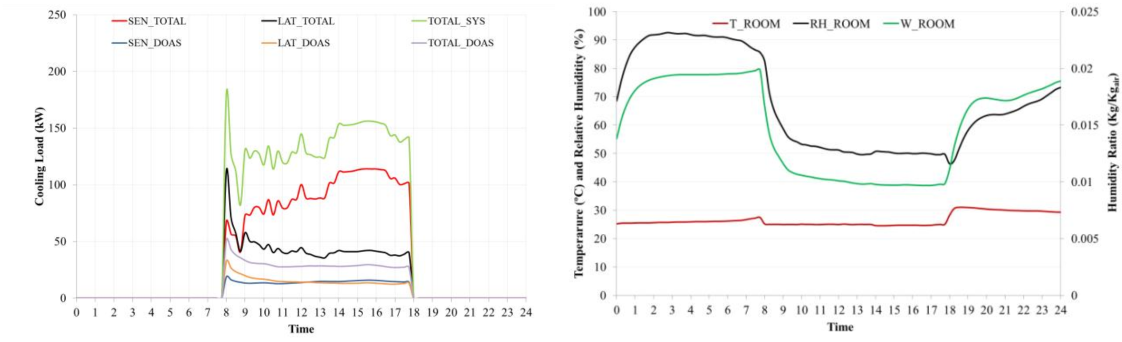
Figure 4.7 shows the simulation results of the cooling loads and space condition of DOAS #3 to which an enthalpy wheel is equipped. It is obvious that both sensible and latent loads of the OA unit decrease. The load at the OA is 40-65 kW$_{th}$; lower by 20-23% from that of DOAS #2. According to Figure 4.7(b), the relative humidity of the room air is 54.4% (0.0107 kg$_w$/kg$_{da}$) on average. The performance to condense moisture from the air is lower for DOAS #3 compared to DOAS #2.



(a) Cooling load (b) Space conditions

Figure 4.7 DOAS #3 (EW) with the office building.

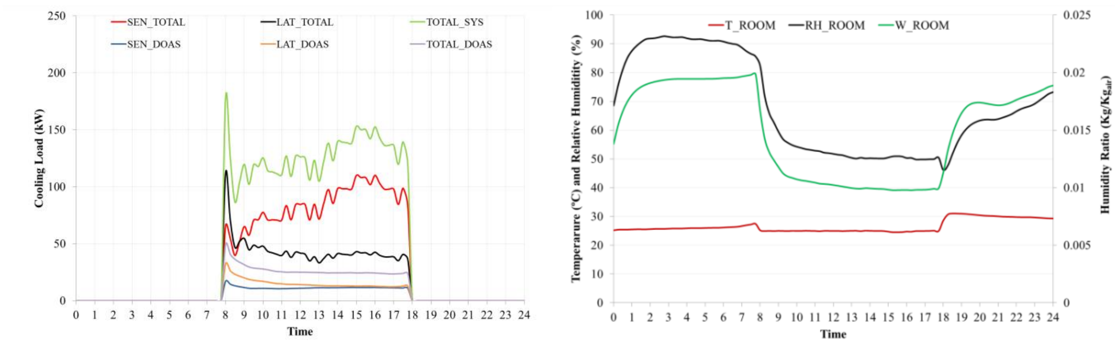
Figure 4.8 shows the cooling load and space condition simulated for DOAS #4. The results show that the sensible wheel reduces the load at the OA unit, but increase the load at the terminal unit. From Figure 4.8(b), the average relative humidity of the indoor air is 53.2% (0.0104 kg$_w$/kg$_{da}$).



(a) Cooling load (b) Space conditions

Figure 4.8 DOAS #4 (EW+SW) with the office building.

Figure 4.9 shows the cooling load and space condition from the operation of DOAS #5. It is observed that the load of the OA unit is about 25-50 kW$_{th}$, being the smallest among different DOAS configurations. As well, DOAS #5, compared to DOAS #4, has slightly smaller sensible load. The plot in Figure 4.9(b) shows that the average relative humidity of the indoor air is 53.6% (0.0106 kg$_{w}$/kg$_{da}$).

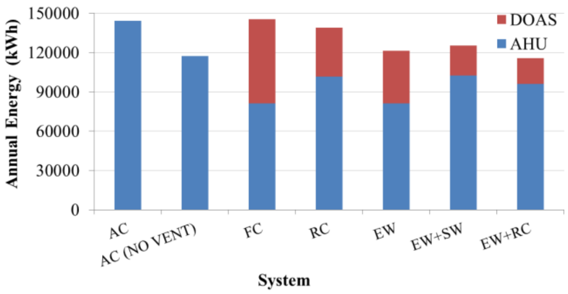


(a) Cooling load (b) Space conditions

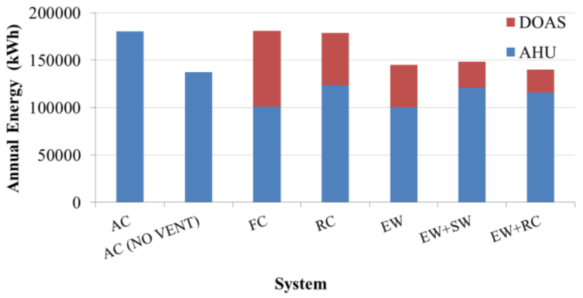
Figure 4.9 DOAS #5 (EW+RC) with the office building.

4.1.3 Annual Energy for Air-Conditioning

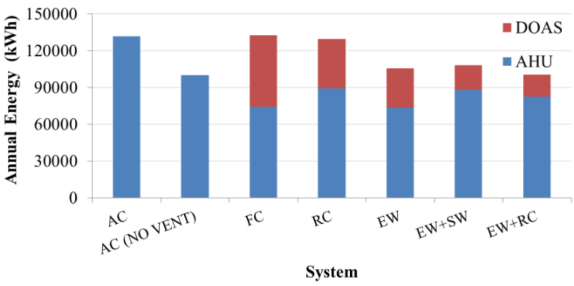
Figure 4.10 shows the comparison of the annual energy for air-conditioning by the conventional system and the DOAS.



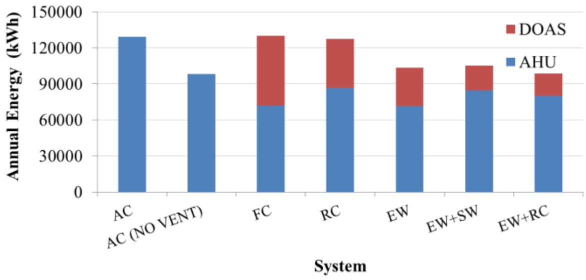
(a) Cold and dry period



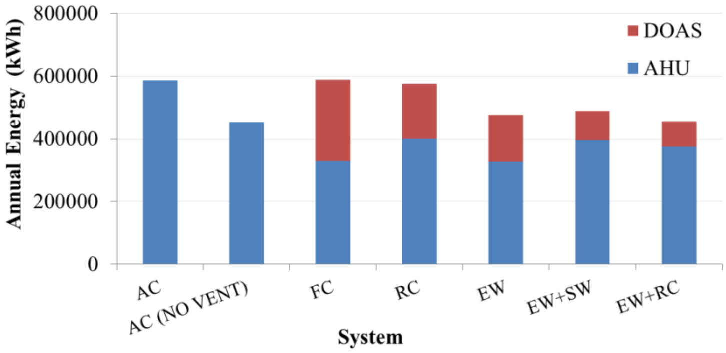
(b) Hot and dry period



(c) Hot and humid period



(d) Cold and humid period



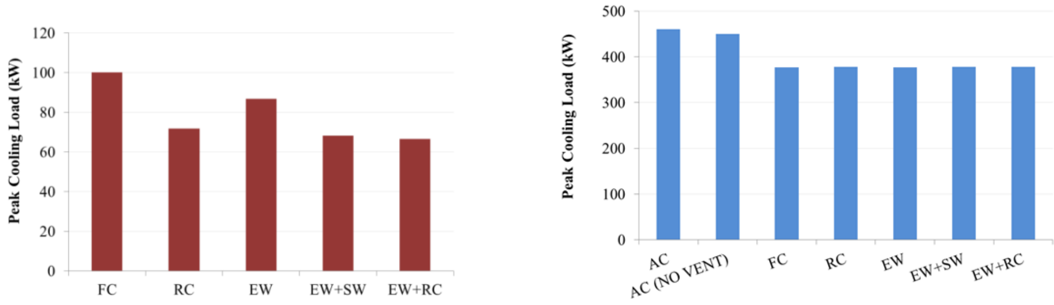
(e) Whole year

Figure 4.10 Annual energy consumption for air-conditioning (Office).

According to the figure, the DOASs equipped with the enthalpy wheel can reduce the cooling load by 19-23% as compared to the conventional system. The cooling load is also not distinct from the case when the ventilation system is turned-off.

4.1.4 Sizes of the DOAS and Space Terminal Unit

Figure 4.11 shows the sizes of the DOAS and the space terminal unit obtained from the peak loads of the equipment from the annual simulations.



(a) DOAS unit (b) Space terminal unit

Figure 4.11 The sizes of the DOAS unit and the terminal unit (Office).

In Figure 4.11, DOAS #1 possesses the largest size, as it has no heat recovery equipment and features. The cooling coil takes the largest load in order to achieve the dew point temperature of the outdoor air at 12ºC. The cooling coil size of DOAS #3 comes with the second largest. The enthalpy wheel can recover the coolness from the exhaust air and the transfer it to the outdoor air, thus reducing the cooling requirement. However, it should be noted that the wheel operates only when the exhaust air has lower enthalpy than the outdoor air.

For DOAS #4 equipped with a sensible wheel, the sensible wheel enhances the transfer of the cool energy from the exhaust air to the outdoor air, thus its cooling coil size is smaller than DOAS #3.

For DOAS #2 and #5, their OA unit sizes are smaller than that of DOAS #3, but comparable to that of DOAS #4. The runaround coil pre-cools the outdoor air and reduces its temperature significantly, hence reducing the cooling coil size.

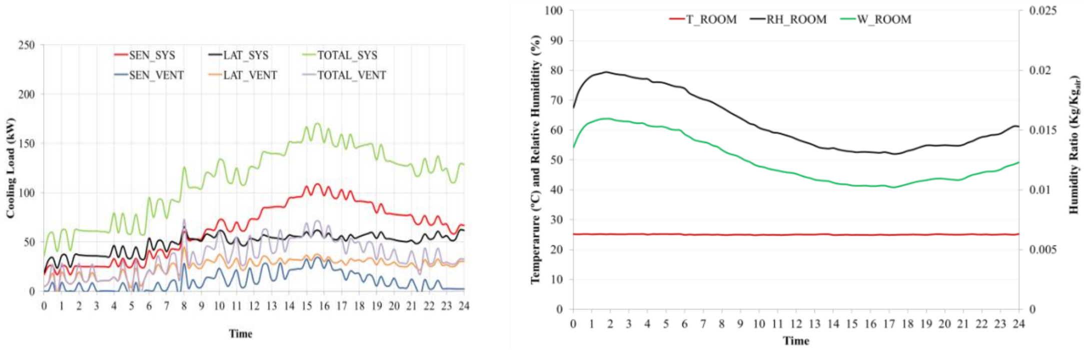
According to Figure 4.11(b), the space terminal unit of the all-air system is the largest, as it has to handle all loads from the space and ventilation. It is also observed that even though turning off the ventilation system can reduce the system cooling load but the size of the terminal unit is not reduced. For all DOASs, the terminal unit size is definitely smaller, as the OA unit shares some of the latent and sensible loads of the space. However, among the DOAS, the terminal unit of those with runaround has larger size, as the sensible cooling capacity is reduced by the reheating process.

4.2 DOASs with the Hotel Building

4.2.1 Cooling load

For hotels, the system and ventilation load patterns are distinct from that of the office building due to the building function and the system operating time.

Figure 4.12(a) shows the cooling load and space condition for the conventional all-air system. It is observed that the system cooling load varies in a range of 35–170 kW_{th}. Compared to the office case, the peak cooling load is smaller due to the lower occupant density. Moreover the air-conditioning system also operates during the night when the cooling is relatively low.



(a) Cooling load (b) Space conditions

Figure 4.12 Simulation results of the all-air air-conditioning system (Hotel).

Figure 4.12(b) shows the indoor condition of the space. The temperature of the indoor air can be maintained at the set point temperature 25ºC all the time. However, the humidity of the room air varies largely from 50-80%. The average humidity of the room air is 62.8% (or 0.0125 kg_w/kg_{da}).

Examining Figure 4.13, it can be seen that the cooling load share of the hotel building and of the office building are rather identical in which the loads from the ventilation and envelope are the two largest loads at the shares of 29.5% and 26.2% of the total system load, respectively. The slightly higher ventilation load than the envelope load would result from the 24 hr. system operation time.

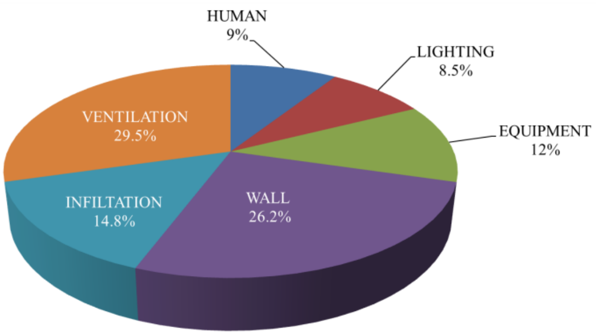
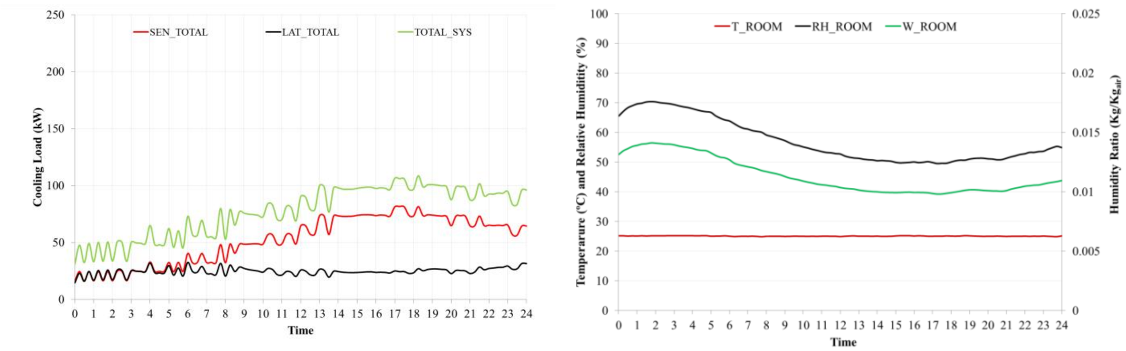


Figure 4.13 The cooling load share of the modeled hotel building.

Figure 4.14 shows the simulation results when the ventilation system of the building is turned off. The system load varies between 30-110 kW$_{th}$, reducing by 30% from the previous case. The average of the relative humidity of the room air is about 57.0% (or 0.1133 kg$_w$/kg$_{da}$).

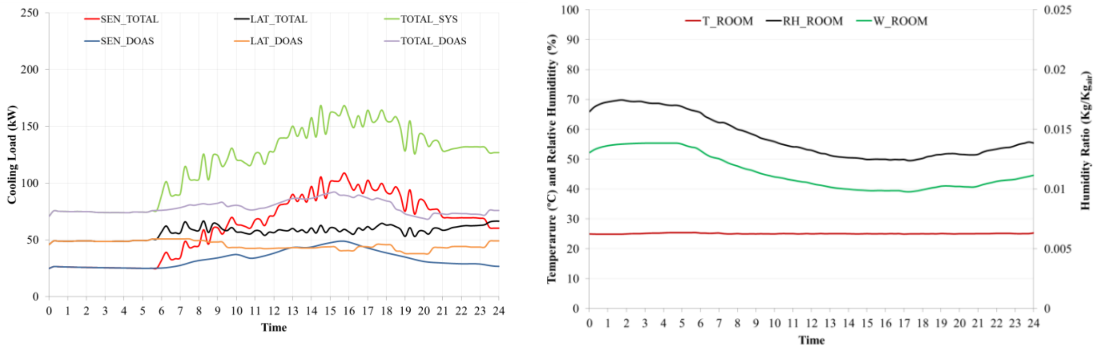


(a) Cooling load (b) Space conditions

Figure 4.14 Simulation results of the all-air air-conditioning system (Hotel with no ventilation).

4.2.2 Air-Conditioning by the DOAS

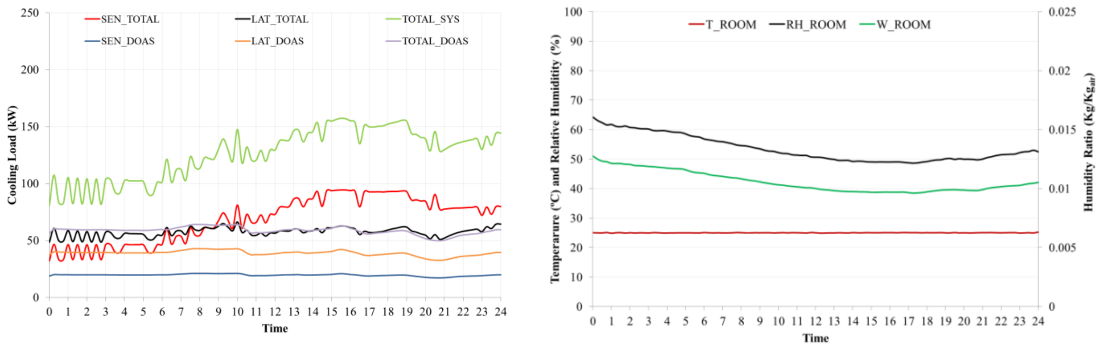
The simulations of the DOAS for the hotel are performed in the same way as use for the office building. Figure 4.15 shows the cooling load and the space condition when DOAS #1 is used for air-conditioning. Similar to the office case, the cooling load of DOAS #1 is slightly higher than the conventional system. However, it is observed that during 0:00-6:00, the DOAS can remove all the latent loads from the ventilation and space. The cooling load of the OA unit contributes up to 65% of the total system load. From the simulations, the average relative humidity of the room air is 57.4% (0.0114 kg$_w$/kg$_{da}$), indicating a better indoor air control of DOAS #1.



(a) Cooling load (b) Space conditions

Figure 4.15 DOAS #1 (FC) with the hotel building.

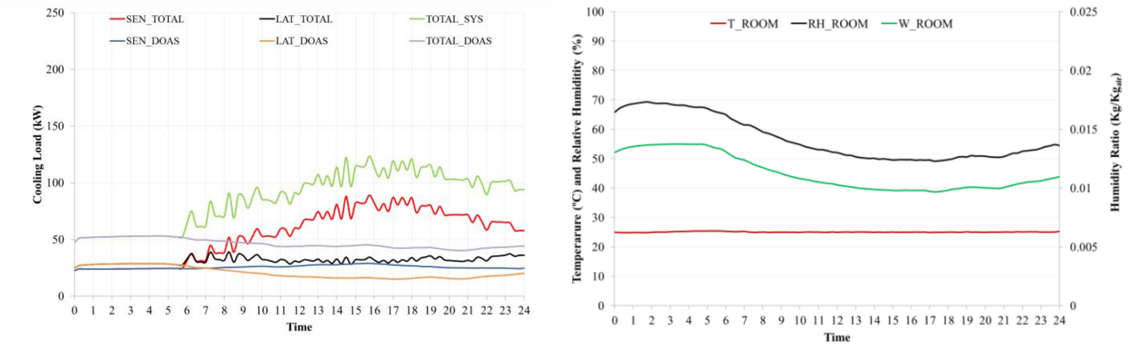
Figure 4.16 shows the simulation results of DOAS #2. It is obvious that with the runaround coil, the DOAS load is about 50-65 kW$_{th}$, reducing by 25% compared to DOAS #1. The average relative humidity of the room air is 53.5% (0.0106 kg$_w$/kg$_{da}$). DOAS #2 has a comparable system load to DOAS #1, as the cooling load at the space terminal unit is larger.



(a) Cooling load (b) Space conditions

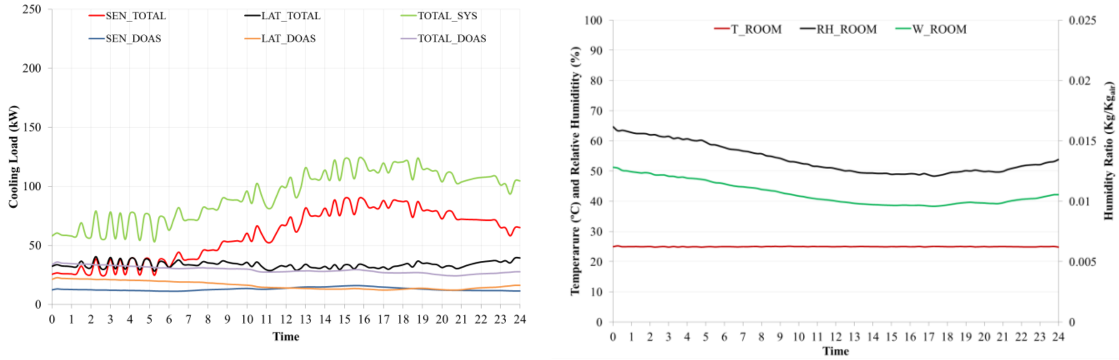
Figure 4.16 DOAS #2 (RC) with the hotel building.

Figures 4.17-4.19 show the simulation results of DOAS #3, #4, and #5. All DOAS have the enthalpy wheel as its system component. Similar to the office case, the sensible and latent load of the system reduce significantly by the heat recovery of the enthalpy wheel.



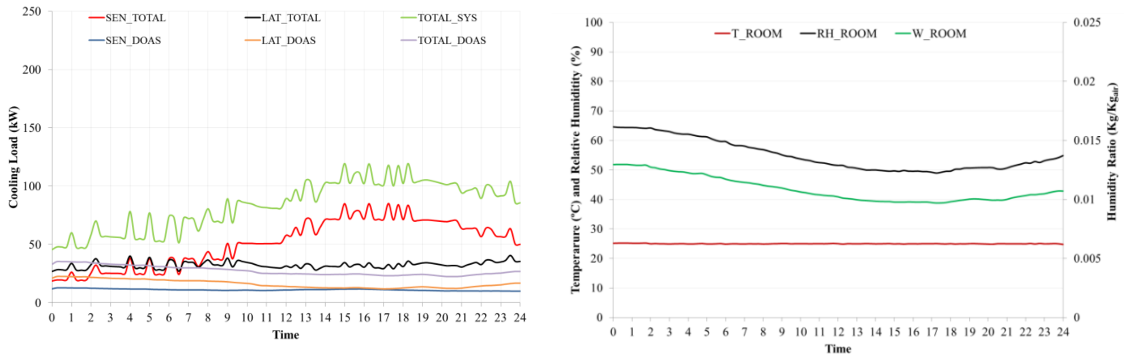
(a) Cooling load (b) Space conditions

Figure 4.17 DOAS #3 (EW) with the hotel building.



(a) Cooling load (b) Space conditions

Figure 4.18 DOAS #4 (EW+SW) for hotel building.

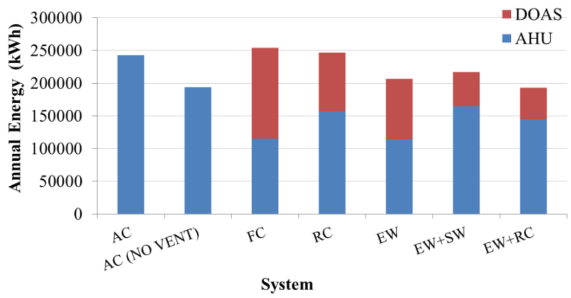


(a) Cooling load (b) Space conditions

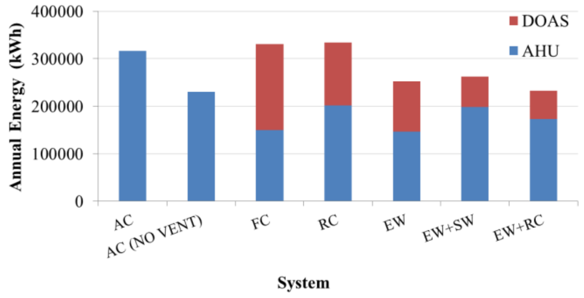
Figure 4.19 DOAS #5 (EW+RC) with the hotel building.

4.2.3 Annual Energy for Air-Conditioning

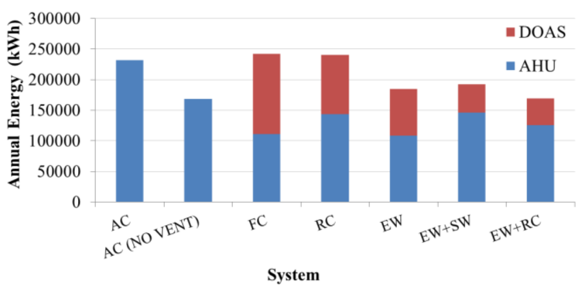
Figure 4.20 shows the energy consumption of air-conditioning in different weather periods and for the whole year. Compared to the office building, the hotel consumes much more air-conditioning energy, as the system operating time is much longer. However, the simulation results show a similar trend with the DOAS that has cooling coil, enthalpy wheel, and runaround coil perform the best performance. This configuration consumes 15-25% lower energy consumption than the conventional system.



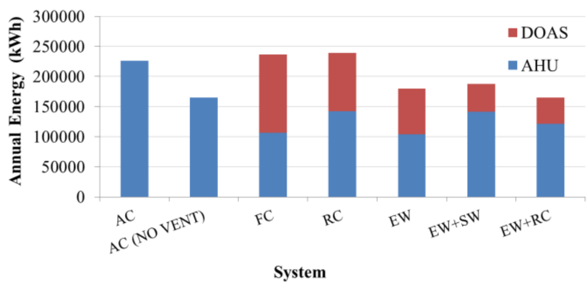
(a) Cold and dry period



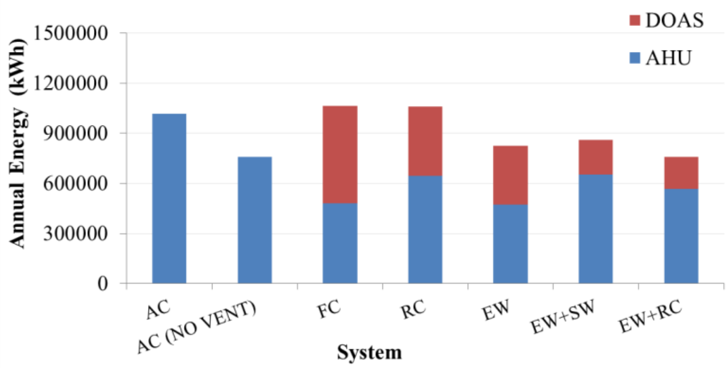
(b) Hot and dry period



(c) Hot and humid period



(d) Cold and humid period

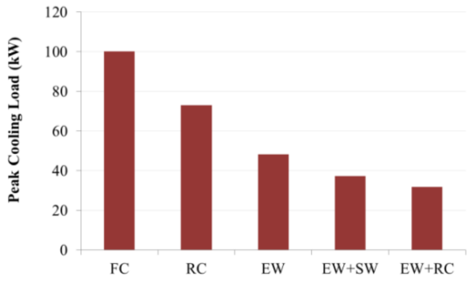


(e) Whole year

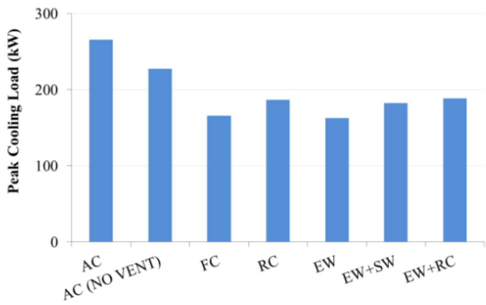
Figure 4.20 Annual energy consumption for air-conditioning (Hotel).

4.2.4 Sizes of the DOAS and Space Terminal Unit

Figure 4.21 shows the sizes of the DOAS unit and space terminal unit. The unit sizes are obtained from the peak load of the unit appearing during the year. Compared to the case of the office building, the components of the DOAS unit have more influence on the unit size. The heat recovery equipment of enthalpy wheel and sensible wheel can reduce the DOAS unit size to 40-50% of the DOAS #1. The space terminal units of all DOAS configuration are smaller than the conventional all-air system.



(a) DOAS unit



(b) Space terminal unit

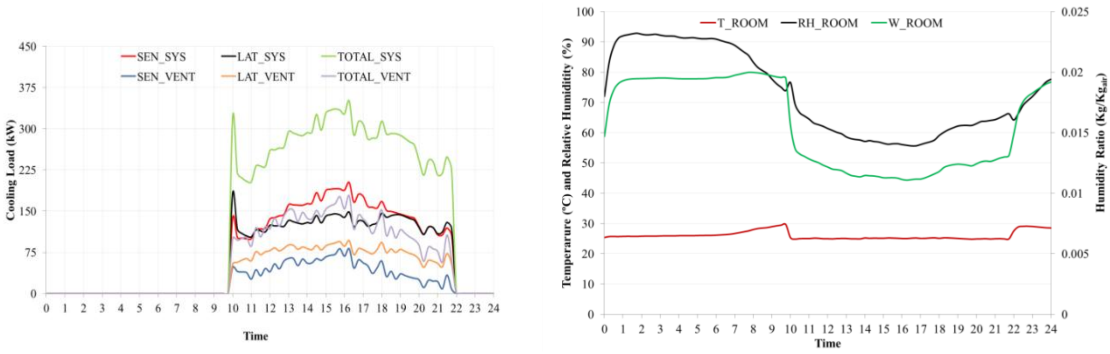
Figure 4.21 The sizes of the DOAS unit and the terminal unit (Hotel).

4.3 DOASs with the Department Store Building

4.3.1 Cooling Load

For the department store, the air-conditioning system is operated between 10.00 and 22.00. In this section, the cooling load and space condition on the hot and humid day are plotted for the conventional air-conditioning system and for the DOAS.

Figure 4.22(a) shows the simulation results from the conventional all-air system. It is observed that the system cooling load is peak at 350 kW_{th}, higher than that of office building (200 kW_{th}) and hotel (170 kW_{th}). This is because the department store has higher occupancy density and equipment power. Moreover, the department requires more ventilation as the number of occupants in the building is higher. In the plot, the ventilation load varies during 60-175 kW_{th}



(a) Cooling load (b) Space conditions

Figure 4.22 Conventional air-conditioning system (Department store).

In Figure 4.22(b), the space condition in the department store is shown. The relative humidity of the indoor air is about an average of 60.8% (0.0122 kg_w/kg_{da}).

Figure 4.23 shows the cooling load share of the department store. It can be observed that the load from the ventilation is as high as 36.5% and larger than that of other buildings.

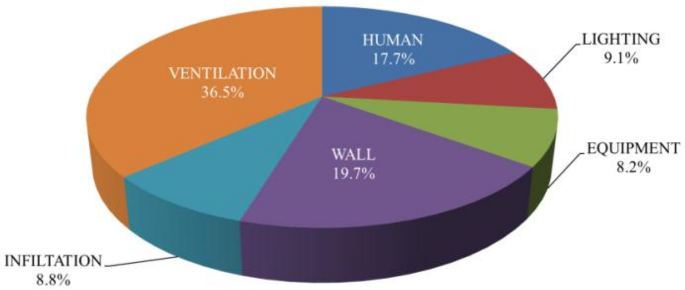
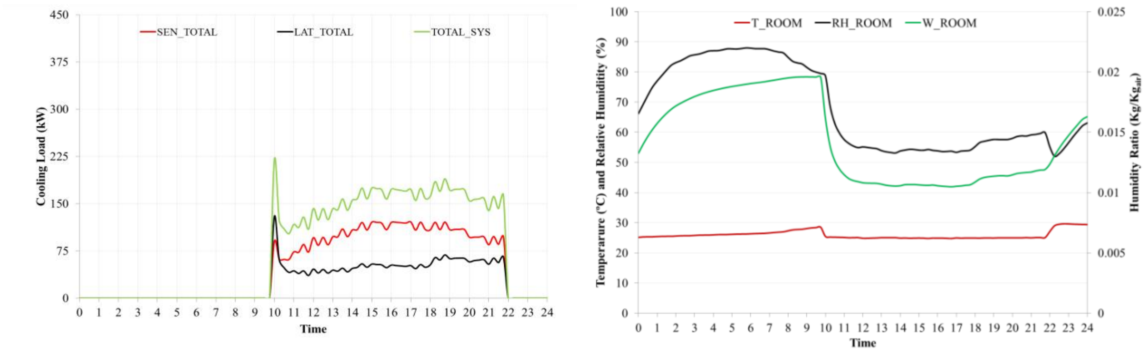


Figure 4.23 The cooling load share of department store.

Figure 4.24 shows the cooling load and space condition when the ventilation system is turned-off. The system load is found to significantly drop due to no ventilation load.

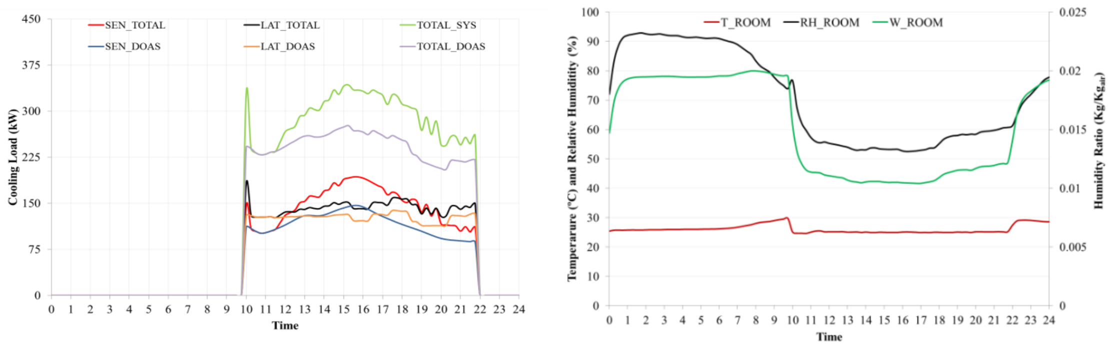


(a) Cooling load (b) Space conditions

Figure 4.24 Conventional air-conditioning system with no ventilation (Department store).

4.3.2 Air Conditioning by the DOAS

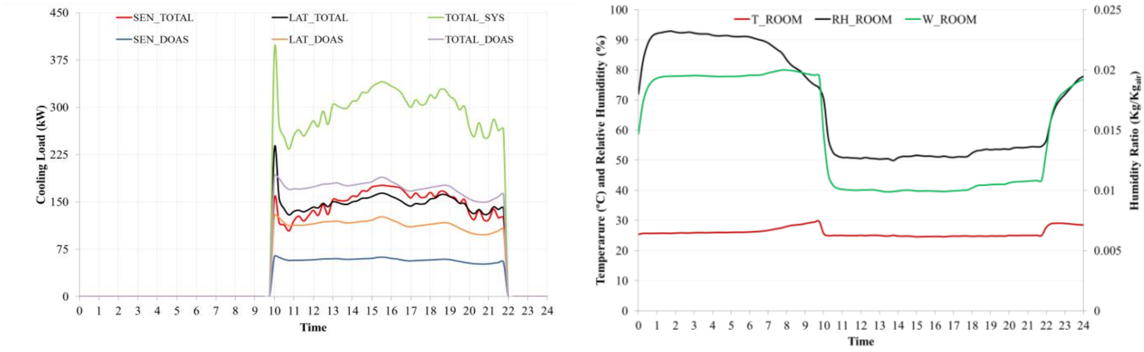
Figure 4.25 shows the cooling load and space condition for DOAS #1. It is observed that the cooling load is 5% higher than that of the conventional system. The cooling load of the DOAS also shares up to 70-80% of the total system load. In some period, the DOAS can take all system loads (both sensible and latent parts). The relative humidity of the indoor air is 56.6% (0.0113 kg_w/kg_{da}) on average. The DOAS control the indoor humidity better than the conventional system.



(a) Cooling load (b) Space conditions

Figure 4.25 DOAS #1 (FC) with the department store building.

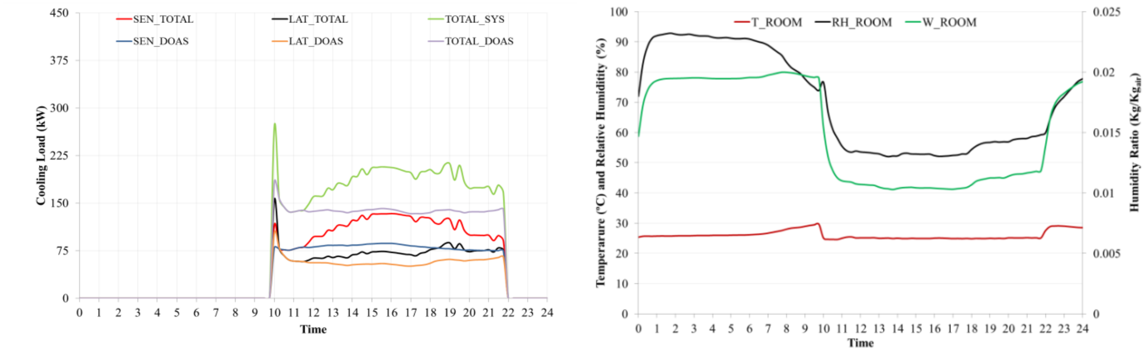
Figure 4.26 shows the simulation results of DOAS #2. The cooling load of this configuration is 150-190 kW_{th}; 30% smaller than that of DOAS #1. The smaller DOAS thus requires the larger space terminal unit. The indoor air relative humidity is 52.6% on average that is lower than that of DOAS #1.



(a) Cooling load (b) Space conditions

Figure 4.26 DOAS #2 (RC) with the department store building.

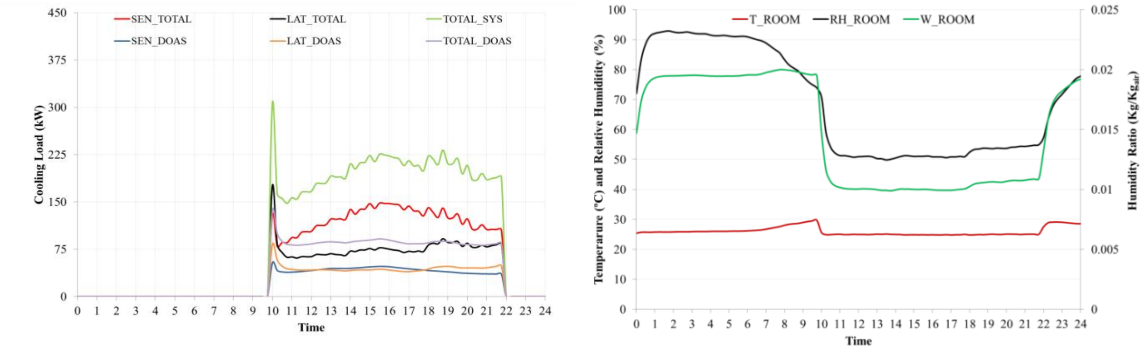
Figure 4.27 shows the simulation results of DOAS #3. In the figure, the cooling load of the DOAS varies from 130-180 kW$_{th}$ that is smaller than that of DOAS #2. The system load also reduces from the heat recovery of the enthalpy wheel. For this DOAS configuration, the average value of the relative humidity of the indoor air is 55.5% (0.0111 kg$_w$/kg$_{da}$)



(a) Cooling load (b) Space conditions

Figure 4.27 DOAS #3 (EW) with the department store building.

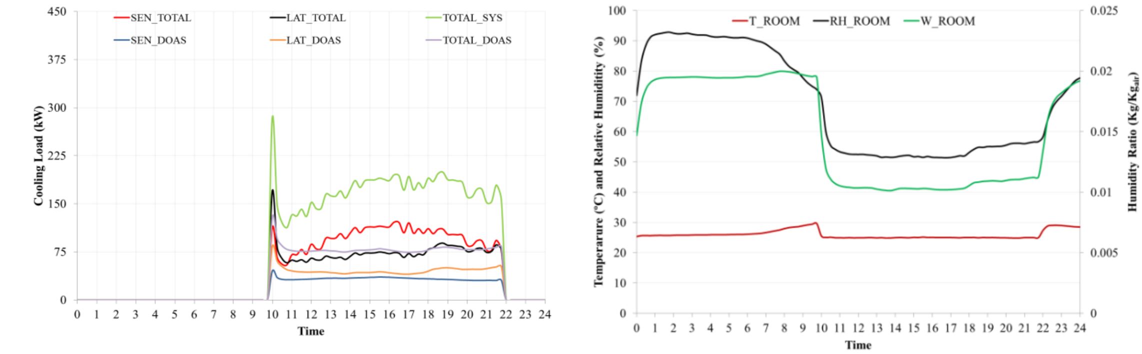
Figure 4.28 shows the simulation results of DOAS #4. For this configuration, the system load is 8% lower than that of DOAS #3. The sensible wheel enhances the heat recovery from the exhaust air. The cooling of this DOAS is 80-135 kW$_{th}$ and the relative humidity of the indoor air is 52.6% (0.0104 kg$_w$/kg$_{da}$) on average.



(a) Cooling load (b) Space conditions

Figure 4.28 DOAS #4 (EW+SW) with the department store building.

Figure 4.29 shows the simulation results of DOAS #5. It is obvious that this DOAS configuration has the smallest cooling load. The system cooling load is also smallest. The runaround coil slightly improves the capacity to condense moisture from the outdoor air. From the figure, the average of the indoor air relative humidity is 53.9% (0.0107 kg_w/kg_{da})

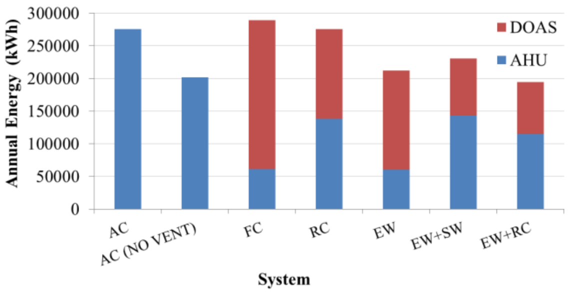


(a) Cooling load (b) Space conditions

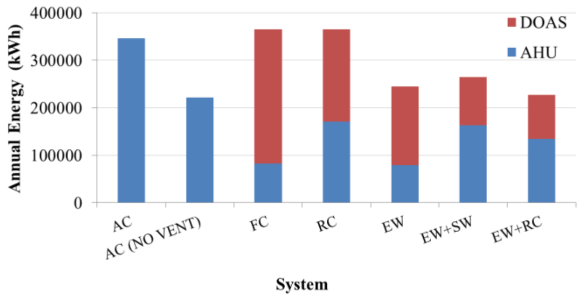
Figure 4.29 DOAS #5 (EW+RC) with the department store building.

4.3.3 Annual Energy for Air-Conditioning

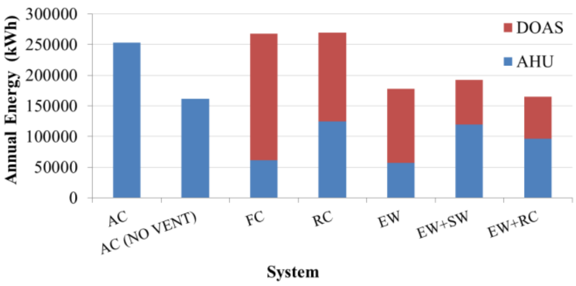
Figure 4.30 shows the energy consumption of air-conditioning in the department store. Comparing to the office and hotel buildings, the department store has larger energy consumption for the air-conditioning. A major reason is the higher amount of the ventilation air for the building occupants. With the very ventilation, the presence of the enthalpy wheel as a system component can save 22-33% of the air-conditioning energy from the conventional all-air system.



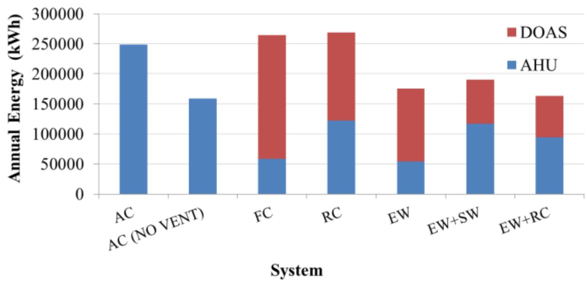
(a) Cold and dry period



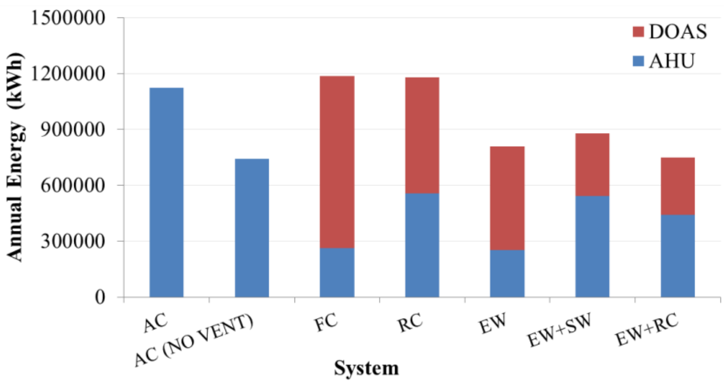
(b) Hot and dry period



(c) Hot and humid period



(d) Cold and humid period

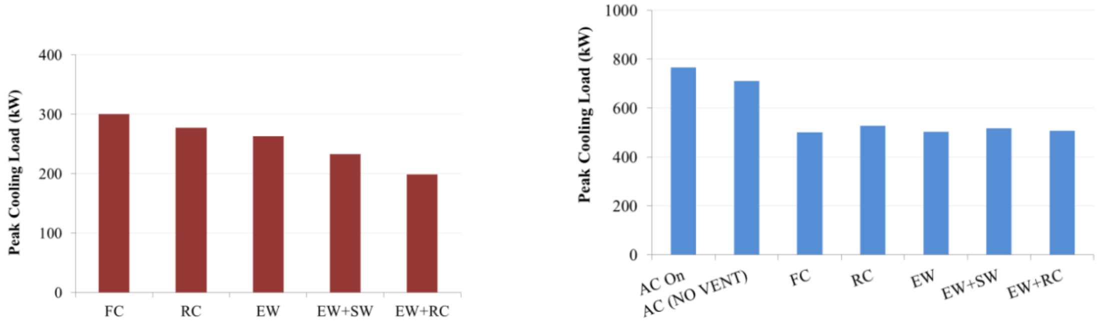


(e) Whole year

Figure 4.30 Annual energy consumption for air-conditioning (Department store).

4.3.4 Sizes of the DOAS and Space Terminal Unit

Figure 4.31 shows the sizes of the DOAS and the space terminal unit obtained from the peak loads of the equipment from the annual simulations. It is obvious that DOAS #5 has the smallest sizes of both the DOAS unit and the space terminal unit.



(a) DOAS unit (b) Space terminal unit

Figure 4.31 The sizes of the DOAS unit and the terminal unit (Department store).

4.4 Life Cycle Costing

In the section, the analysis of the life cycle costs of the DOAS is presented. The costs include the equipment cost and the energy cost through the system life time. The equipment cost based on the cooling capacity of the equipment while the energy cost based on the system cooling load. The equipment cost and the energy cost obtained by correlation with the equipment cost from literature review [9] and calculated according to the service rated of Metropolitan Electricity Authority (MEA) [22].

To determine the electrical energy cost, the electricity tariff of each building is applied. It is also assumed that the interest rate is 7% per annum and the escalation of the electricity price is 3% per annum. Table 4.1, 4.2, and 4.3 show the life cycle costs of the DOAS for the three building types.

Table 4.1 Life cycle cost of the DOAS for office building**.**

Configuration	Equipment cost (Baht)	Energy cost (Baht/year)	Net Present Value (Baht)	Annual Worth Value (Baht/year)
Conventional	97,000	780,167	8,842,672	970,925
DOAS #1	295,000	784,334	9,087,394	997,795
DOAS #2	338,500	765,655	8,921,498	979,580
DOAS #3	565,000	633,183	7,662,983	841,395
DOAS #4	781,000	649,021	8,056,535	884,607
DOAS #5	601,500	607,094	7,407,032	813,292

Table 4.2 Life cycle cost of the DOAS for hotel building**.**

Configuration	Equipment cost (Baht)	Energy cost (Baht/year)	Net Present Value (Baht)	Annual Worth Value (Baht/year)
Conventional	58,000	1,283,970	14,451,307	1,586,753
DOAS #1	252,500	1,341,698	15,292,934	1,679,164
DOAS #2	288,500	1,335,114	15,255,134	1,675,013
DOAS #3	528,500	1,036,053	12,142,665	1,333,264
DOAS #4	737,000	1,082,143	12,867,827	1,412,887
DOAS #5	552,500	958,418	11,296,375	1,240,342

Table 4.3 Life cycle cost of the DOAS for department store building.

Configuration	Equipment cost (Baht)	Energy cost (Baht/year)	Net Present Value (Baht)	Annual Worth Value (Baht/year)
Conventional	175,000	1,472,374	16,680,315	1,831,498
DOAS #1	407,500	1,550,695	17,790,795	1,953,429
DOAS #2	507,000	1,544,497	17,820,812	1,956,725
DOAS #3	880,000	1,068,041	12,852,750	1,411,232
DOAS #4	1,132,700	1,153,568	14,064,203	1,544,249
DOAS #5	978,000	989,708	12,072,630	1,325,574

The results in the tables show that DOAS #1 and #2 possess the life cycle cost of 1.0-2.8% higher than the conventional all-air system. For DOAS #3, #4, and #5, which have an enthalpy wheel as the system component for heat recovery, the life cycle costs are lower than the conventional system by 9-16% for office, 11-22% for hotel, and 16-28% for department store.

CHAPTER 5
CONCLUSION

In this study, the performance of the five DOAS configurations was evaluated under the hot and humid climate of Thailand by simulations using TRNSYS software. The simulations were undertaken for three building types i.e. office, hotel and department store. The results can be concluded as follows:

- All DOAS are able to control the humidity of the indoor air to meet the standard requirements better than the conventional all-air system,
- The enthalpy wheel is the key component making the DOAS more energy efficient than the all-air system. The DOAS equipped with an enthalpy wheel can save energy 19-23% for office, 15-25% for hotel, and 22-33% for department store,
- In terms of the life cycle costs, the DOAS with an enthalpy wheel has the lower costs than the all-air system by 9-16% for office, 11-22% for hotel, and 16-28% for department store.

The most energy-efficient configuration of DOAS is the combination of a cooling coil, an enthalpy wheel and a runaround coil. This configuration also offers the lowest life cycle costs.

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APPENDIXES

APPENDIX A: Schematics Diagram of Simulated System in TRNSYS

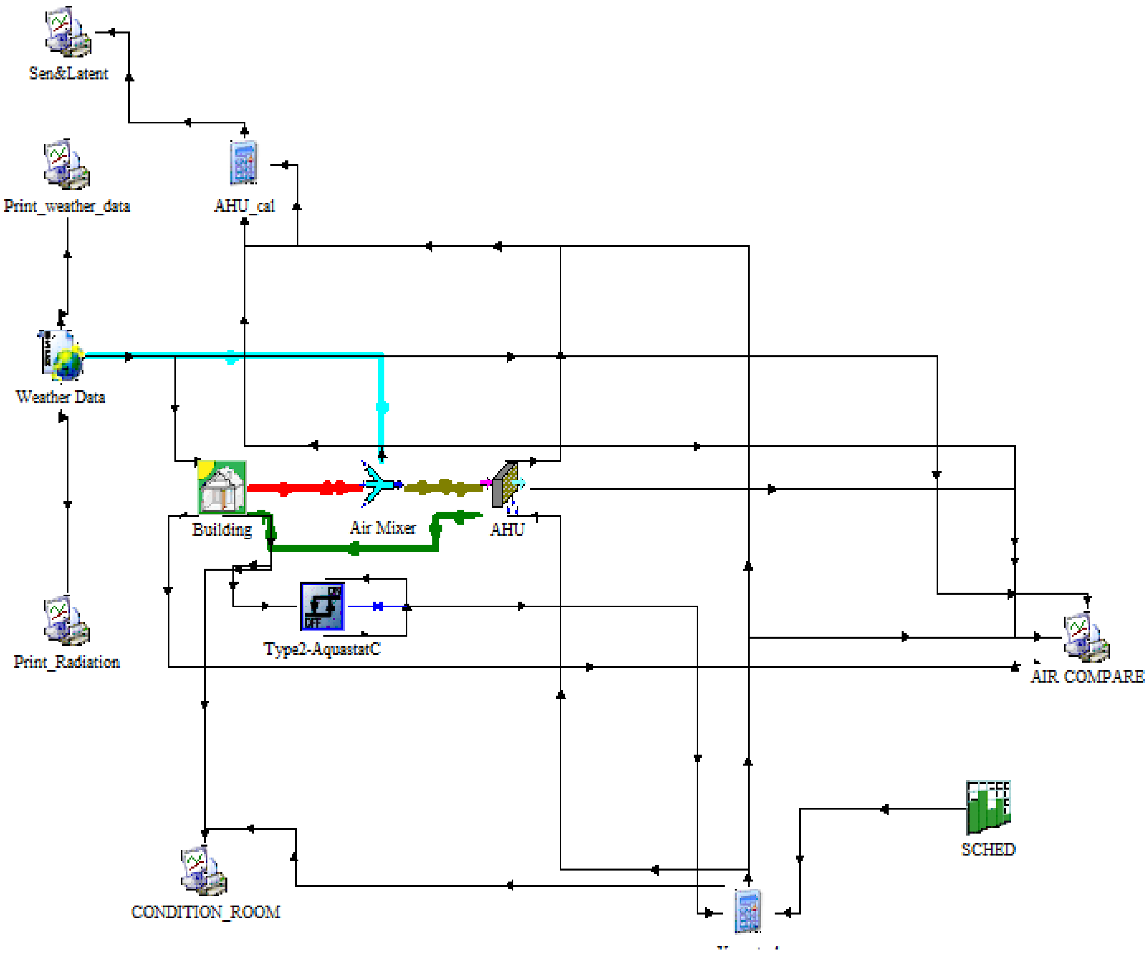


Figure A.1 Schematic diagram of the conventional air-conditioning system.

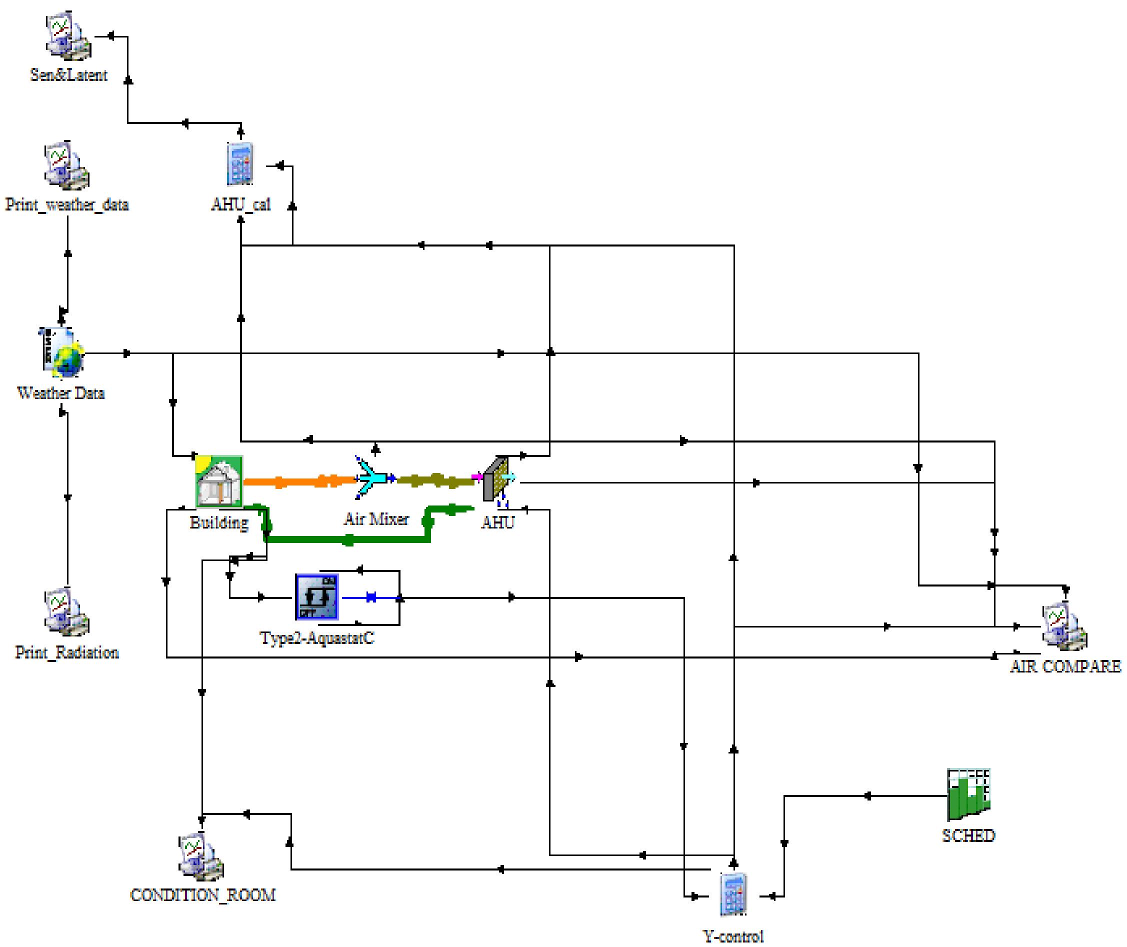


Figure A.2 Schematic diagram of the conventional air-conditioning system with no ventilation.

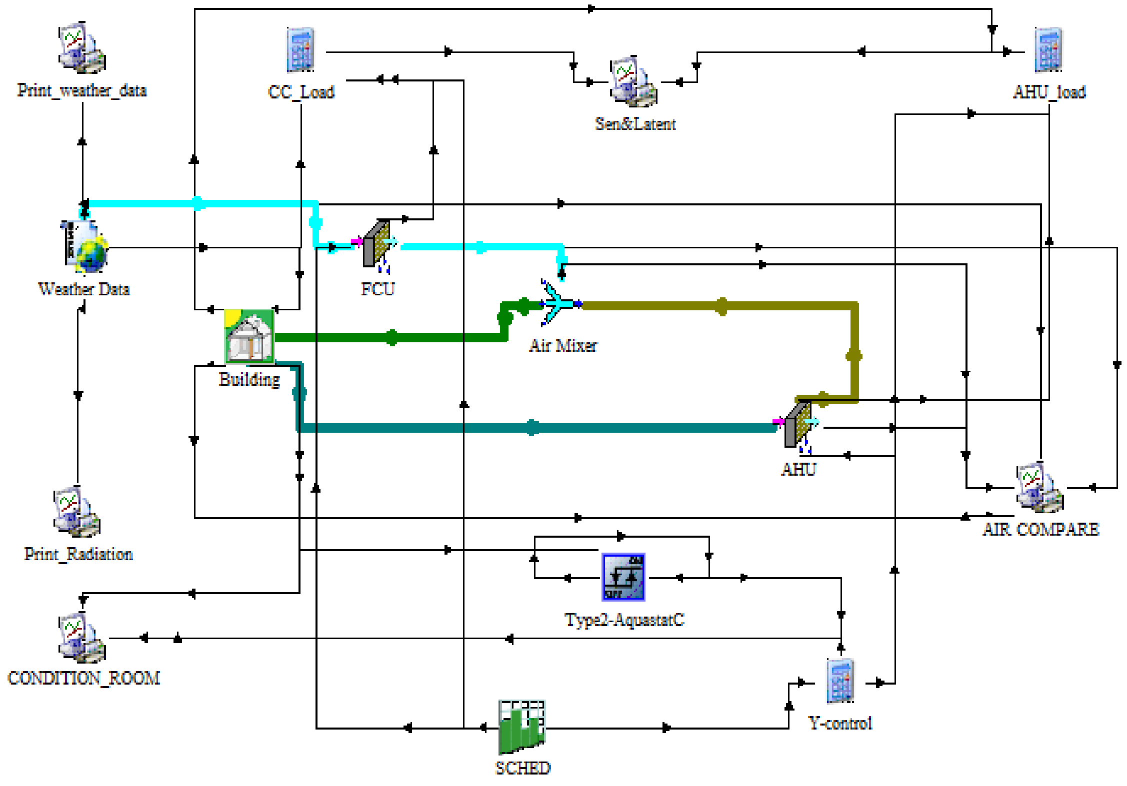


Figure A.3 Schematic diagram of DOAS#1 (FC).

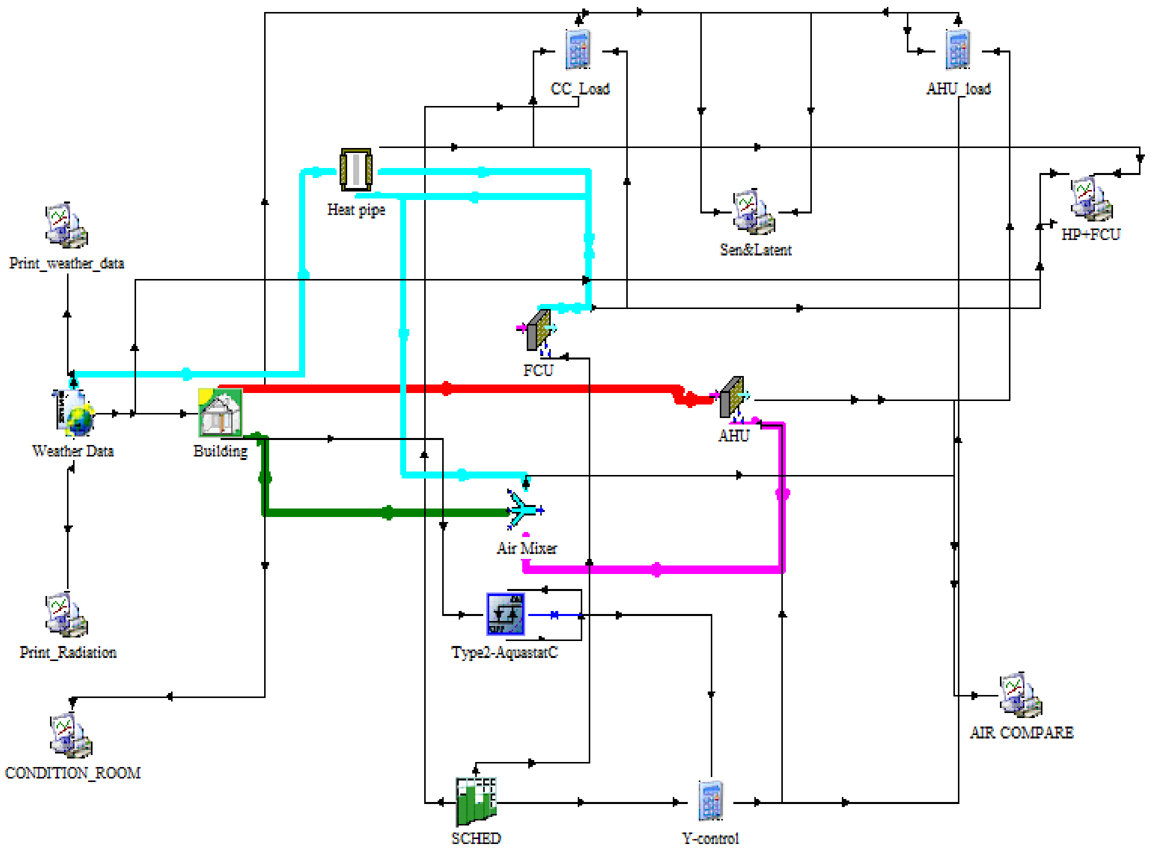


Figure A.4 Schematic diagram of DOAS#2 (RC).

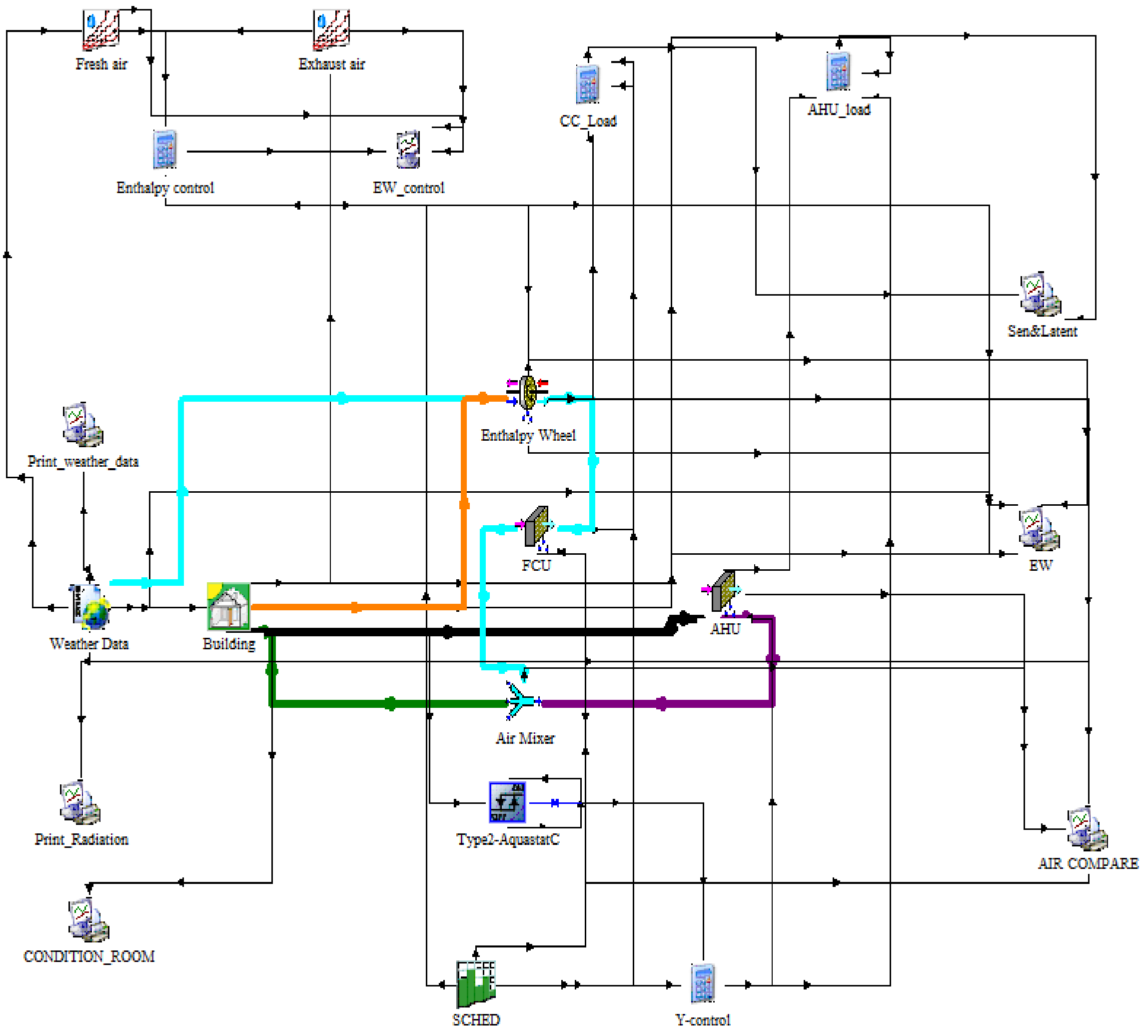


Figure A.5 Schematic diagram of DOAS#3 (EW).

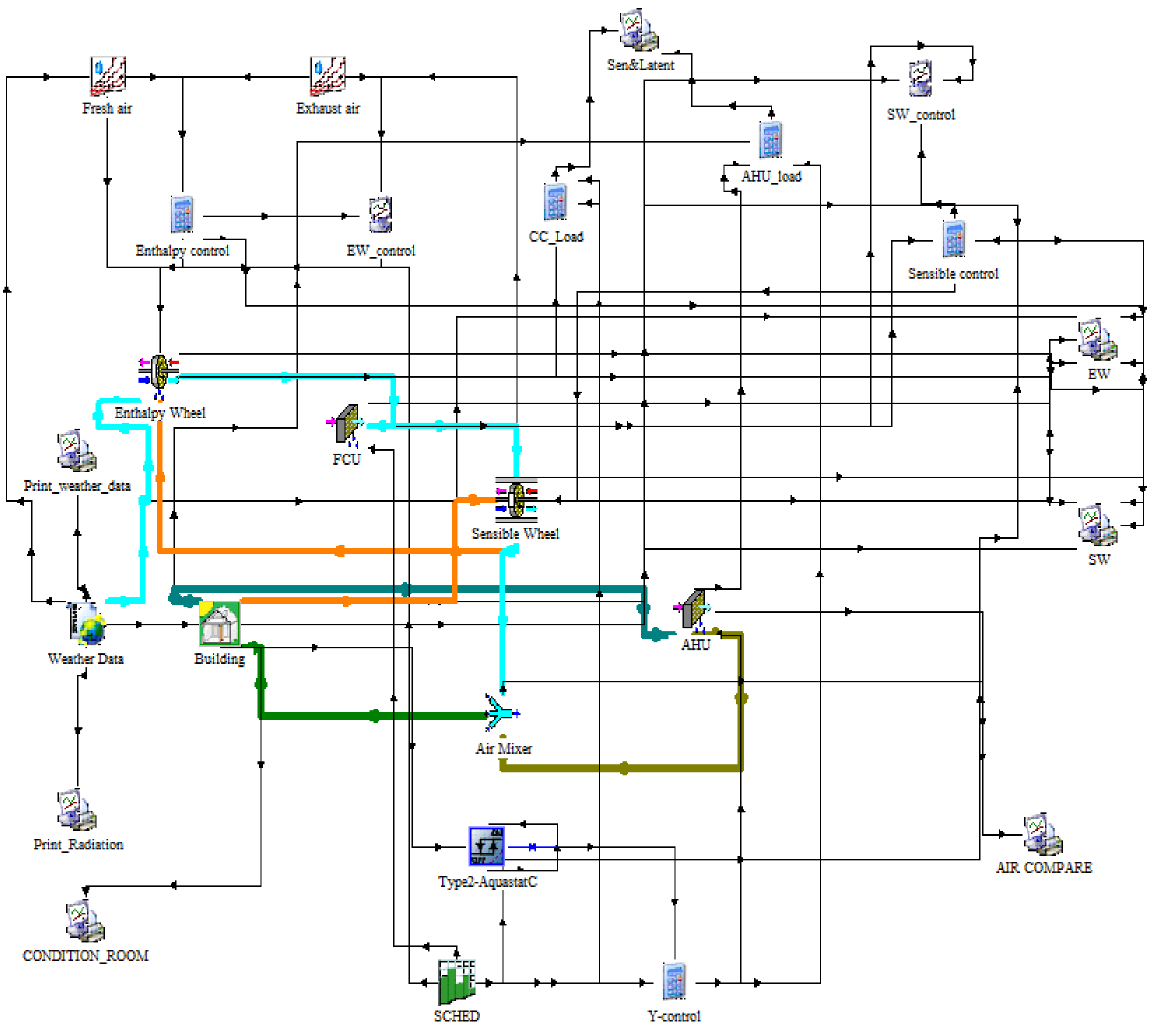


Figure A.6 Schematic diagram of DOAS#4 (EW+SW).

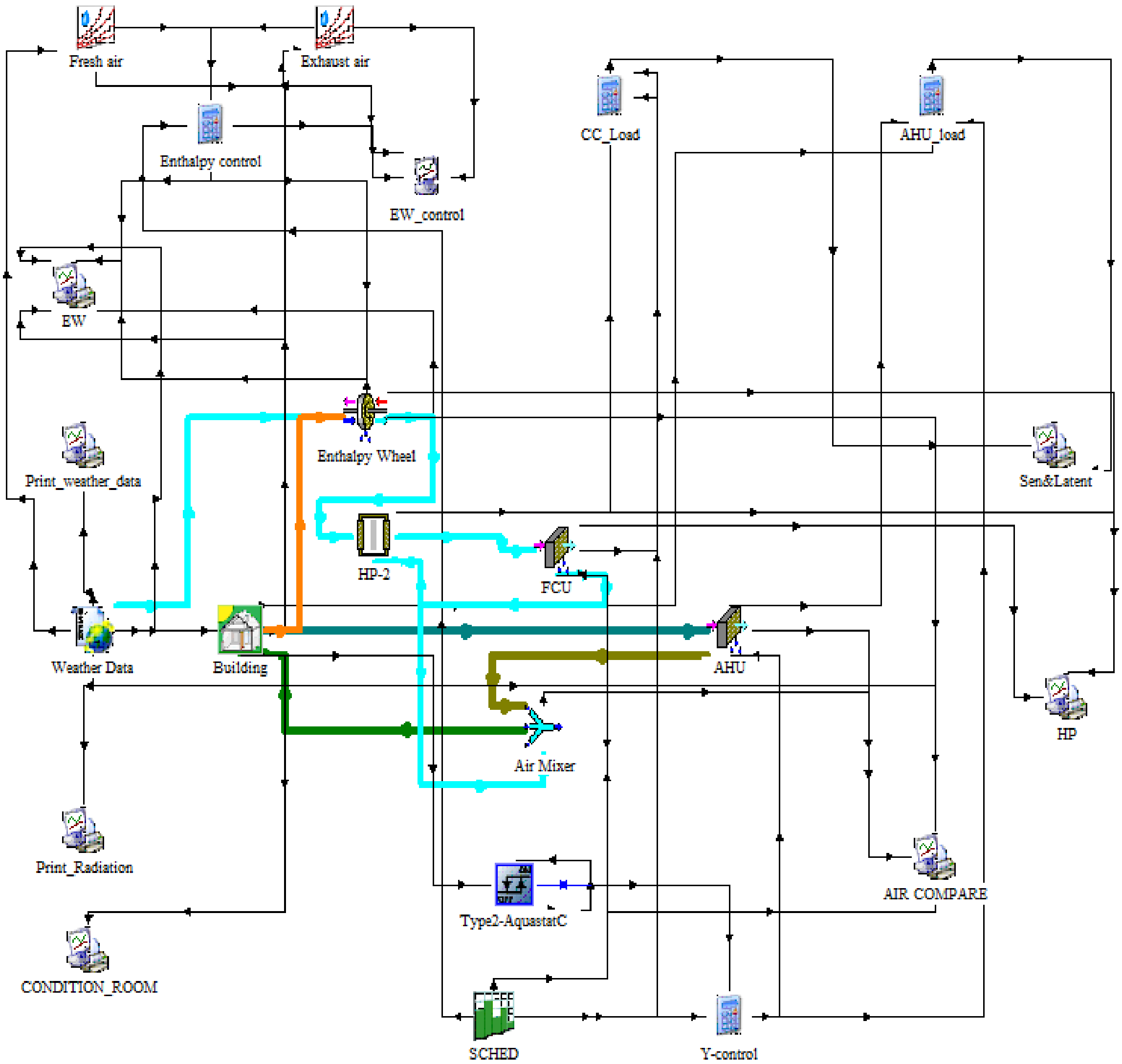


Figure A.7 Schematic diagram of DOAS#5 (EW+RC).

APPENDIX B: DOAS on Cold and Dry Day

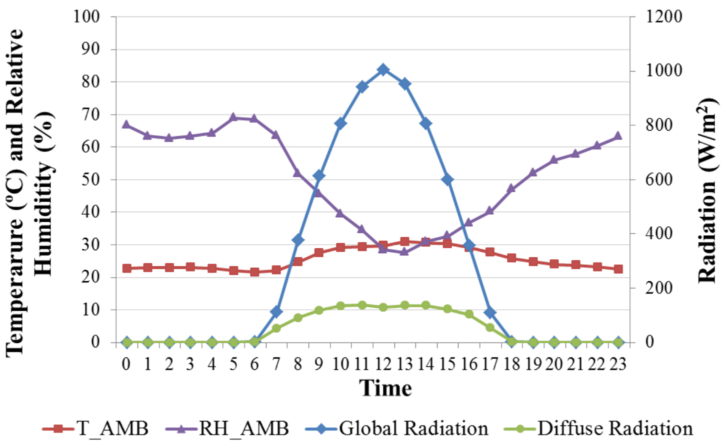
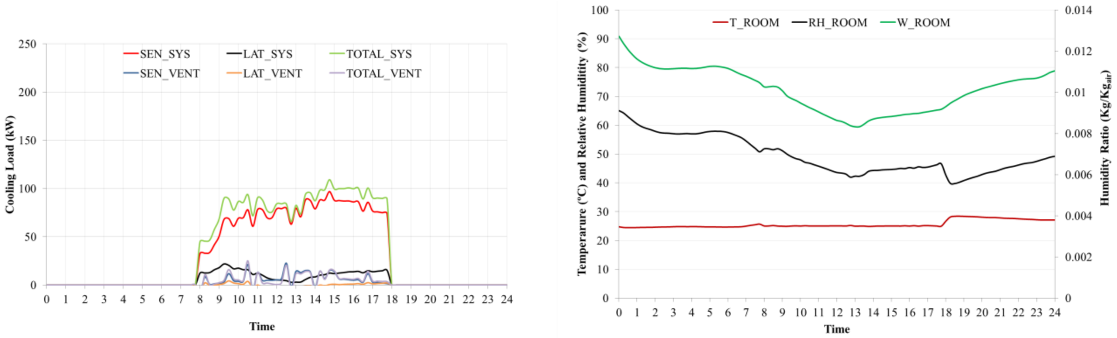


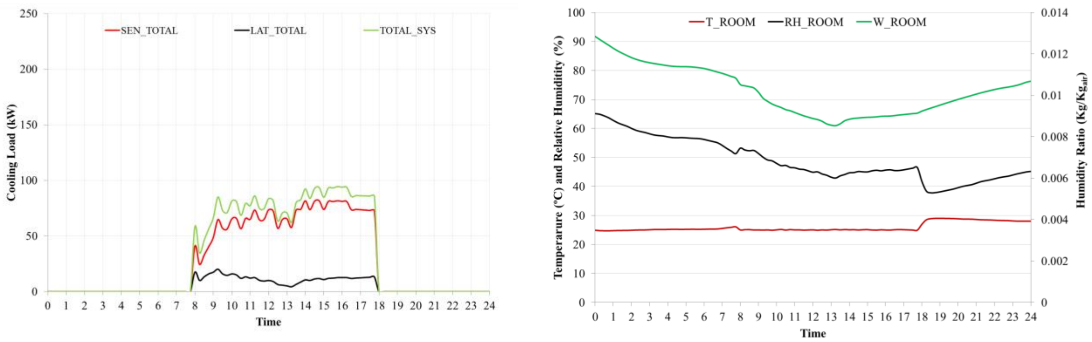
Figure B.1 The solar radiation and ambient air condition on cold and dry day (11 November).

Office Building



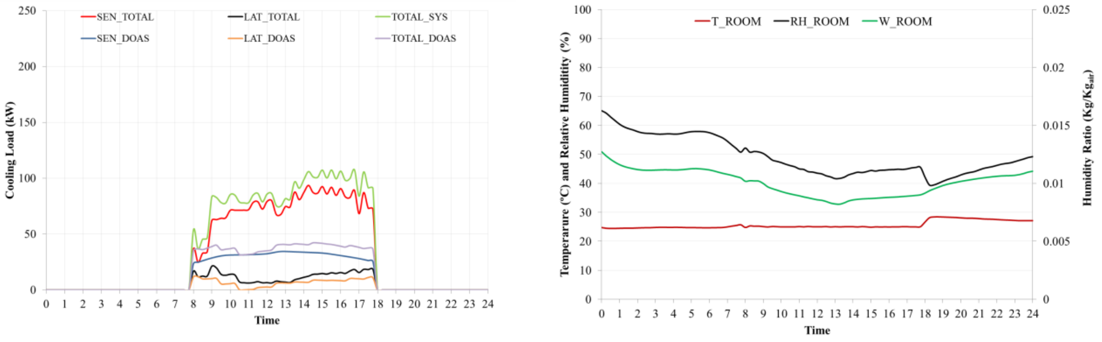
(a) Cooling load (b) Space conditions

Figure B.2 Conventional air-conditioning system on cold and dry day (Office).



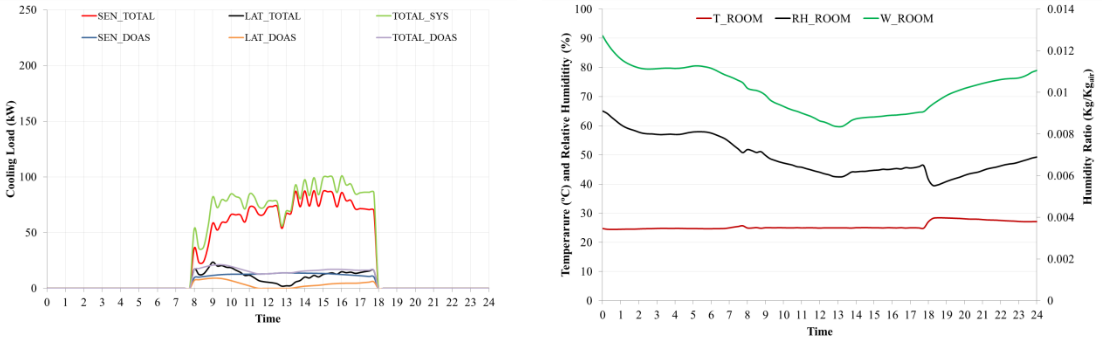
(a) Cooling load (b) Space conditions

Figure B.3 Conventional air-conditioning system with no ventilation on cold and dry day (Office).



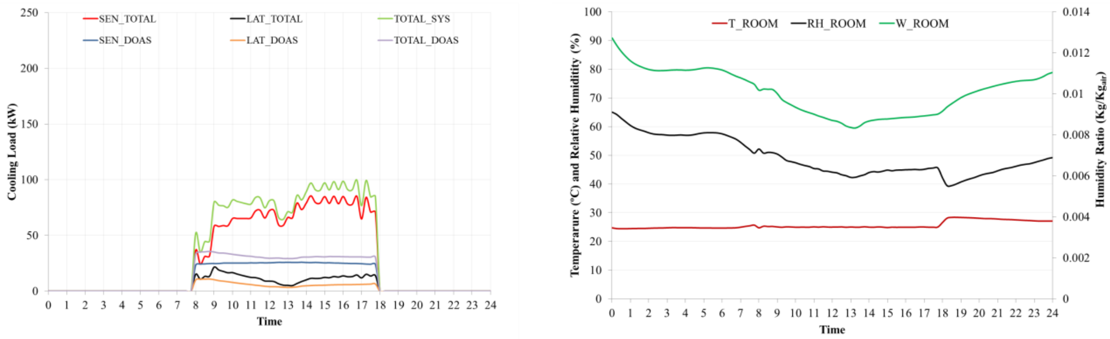
(a) Cooling load (b) Space conditions

Figure B.4 DOAS#1 (FC) on cold and dry day (Office).



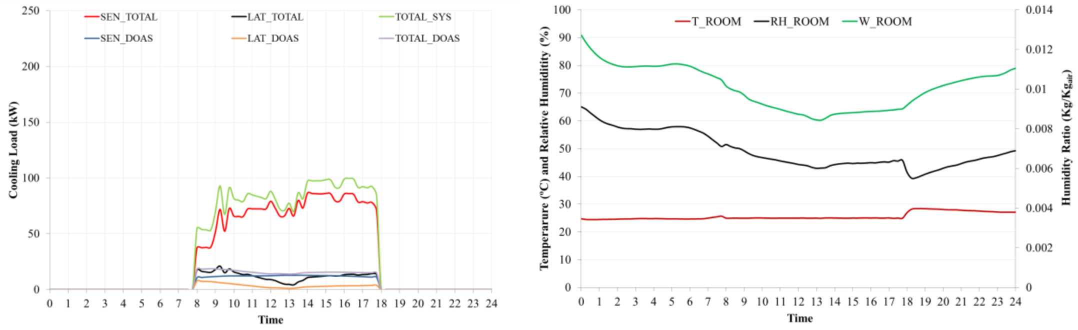
(a) Cooling load (b) Space conditions

Figure B.5 DOAS#2 (RC) on cold and dry day (Office).



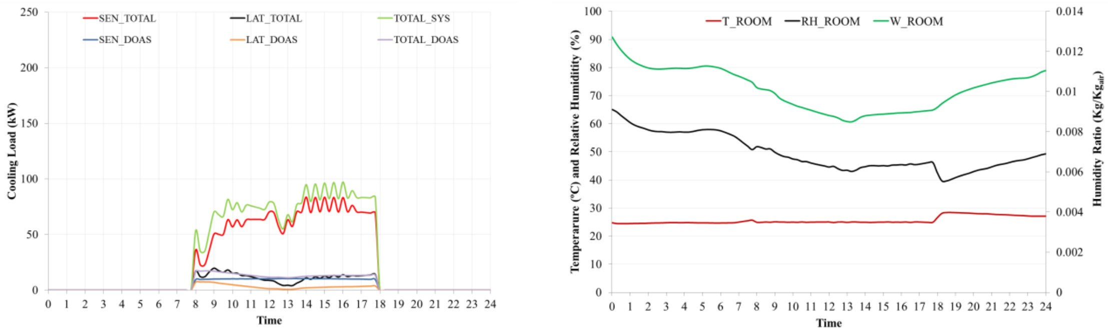
(a) Cooling load (b) Space conditions

Figure B.6 DOAS#3 (EW) on cold and dry day (Office).



(a) Cooling load (b) Space conditions

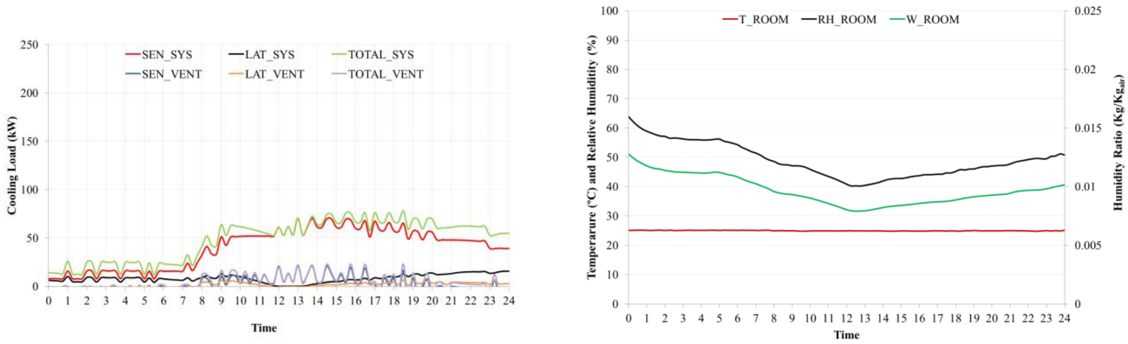
Figure B.7 DOAS#4 (EW+SW) on cold and dry day (Office).



(a) Cooling load (b) Space conditions

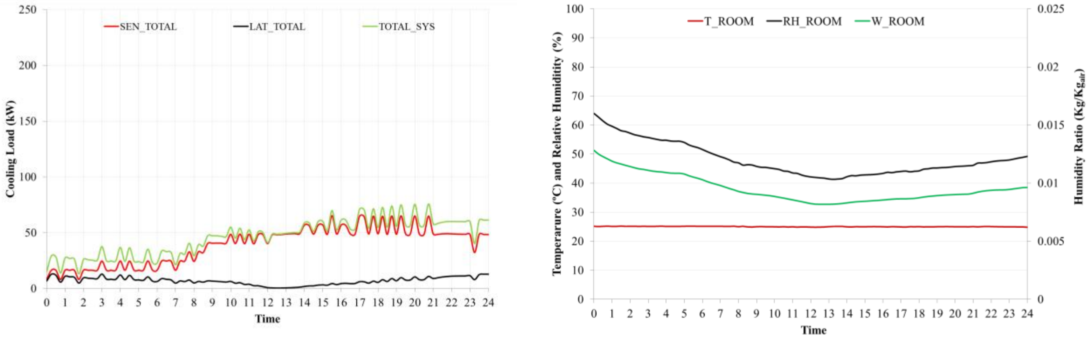
Figure B.8 DOAS#5 (EW+RC) on cold and dry day (Office).

Hotel Building



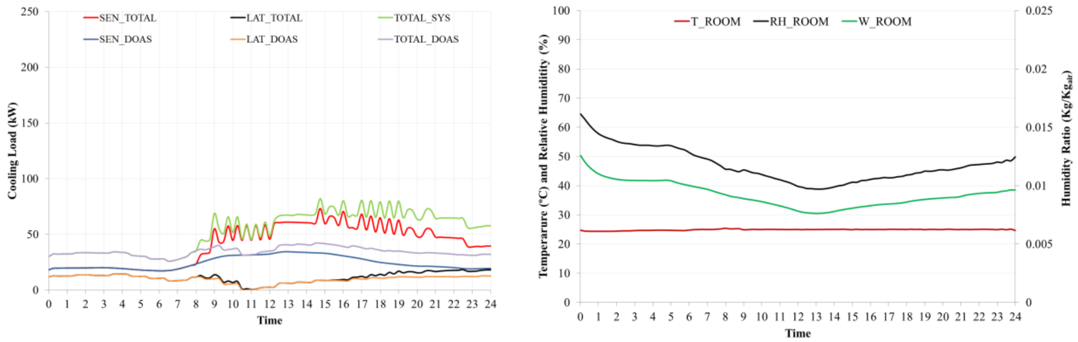
(a) Cooling load (b) Space conditions

Figure B.9 Conventional air-conditioning system on cold and dry day (Hotel).



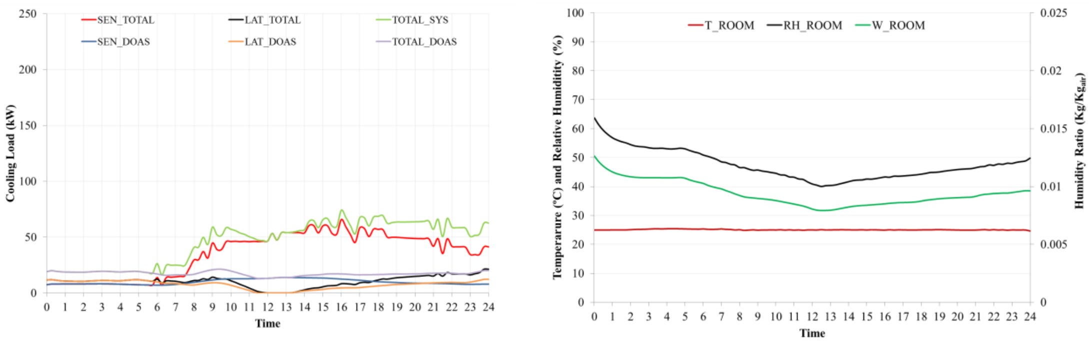
(a) Cooling load (b) Space conditions

Figure B.10 Conventional air-conditioning system with no ventilation on cold and dry day (Hotel).



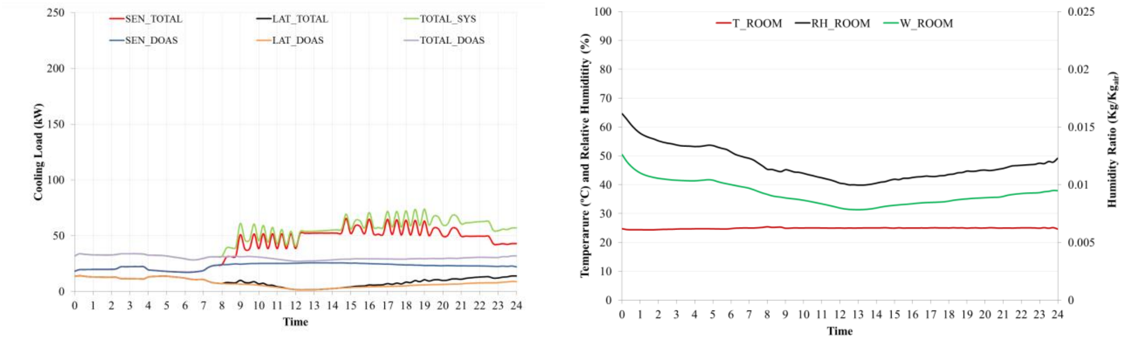
(a) Cooling load (b) Space conditions

Figure B.11 DOAS#1 (FC) on cold and dry day (Hotel).



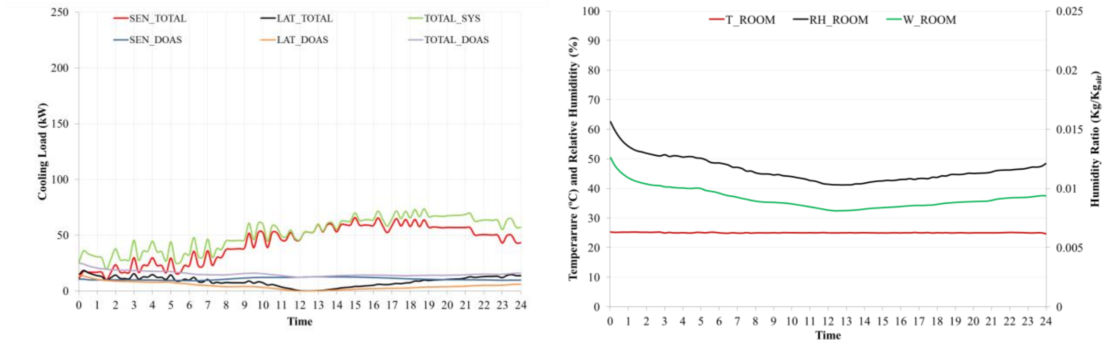
(a) Cooling load (b) Space conditions

Figure B.12 DOAS#2 (RC) on cold and dry day (Hotel).



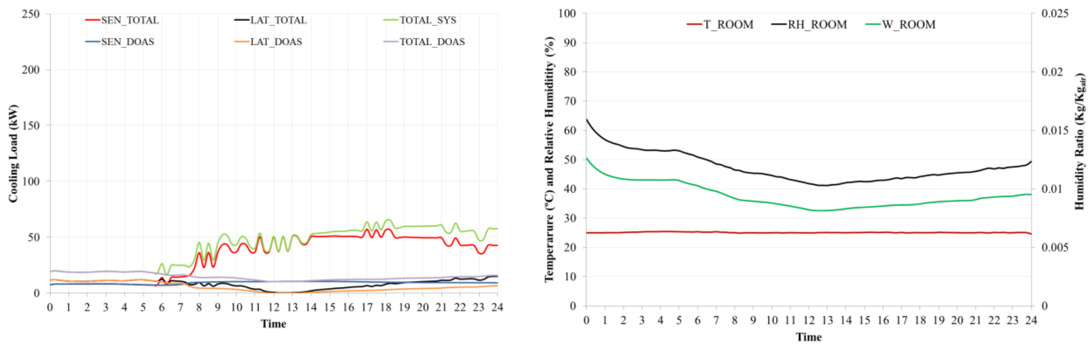
(a) Cooling load (b) Space conditions

Figure B.13 DOAS#3 (RC) on cold and dry day (Hotel).



(a) Cooling load (b) Space conditions

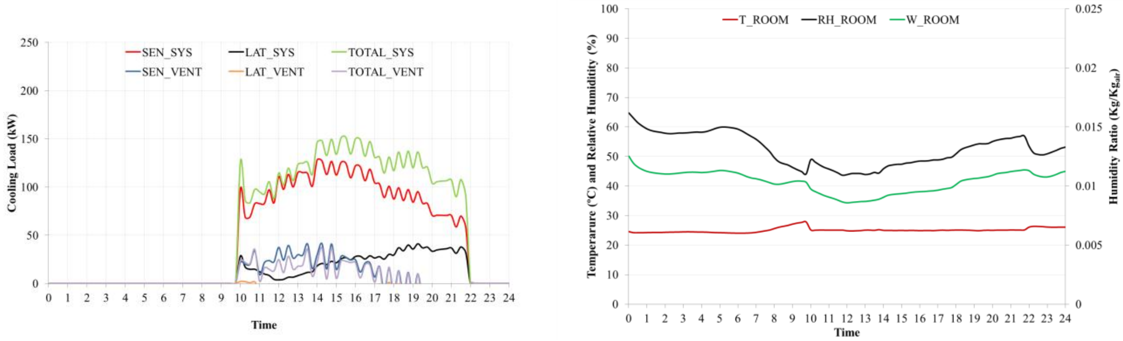
Figure B.14 DOAS#4 (EW) on cold and dry day (Hotel).



(a) Cooling load (b) Space conditions

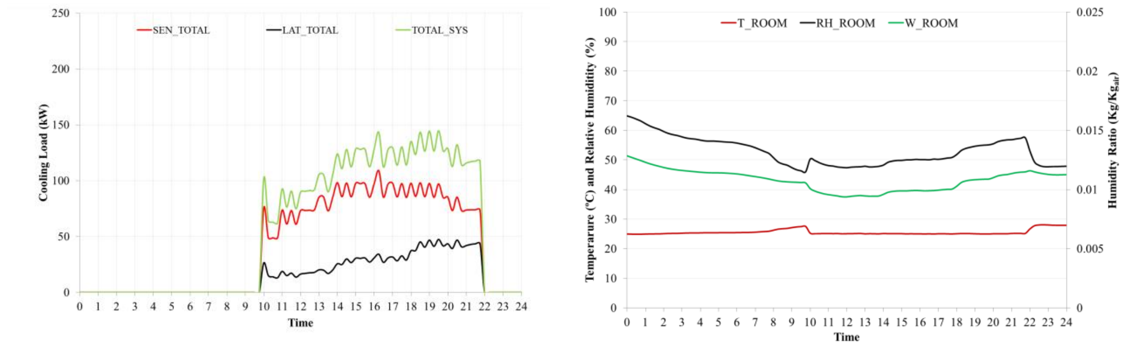
Figure B.15 DOAS#5 (EW+RC) on cold and dry day (Hotel).

Department Store Building



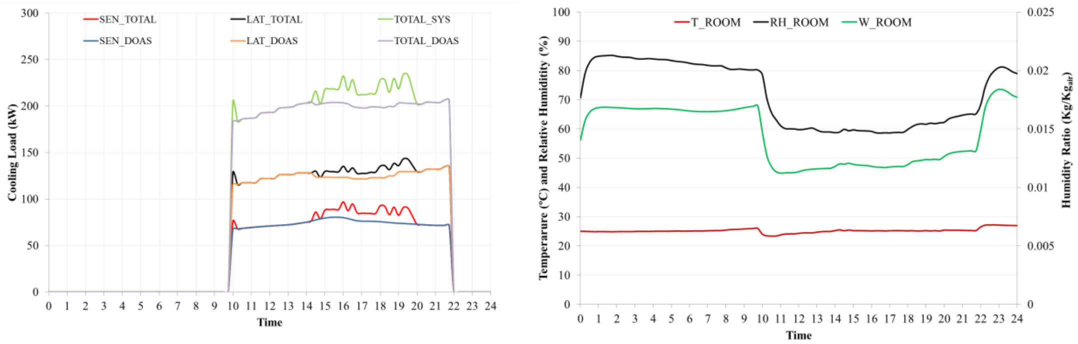
(a) Cooling load (b) Space conditions

Figure B.16 Conventional air-conditioning system on cold and dry day (Department store).



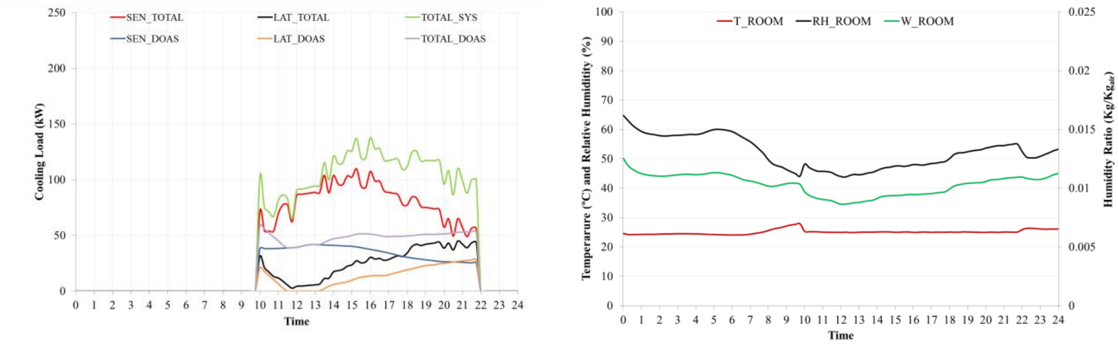
(a) Cooling load (b) Space conditions

Figure B.17 Conventional air-conditioning system with no ventilation on cold and dry day (Department store).



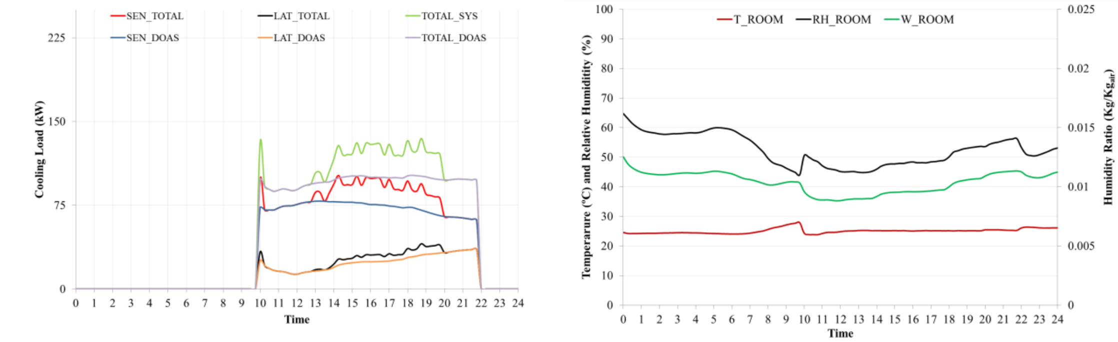
(a) Cooling load (b) Space conditions

Figure B.18 DOAS#1 (FC) on cold and dry day (Department store).



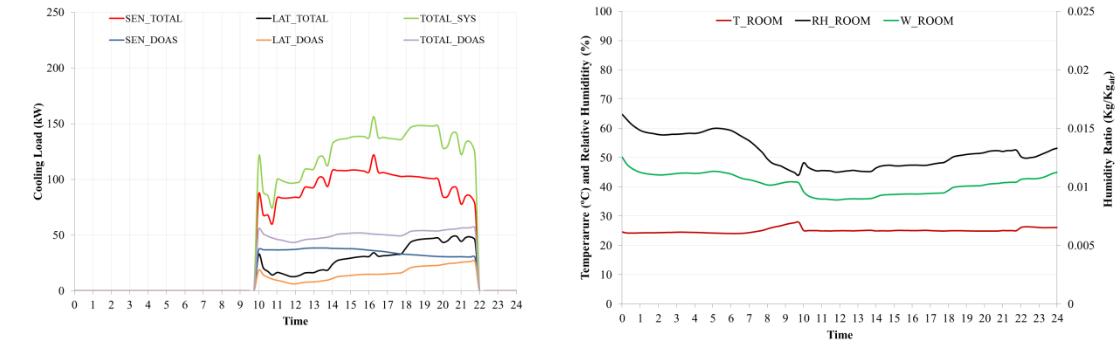
(a) Cooling load (b) Space conditions

Figure B.19 DOAS#2 (RC) on cold and dry day (Department store).



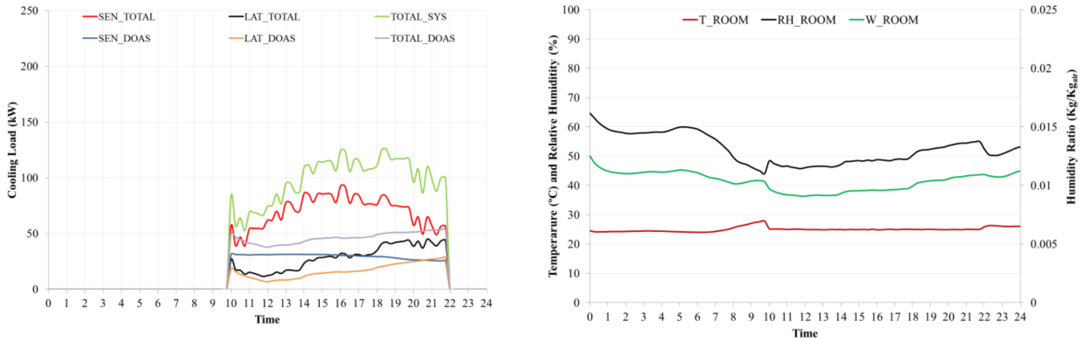
(a) Cooling load (b) Space conditions

Figure B.8 DOAS#3 (EW) on cold and dry day (Department store).



(a) Cooling load (b) Space conditions

Figure B.9 DOAS#4 (EW+SW) on cold and dry day (Department store).



(a) Cooling load (b) Space conditions

Figure B.10 DOAS#5 (EW+RC) on cold and dry day (Department store).

APPENDIX C: DOAS on Hot and Dry Day

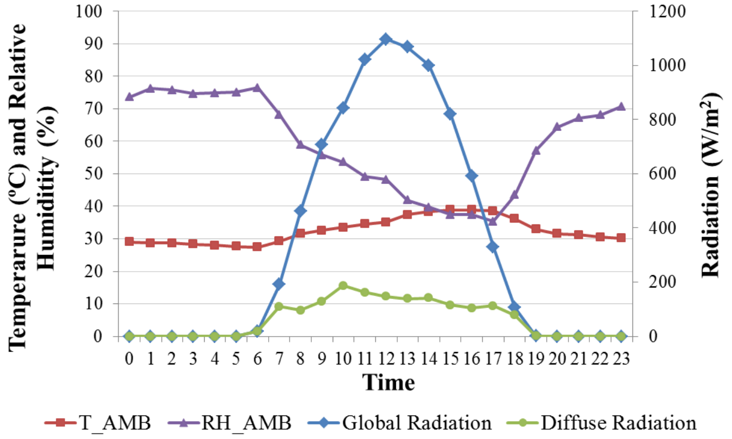
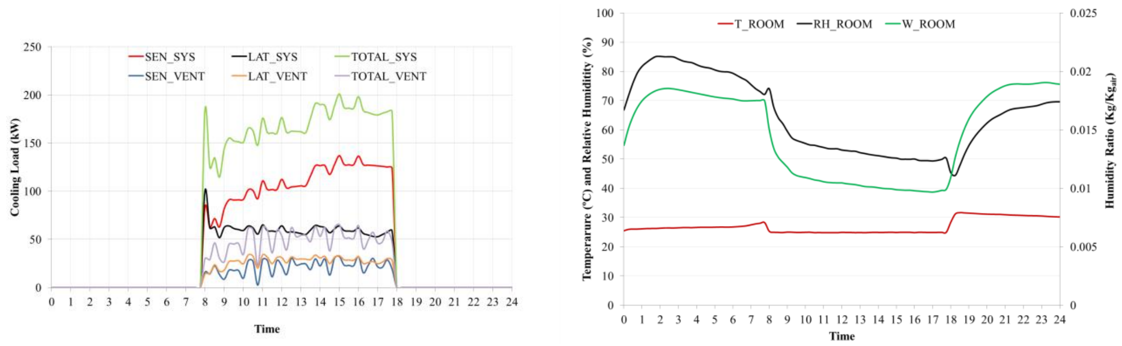


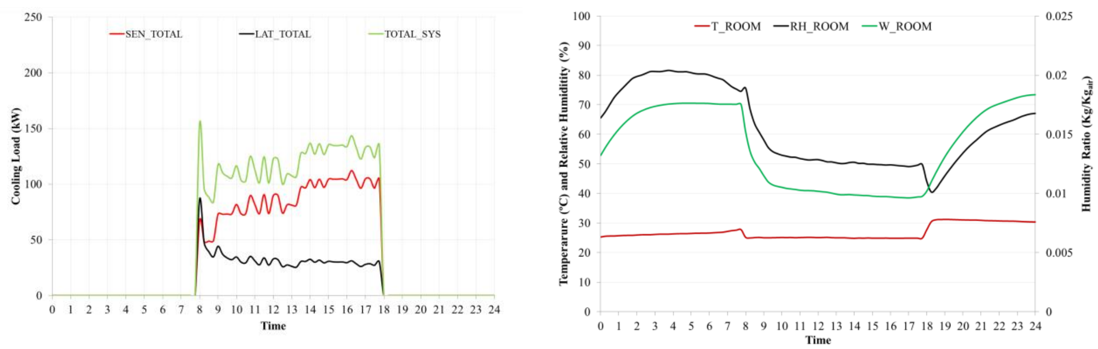
Figure C.1 The solar radiation and ambient air condition on hot and dry day (29th May).

Office Building



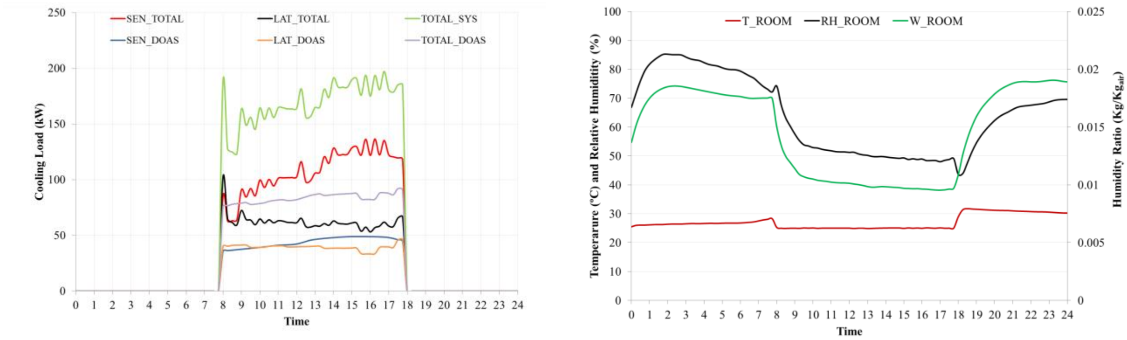
(a) Cooling load (b) Space conditions

Figure C.2 Conventional air-conditioning system on hot and dry day (Office).



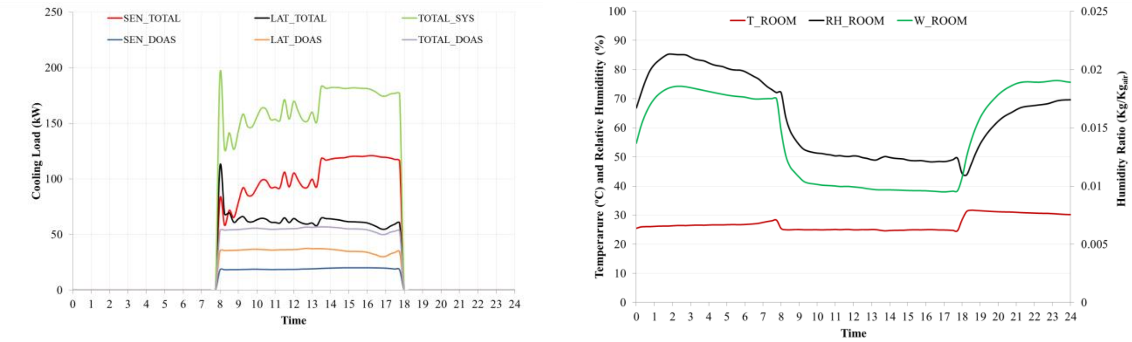
(a) Cooling load (b) Space conditions

Figure C.3 Conventional air-conditioning system with no ventilation on hot and dry day (Office).



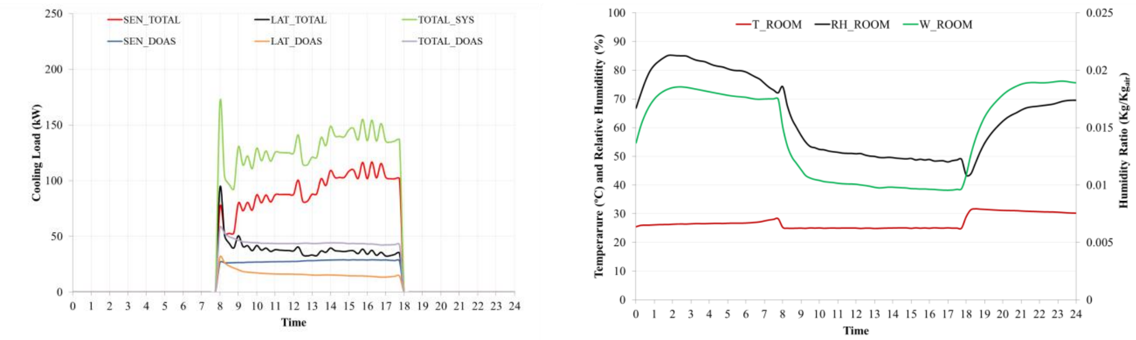
(a) Cooling load (b) Space conditions

Figure C.4 DOAS#1 (FC) on hot and dry day (Office).



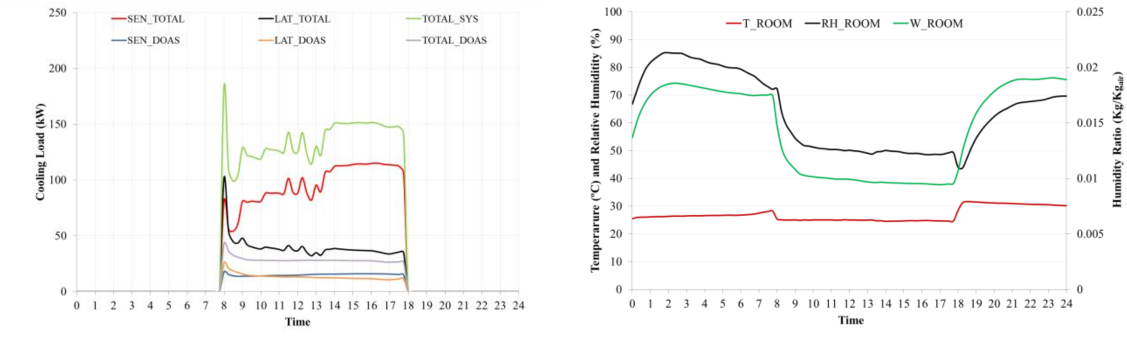
(a) Cooling load (b) Space conditions

Figure C.5 DOAS#2 (RC) on hot and dry day (Office).



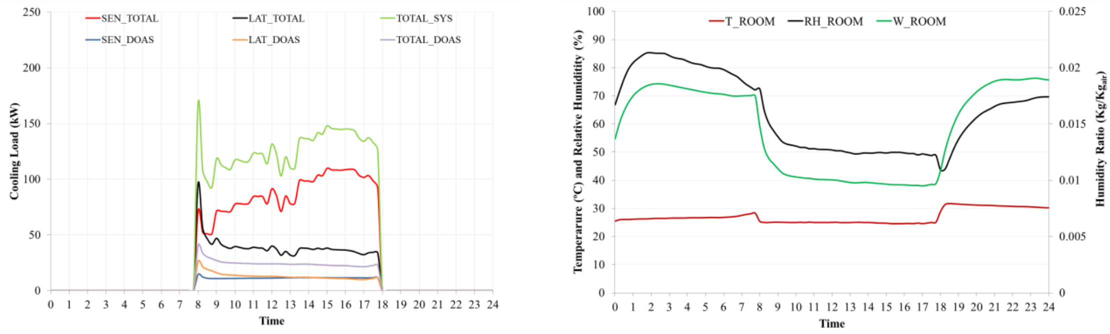
(a) Cooling load (b) Space conditions

Figure C.6 DOAS#3 (EC) on hot and dry day (Office).



(a) Cooling load (b) Space conditions

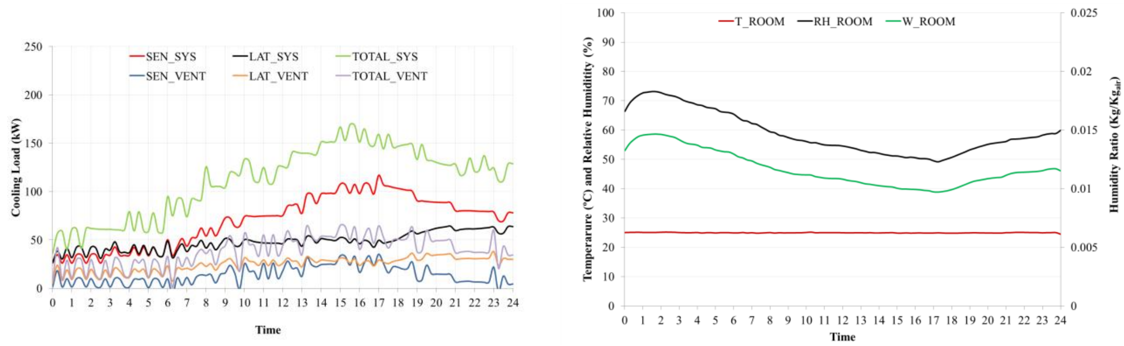
Figure C.7 DOAS#4 (EW+SW) on hot and dry day (Office).



(a) Cooling load (b) Space conditions

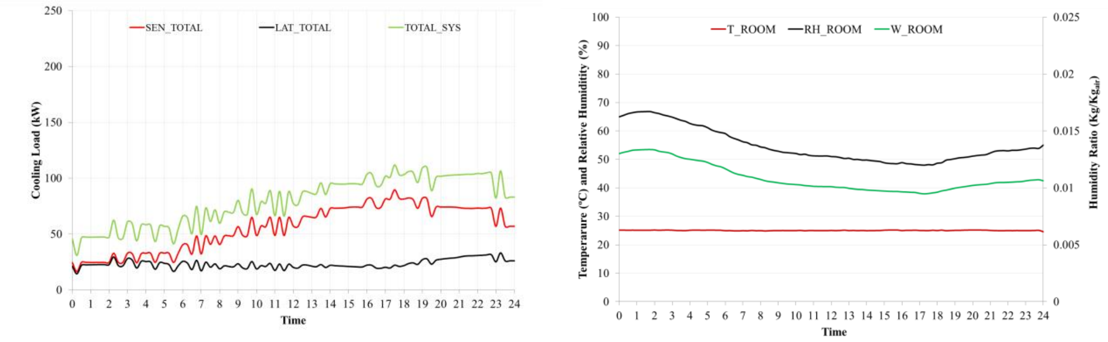
Figure C.8 DOAS#5 (EW+RC) on hot and dry day (Office).

Hotel Building



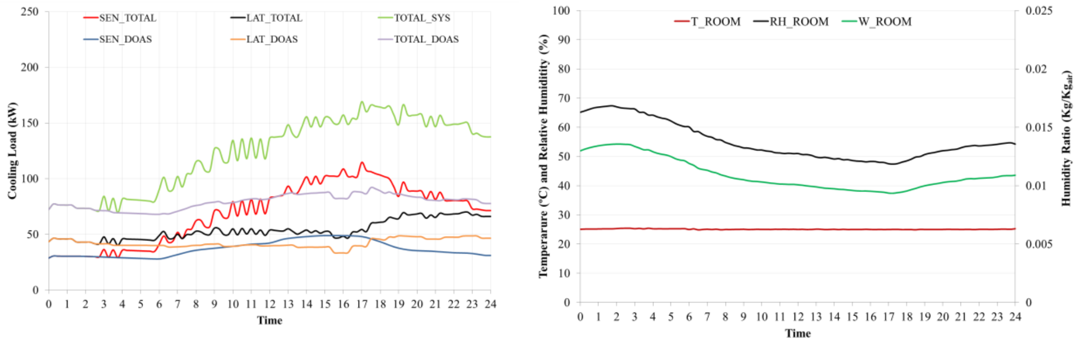
(a) Cooling load (b) Space conditions

Figure C.9 Conventional air-conditioning system on hot and dry day (Hotel).



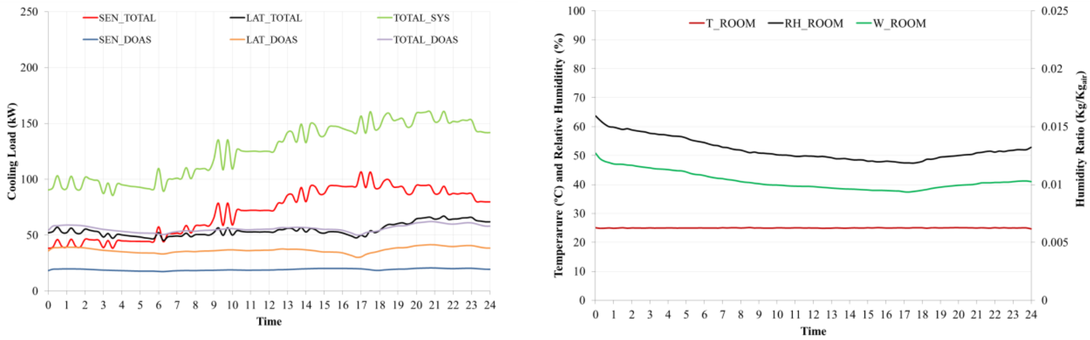
(a) Cooling load (b) Space conditions

Figure C.10 Conventional air-conditioning system with no ventilation on hot and dry day (Hotel).



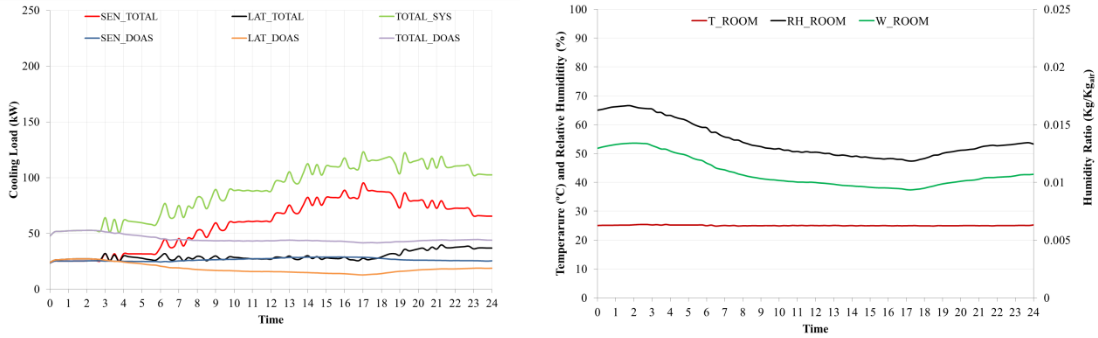
(a) Cooling load (b) Space conditions

Figure C.11 DOAS#1 (FC) on hot and dry day (Hotel).



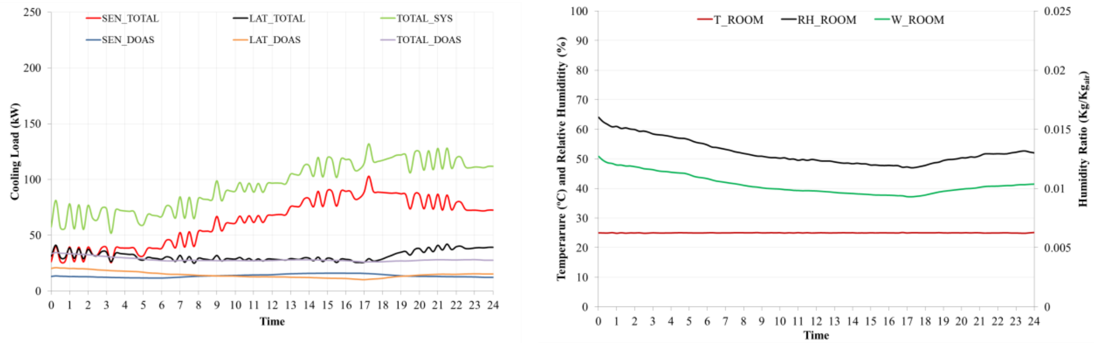
(a) Cooling load (b) Space conditions

Figure C.12 DOAS#2 (RC) on hot and dry day (Hotel).



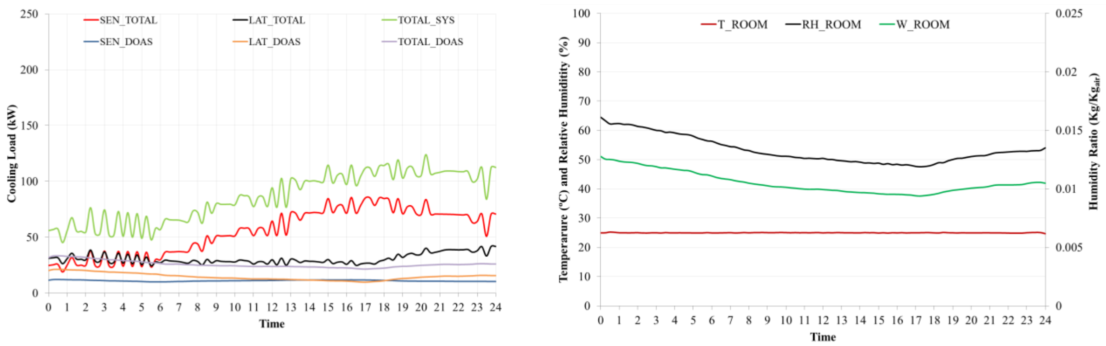
(a) Cooling load (b) Space conditions

Figure C.13 DOAS#3 (EW) on hot and dry day (Hotel).



(a) Cooling load (b) Space conditions

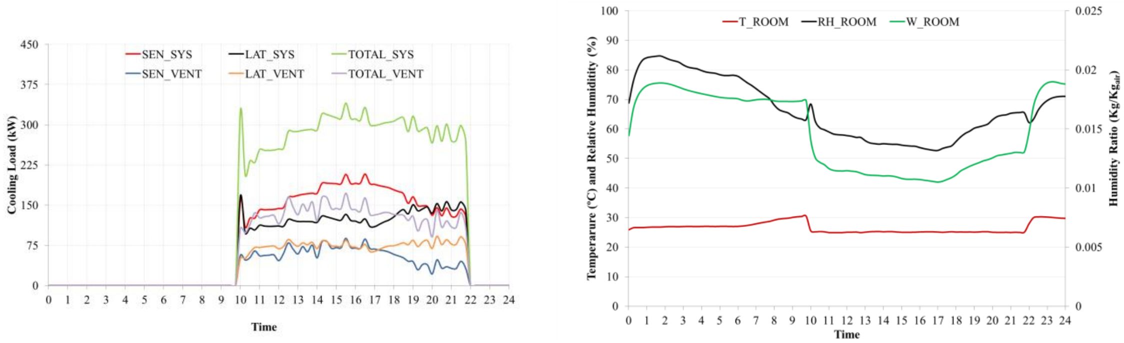
Figure C.14 DOAS#4 (EW+SW) on hot and dry day (Hotel).



(a) Cooling load (b) Space conditions

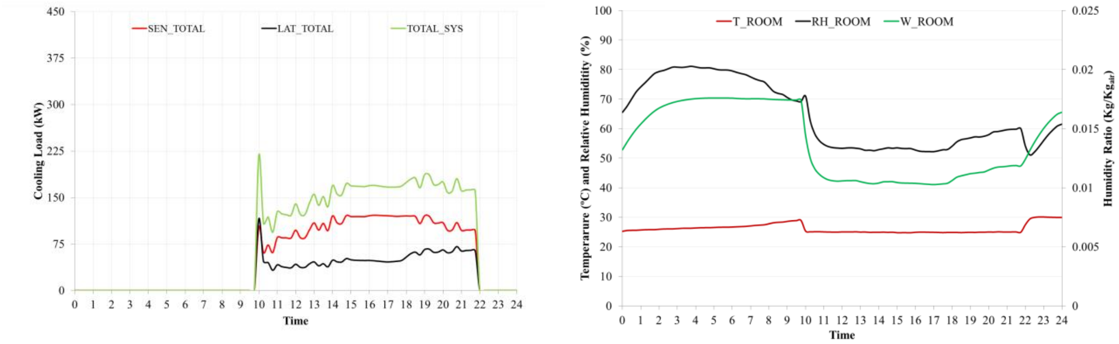
Figure C.15 DOAS#5 (EW+RC) on hot and dry day (Hotel).

Department Store Building



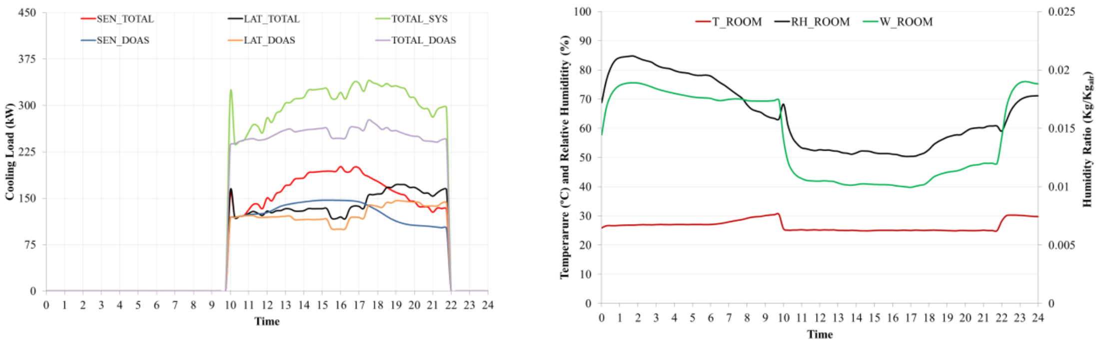
(a) Cooling load (b) Space conditions

Figure C.16 Conventional air-conditioning system on hot and dry day (Department store).



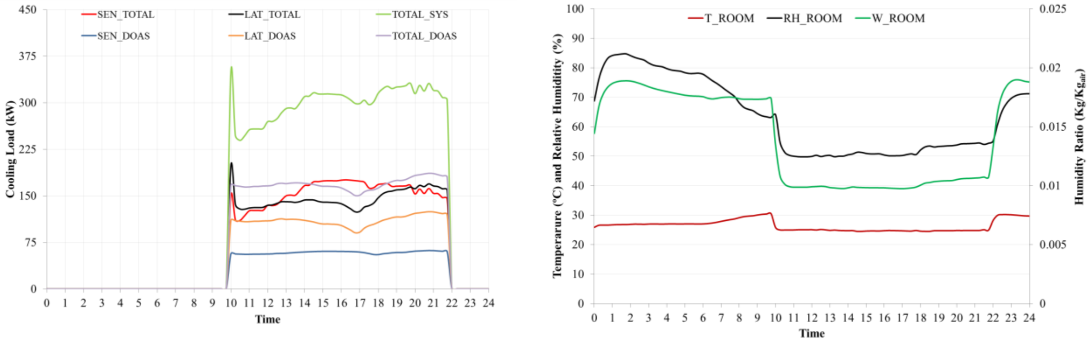
(a) Cooling load (b) Space conditions

Figure C.17 Conventional air-conditioning system with no ventilation on hot and dry day (Department store).



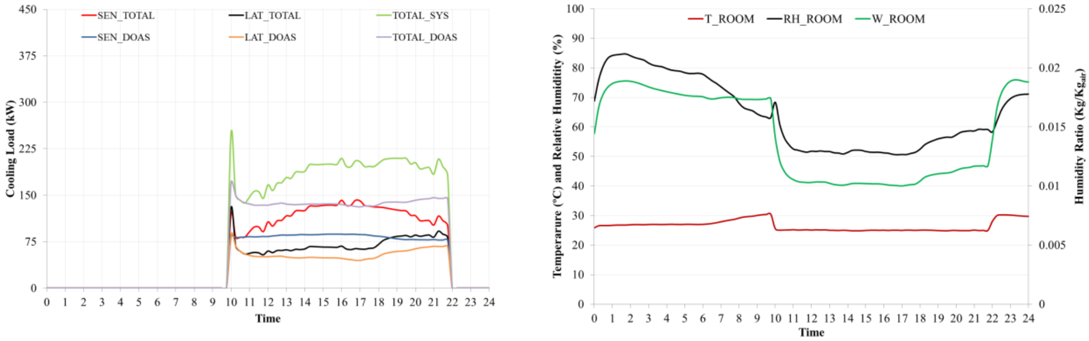
(a) Cooling load (b) Space conditions

Figure C.18 DOAS#1 (FC) on hot and dry day (Department store).



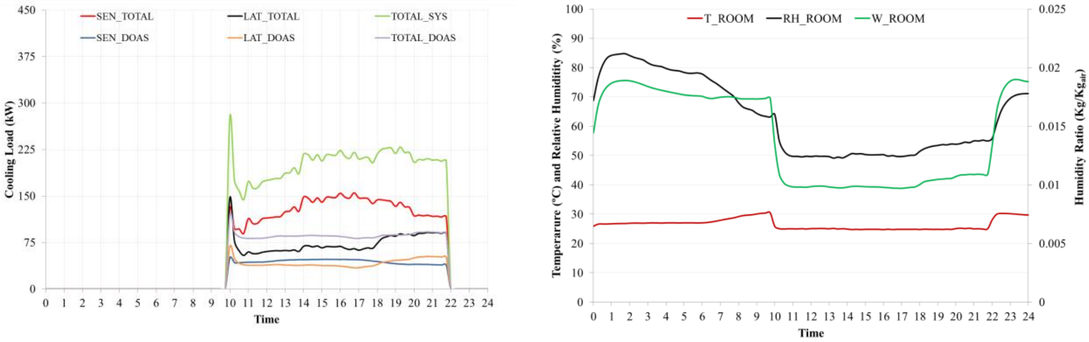
(a) Cooling load (b) Indoor conditions

Figure C.19 DOAS#2 (RC) on hot and dry day (Department store).



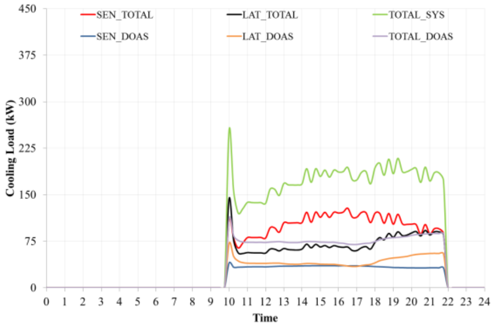
(a) Cooling load (b) Space conditions

Figure C.20 DOAS#3 (EW) on hot and dry day (Department store).

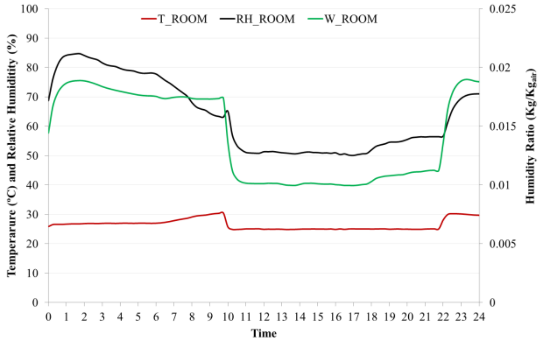


(a) Cooling load (b) Space conditions

Figure C.21 DOAS#4 (EW+SW) on hot and dry day (Department store).



(a) Cooling load



(b) Indoor conditions

Figure C.22 DOAS#5 (EW+RC) on hot and dry day (Department store).

APPENDIX D: DOAS on Late Rain Day (Cold and Humid)

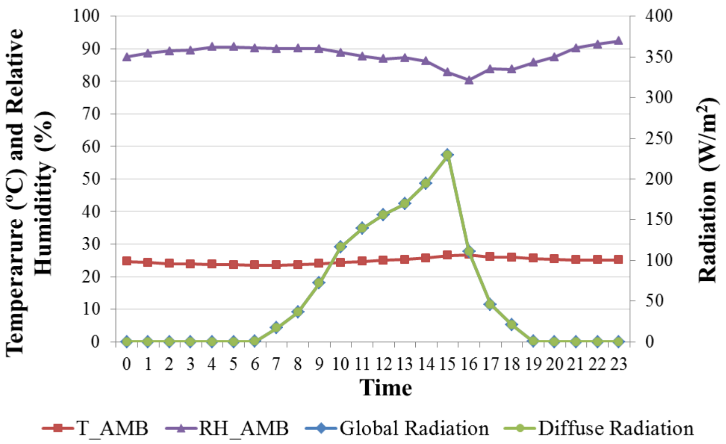
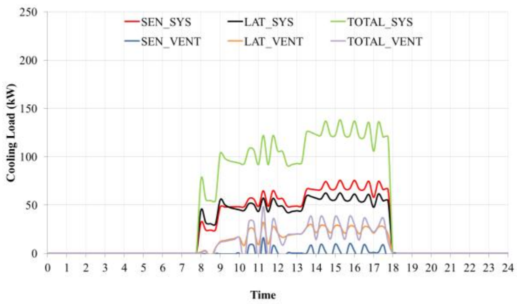


Figure D.1 The solar radiation and ambient air condition on cold and humid day (24th August).

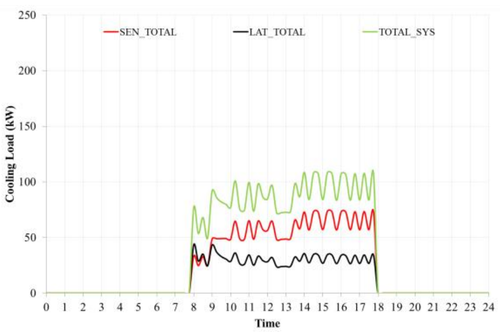
Office Building



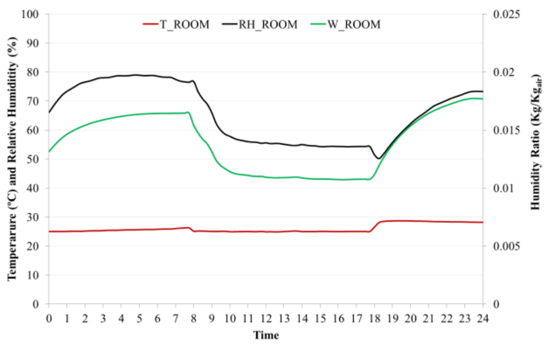
(a) Cooling load

(b) Space conditions

Figure D.2 Conventional air-conditioning system on cold and humid day (Office).

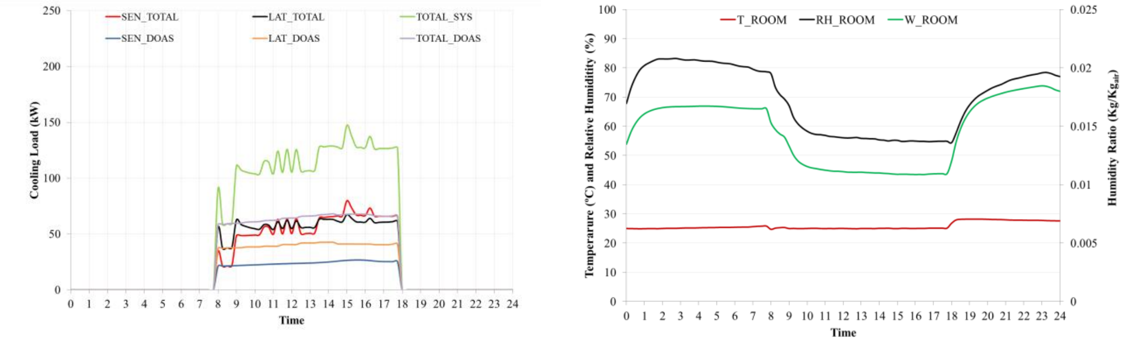


(a) Cooling load



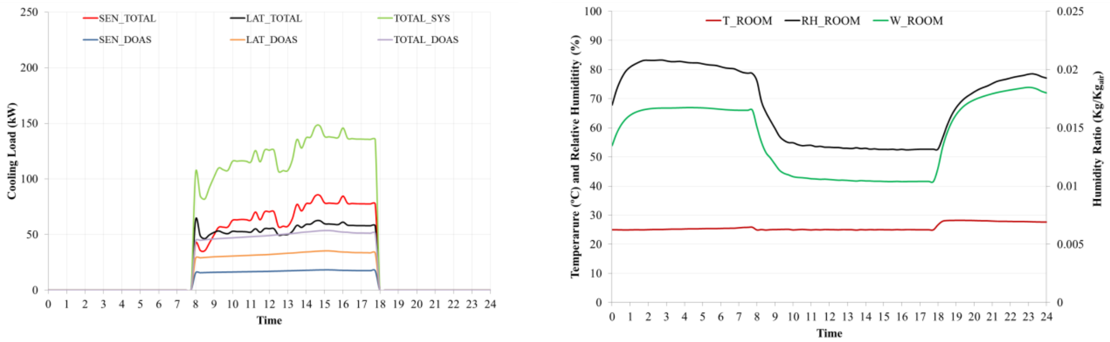
(b) Space conditions

Figure D.3 Conventional air-conditioning system with no ventilation on cold and humid day (Office).



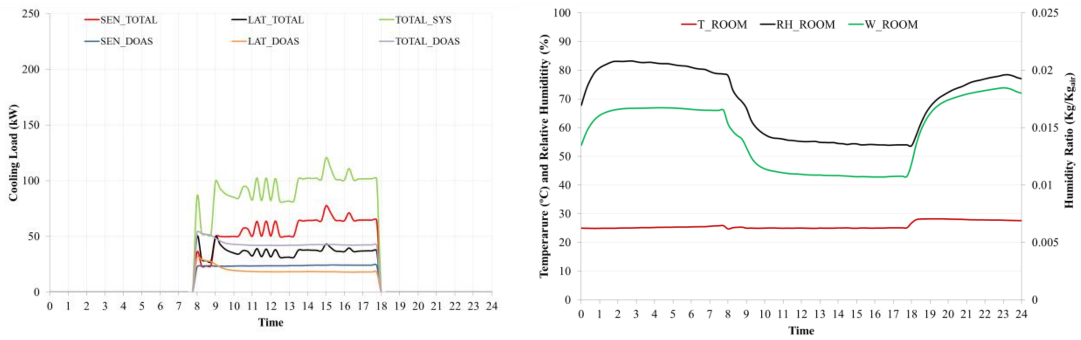
(a) Cooling load (b) Space conditions

Figure D.4 DOAS#1 (FC) on cold and humid day (Office).



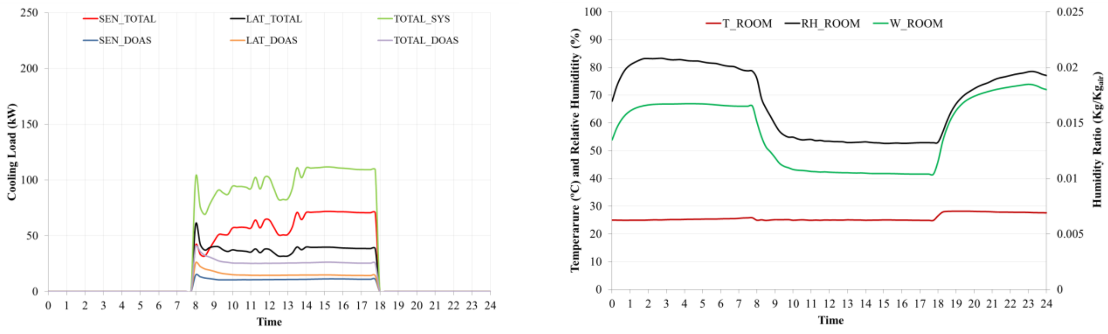
(a) Cooling load (b) Space conditions

Figure D.5 DOAS#2 (RC) on cold and humid day (Office).



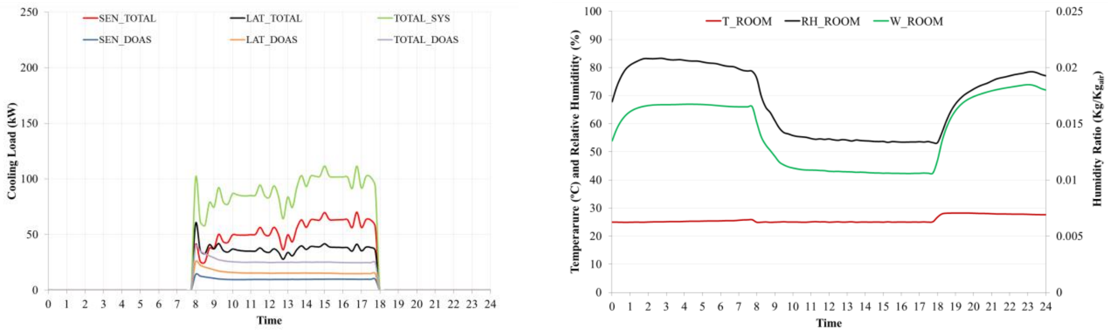
(a) Cooling load (b) Space conditions

Figure D.6 DOAS#3 (EW) on cold and humid day (Office).



(a) Cooling load (b) Space conditions

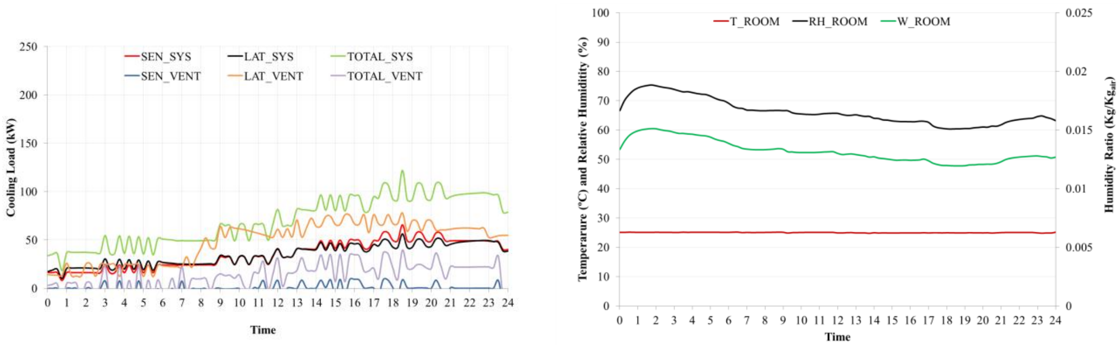
Figure D.7 DOAS#4 (EW+SW) on cold and humid day (Office).



(a) Cooling load (b) Space conditions

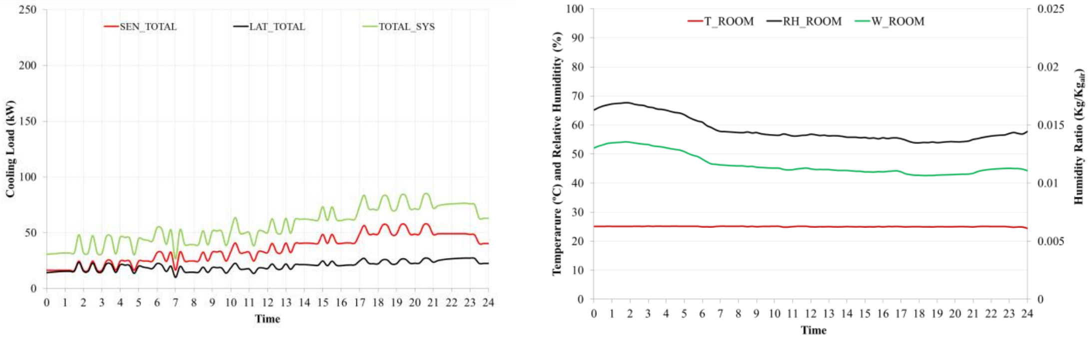
Figure D.8 DOAS#5 (EW+RC) on cold and humid day (Office).

Hotel Building



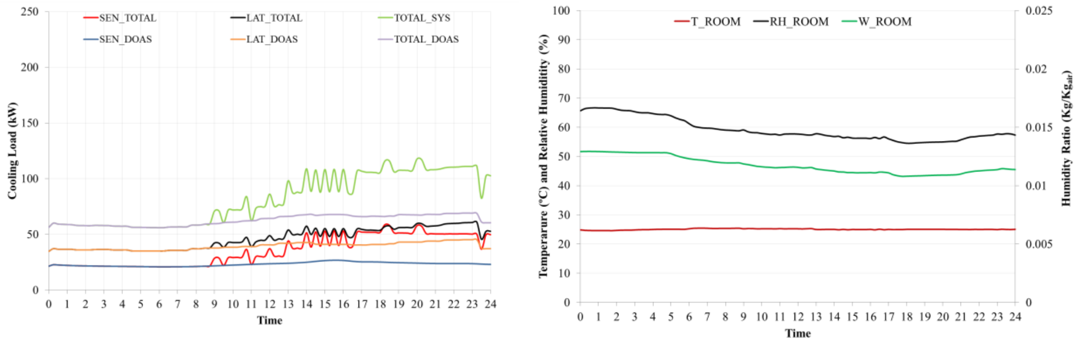
(a) Cooling load (b) Space conditions

Figure D.9 Conventional air-conditioning system on cold and humid day (Hotel).



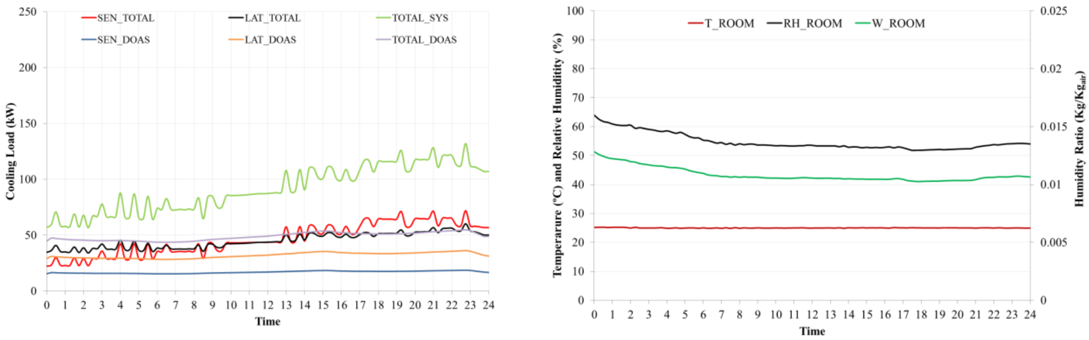
(a) Cooling load (b) Space conditions

Figure D.10 Conventional air-conditioning system with no ventilation on cold and humid day (Hotel).



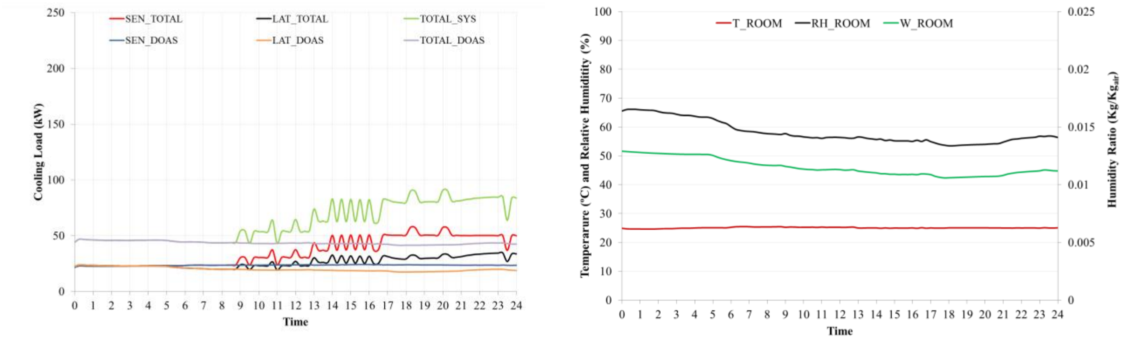
(a) Cooling load (b) Space conditions

Figure D.11 DOAS#1 (FC) on cold and humid day (Hotel).



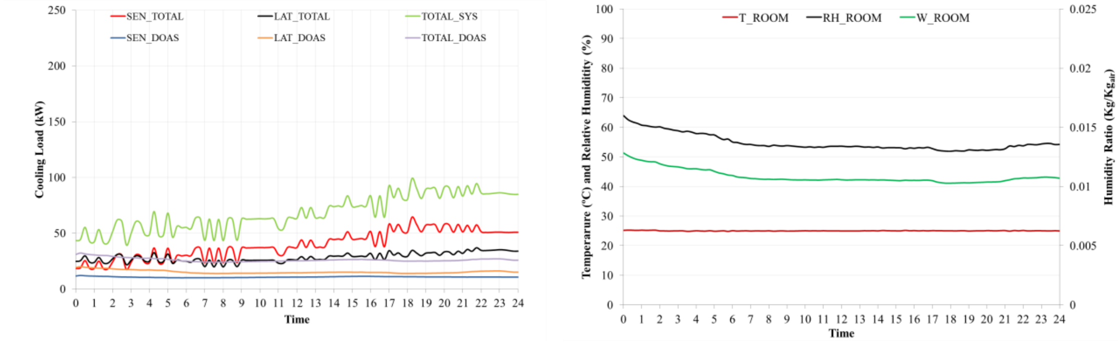
(a) Cooling load (b) Space conditions

Figure D.12 DOAS#2 (RC) on cold and humid day (Hotel).



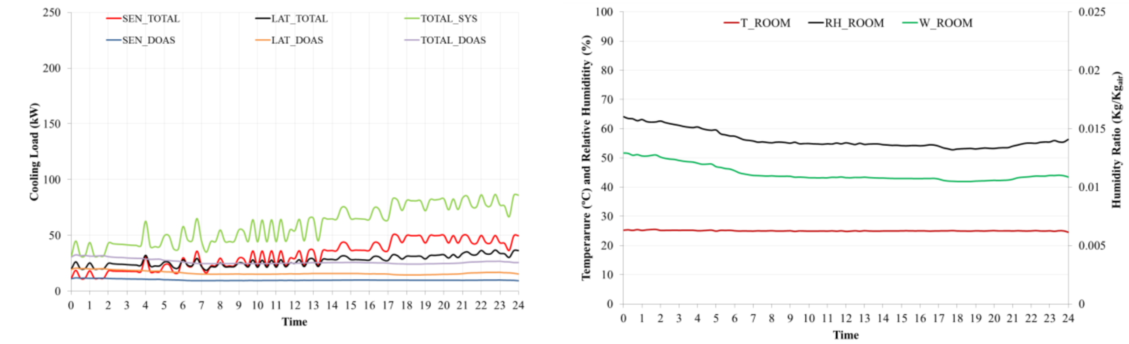
(a) Cooling load (b) Space conditions

Figure D.13 DOAS#3 (EW) on cold and humid day (Hotel).



(a) Cooling load (b) Space conditions

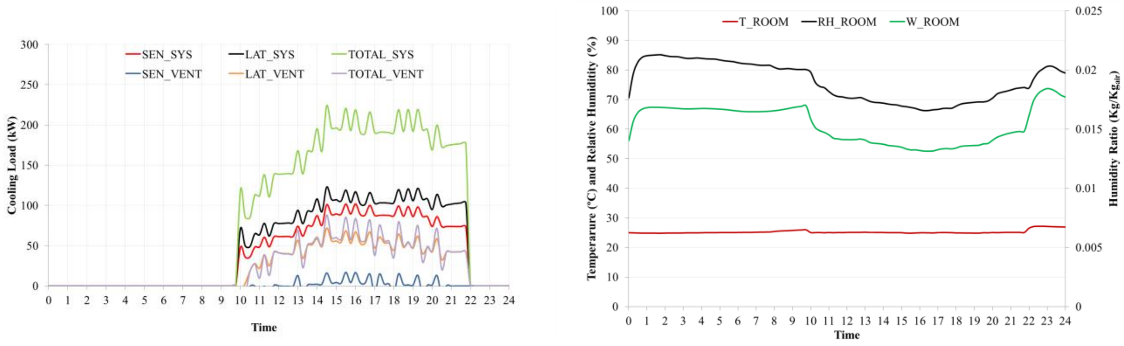
Figure D.14 DOAS#4 (EW+SW) on cold and humid day (Hotel).



(a) Cooling load (b) Space conditions

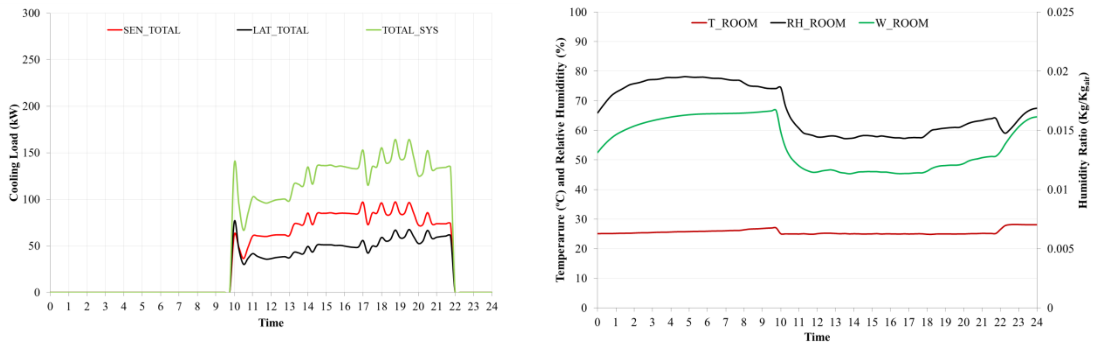
Figure D.15 DOAS#5 (EW+RC) on cold and humid day (Hotel).

Department Store Building



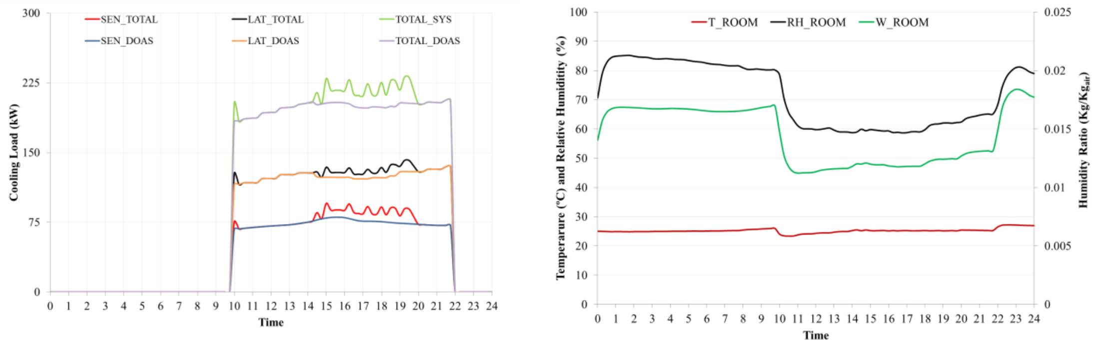
(a) Cooling load (b) Space conditions

Figure D.16 Conventional air-conditioning system on cold and humid day (Department store).



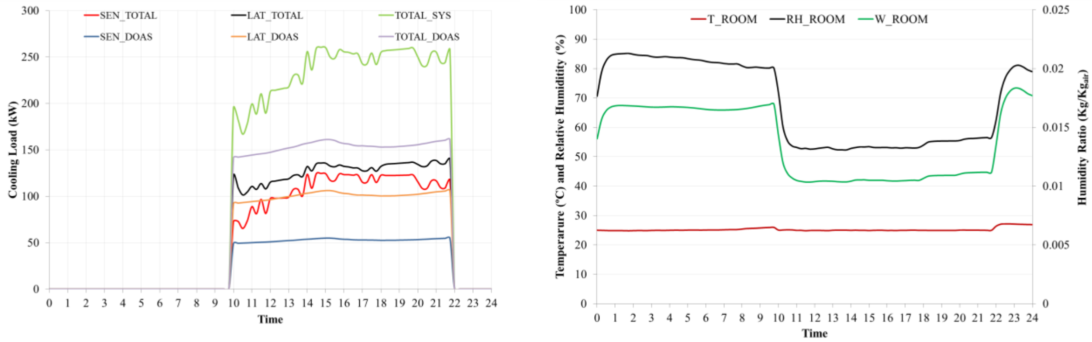
(a) Cooling load (b) Space conditions

Figure D.17 Conventional air-conditioning system with no ventilation on cold and humid day (Department store).



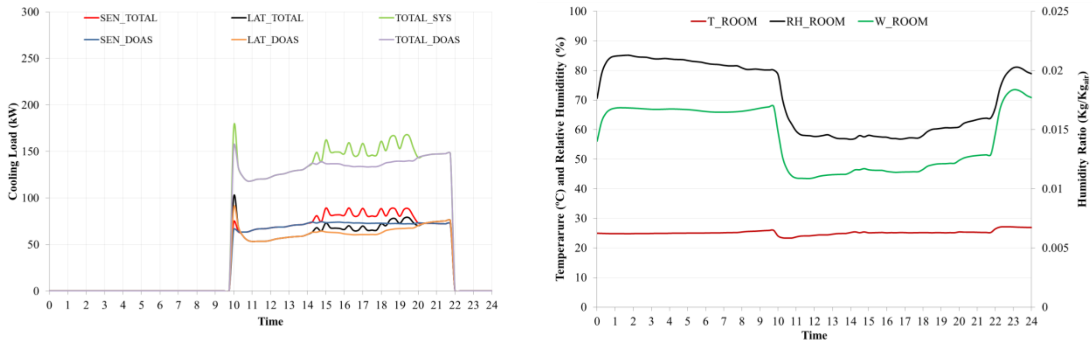
(a) Cooling load (b) Space conditions

Figure D.18 DOAS#1 (FC) on cold and humid day (Department store).



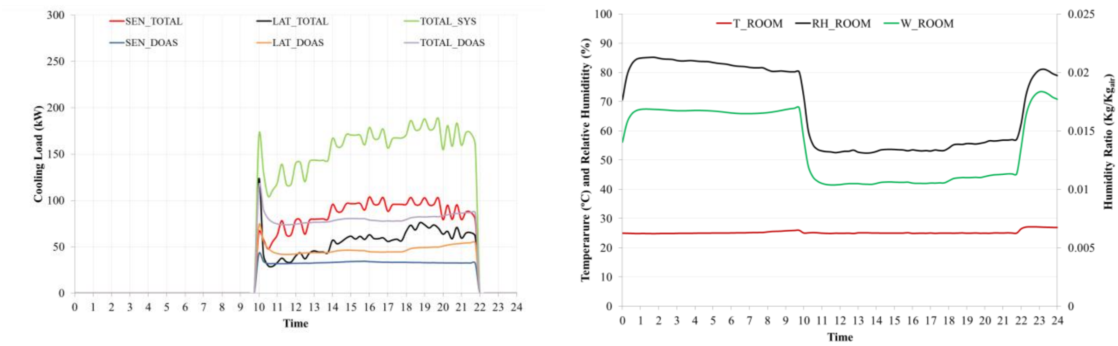
(a) Cooling load (b) Space conditions

Figure D.19 DOAS#2 (RC) on cold and humid day (Department store).



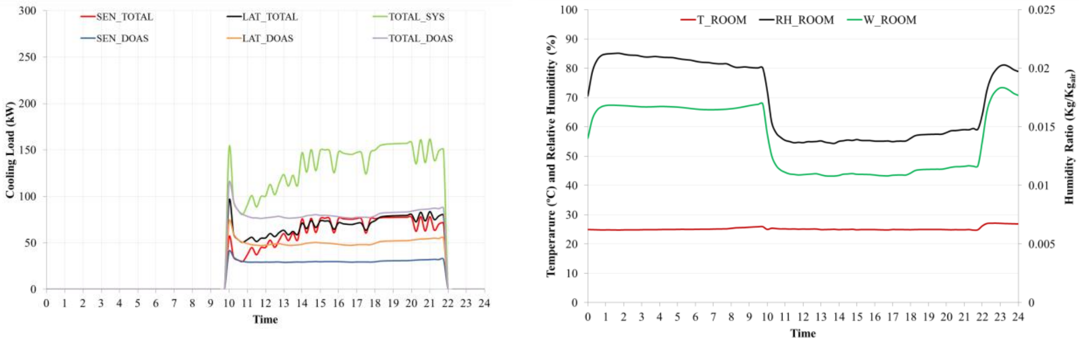
(a) Cooling load (b) Space conditions

Figure D.20 DOAS#3 (EW) on cold and humid day (Department store).



(a) Cooling load (b) Indoor conditions

Figure D.21 DOAS#4 (EW+SW) on cold and humid day (Department store).



(a) Cooling load (b) Indoor conditions

Figure D.22 DOAS#5 (EW+RC) on cold and humid day (Department store).

Appendix E: Evaluation of Cost Effectiveness

An Example of Life Cycle Cost Calculation

An example of life cycle cost calculation refers to the conventional air-conditioning system for the office building. The payment is shown in Table E.1.

The payment of this system can be allocated as 2 parts:

(1) The investment cost for air handling unit (AHU) 97,000 baht (PV),

(2) The operation cost 780,167 baht/year (A).

The analysis of life cycle cost has defined parameters as follows:

(1) Project time period 15 years (n),

(2) Discount rate 7% (d),

(3) Cost escalation 3% (e).

From the information above, the overall information of this system in term of cash flow diagram can be drawn as Figure E.1.

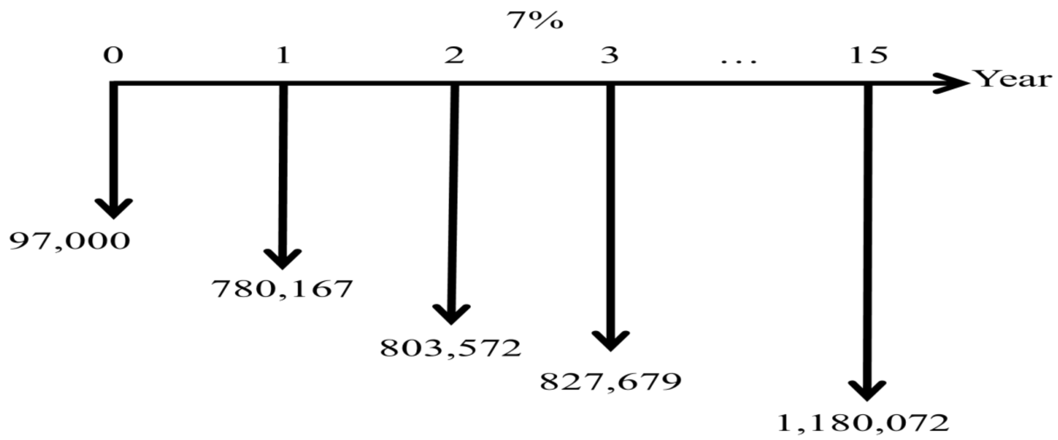


Figure E.1 Cash flow diagram of the conventional air-conditioning system (Office).

The life cycle cost in terms of present value can be calculated according to Equation 3.1 and the net present value (NPV) is a summation of the present value that originated from any period of time.

$$PV = \mathrm{A}\,(\frac{1+e^{n}}{d-e})(1-\frac{(1+e)^{n}}{(1+d)^{n}})$$

Substitute

$$PV = 780{,}167\left(\frac{1+0.03}{0.07-0.03}\right)\left(1-\frac{(1+0.03)^{15}}{(1+0.07)^{15}}\right)$$

$$PV = 8{,}745{,}672$$

hence

$$NPV = PV_1 + PV_2$$

where

PV_1 = investment cost

PV_2 = operation cost in term of present

thus

$$NPV = 97{,}000 + 8{,}745{,}672$$

$$NPV = 8{,}842{,}672$$

Annual worth (AW) can be calculated by distributing the net present value (NPV) into the annual costs (A) as follows Equation 3.2.

$$A = PV\left(\frac{d(1+d)^n}{(1+d)^n-1}\right)$$

$$A = (PV_1 + PV_2)\left(\frac{d(1+d)^n}{(1+d)^n-1}\right)$$

$$A = (NPV)\left(\frac{d(1+d)^n}{(1+d)^n-1}\right)$$

substitute

$$A = (8{,}842{,}672)\left(\frac{0.07(1+0.07)^{15}}{(1+0.07)^{15}-1}\right)$$

$$A = 970{,}925$$

therefore

The net present value (NPV) and annual worth (AW) of the conventional air-conditioning system for office building are equal to 8,842,672 baht and 970,925 baht/year, respectively.

Office Building

Table E.1 The payment for conventional air-conditioning system (Office).

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
AHU	97,000	780,167

Table E.2 The payment for DOAS#1 (FC) for office building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	200,000	784,334
AHU	95,000	

Table E.3 The payment for DOAS#3 (RC) for office building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	150,000	765,655
Runaround Coil	72,500	
AHU	116,000	

Table E.4 The payment for DOAS#3 (EW) for office building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	120,000	633,183
Enthalpy Wheel	350,000	
AHU	95,000	

Table E.5 The payment for DOAS#4 (EW+SW) for office building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	75,000	649,021
Enthalpy Wheel	350,000	
Sensible Wheel	245,000	
AHU	111,000	

Table E.6 The payment for DOAS#5 (EW+RC) for office building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	65,000	607,094
Enthalpy Wheel	350,000	
Runaround Coil	72,500	
AHU	114,000	

Table E.7 The net present value and annual worth value (Office).

Configuration	Investment Cost (Baht)	Net present value: NPV (Baht)	Annual worth value: AW (Baht/year)
Conventional	97,000	8,842,672	970,925
DOAS #1	295,000	9,087,394	997,795
DOAS #2	338,500	8,921,498	979,580
DOAS #3	565,000	7,662,983	841,395
DOAS #4	781,000	8,056,535	884,607
DOAS #5	601,500	7,407,032	813,292

Hotel Building

Table E.8 The payment for conventional air-conditioning system (Hotel).

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
AHU	58,000	1,283,970

Table E.9 The payment for DOAS#1 (FC) for hotel building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	200,000	1,341,698
AHU	52,500	

Table E.10 The payment for DOAS#2 (RC) for hotel building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	150,000	1,335,114
Runaround Coil	72,500	
AHU	66,000	

Table E.11 The payment for DOAS#3 (EW) for hotel building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	126,000	1,036,053
Enthalpy Wheel	350,000	
AHU	52,500	

Table E.12 The payment for DOAS#4 (EW+SW) for hotel building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	76,000	1,082,143
Enthalpy Wheel	350,000	
Sensible Wheel	245,000	
AHU	66,000	

Table E.13 The payment for DOAS#5 (EW+RC) for hotel building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	70,000	958,418
Enthalpy Wheel	350,000	
Runaround Coil	72,500	
AHU	60,000	

Table E.14 The net present value and annual worth value (Hotel).

Configuration	Investment Cost (Baht)	Net present value: NPV (Baht)	Annual worth value: AW (Baht/year)
Conventional	58,000	14,451,307	1,586,753
DOAS #1	252,500	15,292,934	1,679,164
DOAS #2	288,500	15,255,134	1,675,013
DOAS #3	528,500	12,142,665	1,333,264
DOAS #4	737,000	12,867,827	1,412,887
DOAS #5	552,500	11,296,375	1,240,342

Department Store Building

Table E.15 The payment for conventional air-conditioning system (Department Store).

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
AHU	175,000	1,472,374

Table E.16 The payment for DOAS#1 (FC) for department store building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	245,000	1,550,695
AHU	162,500	

Table E.17 The payment for DOAS#2 (RC) for department store building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	166,000	1,544,497
Runaround Coil	93,500	
AHU	247,500	

Table E.18 The payment for DOAS#3 (EW) for department store building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	150,000	1,068,041
Enthalpy Wheel	575,000	
AHU	155,000	

Table E.19 The payment for DOAS#4 (EW+SW) for department store building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	89,200	1,153,568
Enthalpy Wheel	575,000	
Sensible Wheel	402,500	
AHU	246,000	

Table E.20 The payment for DOAS#5 (EW+RC) for department store building.

Equipment list	Investment Cost (Baht)	Operational Cost (Baht)
DOAS	82,000	989,708
Enthalpy Wheel	575,000	
Runaround Coil	93,500	
AHU	227,500	

Table E.21 The net present value and annual worth value (Department Store).

Configuration	Investment Cost (Baht)	Net present value: NPV (Baht)	Annual worth value: AW (Baht/year)
Conventional	175,000	16,680,315	1,831,498
DOAS #1	407,500	17,790,795	1,953,429
DOAS #2	507,000	17,820,812	1,956,725
DOAS #3	880,000	12,852,750	1,411,232
DOAS #4	1,132,700	14,064,203	1,544,249
DOAS #5	978,000	12,072,630	1,325,574