desi | grated

Desiccant **I**ntegrated **F**acade **S**ystem

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'Desi-grated'
Desiccant Integrated Facade System

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and the Built Environment
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Acknowledgment

The thesis marks the last milestone to conclude my two years of the master program. Nevertheless, it is just merely a beginning for future endeavors. Throughout these two years, I've met numerous intelligent colleagues, made great friendships, exchanged uncountable ideas, and learn a lot of things. And I would like to express my sincere appreciation to everyone.

First of all, I would like to thank you both my first and second mentors, Alejandro Prieto Hoces and Christien Janssen who guided me throughout my project. I really learned a lot from your expertise. It was a pleasure being under your supervision.

I want to thank you, my family, who is always supportive of all circumstances if it were not because of you guys I wouldn't be here in the first place. To Paan, I am grateful to have you here, as you have always been my support mentally and academically. Thank you for being there reflecting my clouded thoughts. Lastly, a shout out to the lovely Thai community, the North and South group who welcomed me into their family, and all my BTs friends, it wouldn't be such an amazing journey without you guys.

To everyone I've crossed path with in this journey, I would like to say…

Thank you.

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1 Introduction

“Climate change is real.” (IPCC 2001) It is an inevitable fact, which had been presented in numerous peer-reviewed scientific journals in which over 97% of climate scientists agreed upon while the results of their findings had shown that global warming is real and primarily cause by human. (Cook et al. 2016)

NASA has presented a list of evidence to depict the seriousness of climate change which includes warming oceans, shrinking ice sheets, glacial retreats, decreased snow cover, sea level rise, declining Arctic sea ice, ocean acidification, and the most tangible one – global temperature rises. According to NASA’s research, the planet’s average surface temperature has risen by approximately 0.9C since the 19th century. While 17 out 18 warmest years in the 136 years record have occurred since 2001. (NASA/GISS)

In parallel to the rising temperature, the use of energy for space cooling has been tripled within the last twenty years; the rising demand has driven up carbon emission by three times since 1990. The growing demand is essentially influenced by the high temperature as well as economic and population growth. Due to the limited effect of government policies throughout the world, it is expected that the current trend of cooling demand will continue growing in the following years. (OCED&IEA, 2018)

We are stuck in a paradoxical situation, where higher demand for cooling is required, while carbon emission needs to be controlled. Therefore, this thesis aims to ease the entangled situation, by means of exploring the potential of the existing cooling technologies in reducing electricity usage as well as carbon emission. At the same time, a scheme to assemble the applicable technologies would be developed. It would offer an integration strategy into the built environment, providing an alternative to the conventional cooling system. The thesis has targeted an area with a hot and humid climate, where population and economic growth profoundly influenced the cooling demand such as Bangkok, Thailand.

1.1 Problem Statement

Bangkok is Thailand's largest and most important city. It is the capital city both in terms of governance and commercial. Bangkok is a rapidly expanding city; it is currently ranked 28 in the world population rank with an average growth rate of 2.0% (encyclopedia.com). Currently, it has a population of 10,156,316 people and is predicted to reach 12,679,614 in 2030. (worldpopulationreview.com)

Bangkok is the center of Thailand's economy, housing over one-third of Thailand's bank as well as the stock exchange. Over the last four decades, Thailand has made exceptional progress in social and economic development. After a long slow average growth, Thailand's economic growth is increasing its pace. Its economic growth had reached 4.8% in the first quarter of 2018 the highest pace since 2013. Economic growth is reflected in the substantial growth of office buildings. Collier International Thailand had reported a growing demand for office spaces in Bangkok due to limited supplies and government policy to attract foreign investment. Currently, the office supply in Bangkok is at 8,618,862 sq.m, and it is predicted to increase by 662,676 sq.m within 2019-2022. (Collier, 2017)

Bangkok's hot and humid climate context as well as it's growing population and economics has imposed a pressing challenge in resolving global warming, as cooling demand is accounted for 50% of the total electricity usage in office buildings. (Qi, 2006) According to the Eco-Business report, without new cooling technologies or strategies, air conditioning alone would be accounted for 40% of South East Asia's electricity demand. (Hill, 2018)

As a result of the growing demand for office spaces, several new office buildings will be added to Bangkok's rectilinear fully glazed skyline. Throughout the past decades, Bangkok's skyscraper had adopted the international façade typology which reflects the Modernism approach of "predominant glass-and-steel aesthetic". (Wood, 2015) However, like many other international façade typologies, the fully glazed buildings failed to adapt themselves to the context; both physically, culturally, and most importantly, environmentally. (Wood, 2015) With the high temperature and solar radiation of Bangkok, the glazed façade had shown limited insulation characteristic; resulting in a comparative higher cooling demand for space cooling.

With a hot and humid climate, rapidly growing population and economics, as well as an acontextual development in the built environment, Bangkok deliberately requires a strategy to tackle the rising of cooling demands in the imminent future. To tackle this issue, a desiccant integrated façade system would be proposed. Due to its efficient control over humidity and low energy consumption, the solid desiccant cooling system has presented itself as a favorable alternative to the conventional vapor compression air conditioning systems. With recent developments, the system could be coupled or integrated with other means of cooling technologies to increase its efficiency. Additionally, due to abundant solar radiation in Bangkok, the usage of solar energy harvesting technologies would be integrated as a thermal source for the regeneration process, while serving as a part of the heat prevention strategy. The proposed façade system aims to reduce the cooling loads of office buildings to facilitate the building's HVAC system.

1.2 Research Objective

With such an imminent threat, the research objective aims to provide a strategy to ease the pressing problem of cooling demands which Bangkok is encountering at the moment through the following objectives:

- **Elaborate the challenges imposed by the hot and humid climate context** in order to create a sustainable connection between the buildings and the built environment while providing sufficient thermal comfort for its occupants.

- **Explore the potential of existing dehumidification technologies, cooling technologies, and heat prevention strategies** which is suitable for a hot and humid climate to provide an alternative with less electricity demands and carbon emission.

- **Develop a façade scheme by assembling the applicable systems** in order to offer an alternative system to ease the loads of the conventional cooling system and integrating the usage of solar energy.

- **Develop and evaluate an integration strategy of the façade scheme** to facilitate the building system and justify its efficiency and possible development in the future.

1.3 Research Question

"How can the **desiccant cooling facade system** be integrated into the built environment to reduce the cooling load of **office buildings in a hot and humid climate**?"

1.4 Research Sub-Question

To be able to answer the main research question, the following sub-questions have been formulated. They can be categorized into 4 main categories as follows:

Site Context
- How does the climate context of Bangkok affect the design variable of façade design?
- What challenges does Bangkok's climate imposed on the existing façade typology?

Thermal Comfort
- What is the optimal temperature range for Bangkok office spaces?

Cooling Technologies
- Which passive cooling strategies would be applicable for Bangkok's context?
- How can dehumidification improve the performances of the cooling system?
- Which cooling system would have the most potential to provide sufficient thermal comfort while reducing the electricity demand for Bangkok?
- How can solar energy be implemented to facilitate the facade cooling system?

Building Integration
- What are the required components for developing a desiccant cooling façade?
- What are the possible configuration to integrate the desiccant and cooling system?
- How efficient is the proposed system in relative to the conventional cooling system?

1.5 Research Methodology

The main objective of this thesis is to develop a desiccant integrated façade system which is suitable for the context of a hot and humid city like Bangkok. The integrated façade system aims to provide an alternative to the conventional vapor compression – air conditioning system to reduce the electricity demands and carbon emission. Therefore, a theoretical framework has been formulated. It could be categorized into two main parts: contextual analysis and research on the existing cooling systems.

Contextual research has been conducted to provide background information about Bangkok. Research and analysis on the climate context and the built environment have been conducted in parallel, to elaborate the required parameters for designing a facade system. The studies have been carried out through the means of literature reviews from journals and statistical data from the related agency. Furthermore, research on adaptive thermal comfort has also been conducted via comparative literature reviews between surveys conducted by two journals to define a thermal comfort range of offices in Bangkok to be used in further stages.

Research on the potential of available cooling systems has also been conducted. Energy saving potentials, system efficiency, and applicability to the hot and humid climate are the parameters used in researching and defining the potential of the systems. The research approach can be defined as disassembly, design, and integration as the available systems are first disassembled to investigate the suitable components. Then during the design stage, they are re-assembled for optimal efficiency, and finally, they are integrated into the built environment. An extensive comparative review between desiccant technologies, cooling systems, and regeneration method had been performed to define the suitable components and configuration of the integrated system. The study is based on literature reviews, patent records, company fact sheets, along with experimental data conducted by various authors. Moreover, case studies regarding developing technologies, experiments based in a similar context, and laboratory experiments will also be discussed in this report.

As a mean of evaluating the proposed system, a benchmark would be established by collecting data regarding energy performances of an existing building with a fully glazed typology. Software simulation will be used as an evaluating tool to compare the performance of the proposed system to the existing benchmark. The comparison will be based on the efficiency of the system regarding energy consumption and thermal comfortability.

1.6 Thesis Structure

The introduction has already given an overview of the research objective, which includes the main questions following by its sub-questions and the methodology to achieve the goal of the research. The structure of this thesis is categorized into 3 main sections representing the process of the research in the chronological order. The first part includes the preliminary research framed by the theoretical framework, which includes the site context, thermal comfort, and the cooling strategy. It is followed by the design implementation where the data extracted from the research would be implemented in the façade's development. The system's configuration and integration would also be discussed in this section. In the last section, the thesis would present the evaluation of the proposed system to justify its performance in comparison to a benchmark situation. Lastly, the thesis will be concluded with a discussion for further development and a reflection towards the process.

“How can the **desiccant cooling facade system** be integrated into the built environment to reduce cooling load of **office buildings in a hot and humid climate**?”

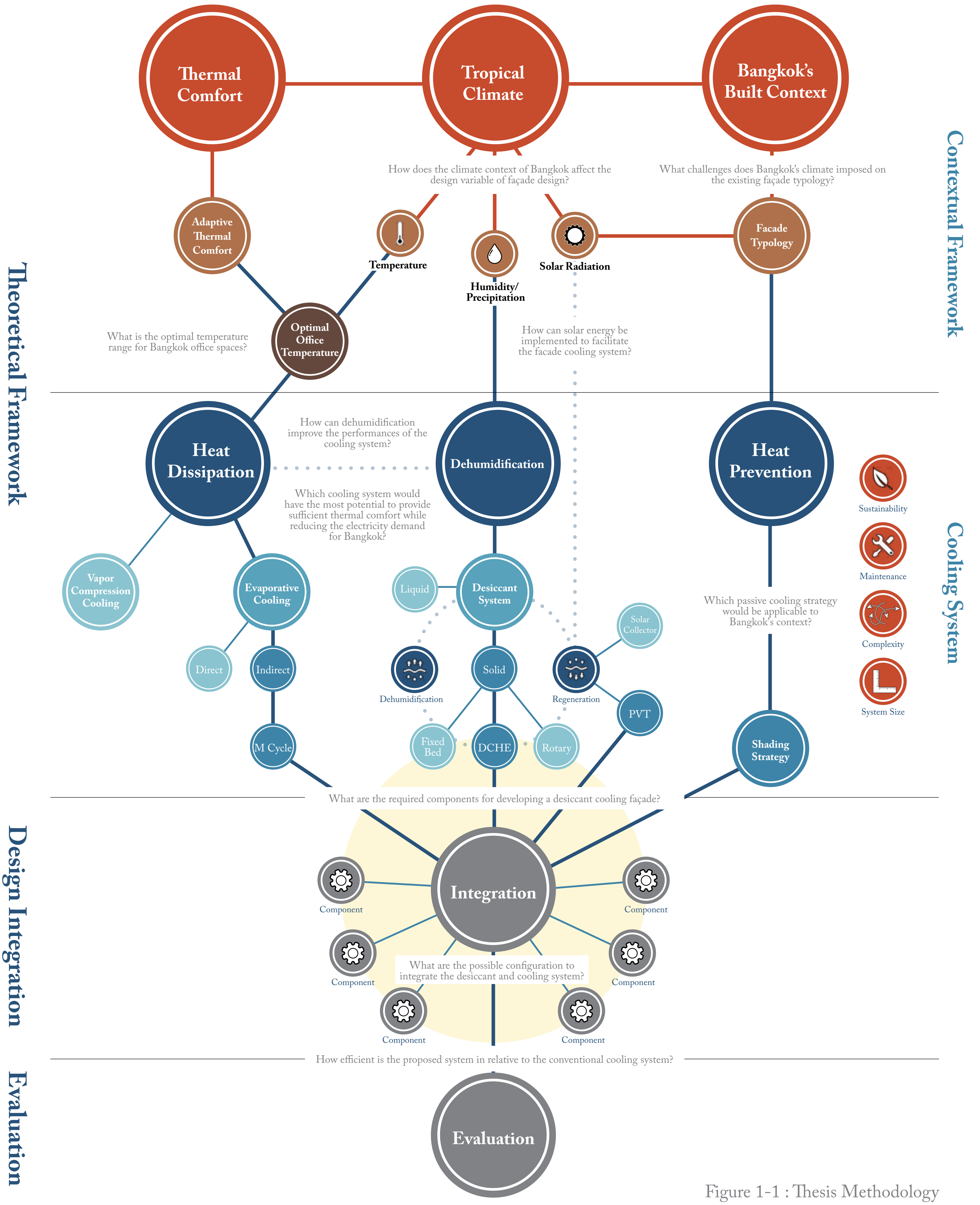


Figure 1-1 : Thesis Methodology

Part I

Disassembly | Background Researches

2 Site Context
Background Research

Figure 2-1 : Bangkok's built environment

TWStock. "The Golden Grand Palace of Bangkok. with skyscraper view or cityscape at sunrise time. The most favorite landmark or travel destination of asia. Best of amazing beautiful scene of Thailand." Shutterstock, https://www.shutterstock.com/nl/image-photo/golden-grand-palace-bangkok-skyscraper-view-300284237?src=UO0znbuVMn3ID80M9J8OGw-1-0

The environmental and social context primarily influences growing demand in space cooling. (OCED&IEA, 2018) Therefore, this chapter is dedicated to providing background information of the hot and humid climate of Bangkok, as well as its built environment. An overview of the cooling degree days, along with the characteristics of the tropical climate, would be discussed. Furthermore, an insight of the specific climate context of Bangkok would also be presented; to describe the relationship between the climate context's effect on Bangkok's built environment and determining a parameter to be used in designing an integrated façade. Moreover, an overview of Bangkok's built context would also be presented, portraying the development of the facades typology in office buildings in the past decades. The characters of the facades will be analyzed, while its development trends will be predicted. In addition, Thailand's Building Energy Code would also be discussed to present the government's approach to providing a standard for energy conservation for new buildings.

2.1 Cooling Degree Days (CDDs)

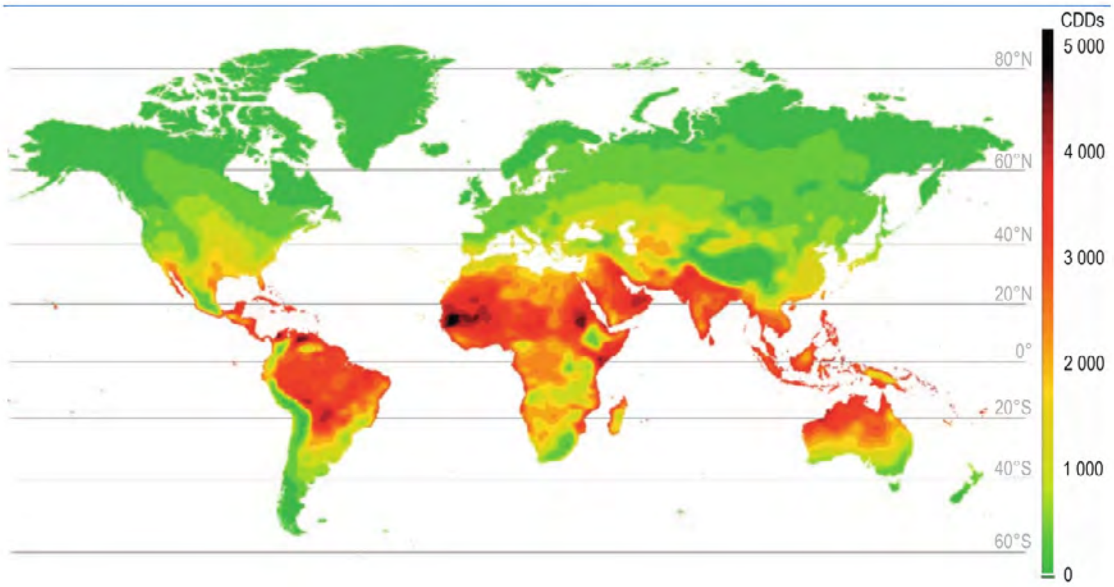


Figure 2-2 : Annual average CDDs between 2007-2017 (OCED&IEA, 2018)

Cooling Degree Days (CDD) is used to determine how cold or how warm a location is by comparing the mean of the high and low outdoor temperature of a location to a standard temperature. The CDDs depict how much the daily mean temperature exceeds the standard temperature over a period of time. In addition to air temperature, humidity also plays an essential role in increasing the cooling demands, while also affecting the CDDs. Humidity affects the human perception of temperature. As higher relative humidity makes it more difficult to sweat, hence it feels hotter at the same temperature. Therefore, a heat index has been considered to incorporate the influence of humidity. The IEA has presented a map depicting an annual average CDD between 2007-2017. [Figure 2-2]

It could be observed that the countries with higher CDDs lie within the region in parallel with the equator, covering the tropics and sub-tropics. In these areas, the CDDs could be 10 times higher than the reference case. As a result of climate change, the IEA has predicted a 1˚C increase in global average temperature, which would lead to a 25% average increase of CDDS globally by 2050. (OCED&IEA, 2018)

2.2 Tropical Climate Characteristics

According to Koppen's climate classification, the tropical climate is classified as Type A climate. The tropical region forms a uniformed belt in parallel to the equator within 15˚ N and S latitudes. Due to its location, it possessed a constant net solar radiation throughout the year, resulting in relatively high temperature over 18˚C annually, which is reflected in the CDDs map mentioned in the earlier section. Furthermore, the temperature differences between day and night are typically higher than between the seasons. In correspond to Koppen's climate classification, the tropical climate could be categorized into Wet equatorial climate (Af), Tropical monsoon and trade-wind littoral climate (Am) and Tropical wet-dry climate (Aw). (Arnfield, 2018)

A.Tropical

f Wet equatorial climate

m Tropical monsoon and trade-wind littoral climate

w Tropical wet-dry climate

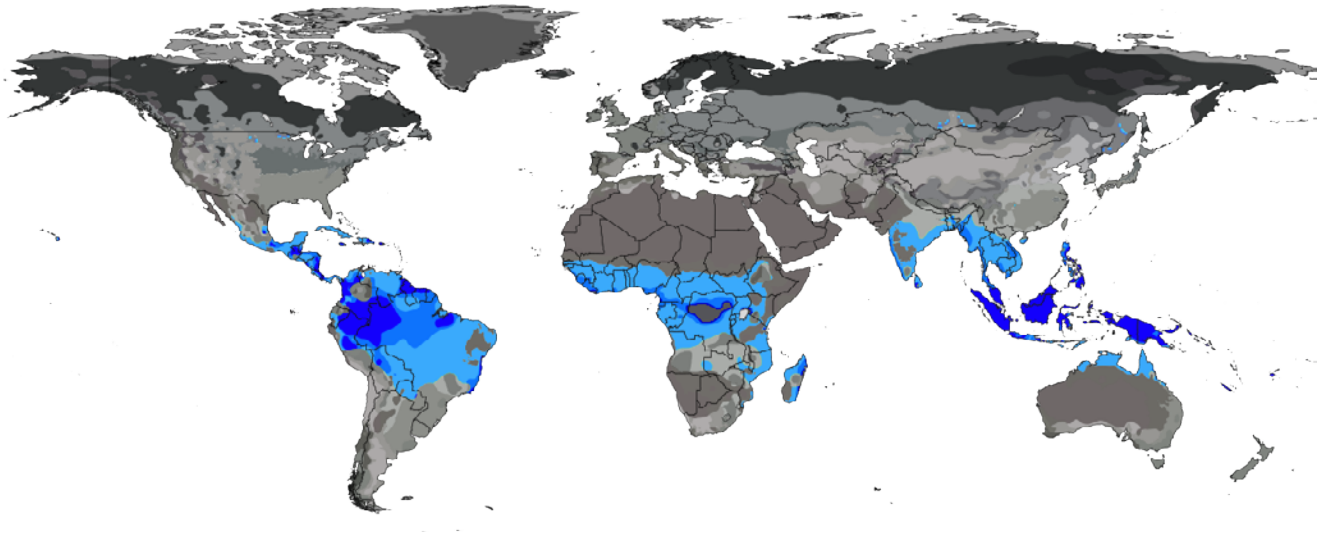


Figure 2-3 : Koppen 's Classification : Tropical Climate Zone

Peel, M.C. and Finlayson, B.L and MacMahnon, T.A.. "World map of Koppen-Geiger climate classification" Updated Köppen-Geiger climate map of the world. 2007, https://people.eng.unimelb.edu.au/mpeel/koppen.html

Wet equatorial climate (Af)

The Wet equatorial climate could be defined by a consistently high temperature ranging around 30°C, high precipitation level varying around 150-1,000cm, heavy cloud cover, high humidity, and little annual temperature variation. The Wet equatorial regions lie within the intertropical convergence zone (ITCZ). With the influence of the ITCZ, dry months occurs when it moves away. Rainfall usually occurs in the late afternoon or early evening when the atmosphere is prone to thunderstorms. (Arnfield, 2018)

Tropical monsoon and trade-wind littoral climate (Am)

The tropical monsoon and trade-wind littoral climate are located primarily in southern and southeastern Asia. It is characterized by its small annual temperature ranges, high temperature, and a higher precipitation level than Wet equatorial climate. Moreover, the Am region usually faces a short dry season in winter. The Asian monsoon influences the climate in the southern and southeastern Asian. It brings convective and precipitation in the summer when the tropical air from the ocean meets the low-pressure zone of the Himalayas. On the other hand, the Siberian anticyclone brings fresh and dry air in winter. The trade wind, on the other hand, carries precipitation to the coastal strip in the Americas and Africa. The seasonal movement and changes in the intensity of the trade-wind would result in a short and moderate dry season, while tropical disturbances intensify summer precipitation.(Arnfield, 2018)

Tropical wet-dry climate (Aw)

The tropical wet-dry climate is distinguished by the definite wet and dry season. In the tropical wet-dry region, the temperature varies between 19-20°C in winter and 24-27°C in summer. Its annual rainfall levels are relatively less than Af and Am ranging around 50-175cm and would usually occur during summer. The tropical circulation defines the seasonal cycle in a tropical wet-dry climate. In the summer, the Intertropical convergence zone (ITCZ) moves towards the equator stimulating convective rainfall in the region. In winter, the ITCZ is replaced by the subtropical anticyclone resulting in a period of dry weather. (Arnfield, 2018)

2.3 Bangkok Climate Characteristic

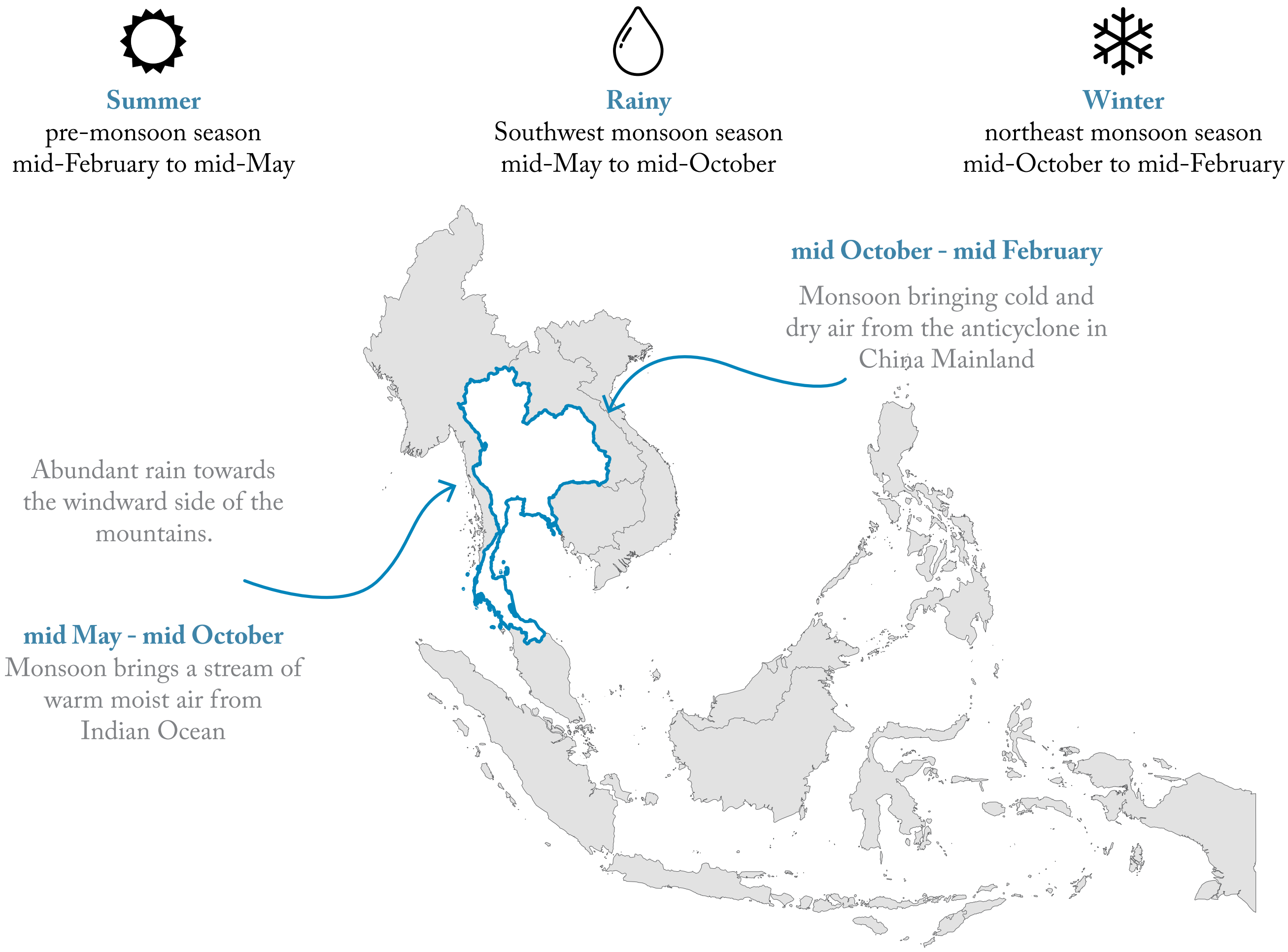


Figure 2-4 : Influence on Thailand's climate

Thailand is located in the tropical region between the latitude 5˚37 N to 20˚27 N and longitude 97˚ 22 E to 105˚ 37 E. The country could be divided into 5 regions according to its climate pattern and conditions namely: Northern, Northeastern, Central, Eastern and Southern region. Bangkok is strategically located in the center of the country, in the low-level plain of the central region. It is characterized by the Koppen's classification as Tropical wet-dry climate (Aw).

The climate of Thailand is influenced by seasonal monsoon wind. Starting in May, the southwest monsoon would bring a stream of warm moist air from the Indian Ocean. The southwest monsoon is accompanied by the Inter-Tropical Convergence Zone (ITCZ) and tropical cyclones resulting in abundant rainfall throughout the country. The ITCZ arrives in the Southern part in May and rapidly moves toward the north towards southern China between June and early July causing a dry spell in the Northern Region. Then in August, it moves southwards to the Northern and Northeastern region of Thailand before it reaches the Central region in September. Finally, it moves downwards to the Southern Part in October. Furthermore, Thailand's climate is also influenced by the northeast monsoon, which brings cold and dry air from the anticyclone in China in October. This causes the temperature to drop in the high-altitude areas of the Northern and Northeastern parts. However, it also causes abundant rain along the eastern coastal area of the Southern Region. (Meteorological Development Bureau) Since it is located in the Central region, Bangkok doesn't face drastic temperature changes. However, precipitation levels are essentially being influenced by the arrival of ITCZ.

2.3.1 Precipitation

Due to the southwest monsoon, intensive rainfall starts from mid-May until early October. During the period, the weather is often cloudy, and daily rainfall could be expected. In 2017, May had the highest rainfall at 458mm. The annual rainfall is accumulated to 1585.6mm over 140 days. (Meteorological Development Bureau)

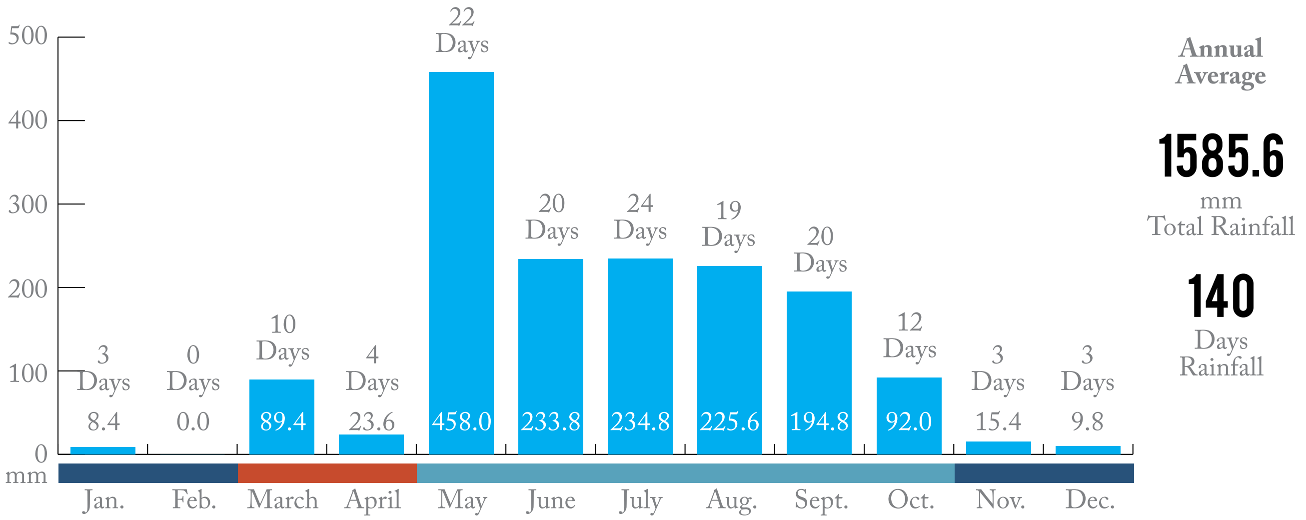


Figure 2-5 : Bangkok's annual precipitation (2017)

2.3.2 Humidity

Bangkok is not as hot as the north-central inland areas; however, due to its proximity to the coastal area, it is more humid. Maximum humidity level could reach to 100% almost all year round. During winter and summer (November – April) the average relative humidity lies between 60-70%. However, as the monsoon arrives, in May the humidity drastically rises to the range between 70-80%. (Meteorological Development Bureau)

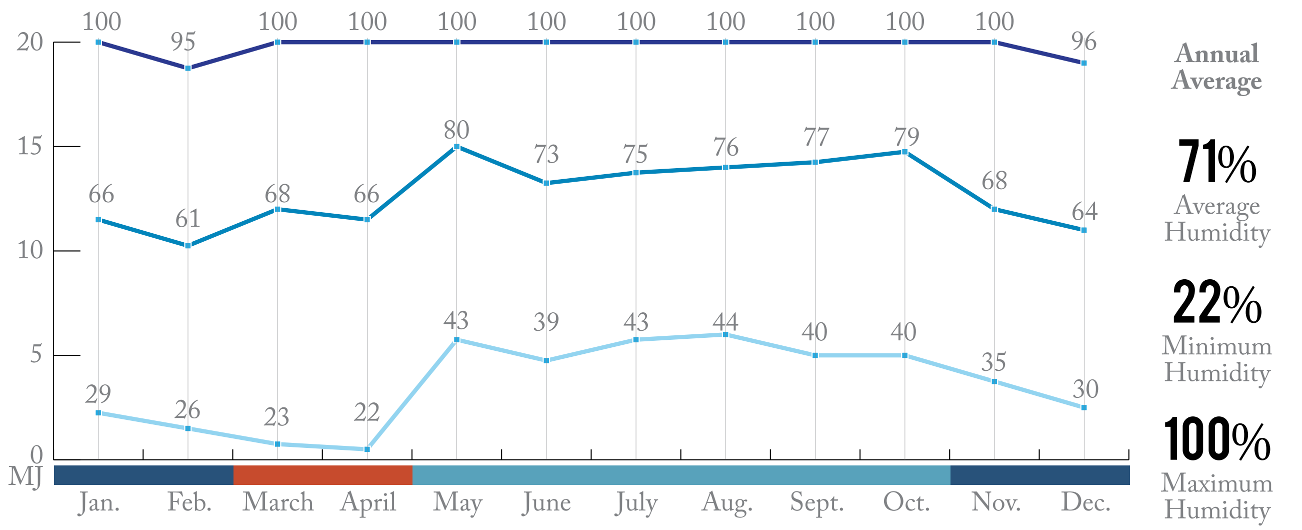


Figure 2-6 : Bangkok's annual relative humidity (2017)

2.3.3 Temperature

As a characteristic of the Tropical wet-dry (Aw) climate, the variation of the annual temperature of Bangkok is minimal; ranging between 25.2-30.6°C. However, during the hottest month of the year between March and May, the temperature could reach up to 38.7°C. Due to the arrival of the monsoon in May the temperature gradually decreases with an average slightly below 30°C. Unlike the Northern region, which the temperature drastically drops to nearly zero degrees in winter; the arrival of the anticyclone from China has a moderate effect on Bangkok's temperature. The coldest month in Bangkok lies between December and January, with the average temperature of 25-27°C; however, in some days, it could be as low as 15°C. (Meteorological Development Bureau)

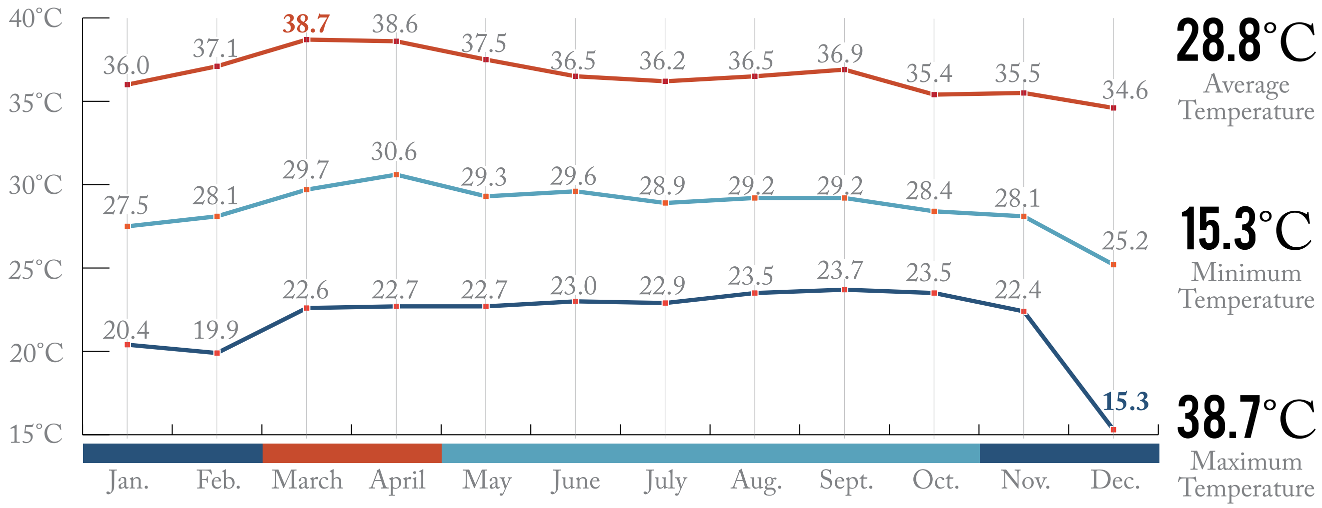


Figure 2-7 : Bangkok's annual temperature (2017)

2.3.4 Solar Radiation

During winter and summer (November-March) cloud cover is minimal. The clear skies resulted in high temperature and solar radiation levels, especially in March to May. Solar radiation level could reach up to 5.00 kWh. Despite the overcast sky during the monsoon season, the radiation remains around 4 kWh. With prominent solar radiation level at an average of 1785 kWh/m^2, the potential for solar energy harvesting in Bangkok is exceedingly feasible.

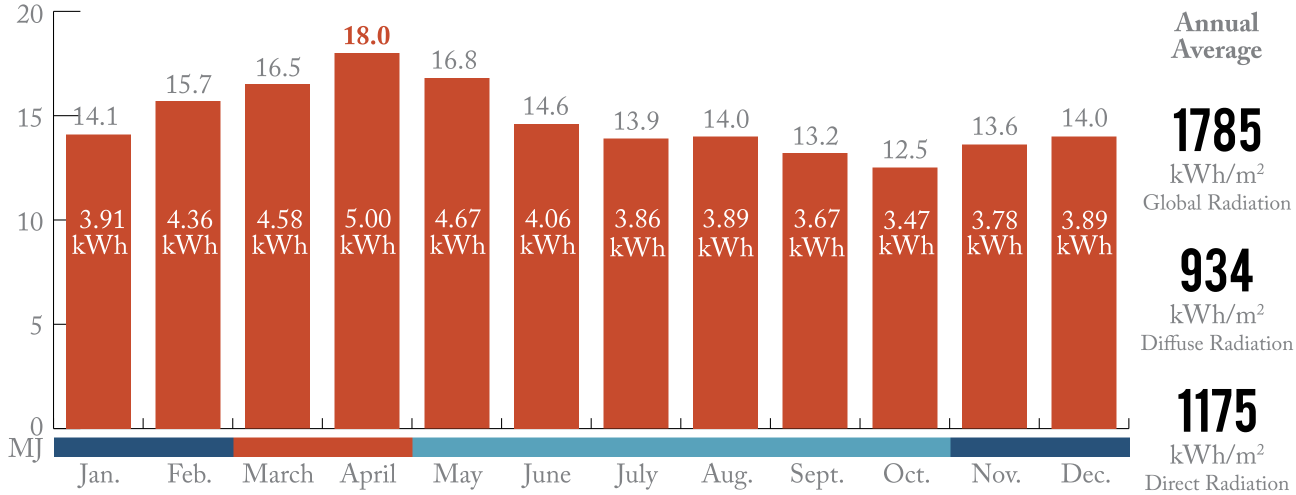


Figure 2-8 : Bangkok's annual average solar radiation (2017)

2.3.5 Sunpath and Sun Angle

Due to Thailand's location near the equator, the sun angle in Bangkok is quite high. During summer, autumn, and equinox solstice, the sun's altitude angle at noon varies between 75°-80°. However, in the winter solstice, it lowers down to approximately 50°. Therefore, horizontal surfaces would receive higher solar exposure.

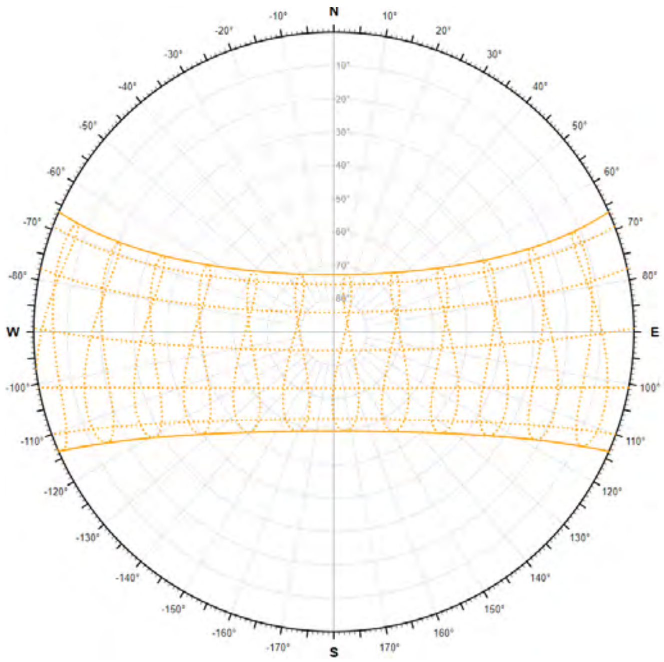


Figure 2-9 : Bangkok's annual temperature (2014)

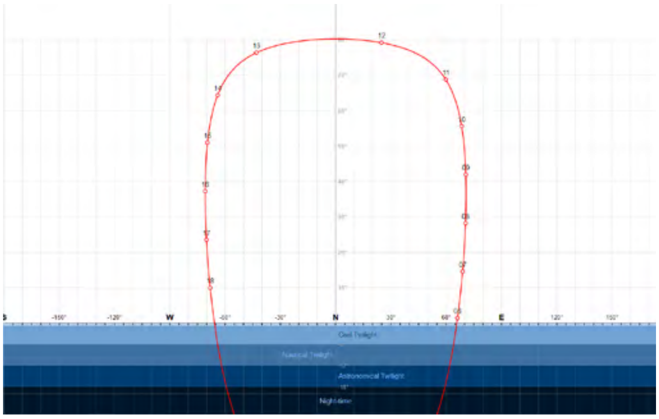


Figure 2-10 : Summer Solstice 21 June 2014

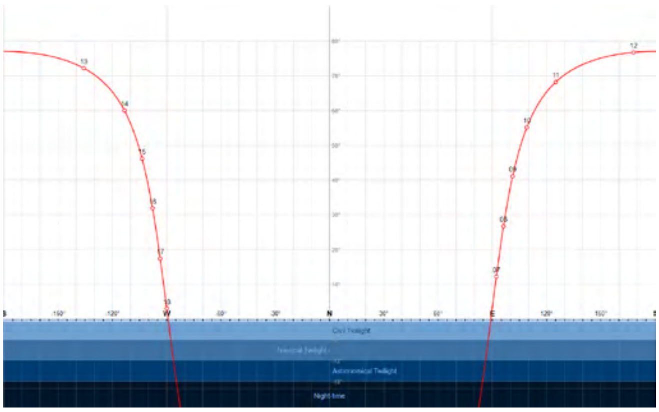


Figure 2-11 : Autumn Equinox Solstice 21 September 2014

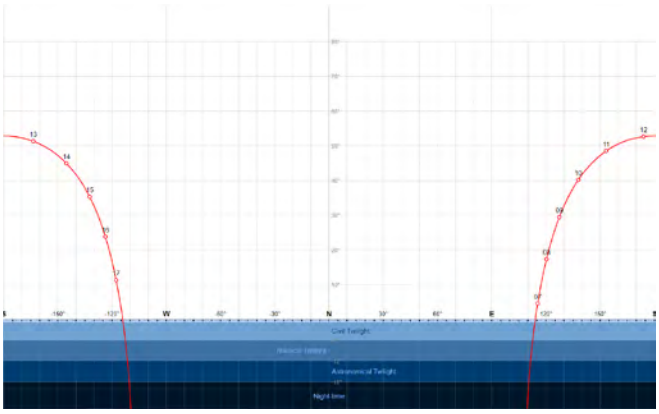


Figure 2-12 : Winter Solstice 21 December 2014

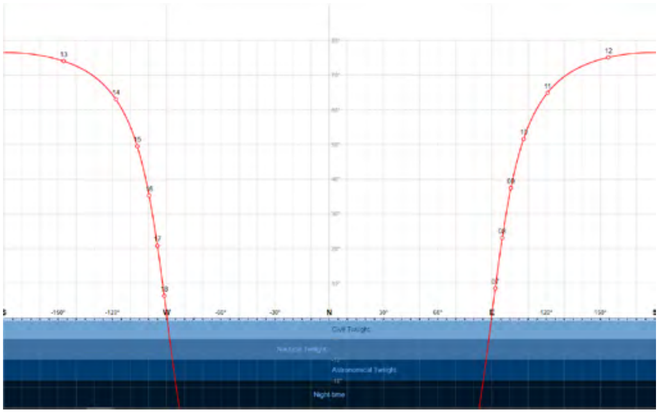


Figure 2-13 : Spring Equinox Solstice 21 March 2014

Marsh.Andrew. "Sun Path / Chart" SunPath 2D, http://andrewmarsh.com/apps/releases/sunpath2d.html

Figure 2-14 : Traditional Thai house resembling vernacular approach.

Images.drstockphoto, "Traditional Thai House", https://images.drstockphoto.com/0/w1024/traditional-thai-house.jpg

Figure 2-15 : Sathorn district : One of major central business district of Bangkok

Chatnara. "Buildings landmark of Sathorn and Silom district, This area is an important business and banking center in Bangkok. With warm sunlight effect on sunset time create highlight and shadow at building.." Shutterstock, https://www.shutterstock.com/nl/image-photo/buildings-landmark-sathorn-silom-district-this-436534558?src=Z49LGeO7ZdgT0g1DT7X4Jw-1-0

2.4 Bangkok's Built-Environment

Vernacular architecture is a reflection of the climate and geography of a local context. In Thailand, traditional Thai houses were designed and built to cope with the tropical climate context of high temperature and humidity as well as heavy rainfall. They were designed with elevated ground floor to be resilience with flooding, while during the dry period space can be used as an outdoor recreational space. At the same time, elevated floors also facilitate ventilation, as hot air rises through the thatched roof it is being replaced by cool air drawn through the wooden floor planks. To adapt with high solar radiation, Thai houses are oriented to face north, to avoid direct sunlight exposure on the roof. (Boonjub, 2009)

However, looking into Bangkok's built context in the modern days, traditional Thai architecture and its application are rarely seen. Bangkok's skyline is majorly influenced by the International Style proposed during the 1920s. Due to densification and rapid population growth, new developments tend to grow vertically. According to the Council of Tall Buildings and Urban Habitat, Bangkok possessed 150 skyscrapers over 100m. (http://www.skyscrapercenter.com) Dating back to the Asian investment boom between the 1980s to 1990s, several overseas corporations set up their regional headquarters in Bangkok, turning it into a regional economic hub. This provokes the demand for office spaces. Currently, the office supply in Bangkok is at 8,618,862 sq.m, however, researches by Collier International Thailand had shown that it would eventually grow by 662,676 sq.m within 2022. (Collier, 2017) With this rapid growth, office and mixed-used functions are accounted for 25% of the buildings over 150m. (http://www.skyscrapercenter.com)

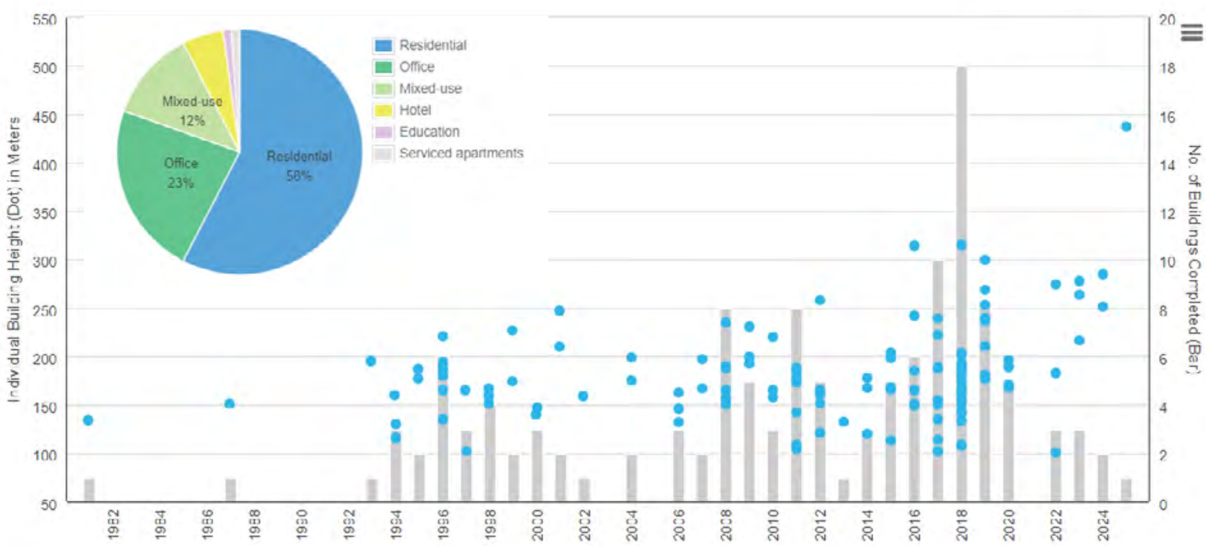


Figure 2-16 : Bangkok's Building Completion Timeline of Building over 100m.

Skyscrapercenter. "Building Completions Timeline (Last 50 Years, 100 m+)." Skyscrapercenter, https://www.skyscrapercenter.com/city/bangkok

2.4.1 Facade Classification

Building facades acts as a part of the building envelop functioning as an intermediary layer between the outdoor and the indoor spaces. It plays a vital role in regulating the comfortability of the occupants both in terms of visual, thermal, and acoustical. According to Hyde, building façades can be classified into 5 types: Thin Skin, Thick Skin, Inclined Skin, Buffering Skin, and Valve Effect Skin. (Hyde, 2001)

Thin Skin
Thin skin facades are determined by the way a façade uses its material to regulates the indoor climate. It primarily relies on the characteristics of the materials rather than external buffering to provide shading. Thin skin façade is usually distinguished by the large, unshaded glazing areas and deep open spaces.

Thick Skin
Thick skin facades provide shading for the interior spaces through the use of its depth and projections. It could be distinguished by the shade providing components such as balconies, exterior shading devices, and perforated walls.

Inclined Skins
As its name suggested, Inclined Skin façade is characterized by the use of inclined wall elements or glass panes to shield the interior space from direct solar radiation.

Buffering Skins
A Buffering Skin façade incorporates its environment such as trees, verandas and surrounding structure as a shading strategy to minimize direct solar exposure.

Valve Effect Skins
The Valve Effect Skins regulates the indoor environment through the means of an operable façade system. The apertures allow the incoming air and light to be filtered and controlled.

The classification defined by Hyde will be used in elaborating the typology of Bangkok's built context.

2.4.1 Bangkok's Office Facade Typology

Bangkok's skyline is a composed mainly of the thick and thin skin typology. However, the trend is shifting, as thin skin is becoming more dominant in the past decade especially in office buildings and mixed use programs. To depict this change, a timeline presenting office and mixed used skycrapers with the height of over 100m will be presented in this chapter.

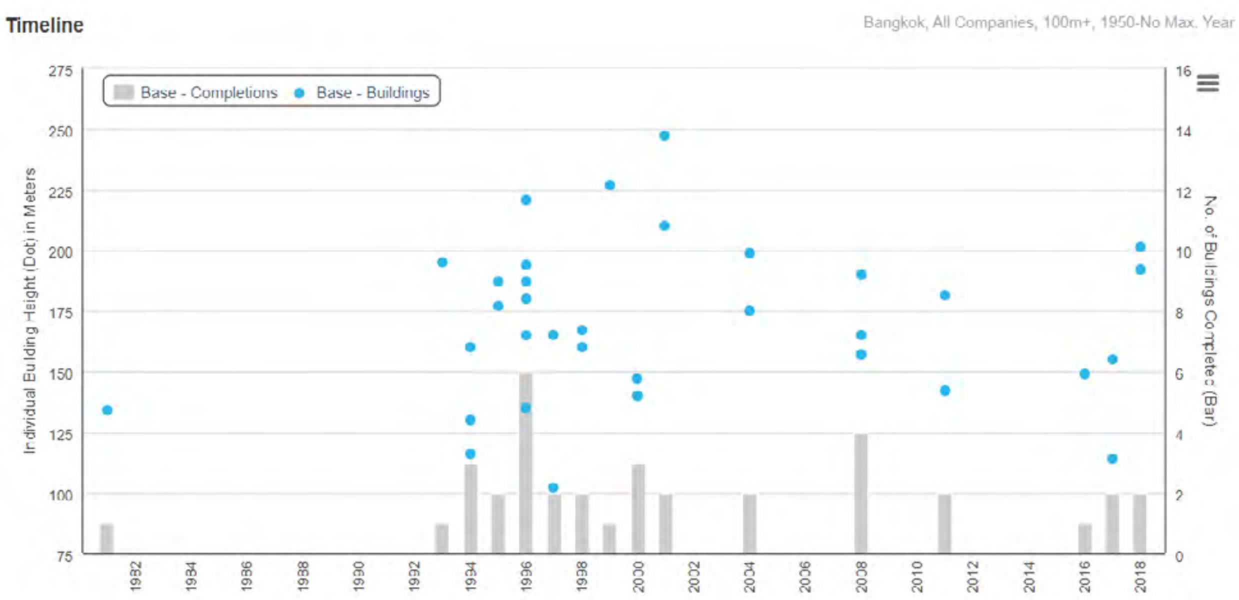


Figure 2-17 : Bangkok's Office Building Timeline of Building over 100m.
Skyscrapercenter. "Building Completions Timeline (Last 50 Years, 100 m+)." Skyscrapercenter, https://www.skyscrapercenter.com/city/bangkok

1982

Fig. 2-18 Bangkok Bank Tower

1993

Fig. 2-19 Sinn Sathorn Tower

1994

Fig. 2-20 TMB Tower

Fig. 2-21 Sathorn City Tower

Fig. 2-22 Vanity Tower

1995

Fig. 2-23 K-Bank Tower

Fig. 2-24 United Center Tower

Skyscrapercenter. "Building Completions Timeline (Last 50 Years, 100 m+)." Skyscrapercenter, https://www.skyscrapercenter.com/city/bangkok

1996


Fig. 2-25 RS Tower


Fig. 2-26 Siam Bank Tower


Fig. 2-27 Tipco Tower

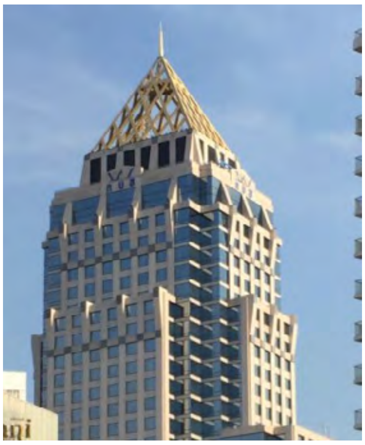


Fig. 2-28 Abdulrahim Place


Fig. 2-29 Thai Wah Tower


Fig. 2-30 Jewelry Trade Tower

1997


Fig. 2-31 Ital Thai Tower


Fig. 2-32 Elephant Tower

1998


Fig. 2-33 U Chu Liang Tower


Fig. 2-34 Emporium Suites Tower

Skyscrapercenter. "Building Completions Timeline (Last 50 Years, 100 m+)." Skyscrapercenter, https://www.skyscrapercenter.com/city/bangkok

1999

Fig. 2-35 Sathorn House Tower

2000

Fig. 2-36 Capitol Tower

Fig. 2-37 Shinawatra III Tower

2001

Fig. 2-38 CRC Tower

Fig. 2-39 State Tower

2004

Fig. 2-40 Office at Central World

Fig. 2-41 Exchange Tower

Skyscrapercenter. "Building Completions Timeline (Last 50 Years, 100 m+)." Skyscrapercenter, https://www.skyscrapercenter.com/city/bangkok

2008


Fig. 2-42 Cyber World Tower


Fig. 2-43 Chamchuri Square Tower

2011


Fig. 2-44 Park Venture Tower


Fig. 2-45 Sathorn Square Tower

2016


Fig. 2-46 Pearl Tower

2017

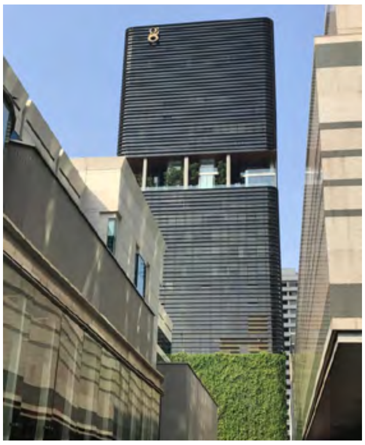


Fig. 2-47 Gaysorn Tower


Fig. 2-48 Square Tower

Skyscrapercenter. "Building Completions Timeline (Last 50 Years, 100 m+)." Skyscrapercenter, https://www.skyscrapercenter.com/city/bangkok

2.4.1 Bangkok's Office Facade Trend

According to Thailand industry outlook 2017-19 by Krungsri Research, over the next few years, the demand for office spaces is expected to continue growing as a result of the recovering economy and increasing demands due to Bangkok's strong regional connections. (Krungsri Research, 2017) Publicized renderings and plans of the prospect developments have reflected the on-going trend of thin-skin façade typology. This global trend is increasing its popularity in the tropical context, due to the improvement of building materials and construction techniques, especially the improvement of window's U-Value. Furthermore, it is due to the shift of the architectural trend, which is strongly influenced by the International Style. (Dinapradipta,2015)

2018

Fig. 2-49 Krunsri Ploenchit Tower

Fig. 2-50 Singha Complex Tower

Fig. 2-51 T1 Tower

Fig. 2-52 Samyan Mitrtown

Fig. 2-53 Ari Hill Tower

2019

Fig. 2-54 MS Siam Tower

2.4.1 Bangkok's Building Code and Regulations regarding energy conservation

In order to deal with the global issue of global warming and climate change, the government of Thailand had legislated several laws regarding energy efficiency which includes:

- **Energy Conservation Promotion ACT (ECP Act) (1992)**
- **Royal Decree 1995** – Identifying designated buildings (DBs)
- **Ministerial Regulation No.44 (1995)** - Regulating that designated buildings should audit their energy consumption
- **Ministerial Regulation (2009)** – Identifying designated building and building size and regulating standards and procedures in designing for energy conservation

Ministerial Regulation, 2009

As the most recent and matured regulation regarding energy conservation standards and design requirements, the Ministerial Regulation (2009) will be discussed in the following chapter. Moreover, since office buildings are the main focus of this thesis, only requirements regarding offices will be discussed. Office buildings are categorized as type 1 according to the regulation, as it is considered to be building with operation hours below 8.

Section 1

According to section 1 of the regulation, buildings with an area over 2,000 sq.m and is characterized by the following function shall be regulated by the mandatory requirment for energy conservation.

1. Educational
2. Office
3. Theater
4. Department Store
5. Entertainment Spot
6. Auditorium
7. Hotel
8. Hospital
9. Apartments/Condominium

Section 2

The requirements are divided into 6 categories namely : Building envelope, Lighting System, HVAC system, Heating System, Overall energy performance, and Renewable energy.

Building Envelope	W/m^2
Overall Thermal Transfer Value (OTTV)	**<50**
Roof Thermal Transfer Value (RTTV)	**<15**

Lighting System	W/m^2
Lighting Power Density (LPD)	**<14**

HVAC System

HVAC System	Coefficient of Performance (COP)	Energy Efficiency Ratio (EER)
Unitary System <12,000 W	>3.22	11

Central System	Pressure Type	Refrigeration Pressure	kW/Tr
Air Cooled	All	<300	1.33
		>300	1.31
Water Cooled	Piston	All Sizes	0.89
	Rotary	<150	0.78
	Rotary	>150	0.76
	Centrifugal	<500	0.76
	Centrifugal	>500	0.62

Absorption Chiller	Coefficient of Performance (COP)	Energy Efficiency Ratio (EER)
Single Effect Absorption Chiller	3.22	11
Double Effect Absorption Chiller	3.22	11

Heating System

Boilers	Minimum Efficiency (%)
Oil Fired Steam Boiler	85
Oil Fired Hot Water Boiler	80
Gas Fired Steam Boiler	80
Gas Fired Hot Water Boiler	80

Heat Pump	Inlet Temp. (C°)	Outlet Temp. (C°)	Air Temp. (C°)	Minimum COP
Type 1	30	50	30	3.5
Type 2	30	60	30	3.0

Overall Energy Consumption

There are two options in fulfilling the requirements:

1. Meeting the standards in section 1-4
2. Overall energy consumption = **<300,000 kW-hr/year**

Renewable Energy

Electrical consumption for lighting system can be subsidized by the following strategies:

- On/Off lighting switches should be seperated along the building boundary.
- Effective shading coefficient >0.3
- Glasses light to solar gain ratio = >1.0
- Natural energy source such as solar energy could be used in subsidizing electrical consumption.

2.5 Conclusion

Being in a Tropical Wet-Dry region (Aw), Bangkok's climate is characterized by high solar radiation, a high temperature which could reach 38°C in the summer, relatively high humidity with an average of 70%, and abundant precipitation year-round. Vernacular strategies of traditional Thai houses had reflected a successful approach in coping with the hot and humid climate. However, due to the rapidly growing population, urban densification, and globalization, new developments are being stacked vertically, at the same time, the architectural trends had shifted away from the vernacular approach.

Being heavily influenced by the International Style, Bangkok's skyscrapers are being dominated by thin skin typology. Despite technological advancement, and innovative building techniques, the rectilinear fully glazed façade fails to efficiency adapt to the hot and humid context. As a mean of regulating energy efficiency and energy conservation, the Thai government has legislated several laws and regulations regarding the standards of new buildings. The regulations aim to control thermal insulation values, the energy consumption of the building systems as well as promoting the use of renewable energy.

However, further actions could be implemented to comply with the climate context efficiently. As the simulation by Fraunhofer Institute, [Figure 2-55] had shown that double glazing windows possessed the lowest energy demand reduction when compared with other passive and active strategies. (Fraunhofer Institute, 2017) Therefore, adaptive thermal comfort, as well as cooling techniques and its implementation, will be discussed in the following chapters.

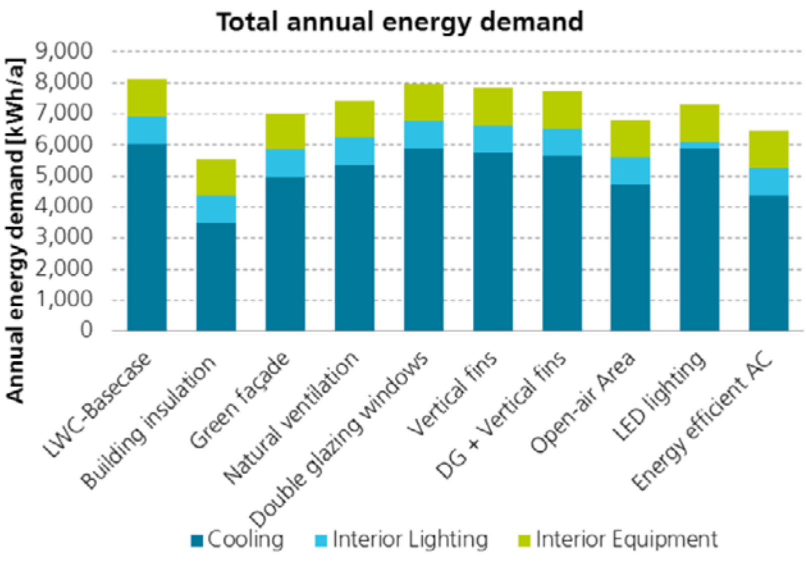


Figure 2-55 : The comparison of energy consumption of the active and passive strategies simulation (Fraunhofer Institute,2017)

3 Thermal Comfort
Background Research

Figure 3-1 : Thermal comfort in office.

Jayna. "Increased Office Productivity? Sounds Cool…." Michaelwardonline, https://www.michaelwardonline.co.uk/news/air-conditioning-higher-office-productivity/

Thermal comfort is a condition of mind which expresses satisfaction with the environment, both in terms of the physiological and psychological aspect. (ASHRAE 55-2004) Researches have shown that the definition of comfort varies from person to person, making it almost impossible to satisfy everyone in space. Therefore, statistical data from extensive laboratory and field work has been collected to determine a standard to define a condition that a specified percentage of occupants will find thermally comfortable. Amongst the standards, one of the international standard widely used is ASHRAE Standard 55 (2004). However, since ASHRAE's method only offers an overview of the temperature range, two studies conducted in office buildings in Thailand will also be discussed.

3.1 ASHRAE 55

The American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) Standard 55 (2004), on 'Thermal Environmental Conditions for Human Occupancy' is a standard which aims to specify the combination of thermal environmental factors and personal factors that will produce a thermal environmental condition which is acceptable to the majority of the occupants of a space. The standard was developed to be used in the design, commissioning and testing of buildings and their HVAC systems as well as evaluation of thermal environments. However, since it is not possible to control the metabolic rate of occupants and their clothing levels, operating setpoints for buildings cannot be mandated by the standard.

According to ASHRAE 55 (2004), six primary factors are required to be addressed when defining the condition of thermal comfort. The six factors include:

1. Metabolic rate
2. Clothing insulation
3. Air temperature
4. Radiant temperature
5. Air speed
6. Humidity

Section 5.2 in ASHRAE 55 (2004) offers the method to determine the acceptable thermal condition in an occupied space. A comfort zone may be determined through the values of humidity, airspeed, metabolic rate, and clothing insulation. It could be determined in terms of a range of operative temperature or of the combination of air temperature and mean radiant temperature that people find thermally acceptable.

Graphical Method

For a typical situation, a graphical method could be used to determine the comfort zone. However, it is limited to the condition that activity levels in the metabolic rates should be between 1.0-1.3 met and clothing worn is between 0.5-1.0 clo. Additionally, air speeds should not be greater than 0.20 m/s. The operative temperature range allowed for intermediate values of clothing insulation may be determined by linear interpolation between the limits for 0.5 clo and 1.0 clo.

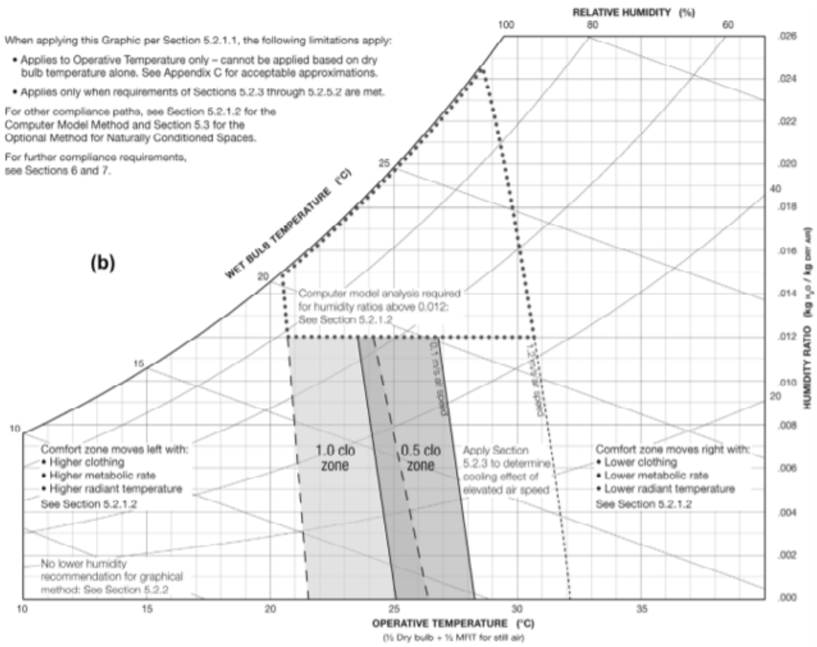


Figure 3-2 : Psychrometric Chart depicting thermal comfort zone. (ASHRAE 55)

Computer Model Method for General Indoor Application

The computer model method for general indoor application is based on a heat balance model to determine the comfort zone for a wider range of application. The method is applicable in a situation where the occupants have an average metabolic rate between 1.0-2.0 met and clothing worn has 1.5 or less clo values of insulation.

To quantify people's thermal sensation, ASHRAE had developed a thermal sensation scale which defines the user's sensation from:

+3 hot +2 warm +1 slightly warm 0 neutral -1 slightly cool -2 cool -3 cold

The predicted mean vote (PMV) model uses heat balance principle to relate the six key factors to the average response of the users regarding the thermal sensation scale. The model is based on the assumption that people voting warm, hot, cool, or cold are dissatisfied. Therefore, for typical applications the PPD should be <10% while the PMV ranges between -0.5 < PMV < + 0.5.

3.2 Case Study 1 : John F. Busch

In order to justify whether the comfort standard developed for offices in temperate climate would be suitable for the ones in a tropical climate; Busch had conducted a field study in Bangkok, Thailand. The study was conducted through a questionnaire which was responded by 1146 office workers in four building during two seasons. Six hundred responses were obtained in the hot season and 546 in the wet season. The samples were categorized into the air-conditioned environment, which is accounted for 770 occupants, and 376 occupants were taken from office workers in a naturally ventilated office.

The questionnaire included the subjective thermal rating scale based on the ASHRAE 55 thermal sensation scale, which takes the thermal impacts of humidity, radiant temperature, air velocity, and clothing levels into account. In addition, two further seven-point scales addressing the comfortability regarding airflow and humidity were also included.

Physical quantities of the environment measured include dry-bulb temperature, relative humidity, globe temperature, and air velocity. While the metabolic rate of the occupant was considered to be 1.1 met which is a typical level for light office activities. Clothing insulation was lower in a natural ventilated building at 0.49 clo while clothing insulation of the air-conditioned building is slightly higher at 0.56 clo. The differences are the result of differing standard for office uniform, where the natural ventilated building was occupied by the government sector while the air-conditioned offices by the private sector.

With the result gathered ASHRAE's Effective temperature (ET), or temperature at 50% relative humidity, and mean radiant temperature equal to air temperature, had been calculated to depict the thermal sensation of the actual environment. The result had shown that the mean ET for the air-conditioned offices is at 24.7˚C significantly lower than the natural ventilated building, which lies at 31.5˚C. Along with the Effective Temperature, the standard effective temperature (SET) was also calculated. The standard effective temperature (SET), is an extended version of Effective Temperature which considered 50% relative humidity and air velocity of 0.1m/s but also incorporates clothing insulation and metabolic rate into consideration. The values showed a slight drop in the average temperature when compared to the ET. The average value of the air-conditioned offices was at 24.3˚C while the average temperature in the natural ventilated building was at 31.5˚C respectively.

The result of Busch findings was interpreted through the distribution of the collected responses in the ASHRAE Thermal sensation scale and the McIntyre scale. To determine the expected temperature at which the occupant would vote "neutral", a simple linear regression was performed on the ASHRAE Thermal sensation scale as a function of ET and SET. The results have shown that as a function of ET, the neutral temperature of an Air-conditioned office could be defined at 24.7˚C while the natural ventilated building it is defined at 27.4˚C. On the other hand, as a function of SET, the neutral temperature was defined at 24.4˚C for airconditioned offices, and 22.8˚C for the natural ventilated buildings.

To determine the thermal acceptability, the standard proposed by ASHRAE 55-81 that the sensation votes within the central three categories of the seven-point scales indicate ASHRAE comfort standards which aims to satisfy the thermal comfort of 80% of the building occupants, was applied. The percentage of votes at the thermal acceptability range was plotted over the ET temperature. The study had concluded that the lower boundary of the comfort zone in the air-conditioned building is around 22˚C while its upper boundary reaches 28˚C. For the natural ventilated building, the lower boundary of the comfort zone is undefined, but the upper boundary is estimated at 31˚C.

3.3 Case Study 2 : Yamtraipat et al.

Typically, researches regarding indoor thermal comfort would be considered through the means of quantifiable variables such as temperature and humidity. However, according to Yamitraipat et al., the state of comfort is also dependent of the "non-quantifiable" aspect such as mental states, habits, education, as well as their acclimatization to a particular climate. Therefore, the preferences in terms of comfort would vary according to their long-term experiences in a location. (Yamtraipat et al., 2005)

Yamtraipat et al. had conducted a field investigation in the form of a survey from 1520 Thai volunteers working in an air-conditioned office(light activity level or office work was about 1.2 met) from different climatic zones of Thailand. The climatic zones are categorized into 3 zones according to their dry bulb temperature and 4 zones according to their relative humidity. The questionnaire consists of 2 parts. The first part includes questions regarding the so-called 'non-quantifiable factors' such as

Age
Gender
Weight
Height
Education level
Type of clothes the subjects wore on the day of investigation
Use of home air-conditioners.

While in the second part, the questions used was referred to the ASHRAE thermal and humidity sensation scale. The volunteers were to rate their feeling according to the given scale.

-3 = cold -2 = cool -1 = slightly cool 0= neutral 1 = slightly warm 2 = warm 3 = hot
-2 = dry -1 = slightly dry 0= just right 1 = slightly humid 2 = humid
1 = too still 2 = just right 3 = breezy 4 = too breezy

From the result, it is evidence that high indoor air relative humidity had a significant effect on sensation vote only when the air temperature was relatively high (26°C -27°C). To determine the mean neutral sensation indoor air temperature, a simple linear regression analysis was mapped. The neutral indoor air temperature for Bangkok's climatic area was concluded at 25.6°C while the country's neutral mean lies at 25.9°C.

To define the thermal acceptability range of the subjects, the vote for both slightly warm and slightly cool sensation had been taken into consideration; since each volunteer would have a different interpretation of their senses. Therefore, the conclusion is based on the result which 80% of the occupants have thermal sensations between "slightly cool" and "slightly warm." It could be concluded that the thermal acceptability for Bangkok ranges between 24.5-27.4°C. In terms of preferred relative humidity range for temperature varying between 25°C and 27°C, the relative humidity between 50-60% is voted "just right" in every zone. While the majority preferred the average body air velocity at 0.2 m/s as it was voted "just right" in every zone.

In their study, Yamtraipat et al. also took the "non-quantifiable" variable such as air-conditioner acclimatization and education level into consideration. The results have presented a neutral temperature difference between the subject who used the air conditioner at home and the ones who don't at 25.4°C and 26.3°C respectively.

Moreover, the results regarding the education level have shown that the neutral temperature of post-graduate degree is lowest at 25.3°C while the other education level lies around 26°C. The fluctuation of the temperature between the subject with Air conditioning at home is relatively small at 0.3°C. However, the difference between the non-air-conditioned group reaches 0.9°C. The authors concluded that Thai people with higher education level prefer lower indoor temperature. In addition to personal preference, this could be elaborated on the grounds of Thai office culture, where higher ranking officers wear thicker attires such as suits.

From the study of Yamtraipat et al., it could be concluded that the thermal acceptability temperature for the occupants of offices range between 24.5-27.4°C, while the relative humidity is preferred at 50-60%, and the air velocity at 0.2 m/s. However, in higher ranked offices, the mean neutral temperature would be lower with an average of about difference of 1°C.

3.4 Conclusion

To determine the thermal comfort of air-conditioned offices in Bangkok, Thailand, two literatures conducted in Thailand has been studied. The researches are chosen based on their approaches to defining the range of thermal acceptability through the means of ASHRAE 55's thermal sensation scale in their field study. Both literatures aimed to determine the range of thermal acceptability of office workers in Bangkok and Thailand to justify the idea of climatization and thermal adaptivity.

According to their field study, Busch had concluded the thermal acceptability range of Bangkok air-conditioned office to be 22°C to 28°C at the relative humidity level of 50% and the air velocity of 0.1 m/s. However, Yamtraipat et al. had provided a slightly increasing lower bound of 24.5°C while the upper bound is marginally lower at 27.4°C. Yamtraipat et al. had further inferred that the preferred relative humidity level is at 50-60% and air velocity at 0.2 m/s. The range used as a parameter in this thesis would be derived from the mean of the two studies. Therefore the thermal acceptability will be considered at 23.5°C-27.7°C.

4 Cooling Strategy
Background Research

Figure 4-1 : A normal scene of CDU units of split type air-conditioned system in tropical countries

Zhuang.Justin. "Air-conditioned Nation" Flickr, https://www.flickr.com/photos/zmackid/332732312/

In order for a cooling system to perform efficiently in Bangkok's climate context, the system should incorporate strategies and techniques to cope with the high level of solar radiation, high temperature as well as high humidity. Therefore, this chapter will present an overview of the potential strategies and technologies which are suitable for Bangkok's context. It will be categorized into three main strategies: Heat prevention, dehumidification, and heat dissipation. Since the technologies would be integrated as a cooling façade system in the latter stage of this thesis, the parameters of performance, ease of maintenance, system complexity, sustainability, system size, and configurations would be discussed. Moreover, case studies regarding developing technologies, experiments conducted in similar climate context, and results of laboratory experiments would also be addressed to evaluate the potential of the technologies.

4.1 Passive Cooling strategy

Numerous studies have concluded to a common agreement that passive strategies should be the initial step in designing an energy efficient building, prior to consideration of active approaches such as mechanical equipment driven by fossil fuels. (Prieto et al., 2018) Therefore, this chapter would provide an extensive overview of the typical passive strategies used in the hot and humid climate region, which includes: Shading Strategy, Ventilation, Glazing type, and Window to Wall ratio while the comparative reviews of the strategies conducted by Prieto et al. will also be discussed.

Shading Strategy

The most efficient shading strategy is to intercept direct solar radiation before it reaches the transparent component of the façade. The shading system could be categorized briefly according to its location as external or internal. The external shading strategy is more efficient as it prevents the interior space from solar exposure; however, it requires higher maintenance. Shading system could also be further categorized by the control possibilities as either movable or fixed systems. The movable systems would provide higher adaptability to the context; however, higher maintenance is required. Furthermore, orientation is a crucial parameter used in determining the shading type. For instance, a horizontal screening louver would block direct sunlight on the south side with minimal visual interference, while cantilevers shading device would block high angle sun rays in the summer. Vertical louvers are more suitable for the east and west facades due to the low altitude sun angle.(Konstantinou&Prieto,2018) According to the literature review by Prieto et al., shading strategy in warm and humid climate shows a comparatively high mean and median values when compared with the glazing type or window to wall ratio strategy. While the maximum values reach 55.6% based on a study conducted in Bangkok. Moreover, the authors had concluded that equator-facing offices have larger cooling saving potential due to the high solar incidence angle in the north and south façade. Additionally, it is also implied that louvers and screens show a higher cooling saving potential due to the amount of exposed window area. (Prieto et al. 2018)

Ventilation

Ventilation is one of the common passive strategies to dissipate heat. It uses air as a heat reservoir in reducing the indoor temperature. Ventilation strategy can be categorized based on its principle as diurnal ventilation or comfort ventilation and nocturnal ventilation or night-flush ventilation. Diurnal ventilation functions during the peak demands, while nocturnal ventilation flushed out stored head during the night to cool down the building. (Konstantinou&Prieto,2018) In their studies, Prieto et al. have concluded that ventilation strategies achieved the highest cooling demand savings. The mean and median values observed for a warm and humid climate is 33% and 30% respectively. However, the authors also pointed out that the warm and humid climate imposed a challenge on the application of natural ventilation strategy. As a high level of humidity needs to be controlled to regulate indoor comfortability and preventing deterioration of building components due to condensation. (Prieto et al. 2018)

Glazing Type

Due to the recent building trends, openings are one of the main components of the building envelope. As skyscrapers tend to have transparent façades to provide optimal views, daylighting and sometimes ventilation. Glass itself is characterized by its relatively low thermal properties. However, new technologies allow integration of Low-E coating and air-spaces insulations providing better insulation for the glazed façade. (Konstantinou&Prieto,2018) Despite the technological improvements of coatings and gas-infused air cavity, according to Prieto et al. the use of different glazing types shows the lowest energy saving potentials. In warm-humid climates, the mean and median values are at 12% and 10%. The maximum energy saving potential was found in the humid subtropical (Cfa) climate of Milan with the values of 39%. The authors had concluded that the increasing numbers of glass layers and changes in color properties doesn't have many effects on cooling demand saving. However, when both parameters are combined, it has a higher saving potential. (Prieto et al. 2018)

Window to Wall ratio

While transparent materials provide views and daylighting, it also introduces heat into the interior spaces. Therefore, by regulating the window to wall ratio, the thermal performance of the façade could be improved. As a material with high thermal resistance or insulations can be integrated as a component of the opaque wall to reduce transmission heat losses. (Konstantinou&Prieto,2018) The data gathered by Prieto et al. had shown that a mean of 18% and a median value of 14% energy saving potential in a warm and humid climate. The maximum energy saving potential is reported in Shanghai at 43.7% and a tropical savanna climate of Malina at 41.1%. In conclusion, the authors concluded that a cooling demand saving tends to be higher as the relative window sizes decrease. (Prieto et al. 2018)

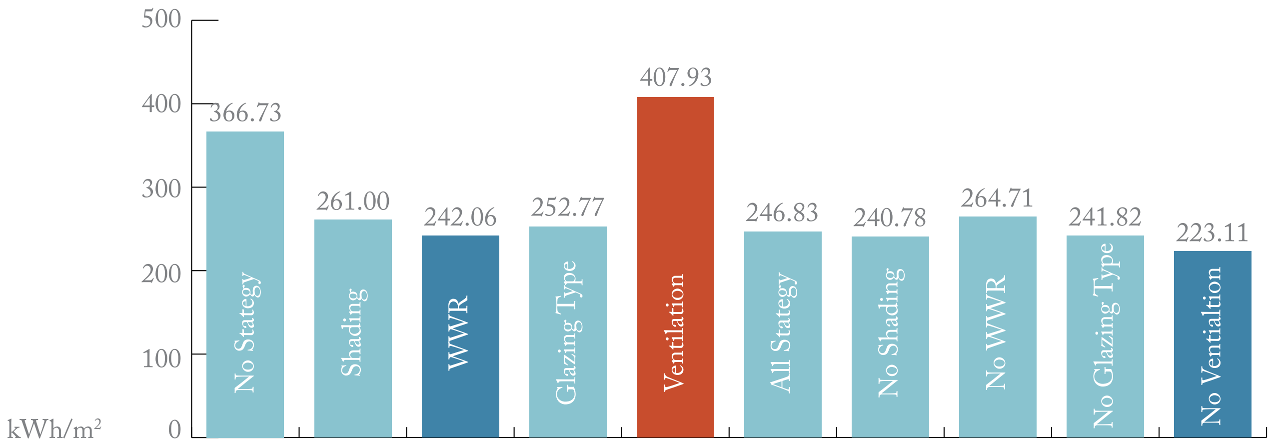


Figure 4-2 : Summary of simulated energy consumption reduction strategies based on Singapore's climate (Prieto et a., 2018)

Conclusion

Figure 4-2 depicts the simulation conducted by Prieto et al. based on Singapore hot and humid climate. In comparison to the benchmark situation of with no heat prevention strategies, shading, window-to-wall ratio, and glazing type have a similar cooling demand reduction potential. However, it is evidenced that ventilation strategy is not suitable for a warm-humid climate as it reflects a higher cooling demand relative to the base scenario due to the high latent loads. However, from the scenarios of the combined strategies, the implementation of window-to-wall ratio strategy show cooling demand savings in all cases. Therefore, the authors had concluded that smaller glazed area is highly recommended in a warm humid climate. While the shading and glazing type should be considered according to the façade requirements since they are working on a similar principle and could impose adversary effect. (Prieto et al. 2018)

4.1.2 Shading Strategies

According to Prieto et al.'s studies, it could be inferred that shading strategy has a viable potential to be applied in a warm and humid context. While ventilation seems to have a better performance according to the literature, but due to its limitation in the hot and humid context, shading strategy has been chosen for further exploration. Moreover, due to one of the design parameter to minimize maintenance in the design phase, only variables regarding external-fixed shading system would be discussed.

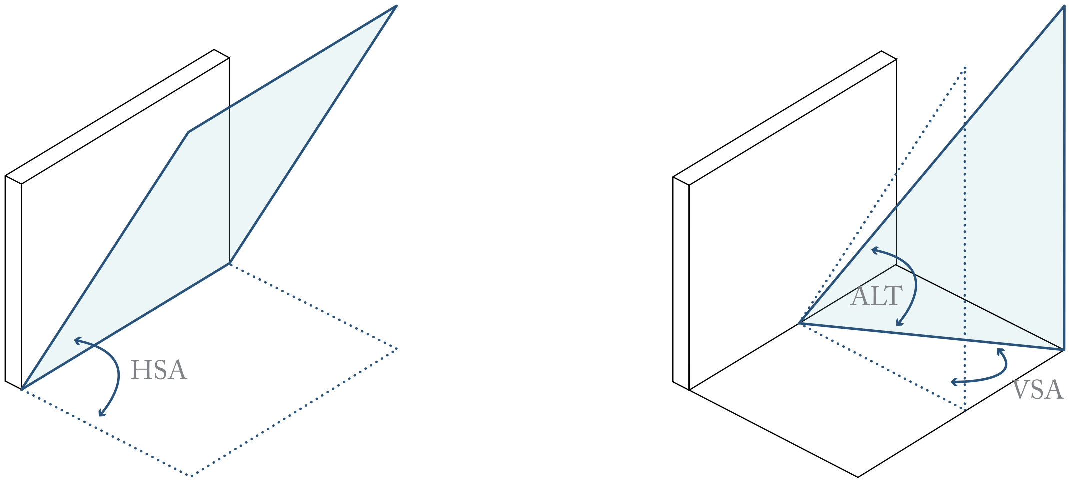


Figure 4-3 : Horizontal Solar Angle (HSA) and Vertical Solar Angle (VSA)

As briefly discussed in the previous chapter, external shading is the most effective shading strategy, as it prevents direct sunlight during summer with high incidence angle. However, if it is not designed effectively, it would reduce the natural lighting in the interior space. (Al-Tamimi&Fadzil, 2011) Therefore, orientation should be taken into consideration. Factors such as Horizontal sun angle (HSA) and Vertical sun angle (VSA) should be analyzed for optimal performance. Horizontal solar angle (HSA) can be cut-off by the integration of vertical fins, while vertical solar angle (VSA), at critical hours, can be cut-off by horizontal fins. Therefore, typically, shading devices should be applied to the south façade permanently, while vertical fins or moveable shading system are suitable for the east and west façade. (high-performancebuildings.org) However, to achieve optimal performance, detail analysis of the specific site context should be conducted, to obtain a specific design and shading pattern.

4.2 Dehumidification Strategy

Due to the high humidity of the tropical climate, a conventional vapor compression system such as an air-conditioning system requires higher cooling energy.; first to dehumidify the air then to decrease its sensible load. To be able to increase the efficiency of the cooling process, dehumidification strategies need to be implemented either as a supplementary system to the conventional system or as a standalone system. Moisture content in the air can either be regulated by condensing the water vapor as done in a conventional compression system or by using a suitable adsorbent. Therefore, the use of desiccant cooling systems as a dehumidification strategy is a viable alternative to the conventional vapor compression system, as they do not require coolants or refrigerants and consume less energy. (Sahlot&Riffat, 2015)

The principle of a desiccant system is based on the principle of moisture transfer due to the difference of vapor pressure between the air and the adsorbing material. The dehumidification process occurs as the cool desiccant with low moisture content absorb moisture from the air until its vapor pressure is in equilibrium with the air. However, when it reaches the equilibrium stage, the desiccant needs to be regenerated. The desiccant must be heated so that the vapor pressure on the desiccant surface is higher than the surrounding air, enabling the desiccant to remove the moisture. Due to the low regeneration temperature requirement, solar energy and waste heat could be used. (Misha et al., 2012) The desiccant system can be classified by the type of desiccant used in the system, commonly as liquid or solid.

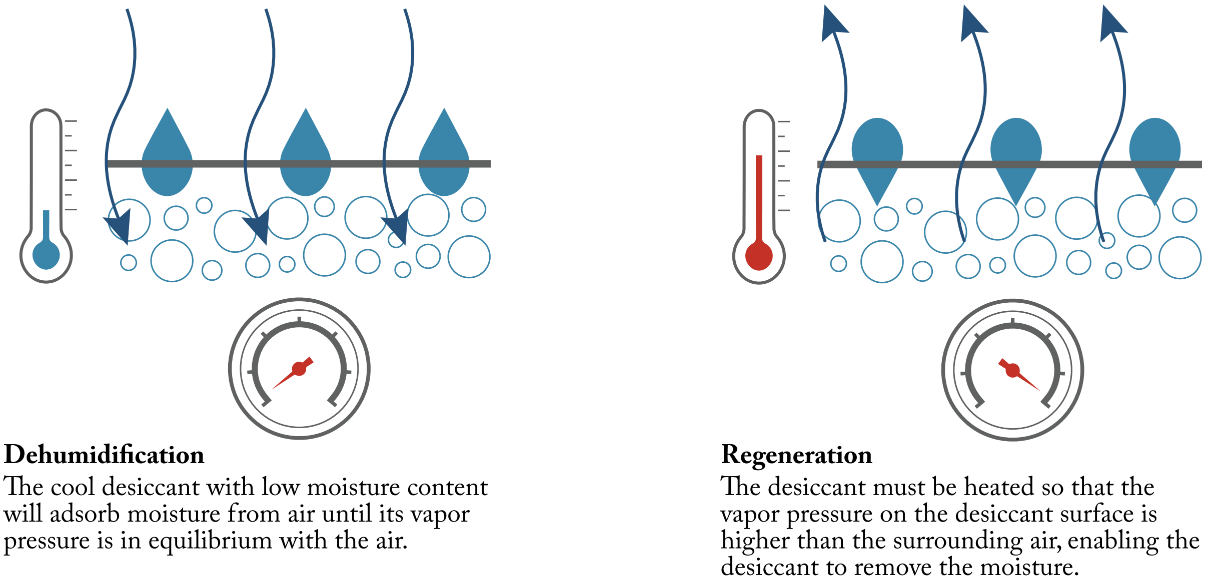


Figure 4-4 : Principle of a desiccant system.

4.2.1 Liquid Desiccant

The principle of the liquid desiccant system is presented in Figure 4-5. A typical liquid desiccant system is primarily composed of a dehumidifier and a regenerator.

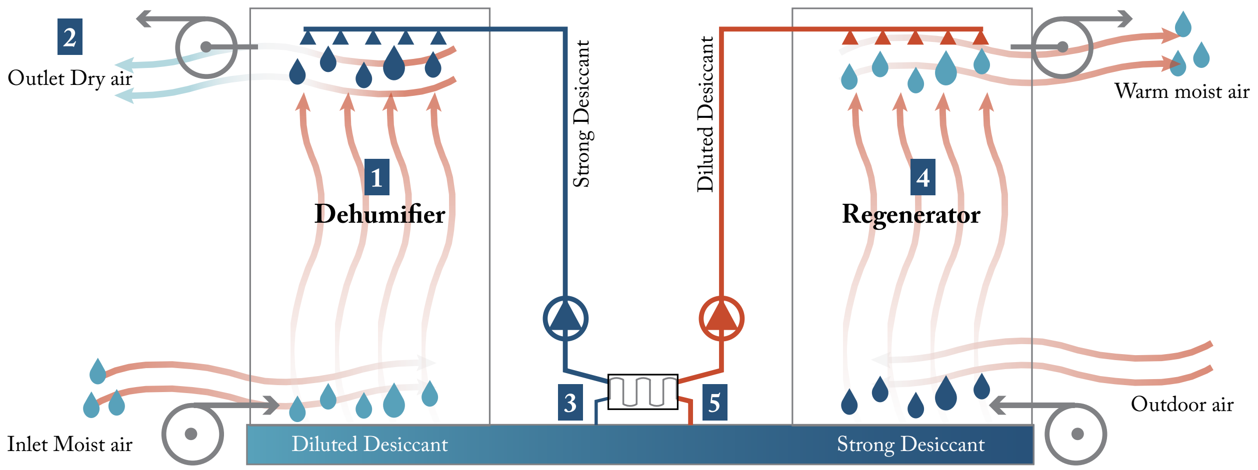


Figure 4-5 : Liquid desiccant system scheme

1. Moisture of the inlet air is removed in the dehumidification unit, as the liquid adsorbent absorbs the water vapor. Due to the difference in vapor pressure, mass transfer occurs and heat is released as water condenses and heat exchange during the mixing process

2. After dehumidification, the air is either provided into the conditioned space or sends to another cooling process. While the diluted desiccant is send back to the regenerator.

3. Prior to entering the regenerator, the solution is passed through a liquid to liquid sensible heat exchanger and then a heating coil to preheat the solution.

4. During regeneration, the diluted solution is exposed to the regenerative air and moisture is transferred from the diluted solution to air due to vapor pressure differences.

5. Then the regenerated solution passes through a liquid to liquid heat exchanger and a cooling coil before it enters the dehumidification unit.

In comparison to a solid desiccant system, the liquid desiccant system has more complexity. At the same time, the system has high flexibility. Due to the liquid property, alteration of the system configuration could be made, allowing the dehumidifier and regenerator to be separated. This would make the system more compact. (Misha et al., 2012) Furthermore, the system imposed a low-pressure drop making it suitable for low regeneration temperature, and the desiccant itself could be stored and use when the heat source is not available. (Sahlot&Riffat, 2015) Furthermore, liquid desiccant has the potential to absorb organic and inorganic contaminant, and it is non-toxic and odorless. (Misha et al., 2012) Nevertheless, while it is non-toxic, several liquid desiccants such as Lithium chloride and lithium bromide and other salts are corrosive and could face the problem of crystallization, which reduces the life span of the system. Moreover, there is a possibility of carry-over of liquid desiccant into the conditioned space, which could harm the health of the occupants. (Sahlot&Riffat, 2015)

4.2.2 Solid Desiccant

A solid desiccant system is defined by the used of solid desiccant material to absorb moisture from incoming air. The solid desiccant system operation is straightforward; as the material do not react chemically with the moisture content of the treated air. In comparison to the liquid desiccant system, the solid desiccant system has a higher drying capacity, and it can withstand various ranges of temperature variation. Moreover, due to an uncomplicated system configuration, the solid desiccant system has a lower maintenance requirement, and it consumes lesser operational energy. (Baniyounes et al., 2013) However, the system also has some drawbacks, as it imposes a higher pressure drop and higher regeneration temperature in comparison to the liquid desiccant system. (Misha et al., 2012)

Nevertheless, due to the attractive low maintenance, economical, and uncomplicated characteristic of the solid desiccant system, it presents itself as a viable option for facade integration. Therefore, further investigation regarding the material package and system configuration will be carried out in the following chapters. The solid desiccant system can be commonly categorized by the packaging configuration of the material as a rotary system or fixed bed system. However, new techniques such as desiccant coating also show potential in increasing system efficiency and compactness. Therefore, the following chapter will elaborate on the potential each system possessed in search of a suitable system for façade integration.

4.2.2.1 Rotary Wheel System

A rotary desiccant wheel system acts as an energy recovery heat exchanger that reduces the relative humidity. The walls of the rotary wheel are commonly fabricated in the shape of honeycomb or corrugated stacks to create a matrix structure which maximizes the area. (O'Connor et al., 2016) The wheel is divided into two main sections. In one part of the wheel, the moist air gets dehumidified; on the other section, hot air is supply to regenerate the wheel.

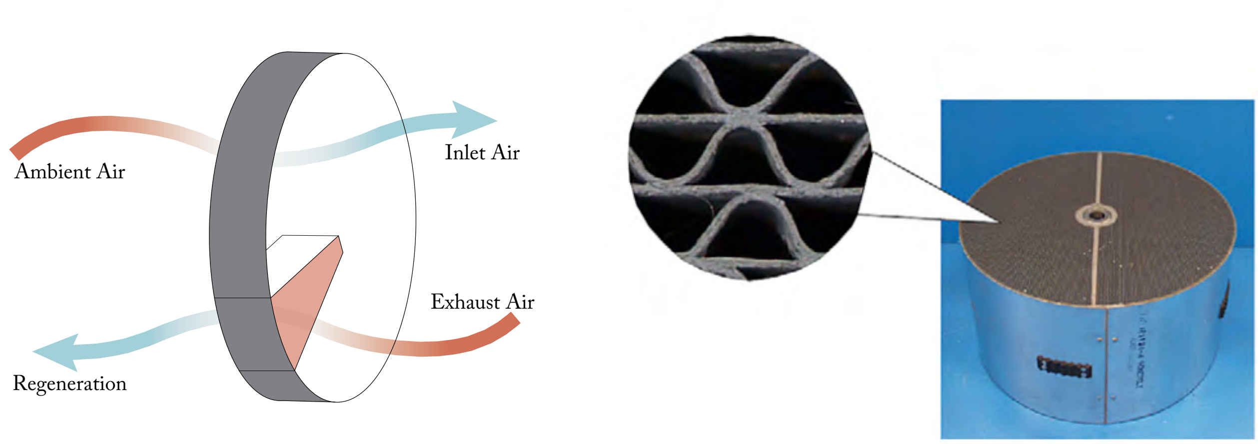


Figure 4-6 : Rotary wheel used in desiccant cooling system. (Jeong et al.,2011)

During operation, the wheel rotates at a constant speed, making the dehumidification process continuous. Studies have shown that by packing the desiccants on a wheel, the contact between the air molecules and the desiccant matrix improved. Additionally, a rotary desiccant wheel is claimed to be relatively compact, less subjected to corrosion, and can work constantly. (Li et al., 2010) However, with the rotary wheel system, the adsorption capacity is reduced, due to the heat released during the adsorption process. As its temperature increases, its adsorption capacity would decrease. Furthermore, since regeneration is heated by hot air, two heat transfer resistances occur. The first from the heat source to the air, and secondly from the air to the desiccant. (Vivekh et al., 2018)

4.2.2.2 Fixed Bed System

A fixed bed system incorporates the desiccant matrix by packing it in a stationary bed. Being stationary, the fixed bed system is relatively free from mechanical problems, while offering a high ratio of solid surface area to volume (Yeboah&Darkwa, 2016) Additionally, since the packed bed operates without dust and pollution, and adsorption heat could simply be removed through an inner cooling dehumidification process, the solid desiccant fixed bed is a viable option for the solid desiccant system.

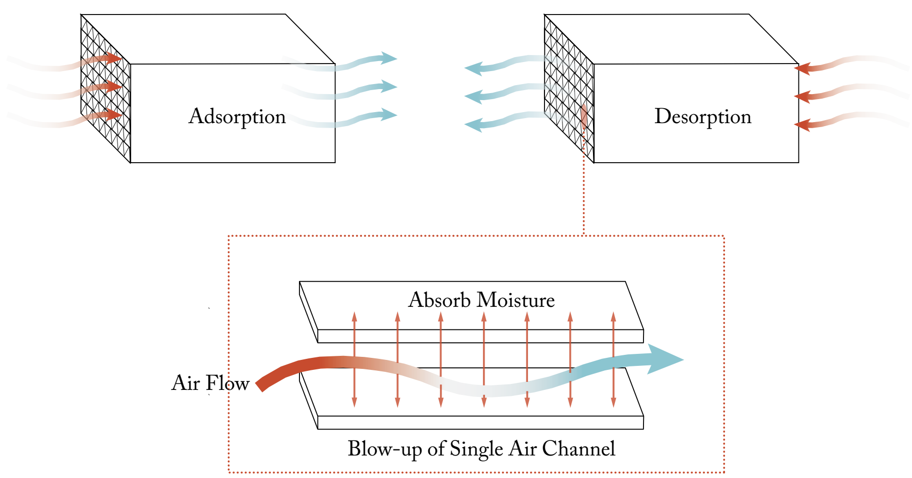


Figure 4-7 : Fixed-bed schematic diagram (Jeong et al., 2011)

In operation, the moist air or the process air would flow through the bed. During the dehumidification process, the desiccant would absorb the moisture of the process air, producing dry air. Alternately, during the regeneration process, hot air is circulated through the desiccant matrix to reactivate the desiccant. However, the system would require the regeneration and dehumidification process to operate alternatively. This limited the continuous adsorption process. Therefore, to achieve continuous dehumidification, several beds are installed. Hence, a large surface area per cooling capacity is required. While the structure of the fixed bed system is simple and easy to operate, it has a relatively low heat transfer coefficient due to poor contact between the bed and the desiccant matrix. Therefore, it will affect the adsorption capacity and cooling performance of the fixed bed desiccant system. (Vivekh et al., 2018)

4.2.2.3 Desiccant Coated Heat Exchanger (DCHE)

Desiccant Coated Heat Exchanger (DCHE) system was developed to deal with the limitations of the conventional desiccant packages such as the fixed-bed and the rotary system. DCHE is fabricated by coating the desiccant material directly onto the fin side of a conventional fin tube heat exchanger.

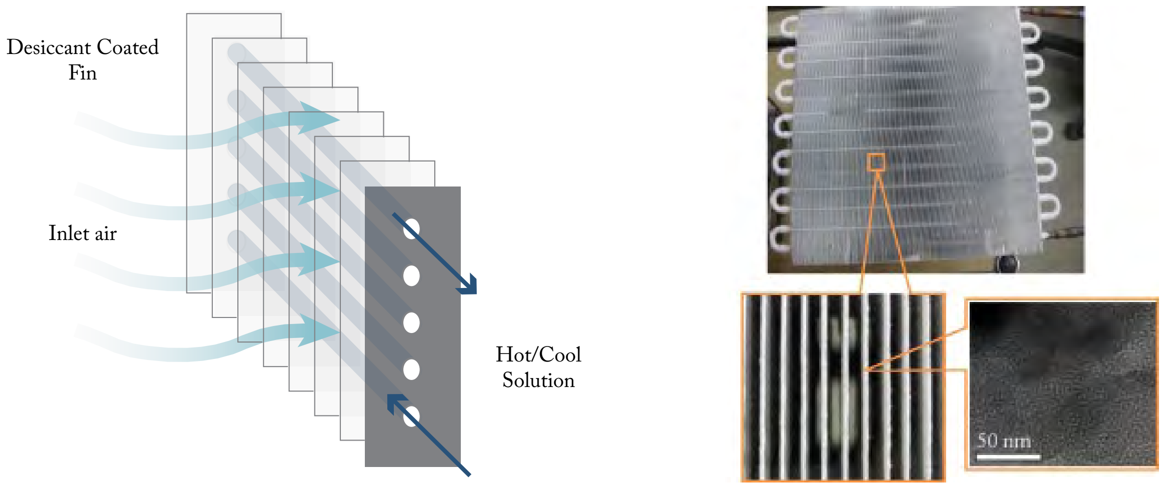


Figure 4-8 : Desiccant coated heat exchanger schematic diagram (Vivekh et al.,2018)

The principle of the system increases the desiccant's adsorption capacity by removing the adsorption heat. During humidification, the cooling fluid passed through the tubes removing the sorption heat, as a large portion of the latent load is removed from the air; hence, the total cooling load is reduced. Through the means of the tube heat exchanger, regeneration could be done with relatively higher efficiency since metal fin with higher thermal conductivity facilitates the heat transfer process. Nevertheless, a single DCHE system alone cannot provide continuous dehumidification. (Vivekh et al., 2018)

As a result of the alteration between dehumidification and regeneration mode, cooling and humidity level don't synchronize. In dehumidification mode, because of a significant temperature difference, the heat transfer rate is relatively fast as the process air gradually decreases to a constant state. Similarly, the humidity ratio of the process air also decreases quickly to its minimum state, then it gradually increases to the ambient condition, due to the limitation of the desiccant adsorption state. For efficient heat transfer, high thermal capacity materials such as aluminum and coppers are used in fabricating the heat exchanger. This caused the temperature to react slower than the humidity ratio. Therefore, when humidity is low, the corresponding temperature is relatively high. (Ge et al., 2017)

4.3 Cycle Schematic

Low-exergy cooling is not a relatively new concept. The concept of evaporative cooling can be dated back to the 1930s, while indirect evaporative cooling is developed as an improvement system a few years later. With technological developments, the systems are applicable in a broad climatic range. However, they still failed to work efficiently in humid climates. However, the challenge was overcome by Pennington, as he proposed a system configuration which a dehumidifier with a heat source is coupled to a double regenerative evaporative cooler. (Penney&McClain, 1985) Throughout the years, several attempts had been made to develop the system. Therefore, the developed configurations will be discussed in this chapter.

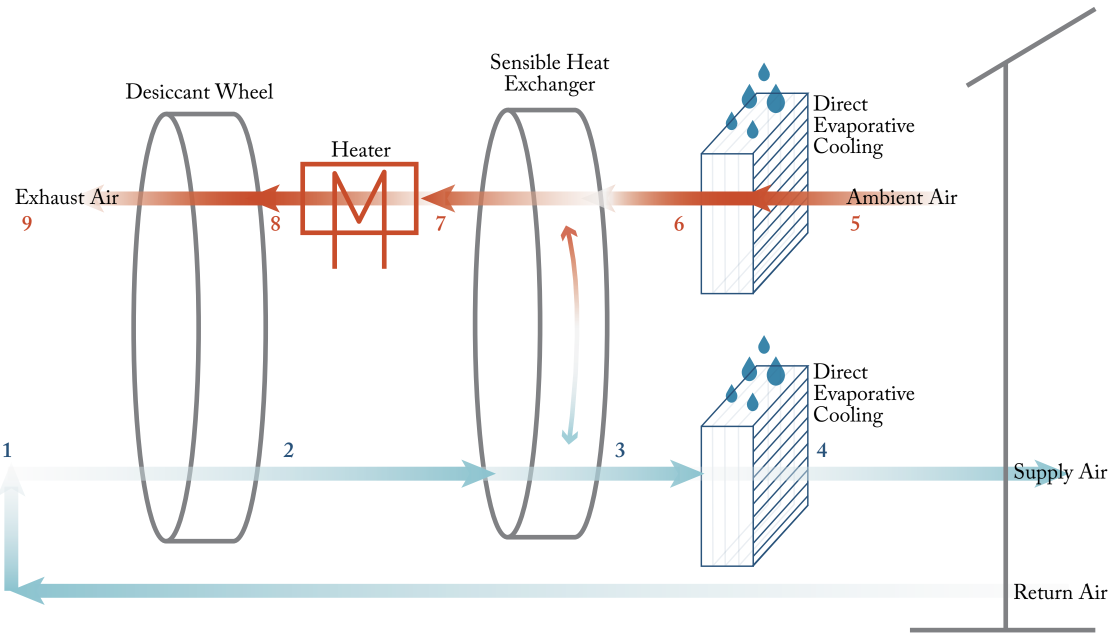


Figure 4-9 : Pennington Cycle schematic diagram (Dai et al., 2010) [Redrawn by Author]

Pennington Cycle

Dehumidification

1. Ambient air passes through the desiccant wheel where moisture is being absorb.
2. Air with lower humidity but higher temperature
3. Air is cooled in the air sensible heat exchanger
4. Cooled by a direct evaporative cooler

Regeneration

5. Return air is cooled and humidify in a direct evaporative cooler
6. The air is then pre heat through the sensible heat exchanger
7. The air is heat up further in order to regenerate the desiccant wheel
8. The hot air is used to regenerate the desiccant wheel
9. It is exhaust to the environment

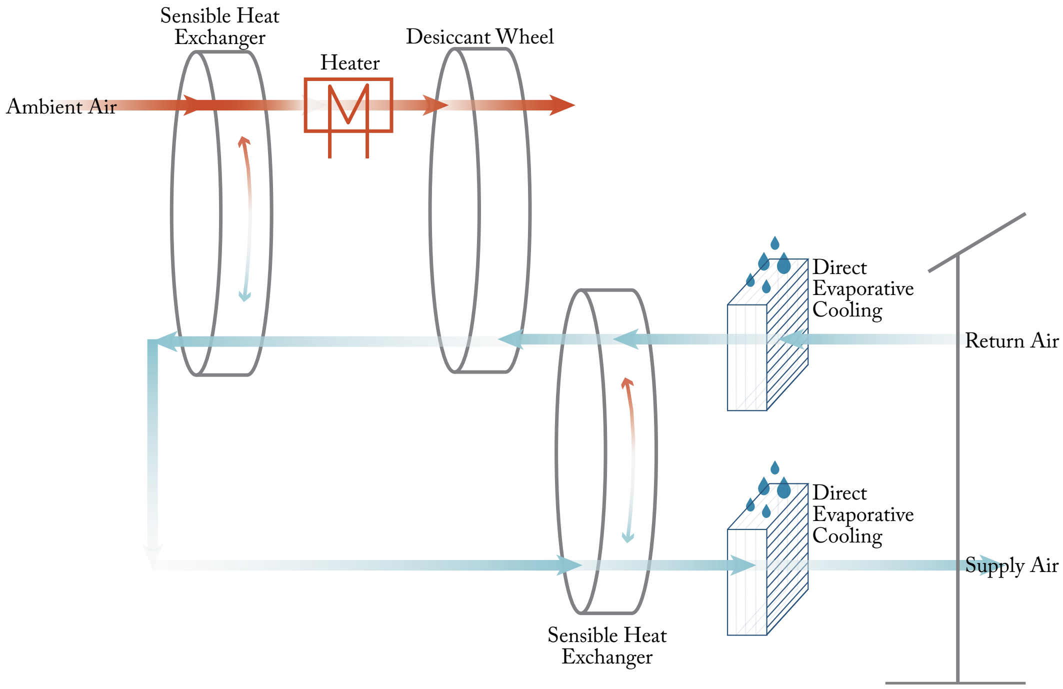


Figure 4-10 : Dunkle Cycle schematic diagram (Dai et al., 2010) [Redrawn by Author]

Dunkle Cycle

Dunkle cycle [Figure 4-10] proposes a development to the Pennington cycle by combining the quality of the ventilation cycle to supply fresh air at relatively low temperature with the recirculation cycle to provide higher efficiency. It incorporates an additional sensible heat exchanger to generate a lower temperature process air for the heat exchanger. (Jani et al., 2016)

SENS Cycle

While ventilation of fresh air enhances a more comfortable and healthier indoor environment, it also increases the cooling loads. Therefore, fresh air proportion should be regulated and maintained at the required level to provide favorable indoor air quality while maintaining system efficiency. Therefore, the SENS cycle was proposed by MacLaine-cross. (Jani et al., 2016) The principle of the SENS cycle is depicted in Figure 4-11.

1. The ambient air is dehumidified by the desiccant wheel
2. It is sensibly cooled by two-sensible heat exchanger connected in a series
3. The process air is mixed with the room return air
4. Then it is further cooled in a cooling coil through which the heat is exchanged with the cooling tower
5. The supply air is divided into two parts
6. Redirected to the cooling tower, and exhausted into the ambient after exchanging heat with the process air
7. Supplies to the conditioned space

With the series of sensible heat exchanger, the SENS cycle can achieve the highest thermal COP. However, it didn't receive as much attention due to the high complexity of the system. (Dai et al., 2010)

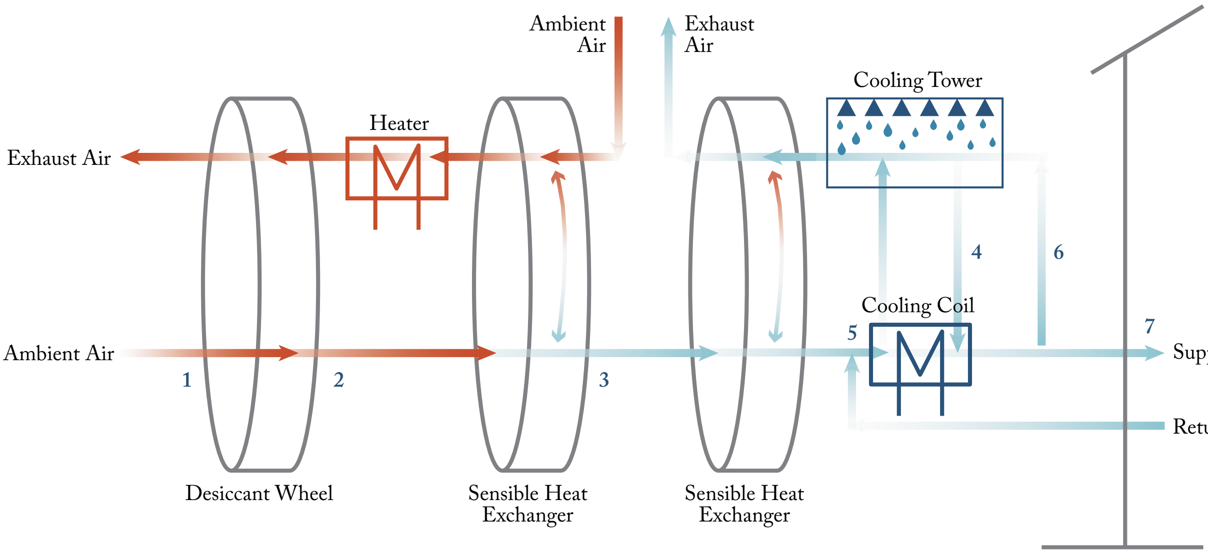


Figure 4-11 : SENS Cycle schematic diagram (Dai et al., 2010) [Redrawn by Author]

DINC cycle

The direct-indirect evaporative cooling (DINC) cycle was proposed as a modification to simplify the complexed SENS cycle. The modification was done by replacing the sensible heat exchanger, cooling tower and cooling coil with a series of an indirect evaporative cooler and a direct evaporative cooler. The configuration provides the system with a COP of 1.6 (Dai et al., 2010)

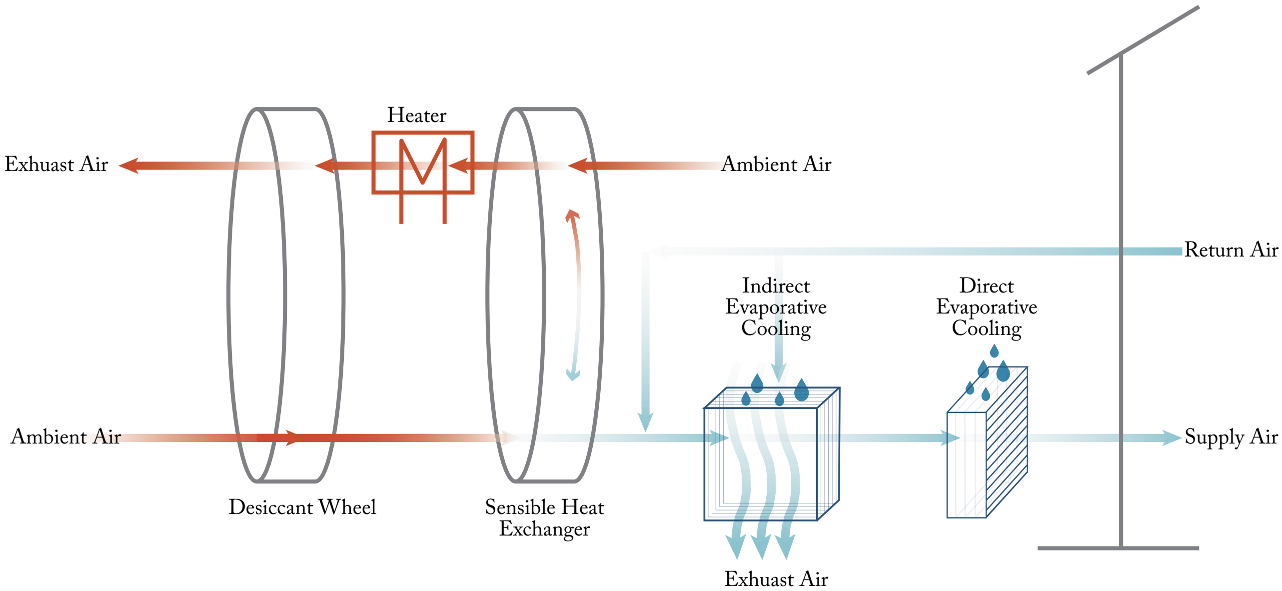


Figure 4-12 : DINC Cycle schematic diagram (Dai et al., 2010) [Redrawn by Author]

4.3.1 System Integrations and configurations

Pennington had shown the potential of the integration of the desiccant system with an evaporative cooling system [Figure 4-13] to expand the possibility of a low ex cooling system in the hot and humid climate. Due to the advancement of recent technologies, the desiccant system shows excellent prospect to be coupled or integrated with other cooling and regenerating system providing higher efficiency. Therefore, this chapter would discuss the various potential to incorporate a desiccant system with different cooling and regenerating systems such as vapor compression cooling and solar collectors. (Jani et al., 2016)

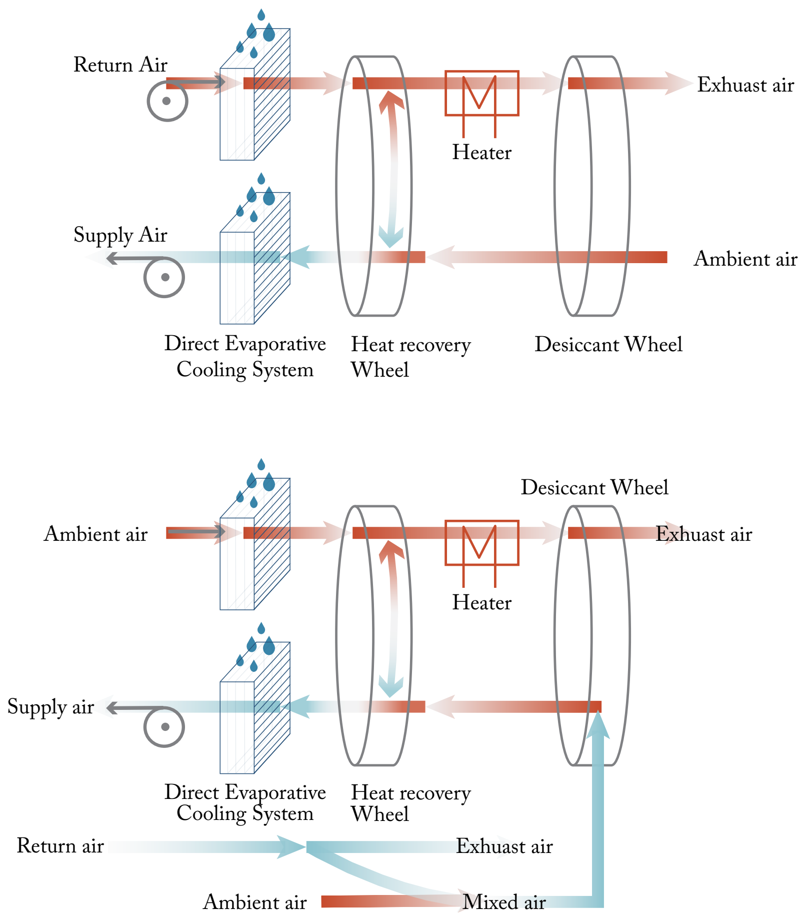


Figure 4-13 : Desiccant system integration with evaporative cooling system (Jani et al., 2016) [Redrawn by Author]

Integration of desiccant and evaporative cooling system

The schematic of the system configuration of a desiccant system with an evaporative cooling system discussed in the previous chapter is depicted in Figure 4-13. It portrays both the configuration of the ventilation mode along with the recirculation mode. In the ventilation mode, 100% of fresh air is being drawn into the system, while the recirculation mode mixed the fresh air with the recirculated air returned from the conditioned space. Jani et al. had concluded that the recirculation configuration would require higher energy for regeneration due to the lower temperature from the return air. However, in comparison to the cooling demand to cool down the ambient temperature, the higher regeneration energy becomes insignificant. Therefore, it could be concluded that the COP of the recirculated configuration remains higher than the ventilation configuration. (Jani et al., 2016)

Integration of desiccant and solar collector system

Solar collector integration with the desiccant cooling system [Figure 4-14] was developed as one of the prospect alternatives to the typical evaporative integrated desiccant system, as it consumes less electrical power for regeneration due to the use of solar energy. The principle of the system remains unchanged. However, the electric heater is replaced with a solar heating system. The solar heating system is composed of a solar collector or a PVT panel, back up heater, storage tank, circulating pump and a liquid to air heating coil. The system operates to convert solar radiation to thermal energy to be used in the regeneration of the desiccant system. However, when there is insufficient solar radiation, the backup heater will be used to generate the required thermal energy. (Jani et al., 2016)

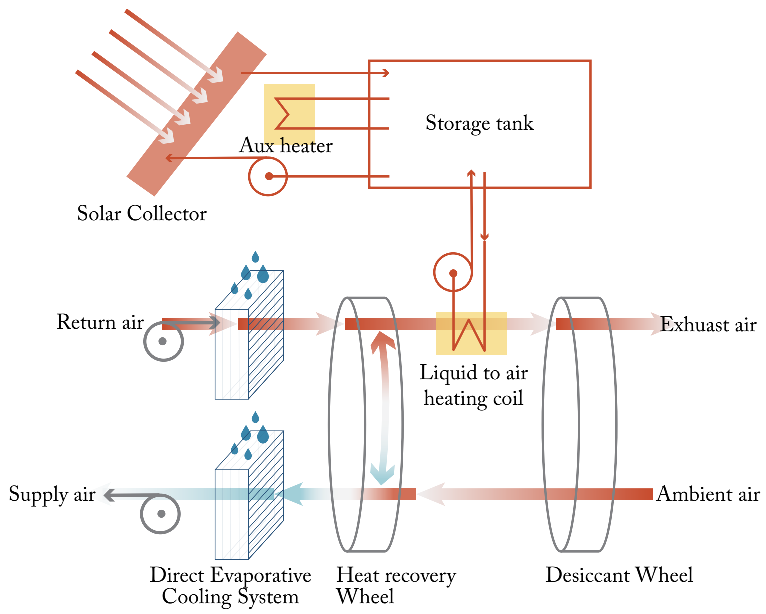


Figure 4-14 : Desiccant system integration with solar collector (Jani et al., 2016) [Redrawn by Author]

Integration of desiccant and vapor compression system

Vapor compression system presents itself as a viable system for the situation in the hot and humid climate where the evaporative system fails to efficiently handle the sensible heating load due to the exceeding amount of humidity. With the integration of the desiccant dehumidification system with the vapor compression system, latent load and sensible heat load are being handled separately. Therefore, the performance of both system would improve significantly. (Jani et al., 2016) The hybrid system studied by Yadav and Kaushik is depicted in Figure 4-14. (Yadav&Kaushik, 1991)

Dehumidification

1. The air from ambient space is mixed with a portion of return air
2. The mixed air is processed through a desiccant dehumidifier, while the rest of the return air is exhausted to the atmosphere
3. The dehumidified air is sensibly cooled by an indirect evaporator
4. Further sensible cooling is carried out by the vapor compression cooling system

Regeneration

5. Waste condenser heat is used to preheat the ambient air
6. The preheated air is heated by an auxiliary heater to the required regeneration temperature
7. After regeneration, the heat is released to the ambient environment

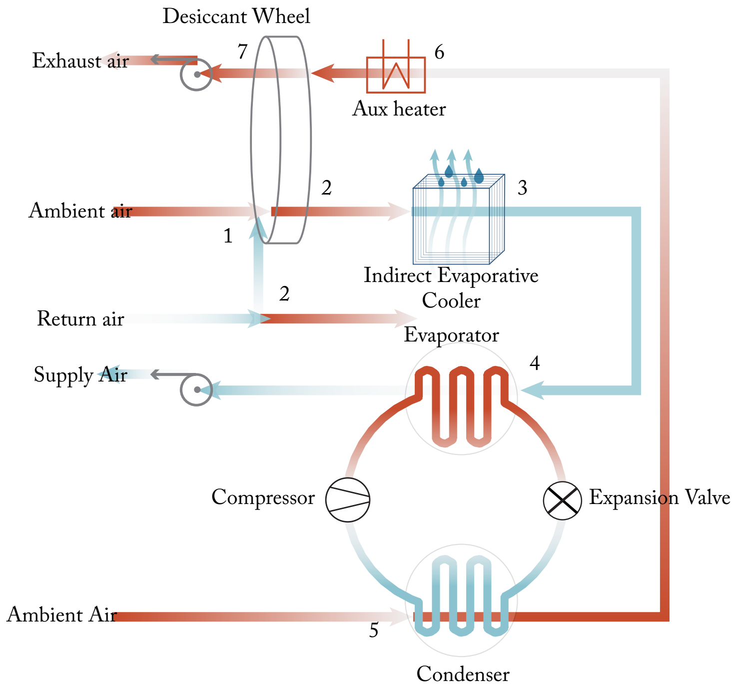


Figure 4-15 : Desiccant system integration with vapor compression system (Jani et al., 2016) [Redrawn by Author]

Integration of desiccant and vapor compression cooling system

The integration of a solar collector to the desiccant and vapor compression system provides a further reduction on both electricity consumption and carbon emission. The principle of the system is similar to the scheme of the solar collector to evaporative cooling and desiccant system. However, the evaporative cooling system is replaced by a vapor compression system. (Jani et al., 2016)

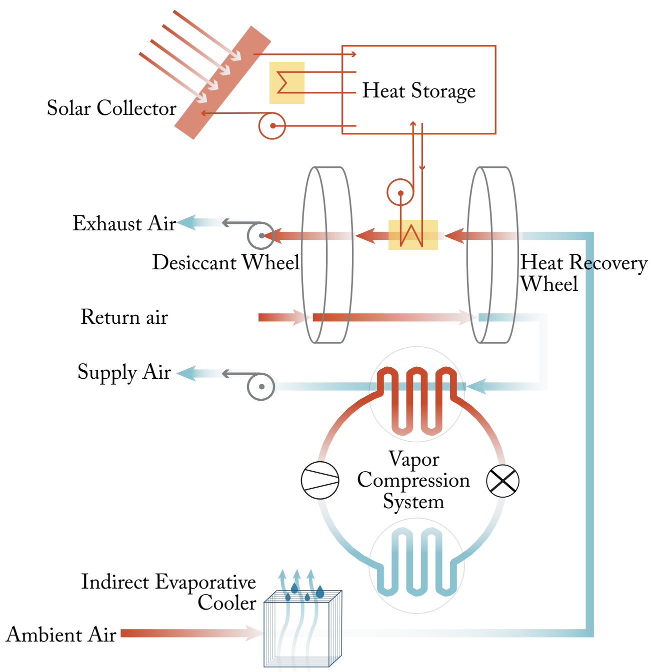


Figure 4-16 : Desiccant system integration with vapor compression system and solar collector (Jani et al., 2016) [Redrawn by Author]

4.4 Regeneration System

Regeneration is a part of the process which allows the desiccant dehumidifying system to operate continuously. As the desiccant continues to absorb moisture from the air, its efficiency gradually decreases. Therefore, water vapor absorbed by the desiccant material must be driven out to enable the system to absorb moisture in the next cycle. Regeneration could be done by heating the desiccant material to the regeneration temperature, which varies upon the type of the desiccant used. (Jani et al., 2016) Vivekh et al. had summarized the regenerative temperature required by the common desiccant material in his study. While there are numerous methods to regenerate the desiccant materials; only sources from solar energy would be discussed in this report.

4.4.1 Solar Collector

Solar collector is a type of heat exchanger that transform solar radiation energy into internal energy of a transport medium such as air or water. Solar collector could be categorized into two main groups: non-concentrating and concentrating. Non-concentrating solar collectors are stationary and consist of the same area for absorbing solar radiation. Concentrating solar collectors, on the other hand, usually, have a concave surface to collect solar radiation. (Tyagi et al., 2012)

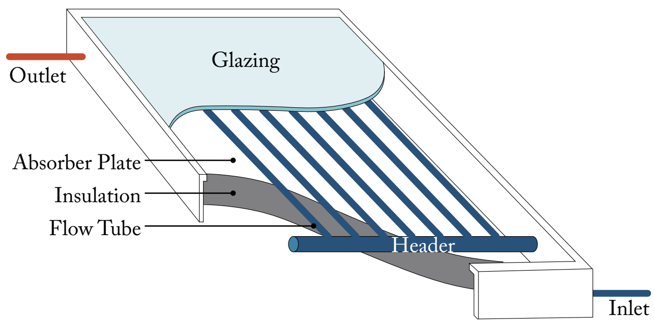


Figure 4-17 : Flat plate solar collector (Tyagi et al., 2012)

Flat plate collector (FPC)

Flat plate collector (FPC) is a type of non-concentrating solar collector. With its simplified mechanical configuration, flat plate collector is the most typical type of solar collector which are used for both domestic and industrial purposes. The FPC consists mainly of a flat blacken plate which absorbed the solar radiation, converting it into thermal energy and several tubes are attached to the absorber plates to circulate the harvested energy. Furthermore, it is insulated at the back and all sides to minimize heat losses. While a transparent cover is placed on top of the absorbing plate to reduce upward convection and radiation heat loss. The FPC is designed to operate under either a low-temperature range <60°C or a medium temperature range <100°C. (Tyagi et al., 2012)

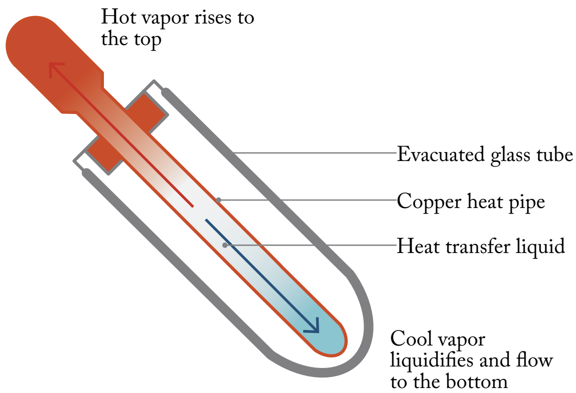


Figure 4-18 : Evacuated tube solar collectors (ETC) (Tyagi et al., 2012)

Evacuated tube solar collectors (ETC)

Evacuated tube solar collectors (ETC) works on a similar principle as a flat plate collector. However, ETC uses liquid-vapor phase change materials to transfer heat. The heat pipes are made of sealed copper pipe attached to a black copper fin and are placed inside a vacuum-sealed tube. Within the pipeline, a small amount of fluid would undergo an evaporating-condensing cycle in which the liquid would evaporate while the vapor would travel to the heat sink where it condenses – releasing its latent heat. Due to its highly selective surface coating and vacuum insulation, the absorber element has a relatively high heat extraction efficiency in the temperature range of 80C. (Tyagi et al., 2012)

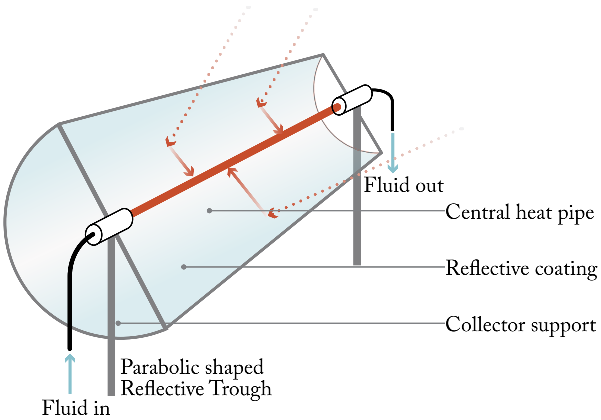


Figure 4-19 : Concentrating solar collector (Tyagi et al., 2012)

Concentrating solar collector

The concentrating solar collector usually incorporates the ability to track the sun to re-direct solar radiation through an aperture into the absorber. The solar radiation is optically concentrated before it is converted into heat. Therefore, the concentrating solar collector system can achieve relatively higher temperature thermal energy in comparison to a non-concentrating solar collector, at the same solar radiation. Due to minimize heat loss, the concentrating collectors also have higher thermal efficiency. Moreover, owing to its reflecting surfaces, it also requires lesser materials and is structurally simpler. However, the concentrator system is able to collect lower diffuse radiation, and despite its structural simplicity, it usually requires an advanced tracking system to achieve its optimal efficiency. (Tyagi et al., 2012)

4.4.2 Photovoltaic -Thermal (PVT) collector

Photovoltaic -Thermal (PVT) collector integrates both electricity production and thermal energy production into a single module. A conventional Photovoltaic (PV) cells only utilize a portion of the solar radiation to produce electricity while the remainder is transformed into waste heat. This reduces the efficiency of the PV cells due to the rising temperature. On the other hand, a PVT module is integrated with either air, water, or evaporative collectors to recover heat for alternative applications, such as regeneration of a desiccant cooling system. (Tyagi et al., 2012)

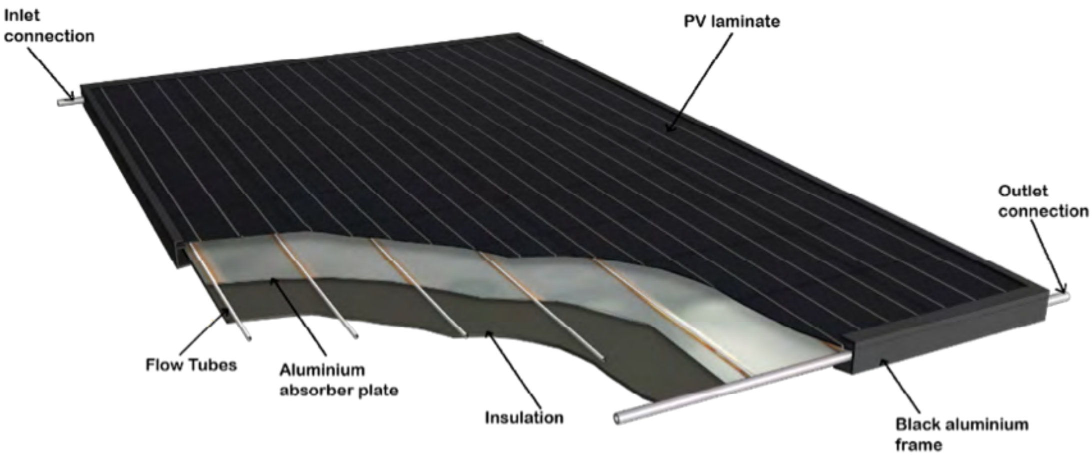


Figure 4-20 : Photovotaic thermal collector (Rawat&Dhiran, 2017)

Liquid PVT collector

The liquid PVT collect shares a similar principle with the flat plate collector. However, the configuration usually varies in a domestic and industrial application. In a local scale, the flat plate collectors work in parallel connection and run automatically with thermos-siphon. While in industrial application, industrial water heating system, a photovoltaic driven water pump is used to maintain the water flow. Rawat and Dhiran had conducted a study to compare the efficiency between the PVT/Liquid collector and PVT/Air collector. The authors had concluded that the PVT/liquid collector has a daily electrical efficiency of 7.54% and a thermal efficiency of 70% respectively. (Rawat&Dhiran, 2017)

Air PVT collector

In air-PVT collector, the air is used as a medium to recover the heat from the PV system. However, in comparison to water, the air has a lower thermos-physical property making it less efficient. Nevertheless, Air/PVT collector required low construction, minimal used of material, and relatively lower operating cost. Sukamongkokl et al. had experimented with verifying the dynamic performance of a condenser heat recovery with PVT/air collector in tropical climates. The authors had presented a result which shows that the system is capable of generating dry air with the temperature of 53C and 23% relative humidity. While 6% of the daily solar radiation could be obtained from the PVT system. (Sukamongkol et al., 2010)

According to the study of Rawat and Dhiran, the efficiency of the PVT/Air system has an electrical efficiency of 6.56% and a thermal efficiency of 56.47%. Additionally, the study had concluded that the system's total efficiency reaches 62.57%. (Rawat&Dhiran, 2017)

4.5 Cooling Strategy

4.5.1 Vapor Compression System

The vapor compression refrigeration system has been in practice for over 100 years; it is currently dominating the air conditioner market. The system is composed of 4 main components: an evaporator, a condenser, a compressor, and an expansion valve. Within the cycle, refrigerant, a phase changing material, is circulated. (Duan et al. 2012)

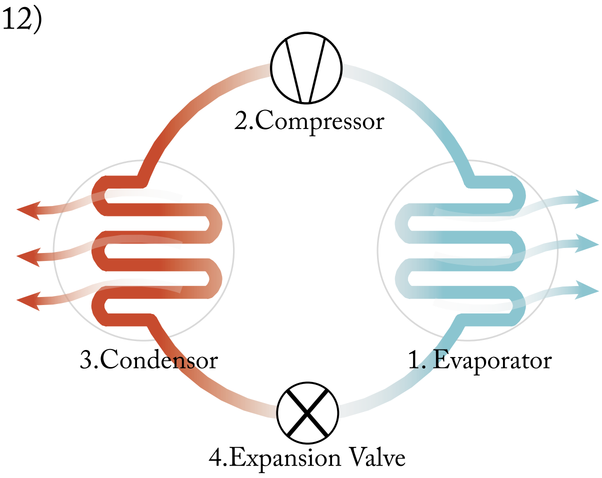


Figure 4-21 : Vapor compression cycle

1. In the evaporator, the refrigerant absorbs heat from the surrounding transforming itself from liquid to vapor – subsequently cooling the surrounding medium.
2. Then the refrigerant circulates to the compressor. Through the usage of electrical power, the compressor generates a high pressure, and saturated refrigerant vapor.
3. The high pressured, and saturated refrigerant vapor then enters the condenser. It loses the heat to the surrounding medium, hence resulting in the condensation of the refrigerant vapor.
4. The refrigerant then is led into the expansion valve, which through the throttle effect reduces the refrigerant pressure. Then the cycle restarts at the evaporator.

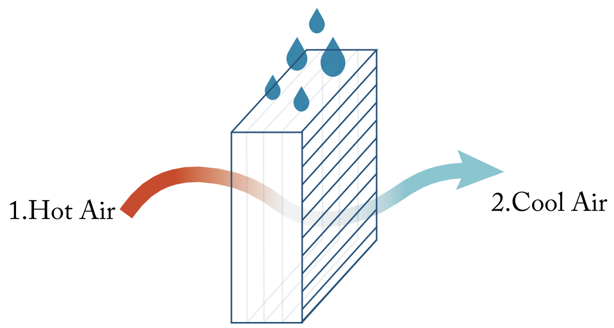
Due to its long history, the system is highly developed. Therefore, it presented a stable performance, low cost, long life cycle and reasonable energy performance with a COP in the range of 2-4 (Duan et al. 2012) However, vapor compression refrigeration system used in air condition system is still considered to be the primary contributor to ozone depletion and global warming. (Lu, 2007) Furthermore, the process efficiency is low, since the cooling coil is reducing both the latent and sensible heat at the same time. Additionally, due to the used of mechanical compressors, high-grade electrical energy is required as an input source. (Chua et al. 2012)

4.5.2 Evaporative Cooling

The principle of evaporative cooling is making air cool by increasing its water vapor content. As water is evaporated, sensible heat is being converted into latent heat, which resulted in the reduction of ambient temperature. (Tale&Gawali, 2017)

In recent years, evaporative cooling technology is gaining its popularity in the HVAC industry, due to its simple structure and usage of natural energy in the ambient. (Duan et al. 2012) The technology also is claimed to have a high-efficiency range of COP at 15-20 (Duan et al. 2012), low retrofit cost, low maintenance, and attractive payback period. At the same time, due to its usage of natural energy, it serves as a promising strategy to reduce carbon emissions and energy consumption. (Cuce&Riffat, 2015) Conair claimed that some humidifier could use 1,000kg of moisture could provide 680kW of evaporative cooling per hour while operating with less than 0.3 kW of electricity. (Conair.uk)

However, since the driving force of the system is being derived from the temperature difference between the dry-bulb and wet bulb temperature of the process air, the system would be more viable in a hot and arid climate. Since the dry-bulb and wet-bulb temperature in a humid or mild climate region is limited, the system cooling capacity would not be as efficient. (Duan et al. 2012)



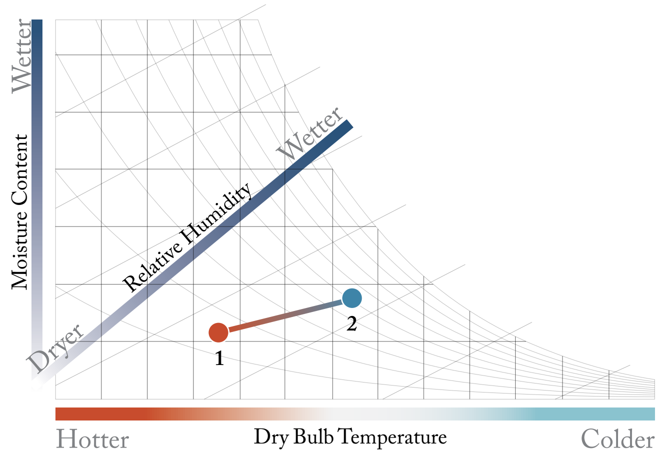


Figure 4-22 : Direct evaporative cooling schematic

4.5.2.1 Direct Evaporative Cooling

The principle of direct evaporative cooling is to keep the primary in direct contact with water. This causes the water to evaporate, and vapor is added into the air reducing the air temperature. (Duan et al. 2012)

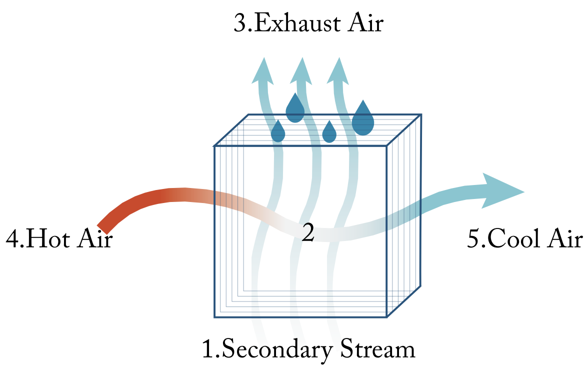
A typical direct evaporative system consists of large porous wetted pads, water sump, an electric motor driven fan, a water pump, and water distribution. The system is operated by drawing air through the porous wetted pads, allowing the sensible heat to evaporate some water. The heat and mass transfer between the air and water reduces the dry-bulb temperature of the air while increasing the humidity at a constant wet bulb temperature. The heat and mass transfer would stop when the dry-bulb temperature is air approaches the ambient air wet bulb temperature. (Jaber&Ajib,2010) With a honeycomb evaporative pad, the efficiency of the system could reach 90% decreasing air temperature of up to 8-10°C with lower wet bulb temperature. However, the humidity could be increased up to 70% (Tale&Gawali, 2017)

Adequate cooling capacity could be achieved with the system. However, the increasing level of air humidity could affect occupants' comfortability. (Cuce&Riffat, 2015) In addition to air quality, the evaporative cooling system also possessed certain disadvantages such as noise, and difficulty in controlling interior temperature. Furthermore, Bom et al. have presented a study which shows that if the ambient wet-bulb temperature is higher than 21°C, the cooling capacity will not be sufficient for indoor comfort cooling. (Bom et al., 2010)

4.5.2.2 Indirect Evaporative Cooling

As its name stated, in the indirect evaporative cooling system, water is being separated from the primary air source by a heat exchanger. The indirect process is a sensible cooling process which follows along with a constant humidity ratio since no moisture is introduced. (Jaber&Ajib,2010)

In a typical flat plate structure, a dry channel is positioned in adjacent to a wet channel. During operation, the primary air enters the dry channel while the secondary air enters the wet channel. The evaporation of the water would reduce the temperature of the plate, allowing a sensible heat transfer between the primary air and the plate. The operation allows the primary air to be cooled with constant moisture content with a similar wet-bulb temperature the secondary air. (Duan et al. 2012)



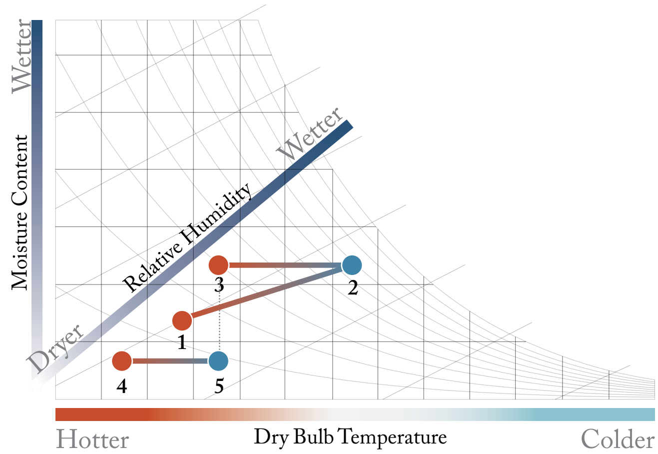


Figure 4-23 : Indirect evaporative cooling schematic

While it can cope with the moisture issue, the cooling performance of the indirect evaporative cooling is comparatively lower than the direct evaporative cooling system. (Cuce&Riffat, 2015) Considering air movement with a counter flow manner, only 40-80% of the wet-bulb temperature could be achieved. This is due to the limited heat-exchanging surface area, none pure counter flow pattern is unachievable, and the difficulty in obtaining a uniformed water distribution over the wet channel. (Duan et al. 2012)

However, with the development of a geometrical configuration for air distribution known as the Maisotsenko-cycle or the M-Cycle, the system could produce cold air of temperature lower than the wet-bulb ambient air temperature. (Rogdakis&Tertipis, 2015) A study conducted by Pacific Gas and Electric Company has shown that an M-cycle based heat exchanger could obtain wet bulb effectiveness of 81-91% and a dew point effectiveness of 50-60% which is 10-30% higher than of a conventional indirect evaporative cooling system. (Elberling, 2006)

4.5.2.3 Conclusion

In consideration of the cooling system to be implemented in a tropical climatic context such as Bangkok, Thailand, two main cooling strategies widely used, namely vapor compression cooling system and evaporative cooling had been studied. The parameters used in decision making considers both the performance aspects, environmental aspects, as well as the prospect of integrating the system with an additional system.

While the vapor compression refrigeration system presented itself as a highly developed system with low initial cost and stable performance, it contributes a negative impact on the environment. This is mainly due to the refrigerant used in the system as well as the high demand for electricity usage.

On the contrary, with the usage of water for cooling, the evaporative cooling system presented itself as a more environmentally friendly option. Furthermore, it offers a high-efficiency range of COP at 15-20, comparatively higher than the vapor compression refrigeration system of COP 2-4. At the same time, it has a low retrofit cost, low maintenance, and an attractive payback period. However, with the small difference between dry-bulb and wet-bulb temperature in a hot and humid climate, the system performance is limited. Nevertheless, by coupling the system with a dehumidification system, moisture content could be reduced before entering the evaporative cooling system.

In conclusion, with recent developments to improve its efficiency and performance, the indirect evaporative cooling system becomes a viable option as the developments allowed the air to be cooler than the ambient wet-bulb temperature while maintaining a constant humidity level and at the same time due to its usage of natural energy, the system is a promising strategy in tackling carbon emission and energy consumption issues.

4.6 Indirect Evaporative Cooling - Maisotsenko-cycle

Maisotsenko-Cycle (M-Cycle) or the dew point evaporative cooling system, was proposed by Professor Valeriy Maisotenko as an approach in developing a more efficient heat exchanger configuration. With the means of a flat plate, cross-flow and perforated heat exchanger, the M-cycle combines the thermodynamic process of heat transfer allowing the temperature of the product air to approach ambient dew point temperature. (Duan et al. 2012) The most popular application of the M-cycle lies in the air conditioning system. Studies had shown that the system could reduce energy consumption by up to 90% in comparison to the conventional system. (Pandelidis et al., 2018)

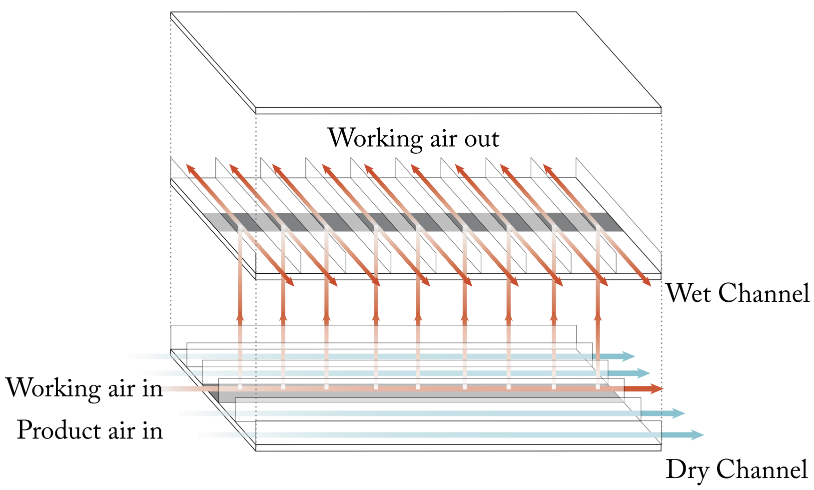


Figure 4-24 : Maisotenko Cycle schematic diagram

4.6.1 System Configuration

Similar to an indirect evaporative cooling system, the M-Cycle is composed of a wet and a dry channel. As the ambient air is drawn into the dry channel, it loses sensible heat to the wet channel, lowering the temperature of the outlet air. Therefore, the M-Cycle diverts a fraction of the outlet air to act as the working air in the wet channel. In the wet channel, the working air is humified as it absorbs the heat from the dry channel; at the end, it is rejected to the atmosphere. This cycle allows the reduction of the outlet air below the ambient wet-bulb temperature with constant humidity. Theoretically, the outlet temperature can be decreased toward the dew point temperature of the ambient air. (Riangvilaikul &Kumar, 2009)

Riangvilaikul and Kumar proposed the equation of the relationship between out air temperature as:

$$t_2 = t_1 - r/c_{pm}(h_3-h_2)$$

In which r is the mass ratio of the working air to the intake air and c is the specific heat of the moist air and the enthalpy difference of the working air in the wet channel. While the second term refers to the temperature drop between the inlet and outlet air of the dry passage, it could be concluded that the increase of working air to intake air ratio can lower the outlet air temperature. However, this may lead to the reduction of air flow rate supplied to space.

Therefore, to achieve optimum operating condition, the outlet air temperature requires, outlet air flow rate, type of cooling load, inlet air condition, channel configuration and energy used for fan and water pump should be taken into consideration. (Riangvilaikul&Kumar, 2009)

4.6.2 Case Study : Riangvilaikul and Kumar

Riangvilaikul and Kumar have proposed a vertical configuration of the dew point evaporative cooling system with the use of countercurrent arrangement for the flowing fluids. Their design intention was to obtain a high heat and mass transfer of all streams both air and water, avoid corrosion. Therefore, water is used in the cooling process, saturate the wet channel surface evenly, and extract a certain fraction of the outlet air as the working air in the cooling process.

They set-up an experiment to evaluate the system in Bangkok, Thailand. The system set-up is composed of several dry and wet channels separated by polymer sheets to avoid infiltration of water and the water feeding system. Vertical configuration is implemented to ensure even distribution of water in the damp surface. It is regulated precisely by a control valve, and a fan is used in distributing the intake and working air. The wet and dry channel has a gap of 80mm in width and 1200mm in height. The walls are stacked with 5mm spacing to form a rectangular configuration of heat and mass exchanger with four dry channels and five wet channels. This would ensure that the walls of the dry channel are enclosed with the wet channels so that the intake air is sensibly cooled between the two parallel surfaces along the flow path.

Two types of experiments were conducted to investigate the performance of the dew-point evaporative cooling system. The static studies were conducted to evaluate the relationship between the influence of the inlet air temperature (ranging from 25C to 45C) and humidity ratio (ranging from 6.9 to 26.4 g/kg) towards the outlet air temperature. As well as the influence of air velocity on outlet air temperature, while the dynamic studies aimed to evaluate the performance of the outlet temperature under the ambient condition of an actual summer day in Thailand. The result of the static experiment has shown a constant linear correlation between the inlet and outlet air. As the inlet air with lower temperature would result in an outlet air with lower temperature. However, the result has shown a strong influence of humidity ratio towards the outlet air temperature; as the humidity ratio decreased by 10g/kg, the outlet air temperature would decrease by approximately 6-8C with an

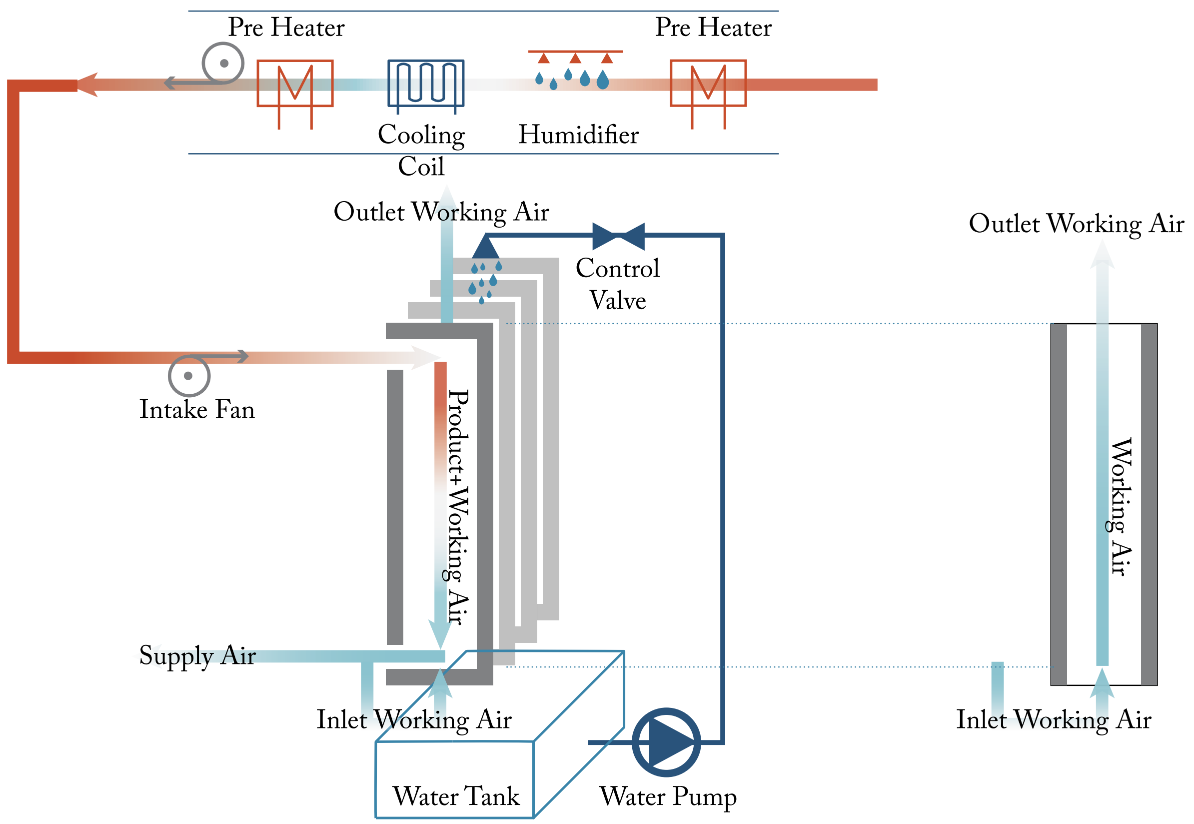


Figure 4-25 : Riangvilaikul and Kumar proposed system (Riangvilaikul&Kumar, 2009) [Redrawn by Author]

identical inlet air temperature. Similarly, when velocity is increased, the outlet temperature also increases. However, the system is still capable of achieving an outlet temperature below the wet bulb condition when the air velocity is below 2.5 m/s. The authors have summarized the result with an equation expressing the performance of the outlet air from the system:

$$t_{outlet}\,(^{\circ}C)=7.65+0.152t_{(in,ambient)}(^{\circ}C)+681w_{(in,ambient)}(kg/kg)$$

Furthermore, the experiment has presented a satisfying result in reducing the outlet air temperature below the inlet wet bulb condition. At the condition of inlet temperature between 30-45C the wet bulb effectiveness ranges from 100 to 115% while the dew point effectiveness varies between 63 to 85%. From the result, it could be concluded that the higher inlet temperature would lead to greater wet bulb effectiveness. The dynamic studies were conducted under the actual ambient condition of Bangkok, with the ambient temperature varying from 30.5C to 34C. The inlet humidity ratio ranged from 20.5 g/kg in the morning to 18 g/kg in the nighttime. However, the outlet temperature shows only a slight variation from 25.7 C to 27.8C approaching the ambient wet bulb temperature. The system has presented effectiveness ranged from 101 to 104% with the dew point effectiveness at 75 to 79%. The authors had concluded that the system would be able to perform on its own under the condition with an inlet temperature of below 45C and a humidity ratio of 11.2 g/kg. (Riangvilaikul&Kumar, 2009)

4.7 Case Study | Precedents

4.7.1 Case Study 1 : FREESCO

Solar invent had patented a compact solar air conditioning system designed for ventilation, dehumidification and cooling for residential and office sectors. With different type of configuration, the system could be applied on both a flat roof, sloped roof or façade integration.

Figure 4-26 : Prototype of the solar air conditioner by SolarInvent (Finocchiaro et al., 2014)

The system is composed of 2 fixed packed silica gel desiccant bed operating in batch process, and two wet evaporative heat exchangers connected in a series. The desiccant bed is designed like a fin and tube heat exchanger, as the spaces between the fins are filled with silica gel grains. Therefore, the silica gel could be cooled by means of a water loop in the heat exchanger which is connected to a heat sink. Furthermore, the desiccant bed also operates as latent energy storage. Due to the high adsorption capacity of the desiccant material, solar energy could accumulate when solar radiation is available. To ensure a continuous regeneration process, a system of air dumpers is integrated between the two-desiccant bed. Regeneration is carried out using solar air collector, while the cooling tower is also integrated into the system to be used in rejecting the adsorption heat generated during desiccant dehumidification. 40% of the airflow over the desiccant bed is delivered into the conditioned space while the cooling power could be controlled through various fan speed. (Finocchiaro et al., 2014)

4.7.1.1 System Principle

As the system operates, the ambient air is being drawn through one of the desiccant bed to be dehumidified and partially cooled. Due to the simultaneous moisture and heat exchange, the dehumidification process can be carried out at almost constant temperature. Then the dehumidified air is mixed with the return air from the building. The mixed air with a flow rate of 140% of the air flow rate enters the wet heat exchanger before being supply to the building.

On the other hand, 40% of the air from the second wet heat exchanger is being drawn to the secondary side. The water loop carries the heat rejected from the desiccant bed to the connected cooling tower. The air flowing through the cooling tower either be from the secondary side of the wet heat exchanger or the ambient air. However, if the air is taken from the heat exchanger, an additional fan is required.

4.7.1.2 System Size and Performance Version 1

The two desiccant bed used in the system have a dimension of 0.55m x 0.6m x 0.1m, each containing 15kg of silica gel grains. The flow rate of the system into the conditioned room is at 500 m3, producing a maximum cooling power at 2.2 kW at the ambient condition of 35C with 50% relative humidity and the room condition at 27C with 50% relative humidity. The cooling power delivered to the building is approximately 1 kW as calculated by the enthalpy difference between the return and supply air. Moreover,

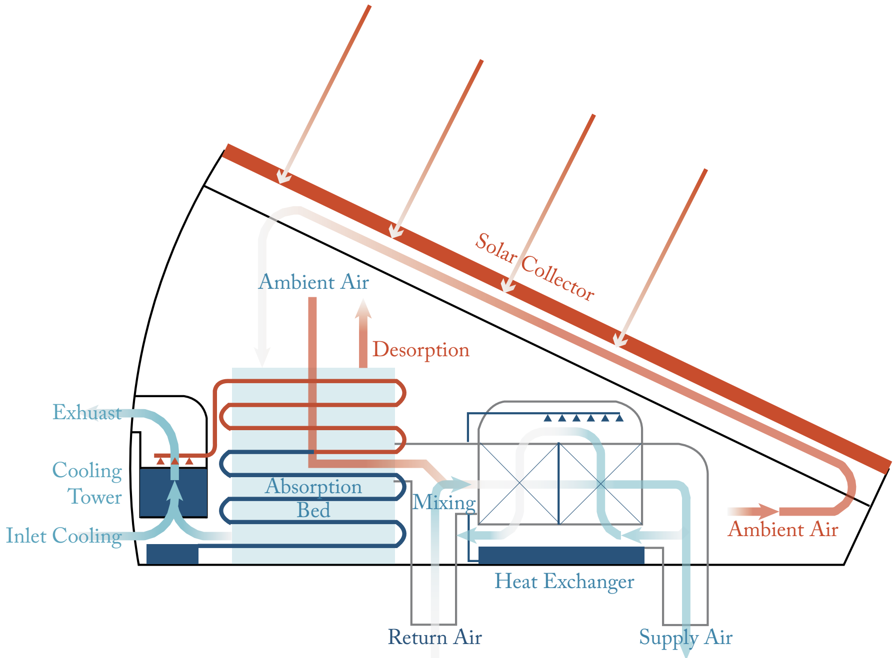


Figure 4-27 : Schematic diagram of solar air conditioner by Solar Invent (Finocchiaro et al., 2014) [Redrawn by Author]

the electric power requires approximately 200 W due to the operation of two DC variable speed fans and two DC pumps.

The experiment result had shown how two desiccant bed could work together to maintain a sufficient humidity ratio and temperature. When the desired humidity ratio cannot be reached, the control would communicate to the other bed, switching their roles. The result of the experiment had shown that the system is capable of supplying air at the average temperature of 22.5˚C even when the ambient temperature reaches 30˚C. When the wet heat exchanger pump is on, the temperature difference over the wet heat exchanger is approximately 5˚C; corresponding to the efficiency of the wet heat exchanger ranging from 45-48%. The efficiency is limited due to the flow rate mass ratio between the secondary and primary heat exchanger at 0.28. The regeneration temperature ranges between 40-60˚C, which is lower than what is used in a conventional desiccant rotary based system. The thermal COP is 1.21, where 38% of the cooling power is related to the enthalpy difference in the adsorption bed, while 62% is due to the wet heat exchanger. (Finocchiaro et al., 2014)

4.7.1.3 System Size and Performance Version2

Solarinvent has developed a second generation of the FREESCOO system to enable a stand-alone operation. Several alterations had been made including: integrating a PVT air collector instead of a solar air collector, and integrating batteries.

Figure 4-28 : Image of the prototype setted up at ENEA research center (left) and University of Parlemo (Right) (Finocchiaro et al., 2014)

Two sets of systems had been tested. A smaller system in ENEA research center and a larger system at the University of Palermo. Both room sizes are 46 sq.m; however, the room at ENEA research center has a volume of 135 m3 while the room in Palermo is at 190m3.

The system configuration at ENEA research center includes 13 kg silica gel grains, with a flow rate of 500m3/hr. and a maximum cooling temperature of 2.7kW. 2.4 m2 of PVT panels had been integrated, providing approximately 170 W for one module, and a battery of 65 Ah was also integrated. The room temperature was kept at 25C by the internal fan coils connected to the auxiliary cooling system. The result discussed by the authors summarized the energy production and the electricity consumption (without the electricity production of the PV) of the system. Out of the 15 testing days, the system was able to operate in a stand-alone mode for seven days, which reflects the added value due to the integration of the PVTs. In consideration efficiency of electricity usage, the system was reported to perform with 30.7 Energy Efficiency Ratio (EER).

On the other hand, the system at the University of Palermo is composed of 25 kg silica gel grans, a flow rate of 1000m3/hr, and maximum cooling power at 5.5kW, under the ambient condition of 35C with 50% relative humidity and the room condition at 27C with 50% relative humidity. 4.8m2 of PVT panels had been installed providing approximately 340 W, with a 100Ah. The room temperature was kept at 26C connected to the auxiliary cooling system. The result had shown the performance of the system in operation after 16.00. The ambient air could be dehumidified by 5-6 g/kg. Therefore, the supply humidity ratio lies within the range of 10g/kg and room humidity ratio of 11 g/kg even after sunset. The supply temperature to the building ranged between 19 C to 22C.

In conclusion, the system had shown great potential in using the desiccant material as latent heat storage, allowing the cool air to be supply even after sunset. Furthermore, the integration of the PVT modules the system has the potential of working as a stand-alone unit. (Finocchiaro et al., 2014)

4.7.2 Case Study 2 : BSRC - Thailand

While the desiccant air-conditioning system has rapidly gained its popularity as an energy saving technique, however, there is yet an actual system that is implemented in Thailand. Therefore, the Building Scientific Research Center (BSRC) had conducted an experimental investigation to evaluate the actual performance of desiccant air-conditioning under Thailand's context

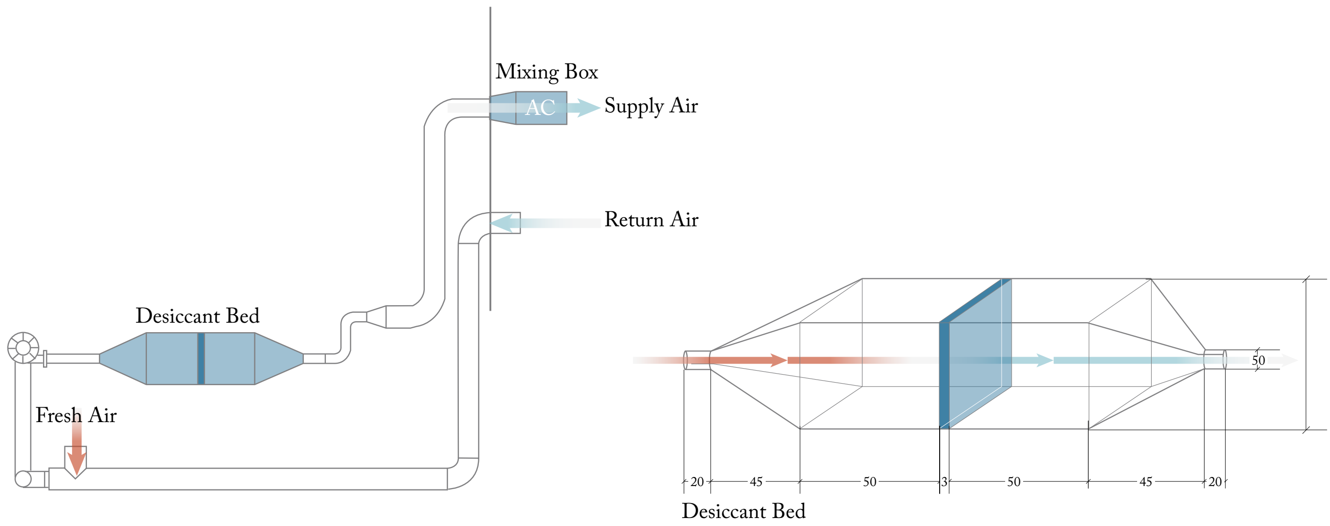


Figure 4-29 : Schematic Diagram of desiccant bed and air conditioning system by BRSC (Hirunlabh, J. et al., 2006)

4.7.2.1 System Configuration

The experimented room is located on the sixth floor of an 11 stories building. Only its North-façade is in contact with the ambient with a 1.5m. overhand shading device. The conditioned room has an area of 76.8m3 and the operating time of the experiment will be from 9.00am to 4.00pm.

The system set up is composed of two main components: 2 desiccant bed and a 1.5-ton split type air conditioning system. The desiccant beds have the dimension of 0.5m x 0.5m with 5 cm bed thickness. The desiccant beds contain 5kg and 11kg silica gel, respectively. The commercial silica gel has the diameter of 3-5mm diameter and is contained in a wire mesh box. While the bed casing was made of 3mm thick stainless steel. A 1 hp blower with an airflow regulator was integrated to blow air through the desiccant bed. As an integration strategy, the ducting system was designed to mix the return and fresh air prior to entering the desiccant bed. After the air leaves the desiccant bed, it turns into dry dir. Then the dry air is mixed with the indoor air in the mixing channel before it enters the air conditioner's evaporator, which provides the air flow rate of 1000m3/hr. (Hirunlabh, J. et al., 2006)

4.7.2.2 Experiment Result

At the beginning of the operation, it is observed that high-temperature air exits the desiccant bed. This is due to the adsorption process of moisture by the silica gel. Therefore, the authors recommended that a minimum of the half-hour is needed to reject this high temperature into the ambient. Since higher energy is consumed by the air-conditioning system if higher temperature air is treated. Furthermore, a

test on the thickness of the silica gel has been conducted. The result had shown that the pressure-drop is an insufficient trade-off when compared to the adsorption capability reduction when using a thinner bed. \
The result of various operation conditions such as ratios of indoor air, return and fresh air had been compiled. The author had concluded that the optimum operating ratio is at 70% indoor air mixed with 30% dry air, which is composed of 50% ambient air. The condition would provide 24% electricity saving when compared to a conventional air-conditioning system. Furthermore, in terms of economic analysis, it is recommended that the system would be a feasible alternative for large cooling capacity over 100tons. As the payback period is approximately four years, with an internal rate of return at 24.5% (Hirunlabh, J. et al., 2006)

4.7.2 Case Study 3 : Desiccant Coated Heat Exchanger

To resolved the unsynchronized relation between the humidity ratio and temperature due to the alteration between the dehumidification and regeneration process of a desiccant-coated heat exchanger (DCHE), Ge et al. have proposed a configuration in which desiccant coated heat exchanger, and sensible heat exchanger is operating in a series. Ideally, the outlet process air with low temperature and low humidity ratio are expected, so higher cooling power could be produced. (Ge et al., 2017)

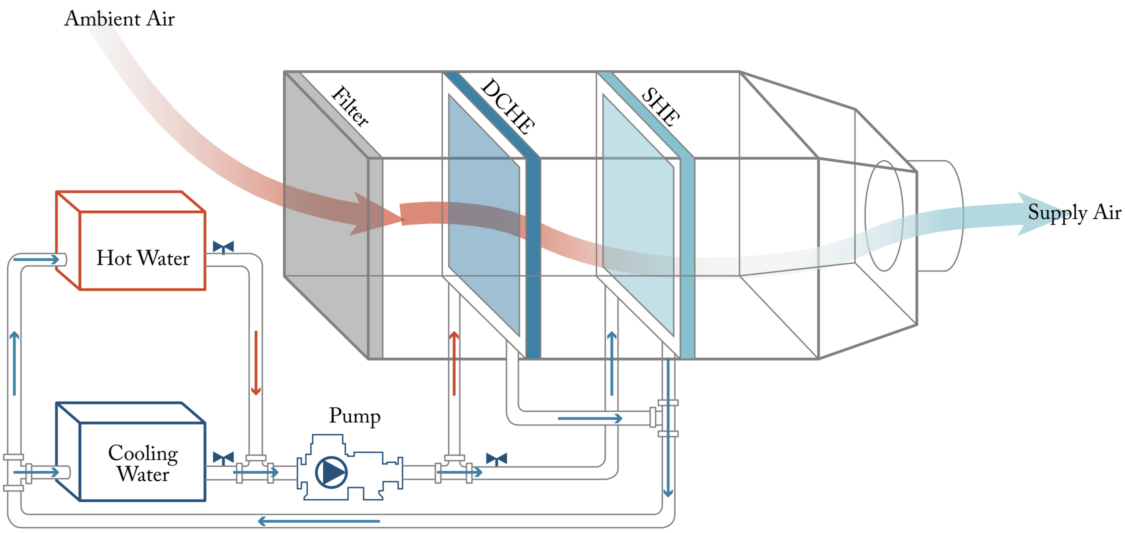


Figure 4-30 : Schematic Diagram of the experiment set up by Ge et al. (Ge et al., 2017) [Redrawn by Author]

4.7.2.1 Experiment Set up

To investigate the performance of the system, an experiment had been conducted. The system is composed of two main components: an air tunnel and a water loop.

The stainless-steel sheet air tunnel has a cross section of 230mm x 200mm with the length of 2.5m. An air filter is installed in the inlet channel, while a DCHE and a SHE are placed with a spacing of 500mm is integrated inside the wind tunnel. With a frequency conversion fan at the end of the tunnel, the maximum airflow rate controlled is at 30W / 360m3/hr. Two desiccant materials had been tested, a conventional silica gel and a composite desiccant based on silica gel and lithium chloride.

The water loop consists of two thermostatic water baths operating in parallel to produce hot and cold water for thermal and cooling source. Two water valves are integrated to enable the switching between hot and cold water. The volume of the water bath is 30 L, with a corresponding cooling and heating of 2.7 kW and 3kW respectively.

The ambient conditions are simulated in a 3m (L) x 3m (W) x 2.45m (H) experimental chamber with constant temperature and humidity ranges from -10C to 40C and 30% to 90% relative humidity ratio. (Ge et al., 2017)

4.7.2.2 Experiment Result

The result of the experiment could be categorized into the dynamic outlet state and cooling power performance. Ge et al concluded that the experimental results agree with the theoretical analysis. While it takes approximately 400 seconds for both the system without a sensible heat exchanger to obtain a constant temperature, only 200 seconds is required for the other two systems operating in series with the heat exchanger. In terms of outlet temperature, the conventional desiccant coated heat exchangers can produce outlet temperature at 27C while in the systems integrated with the sensible heat exchanger the temperature remains stable at 24C. While SHE is not needed in regeneration, results had shown that both variable temperature rate and final outlet temperature also increased in the process. It could be validated that by integrating the SHE, the system obtained an improved temperature variable rate.

Since the SHE doesn't have a role in humidity ratio change, the humidity still decreases rapidly to the minimum before it gradually increases to reach inlet condition in dehumidification mode. In contradiction, the humidity would reach to the maximum before it subsides. In comparison between the Composite coated heat exchanger (CCHE) and Silica gel coated heat exchanger (SGCHE), the result shows that CCHE can obtain a higher dehumidification capacity due to the added chemical adsorption.

With additional parameters, the authors had elaborated that, despite the increasing inlet air temperature, a lower outlet air temperature can be obtained by reducing the temperature of the cooling water. The result had shown that when the ambient air temperature had been increased to 34C, and cooling water has been decreased to 15C the outlet temperature remains constant at 24C in the system with sensible heat and 28C in the systems without sensible heat respectively.

In the cooling performance aspect, the result had shown that the integration of SHE increases the COP and the cooling power of the system. In SGCHE the cooling power increased from 0.45kW to 0.8kW while its CP rises from 0.75 to 0.85. Furthermore, it could also be concluded that 4C of lower outlet temperature can be obtained. Additionally, to enhanced the efficiency of the system, lower temperature cooling water, and lower air velocity should be implemented. (Ge et al., 2017)

4.8 Conclusion

Due to high level of humidity, temperature, and solar radiation of Bangkok; heat prevention strategies, dehumidification and heat dissipation technologies had been discussed in the previous chapter. A comparative overview and potentials of the strategies and technologies had been presented. However, through the parameters of performance, ease of maintenance, system complexity, sustainability, system size and configurations, certain technologies and strategies had presented themselves as a more suitable option for a façade system in Bangkok's context.

Heat Prevention

As recommended by Prieto et al. in their studies, window-to-wall ratio strategy should be prioritized in a hot and humid climate. As smaller glazed are shows higher cooling demand savings. Furthermore, the authors also commented that the used of shading and glazing type strategy could impose an adverse effect. Therefore, only the window-to-wall ratio and shading strategy would be taken into further consideration. Due to the high solar incidence angle in the north and south façade, equator-facing offices eventually have a higher cooling potential with the implementation of shading strategy. Despite its high cooling demand reduction potential, further consideration regarding its orientation should be further developed. Factors such as horizontal sun angle (HSA) and vertical sun angle (VSA) should be analyzed while designing the façade for optimal performance.

Dehumidification

Due to high humidity, the conventional vapor compression system requires higher cooling energy to be dehumidified and decrease the sensible load. Therefore, the desiccant cooling system is a more viable alternative as they do not require refrigerants and consume less energy. Amongst the desiccant technology, the desiccant-coated heat exchanger (DCHE) system out-perform the others, as it was developed to deal with the limitation of the conventional desiccant packages. The experiment by Ge et al. had shown the potential of connecting DCHE with SHE into a series to produce outlet air with low temperature and low humidity efficiently.

Regeneration

Despite the fact that solar collector served as a source which provides higher temperature for regeneration, it generates more than what is required. On the contrary, the PVTs system might not be able to generate such high level of heat; however, according to Rawat and Dhiran with the current technologies, the PVTs system is able to achieve a daily electrical efficiency of 7.54% and a thermal efficiency of 70% respectively. The thermal energy production is sufficient for regenerating the desiccant system, while the electricity generated could be further used by auxiliary components of the façade system. Furthermore, the PVT panels could be integrated as a shading strategy since it required exposure to direct solar radiation.

Heat Dissipation

Despite being highly developed and being used commonly worldwide, the vapor compression systems are one of the primary contributors to global warming. This is due to is low process efficiency since it needs to handle both latent and sensible load simultaneously. Additionally, due to the used of mechanical compressors, high electrical energy is required. With recent technologies developments, the indirect evaporative cooling system becomes a more favorable option in tackling carbon emission and energy consumption issues. With the M-Cycle configuration, studies by Pandelidis et al. shows that it has the potential to reduce the energy consumption up to 90% in comparison to the conventional system.

The research can be summarized by the following diagram in Figure 84. This depicts the possible configuration in integrating the systems as a façade module. As the system had been disassembled, several components could be shared or combined as a centralized system. The other components could be reassembled and fitted in a façade module with consideration to the window to wall ratio, while components such as the PVT panels will be integrated as a part of the shading strategy. Additional configuration possibilities and technical details will be further elaborated in the following chapters.

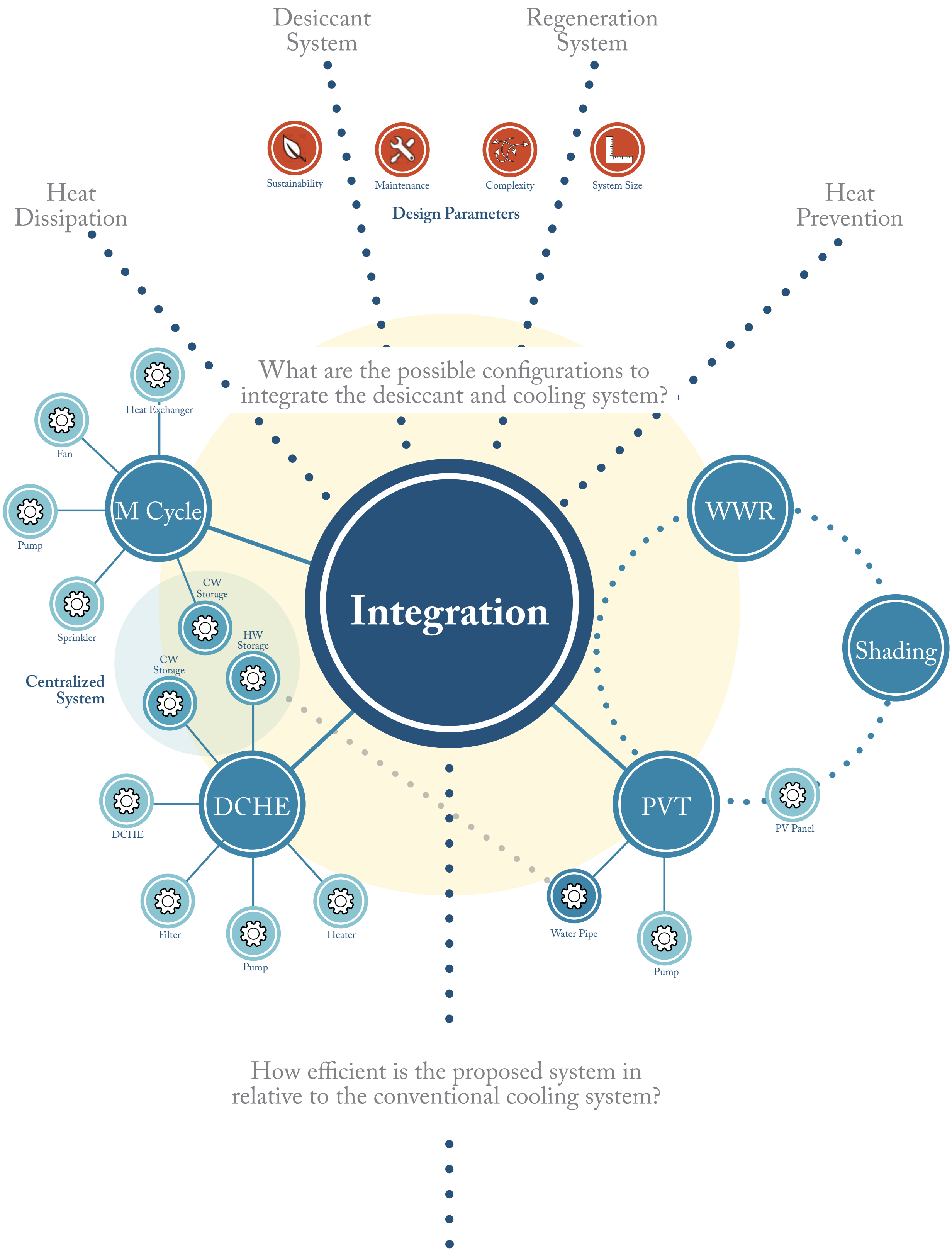


Figure 4-31 : System components and possible configuration

Part II

Reassembly | Design Concept

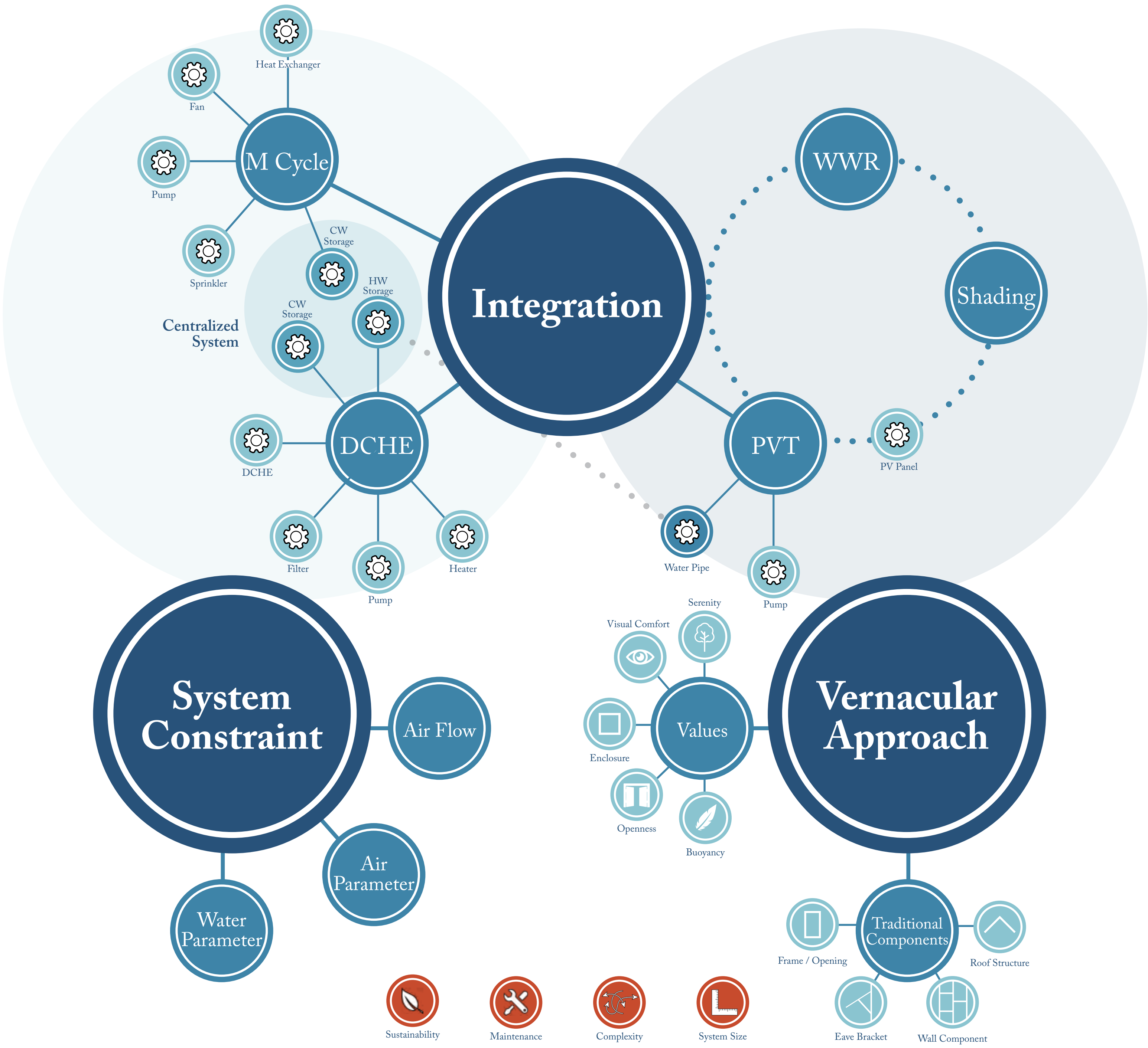


Figure 5-1 : Design concept development strategy

5 Design Concept

'Desigrated' aims to integrate heat prevention strategies with low-ex cooling technologies, namely the desiccant and m-cycle evaporative cooling technology, in the form of a façade system for high rise office buildings. The project targets to provide an alternative cooling solution for a hot and humid climate context of Bangkok. To achieve its goals, the project's design approach is branched into two aspects: the vernacular approach and the constraints of the system.

As discussed earlier, the fully glazed buildings dominating the skylines of metropolitan cities around the world fail to adapt both physically and culturally. (Wood, 2015) Therefore, the vernacular approach is implemented as one of the main strategies in developing the façade. The characteristics of traditional Thai architectures would be discussed both in terms of the implied values and iconic components. While the potentials in implementing them on the façade of high rises building would also be investigated in this chapter.

At the same time, in terms of the systems' performances, the results from experiments by various researchers and literature are used as assumptions in developing the system. Being one of the prominent dehumidification technologies, the Composite Silica Gel Heat Exchanger (CCHE) will be implemented as a primary part of the façade system. While the M-cycle technology would also be implemented as a secondary cooling technique to cool down the supply air. The constraints derived from the data will then be used in the preliminary sizing and system configuration.

5.1 Venecular Approach

Figure 5-2 : Traditional Thai house on pilotis within a natural setting.
ZionStars. "Thailand Houses" ZionStar,https://zionstar.net/thailand-houses/

Figure 5-3 : Multi-tier roof of a temple.
Webphra. "Thai Temples" Webphra, https://www.web-pra.com/auction/show/8283836

Figure 5-4 : 'Chaarn' or large balcony connecting a group of houses.
Chulalongkorn University. "Chulalongkorn Thai Building" Chulalongkorn University, http://www.prm.chula.ac.th/cen046.html

5.1.1 Characteristic of Thai Architecture

Traditional Thai architecture accurately depicts the vernacular design approach, which successfully integrates the influence of culture, lifestyle, and religious belief with the natural context. These aspects are implied in the form of abstracted values, aesthetic values, and symbolic values which reflected the 'Thainess' characteristic. To provide an overview of the traditional Thai architecture approach, these characteristics can be generally characterized with the following physical aspects reflecting the attempt to achieve the abstracted quality such as buoyancy, openness, enclosure, visual comfort, and serenity. (Boonjub, 2009; Horayangkura, 2017)

Physical Appearance / Site Planning

Traditional Thai houses are usually situated amongst a natural setting with various trees and a water body, allowing the users to dwell in a shaded and serene environment. The houses are designed and built with a Trilogical Order, consisting of 3 main parts: a ground level, a primary level with trapezoidal form due to a slightly leaned in walls sitting on stilts, and a curve gable structure with extended eaves. Small columns or stilts are being used as an attempt to create a levitating effect portraying the values of lightness. Other strategies such as imposing concave curvature shaped roof and multi-layered roof as seen in temples also depicts the attempts to tone-down the bulky appearance of the large roof structure.

Zoning

Traditional Thai houses are designed with a collective setting consisting of small units which are connected by a large balcony or 'Chaarn'. The strategy allows the smaller units to be well ventilated due to a large amount of open space. While the distribution of the units enables the houses to be built with a smaller structure and spaces for further extension. Through this strategy, the building cluster achieves a sense of openness, while space is being enclosed by the orientation of the small units.

Materiality and Modularity

A standard traditional Thai house would consist of a single level raised above the ground on stilts. The structural system is made up of columns and beams with an odd number of spans (either 3 or 5). While other components such as walls are designed as modules allowing the house to be taken down and reassemble in a new site.

Ventilation

In addition to the shades from trees and extended roof eaves, traditional Thai houses are designed with an effective ventilation system through the following strategy:

- Designed in a collective manner with small units and a communal open space
- Raised above ground with ventilatable floor openings between the steps from the balcony to the main building
- Large windows and openings
- Operatable walls openings or 'Fa Lai' allowing the users to control the amount of ventilation
- Ventilation grill above the window
- Well ventilated areas under the roof

Natural lighting

Lighting hierarchy between the exterior and interior is an effort to create visual comfort for the users, gradually dimming the amount of light into the interior space. The lighting atmosphere help shaped the zoning hierarchy gradually creating a transition between a serene private space to a well-lit communal area.

Figure 5-5 : Chulapat 13 Building

Yamakun. "Chulapat 13 Building" Pantip, https://pantip.com/topic/34619849

5.1.2 Strategy to Apply Thai Characteristic in Contemporary Context

Like many vernacular design approaches, the knowledge of Thai architecture is being passed down from generation to generation. Therefore, it reflects upon the tradition and cultural values of the past. (Chareonsupakul, 1998) However, it is undeniable that the elements which hold an iconic value of the past is being ingested by the fast-changing world and globalization. Hence, these elements are being depicted as a universal emblem which defines 'Thainess' rather than portraying its specific values. (Panin, 2000). Therefore, to keep up with the situation, the core of the architectural concept should be maintained, while its element could be adapted to the modern context. Horayangoor (1996) suggested that the revival or integration of traditional Thai architecture could be developed through the following approach.

Extensive Revival
The approach aims to completely revive the traditional method, developing a design which incorporates the true essence of Thai traditional architecture.

Implementing a Specific Component or Characteristic of Traditional Architecture
Applying a certain specific traditional architecture element onto the design to emphasize the iconic value of Thai architecture.

Adaptation of Abstracted Characteristic
Adjustments on forms, shapes, or details could be made or mixed with the newly developed design, but still, reflect on the core idea behind the element. This strategy was incorporated by the design of Chulapat 13 building in Bangkok. The building consists of several elements which reflect the essence of Thai architecture, such as the open ground floor, extensive roof eaves, being well shaded, and an attempt to break down the bulky mass of the building.

5.1.3 Integration Strategy

It is evident that traditional Thai architecture is mainly developed in the form of low rises or medium rises; making its immediate application for high rises and skyscraper limited. However, as mention in the previous chapter, several approaches had been suggested throughout the years as the means to cope with the fast-changing trend, offering further potential in adapting Thai architecture into a modern context.

Therefore, the project aims to revive and reintroduce the essence of Thai architecture back into Bangkok's skyline. The design strategy in developing the desiccant cooling system aims to adopt the vernacular strategy to tackle the hot and humid tropical climate while reflecting on the core characteristics of Thai architecture. Both the abstracted values and physical components, which resemble the so-called universal emblem would be deconstructed and adapted as an integral part of the façade system. The potential in integrating elements such as the extended roof eaves, eaves brackets, window frames, and opening, as well as the Pakon wall would be elaborated in this chapter.

Figure 5-6 : Extended Roof of a traditional Thai house.
Thai Dhupp. "Thai House" Teakdoor, http://teakdoor.com/building-in-thailand-famous-threads/174969-thai-dhupp-princess-joys-thai-house-51.html#post3786156

5.1.3.1 Roof Structure

Extensive roof eave is one of the primary components that is being thought of when thinking of Thai architecture. Due to the vast amount of solar radiation and rain in Thailand, extensive roof eave had been implemented as a shading strategy to prevent excessive heat and weather penetration. Furthermore, the iconic multi-tier roofs design in temples can be implemented as a method to 'lighten' a bulky or monolithic structure. (Boonjub, 2009)Therefore, the extensive roof eave has the potential to be integrated as a façade component to reduce the excessive heat gain from direct solar radiation, while the multi-tier approach could be incorporated to tone down a bulky appearance.

Figure 5-7 : The different style of eave brackets.

Sookjai. "KanTuay" Sookjai, http://www.sookjai.com/index.php?action=printpage;topic=199218.0
BazaarPlanet. "Eavesbracket" BazaarPlanet, http://www.bazaarplanet.com/south_asia/42_bangkok_thailand.html

5.1.3.2 Eave Bracket

To tackle the abundant rain of the tropical climate, Thai houses are designed with lower roof eave. Therefore, the extended eave requires additional support from eaves brackets or columns. Eave brackets have been used to enhance the quality of lightness, allowing the roof to extend out with a relatively slender structure. (Boonjub, 2009)

Figure 5-8 : Rectangular window frame.

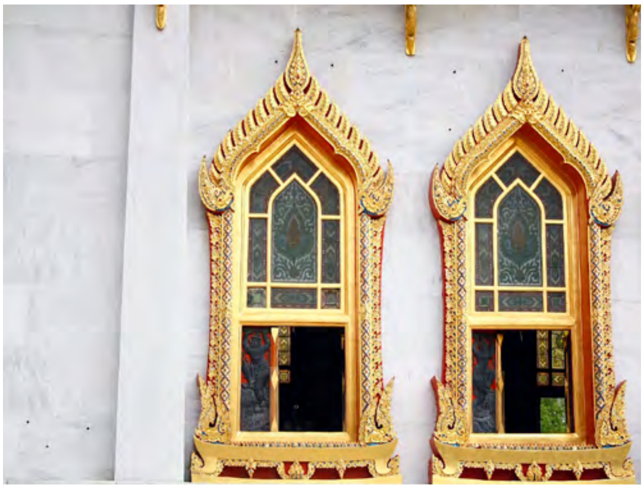


Figure 5-9 : Decorated window frame of a temple.

Thai Wood House. "Thai House Components" Thai Wood House, http://thai-wood-house.blogspot.com/2016_07_03_archive.html
Enfaye. "Temple Opening" Enfaye Blogspot, http://enfaye.blogspot.com/2012/03/professional-practice-assignment-3.html

5.1.3.3 Window Frame and Openings

One of the main ventilation strategies in Thai architecture is to create a large opening allowing fresh air to ventilate directly into the interior space. To emphasize the openings, windows and doors are incorporated with ornamented frames. Ornaments on window and door frames express a hierarchical value in which buildings with higher hierarchy such as temples and palaces would be decorated with a higher amount of ornamentations.

Figure 5-10 : 'Pakon' pattern of a house wall.
Goldenteakwood. "Pakon" Goldenteakwood, http://goldenteakwood.blogspot.com/2015/02/14.html

5.1.3.2 Wall components | 'Fa Pakon'

One of the characteristics of traditional Thai houses is its modularity. Wall panels are designed as modules. They are pre-assembled and could be fixed between the columns on site. One of the most common patterns of the modular wall is the rectangular grid pattern known as the "Pakon" wall pattern. The pattern could be considered one of the universal emblem which represents Thai architecture, as it is one of the most common patterned used in traditional houses.

5.2 System Constraints

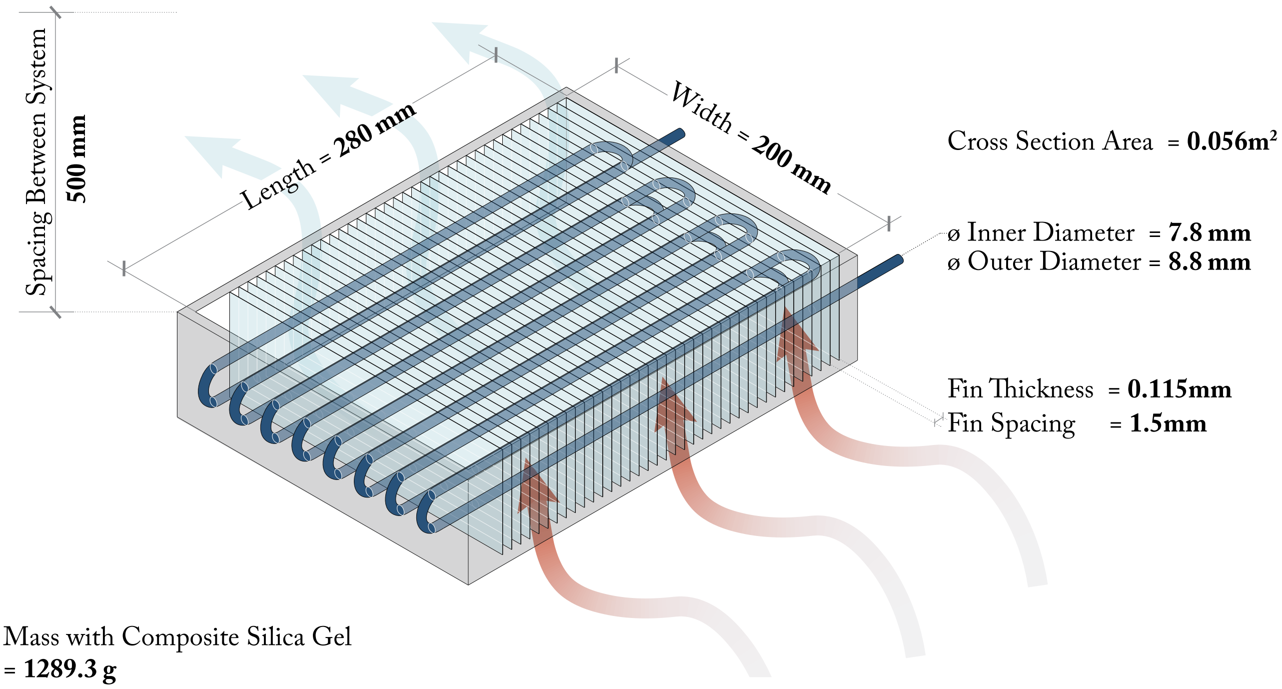


Figure 5-11 : Schematic diagram of a Composite silica gel heat exchanger.

1.25
kW
Heat Transfer Value

3.9
g kg
Humidity Difference

Table 5-1 : CCHE System Constraints

Water Parameter			Air Parameter			
Cooling Water (˚C)	Heating Water (˚C)	Water Volume (L/h)	Inlet Air Temperature (˚C)	Relative Humidity (%)	Flow Velocity (m/s)	Cycle Switch Period (mins)
15	**50**	**200**	**30**	**70**	**1.54**	**10**

(Ge et al, 2017)

5.2.1 CCHE System Constraints

The Composite Silica Gel Heat Exchanger (CCHE) works as a primary component of the system to dehumidify the ambient air condition; reducing the latent load the cooling system needs to process. The CCHE configuration used in this project is based on an experiment model by Ge et al., 2017 depicted in Figure 5-11. The system constraints [Table 5-1] are referenced to the results based on the inlet air condition at 30˚C|70% RH, which is closest to Bangkok's average ambient temperature of 28˚C|70%RH. Based on this assumption, the system is expected to perform under a cooling capacity of 1.25 kW and a dehumidification rate of 3.9 g kg.

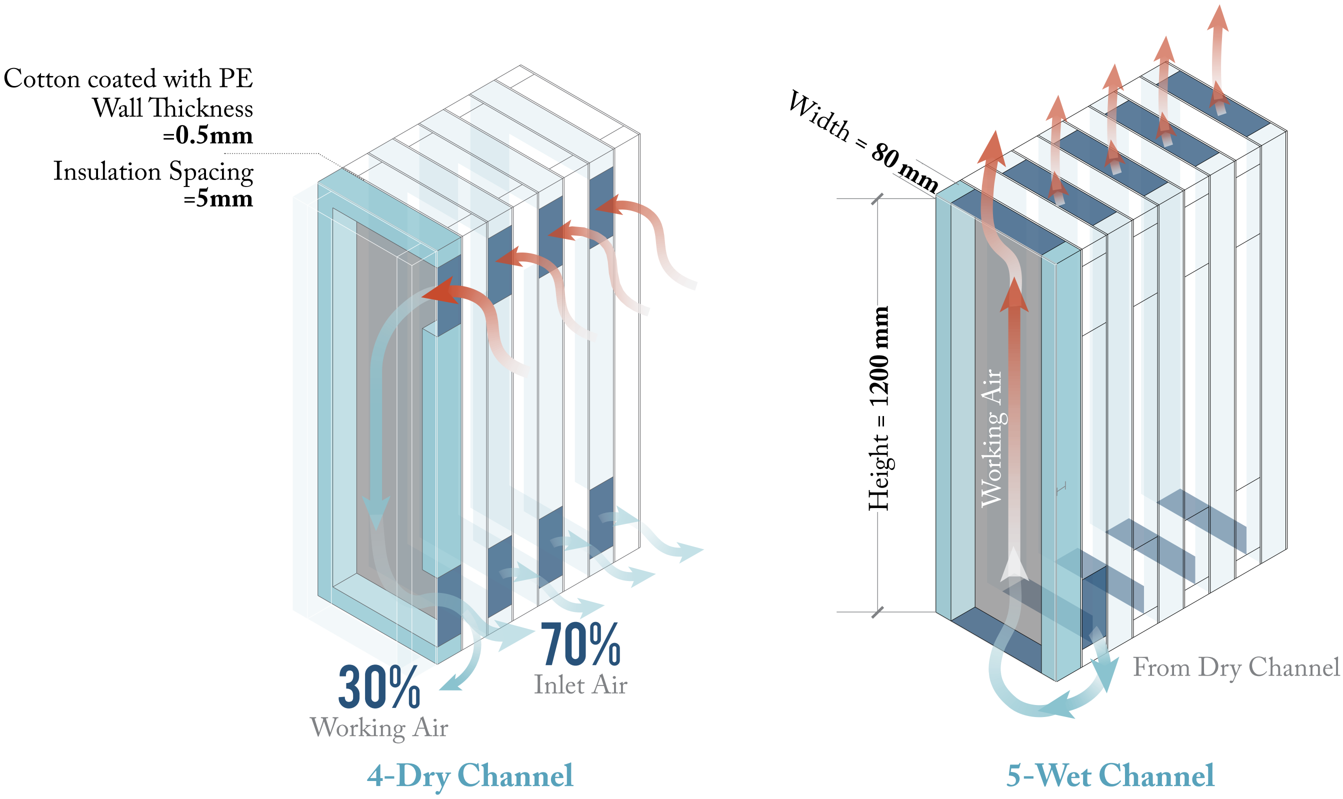


Figure 5-12 : Schematic diagram of a M-Cycle evaporative cooling system.

101-104% Wet Bulb Effectiveness

75-79% Dew-point Effectiveness

Table 5-2 Dewpoint Indirect Evaporative System Constraints

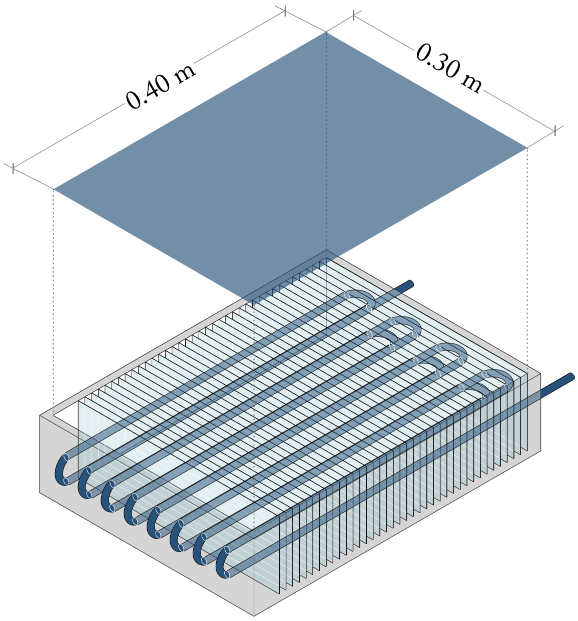
Water Parameter	Air Parameter	
Water Volume (g/h)	**Flow Velocity** (m/s)	**Working Air** (kg/kg)
60	**<2.4**	**0.33**

(Riangvilaikul&Kumar, 2009)

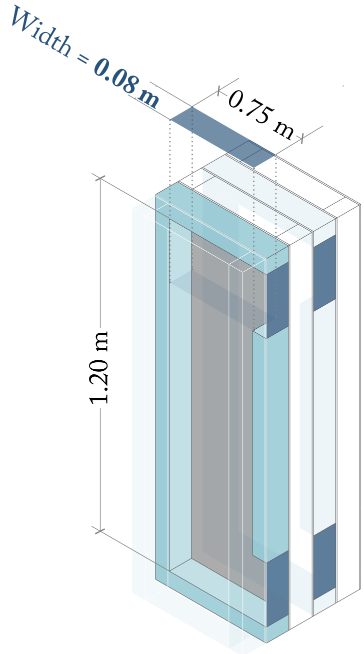
5.2.2 Dewpoint Indirect Evaporative System Constraints

The M-Cycle system served as a secondary element to cool down the air after it is dehumidified by the CCHEs. The dry air would enter through the dry channel counterflowing the water flow in the wet channel. The configuration would allow high heat and mass transfer for both the streams of air and water. However, it also requires 30% of the outlet air to serve as the working air in the cooling process. The system's schematic design and its performance are based on a dynamic study conducted under the actual ambient condition of Bangkok by Riangvilaikul&Kumar (2009). [Figure 5-12] According to the literature, the system can provide wet bulb effectiveness of 101-104% and a dew point effectiveness of 75-79%.

5.3 System Sizing



Cross Section Area = **0.12 m²**



Cross Section Area = **0.06 m²**

No. of Dry Channel = **2**

Total Area = **0.12 m²**

Figure 5-13 : System cross-section area.

Table 5-3 : Mass Flow Rate

$m = \rho v A$

0.22 kg/s Mass Flow Rate		ρ Air Density (kg/m³)	v Air Velocity (m/s)	A Cross Section Area (m²)
	CCHE	**1.17**	**1.54**	**0.12**
	M-Cycle	**1.17**	**1.54**	**0.12**

Table 5-4 : Volume Flow Rate

$Q = v A$

0.18 m³/s Volume Flow Rate		v Air Velocity (m/s)	A Cross Section Area (m²)
	CCHE	**1.54**	**0.12**
	M-Cycle	**1.54**	**0.12**

5.3.1 Air Flow Rate

The first step in designing the façade system is to define the system's size, specifically its cross-section. The cross-section plays a crucial role in determining the mass rate and volume flow rate of the inlet and outlet air. The preliminary assumption is based on a similar cross-section between both the system with an area of $0.12 m^2$, to achieve a constant air flow as it is being processed. With $0.12 m^2$ cross-section area, the system can provide a constant mass flow rate of 0.22 kg/s and a volume flow rate of $0.18 m^3/s$.

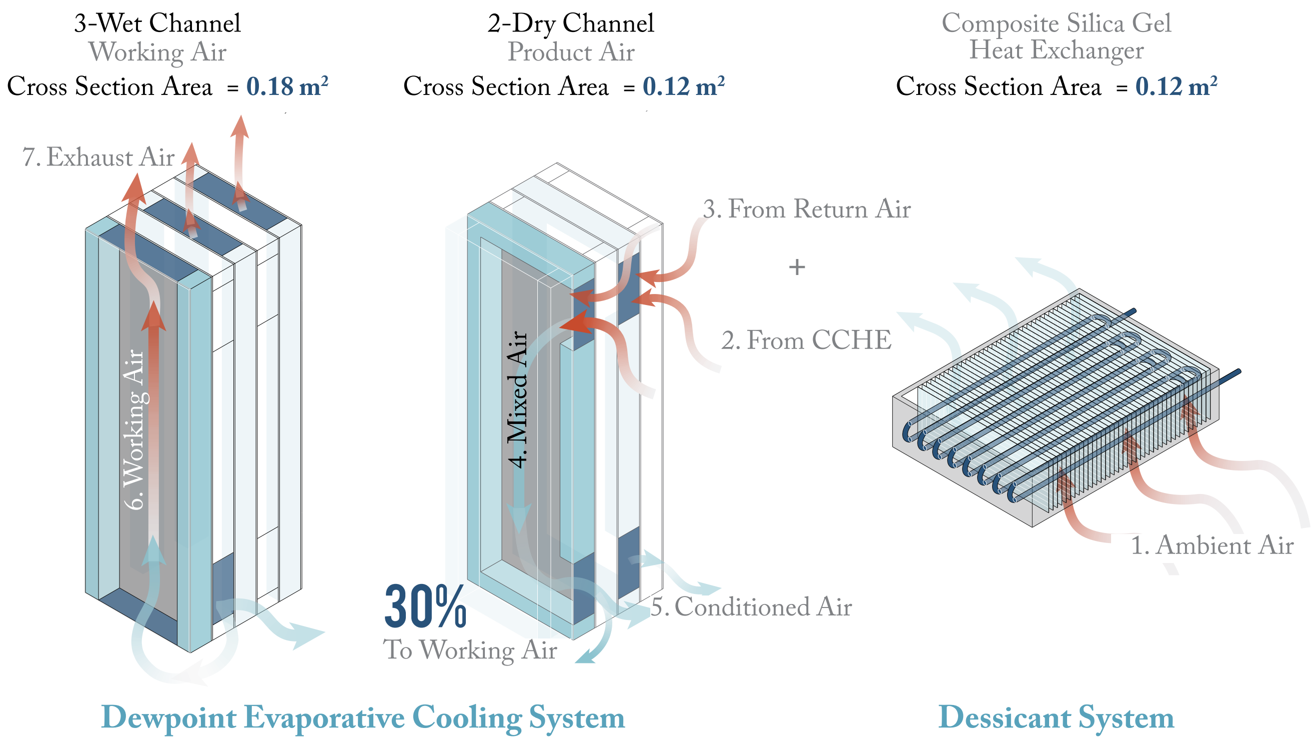


Figure 5-14 : Schematic configuration for fresh air ratio.

Table 5-5 : Fresh Air Ratio

	100% Fresh Air		70 : 30		30 : 70		30 : 70*	
	Volume Flowrate (m^3/s)	Air Velocity (m/s)	Volume Flowrate (m^3/s)	Air Velocity (m/s)	Volume Flowrate (m^3/s)	Air Velocity (m/s)	Volume Flowrate (m^3/s)	Air Velocity (m/s)
2. CCHE	0.18	1.54	0.18	1.54	0.09	1.08	0.18	1.54
3. Return Air	-	-	0.08	0.67	0.20	1.67	0.42	1.67
4. Mixed Air	0.18	1.54	0.26	2.16	0.29	2.40	0.60	2.40
5. Conditioned Air	0.13	0.97	0.18	1.54	0.20	1.08	0.42	1.68
6. Working Air	0.05	0.27	0.08	0.44	0.09	0.5	0.18	0.48
7. Exhaust Air	0.05	0.27	0.08	0.44	0.09	0.5	0.18	0.48

*M-Cycle Cross Section:
= **0.25m²**

5.3.2 Fresh Air Ratio

Once the system's volume flow rate could be defined, it is used in evaluating the fresh air intake ratio. Due to the 30% working air requirement, 4 schemes are proposed in Table 5-5. The first 3 schemes considered the 0.12m^3 cross-section area as its constraints. On the other hand, the 4th scheme has taken the air velocity of each system as its constraints. Based on the results, it could be concluded that the 4th scheme is able to provide a higher volume of supply air at 0.42m^3/s, under the condition that the M-cycle system cross-section needs to be increased to 0.25m^2.

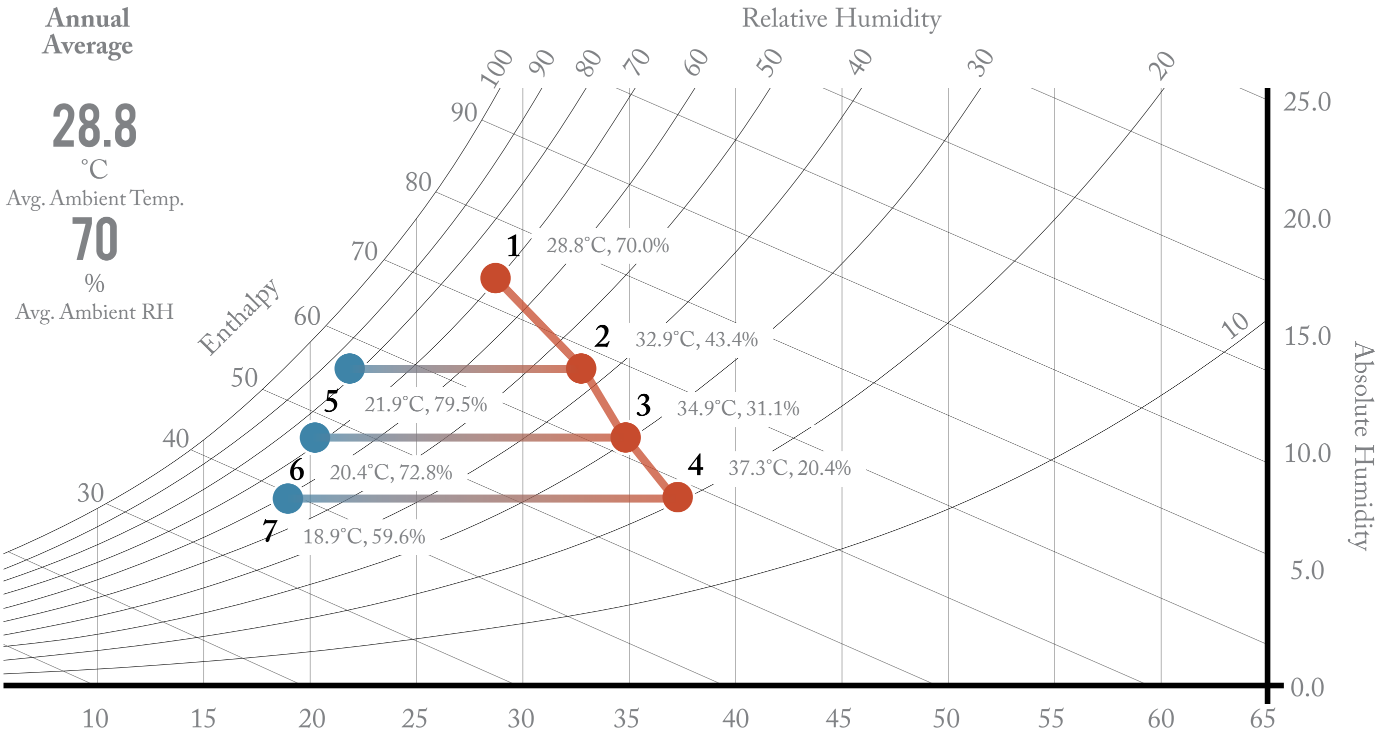


Figure 5-15 : Psychometric chart

Table 5-6 : CCHE Performance

$Q = M\,(Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{out}) (kJ kg)	Abs. Humidity (g kg)
2	28.8	70.0	73.7	17.5	**32.9**	43.4	67.9	13.6
3	32.9	43.4	67.9	13.6	**34.9**	31.1	63.1	10.6
4	34.9	31.1	63.1	10.9	**37.3**	20.4	58.3	8.1

Table 5-7 : M-Cycle Performance

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air				Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Absolute Humidity (w) (g kg)	Wetbulb Temperature (°C)	Air Temperature (°C)	Relative Humidity (%)	Absolute Humidity (g kg)	Wetbulb Efficiency (%)
5	32.9	43.4	13.6	23.0	**21.9**	82.5	13.6	111
6	34.9	31.1	9.7	21.7	**20.4**	72.8	9.7	110
7	37.3	20.4	8.1	20.4	**18.9**	59.6	8.1	109

5.3.3 Air Flow Rate

To illustrate the performance of the system, a series of calculations has been conducted and plotted on the psychometric chart [Figure 5-15]. The equation used in defining the CCHE's performance is extracted from the literature by Ge et al., while the equation expressing the performance of the M-cycle is referenced to Riangvilaikul&Kumar's experiment. The inlet air condition is based on Bangkok's average ambient temperature at 28.8°C|70%RH. To thoroughly evaluate the performance of the systems, the air condition is being calculated at every procedure. It could be observed that, while the desiccant could dehumidify the air by 3.9 g kg, the relative humidity increases drastically after the M-cycle system processed it due to the immense changes in temperature. Therefore, the system would require 3 CCHEs to provide the supply air condition with relative humidity below 55%.

5.4 System Configuration

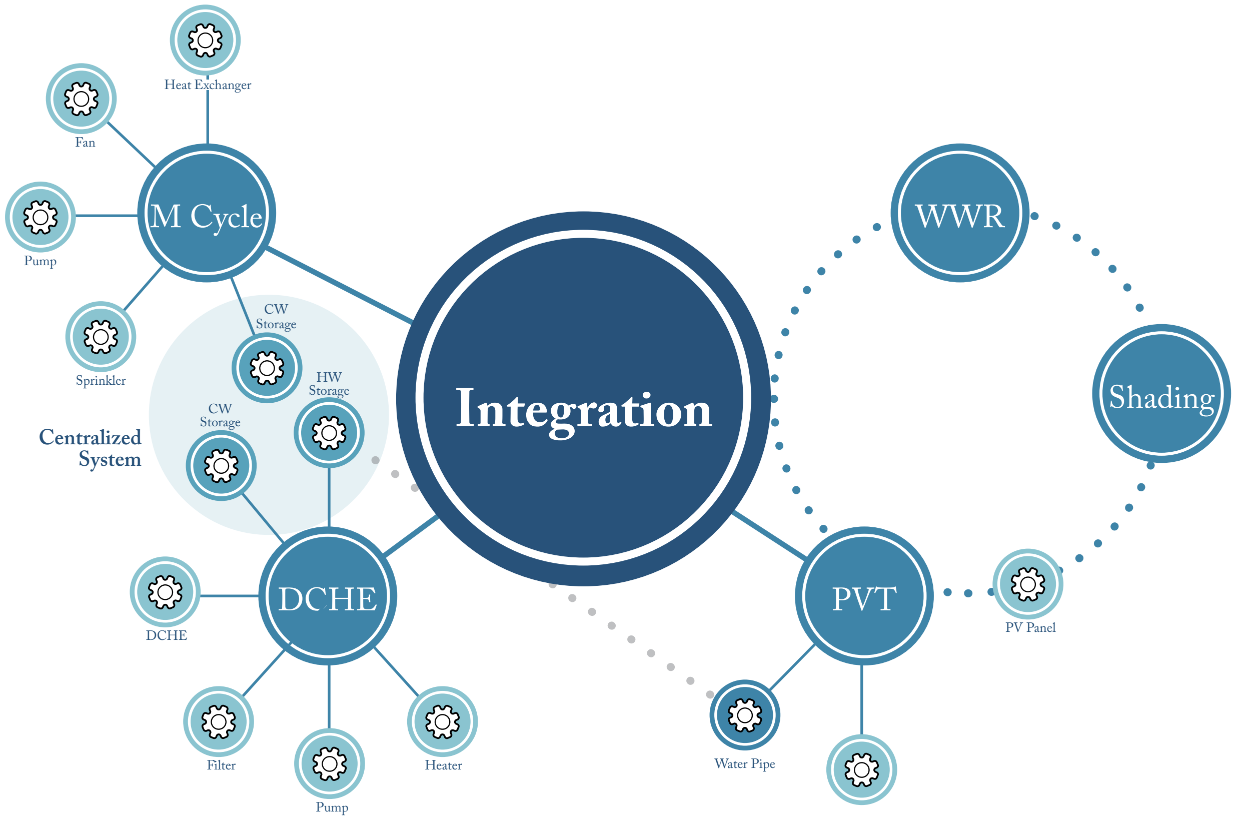


Figure 5-16 : System configuration flowchart

5.4.1 Design Approach

The main objective of 'Desi-grated' is to develop a façade system which is suitable for a hot and humid climate of Bangkok to reduce the cooling demands of office buildings while reflecting on the cultural aspects and identities. To achieve such goals, the design of the façade is approached with the 'Adaptation and Abstracted Characteristic' strategy. Iconic Thai architectural components and shading strategy mention in the previous chapter, such as the extended roof, eave brackets, and window frame will be abstracted and implemented as a part of the façade design. While the essence of buoyancy and lightness should also be preserved. At the same time, the façade will be designed and configured with the constraints imposed by the CCHE and M-cycle components.

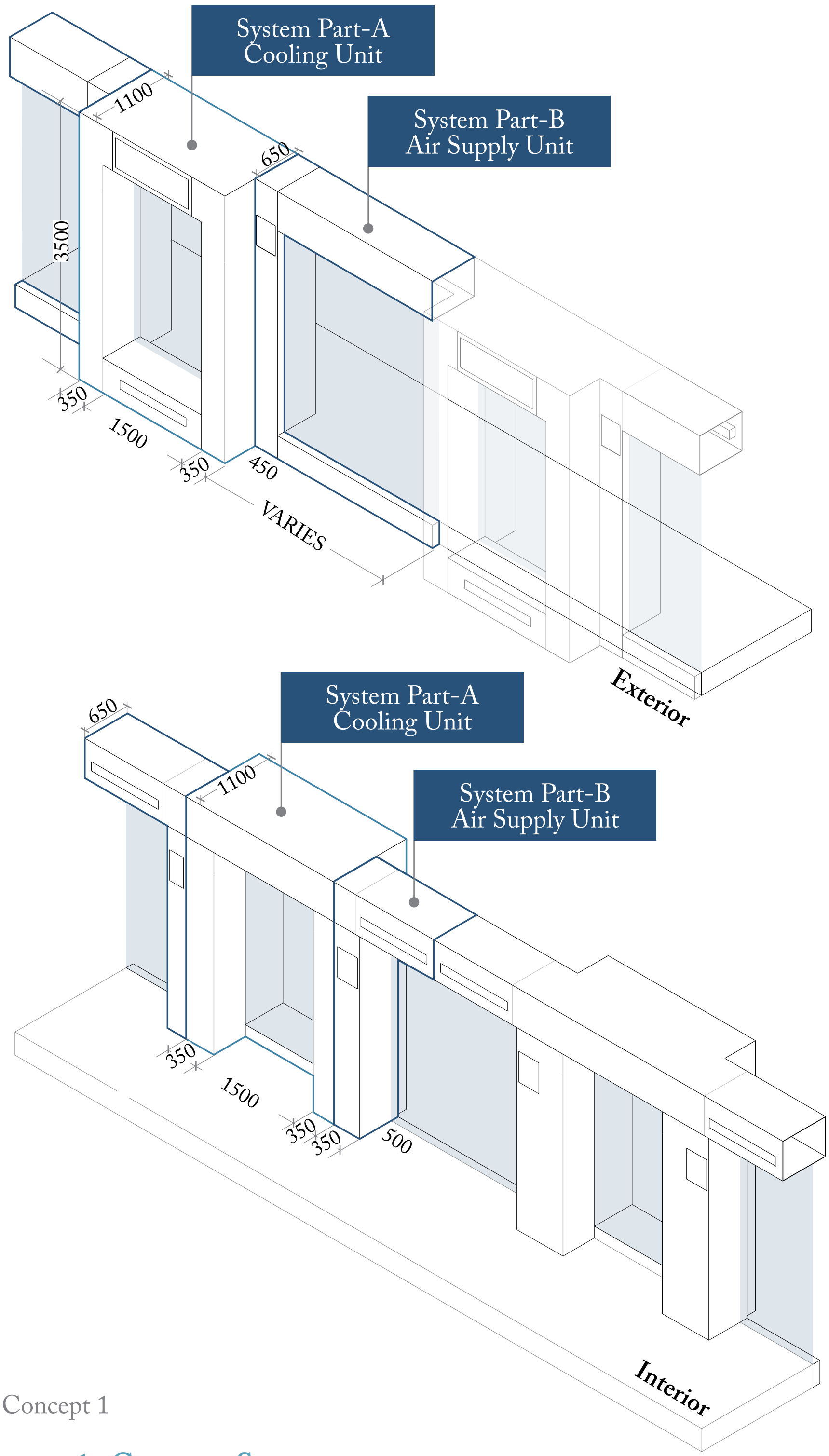


Figure 5-17 : Design Concept 1

5.4.2 Design Concept 1 : Compact System

The first design concept is developed as a stand-alone, compact system. The system consists of 2 parts, a cooling unit, and an air supply unit. The cooling unit consists of an air intake unit on the bottom, two sets of 3-CCHEs units running parallelly on the sides, and an M-cycle system on top. The components form a frame-like structure wrapping around the window opening. The outdoor air is taken in through the air inlet at the base of the frame, it is then being dehumidified by the series of CCHEs before being cooled down by the M-Cycle cooling system and sent to the air supply unit. The air supply unit rests on top of the adjacent window next to the cooling unit, allowing the conditioned air to be supply from the ceiling level. Furthermore, the return air duct is also incorporated as a part of the supply air unit running on the side of the window opening, forming an interior frame.

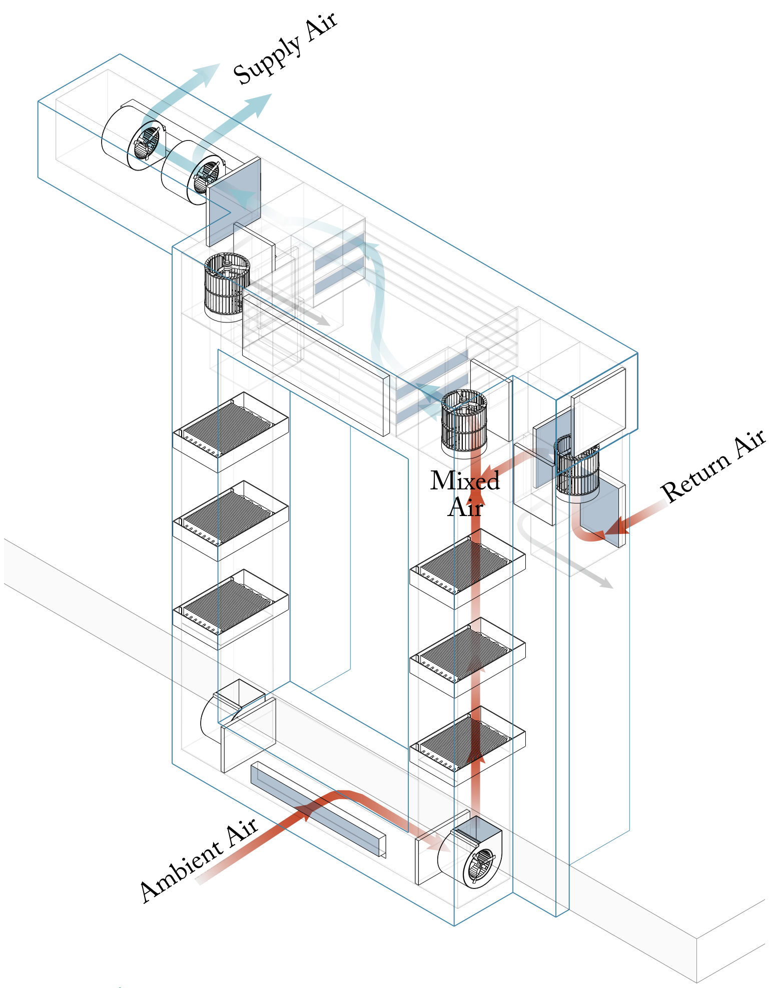


Figure 5-18
Supply Air
Schematic Diagram

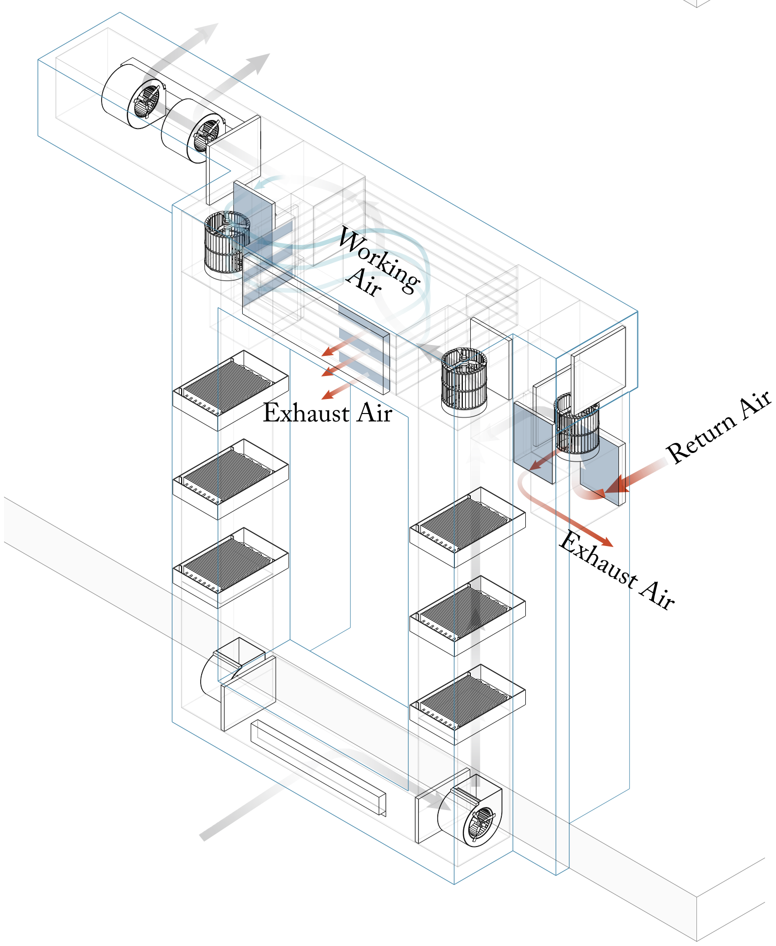


Figure 5-19
Exhaust Air
Schematic Diagram

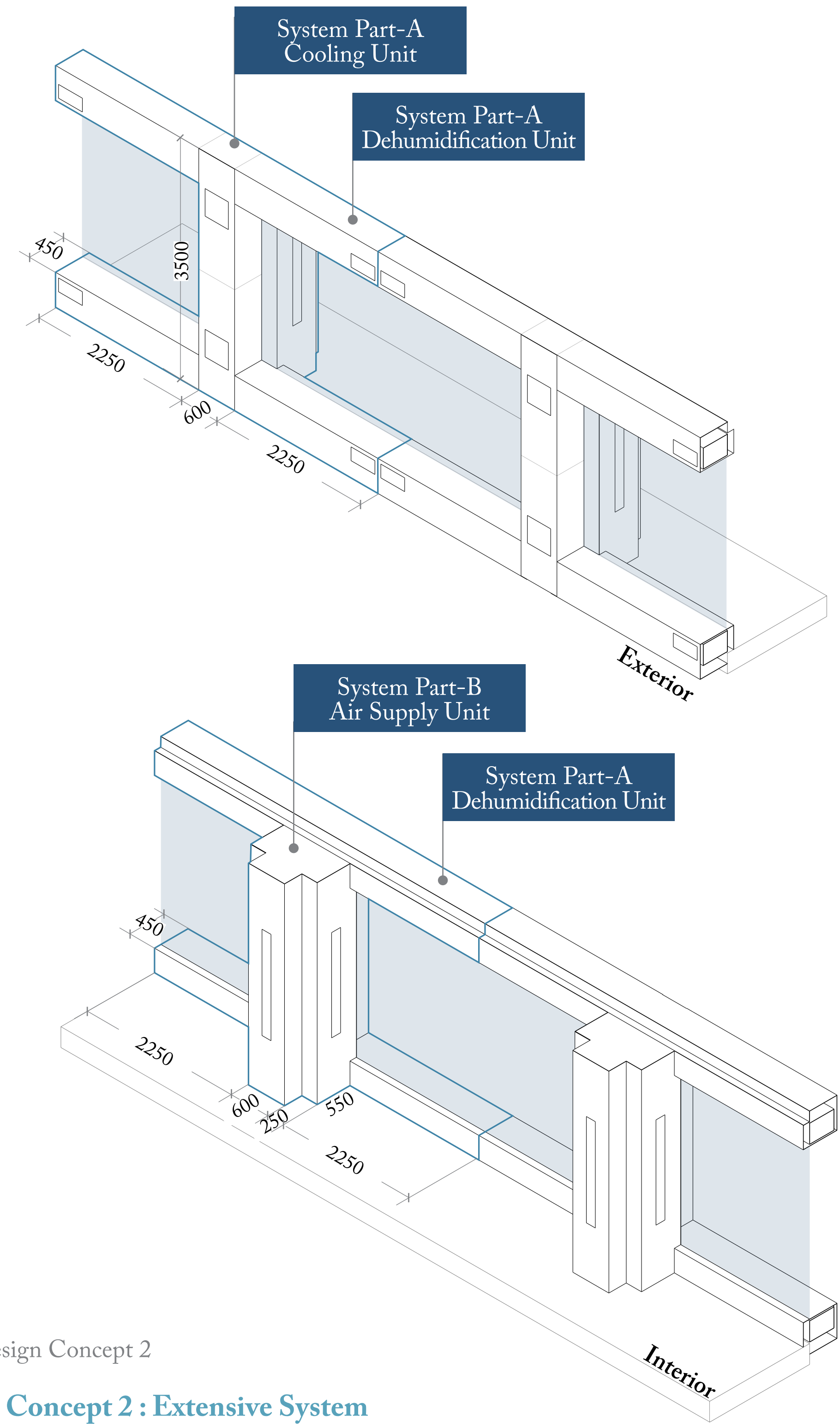


Figure 5-20 : Design Concept 2

5.4.3 Design Concept 2 : Extensive System

The second design concept is developed to provide higher supply air volume, reducing the numbers of systems require. The system is composed of 2 main parts, the dehumidification unit, and the supply air unit. The dehumidification unit consists of a 3 CCHEs system aligned horizontally on the top and the bottom of the floor slab. It is connected to the supply air unit which is positioned vertically at the end of the dehumidification unit. The supply air unit is composed of a vertically positioned m-cycle system which is connected directly to the supply inlet at body level. The configuration allows two sets of dehumidification unit to be coupled to the supply air unit simultaneously providing a higher supply air flow rate.

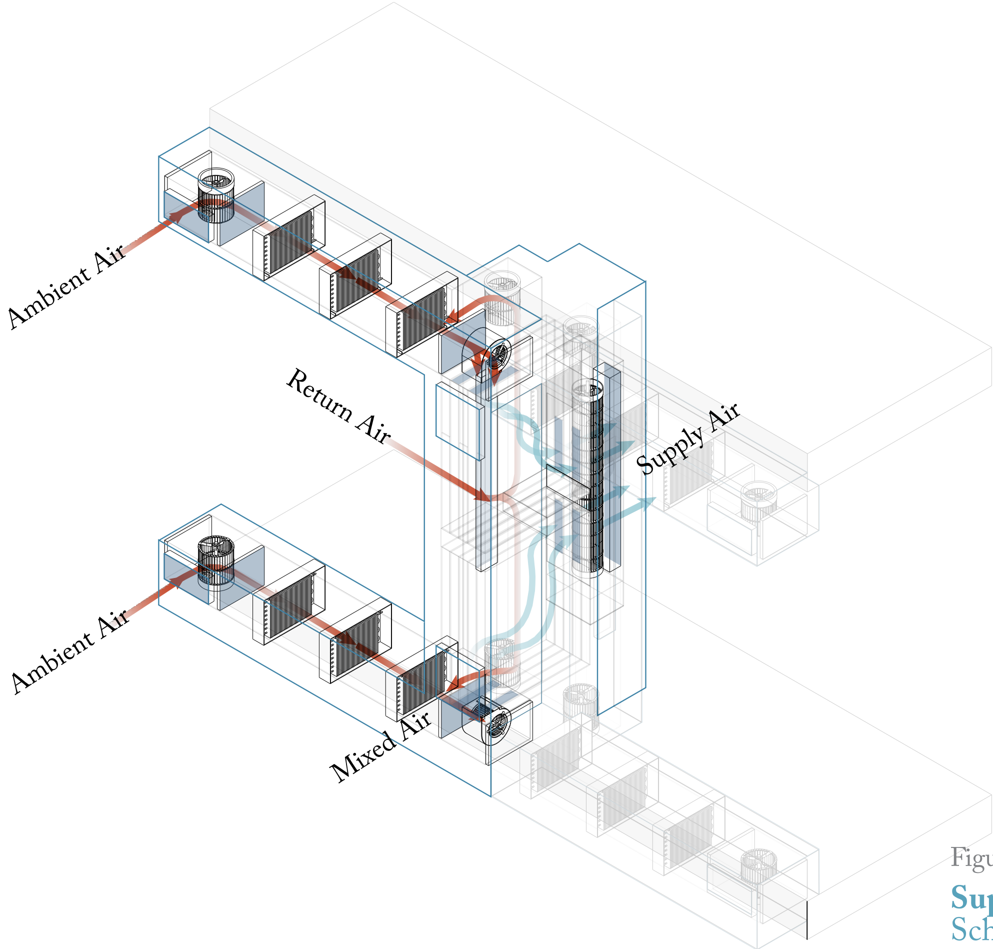


Figure 5-21
Supply Air
Schematic Diagram

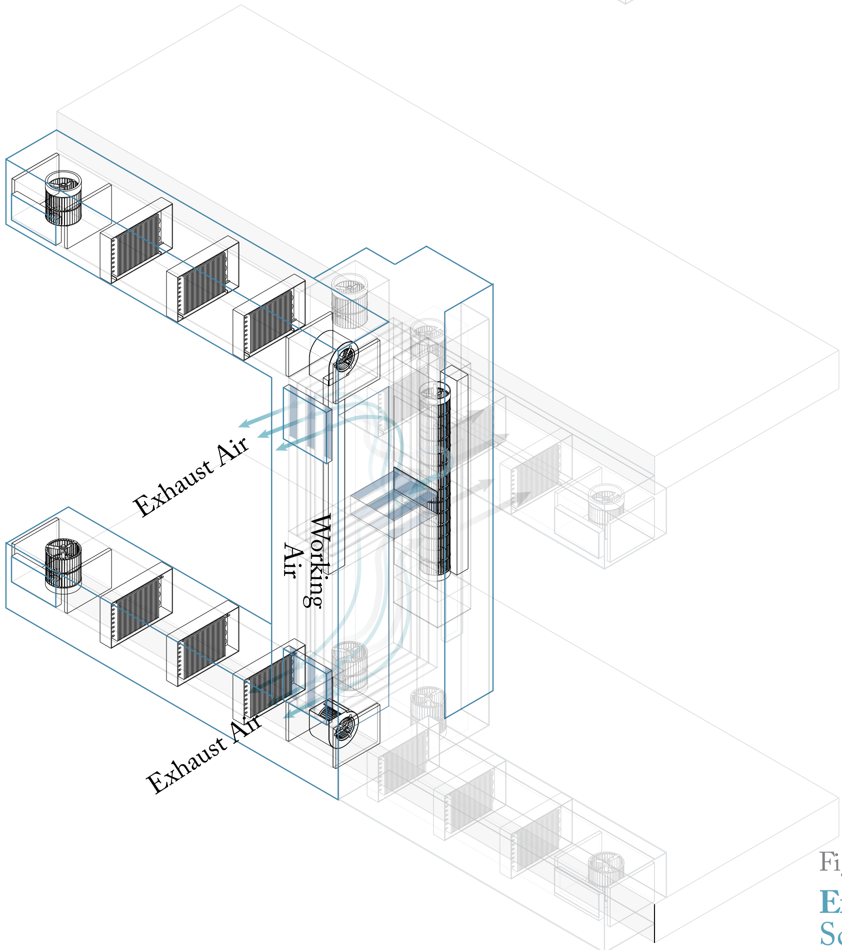


Figure 5-22
Exhaust Air
Schematic Diagram

5.4.4 Design Concept Assessment

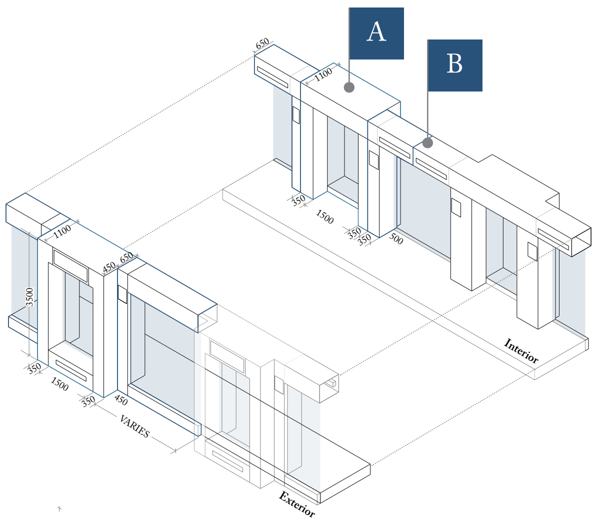


Figure 5-23 : Design Concept 1

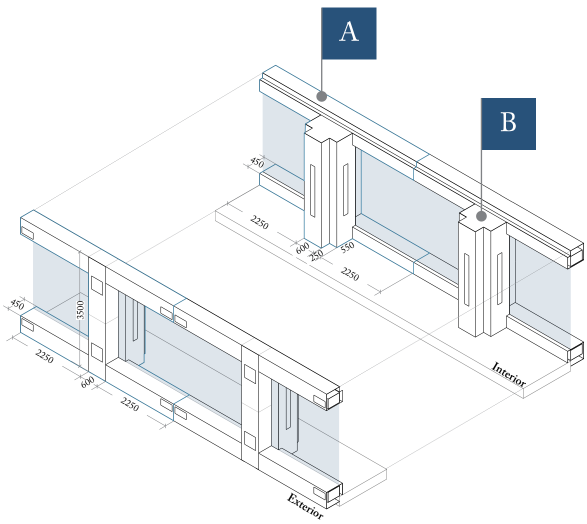


Figure 5-24 : Design Concept 2

Table 5-8 : Design Concept 1 Summary

Performance Aspect

Maxium Air Supply Capacity (m^3/s)	Maximum Return Air Capacity (m^3/s)	Maximum Fresh Air Capacity (m^3/s)
0.42	0.18	0.18

Application Aspect

	Exterior Dimension WxLxH (mm)	Interior Dimension WxLxH (mm)	Air Supply Position
A	450 x 2200 x 3500	500 x 2200 x 3500	-
B	150 x 2200 x 3500	500 x 1800 x 3500	At Ceiling Level

Window to Wall Aspect

	Facade Area Occupied (m^2)	Window Area	Wall Area	Window | Wall Percentage
1	7.7 m^2	3.8 m^2	3.9 m^2	49%
2	6.45 m^2	3.15 m^2	3.30 m^2	49%

Table 5-9 : Design Concept 2 Summary

Maxium Air Supply Capacity (m^3/s)	Maximum Return Air Capacity (m^3/s)	Maximum Fresh Air Capacity (m^3/s)
0.84	0.36	0.36

Application Aspect

	Exterior Dimension WxLxH (mm)	Interior Dimension WxLxH (mm)	Air Supply Position
A	450 x 5100 x 3500	-	-
B	-	840 x 1100 x 3500	At Body Level

Window to Wall Aspect

Facade Area Occupied (m^2)	Window Area	Wall Area	Window | Wall Percentage
21.0 m^2	12 m^2	9 m^2	57%

*Based on 2 units

Since each of the concepts possessed their pros and cons, an assessment has been conducted to summarize and compare the characteristics and potentials of the proposed concepts. The assessment is conducted under three main aspects: performance, application, and window to wall ratio aspect. Based on the summary, it is evident that concept 2 can supply higher conditioned air and fresh air. However, it also requires a larger area. When the systems are connected, it will occupy up to 21m2. Furthermore, due to its configuration, concept 2 creates a large window opening, with a window to wall ratio percentage of up to 57%. While concept 1 couldn't provide as much supply air as concept 2, it requires lesser space and has a relatively lower window to wall ratio. Moreover, the simple frame-liked appearance and modularity, along with the possibility of duct connection due to its air supply position; Concept 1 provides a broader application possibility. Owing to its compactness and versatility, concept 1 will be used as a base for further design development.

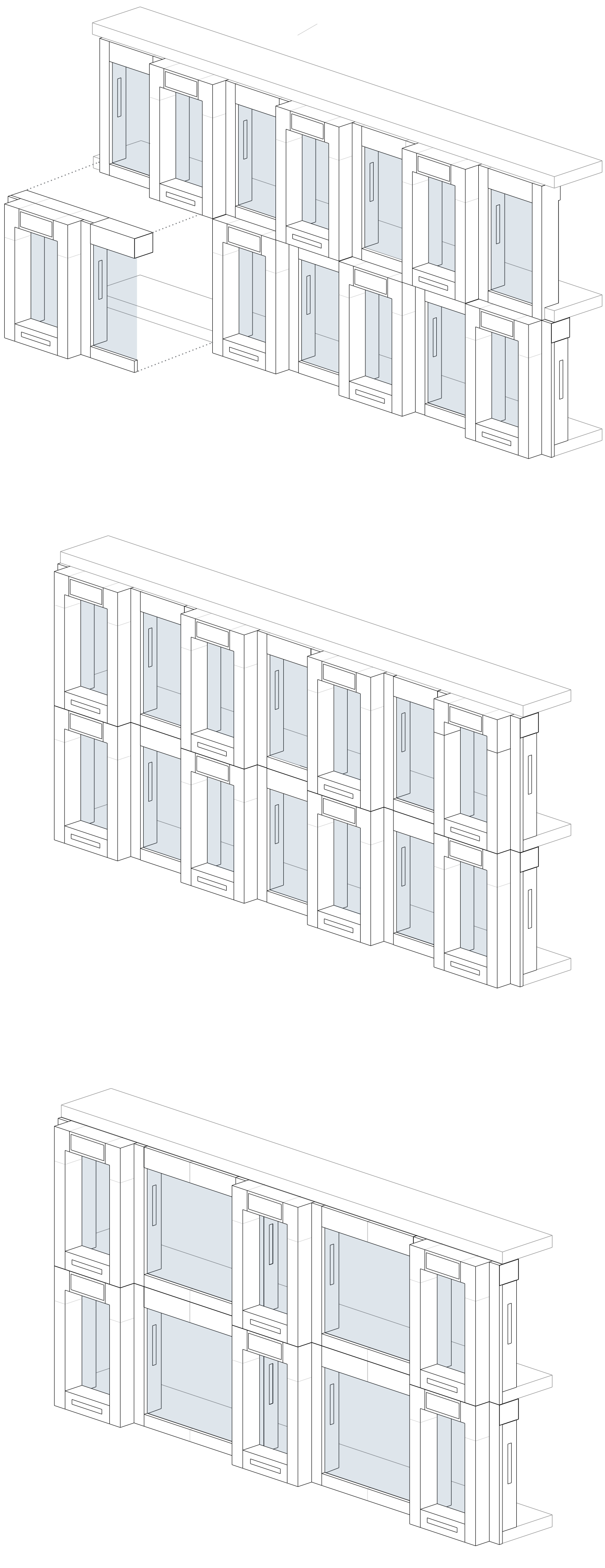


Figure 5-25 : Design options with Concept 1

5.5 System Requirement

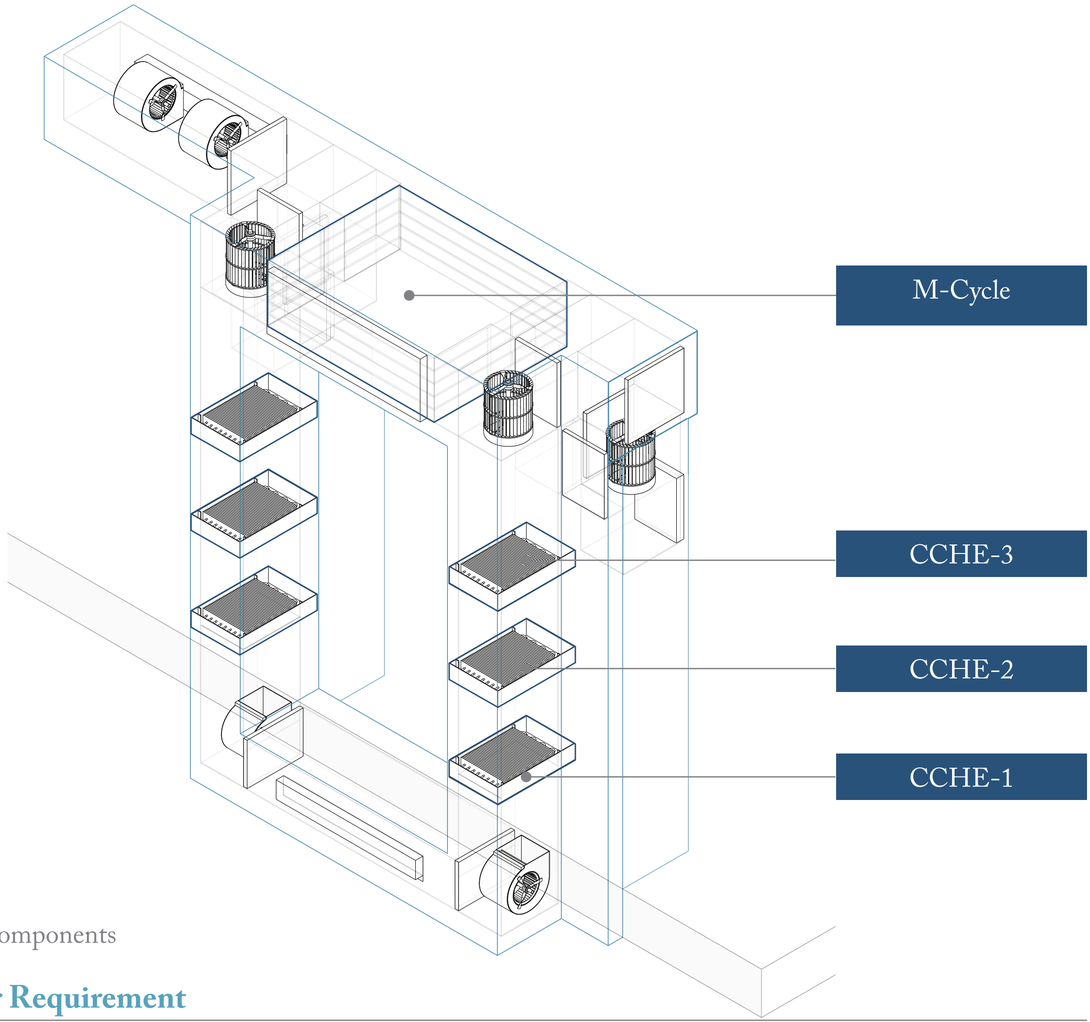


Figure 5-26 : System components

Table 5-10 : Water Requirement

	Cold Water (L/hr) Riangvilaikul&Kumar (2009)			Hot Water (L/hr)		
M-Cycle	0.06			-		
	Case 1 Jiang et al., 2014	Case 2 Ge et al., 2017	Case 3 Ge et al., 2017	Case 1 Jiang et al., 2014	Case 2 Ge et al., 2017	Case 3 Ge et al., 2017
CCHE-1	200	155	60	200	155	60
CCHE-2	200	155	60	200	155	60
CCHE-3	200	155	60	200	155	60
	600.06	**465.06**	**180.06**	**600**	**465**	**180**
	$0.6 m^3/hr$	$0.47 m^3/hr$	$0.18 m^3/hr$	$0.6 m^3/hr$	$0.47 m^3/hr$	$0.18 m^3/hr$

Table 5-11 : Heating Requirement

$$Q = V' C_p (\Delta T)$$

	V Volume Flowrate (m^3/hr)	V' Volume Flowrate (m^3/day)(11 Hours)	ΔT Temperature Difference (°C)	Q Energy Required (kWh/day)
Case 1	0.60	6.60	5.7	**43.6**
Case 2	0.47	5.17	5.7	**33.8**
Case 3	0.18	1.98	5.7	**13.1**

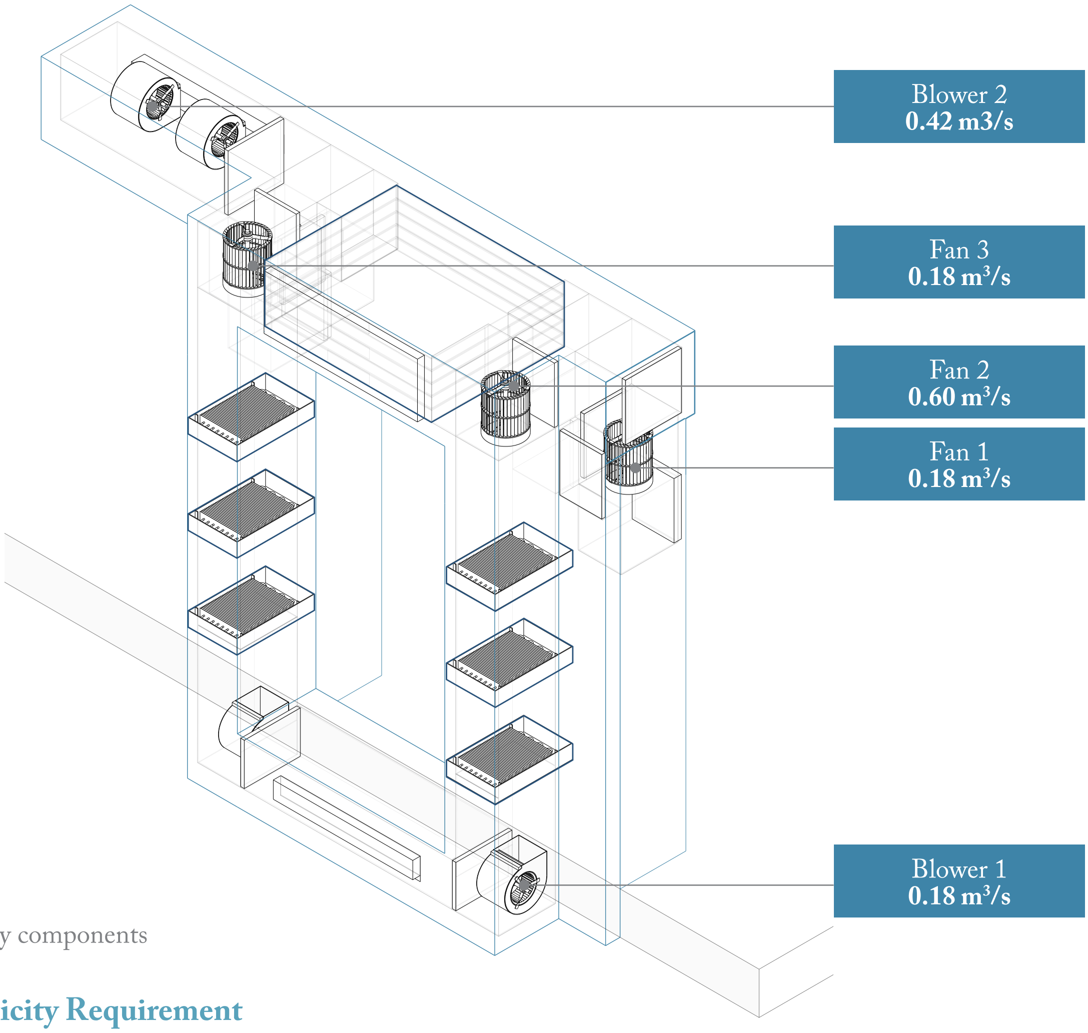


Figure 5-27 : Air supply components

Table 5-12 : Electricity Requirement

	Integrated System			
	Volume Flowrate (m^3/s)	Volume Flowrate (cfm)	Energy Consumption (kWh)	Energy Consumption (kWh/day) (11 Hours)
Blower 1	0.18	381	0.128	1.4
Blower 2	0.42	889	0.171	1.9
Fan 1	0.18	381	0.037	0.4
Fan 2	0.60	1271	0.117	1.3
Fan 3	0.18	381	0.037	0.4
				5.4 kWh/day

In order to develop the system further, the resources required for operation must be established. The requirements can be categorized into three categories, which includes, cold water for both cooling and dehumidification, hot water for regenerating the CCHEs, and electricity for the fans and blowers, which can be found in Table 5-10 to 5-12. The cold-water requirement for the M-cycle system is based on the experiment by Riangvilaikul&Kumar (2009). However, the water flow rate required for the CCHEs is derived from 3 different experiments by Jiang et al. (2014) and Ge et al. (2017). The energy needed for heating the hot water is being calculated under the assumption that the water would lose 5.7°C after the CCHEs process it. While the electrical required by the fans and blowers are referenced to Ebmpapst's Centrifugal fan and blowers version 7 catalog.

5.6 PVT System Integration

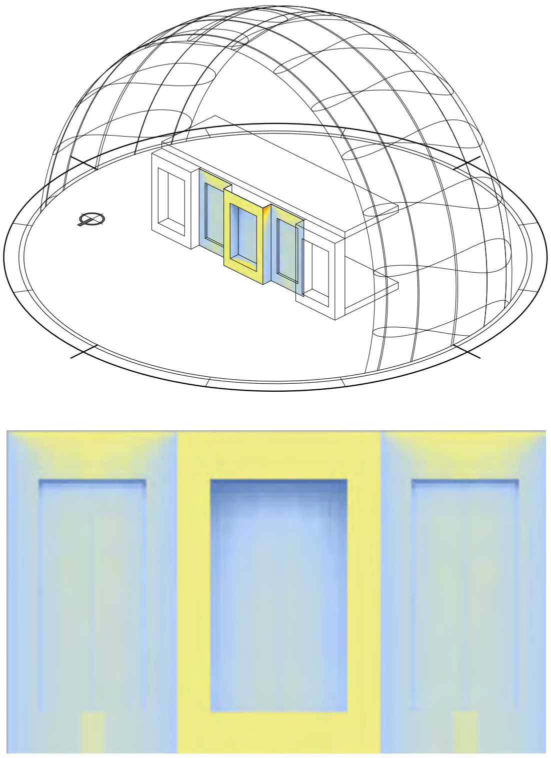


Figure 5-28 : North facade

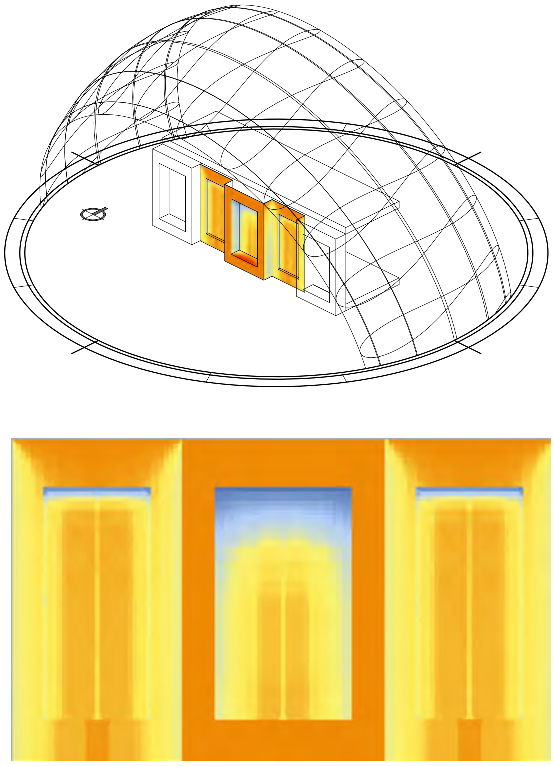


Figure 5-29 : South facade

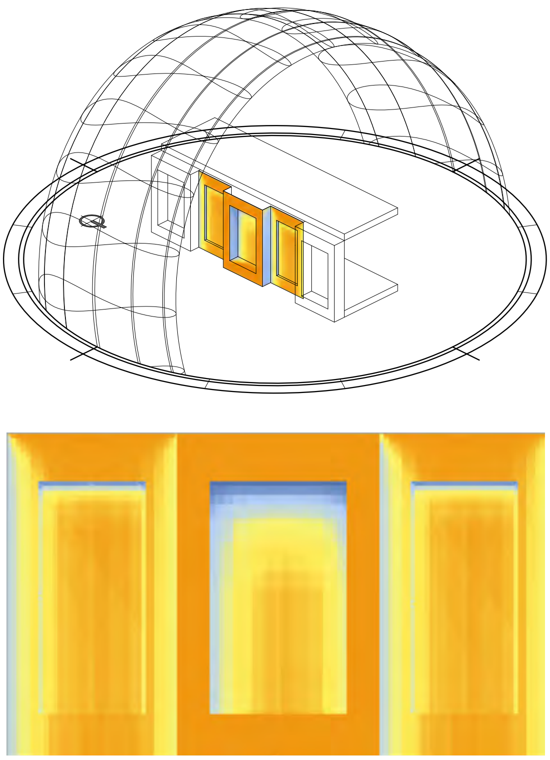


Figure 5-30 : East facade

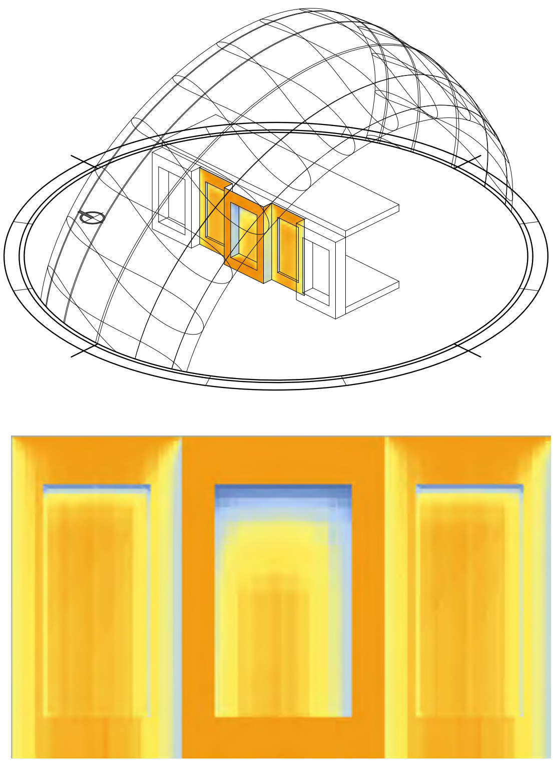


Figure 5-31 : West facade

5.6.1 Solar Radiation Studies

As the first step in integrating the PV(T) system to the façade, the solar radiation study has been simulated with Ladybug simulation, illustrating the relationship between the orientation of the façade and solar exposure. From the results [Figure 5-28 to 5-31], a conclusion can be drawn that the South facing façade will receive the highest solar radiation exposure annually. The east and west façade shares a similar exposure level with the highest solar radiation level 728 kWh/m2, while the north façade would be least exposed to solar radiation. The result corresponds with Bangkok's location, which is slightly above the equator; therefore, the sun would move from East to West due South, leaving the north façade with minimal exposure.

Furthermore, the extended frame structure fails to provide sufficient shading for the interior space except for the North facade. However, it can prevent direct solar exposure of the top side of façade, due to the high incident angle of the sun. It should also be noted that the highest solar radiation falls upon the base of the extended frame structure. The nature of the sun's movement will be used in developing and test the suitable position and tilt angle for the PV(T) System.

In order to explore the suitable approach to integrate the PV(T) system, four concepts had been developed. The concepts are developed based on the position and tilt angle of the PV(T), which was optimized through the means of Galapagos and Ladybug. The fixed parameter includes the size of the panels and the rotation point, which is positioned at the center of the panels. The parameter only allows the optimization to rotate the panels to a certain extent in which the panel wouldn't touch the main façade component. The optimization also includes movements in the z-axis to minimize the shadow of the upper panel from being cast onto the bottom one. Their performances are based on the average daily Alternating Current (AC) energy production and average daily thermal energy production. The production results are calculated with the PV's efficiency of 15% and Thermal energy production of 70%.

5.6.2 Integration Concept

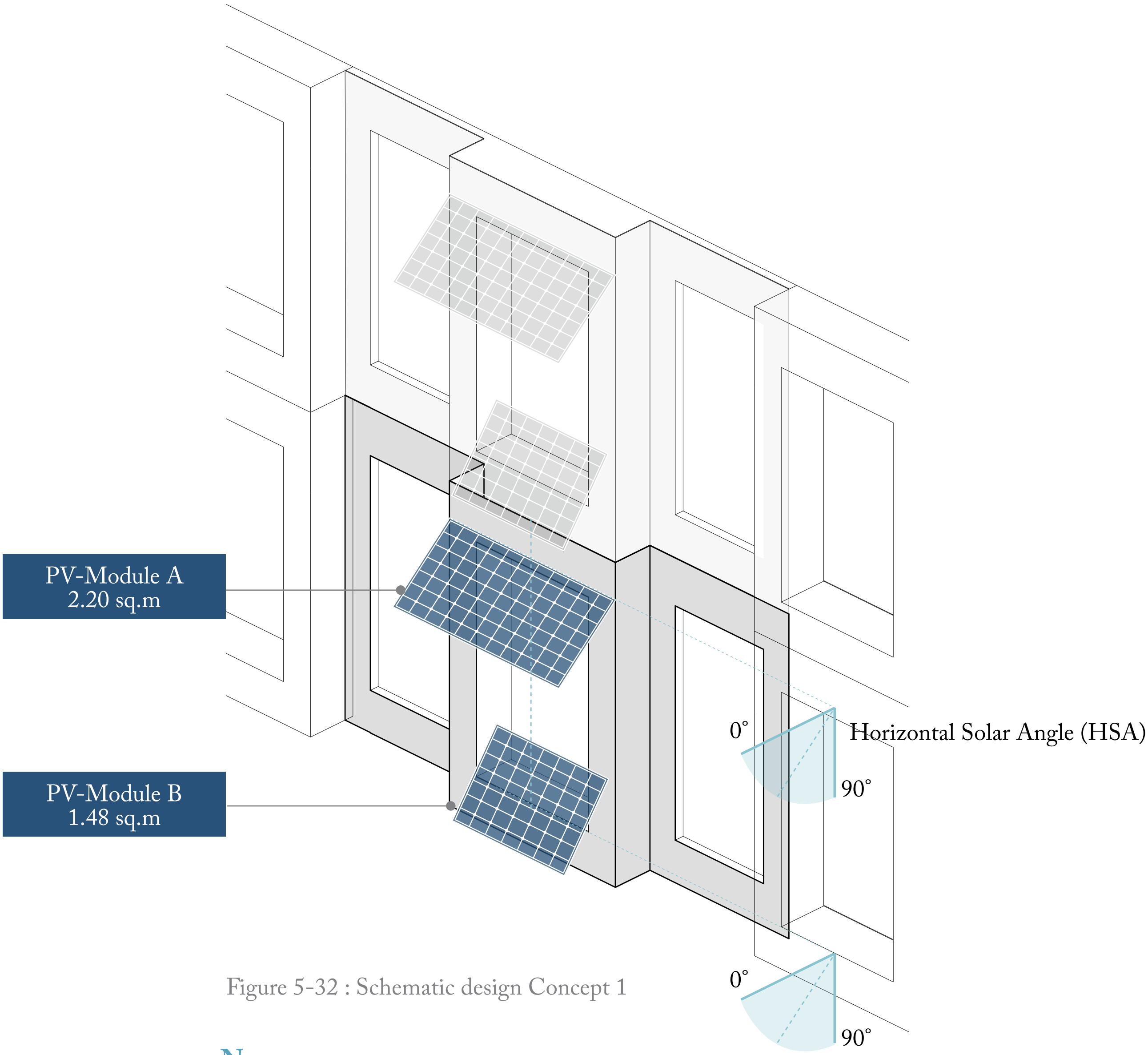
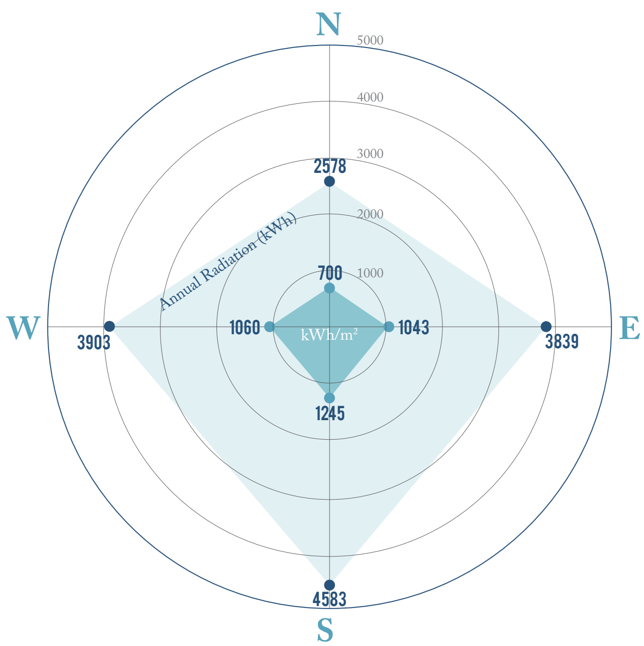


Figure 5-32 : Schematic design Concept 1



5.6.2.1 Concept 1 :Horizontal Orientation

Total Area	3.68 sq.m			
Optimal HSA (˚)	**N**	**E**	**S**	**W**
Module A	27	41	34	42
Module B	56	57	48	54
Average Daily Radiation (kWh)	7.1	10.7	12.6	10.5

PVT Performance	N	E	S	W
Average Daily AC Energy (kWh)	1.2	1.4	1.6	1.4
Average Daily Thermal Energy (kWh)	4.9	7.5	8.8	7.4

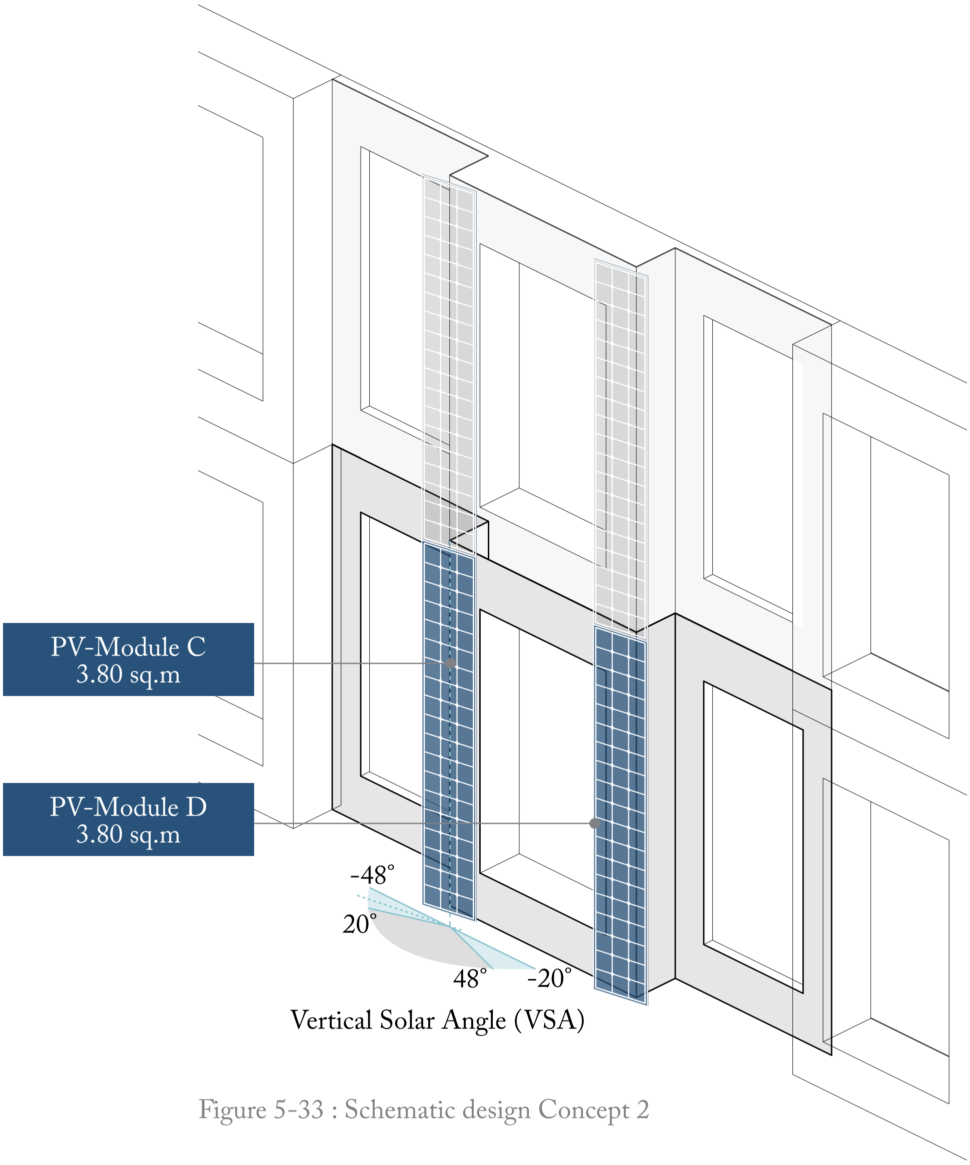
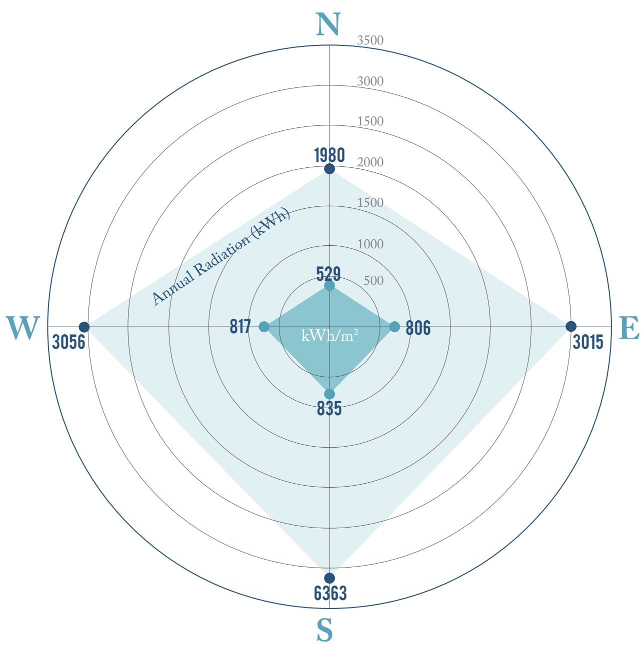


Figure 5-33 : Schematic design Concept 2



5.6.2.2 Concept 2 : Vertical Orientation

Total Area	3.74 sq.m			
Optimal HSA (°)	**N**	**E**	**S**	**W**
Module C	312	109	161	243
Module D	48	115	158	251
Average Daily Radiation (kWh)	5.4	8.4	8.6	8.3

PVT Performance	N	E	S	W
Average Daily AC Energy (kWh)	0.7	0.9	0.9	0.9
Average Daily Thermal Energy (kWh)	3.8	5.9	6.0	5.8

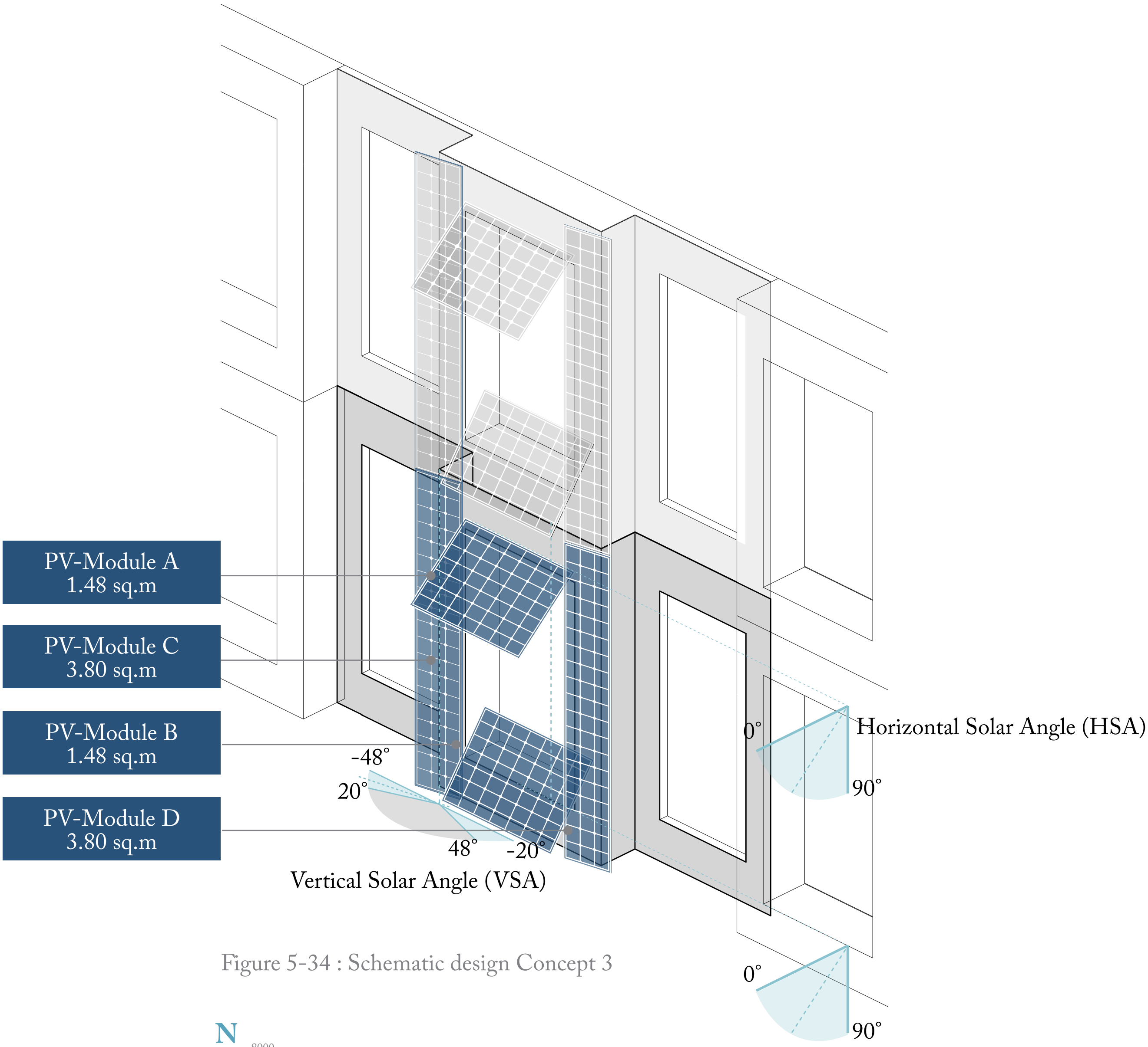
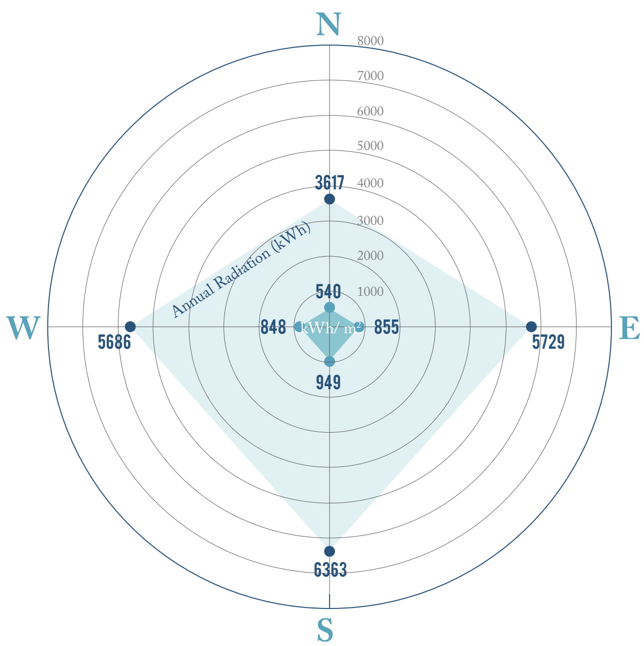


Figure 5-34 : Schematic design Concept 3



5.6.2.3 Concept 3 : Horizontal + Vertical Orientation

Total Area	**6.70 sq.m**			
Optimal HSA (°)	**N**	**E**	**S**	**W**
Module A	35	47	32	41
Module B	56	51	56	61
Module C	48	111	199	262
Module D	312	110	168	247
Average Daily Radiation (kWh)	9.9	15.7	17.4	15.6

PVT Performance	N	E	S	W
Average Daily AC Energy (kWh)	1.6	2.0	2.1	1.9
Average Daily Thermal Energy (kWh)	6.9	11.0	12.2	10.9

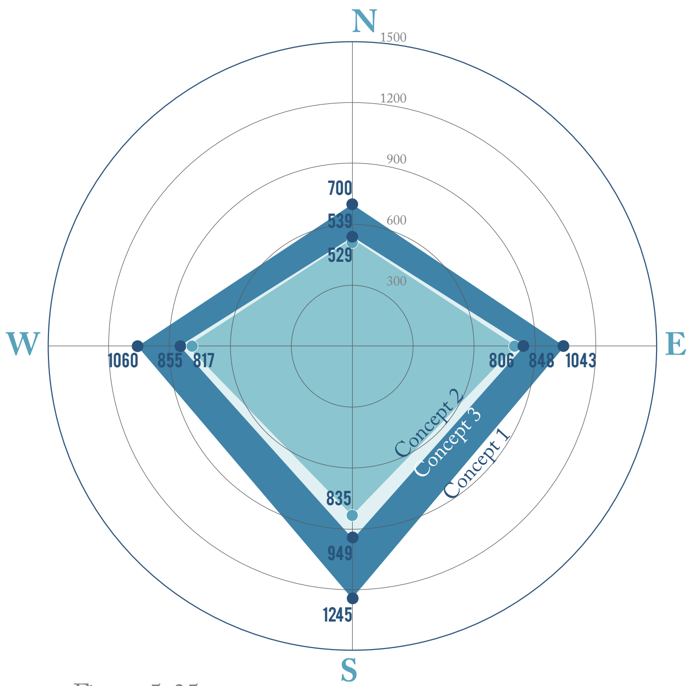


Figure 5-35
Annual Average Solar Radiation per sq.m (kWh/m²)

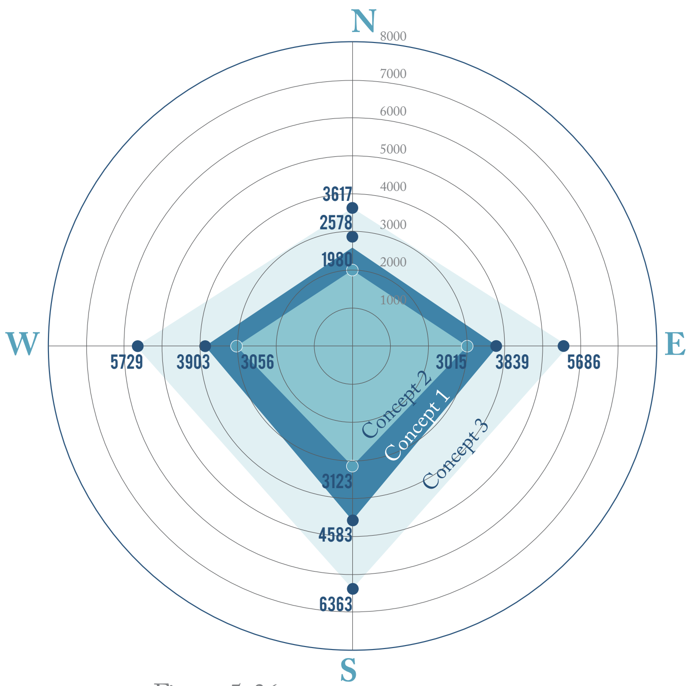


Figure 5-36
Annual Total Solar Radiation (kWh)

Table 5-13 : Design Assessment

	Concept 1 Horizontal Orientation				Concept 2 Vertical Orientation				Concept 3 Horizontal + Vertical Orientation			
Total Area	3.68 sq.m				3.74 sq.m				6.70 sq.m			
	N	E	S	W	N	E	S	W	N	E	S	W
Average Daily Radiation (kWh)	7.1	10.7	12.6	10.5	5.4	8.4	8.6	8.3	9.9	15.7	17.4	15.6
Average Daily Radiation per m² (kWh/m²)	1.9	2.9	3.4	2.9	1.4	1.1	1.1	1.1	1.5	2.3	2.6	2.3

Table 5-14 : PVT Performance

	N	E	S	W	N	E	S	W	N	E	S	W
Average Daily AC Energy (kWh)	1.2	1.4	1.6	1.4	0.7	0.9	0.9	0.9	1.6	2.0	2.1	1.9
Average Daily Thermal Energy (kWh)	4.9	7.5	8.8	7.4	3.8	5.9	6.0	5.8	6.9	11.0	12.2	10.9

5.6.3 Design Assessment

To assess the performance of the PV systems of each orientation, the annual average solar radiation exposure per m2 and the annual total solar radiation exposure has been plotted in Figure 5-35 and 5-36, while the daily performances in terms of AC energy and thermal energy production are summarized in Table 5-13,14. The results have portrayed a corresponding conclusion with the solar study, in which South façade would receive the highest solar radiation exposure, the west and east façade shares a similar exposure level, while the North façade exposure is minimal. Based on the results, it is evident that Concept 3 has the highest total solar radiation exposure due to its largest surface area coverage. However, when the results have been distributed evenly amongst the surface area, Concept 1 with horizontal configuration shows higher exposure level. A conclusion could be drawn that, due to the high solar incident angle, horizontal orientation is the most suitable concept to integrate the PV system onto the façade. Therefore, further design development could be generated based on the horizontal orientation scheme.

5.6.3 Design Assessment

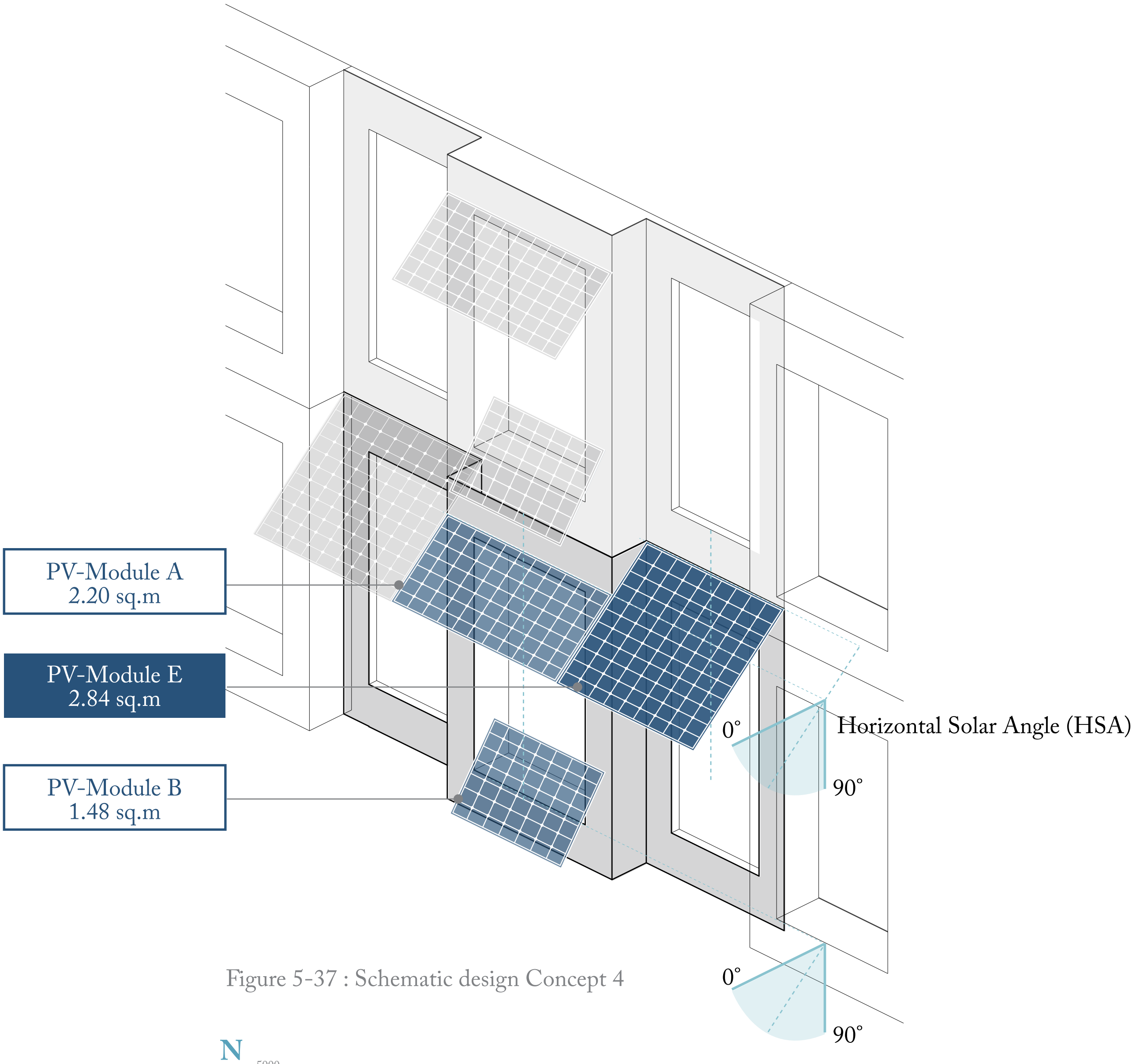
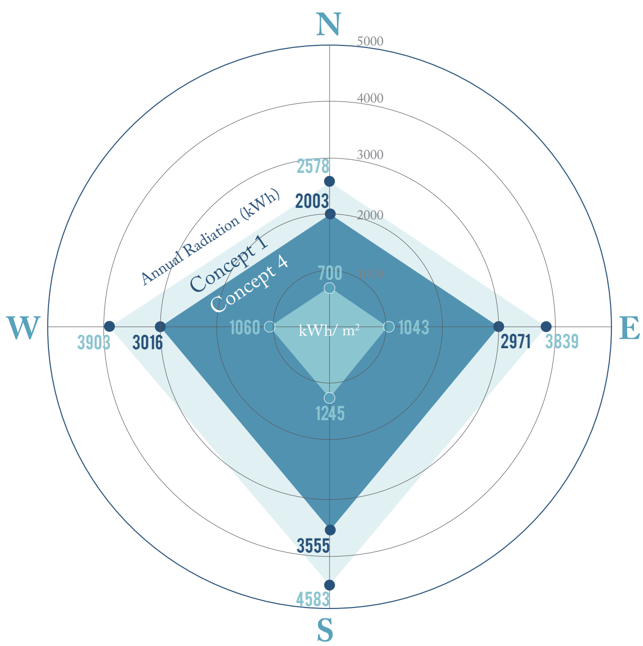


Figure 5-37 : Schematic design Concept 4



5.6.3.1 Concept 4 : Additional Horizontal Orientation

Total Area	2.84 sq.m (6.52 sq.m)			
HSA (°)	N	E	S	W
Module E	27	41	34	42
Average Daily Radiation (kWh)	5.5	8.3	9.7	8.1

PVT Performance	N	E	S	W
Average Daily AC Energy (kWh)	1.1	1.1	1.3	1.1
Combined with Concept 1 (kWh)	2.3	2.5	2.9	2.5
Average Daily Thermal Energy (kWh)	3.8	5.8	6.8	5.7
Combined with Concept 1 (kWh)	8.7	13.3	15.6	13.1

	Case1 : 200 L/hr	Case2 : 155 L/hr	Case3 : 60 L/hr
5.4 kW/day Electricity Requirements	**43.6** kW/day Thermal Energy Requirements	**33.8** kW/day Thermal Energy Requirements	**13.1** kW/day Thermal Energy Requirements

Table 5-15 : Energy Assessment

	Thermal Energy Requirements (kW/day)			
	N	**E**	**S**	**W**
Energy Production	8.7	13.3	15.6	13.1
Concept 1 + 4	1.34 kW/m²	2.04 kW/m²	2.39 kW/m²	2.00 kW/m²
Case 1 Energy Required:	**-34.9**	**-30.3**	**-28**	**-25**
Case 1 Area Required:	**+26.04 m²**	**+14.85 m²**	**+11.72 m²**	**+12.5 m²**
Case 2 Energy Required:	**-25.1**	**-22.5**	**-18.2**	**-20.7**
Case 2 Area Required:	**+18.73 m²**	**+6.52 m²**	**+6.53 m²**	**+10.35 m²**
Case 3 Energy Required:	**-4.4**	**+0.2**	**+2.5**	**+0.0**
Case 3 Area Required:	**+3.28 m²**	-	-	-

	Electricity Requirements (kW/day)			
	N	**E**	**S**	**W**
Energy Production	2.3	2.5	2.9	2.5
	0.35 kW/m²	0.39 kW/m²	0.46 kW/m²	0.39 kW/m²
Energy Required:	**-3.1**	**-2.9**	**-2.5**	**-2.9**
Area Required:	**+7.95 m²**	**+7.44 m²**	**+5.4 m²**	**+7.44 m²**

5.6.3.2 Energy Assessment

Concept 4 is developed based on the assumption that horizontal orientation could obtain the highest solar radiation exposure per m2. Therefore, an additional set of PV(T) system with an area of 2.84m2 has been installed on the air supply unit part, hovering over the adjacent window. The additional system would be connected to the top panel of the system proposed in Concept 1 therefore, sharing a similar tilt angle and position. With the additional system, thermal energy production increases with an average of 77% in comparison to Concept 1, while the AC current production increases with an average of 83%. Furthermore, despite having a slightly lesser coverage area, its thermal productivity is 124% higher on average, while its AC productivity is 135% higher in comparison to Concept 3.

With the drastic improvement in terms of energy production, Concept 4 has been assessed with the thermal and electrical energy requirements of the system. From results in Table 5-15, it could be concluded that, despite having a higher production rate, Concept 4 fails to provide sufficient heating energy for Case 1 (200L/hr) and Case 2 (155L/hr). Therefore, additional PV(T) system should be implemented. However, the scheme can provide sufficient energy for Case 3 (60L/hr) from the majority of its orientation except for the North façade, which requires an additional area of 3.28m2. Nevertheless, due to the relatively low-efficiency rate of the PV panels, the scheme fails to satisfy the electrical requirements of the façade system, being able to supply an average of 47% of what is required.

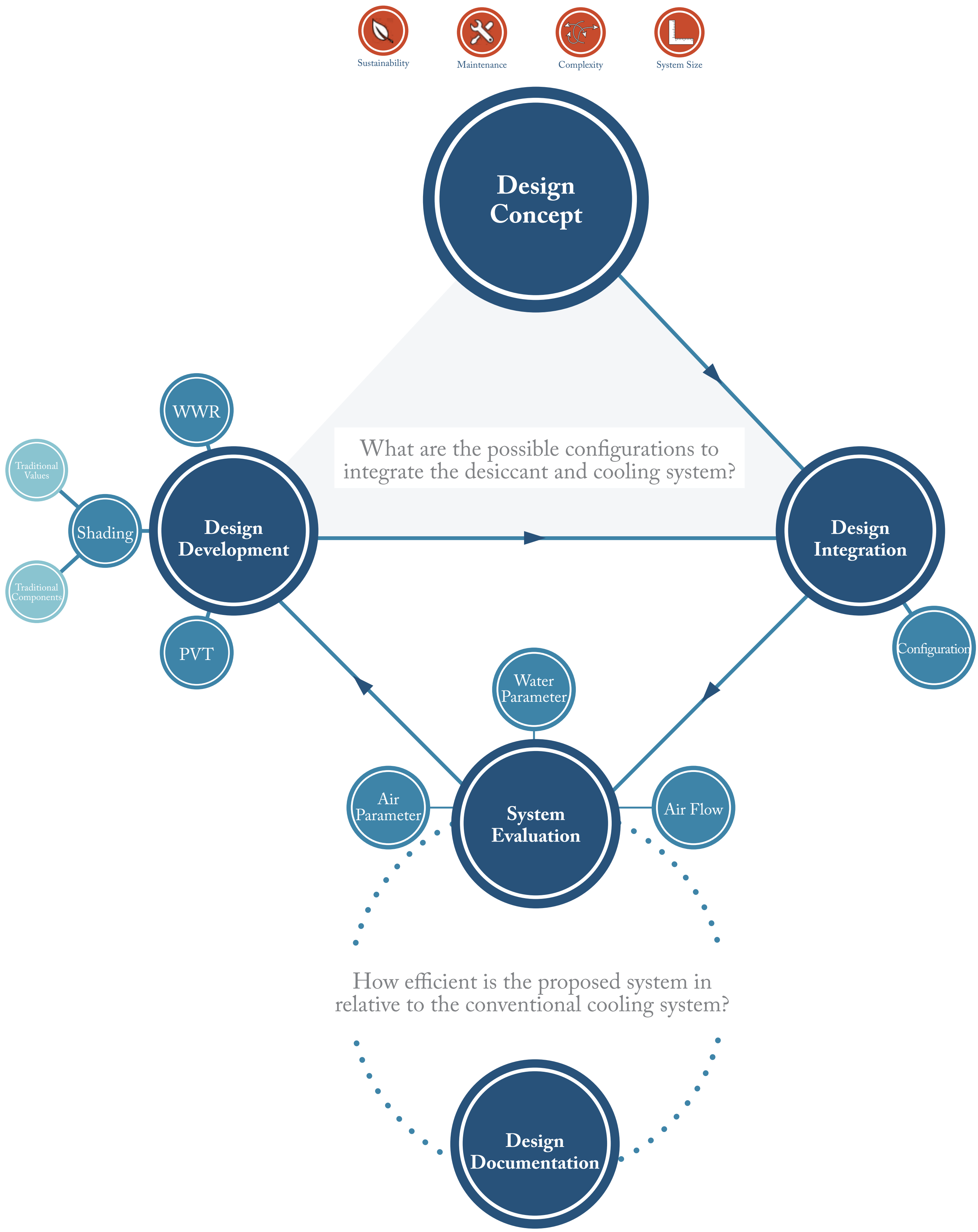


Figure 5-38 : Design development strategy flowchart

5.7 Design Development Strategy

To determine and optimize the possible configurations to integrate the desiccant and cooling system as a façade component, the scheme in Figure 5-38 will be implemented as the strategy in the façade's design development. An office building which resembles the typical high-rise office typology in Bangkok will be analyzed and use as a benchmark for the project. The design concept will be integrated into the benchmark allowing preliminary assessment in terms of cooling capacity and air flow rates to be evaluated. The evaluation will provide further insights on the number of façade system requires for a floor area and additional development agendas.

To validate the performance of the 'Desigrated' façade system, a thorough system evaluation process will be conducted to elaborate the extend to the system's performance under Bangkok's condition. A series of calculation will be made to determine the supply air condition based on the different ambient air ranges. The evaluation would determine the number of operable hours and resources required by the system. Therefore, an assessment regarding energy consumption could be made and compare to the conventional air conditioning system. The results from the assessment will provide additional resources to be used in the design development stage. It would allow further discussions to be made regarding the feasibility of PV and PVT integration to the shading system. Moreover, the system's insulation details, installation procedure, and maintenance will also be further discussed.

Figure 6-1 : Krungthai Bank - Sukhumvit building

6 System Application

Krungthai Bank - Sukhumvit Building

General Information	
Building Function	**Office Building**
Operation Hours per Day	**8**
Operation Days per Year	**246**

Area Distribution	**m^2**
Total Floor Area	**38,857**
Floor Area	**27,731**
Air Conditioned Floor Area	**25,278**
Un-Conditioned Floor Area	**2,093**
Parking Area	**11,486**

6.1 Benchmark

A benchmark has to be established to evaluate the 'Desigrated' system under Bangkok's context. Therefore, the Krungthai Bank (KTB) -Sukhumvit Building has been selected as the benchmark building for the system's application and further evaluation. The building is located on Sukhumvit road, one of the major business arteries of Bangkok. The KTB-Sukhumvit Building opened its door in 2005. It functions as a high-rise office serving its purpose as the headquarter for Krung Thai Bank. With its full glazed façade and rectilinear form, the building's style and appearance reflect the modernism design approach, resembling the typical high-rise typology on Bangkok's skyline.

The high-rise stands at 114m, consisting of 30 levels with two additional rooftop levels, and two basement levels. It has a total coverage area of 38,857 m^2, in which 27,371 m^2 served as an operable area while 11,486 m^2 served as parking spaces. Out of the operable area, 25,278 m^2 (92%) is operated with an air conditioning system. The building operates 8 hours per day, accounted to 246 days per year.

In this chapter, the building will be analyzed in terms of Façade system, Solar radiation exposure, and the HVAC system. Furthermore, a typical floor plan will be single out to be analyzed further as a site for the proposed façade system installation. The energy consumption data and simulation would be assessed to be used in evaluating the system.

Figure 6-2 : Krungthai Bank - Sukhumvit Building's East facade

6.1.1 Facade System

Despite being fully glazed, the building still concurs to the Ministerial Regulation (2009) regarding energy conservation standards and design requirements by having the overall thermal transfer value of 44.75 w/m2. Due to minimal shading and insulation, the façade relies mainly on the performance of the insulating glazing units. The façade is made up of 4 main types of glazing. It is composed primarily of GL1 or the 24.38mm thick laminated insulated glazing unit, which has been used in both the large and small clear window panes of the façade. GL3 and GL4 shares a similar thickness of 12mm and are used at the slab edge part of the façade, while GL5 could be found in the lower parts of the building which primarily serves as a public area. Moreover, the façade is integrated with vertical aluminum fin components which run vertically over the elevation in a symmetrical manner.

Figure 6-3 : Building's facade section

Table 1 : Glazing Unit Type

GL1	Laminated Insulated Unit 24.38mm
	Annealed Reflective Tinted glass 6mm + PVB Interlayer 0.38mm + Clear Float 6mm + Air gap 6mm + Clear Float glass 6mm
GL3	Laminated Insulated Unit 12.38mm
	Annealed Reflective Tinted glass 6mm + PVB Interlayer 0.38mm + Air gap 6mm + Clear Float glass 6mm
G4	Laminated Insulated Unit 12.38mm
	Annealed Reflective Tinted glass 6mm + PVB Interlayer 0.38mm + Air gap 6mm + Tinted Float glass 6mm
GL5	Laminated Insulated Unit 24.38mm
	Annealed Reflective Tinted glass 6mm + PVB Interlayer 0.38mm + Clear Float 6mm + Air gap 6mm + Clear Low-E 6mm

Building Envelope Performance

	W/m²
Overall Thermal Transfer Value (OTTV)	**44.75**

Lighting System

	W/m²
Lighting Power Density (LPD)	**12.15**

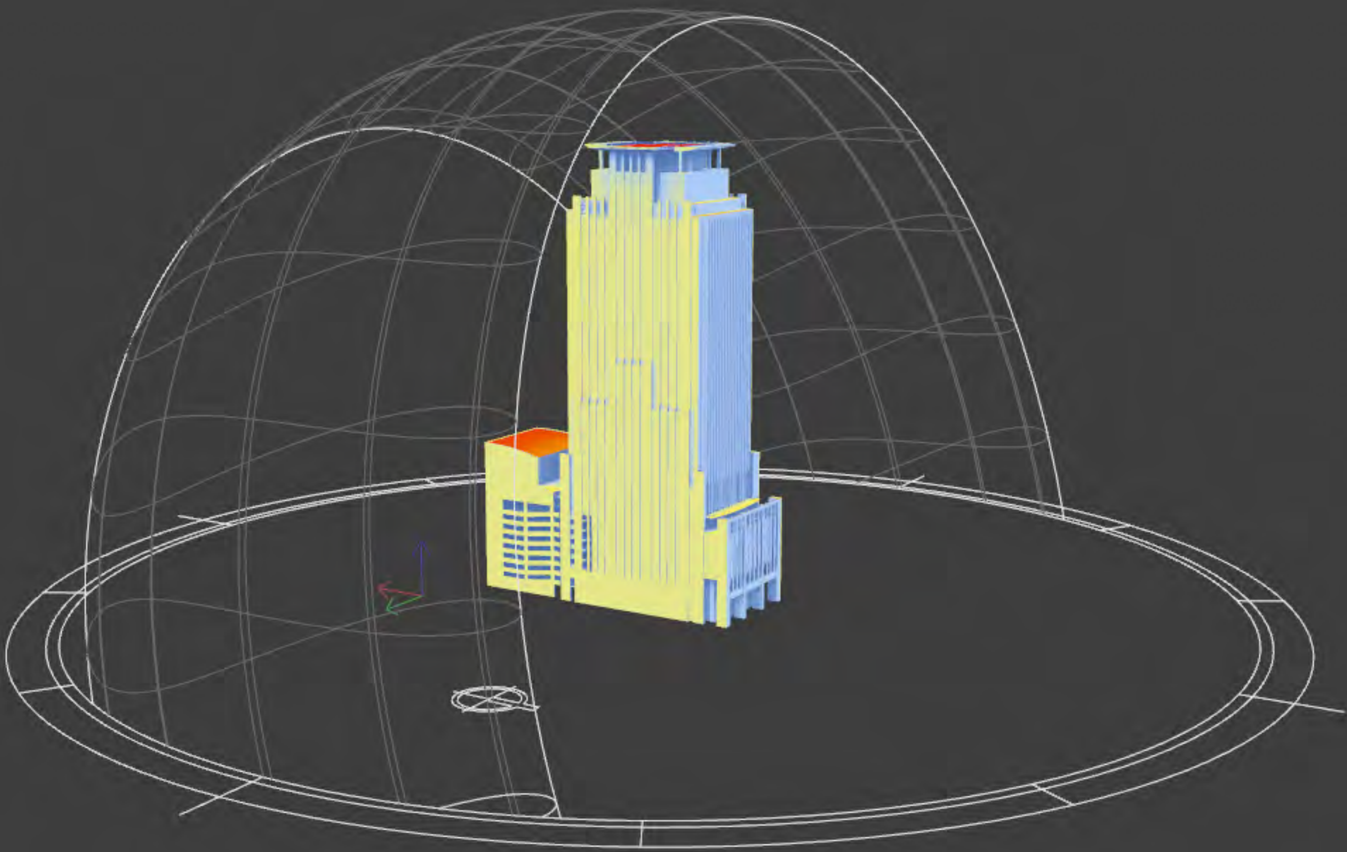


Figure 6-4 : Krungthai Bank - Sukhumvit Building's solar radiation simulation

6.1.2 Solar Radiation Analysis

To evaluate the effect which the facade component and orientation have on the building's solar radiation exposure, a solar radiation analysis has been simulated. A simplified model of the building was constructed and used for the simulation in Ladybug software. As expected, the results have shown minimal exposure on the north façade, while exposure into the window panes is visible in the other orientations. It is evident that the vertical aluminum fin and cladding have minimal effect in preventing solar exposure. This is due to the depth of the fins, which only extends 10cm out from the window pane. Moreover, due to the high incident angle, the vertical fins fail to restrain direct solar radiation into the building.

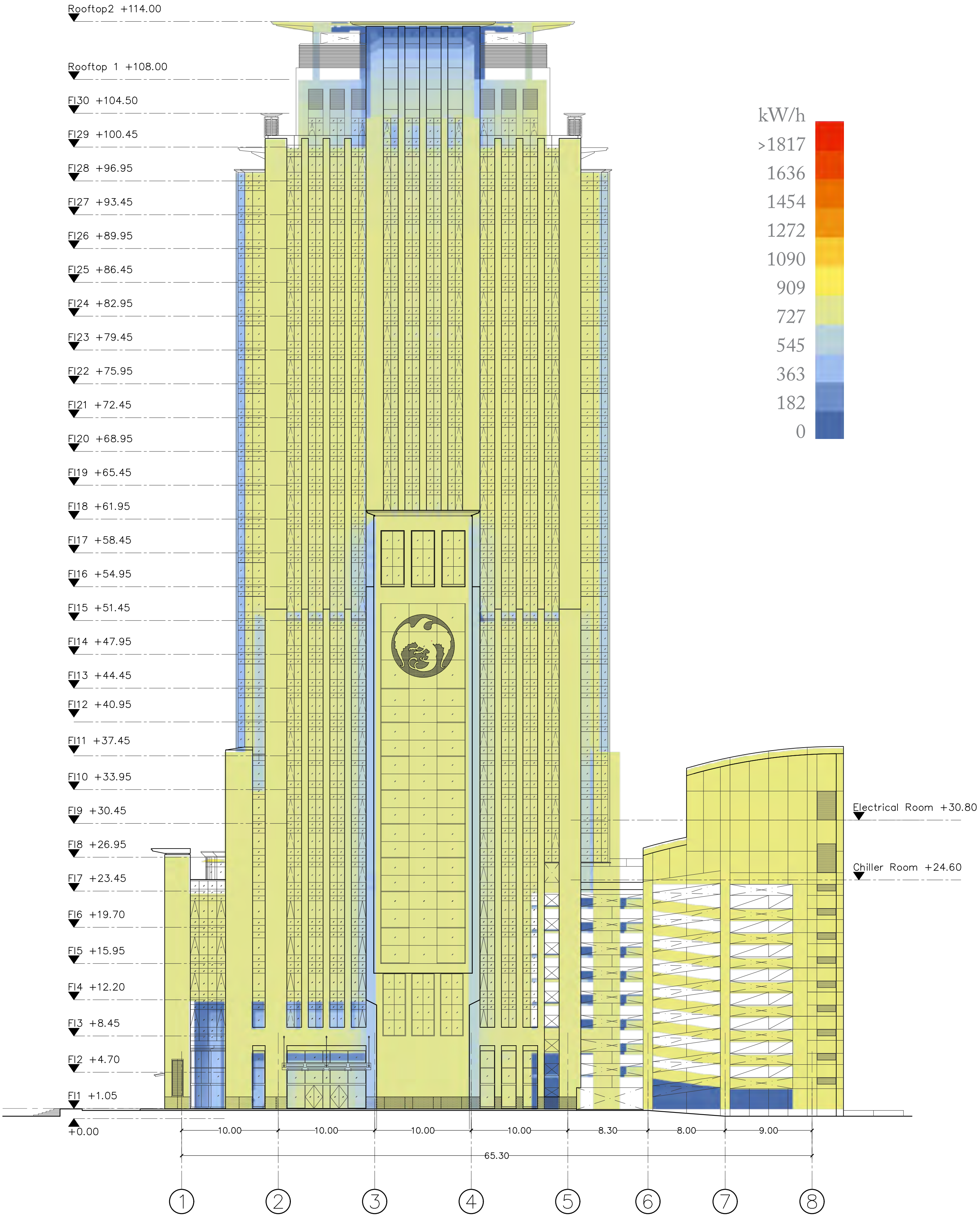


Figure 6-5 : East Facade

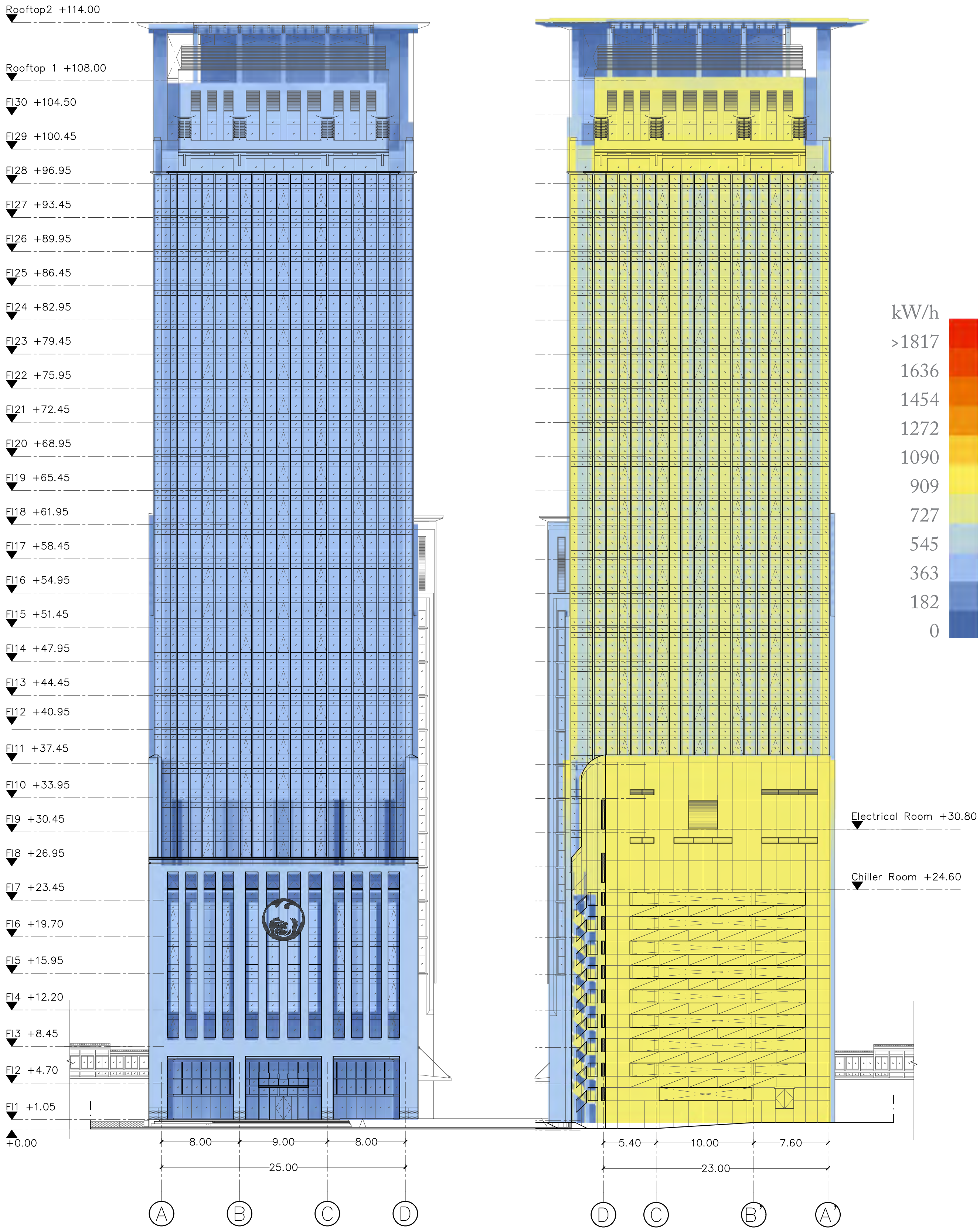


Figure 6-6 : North Facade

Figure 6-7 : South Facade

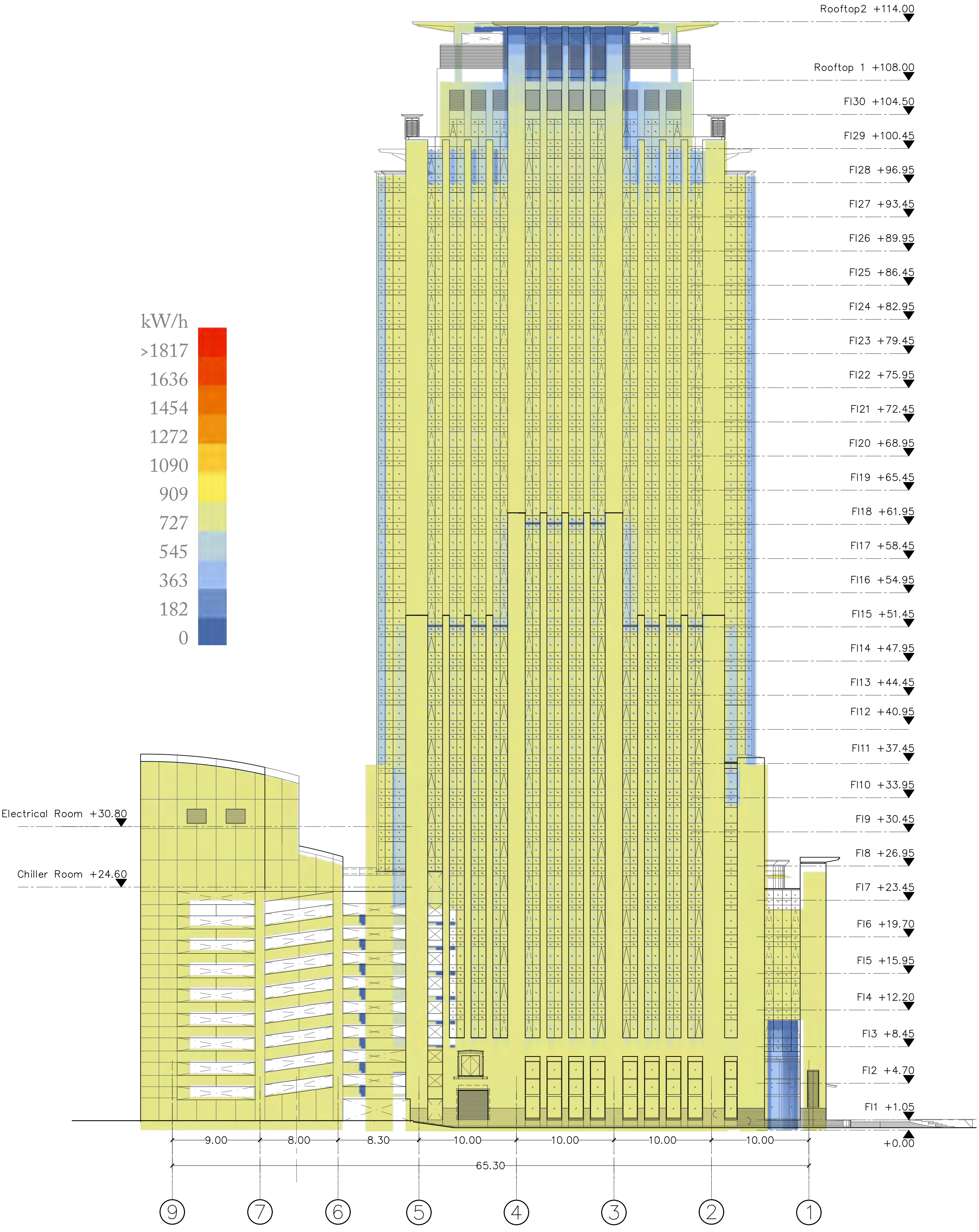


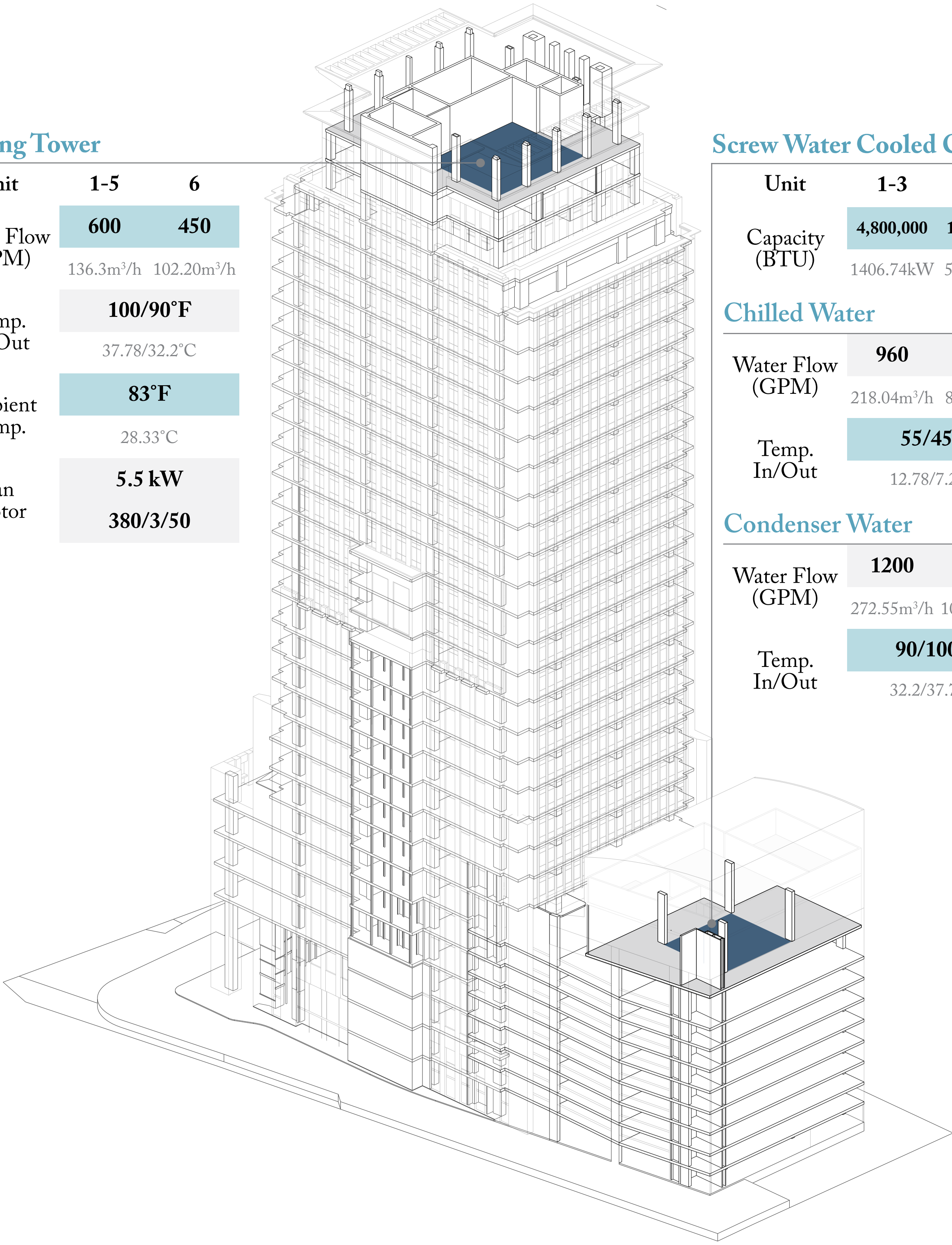
Figure 6-8 : West Facade

6.1.3 HVAC System - Cooling Tower / Screw Water Cooled Chiller

The building consists of 6 Cooling towers located on the rooftop and 4 Screw water cooled chillers located on the rooftop of the car park building. 2 out of 3 chillers will be simultaneously working during the weekdays, while the 4th one only operates during the weekends. It should be addressed that the relatively low temperature of the chilled return water could be used to serve both the desiccant and m-cycle system.

Cooling Tower

Unit	1-5	6
Water Flow (GPM)	600	450
	136.3m^3/h	102.20m^3/h
Temp. In/Out	100/90˚F	
	37.78/32.2˚C	
Ambient Temp.	83˚F	
	28.33˚C	
Fan Motor	5.5 kW	
	380/3/50	



Screw Water Cooled Chiller

Unit	1-3	4
Capacity (BTU)	4,800,000	1,800,000
	1406.74kW	527.52kW

Chilled Water

Water Flow (GPM)	960	360
	218.04m^3/h	81.76m^3/h
Temp. In/Out	55/45˚F	
	12.78/7.22˚C	

Condenser Water

Water Flow (GPM)	1200	450
	272.55m^3/h	102.20m^3/h
Temp. In/Out	90/100˚F	
	32.2/37.78˚C	

Figure 6-9 : Schematic diagram of centralized cooling system

6.1.4 HVAC System - Pre Cooled AHU / Service Area

Pre-Cooled AHU - 1

Cooling Capacity

Sensible (BTU)	**737,500** 216.14kW
Total (BTU)	**281,700** 82.56kW
Water Flow (GPM)	**75.3** 17.10m³/h
Supply Fan (CFM)	**7,800**

Pre-Cooled AHU - 2

Cooling Capacity

Sensible (BTU)	**2,516,720** 737.58kW
Total (BTU)	**961,370** 281.75kW
Water Flow (GPM)	**410** 93.12m³/h
Supply Fan (CFM)	**29,690**

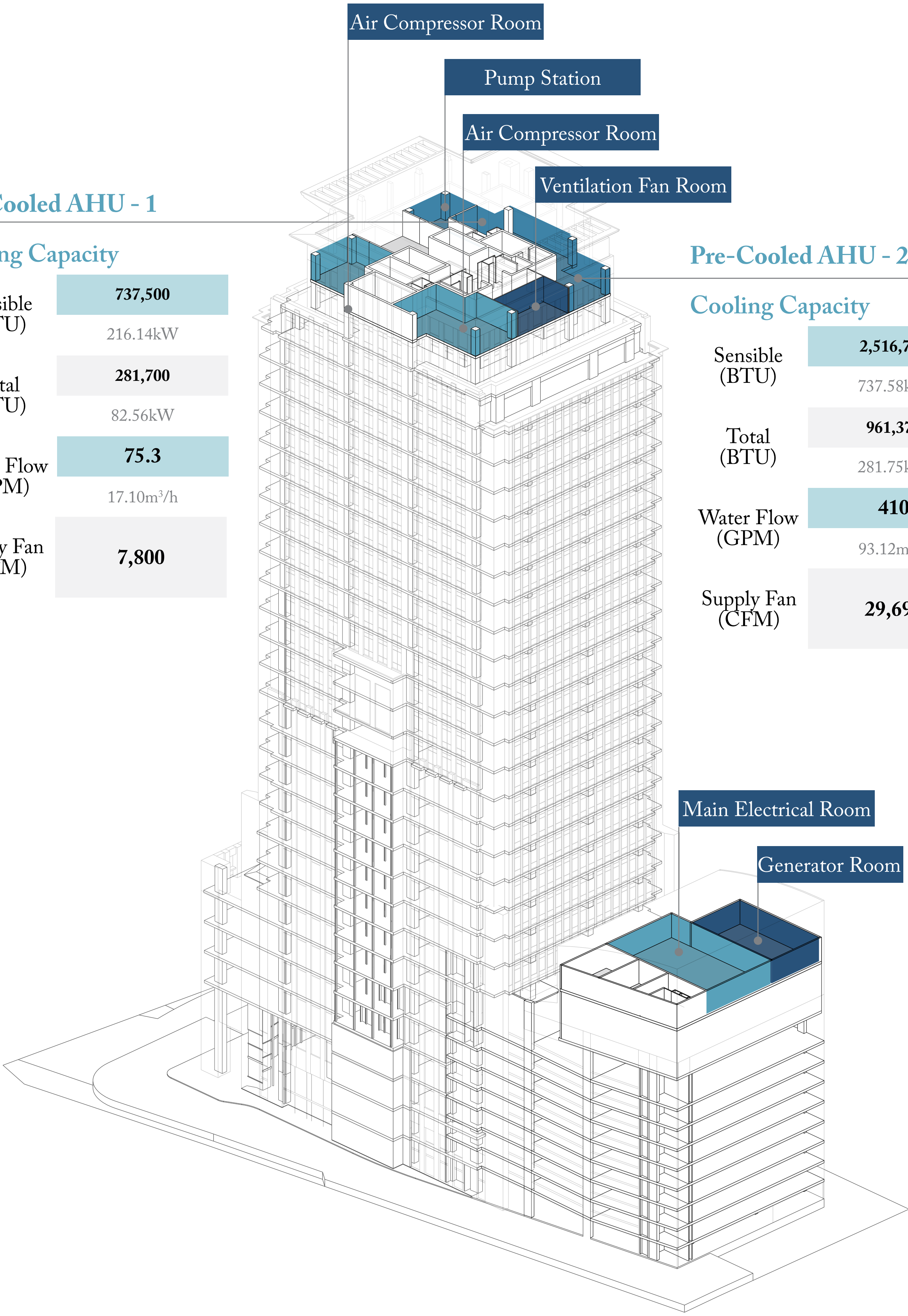


Figure 6-10 : Schematic diagram of pre-cooling system and service areas

6.2 Typical Floor Detail

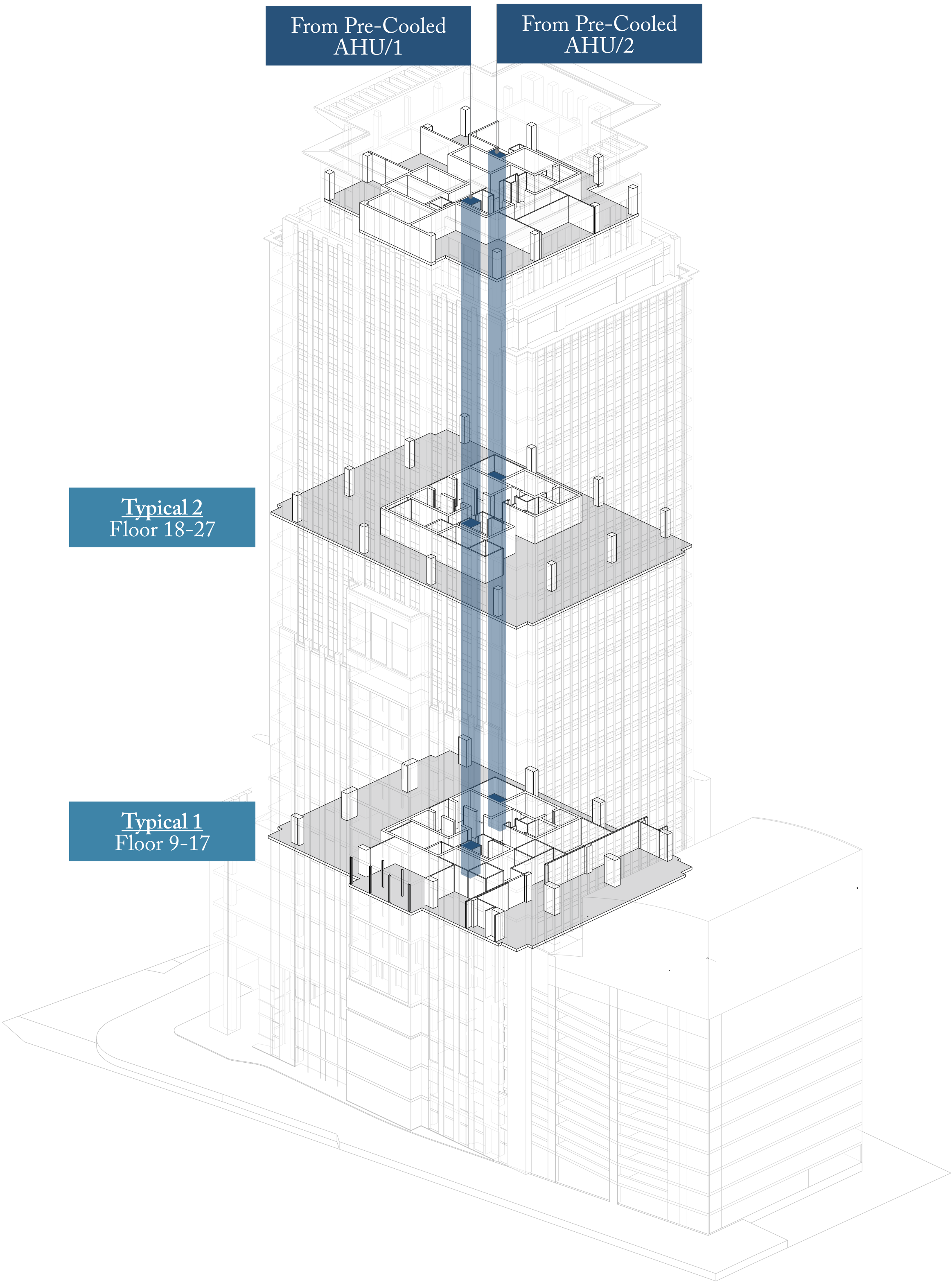


Figure 6-11 : Schematic diagram of centralized cooling system

Figure 6-12 : 19th Floor Plan

Table 2 : 19th Floor : Area Distribution	m^2
Main Office Area	**695.0**
Secondary Office Area	**29.4**
Floor Entrance Foyer	**36.6**
Service Area	**172.0**
MEP Shaft	**7.0**
AHU Room	**24.1**
Fresh air Shaft	**2.4**
Electrical and Computer Room	**8.5**

6.2.1 Floor Plan - Function

The building possessed two types of typical floor area. Typical 1 covers the area from level 9-17 which includes a center service core and an external elevator shaft, while Typical 2 covers the area from level 18-27 with only one center service core. The Typical 2 floor plan of the 19th floor is selected to be used as the benchmark, as it has a large open floor area with only a single service core in the center. The main office floor area which is subjected to simulation occupies 695 m^2. With 3.5m floor to floor height, it covers a volume of 2430m^3.

6.2.2 Floor HVAC System - AHU

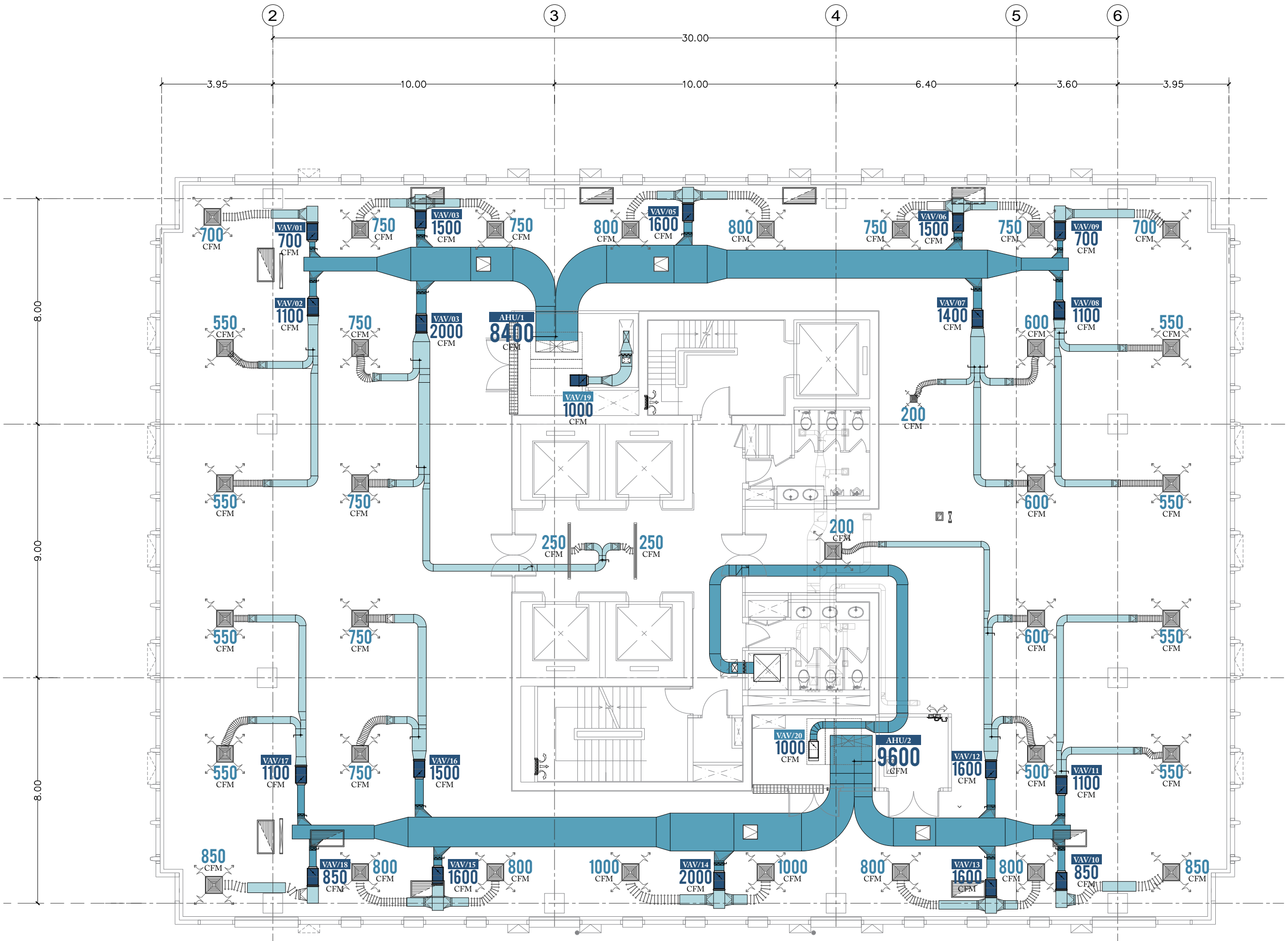


Figure 6-13 : 19th floor AHU and VAV Plan

Table 6-3 : Air Handling Unit Schedule

Unit No.	Cooling Capacity (BTU/hr)		Entering Air Temp. (°F)		AirFlow (CFM)	Supply Fan	
	Total	Sensible	Dry Bulb	Wet Bulb		Motor (kW)	Electricity
AHU/1	**238,810**	**143,000**	**73.9**	**66.5**	**8,400**	**11.0**	**380/3/50**
	69.98 kW	41.99kW	23.28°C	19.17°C			
AHU/2	**272,520**	**166,000**	**74.1**	**66.5**	**9,600**	**11.0**	**380/3/50**
	79.87 kW	48.65kW	23.39°C	19.17°C			

Table 6-4 : AHU Chilled Water Data

Unit No.	Flow Rate (GPM)	Inlet Temp. (°F) EWT/LWT	Pressure Drop (Ft. wg)
AHU/1	**47.76**	**45/55**	**10.67**
	10.85m³/hr	7.2/12.8°C	31.89kPa
AHU/2	**54.50**	**45/55**	**6.50**
	12.38m³/hr	7.2/12.8°C	19.43kPa

Table 6-5 : Air Handling Unit Schedule - 19th Floor

Unit No.	Chilled Water Requirement	Energy Consumption	
	GPM	kWh/Year	kWh/day (246days)
AHU/1	47.76	17,670	71.83
	10.85 m^3/hr		
AHU/2	54.5	13,732	55.82
	12.38 m^3/hr	**Total Consumption** **31,402** kWh/year	**Avg. Daily Consumption** **127.65** kWh/day

Table 6-6 : Centralized System Energy Consumption - 19th Floor

Considering 10.6% of Energy Consumption

System	Average CDS | CHS	Total Avg. Energy Consumption	19th Floor Avg. Energy Consumption
	GPM	kWh/day	kWh/day
Screw Water Cooled Water Chiller	960	2240.89	237.5
	10.85 m^3/hr		
Cooling Tower	570	58.67	6.22
	10.85 m^3/hr	**Total Consumption** **2,299.56** kWh/day	**Total Consumption** **244.95** kWh/day

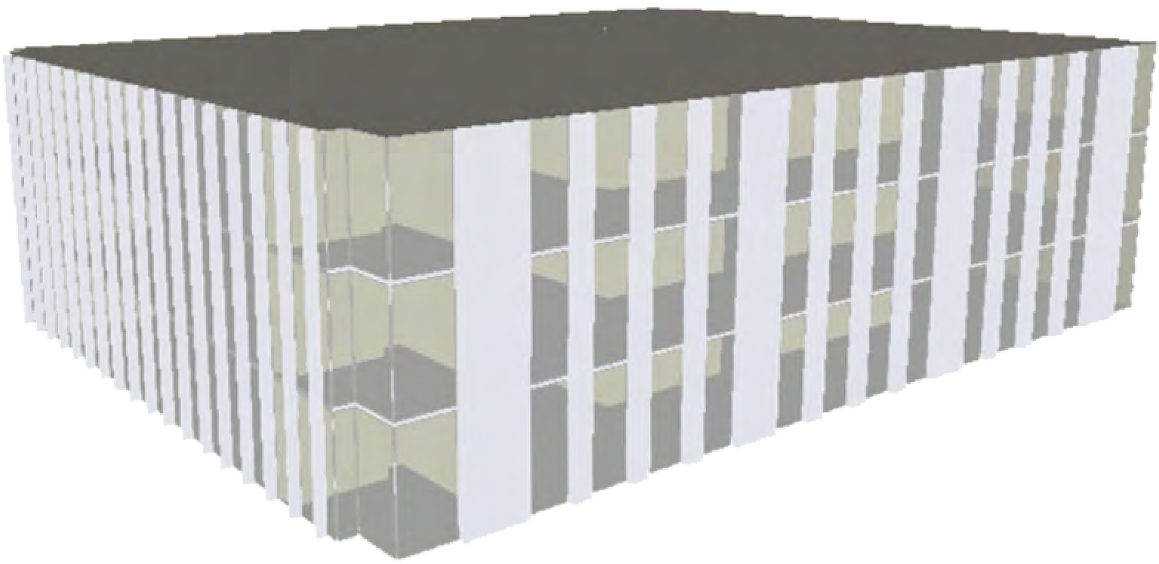
Air Handling Units	Centralized Cooling Systems	Total Energy Consumption
127.65	**244.95**	**372.60**
kWh/day	**kWh/day**	**kWh/day**

6.2.3 HVAC Energy Consumption

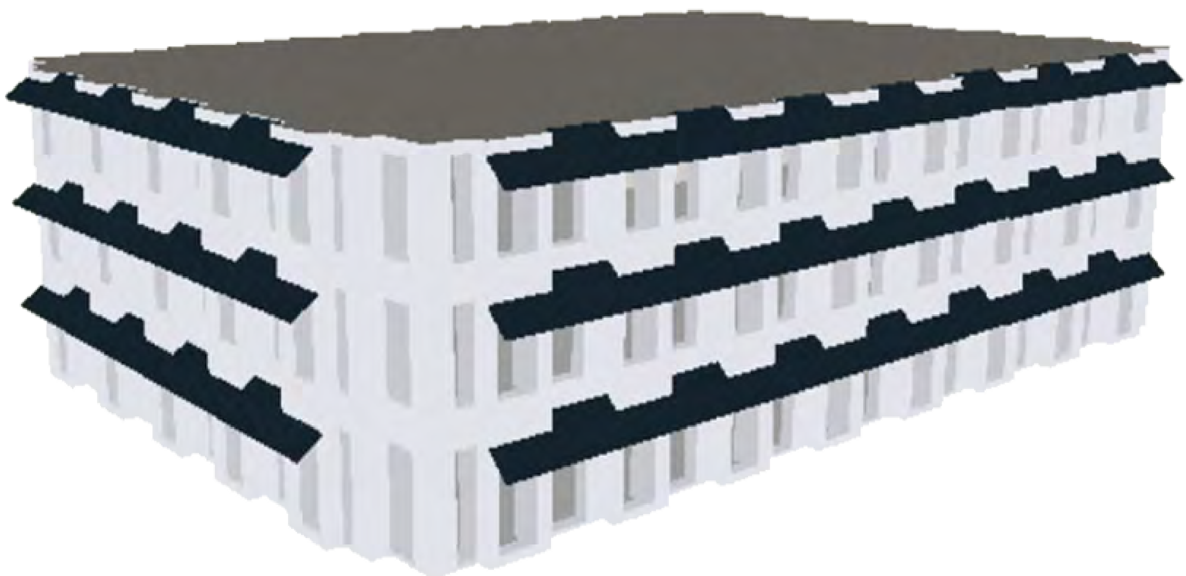
The energy consumption calculation is based on the energy audit data of KTB – Sukhumvit building. The calculation aims to determine the electrical energy consumption of the 19th floor. It is calculated based on two main categories: Air handling units and the centralized system, which includes the water chiller system and cooling tower. The 19th floor accommodates two AHU system with the air flow rate of 8400 CFM and 9600 CFM, respectively. The total energy consumption of the two system cumulates to 31,402 kWh per year, which averages out to 127.56 kWh/day.

The energy consumption of the centralized system for the 19th floor is derived from the chilled water proportion required by the 19th floor. Based on the system requirements, the 2 AHU requires 102.26 gallons of chilled water per minute, which is accounted for 10.6% of the chilled water produced by the chillers. Considering the average energy consumption by the four chillers, the 19th floor requires 237.5 kWh of electricity per day. The cooling tower would need 6.22 kWh/day, therefore, the energy requirement of the cooling system of the 19th floor would sum up to 372.60 kWh/day.

Existing Building Model



Proposed Application Model



North & South Facade
5 Systems/side

East & West Facade
8 Systems/side

Maximum no.
of Facade Integration
26
Systems

Figure 6-14 : Simplified model used in Design Builder

Table 6-7 : Preliminary Simulation Results - Existing Condition

Supply Air Condition			Cooling Load	
Min. Supply Air temp. (°C)	Supply Humidity Ratio (g/g)	Air Change Rate (ach)	Design Capacity (kW)	Design Flowrate (m^3/s)
12.0	0.077	1.75	**134.31**	**6.51**

Table 6-8 : Preliminary Simulation Results - Proposed System

Supply Air Condition			Cooling Load	
Min. Supply Air temp. (°C)	Supply Humidity Ratio (g/g)	Air Change Rate (ach)	Total Cooling Load (kWh)	Design Flowrate (m^3/s)
18.9	0.0081	-	**84.24**	**4.42**

Required Design Flowrate	System Flowrate	No. of Facade System required
4.42 m^3/s	**0.42** m^3/s	**11** System

6.2.4 Design Simulation

To be able to define the number of systems require in cooling the floor, the design air flow rate needs to be determined. Therefore, Design Builder software has been used in simulating the Design Cooling load and Design Flowrate. The KTB-Sukhumvit has been modeled and simplified to be used for the simulation. The parameters used in the simulation is listed in Appendix C. The occupancy rate is set at 0.11 people per m2 (approximately 70 people) while the cooling setpoint / operative temperature is set at 25.5°C based on the average of the comfortable temperature from the literature reviews. It should also be noted that the dehumidification control is set to constant humidity ratio while the cooling limit type is set as limit flowrate/capacity.

Two simulations have been conducted to compare the results of the conventional A/C system and the 'Desigrated' system. With the conventional A/C system, the supply air temperature is set at 12.0°C with 0.0077 g/g humidity ratio. Furthermore, the air change rate is fixed at 1.75 ach based on the minimum standards in Thai's regulation. The simulation has shown that the building requires a design cooling capacity of 134.31 kW with 6.51m3/s flow rate. The results are slightly lower than the actual designed value but are within the acceptable range. On the other hand, the supply air temperature of the proposed system is fixed at 18.9°C with 0.0081 g/g humidity ratio based on the preliminary calculations in the design concept chapter. While the mechanical ventilation is turned off since 30% of fresh air has already been mixed with the supply air. Due to the mechanical ventilation being turned off, the design cooling capacity drops slightly to 84.24 kW, and the Design flowrate is reduced to 4.42 m^3/s. Considering the 'Desigrated' system design flow rate at 0.42m^3/s, 11 systems are needed for cooling the 19th floor.

6.3 System Application and Performances

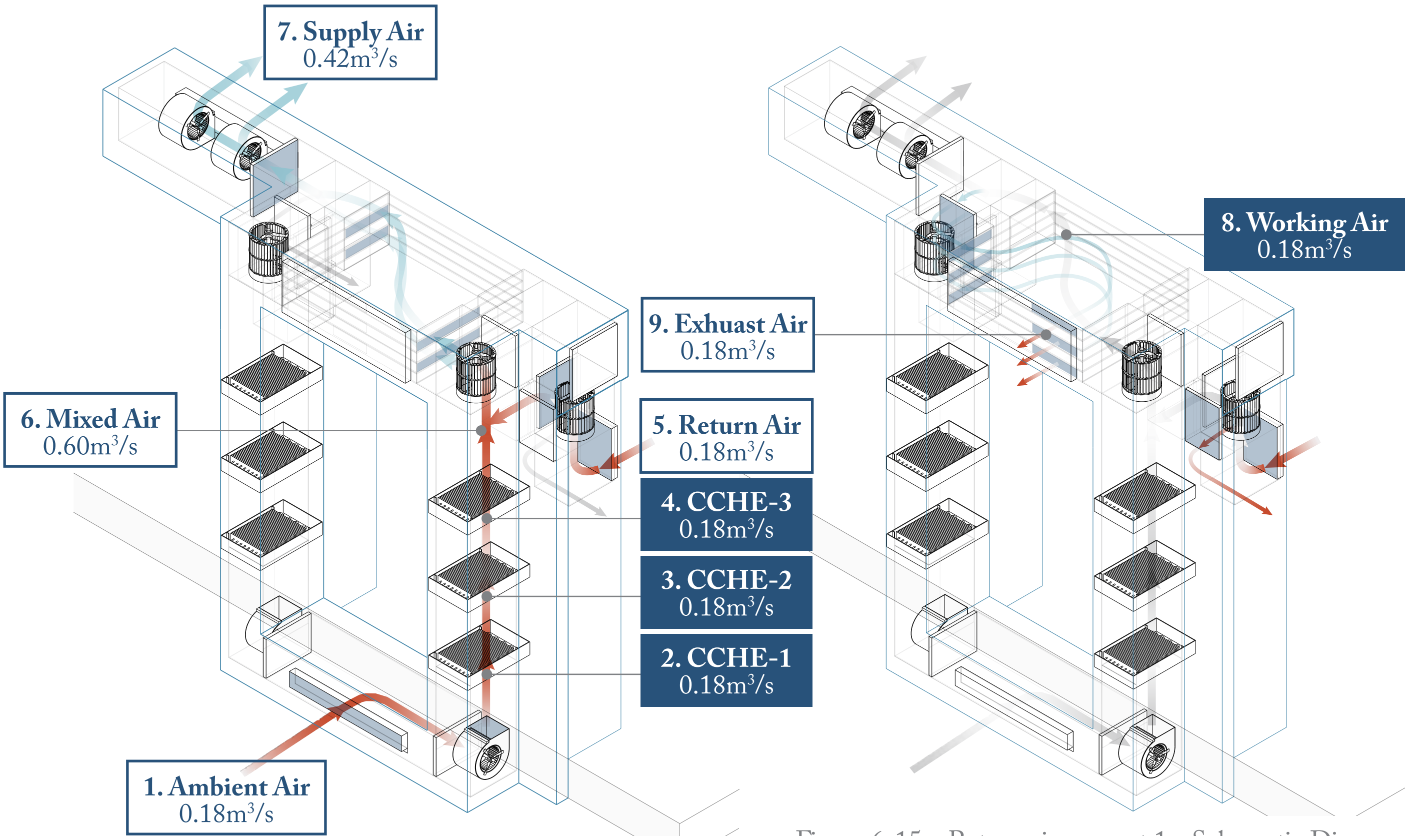


Figure 6-15 : Return air concept 1 - Schematic Diagram

Table 6-9 : Moisture from Occupants

$\Delta C = P/n^*v$

0.37 g kg
Additional Moisture

		P	n	V
Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m³)
40	70	2800	3.12	2430

Table 6-10 : Return Air

$(T_{supply} - T_{return}) = Q/V\ C\rho$

28.3 ˚C
Return Temperature
Abs.Humidity : 8.5 g kg | RH : 35.5%

T_{supply}	Q	V
Supply Air Temperature (˚C)	Simulated Cooling Capacity (kW)	Volume Flowrate (m³/s)
18.9	87.21	4.92

Table 6-11 : Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$ $T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$

31.0 ˚C
Return Temperature
Abs.Humidity : 8.4 g kg | RH : 30.0%

Air Volume		Air Temperature		Humidity Ration	
Q_{in}	Q_{re}	T_{in}	T_{re}	X_{in}	X_{re}
Inlet Air (m³/s)	Return Air (m³/s)	Inlet Air (˚C)	Return Air (˚C)	Inlet Air (kg kg)	Return Air (kg kg)
0.18	0.42	37.3	30.5	8.1	8.5
30%	70%				

Table 6-12 : Supply Air

$T_{out} = 7.65 + 0.152\ T_{in} + 681\ w$

18.1 ˚C
Supply Temperature
Abs.Humidity : 8.4 g kg | RH : 65.0%

From Mixed Air			
Air Temperature (˚C)	Relative Humidity (%)	Absolute Humidity (g kg)	Wetbulb Temperature (˚C)
31.0	30.0	8.4	18.7

Figure 6-16 : Return air concept 1 Psychometric Diagram

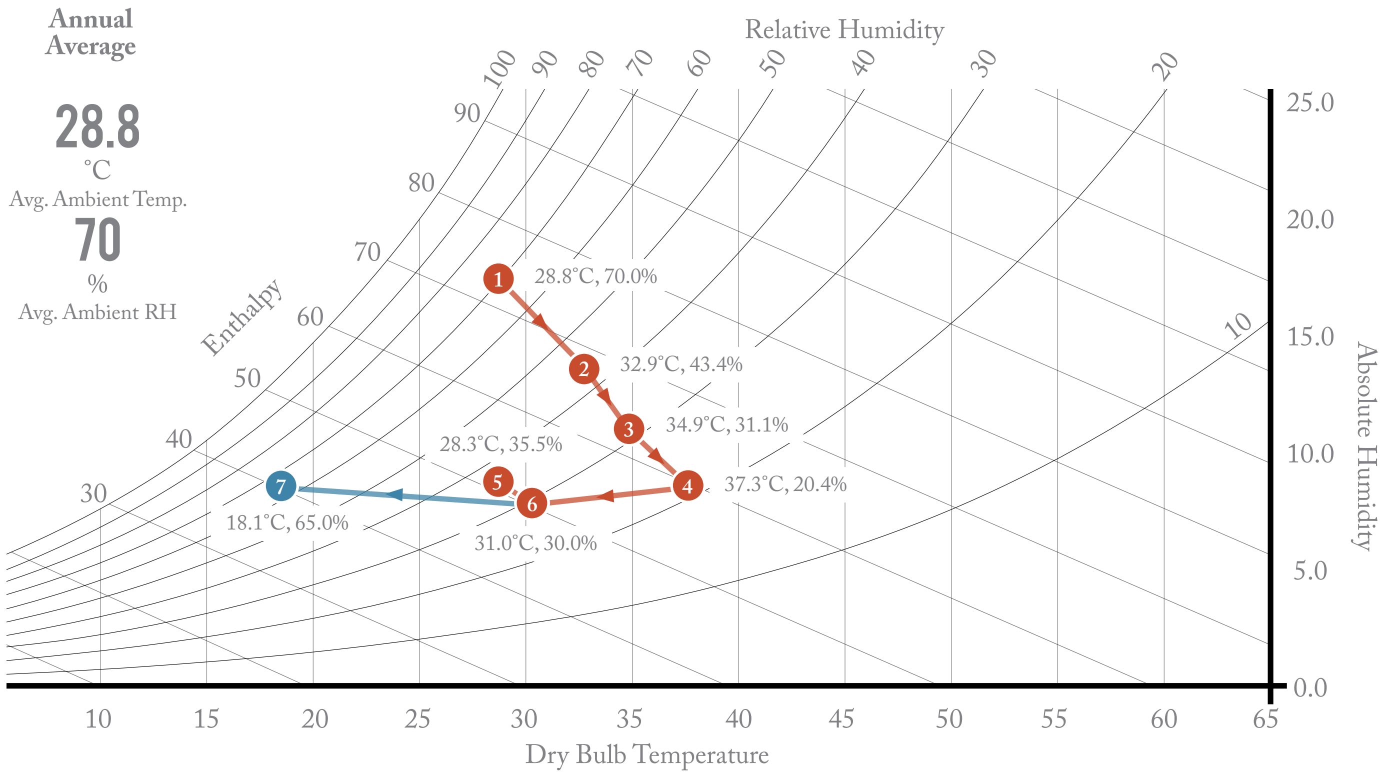


Table 6-13 : Concept 1 - Processed air condition

	Air Condition		
	Air Temperature (°C)	Relative Humidity (%)	Humidity Ratio (g kg)
1. Ambient Air	28.8	70	17.5
2. CCHE-1	32.9	43.4	13.6
3. CCHE-2	34.9	31.1	10.9
4. CCHE-3	37.3	20.4	8.1
5. Return Air	28.3	35.5	8.5
6. Mixed Air	31.0	30.0	8.4
7. Supply Air	**18.1**	**65.0**	**8.4**

6.3.1 Return Air Concept 1

Two concepts have been proposed to elaborate on the impact of the position of the mixed air chamber. The return air is calculated under the assumption that the room is occupied by 70 people but un-conditioned. The first supply air is considered at 18.9°C based on the calculation during the design concept chapter. Under the criteria, the return air is assumed to be at 28.3°C with 8.5 g kg humidity ratio. In the first concept, the ambient air is processed by three series of CCHEs before being mixed with the return air from the room. The mixed air is then processed by the M-Cycle system and supply to the room as conditioned air. This results in a supply air temperature of 18.1°C. While the supply air temperature reduces slightly by 0.8°C, the relative humidity ratio rises to 65%, exceeding the comfortable relative humidity ratio by 10%.

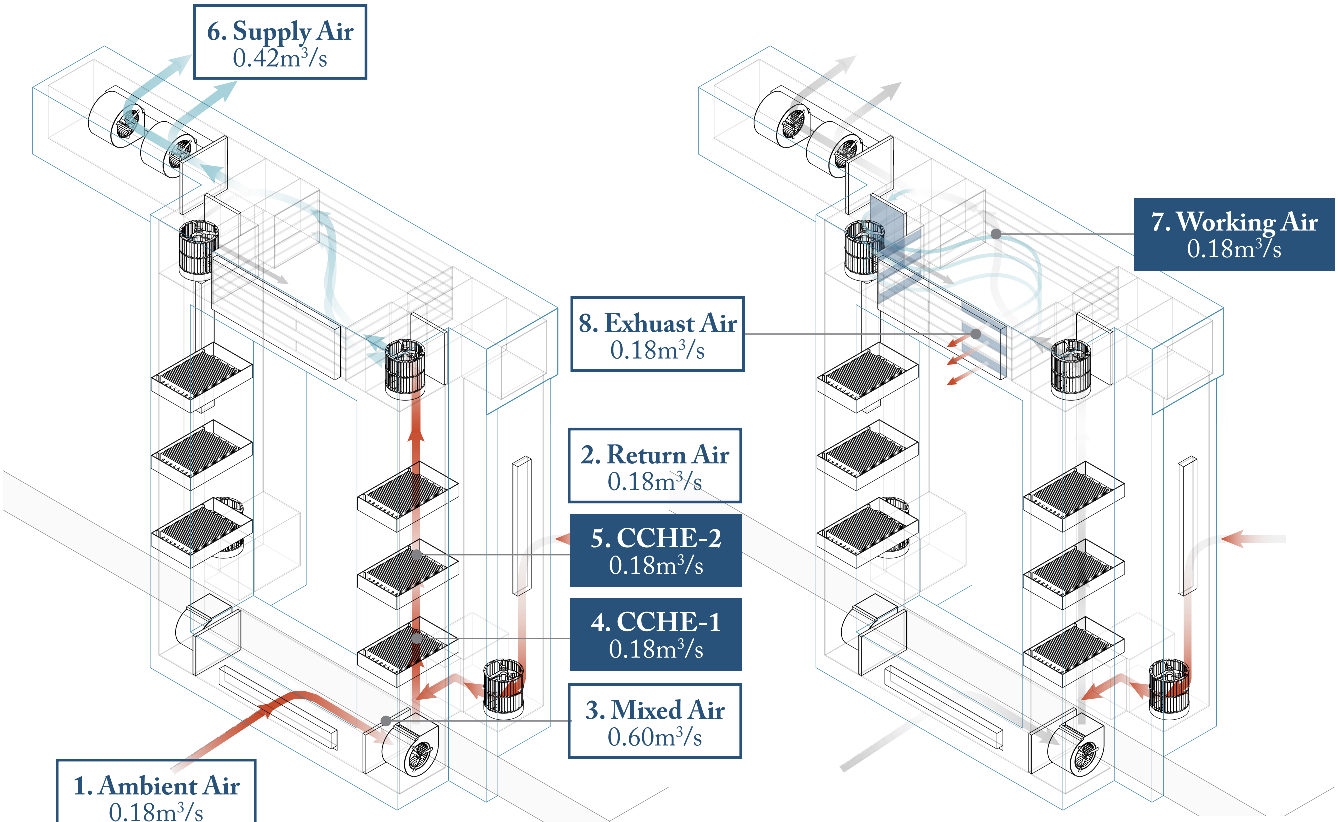


Figure 6-17 : Return air concept 2 - Schematic Diagram

Table 6-14 : Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in}+Q_{re})$ $T_{mixed}=(Q_{in}T_{in}+Q_{re}T_{re})/Q_{in}+Q_{re}$

28.5
°C
Return Temperature
Abs.Humidity : 11.2 g kg | RH : 46.0%

Air Volume		Air Temperature		Humidity Ratio	
Q_{in} Inlet Air (m³/s)	Q_{re} Return Air (m³/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)
0.18	0.42	28.8	28.3	17.5	8.5
30%	70%				

Table 6-15 : CCHE Performance

$Q = M\,(Ha_{in} - Ha_{out})$ M = 0.22 kg/s

	Inlet Air				Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{out}) (kJ kg)	Abs. Humidity (g kg)
1	28.5	46.0	57.3	11.2	**30.7**	32.4	53.6	8.9
2	30.7	32.4	53.6	8.9	**32.8**	21.4	49.9	6.6

Table 6-16 : Supply Air

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

17.1
°C
Supply Temperature
Abs.Humidity : 6.6 g kg | RH : 54.6%

From Mixed Air		
Air Temperature (°C)	Relative Humidity (%)	Absolute Humidity (g kg)
32.8	21.4	6.6

Figure 6-18 : Return air concept 2 Psychometric Diagram

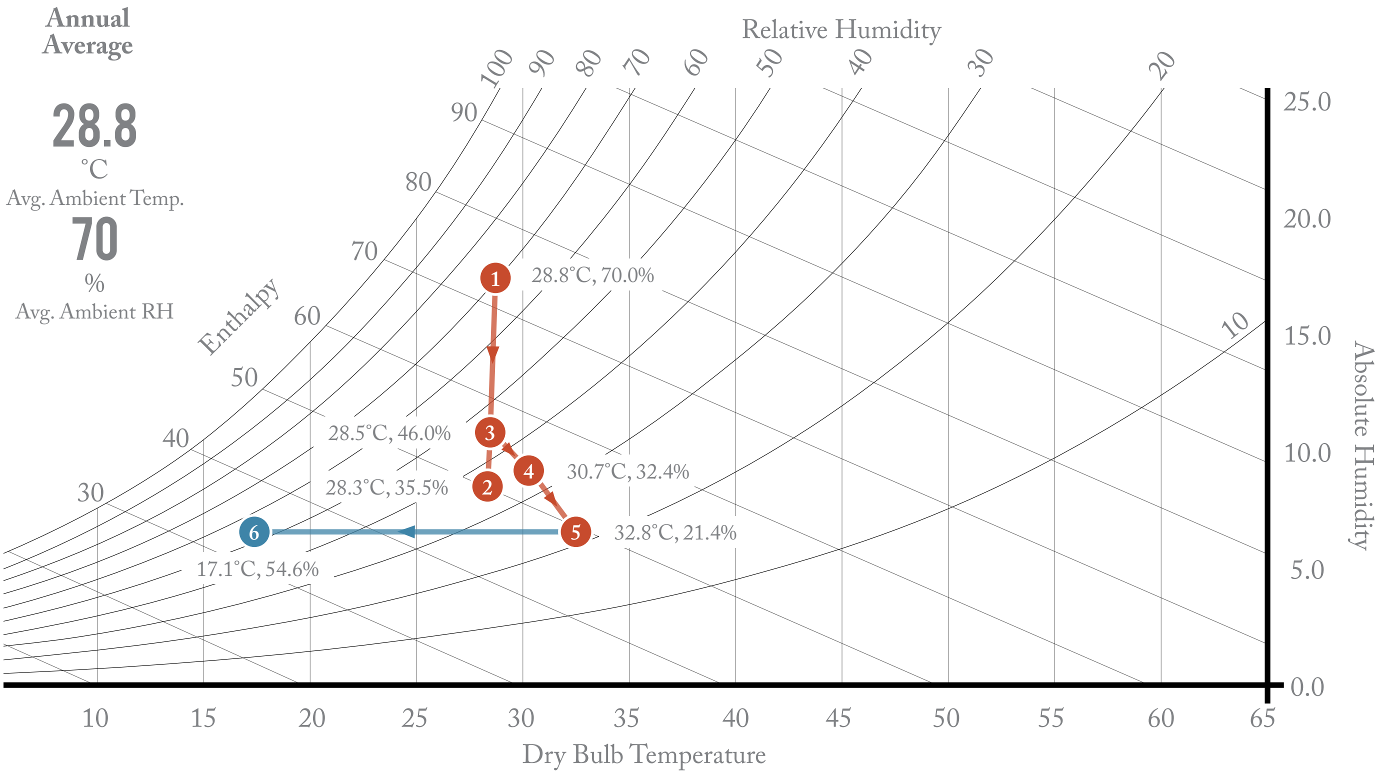


Table 6-17 : Concept 2 - Processed air condition

	Air Condition		
	Air Temperature (°C)	Relative Humidity (%)	Humidity Ratio (g kg)
1. Ambient Air	28.8	70	17.5
2. Return Air	28.3	35.5	8.5
3. Mixed Air	28.5	46.0	9.8
4. CCHE-1	30.7	32.4	8.9
5. CCHE-2	32.8	21.4	6.6
6. Supply Air	**17.1**	**54.6**	**6.6**

6.3.2 Return Air Concept 2

The assumption used in calculating the supply air in Concept 1 is also applied in the calculation of Concept 2. However, in this concept, the mix air chamber is positioned before the desiccant systems. Therefore, the ambient air and the return air will be mix before it gets dehumidified. After the dehumidification process, the process air follows the same routine of entering the M-Cycle system before being supply into the room. With this configuration, only 2 CCHEs is required to dehumidify the incoming air to an acceptable level of relative humidity below 55%. In comparison to the first concept, the second concept shows higher efficiency in cooling down the inlet air, reducing the temperature to 17.1°C, with a relative humidity level of 54.6%. Due to the potential in reducing the number of CCHEs required and higher cooling capacity, the system configuration will be based on the schematic propose in Concept 2.

Part III

Integration | Design Implementations

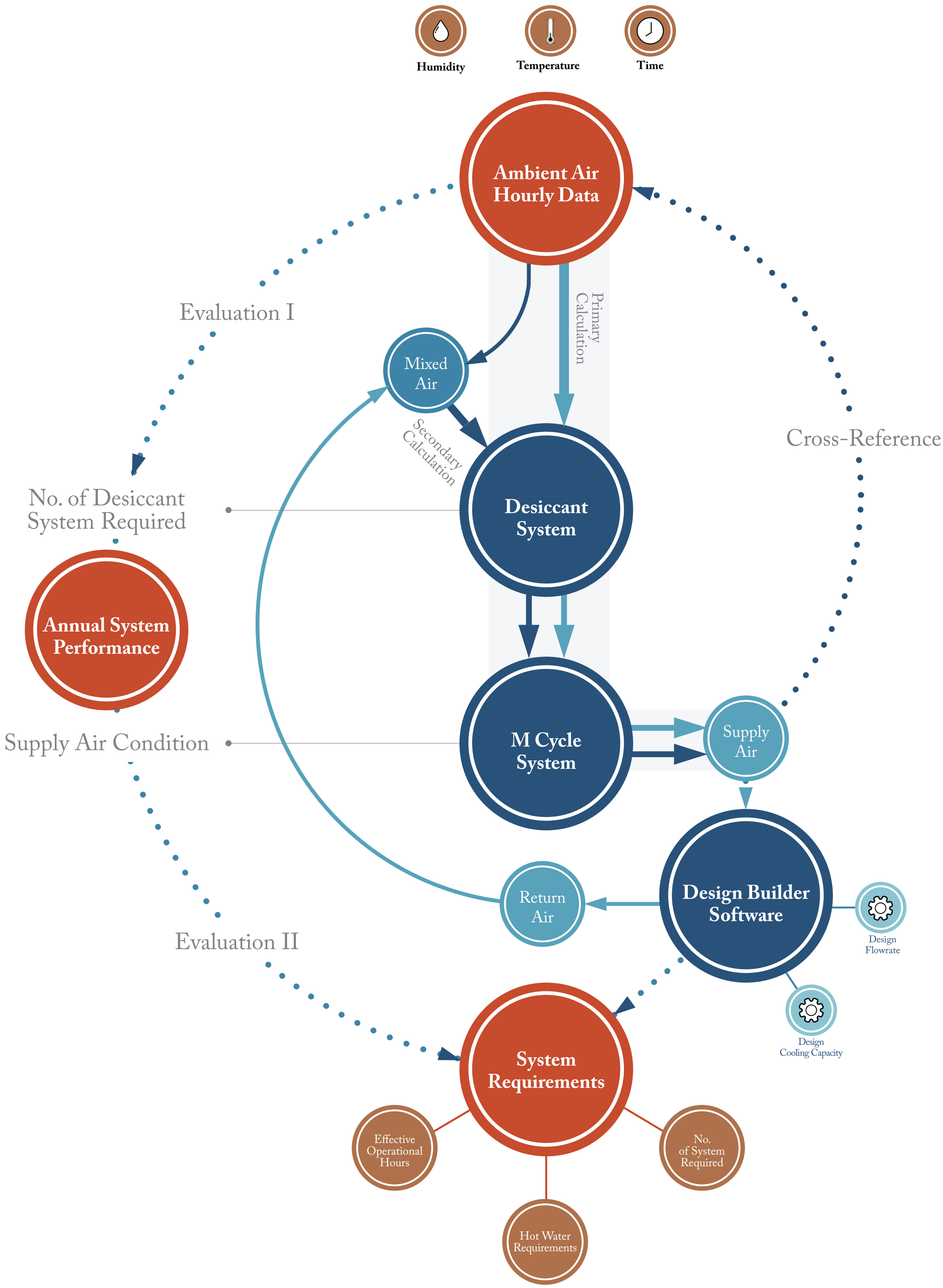


Figure 7-1 : Supply air calculation strategy

7 System Evaluation

While the preliminary calculation has proven that the proposed system is operable under the average ambient condition of Bangkok, further investigation has been conducted to elaborate the extent of its performance based on the variation of the ambient conditions. The strategy used in evaluating the performance of the proposed system is depicted in Figure 7-1. Hourly weather data of Bangkok between 8.00 – 18.00 was extracted and categorized into 25 categories based on the temperature and humidity levels. The data are used as input ambient air condition for the system. The evaluation is based on two cycles of calculations.

Under the assumption that the simulated room is unconditioned, the primary calculation is used to initiate the process. The supply air condition of the primary calculation is based on Bangkok's ambient air condition. Based on the equations extracted from literature reviews, the supply air condition is being calculated as the ambient air is being dehumidified by 3 CCHEs before being process by the M-cycle cooling system to achieve the optimal air condition. Through the means of Design Builder simulation, the supply air condition is also used to determine the cooling loads and volume flow rates required by the room to set up an operative temperature of 25.5°C. Hence, the return air condition could be determined.

The return air is being mixed with ambient air with a 70:30 ratio as proposed in the design concept phase and used as the supply air in the secondary calculation. The secondary calculation is conducted to assess the supply air condition from the mixture of air from the conditioned room and the ambient air, while also determining the effective number of CCHEs required. The process undergoes a similar calculation cycle. However, the supply air condition is being determined after every CCHE dehumidification process to evaluate the optimal number of CCHE required. The supply air condition is considered valid when its relative humidity level is under 55.5%, as previously discussed. The supply air condition is then cross-referenced with the hourly weather data of Bangkok to evaluate the effectiveness of the system considering the number of operable hours. Finally, the supply air conditions are used to determine the cooling loads and volume flow rates to be used as the design parameters for the design development process.

Table 7-1 : Hourly Ambient Air Data based on 8.00 - 18.00

Humidity Range	<22.5°C	25°C ≥22.5 - < 27.5 (°C)	30°C ≥27.5 - < 32.5 (°C)	35°C ≥32.5 - ≤37.5 (°C)	>37.5°C	
	Temperature Range					
>75%	6	300	329	-	-	635
70% ≥65 - ≤75%	12	99	806	13	-	918
60% ≥55 - <65%	22	72	750	374	-	1196
50% ≥45 - <55%	7	55	307	537	-	899
<45%	-	29	136	154	7	319
	6	555	2328	1078	7	

7.1 Categorization of Hourly Ambient Data

Hourly ambient data of Bangkok between 8.00-18.00 has been extracted based on the daily operating hours of the Krungthai Bank - Sukhumvit Building. 4015 hours of data are categorized into 25 categories based on the temperature and relative humidity level. The categories are in reference to the parameters used in determining the cooling capacity and dehumidification performance of the desiccant system in an experiment conducted by Ge et al., 2017 [Appendix D]. Due to the limited test results, some values are based on the proportional variation based on the experiment results.

The temperature under the three main temperature categories (25,30,35) are grouped based on a ± 2.5°C differences, while the three main temperature categories (50,60,70) are grouped based on a ±5% differences. From Table 7-1, it is evident that the majority of the ambient temperature falls under the 30°C/70%RH, which corresponded with the average temperature parameter used in the preliminary calculations.

7.2 Primary Calculation

Figure 7-2 : Primary calculations -Supply air temperature | Unconditioned Room

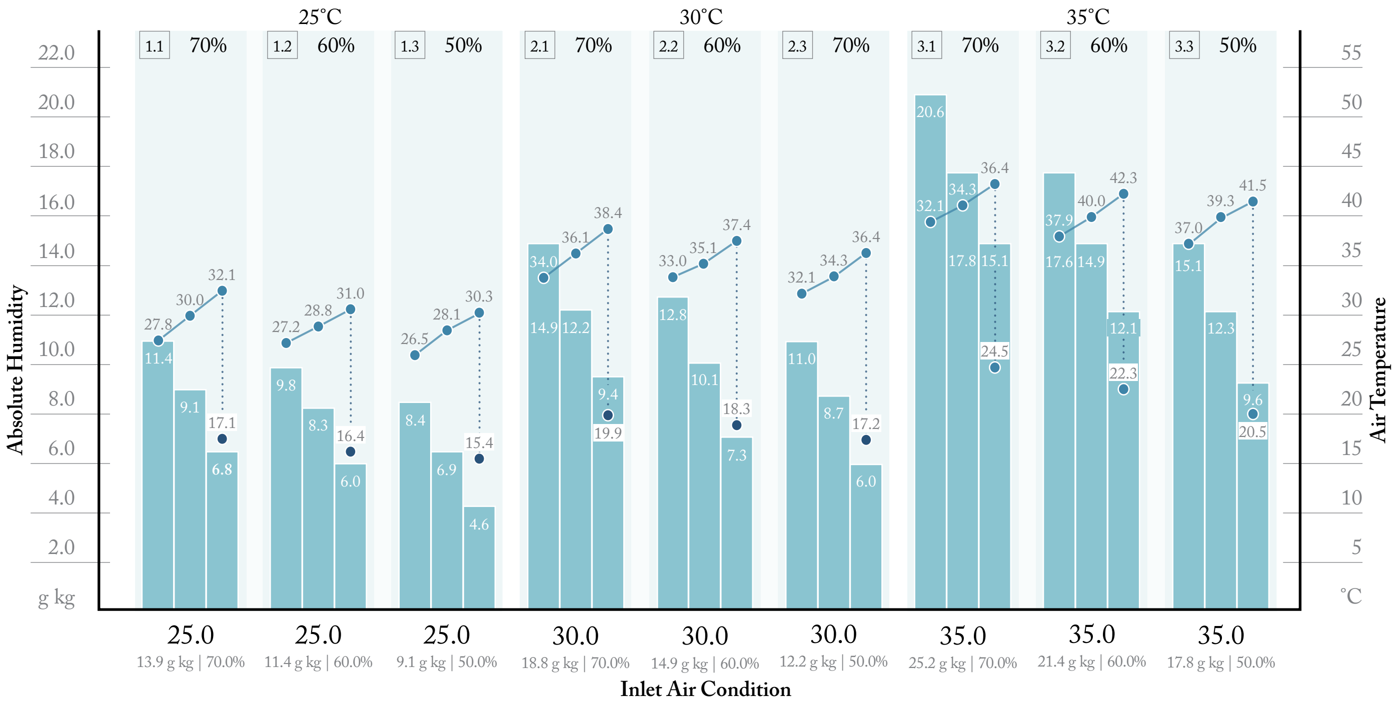


Table 7-2 : Supply Air Condition

	Supply Temperature (°C)	Dehumidification: Process by CCHE-1 (°C)	Process by CCHE-2 (°C)	Process by CCHE-3 (°C)	M-Cycle Cooling: From CCHE-1 (°C)	From CCHE-2 (°C)	From CCHE-3 (°C)
1.1	25.0 13.9 g kg | 70.0%	27.8°C 11.4 g kg | 48.7%	30.0°C 9.1 g kg |34.5%	32.1°C 6.8 g kg | 22.9%	19.6°C 11.4 g kg | 82.0%	18.4°C 9.1 g kg |69.0%	17.1°C 6.8 g kg | 56.2%
1.2	25.0 9.1 g kg | 50.0%	27.2°C 9.8 g kg | 43.5%	28.8°C 8.3 g kg | 33.6%	31.0°C 6.0 g kg | 21.6%	18.5°C 9.8 g kg | 73.8%	17.7°C 8.3 g kg | 65.9%	16.4°C 6.0 g kg | 51.9%
1.3	25.0 9.1 g kg | 50.0%	26.5°C 8.4 g kg | 38.9%	28.1°C 6.9 g kg | 29.2%	30.3°C 4.6 g kg | 17.3%	17.4°C 8.4 g kg | 67.9%	16.6°C 6.9 g kg | 58.8%	15.4°C 4.6 g kg | 42.5%
2.1	30.0 18.8 g kg | 70.0%	34.0°C 14.9 g kg | 44.5%	36.1°C 12.2 g kg |32.7%	38.4°C 9.4 g kg | 22.3%	23.0°C 14.9 g kg | 84.3%	21.4°C 12.2 g kg | 76.5%	19.9°C 9.4 g kg | 64.9%
2.2	30.0 14.9 g kg | 60.0%	33.0°C 12.8 g kg | 40.6%	35.1°C 10.1 g kg | 28.6%	37.4°C 7.3 g kg | 18.3%	21.4°C 12.8 g kg | 73.8%	19.8°C 10.1 g kg | 70.1%	18.3°C 7.3 g kg | 55.9%
2.3	30.0 12.2 g kg | 50.0%	32.1°C 11.0 g kg | 36.7%	34.3°C 8.7 g kg | 25.8%	36.4°C 6.0 g kg | 15.9%	20.0°C 11.0 g kg | 67.9%	18.8°C 8.7 g kg | 64.4%	17.2°C 6.0 g kg | 49.3%
3.1	35.0 25.2 g kg | 70.0%	39.0°C 20.6 g kg | 46.5%	41.3°C 17.8 g kg | 35.7%	43.5°C 15.1 g kg | 27.1%	27.6°C 20.6 g kg | 87.9%	26.1°C 17.8 g kg | 83.3%	24.5°C 15.1 g kg | 78.1%
3.2	35.0 21.4 g kg | 60.0%	37.9°C 17.6 g kg | 42.2%	40.0°C 14.9 g kg | 32.1%	42.3°C 12.1 g kg | 23.1%	25.4°C 17.6 g kg | 85.9%	23.9°C 14.9 g kg | 79.9%	22.3°C 12.1 g kg | 68.4%
3.3	35.0 17.8 g kg | 50.0%	37.0°C 15.1 g kg | 38.3%	39.3°C 12.3 g kg | 27.6%	41.5°C 9.6 g kg | 19.3%	23.5°C 15.1 g kg | 82.9%	22..0°C 12.3 g kg | 74.3%	20.5°C 9.6 g kg | 63.8%

7.2.1 Supply Air Condition

To initiate the process, the primary calculation is conducted. The variation of Bangkok's ambient temperature is being used as the inlet temperature to evaluate the system's supply temperature. From Figure 7-2, it is evident that the air temperature gradually increases, while the absolute humidity level decreased as the CCHE is processing the air. Furthermore, it could be concluded that the M-cycle performed more effectively as the air is dehumidified. The results of the calculations had shown that a lower temperature supply air could be achieved from the inlet air with relatively more moderate temperature and humidity. With the ambient air condition of Bangkok, the supply air temperature ranges between 15.4°C to 24.5°C.

Figure 7-3 : Primary calculations -Cooling Load/Volume Flowrate | Unconditioned Room

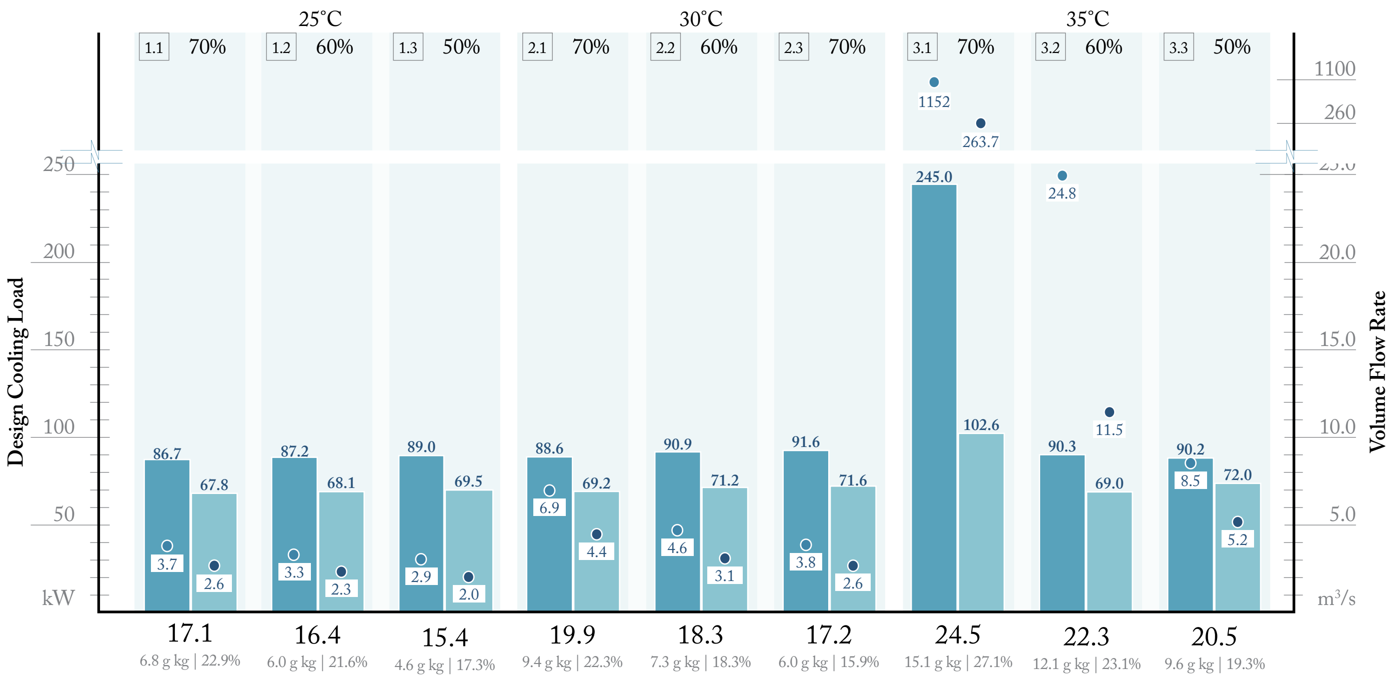


Table 7-3 : Cooling Load | Volume Flow Rate

	Supply Temperature (°C)	Existing Condition: Design Cooling Load (kW)	Existing Condition: Volume Flowrate (m³/s)	Existing Condition: Air Change per Hour (ach)	Integrating Condition: Design Cooling Load (kW)	Integrating Condition: Volume Flowrate (m³/s)	Integrating Condition: Air Change per Hour (ach)
1.1	17.1 6.8 g kg | 22.9%	86.7	3.7	3.68	67.8	2.6	1.62
1.2	16.4 6.0 g kg | 21.6%	87.2	3.3	2.92	68.1	2.3	1.46
1.3	15.4 4.6 g kg | 17.3%	89.0	2.9	2.60	69.5	2.0	1.28
2.1	19.9 9.4 g kg | 22.3%	88.6	6.9	5.74	69.2	4.4	2.79
2.2	18.3 7.3 g kg | 18.3%	90.9	4.6	4.00	71.2	3.1	1.97
2.3	17.2 6.0 g kg | 15.9%	91.6	3.8	3.30	71.6	2.6	1.65
3.1	24.5 15.1 g kg | 27.1%	245	1152.0	731.13	102.6	263.7	167.36
3.2	22.3 12.1 g kg | 23.1%	90.3	24.8	15.74	69.0	11.5	7.30
3.3	20.5 9.6 g kg | 19.3%	92.2	8.5	5.39	72.0	5.2	3.30

7.2.2 Cooling Load | Volume Flowrate Simulation

Design Builder Software has been used to simulate the cooling load and volume flow rate demand of the designated floor of the Krungthai-Sukhumvit Building. Both the existing condition of the building and the proposed façade design has been simulated. The two scenarios are mapped against each other in Table 7-3. The results have shown that the proposed scenarios have lowered the cooling load of the room by an average of 19.7 kW, except for the irregularity trend which occurs at scenario 3.1 due to the relatively high supply temperature. At the same time, the integration of the proposed scenarios has presented a similar pattern regarding the volume flow rate requirements. An average decreasing trend of 1.4m3/s could be concluded from the first two scenarios. However, due to the relatively high supply temperature, irregular trends also occur in scenario 3 in which the decreasing rate varies from 888 – 3.3m3/s.

Figure 7-4 : Primary calculations -Return /Mixed air temperature | Unconditioned Room

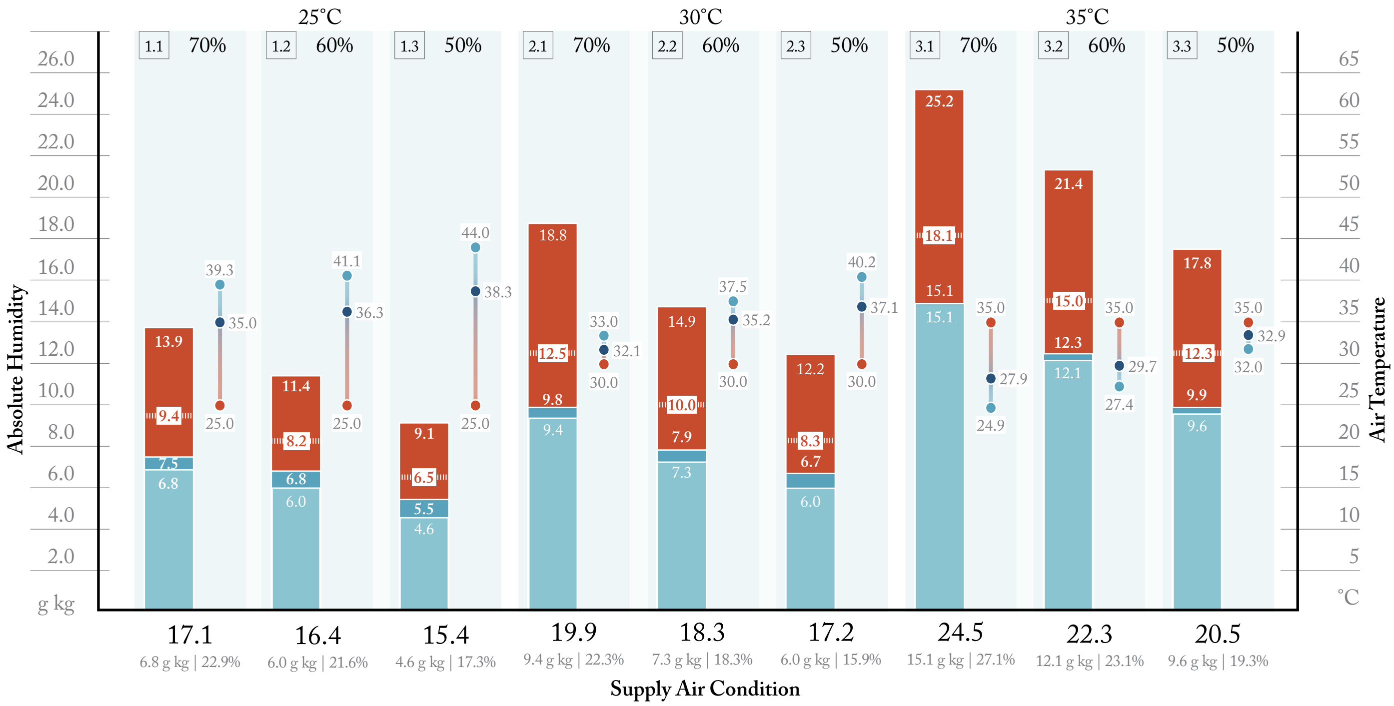


Table 7-4 : Return air | Mixed air condition

		Additional Humidity		Return Air Condition		Mixed Air Condition	
	Supply Temperature (°C)	Humidity Difference (g kg)	Absolute Humidity (g kg)	Temperature Difference (°C)	Return Temperature (°C)	Ambient Temperature (°C)	Mixed Air Result (°C)
1.1	17.1 6.8 g kg | 22.9%	+0.71	7.5	+22.2	39.3	25.0 13.9 g kg | 70.0%	35.0 9.4 g kg | 26.8%
1.2	16.4 6.0 g kg | 21.6%	+0.78	6.8	+24.7	41.1	25.0 11.4 g kg | 60.0%	36.3 8.2 g kg | 21.8%
1.3	15.4 4.6 g kg | 17.3%	+0.89	5.5	+28.7	44.0	25.0 9.1 g kg | 50.0%	38.3 6.5 g kg | 15.5%
2.1	19.9 9.4 g kg | 22.3%	+0.41	9.8	+13.1	33.0	30.0 18.8 g kg | 70.0%	32.1 12.5 g kg | 41.7%
2.2	18.3 7.3 g kg | 18.3%	+0.58	7.9	+19.1	37.5	30.0 14.9 g kg | 60.0%	35.2 10.0 g kg | 28.2%
2.3	17.2 6.0 g kg | 15.9%	+0.70	6.7	+23.0	40.2	30.0 12.2 g kg | 50.0%	37.1 8.3 g kg | 21.1%
3.1	24.5 15.1 g kg | 27.1%	+0.007	15.1	+0.3	24.9	35.0 25.2 g kg | 70.0%	27.9 18.1 g kg | 76.2%
3.2	22.3 12.1 g kg | 23.1%	+0.20	12.3	+5.0	27.4	35.0 21.4 g kg | 60.0%	29.7 15.0 g kg | 57.2%
3.3	20.5 9.6 g kg | 19.3%	+0.30	9.9	+11.5	32.0	35.0 17.8 g kg | 50.0%	32.9 12.3 g kg | 39.2%

7.2.3 Return air and Mixed Air Condition

The cooling load derived from the simulation is used in calculating the room's air change rate to determine the temperature increase of the return air. The volume flow rate is used in calculating the room's air change rate to assess the additional moisture from the room. The return air is then mixed with the ambient inlet air with a 70:30 ratio, as discussed in the conceptual design phase. The calculation has depicted a trend in which the return air's temperature tends to be relatively lower when the supply air temperature is higher. At the same time, a relatively lower volume flow rate would result in a higher additional moisture rate.

7.3 Secondary Calculation

Figure 7-5 : Secondary calculations -Supply air temperature | Conditioned Room

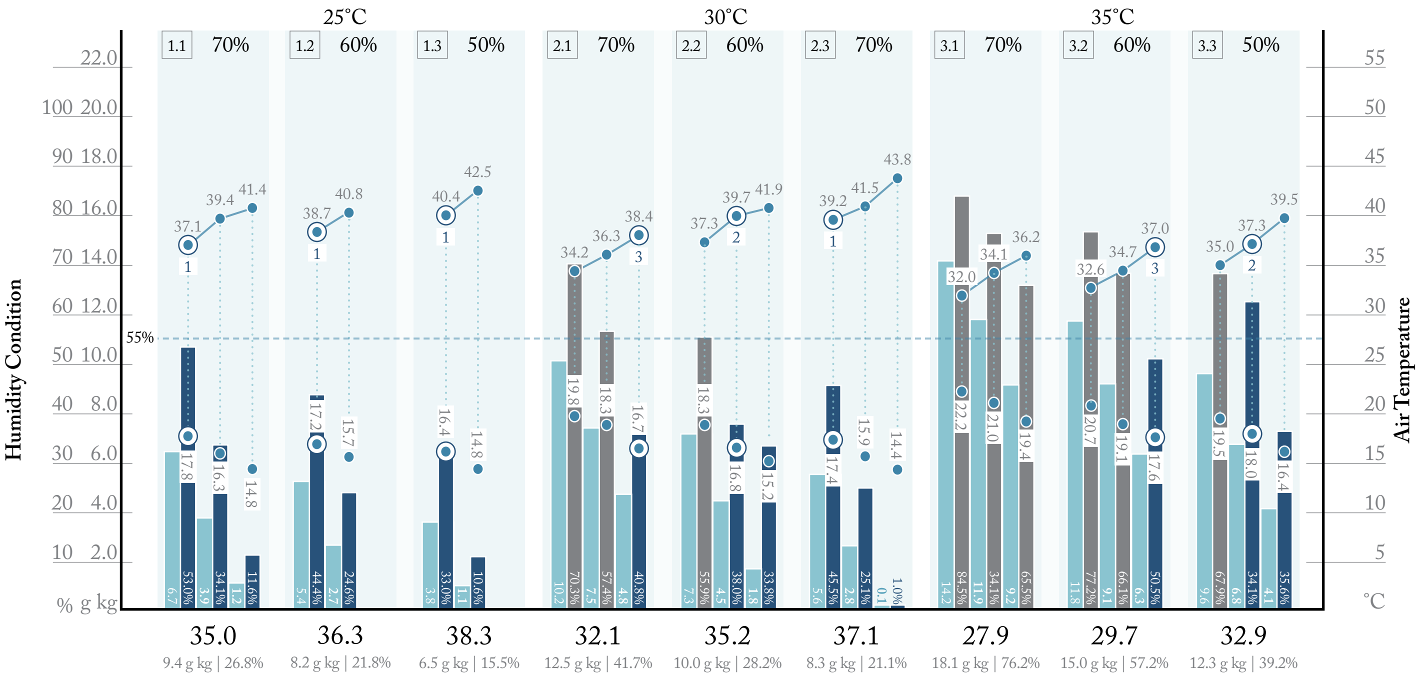


Table 7-5 : Supply Air Condition

	Supply Temperature (°C)	Dehumidification: Process by CCHE-1 (°C)	Process by CCHE-2 (°C)	Process by CCHE-3 (°C)	M-Cycle Cooling: From CCHE-1 (°C)	From CCHE-2 (°C)	From CCHE-3 (°C)
1.1	35.0 9.4 g kg | 26.8%	37.1°C 6.7 g kg |17.1%	39.4°C 3.9 g kg |8.8%	41.7°C 1.2 g kg | 2.4%	17.8°C 6.7 g kg |53.0%	16.3°C 3.9 g kg |34.1%	14.8°C 1.2 g kg | 11.6%
1.2	36.3 8.2 g kg | 21.8%	38.7°C 5.4 g kg | 12.7%	40.8°C 2.7 g kg | 5.7%	-	17.2°C 5.4 g kg | 44.4%	15.7°C 2.7 g kg | 24.6%	-
1.3	38.3 6.5 g kg | 15.5%	40.4°C 3.8 g kg | 8.2%	42.5°C 1.1 g kg | 2.1%	-	16.4°C 3.8 g kg | 33.0%	14.8°C 1.1 g kg | 10.6%	-
2.1	32.1 12.5 g kg | 41.7%	34.2°C 10.2 g kg |30.3%	36.3°C 7.5 g kg |19.9%	38.4°C 4.8 g kg | 11.4%	19.8°C 10.2 g kg |70.3%	18.3°C 7.5 g kg |57.4%	16.7°C 4.8 g kg | 40.8%
2.2	35.2 10.0 g kg | 28.2%	37.3°C 7.3 g kg | 18.4%	39.7°C 4.5 g kg | 10.0%	41.9°C 1.8 g kg | 0.2%	18.3°C 7.3 g kg | 55.9%	16.8°C 4.5 g kg | 38.0%	15.2°C 1.8 g kg | 33.8%
2.3	37.1 8.3 g kg | 21.1%	39.2°C 5.6 g kg | 12.8%	41.5°C 2.8 g kg | 5.7%	43.8°C 0.1 g kg | 15.9%	17.4°C 5.6 g kg | 45.5%	15.9°C 2.8 g kg | 25.1%	14.4°C 0.1 g kg | 1.0%
3.1	27.9 18.1 g kg | 76.2%	32.0°C 14.2 g kg | 47.6%	34.1°C 11.9 g kg | 35.6%	36.2°C 9.2 g kg | 24.6%	22.2°C 14.2 g kg | 84.5%	21.0°C 11.9 g kg | 76.5%	19.4°C 9.2 g kg | 65.5%
3.2	29.7 15.0 g kg | 57.2%	32.6°C 11.8 g kg | 38.4%	34.7°C 9.1 g kg | 26.5%	37.0°C 6.3 g kg | 16.2%	20.7°C 11.8 g kg | 77.2%	19.1°C 9.1 g kg | 66.1%	17.6°C 6.3 g kg | 50.5%
3.3	32.9 12.3 g kg | 39.2%	35.0°C 9.6 g kg | 27.4%	37.3°C 6.8 g kg | 17.2%	39.5°C 4.1 g kg | 9.2%	19.5°C 9.6 g kg | 67.9%	18.0°C 6.8 g kg | 53.1%	16.4°C 4.1 g kg | 35.6%

7.3.1 Supply Air Condition

The secondary calculation is conducted to assess the adequate number of CCHEs required to process the mixed air to achieve a supply air with a relative humidity level under 55.5%. The results of the calculation have shown that the system is operable in all ambient condition except for scenario 3.1. Despite being process through 3 CCHE, the relative humidity levels is still relatively high at 65.5%. On the other hand, 4 scenarios could be satisfied with only 1 CCHE requires, 2 scenarios with 2 CCHE requires, and 2 scenarios with 3 CCHE requires respectively. It should also be noted that the variation between the highest supply temperature and the lowest supply temperature is at ±1.6°C from the temperature ranges of 16.4-18.0°C; therefore, allowing the supply air to be supplied at a relatively constant temperature.

Figure 7-6 : Secondary calculations -Cooling Load/Volume Flowrate | Conditioned Room

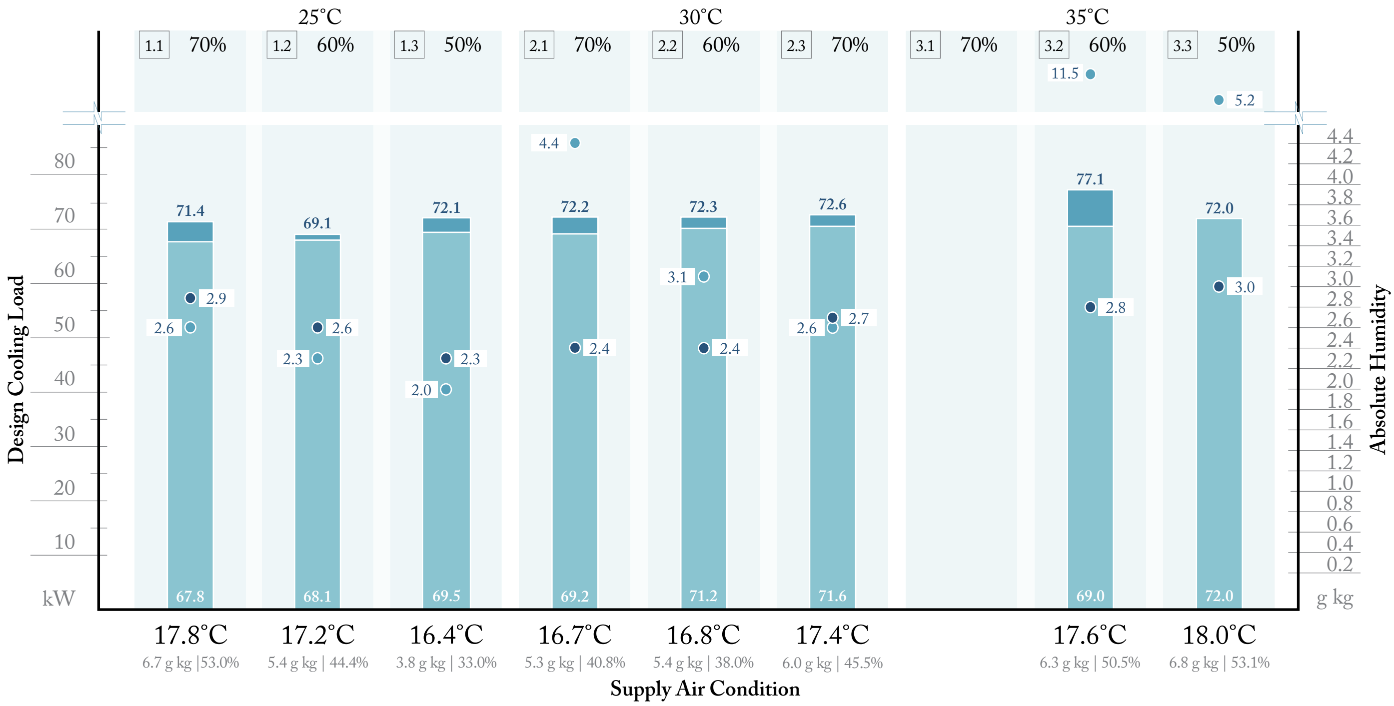


Table 7-6 : Cooling Load | Volume Flow Rate

	Secondary Condition				Primary Calculation			
	Supply Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m³/s)	Air Change per Hour (ach)	Supply Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m³/s)	Air Change per Hour (ach)
1.1	17.8°C 6.7 g kg |53.0%	71.4	2.9	1.84	17.1 6.8 g kg | 22.9%	67.8	2.6	1.62
1.2	17.2°C 5.4 g kg | 44.4%	69.1	2.6	1.65	16.4 6.0 g kg | 21.6%	68.1	2.3	1.46
1.3	16.4°C 3.8 g kg | 33.0%	72.1	2.3	1.46	15.4 4.6 g kg | 17.3%	69.5	2.0	1.28
2.1	16.7°C 5.3 g kg | 40.8%	72.2	2.4	1.52	19.9 9.4 g kg | 22.3%	69.2	4.4	2.79
2.2	16.8°C 5.4 g kg | 38.0%	72.3	2.4	1.52	18.3 7.3 g kg | 18.3%	71.2	3.1	1.97
2.3	17.4°C 6.0 g kg | 45.5%	72.6	2.7	1.71	17.2 6.0 g kg | 15.9%	71.6	2.6	1.65
3.1	-	-	-	-	24.5 15.1 g kg | 27.1%	102.6	263.7	167.36
3.2	17.6°C 6.3 g kg | 50.5%	77.1	2.8	1.78	22.3 12.1 g kg | 23.1%	69.0	11.5	7.30
3.3	18.0°C 6.8 g kg | 53.1%	72.0	3.0	1.90	20.5 9.6 g kg | 19.3%	72.0	5.2	3.30

7.3.2 Cooling Load | Volume Flowrate Simulation

The supply air condition is being used as an input parameter to re-evaluate the proposed façade's room condition. Due to the relatively constant supply temperature, the design cooling load requires has minimal differences, ranging from 69.1 – 77.1 kW. In comparison to the primary calculation, it could be concluded that the design cooling load corresponded relatively with the increasing supply temperature. Furthermore, a similar conclusion could be drawn regarding the volume flow rate, as it is relatively corresponding with the air's absolute humidity. In which the volume flow rate would decrease as the absolute humidity decreases. This resulted in a volume flow rate requirements between 2.3-3.0 m3/s. Due to the lower volume flowrate requirement, the air change rate also decreases; therefore, additional fresh air ventilation is required in cases where the air change rate falls below 1.75ach.

Figure 7-7 : Secondary calculations -Return air/Mixed air Condition | Conditioned Room

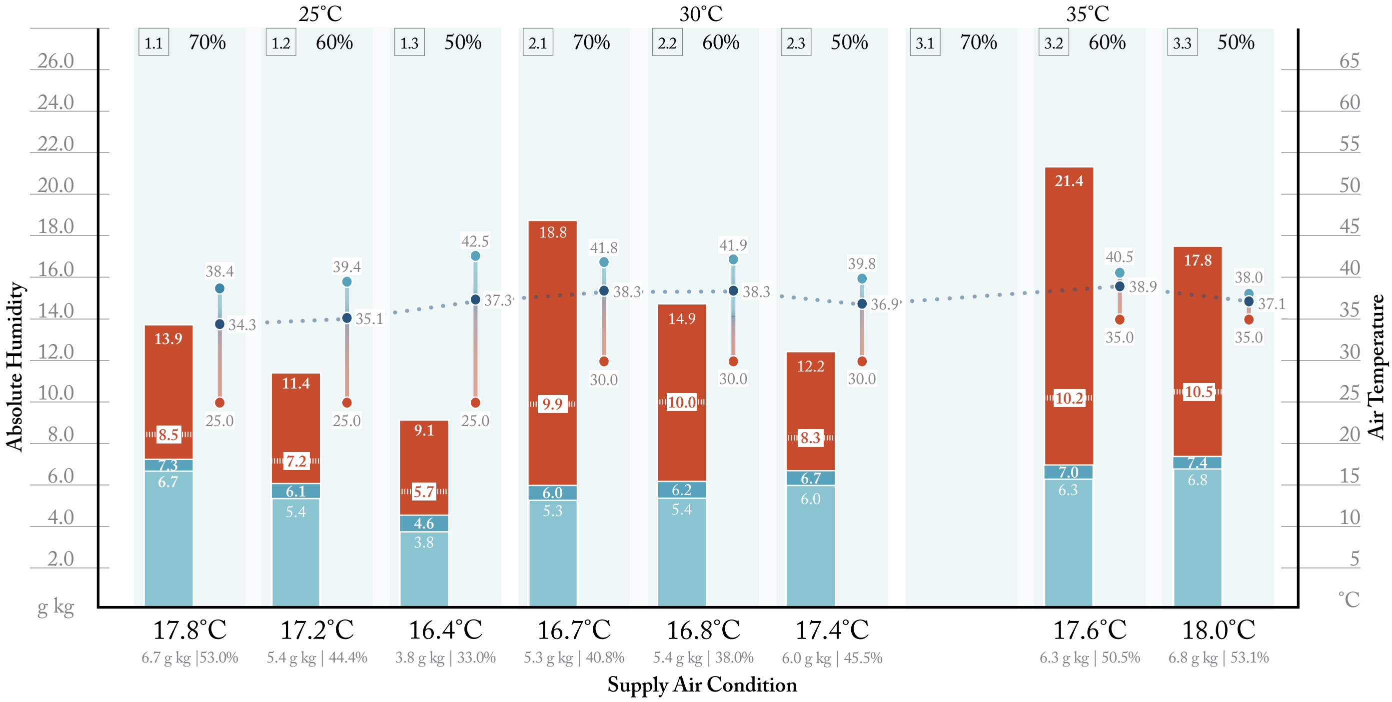


Table 7-7 : Return air | Mixed air condition

		Additional Humidity		Return Air Condition		Mixed Air Condition	
	Supply Temperature (°C)	Humidity Difference (g kg)	Absolute Humidity (g kg)	Temperature Difference (°C)	Return Temperature (°C)	Ambient Temperature (°C)	Mixed Air Result (°C)
1.1	17.8°C 6.7 g kg |53.0%	+0.63	7.3	+20.5	38.4	25.0 13.9 g kg | 70.0%	34.3 8.5 g kg | 25.2%
1.2	17.2°C 5.4 g kg | 44.4%	+0.70	6.1	+22.1	39.4	25.0 11.4 g kg | 60.0%	35.1 7.2 g kg | 20.5%
1.3	16.4°C 3.8 g kg | 33.0%	+0.79	4.6	+26.1	42.5	25.0 9.1 g kg | 50.0%	37.3 5.7 g kg | 14.4%
2.1	16.7°C 5.3 g kg | 40.8%	+0.76	6.0	+25.1	41.8	30.0 18.8 g kg | 70.0%	38.3 9.9 g kg | 23.6%
2.2	16.8°C 5.4 g kg | 38.0%	+0.76	6.2	+25.1	41.9	30.0 14.9 g kg | 60.0%	38.3 8.8 g kg | 21.0%
2.3	17.4°C 6.0 g kg | 45.5%	+0.67	6.7	+22.4	39.8	30.0 12.2 g kg | 50.0%	36.9 8.3 g kg | 21.4%
3.1	-	-	-	-	-	-	-
3.2	17.6°C 6.3 g kg | 50.5%	+0.65	7.0	+22.9	40.5	35.0 21.4 g kg | 60.0%	38.9 10.2 g kg | 23.5%
3.3	18.0°C 6.8 g kg | 53.1%	+0.60	7.4	+20.0	38.0	35.0 17.8 g kg | 50.0%	37.1 10.5 g kg | 26.6%

7.3.3 Return air and Mixed Air Condition

A similar procedure as the primary calculation has been conducted to calculate the return air and the mix air condition for the next cycle. Due to a more constant cooling load and volume flow rate requirements, the temperature variation of the return air reduces, as the return temperature ranges from 38.0 – 41.9°C. Therefore, the mixed air temperature also shows minimal variations, ranging from 34.3°C- 38.3°C. Moreover, it should be noted that, in comparison to the primary calculations, the mixed air temperature rose up noticeably, while the absolute humidity levels are reduced due to the supply air condition with lower temperature and humidity levels.

7.4 Performance Assessment

Table 7-8 : CCHE System Required

Ambient Humidity Range	25°C ≥22.5 - < 27.5 (°C)		30°C ≥27.5 - < 32.5 (°C)		35°C ≥32.5 - ≤37.5 (°C)	
70% ≥65 - ≤75%	1	17.8°C 6.1 g kg | 53.0%	3	17.2°C 5.3 g kg | 40.8%	3	16.4°C 9.2g kg | 65.5%
60% ≥55 - <65%	1	16.7°C 6.6 g kg | 44.4%	2	16.8°C 5.4 g kg | 38.0%	3	17.4°C 6.3 g kg | 50.5%
50% ≥45 - <55%	1	19.4°C 5.0 g kg | 33.0%	1	17.6°C 6.0 g kg | 45.5%	2	18.0°C 6.8 g kg | 53.1%

Ambient Temperature Range

Table 7-9 : Cross Reference with Hourly Ambient Data

Temperature Range

Humidity Range	<22.5°C		25°C ≥22.5 - < 27.5 (°C)		30°C ≥27.5 - < 32.5 (°C)		35°C ≥32.5 - ≤37.5 (°C)		>37.5°C	
>75%	-	6	-	300	-	329	-	-	-	-
70% ≥65 - ≤75%	-	12	2	99	3	806	-	13	-	-
60% ≥55 - <65%	-	22	1	72	2	750	3	374	-	-
50% ≥45 - <55%	-	7	1	55	1	307	2	537	-	-
<45%	-	-	1	29	1	136	2	154	-	7

No. of Desiccant System : **1**	No. of Desiccant System : **2**	No. of Desiccant System : **3**	Inoperable / Limited Info.
14.9%	**53.3%**	**82.7%**	**17.3%**
599 Operable Hours	**2139** Operable Hours	**3319** Operable Hours	**696** Operable Hours

7.4.1 Cross Referencing with Hourly Ambient Data

The results of the calculation have shown that the proposed system is operable under most ambient scenarios except for the ambient condition at 35°C|70% RH due to the exceeded relative humidity of the supply air at 65.5%. To evaluate the performance of the system, the number of the required desiccant system is cross-referenced with the ambient hourly data in Table 7-9. Due to limited experimental data regarding the performance of the desiccant system in some scenarios, the data are categorized under inoperable/ limited information. From the results, it could be assessed that with 1 CCHE, the system is effective 14.9% annually. However, due to the hot and humid ambient condition of Bangkok's climate, it requires additional dehumidification and cooling. Therefore, with an additional CCHE coupled, the effectiveness rate drastically increases by 40.4%.Moreover, since the majority of Bangkok's ambient condition falls under the 30°C | 70% RH category when 3 CCHEs is coupled the system effectiveness can satisfy 82.7% of the ambient condition annually. Based on the calculation, it could be further predicted that ambient condition with low temperature and humidity (highlighted in blue) could also be processed by the system. Under this assumption, with 3 CCHEs components, the system will be able to handle 347 additional hours, satisfying 91.3% of the ambient condition. Base on this assumption, it is recommended that the system should be equipped with 3 CCHEs to be able to handle Bangkok's condition all year round.

Table 7-10 : Water Requirement

	Cold Water (L/hr)			Hot Water (L/hr)		
M-Cycle	0.06			-		
	Case 1	Case 2	Case 3	Case 1	Case 2	Case 3
CCHE-1	60	60	60	60	60	60
CCHE-2	-	60	60	-	60	60
CCHE-3	-	-	60	-	-	60
	60	**120**	**180**	**60**	**120**	**180**
	$0.06m^3/hr$	$0.12m^3/hr$	$0.18m^3/hr$	$0.06m^3/hr$	$0.12m^3/hr$	$0.18m^3/hr$

Table 7-11 : Heating Requirement per CCHE

$Q = V' \, Cp \, (\Delta T)$

	V Volume Flowrate (m^3/hr)	V' Volume Flowrate (m^3/day)(11 Hours)	ΔT Temperature Difference (°C)	Q Energy Required (kWh/day)
Case 1	0.06	0.66	5.7	**4.4**
Case 2	0.12	1.32	5.7	**8.7**
Case 3	0.18	1.98	5.7	**13.1**

1 - CCHE	2 - CCHE	3 - CCHE
4.4	**8.7**	**13.1**
kW/day	kW/day	kW/day
Thermal Energy Requirements	Thermal Energy Requirements	Thermal Energy Requirements

Table 7-12 : Energy Assessment - PVT system Requirements

	Thermal Energy Requirements (kW/day)			
	N	**E**	**S**	**W**
Energy Production per m^2	1.34 kW/m^2	2.04 kW/m^2	2.39 kW/m^2	2.00 kW/m^2
Case 1 Area Required:	3.28 sq.m	2.16 sq.m	1.84 sq.m	2.20 sq.m
Case 2 Area Required:	6.49 sq.m	4.26 sq.m	3.64 sq.m	4.35 sq.m
Case 3 Area Required:	9.77 sq.m	6.42 sq.m	5.48 sq.m	6.55 sq.m

7.4.2 PVT System Thermal Energy Assessment

Hot water is one of the critical elements of the system, as it is required by the system for regeneration process. The requirements are based on three different scenarios depending on the number of CCHE required. The water flow rate is referenced to case 3 in the previous chapter, which accounted for 0.06m3/s volume flowrate. The calculations are generalized; as they are calculated based on a daily basis, not hourly. It could be concluded that in the worst-case scenario with 3CCHEs required, the system required 13.1 kW of thermal energy to satisfy the heating demand. With this data, the parameters for designing solar collector could be derived. From Table 7-12, it could be observed that energy production is highest in the south orientation, followed by East, West, and North orientation. Therefore, the area required to fulfill the heating demand is highest in the North façade. The calculated areas would then be used in integrating solar collectors the design in the following chapters.

Table 7-13 : Heating Energy Requirement Based on Operable Hours

Component	No. of Operable Hours	Energy Consumption		
	Hours	kWh/day (11 Hours)	kWh	kWh/year
1 - CCHE	599	4.4	0.128	239.6
2 - CCHE	1540	8.7	0.171	1218.0
3 - CCHE	1180	13.1	0.037	1405.3
Total	3319		**Avg . Total Consumption** **9.5** kW/day	

Table 7-14 : Heating with Air to Water Heat Pump Energy Consumption

COP = 3.5

Component	Energy Required	Energy Consumption
	kWh/year	kWh/year
1 - CCHE	239.6	68.5
2 - CCHE	1218.0	348.0
3 - CCHE	1405.3	401.5
Total	2862.9	818
		Avg . Total Consumption **2.7** kW/day

Table 7-15 : Facade Integrated Components Energy Consumption

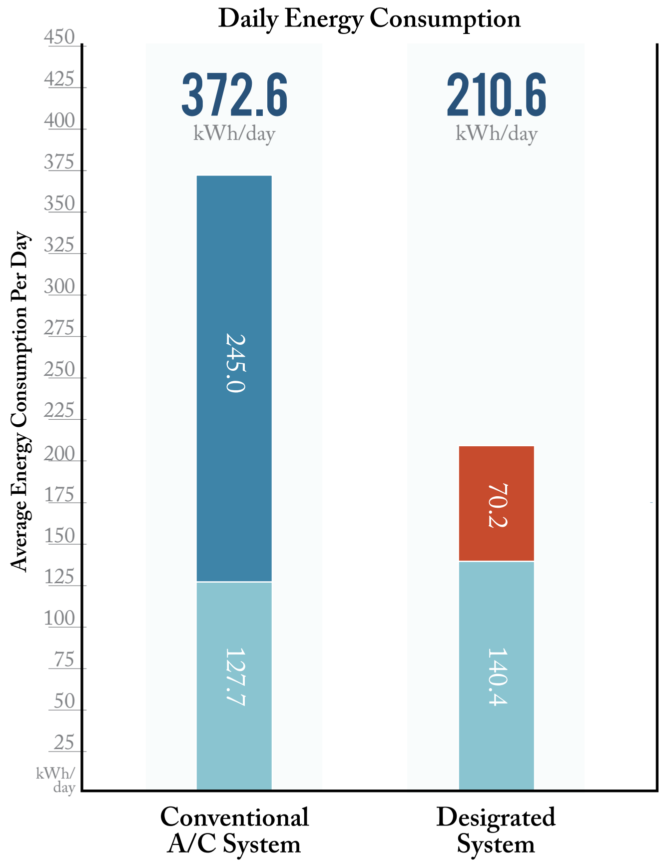
Component	Volume Flowrate	Energy Consumption	
	m^3/s	kWh	kWh/day (11 Hours)
Blower 1	0.18	0.128	1.4
Blower 2	0.42	0.171	1.9
Fan 1	0.18	0.037	0.4
Fan 2	0.6	0.117	1.3
Fan 3	0.18	0.037	0.4
			Total Consumption **5.4** kW/day

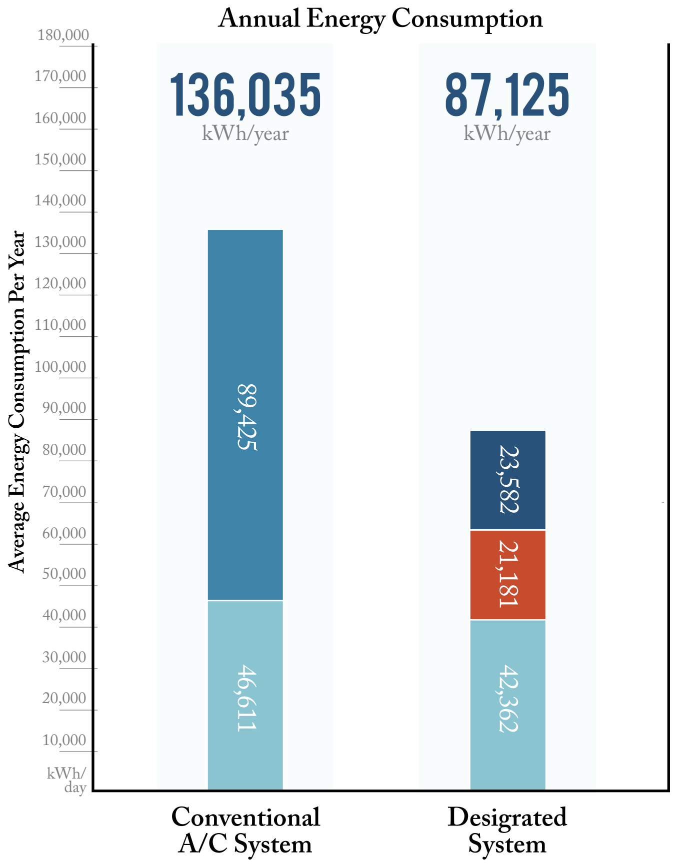
No. of Integrated Facade System	Facade Integrated Components	Air to Water Heat Pump	Total Electricity Requirements
26	**140.4**	**70.2**	**210.6**
Systems	**5.40** kWh/day	**2.70** kWh/day	kWh/day

7.4.3 Total System's Energy Consumption

To elaborate on the annual thermal energy requirements of the system, the annual thermal energy requirements based on the number of operable hours has been assessed. The number of the required desiccant system is cross-referenced with ambient hourly data. With 3319 operable hours, the system requires 2,862.9 kWh/year which could be averaged to 9.5 kWh/day. Further calculations have been made to determine the energy needed for the water to be heated up with a heat pump system. Owing to its high COP of 3.5, the total energy consumption required annually is accounted to 818kW/day, lowering its daily requirement to 2.7kW/day. When combined with the electricity necessary to operate the façade system, the required energy will be summed up to 8.1kWh/day per system. Based on the benchmark floor area, a maximum of 26 façade system could be integrated into the floor. Therefore, the total energy required to operate the system would be accounted to 210.6 kWh/day.

Figure 7-8 : System's energy consumption based on daily and annual usage.





Daily Energy Consumption

Component	Conventional A/C System	Desigrated System
	kWh/day	kWh/day
AHU	127.7	-
	11.6 kWh	
Facade Integrated	-	140.4
		12.8 kWh
Heat Pump	-	70.2
		6.4 kWh
Centralized Cooling	245.0	Uses return Chilled Water
	22.3 kWh	
Total Consumption	**372.7** kW/day	**210.6** kW/day
	33.9 kWh	19.1 kWh

Annual Energy Consumption

Component	Conventional A/C System	Desigrated System
	kWh/year	kWh/year
AHU	46,611	8,080
	4015 hours	696 hours
Facade Integrated	-	42,362
		3319 hours
Heat Pump	-	21,181
		3319 hours
Centralized Cooling	89,425	15,502
	4015 hours	696 hours
Total Consumption	**136,035** kW/year	**87,125** kW/year

Energy Consumption Reduction in comparison to AHU

+64.9%

kWh/day

Energy Consumption Reduction in comparison to Conventional System

-43.4%

kWh/day

Energy Consumption Reduction in comparison to Conventional System

-36%

kWh/year

7.4.4 Energy Reduction Potential

To evaluate the 'Desigrated' system efficiency, its energy consumption has been assessed and compared with the energy required by the conventional air condition system. The evaluation is based on three main aspects, the air handling unit (AHU), heating system, and the centralized cooling system (cooling tower and chiller system). Considering the 19th floor, the conventional A/C system operates with 2 AHU which requires 127.7 kWh/day for supplying the air. Moreover, the centralized system required 245 kWh/day to provide chilled water for cooling down the air. Therefore, the total energy needed by the conventional system would sum up to 372.7 kWh/day.

On the other hands, with 26 proposed façade systems, the 'Desigrated' system require 210.6kWh/day to operate. The energy required by façade integrated components (fans and blowers) is accounted for 140.4 kWh/day. Moreover, since hot water is necessary for regenerating the CCHEs, 70.2 kWh/day is consumed by the heat pump. It should be addressed that since the cold water required by the system, it will be taken from the return chilled water. Therefore, no energy is required by the centralized system.

Two major conclusions could be drawn from the results. When considering solely on the energy consumption of the air supply components (AHU and Façade integrated components), the results have shown that the 'Desigrated' system has 64.9% higher energy consumption. However, when the whole system is being assessed, the calculation has shown a 43.4% energy reduction. The reduction is due to the usage of return chilled water from the conventional system, which saves up to 245 kWh/day.

To further investigate the annual energy consumption of the 'Desigrated' system, the consumption is based on the operable hours has been calculated. With 3319 operable hours, 42,362 kWh/year is required for both the façade integrated components and the heat pump. However, additional energy of 23,582 kWh/year will be required for the back-up A/C system for the 696 hours, which is inoperable due to the exceeding level of ambient relative humidity. Due to the use of the back-up A/C system, the total annual energy required is summed up to 87,125 kWh/year, which slightly reduces the energy reduction from 43.4% to 36%.

Figure 8-1 : Multi-tier roof design technique used to create the buoyancy effect in Thai temple.

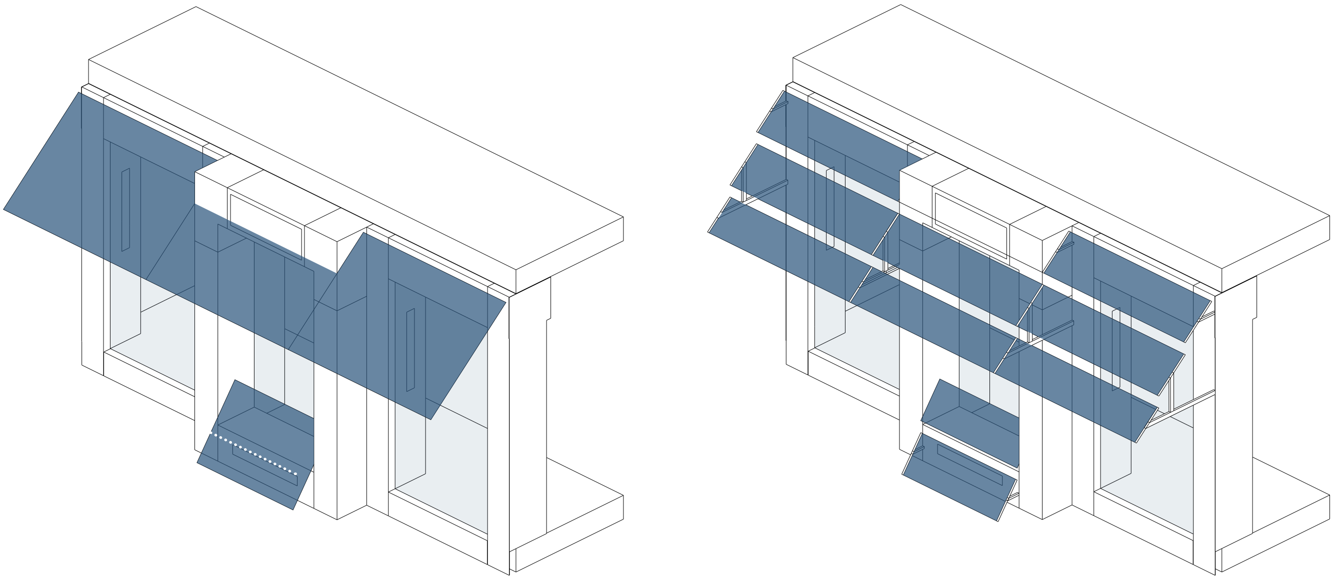


Figure 8-2 : Design development concept inspired by multi-tier design approach.

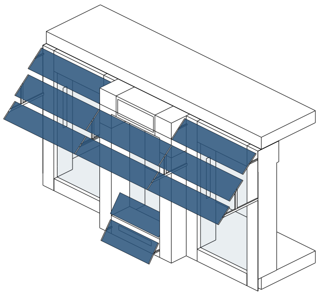
8 Design Development &Documentation

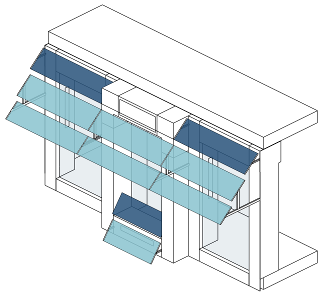
To further explore the design possibilities proposed in the earlier stages, the design concepts will be developed to emphasize the Thainess characteristic. The characteristics such as buoyancy and lightness would be abstracted based on the traditional design techniques discussed in the earlier chapters and implemented as a strategy for PVT integration. Moreover, design development would involve providing variation of the PVTs system configuration, balancing its operation with the heat pump system. The discussions would be based on the percentage of the coverage areas of the PV/PVTs system, which would also affect the shade and shadow cast on the façade. The façade system will be integrated into the benchmark building, and its effects on the appearance would also be further discussed. Moreover, the construction details in terms of insulation and assembly would also be elaborated in this chapter.

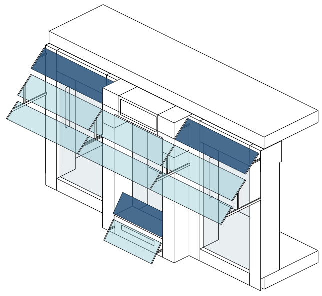
8.1 Achieving Buoyancy

The design concept was developed with the primary strategies and components reflecting the Thai architecture qualities. The base of the integrated façade system was designed as a modular, frame-like structure emphasizing on the openings. While the PVTs system was integrated as an extended roof structure positioned to obtain high solar radiation, preventing it from penetrating the interior space directly. This resulted in a large and bulky structure due to the high coverage area requirements. Therefore, to tone down the bulky structure, a traditional Thai design technique has been proposed. Inspired by the multi-tier roof of Thai temple, the bulky PVT structure will be divided into smaller strips and connected with its adjacent neighbor while eave brackets will be integrated to support the cantilever structure.

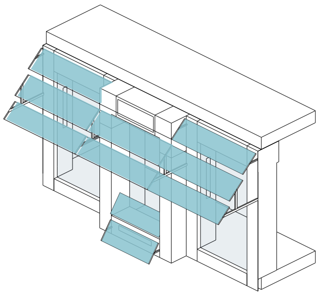
Table 8-1 : PV | PVT Configuration Concepts

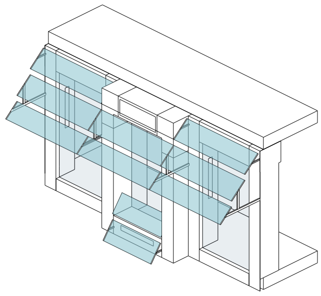


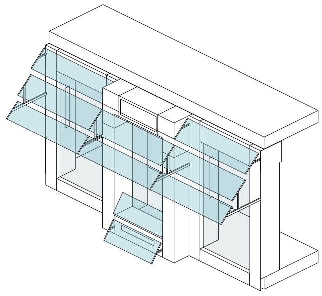




	Concept 1		Concept 2		Concept 3	
	PVT		PVT	PV	PVT	PV
Coverage Area	7.7 m^2		2.0 sq.m	5.7 m^2	2.0 sq.m	5.7 sq.m
Coverage %	100%		100%		100%	50%
Orientation	Thermal Production/m^2	Electricity Production/m^2	Thermal Production/m^2	Electricity Production/m^2	Thermal Production/m^2	Electricity Production/m^2
N	51.6	13.5	25.8	6.7	13.4	5.0
E	125.7	24.0	62.8	12.0	32.6	8.9
S	92.0	17.7	46.0	8.9	23.9	6.6
W	123.2	24.0	61.6	12.0	32.0	8.9
Total Energy Production kWh/day	**392.5**	**79.2**	**196.2**	**39.6**	**101.9**	**29.3**
Heat Pump Requirement	-		**41.4** kWh/day		**41.4** kWh/day	
Electricity Reduction Rate	**71%** 61.2 kWh/day		**42%** 123.2 kWh/day		**28%** 152.5 kWh/day	







	Concept 4	Concept 5	Concept 6
	PV	PV	PV
Coverage Area	7.7 m^2	7.7 m^2	7.7 m^2
Coverage %	100%	89%	50%
Orientation	Electricity Production/m^2	Electricity Production/m^2	Electricity Production/m^2
N	10.0	8.7	5.0
E	17.8	15.5	8.9
S	13.1	11.4	6.6
W	17.8	15.5	8.9
Total Energy Production kWh/day	**58.7**	**51.0**	**25.5**
Heat Pump Requirement	**70.2** kWh/day	**70.2** kWh/day	**70.2** kWh/day
Electricity Reduction Rate	**28%** 151.9 kWh/day	**24 %** 159.6 kWh/day	**14%** 181.3 kWh/day

8.2 PV | PVT Configuration Concept

The results from the evaluation have shown that the 'Desigrated' system using the heat pump as a hot water source has the potential to reduce 43.4% of energy consumption in comparison to the conventional air-conditioning system. However, its efficiency could be increased by acquiring thermal and electrical energy through the means of PV and PVTs system. 6 Design Concepts has been introduced in this chapter. The variation of the schemes is based on three main aspects including the balance between the usage of PV and PVT's system, Cell coverage density, and the balancing the operation between the PV/PVTs system and the heat pump system. The variation offers a wide range of options for the user to pursue, from a net-zero approach or the freedom to create the desired pattern according to the intention of the designer. The energy production is based on the calculation from the previous chapter. Considering that five systems are integrated on the north and south orientation and eight systems on the east and west orientation.

The first three concepts are explored with the implementation of the PVTs system, allowing thermal and electrical energy to be generated by the façade With 100% PVT coverage in Concept 1, 392.5kWh thermal energy could be produced daily. This exceeds the required energy of 247kWh by 146 kWh per day. However, due to the limited efficiency in terms of electricity production, 61.2kWh/day is still needed. Nevertheless, Concept 1 shows the highest energy reduction potential with the potential to reduce energy consumption further by 71%. Concept 2 and 3 shares a similar approach, in which the PVTs will only be installed at the bottom of the façade frame and on top of the air supply unit, while PV system will be installed in the other areas. The approach fails to provide sufficient thermal energy; therefore, 41.4 kWh/day is required by the heat pump. This reduces the efficiency of the system, as its energy consumption rate decreases to 41% and 28% respectively. However, the concepts possessed higher transparency and flexibility in terms of design as the designers can adjust the cells patterns according to their design intention.

Concept 3 to 6 shares a similar approach in which only PV panels will be integrated, aiming to provide higher transparency and design versatility. It should be addressed that Concept 5 with 89% cell coverage, is introduced to investigate the energy reduction potential if sufficient electrical energy could be generated to operate the heat pump. In corresponding to the elimination of PVTs system in Concept 3 to 6, the efficiency of the system reduces drastically. With 100% cell coverage, the PV system can reduce energy consumption by 28% and still requires 151.9kWh/day for its operation. However, further consideration in terms of transparency and shading should be considered as minimizing the PV coverage area would reduce the shading capacity of the shading components.

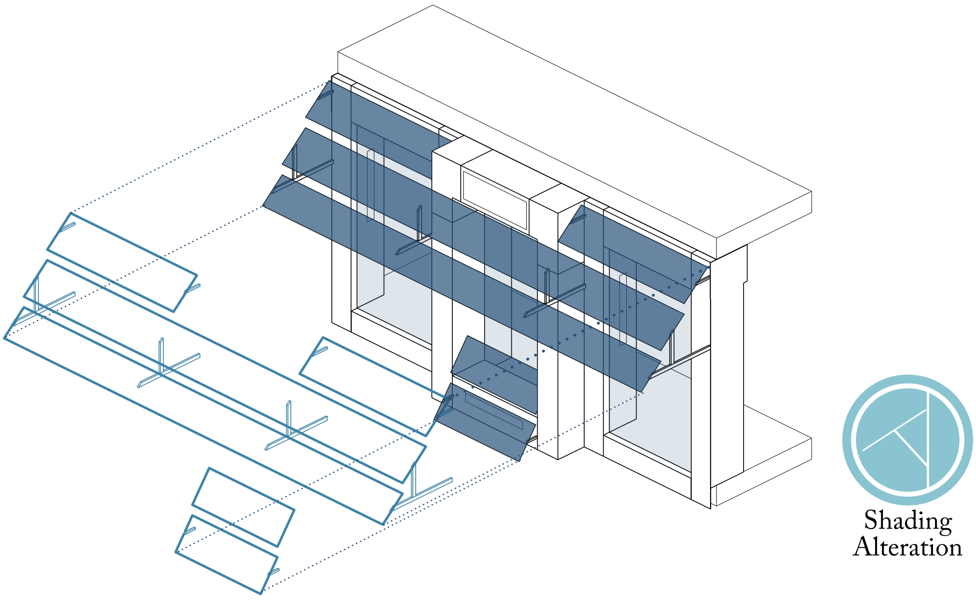


Figure 8-3 : Design configuration with alteration of shading components

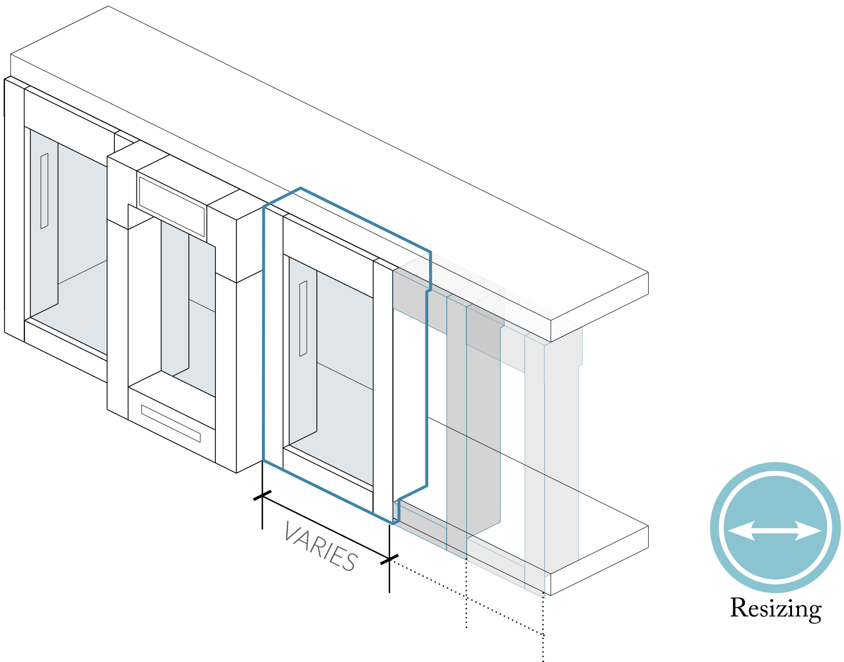
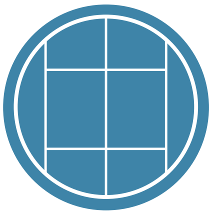


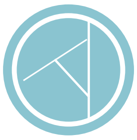
Figure 8-4 : Design configuration with resizing of system component

8.3 System Component Alteration

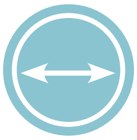
To enhance the versatility in the system's application, the 'Desigrated' system can be adjusted in two main aspects: shading alteration and air supply component resizing. As mentioned in the previous chapters, the system could operate solely with heat pumps. Therefore, shading components can be manipulated depending on the intention of the designers. These adjustments include choosing between PV or PVT, PV cells density, and the number of shading on each façade component. At the same time, the air supply component could also be modified horizontally, providing the designer with the freedom to either create a larger opening or minimize the window to wall ratio. To present the possibilities of the modification, three concepts with the constraint of having seven systems on the East/West facade and five systems on North/South facade will be presented in this chapter.



Symmetrical Array Pattern



Every Floor



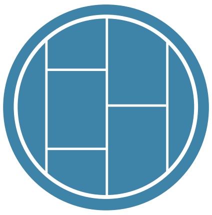
Standard Supply Unit

Figure 8-5 : Design Application - Symmetrical Array Pattern

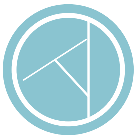
8.3.1 System Application Potential : Concept 1

Concept 1 portrays the most typical application of the system as the building's façade is designed with symmetrical array pattern. Therefore, the shading component is installed on every component at all floor levels, while the standard sizing of the air supply unit is being used.

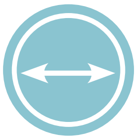
Figure 8-6 : Design Application influenced by Pakon pattern



'Pakon' Alternating Pattern



Every other Floor



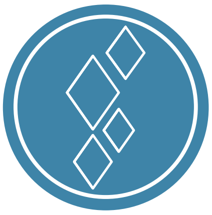
Resized - Supply Unit

Figure 8-7 : Design Application - 'Pakon Alternating Pattern'

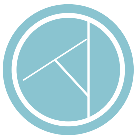
8.3.2 System Application Potential : Concept 2

Concept 2 is an attempt to integrate the traditional 'Pakon' wall pattern onto the façade. Therefore, alternating and shifting pattern is being projected. To achieve a continuous shifting pattern, the air supply unit is being expanded. Furthermore, the shading component is installed on every other floor to emphasize the shifts between the patterns.

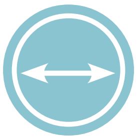
Figure 8-8 : Design Application influenced by traditional Thai Pattern



Abstraction of Thai Pattern



Placement
according to design



Non-modular

Figure 8-9 : Design Application - Abstraction of Thai Pattern

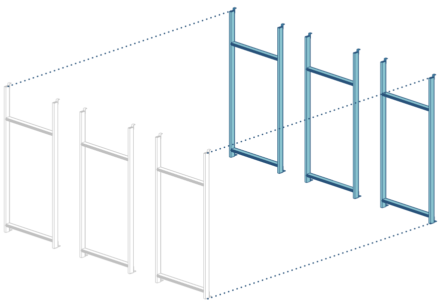
8.3.2 System Application Potential : Concept 3

Concept 3 aims to portrays the freedom and versatility of the system as the application is an attempt to project an abstracted Thai pattern onto the façade. To emphasize the pattern, the air supply unit has been resized, while the cladding color on the pattern position has been emphasized. It should also be addressed that this will make the air supply sizing non-modular. Furthermore, to enhance the effect of the pattern, the shading component is only installed in areas outside the triangular pattern.

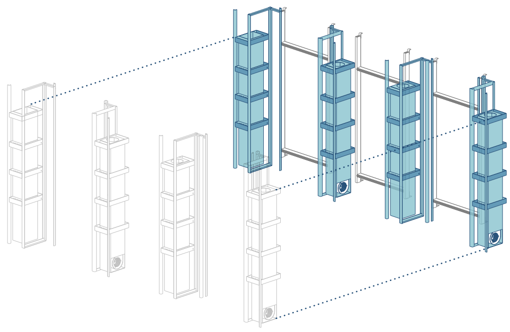
8.4 System Assembly

8.4.1 Prefabrication

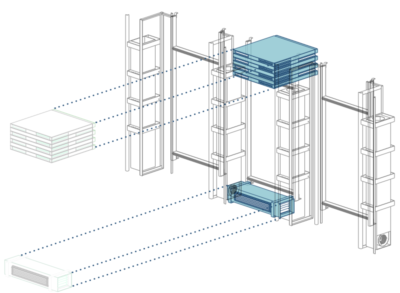
Due to the complexity of the 'Desigrated' façade system, the façade components will be prefabricated to facilitate on-site assembly. The diagram below illustrates the fabrication process. Firstly, the structural components such as mullion and the dehumidification units will be assembled. It is then followed by the M-Cycle System and Air Supply Unit, forming the essential core of the system. Then, the glazing unit, insulations, and piping system will be integrated. Finally, the base component will be cladded by the cladding materials.



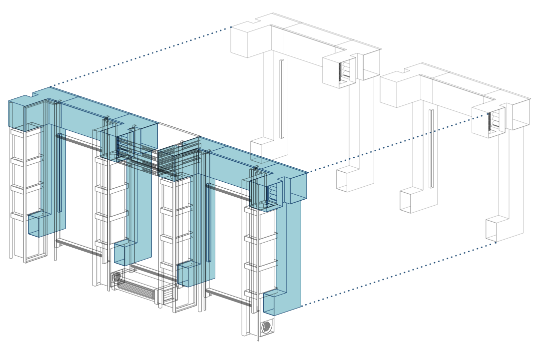
1. Mullion Structure



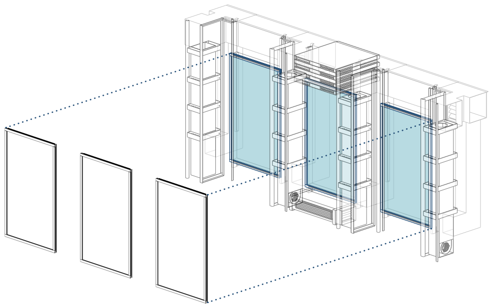
2. Dehumidification Unit



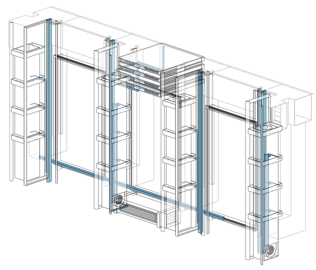
3. M-Cycle Unit



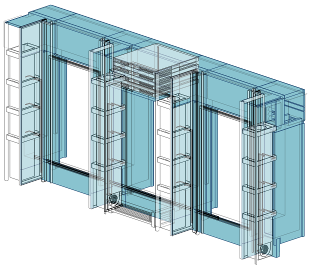
4. Air Supply Unit



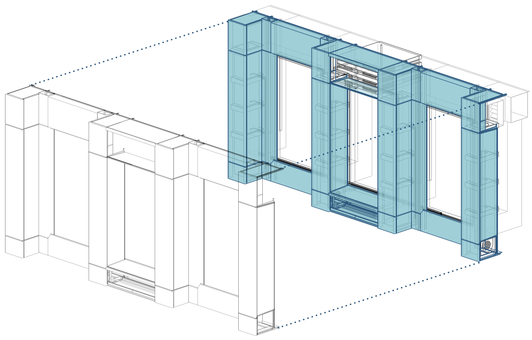
5. Glazing Units



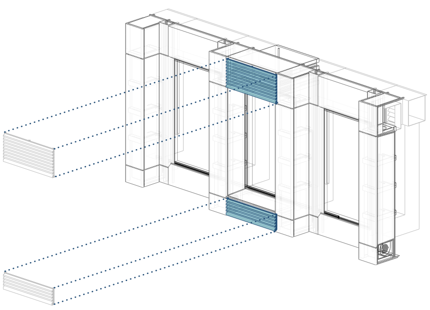
6. Piping System



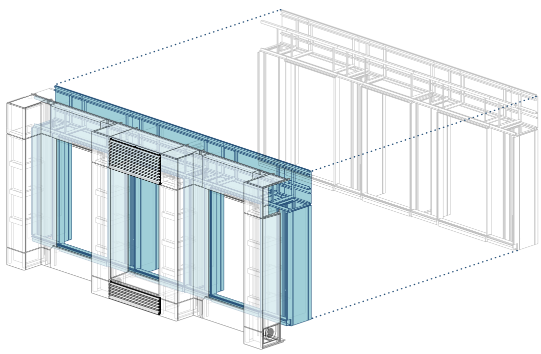
7. Insulation



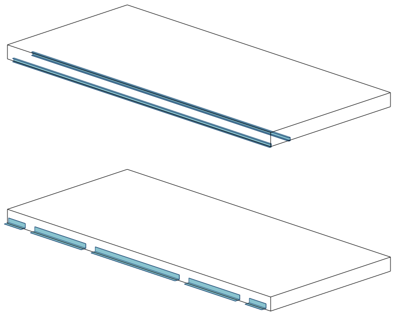
8. Exterior Cladding



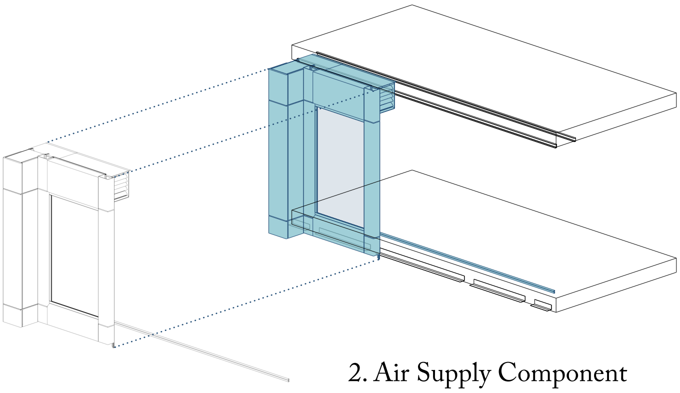
9. Facade Grill



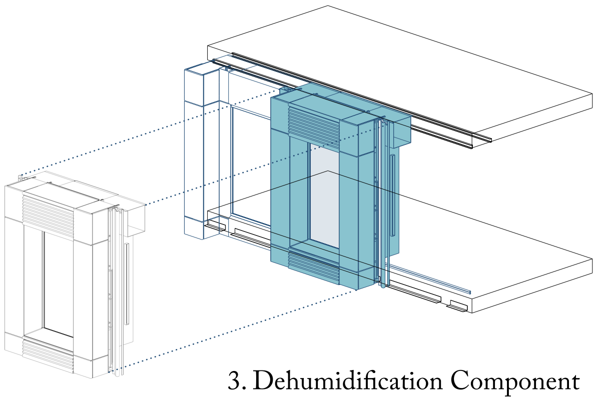
10. Interior Cladding



1. Facade Bracket



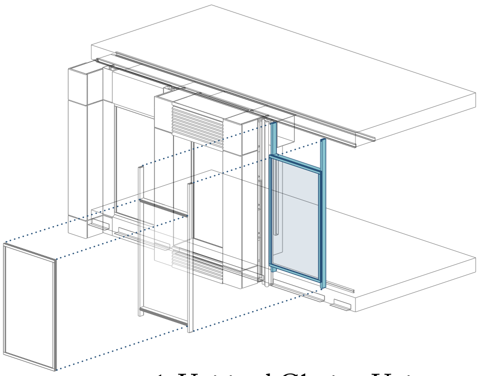
2. Air Supply Component



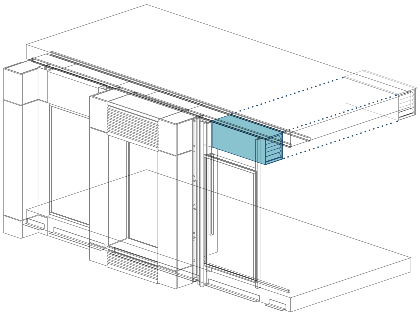
3. Dehumidification Component

8.4.2 On-Site Assembly : Prefabricated Components

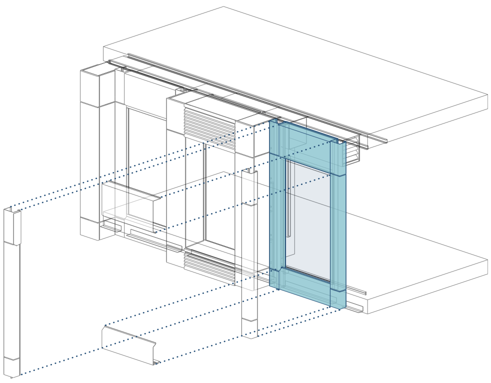
With the prefabricated components, the façade can be assembled with minimal sequence. Prefabrication allows the façade to be manufactured as either a large or a small module, which can be installed on site onto the pre-installed façade bracket.



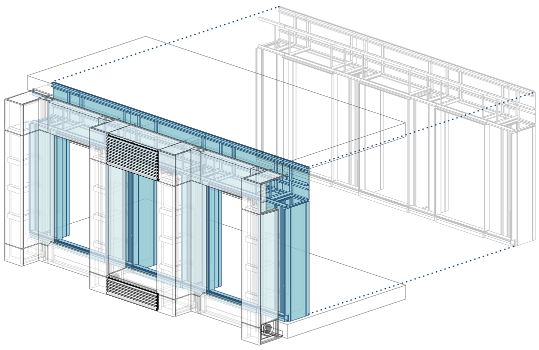
1. Unitized Glazing Unit



2. Air Supply Duct



3. Exterior Cladding



4. Interior Cladding

8.4.3 On-Site Assembly : On-Site Adjustment

In certain cases where the design of the façade design is not modular, the air supply component of the system could be manually adjusted on-site. With a similar procedure to the prefabrication process, the resized mullion structure will be assembled first, followed by the air supply component, and the exterior and interior cladding materials.

8.5 System Feasibility

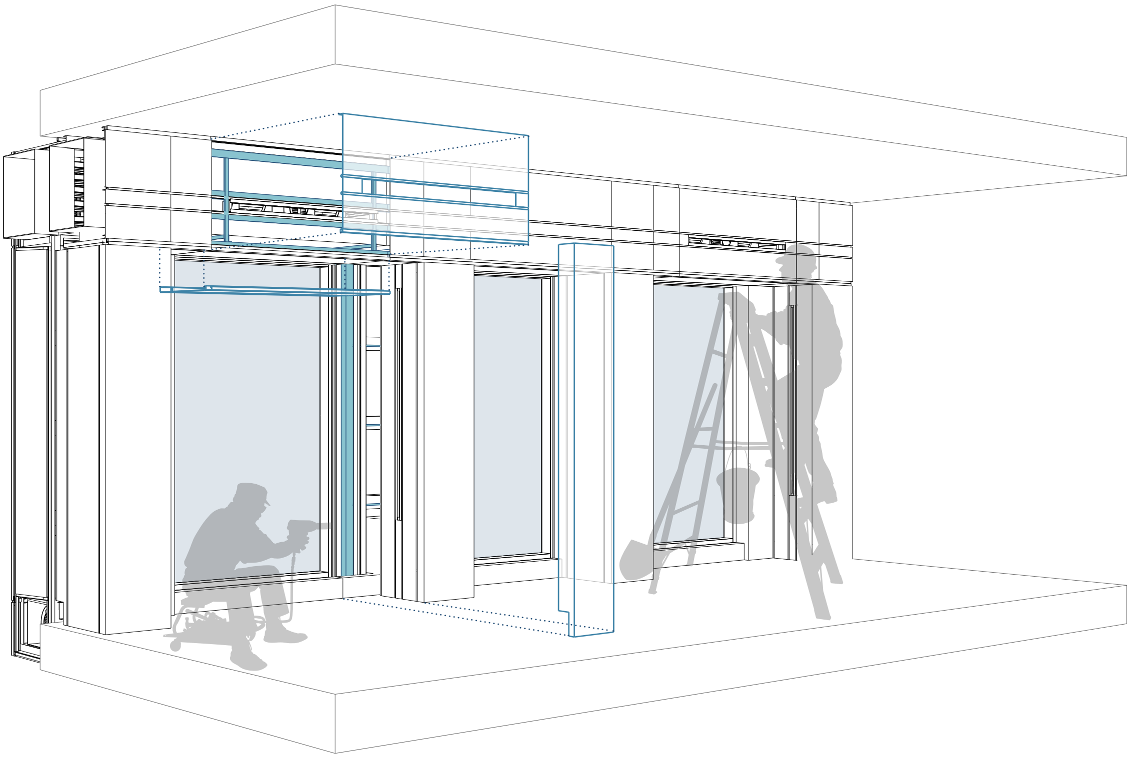


Figure 8-10 : Interior maintenance routine

8.5.1 System Maintenance

Similar to a conventional air conditioning system, the 'Desigrated' system required annual system maintenance and service. However, the maintenance required is relatively more complicated, as the system is being decentralized and requires both interior and exterior maintenance. The interior service requires minimal effort since it only deals with removing the cladding to clean out the air supply filters. While a service shaft is also provided for piping service in case of pipe blockage or leakage. On the other hand, exterior maintenance would require extra precautions. Unlike the centralized, conventional, air conditioning system in high rises, the 'Desigrated' system is designed to be decentralized; therefore, requiring higher maintenance. Sophisticated gears and machinery such as hanging lift will be necessary for the service of the exterior components, as the outer inlet air filter along with CCHEs unit also needed to be serviced. Furthermore, the shading components will also impose additional complexity during maintenance.

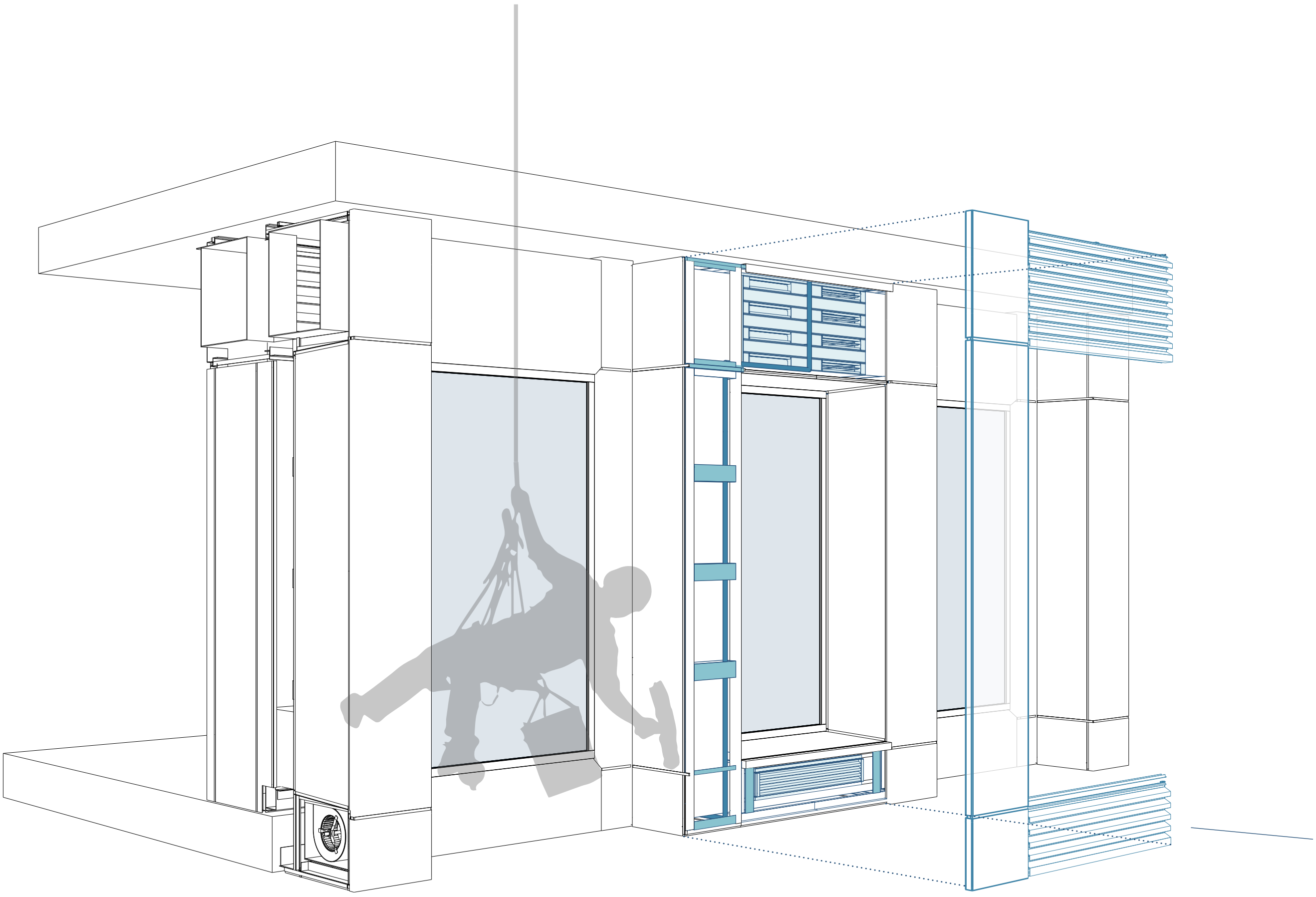


Figure 8-11 : Exterior maintenance routine

8.5.2 Initial Investment | Feasibility

Due to the limited usage of desiccant system and m-cycle system in building levels, the feasibility of the system will be discussed qualitatively based on similar product types available in the market. With a similar unitized façade system component, only the additional main components of the desiccant heat exchangers and an m-cycle system will be discussed. Based on heat exchanger whole sales values in the Asian market, the price of a heat exchanger varies around 100€ ("Oem Water to Air Plate Heat Exchanger Price"). However, when ordered in large amounts the price could be as low as 45€. per unit. ("Outdoor Water To Air Heat Exchanger") The cost of the M-cycle system can be referred to the Coolerado Cooler system with a cost parameter ranging from \$900-\$1,100 (795€-970€) per ton, falling within a similar price range to a conventional air conditioning system of \$900-1,800 (795€-1580€) per ton. (Robichaud, 2007) Moreover, due to its uncommon configuration and assembly, additional labor and structural cost should also be anticipated. With 6 CCHE components and an M-Cycle system integrated, the extra material cost will be accounted for approximately 1,480€ additional investment. However, the initial investment can be drastically reduced during mass production.

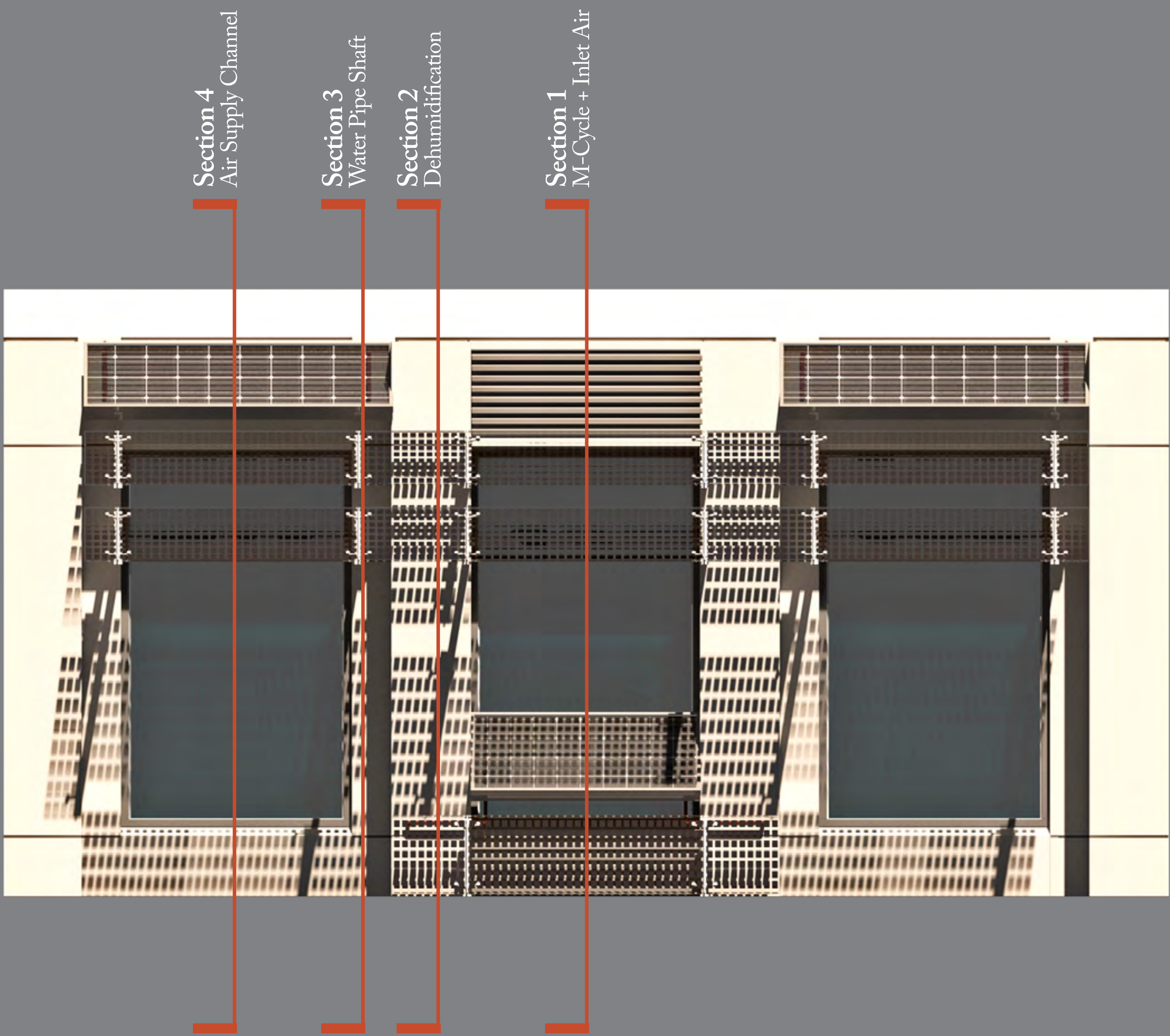
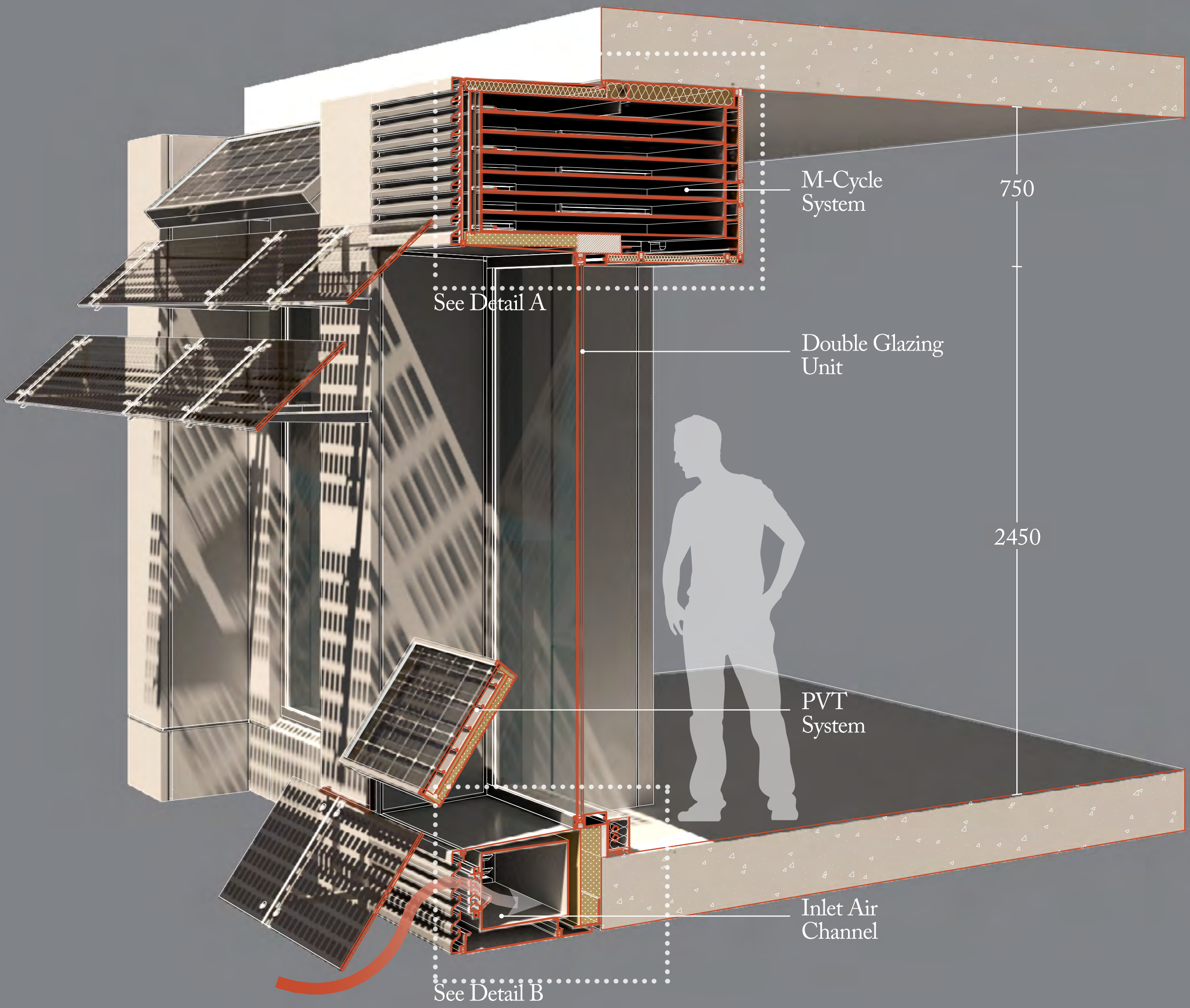


Figure 8-12 : 'Desigrated' System Elevation

8.6 Design Documentation

The design concept, evaluation, and design possibilities have already been discussed in the previous chapters. Therefore, sectional details and fittings of the systems will explore further in this chapter. As it will provide a more elaborated insight on the air flow and piping configurations. Four main sections through m-cycle + inlet air component, dehumidification channel, water pipe shaft, and air supply channel will be presented and elaborated.

Section 1
M-Cycle + Supply Air Channel



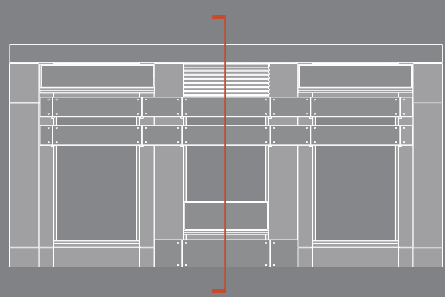
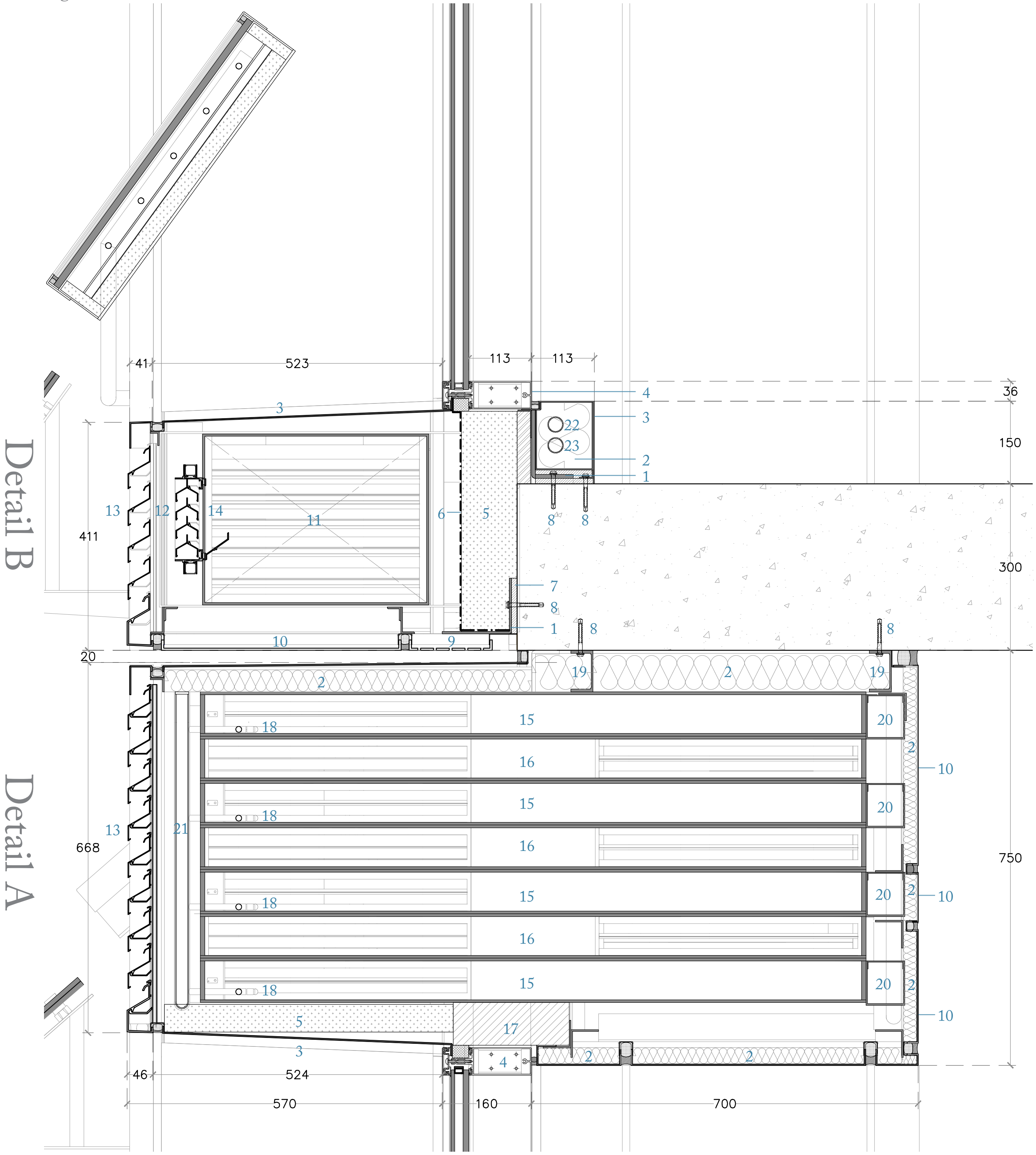


Figure 8-13 : 'Desigrated' System - Section 1

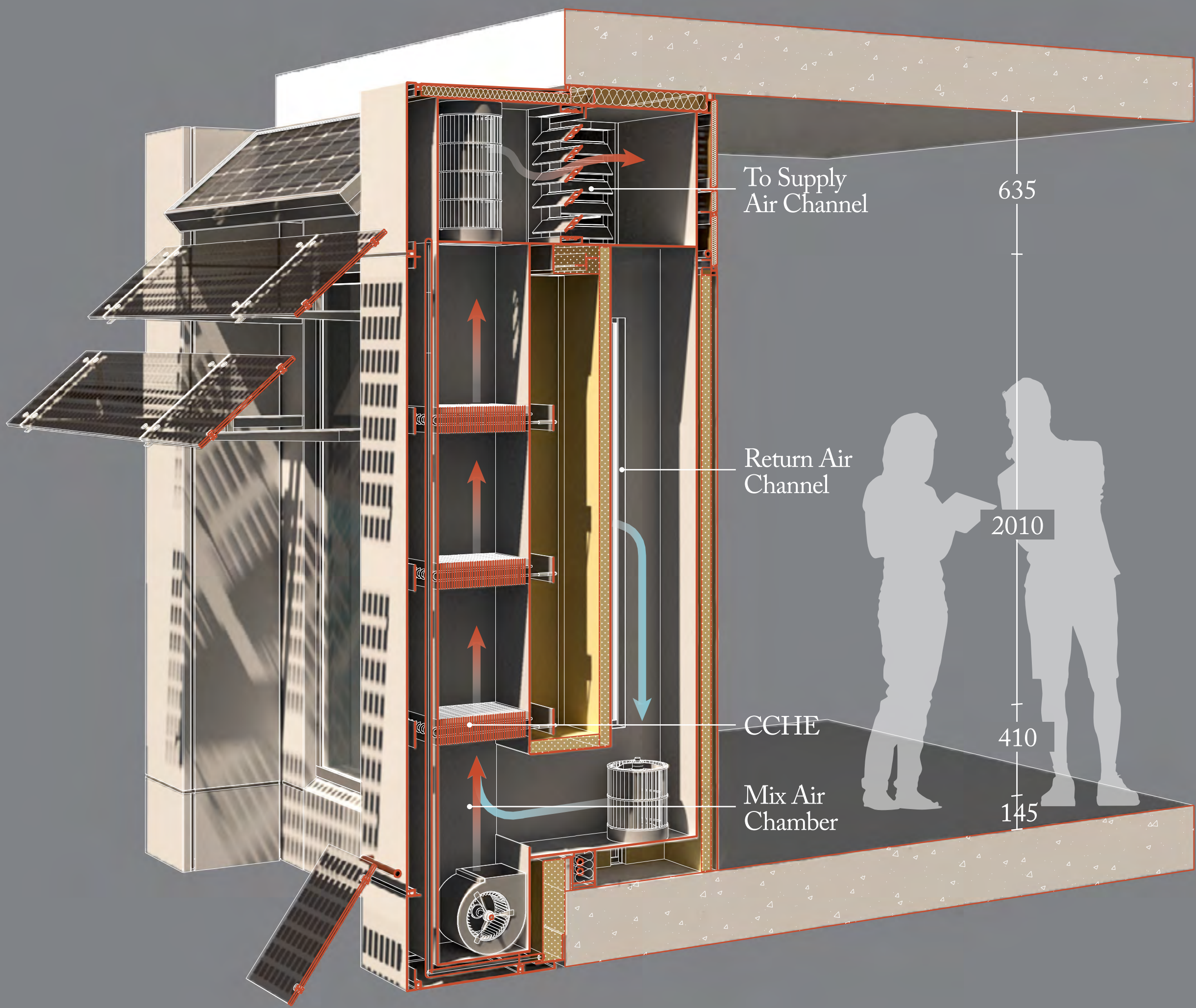
Connection Detail | M-Cycle Component

Figure 8-14



1. L-Bracket
2. Fiberwool Insulation
3. Aluminium Cladding
4. Transom
5. PUR Insulation
6. Waterproof Membrane
7. Thermalfoot
8. 10mm Concrete Bolt
9. Perforated Aluminium Cladding
10. Aluminium Composite Cladding
11. Inlet Air Channel
12. Inlet Aluminium Air Grill
13. Aluminium Facade Grill
14. Air Filter
15. M-cycle : Wet Channel
16. M-cycle : Dry Channel
17. Promatect Insulation
18. Water Sprinkler
19. C-Channel Bracket
20. Return Water Gutter
21. Inlet Cold Water Pipe
22. Return Main Hot Water Pipe
23. Supply Main Hot Watter Pipe

Section 2
Dehumidification Channel



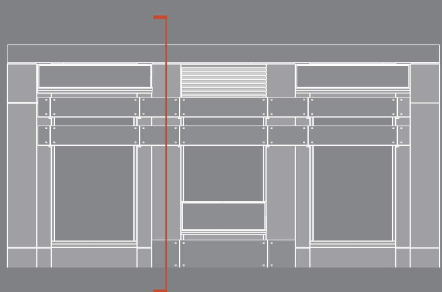
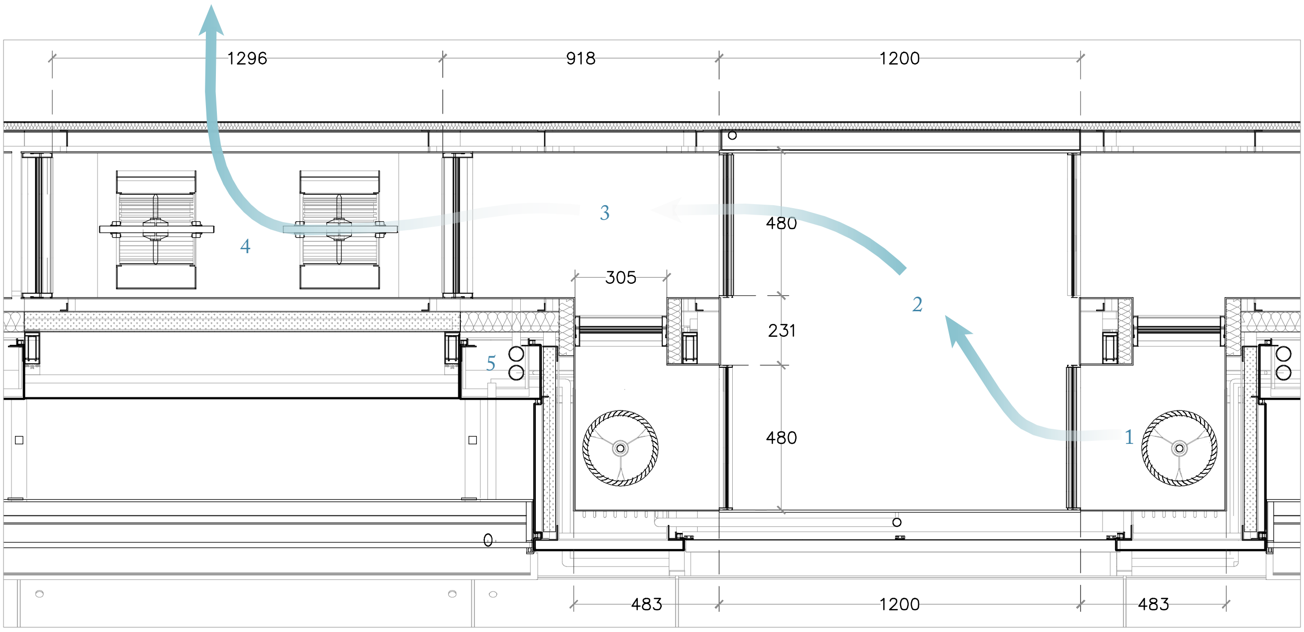


Figure 8-15 : 'Desigrated' System - Section 2

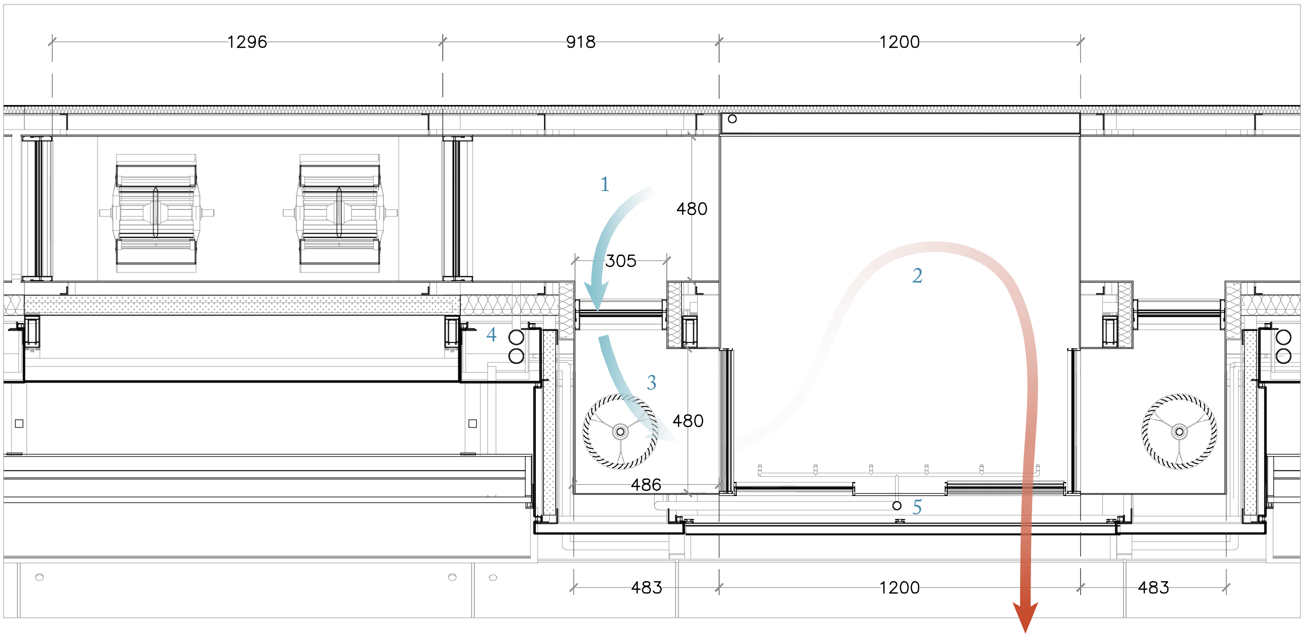
M-Cycle Component | Air Flow Schematic



M-Cycle : Supply Air Scheme

Figure 8-16

1. Inlet Air Chamber from CCHE
2. M-Cycle : Dry Channel
3. Air Channel
4. Supply Air Channel
5. Water Pipe Shaft

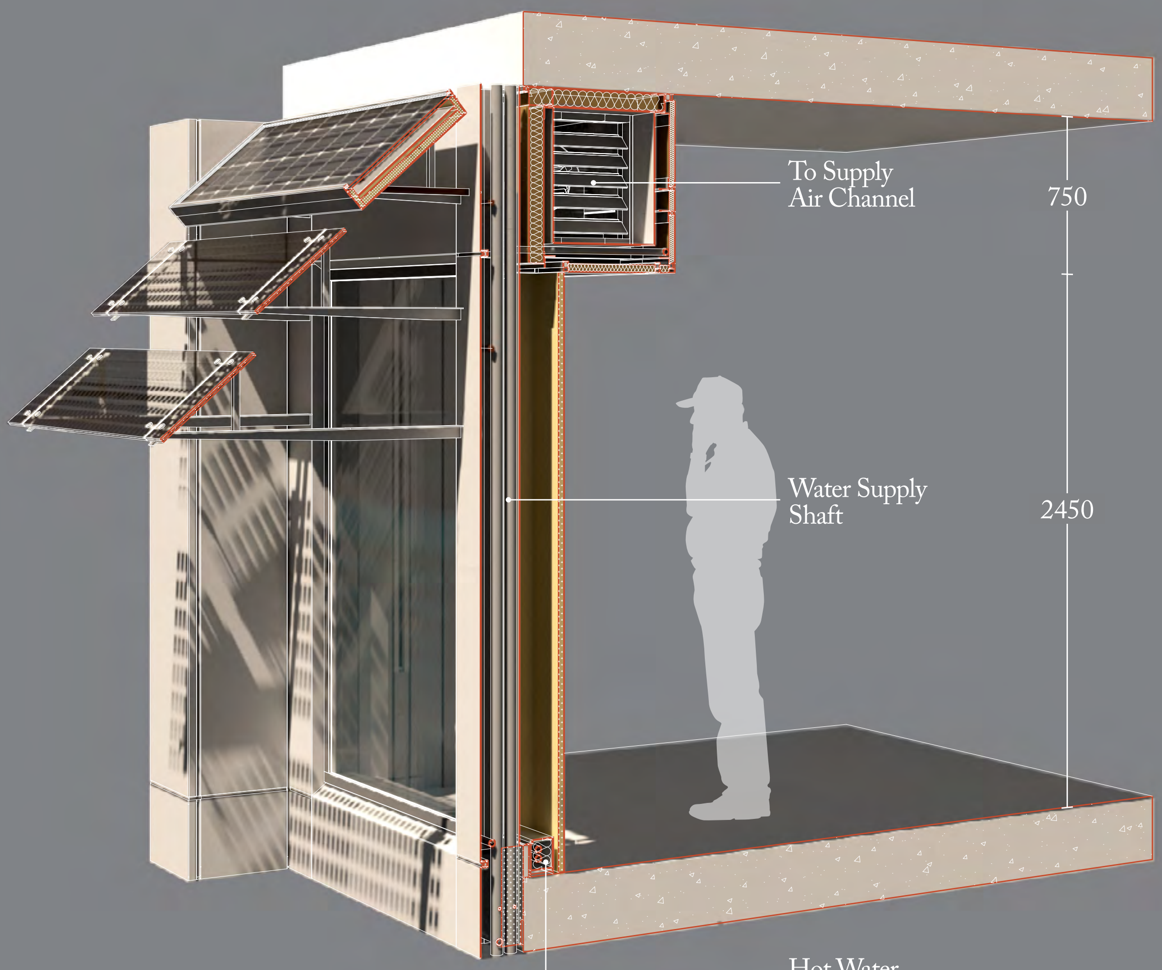


M-Cycle : Working Air Scheme

Figure 8-17

1. Inlet Air Chamber from Dry Channel
2. M-Cycle : Wet Channel
3. Air Channel
4. Water Pipe Shaft
5. Cold Water Inlet Pipe

Section 3
Water Pipe Shaft



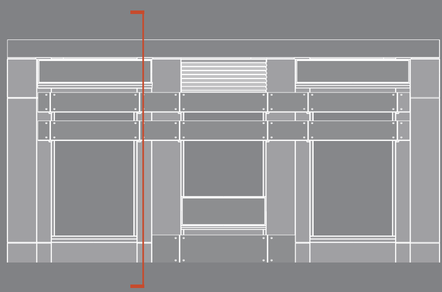


Figure 8-18 : 'Desigrated' System - Section 3

Water Pipe Schematic

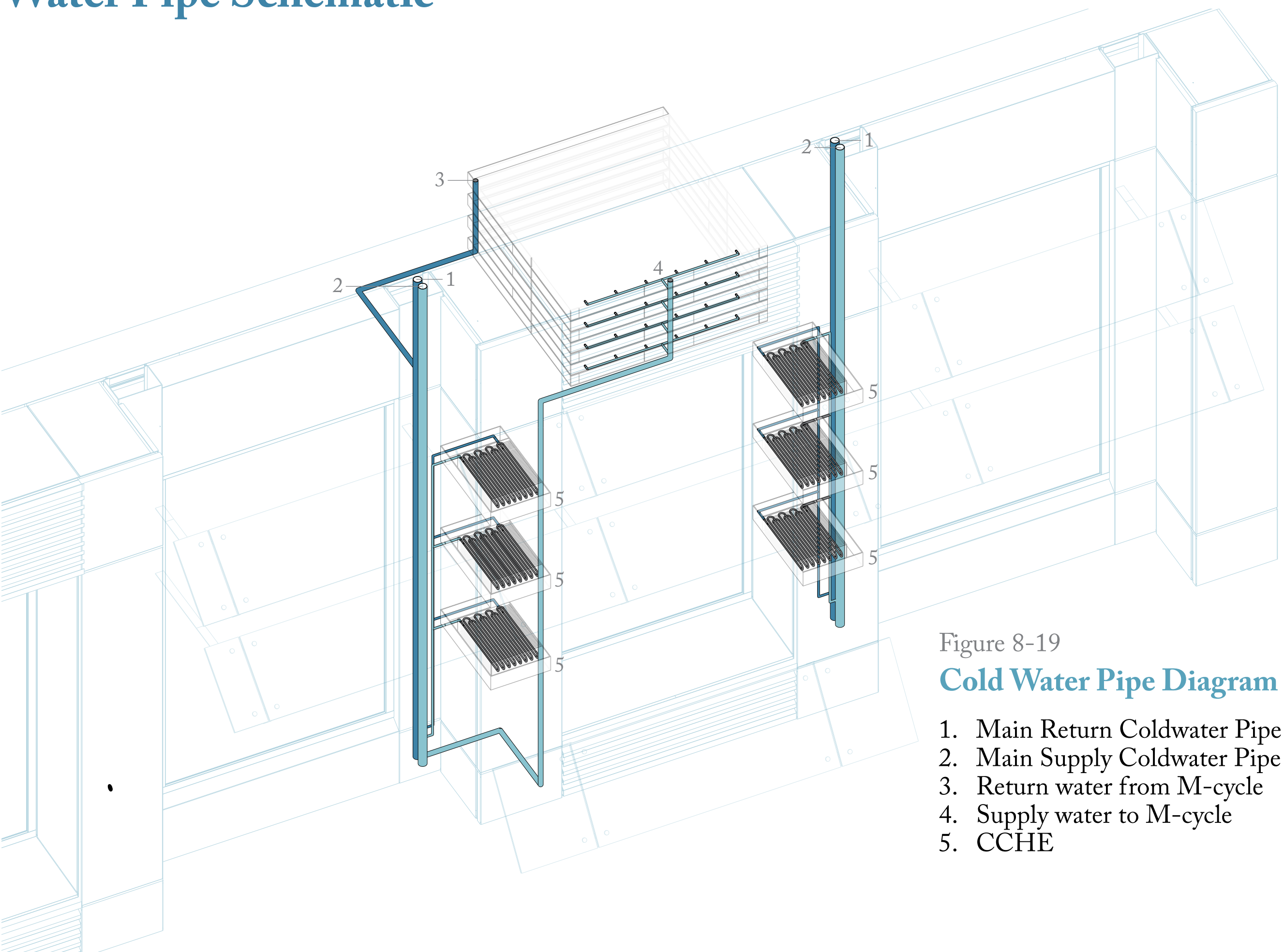


Figure 8-19
Cold Water Pipe Diagram

1. Main Return Coldwater Pipe
2. Main Supply Coldwater Pipe
3. Return water from M-cycle
4. Supply water to M-cycle
5. CCHE

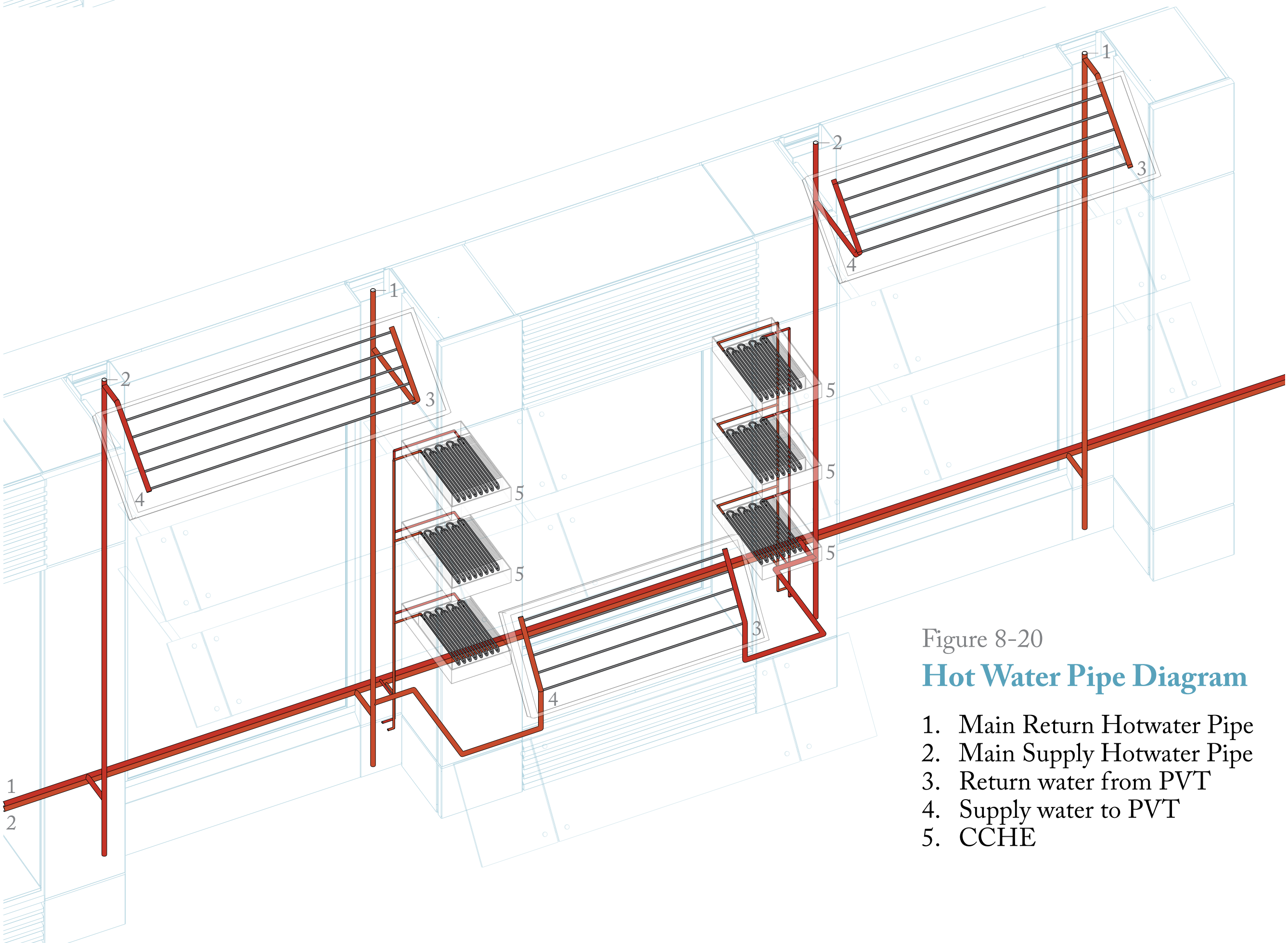
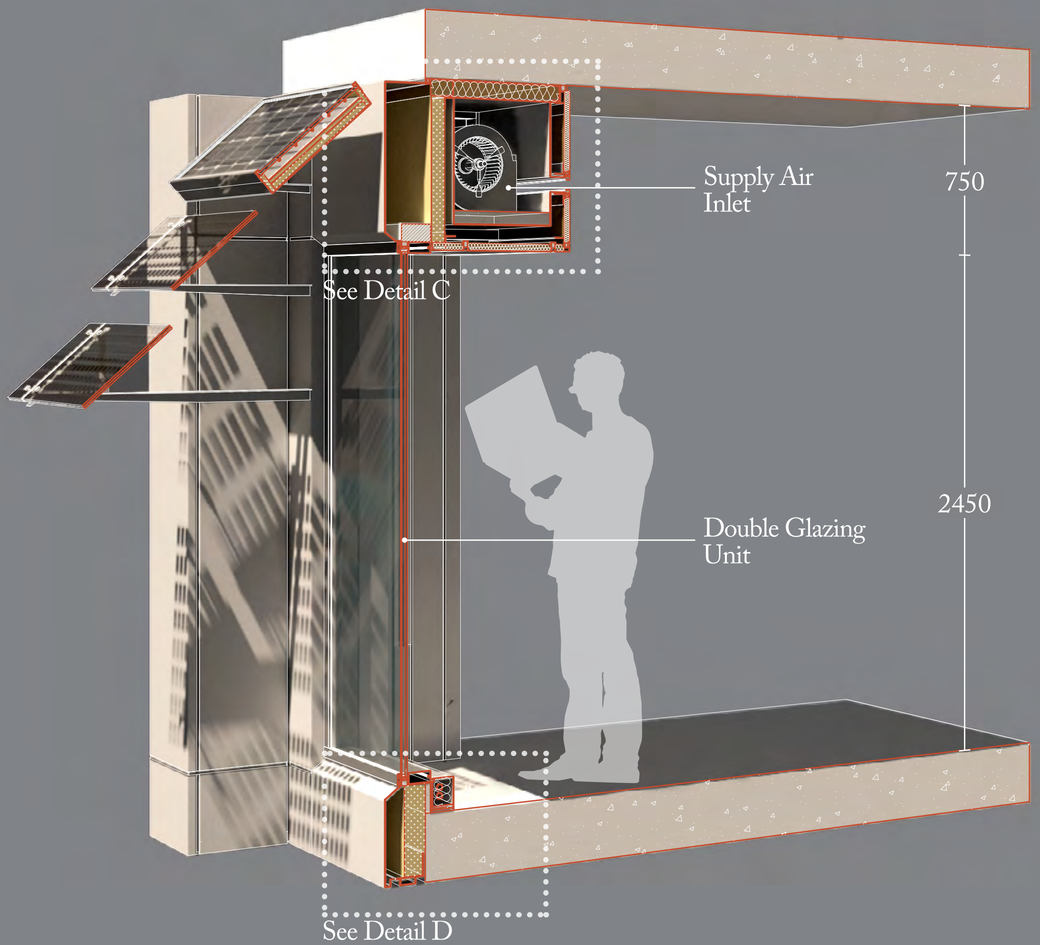


Figure 8-20
Hot Water Pipe Diagram

1. Main Return Hotwater Pipe
2. Main Supply Hotwater Pipe
3. Return water from PVT
4. Supply water to PVT
5. CCHE

Section 4
Supply Air Channel



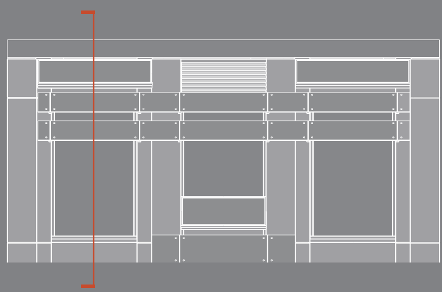
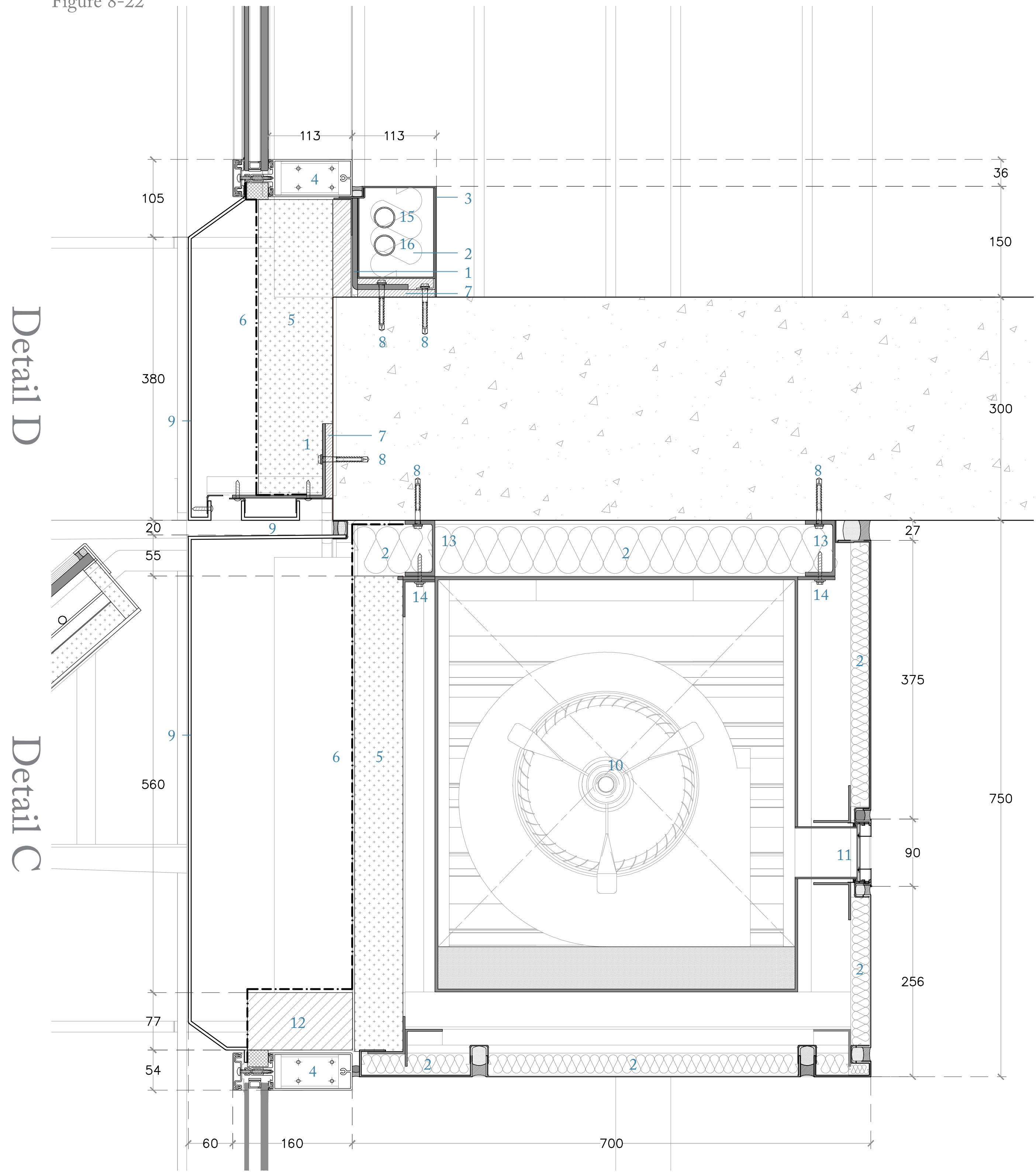


Figure 8-21 : 'Desigrated' System - Section 4

Connection Detail | Air Supply Component

Figure 8-22



1. L-Bracket
2. Fiberwool Insulation
3. Aluminium Cladding
4. Transom
5. PUR Insulation
6. Waterproof Membrane
7. Thermalfoot
8. 10mm Concrete Bolt
9. Aluminium Composite Cladding
10. Supply Air Channel
11. Aluminium Supply Air Slot
12. Promatect Insulation
13. C-Channel Bracket
14. Metal Screw
15. Main Return Hot Water Pipe
16. Main Supply Hot Water Pipe

9 Conclusion

“How can the **desiccant cooling facade system** be integrated into the built environment to reduce cooling load of **office buildings in a hot and humid climate**?”

Through this thesis, several possibilities have been proposed to illustrate the potential to integrate the desiccant system as a façade component. The core components of the ‘Desigrated’ system includes three Composite Silica get Coated Heat Exchanger (CCHE) connected in a series and an M-Cycle Indirect Evaporative System. The dehumidification and cooling capacity of the coupled system along with the shading strategy and insulation has been proven to provide a reduction of cooling load and sufficient air supply condition. The results of the simulation have shown that the proposed system has lowered the cooling load of the room by an average of 19.7 kW. Furthermore, with 3 CCHEs, the system’s effectiveness can satisfy up to 82.7% of the ambient air condition, in relative to the operational hours of the building between 8.00-18.00 with an energy reduction potential of up to 36% annually. However, due to the current system's efficiency, the ‘Desigrated’ system fails to achieve its ultimate goal in eliminating the use of refrigerants. Since cold water at 15°C is required, the system is still dependent on returning water of a chiller system. Nevertheless, it shows a positive potential in its application as a transition tool, gradually paving the way for a sustainable future.

9.1 Part I : Disassembly
Site Context | Thermal Comfort

Being heavily influenced by the International Style, Bangkok’s skyscrapers are being dominated by thin skin typology. Despite technological advancement, and innovative building techniques, the rectilinear fully glazed façade fails to efficiency adapt to the hot and humid context.

Bangkok’s climate is characterized by Tropical Wet-Dry climate (Aw), resulting in high solar radiation, a high temperature which could reach 38°C in the summer, relatively high humidity with an average of 70%, and abundant precipitation year-round. Vernacular strategies of traditional Thai houses have presented itself as an effective precedent in coping with the hot and humid climate. Through the technique of ‘adaptation of abstracted characteristics’ applicable characteristics of Thai architecture has been abstracted and adapted to the design concept of the ‘Desigrated’ system.

Based on two literatures reviews conducted in with an approach of defining the range of thermal acceptability through the means of ASHRAE 55’s thermal sensation scale in their field study. The optimal temperature range referred to in this thesis is based on the average value between the two studies, considering the thermal acceptability range at 23.5°C-27.7°C. with the preferred relative humidity at 50-60% and air velocity at 0.2 m/s

9.2 Part II : Assembly
Cooling Strategies

The result from the research has shown that the most appropriate passive strategies for a hot and humid climate such as Bangkok, is the prevention of direct solar exposure into the interior space. On the other hand, it is evidence that ventilation strategy is inapplicable in Bangkok's context due to the increase of latent loads. Therefore, it could be concluded that the most suitable passive strategy is the implementation of a minimized window-to-wall ratio, along with the shading strategy. Through the abstraction of Thai's architecture's 'iconic emblems' such as extensive roof structures and eave brackets as shading components, along with the emphasis on window opening and 'Pakon' Modular wall, the 'Desigrated' system is incorporated with features that allows it to adapt to the climate condition of Bangkok, while expressing Thai's cultural value. The design concept is translated into a simple frame-liked modular structure which provides a broader application possibility.

Composite Silica Gel Coated Heat Exchanger (CCHE) works as a primary component of the system to dehumidify the ambient air condition as it reduces the latent load needed to be processed by the cooling system. Based on the studies, under the average ambient condition of Bangkok, the CCHE can provide a cooling capacity of 1.25kW with a dehumidification rate of 3.9 g kg. However, the CCHE alone cannot provide sufficient cooling capacity to cool down the supply air to the required condition.

The system configuration is consists of two main components; the Desiccant coated heat exchanger and the M-Cycle. The constraints used in the calculations are based on literature reviews and experiment results or Ge et al., and Riangvilaikul&Kumar, respectively. Factors which influences the performance of the system, including recommended flow velocity and 30% working air ratio has been assessed and used in determining the system sizing and fresh air ratio scheme. As a result, the optimal airflow cross section is fixed at 0.25m2 with a fresh air ratio of 30:70. A further preliminary calculation has shown that the system requires 3-CCHE to provide supply air with a relative humidity of 55%.

The integration of the PVT and PV system integration has been studied. Results of the simulation and calculation have shown that the horizontal orientation is the optimal orientation for the PV/PVT system integration. However, the tilt angle varies is varied according to the orientation of the façade. The PVT component coverage area of 6.5 m2 is sufficient thermal energy for the minimal hot water requirement of 60L/hr on all orientation except the North façade. However, due to the low-efficiency rate of the PV panels, the scheme fails to satisfy the electrical requirements of the façade system, being able to supply an average of 47% of what is required.

9.3 Part III : Integration
Building Integration

The system configuration has been developed with a modular, frame appearance which provides broader application possibility. With this configuration, 3 CCHE components and an M-Cycle system require at least 180m3/hr. of hot and cold water to operate, while the fan and blower system requires 5.4 kWh/day. Since PVT integration fails to provide sufficient thermal energy for hot water, the investigation on the implementation of heat pump has been conducted. Owing to its high COP of 3.5, the total energy consumption required annually is accounted to 818kW/year, lowering its daily requirement to 2.7kW/day. When combined with the electricity needed to operate the façade system, the required energy will be summed up to 8.1kWh/day per system.

To evaluate the 'Desigrated' system efficiency, its energy consumption has been assessed and compared with the energy required by the conventional air condition system. The benchmark, at the 19th floor of the Krungthai-Sukhumvit Building with a conventional A/C system with 2 AHU, requires 372.7 kWh/day for the operation. While with 26 proposed façade systems, the 'Desigrated' system requires 210.6kWh/day to operate. The energy required by façade integrated components (fans and blowers) is accounted for 140.4 kWh/day. Moreover, since hot water is necessary for regenerating the CCHEs, 70.2 kWh/day is consumed by the heat pump.

Two significant conclusions could be drawn from the results. When considering solely on the energy consumption of the air supply components (AHU and Façade integrated components), the results have shown that the 'Desigrated' system has 64.9% higher energy consumption. However, when the whole system is being assessed, the calculation has shown a 43.4% energy reduction. The reduction is due to the usage of return chilled water from the conventional system, which saves up to 245 kWh/day. However, when evaluating the system performance based on annual operating hours, the results vary slightly. With 3319 operable hours, 42,362 kWh/year is required for both the façade integrated components and the heat pump. However, additional energy of 23,582 kWh/year will be required for the back-up A/C system for the 696 hours, which is inoperable. Due to the use of the back-up A/C system, the total annual energy required is summed up to 87,125 kWh/year, which slightly reduces the energy reduction from 43.4% to 36%.

To further explore the design possibilities proposed in the earlier stages, the design concepts have been developed to emphasize on the Thai characteristic of buoyancy and lightness and implemented as a strategy for PVT integration. Six Design Concepts has been introduced based on three main aspects including the balance between the usage of PV and PVT's system, Cell coverage density, and the balancing the operation between the PV/PVTs system and the heat pump system. The variation offers a wide range of options for the user to pursue, from a net-zero approach or the freedom to create the desired pattern according to the intention of the designer. Concept 1 with 100% PVT coverage, shows the highest energy reduction with the potential to reduce energy consumption further by 71%, but due to the limited efficiency in terms of electricity production, 61.2kWh/day is still required. Moreover, to provide sufficient electrical energy requirement for the heat pump, a PV system with 89% cell coverage is needed.

It should be addressed that during the architectural design development consideration towards transparency and shading should be considered. As minimizing the PV coverage area would reduce the shading capacity of the shading components. Therefore, other means of solar prevention, such as tinted glass or louver components should be thoroughly considered while designing.

9.4 Further Recommendations and Limitations

The 'Desigrated' thesis discussed the possibilities of integrating the desiccant system and M-cycle system as a façade component. The results have shown that it has the potential in reducing cooling loads and energy consumption of a specific benchmark building with a general possibility for implementation in other building in a hot and humid climate. However, there are numerous limitations and recommendations which could be further elaborated and discussed. These aspects include Limitations of performance data, System operation scheduling, Solar radiation analysis, Cost consideration, Maintenance aspects, Architectural Application.

Limitations of performance data

The calculation in this thesis is based on an assumption of general system scheduling, focusing mainly on using either conventional air condition or 'Desigrated' system individually. Therefore, further analysis and calculation could be made to analyze the optimal operation schedule where both systems could operate simultaneously. Due to the current efficiency of the systems, the 'Desigrated' system still fails to achieve its ultimate goal in eliminating the use of the refrigerant system. To operate under high temperature and humidity, the system still requires cold water (15°C) to operate efficiently. Therefore, further studies could be investigated to achieve a net-zero operation.

System operation scheduling

The calculation in this thesis is based on an assumption of general system scheduling, focusing mainly on using either conventional air condition or 'Desigrated' system individually. Therefore, further analysis and calculation could be main to analyze the optimal operation schedule where both systems could operate simultaneously.

Solar radiation analysis

Currently, solar radiation analysis is based on an average result of annual solar exposure. However, in reality, solar exposure varies daily. Therefore, to achieve more precise results, studies of solar radiation on a daily basis should be investigated. This result could be used to optimize the dependency of a heat pump, which could lead to a net-zero application in the future.

Cost consideration

Cost consideration discussed in this thesis is mainly based on qualitative speculations. It generally provided an overview of the construction and labor cost in comparison to a conventional unitized façade system. Therefore, through the means of prototyping, feasibility studies, and life cycle assessments, an extensive insight into the system feasibility could be further investigated.

Maintenance aspects

As discussed in the report, the maintenance of the system is relatively sophisticated, due to its limited accessibility to the exterior components and decentralized approach. Further studies and holistic design approach with the architect during the building design stage is highly recommended. With either an opening to a service corridor or future development to provide accessibility to the exterior components, the maintenance issues could be resolved.

Architectural Application

The application possibility tackled in this thesis only provides a few general ideas of how the façade can be implemented. To minimize on-site complications, thorough consideration should be made during the building design stage to generate a modular solution. Furthermore, attention on pv/pvt coverage area should be analyzed. As minimizing the PV coverage area would reduce the shading capacity of the shading components. Therefore, other means of solar prevention, such as tinted glass or louver components should be thoroughly considered while designing.

10 Reflection

Growing up in a hot and humid context of a fast-growing metropolitan of Bangkok, Thailand is one of the major drives to pursue my graduation project - *'Desigrate'*. Living in such context, air conditioning system becomes one of the necessities to provide indoor comfortability. Due to economical growth and rising temperature especially in summer, the growing demand for cooling is rapidly growing. This is visually evidenced from the condensing unit boxes hanging on the façade of houses and offices. We are stuck in a paradoxical situation, where greater demand of cooling is required, while carbon emission needs to be controlled.

'Desigrated' was developed under the COOLFACADE studio. The studio acts as a follow up on Alejandro Prieto's PHD thesis "COOLFACADE, Architectural integration of solar cooling technologies in the building envelope'. In his thesis Alejandro has conducted an extensive research to explore the possibilities and constraints for architectural integration of solar cooling systems in building facades to facilitates the design of architectural components for office buildings. The research has laid a solid groundwork for my project as it offers a comprehensive study regarding the solar cooling technologies in terms of its applications, constraints, and performances. It plays a crucial role in shaping the direction of my graduation project to explore the potential of a sustainable approach to provide an alternative system to facilitate or replace the conventional air conditioning system. The system is designed to serve as a façade component of office buildings. Therefore, the building technology aspect of climate, façade detailing, and façade assembly is thoroughly discussed, optimized and evaluated to provide a product which acts efficiently as a cooling system and enhances the insulation of the building. Hence serving as a cooling strategy both actively and passively. Furthermore, as the façade system is specifically designed for the hot and humid climate of Bangkok, both climate and cultural context are crucial aspects in designing the system. The design concept of the façade system is based on a vernacular approach to enhance the cultural value of the product which aims to reflects the "Thainess" identity to the built environment.

The project was developed with the 'disassembly – reassembly – integrating' strategy. The strategy is based on the line of inquiry imposed by the project main research question "How can the desiccant cooling facade system be integrated into the built environment to reduce cooling load of office buildings in a hot and humid climate?" and the COOLFACADE mentality to explore the potential of solar cooling technologies.

Disassembly

To bridge-in the knowledge gap regarding the solar cooling technologies and techniques, the research started off with the disassembly process. Primarily, the research is a comparative review based on literatures, experiments, case studies and market available products to understand the configuration of the different systems, and their performances based on different climate contexts.

Reassembly
While cooling and dehumidification techniques such as desiccant system and evaporative cooling has been widely use in industrial sectors for drying crops and humidity control in textile mills, it has a limited usage in building level, especially as a façade. Therefore, to reassemble the applicable systems as a façade component, literature reviews and researches become one of the most crucial tools in selecting the suitable system and strategy in designing the façade system. A comprehensive research base on the criteria of performances, applicability in hot and humid climate, and system configuration has been conducted. The constraints and results from experiments has been used as a framework in designing the façade system.

Integration
The final part of the research takes the form of a design process to integrate the façade system into a selected building which acts as a benchmark for evaluation. The performances of the systems are based on the assumption that the system would work under a similar condition and constraints of the experiments. The formulas and general calculations derived from the experiments were also used as a means of evaluating the system. Furthermore, to re-evaluate and verify the system performance, simulation software such as Ladybugs and Design Builders are used to generate the condition produces by the proposed system.

The project aims to explore the possibility of integrating a low-energy consumption technology such as the desiccant system and the dew-point indirect evaporative system into a façade component to offer an alternative cooling strategy to facilitates or ultimately replace the conventional cooling system. Therefore, it presents the possibility a system which would consume less energy and eliminating the use of refrigerants. It has the potential to resolve the growing carbon emission issues while providing serving the cooling demands. While the results and conclusion are merely based on assumptions and experimental constraints, the project could be use as a baseline for future prototypes in terms of system configuration and integration.

Due to the current efficiency of the systems, the 'Desigrated' system still fails to achieve its ultimate goal in eliminating the use of refrigerant system. In order to operate under high temperature and humidity the system still requires cold water (15°C) to operate efficiently. Moreover, further studies of solar radiation on a daily basis should be investigated to optimize the dependency of a heat pump. These additional researches could lead to a net-zero application in the future. Furthermore, due to the limited usage of desiccant system and dew-point evaporative system in building levels, most of the results and data are based on experiment results which are conducted in a small scale. Consequently, it doesn't guarantee that the system would perform as predicted in real life. Therefore, a prototype and further experiment is recommended.

APPENDIX – A

Grasshopper Script for Solar Radiation Simulation

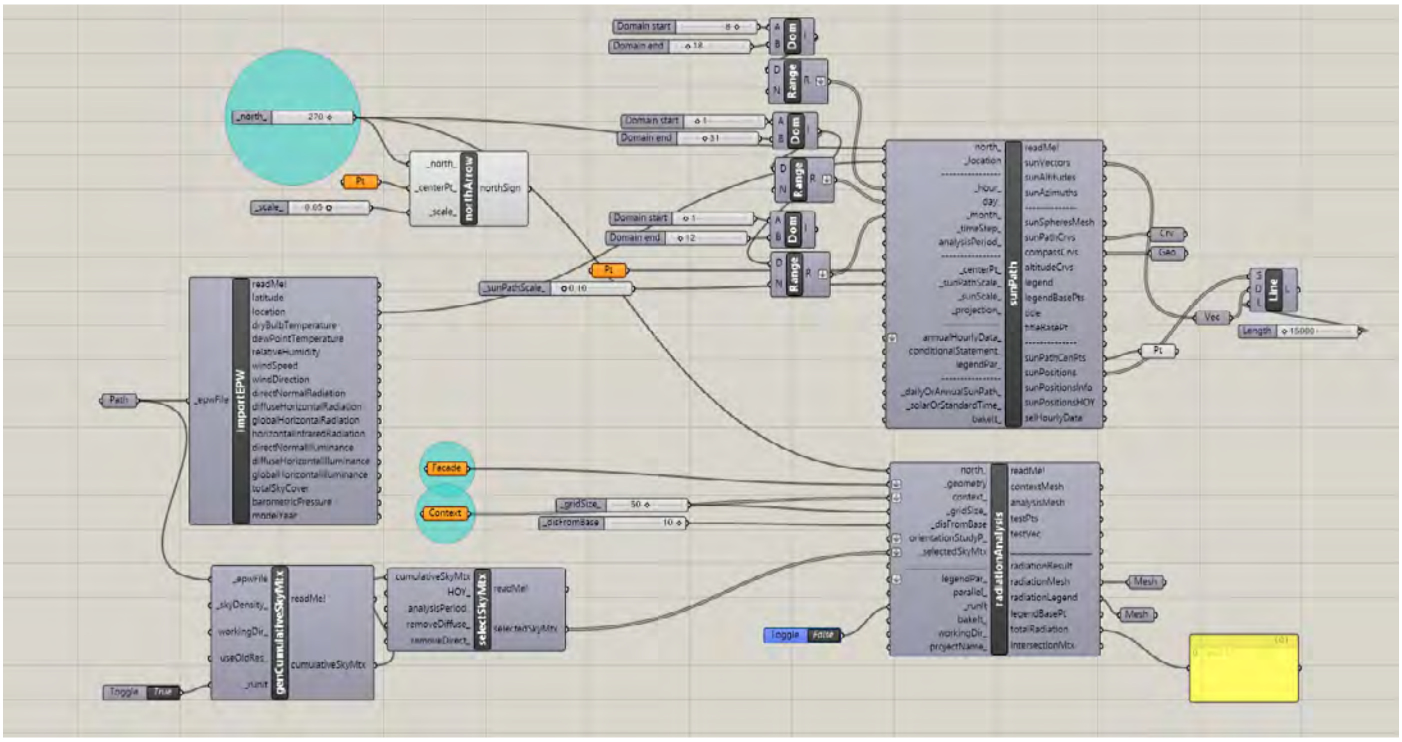


Figure A-1 : Primary Solar Radiation Simulation

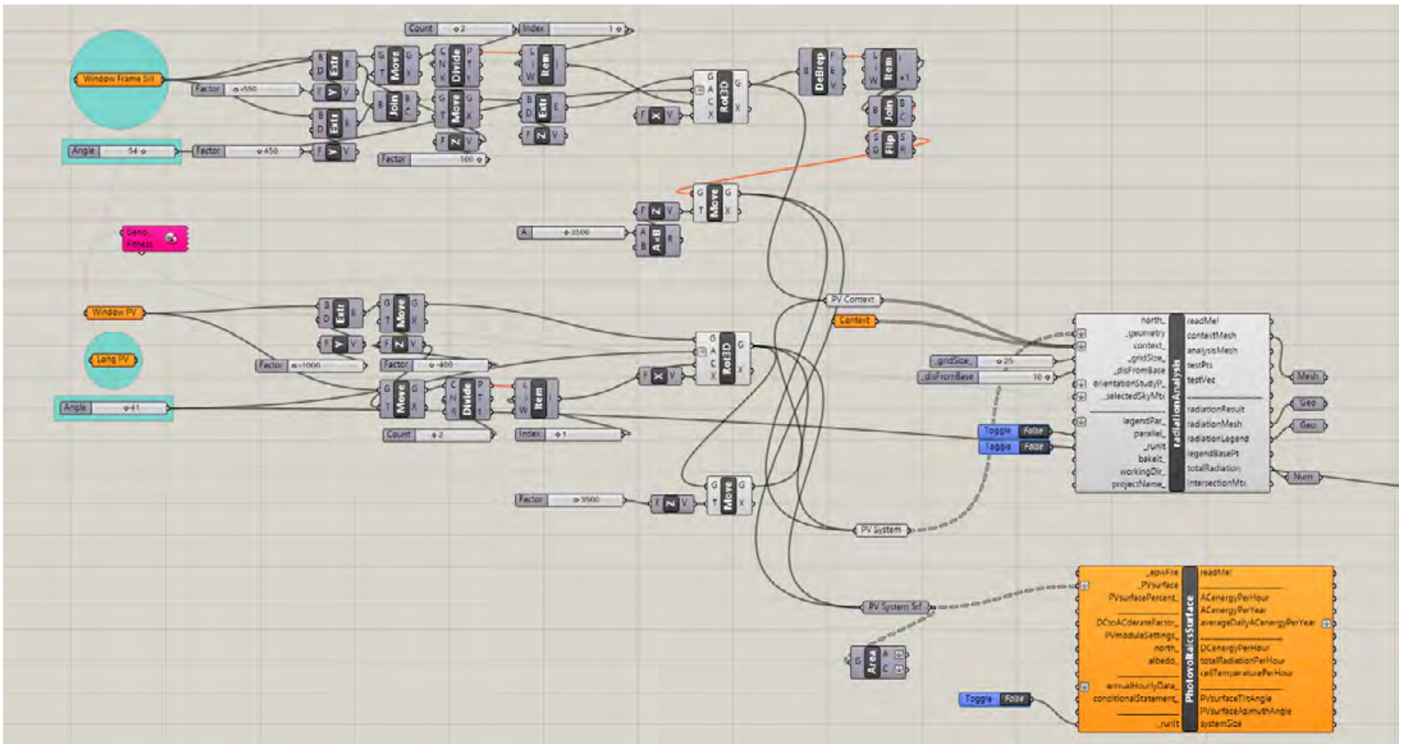


Figure A-2 : Horizontal PV configuration with Galapagos

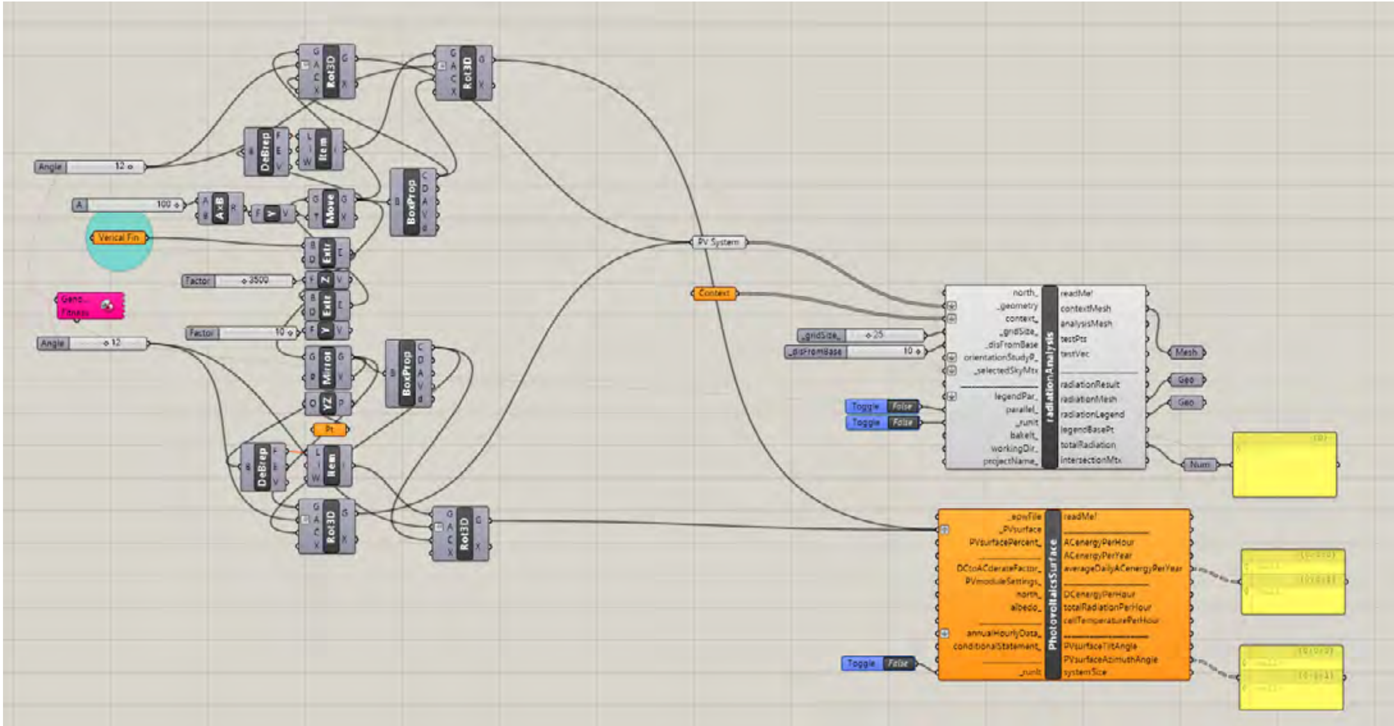


Figure A-3 : Vertical PV configuration with Galapagos

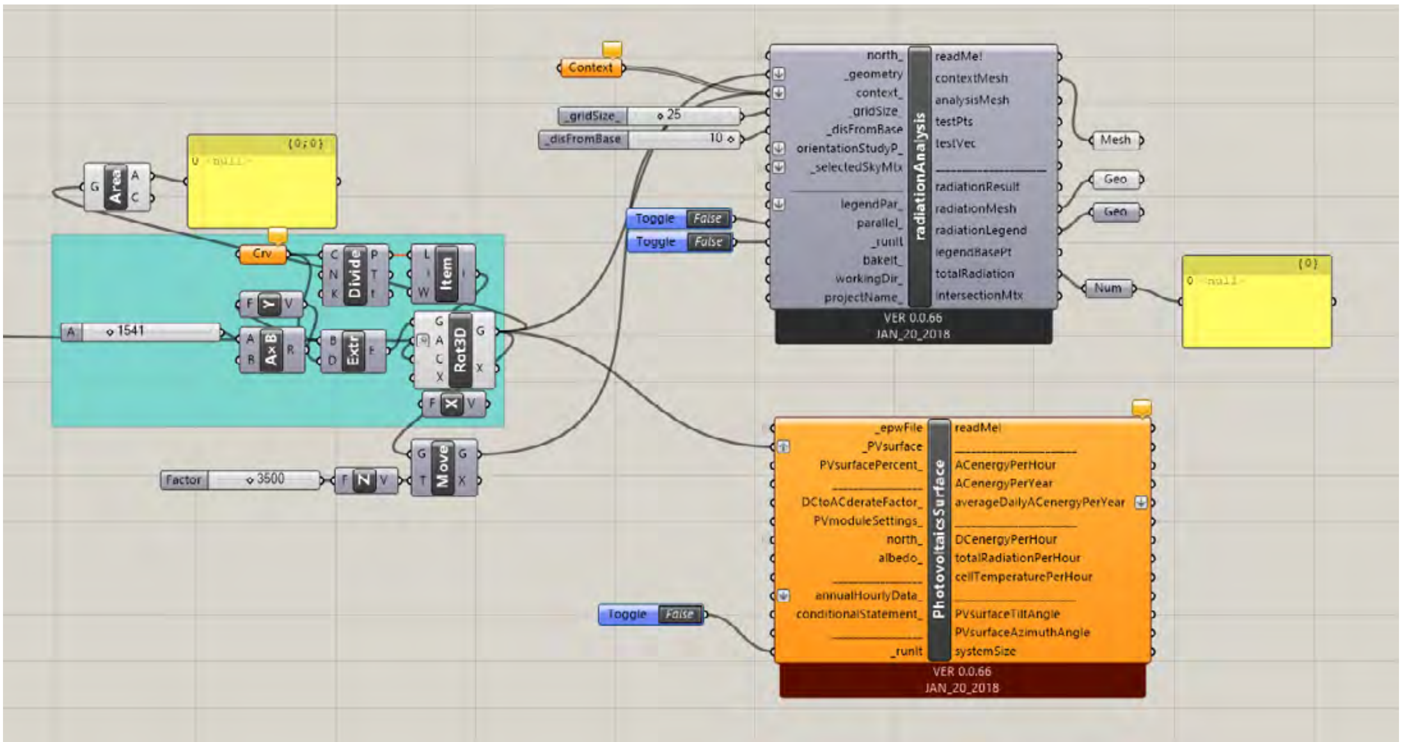


Figure A-4 : Performance of PV Panels

APPENDIX – B

Centralized System Requirement Schedule

Table B-1 : Screw Water Cooled Water Chiller Schedule

2 system working simoutaneously during operation

Unit No.	Energy Consumption		Chilled Water Production	
	kWh/Year	kWh/day (146days)	GPM	m^3/hr
CH/1	333,616	2,034.24	960	218.04
CH/2	408,102	2,488.43	960	218.04
CH/3	360,798	2,199.99	960	218.04
Average Consumption		**2,240.89 kWh/day**		

Table B-2 : Cooling Schedule Schedule

Unit No.	Energy Consumption		Chilled Water Requirement	
	kWh/Year	kWh/day (146days)	GPM	m^3/hr
CT/1	10,223	62.34	600	136.4
CT/2	9,813	59.84	600	136.4
CT/3	9,403	57.34	600	136.4
CT/5	9,698	59.13	600	136.4
CT/6	8,976	54.73	450	102.3
Average Consumption		**58.67 kWh/day**		**129.5 m^3/hr**

APPENDIX - C

Design Building Simulation Parameters

Table C-1 : Simulation Fixed Parameter

Overview			
Floor Area (m^2)	Floor Volume (m^3)	Occupancy (people / m^2)	Cooling Setpoint Operative Temp. (°C)
695	2430	0.11	25.5

Activity			Fixed Air Parameter	
Metabolic Rate (met)	Office Equipments (W/m^2)	Lighting (W/m^2)	Humidity Control (Dehumidification)	Cooling Limit Type
0.9	11.77	2.5	Constant Supply Humidity Ratio	Limit Flowrate|Capacity

Table C-2 : Existing Building Construction Parameter

Exterior Wall

Layer	Material	Thickness (mm)
Layer 1	Aluminium Cladding	0.4
Layer 2	Insulation with Metal Framing	100
Layer 3	Air Gap	300
Layer 4	Insulation with Metal Framing	100
Layer 5	Cement Board Panel	200
Layer 6	Aluminum Cladding	0.4

Opening

Layer	Material	Thickness (mm)
Layer 1	12mm Clear Float Glass	12
Layer 2	Air Gap	6
Layer 3	12mm Clear Float Glass	12

Table C-3 : Proposed Facade System Construction Parameter

Exterior Wall

Layer	Material	Thickness (mm)
Layer 1	Aluminium Cladding	0.4
Layer 2	Insulation with Metal Framing	200
Layer 3	Concrete Panel	100

Opening

Layer	Material	Thickness (mm)
Layer 1	12mm Clear Float Glass	12
Layer 2	Air Gap	6
Layer 3	12mm Clear Float Glass	12

APPENDIX - D

System Performance Calucations

Table D-1 : CCHE Performance Based on Difference Scenario

(Ge et al, 2017)

	Inlet Air Condition		Desiccant Performance	
	Air Temperature (°C)	Relative Humidity (%)	Cooling Capacity (Qa) (kW)	Dehumidification Performance (g kg)
1.1	25	70	0.75	2.5
1.2	***25***	***60***	***0.67***	***2.1***
1.3	25	50	0.47	1.5
2.1	***30***	***70***	***1.25***	***3.9***
2.2	***30***	***60***	***1.12***	***3.2***
2.3	***30***	***50***	***0.80***	***2.3***
3.1	35	70	1.63	4.6
3.2	***35***	***60***	***1.47***	***3.8***
3.3	35	50	1.00	2.7

Remark

The performance of the CCHEs is based on the experiments conducted by Ge.et al. (2017). However, due to the limited information on the variation of the studies, the data taken directly from the experiments are highlighted, while the other data are based on an assumption in proportion to the experiment results. These data are used in the calculations of CCHEs calculation in this thesis, including the one presented in Appendix D.

Appendix D-2 : Existing Building | Primary Calculation - Ambient 25°C | Unconditioned Room

CCHE Performance 25°C

$Q = M\,(Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	25.0	70.0	60.6	13.9	0.75	2.5	11.4	57.1	27.8	48.7
	27.8	48.7	57.1	11.4	0.80	2.3	9.1	53.4	30	34.5
	30	34.5	53.4	9.1	0.80	2.3	6.8	49.7	32.1	22.9
1.2	25.0	60.0	55.5	11.9	0.67	2.1	9.8	52.4	27.2	43.5
	27.2	43.5	52.4	9.8	0.47	1.5	8.3	50.2	28.8	33.6
	28.8	33.6	50.2	8.3	0.80	2.3	6.0	46.5	31.0	21.6
1.3	25.0	50.0	50.3	9.9	0.47	1.5	8.4	48.1	26.5	38.9
	26.5	38.9	48.1	8.4	0.47	1.5	6.9	45.9	28.1	29.2
	28.1	29.2	45.9	6.9	0.80	2.3	4.6	42.2	30.3	17.3

M-Cycle Performance 25°C

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	27.8	11.4	48.7	**19.6**	82
	30	9.1	34.5	**18.4**	69
	32.1	6.8	22.9	**17.1**	56.2
1.2	27.2	9.8	43.5	**18.5**	73.8
	28.8	8.3	33.6	**17.7**	65.9
	31	6.0	21.6	**16.4**	51.9
1.3	26.5	8.4	38.9	**17.4**	67.9
	28.1	6.9	29.2	**16.6**	58.8
	30.3	4.6	17.3	**15.4**	42.5

Moisture from Occupants

$\Delta C = P/n{*}v$

			P	n	V	ΔC	Abs. Humidity
	Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m^3)	Humidity Difference (g kg)	Absolute Humidity (g kg)
1.1	40	70	2800	2.35	2430	0.49	7.2
1.2	40	70	2800	2.09	2430	0.55	6.5
1.3	40	70	2800	1.84	2430	0.63	5.2

Return Air

$(T_{supply} - T_{return}) = Q/V\,C\rho$

	T_{supply}	Q	V	ΔT	Return Air
	Supply Air Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m^3/s)	Temperature Difference (°C)	Temperature (°C)
1.1	17.1	86.7	3.7	19.5	36.7
1.2	16.4	87.2	3.3	22.0	38.5
1.3	15.4	89.0	2.9	25.6	40.9

Appendix D-3 : Existing Building | Primary Calculation - Ambient 25°C | Conditioned Room

Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in}+Q_{re})$ $T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in}+Q_{re}$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m³/s)	Q_{re} Return Air (m³/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
1.1	0.18	0.42	25.0	36.7	13.9	7.2	33.2	9.2	29.0
1.2	0.18	0.42	25.0	38.5	11.4	6.5	34.4	8.0	23.6
1.3	0.18	0.42	25.0	40.9	9.1	5.2	36.2	6.4	17.2

1.1 **33.2** °C
Mixed Air Temperature
Abs.Humidity :9.2 g kg | RH : 29.0%

1.2 **34.4** °C
Mixed Air Temperature
Abs.Humidity : 8.0 g kg | RH : 23.6%

1.3 **36.2** °C
Mixed Air Temperature
Abs.Humidity : 6.4 g kg | RH : 17.2%

CCHE Performance 25°C - Mixed Air

$Q = M (Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	33.2	29.0	57.0	9.2	1.00	2.7	6.5	52.4	35.5	18.1
	35.5	18.1	52.4	6.5	1.00	2.7	3.8	47.7	37.7	9.4
	37.7	9.4	47.7	3.8	1.00	2.7	1.0	43.1	40.3	2.2
1.2	34.4	23.6	55.1	8.0	1.00	2.7	5.2	50.5	36.9	13.4
	36.9	13.4	50.5	5.2	1.00	2.7	2.5	45.8	39.1	5.8
	39.1	5.8	45.8	2.5	1.00	2.7	0.5	41.2	-	-
1.3	36.2	17.2	52.8	6.5	1.00	2.7	3.8	48.2	38.2	9.2
	38.2	9.2	48.2	3.8	1.00	2.7	1.0	43.5	40.7	2.1
	40.7	2.1	43.5	1.0	1.00	2.7	-1.7	38.9	-	-

M-Cycle Performance 25°C - Mixed Air

$T_{out} = 7.65 + 0.152\, T_{in} + 681\, w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	35.5	6.5	18.1	**17.5**	52.4
	37.7	3.8	9.4	**15.9**	34.1
	40.3	1.0	2.2	**14.5**	9.8
1.2	36.9	5.2	13.4	**16.8**	43.9
	39.1	2.5	5.8	**15.3**	23.3
1.3	38.2	3.8	9.2	**16.0**	33.8
	40.7	1.0	2.1	**14.5**	9.8

	1.1	1.2	1.3
No. of Desiccant System	1	1	1
Mixed Air Temperature (°C)	17.5	16.8	16.0
Abs.Humidity	6.5 g kg	5.2 g kg	3.8 g kg
RH	52.4%	43.9%	33.8%

Moisture from Occupants

$$\Delta C = P/n*v$$

	Moisture from Occupant (g/hr)	Total Occupant (person)	P Vapour Production (g/h)	n Ventilation Rate (ach)	V Room Volume (m³)	ΔC Humidity Difference (g kg)	Abs. Humidity Absolute Humidity (g kg)
1.1	40	70	2800	2.51	2430	0.46	7.0
1.2	40	70	2800	2.23	2430	0.52	5.8
1.3	40	70	2800	1.97	2430	0.58	4.3

Return Air

$$(T_{supply} - T_{return}) = Q/V\,C\rho$$

	T_{supply} Supply Air Temperature (°C)	Q Design Cooling Load (kW)	V Volume Flowrate (m³/s)	ΔT Temperature Difference (°C)	Return Air Temperature (°C)
1.1	17.5	90.6	4.0	19.1	36.6
1.2	16.8	93.4	3.5	22.2	39.0
1.3	16.0	95.9	3.1	25.7	41.7

Mixed Air

$$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$$

$$T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m³/s)	Q_{re} Return Air (m³/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
1.1	0.18	0.42	25.0	36.6	11.4	7.0	33.1	8.3	26.4
1.2	0.18	0.42	25.0	39.0	9.8	5.8	34.8	7.0	20.3
1.3	0.18	0.42	25.0	41.7	8.4	4.3	36.7	5.6	14.6

Appendix D-5 : Existing Building | Primary Calculation - Ambient 30°C | Unconditioned Room

CCHE Performance 30°C

$Q = M(Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	30.0	70.0	78.2	18.8	1.25	3.9	14.9	72.4	34.0	44.5
	34.0	44.5	72.4	14.9	1.00	2.7	12.2	67.6	36.1	32.7
	36.1	32.7	67.6	12.2	1.00	2.7	9.4	62.8	38.4	22.3
1.2	30.0	60.0	71.2	16.0	1.12	3.2	12.8	66.0	33.0	40.6
	33.0	40.6	66.0	12.8	1.00	2.7	10.1	61.2	35.1	28.6
	35.1	28.6	61.2	10.1	1.00	2.7	7.3	56.4	37.4	18.3
1.3	30.0	50.0	64.2	13.3	0.80	2.3	11.0	60.5	32.1	36.7
	32.1	36.7	60.5	11.0	0.80	2.3	8.7	56.8	34.3	25.8
	34.3	25.8	56.8	8.7	1.00	2.7	6.0	52.0	36.4	15.9

M-Cycle Performance 30°C

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	34	14.9	44.5	**23.0**	84.3
	36.1	12.2	32.7	**21.4**	76.5
	38.4	9.4	22.3	**19.9**	64.9
1.2	33.0	12.8	40.6	**21.4**	73.8
	35.1	10.1	28.6	**19.8**	70.1
	37.4	7.3	18.3	**18.3**	55.9
1.3	32.1	11.0	36.7	**20.0**	67.9
	34.3	8.7	25.8	**18.8**	64.4
	36.4	6.0	15.9	**17.2**	49.3

Moisture from Occupants

$\Delta C = P/n{*}v$

			P	n	V	ΔC	Abs. Humidity
	Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m^3)	Humidity Difference (g kg)	Absolute Humidity (g kg)
1.1	40	70	2800	4.38	2430	0.26	9.7
1.2	40	70	2800	2.92	2430	0.39	7.7
1.3	40	70	2800	2.41	2430	0.48	6.4

Return Air

$(T_{supply} - T_{return}) = Q/V\,C\rho$

	T_{supply}	Q	V	ΔT	Return Air
	Supply Air Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m^3/s)	Temperature Difference (°C)	Temperature (°C)
1.1	19.9	88.6	6.9	10.7	30.6
1.2	18.3	90.9	4.6	16.5	34.8
1.3	17.2	91.6	3.8	20.1	37.3

Appendix D-6 : Existing Building | Primary Calculation - Ambient 30°C | Conditioned Room

Mixed Air - 30°C

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$

$T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m^3/s)	Q_{re} Return Air (m^3/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
2.1	0.18	0.42	30.0	30.6	18.8	9.7	30.4	12.4	45.6
2.2	0.18	0.42	30.0	34.8	14.9	7.7	33.4	9.9	30.8
2.3	0.18	0.42	30.0	37.3	12.2	6.4	35.1	8.2	23.3

2.1 **30.4** °C
Mixed Air Temperature
Abs.Humidity : 12.4 g kg | RH : 45.6%

2.2 **33.4** °C
Mixed Air Temperature
Abs.Humidity : 9.9 g kg | RH : 30.8%

2.3 **35.1** °C
Mixed Air Temperature
Abs.Humidity : 8.2 g kg | RH : 23.3%

CCHE Performance 30°C - Mixed Air

$Q = M (Ha_{in} - Ha_{out})$

$M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
2.1	30.4	45.6	62.3	12.4	0.80	2.3	10.1	58.6	32.5	33
	32.5	33.0	58.6	10.1	0.80	2.3	7.8	54.9	34.7	22.7
	34.7	22.7	54.9	7.8	1.00	2.7	5.1	50.2	36.9	13.2
2.2	33.4	30.8	59.0	9.9	0.80	2.3	7.6	55.3	35.6	21.0
	35.6	21.0	55.3	7.6	0.80	2.3	5.3	51.6	37.8	13.1
	37.8	13.1	51.6	5.3	1.00	2.7	2.5	46.9	40.2	5.4
2.3	35.1	23.3	56.3	8.2	0.80	2.3	5.9	52.6	37.2	15.0
	37.2	15.0	52.6	5.9	1.00	2.7	3.1	47.9	39.7	6.9
	39.7	6.9	47.9	3.1	1.00	2.7	0.4	43.3	42.0	0.8

M-Cycle Performance 30°C - Mixed Air

$T_{out} = 7.65 + 0.152\, T_{in} + 681\, w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
2.1	32.5	10.1	33.0	**19.5**	71.4
	34.7	7.8	22.7	**18.3**	59.7
	36.9	5.1	13.2	**16.7**	43.3
2.2	35.6	7.6	21.0	**18.2**	58.5
	37.8	5.3	13.1	**17.0**	44.2
	40.2	2.5	5.4	**15.5**	49.5
2.3	37.2	5.9	15.0	**17.3**	48.2
	39.7	3.1	6.9	**15.8**	28.0
	42.0	0.4	0.8	**14.3**	4.0

	No. of Desiccant System : 3	No. of Desiccant System : 2	No. of Desiccant System : 1
	1.1	**1.2**	**1.3**
Mixed Air Temperature	16.7 °C	17.0 °C	17.3 °C
	Abs.Humidity : 5.1 g kg | RH : 43.3%	Abs.Humidity :5.3g kg | RH : 44.2%	Abs.Humidity : 5.9 g kg | RH : 48.2%

Moisture from Occupants

$$\Delta C = P/n*v$$

			P	n	V	ΔC	Abs. Humidity
	Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m^3)	Humidity Difference (g kg)	Absolute Humidity (g kg)
1.1	40	70	2800	2.22	2430	0.52	5.6
1.2	40	70	2800	2.30	2430	0.50	5.8
1.3	40	70	2800	2.42	2430	0.48	6.3

Return Air

$$(T_{supply} - T_{return}) = Q/V\,C\rho$$

	T_{supply}	Q	V	ΔT	Return Air
	Supply Air Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m^3/s)	Temperature Difference (°C)	Temperature (°C)
1.1	16.7	93.4	3.5	22.2	39.0
1.2	17.0	94.1	3.6	21.7	38.7
1.3	17.3	92.7	3.8	20.3	37.6

Mixed Air

$$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$$

$$T_{mixed} = (Q_{in}T_{in} + Q_{re}T_{re})/Q_{in} + Q_{re}$$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in}	Q_{re}	T_{in}	T_{re}	X_{in}	X_{re}	T_{mixed}	X_{mixed}	RH
	Inlet Air (m^3/s)	Return Air (m^3/s)	Inlet Air (°C)	Return Air (°C)	Inlet Air (kg kg)	Return Air (kg kg)	Mixed Air (°C)	Mixed Air (g kg)	Relative Humidity (%)
1.1	0.18	0.42	30.0	39.0	18.8	5.6	36.3	9.6	25.5
1.2	0.18	0.42	30.0	38.7	14.9	5.8	36.1	8.5	22.8
1.3	0.18	0.42	30.0	37.6	12.2	6.3	35.3	8.1	22.8

CCHE Performance 35°C

$Q = M\,(Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	35.0	70.0	99.8	25.2	1.63	4.6	20.6	92.2	39.0	46.5
	39.0	46.5	92.2	20.6	1.00	2.7	17.8	87.4	41.3	35.7
	41.3	35.7	87.4	17.8	1.00	2.7	15.1	82.7	43.5	27.1
1.2	35.0	60.0	90.2	21.4	1.47	3.8	17.6	83.4	37.9	42.2
	37.9	42.2	83.4	17.6	1.00	2.7	14.9	78.6	40.0	32.1
	40.0	32.1	78.6	14.9	1.00	2.7	12.1	73.8	42.3	23.1
1.3	35.0	50.0	80.8	17.8	1.00	2.7	15.1	76.0	37.0	38.3
	37.0	38.3	76.0	15.1	1.00	2.7	12.3	71.2	39.3	27.6
	39.3	27.6	71.2	12.3	1.00	2.7	9.6	66.5	41.5	19.3

M-Cycle Performance 35°C

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	39.0	20.6	46.5	**27.6**	87.9
	41.3	17.8	35.7	**26.1**	83.3
	43.5	15.1	27.1	**24.5**	78.1
1.2	37.9	17.6	42.2	**25.4**	85.9
	40.0	14.9	32.1	**23.9**	79.9
	42.3	12.1	23.1	**22.3**	68.4
1.3	37.0	15.1	38.3	**23.5**	82.9
	39.3	12.3	27.6	**22.0**	74.3
	41.5	9.6	19.3	**20.5**	63.8

Moisture from Occupants

$\Delta C = P/n*v$

			P	n	V	ΔC	Abs. Humidity
	Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m^3)	Humidity Difference (g kg)	Absolute Humidity (g kg)
1.1	40	70	2800	737.1	2430	0.001	15.1
1.2	40	70	2800	15.7	2430	0.1	12.2
1.3	40	70	2800	5.4	2430	0.2	9.7

Return Air

$(T_{supply} - T_{return}) = Q/V\,C\rho$

	T_{supply}	Q	V	ΔT	Return Air
	Supply Air Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m^3/s)	Temperature Difference (°C)	Temperature (°C)
1.1	24.5	216.0	1152.0	0.2	24.7
1.2	22.3	68.0	24.8	3.0	25.4
1.3	20.5	70.0	8.5	9.0	29.5

Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in}+Q_{re})$ $T_{mixed}=(Q_{in} T_{in}+Q_{re} T_{re})/Q_{in}+Q_{re}$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m³/s)	Q_{re} Return Air (m³/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
1.1	0.18	0.42	35.0	24.7	25.2	15.1	27.8	18.1	76.6
1.2	0.18	0.42	35.0	25.4	21.4	12.2	28.3	15.0	62.0
1.3	0.18	0.42	35.0	29.5	17.8	9.8	31.2	12.2	42.9

1.1 **27.8** °C
Mixed Air Temperature
Abs.Humidity : 18.1 g kg | RH : 76.6%

1.2 **28.3** °C
Mixed Air Temperature
Abs.Humidity : 15.0 g kg | RH : 62.0%

1.3 **31.2** °C
Mixed Air Temperature
Abs.Humidity : 12.2 g kg | RH : 42.9%

CCHE Performance 35°C - Mixed Air

$Q = M (Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	27.8	76.6	74.2	18.1	1.25	3.9	14.2	68.4	31.9	47.9
	31.9	47.9	68.4	14.2	0.80	2.3	11.9	64.7	34.0	35.7
	34.0	35.7	64.7	11.9	1.00	2.7	9.2	60.0	36.2	24.6
1.2	28.3	62.0	66.8	15.0	1.12	3.2	11.8	61.6	31.2	41.4
	31.2	41.4	61.6	11.8	0.80	2.3	9.5	57.9	33.4	29.7
	33.4	29.7	57.9	9.5	1.00	2.7	6.7	53.2	35.8	18.4
1.3	31.2	42.9	62.6	12.2	0.80	2.3	9.9	58.9	33.3	31.0
	33.3	31.0	58.9	9.9	1.00	2.7	7.2	54.2	35.5	20.0
	35.5	20.0	54.2	7.2	1.00	2.7	4.4	49.6	38.1	10.7

M-Cycle Performance 35°C - Mixed Air

$T_{out} = 7.65 + 0.152\, T_{in} + 681\, w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	31.9	14.2	47.9	**22.2**	84.5
	34.0	11.9	35.7	**20.9**	76.9
	36.2	9.2	24.6	**19.4**	65.5
1.2	31.2	11.8	41.4	**20.4**	78.7
	33.4	9.5	29.7	**19.2**	68.5
	35.8	6.7	18.4	**17.7**	53.3
1.3	33.3	9.9	31.0	**19.5**	70.0
	35.5	7.2	20.0	**17.9**	56.5
	38.1	4.4	10.7	**16.5**	37.5

	No. of Desiccant System : 3		No. of Desiccant System : 3		No. of Desiccant System : 3
3.1	**19.4** °C Mixed Air Temperature Abs.Humidity : 9.2 g kg | RH : 65.5%	**3.2**	**17.7** °C Mixed Air Temperature Abs.Humidity : 6.7 g kg | RH : 53.3%	**3.3**	**16.5** °C Mixed Air Temperature Abs.Humidity : 4.4 g kg | RH : 37.5%

Moisture from Occupants

$\Delta C = P/n*v$

	Moisture from Occupant (g/hr)	Total Occupant (person)	**P** Vapour Production (g/h)	**n** Ventilation Rate (ach)	**V** Room Volume (m^3)	**ΔC** Humidity Difference (g kg)	**Abs. Humidity** Absolute Humidity (g kg)
3.3	40	70	2800	2.60	2430	0.44	7.2
3.3	40	70	2800	2.60	2430	0.44	4.9

Return Air

$(T_{supply} - T_{return}) = Q/V C\rho$

	T_{supply} Supply Air Temperature (°C)	**Q** Design Cooling Load (kW)	**V** Volume Flowrate (m^3/s)	**ΔT** Temperature Difference (°C)	**Return Air** Temperature (°C)
3.3	17.7	90.68	4.1	18.4	36.1
3.3	16.5	90.69	4.1	18.4	34.9

Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$ $T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$

	Air Volume		**Air Temperature**		**Humidity Ratio**		**Mixed Air**		
	Q_{in} Inlet Air (m^3/s)	**Q_{re}** Return Air (m^3/s)	**T_{in}** Inlet Air (°C)	**T_{re}** Return Air (°C)	**X_{in}** Inlet Air (kg kg)	**X_{re}** Return Air (kg kg)	**T_{mixed}** Mixed Air (°C)	**X_{mixed}** Mixed Air (g kg)	**RH** Relative Humidity (%)
3.3	0.18	0.42	35.0	36.1	17.8	7.2	35.8	10.4	28.3
3.3	0.18	0.42	35.0	34.9	17.8	4.9	34.9	8.8	15.3

CCHE Performance 25°C

$Q = M\,(Ha_{in} - Ha_{out})$ M = 0.22 kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	25.0	70.0	60.6	13.9	0.75	2.5	11.4	57.1	27.8	48.7
	27.8	48.7	57.1	11.4	0.80	2.3	9.1	53.4	30	34.5
	30	34.5	53.4	9.1	0.80	2.3	6.8	49.7	32.1	22.9
1.2	25.0	60.0	55.5	11.9	0.67	2.1	9.8	52.4	27.2	43.5
	27.2	43.5	52.4	9.8	0.47	1.5	8.3	50.2	28.8	33.6
	28.8	33.6	50.2	8.3	0.80	2.3	6.0	46.5	31.0	21.6
1.3	25.0	50.0	50.3	9.9	0.47	1.5	8.4	48.1	26.5	38.9
	26.5	38.9	48.1	8.4	0.47	1.5	6.9	45.9	28.1	29.2
	28.1	29.2	45.9	6.9	0.80	2.3	4.6	42.2	30.3	17.3

M-Cycle Performance 25°C

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	27.8	11.4	48.7	**19.6**	82
	30	9.1	34.5	**18.4**	69
	32.1	6.8	22.9	**17.1**	56.2
1.2	27.2	9.8	43.5	**18.5**	73.8
	28.8	8.3	33.6	**17.7**	65.9
	31	6.0	21.6	**16.4**	51.9
1.3	26.5	8.4	38.9	**17.4**	67.9
	28.1	6.9	29.2	**16.6**	58.8
	30.3	4.6	17.3	**15.4**	42.5

Moisture from Occupants

$\Delta C = P/n{*}v$

			P	n	V	ΔC	Abs. Humidity
	Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m^3)	Humidity Difference (g kg)	Absolute Humidity (g kg)
1.1	40	70	2800	1.62	2430	0.71	7.5
1.2	40	70	2800	1.46	2430	0.78	6.8
1.3	40	70	2800	1.28	2430	0.89	5.5

Return Air

$(T_{supply} - T_{return}) = Q/V\,C\rho$

	T_{supply}	Q	V	ΔT	Return Air
	Supply Air Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m^3/s)	Temperature Difference (°C)	Temperature (°C)
1.1	17.1	67.8	2.6	22.2	39.3
1.2	16.4	68.1	2.3	24.7	41.1
1.3	15.4	69.5	2.0	28.7	44.0

Appendix D-12 : Proposed Integration | First Assumption Calculation - Ambient 25°C | Conditioned Room

Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$ $T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m^3/s)	Q_{re} Return Air (m^3/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
1.1	0.18	0.42	25.0	39.3	13.9	7.5	35.0	9.4	26.8
1.2	0.18	0.42	25.0	41.1	11.4	6.8	36.3	8.2	21.8
1.3	0.18	0.42	25.0	44.0	9.1	5.5	38.3	6.5	15.5

1.1 **35.0** °C
Mixed Air Temperature
Abs.Humidity : 9.4 g kg | RH : 26.8%

1.2 **36.3** °C
Mixed Air Temperature
Abs.Humidity : 8.2 g kg | RH : 21.8%

1.3 **38.3** °C
Mixed Air Temperature
Abs.Humidity : 6.5 g kg | RH : 15.5%

CCHE Performance 25°C - Mixed Air

$Q = M (Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	35.0	26.8	59.3	9.4	1.03	2.7	6.7	54.5	37.1	17.1
	37.1	17.1	54.5	6.7	1.03	2.7	3.9	49.7	39.4	8.8
	39.4	8.8	49.7	3.9	1.03	2.7	1.2	45.0	41.7	2.4
1.2	36.3	21.8	57.6	8.2	1.03	2.7	5.4	52.8	38.7	12.7
	38.7	12.7	52.8	5.4	1.03	2.7	2.7	48.0	40.8	5.7
	40.8	5.7	48.0	2.7	1.03	2.7	-0.1	43.3	-	-
1.3	38.3	15.5	55.2	6.5	1.03	2.7	3.8	50.4	40.4	8.2
	40.4	8.2	50.4	3.8	1.03	2.7	1.1	45.6	42.5	2.1
	42.5	2.1	45.6	1.1	1.03	2.7	-1.7	40.9	-	-

M-Cycle Performance 25°C - Mixed Air

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	37.1	6.7	17.1	**17.8**	53.0
	39.4	3.9	8.8	**16.3**	34.1
	41.7	1.2	2.4	**14.8**	11.6
1.2	38.7	5.4	12.7	**17.2**	44.4
	40.8	2.7	5.7	**15.7**	24.6
1.3	40.4	3.8	8.2	**16.4**	33.0
	42.5	1.1	2.1	**14.8**	10.6

Appendix D-13 : Proposed Integration | Secondary Calculation - Ambient 25°C | Conditioned Room

	3.1	3.2	3.3
No. of Desiccant System	2	1	1
Mixed Air Temperature (°C)	17.8	17.2	16.4
Abs.Humidity	6.1 g kg	6.6 g kg	5.0 g kg
RH	53.0%	44.4%	33.0%

Moisture from Occupants

$$\Delta C = P/n*v$$

	Moisture from Occupant (g/hr)	Total Occupant (person)	P Vapour Production (g/h)	n Ventilation Rate (ach)	V Room Volume (m³)	ΔC Humidity Difference (g kg)	Abs. Humidity Absolute Humidity (g kg)
1.1	40	70	2800	1.84	2430	0.63	7.3
1.2	40	70	2800	1.65	2430	0.70	6.1
1.3	40	70	2800	1.46	2430	0.79	4.6

Return Air

$$(T_{supply} - T_{return}) = Q/V\,C\rho$$

	T_{supply} Supply Air Temperature (°C)	Q Design Cooling Load (kW)	V Volume Flowrate (m³/s)	ΔT Temperature Difference (°C)	Return Air Temperature (°C)
1.1	17.8	71.4	2.90	20.5	38.4
1.2	17.2	69.1	2.60	22.1	39.4
1.3	16.4	72.1	2.30	26.1	42.5

Mixed Air

$$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re}) \qquad T_{mixed} = (Q_{in}T_{in} + Q_{re}T_{re})/Q_{in} + Q_{re}$$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m³/s)	Q_{re} Return Air (m³/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
1.1	0.18	0.42	25.0	38.4	11.4	6.7	34.3	8.5	25.2
1.2	0.18	0.42	25.0	39.4	9.8	7.3	35.1	7.2	20.5
1.3	0.18	0.42	25.0	42.5	8.4	5.8	37.3	5.7	14.4

Appendix D-14 : Proposed Integration | Primary Calculation - Ambient 30°C | Unconditioned Room

CCHE Performance 30°C

$Q = M\,(Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	30.0	70.0	78.2	18.8	1.25	3.9	14.9	72.4	34.0	44.5
	34.0	44.5	72.4	14.9	1.00	2.7	12.2	67.6	36.1	32.7
	36.1	32.7	67.6	12.2	1.00	2.7	9.4	62.8	38.4	22.3
1.2	30.0	60.0	71.2	16.0	1.12	3.2	12.8	66.0	33.0	40.6
	33.0	40.6	66.0	12.8	1.00	2.7	10.1	61.2	35.1	28.6
	35.1	28.6	61.2	10.1	1.00	2.7	7.3	56.4	37.4	18.3
1.3	30.0	50.0	64.2	13.3	0.80	2.3	11.0	60.5	32.1	36.7
	32.1	36.7	60.5	11.0	0.80	2.3	8.7	56.8	34.3	25.8
	34.3	25.8	56.8	8.7	1.00	2.7	6.0	52.0	36.4	15.9

M-Cycle Performance 30°C

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	34	14.9	44.5	**23.0**	84.3
	36.1	12.2	32.7	**21.4**	76.5
	38.4	9.4	22.3	**19.9**	64.9
1.2	33.0	12.8	40.6	**21.4**	73.8
	35.1	10.1	28.6	**19.8**	70.1
	37.4	7.3	18.3	**18.3**	55.9
1.3	32.1	11.0	36.7	**20.0**	67.9
	34.3	8.7	25.8	**18.8**	64.4
	36.4	6.0	15.9	**17.2**	49.3

Moisture from Occupants

$\Delta C = P/n{*}v$

			P	n	V	ΔC	Abs. Humidity
	Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m^3)	Humidity Difference (g kg)	Absolute Humidity (g kg)
1.1	40	70	2800	2.79	2430	0.41	9.8
1.2	40	70	2800	1.97	2430	0.58	7.9
1.3	40	70	2800	1.65	2430	0.70	6.7

Return Air

$(T_{supply} - T_{return}) = Q/V\,C\rho$

	T_{supply}	Q	V	ΔT	Return Air
	Supply Air Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m^3/s)	Temperature Difference (°C)	Temperature (°C)
1.1	19.9	69.2	4.4	13.1	33.0
1.2	18.3	71.2	3.1	19.1	37.5
1.3	17.2	71.6	2.6	23.0	40.2

Mixed Air - 30°C

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$

$T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m^3/s)	Q_{re} Return Air (m^3/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
2.1	0.18	0.42	30.0	33.0	18.8	9.8	32.1	12.5	41.7
2.2	0.18	0.42	30.0	37.5	14.9	7.9	35.2	10.0	28.2
2.3	0.18	0.42	30.0	40.2	12.2	6.7	37.1	8.3	21.1

2.1 **32.1** °C
Mixed Air Temperature
Abs.Humidity : 12.5 g kg | RH : 41.7%

2.2 **35.2** °C
Mixed Air Temperature
Abs.Humidity : 10.0 g kg | RH : 28.2%

2.3 **37.1** °C
Mixed Air Temperature
Abs.Humidity : 8.3 g kg | RH : 21.1%

CCHE Performance 30°C - Mixed Air

$Q = M (Ha_{in} - Ha_{out})$

$M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
2.1	32.1	41.7	64.3	12.5	0.80	2.3	10.2	60.6	34.2	30.3
	34.2	30.3	60.6	10.2	1.03	2.7	7.5	55.8	36.3	19.9
	36.3	19.9	55.8	7.5	1.03	2.7	4.8	51.0	38.4	11.4
2.2	35.2	28.2	61.1	10.0	1.03	2.7	7.3	56.3	37.3	18.4
	37.3	18.4	56.3	7.3	1.03	2.7	4.5	51.5	39.7	10.0
	39.7	10.0	51.5	4.5	1.03	2.7	1.8	46.8	41.9	3.6
2.3	37.1	21.1	58.6	8.3	1.03	2.7	5.6	53.8	39.2	12.8
	39.2	12.8	53.8	5.6	1.03	2.7	2.8	49.0	41.5	5.7
	41.5	5.7	49.0	2.8	1.03	2.7	0.1	44.3	43.8	0.2

M-Cycle Performance 30°C - Mixed Air

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
2.1	34.2	10.2	30.3	**19.8**	70.8
	36.3	7.5	19.9	**18.3**	57.4
	38.4	4.8	11.4	**16.7**	40.8
2.2	37.3	7.3	18.4	**18.3**	55.9
	39.7	4.5	10.0	**16.8**	38.0
	41.9	1.8	3.6	**15.2**	33.8
2.3	39.2	5.6	12.8	**17.4**	45.5
	41.5	2.8	5.7	**15.9**	25.1
	43.8	0.1	0.2	**14.4**	1.0

Appendix D-16 : Proposed Integration | Secondary Calculation - Ambient 30°C | Conditioned Room

	No. of Desiccant System : 3	No. of Desiccant System : 2	No. of Desiccant System : 1
	1.1 16.7 °C	**1.2** 16.8 °C	**1.3** 17.4 °C
	Mixed Air Temperature	Mixed Air Temperature	Mixed Air Temperature
	Abs.Humidity : 5.3 g kg | RH : 40.8%	Abs.Humidity : 5.4 g kg | RH : 38.0%	Abs.Humidity : 6.0 g kg | RH : 45.5%

Moisture from Occupants

$\Delta C = P/n*v$

	Moisture from Occupant (g/hr)	Total Occupant (person)	P Vapour Production (g/h)	n Ventilation Rate (ach)	V Room Volume (m^3)	ΔC Humidity Difference (g kg)	Abs. Humidity Absolute Humidity (g kg)
1.1	40	70	2800	1.52	2430	0.76	6.0
1.2	40	70	2800	1.52	2430	0.76	6.2
1.3	40	70	2800	1.71	2430	0.67	6.7

Return Air

$(T_{supply} - T_{return}) = Q/V\,C\rho$

	T_{supply} Supply Air Temperature (°C)	Q Design Cooling Load (kW)	V Volume Flowrate (m^3/s)	ΔT Temperature Difference (°C)	Return Air Temperature (°C)
1.1	16.7	72.2	2.4	25.1	41.8
1.2	16.8	72.3	2.4	25.1	41.9
1.3	17.4	72.6	2.7	22.4	39.8

Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$ $T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m^3/s)	Q_{re} Return Air (m^3/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
1.1	0.18	0.42	30.0	41.8	18.8	6.0	38.3	9.9	23.6
1.2	0.18	0.42	30.0	41.9	14.9	6.2	38.3	8.8	21.0
1.3	0.18	0.42	30.0	39.8	12.2	6.7	36.9	8.3	21.4

Appendix D-17 : Proposed Integration | Primary Calculation - Ambient 35°C | Unconditioned Room

CCHE Performance 35°C

$Q = M (Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	35.0	70.0	99.8	25.2	1.63	4.6	20.6	92.2	39.0	46.5
	39.0	46.5	92.2	20.6	1.00	2.7	17.8	87.4	41.3	35.7
	41.3	35.7	87.4	17.8	1.00	2.7	15.1	82.7	43.5	27.1
1.2	35.0	60.0	90.2	21.4	1.47	3.8	17.6	83.4	37.9	42.2
	37.9	42.2	83.4	17.6	1.00	2.7	14.9	78.6	40.0	32.1
	40.0	32.1	78.6	14.9	1.00	2.7	12.1	73.8	42.3	23.1
1.3	35.0	50.0	80.8	17.8	1.00	2.7	15.1	76.0	37.0	38.3
	37.0	38.3	76.0	15.1	1.00	2.7	12.3	71.2	39.3	27.6
	39.3	27.6	71.2	12.3	1.00	2.7	9.6	66.5	41.5	19.3

M-Cycle Performance 35°C

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	39.0	20.6	46.5	**27.6**	87.9
	41.3	17.8	35.7	**26.1**	83.3
	43.5	15.1	27.1	**24.5**	78.1
1.2	37.9	17.6	42.2	**25.4**	85.9
	40.0	14.9	32.1	**23.9**	79.9
	42.3	12.1	23.1	**22.3**	68.4
1.3	37.0	15.1	38.3	**23.5**	82.9
	39.3	12.3	27.6	**22.0**	74.3
	41.5	9.6	19.3	**20.5**	63.8

Moisture from Occupants

$\Delta C = P/n*v$

			P	n	V	ΔC	Abs. Humidity
	Moisture from Occupant (g/hr)	Total Occupant (person)	Vapour Production (g/h)	Ventilation Rate (ach)	Room Volume (m^3)	Humidity Difference (g kg)	Absolute Humidity (g kg)
1.1	40	70	2800	167.36	2430	0.007	15.1
1.2	40	70	2800	7.3	2430	0.2	12.3
1.3	40	70	2800	3.3	2430	0.3	9.9

Return Air

$(T_{supply} - T_{return}) = Q/V\,C\rho$

	T_{supply}	Q	V	ΔT	Return Air
	Supply Air Temperature (°C)	Design Cooling Load (kW)	Volume Flowrate (m^3/s)	Temperature Difference (°C)	Temperature (°C)
1.1	24.5	102.6	263.7	0.3	24.9
1.2	22.3	69.0	11.5	5.0	27.4
1.3	20.5	72.0	5.2	11.5	32.0

Appendix D-18 : Proposed Integration | Primary Calculation - Ambient 35°C | Conditioned Room

Mixed Air

$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$ $T_{mixed} = (Q_{in} T_{in} + Q_{re} T_{re})/Q_{in} + Q_{re}$

	Air Volume		Air Temperature		Humidity Ratio		Mixed Air		
	Q_{in} Inlet Air (m^3/s)	Q_{re} Return Air (m^3/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	RH Relative Humidity (%)
1.1	0.18	0.42	35.0	24.9	25.2	15.1	27.9	18.1	76.2
1.2	0.18	0.42	35.0	27.4	21.4	12.3	29.7	15.0	57.2
1.3	0.18	0.42	35.0	32.0	17.8	9.9	32.9	12.3	39.2

1.1 **27.9** °C
Mixed Air Temperature
Abs.Humidity : 18.1 g kg | RH : 76.2%

1.2 **29.7** °C
Mixed Air Temperature
Abs.Humidity : 15.0 g kg | RH : 57.2%

1.3 **32.9** °C
Mixed Air Temperature
Abs.Humidity : 12.3 g kg | RH : 39.2%

CCHE Performance 25°C - Mixed Air

$Q = M (Ha_{in} - Ha_{out})$ $M = 0.22$ kg/s

	Inlet Air				Performance		Outlet Air			
	Air Temperature (°C)	Relative Humidity (%)	Air Enthalpy (Ha_{in}) (kJ kg)	Abs. Humidity (g kg)	Cooling Capacity (kW)	Desiccant Performance (g kg)	Abs. Humidity (g kg)	Air Enthalpy (Ha_{out}) (kJ kg)	Air Temperature (°C)	Relative Humidity (%)
1.1	27.9	76.2	74.3	18.1	1.25	3.9	14.2	68.5	32.0	47.6
	32.0	47.6	68.5	14.2	0.80	2.3	11.9	64.8	34.1	35.6
	34.1	35.6	64.8	11.9	1.03	2.7	9.2	60.0	36.2	24.6
1.2	29.7	57.2	68.2	15.0	1.12	3.2	11.8	63.0	32.6	38.4
	32.6	38.4	63.0	11.8	1.03	2.7	9.1	58.2	34.7	26.5
	34.7	26.5	58.2	9.1	1.03	2.7	6.3	53.4	37.0	16.2
1.3	32.9	39.2	64.6	12.3	1.03	2.7	9.6	59.8	35.0	27.4
	35.0	27.4	59.8	9.6	1.03	2.7	6.8	55.0	37.3	17.2
	37.3	17.2	55.0	6.8	1.03	2.7	5.1	50.3	39.5	9.2

M-Cycle Performance 35°C - Mixed Air

$T_{out} = 7.65 + 0.152\,T_{in} + 681\,w$

	Inlet Air			Outlet Air	
	Air Temperature (°C)	Absolute Humidity (w) (g kg)	Relative Humidity (%)	Air Temperature (°C)	Relative Humidity (%)
1.1	32.0	14.2	47.6	**22.2**	84.5
	34.1	11.9	35.6	**21.0**	76.5
	36.2	9.2	24.6	**19.4**	65.5
1.2	32.6	11.8	38.4	**20.7**	77.2
	34.7	9.1	26.5	**19.1**	66.1
	37.0	6.3	16.2	**17.6**	50.5
1.3	35.0	9.6	27.4	**19.5**	67.9
	37.3	6.8	17.2	**18.0**	53.1
	39.5	4.1	9.2	**16.4**	35.6

	No. of Desiccant System : 3		No. of Desiccant System : 3		No. of Desiccant System : 2
3.1	**19.4** °C	**3.2**	**17.6** °C	**3.3**	**18.0** °C
	Mixed Air Temperature		Mixed Air Temperature		Mixed Air Temperature
	Abs.Humidity : 9.2 g kg | RH : 65.5%		Abs.Humidity : 6.3 g kg | RH : 50.5%		Abs.Humidity : 6.8 g kg | RH : 53.1%

Moisture from Occupants

$$\Delta C = P/n*v$$

	Moisture from Occupant (g/hr)	Total Occupant (person)	**P** Vapour Production (g/h)	**n** Ventilation Rate (ach)	**V** Room Volume (m^3)	**ΔC** Humidity Difference (g kg)	**Abs. Humidity** Absolute Humidity (g kg)
3.3	40	70	2800	1.78	2430	0.65	7.0
3.3	40	70	2800	1.90	2430	0.60	7.4

Return Air

$$(T_{supply} - T_{return}) = Q/V\,C\rho$$

	T_{supply} Supply Air Temperature (°C)	**Q** Cooling Capacity (kW)	**V** Volume Flowrate (m^3/s)	**ΔT** Temperature Difference (°C)	**Return Air** Temperature (°C)
3.2	17.6	77.1	2.8	22.9	40.5
3.3	18.0	72.0	3.0	20.0	38.0

Mixed Air

$$X_{mixed} = (Q_{in} * X_{in} + Q_{re} * X_{re})/(Q_{in} + Q_{re})$$

$$T_{mixed} = (Q_{in}T_{in} + Q_{re}T_{re})/Q_{in} + Q_{re}$$

	Air Volume		**Air Temperature**		**Humidity Ratio**		**Mixed Air**		
	Q_{in} Inlet Air (m^3/s)	Q_{re} Return Air (m^3/s)	T_{in} Inlet Air (°C)	T_{re} Return Air (°C)	X_{in} Inlet Air (kg kg)	X_{re} Return Air (kg kg)	T_{mixed} Mixed Air (°C)	X_{mixed} Mixed Air (g kg)	**RH** Relative Humidity (%)
3.2	0.18	0.42	35.0	40.5	17.8	7.0	38.9	10.2	23.5
3.3	0.18	0.42	35.0	38.0	17.8	7.4	37.1	10.5	26.6

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